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# Design and Fabrication of a Stirling Engine

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# Design and Fabrication of a Stirling Engine

*A Major Qualifying Project Report*

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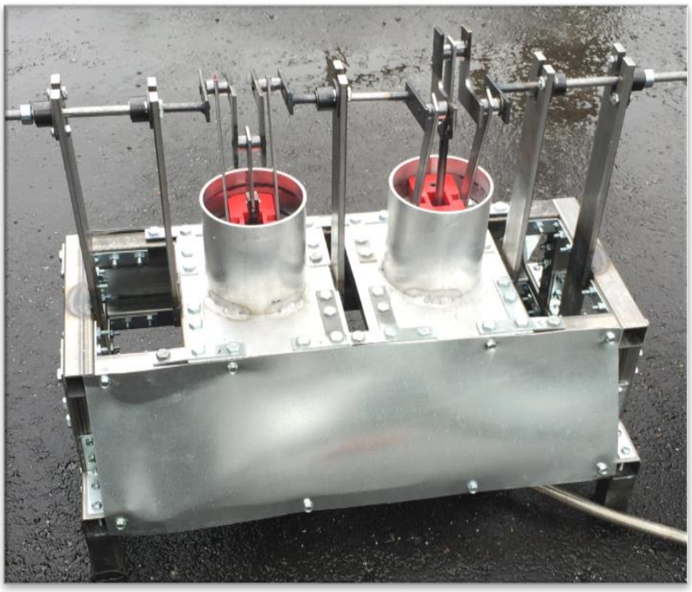

*Approved by:*

Professor John Sullivan (Advisor)

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# Abstract

In this Major Qualifying Project (MQP), the team designed and fabricated a Stirling engine, an external combustion engine that operates due to expansion and compression of air associated with an external heat source and converting that energy into mechanical work. The team designed a dual beta-type Stirling configuration and modeled it using computer aided design (CAD) software. The engine was analyzed thermodynamically and subsequently fabricated, through a sequence of design iterations to reach the final prototype assembly. Computer aided manufacturing (CAM) software and computer numerical control (CNC) machines were used to create complex parts. The dual nature of the design reduced the need for a large flywheel to store inertial energy. A unique piston design and ring system were implemented to reduce friction in the system. The prototype was analyzed through rotation and thermal testing. Results and recommendations to improve this design are provided.

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# 1.0 Introduction

The goal of this Major Qualifying Project (MQP) was to design and fabricate a functional Stirling Engine with a 150 Watt output. A Stirling Engine is an external combustion engine operating via air compression and expansion inside of a cylinder. This compression and expansion of air is used to yield appropriate mechanical work as an output of heat energy into the system. The Stirling Engine was first developed in the early 1800s by Robert Stirling in an attempt to compete with the steam engine. At the time, the Stirling Engine proved to be more attractive than its steam counterpart for various reasons. The main driving force behind designing and producing the Stirling Engine was its increase in efficiency and decrease in noise during operation. Today, the desire to improve, and revitalize the Stirling Engine is growing since it is able to use external energy as fuel from a variety of renewable and waste heat sources.

Moving forward from a previous Major Qualifying Project (MQP), Green Stirling Engine Power Plant, this project worked to design and fabricate a new engine. The previous team's engine had complications with maintaining operation due to a lack of heat transfer between the caps and the working fluid. This team was not able to develop a large enough temperature difference, and therefore energy, difference between cylinders to overcome the effects of friction without heating the engine to the point where the internal clearances become too small and engines components start colliding. Stirling engines are challenging to get to run for actual energy conversion, but this project attempted to combat these common issues with a new design.

By including a larger heat source and cooling system, our team aimed to overcome these limits and develop an operating engine. This report aims to show the development of the engine through the entire design process. Ultimately, recommendations were included to further improve this design and ultimately transfer it into green applications by using renewable or waste energy as the heat source.

# 2.0 Background Research

## 2.1 How Stirling Engines Work

Stirling Engines, though at first glance can seem complex, are fascinating examples of the work that can be done by simple thermodynamic processes on mechanical devices. The outcome is an efficient method for taking energy in the form of heat and turning it into mechanical energy.

There are several different Stirling Engine design configurations. However, the Stirling cycle is similar through all configurations and can be broken down into four ideal processes: isothermal compression, constant volume heat addition, isothermal expansion, and constant volume heat removal. The processes are modeled below by pressure vs. volume (P-V) and temperature vs. entropy (T-S) diagrams in Figure 1:

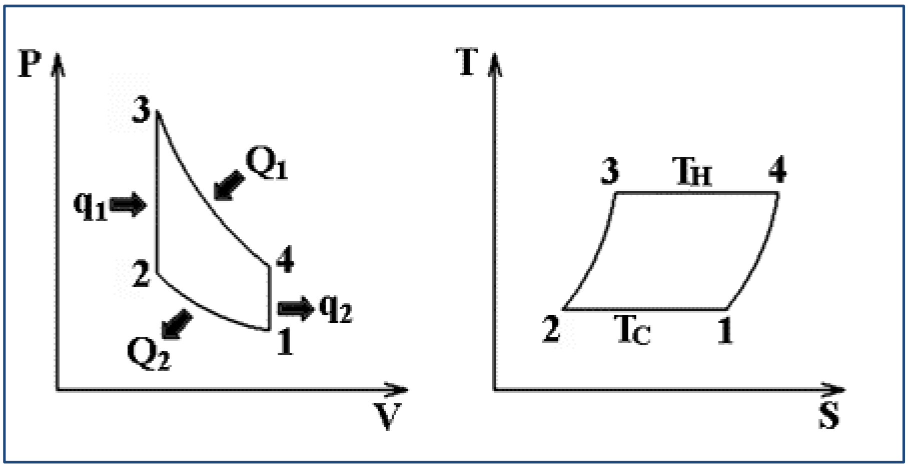


*Figure 1: P-V and T-S diagrams of the Stirling Cycle*

Process 1-2: Isothermal Compression- The heat from the working fluid is dissipated through the cold space at the minimum temperature[1]

Process 2-3: Isochoric Displacement- The working fluid is moved to the heating space while temperature increases from the regenerator[2]

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[1] Walker G., Senft J.R., (1985). Thermodynamics of the Stirling Cycle. *Free Piston Stirling Engines*. Berlin: Springer-Verlag.

[2] Walker G., Senft J.R., (1985).

Process 3-4: Isothermal Expansion- Heat is transferred from the external heating element to the working fluid while at the maximum temperature.[3]

Process 4-1: Isochoric Displacement- The working fluid is moved to the cooling space while temperature decreases due to the regenerator absorbing heat[4]

Since all of the configurations use the same thermodynamic principles, they can all be simplified into the same basic model, as shown in Figure 2. This model consists of two opposing pistons in the same cylinder with a regenerator in the middle. A working fluid is contained in-between the pistons with a constant mass. The cylinder is divided into five zones: compression and expansion spaces where the pistons are located, a heating space where heat is added by an external source, a cooling space where heat is dissipated, usually via heat fins, and a regenerator space[5]. The regenerator helps facilitate the heating and cooling of the working fluid as it passes from one side to the other.

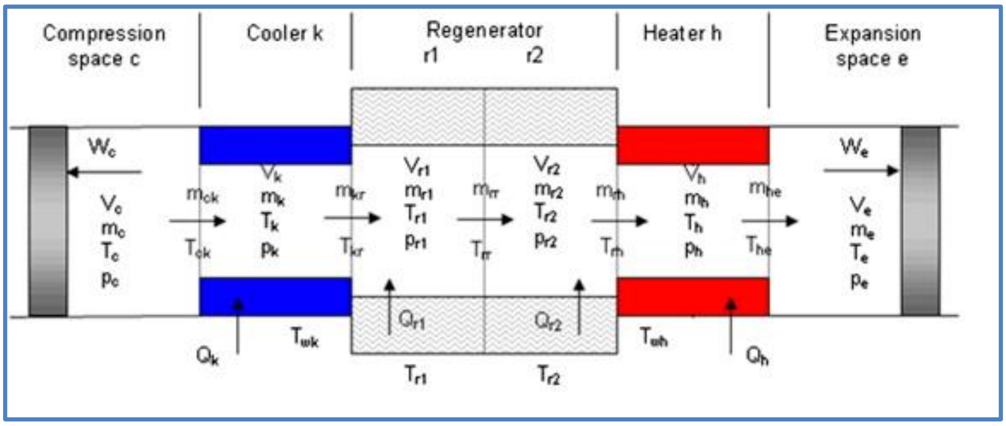


*Figure 2: Simplified visual model of a Stirling Engine[6]*

In this model, the working fluid starts in the compression chamber on the left next to the compression piston and moves to the right to do work on the expansion piston. The cycle

[3] Walker G., Senft J.R., (1985).

[4] Walker G., Senft J.R., (1985).

[5] Elsner M., Parlak N., Soyhan H.S., Wagner A. (2009). Thermodynamic Analysis of a Gamma Type Stirling Engine in Non-ideal Adiabatic Conditions. *Renewable Energy*, 34 (1), 266-273. Retrieved from http://www.sciencedirect.com/science/article/pii/S09601481080008 15.

[6] Elsner M., Parlak N., Soyhan H.S., Wagner A. (2009).

begins with the compression piston all the way to the left at its outer dead point, and the expansion piston is at its inner dead point, just to the right of the regenerator. At this point in time, the working fluid is in state 1 with a maximum volume while temperature and pressure are at a minimum.[7] The working fluid now undergoes process 1-2. Due to the low pressure to its right, the compression piston moves to the right, increasing pressure. Since this takes place in the cooling space, temperature is held constant. Process 2-3 now occurs. While maintaining constant volume, the gas forces the expansion piston to the right due to the increase in pressure from the compression piston. The working gas passes through a regenerator, which starts to introduce heat and increase temperature as the fluid approaches state 3. Because of this temperature increase, the pressure also starts to rise[8]. Eventually the compression piston reaches its inner dead point and the expansion piston approaches its outer dead point and the fluid reaches state 3. The working fluid is now in the heating space of the cylinder and begins process 3-4; the gas now expands pushing the expansion piston to its outer dead point, all the way to the right[9]. State 4 is now reached and volume is a maximum. Process 4-1 occurs as an external force pushes both pistons to their original locations, decreasing temperature as the fluid passes through the regenerator. This also causes pressure to return to its original state as well.

The force that causes process 4-1 is a mechanical kinetic energy storage element, most commonly a flywheel. The linear motion of the expansion piston powers the flywheel. The flywheel is important to this system in that it moves both pistons back to their original

[7] Kongtragool B., Wongwises S., (2006, March). Thermodynamic Analysis of a Stirling Engine Including Dead Volumes of Hot Space, Cold Space and Regenerator. *Renewable Energy*, 31 (3), 345-359. Retrieved from http://www.sciencedirect.com/science/article/pii/ S0960148105000765.

[8] Kongtragool B., Wongwises S., (2006, March)

[9] Kongtragool B., Wongwises S., (2006, March)

position. Without the force of the flywheel, the conditions of the gas would not be maintained and the engine would not recycle.

## 2.2 Types of Stirling Engines

Stirling Engine designs can be divided into three categories based on configuration and how the working fluid is displaced within its closed system. Each design involves two pistons/displacers out of phase by 90 degrees as they simultaneously move back and forth. The three types are referred to as alpha (α), beta (β), and gamma (γ). All three designs come with advantages and disadvantages that must be taken into account when deciding on which one of the three to construct.

### 2.2.1 Alpha (α)

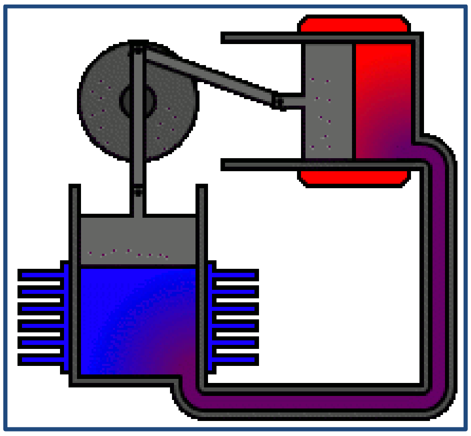

*Figure 3: Alpha Stirling Engine Design[10]*

The α type (Figure 3) is a closed system between two separate cylinders positioned 90 degrees apart along with the engine phasing. One cylinder is connected to an external thermal device creating a hot side while the other cylinder is designed to dissipate heat and thus referred to as the cool side. Insulation techniques may include increasing the surface area for additional heat exchange as well as radial cooling.

