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Design and Demonstration of a Heat Exchanger for a Compact Natural Gas Compressor

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Design and Demonstration of a Heat Exchanger for a Compact Natural Gas Compressor

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Abstract

Natural gas is an important fossil fuel. In order to optimize the transportation of natural gas from the source to the consumers, it must first be compressed. As natural gas is compressed, the temperature increases significantly and must be cooled before it can be processed further. However, current heat exchangers for this application are large and inefficient for a natural gas compression skid. As these devices must operate in locations where there is a limited amount of space, such as offshore oil platforms, it is crucial to reduce the footprint as much as possible. The goal of this project was to design a more efficient and compact heat exchanger to be used in conjunction with OsComp's rotary, positive-displacement natural gas compressor. The design incorporated heat pipes as a means to improve the efficiency of the overall heat transfer. The design has a total footprint of 0.871 m^2, which is 21.5% smaller than an industry standard heat exchanger of the same specification (1.1 m^2). To demonstrate the feasibility of a heat-pipe forced convection cooler; a scaled-down test stand was manufactured and tested.

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1. Introduction

Natural gas is found in oil or gas wells and consists primarily of methane (85% to 95% by volume) in addition to trace amounts of other gases. Natural gas is used in many applications such as power generation and running industrial equipment. Compression of this gas is necessary to maximize the amount that can be stored and transported. Traditional natural gas compression systems require multiple compression and cooling stages to achieve high pressure ratios and reduce the temperature increases caused by compression, respectively. Since the gas is at different pressures after each stage, multiple pieces of equipment are required which greatly increases the amount of space the equipment occupies. If the natural gas could be compressed in fewer cycles, the size of the compression platform could be greatly reduced. This is particularly important on oil rigs and other locations where space is limited.

OsComp Systems, founded in 2009 by Pedro Santos, has created a new compressor that uses a single stage to achieve pipeline pressures at higher efficiency and energy density, leading to a smaller size. Their compressor uses dynamic liquid injection to achieve near-isothermal compression. Since the gas is cooled within the compressor, the entire compression can be done in one cycle, and this allows the compressor to be packaged on a smaller frame.

Currently, the largest component on the compressor platform is the cooler. The compressor skid holds the compressor and all other components required to keep it running. The cooler is used to cool the engine fluids, from the engine running the compressor, the natural gas coolant, and the natural gas itself. Current industry standard coolers are approximately 7x3x4 meters [84 m^3 in volume] and have a flow rate of 378 SCFM. If the cooler could be redesigned as a more efficient component, the size could be reduced, allowing for a smaller compression skid. This is important because transporting the heavy, bulky skids is expensive and difficult. This becomes even more important in off-shore and under-sea applications where optimizing footprint is of the utmost importance. The goal of this project is to design a cooler that is more efficient and compact than industry standard to be used in conjunction with OsComp's rotary, positive-displacement natural gas compressor. The objective of our project is to design a working cooler section and to build a test model. This test model will be a scale version of the full unit.

2. Background

2.1 Thermodynamic and Heat Transfer Concepts

The two main modes of heat transfer associated with this cooler are convection and conduction. Convection refers to heat transfer between a surface and a moving fluid (Atlas Copco, 2010). Here, a fluid can be considered either a gas or a liquid. This type of heat transfer is characterized by a convection coefficient h. Conduction is heat transfer through the direct contact of particles, whether it is a solid or a stationary fluid (Atlas Copco, 2010). It is characterized by a thermal conductivity k. Each type of heat transfer provides a thermal resistance, which is helpful in calculating the amount of heat that is transferred in a system.

In a heat exchanger, heat is first transferred from the hot fluid to the tube or pipe wall by convection. Next, heat is transferred through the wall by conduction. Finally, through convection, heat is transferred from the wall to the cold fluid (Cengel et al., 2008). Using the subscript i to identify variables inside the tube and the subscript o to identify variables outside the tube, the thermal resistance for all three modes of heat transfer taking place can be defined, along with an equivalent thermal circuit shown in Figure 1. The equations for each thermal resistance are described in Equations (1) through (3). Equation (4) is the equivalent resistance for the entire tube.

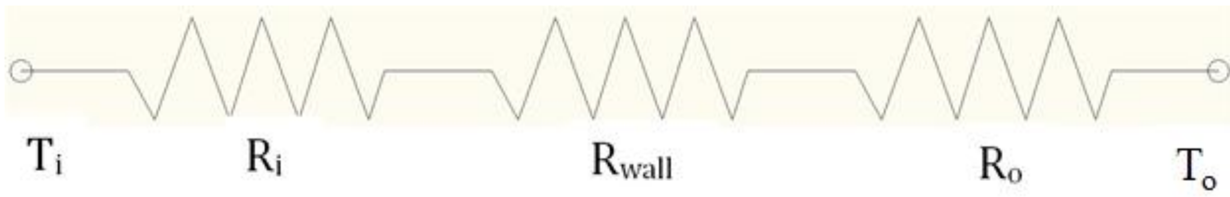

Figure 1: Thermal resistance for a tube

$$R_i = \frac{1}{h_i \times A_i} \tag{1}$$

$$R_{wall} = \frac{\ln\frac{D_o}{D_i}}{2 \times \pi \times k \times L} \tag{2}$$

$$R_o = \frac{1}{h_o \times A_o} \tag{3}$$

$$R_{total} = R_i + R_{wall} + R_o = \frac{1}{h_i \times A_i} + \frac{\ln\frac{D_o}{D_i}}{2 \times \pi \times k \times L} + \frac{1}{h_o \times A_o} \tag{4}$$

The thermal resistance for convection is dependent on the convection coefficient h and the surface area A, on which the heat transfer is to occur. For the inner surface area of circular tubes, $A_i = \pi \times D_i \times L$, where D_i is the inner diameter of the tube and L is the length of the tube. Similarly, for the outer surface area of a tube, $A_o = \pi \times D_0 \times L$. The thermal resistance for conduction is a function of the thermal conductivity k, the length of the tube L, and the ratio between the outer and inner diameters, $\frac{D_o}{D_i}$. Since the heat transfer modes act in series, the total thermal resistance is equal to the sum of these three resistances. From this resistance, the rate of heat being transferred in this process can be determined, using equation (5).

$$Q = \frac{\Delta T}{R_{total}} = U \times A \times \Delta T \tag{5}$$

In Equation (5), ΔT is the temperature difference between the outlet and inlet, and U is the overall heat transfer coefficient. To simplify calculation of the overall heat transfer coefficient, the thermal resistance in the wall may be approximated as zero, since the thickness of the wall is very small. The inner and outer surface areas of the pipe are also considered to be equal. As a result, the overall heat transfer coefficient can be calculated from the two convection coefficients:

$$\frac{1}{U} = \frac{1}{h_i} + \frac{1}{h_o} \tag{6}$$

The values of the overall heat transfer coefficient will range from approximately 10 $W/m^2 * ℃$ for gas-to-gas heat exchangers to approximately 10,000 $W/m^2 * ℃$ for heat exchangers that involve phase changes (Cengel et al., 2008).

Because heat exchangers are made to operate for long periods of time without any change in their operating conditions, they can be analyzed as steady-flow devices, where the mass flow rate in each fluid will remain constant (Cengel et al., 2008). Fluid properties including temperature and velocity at inlets or outlets will also remain constant. Fluid streams experience little to no change in velocities or elevations, meaning that both kinetic and potential energy can be considered negligible.

The first law of thermodynamics states that energy cannot be created nor destroyed; therefore energy must be conserved in a closed system (Atlas Copco, 2010). For a heat exchanger, this means that the rate of heat transfer from the hot fluid must be equal to the rate of heat transfer to the cold fluid. Using c as a subscript to describe the cold fluid and the subscript h as an indicator for the hot fluid, the following equations can be written for the rate of heat transfer for the system. Using the mass flow rates, $\dot{m}$, the specific heats, c_p, and the inlet and outlet temperatures, T_{in} and T_{out}:

$$\dot{Q} = \dot{m} \times c_{p,c} \times \left(T_{c,out} - T_{c,in}\right) \tag{7}$$

$$\dot{Q} = \dot{m} \times c_{p,h} \times \left(T_{h,in} - T_{h,out}\right) \tag{8}$$

Equations (7) and (8) can be rewritten and simplified in terms of the heat capacity rate C. The heat capacity rate is the product of the mass flow rate and the specific heat of the fluid. It represents the rate of heat transfer required to change the temperature of a fluid stream by 1°C. In a heat exchanger, fluids with a large heat capacity rate will experience a smaller temperature change (Cengel et al., 2008). Conversely, there will be a larger temperature change in fluids with a small heat capacity.

2. 2 Compressors

Various techniques for compression are used in products on the market today. Cooling liquid-injected compression, which is utilized by OsComp Systems, is the process of injecting liquid into a gas during compression. The coolant absorbs heat generated during the compression process, sometimes evaporating as a result. This cools the gas during compression, allowing for high pressure ratios in a single stage and eliminating the need for inter-stage cooling sections in the system. This is different from industry standard natural gas compressors or reciprocating piston compressors, as they will fail catastrophically when liquids are included in the gas stream. OsComp's compressor utilizes a novel design which uniquely allows for liquid injection and wet gas compression.

2.3 Heat Exchangers

A heat exchanger is a device that is used to transfer thermal energy between two or more fluids. Heat exchangers are classified according to the heat transfer process, number of fluids, construction, surface compactness, flow arrangements, and heat transfer mechanisms. Two fluids are commonly used in heat exchangers, with one fluid being cooled and another fluid acting as a coolant. However, "as many as twelve fluid streams have been used in some chemical processes (e.g., air separation systems and purification/liquefaction of hydrogen) (Shah 2003)."

2.3.1 Heat Transfer Process

Heat exchangers can be classified as devices that transfer heat either directly or indirectly. A direct contact heat exchanger transfers energy directly from one fluid to another, usually separated by a barrier. Indirect contact heat exchangers utilize an intermediate medium to transfer the thermal energy between fluids. There are also different classifications within each of these types of exchangers, as illustrated below in Figure 2.

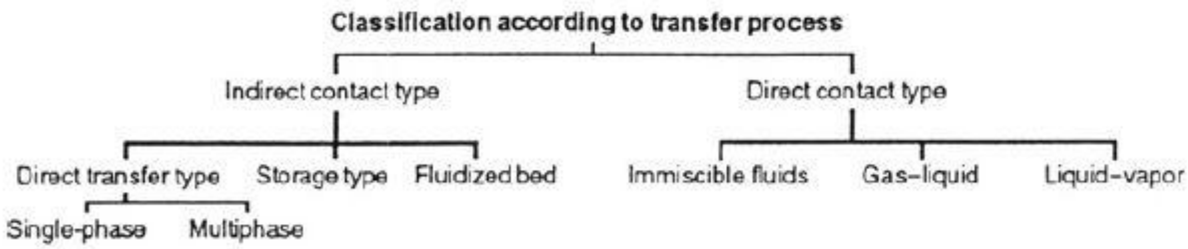

Figure 2: Heat exchanger classification according to transfer process

There are three arrangements within the indirect-contact heat exchanger: direct transfer, storage, and fluidized bed. In the direct transfer setup, heat is transferred continuously from the hot fluid to a dividing wall and from the dividing wall to the cold fluid. Although two (or more) fluids flow simultaneously, the fluids never mix because each fluid is flowing in a separate passage. Examples of direct-transfer heat exchangers are tubular, plate-type, and extended surface (fin) exchangers. “In the storage type exchanger, both fluids flow alternatively through the same flow passages, creating intermittent heat transfer. In a fluidized-bed heat exchanger, one side of a two-fluid exchanger is immersed in a bed of finely divided solid material, such as a tube bundle immersed in a bed of sand or coal particles.” (Shah 2003)

In direct-contact heat exchangers, the fluids come into direct contact when transferring heat. The fluid types associated with these heat exchangers can be two immiscible fluids, a gas and a liquid, or a liquid and a vapor. Typical applications involved heat and mass transfer, such as in evaporative cooling. It is rare that applications only involve sensible heat transfer. With respect to indirect- contact heat exchangers, direct-contact can achieve very high heat transfer rates, are relatively inexpensive, and do not suffer from fouling since there is no wall between the two fluids (Shah 2003).

2.3.2 Construction

There are four main construction types of heat exchangers: tubular, plate-type, extended surface (fin), and regenerative. There are other constructions available that may be classified as one of these types but have some unique features compared to the conventional type of exchanger.

Tubular heat exchangers are generally built of circular tubes. However, elliptical, rectangular, and round/flat twisted tubes can also be used. There is considerable flexibility in this design as the core geometry can be varied easily. This is done by changing the tube diameter, length, and arrangement. These exchangers can be designed for high pressures relative to the environment as well as high pressure differences between fluids. Primary heat transfer applications include liquid-to-liquid and liquid-to-phase change (condensing or evaporating).Tubular heat exchangers can be used for gas-to-liquid and gas-to-gas heat transfer applications when the operating temperature and/or pressure is very high, or if fouling is a severe problem on at least one fluid side. (Shah 2003)

The plate heat exchanger, shown in Figure 3 below, is often used for two liquid streams. This type of exchanger consists of many individual plates that are stacked together, refer to Figure 3. The plates are corrugated, forming flow channels between the adjacent plates. This type of heat exchanger is compact and easy to disassemble for cleaning. It is also relatively easy to increase or decrease their size as needed by adding or removing plates.

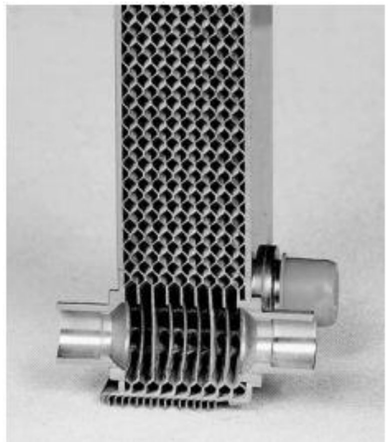

Figure 3: Plate heat exchanger

Fins are used to increase the heat exchanger surface area. They are usually "placed on the gas side of a gas-to-liquid configuration, compensating for the low convection heat transfer coefficients that are typical for forced convection with a gas (Nellis 2009)." The fin configuration can vary depending on the application. The most common applications are for automobile radiators and heat sinks for computers.

A shell-and-tube heat exchanger has one fluid flowing through a bank of tubes, which is inside a larger shell. The cooling fluid flows within the shell and around the outer surface of the tubes. Baffles, interior plates used to deflect or regulate flow, are usually added in the shell to force the shell-side flow to pass across the tubes, in a serpentine pattern (Figure 4).

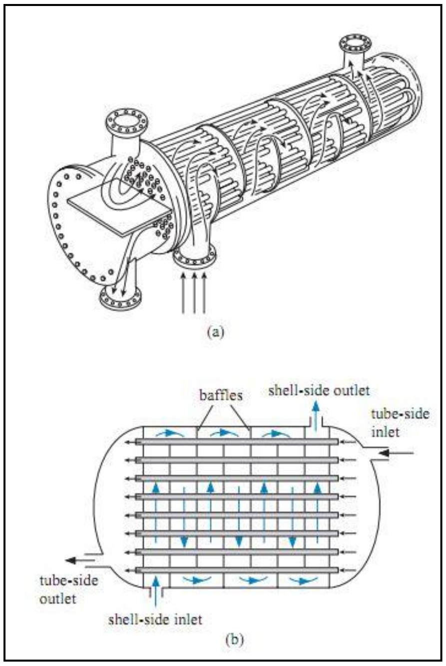

Figure 4: Shell-and-tube heat exchanger

2.3.3 Heat Transfer Mechanisms

The basic heat transfer mechanisms incorporated in heat exchangers are single-phase convection, two-phase convection, and combined convection and radiation heat transfer. The convection can either be free or forced in any case. Any of these mechanisms can be active on each fluid side of the heat exchanger, either individually or in combination, depending on the configuration (Shah, 2003).

2.4 Heat Pipes

Heat pipes are used to transport large quantities of heat from one location to another without the use of electricity. Utilizing a closed, two-phase, fluid cycle with a hot interface (evaporator) and a cold interface (condenser), heat pipes utilize the properties of gravity or capillary action, provided by a wick, and pressure differential to transport liquid and vapors. Heat pipes are extensively used for cooling systems in spacecraft and for cooling various modern computer system components, including central and graphical processing units.

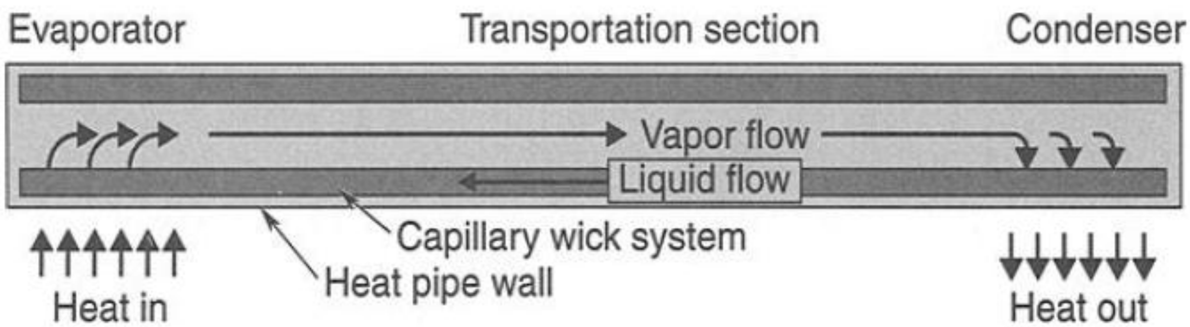

Figure 5: Heat Pipe Operation (Gilmore, 2002)

As illustrated above in Figure 5, a heat pipe consists of a wick and a vapor space. The wick contains the liquid phase and the vapor space contains the vapor phase, where both are at saturation. At the evaporator, the incoming heat raises the temperature inside and the liquid evaporates, travelling to the other end. This lowers the pressure in the evaporator end because less liquid remains. The local vapor pressure at the condenser end raises, as it must remain in saturation with the heated liquid. This pressure difference is sufficient enough to draw liquid from the condenser toward the evaporator, while the heated vapors in the evaporator flow toward the condenser, which is now at a lower pressure. The vapors then condense when they come in contact with the condenser's lower surface temperature. This cycle is repeated to transfer heat (Gilmore, 2002).

2.4.1 Types of Heat Pipes

There are various types of heat pipes available, ranging from constant-conductance heat pipes and diode heat pipes to variable-conductance heat pipes (VCHPs) and hybrid systems. Constant-conductance heat pipe designs are among the most basic and are categorized according to their 4 types of wick structure: groove, which has many small slots around the inside of the pipe to carry liquid back; mono-groove, which uses one larger groove instead of many smaller grooves; composite, which layers material around the inside of the pipe to carry the liquid back; and artery & tunnel, which provides one or more extra unrestricted liquid flow paths in addition to composites.

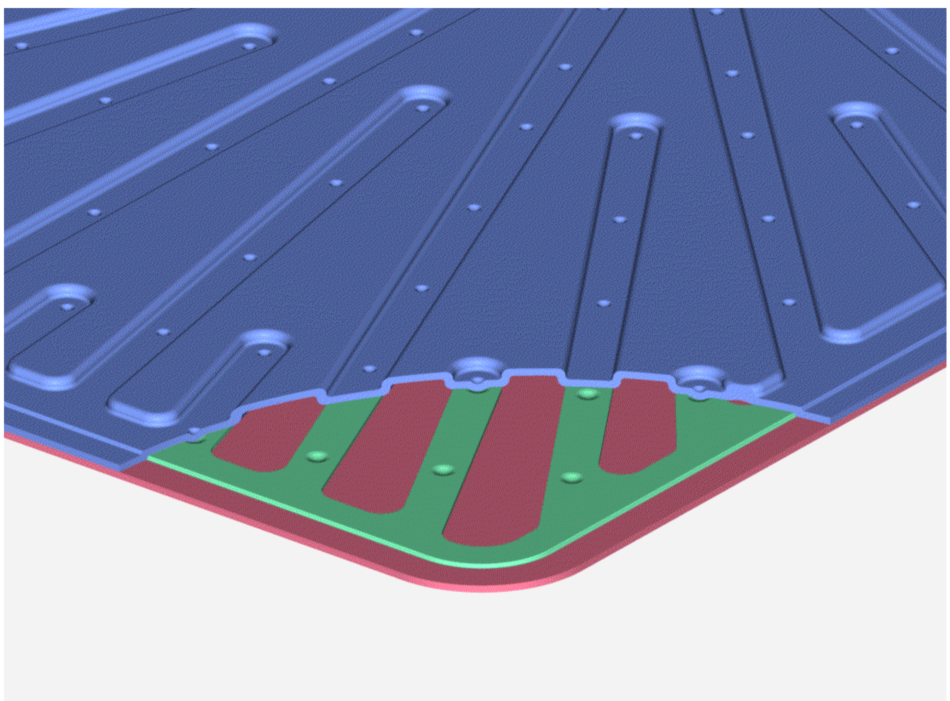

Figure 6: Flat Heat Pipes

Heat pipes do not need to have a conventional cylindrical shape; they can also be designed as flat plates as seen above in Figure 6 (Gilmore, 2002). Diode heat pipes are similar to constant-conductance heat pipes, except they only operate in one direction. The three most common diode heat pipes are: liquid trap, which uses a reservoir next to the evaporator to block vapor travel; liquid blockage, which uses a reservoir by the condenser which fills and empties with liquid to block vapor travel; and gas blockage which also uses a reservoir next to the evaporator, but is filled with a non-condensable gas to block vapor travel (Gilmore, 2002). Variable-conductance heat pipes, as shown in Figure 7 below, utilize gas reservoirs.

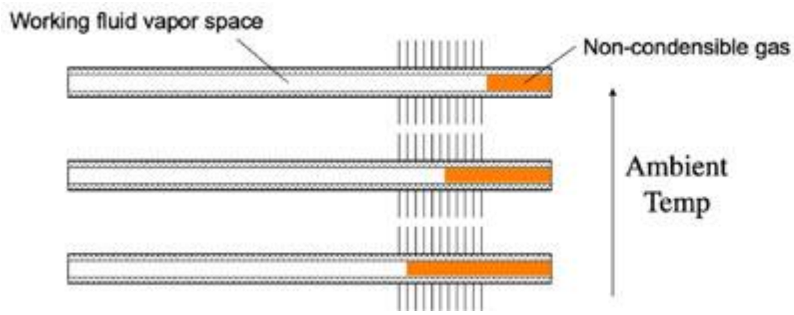

Figure 7: Variable-Conductance Heat Pipes

These reservoirs are filled with a non-condensable gas to control the operating area of the condenser based on the evaporator temperature. The benefit of this type of heat pipe is to reduce the volume of control gas and open up more area of the condenser for heat pipe operation (Gilmore, 2002). Hybrid systems are extensions of capillary-pumped loop designs. Coolant is vaporized as part of this, and major benefits of this system are its ability to be operated without a separate driving unit and being suitable for light-weight miniaturized electronics (Gilmore, 2002).

3. Methodology

3.1 Design Requirements

With the goal of the project to design a more efficient and compact heat exchanger to be used in conjunction with OsComp's rotary, positive-displacement natural gas compressor, the next step was to create a list of design specifications and requirements to follow as initial design ideas took shape. After speaking with our sponsor and examining their natural gas compressor, we learned that the temperature of the natural gas entering our heat exchanger is 104º Celsius and the desired exit temperature is 60 ºC. The design must handle the pressure of the compressed natural gas at 10.55 MPa and be able to continuously run for months at a time. The natural gas flows at a rate of 0.473m^3/sec and has a density of 53.33 kg/m^3. Our sponsor is interested in seeing innovative or novel ideas, encouraging us to stay away from more traditional designs such as shell-and-tube and plate heat exchangers. Finally, the design is desired to be smaller than commercial natural gas heat exchangers currently available on the market. For comparison, we used an industry standard after-cooler section quote, given to us by our sponsor, which has a total volume of 0.55 m^3, broken up into a length of 1.619 m, a width of 0.343 m, and a height of 0.991 m.

