

AN INVESTIGATION INTO
THERMOELECTRIC COOLERS

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Master of Science
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by
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AN INVESTIGATION INTO THERMOELECTRIC COOLERS

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ABSTRACT

A thermoelectric cooler (TEC) is a type of cooler that uses the electricity between the junction of two materials to cause a temperature drop. These coolers operate using the Peltier effect. Effectively, a TEC uses electrical energy to pump heat from one area to another. An understanding of the science of how TECs operate as well as their design will be investigated and finally, a unique design for a TEC with a high efficiency is presented. Current TECs do not operate at high efficiency and cannot generate as much cooling as desired. This report creates a design of a high efficiency cooler that operates at a high coefficient of performance. The design will also include phase change material that will differentiate it from its competition and allow for superior cooling and efficiency. The design will also feature vacuum-sealed chambers that will provide very good insulation and allow for optimal cooling for long periods. This report presents a thermal model of a TEC that will investigate the different thermal systems acting on the cooler and provide a better understanding of the cooler at equilibrium once it reaches its coolest position. A unique heat sink is designed to optimally dissipate the heat that is removed by the TECs. The model can also examine the variables that affect the thermal system that are controllable in the design of the cooler. Given the new design, the cooler in this report was able to cool up 14 cans of soda from 25°C to 12°C in approximately 86 minutes given a constant 12 V power supply. The theoretical analysis concludes that this design is sufficient at creating a cooler with a high efficiency that can cool to the desired low temperature.

An Investigation into Thermoelectric Coolers

1. Chapter 1: INTRODUCTION

1.1 INTRODUCTION

Refrigeration and cooling of food and beverages has become a necessary need in the world today. Without proper temperature control, fresh food will go bad, drinks will be lukewarm, and people will be unhappy. For these reasons, when away from the home, coolers are very popular for keeping food and beverages cool on the go. Whether being when out camping, tailgating at a sporting event, or on a road trip in a car or RV, people will utilize coolers to keep their food and beverages at the perfect desired temperature. This luxury has become expected and a part of everyday life. One aspect of these standard coolers are that in reality they are really just good insulators, they do not actually provide any cooling. Instead, ice is required to be put in the cooler to cool the food and beverages. While this is typically convenient and cheap, it would be nice if there were a way to cool food and beverages on the go without the need of ice. This is where a thermoelectric cooler (TEC) is beneficial. A TEC does not require ice to cool products but instead it cools using electrical energy. This application would allow for food and beverages to be cooled at any time without having to rely on ice. With only the requirement of electrical energy, batteries can be taken for a determined amount of time and the possibility of cool food and beverages is available anywhere at any time.

1.2 THERMOELECTRIC EFFECT

A thermoelectric cooler is a type of insulated cooler that uses the electrical power between the junction of two materials to cause a temperature drop. These coolers operate using the Peltier effect. The Peltier effect states that there is a presence of heating or cooling at an electrified junction of two different conductors. Jean Charles Athanase Peltier discovered this effect in 1834 and it is still being researched. As a current is flowing through a junction of two conductors, heat can be created or dissipated at the junction. The Peltier effect can show the amount of heat created or removed per unit time, \dot{Q} , using Eq. 1, i.e.,

$$\dot{Q} = (\pi_A - \pi_B) I \quad (1)$$

where π_A is the Peltier coefficient of material A, π_B is the Peltier coefficient of material B, and I is the electric current from A to B. The Peltier effect is useful in understanding how much heat is carried per unit charge. The Peltier effect is often considered the back-action counterpart to the Seebeck effect.

The Seebeck effect is the conversion of heat directly into electricity at the junction of two conductors. This type of relationship is caused by the creation of an electromagnetic field, which can be predicted by

$$E_{emf} = -S\nabla T \quad (2)$$

where S is the Seebeck coefficient and ∇T is the temperature gradient. As the Seebeck effect is conversion of heat directly into electricity, through the Peltier effect, electricity can create or remove heat.

Also referred to as a Peltier cooler, a thermoelectric cooler is a solid-state active heat pump that transfers heat from one side of the device to the other while consuming electrical energy. TECs are

much less commonly used than traditional vapor-compression refrigeration for cooling. However, TECs do have a few advantages over vapor-compression refrigeration which are that they have no moving parts or circulating liquid, provide great longevity, are invulnerable to leaks, and are small. The disadvantages of TECs are their high cost and poor power efficiency.

Figure 1 describes a typical TEC that it is an object with two sides that are both conductors. DC electricity is sent through the object, bringing heat from one side to the other. The warmer side is connected to a heat sink causing it to stay near ambient temperature. This transfer of heat causes the cooler side to drop below room temperature and can be used as a cooler.

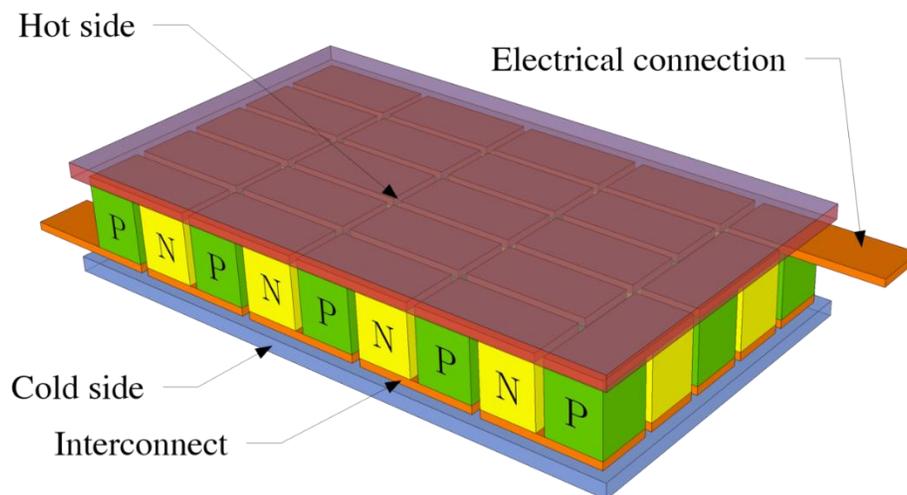


FIG. 1: Schematic of thermoelectric module

Between the cold and hot sides are semiconductors, one n-type and one p-type as it is necessary they have different electron densities. These semiconductors are placed such that they are thermally in parallel and electrically in series with each other. A voltage is applied to the open ends of the semiconductors causing a flow of DC current over the junction of the semiconductors resulting in a temperature difference. The cooling plate absorbs heat and moves it to the other side of the TEC where the heat sink is. A common practice is to connect TEC side by side and put them between two ceramic

plates. Advantages of TECs are that they have no moving parts and require infrequent maintenance, they do not use chlorofluorocarbons, they have very accurate temperature control, they can be used in environments in which traditional refrigerators cannot, and they are controllable by changing the input voltage or current. While these advantages make TEC favorable in many situations, the shortcomings regarding heating and cooling performance still makes TECs not the most viable solution. When using TECs, there is only a limited amount of heat flux that can be dissipated. Also in comparison to traditional cooling systems, vapor-compression, TECs are not nearly as efficient. TECs provide an efficiency of 10-15% of the ideal Carnot cycle refrigerator compared to the 40-60% of traditional refrigeration cycles (Rankine cycles using compression and expansion). Because of these efficiency differences, TECs are typically only found in application in instances in which the nature of the cooler (low maintenance, no moving parts, and compact size) is preferred over pure efficiency.

One issue with TECs is that the more heat they generated by Joule, the less efficient they become. The amount of heat that can be absorbed can be determined by

$$Q = PIt \quad (3)$$

where P is the Peltier Coefficient, I is the current, and t is time. The Peltier Coefficient is determined by the ambient temperature and the materials that the TEC is made of as well as the heat transfer performance, the Peltier module geometry, and Peltier electrical parameters. Selection of materials for the sides of the TEC are dependent on their thermoelectric capabilities. The semiconductors should have narrow band-gaps because they operate in room temperature conditions. Materials that are heavy and have high mobility with low thermal conductivity are ideal. The most common thermoelectric material used currently is bismuth telluride with lead telluride and silicon germanium also being used.

1.3 THERMOELECTRIC MATERIALS & APPLICATIONS

When selecting a material to be used as a thermoelectric material, the two main factors are the material's device efficiency and power factor. These two metrics are determined by the material's electrical conductivity, thermal conductivity, Seebeck coefficient, and how the materials react to temperature change. Given these factors, to obtain a good efficiency, high electrical conductivity, low thermal conductivity, and high Seebeck coefficients are desired. The efficiency for a thermoelectric device is found by dividing the initial energy provided to the load divided by the heat energy absorbed at the hot junction. Any energy that is absorbed at the hot junction and not dissipated into the air by the heat sink will cause the temperature of the hot junction to rise as well as the temperature of the entire thermoelectric device, minimizing the cooling effects. A material's power factor can be determined by

$$\text{Power Factor} = \sigma S^2 \quad (4)$$

where σ is the electrical conductivity and S is the Seebeck Coefficient. A material with a high power factor is able to move more heat from the temperature difference than those with lower power factors. While this is beneficial, it does not necessarily move this heat more efficiently than a material with a lower power factor. Metals typically have low Seebeck Coefficients and are not ideal thermoelectric materials. Conversely, the main materials of interest in thermoelectric materials are semiconductors. Another factor in determining the efficiency is the thermal conductivity of a material. This thermal conductivity is found by the sum of the thermal conductivity of the materials electrons and phonons. The thermal conductivity of the phonons is usually poor and it is important to try and minimize the effects of the phonons. In an attempt to accomplish this and improve thermoelectric materials, the materials are typically studied in three different categories. First are alloys which create point defects or vacancies to help scatter phonons in the unit cell. Second, complex crystals are being studied to try and separate the phonon glass from the electron crystal similar to how the process is done in

superconductors. Third, multiphase nanocomposites are attractive as they scatter phonons at the interfaces of nanostructured materials.

Using the Peltier effect and thermoelectric process, there are many different applications that benefit from this technology. From simple food and beverage coolers to advanced electronic cooling, thermoelectric systems provide cooling below ambient unlike other cooling methods. A simple heat sink dissipates heat into the air in an attempt to return the given product to ambient temperature. Unlike this, a thermoelectric system can provide temperatures below ambient as it is an active cooling method compared to a standard heat sink being a passive cooling method. Another benefit of thermoelectric cooling is that it is controllable. Depending on the amount of current and voltage supplied, a temperature can be held constant to high accuracy. A standard heat sink can only slowly reduce the temperature to ambient and is not able to be controlled. One application of the thermoelectric process is thermoelectric generators. These generators can use this technology to mimic the effects of a heat engine. Thermoelectric generators can be better than heat engines as they are less bulky and do not have any moving parts. However, unfortunately, thermoelectric generators are more expensive and less efficient than a heat engine. Another use of these thermoelectric generators is converting waste heat to electrical power as a mean of recycling energy. A different application of the thermoelectric process is in measuring temperature via thermocouples. Thermocouples use the Seebeck effect to measure the temperature between two objects by keeping the temperature at one junction a constant known temperature and measuring the other junction. A third application of the thermoelectric process is in the cooling of electronics and computer hardware on a microscale. Microprocessors and semiconductors that may otherwise easily over heat can be cooled by the thermoelectric cooling process and allow the electronics to continue running smoothly. These applications are already in use but research is being done to find new applications and continue to improve upon thermoelectric

efficiency. One example of this research is an attempt to create lighting that can be powered by using human body heat as a source.