The primary advantages of the α-type are its efficiency, power density, and manufacturability. When all components are working correctly the engine provides roughly

[10] (2010). Stirling Engines - Mechanical Configurations. Retrieved from https://www.ohio.edu/mechanical/stirling.

double the power density due to its two power pistons as opposed to one[11]. Additionally, the lack of moving a displacer piston, which can found in the other designs, allows the extra power to increase the power density. The ease of maintenance and manufacturability attributes to this design as well[12].

The major disadvantage to the α-type is that its closed system has two moving seals where the pistons move back and forth[13]. If either of these seals are broken, then the integrity of the Stirling cycle is lost and the engine fails to perform. The hot cylinder provides the biggest threat to breaking a seal due to the increased temperature and friction[14]. Another disadvantage associated with this design is its amount of dead space. Due to two cylinders being present it will inevitably have more volume than just one cylinder, reducing the maximum pressure of the system.

### 2.2.2 Beta (β)

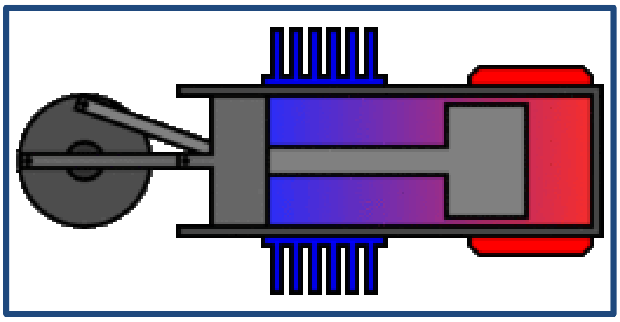

*Figure 4: Beta Stirling Engine Design[15]*

The β style (Figure 4) differs from the α design in a few ways. The most prominent of these differences would be that the temperature differential occurs on two ends of one cylinder as opposed to separate hot and cold cylinders, with the hot side of the cylinder being

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[11] (2010). Stirling Engines - Mechanical Configurations.

[12] Comiskey P., Finkes D., Scanlon C., Weise A. *Stirling Engine Microgrid Generator* [PDF document]. Retrieved from https://www.msoe.edu/servlet/JiveServlet/previewBody/2694-102-1-2826/Senior%20Design%20Report%20Compressed.pdf.

[13] (2010). Stirling Engines - Mechanical Configurations.

[14] Comiskey P., Finkes D., Scanlon C., Weise A.

[15] (2010). Beta Type Stirling Engines. Retrieved from https://www.ohio.edu/mechanical/stirling/engines/beta.html.

furthest from the flywheel in this case. The process includes a power piston pushed by isothermal expansion with another piston (referred to as a displacer piston) slides back and forth in the cylinder displacing the working fluid. The working fluid flows towards the cool side as it drives a power piston attached to the flywheel, allowing it to rotate with the system.

An advantage the β has over the alpha design is that there is no need for a hot seal, just the cooler side adjacent to the power system. This will reduce the risk of leaks to the closed system. Other advantages of the design include reduced dead space, engine size, and cost. The lack of two cylinders means the working fluid requires no medium to travel between the two temperatures. This lack of medium reduces the space the working fluid needs to travel from side to side (dead space) and allows for more pressure to build within the system[16].

Disadvantages to this type involve the lack of insulation between the two different temperature cylinders. Heating the hot side so closely to the cold side might reduce the temperature differential and potential work. Additional manufacturing is required in creating the displacer piston. The power piston and the displacer to share the same axis, which requires the displacer to travel through the power system without compromising the closed system.

### 2.2.3 Gamma (γ)

The γ design (Figure 5) Stirling Engines are very similar to β type designs except for one major difference. The heat exchange system and displacer piston are in a separate area from the power piston, creating more dead space for the working fluid[17]. Due to this the power density and efficiency suffer opposed to the β[18].

[16] (2010). Beta Type Stirling Engines.
[17] (2010). Gamma Type Stirling Engines. Retrieved from https://www.ohio.edu/mechanical/stirling/engines/gamma.html.
[18] Comiskey P., Finkes D., Scanlon C., Weise A.

The advantage that the γ design offers is avoiding intersecting linkages seen in the β design. This allows for easier manufacturability but it sees numerous drawbacks.

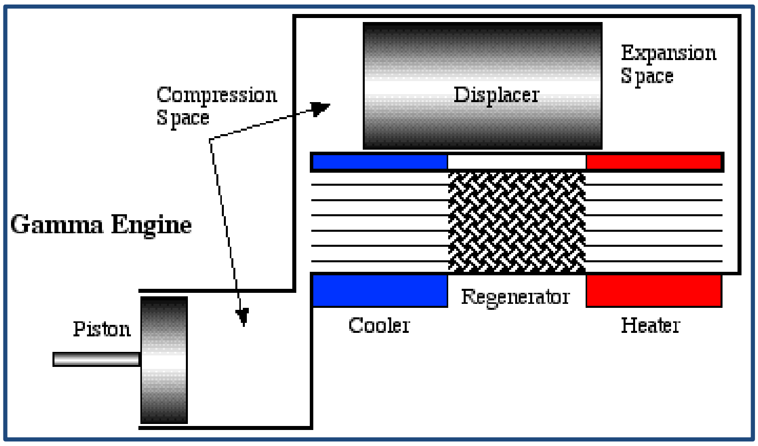

*Figure 5: Gamma Stirling Engine Design*[19]

Negative impacts from moving the power piston to a different axis include increased dead space, lower efficiency, and lower power density. One significant drawback to the power output is the lack of heat exchange in the power piston cavity of the engine. The working fluid travels through dead space before becoming in contact with the piston, which decreases pressure and power.

## 2.3 History & Applications of the Stirling Engine

The development of the Stirling Engine began in the early 19th century by Scottish clergyman, Rev. Robert Stirling.  In September 1816, Robert Stirling applied for a patent for his first hot air engine[20]. The machine operated under two principles: its closed-cycle and its regenerator which Stirling called the "economizer". A Stirling Engine has a closed-cycle since the working fluid is contained within the machine. In contrast, the Otto engine vents its working fluid into the atmosphere and thus has an open-cycle[21]. The regenerator separates the hot and cold chambers, and as the working fluid passes through, heat is transferred to and from the regenerator.

[19] (2010). Gamma Type Stirling Engines.
[20] Bauer R. (2009). The Stirling Engine: The 'Cyclical Life' of an Old Technology. *International Committee for the History of Technology, 15.* Retrieved from http://www.jstor.org/stable/23787099.
[21] Bauer R. (2009).

Despite 27 years of redesigning and improving performance, Robert Stirling did not receive any commercial success. Issues with the four Stirling Engines he built were related to inadequate materials[22]. The cast-iron sides of the cylinders used as heat exchangers could not stand the thermal stress caused by the coal fire heat source. Leather gaskets quickly wore out and could not provide enough seal and therefore pressurization[23]. Perceiving his work as failure, Stirling said, "If Bessemer iron and steel had been known thirty-five or forty years ago there is scare a doubt that the air engine would have been a great success…It remains for some skilled and ambitious mechanist in a future age to repeat it under more favorable circumstances and with complete success"[24].

The true first application of the Stirling Engine was to pump water from a local quarry. While this is not the common application of a Stirling Engine by today's standards, using a Stirling Engine to pump water was vital for collecting data and testing the engine. For the two years this Stirling Engine was in operation, it produced 1.5kW of power. The life span of this engine was cut short due to poor metal quality as well as a lack of technology.

There is a wide range of modern applications for the Stirling Engine as well as an everlasting search for new applications. Some of the main modern applications include powering submarines, auxiliary power generators, large-scale solar power, or as small models and toys. While most of these applications are not commercialized or marketed, some companies, like Kontax, insist that Stirling Engines do have the potential to be introduced and manufactured for the large market and use in commercial settings[25].

Since the 1980's a Swedish ship builder, Kockums, has been developing the application of Stirling power in submarines and has proven just how effective the can be.

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[22] Bauer R. (2009).
[23] Bauer R. (2009).
[24] Bauer R. (2009).
[25] (2015) Kontax Stirling Engines. Retrieved from http://www.stirlingengine.co.uk/History.asp.

Kockum's eight successful Stirling-powered submarines carry compressed oxygen to allow fuel combustion submerged, providing heat for the Stirling Engine. They are currently used on submarines of the Gotland and Sodermanlan classes. They are the first submarines in the world to feature Stirling air-independent propulsion (AIP), which extends their underwater endurance from a few days to several weeks; this 30 capability has previously only been available with nuclear-powered submarines[26]. As displayed by Kockums, with the right design and proper consideration of application the Stirling Engine has great potential for very effective applications.

As for vehicle application, or modes of transportation on land, the Stirling Engine has not been well received. Stirling Engines lack several key components to being applicable in the automotive industry. For example, one of the biggest issues with using a sterling engine in a vehicle is time. Stirling Engines take time in that they have a slow start up and shut down time, rendering them impractical for a short drive to the store. Also, another major drawback of the Stirling Engine is its lack of acceleration control. To be able to control the acceleration of these engines, there is a need for more heat and again, more time. The acceleration response, if any, is slow and hard to control, denying it entry into the automotive realm.

However, a common theme in the progression of the Stirling Engine is advancement. With each new design is an opportunity to advance the engine. Since the development of the Stirling Engine in the 1800s the fundamental concepts have remained the same, but the advancement of the engine for specific applications if continuing to grow. According to Hacsi, a modified Stirling Engine has taken design ideas from internal combustion engines

[26] Costa N., Donnan B., Ephraim D., Graff E., Haveles A., Larsen A., Morozov M., Rolon, M. (2015). Green Stirling Engine Power Plant. (Undergraduate Major Qualifying Project No. E-project-043015-131215). Retrieved from Worcester Polytechnic Institute Electronic Projects Collection: ttps://www.wpi.edu/Pubs/E-project/Available/E-project-043015-131215/unrestricted/Green_Stirling_Engine_Power_Plant.pdf

with sidewall combustion chambers to address the inherent Stirling Engine drawbacks of power-density and slow acceleration response[27]

Stirling Engines are not only a promising application for providing a propulsion system but are also a breakthrough for renewable energy and power plant settings. All Stirling Engines need to be able to run is heat. Large-scale solar Stirling Engine power plant applications are being looked into all over the world and especially in the United States by Stirling Engine Systems (SES), a systems integration and project management company. SES develops utility-scale sized renewable energy power plants and electrical generators and has the potential to become a premier renewable energy company. The use of Sterling engines is becoming a more sought after idea and their potential for change in the world's power generation is immense.

## 2.4 Thermodynamics in Stirling Engines

The work performed by a Stirling Engine is a directly related to its Stirling cycle. As mentioned previously, the working fluid is constantly switching states of temperature, pressure, and volume to provide the power piston the energy required to spin the crankshaft. The two forms of energy Stirling Engines manipulate are thermal energy and mechanical energy. Thermal energy is transferred into the system via the hot side of the cylinder and is converted into mechanical energy through the cyclic motion of the power piston. The Stirling cycle is a reversible thermodynamic cycle that includes a regenerator in its closed system.

The regenerator must be placed in between the path of the hot and cold sides of the engine cylinder to increase the thermal efficiency as the working fluid rapidly changes temperature. Common types of regenerators found in Stirling Engines are pockets of steel wool that come into contact with the working fluid as it travels between the two sides of the

[27] Hacsi, J. S. (2008). *Internal combustion engine with sidewall combustion chamber and method.* U.S. Patent 7387093 B2. Retrieved from http://www.google.com/patents/US7387093.

cylinder. In theory, as the working fluid undergoes isothermal compression, thermal energy is left behind to increase the temperature differential. Consequently, during isothermal expansion it will pass through the regenerator and recycle the thermal energy it left behind during the process of isothermal compression[28].