3.2 Design Process

Each member of the design team came up with a preliminary design for a more compact method of cooling natural gas. These concepts were each analyzed in great detail to help determine which concepts we would continue to explore. First, we used a decision matrix to narrow our design options down to two concepts. Then, we conducted further analysis to compare both final concepts and make a decision on which path to pursue. We started with five unique designs, detailed below, and ended up pursuing a heat pipe based design with copper cradles.

3.2.1 Concepts Considered

Spiral Heat Exchanger

Spiral heat exchangers feature two fluids in counter flow. In this set-up, the hot fluid is cooled by a colder fluid. They are often used in paper processing and refineries, and are beneficial due to their anti-fouling designs and compact design. Some challenges with using spiral heat exchangers are that they can only cool one fluid, and that the coolant flow must be a forced flow. We chose not to use a spiral heat exchanger due to the lack of scalability and the fact that only one fluid can be cooled in each heat exchanger. A diagram of the spiral heat exchanger model is shown in Figure 8.

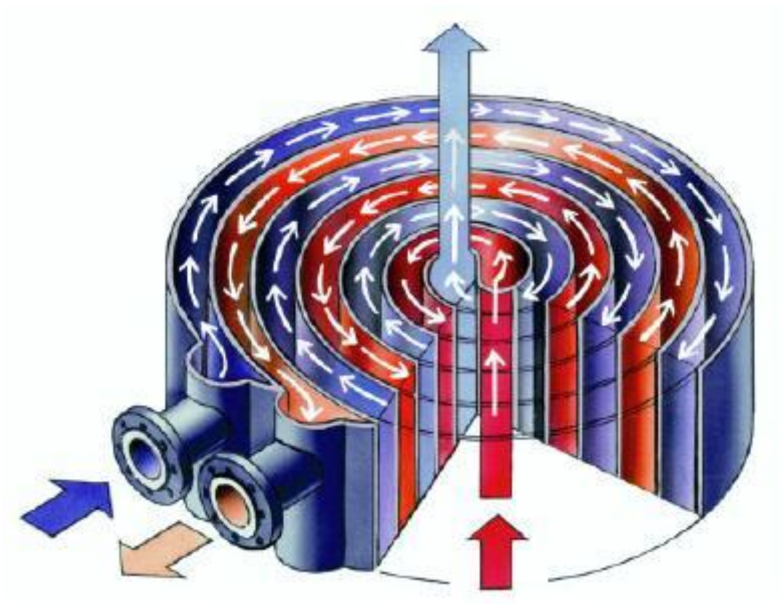

Figure 8: Spiral heat exchanger (Cesco, 2011)

Two Section Heat Exchanger

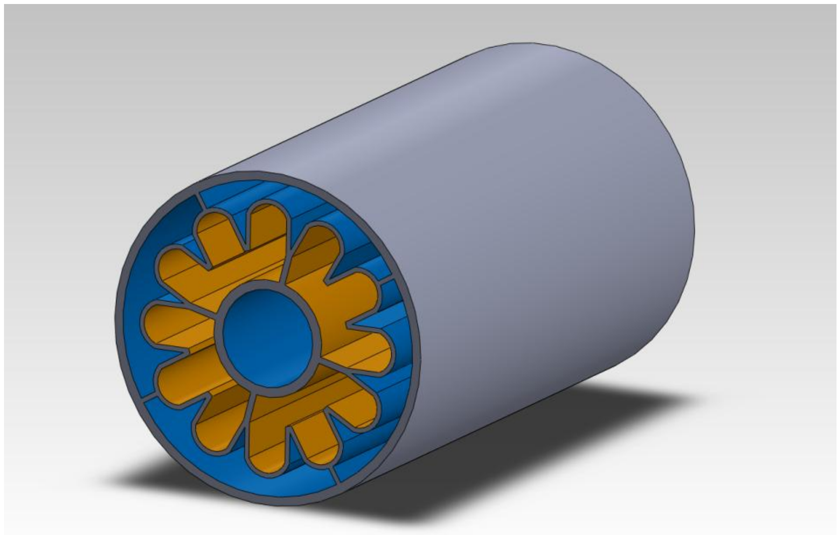

Figure 9: Two section heat exchanger

A two section heat exchanger, shown in Figure 9, was developed as another possible cooler design. Made out of aluminum, this design operates by running the hot gas through the yellow sections and coolant through the blue sections. The pipe bends back and forth to create a stacked, staggered pattern in order to reduce the overall size. Baffles within the blue section of the tube help support the yellow section and direct the flow of coolant. In order to manufacture, wire EDM or aluminum extrusion should be considered. The advantages of this design are that the multiple passes allow for longer duration of contact with coolant and provide a large heat transfer surface area. However, this design was not chosen for further analysis because it became apparent that the length needed to sufficiently cool the natural gas would make the design too large.

Thermoelectric Cooler

An additional method for cooling the natural gas that was considered was the use of thermoelectric coolers. These coolers utilize the thermoelectric effect (also known as the Peltier effect), in which an induced voltage between two different metals creates a temperature difference. Although they contain no moving parts, thermoelectric coolers are characterized by poor efficiency ratings (Avallone, 2007). Ultimately, we did not continue with the thermoelectric concept for our design because of the high amount of electrical energy required to operate it, the relatively high cost for individual coolers, and its poor scalability. Mounting the coolers to a cylindrical pipe would also be challenging, especially with a large heat sink attached to each cooler. Figure 10 and Figure 11 below show how a thermoelectric cooler operates and a sample thermoelectric cooler, respectively.

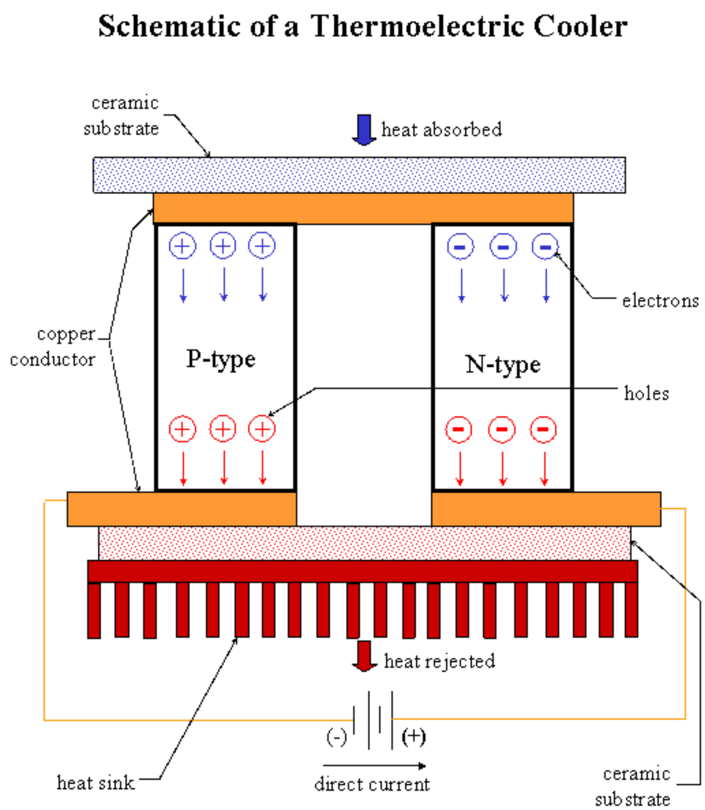

Figure 10: Thermoelectric Cooler Operation (TE Technology Inc., 2010)

Figure 11: Sample Thermoelectric Cooler (Snake Creek Laser, 2011)

Compact Heat Exchanger

When exploring the different types of heat exchanger, there was one particular classification that seemed to fit well for our application. The idea of a compact heat exchanger was intriguing because of the ability to cool a fluid over a large heat transfer surface area while occupying a relatively small amount of space. They are commonly used in applications that have limitations on weight and volume.

The original compact cooler concept, inspired by a thermal-fluid textbook, *Fundamentals of Thermal-Fluid Sciences* by Y.A. Cengel, R. H. Turner, and J.M. Cimbala, consists of multiple tubes passing through a large amount of thin plates. Air is forced through the plates using a fan. The plates increase thermal conductivity, meaning that the greater number of plates used in a design, the greater the amount of heat can be removed from the system. After several iterations, the final compact cooler design that was developed featured one tube that is looped back and forth between each end of the cooler, with each pass going through thin aluminum plates. This configuration results in a tube length long enough so that the natural gas can be cooled to the necessary temperature. At the same time, by looping the tube, size and footprint are both minimized. Figure 12 shows the configuration of the model with the looped tubes.

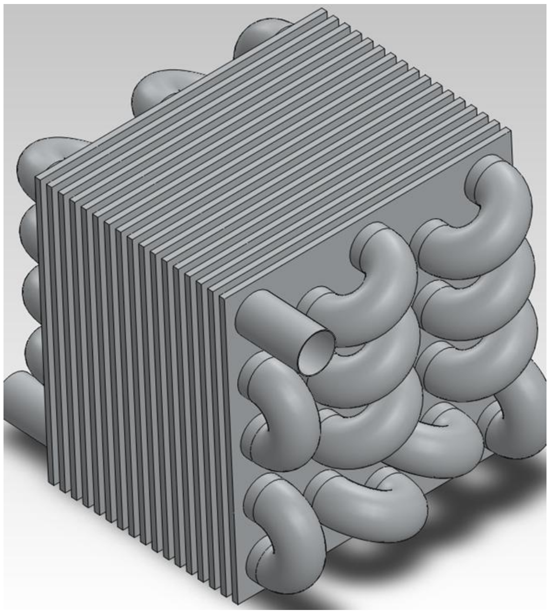

Figure 12: Compact cooler

This concept presented several manufacturing challenges, and after further analysis, it was realized that this design, even with the pipe looped back and forth, would still have to be quite large if it were to cool the natural gas to the desired temperature. Due to these reasons, we ultimately did not further pursue this design concept.

Deployable Radiator

Another concept we developed was inspired by the deployable radiators found on the International Space Station. Our initial background research confirmed that aircraft and spacecraft required compact thermal control devices since they had limited space available.

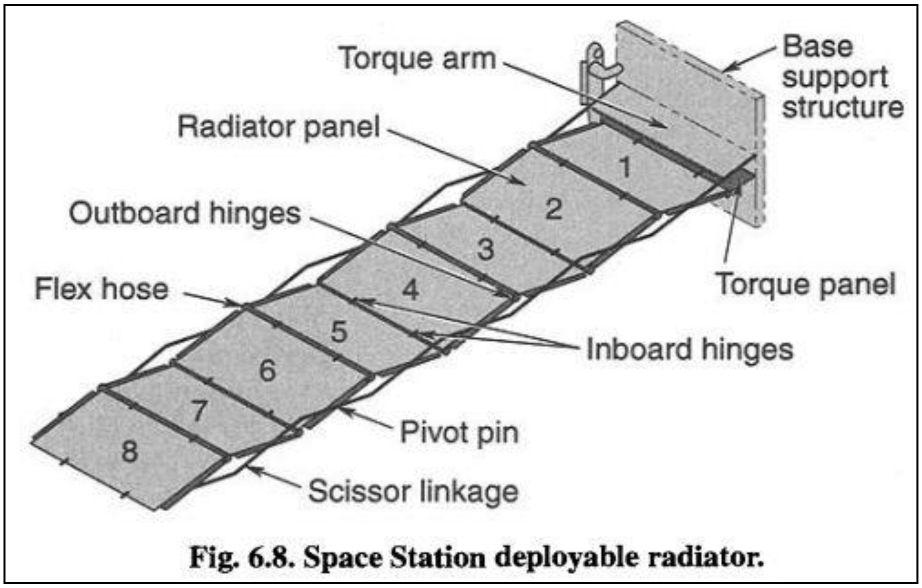

Figure 13: Deployable radiator preliminary concept (Gilmore)

In order to reduce the large footprint of a typical heat exchanger, we modified the deployable radiator concept, shown in Figure 13 to be used on Earth. The base would provide a smaller footprint and have numerous panels extend upwards. Each panel would be equipped with heat pipes to further increase the heat transfer rate. The panels themselves help increase the surface area available to mount the heat pipes as well as dissipate larger amounts of heat. The panels also would have the ability to retract when not in use, which would make transportation and instillation easier. However, we learned from OsComp that this feature was not necessary or very beneficial.

Panels with Heat Pipes

The deployable radiator concept was further modified to improve the structural integrity as well as ensure an easier manufacturing process. The panels would be mounted horizontally and stacked vertically. Each panel would have heat pipes mounted on its surface, with the condenser region extending off the edge of the panel. The compressed natural gas would enter and exit at the base of the cooler and would snake through the panels. Schematics of this design are described below in Figure 14 and Figure 15.

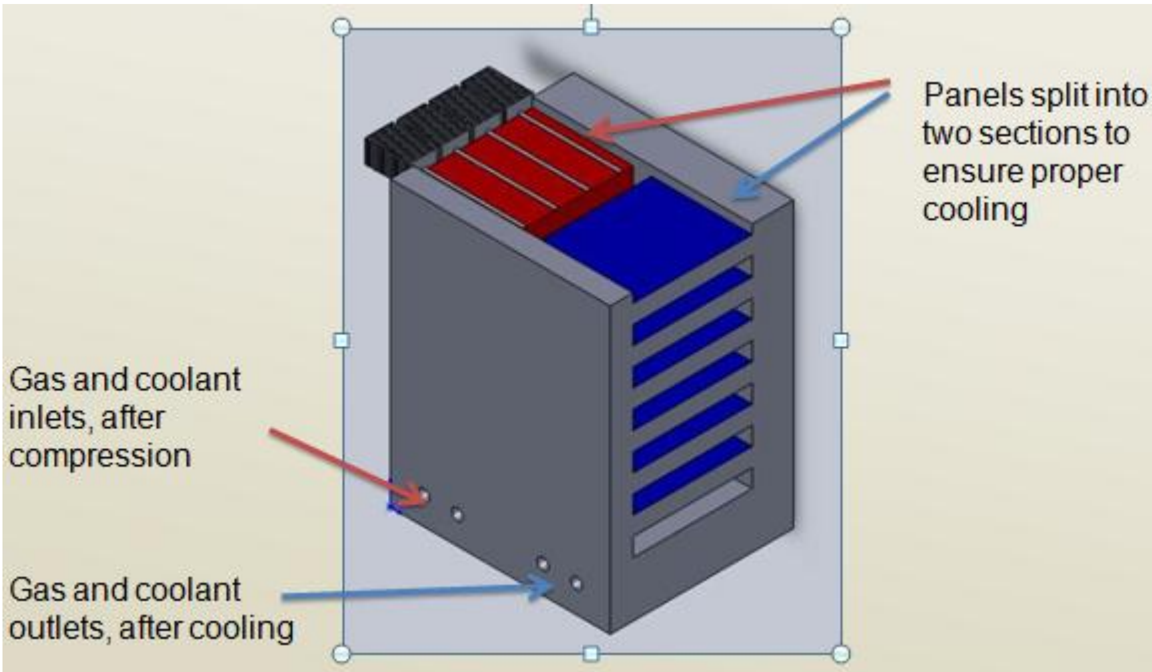

Figure 14: "Panels with heat pipes" preliminary concept

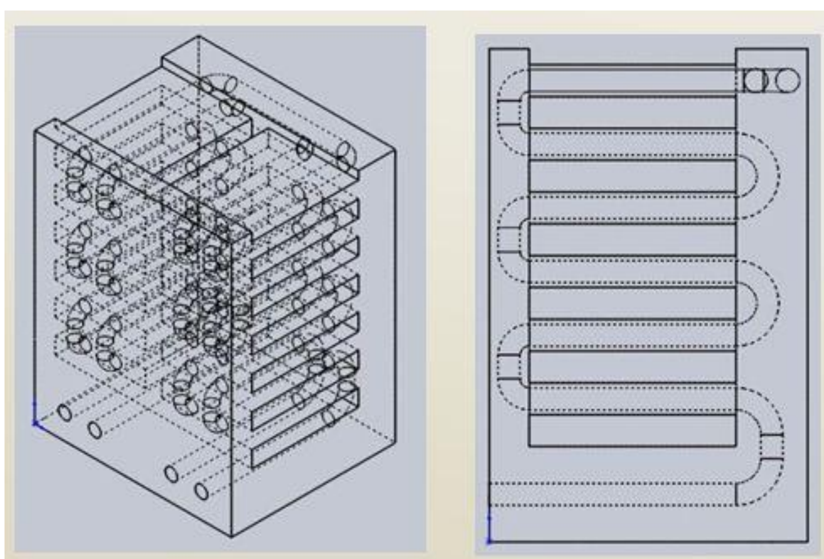

Figure 15: "Panels with heat pipes" preliminary design concept, inside view

Similar to the deployable radiator concept, the panels are meant to increase the surface area and allow the heat pipes to be mounted to them. However, this design had a few potential issues. First, there was concern with the pressure loss in the system from the natural gas pipes snaking within the panels. Second, there would be a significant masking effect due to the panels and heat pipes being stacked directly over each other. This would not allow the cooler to be as effective. Lastly, it would cost a significant amount of money to manufacture due to the intricate piping throughout the heat exchanger.

Thermacore's Heat Exchangers

The team researched current heat exchangers on the market to determine if they could somehow be integrated in our design concepts. Thermacore Thermal Management Solutions' air-to-air heat exchangers presented a few concerns. The primary concern was that the heat exchanger had to be mounted in such a manner where the natural gas flows through the lower half of the heat exchanger. This presented a challenge in terms of manufacturing our design concept. Also, the heat exchangers are not meant for compressed natural gas, which meant that corrosion could occur over time. Lastly, the unit's performance was insufficient for our application. Thermacore's liquid-to-air heat exchanger also could not handle the volumetric flow rate required for the coolant. Refer to Figure 16 for examples of Thermacore's current air-to-air and liquid-to-air heat exchangers.

Figure 16: Thermacore's heat exchangers (Thermacore, 2009)

3.2.2 Decision Matrix

After the deployable radiator concept was discontinued, our five remaining preliminary designs were evaluated using the following decision matrix (Table 1). From this, we were able to eliminate the spiral heat exchanger, two section heat exchanger, and thermoelectric cooler from final design contention.

Table 1: Decision matrix

RATINGS		Compact Heat Exchanger	Thermoelectric Cooler	Spiral Heat Exchanger	Two Section Heat Exchanger	Panels with Heat Pipes
Size		4	3	4	2	4
Footprint		4	4	5	2	5
Power Consumption		4	2	4	3	5
Manufacturability		5	4	4	3	4
Scalability		5	3	1	2	4
Cost		4	2	2	2	3
SCORING	Weight	Compact Heat Exchanger	Thermoelectric Cooler	Spiral Heat Exchanger	Two Section Heat Exchanger	Panels with Heat Pipes
Size	25	100	75	100	50	100
Footprint	25	100	100	125	50	125
Power Consumption	20	80	40	80	60	100
Manufacturability	10	50	40	40	30	40
Scalability	10	50	30	10	20	40
Cost	10	40	20	20	20	30
SUM	**100**	**420**	**305**	**375**	**230**	**435**

Since the compact heat exchanger and the panels with heat pipes had the two highest scores, they were the two designs we selected for further analysis. The conclusion of this analysis was that a heat pipe based approach would be the least expensive, most innovative, and best way to reduce size and increase efficiency for a natural gas cooler model. Figure 41 and Figure 42, in Appendix D - CAD Drawings and Pictures, are drawings of our preliminary design featuring heat pipes in cradles, which is the same concept used in our final design.

3.3 Final Design

Using the heat pipe in cradles concept for our final design, the heat pipes we used are 8mm in diameter and 150 mm long, utilize a sintered-powder wick, and are filled with water as their working fluid. At the top of each heat pipe are 18 aluminum fins, 20 mm long by 25 mm wide and 0.4 mm thick with a space of 2.5 mm between each fin. The stainless steel gas pipes are connected in parallel with each other and the natural gas flows through the pipes simultaneously. Forced air convection is used to cool the heat pipes by means of a fan blowing across the three rows of pipes. Below in Figure 17 is a picture of the full design.

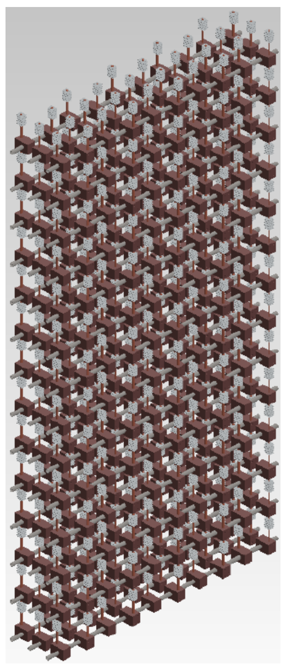

Figure 17: Full Model

Utilizing heat pipes spread out along multiple 5-ft long sections of steel pipe, each heat pipe is inserted into a copper cradle, which is attached around the gas pipe. Thermal grease is applied between the heat pipe & cradle, and the cradle & steel gas pipe to enhance the heat transfer between the separate parts. Figure 18 below is a picture of the cradle with the heat pipe and steel gas pipe inserted.

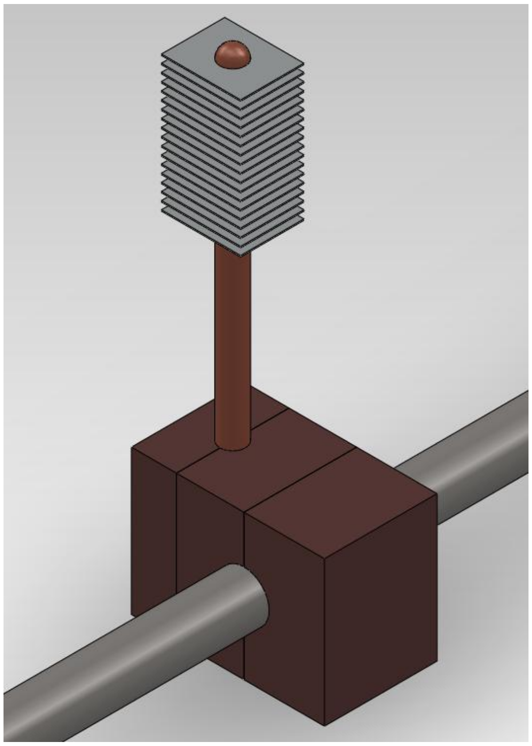

Figure 18: Cradle with Heat Pipe

There are a total of 419 cradles and heat pipes in the full design and forty-two 5-ft section gas pipes; with 10 cradles and heat pipes on each. These gas pipes are arranged in 3 columns of 14 rows to minimize the masking effect caused by the heat pipes blocking each other as the fan blows across. The length of the design is 1.524 m and the width of the design is 0.572 m, resulting in a footprint of 0.871 m^2. The height of the design is 2.393 m, which results in a total volume of 2.084 m^3.

3.4 Manufacturing

To run our experiment, we constructed a scale demonstrator. There was one component, the copper cradles, which we had to manufacture ourselves and another, the heat pipes with fins, which we had to have an outside company manufacture for us.

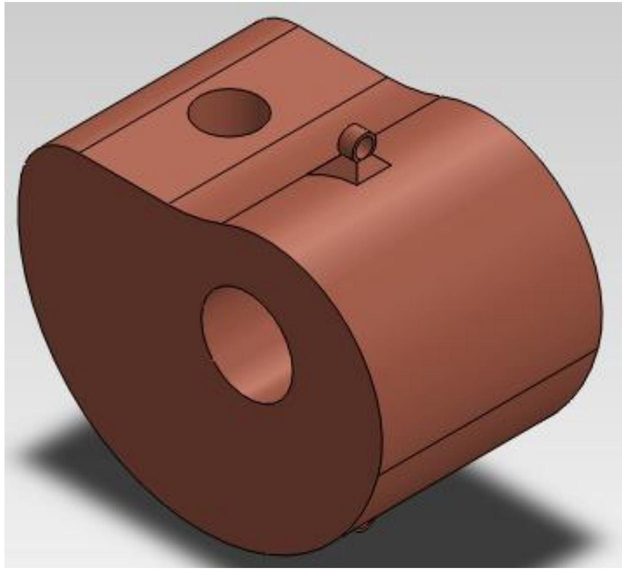

Figure 19: Cradle (ideal)

Our original design, illustrated above in Figure 19, was designed for curved heat pipes and to minimized weight. We discovered that the heat pipes could not be bent the way we wanted and had to change the design to utilize straight heat pipes. The design was also changed to minimize manufacturing time as opposed to weight. These changes resulted in the final design shown below in Figure 20. Our final design is not the ideal size as the stock was purchased before the design changes so it was made using 1.5-in square stock instead of 5/8-in square stock.