ADVANTAGES OF THERMOELECTRIC COOLING

- **No moving parts** – virtually maintenance free
- **Small size** – much smaller than traditional cooling
- **Can cool below ambient** – better than traditional heat sink
- **Can heat and cool with same module** – switching polarity of DC power input can change between heating and cooling
- **Temperature Control** – can control temperature by voltage and current input
- **Very Reliable** – solid state construction can lead to life of over 200,000 hours
- **Quiet** – makes no noise during operation
- **Orientation independent** – can be used even in non-gravity environments
- **Environmentally Friendly** – does not generate any gases
- **Can generate electrical power** – running process in reverse will generate DC power

1.4 TEC COOLERS & COEFFICIENT OF PERFORMANCE

One popular application of the thermoelectric process is in cooling food and beverages. By using the thermoelectric process in a cooler, it can eliminate the need for ice to cool food and beverages. While thermoelectric coolers (TEC) are not as effective as standard coolers and ice, for certain applications they can be better. Whenever the primary needs for a cooler are not necessarily pure performance but any of the advantages listed above, a TEC can be a worthy option. Popular uses for TECs are in cars and RVs or while camping.

Currently, there are TECs made that provide cooling but have poor Coefficients of Performance (COP). One example of a current TEC on the market is the “Coleman PowerChill Thermoelectric Cooler” as shown in Fig. 2. This Coleman cooler holds 40 quarts and retails for \$100. A thermoelectric system is located in the cooler along with a fan to aide in the dissipation of heat by the heat sink. The cooler is powered by the standard cigarette lighter receptacle of any vehicle. A standard cigarette lighter receptacle is powered by a car’s 12 volt battery and can draw a current of 10 or more Amps. This cooler can store up to 44 12 oz aluminum cans for cooling. Through the thermoelectric process the cooler is able to cool to 40 degrees below the ambient temperature.



FIG. 2: Photo of competitor Coleman’s PowerChill Thermoelectric Cooler

While some thermoelectric coolers like the one show in Fig. 2 can cool well, they have low coefficients of performance (COP). A design that can provide a better COP would be preferred and will

allow for better cooling. Coefficient of Performance is determined by the amount of cooling done for a given amount of work done by a heat pump which can be calculated by

$$COP = \frac{Q_{cooling}}{W} \quad (5)$$

where $Q_{cooling}$ is the amount of heat removed by the heat pump and W is the amount of Work supplied to the heat pump. By increasing the amount of cooling with more efficient thermoelectric materials and thermoelectric cooler designs, a cooler will have a higher COP. More efficient materials and designs will also allow for cooling to occur with less work supplied to the heat pump. The amount of heat removed by the heat pump in a thermoelectric cooler will also be affected by how well insulated the cooler is.

1.5 SCOPE OF TEC USE

Over time as TECs continue to improve with technology, their viability for use continues to grow. Given the design benefits that come with a TEC as compared to traditional cooling strategies, they are typically used when the cooling ability is not the most important constraint. Most commonly, a TEC is used because of its size, energy input of electricity to provide cooling, or because of its robust design. Currently, there is already a very wide scope of uses for TECs ranging from household items to aerospace applications to military products.

One of the most common uses of a TEC, and what is being explored in this thesis, is for use in portable and camping coolers. TECs have ideal size and performance parameters to provide necessary cooling on the go. Traditional portable and camping coolers require ice at all times to cool food and beverages. TECs require electricity which at times can be easier to obtain than ice and also can be much more sustainable for long periods of time. Another use that is popular for consumers is a thermoelectric wine cooler. Essentially, a thermoelectric wine cooler is simply a miniature refrigerator that uses a TEC to cool the wine as opposed to a traditional compressor system. The benefits of a thermoelectric wine

cooler are that it is more energy efficient, quieter, and does not vibrate compared to a traditional refrigeration system.

While TECs have simple applications in household items, the same technology can be used in extremely complicated military and aerospace applications. Given the TECs design advantages of being small, having no moving parts, relatively infinite life span, and electric input, TECs are ideal for cooling applications in space. They are used to cool and power electronics on board a space craft. Satellites and spacecraft use TECs to reduce the temperature of direct sunlight by dissipating the heat over the cold shaded side and dissipating excess heat as thermal radiation to space. Another reason TECs are useful in space and military applications is because of TECs ability to very accurately control temperature through the input voltage. A TEC can typically control the temperature within ± 0.01 ° C and because of this can be used in very complicated temperature control systems in missiles and space vehicles

Another common use of a TEC is as a power generator. Similar to how the Peltier effect is used to remove heat when subjected to an input voltage, the reverse of this process known as the Seebeck effect can be used to produce electricity. A TEC that is being used in this manner will be designed slightly differently to the goal of producing electricity when subjected to a heat input.

1.6 TEC REVIEWS

Rawat et al. [1] conducted an experiment investigating the cooling power of a thermoelectric module for a 1 liter insulated cabinet. They connected a controllable DC power supply to the thermoelectric module to be able to control the input voltage. A finned heat sink was in contact with the hot side of the thermoelectric module and a fan was used to blow over the heat sink and aide in dissipating the heat. They were able to achieve a temperature change of 11°C with nothing inside the cabinet and a difference of 9°C with 100 mL of water inside the cabinet within 30 minutes. Their results show that it is beneficial to connect a heat sink to the hot side of the TEC to help dissipate heat and

increase the COP of the TEC. Also this information can help to show that as more drinks or objects to be chilled are inserted into the cooler, it will take much more time to reach maximum cooling. Chen et al. [2] ran an experiment comparing the performance of a thermoelectric chiller, or a plate that can chill a cup of water, when it was powered by a constant DC power source and when it was powered by a solar cell. They were interested to see how differing power supplies could affect the temperature difference as well as the COP. They hooked up halogen lights to mimic solar energy and used the power created by the solar panels to power a thermoelectric cooler underneath the cup of water. By running the experiment for the constant DC power supply and the solar cells, they found that the DC voltage provided higher cooling at a rate of 11.9 W compared to 11.2 W of the solar cells. However, even though the solar cells did not cool as quickly as the DC voltage input, they operated at a higher COP due to its lower operating current. This knowledge is needed as the operating current of this desired design will need to be evaluated to maximize COP. Another way of trying to maximize COP is shown by Han et al. [3] when they analyzed how different styles of heat sinks would affect the efficiency of a TEC. They compared a louver style heat sink, an offset strip fin heat sink, and a plate fin heat sink. In the experiment, a TEC was placed beneath a volume of water and a heat sink was put in contact with the hot side of the TEC. The TEC was then ran for a period of time with the three varying heat sink designs and also varying designs of those heat sinks such as fin pitch, fin height, and heat sink length and size. It was found that to maximize COP and cooling of the heat sink, the plate fin heat sink performed best. It was also found that the main goal in design of the heat sink was to decrease the amount of convective thermal resistance. Given the similarity in design of this experiment and this thesis' design, this information will help drive the design of the heat sink near the bottom of the cooler.

A plate fin heat sink will be used to dissipate the heat from the hot side of the TEC. This plate fin heat sink will contain a circular array of fans centered around a fan blowing air over the fins to aide in

heat dissipation. The TECs will be used to remove heat from a cylindrical cooler that will provide cooling to beverages inside. The cooler designed will be discussed in the following section.

2. CHAPTER 2: THEORHETICAL ANALYSIS

2.1 THERMOELECTRIC COOLER (TEC) USED IN THE SYSTEM

A new design for a thermoelectric cooler (TEC) with a better COP will be the focus of this thesis. The goal of the cooler is to utilize the cooling abilities of a thermoelectric cooler to remove heat from the interior of the cooler. The heat that is removed by the thermoelectric coolers will then be dissipated by a heat sink located at the bottom of the cooler. The design utilizes two thermoelectric coolers which each remove 30 W of heat from the cold side. With a COP of about 0.5, the heat to be removed from the hot side will be about 60 W for each thermoelectric cooler and two thermoelectric coolers will generate a total of about 120 W of heat. The dimensions of the thermoelectric coolers that are being used are 40 mm x 40 mm x 15 mm. The thermoelectric coolers are powered by the 12 V input source. A centrally located fan will be located underneath the heat sink and force air through the heat sink to aid in the heat dissipation.

2.2 INITIAL DESIGN

Figure 3 shows a circular thermoelectric cooler with the capability of cooling 14 standard 12 oz cans of soda to be designed. The cooler will have the ability to cool beverages to around 13 degrees Celsius below ambient temperature. The circular body will feature a vacuum sealed chamber to provide very strong insulation. Mounted inside of the vacuum chamber at the bottom of the cooler will be two TECs. These TECs will remove heat from the cooler as explained in Chapter 1. The base of the cooler is a heat sink which is a key aspect of the TEC and allows for heat to continually be drawn from the cooler and dissipated into the ambient room.

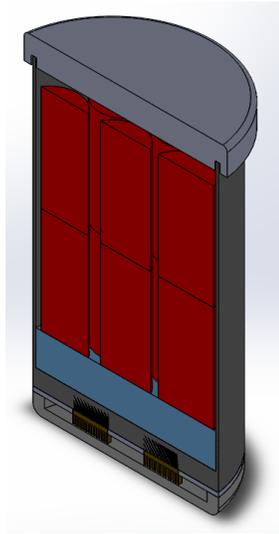


FIG. 3: Cross section of new thermoelectric cooler design

Located at the bottom of the inside chamber of the cooler right above the TECs will be a 4 cm thick pocket of phase change material (PCM). PCM has a very high latent heat of fusion which means its enthalpy changes easily when subjected to a change in energy. For the application in this cooler, it is beneficial because as the TECs help to cool the inside of the cooler, the layer of PCM will also cool and help to lower the temperature of the beverages inside. Also, the PCM will become cold and have stored energy until the TEC is turned off. This means that after the TEC is turned off and no longer supplied power, the layer of PCM will continue to keep the interior of the cooler at a low temperature for a duration. The heat that is removed by the thermoelectric coolers will then be dissipated by a heat sink located at the bottom of the cooler. A centrally located fan will be located underneath the heat sink and force air through the heat sink to aid in the heat dissipation.