## 2.5 Heat Transfer in Stirling Engines

Fundamental knowledge of heat transfer is essential in understanding Stirling Engines. Heat transfer is the exchange if thermal energy due to temperature differential. Whenever a temperature difference exists in a medium or between a media, heat transfer must occur[29]. The three modes of heat transfer include conduction, convection, and radiation. Conduction is the heat transfer across a medium, while convection is the heat transfer between a surface and a moving fluid. Radiation involves the heat transfer in the form of electromagnetic waves, which occurs with all surfaces at different temperatures. Heat transfer analysis between the heat source and the bottom of the cylinder will be crucial for determining the power output from the heat source.

[28] Stirling Engine. (n.d.). Retrieved December 17, 2015, from http://www.robertstirlingengine.com/regenerator.php

[29] Incropera, F., & Bergman, T. (2007). Fundamentals of Heat and Mass Transfer (7th ed.). Hoboken, NJ: John Wiley.

# 3.0 Design Methodology

The goal of this Major Qualifying Project (MQP) was to design and prototype a Stirling Engine with significant power output. Accomplishing this goal aimed to improve upon the previous MQP, Green Stirling Engine Power Plant, by installing a working engine that can use waste energy as its power source. The following tasks were developed to guide the design process for this Stirling Engine:

Task 1: Select Type of Engine

Task 2: Select Heat Source

Task 3: Design of the Cylinders

Task 4: Design of the Pistons

Task 5: Design of the Diffusers

Task 6: Design of the Crankshaft

Task: Design of the Cooling System

In this chapter, the methodologies used to complete each task will be discussed. Dimensioned drawings for all machined parts can be found in Appendix B.

## 3.1 Type of Engine Selection

As discussed in the background chapter, there are different types of Stirling Engines and each orientation has its pros and cons. Many factors were considered in the selection of the type of Stirling Engine that would be designed and prototyped in this project. The team established which objectives were most important in design, which included power output, manufacturability, and cost. Preliminary designs of both an alpha and beta type were compared. The first design was a standard alpha (Figure 6) where the heat source is applied to the lower horizontal cylinder while heat is dissipated through the fins on the vertical cylinder. The cylinders are connected through a small pipe with a regenerator on it. This helps facilitate the induction and dissipation of heat into and out of the fluid.

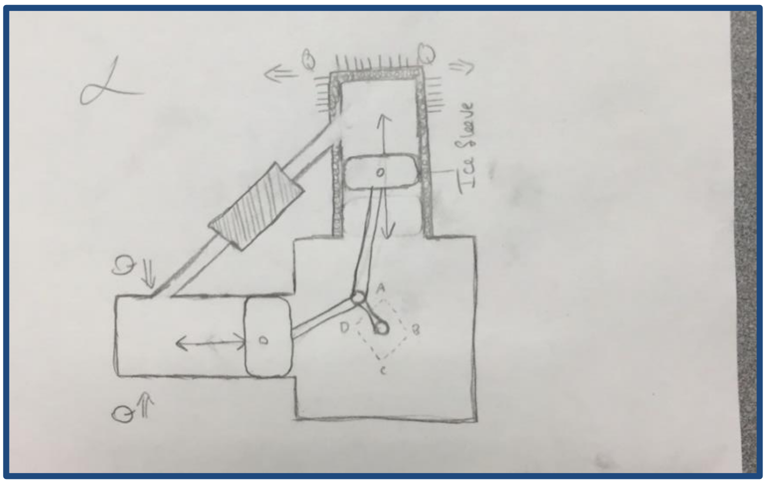



*Figure 6: Preliminary Alpha Design*

The alpha engine design was analyzed through work and pressure calculations using Mathcad. Figure 7 shows change in volume in figures 7 using radius of the cylinder ($r_c$) and regenerator ($r_r$). Initial pressure and temperatures were represented by $P_0$ and $T_0$ to determine the mass of the working fluid. $T_1$ and $T_2$ represent the maximum and minimum temperatures the working fluid will reach as it cycles through the engine. $H_1$ and $H_2$ properly represent the dead space length and the stroke length of the power pistons in both the hot and cold cylinders. $L_r$ is depicted as the length of the of the regenerator tube connecting the cylinders while $S_c$ is an arbitrary frequency chosen to compare the alpha and beta designs at the same revolutions per minute. Using the kinematic understanding of where the pistons will be at any given time, the maximum and minimum volumes were calculated to yield a change in volume that will be used to determine the work of the engine cycle. $V_1$ is the minimum volume when the power piston in the hot side is at bottom dead center (BDC), while $V_2$ is the maximum volume when the same piston is at top dead center (TDC).

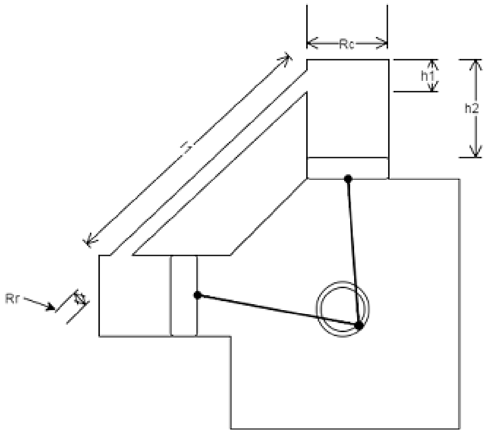
## Alpha Design

$T_0 := 298K$ $\quad r_c := 1.625in$ $\quad r_r := 0.375in$ $\quad P_0 := 1atm$

$T_1 := 373K$ $\quad h_1 := 1in$ $\quad l_r := 1ft$ $\quad S_c := 8.33Hz$

$T_2 := 453K$ $\quad h_2 := 3.5in$ $\quad R := 0.082 \frac{L \cdot atm}{K \cdot mol}$

$M := 28.97 \cdot \frac{gm}{mol}$




## Max Volumes

$V_1 = V_{C1} + V_R + V_{H1}$

$V_2 = V_{C2} + V_R + V_{H2}$

$V_R := (r_r)^2 \cdot \pi \cdot l_r = 0.087 L$

$V_{C1} := (r_c)^2 \cdot \pi \cdot h_1 = 0.136 L$ $\quad V_{H1} := (r_c)^2 \cdot \pi \cdot h_1 = 0.136 L$

$V_{C2} := (r_c)^2 \cdot \pi \cdot h_2 = 0.476 L$ $\quad V_{H2} := (r_c)^2 \cdot \pi \cdot h_2 = 0.476 L$

$V_1 := V_{C1} + V_R + V_{H1} = 0.359 L$

$V_2 := V_{C2} + V_R + V_{H1} = 0.699 L$

$\Delta V := V_2 - V_1 = 0.34 L$

*Figure 7: Alpha Design Volumes*

## Pressure from Working Fluid (air)

$P \cdot V = \left(\frac{m}{M}\right) \cdot R \cdot T$

$P_0 \cdot V_2 = n \cdot R \cdot T_0 \text{ solve}, n \rightarrow \frac{0.028589776145192789327 \cdot L \cdot mol}{L}$

$n := 0.02858 \cdot mol$

$n \cdot M = 8.28 \times 10^{-4} kg$ air

$P_3 := \frac{n \cdot R \cdot T_2}{V_1} = 2.959 \cdot atm$ $\quad P_4 := \frac{n \cdot R \cdot T_2}{V_2} = 1.52 \cdot atm$

$P_2 := \frac{n \cdot R \cdot T_1}{V_1} = 2.437 \cdot atm$ $\quad P_1 := \frac{n \cdot R \cdot T_1}{V_2} = 1.251 \cdot atm$

*Figure 8: Alpha Design Pressure*

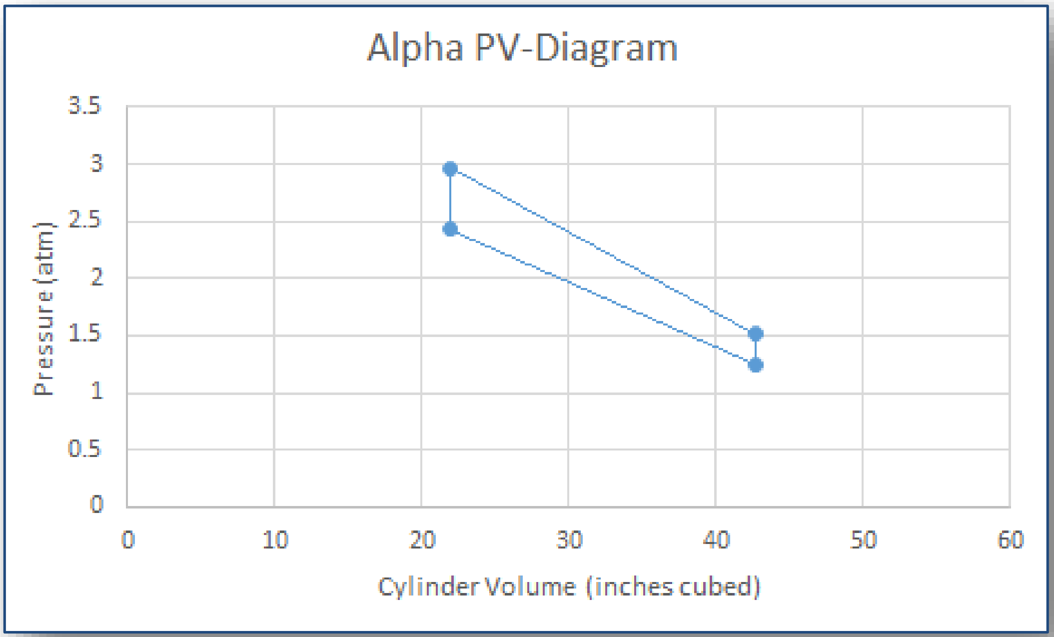



*Figure 9: Alpha Design Pressure vs. Volume Diagram*

**Work done by Thermal Expansion**

$$W = P \cdot \Delta V$$

$$W_1 := P_3 \cdot \Delta V = 101.902\,J$$

$$W_2 := P_1 \cdot \Delta V = 43.088\,J$$

$$Work_{Net} := W_1 - W_2 = 58.814\,J \quad \text{per cycle}$$

At 500 RPM

$$Power := Work_{Net} \cdot S_c = 489.919\,W$$

*Figure 10: Alpha Design Work*

The second design concept is a beta type Stirling Engine which includes two cylinders that are connected in series with one crankshaft (Figure 11). The idea is that two cylinders would produce more power. Each cylinder would be positioned to have the heat source coming from underneath the hot sides of their cylinders. Heat fins, used as a preliminary cooling concept will be on the cold side of the cylinder. The two sides for the cylinder will be separated by a heat shield in order to maintain the highest amount of temperature differential as the heat rises from the bottom.

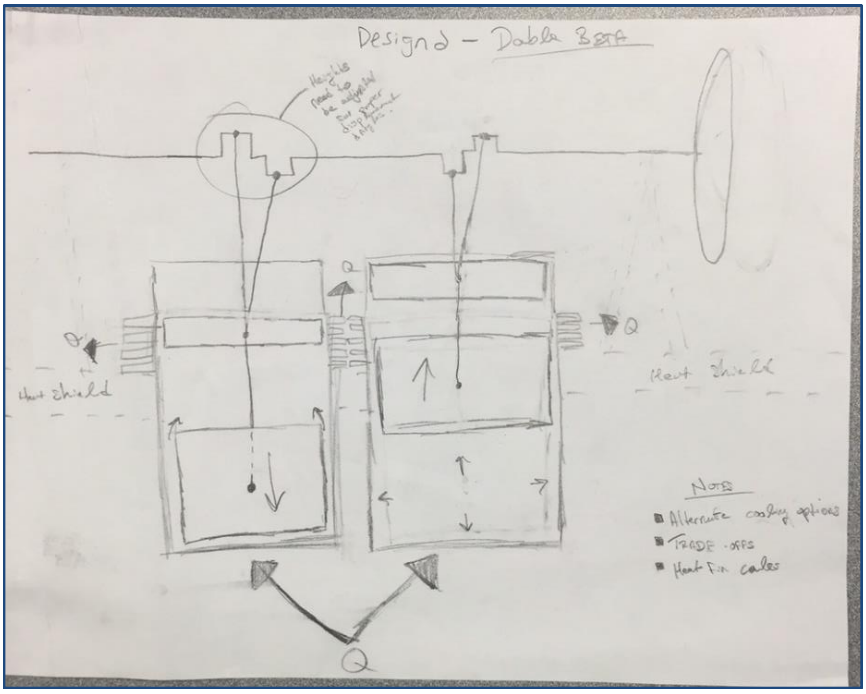



*Figure 11: Preliminary Beta Design*

The beta design's work and pressure were analyzed in Mathcad in figures 12 through 15. The same variables from the alpha design apply to the beta volume calculation except for the regenerator volume which does not apply in this design. The change in volume and potential work output for the beta design can be seen in figures 12 thru 15.