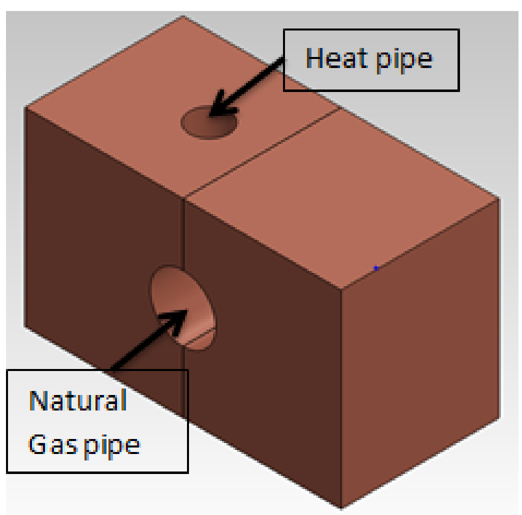

Figure 20: Cradle (modified)

Each cradle is made of two pieces of 1.5-in square stock and mated around the natural gas pipes as shown in Figure 20. All the cradles were machined on a HAAS VF4 CNC machine. There are two different milling methods that we could have tried to make our cradle pieces, drilling and surfacing. We used the drilling method, which uses only drilling operations, to make the cradles for our test unit. The other method is surfacing, which slowly removes material off the surface of the part. Although we did not use this method because it is slower, it could have allowed us to use fewer drill bits and is slightly more reliable.

There are many challenges to milling copper. The major challenges are that copper is soft, has a low melting point, and work-hardens. Work-hardening is when a material gets stronger and more resistant to deformation as it is deformed or worked. This means that any material removal tool will not last as long as it would with a non-work-hardening material. Due to those challenges and the costs of copper, it is recommended to look into various forms of casting. Casting would reduce material waste, and for a full heat exchanger, could greatly reduce costs.

4. Testing

4.1 Heat Pipe Experiments

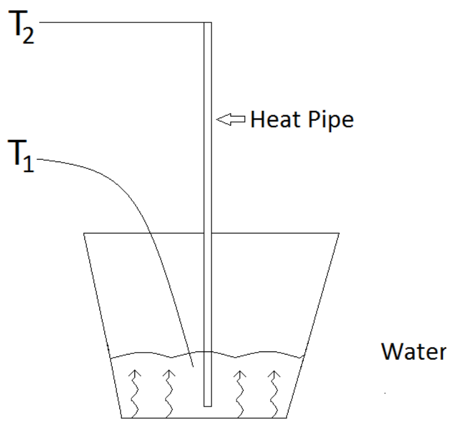

Figure 21: Heat pipe experimental set-up

In order to verify the capabilities of a heat pipe, we set up an experiment. A sketch of the set up for our heat pipe experiments, with the heat pipe in a vertical orientation, is shown above in Figure 21. We filled a beaker with water and placed it on a hot plate. One thermocouple was placed in the water, while the other thermocouple was attached to the end of a heat pipe. The heat pipe was then placed in the water, and thermocouple readings were taken for one minute using a LabView virtual instrument. From these readings, we were able to determine the temperature drop between the two ends of the pipe. By determining the conduction through the water, we determined the thermal resistance of the heat pipe.

Table 2, shown below, compares the thermal resistances of the 10x140 mm heat pipe with various applied heat loads.

Table 2: Heat load vs. thermal resistance of heat pipes

q_avg (W)	R_avg (°C/W)
4.11	2.71
6.88	3.03
8.12	3.06
10.16	3.36
10.33	3.49
11.86	3.61
12.66	3.29
13.70	3.34

According to data, shown in Figure 22, provided by Enerton, a heat pipe manufacturer, a 6-mm diameter heat pipe with an applied heat load of 10 W in a vertical orientation should have a thermal resistance of approximately 3°C/W.

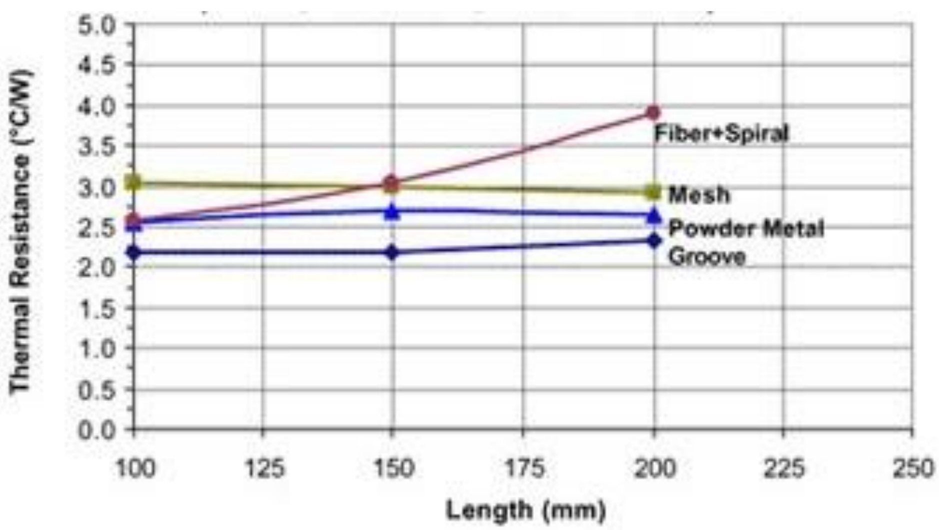

Figure 22: Thermal resistance vs. Heat Pipe Length (Enerton, 2011)
(Q = 10W, Diameter = 6mm, Vertical Orientation)

Figure 23 plots the experimental average resistances found at certain heat loads. These results were compared to the expected resistance of 3 °C/W for a 6-mm pipe with a heat load of 10 W. The results here suggest that as diameter increases, thermal resistance also increases.

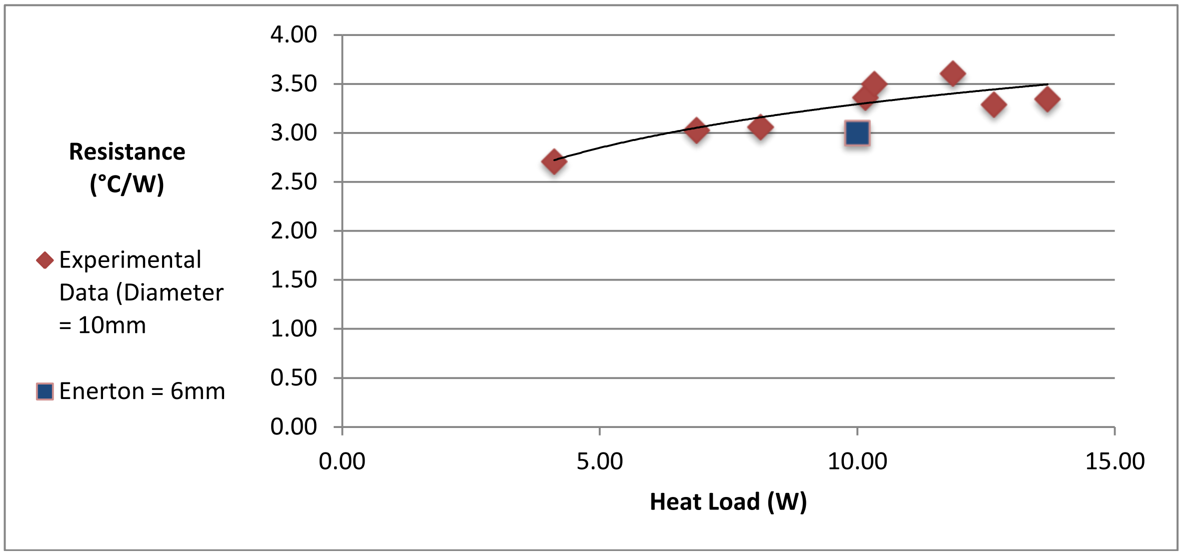

Figure 23: Average heat load vs. thermal resistance of a heat pipe (Diameter = 10mm, Vertical Orientation)

Figure 24 shows the thermal resistances of the 10 x 140-mm pipe with an average applied heat load of 10.2 W over a one minute interval. The downward slope suggests that with time, the thermal resistance of the heat pipe decreased. This trend was consistent throughout each of our experimental trials. After about 50 seconds, the thermal resistance approached a steady state of about 3°C/W.

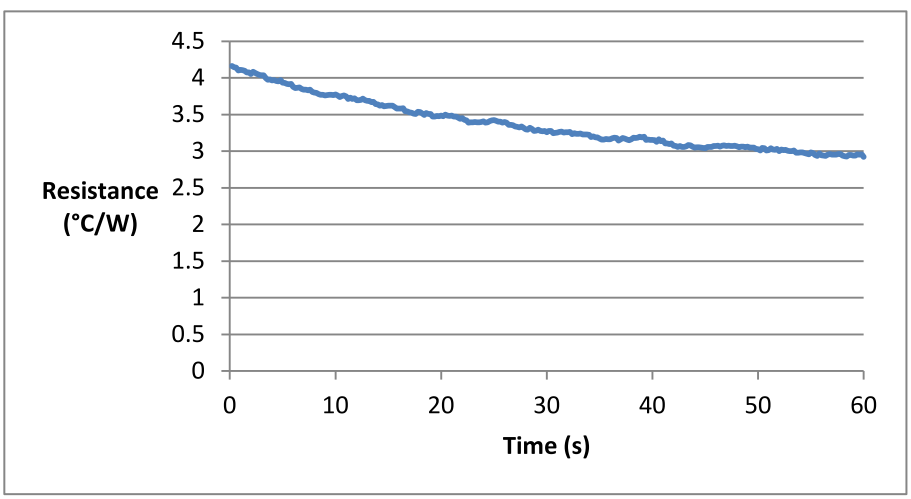

Figure 24: Thermal resistance over time

4.2 Scale Model Experiments

Due to the availability of fluid sources, limited access to manufacturing facilities, and time constraints, a full-scale cooler could not be produced. We constructed a demonstrator to test the cooling capacity for a small section of pipe to compare to the full-scale cooler.

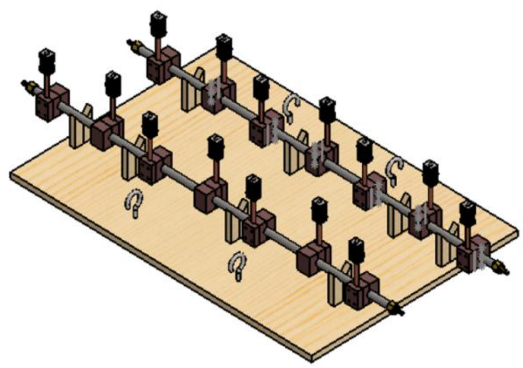

Figure 25: CAD model of demonstrator

4.2.1 Assembly of Demonstrator

The following steps were used to construct the demonstrator:

1. Steel piping with a diameter of 5/8 inches was used to carry the hot fluid.
2. 90° elbow pipe connections were used to attach two piping segments for a parallel flow, and T-style pipe connections were attached between the parallel pipe segments to divide and later combine the inlet and outlet flows for the demonstrator.
3. The length of the demonstrator was 41 ½ in (including the 90° elbow connections on each end).
4. Fourteen copper cradles (shown in Figure 18 above) (two pieces each) were manufactured to attach the heat pipes to these steel pipes.
5. Thermal grease was applied between the steel pipes and these copper cradles as well as between the heat pipes and the cradles to improve heat transfer in these regions.

6. The spacing between cradles along the sections of the cooler ranged from 2 inches to 4 ½ inches, and the two cradle pieces for each were clamped together around the steel pipe to hold them in place.

Diagrams of the demonstrator cradle layouts are shown in Figure 26 and Figure 27 below. The initial tests consisted of 7 heat pipes and cradles in total, while 14 heat pipes and cradles were used for the tests after additional cradles were manufactured. Figure 26 shows the demonstrator arrangement with 7 cradles and heat pipes (all dimensions in inches), while Figure 27 shows the demonstrator arrangement with 14 cradles and heat pipes.

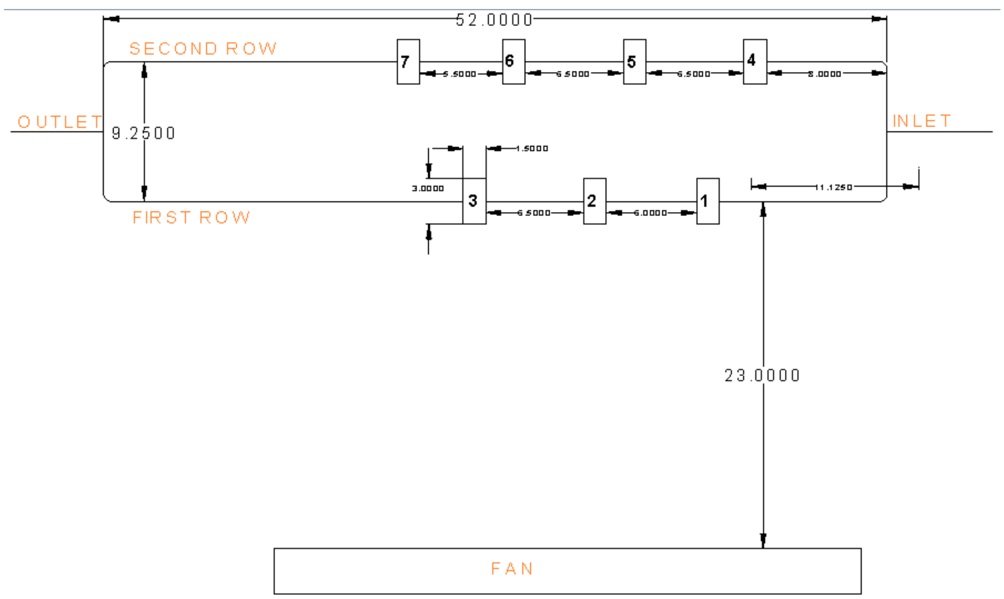

Figure 26: 7 Cradle Demonstrator Layout

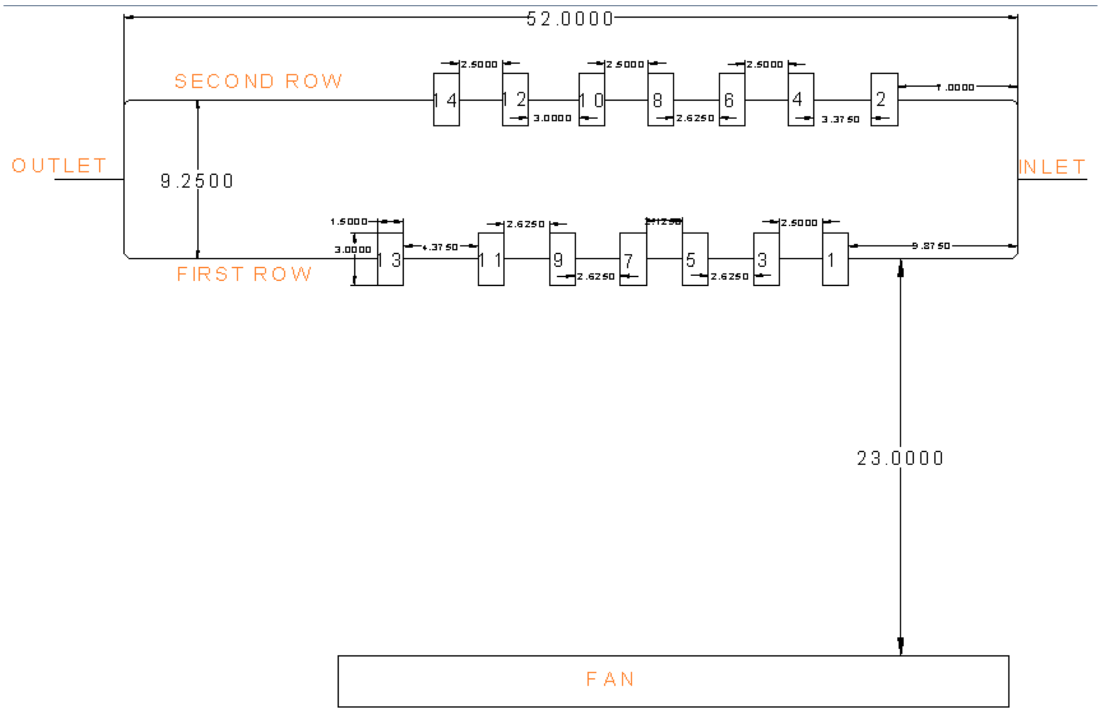

Figure 27: 14 Cradle Demonstrator Layout

7. Seven cradles with heat pipes were attached to each parallel section of steel pipe.
8. The heat pipes were inserted into a hole in each cradle and allowed to rest on a wooden support beneath the cradles.
9. The heat pipes were mounted in a vertical orientation.

4.2.2 Assembly of Test Apparatus

The experiments were conducted in the welding shop of Washburn Shops to provide sufficient ventilation for the internal combustion engine. The following steps were used to set up the test assembly:

1. A Briggs and Stratton 675 series engine mounted on a lawnmower (with the blade removed) was used for the hot fluid source for the testing.
2. The test assembly was mounted on a table, and the lawnmower was clamped to the table for stability.

3. The muffler of the engine was removed to improve flow, and a fitting was welded to attach the engine exhaust (at 7/8 inches) to the demonstrator inlet (at 5/8 inches).
4. A wooden frame was built to support the demonstrator piping sections during the experiments.
5. A 3-foot diameter Utilitech industrial fan (Model Number HVD-36A), operated on the “Low” setting, was placed 23 inches away from the demonstrator to match the air flow velocity (approximately 4.5 mph) of the full-scale cooler.

   (120V AC 60 Hz 3.8A 420W)

The final demonstrator setup is shown in Figure 28 below. Additional pictures of the demonstrator setup may be found in Appendix E - Experimental Setup Pictures.

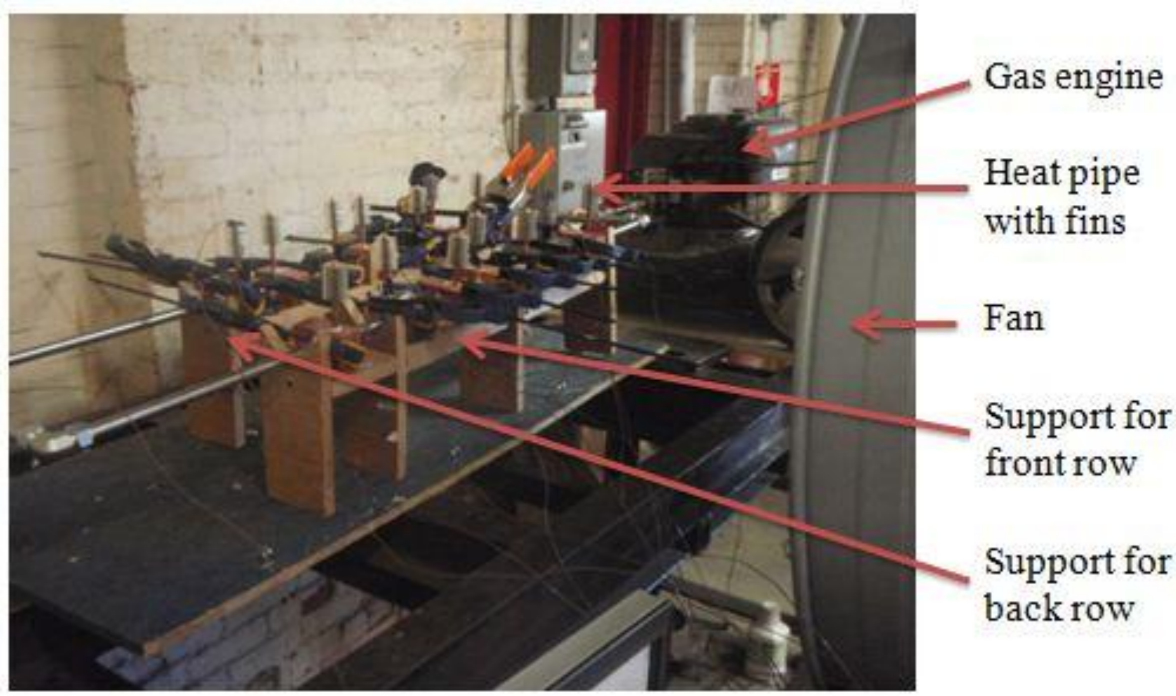

Figure 28: Final Demonstrator

4.2.3 Experimental Setup

1. The computer software LabView was used to create a data acquisition instrument for the test. Screenshots of the software setup may be found in Appendix F - LabView Setup for Data Acquisition.
2. Samples were taken at a rate of 6 Hz.
3. 11 Type-K thermocouples were mounted along the test apparatus to measure temperature changes
    a. 1 thermocouple was mounted at the engine exhaust to measure the temperature at the cooler inlet. This thermocouple could not be soldered because the temperatures at the exhaust were high enough to melt the solder, so the leads were twisted together to form a connection.
    b. 1 thermocouple was mounted at the demonstrator outlet after the flows were combined.
    c. 7 thermocouples were mounted on the bases of heat pipes on alternating cradles.
    d. 1 thermocouple was mounted at the top of the second heat pipe in the row closest to the fan.
    e. 1 thermocouple was mounted at the top of the final heat pipe in the second row.
4. 1 Hedland (MFG H671A-150) flow meter was mounted on the demonstrator outlet to measure the flow rate through the demonstrator.

4.2.4 Scaling Procedure

After performing our experiments, we needed to interpret our results and determine how to compare them with the full scale cooler model. Essentially, we must know the requirements needed for our model to able to cool natural gas to a desired temperature. We developed a method in which we can determine the length of the natural gas pipe and the number of heat pipes that would be necessary in order to effectively cool the gas to a temperature of 60°C.

1. First, we ran each experiment using our demonstrator and evaluate the average change in temperature (ΔT) between the beginning and the end of the pipe. We also determined a density for the fluid from the inlet temperature. Flow rate for the working fluid and the air across the fins will remain constant throughout the entire experiment.

2. Using our thermal resistance model, we calculated the total thermal resistance of the demonstrator using exhaust properties. This thermal resistance was called R_{ex}.

3. Next, we applied the same thermal resistance model to our full scale analytical model. The length and number of heat pipes were changed in the calculation to match the parameters determined from our analytical model. The resulting thermal resistance was labeled as R_{full}.

4. To compare the theoretical and experimental heat loads, we created a ratio R in which $R = R_{ex}/R_{full}$. This ratio will allow a comparison between the thermal resistance from the demonstrator and our analytical model.

5. We then calculated the thermal resistance for our demonstrator once again, only this time, we used the parameters of natural gas as the hot fluid. This resistance was named R_{ng1}.

6. Using the ratio R, which was previously calculated, we found the thermal resistance that would be required for a full size model to be able to cool the natural gas. This final thermal resistance was called $R_{ngfinal}$, and was solved using the equation $R_{ngfinal} = R_{ng1}/R$.

7. Two of the variables associated with our thermal resistance model that we can control are the total length of the cooler and the number of heat pipes being used. We created a table comparing both values, analyzed their effect on the total thermal resistance, and then found the minimum

values which can yield the proper change in temperature. The appropriate length and number of heat pipes were arranged in a fashion so the size and footprint of the cooler would be minimized.