2.2.1 TEC DESIGN

As mentioned before, the design will use two TECs to remove heat from the inside of the cooler. The dimensions of the thermoelectric coolers that are chosen are 40 mm x 40 mm x 15 mm and remove

about 30 W from the cold side of the TEC and with a COP of about 0.5. 60 W of heat is generated from the hot side. The design places the two TECs between the layer of PCM and the heat sink. The idea of this design is that the thermoelectric coolers will remove heat from the inside of the body of the cooler and be able to dissipate the heat to the heat sink. The location of the two thermoelectric coolers can be seen below in Fig. 4. The figure is a cross section and the cans and PCM sections have been removed as to better see the location of the thermoelectric coolers. The thermoelectric coolers are the black squares located near the bottom of the design right above the heat sink.

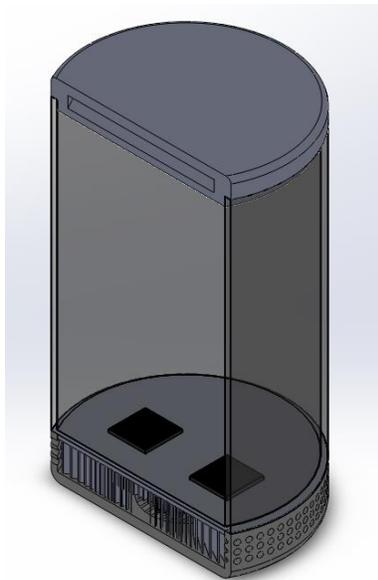


FIG. 4: Cross section of new thermoelectric cooler design

2.2.2 BODY DESIGN

Given the design objective of cooling 14 standard size cans, the size of the can was the driving factor in the dimensioning of the cooler. A standard soda can has a diameter of 6.6 cm and a height of 12.2 cm. The circular cooler will allow cans to be packed in two tiers of 7 cans. A hexagonal shape with a can in the middle will allow for 14 cans to be cooled in the most volume efficient way. Given that three

cans would be lined up in a row, the inner diameter of the cooler needed to be larger than the diameter of the three cans. With this in mind, an inner diameter of 21 cm was chosen. Similarly, the height of the inside of the cooler needed to be at least as tall as two cans stacked so a height of 27 cm was chosen. The body of the cooler will be made of 304 stainless steel and will be formed into a hollow chamber. 304 stainless steel will be chosen as the material due to its high strength and toughness, its high corrosion resistance, its relatively low price, and its ease of manufacturing. Inside this chamber will be a vacuum chamber that will provide excellent insulation as nearly all gas will be removed allowing for little to no heat loss or transfer

2.2.3 FAN DESIGN

Given the overall initial design, it is better to use a centrally located fan would be used for the design. The fan must run from the input 12 V supply from the car cigarette lighter. The fan design was very important as it also played a large part in the design of the heat sink. The velocity of the air flowing from the fan affects the heat transfer coefficient which in turn affects the performance of the heat sink. Because of this, it is desired the fan to push air through the heat sink channels as fast as possible. However, fans are not rated based on the velocity of the air they expel but rather by the volumetric flow rate of the air they expel. For this reason, the geometry of the channels of the heat sink were important to determining the fan that would be ideal for the design.

It was determined that any air velocity moving through the channels greater than 2 m/s would be too loud. Air velocity traveling at 2 m/s creates 35 dB of noise which is too much. Therefore, the fan was determined by finding the maximum amount of cfm that would not create an air velocity over 2 m/s.

2.2.4 PHASE CHANGE MATERIAL DESIGN

A layer of phase change material is placed at the bottom of the inside of the cooler body so that it can store energy and continue to cool the beverages even after the cooler is no longer connected to the power source. The mass of phase change material will be optimized. The phase change material has a latent heat of fusion of 200,000 J/kg so it can absorb and store a large amount of energy. The phase change material that was chosen is 8°C phase change material. This means that at 8°C the material will change from a liquid to a solid and stores the energy from the removed heat by the thermoelectric cooler. Then after the cooler is turned off, the phase change material will cool the beverages as it melts back into a liquid.

2.2.5 MATERIAL SELECTION

The material chosen for the body of the cooler is 304 Stainless Steel. This material was chosen because it is strong and easy to manufacture. It is easy to control the thickness and obtain the design dimensions that are desired. For the heat sink, it will be made of copper as copper is very conductive and will dissipate the heat removed from the thermoelectric cooler effectively.

2.2.6 HEAT SINK DESIGN

The design of the heat sink begins with the conceptual design before any calculations were made. Given the overall design of the cooler, it was chosen that there would be one circular fan placed underneath the heat sink to force air through the channels to increase the heat dissipation. Given the calculated cfm requirements, the power source limitations, and the desired performance of the heat sink, a circular array of fins was determined to be optimal for the heat sink design. A circular array of fins was the best way to maximize heat sink surface area and fit in the body of the cooler. The design was chosen to have a long fin located every six degrees and a short fin every six degrees. The short fins are

offset by three degrees and set back by 35 mm to fill the void left as the long fins expand further and further apart from the center of the heat sink. The choices for the dimensions of the heat sink fins began with considerations of traditional heat sink dimensions and altered to obtain the desired surface area.

For a typical heat sink, the performance of a heat sink is largely dependent on the design of the fins. In this thesis, the calculations and design for a single channel will be found and then those results can be used for the entirety of the heat sink. The main design constraints of the heat sink are the length, height, and thickness of each individual fin as well as the spacing between the fins. These dimensions are shown below

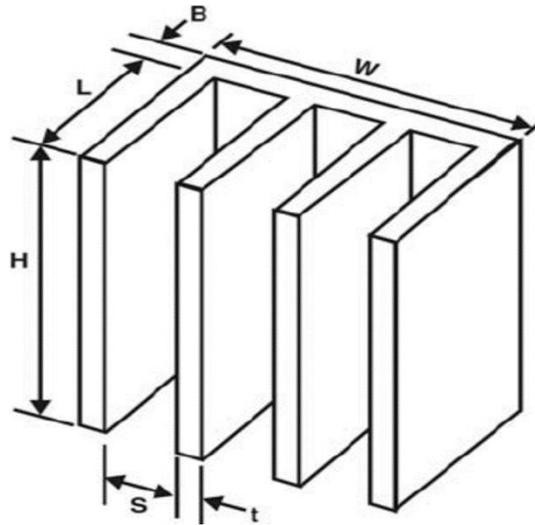


FIG. 5: Heat sink fin dimensions

Certain properties of heat sinks are known and will be examined in the design process. It is known that the longer, thicker, and taller that each individual fin is, the more heat it will dissipate. It is also known that the further apart the fins are from one another, the more heat they will dissipate. Given these fundamental understandings of heat sinks, a standard design of a functional heat sink was found. Using this initial design, multiple variables were tested by altering one dimension while keeping all other dimensions constant to see the affect that each dimensional change would have. For instance, while

keeping the height and length constant, but testing various thicknesses, and repeating for the other dimensions, it was possible to determine which variables played the biggest roles in the dissipation of heat.

For this design and these calculations there were many assumptions that needed to be made. One of which was that the air flowing through the channels was viewed as forced convection over a flat plate. This allowed for the Reynolds and Nusselt correlations to be used to estimate the heat transfer coefficient. Next, it was important to consider that the base of the plate is a constant temperature throughout and that it evenly distributes and dissipates heat. This will allow for calculation of the temperature gradient as well as simplifying calculation.

Figure 6 shows the physical model of the heat sink to be used for the project. The heat sink is a circular array of fins evenly spaced around the middle of the heat sink base with a cut out in some of the fins to allow room for the circulating fan. Using this physical design, a simplified model that is isolated to include only one of the channels between fins is shown below in Fig. 7. By taking a cross section of the single channel above as indicated by the A-A cut, it is possible to conduct analysis on a single channel of the design that can be scaled for the entire heat sink. The cross section of the heat sink is shown below in Fig. 8. The cross section is turned on its side to fit the orientation of the page.