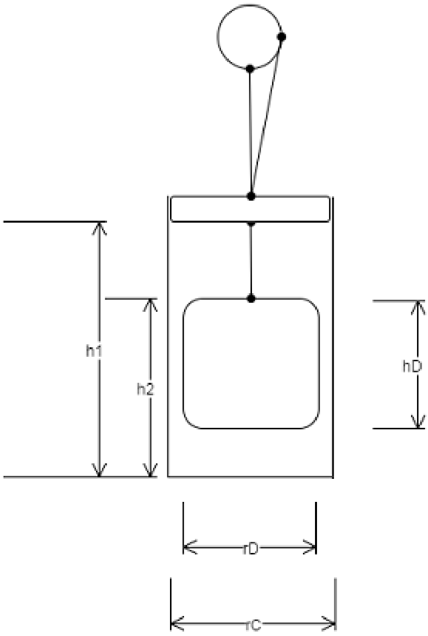
## Beta Design

$T_0 := 298K$ $\quad r_c := 1.625in$ $\quad r_D := 1.5625in$ $\quad P_0 := 1atm$

$T_1 := 373K$ $\quad h_1 := 8in$ $\quad h_D := 3in$ $\quad S_c := 8.33Hz$

$T_2 := 453K$ $\quad h_2 := 5.5in$ $\quad R := 0.082\frac{L \cdot atm}{K \cdot mol}$

$M := 28.97 \cdot \frac{gm}{mol}$




Max Volumes

$$V_1 := (r_c)^2 \cdot \pi \cdot h_2 - r_D{}^2 \cdot \pi \cdot h_D = 0.371 L$$

$$V_2 := (r_c)^2 \cdot \pi \cdot h_1 - r_D{}^2 \cdot \pi \cdot h_D = 0.71 L$$

$$\Delta V := V_2 - V_1 = 0.34 L$$

*Figure 12: Beta Design Volumes*

## Pressure from Working Fluid (air)

$$P \cdot V = n \cdot R \cdot T$$

$$P_0 \cdot V_2 = n \cdot R \cdot T_0 \text{ solve}, n \rightarrow \frac{0.029075325020427770503 \cdot L \cdot mol}{L}$$

$$n := 0.029075 \cdot mol$$

$$n \cdot M = 8.423 \times 10^{-4} kg \quad \text{air}$$

$$P_3 := \frac{n \cdot R \cdot T_2}{V_1} = 2.914 \cdot atm \qquad P_4 := \frac{n \cdot R \cdot T_2}{V_2} = 1.52 \cdot atm$$

$$P_2 := \frac{n \cdot R \cdot T_1}{V_1} = 2.399 \cdot atm \qquad P_1 := \frac{n \cdot R \cdot T_1}{V_2} = 1.252 \cdot atm$$

*Figure 13: Beta Design Pressures*

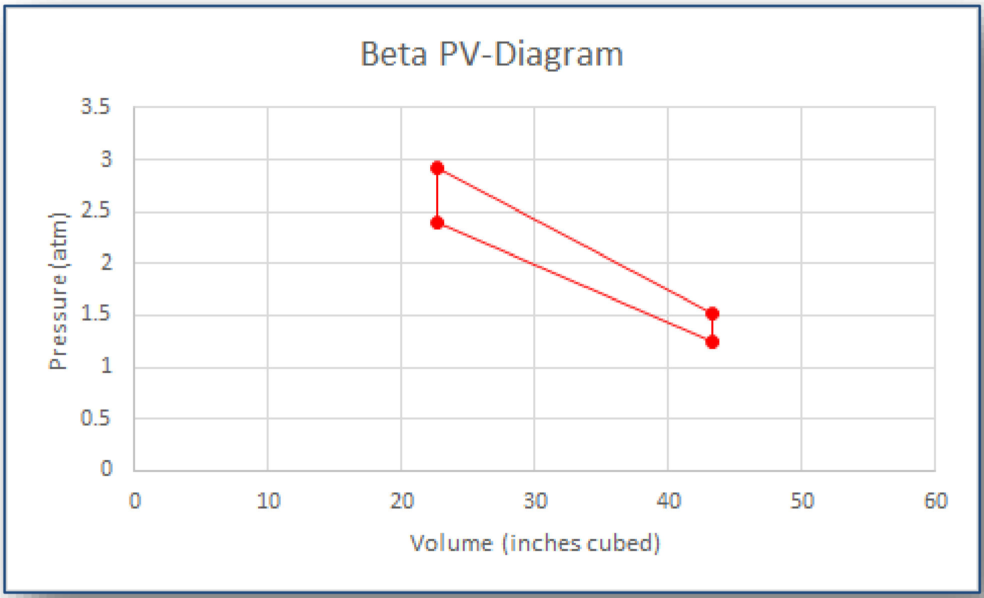



*Figure 14: Beta Design Pressure vs. Volume Diagram*

**Work done by Thermal Expansion**

$$W = P \cdot \Delta V$$

$$W_1 := P_3 \cdot \Delta V = 100.348\,J$$

$$W_2 := P_1 \cdot \Delta V = 43.102\,J$$

$$Work_{Net} := W_1 - W_2 = 57.246\,J$$

At 500 RPM

$$Power := Work_{Net} \cdot S_c = 476.857\,W$$

*Figure 15: Beta Design Work*

These calculations were performed in order to compare the potential work output and internal pressure of both the preliminary alpha and beta designs. For both designs, the initial calculation was to find the system’s maximum and minimum volumes. By designating equivalent dimensions of cylinder diameter and piston stroke, the team was able to calculate the working volumes of both systems. Next the mass of the working fluid was determined by using the Ideal Gas Law:

$$P * V = \frac{m}{M} * R * T$$

The working fluid mass is established by the pressure, volume, and temperature before any heat is added to the system. Once the mass of the working fluid was calculated, the moles (of air) were plugged back into the Ideal Gas Law to calculate differing pressures with respect to changing temperatures and volumes. At the third phase of the Stirling cycle, the system experiences the smallest volume and the highest temperature, resulting in a maximum pressure. Contrary to phase 3, phase 1 has the largest volume with the lowest temperature, resulting in the lowest pressure. From here the work of the gas expansion was calculated using the following equation for work:

$$W = P\Delta V$$

After calculating minimum and maximum pressures, the work done by thermal expansion was found. Since the analytics of regeneration are too complex, a simpler form of work was used to get an idea of the work output from each type of engine. The maximum and minimum pressures were plugged into the above equation, while the $\Delta V$ remained constant for each type. The difference of the maximum and minimum work done by the gas to expand and cool designates the net work done by the gas in the system. This result was then multiplied by an arbitrary frequency of 500 rpm to see which engine would yield more power at the same speed. The alpha design was found to yield approximately 20 W more than the beta design.

In addition to theoretical work output and internal pressure, manufacturability of the chosen design is a high priority. The goal of having a working Stirling Engine with significant power output cannot come to fruition if the engine is too complex to build, therefore manufacturability will be the most important objective in choosing which design to build. Considering cost of the design is also necessary as the team is working with a budget of about $800 for materials.

A weighted decision matrix was developed in order to compare the two options. By weighing the objectives and ranking the design on each objective, a decision can be made on which design should be further developed and built.

*Table 1: Weighted Decision Matrix*

| | Power Output (4) | Manufacturability (5) | Low Cost (3) | Total |
|---|---|---|---|---|
| Design 1 - Alpha Design | 4x(4) | 3x(5) | 3x(3) | 40 |
| Design 2 - Double Beta Design | 3x(4) | 5x(5) | 3x(3) | 46 |

In table 1, the three main objectives are weighted on scale of 1-5, with 5 being the most important. Manufacturability was assigned a 5, power output a 4, and cost a 3. Next, the designs were evaluated on all three objectives on a scale of 1-5, with 5 being the design completely satisfies the objective. The alpha design had the highest power so it was ranked as 4, while the beta was ranked as a 3. Since the beta design required less material and machining it was ranked 5 for manufacturability and the alpha was ranked 3. The cost of each design was based on amount of material expected. The two designs are expected to have a similar cost and were both assigned a ranking of 3. The beta design had the highest total and thus was chosen as the design to be built.

## 3.2 Heat Source Selection

The engine's heat source was chosen based on available supplies and ability to reach a high temperature differential. Two team members already owned propane burners similar to the one pictured in figure 16. This was a convenient and cost effective solution to achieving the design objectives.

*Figure 16: Heat Source – Propane Burner*

An experiment was performed in order to calculate the approximate heat transfer from this burner. Water was placed into a cylinder of similar dimensions to our preliminary design. The cylinder was then heated to measure the temperature differential in a given amount of time. The experiment is outlined below:

Objective: Find the heat transfer into a metal cylinder.

Procedure:

1. Cut a small hole in a cylinder and empty its contents (in this case we used a steel soup can)
2. Record mass of the cylinder
3. Fill cylinder with water
4. Record mass of cylinder + water
5. Set up DAQ box input with thermocouple (type K)
6. Place thermocouple in the water – make sure it is only in the water and not touching the sides of the cylinder
7. Start recording

8. Turn on propane burner

9. Heat and record until water reaches 80°C

$$Q = \frac{m * C_p * \Delta T}{t} = \frac{(806\ g)\left(4.184\frac{J}{g℃}\right)(36℃)}{160s} = 759\ \frac{J}{s} = 759\ Watts$$

From this experiment, the calculated Qin of the heat source was found to be 759 Watts. This power estimate will be an important figure in designing the remainder of the system.

## 3.3 Design of the Cylinders

The cylinder orientation, size, and material were greatly considered as these design features will influence the work output. The heat source will be placed below the cylinder in order to receive direct heat to the hot side of the cylinder (bottom). The dimensions of each cylinder were chosen to be 3 ½" (8.89 cm) outer diameter and 3 ¼" (8.255 cm) inner diameter, thus the wall thickness will be 1/8" (0.3175 cm) (Figure 13). The wall thickness must be thin enough to maximize heat transfer while still maintaining the structural integrity of the system. The material of the cylinder was selected to be aluminum to reduce cost. The height of the cylinder was chosen to be 9" (22.86 cm) through the design and analysis of the crankshaft. Figure 17 shows a model of the cylinder and its flange.

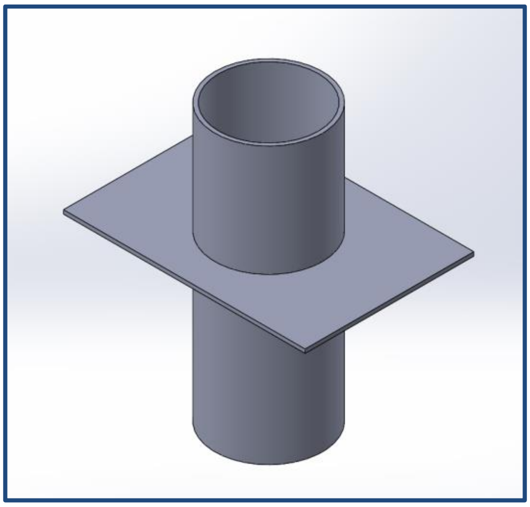


*Figure 17: Cylinder Model*

The cylinder cap will be in direct contact with the heat source and must maximize heat transfer into the system. To compare thermal conductivity of materials, the following calculations were performed for a copper and aluminum cap with a diameter of 3 ¼" (8.255 cm) and a thickness of 1/8" (0.3175 cm).

$$A = \pi \left( \frac{3.25"}{2} \right)^2 = 8.296\ in^2 = 0.058\ ft^2$$

$$Thermal\ Resistance = \frac{l}{kA}$$

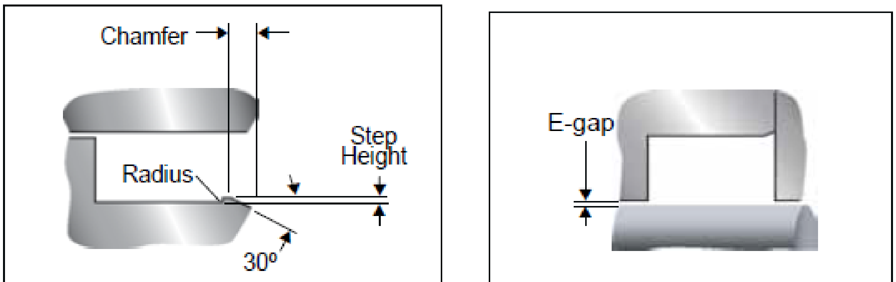
| Material | Thermal Conductivity (BTU/ft·°F·hr) | Thermal Resistance (ft$^2$·°F·hr)/Btu |
|---|---|---|
| Copper | 231 | $7.9*10^{-5}$ |
| Aluminum | 136 | $1.3*10^{-4}$ |

Table 2: Cap Material Thermal Resistance Comparison

The thermal resistance of each cap was determined. As expected, there is a lower thermal resistance for a copper cap, almost twice as conductive as the aluminum cap. This change in resistance must be considered in purchasing materials. The additional cost of the copper cap may not be worth the slight improvements in thermal conductivity, for this reason aluminum was the appropriate choice.