5. Analysis

5.1 Performance

Eight different experiments were performed using our demonstrator, with engine exhaust from a lawnmower as our hot fluid. Each experiment had variations in number of heat pipes and cradles, type of convection (free or forced), flow rate, and time duration. The eight different experiments run with the demonstrator were:

1. Without heat pipes and with the fan off (free convection)
2. Without heat pipes and with the fan on (forced convection)
3. With 7 heat pipes and with the fan off
4. With 7 heat pipes and with the fan on
5. With 7 heat pipes, the fan off, an increased flow rate
6. With 14 heat pipes and with the fan off
7. With 14 heat pipes and with the fan on
8. With 14 heat pipes and with the fan on over a long duration

The long-duration test was run for 33 minutes, while the rest were run for approximately 7-8 minutes. Flow rate, measured at 0.002 m^3/s, was kept constant for all tests except one in which the throttle was adjusted to increase the flow rate. The throttle was adjusted, but the flow meter used was unable to measure the flow and failed catastrophically, likely due to the highly unsteady flow produced by the lawnmower at this higher flow rate.

5.1.1Temperature Profiles

For the six experiments which were run with heat pipes, thermocouples were utilized to gain a temperature profile across the cooler. For the 7 heat pipe set up, a thermocouple was placed on every cradle, and additional thermocouples were attached to two of the heat pipes. This allowed us to monitor how the heat pipes were performing. For the 14 heat pipe set up, a thermocouple was placed on every other cradle. Again, additional thermocouples were used to evaluate the performance of two of the heat pipes in the model. The temperature at each cradle was determined by averaging the thermocouple readings taken after 200 seconds up until 700 seconds had elapsed.

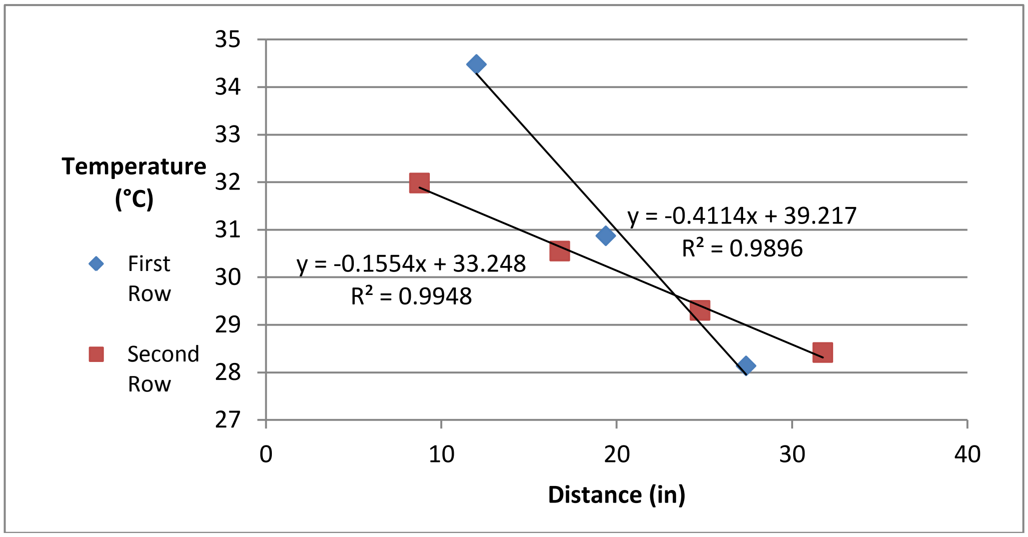

Figure 29: Temperature Profile - Fan off, 7 heat pipes

Figure 29 above shows the temperature profile from the test run with the fan off and 7 heat pipes utilized. The x-axis in this figure represents the distance each cradle location is from the demonstrator's inlet. We can see here that the exhaust is being cooled continuously as it flows along the gas pipe. However, the first row is being cooled at a faster rate than the second row. The reason for this is not clear, since the fan was not turned on. The results from the experiments run with the fan off and 14 heat pipes, shown below in Figure 30, as well as the experiment with the increased flow rate, shown in Figure 31, verify this conclusion.

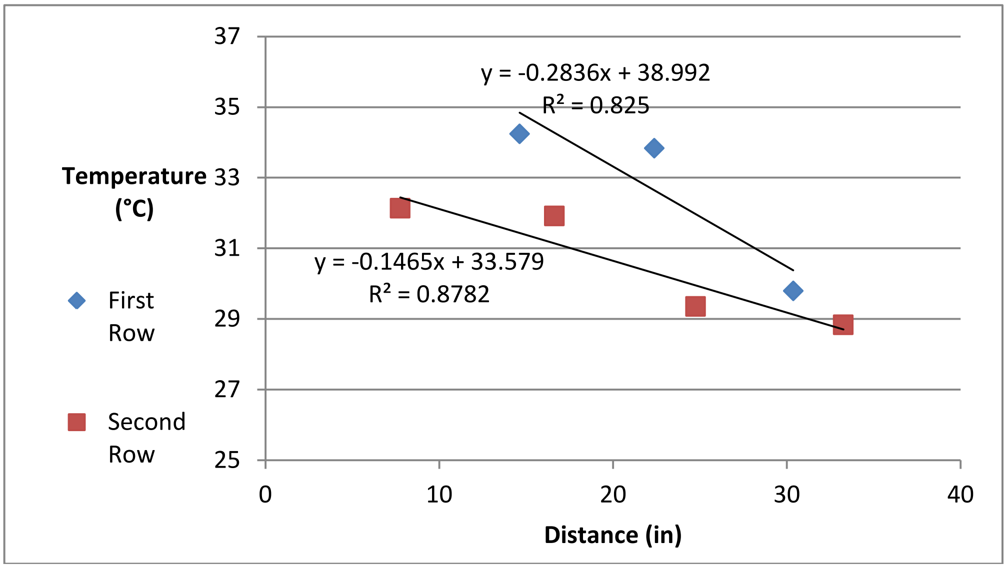

Figure 30: Temperature Profile - Fan off, 14 heat pipes

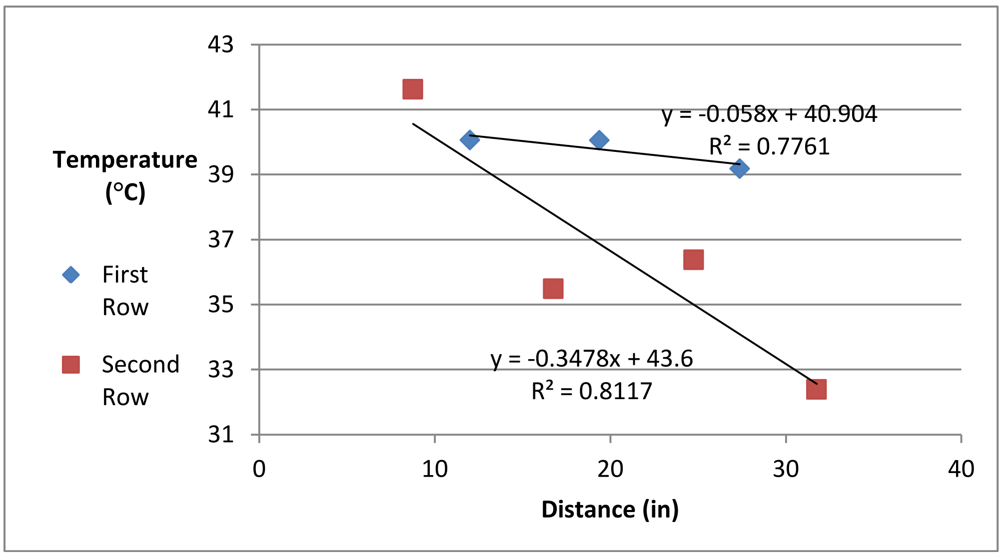

Figure 31: Temperature Profile - Fan off, 7 pipes, increased flow rate

For the first and second row, the temperatures and slopes in Figure 30 are close, which was what we expected. When the flow rate was increased, temperature profile in Figure 31 is opposite to that of Figure 29, with the second row experiencing a sharper decrease in temperature. Also, the temperatures with the increased flow rate were higher than the experiments at lawnmower's normal flow rate.

As we turned the fan on for the tests, we expected that the temperature profile for the first row would be lower than the profile for the second row, since the first row is closer to the fan.

However, the first test ran with the fan on, which featured 7 heat pipes, did not exhibit any significant temperature changes between the two rows until the exhaust reached the end of the pipe, where, contrary to what was expected, the second row experienced the drop in temperature. These results are shown below in Figure 32.

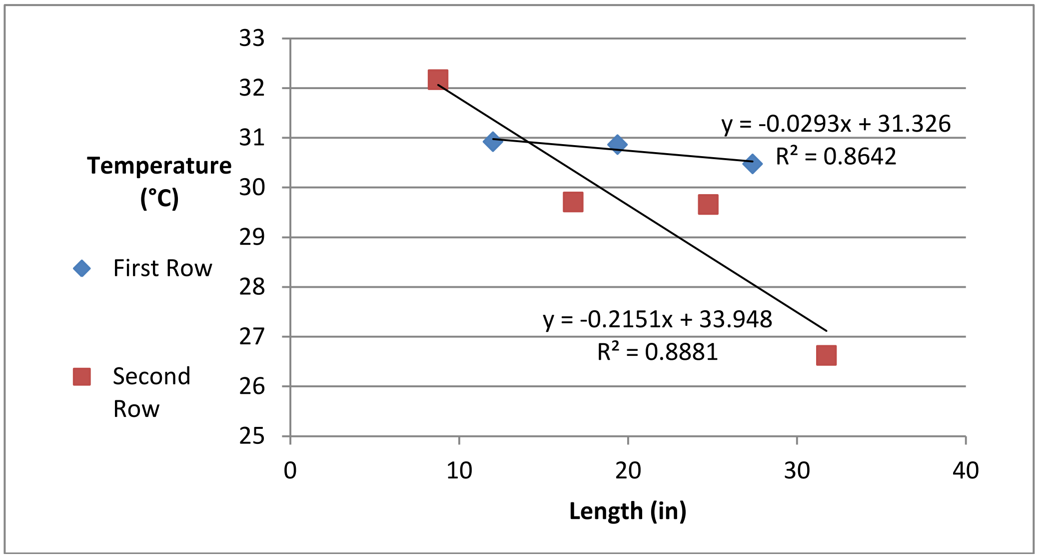

Figure 32: Temperature Profile: Fan on, 7 heat pipes

When the same test was run with 14 heat pipes, the difference in temperature that we were anticipating was experienced. The results for this experiment are shown below in Figure 33.

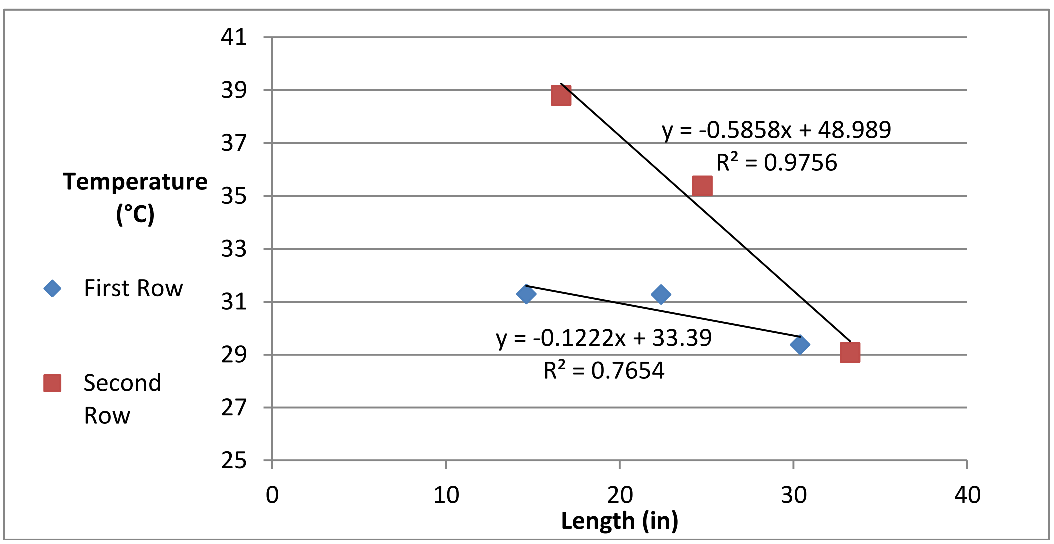

Figure 33: Temperature Profile: Fan on, 14 heat pipes

Overall, the results from the experiments with 7 heat pipes were inconsistent and led to a lot of uncertainty. The results from the experiments with 14 heat pipes were more consistent, like the ones in Figure 30 and Figure 33, and therefore more meaningful.

The temperature profile for the long duration test is shown below in Figure 34. This temperature profile was significant because it best showed the difference the fan has on cooling. The temperatures for the first row were, on average, about 6°C cooler than the temperatures for the second row. The averages were taken after 200 second up until 33 minutes had elapsed.

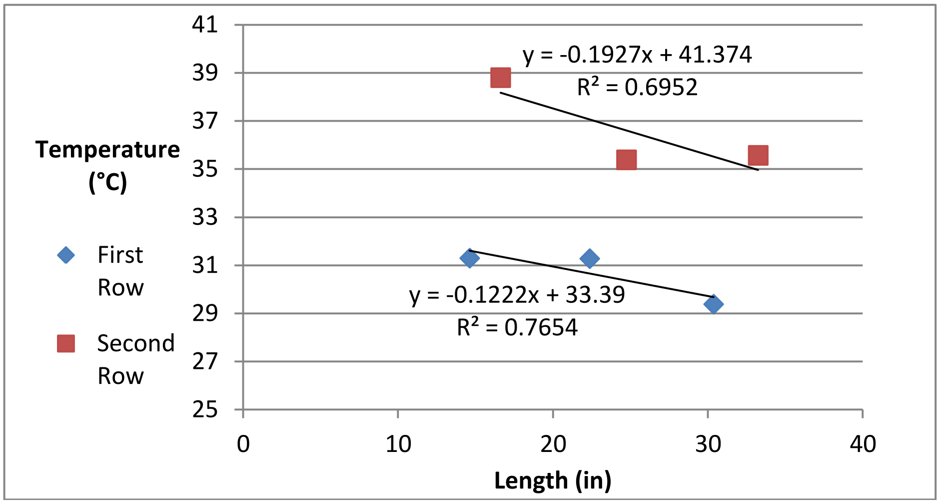

Figure 34: Temperature Profile - Fan on, 14 heat pipes, long duration (33 minutes)

5.1.2 Heat Pipe Performance

For each experiment that included heat pipes, two heat pipes were hooked up to thermocouples. For these two heat pipes, one thermocouple was attached to the base of the heat pipe, while the other was attached to the top of the heat pipe. The intent is to determine the performance of the heat pipes and observe how much heat they remove from the exhaust. From the temperature readings, we were able to see how the heat pipes performed under the variety of conditions provided in our tests. For the 7 heat pipe demonstrator, temperature readings were taken from Heat Pipes 1 and 3 (refer to Figure 26). For the 14 heat pipe demonstrator, temperature readings were taken from Heat Pipes 3 and 14 (refer to Figure 27).

Figure 35 shows the performance of Heat Pipe 1 during the three experiments ran with the 7 heat pipe demonstrator. The performance of Heat Pipe 3 is shown in Figure 36. In general, Heat Pipe 3 experienced a larger temperature difference, implying that the heat pipes at the end of the demonstrator remove more heat.

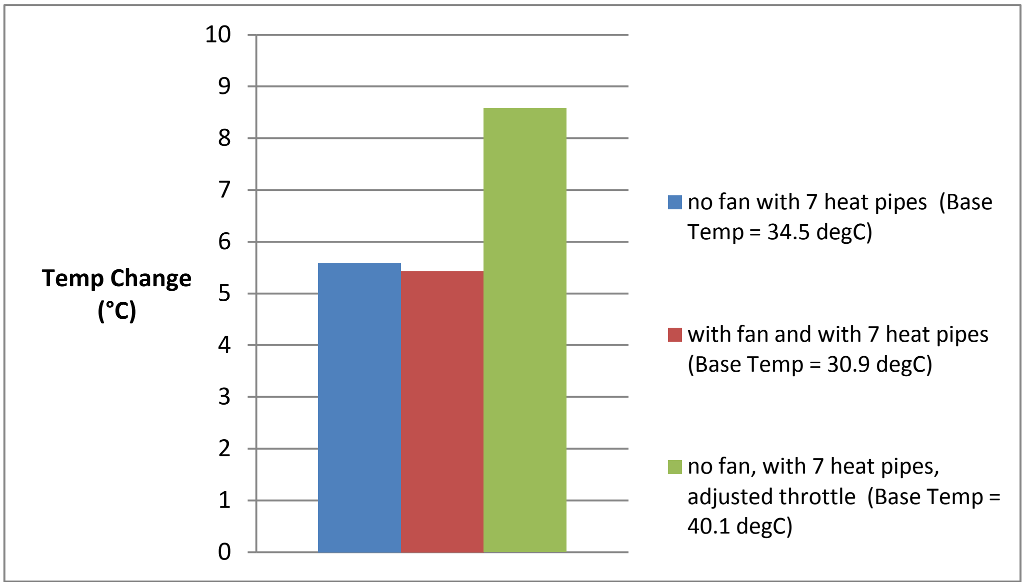

Figure 35: Temperature Change across heat pipe 1 for the 7 heat pipe demonstrator

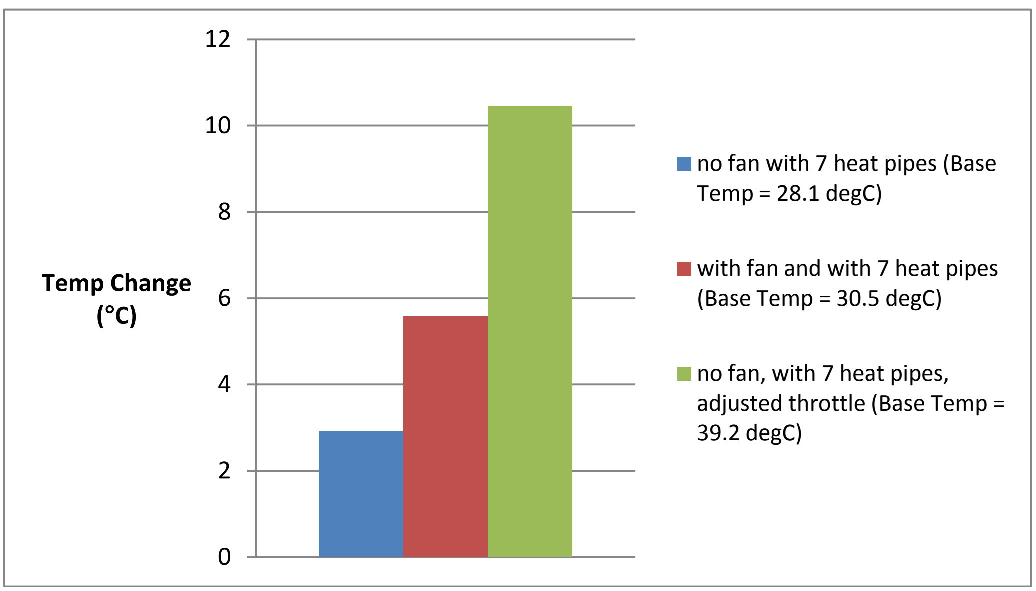

Figure 36: Temperature Change across heat pipe 3 for the 7 heat pipe demonstrator

Figure 35 and Figure 36 also suggest that as the base temperature of the heat pipe increases, the temperature change across the heat pipe increases. To better evaluate the performance of each heat pipe under different conditions, we found the ratio between the change in temperature across the heat pipe to the base temperature of the heat pipe. This variable is labeled as ΔT / T_base in Figure 37, which shows this performance for Heat Pipe 1, and Figure 38, which shows this performance for Heat Pipe 3.

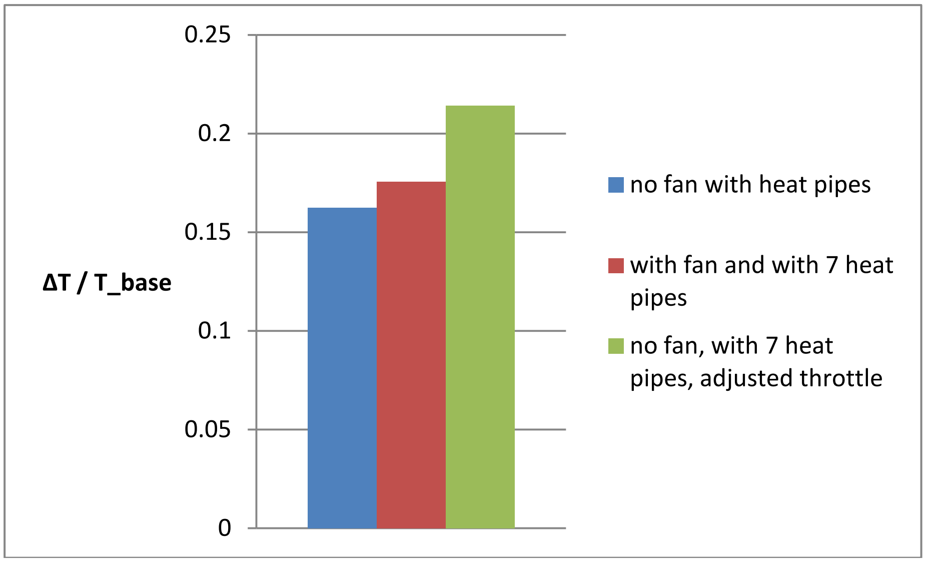

Figure 37: Performance of Heat Pipe 1 for the 7 heat pipe demonstrator

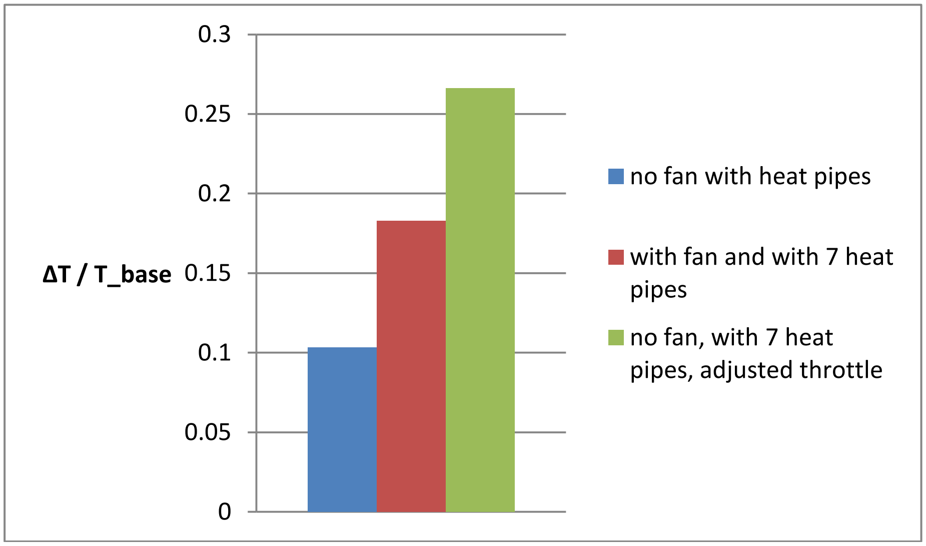

Figure 38: Performance of Heat Pipe 3 for the 7 heat pipe demonstrator

The results shown in Figure 37 and Figure 38 confirm that the heat pipes perform better when exposed to higher temperature, since the base temperature for the test with increased flow rate was almost 10°C higher than the other two tests. We can also see from the results of the two experiments run with the regular flow rate that the heat pipes appear to perform slightly more effectively with the fan turned on.

The results from the analysis of Heat Pipes 3 and 14 on our 14 cradle demonstrator gave us further confirmation on our findings from analyzing the 7 cradle demonstrator. The results from the 14 cradle demonstrator showed that the heat pipes nearest the outlet removed the most heat. Also, as with the 7 cradle demonstrator, when the temperature at the base of the heat pipes increases, the change in temperature across the heat pipes also increased. These results are displayed below in Figure 39 and Figure 40.