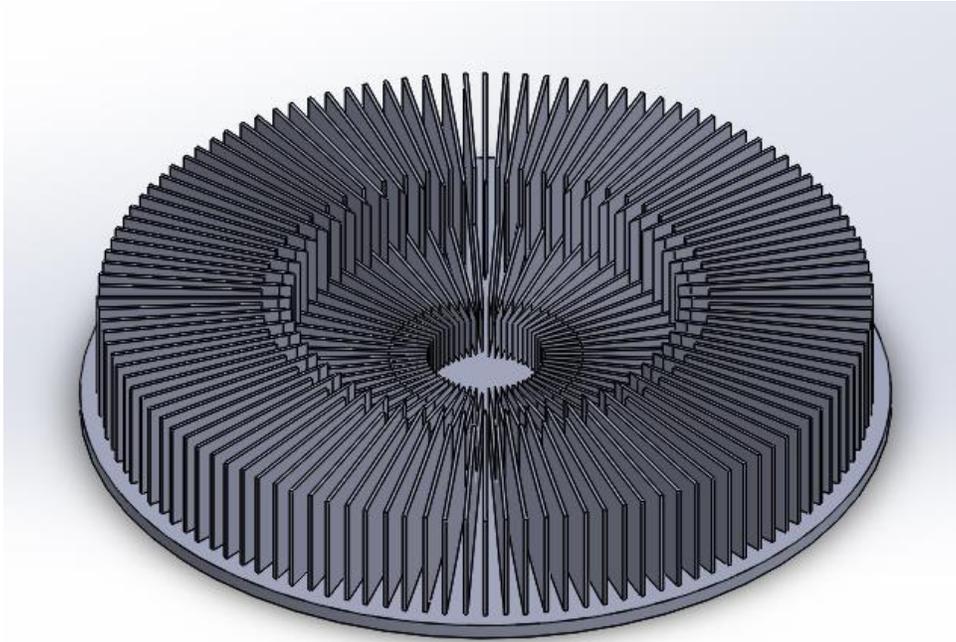


FIG. 6: Heat sink initial design

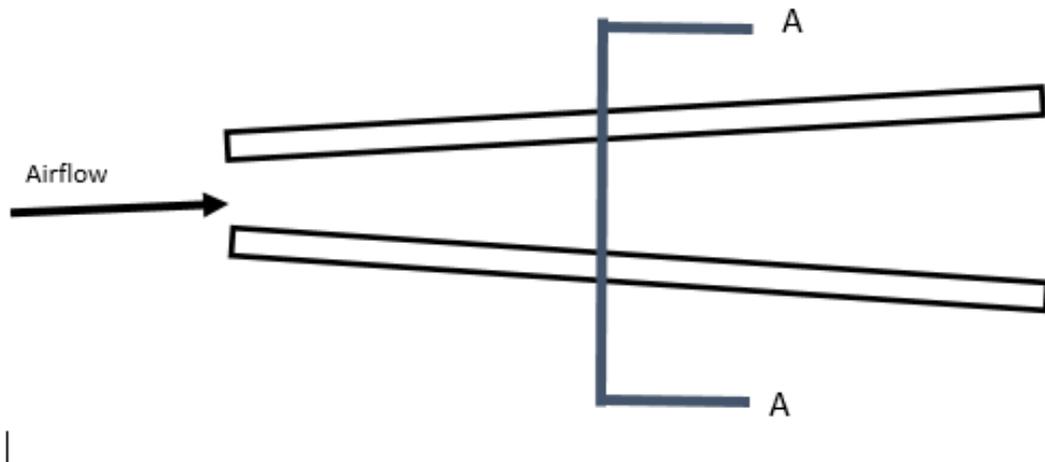


FIG. 7: Schematic of a channel between two fins

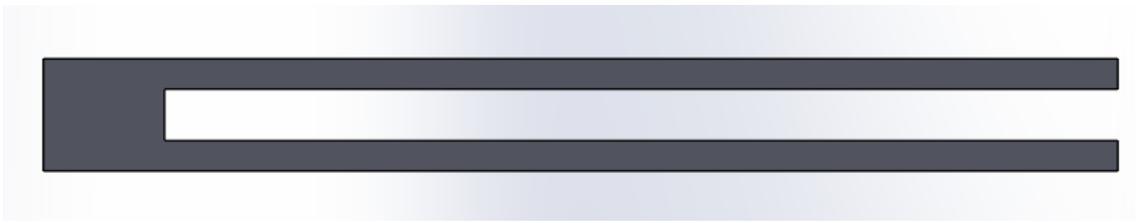


FIG. 8: Cross section of a single channel

2.3 HEAT SINK MATHEMATICAL MODEL

By orienting the fin channel in this way, it is possible to calculate the total rate of heat transfer by convection from the fins by

$$q_t = hA_t \left[1 - \frac{NA_f}{A_t} (1 - \eta_f) \right] \theta_b \quad (6)$$

where h is the heat transfer coefficient, A_t is the total area, N is the number of fins, A_f is the area of the fins, η_f is the fin efficiency, and θ_b is the temperature difference between the base and the ambient.

The total area, A_t , in Eq. (6) can be found by

$$A_t = NA_f + A_b \quad (7)$$

where A_b is the area of the base between the fins. The area of the fins is found by using the equation

$$A_f = 2wL_c \quad (8)$$

where w is the width of the fins and L_c is the characteristic length. The characteristic length can be found using the equation, i.e.,

$$L_c = L + \frac{t}{2} \quad (9)$$

Where L is the length of the fin and t is the thickness. For the condition shown in Fig. 8, the fin efficiency can be calculated by

$$\eta_f = \frac{\tanh mL_c}{mL_c} \quad (10)$$

where m can be found by

$$m = \sqrt{\frac{hP}{kA_c}} \quad (11)$$

where h is the heat transfer coefficient, P is the perimeter of the fin, k is the thermal conductivity of the fin, and A_c is the area of the end of the fin. The perimeter of the fin and the area of the end of the fin can be found using the equations

$$P = 2w + 2t \quad (12)$$

$$A_c = wt \quad (13)$$

The temperature difference between the base and ambient is defined by

$$\theta_b = T_b - T_\infty \quad (14)$$

and the total heat transfer removed by air can be found by

$$Q = C_p \dot{m} \Delta T \quad (15)$$

where Q is the desired amount of heat to be dissipated, C_p is the specific heat of the air, \dot{m} is the mass flow rate, and ΔT is the change in temperature of the air from inlet to outlet. The air mass flow rate in Eq. (15) can be calculated by

$$\dot{m} = \dot{V} \rho \quad (16)$$

where \dot{V} is the volumetric flow rate of the air and ρ is the density of the air. By making this substitution and solving the previous equation the necessary volumetric flow rate needed to dissipate the desired amount of heat can be found. Next, the heat transfer coefficient will be found using the Reynolds and

Nusselt correlations for flow over a flat plate. This assumption is made as through each channel of the heat sink, the air will be flowing as if over a flat plane. The Reynolds number is defined by

$$Re = \frac{u_{\infty} L}{\nu} \quad (17)$$

where u_{∞} is the velocity of the air, L is the length of the channel, and ν is the kinematic viscosity. To find the velocity of the air in a channel, it must be determined how much air is going through each channel. To do find the amount of air going through each channel, the total cfm as found above will be divided by the number of channels. Next, by using the geometry of the channels and the volume of air flowing through the channels, the velocity of the air in each channel can be found. Using the Reynolds number along with a rearrangement of the Nusselt correlation, the heat transfer coefficient can be found by

$$Nu = \frac{hL}{k} = 0.664 Re^{1/2} Pr^{1/3} \quad (18)$$

where k is the thermal conductivity of the air and Pr is the Prandtl number. Now that the heat transfer coefficient has been found, it is possible to find the total rate of heat transfer from convection as shown in the first equation.

Next the temperature distribution throughout the fin will be determined. It is important to note however that certain assumptions have been made for calculation. First, it is assumed that the temperature of the base is uniform throughout. Next, it is assumed the heat sink is perfectly conductive and will distribute heat evenly as expected. The temperature distribution in a fin of uniform cross section can be calculated by

$$\frac{\theta(x)}{\theta_b} = \frac{\cosh m(L-x) + \left(\frac{h}{mk}\right) \sinh m(L-x)}{\cosh mL + \left(\frac{h}{mk}\right) \sinh mL} \quad (19)$$

where L is the length of the fin, x is the desired point of the fin that is being tested, h is the heat transfer coefficient, k is the thermal conductivity, and m is a constant as found above. By multiplying θ_b to the right side of the equation, $\theta(x)$ is found and can be used in the equation below to solve for the temperature of the fin at any distance, x , along the length of the fin

$$T(x) = \theta(x) + T_{\infty} \quad (20)$$

2.4 COOLER THERMAL MODEL

In order to obtain a better understanding of the cooling process of the TEC, a thermal model is useful to understand the many different thermal effects taking place on the Cooler. While the TECs remove heat from the inside of the cooler to cool the beverages, there are other factors acting against the TEC cooling to heat the interior of the cooler. The factors that will be examined in this model are the cooling of the TECs, the heat transferring to the cooler from conduction and radiation, the heat that is absorbed by the water due to its latent heat, and the sensible heat causing transfer to the cooler. For the steady state process, the cooling, $Q_{cooling}$, produced by the cooler is equal to the sum of the heat, Q_H , transferred to the cooler through conduction and radiation, is the heat, Q_{LH} , absorbed by the water due to its latent heat, and the heat, $Q_{s,l}$ and $Q_{s,m}$, transferred to the cooler due to the sensible heat, which can be expressed as

$$Q_{cooling} = Q_H + (Q_{LH} + Q_{s,l} + Q_{s,m})t \quad (21)$$

where $Q_{cooling}$ is the heat transfer rate removed by the TECs, Q_H is the heat transfer rate transferred to the cooler through conduction and radiation, Q_{LH} is the heat transfer rate absorbed by the water due to its latent heat, and $Q_{s,l}$ and $Q_{s,m}$ are the heat transfer rate transferred to the cooler due to the sensible heat. For this model, the value of $Q_{cooling}$ will be assumed as further testing of the physical TEC would

be required to find the exact amount of heat removed in this application. The heat transfer rate to the cooler through conduction and radiation can be expressed as

$$Q_H = \frac{T_H - T_C}{R_{COND,i} + R_{COND,o} + R_{RAD}} \quad (22)$$

where T_H is the temperature outside the cooler, T_C is the temperature inside the cooler, $R_{COND,i}$ is the thermal resistance caused by conduction in the inner layer of the cooler body, $R_{COND,o}$ is the thermal resistance caused by the conduction in the outer layer of the cooler body, and R_{RAD} is the thermal resistance caused by the radiation between the two layers of the cooler body. The conduction resistance can be found by

$$R_{COND} = \frac{\ln \frac{r_2}{r_1}}{2\pi Lk} \quad (23)$$

where L is the length or thickness of the material, k is the thermal conductivity of the material, and A is the surface area of the given material. The radiation thermal resistance, R_{RAD} , can be found by

$$R_{RAD} = \frac{1}{h_{RAD}A} \quad (24)$$

where the radiative heat transfer coefficient, h_{RAD} , can be expressed as

$$h_{RAD} = \frac{\sigma(T_H^4 - T_C^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} \quad (25)$$

where σ is the Stefan-Boltzmann constant, ε_1 and ε_2 are the emissivity of the stainless-steel walls of the cooler. Considering both walls of the cooler are made of the same stainless steel material and will have the same surface finish, ε_1 and ε_2 are equal to each other.

As mentioned in Eq. (21), another factor in the thermal model is Q_{LH} or the heat absorbed by the water during phase change due to its latent heat. Latent heat is the amount of heat that is required

to change a certain material from a solid to a liquid which in our case the material is water. This amount of heat is found by

$$Q_{LH} = \frac{mL_f}{\Delta\tau} \quad (26)$$

where m is the mass of the material, $\Delta\tau$ is the total time, and L_f is the latent heat of fusion. The final variables to be solved from Eq. (21) are $Q_{s,l}$ and $Q_{s,m}$ which are the sensible heats caused by the mass of the water and the mass of the cooler respectively, i.e.,

$$Q_s = mC_p(T_H - T_C)/\Delta\tau \quad (27)$$

where m is the mass of the solid or liquid and C_p is the specific heat of material.

As this is a theoretical model, there are many assumptions that have been made. In this model, only the heat transfer due to conduction and radiation will be considered so heat transfer due to convection will not be included. Another assumption is that the TECs remove heat from the system at a uniform rate. This assumption is not perfect as the amount of heat removed will change depending on the temperature difference between the hot and cold sides of the TEC as well as the performance of the heat sink. Given that the cooler is still in the design process and the design and material selection of the lid of the cooler has not been finalized, the heat transfer through the lid and caused by the lid are not included in this model. Also, as the surface finish of the stainless steel that is the body of the cooler is unknown, the emissivity will be assumed. Although the emissivity is assumed, choosing different emissivity values for design will be examined to understand how they affect the heat transfer due to radiation. In the calculation of the sensible heat for the contents of the cooler, it is dependent on the mass and the specific heat of the contents of the cooler. For this model and for ease of calculation, it is assumed that the contents of the cooler are all water and therefore ignoring the containers that would be holding the water as well as the air surrounding the beverages.

2.5 THERMOELECTRIC COOLER POWER DRAW

The amount of power that will be consumed by the thermoelectric coolers, can be found by

$$Q_{out} = Q_{in} + W \quad (28)$$

where Q_{out} is the heat transfer rate removed by the thermoelectric coolers, W is the power consumed by the thermoelectric cooler, and Q_{in} shown using Eq. (28) can be found by

$$Q_{in} = COP * W \quad (29)$$

where COP is the coefficient of performance. As shown in the equations above the higher the coefficient of performance is, the less power will be consumed by the thermoelectric coolers. The power consumed by TEC can be found by

$$W = IV \quad (30)$$

where I is the current drawn and V is the input voltage.