## 3.4 Design of the Pistons

The pistons were designed with regard to seals, piston rings, and tolerances of the system. In respect to tolerances, depending on what piston rings and seals are being used, there are specific distances between the cylinder wall and the outside diameter of the piston that need to be met. For additional expertise in the area of piston rings, Eriks Seals and Plastics, was contacted for design consultation. From this consultation, the team was able to acquire all of the necessary information for finalizing the design of the piston. The team obtained specific values, such as tolerances for the cylinder wall, groove dimensions and material specifications of the piston rings (Figure 18). After these specifications were verified, a 3D model of the piston was created in Solidworks. The seal between the piston and the cylinder must be held in order to increase pressure and in turn generate power. The design includes an aluminum cylinder with one groove for a double functioning piston ring and seal. The piston ring that will be used incorporates a spring loaded ring to seal the cylinder and a wear band that provides guidance while not applying extra frictional forces onto the cylinder wall.




**Stepped Piston Gland Dimensions**

| Series | Step Height | Max. Radius | Min. Chamfer | Min. Seal ID |
|---|---|---|---|---|
| 000 | .008/.012 | .005 | .025 | .375 |
| 100 | .010/.015 | .005 | .030 | .563 |
| 200 | .015/.020 | .007 | .040 | 1.000 |
| 300 | .017/.023 | .010 | .045 | 1.375 |
| 400 | .022/.028 | .010 | .055 | 2.500 |

**Maximum Extrusion Gap.**

| Series | 300 psi | 1500 psi | 3000 psi | 6000 psi |
|---|---|---|---|---|
| 000 | .008 | .004 | .003 | .002 |
| 100 | .010 | .006 | .004 | .003 |
| 200 | .014 | .008 | .006 | .003 |
| 300 | .020 | .010 | .008 | .004 |
| 400 | .024 | .012 | .010 | .005 |

*Figure 18: Piston Dimensions - Eriks Seals and Plastics*

## 3.5 Design of the Diffusers

The role of the diffuser, similar to a displacer, is to occupy dead volume in the cylinder while shuffling air between the hot and cold side. The design includes a 3” (7.62 cm) tall cylinder with a 3 1/8” (7.9375 cm) diameter. There are four 3/8” (0.9525 cm) through holes in the diffuser which will be paced with steel wool. These tubes will act as the regenerator of the system by circulating air between the hot and cold sides of the cylinder while absorbing and transferring thermal energy. The design of the diffuser can be seen in figure 19.

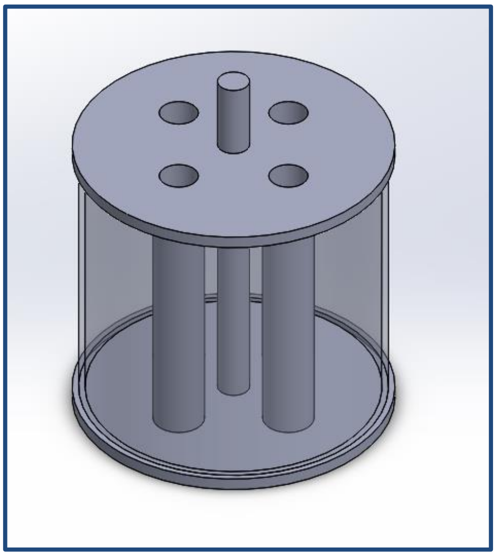


*Figure 19: Diffuser Design*

## 3.6 Design of the Crankshaft

The crankshaft's function is to convert the linear motion of the pistons into rotational motion. The initial crankshaft design, shown in figure 20, utilized lever arms offset by 90 degrees and 180 degrees to achieve proper timing of the two power pistons.

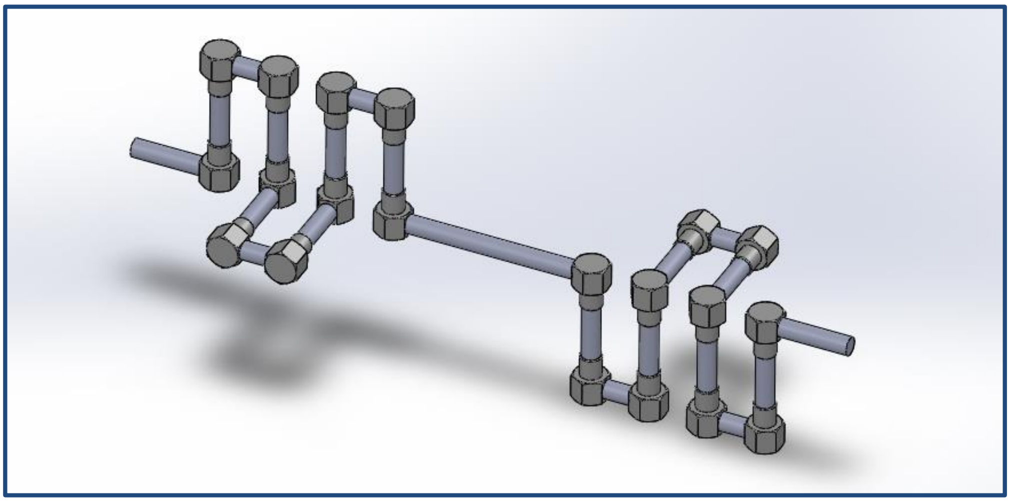


*Figure 20: Crankshaft Design*

Aluminum rods were threaded and connected by 90 degree NPT pipefittings. These specific fittings were chosen for strength as well as their compact configuration (Figure 21).

These pipefittings were imported into the Solidworks file and the connecting rod sizes were determined in order to obtain the desired stroke length of 2.75" (6.985 cm).

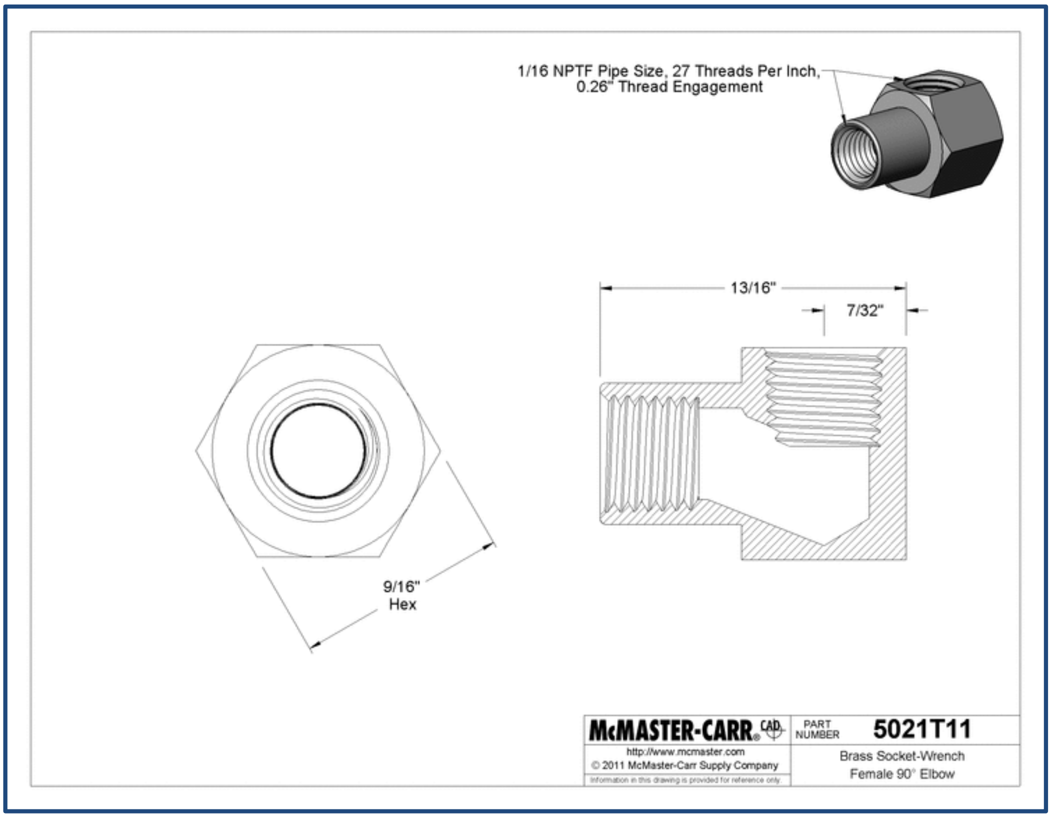



*Figure 21: NPT Threaded Connectors*

The length of the linkages and connecting rods for crankshaft were determined from desired stroke lengths of the power piston and diffuser. The angular displacement of the crankshaft was set as a function of the height of the piston head. Their mathematical relationship was found and graphed in figures 22-25.

| Key Variables | Definition |
|---|---|
| $A_P$ | length of crankshaft link for piston |
| $A_D$ | length of crankshaft link for diffuser |
| $L_P$ | length of connecting link for piston |
| $L_D$ | length of connecting link for diffuser |
| $B_P$ | length of connecting rod for piston |

| | |
|---|---|
| $B_D$ | length of connecting rod for diffuser |
| $t_p$ | thickness of the power piston |
| $\theta_P$ | angle formed between $A_P$ the zero position |
| $\theta_D$ | angle formed between $A_D$ the zero position |
| $h_P$ | distance from crankshaft to top of $B_P$ |
| $h_D$ | distance from crankshaft to top of $B_D$ |
| $y_{Ptop}$ ($y_p$) | distance from crank to top of piston |
| $y_{Pbot}$ | distance from crank to bottom of piston |
| $y_{Dtop}$ ($y_D$) | distance from crank to top of diffuser |