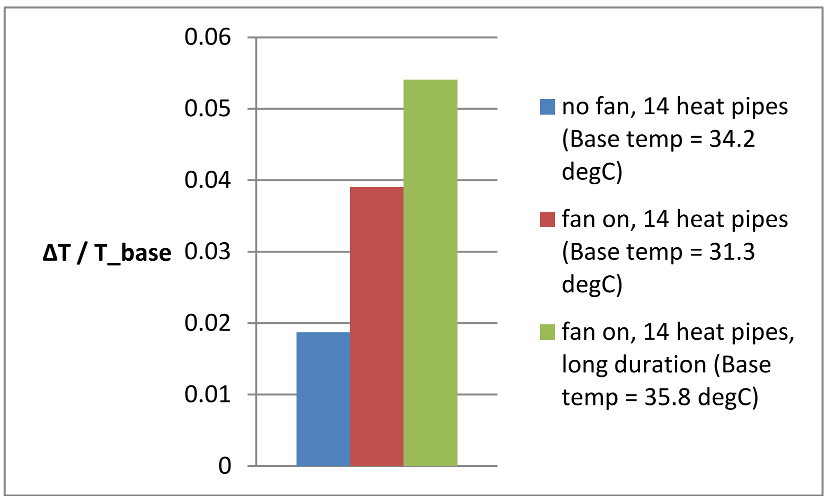

Figure 39: Performance of Heat Pipe 3 for the 14 heat pipe demonstrator

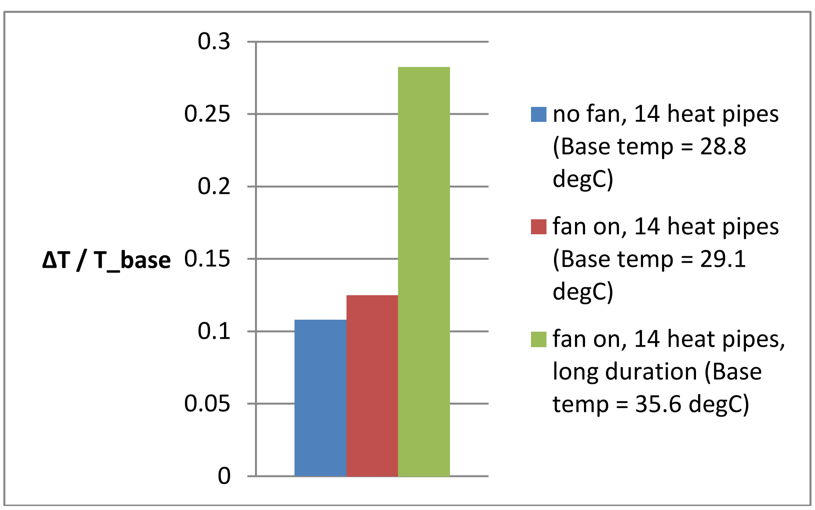

Figure 40: Performance of Heat Pipe 14 for the 14 heat pipe demonstrator

5.1.3 Cooling Performance

The ultimate goal of the demonstrator is to cool the exhaust passed through it. For each test, an average outlet temperature was averaged over the 200s to 700s time interval (with the exception of the long duration trial, in which the outlet temperature was averaged from 200s to the full 33 minutes). These calculations are found below in Table 3, arranged from the highest to lowest temperature. The average inlet temperature for the experiments was 420.9°C.

Table 3: Outlet temperature for each experiment

EXPERIMENT	OUTLET (°C)
increased flow rate	131.8
7 heat pipes, no fan	47.7
no heat pipes, no fan	44.9
no heat pipes, with fan	42.4
7 heat pipes, with fan	38.4
long duration	28.9
14 heat pipes, no fan	26.7
14 heat pipes, with fan	24.6

The first conclusion that can be reasoned from these results is that increasing the flow rate of the hot fluid will decrease the cooler's ability to cool the fluid. This is logical, as the faster the fluid moves, the less time it will have to be exposed to the heat pipes and convection across the pipes. Speaking of convection, another inference that we can make is that forced convection slightly improves the cooler's performance. On average, the addition of the fan lowered outlet temperature by 4.6°C. The most significant conclusion we can draw here is that as the number of heat pipes increase, the cooler performance also increases, a conclusion which we expected to be able to make.

5.1.4 Scaling Analysis

The final step in the analysis of our data was to scale our results to determine the minimum length and number of heat pipes our design must feature in order to sufficiently cool the natural gas. The gas must be cooled from 105°C to 60°C, a change in temperature of 45°C (or 45 K). Table 4 below shows a comparison between results calculated with our analytical model, our scaled experimental results, and the industry standard for natural gas coolers. The calculations used for our analytical model analysis are shown in Appendix G.3 Cradle Heat Pipe Design Analysis, the calculations for our scaled experiment are shown in Appendix G.2 Scaling Calculations, and the properties used to make such calculations are listed in Appendix B – Properties Used.

Table 4: Scaling Analysis

	Analytical Model	Scaled Experiment	Industry Standard
Length (m)	64	42	Could not be determined
Number of Heat Pipes	419	275	N/A
Footprint (m^2)	0.871	0.871	1.1
Volume (m^3)	2.084	1.445	1.1

From our scaling analysis, we were able to shorten our cooler length by 34%, decrease the number of heat pipes by 34%, and shrink the cooler's volume by 31%, in comparison to our analytical model. The footprint remained the same because, as with the analytical model, there would be three columns of tubes, with the tubes remaining at a length of 5 feet. We will also keep the same distance between the heat pipes. However, with this arrangement, we would be able to decrease the height of the cooler. A possible reason for the overestimate in our analytical model may be because we neglected the convective heat transfer from the gas and heat pipes.

5.1.5 Demonstrator Analytical Analysis

In order to support and verify our demonstrator's data, we constructed an analytical model of the demonstrator to compare the experiment's results against. To provide the most accurate comparison as possible, we focused on the 14 heat pipes experiment with the fan running and defined certain variables such as:

- Number of heat pipes = 14
- Number of fins on each heat pipe = 18
- Measured distance of the setup
- Inlet and outlet temperature of the experiment = 420.9 C and 24.6 C respectively
- Specific Heat of engine exhaust = 1.327 kJ/kg*K
- Thermal Conductivity of engine exhaust = 0.034 W/m*K
- Viscosity of engine exhaust = 2.17×10^{-5} kg/m*s
- Molecular Weight of engine exhaust = 30 g/mol

These defined variables allowed us to construct a model which would tell us two things, the required maximum resistance of the demonstrator for this experiment, and the calculated resistance of the demonstrator. To calculate the maximum required resistance of the demonstrator involved defining the number of fins and their size, the number of heat pipes, the entrance and exit temperature of the demonstrator and the overall size and dimensions. Calculating the conductive and convective resistance through the demonstrator gave us the required resistance of 60.035 K/W. To find the calculated resistance of the demonstrator, we defined the specifications of the test engine and the fan, such as the flow rate, the specific heat of the engine exhaust, the viscosity, and the thermal conductivity of the exhaust. Solving for the thermal resistance of the complete system resulted in a calculated resistance of 62.61 K/W. By comparing these two resistances, we can gauge the accuracy of the experimental data against the analytical data, and thus verify the accuracy of the full model analysis. If the analytical model of the demonstrator were completely accurate, then the required maximum resistance and the calculated resistance would be equivalent. As such, this means that there is an error of 4.11% between the experiment's data and the analytical model's results. Refer to Appendix G.4 Demonstrator Analytical Analysis for the full calculation.

5.2 Cost

Based on the sizing calculated for our scaling model, a full heat exchanger unit would cost an estimated $27,725 if made using the drilling method, and $32,593 if made using the surfacing method. These estimates assume a machinist rate of $100 per hour. The materials cost is $15,230. The machining time for the cradles requires 109 hours for drilling, and 166 hours for surfacing. The time is the deciding factor between which method is more expensive. However, both methods are more expensive than the competitor quote of $24,269. These cost estimates are based on the two milling methods we explored. It is likely that using casting would reduce the overall manufacturing cost making it more competitively priced. Table 5 below shows the full breakdown of the equipment and materials costs of these two methods. Refer to Appendix C - Budget for a complete breakdown of the project budget.

Table 5: Equipment and materials cost breakdown

Cost Category	Drilling	Surfacing
Equipment		
8mm drill bits	$157.96	$157.96
5/8in drill bits	$999.46	N/A
#7 drill bits	$25.95	$25.95
10-24 tap	$523.25	$523.25
1/4 ball EM	N/A	$101.29
3/16 EM	N/A	$81.48
Total	$1,706.62	$889.93
Materials		
Copper	$947.92	$947.92
Steel Pipe	$845.88	$845.88
Connectors	$3,843.84	$3,843.84
Heat Pipes	$9,465.75	$9,465.75
Screws	$24.99	$24.99
Total	$15,128.38	$15,128.38
Total	$16,835.00	$16,018.31

6. Conclusions

From our scaling analysis, we were able to conclude that our model will have a significantly smaller footprint (21.5%) compared to the current cooler on the market. This proves that the design is a viable solution for OsComp Systems. However, we would recommend conducting further experiments with the following considerations to further validate our findings.

The ventilation fan in the welding lab appeared to be much stronger than the flow rate presented by our fan. This may explain why the fan had an insignificant effect on the cooler performance. This judgment was simply made by comparing to the strengths of the fans by placing our hand close to the duct and behind the fan we used in our experiment. Due to WPI's safety regulations, we were required to have the ventilation fan on at all times during the experiments. This means that we cannot accurately estimate the effect of the forced convection on the system.

An additional conclusion we can draw here is that as the number of heat pipes increase, the cooler performance also increases, a conclusion which we expected to be able to make. Simply stated, the increase in the number of cradles and heat pipes allowed for more heat to be removed from the exhaust.

7. Recommendations

7.1 Heat Pipe Selection

There are many factors that affect the performance of a heat pipe. The material, working fluid, and wick structure should remain the same for this application, which is copper, water, and sintered powder metal. However, there are additional parameters that could be adjusted to yield a higher performance. Increasing the diameter of the heat pipe would allow for greater power carrying capacity. Increasing the lengths of the evaporator and condenser regions would also improve the heat removal capabilities since this would increase the surface area for the heat to pass through and dissipate.

A heat pipe performs at its best when gravity is assisting the device. Therefore, we recommend the heat pipes stay at 90 degrees, which is in the vertical position with respect to the ground. Also, a higher operating temperature will allow a greater maximum heat removal from the system. We also recommend keeping the heat pipe completely straight. Although heat pipes can still function with bends, this decreases the performance of the device and could bring about leakage problems.

Lastly, we recommend looking into the fin analysis further. Our calculations were based on the fundamental equations found in any heat transfer textbook; refer to Appendix G.1 Fin Analysis and Optimization. However, they seem a little too small compared to the typical heat sinks on the market. We would recommend increasing the length and width of the fins as long as the efficiency is greater than 70%. The fin thickness and spacing should remain the same, as they reflect the industry standard. The number of fins required per heat pipe is dependent on the length of the condenser region and would therefore be adjusted accordingly.

7.2 Manufacturing

Casting the cradles instead of milling them may reduce equipment cost. It will also reduce material use, as less material will be thrown out as chips. It also means the material can be bought in the cheapest form not as the appropriately sized stock. Finally it will be easier to get a better weight balance with casting as curved sections do not significantly increase production difficulty.

Since copper is a soft material, it is possible that the threads on the inside of our design will pull out during practical use. If that is an issue, helicoils could be installed in the holes, or the holes could be changed to through holes and have a nut on the other side. However, we would also recommend looking into other materials for the cradle pieces, possibly a grade of aluminum. Copper was chosen for its strong heat transfer properties but is not an ideal metal to machine.

7.3 Conducting Experiments

Since compressed natural gas is not a safe fluid to work with in an academic setting, we were forced to seek alternative heat sources. However, using CNG would yield more accurate results. The % error would be reduced since the calculations to scale the properties of engine exhaust to CNG would no longer be required in the analysis. We would also recommend using a more consistent fluid source, in terms of its flow rate. The lawnmower engine we used to conduct our experiments has a pulsing flow, which doesn't reflect the ideal conditions for our application.

The unsteadiness in the flow turned out to be stronger than we anticipated, introducing substantial vibrations into the system. Adding fixtures, such as brackets, to securely mount the steel pipes and heat pipes to the base support would reduce this effect. Also, using a higher quality flow meter would yield more accurate results. Between the intermittent flow rate and the vibrations present in the system, our flow meter failed shortly after our initial experiments.

Lastly, increasing the size of the scale model would give a better indication of the cooler's performance. Having additional sections, steel pipes with heat pipes mounted onto them, would allow for the stacking and masking effects to be tested. These alternative test setups would be critical in testing the design further.

8. References

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Appendices

Appendix A - Authorship

Section		Peer Reviewers
Abstract	Team	Team
1, 3.2.1, 3.4, 5.2	Jess	Calvin & Brad
2.1, 3.2.0,4,8, 4.1, 4.2.4, 5.1	Brad	Andy & Chris
2.2, 3.2.3, 4.2.0-3	Andy	Chris & Jess
2.3, 3.2.5-7, 6, 7	Chris	Jess & Calvin
2.4, 3.1, 3.2.2, 3.3, 5.1.5	Calvin	Brad & Andy

Appendix B – Properties Used

The following assumptions were made in our calculations and analysis of our designs:

CNG pipes:
- Material: Aluminum alloy 2024
- 100% heat transfer efficiency from CNG pipes to heat pipes
- Used only for cooling CNG
- Thermal conductivity: 143 W/m*K

Air Properties:
- Density = 1.185 kg/m^3
- Specific Heat = 1.005 kJ/kg*K
- Viscosity = $1.983x10^{-5}$ kg/m*s

CNG Properties:
- Density = 53.334 kg/m^3
- Flow rate = 0.473 m^3/s
- Specific Heat = 2.26 kJ/kg*K
- Thermal Conductivity = 0.035 W/m*K
- Viscosity = $1.1x10^{-5}$ kg/m*s
- Molecular Weight = 16.044 g/mol

Exhaust Properties:
- Specific Heat = 1.327 kJ/kg*K
- Thermal Conductivity = 0.034 W/m*K
- Viscosity = $2.17x10^{-5}$ kg/m*s
- Molecular Weight = 30 g/mol

Appendix C - Budget

Table 6: Project Budget

Vendor	Quantity	Item	Individual Price	Total Price
Enertron	2	Heat Pipes	26.82	53.64
MSC Direct	1	Copper Bar, 7ft	669.01	669.01
MSC Direct	1	Screws, box of 100	31.38	31.38
OnlineMetals	2	Steel Rod, 5ft	30.21	60.42
MSC Direct	1	Copper Bar, 1ft	114.21	114.21
MSC Direct	1	Male Connector	24.28	24.28
MSC Direct	1	Thermalcouple wire	89.97	89.97
MSC Direct	1	Flowmeter	318.52	318.52
Northern Tools	1	Motor (Returned S&H)	20	20
Aavid Thermalloy	15	Heat Pipes w/ fins	54.09	811.35
MSC Direct	6	Jobber Drill Bits	3.59	21.54
MSC Direct	2	Thermal Paste, 8oz tube	27.3	54.6
MSC Direct	4	Flute Taps	14.75	59
McMaster-Carr	4	5/8" 90 degree elbows	21.89	87.56
McMaster-Carr	2	5/8" T-connectors	26.62	53.24
McMaster-Carr	1	1" to 1/2" reducer	90.41	90.41
McMaster-Carr	1	1/2" to 5/8" reducer	37.76	37.76
OnlineMetals	1	1" OD pipe , 1ft	13.2	13.2
OnlineMetals	1	1/2" OD pipe, 1ft	5.08	5.08
OnlineMetals	1	5/8" OD pipe, 2ft	15.26	15.26
MSC Direct	6	5/8in Jobber Drill Bits	23.68	142.08
MSC Direct	4	5/8in Drill Bits	28.71	114.84
MSC Direct	3	5/8in TG collet	24.69	74.07
Total Price				2961.42