3. CHAPTER 3: EXPERIMENTAL INVESTIGATION

3.1 PROTOTYPE CREATION

Due to the difficulty of manufacturing a vacuum sealed cooler and the limitations of this thesis, it was not possible for a prototype to be created. The manufacturing process of creating a vacuum seal is very precise and expensive on a small scale. A custom designed heat sink that was manufactured to the exact specifications of the thermal modeling would also be expensive and difficult to manufacture. Because this thesis is a research based project desiring optimal thermal performance, the manufacturability of each individual part was not the primary concern. Another issue that arose with manufacturing was that in order to properly test the design parameters, multiple different sizes of

coolers and heat sinks would have needed to be created to test the variables that affect the performance. If this project were continued past this report, a prototype would ideally be created and tested to evaluate the thermal models used to estimate the cooler performance.

3.2 THERMAL TESTING

If a prototype were created, thermal testing would be conducted to determine the performance of the prototype. Considering the primary focus of the cooler is temperature control, the testing that would be done is timed temperature tests using thermocouples. The first test would be the inside of the cooler body where the beverages are located. The start of the test would be with the cooler at ambient temperature. Then the cooler would be plugged in and the temperature of the cooler would be recorded until equilibrium was reached. Next, the cooler would be unplugged and the temperature would be recorded until the cooler was back to equilibrium at ambient temperature. This test would be repeated many times as to test the performance of the cooler. This test will allow the cooler performance to be compared to the thermal model. The testing will display whether or not the thermoelectric coolers were able to cool the beverages to the desired temperature in the predicted amount of time. Given the prototype design parameters, the selection of the fan would be the largest possible independent variable that can be tested. Fans with different specifications would be tested as described below.

3.2.1 Varying Fan Testing

The choice of a fan to be used to aide in heat dissipation plays an important role in the performance of the heat sink. The higher the volumetric flow rate the fan can produce, the faster the air will travel through the channels of the heat sink. The speed of the air traveling through the channels helps to determine the heat transfer coefficient. For this reason different fans of varying volumetric flow rates would be tested to see how the heat dissipation was affected. It is shown through modeling that

the larger the volumetric flow rate created by the fan, the quicker heat can be dissipated by the heat sink. Another concern that arises with fan selection is the noise caused by the fan. Because the cooler is meant to be a consumer product, the noise caused by the air traveling through the fan cannot be too loud. Because of this, a decibel meter will be used to monitor the noise created by the different fans. The goal is for the noise from the fan to be kept below 35 dB.

4. CHAPTER 4: RESULTS AND DISCUSSION

4.1 HEAT SINK INITIAL DESIGN

As stated above, using the known desired dimensions of the cooler, an initial heat sink was designed. Given that a circular array of fins would be used, with the length of each fin being near half of the total diameter of the cooler itself, the length of a single fin was determined to be 8.5 cm. Using standard dimensions for typical heat sinks of this nature a fin thickness of 1 mm was chosen. Considering the desired 120 fin heat sink as described previously, a fin height of 3.15 cm was found to be appropriate for approximately dissipating the desired amount of heat. Once the initial dimensions were set for a single fin, next, the heat transfer coefficient of the air flowing through the channel was found. To begin, the velocity of the air moving through the channels will be calculated. To find this value, from Eqs. (15) and (16) it can be found that

$$\dot{V} = \frac{Q}{c_p \rho \Delta T} = \frac{120 \text{ W}}{\left(1 \frac{\text{kJ}}{\text{kgK}}\right) \left(1.225 \frac{\text{kg}}{\text{m}^3}\right) (10 \text{ K})} = 0.00979 \frac{\text{m}^3}{\text{s}} = 20.748 \text{ CFM} \quad (31)$$

Using this value and the known diameter of the rectangular channel that the air will be flowing through it can be found that air moves through individual channel is 6.81 m/s. Using this value and the Reynolds and Nusselt correlations shown in Eqs. (17) and (18), the heat transfer coefficient can be found as

$$h = \frac{\left(0.02624 \frac{\text{W}}{\text{mK}}\right)}{0.085 \text{ m}} 0.664(36909.39)^{\frac{1}{2}}(0.707)^{\frac{1}{3}} = 35.0823 \frac{\text{W}}{\text{m}^2\text{K}} \quad (32)$$

With the dimensions of the fin and the heat transfer coefficient of the air flowing through the channel and considering Eqs. (7)-(14), it can be found that,

$$A_t = NA_f + A_b = 0.00557 \text{ m}^2$$

$$A_f = 2wL_c = 0.00544 \text{ m}^2$$

$$L_c = L + \frac{t}{2} = 0.032 \text{ m}$$

$$\eta_f = \frac{\tanh mL_c}{mL_c} = 0.944$$

$$m = \sqrt{\frac{hP}{kA_c}} = 13.305$$

$$P = 2w + 2t = 0.172 \text{ m}$$

$$A_c = wt = 0.000085 \text{ m}^2$$

$$\theta_b = T_b - T_\infty = 5^\circ\text{C}$$

Using all of these values it is possible to solve Eqs. (19) and (20) to find the temperature distribution throughout the fin. Figure 9 describes the temperature distribution from a fin. This temperature distribution is the temperature of the fin at a given distance, x , from the base of the fin. This distribution helps to display how well the heat is being dissipated by the fin. As shown, the temperature of the fin drops as the point of interest is moved further away from the base of the fin.

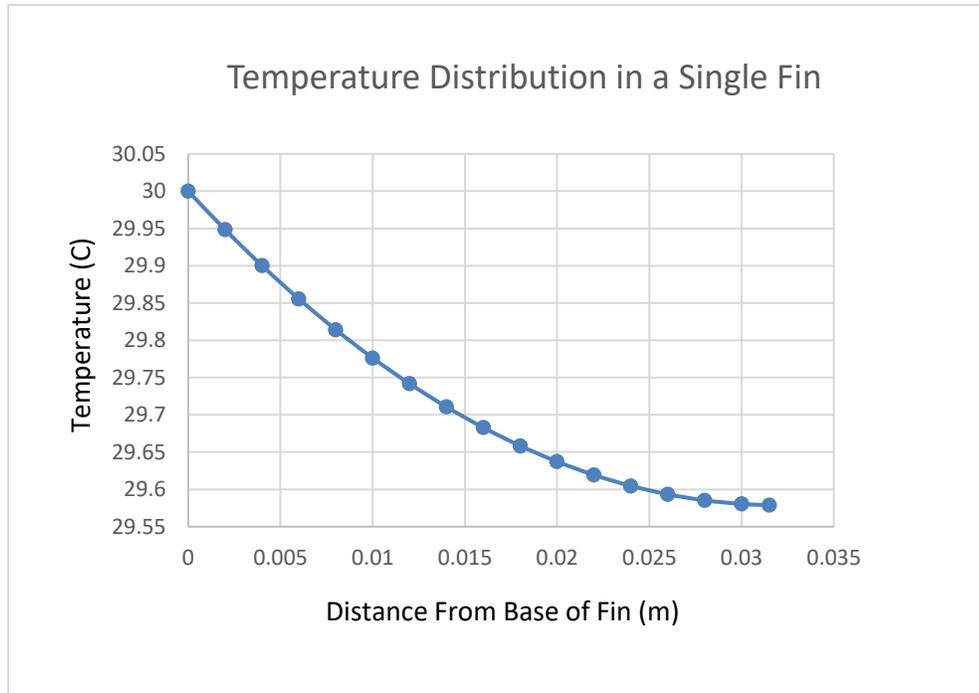


FIG. 9: Length effect of the temperature distribution on a fin

Using the data described above, the total rate of heat transfer due to convection, Eq. 6, can be found for this fin design. The total rate of heat transfer due to convection from one fin can be found as 0.924 W. The number of fins required for the heat sink to dissipate the desired 120 W of heat can then be found by

$$\# \text{ of fins needed} = \frac{120 \text{ W}}{q_t} \tag{33}$$

It is found that this fin design would require 129.8 fins to dissipate the heat removed by the TECs so this design must be altered.

4.2 HEAT SINK DESIGN TESTING

With the results of the initial design providing a good baseline for altering the design to find the ideal design, next the process was to determine which of the design variables most impacted the total

heat sink performance and how they impacted the performance. The four variables that were determined to be tested were fin length, fin height, fin thickness, and heat transfer coefficient of the air flowing through the heat sink. For each variable tested, all other variables were held constant to see how a change in the tested variable would affect the results. With each variable tested, the temperature distribution and the heat transfer rate due to convection will be evaluated.

4.2.1 Varying Fin Length

The initial heat sink design used a fin length of 0.0315 m. In this model, all other variables will be held constant from the initial design but the fin length will be tested from a variety of lengths ranging from 0.024 m to 0.0375 m. Using this range of values for fin length, the length effect on the temperature distribution and heat transfer rate can be found. As shown in Fig. 10, the longer the fin is, the more the heat is distributed along the fin. This is because as the fin grows in length, it has more mass and surface area to evenly dissipate the heat, and therefore, as the fin gets longer, it progressively is at a lower temperature than its shorter fin counterparts. Figure 11 shows how increasing the fin length also increases the rate of heat transfer due to convection. This is important because the goal of the fin is to dissipate as much heat as possible. A fin design that dissipates the heat at a higher rate, will allow the heat sink to cool faster as well as allow for the heat sink design to require fewer fins. Given these results, the ideal design will incorporate a longer fin.