*Figure 22: Table of Kinematic Variables*

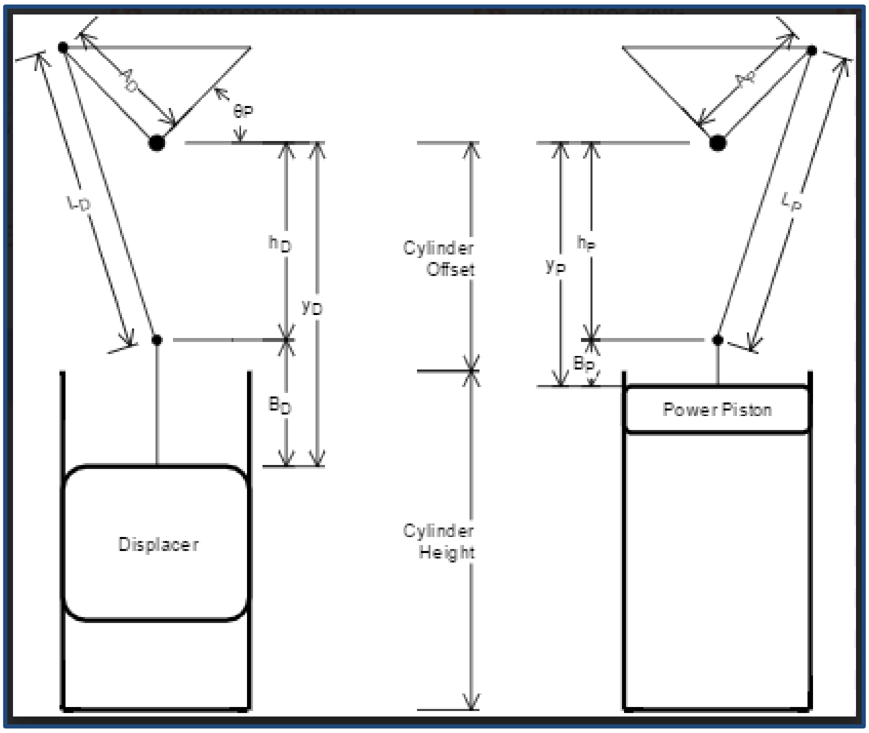



*Figure 23: Variable Diagram for Kinematics*

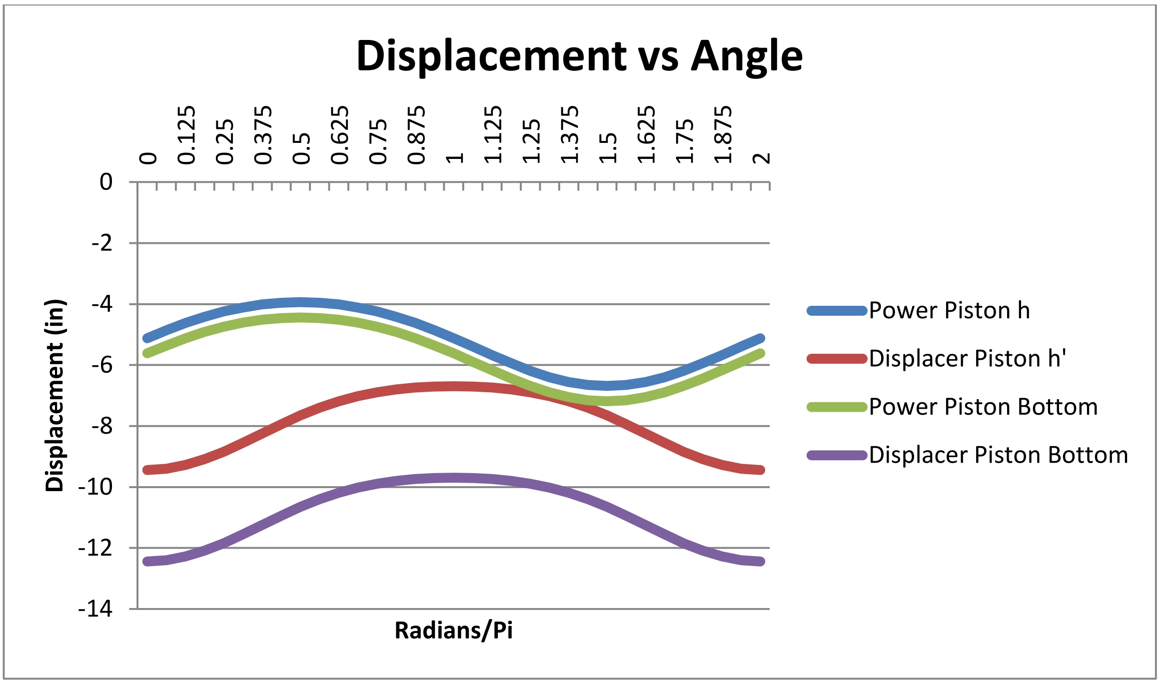



*Figure 24: Displacement vs. Angle*

This graph (Figure 24) is used to predict the locations of the pistons at critical points in the shaft rotation. This data is used to determine the appropriate cylinder height and offset from the crankshaft. Also it indicates the minimum clearance between the piston and the diffuser as well as the diffuser and cylinder cap, which is minimized for optimal engine performance. The graph is generated from the following MatLab calculations in figure 25.

$A_P := 1.375$ $\quad A_D := 1.375$

$L_P := 5$ $\quad L_D := 2.5$

$B_P := .3125$ $\quad B_D := 5.565$

$t_P := .5$

$\theta_P := 0, \frac{\pi}{32} .. 2\pi$

$h_P(\theta_P) := A_P \cdot \sin(\theta_P) - \sqrt{L_P^2 - A_P \cdot \cos(\theta_P)}$

$y_{Ptop}(\theta_P) := h_P(\theta_P) - B_P$

$y_{Pbot}(\theta_P) := y_{Ptop}(\theta_P) - t_P$

$\theta_D(\theta_P) := \theta_P + \frac{\pi}{2}$

$h_D(\theta_P) := A_D \cdot \sin(\theta_D(\theta_P)) - \sqrt{L_D^2 - A_D \cdot \cos(\theta_D(\theta_P))}$

$y_{Dtop}(\theta_P) := h_D(\theta_P) - B_D$

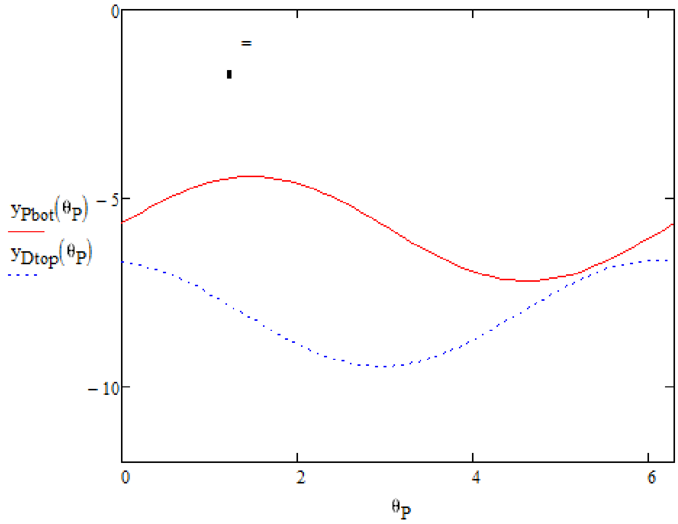



*Figure 25: Kinematic Calculations*

The team met with Professor John Hall to discuss the crankshaft design. Professor Hall provided the team with ideas for measuring the rotational vibration once constructed. He recommended we attach accelerometers on top and beside the bearings in order to measure the natural frequencies of the system. Once analyzed, the shaft can be balanced by adding additional weight to the connectors.

## 3.7 Design of the Cooling System

Incorporating a cooling system into the Stirling Engine design was imperative to increasing the temperature differential and producing the maximum amount of power. Copper tubing was coiled around the top half of both cylinders in order to reduce the temperature in the cool side of the chambers. Cool water will flow from a holding tank through the coiled tube which will be in contact with the outer cylinder walls. A pump will move the water back up into the holding tank when a sensor indicates that the water level is too low in the holding tank.

The design above includes a ¼” (0.635 cm) diameter copper coil that is wrapped around each cylinder 8 times. The wall thickness and thermal conductivity of this copper tube are 0.03” (0.0762 cm) and 385 W/mK respectively. The velocity of the cool water coming out of the tank to a ¼” (0.635 cm) copper tube was calculated using Bernoulli’s equation. State 1 is at the top of the tank and state 2 is at the inlet of the copper pipe:

$$P_1 + \frac{1}{2}\rho(v_1)^2 + \rho g h_1 = P_2 + \frac{1}{2}\rho(v_2)^2 + \rho g h_2$$

$$0 + 0 + \rho g h_1 = 0 + \frac{1}{2}\rho(v_2)^2 + 0$$

$$\rho g h_1 = \frac{1}{2}\rho(v_2)^2$$

$$v_2 = \sqrt{2gh_1} = \sqrt{2\left(9.81\frac{m}{s^2}\right)(0.6096\,m)} = \mathbf{3.46\,m/s}$$

This velocity was used to find the mass and volumetric flowrate of the water at the inlet of the copper pipe:

$$\dot{m} = \rho v A$$

$$\dot{m} = \left(999.97\frac{kg}{m^3}\right)\left(3.46\frac{m}{s}\right)\left(\left(\left(\frac{0.25\,in}{2}\right)\left(\frac{0.0254\,m}{1\,in}\right)\right)^2 \pi\right)$$

$$\dot{m} = \mathbf{0.11\frac{kg}{s}}$$

$$volumetric\ flow\ rate = \frac{0.11\frac{kg}{s}}{999.97\frac{kg}{m^3}}\left(\frac{264.172\,gal}{1\,m^3}\right) = \mathbf{0.03\frac{gal}{s}} = \mathbf{1.8\,gal/min}$$

With this information, the pressure at the beginning and end of the cooling tube were compared. First the pressure at the inlet of the copper tube was calculated:

$$P_{tank\ inlet} = \rho g h = 999.97\frac{kg}{m^3}\left(9.81\frac{m}{s^2}\right)\left(2\ ft\left(\frac{1\ m}{3.28\ ft}\right)\right) = 5981.5\ Pa = 0.868\ psi$$

Next, pressure drop through the length of the cooling tube was considered:

$$\Delta P = f\frac{\rho V^2}{2}\frac{1}{D}L$$

$$f = 0.02\ for\ smooth\ pipe$$

$$\Delta P = (0.02)\frac{\left(999.97\frac{kg}{m^3}\right)(3.46\ m/s)^2}{2}\frac{1}{0.25\ in\ \left(\frac{1m}{39.37\ in}\right)}\left(20\ ft\left(\frac{1m}{3.28\ ft}\right)\right)$$

$$\Delta P = 6.10\ Pa$$

These pressure and volumetric flow rate value determined the necessary pump to move the water bank up into the holding tank. The pump must be able to replace the water flowing out of the tank at 1.8 gal/min. If the tank is 2 meters in the air, it will need a pump with at least 2.85 psi.

# 4.0 Prototype Fabrication

Building the prototype of the Stirling Engine design allowed the team to work through challenges and improve the design. The engineering process involves the need to analyze, assess, and modify to create an overall better design. This was experienced first-hand by the team, working every day to make progress and work through problems provided a great learning experience.

Creating and holding to a build schedule assisted the team with establishing weekly goals and keeping on track with completing the prototype, see Appendix C for the final schedule. This organization was key in looking ahead to make contact with lab assistants and ask for their availability. As the team learned throughout the year that working between four teammate schedules and lab hours was increasingly difficult, but it was important to give plenty of notice to those who went out of their way to help with the project.

A bill of materials is listed in Appendix D, including all parts included in the final prototype. The bill of materials includes the quantity, size, material, manufacturer, part number, and price of every item. A few additional purchased items not used in the final design were not included in the bill of materials.

## 4.1 Building the Cylinders

Beginning the build process, an aluminum cylindrical stock with 3 ½" (8.89 cm) outer diameter and 3 ¼" (8.255 cm) inner diameter was purchased from Speedy Metals. A horizontal band saw was used to cut the stock into two 9" (22.86 cm) cylinders. A 1/16" (0.15875 cm) thick piece of aluminum was then cut to size for the caps. The caps were welded by Adrian to create a seamless cylinder that would allow maximum heat transfer while maintaining structural integrity (Figure 26).

*Figure 26: Welded Cylinder Cap*

The rectangular cylinder flanges were cut out of a 1/8” (0.3175 cm) thick piece of aluminum on the vertical band saw. Then a 3 ½” (8.89 cm) diameter circle was milled out of each flange to fit around the outside of the cylinders (Figure 27). Finally, the flange was welded to the cylinder at an appropriate height to be mounted to the frame.

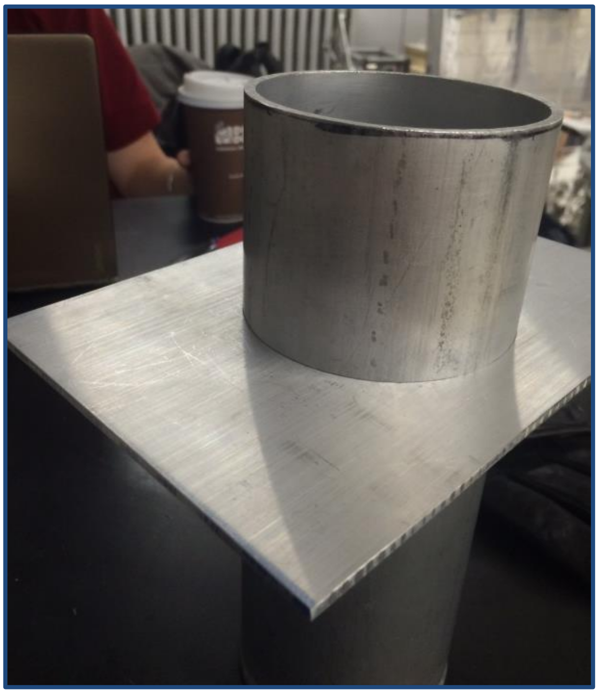


*Figure 27: Cylinder with Flange*

Adjustments were later made to the inside diameter of the cylinders, as there was a tight fit between the pistons and cylinder walls. A honing tool was used to grind down the inside wall.