Appendix D - CAD Drawings and Pictures

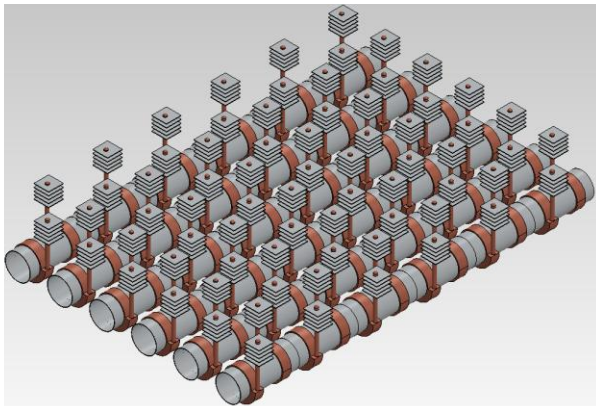


Figure 41: Preliminary design model, using heat pipe concept

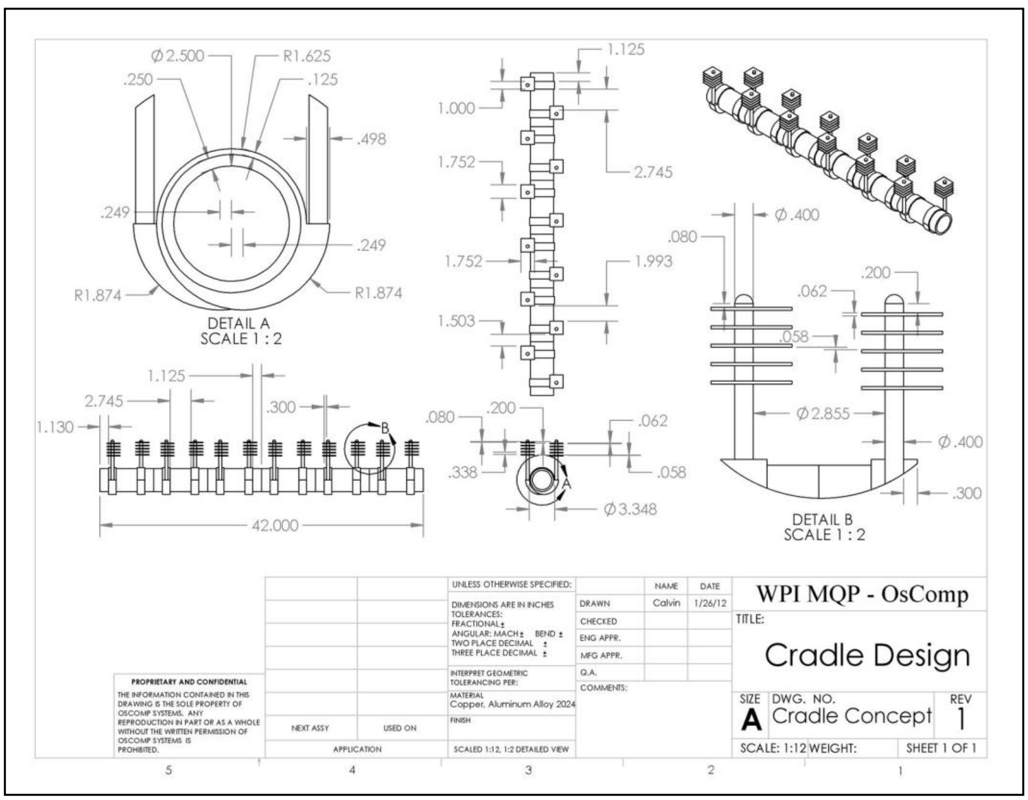


Figure 42: Preliminary design drawing, using heat pipe concept

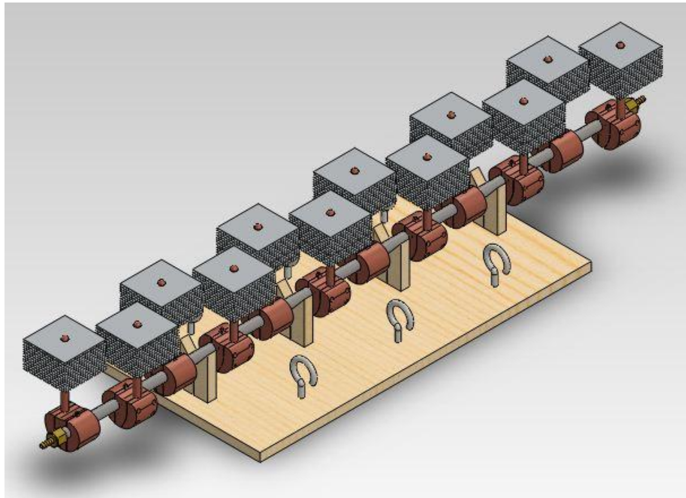


Figure 43: Preliminary demonstrator model

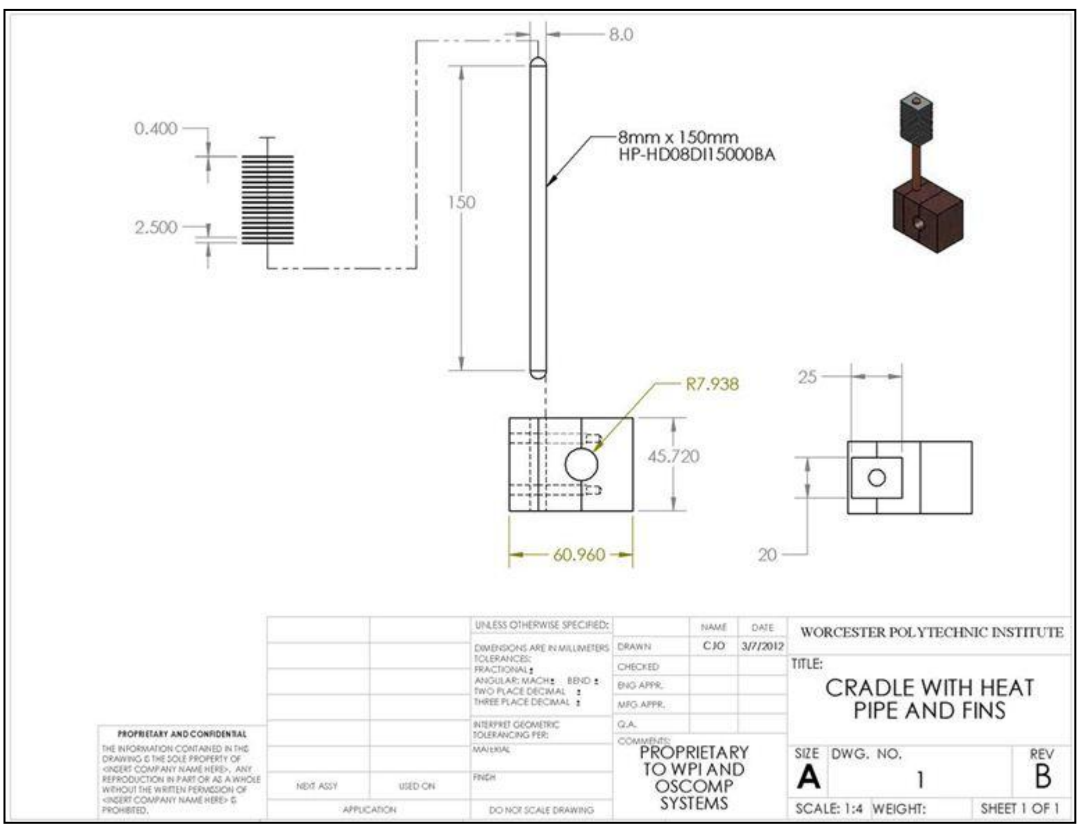


Figure 44: Cradle drawing

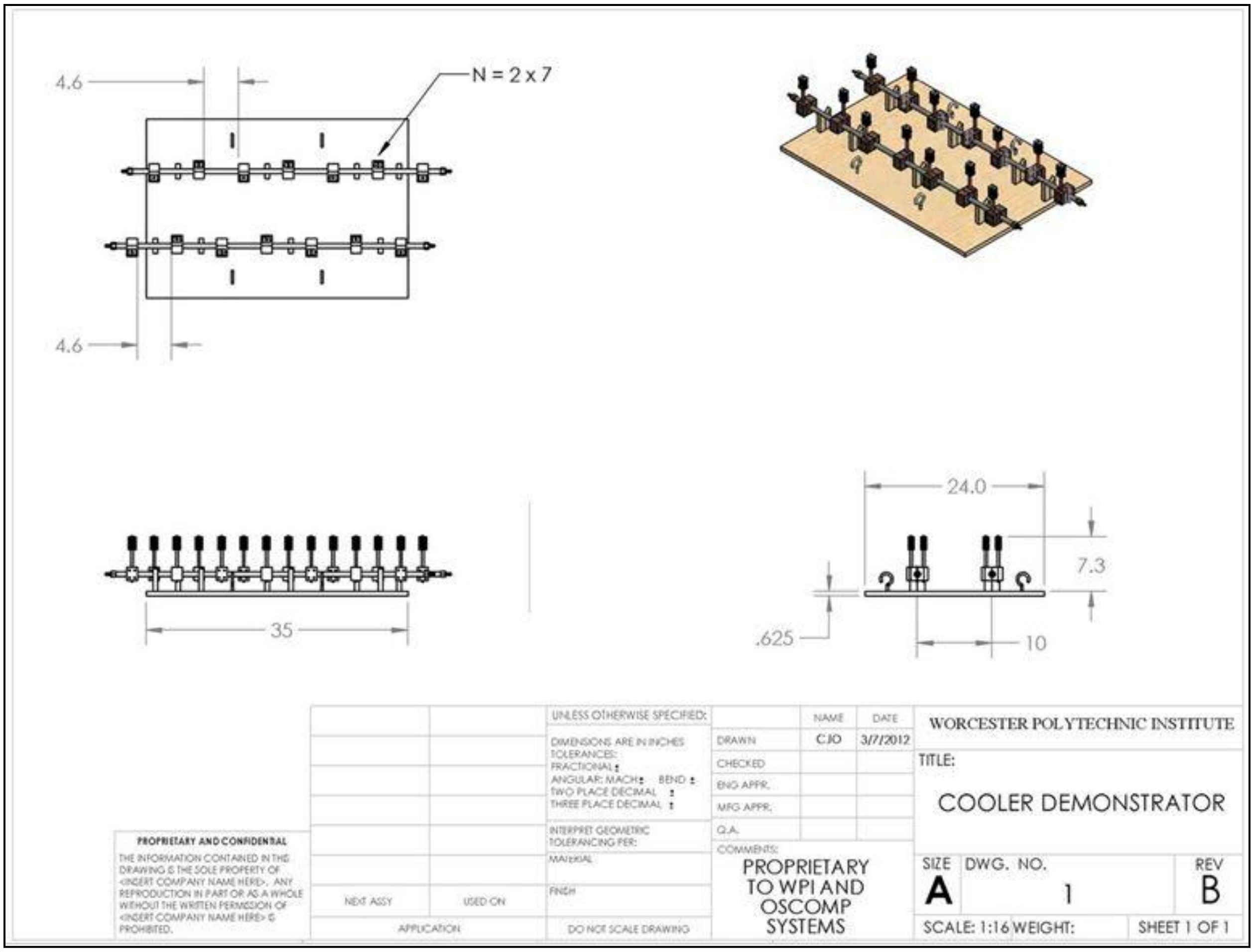


Figure 45: Demonstrator drawing

Appendix E - Experimental Setup Pictures

Figure 46: Close up picture of heat pipe

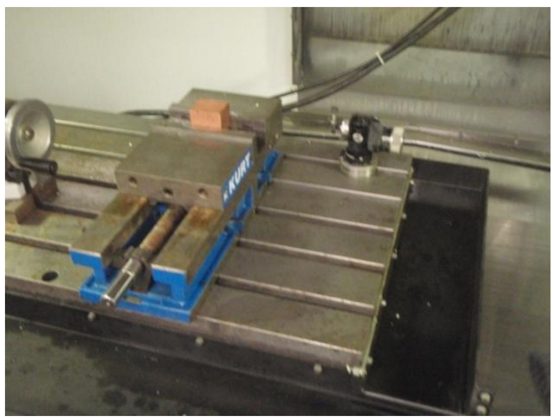


Figure 47: Securing copper for machining

Figure 48: Exhaust outlet connecting to cooler

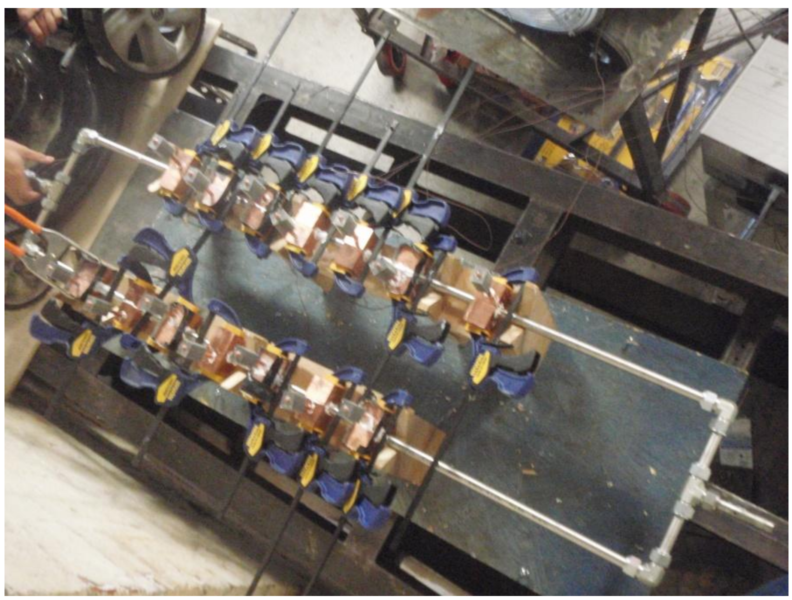


Figure 49: Final setup of demonstrator 1

Figure 50: Final setup of demonstrator 2

Appendix F - LabView Setup for Data Acquisition

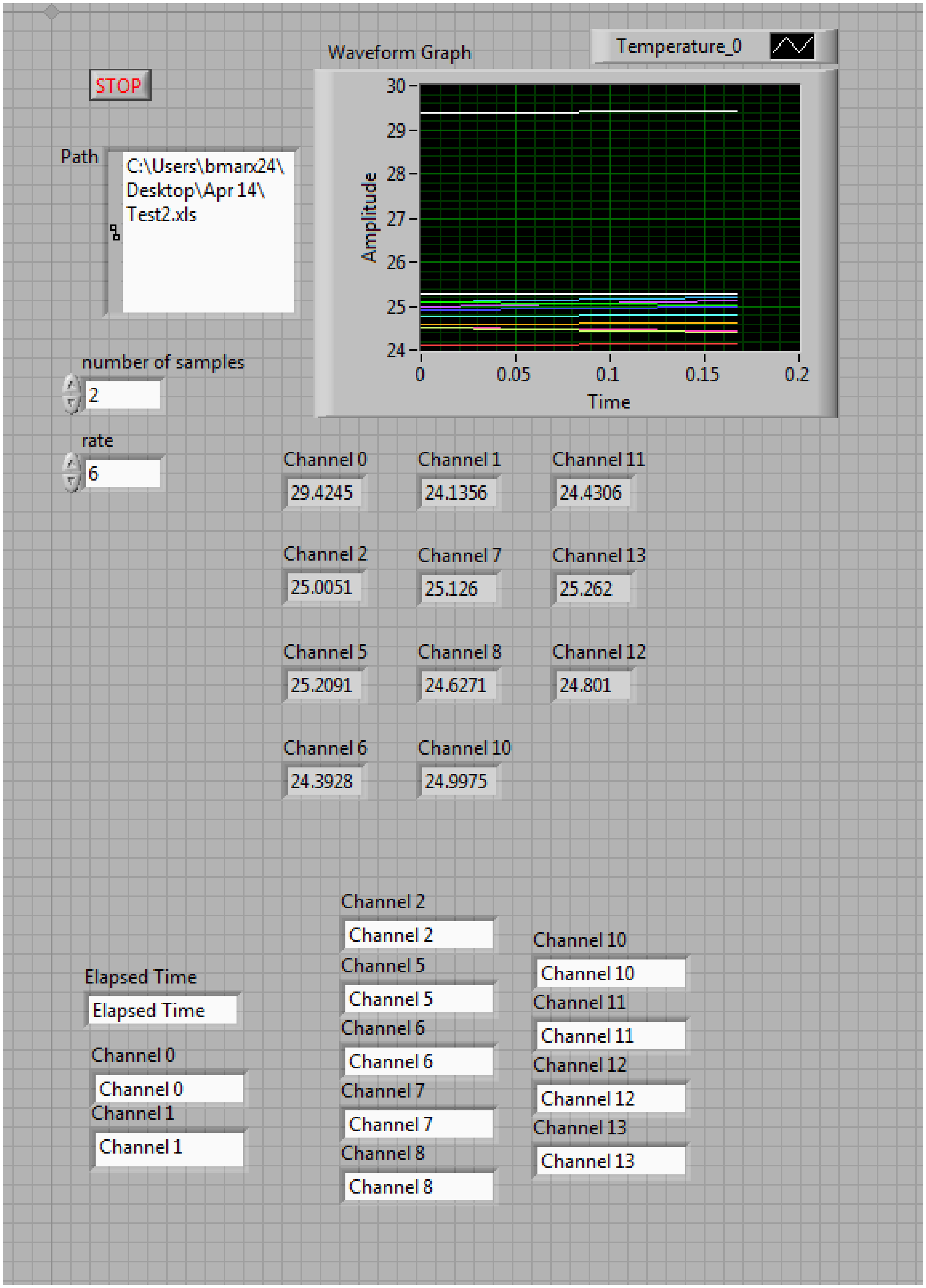


Figure 51: LabView Virtual Instrument

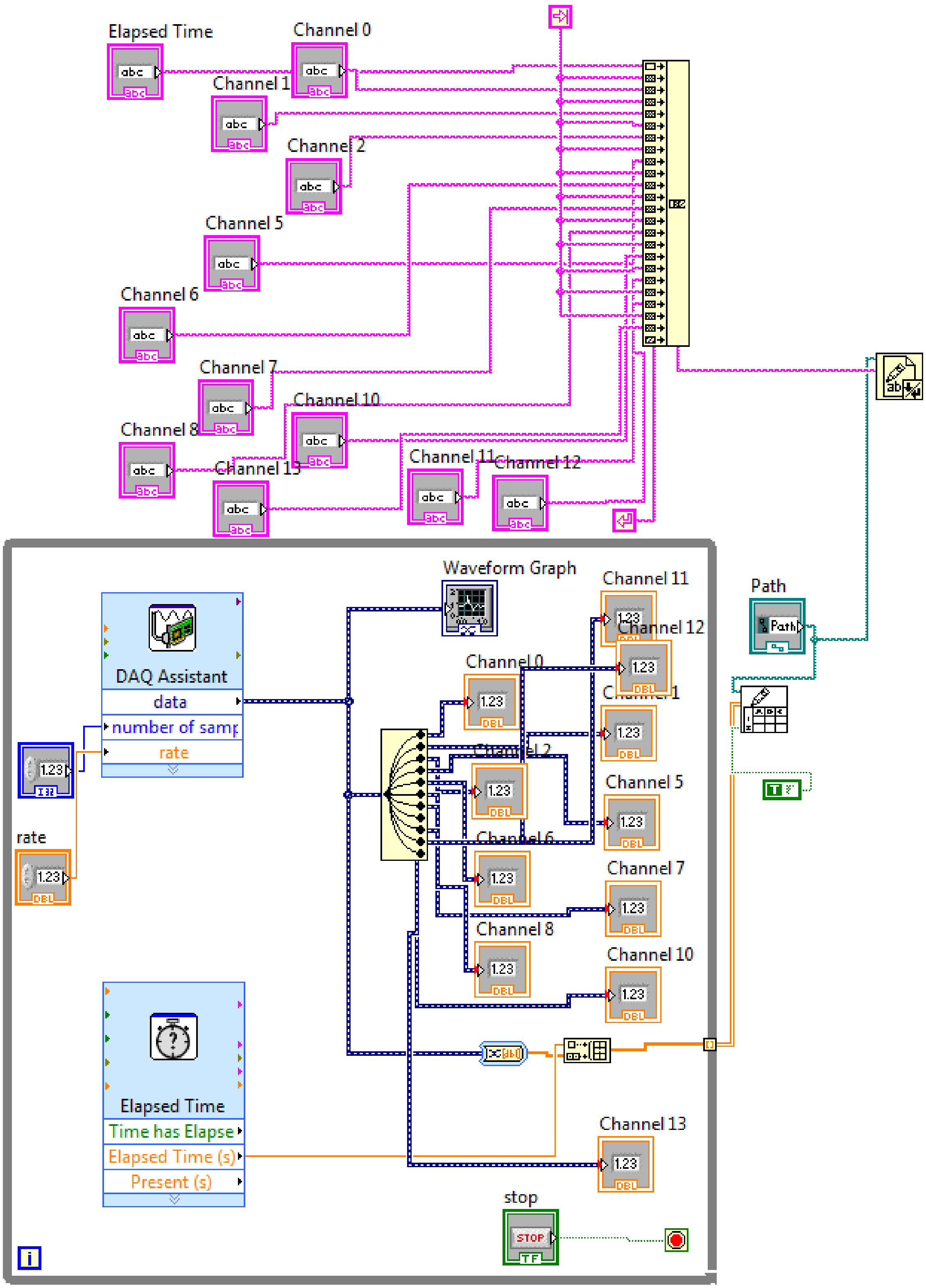


Figure 52: LabView Block Diagram

Appendix G - Mathcad Calculations

G.1 Fin Analysis and Optimization

$d := 8\text{mm}$ diameter of heat pipe

$L_1 := 2\text{cm}$ Length of fin

$w := 2.5\text{cm}$ width of fin

$t := 0.4\text{mm}$ thickness of fin

$T_b := 150\,°C = 423.15\,K$ Base temp of the fin

$k := 134 \frac{W}{m \cdot K}$ thermal conductivity of fin (aluminum, pure @ 150C, which is 300F) **http://www.engineeringtoolbox.com/thermal-conductivity-metals-d_858.html**

$T_{infinity} := 50\,°C = 323.15\,K$ Temp. of the fluid (air)

$h := 100 \frac{W}{m^2 \cdot K}$ Convection coefficient of fluid (air)

a) convection heat transfer

$P := 2 \cdot w + 2 \cdot t$ fin perimeter $\theta_b := T_b - T_{infinity} = 100\,K$

$A_c := w \cdot t$ cross sectional area of fin

$A_f := P \cdot L_1 - 2\pi \cdot \left(\frac{d}{2}\right)^2 + 2w \cdot t$ surface area of the fin

$M := \sqrt{(h \cdot P \cdot k \cdot A_c)} \cdot \theta_b = 8.251\,W$

$m_1 := \sqrt{\frac{(2 \cdot h)}{k \cdot t}} = 61.085 \frac{1}{m}$ Fin parameters

$$q_f := M \cdot \frac{\left[\sinh(m_1 \cdot L_1) + \left(\frac{h}{m_1 \cdot k}\right) \cdot \cosh(m_1 \cdot L_1)\right]}{\cosh(m_1 \cdot L_1) + \left(\frac{h}{m_1 \cdot k}\right) \cdot \sinh(m_1 \cdot L_1)} = 6.961\,W$$ Fin heat transfer rate

$\eta_f := \frac{q_f}{(h \cdot A_f \cdot \theta_b)} = 0.744$ fin efficiency

$\varepsilon_f := \frac{q_f}{h \cdot A_c \cdot \theta_b} = 69.611$ fin effectiveness

$R_{tf} := \frac{\theta_b}{q_f} = 14.366 \frac{K}{W}$ thermal resistance of fin

$$T_1 := \theta_b \cdot \left[\frac{1}{\cosh(m_1 \cdot L_1) + \left(\frac{h}{m_1 \cdot k}\right) \cdot \sinh(m_1 \cdot L_1)}\right] + T_{infinity} = 376.834\,K$$ temp. at the tip of fin

Table 7: Fin Optimization 1

Given Data:			
diameter of heat pipe	8	mm	
Length of heat pipe	150	mm	
thickness of fin	0.4	mm	
fin spacing	2.5	mm	
Assumptions:			
Fin material	aluminum, pure		
thermal conductivity of fin (k)	134	W/mK	http://www.engineeringtoolbox.com/thermal-conductivity-metals-d_858.html
convection coeffcient of fluid (h)	100	W/m^2K	
Base Temp (T.b)	150	C	
Temp. of fluid (air - T.infinity)	50	C	forced convection
Length of condenser region	60	mm	
Length of evaporator region	45	mm	
Number of fins	20		Lc/(spacing+thickness)
Variables:			
Length of fin			
Width of fin			

Table 8: Fin Optimization 2

Length (cm)	Width (cm)	Efficiency	Effectiveness
2	1	0.871	70.428
2	1.25	0.824	70.157
2	1.5	0.796	69.975
2.25	1	0.785	73.66
2	1.75	0.777	69.845
2	2	0.763	69.748
2	2.25	0.752	69.672
2.25	1.25	0.749	73.376
2	**2.5**	**0.744**	**69.611**
2	2.75	0.738	69.561
2	3	0.732	69.52
2.25	1.5	0.726	73.186
2.5	1	0.712	76.13
2.25	1.75	0.711	73.05
2.25	2	0.7	72.948
2.25	2.25	0.692	72.869
2.25	2.5	0.685	72.805

Potential fin arrangements, above line (>70% efficiency)

2.5	1.25	0.684	75.836
2.25	2.75	0.68	72.753
2.25	3	0.676	72.71
2.5	1.5	0.666	75.64
2.5	1.75	0.654	75.499
2.75	1	0.651	77.999
2.5	2	0.645	75.394
2.5	2.25	0.638	75.312
2.5	2.5	0.633	75.246
2.75	1.25	0.628	77.699
2.5	2.75	0.628	75.192
2.5	3	0.625	75.147
2.75	1.5	0.613	77.498
2.75	1.75	0.603	77.354
3	1	0.598	79.405
2.75	2	0.596	77.246
2.75	2.25	0.59	77.161
2.75	2.5	0.586	77.094
2.75	2.75	0.582	77.039
3	1.25	0.579	79.099
2.75	3	0.579	76.993
3	1.5	0.567	78.895
3	1.75	0.558	78.748
3	2	0.552	78.638
3	2.25	0.547	78.552
3	2.5	0.544	78.484
3	2.75	0.541	78.247
3	3	0.538	78.381

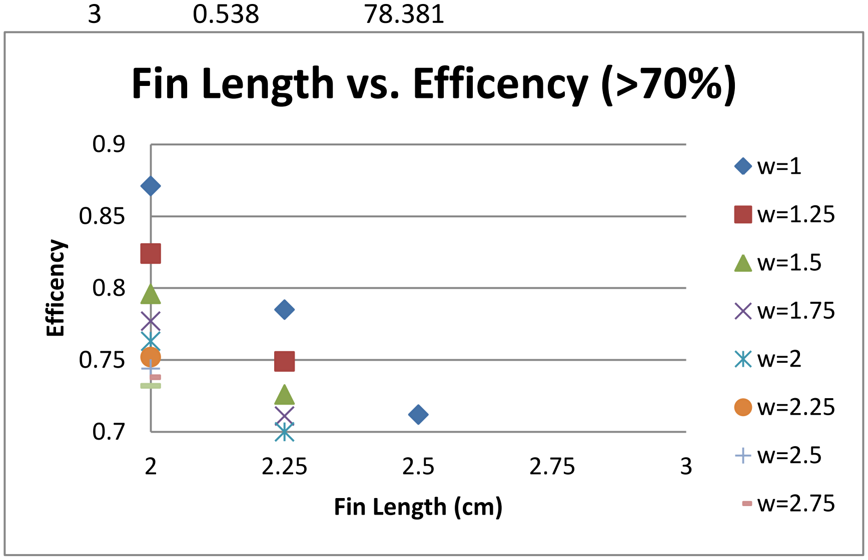


Figure 53: Fin Optimization

G.2 Scaling Calculations

Demonstrator Geometry $kJ := 1000J$

Inner Diameter $D_i := 13.386mm$

Outer Diameter $D_o := 15.875mm$

Inner Radius $r_i := \frac{D_i}{2} = 6.693 \times 10^{-3} m$

Outer Radius $r_o := \frac{D_o}{2} = 7.938 \times 10^{-3} m$

Length $L_{demo} := 52in = 1.321m$

Cross sectional of Area $A_{cross} := \frac{\pi}{4} \cdot D_i^2 = 1.407 \times 10^{-4} m^2$

Surface Area of Pipe $A_{Ldemo} := \pi \cdot D_i \cdot L_{demo} = 0.056m^2$

Full Size Model Geometry

Diameter and cross-sectional area is the same

Length $L_{full} := 64.008m$

Surface Area $A_{Lfull} := \pi \cdot D_i \cdot L_{full} = 2.692m^2$

Determining the density of the exhaust

gas constant: $R := 8.314 \frac{J}{mol \cdot K}$

Measured Parameters:

Inlet Temperature $T_1 := 402K$

Outlet Temperature $T_2 := 29K$

Outlet Pressure $P_2 := 101325Pa$

Measure Outlet Flow rate $Flow_{SCFM} := 40 \frac{ft^3}{min} = 0.019 \frac{m^3}{s}$

Guess $\rho_1 := 0.912 \frac{kg}{m^3}$

$\rho_2 := \rho_1$

Standard Pressure	$P_{Std} := 101325 Pa$
Standard Temperature	$T_{Std} := 273.15 K$
True Outlet Flow rate	$Flow_2 := \frac{P_{Std} \cdot Flow_{SCFM} T_2}{P_2 \cdot T_{Std}} = 2.004 \times 10^{-3} \frac{m^3}{s}$
Inlet Viscosity	$\mu_1 := 2.17 \cdot 10^{-5} \frac{kg}{m \cdot s}$
Mass flow rate	$mass_{flow} := \rho_2 \cdot Flow_2 = 1.828 \times 10^{-3} \frac{kg}{s}$
Inlet Flow rate	$Flow_1 := \frac{mass_{flow}}{\rho_1 \cdot A_{cross}} = 14.242 \frac{m}{s}$
Average Density	$\rho_{avg} := \frac{\rho_1 + \rho_2}{2} = 0.912 \frac{kg}{m^3}$
Average Velocity	$V_{avg} := \frac{Flow_1 + \frac{Flow_2}{A_{cross}}}{2} = 14.242 \frac{m}{s}$
Reynolds number	$Re_D := \frac{\rho_{avg} \cdot V_{avg} \cdot D_i}{\mu_1} = 8.012 \times 10^3$
Surface roughness	$\varepsilon := 0.015 10^{-3} m$
Friction factor	For $Re_D = 8.012 \times 10^3$ and $\frac{\varepsilon}{D_i} = 1.121 \times 10^{-3}$
	$f := \frac{1.325}{\left(\ln\left(\frac{\frac{\varepsilon}{D_i}}{3.7} + \frac{5.74}{Re_D^{0.9}}\right)\right)^2} = 0.035$
Friction loss	$h_f := f \cdot \frac{L_{demo}}{D_i} \cdot \frac{V_{avg}^2}{2 \cdot g} = 35.358 m$
Inlet Pressure	$P_1 := h_f \cdot \rho_{avg} \cdot g + P_2 = 1.016 \times 10^5 Pa$

Inlet Density $\rho_{1mol} := \frac{P_1}{R \cdot T_1} = 30.411 \frac{mol}{m^3}$

Molecular weight $MW := 0.030 \frac{kg}{mol}$

Inlet Density $\rho_1 := \rho_{1mol} \cdot MW = 0.912 \frac{kg}{m^3}$ iterate until density matches