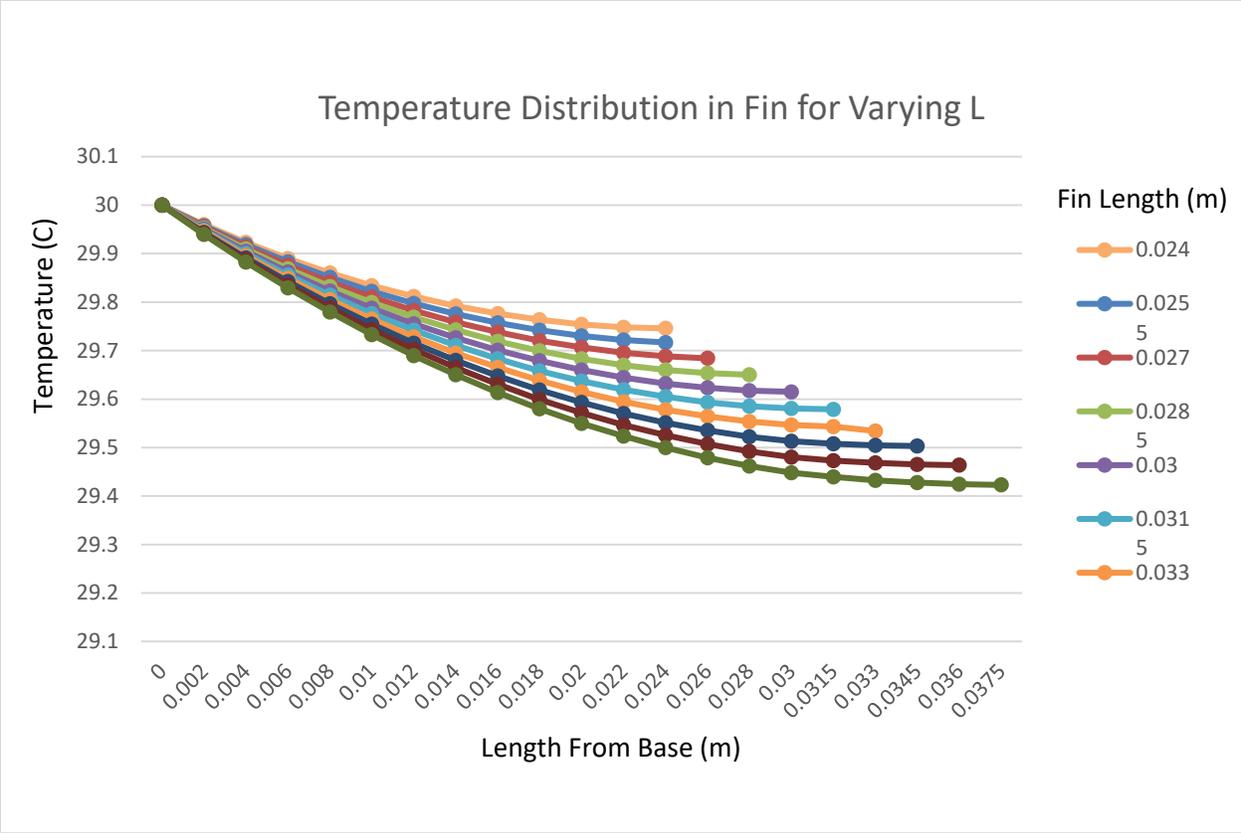


FIG. 10: Length effect on the temperature distribution in a single fin

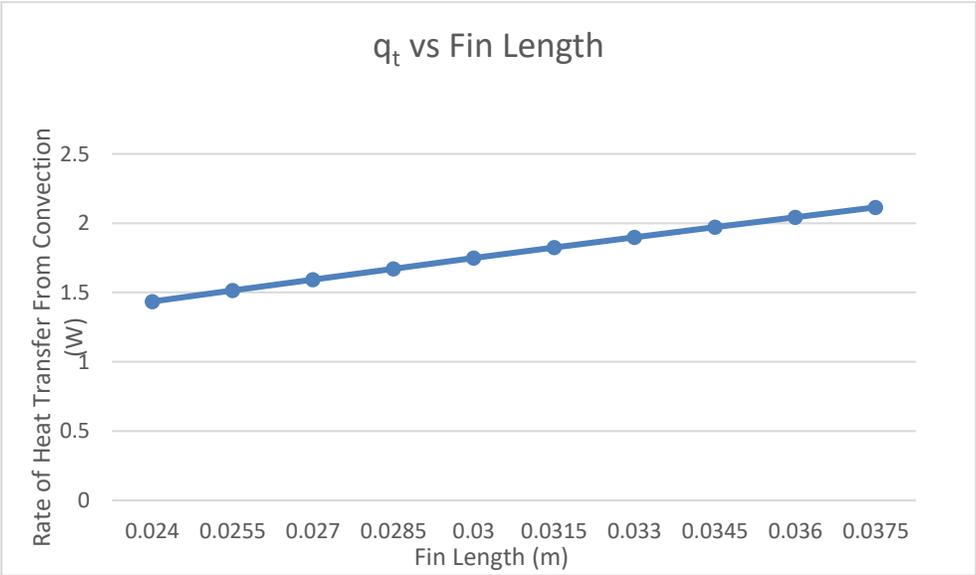


FIG. 11: Fin length effect on the heat transfer rate

4.2.2 Varying Fin Height

The initial heat sink design used a fin height of 0.085 m. In this model, fin heights ranging from 0.06 m to 0.105 m will be evaluated for their effect on the heat transfer rate due to convection and the temperature distribution in the fin. Figure 12 shows that fin height does not affect the temperature distribution because the distance of all parts of the fin from the base remains constant even with varying fin heights. More importantly, Fig. 13 shows that the fin height does play a role in heat transfer rate due to convection. As the fin height increases, the heat transfer rate due to convection also increases. This is because as the fin height increases, it provides more surface area for the fin to dissipate heat. The final heat sink design will incorporate a greater fin height to aide in heat dissipation.

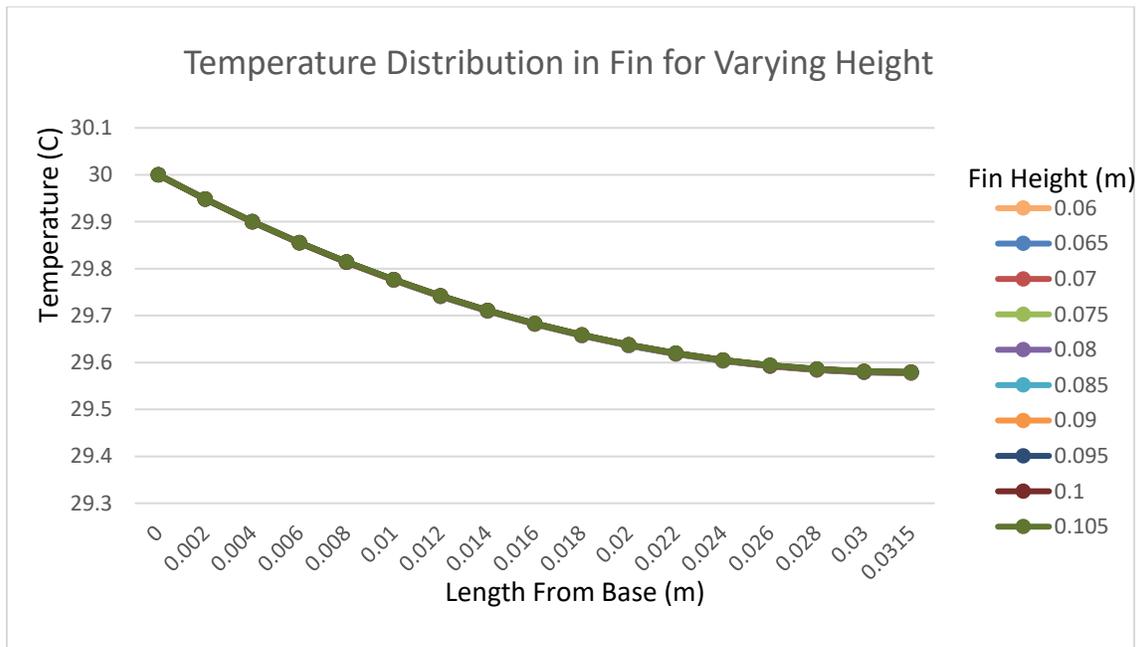


FIG. 12: Fin height effect on the temperature distribution in fin

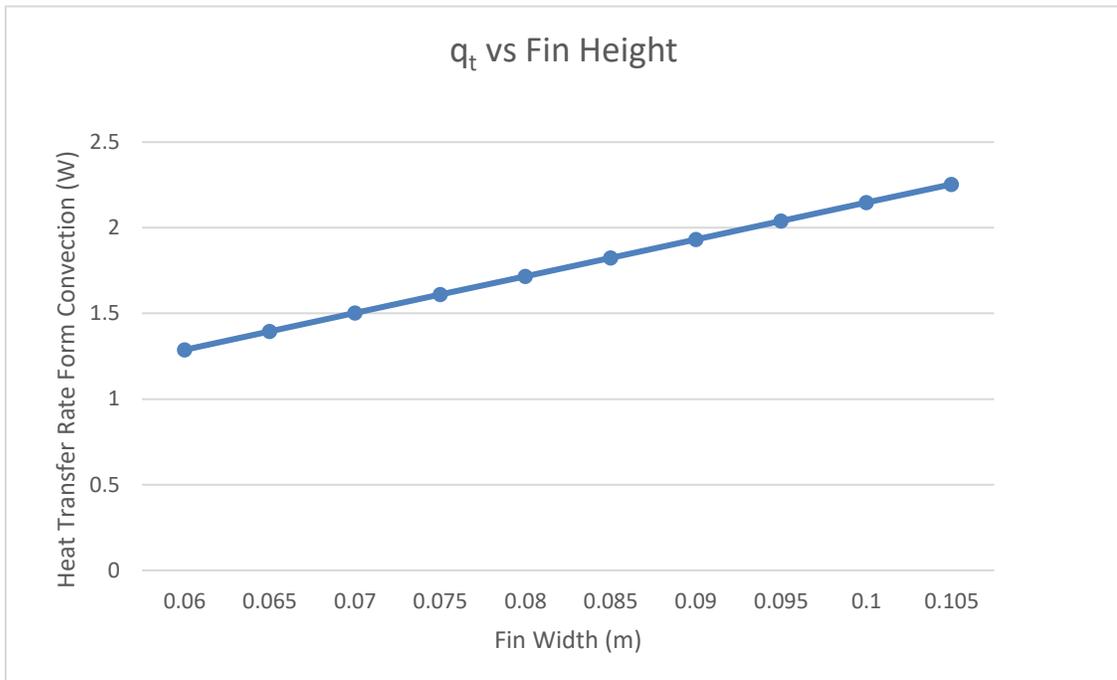


FIG. 13: Fin height effect on the heat transfer rate in a fin

4.2.3 Varying Fin Thickness

The initial heat sink design used a fin thickness of 0.001 m. In this model, fin thicknesses will range from 0.0005 m to 0.002 m. This range of fin thickness will be evaluated on how it effects the temperature distribution in the fin and the rate of heat transfer due to conduction in the fin.