## 4.2 Milling Pistons

The team milled the pistons using a variety of machines and techniques. Beginning the process, the team purchased two 3” (7.62 cm) long solid aluminum cylinders with a diameter of 3 ¼” (8.255 cm). One end of each piece of stock was turned down using the manual lathe in order to securely fit the stock into the 1” (2.54 cm) collet on the ST-10 lathe in Washburn Shops. Using ESPRIT, a computer aided manufacturing model was developed and operations were created to turn grooves into the piston to hold the piston rings and to drill the center hole in the piston for the U-cup seal. With the help of Mik Tan, the model was refined and operations were performed. Next the piece was put in the MiniMill to face off the top and mill out the piston connectors (Figure 28).

*Figure 28: Milling the Pistons*

Finally, the piston was put into a fourth machine, the VM2, to cut the holes in the piston connectors (Figure 29). These holes will guide the clevis pins in connecting the piston to the push rods. Mik Tan again provided great assistance in this operation. This was the first time any of our teammates had operated this machine and the controls are quite different from other CNC machines previously used in fabrication of this project.


*Figure 29: Piston Pin Holes cut in the VM2*

The piston rings were outsourced to Eriks Seals and Plastics, Inc. The company representatives provided great insight on tolerance issues as well as material selection that will be best suited for this high temperature and high friction application. Additionally, the piston rings work well in both sealing the cylinder and guiding the piston during its reciprocating motion due to the outward pressure of the spring inside the ring onto the cylinder walls. Figure 30 and 31 show the piston rings which are seamless and made to fit into the grooves of the piston. The material of the piston rings is Polytetrafluoroethylene (PTFE) due to low coefficient of friction and high temperature resistance. Additionally, the Eriks SE Design catalog states, “the graphite filler in this blend acts as a dry lubricant making it a good choice in dry running services".

*Figure 30: Piston Rings*

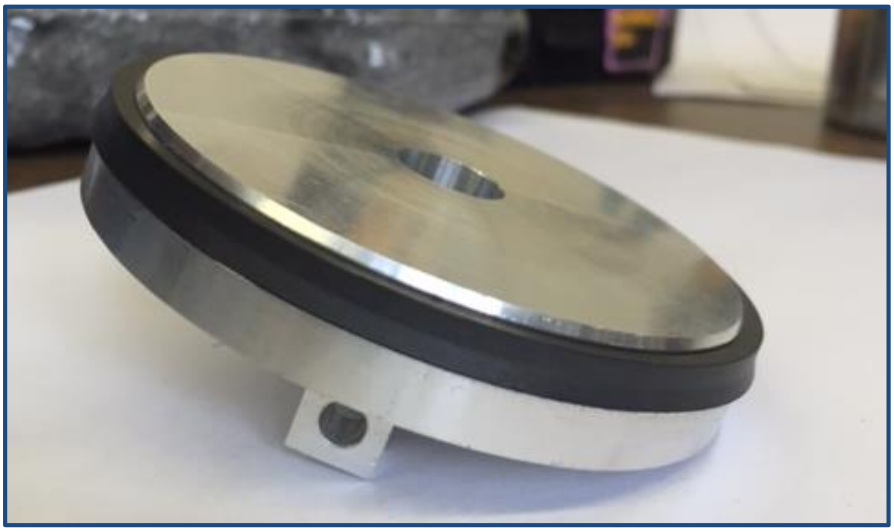


*Figure 31: Piston Rings in Piston Grooves*

The last piece to be added were piston guides. The piston guides were designed to reduce the binding or lateral tilting motion of the pistons along their stroke. The overall design consisted of a clearance hole in the middle for the diffuser push rod, two mounting holes for socket cap screws to thread into the top of the power piston, and two small slots to house individual stainless steel bearings (figure 32). The radius of the curved edge is equivalent to that of the power piston and the piece fits snuggly around the connection bosses of the power piston push rods. The stainless steel bearings were incorporated to provide a guidance support with less friction. A height of 0.75in was chosen where there would be enough material to fully support the piston horizontally while at the same time not interfering with other parts of the assembly. These piston guides were then 3D printed on the MakerBot Replicator 2X in Washburn Shops. Next, they were sanded for a better fit and mounted to the top of the piston using two #10-24 socket head cap screws.

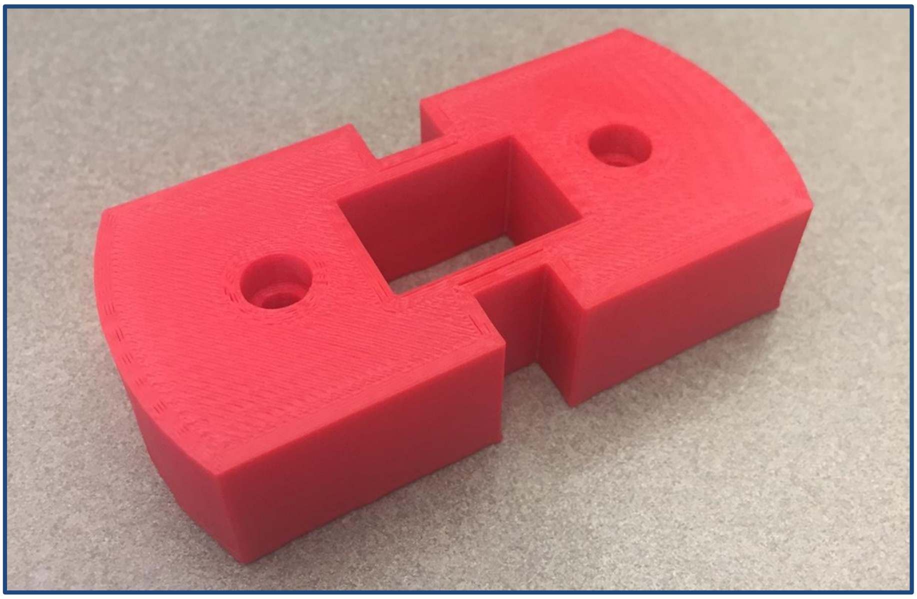


*Figure 32: Piston Guides*

## 4.3 3D Printing Diffuser

Additive manufacturing was explored to prototype this unique shape in a cost effective and efficient manner. Since the diffuser will be experiencing high temperatures, up to 200°C, thermal properties of the material were greatly considered. 3D printing a material able to withstand these temperatures was a large concern, but the team decided to print a few pieces to test. The MakerBot Replicator 2X in Washburn Shops was used to print a diffuser and two pistons (Figure 33).

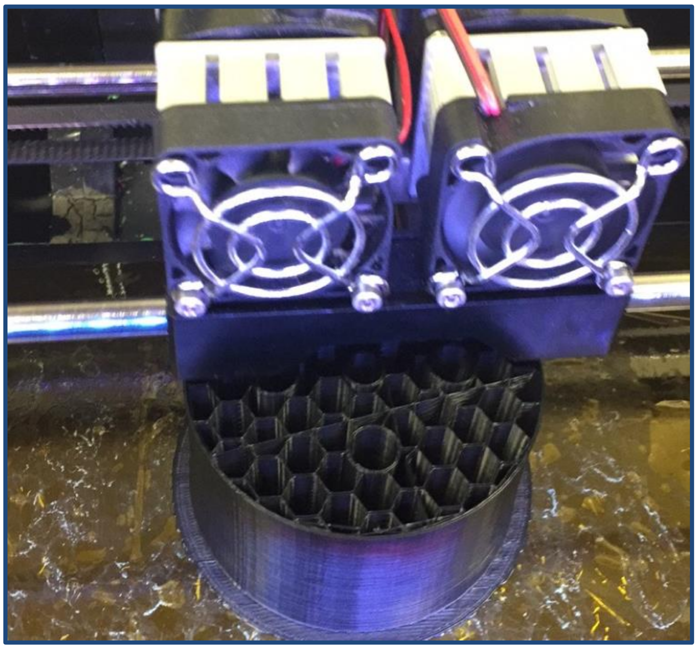


*Figure 33: 3D Printing the Diffuser*

A thermal protectant spray was applied in order to help prevent deformation and melting. DEI High-Temp Silicone Coating Spray was purchased and applied to a few test pieces. The spray was advertised to provide heat protection up to 1500°F, or 815°C. The coating was to be cured at about 400°F, or 200°C. The result of the curing process was extreme deformation, with parts shrinking significantly in size (Figure 34). This led to the conclusion that 3D printed parts would not be able to sustain thermal loads encountered in this Stirling Engine. Not only is there a risk of melting, but parts wouldn't keep their size and would change the volume differentials that the system was designed upon.

*Figure 34: Deformed 3D Printed Parts*

## 4.4 Redesign and Construction of Diffuser

Due to the significant deformation of the 3D printed diffuser at 200°C, the decision to machine the diffuser out of aluminum had to be made. A hollow aluminum cylinder with an outer diameter of 3 ¼” (8.255 cm) and a wall thickness of 1/8” (0.3175 cm) was chosen to be the base cylindrical walls of the diffuser. To avoid suffocating the working fluid in the heated section as the diffuser reciprocates, the outer diameter of the hollow cylinder needed to be turned down to allow a reasonable clearance between the wall of the diffuser and the inside wall of the engine cylinders. A clearance of 1/16” (0.0625 cm) was chosen based on previous volume calculations that reduced the maximum amount of dead space within the cylinders. The hollow cylinder was next cut to the correct equal lengths of 3” (7.62 cm) per diffuser with the horizontal band saw and was ready for the manual lathe. The diffuser wall cylinder was turned down from 3 ¼” (8.255 cm) outer diameter to 3 1/8” (7.9375 cm) outer diameter through the use of the manual lathe in the Higgins machine shop. A challenging part of this process was determining the correct amount of material the lathe should remove during each of its passes. Taking off very small amounts at a time was a bit time consuming but necessary to preserve the part while it was being machined.

The next step in the construction of the aluminum diffuser was machining the aluminum caps to close the diffuser as well as provide guides for the regeneration tubes. A 1/16” (0.15875 cm) sheet of aluminum stock was purchased and divided into 4 equal squares for the top and bottom caps of both diffusers. Next, the center of each of those squares was found through simple geometric diagonals and each center was marked with a small metal punch tool. A concentric circle was drawn inside of each square to a diameter of 3 1/8” (7.9375 cm) to match the outer diameter of the diffuser walls and was cut out using the vertical band saw. Using the already punched centers of these circular caps, the four holes for the regeneration tubes were drawn into each of the four caps and drilled to desired diameter of 3/8" (0.9525 cm). Following that, the determined center for each top cap was drilled to a diameter of 5/16" (0.79375 cm) to match the size of the diffuser pushrod connecting to the crankshaft.

The last pieces to prepare for the assembly of the diffuser was a long aluminum pipe with a diameter of 3/8" (0.9525 cm) to be used as the regeneration tubes and a long steel 5/16" (0.79375 cm) for the pushrods. The pipe was cut to 8 equal sections of 3" (7.62 cm) a piece. The next machining process was to create the pushrod. The main challenge faced by the diffuser pushrod was how to fix it to the diffuser to allow for complete vertical reciprocation. The decision was made to weld an aluminum nut onto the one of the two caps for each diffuser and then thread the steel pushrod to ¼" x 20 thread to match the nut. The correct determination of the centers of each diffuser cap was pivotal for this process to work efficiently. The steel pushrod was formed from a long 5/16" (0.79375 cm) steel rod that was cut to length of machined down to 5/16" (0.79375 cm) with a tolerance of ±0.005" to properly reciprocate through the U-cup seal in the piston.