	Exhaust Characteristics	Natural Gas Characteristics
Density	$\rho_{ex} := \rho_1 = 0.912 \frac{kg}{m^3}$	$\rho_{ng} := 53.334 \frac{kg}{m^3}$
Flow rate	$V_{ex} := Flow_2 = 2.004 \times 10^{-3} \frac{m^3}{s}$	$V_{ng} := 0.473 \frac{m^3}{s}$
Mass flow rate	$m_{ex} := \rho_{ex} \cdot V_{ex} = 1.829 \times 10^{-3} \frac{kg}{s}$	$m_{ng} := \rho_{ng} \cdot V_{ng} = 25.227 \frac{kg}{s}$
Specific Heat	$c_{pex} := 1.327 \frac{kJ}{kg \cdot K}$	$c_{png} := 2.26 \frac{kJ}{kg \cdot K}$
Thermal Conductivity	$k_{ex} := 0.034 \frac{W}{m \cdot K}$	$k_{ng} := 0.035 \frac{W}{m \cdot K}$
Viscosity	$\mu_{ex} := \mu_1 = 2.17 \times 10^{-5} \frac{kg}{m \cdot s}$	$\mu_{ng} := 1.1 \cdot 10^{-5} \frac{kg}{m \cdot s}$
Prandtl Number	$Pr_{ex} := \frac{c_{pex} \cdot \mu_{ex}}{k_{ex}} = 0.847$	$Pr_{ng} := \frac{c_{png} \cdot \mu_{ng}}{k_{ng}} = 0.71$
Reynolds Number	$Re_{ex} := Re_D = 8.012 \times 10^3$	$Re_{ng} := \frac{\rho_{ng} \cdot V_{ng} \cdot D_i}{A_{cross} \cdot \mu_{ng}} = 2.181 \times 10^8$
Nusselt Number	$Nu_{ex} := 0.023 Re_{ex}^{0.8} \cdot Pr_{ex}^{0.3}$	$Nu_{ng} := 0.023 Re_{ng}^{0.8} \cdot Pr_{ng}^{0.33}$
	$Nu_{ex} = 29.046$	$Nu_{ng} = 9.631 \times 10^4$
Convection Coefficient	$h_{ex} := \frac{Nu_{ex} \cdot k_{ex}}{D_i} = 73.775 \frac{W}{m^2 \cdot K}$	$h_{ng} := \frac{Nu_{ng} \cdot k_{ng}}{D_i} = 2.518 \times 10^5 \cdot \frac{W}{m^2 \cdot K}$
Change in temp	$\Delta T_{ex} := T_1 - T_2 = 373K$	$\Delta T_{full} := 45K$

Thermal Resistance Model: Demonstrator with exhaust

Convection $$R_{convex} := \frac{1}{h_{ex} \cdot A_{Ldemo}} = 0.244 \frac{K}{W}$$

Conduction $$R_{condex} := \frac{\ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot L_{demo} \cdot k_{ex}} = 0.604 \frac{K}{W}$$

Number of heat pipes $$n_{hp} := 14$$

Number of fins $$n_{fin} := 18$$

Heat Pipe Resistance $$R_{hp} := 3 \frac{K}{W}$$

Fin resistance $$R_{fin} := 12.293 \frac{K}{W}$$

$$R_{fintot} := \frac{R_{fin}}{n_{fin}} = 0.683 \frac{K}{W}$$

Total HP Resistance $$R_{hptotex} := \frac{1}{\frac{n_{hp}}{R_{hp}} + \frac{n_{hp}}{R_{fintot}}} = 0.04 \frac{K}{W}$$

Total Resistance $$R_{ex} := R_{convex} + R_{condex} + R_{hptotex} = 0.888 \frac{K}{W}$$

$$R_{nohp} := R_{convex} + R_{condex} = 0.848 \frac{K}{W}$$

Thermal Resistance Model: Full size model with exhaust

Convection $$R_{convfull} := \frac{1}{h_{ex} \cdot A_{Lfull}} = 5.036 \times 10^{-3} \cdot \frac{K}{W}$$

Conduction $$R_{condfull} := \frac{\ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot L_{full} \cdot k_{ex}} = 0.012 \frac{K}{W}$$

Number of heat pipes $$n_{hpfull} := 419$$

Number of fins $$n_{fin} = 18$$

Heat Pipe Resistance $$R_{hp} = 3 \frac{K}{W}$$

Fin resistance $$R_{fin} = 12.293 \frac{K}{W}$$

$$R_{fintot} = 0.683\frac{K}{W}$$

Total HP Resistance
$$R_{hptotfull} := \frac{1}{\frac{n_{hpfull}}{R_{hp}} + \frac{n_{hpfull}}{R_{fintot}}} = 1.328\times 10^{-3}\cdot\frac{K}{W}$$

Total Resistance
$$R_{full} := R_{convfull} + R_{condfull} + R_{hptotfull} = 0.019\frac{K}{W}$$

Ratio between demonstrator and full size model
$$R := \frac{R_{ex}}{R_{full}} = 47.155$$

<u>Thermal Resistance Model: Demonstrator with natural gas</u>

Convection
$$R_{convng} := \frac{1}{h_{ng}\cdot A_{Ldemo}} = 7.149\times 10^{-5}\cdot\frac{K}{W}$$

Convection
$$R_{condng} := \frac{\ln\left(\frac{r_o}{r_i}\right)}{2\cdot\pi\cdot L_{demo}\cdot k_{ng}} = 0.587\frac{K}{W}$$

Number of heat pipes:
$$n_{hpng1} := n_{hp} = 14$$

Number of fins:
$$n_{fin} = 18$$

Heat Pipe Resistance
$$R_{hp} = 3\cdot\frac{K}{W}$$

Fin resistance
$$R_{fin} = 12.293\frac{K}{W}$$

$$R_{fintot} = 0.683\frac{K}{W}$$

Total HP Resistance
$$R_{hptotng} := \frac{1}{\frac{n_{hpng1}}{R_{hp}} + \frac{n_{hpng1}}{R_{fintot}}} = 0.04\frac{K}{W}$$

Total Resistance
$$R_{ng1} := R_{convng} + R_{condng} + R_{hptotng} = 0.627\frac{K}{W}$$

RESISTANCE FOR FULL MODEL WITH NATURAL GAS
$$R_{ngfinal} := \frac{R_{ng1}}{R} = 0.013\frac{K}{W}$$

With 42 meters in length and 275 heat pipes, the thermal resistance of our model would be 0.013 K/W

G.3 Cradle Heat Pipe Design Analysis

Unit definition	$kJ := 1000J$	
Inside/Outside Diameter of natural gas Tube	$D_o := 15.875mm = 0.625 \cdot in$	$D_i := 13.386mm = 0.527 \cdot in$
Wall Thickness	$T_p := 1.27mm = 0.05 \cdot in$	
Total Length of Gas Pipe	$L_i := 64.008m = 210 \cdot ft$	Forty-two tubes at 5 ft for 210ft total. 3 rows of 14 pipes stacked.
Breakdown of Design	$L_{tube} := 1.524m = 5 \cdot ft$	
	$W_{tube} := .2286m = 0.75 \cdot ft$	$H_{tube} := 2.32932m = 7.642 \cdot ft$
Surface area of inside of pipe	$A_{in} := \pi \cdot D_i \cdot L_i = 2.692m^2$	
inner/outer pipe radius	$r_i := \frac{D_i}{2} = 6.693mm$	$r_o := \frac{D_o}{2} = 7.938mm$
Temperature of Natural Gas In/Out	$T_{gasin} := 377.928K$	$T_{gasout} := 333.15K$
Flow Rate of Gas	$Flow_{gas} := 0.005459\frac{m^3}{s}$	
Gauge pressure of water in heat pipe	$P_g := 20.265kPa = 0.02 \cdot MPa$	
Standard Pressure	$P_s := 101.325kPa$	
Absolute pressure of water in heat pipe	$P_a := P_s + P_g = 0.122 \cdot MPa$	
Gas Constant, Methane	$R := 518.3\frac{J}{kg \cdot K}$	

http://www.engineeringtoolbox.com/individual-universal-gas-constant-d_588.html

Standard volumetric flow rate of gas	$V_{scfm} := 870\frac{ft^3}{min} = 0.411\frac{m^3}{s}$	
Standard Pressure	$P_{std} := 101325Pa = 0.101 \cdot MPa$	
Standard Temperature	$T_{std} := 273.15K$	
Natural Gas Volume Flow	$\boxed{V_{actual} := \frac{\left(P_{std} \cdot V_{scfm} \cdot T_{gasin}\right)}{P_a \cdot T_{std}} = 0.473\frac{m^3}{s}}$	

Gauge Pressure from competitor cooler $P := 1515psi = 10.446MPa$

Absolute Pressure from competitor cooler $P_{ap} := P + P_s = 10.547 \cdot MPa$

Universal Gas Constant $R := 8.3144126 \frac{J}{mole \cdot K}$

http://en.wikipedia.org/wiki/Gas_constant

Molecular Weight of Methane $M := 16.044 \frac{gm}{mol}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Density of Compressed Natural Gas at Enter (Calc): $d_{in} := \frac{P \cdot M}{R \cdot T_{gasin}} = 53.334 \frac{kg}{m^3}$

Density of Compressed Natural Gas at Exit (Calc): $d_{out} := \frac{P \cdot M}{R \cdot T_{gasout}} = 60.502 \frac{kg}{m^3}$

Chosen Density of CNG $\rho_g := d_{in}$

Mass Flow Rate $mflow_{NG} := \rho_g \cdot V_{actual} = 25.249 \frac{kg}{s}$

Specific Heat of Gas $C_{pNG} := 2.226 \frac{kJ}{kg \cdot K}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Viscosity of natural gas $\mu_{NG} := 0.000110 poise = 1.1 \times 10^{-5} \frac{kg}{m \cdot s}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Thermal Conductivity of Gas $k_{gas} := 0.035 \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Thermal Conductivity of Stainless Steel $k_m := 30 \cdot \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html

Thermal Conductivity of Copper (estimated) $k_{cradle} := 401 \cdot \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html

Cradle Specification

$L_{cradle} := 63.5mm = 2.5 \cdot in$ $H_{cradle} := 63.5mm$

$W_{cradle} := 38.1mm = 1.5 \cdot in$ $T_{cradle} := 38.1mm$

$$A_{cradle} := \left(2L_{cradle} \cdot H_{cradle}\right) + \left(2H_{cradle} \cdot W_{cradle}\right) + \left(2L_{cradle} \cdot W_{cradle}\right) = 0.018m^2$$

Prandtl Number $Pr := \frac{\mu_{NG} \cdot C_{pNG}}{k_{gas}} = 0.7$

Reynolds Number $Re_D := \frac{4 \cdot mflow_{NG}}{\pi \cdot D_i \cdot \mu_{NG}} = 2.183 \times 10^8$ Turbulent if > 2300

Masking Effect

distance along pipe $S_T := 1.524m$ 1.524 meters = 5 ft

distance across pipes $S_L := 1.067m$ 1.067 mm = 3.5 ft

tube diameter $D := 5.875mm$

dimensionless longitudinal distance $s_l := \frac{S_L}{D} = 181.617$

dimensionless tangential distance $s_t := \frac{S_T}{D} = 259.404$

Nusselt Number $Nu := 0.023 Re_D^{0.8} \cdot Pr^{0.33} = 9.59 \times 10^4$

Heat Transfer Coefficient of Gas $h_{NG} := \frac{k_{gas}}{D_i} \cdot Nu = 2.507 \times 10^5 \cdot \frac{W}{m^2 K}$

convective resistance from NG to pipe $R_{ng} := \frac{1}{h_{NG} \cdot \pi \cdot D_i \cdot L_i} = 1.482 \times 10^{-6} \cdot \frac{K}{W}$

conductive resistance through the pipe $R_{pipe} := \frac{\ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot k_m \cdot L_i} = 1.413 \times 10^{-5} \cdot \frac{K}{W}$

conductive resistance through cradle $R_{cradle} := \frac{T_{cradle}}{A_{cradle} \cdot k_{cradle}} = 5.355 \times 10^{-3} \cdot \frac{K}{W}$

http://en.wikipedia.org/wiki/Thermal_conductivity

FIN DESIGN

fin thickness $thickness := .4mm$ $t := thickness$ assume square fins

$.4mm = 0.016in$ $\frac{1}{16} in = 0.063in$

gap between fins $gap := 2.5mm$

$2mm = 0.079in$ $\frac{1}{8} in = 0.125in$

thermal conductivity of air $k_{air} := 0.0278 \frac{W}{m \cdot K}$ Taken at a temperature of 50 C

http://www.engineeringtoolbox.com/air-properties-d_156.html

cross sectional area of fin $A_c(w_f) := w_f \cdot t$

specific heat of air $C_{pNGair} := 1.005 \frac{kJ}{kg \cdot K}$

http://www.engineeringtoolbox.com/air-properties-d_156.html

Viscosity of air $\mu_{air} := 2.02910^{-5} \frac{kg}{m \cdot s}$ Taken at a temperature of 325K

http://www.engineeringtoolbox.com/air-absolute-kinematic-viscosity-d_601.html

density of air $\rho_{air} := 1.097 \frac{kg}{m^3}$ Taken at a temperature of 50 C

http://www.engineeringtoolbox.com/air-properties-d_156.html

volumetric flow rate of air $uncorrectedflow_{air} := 59933 \frac{ft^3}{min} = 28.285 \frac{m^3}{s}$

flow rate correction factor (based on heat load) $flow_{correction} := \frac{176}{1969 + 483 + 650 + 119 + 419 + 176} = 0.046$

flow rate correction factor (based on area of flow) $Cr_{area} := \frac{14}{92 + 56 + 65 + 23 + 56 + 14} = 0.046$

corrected volumetric flow rate of air based on area of flow $flow_{air} := uncorrectedflow_{air} \cdot Cr_{area} = 1.294 \frac{m^3}{s}$

Volumetric flow rate ratio $Flowrate_{ratio} := \frac{flow_{air}}{uncorrectedflow_{air}} = 0.046$

Uncorrected fan power $fanpower_{uncorrected} := 57.2hp = 42.654 kW$

Corrected fan power $\boxed{Fan_{power} := fanpower_{uncorrected} \cdot Flowrate_{ratio} = 1.951 \cdot kW}$

$\boxed{Fan_{power} = 2.617 \cdot hp}$

Fan Size Estimate

From competitor After Cooler Quote

$L_{fan} := 63.75in = 5.312 \cdot ft$

$H_{fan} := 43.125in = 3.594 ft$

$W_{fan} := 13.5in = 1.125 ft$

mass flow rate of air $mflow_{air} := \rho_{air} \cdot flow_{air} = 1.42 \frac{kg}{s}$

ambient air temp

$$T_{inf} := (273.15 + 50)K = 323.15K$$

Prandtl number $$Pr_{air} := \frac{C_{pNGair} \cdot \mu_{air}}{k_{air}} = 0.734$$

average gas temp at surface of pipe $$T_{surf} := \frac{T_{gasin} + T_{gasout}}{2} = 355.539K$$

$$A_{air}(w_f, n) := n \cdot (w_f \cdot gap)$$

air contact perimeter $$p_{air}(w_f, n) := n \cdot (2 \cdot w_f + gap)$$

hydraulic diameter $$D_{hair}(w_f, n) := 4 \cdot \frac{A_{air}(w_f, n)}{p_{air}(w_f, n)}$$

air contact surface area

Reynolds number $$Re_{air}(w_f, n) := \frac{4 \cdot mflow_{air}}{\pi \cdot D_{hair}(w_f, n) \cdot \mu_{air}}$$ turbulent flow

Correction/arrangement factor $$F_a := 1 + 0.1 \cdot s_l + \frac{0.34}{s_t} = 19.163$$

Nusselt number of Air with Correction Factor $$Nu_{air}(w_f, n) := 0.023 Re_{air}(w_f, n)^{0.8} \cdot Pr_{air}^{0.33} \cdot F_a$$

Convective heat transfer coefficient of air $$h_{air}(w_f, n) := \frac{k_{air}}{p_{air}(w_f, n)} \cdot Nu_{air}(w_f, n)$$

perimeter of fin $$p_{fin}(w_f) := 2 \cdot w_f + 2 \cdot t$$

$$m_{fin}(w_f, n) := \sqrt{\frac{2 \cdot h_{air}(w_f, n)}{k_{air} \cdot t}}$$

corrected fin length $$L_c(L_f) := L_f + \frac{t}{2}$$

thermal resistance of fin with convection off end $$R_{fin}(w_f, L_f, n) := \frac{1}{\sqrt{h_{air}(w_f, n) \cdot p_{fin}(w_f) \cdot k_{air} \cdot A_c(w_f)} \cdot \tanh\left(m_{fin}(w_f, n) \cdot L_c(L_f)\right)}$$

for fin with convection off end, p695

Rate of heat transfer out of natural gas required $$Q_{ratemax} := mflow_{NG} \cdot C_{pNG} \cdot (T_{gasin} - T_{gasout}) = 2.517 \cdot MW$$

Maximum resistance to achieve required heat transfer $$R_{tmax} := \frac{T_{surf} - T_{inf}}{Q_{ratemax}} = 1.287 \times 10^{-5} \cdot \frac{K}{W}$$

$$AA := 10 \cdot R_{tmax} = 1.287 \times 10^{-4} \cdot \frac{K}{W}$$

$BB := R_{ng} + R_{pipe} + R_{cradle} = 5.371\times 10^{-3}\cdot\frac{K}{W}$

Using These to check

$AA - BB = -5.242\times 10^{-3}\cdot\frac{K}{W}$

number of heat pipes $\boxed{n_{hp} := 419}$

16 heat pipes per gas pipe

with 15.5 gas pipes

resistance of one heat pipe $R_{ihp} := 0.18\frac{K}{W}$

maximum allowable total fin resistance $R_{finmax} := \left[42\cdot R_{tmax} - \left(R_{ng} + R_{pipe} + R_{cradle}\right)\right]\cdot n_{hp} - R_{ihp} = -2.204\frac{K}{W}$

resistance of single fin $R_{ifin}(n) := n\cdot R_{finmax}$

fin width $\boxed{w := 2.5cm}$ 127 mm = 5 in

fin length $L := 2cm$ resistance independent of L

number of fins per bank $\boxed{n_{fin} := 17}$

maximum resistance of fins $\boxed{R_{ifin}\left(n_{fin}\right) = -37.467\frac{K}{W}}$

calculated resistance per fin, make less than Rifin $\boxed{R_{fin}\left(w, L, n_{fin}\right) = 93.246\frac{K}{W}}$

logic statement to help optimize design and reduce potential for human error

$current := if\left(R_{ifin}\left(n_{fin}\right) < R_{fin}\left(w, L, n_{fin}\right), \text{"more"}, \text{"less"}\right) = \text{"more"}$

$smaller := if\left(R_{ifin}\left(n_{fin} - 1\right) < R_{fin}\left(w, L, n_{fin} - 1\right), \text{"smallest"}, \text{"fewer"}\right) = \text{"smallest"}$

$if(current = \text{"less"} \wedge smaller = \text{"smallest"}, \text{"end"}, \text{"iterate"}) = \text{"iterate"}$

Required condenser region length $condenser_{length} := t\cdot n_{fin} + gap\cdot\left(n_{fin} - 1\right) = 46.8\,mm$ 50mm max

Design Footprint $Footprint := \left(L_{tube}\right)\cdot\left(W_{fan} + W_{tube}\right) = 0.871m^2$

Design Volume $Volume := Footprint\cdot H_{tube} = 2.029m^3$

G.4 Demonstrator Analytical Analysis

Demonstrator Experiment Analysis

Unit definition $kJ := 1000J$

Number of Heat Pipes $\boxed{n_{hp} := 14}$

Inside/Outside Diameter of natural gas Tube $D_o := 15.875mm = 0.625 \cdot in$ $D_i := 13.386mm = 0.527 \cdot in$

Wall Thickness $T_p := 1.27mm = 0.05 \cdot in$

Total Length of Gas Pipe $L_i := 104in = 8.667 \cdot ft$ 2 tubes at 4.3 ft for 8.6ft total. 2rows.

Breakdown of Design $L_{tube} := 52in = 4.333 \cdot ft$

$W_{tube} := 9.25in = 0.771 \cdot ft$ $H_{tube} := 5.91in = 0.492 \cdot ft$

Surface area of inside of pipe $A_{in} := \pi \cdot D_i \cdot L_i = 0.111 m^2$

inner/outer pipe radius $r_i := \frac{D_i}{2} = 6.693mm$ $r_o := \frac{D_o}{2} = 7.938mm$

Temperature of Engine Exhaust In/Out $T_{gasin} := 693.9K$ $T_{gasout} := 297.6K$

Flow Rate of Gas $Flow_{gas} := 0.002\frac{m^3}{s}$

Gague pressure of water in heat pipe $P_g := 20.265kPa = 0.02 \cdot MPa$

Standard Pressure $P_s := 101.325kPa$

Absolute pressure of water in heat pipe $P_a := P_s + P_g = 0.122 \cdot MPa$

Gas Constant, Methane $R := 518.3\frac{J}{kg \cdot K}$

http://www.engineeringtoolbox.com/individual-universal-gas-constant-d_588.html

Standard volumetric flow rate of gas $V_{scfm} := 870\frac{ft^3}{min} = 0.411 \cdot \frac{m^3}{s}$

Standard Pressure $P_{std} := 101325Pa = 0.101 \cdot MPa$

Standard Temperature $T_{std} := 273.15K$

Natural Gas Volume Flow $\boxed{V_{actual} := \frac{(P_{std} \cdot V_{scfm} \cdot T_{gasin})}{P_a \cdot T_{std}} = 0.869\frac{m^3}{s}}$

Gague Pressure from AXH cooler $P := 1515psi = 10.446MPa$

Absolute Pressure from AXH cooler $P_{ap} := P + P_s = 10.547 \cdot MPa$

Universal Gas Constant $R := 8.3144126 \frac{J}{mole \cdot K}$

http://en.wikipedia.org/wiki/Gas_constant

Molecular Weight of Engine Exhaust $M := 30 \frac{gm}{mol}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Density of Compressed Natural Gas at Enter (Calc): $d_{in} := \frac{P \cdot M}{R \cdot T_{gasin}} = 54.316 \frac{kg}{m^3}$

Density of Compressed Natural Gas at Exit (Calc): $d_{out} := \frac{P \cdot M}{R \cdot T_{gasout}} = 126.645 \frac{kg}{m^3}$

Chosen Density of CNG $\rho_g := d_{out}$

Mass Flow Rate $mflow_{NG} := \rho_g \cdot V_{actual} = 110.082 \frac{kg}{s}$

Specific Heat of Gas $C_{pNG} := 1.327 \frac{kJ}{kg \cdot K}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Viscosity of engine exhaust $\mu_{NG} := 2.17 \cdot 10^{-5} \frac{kg}{m \cdot s}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Thermal Conductivity of engine exhaust $k_{gas} := 0.034 \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/methane-d_1420.html

Thermal Conductivity of Stainless Steel $k_m := 30 \cdot \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html

Thermal Conductivity of Copper (estimated) $k_{cradle} := 401 \cdot \frac{W}{m \cdot K}$

http://www.engineeringtoolbox.com/thermal-conductivity-d_429.html

Cradle Specification

$L_{cradle} := 60.96mm = 2.4 \cdot in$ $H_{cradle} := 45.72mm$

$W_{cradle} := 35.56mm = 1.4 \cdot in$ $T_{cradle} := 35.56mm$

$$A_{cradle} := (2L_{cradle} \cdot H_{cradle}) + (2H_{cradle} \cdot W_{cradle}) + (2L_{cradle} \cdot W_{cradle}) = 0.013m^2$$

Prandtl Number $Pr := \frac{\mu_{NG} \cdot C_{pNG}}{k_{gas}} = 0.847$

Reynolds Number $Re_D := \frac{4 \cdot mflow_{NG}}{\pi \cdot D_i \cdot \mu_{NG}} = 4.825 \times 10^8$ Turbulent if > 2300