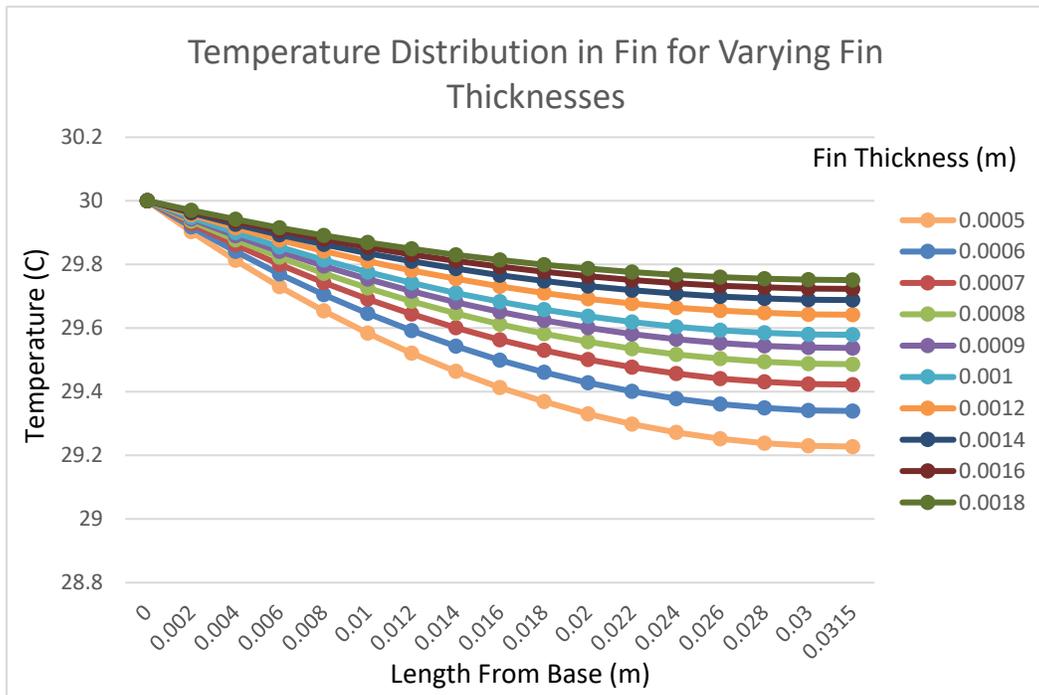


FIG. 14: Temperature distribution in fin for varying fin thicknesses

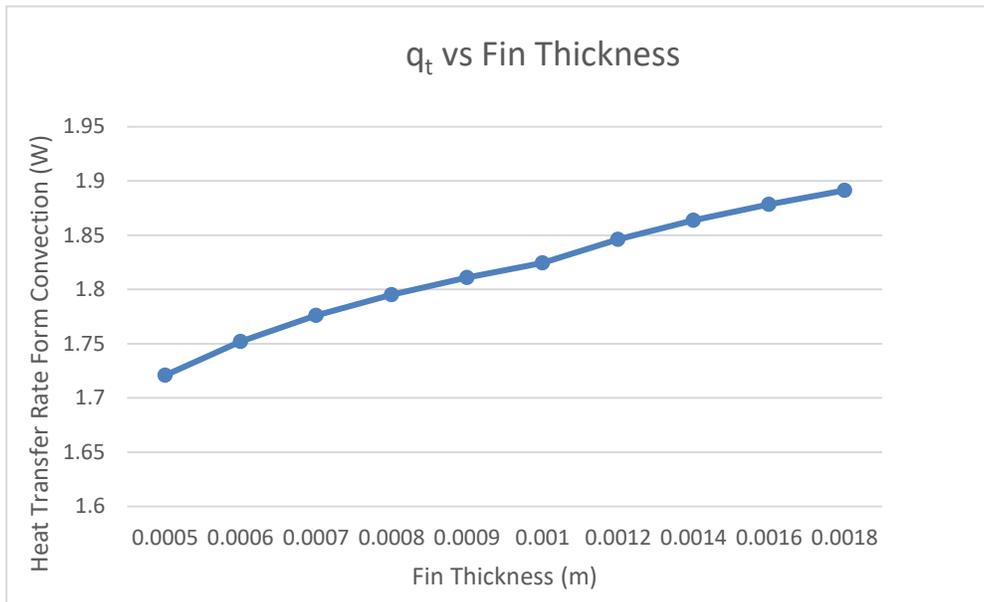


FIG. 15: Heat transfer rate due to convection for varying fin thicknesses

Fig. 14 shows that the temperature distribution is better for the smaller thickness values. This is true in that the further away from the base of the fin the temperature is lower, but this does not mean that the thinner fin is better for dissipating heat. From Fig. 15, it is clear that heat will be dissipated at a faster rate for a thicker fin. This is interesting because although a thicker fin can dissipate heat at a faster rate, the end of the fin from the base is cooler for a thin fin. This phenomena is explained because a thicker fin allows for a more uniform dissipation of the heat. Because the fin is thicker, more heat is able to travel to the end of the fin causing it to be warmer than the thin fins that do not dissipate the heat as evenly. It is important to note that the difference in the fin design that dissipates the most heat and the fin design that dissipates the most heat is not as large for varying thicknesses as it is for other variables. A fin with a thickness of 0.0005 operates at a rate of heat transfer due to convection of 1.72 W, where the fin with the thickness of 0.002 operates at a rate of heat transfer due to convection of 1.90 W. Even increasing the thickness by a magnitude of 4 does not very strongly affect the performance

of the fin. With this in mind, the other variables will be of more importance in the final design than fin thickness.

4.2.4 Varying Heat Transfer Coefficient

The initial design utilized a heat transfer coefficient of $35.08 \frac{W}{m^2K}$. This value was found using the air properties and velocity found from calculating the necessary volumetric flow rate needed to dissipate the desired heat in ideal conditions. In this model, the heat transfer coefficient values will range from $20 \frac{W}{m^2K}$ to $47 \frac{W}{m^2K}$. Figure 16 shows that as the heat transfer coefficient increases, the temperature distribution in the fin increases. This is correct in that as the heat transfer coefficient increases, it allows for heat to be dissipated at a faster rate causing the ends of the fins to be cooler. Fig. 17 displays how the heat transfer rate due to convection is very affected by the heat transfer coefficient of the air flowing through the channels. Of all of the four variables that were tested, the heat transfer coefficient of the air had the greatest impact of determining the rate of heat transfer due to convection. This is expected as the heat transfer coefficient describes the ability for the heat to be transferred over the area that it is flowing through. In this application, heat transfer coefficient is largely dependent on the velocity of the air flowing through the channels.

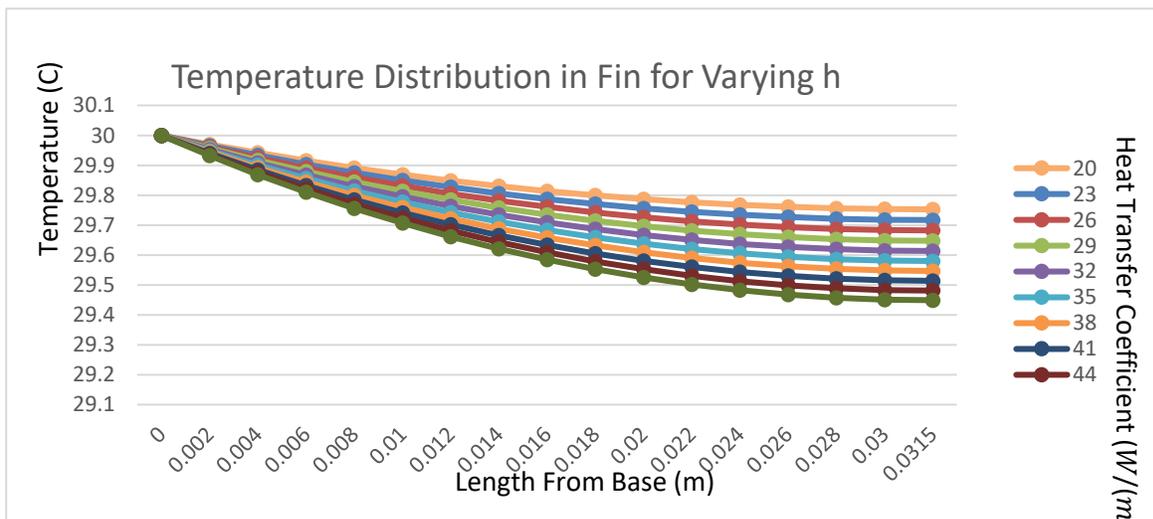


FIG. 16: Heat transfer coefficient effect on the temperature distribution in a fin

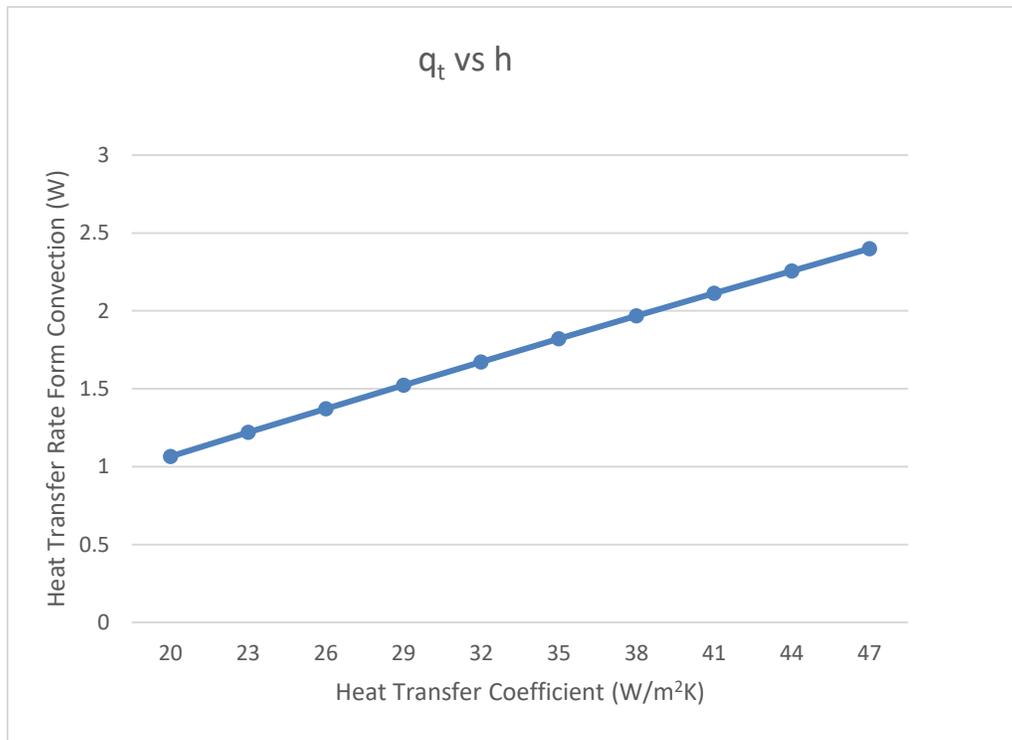


FIG. 17: Heat transfer coefficient effect on the heat transfer rate

4.3 IDEAL HEAT SINK DESIGN

After testing each of the controllable variables independently as well as taking the uncontrollable design constraints into consideration, there is a solid template for design of the ideal heat sink. It was determined that any velocity of air flowing through the channel faster than 2 m/s would be too loud and not appropriate for the application. It was also known that the heat sink would have to fit in the desired diameter of the body of the cooler which is 35 cm. Therefore the fins could be at a maximum of around 15 cm as it is a circular array of fins surrounding a fan. It was determined that with this design, the optimal design would have approximately 60 fins in a circular array with another 60 fins slightly offset but between the original 60 fins for a total of 120 fins.

Along with these design constraints the principles found in the testing of the variables were used to find an optimal design. Fig. 10 showed that the longer the fin the better the rate of heat transfer

due to convection. Fig. 13 showed that a larger fin height yields a higher rate of heat transfer due to convection. Fig. 15 showed that while fin thickness did not play as large of a role in affecting the rate of heat transfer due to convection, the thicker the fin the more the rate was affected. Lastly, Fig. 17 showed that as heat transfer coefficient increased, so too did the rate of heat transfer due to convection.

Given the constraint that the air cannot flow faster than 2 m/s through the channel, it was important to find out the heat transfer coefficient of the air flowing through the channels at that speed. Given the calculations for h presented above, it can be seen that given a determined air speed, the only independent variable is fin length. Considering fin length is in the numerator, as expected, heat transfer coefficient decreases as fin length is increased as shown in Fig. 18

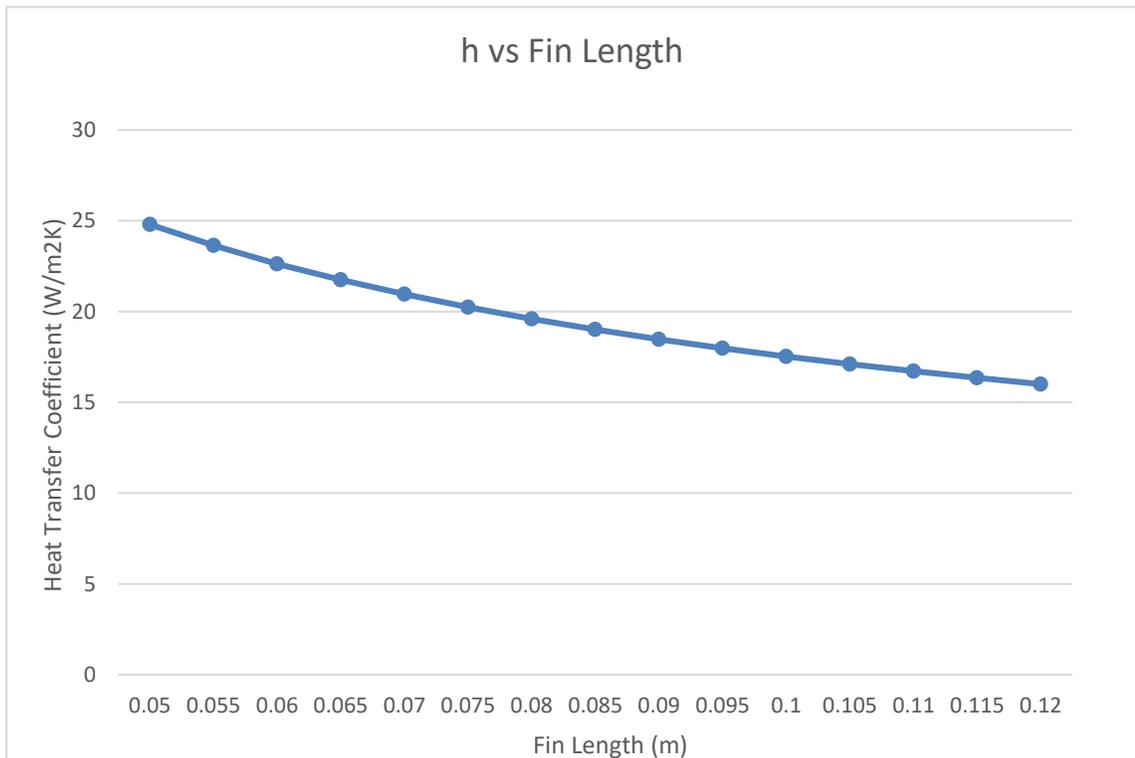


FIG. 18: Heat transfer coefficient for various fin lengths when assuming a constant air speed

As seen above, it is clear that as fin length increases, heat transfer coefficient decreases. With this in mind, one might think that the ideal design will then have a very short fin length as to increase heat transfer coefficient. However, it is also important to remember that as fin length increases, the rate of heat transfer due to convection increases as well.

With this in mind and through trial and error testing it was found that a fin length of 0.11 m, yielding a heat transfer coefficient of $16.7 \frac{W}{m^2K}$ is ideal. Knowing these two dimensions that had the largest effect on the rate of heat transfer due to convection, utilizing Eqs. (6)-(14) and (29) it was possible to determine the ideal fin height, fin thickness, and number of fins. Utilizing this information, the ideal dimensions and design is shown in Table 1.

Table 1. Ideal heat sink design values

Variable	Value
Fin Length	0.057 m
Fin Height	0.11 m
Fin Thickness	0.0014 m
Heat Transfer Coefficient	$16.7 \frac{W}{m^2K}$
# of fins	118.95

4.4 THERMAL MODEL RESULTS

As found in section 2.3 COOLER THERMAL MODEL, the energy equation, Eq. (21) can be solved to determine the amount of time required for the cooler to reach the desired temperature. To complete the model, the values of Eq. (21) must be solved separately before solving the equation. As mentioned above, for this model the value of the amount of heat removed by the TECs will be assumed. Therefore, it is

assumed that the TECs remove heat at a rate of 120 W or 120 J/s. This assumption provides a value of 120 W for $Q_{cooling}$.

Next, Eq. (22)-(25) will be solved to determine Q_H . First Eq. (25) will be solved to determine the radiative heat transfer coefficient, h_{RAD} . For this model, the emissivity of the stainless steel walls is assumed to be 0.1. Using the given values of the Stefan-Boltzmann constant, T_H , T_C , and the emissivity, the radiative heat transfer is found as

$$h_{RAD} = \frac{\sigma(T_H^4 - T_C^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1} = \frac{5.67 \times 10^{-8} (298.15^4 - 285.15^4)}{\frac{1}{0.1} + \frac{1}{0.1} - 1} = 3.851 \frac{W}{m^2K}$$

With the value of h_{RAD} , Eq. 24 can be solved using the surface area of the outer layer of the steel body and the radiation thermal resistance can be found as

$$R_{RAD} = \frac{1}{h_{RAD}A_o} = \frac{1}{(3.851)(0.3)} = 0.865 \frac{K}{W}$$

Next, solutions must be found for the heat lost in conduction for the inner and outer layers of the cooler walls. The conductive thermal resistances for the outer and inner walls can be found by solving Eq. (23), i.e.,

$$R_{COND,i} = \frac{\ln \frac{r_2}{r_1}}{2\pi Lk} = \frac{\ln \frac{0.1106}{0.11}}{2\pi(0.32)(16.2)} = 0.000167 \frac{K}{W}$$

$$R_{COND,o} = \frac{\ln \frac{r_2}{r_1}}{2\pi Lk} = \frac{\ln \frac{0.0125}{0.0119}}{2\pi(0.32)(16.2)} = 0.0015 \frac{K}{W}$$

With all three thermal resistances, it is possible to solve Eq. 22 for Q_H , i.e.,

$$Q_H = \frac{T_H - T_C}{R_{COND,i} + R_{COND,o} + R_{RAD}} = \frac{25 - 12}{0.000167 + 0.0015 + 0.865} = 14.99 W$$

Next is to determine the heat absorbed by the phase change material during its phase change.

Q_{LH} can be found by solving Eq. (26), i.e.,

$$Q_{LH} = mL_f = (1.307)(200000) = 261440 J$$

The value of the mass was found by using the geometry of the phase change material layer of the cooler to determine the volume and multiplying it by the known value of the density of the phase change material. The latent heat of fusion is a known value for the phase change material and is the amount of heat consumed by the phase change material to complete a solid-liquid phase change.

The last values that need to be determined are the sensible heats caused by the steel body and the contents of the cooler. For this the masses of both the steel body and the water beverages that are inside the cooler must be found as well as determining the specific heat of both materials. The solutions for both $Q_{s,l}$ and $Q_{s,m}$ can be found below

$$Q_{s,l} = mC_p(T_H - T_C) = (4.96)(4180)(25 - 12) = 270171.2 J$$

$$Q_{s,m} = mC_p(T_H - T_C) = (2.096)(500)(25 - 12) = 13624 J$$

With all values determined, Eq. (21) can be solved for the time difference by

$$120 W = 14.99 W + (261440 J + 270171.2 J + 13624 J)/\Delta\tau$$

It is important to note that $Q_{cooling}$ is the heat that is being removed by the thermoelectric coolers and that Q_H is the heat that is being added due to conduction and radiation from outside the cooler. It is shown that because $Q_{cooling}$ is so much larger, the cooler will have a net of 105.01 W being removed by the thermoelectric coolers. This shows that it would take approximately $\Delta\tau = 5192.2$ seconds to remove that amount of heat from the cooler. With this amount of cooling, this model could reach cool the temperature inside the cooler from 25°C to 12°C in 5192.2 seconds or 86.5 minutes.

4.6 THERMOELECTRIC COOLER POWER DRAW RESULTS

Using Eq. (28) and (29) with a known value of 0.5 for the coefficient of performance for thermoelectric coolers, and a known value of Q_{out} , the value of W can be found as

$$Q_{in} = COP * W = 0.5W$$

$$Q_{out} = Q_{in} + W = 0.5W + W = 1.5W$$

$$W = \frac{Q_{out}}{1.5} = \frac{120 W}{1.5} = 80 W$$

Next it is possible to determine the current drawn by the thermoelectric coolers by using the known 12 V power, which can be found as

$$I = \frac{W}{V} = \frac{80 W}{12 V} = 6.667 A$$

5. CONCLUSION

A thermoelectric cooler (TEC) is a type of insulated cooler that uses the electricity between the junction of two materials to cause a temperature drop effectively using electrical energy to pump heat from one area to another. In this report the cooling effects of the thermoelectric cooler is used to create a vacuum insulated cooler that can cool beverages to below ambient temperatures. Given the known performance of the thermoelectric coolers, it was paramount to design a heat sink that would be able to dissipate all of the heat being removed from the thermoelectric coolers. A circular array of 120 fins oriented around a centrally located fan was developed to dissipate the heat removed by the thermoelectric coolers. It was found that the performance of the heat sink in dissipating the heat could be increased by utilizing a few key design aspects. It was found that lengthening the fins, increasing the height of the fins, increasing the thickness of the fins, and increasing the speed of the air flowing through the fins would improve the effectiveness of the heat sink in dissipating the heat. A layer of

phase change material located in the base of the inside of the cooler separates this design from competing designs by allowing stored energy to release back into the beverages even after electrical power is removed from the cooler. The thermal modeling developed herein demonstrated that utilizing the cooling power of the thermoelectric coolers is able to cool 14 cans of soda as well as the layer of phase change material from 25°C to 12°C in 86.5 minutes.

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