Masking Effect

distance along pipe $S_T := L_{tube}$

distance across pipes $S_L := W_{tube}$

tube diameter $D := 5.875mm$

dimensionless longitudinal distance $s_l := \frac{S_L}{D} = 39.991$

dimensionless tangential distance $s_t := \frac{S_T}{D} = 224.817$

Nusselt Number $Nu := 0.023 Re_D^{0.8} \cdot Pr^{0.33} = 1.926 \times 10^5$

Heat Transfer Coefficient of Gas $h_{NG} := \frac{k_{gas}}{D_i} \cdot Nu = 4.893 \times 10^5 \cdot \frac{W}{m^2 K}$

convective resistance from NG to pipe $R_{ng} := \frac{1}{h_{NG} \cdot \pi \cdot D_i \cdot L_i} = 1.84 \times 10^{-5} \cdot \frac{K}{W}$

conductive resistance through the pipe $R_{pipe} := \frac{\ln\left(\frac{r_o}{r_i}\right)}{2 \cdot \pi \cdot k_m \cdot L_i} = 3.425 \times 10^{-4} \cdot \frac{K}{W}$

conductive resistance through cradle $R_{cradle} := \frac{T_{cradle}}{A_{cradle} \cdot k_{cradle}} = 6.738 \times 10^{-3} \cdot \frac{K}{W}$

http://en.wikipedia.org/wiki/Thermal_conductivity

FIN DESIGN

fin thickness $thickness := .508mm$ $t := thickness$ assume square fins

gap between fins $gap := 2.5mm$

thermal conductivity of air $k_{air} := 0.0278 \frac{W}{m \cdot K}$ Taken at a temperature of 50 C

http://www.engineeringtoolbox.com/air-properties-d_156.html

cross sectional area of fin $w := 25 \cdot mm$ $A_c := w \cdot t$

specific heat of air $C_{pNGair} := 1.005 \frac{kJ}{kg \cdot K}$

http://www.engineeringtoolbox.com/air-properties-d_156.html

Viscosity of air $\mu_{air} := 2.029 10^{-5} \frac{kg}{m \cdot s}$ Taken at a temperature of 325K

http://www.engineeringtoolbox.com/air-absolute-kinematic-viscosity-d_601.html

density of air $\rho_{air} := 1.097 \frac{kg}{m^3}$ Taken at a temperature of 50 C

http://www.engineeringtoolbox.com/air-properties-d_156.html

volumetric flow rate of air $uncorrectedflow_{air} := 59933 \frac{ft^3}{min} = 28.285 \frac{m^3}{s}$

flow rate correction factor based on heat load $flow_{correction} := \frac{176}{1969 + 483 + 650 + 119 + 419 + 176} = 0.046$

flow rate correction factor based on area of flow $Cr_{area} := \frac{14}{92 + 56 + 65 + 23 + 56 + 14} = 0.046$

corrected volumetric flow rate of air based on area of flow $flow_{air} := uncorrectedflow_{air} \cdot Cr_{area} = 1.294 \frac{m^3}{s}$

Volumetric flow rate ratio $Flowrate_{ratio} := \frac{flow_{air}}{uncorrectedflow_{air}} = 0.046$

Uncorrected fan power $fanpower_{uncorrected} := 420W = 0.42 \cdot kW$

Corrected fan power $\boxed{Fan_{power} := fanpower_{uncorrected} \cdot Flowrate_{ratio} = 0.019 kW}$

$\boxed{Fan_{power} = 0.026 hp}$

Fan Used for Testing Size Estimate

$L_{fan} := 12in = 1 \cdot ft$

$H_{fan} := 36in = 3 \cdot ft$

$W_{fan} := 36in = 3 \cdot ft$

mass flow rate of air $mflow_{air} := \rho_{air} \cdot flow_{air} = 1.42 \frac{kg}{s}$

ambient air temp $T_{inf} := (273.15 + 50)K = 323.15K$

Prandtl number $Pr_{air} := \frac{C_{pNGair} \cdot \mu_{air}}{k_{air}} = 0.734$

average gas temp at surface of pipe $T_{surf} := \frac{T_{gasin} + T_{gasout}}{2} = 495.75K$

air contact surface area $A_{air}(n) := n \cdot (w \cdot gap)$

air contact perimeter $p_{air}(n) := n \cdot (2 \cdot w + gap)$

hydraulic diameter $D_{hair}(n) := 4 \cdot \frac{A_{air}(n)}{p_{air}(n)}$

Reynolds number $Re_{air}(n) := \frac{4 \cdot mflow_{air}}{\pi \cdot D_{hair}(n) \cdot \mu_{air}}$ turbulent flow

Correction/arrangement factor $F_a := 1 + 0.1 \cdot s_l + \frac{0.34}{s_t} = 5.001$

Nusselt number of Air with Correction Factor $Nu_{air}(n) := 0.023 Re_{air}(n)^{0.8} \cdot Pr_{air}^{0.33} \cdot F_a$

Convective heat transfer coefficient of air $h_{air}(n) := \frac{k_{air}}{p_{air}(n)} \cdot Nu_{air}(n)$

perimeter of fin $w := 25 \cdot mm$ $L := 20mm$ $p_{fin} := 2 \cdot w + 2 \cdot L$

$$m_{fin}(n) := \sqrt{\frac{2 \cdot h_{air}(n)}{k_{air} \cdot t}}$$

thermal resistance of fin with convection off end $R_{fin}(w, L, n) := \frac{1}{\sqrt{h_{air}(n) \cdot p_{fin} \cdot k_{air} \cdot A_c} \cdot 2 \cdot \tanh\left(m_{fin}(n) \cdot L\right)}$

for fin with convection off end, p695

Rate of heat transfer out of natural gas required $Q_{ratemax} := \left[mflow_{NG} \cdot C_{pNG} \cdot \left(T_{gasin} - T_{gasout}\right)\right] = 57.891 \cdot MW$

Maximum resisance to achieve required heat transfer $R_{tmax} := \frac{T_{surf} - T_{inf}}{Q_{ratemax}} = 2.981 \times 10^{-6} \cdot \frac{K}{W}$

number of heat pipes $\boxed{n_{hp} := 14}$

resistance of one heat pipe (average) $R_{ihp} := 3.236 \frac{K}{W}$

maximum allowable total fin resistance $R_{finmax} := -1\left[\left[2 \cdot R_{tmax} - \left(R_{ng} + R_{pipe} + R_{cradle}\right)\right] \cdot n_{hp} - R_{ihp}\right] = 3.335 \frac{K}{W}$

resistance of single fin $R_{ifin}(n) := n \cdot R_{finmax}$

fin width $\boxed{w := 25 \cdot mm}$ 25 mm = .984 in fin length $L := 20mm$ 20 mm = .787 in

number of fins per bank $n_{fin} := 18$

Maximum resistance of fins $R_{ifin}(n_{fin}) = 60.035\frac{K}{W}$

calculated resistance per fin, make less than Rifin $R_{fin}(w, L, n_{fin}) = 62.61\frac{K}{W}$

logic statement to help optimize design and reduce potential for human error

$$current := if\left(R_{ifin}(n_{fin}) < R_{fin}(w, L, n_{fin}), \text{"more"}, \text{"less"}\right) = \text{"more"}$$

$$smaller := if\left(R_{ifin}(n_{fin} - 1) < R_{fin}(w, L, n_{fin} - 1), \text{"smallest"}, \text{"fewer"}\right) = \text{"smallest"}$$

$$if(current = \text{"less"} \wedge smaller = \text{"smallest"}, \text{"end"}, \text{"iterate"}) = \text{"iterate"}$$

Required condeser region length $condenser_{length} := t \cdot n_{fin} + gap \cdot (n_{fin} - 1) = 51.644mm$

Design Footprint $Footprint := (L_{tube}) \cdot (W_{fan} + W_{tube}) = 1.518m^2$

Design Volume $Volume := Footprint \cdot H_{tube} = 0.228m^3$

Appendix H - Quotes

H.1 Quote - Current Cooler Competitor

[redacted]

February 14, 2012

Sent Via e-mail
anelson@oscomp-systems.com
(4-Pages Total)

[redacted]

Attn: Andrew Nelson

Re: Case 1 Redesign – Electric Motor Drive; Single Stage
[redacted] Proposal 120082A / Model 36VVF

Dear Andrew:

Following is a description and list of equipment for AXH air-coolers' proposal for your referenced inquiry.

[redacted] **Proposal 120082A; Case 1 – Redesign / Rating Basis 122°F, 2500ft Elevation** [redacted] Model 36VVF forced draft, horizontal air discharge, vertical coils, and motor driven per the attached specification sheet and including:

1932# MAWP AC Section, rated for 870SCFM, cooling from 220.6°F to 140°F,
316L stainless steel headers,
316L stainless steel welded tubes,
[redacted] standard paint system for structure and headers,
Direct Drive by (1) 2HP, 1200RPM, 460/60/3, TEFC, Class I, Div. 2, Grp. D motor,
One hour charted hydro-test,
Automatically controlled louvers by (1) Durastroke Model 60 actuator,
Temperature control by (1) Kimray T12 with thermowell,
Rigid metal bug screens with self tapping screws.

Cooler Net Price…………………$24,269/ea

Pricing: Quoted pricing is valid for 30 days and firm for shipments through the end of August 2012.

Delivery: Please confirm delivery requirements at time of purchase. Current lead time is approximately 8-10 weeks after receipt of order. Delivery is also subject to the availability of raw materials.

Terms & Conditions: 50% at time of order placement with remaining 50% prior to shipment of completed cooler. [redacted]

Sincerely,

[redacted]

	[redacted]						**Proposal / Job No.**	120082A
							Date	2/14/2012
							Page	1 OF 1
1	*Purchaser*	OSCOMP SYSTEMS			*Ultimate User*		OSCOMP SYSTEMS	
2	*Inquiry / PO#*	CASE 1 - REDESIGN			*Destination*			
3	*# Units*	1	*Model*	36VVF	*Reference*		ELECTRIC MOTOR DRIVE; SINGLE STAGE	
4	*Assembly*	PACKAGED	*Draft*	FORCED	*Overall Size (WxL), Ft*		*Est Wt*	1500 LBS

THERMAL & MECHANICAL DESIGN

5	**Service**	AC
6	*Flow*	870SCFM
7	*Fluid*	.8302SPGR
8	*Temp. In / Out, °F*	220.6 / 140.0
9	*Pressure, PSI*	1515PSIG
10	*Pressure Drop, PSI*	0.5
11	*Heat Load, BTU/HR*	176335
12	*True LMTD*	38.4
13	*Overall Rate, U*	84.7
14	*Fouling Factor*	.0020
15	*Surface, Tube / Total, Sq Ft*	58 / 914
16	**Sections, #**	(1)
17	*Design Temp, °F Max / Min*	350 / -20
18	*MWP / Test Press, PSIG*	1932 / 2588
19	*Pass Arrangement*	CROSSFLOW
20	*# Tube Rows*	3
21	*# Tube Passes*	5
22	**Tubes, OD x BWG**	5/8X18
23	*Material*	A249TP316LSS
24	*# Per Section / Length, Ft*	74 / 5
25	*Turbulators*	
26	*Accelerators*	
27	**Fins, Type**	HI-EFF
28	*Material*	AL
29	**Nozzles, Rating / Type**	1500RF
30	*Material*	316LSS
31	*#-Inlets / Size In*	(1) 2
32	*#-Oulets / Size In*	(1) 2
33	**Headers, Type**	BOX
34	*Material*	316LSS
35	*Corrosion Allow, In*	
36	*Grooved Tubesheet*	DBL
37	*Plugs, Type*	SHOULDER
38	*Plugs Material*	304LSS
39	*PWHT*	
40	*ASME Code & Nat'l Board*	YES
41	*CRN*	
42	**Add'l Specs & Options**	
43	*API*	
44	*Louvers / Hail Screen*	AUTO
45	**Inspection / NDT**	

FX= *100% X-Ray of all header seam, attachment & nozzle butt welds.* ***SX=*** *Spot X-Ray of 1 long seam & 1 end closure, per header*
BX= *100% X-Ray of all nozzle butt welds.* ***UT =*** *100% UT of all header seam, attachment & nozzle butt welds.* ***H*** *= Hardness testing.*

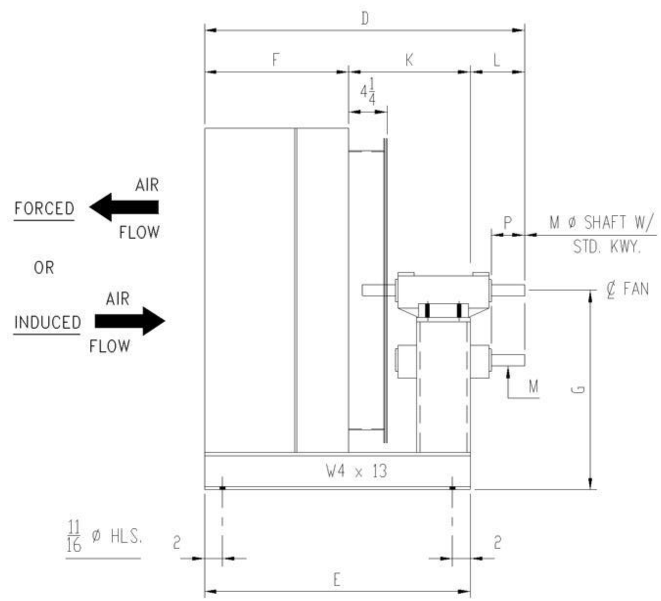
	AIR-SIDE PERFORMANCE		**FAN DATA**		**DRIVER DATA**	**STRUCTURAL**
46	*Ambient Air Temp, In °F*	122	*Fan(s)*	(1) MULTI-WING 5ZL	*Type*	*Guards*
47	*Elevation, Ft*	2500	*Blade Material*	PAG	DIRECT DRIVE BY (1) 2HP, 1200RPM, 460/60/3, TEFC, VFD COMP. MOTOR	
48	*Air Flow, SCFM*	9,276	*HP / Fan*	1.5		
49	*Outlet Air Temp, °F*	139	*Dia, In / # Blades*	36 / 3		
50	*Min Air Temp, °F*	-20	*RPM*	1165		
51			*Tipspeed, FPM*	10979		
52	**Est. Noise Data:**	75 dBA @1m	*Pitch, Deg*	35		

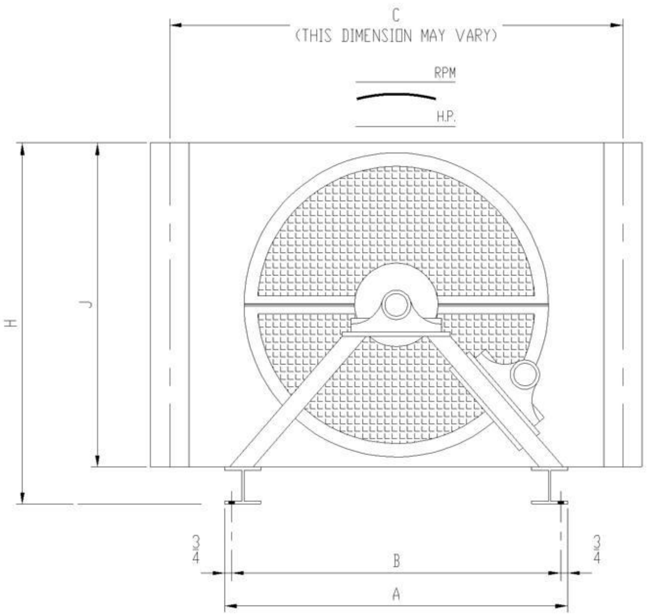
53	**Additional Info.**
54	
55	
56	
57	
58	

MODEL	A	B	C	D	E	F	G	H	J	K	L	M	P
24W	26	24½	39¾	32½	26½	16	17⅝	31⅛	27	10½	6	1³⁄₁₆	4⅛
30W	38	36½	51¾	32½	26½	16	20⅝	37⅛	33	10½	6	1³⁄₁₆	4⅛
36W	50	48½	63¾	37½	31½	18	23⅝	43⅛	39	13½	6	1⁷⁄₁₆	3¹¹⁄₁₆
42W	62	60½	75¾	41½	35½	22	27⅜	50⅝	46½	13½	6	1⁷⁄₁₆	3¹¹⁄₁₆
48W	62	60½	75¾	41½	35½	22	35⅝	67⅛	63	13½	6	1⁷⁄₁₆	3¹¹⁄₁₆

PLEASE INDICATE THE FOLLOWING INFORMATION ON THIS OUTLINE DRAWING IN ORDER THAT A DETAILED CERTIFIED G/A DRAWING CAN BE COMPLETED. NOTE: IF FORMAL G/A DRAWING FOR "APPROVAL" IS ALSO REQUIRED FABRICATION MAY BE DELAYED ACCORDINGLY, OTHERWISE WE WILL PROCEED ON A "RELEASED" FOR FABRICATION BASIS WITHOUT DELAY.

1. INDICATE DESIRED GENERAL ORIENTATION OF BOTH SECTIONS AND NOZZLES.
2. INDICATE FLANGE SIZES, TYPE (PIPE STUB, RF, RTJ) & RATINGS, IF DIFFERENT FROM THE SPECIFICATION SHEET.
3. INDICATE SECTION MWP, IF DIFFERENT FROM THE SPECIFICATION SHEET.
4. INDICATE FANS SPEED (RPM), IF DIFFERENT FROM THE SPECIFICATION SHEET.





[redacted]
SALES DRAWINGS
MODELS 24W THRU 48W

DRAWN: LES	CHECKED M.B.	DRAWING NUMBER W-1	
DATE: 03-22-02	SCALE: NONE		REV 0

REV	REVISION DESCRIPTION	BY	CK	APP	DATE

H.2 Aavid Thermalloy Quote - Heat Pipes

AAVID THERMALLOY

Quote Number	Date	Page
3142012	March 14, 2012	1 of 1

Quoted by:
Michael Beliveau
Phone: (603) 223-1810
e-mail: beliveaum@aavid.com

70 Commercial Street, Suite 200
Concord, NH 03301 USA
(603) 224-9988

Customer:
Worcester Polytechnic Institute
Attn: Christopher O'Brien
100 Institute Rd
Worcester, MA 01609

Contact:
Christopher O'Brien
cjobrien@wpi.edu
781-738-0862

Program	Quote Expiration Date
WPI MQP – Heatpipe with Fins	March 23, 2012

WE ARE PLEASED TO SUBMIT THE FOLLOWING QUOTATION FOR YOUR APPROVAL

Price(s) are quoted in USD

Part Number	Description	Qty	Price	U/M
Heatpipe with Fins	CUSTOM – 8x150MM HP WITH 20x BONDED FINS * .020" FINS BASED ON CURRENT STOCK. * FINS BONDED TO HP WITH THERMAL EPOXY. * THE ABOVE PRICES ARE QUOTED FOB: LACONIA, NH. * THE ESTIMATED LEAD TIME IS 3 WEEKS ARO. * TOOLING IS INCLUDED IN PRICE. * THIS PRODUCT IS NON-CANCELABLE AND NON-RETURNABLE. *	15	$54.09	EACH

STANDARD TOLERANCES:
* Standard commercial extrusion tolerances apply to extruded dimensions (by Aluminum Association of America).
* Low usage extrusion may require a minimum order amount or a set-up charge dependent on material availability at time of order.
* Aavid's typical shipping tolerances may be up to +/- 10%; primarily for custom product.
* Unless otherwise specified, visual criteria per Aavid Thermalloy specification QC-500.

H.3 ACT Quote - Heat Pipes

ADVANCED COOLING TECHNOLOGIES, INC.

ISO9001 & AS9100 Certified

Innovations in Action

Prepared for:
Christopher O'Brien
Worcester Polytechnic Institute '12
Mechanical Engineering
cjobrien@WPI.EDU

Quotation

Number: 20-0922
Date: 3/12/12
Customer number: 00393
Validity period: 30 days from 3/12/12
Division: EP

We deliver according to the following conditions:

Terms of payment:	NT30
Terms of delivery:	FOB Factory
Currency:	USD

Item	Description	Quantity	Price	Price unit	Value
0010	8mm diameter x 150 mm Copper / Water Heat pipe	15	$100 USD	Each	$1,500 USD
0020	8mm diameter x 150 mm Copper / Water Heat pipe, with Al. Fins	15	$780 USD	Each	$11,700 USD
0030	NRE Heat Pipes Only	1	$750 USD	Each	$750 USD
0040	NRE for heat Pipes and Fins	1	$1,250 USD	Each	$1,250 USD

Delivery: HPs Only 3 Weeks ARO, HPs and Fins 6 Weeks ARO

Prepared by: Scott Garner
Vice President
Sales and Marketing
Ph: (717) 295-6088
Fax: (717) 295-6064

cc:

Authorized by: Scott G

1046 New Holland Avenue, Lancaster, PA 17601, USA
717-295-6061 (Phone), 717-295-6064 (Fax)
www.1-ACT.com

H.4 Enertron Inc. Heat Pipe Quote (Unofficial)

Hi Chris,

1. 20 fins with 35x35mm size is manufacturable and recommended.
2. Quote: Total $2,000 including heat pipes, fins, assembly labor, NRE Tooling, for 15 heat pipe assemblies.
   Leadtime: 5~6 weeks;
   Paid in full upfront;
   FOB Enertron Taiwan factory.

Let me know if you have any questions. Thanks

Serena Lin
Thermal Application Engineer
ENERTRON, Inc.
P 480-649-5400 Ext 210
F 480-649-5434
www.enertron-inc.com

H.5 Noren Products Heat Pipe Quote (Unofficial)

Hi Calvin,

I have a couple of questions before we can move forward with a quote for you .
1)we use mesh wick not a sintered powder wick .Do you need a sintered powder wick ?
2)this may be expensive in manufacturing around $ 245 dollars each .Small quantity .
3)We can do best effort to reach 123watts that would bring the price down some .
4) we will need to expand the heat pipe to hold the fins .If we solder now we add a plating cost .

After you answer my questions I can give you a better quote .Let me know how you want to proceed?
Thank you for the opportunity.

Regards
Eric Gaillant
Production Sales Engineer

Appendix I - Contacts

I.1 WPI

- **Simon Evans –** Mechanical Engineering Professor
  - Email: sevans@wpi.edu
  - Office: (508) 831-5462
  - Bio: http://www.me.wpi.edu/People/Evans/
- **Neil Whitehouse** – Lab Machinist II, Higgins Shops
  - Email: nrw@wpi.edu
  - Office: (508) 831-5219
- **Barbara Furhman** – Administrative Assistant VI, Purchasing
  - Email: bfurhman@wpi.edu
  - Office: (508) 831-6046

I.2 OsComp Systems

- **Andrew Nelson** – Mechanical Engineer
  - Email: ANelson@oscomp-systems.com
  - Mobile: (617) 544-7208
- **Pedro Santos** – CEO
  - Email: psantos@oscomp-systems.com

I.3 Heat Pipe Manufacturers

- **Mike Beliveau –** Sales Engineer, Aavid Thermalloy
    - Email: beliveaum@aavid.com
    - Office: (603) 223-1810
    - https://www.aavidthermalloy.com/
- **Serena Lin** – Thermal Application Engineer
    - Email: serena.lin@enertron-inc.com
    - Office: (480) 649-5400 ext 210
    - http://www.enertron-inc.com
- **Ray Balardo** – Sales Manager, Noren Products
    - Email: rayb@norenproducts.com
    - Office: (650) 322-9500 ext 213
    - http://www.norenproducts.com/
- **Scott Garner** – Vice President, Advanced Cooling Technologies Inc.
    - Email: Scott.Garner@1-act.com
    - Office: (717) 295-6088
    - Mobile: (717) 799-6084
    - http://www.1-act.com
- **Walter John Bilski** – Senior Engineer, Thermacore Inc.
    - Email: w.j.bilski@thermacore.com
    - Office: (717) 519-3139
    - http://www.thermacore.com/