*Figure 35: Welded Diffuser*

From here we were ready to weld the diffuser walls, caps, and pushrods together to form once piece (Figure 35). Initially, the diffuser cap without the aluminum nut as well as all four regeneration tubes were welded to the diffuser wall. Next, the bottom cap with the nut was welded on and the rod was fixed into place by screwing it into the aluminum nut. From here the diffusers were fully assembled but still had no means to connecting to the links of the crankshaft. To solve this, a steel fork was deigned to be welded onto the top of the pushrod, which would have a pin connecting it to the crank link.

*Figure 36: Diffuser Rod Connector*

The fork (Figure 36) was milled from a small block of steel and took a total of three operations on the MiniMill in Washburn Shops. The first was to create a hole for a pin to connect the diffuser to the crank. The second operation was to create the slot for the crank link to move between during the engine’s rotation. Lastly, a hole was created on the bottom face to the same diameter of the pushrod so the fork could be welded to the rod.

## 4.5 Manufacturing the Crankshaft

Building the crankshaft was the most challenging aspect of the process due to the high importance of accuracy and balance. The first iteration of building the crankshaft resulted in the shaft not aligning on one axis. The first part of this process was turning all rods on the ST-10 Lathe so that they would be as precise as possible (Figure 37).

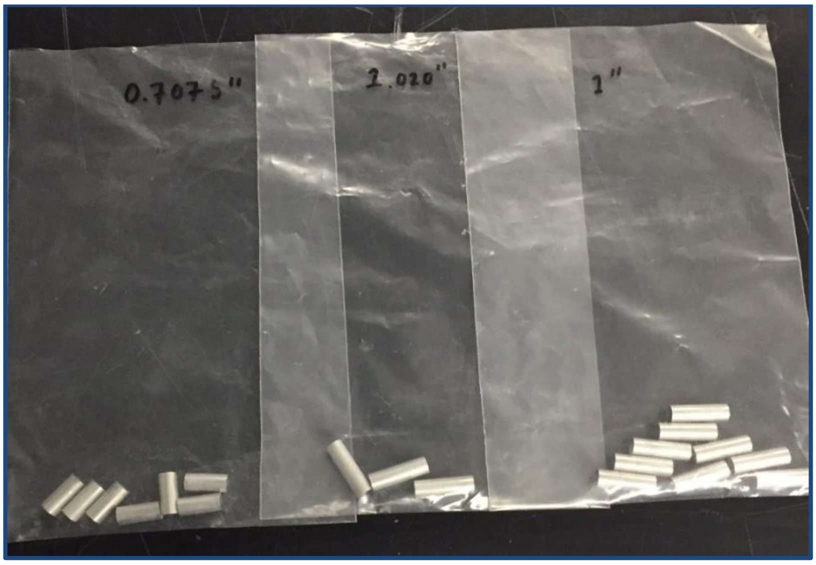



*Figure 37: Crankshaft Links*

Next, the rods were threaded (Figure 38) using a 1/16” (0.15875 cm) NPT die that would allow the rods to securely fit into the 90 degree connectors incorporated into the design.

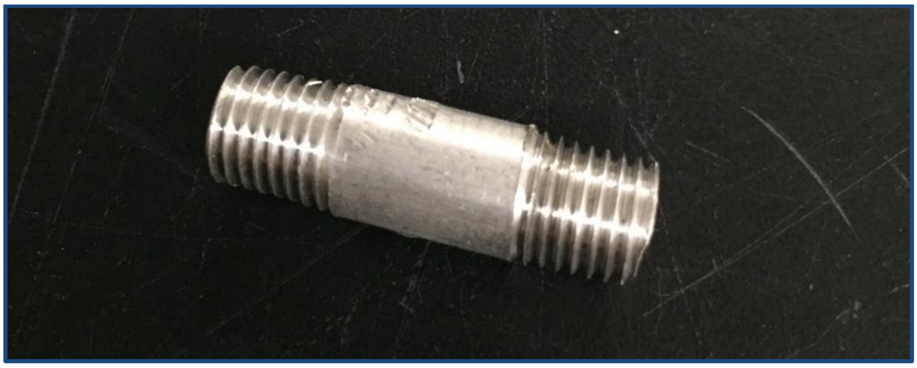


*Figure 38: Threaded Crankshaft Link*

This process allowed the team to see that the 90 degree connectors created more problems than solutions as they added significant mass and required unique NPT threading that was beyond our abilities to thread accurately. Since the threading could not be repeated reliably from rod to rod, the crankshaft was ultimately off centered (Figure 39).

Figure 39: Misaligned Threading & Connectors

The second iteration was very similar to the first but eliminated the need for the 90 degree connectors and NPT threads. Instead steel rods were turned on the ST-10 Lathe in Washburn Shops in order to create a geometry that could be compression fit into place with flat links. Mik Tan assisted the team in creating the computer aided manufacturing files and well as operating the machines. The compression fit rods were then threaded and reinforced onto the links with washers and nuts (Figure 41).

Figure 40: Compression Fit Rods

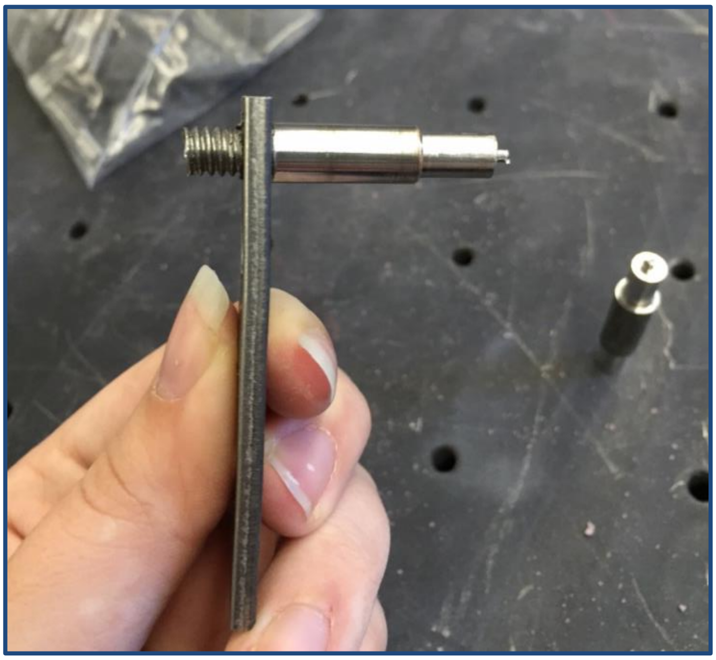


*Figure 41: Rods & Flat Links*

Additionally, more steel links were machined by Justin Fahie's former co-worker, Rick Stanley, at his machine shop in Northbridge, MA. This allowed the team to get some outside expertise and continue working on the project while Washburn Shops was closed over winter break. The links can be seen in figure 42 and dimensioned drawings of these pieces can be found in Appendices B10 and B11.

The final step in building the crankshaft involved reworking the links again as the team discovered the compression fits would not be sufficient enough to prevent the pieces from turning in place. The ends of each rod were cut to a shorter length in order to increase clearance for the push rods, and then welded into place by Adrian. Figure 42 shows the crank in process of being assembled. This assembly challenged the team as it was necessary to ensure each piece was assembled correctly before permanently fixing them in place. In order to do this, the team used a variety of magnets and squares to secure and align each piece in its proper position. The first step in the process was to use the arbor press to press each link into their respective links. Then, the link was repositioned and secured with magnets at 90 degrees to the link and welded in place. Before the next link was welded onto the opposite side of each link, the push rods for the diffuser and power pistons were attached. Then the process was repeated for welding each additional rod and link in place. The process was very tedious in regard to checking and confirming position, however, also a very necessary and vital process to the manufacturing the crankshaft as a whole.

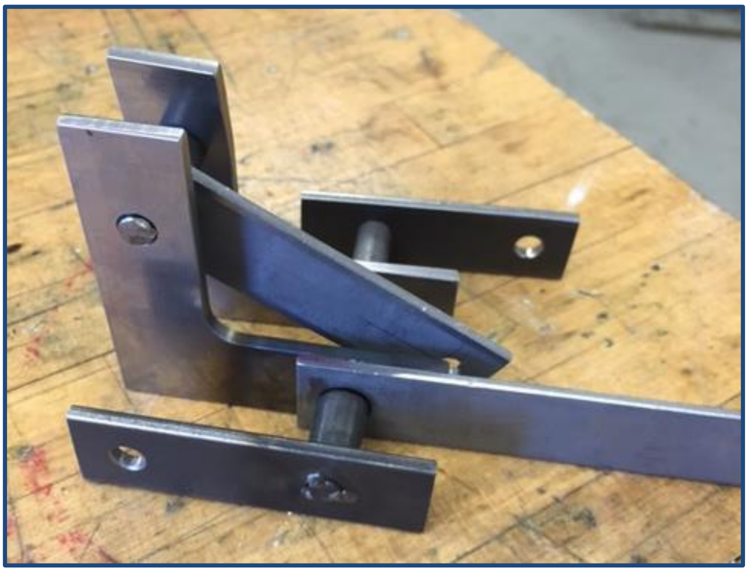


*Figure 42: Constructing the Crank*

## 4.6 Frame and Armature

The frame (Figure 43) was designed around the other components to ensure it supported all the engines requirements. The cylinders had to be a certain distance apart to align with the crank as well as a certain distance from the rotating axis of the crank to account for the right stroke length. Legs were added to the frame to help reduce rocking and provided space for the burner to fit underneath the bottom of the cylinders.

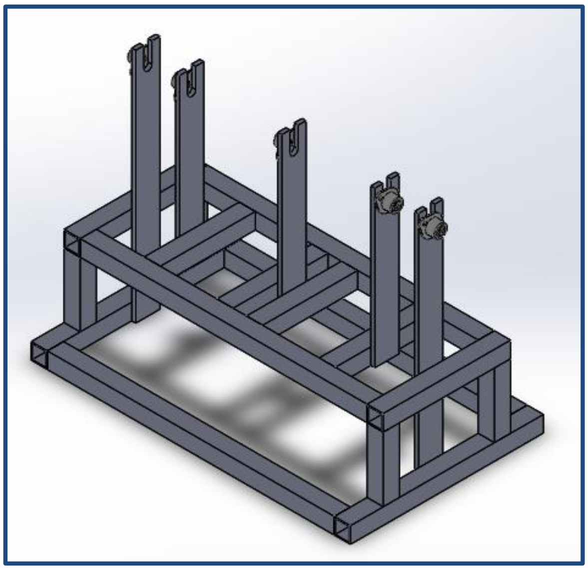


*Figure 43: Frame Design*

Square steel tubes were selected as the material for the frame as it is strong and readily available at Home Depot. This material ensured that the frame was both rigid and heavy so that the engine would not move or deform during operation. Other properties considered included thermal resistance since the frame will be directly subjected to the burner. Also it is important that the material is easy to work with and machine.

Finally, it was determined that the frame should be able to disassemble. The purpose of this requirement is so that during the building process the frame needs to be put together and taken apart multiple times to ensure alignment while machining certain components. Because of this the frame could not be welded. Instead, brackets were bolted at each junction and secured with nuts (Figure 44).

*Figure 44: Disassembled Engine*

Construction started with taking the steel square tubes and cutting each one to the required length as per the team's designs. Next, holes were drilled at appropriate distances along the tubes in order to connect the pieces with 90 degree brackets. All pieces were attached with bolts and nuts. The four legs were welded to the bottom of the frame for added stability. Arms were cut to length from steel flats, and slots were milled in order to allow the crank to slide in and out of the system. The arms were then welded to the frame in a way to ensure alignment and stability, as seen in figure 45. This was the most challenging part of this process as all five arms had to be aligned to one another and at an appropriate height to support the crank in the right position (Figure 46). Finally, rubbed pads were added to the bottom of the legs to prevent sliding.

*Figure 45: Constructed Frame*

*Figure 46: Aligning Arms*

# 5.0 Testing

Two simple tests were conducted to gain performance data. An applied torque of 115 in-lbs (13 Nm) was added to start the engine and then observe both its rotation and temperature while in operation.

## 5.1 Rotation Test

A tachometer was used to measure rotation of crank with and without the heat source, as see in figure 47. Rotation increased 27%, from 71 RPM to 90 RPM, with the added heat source.

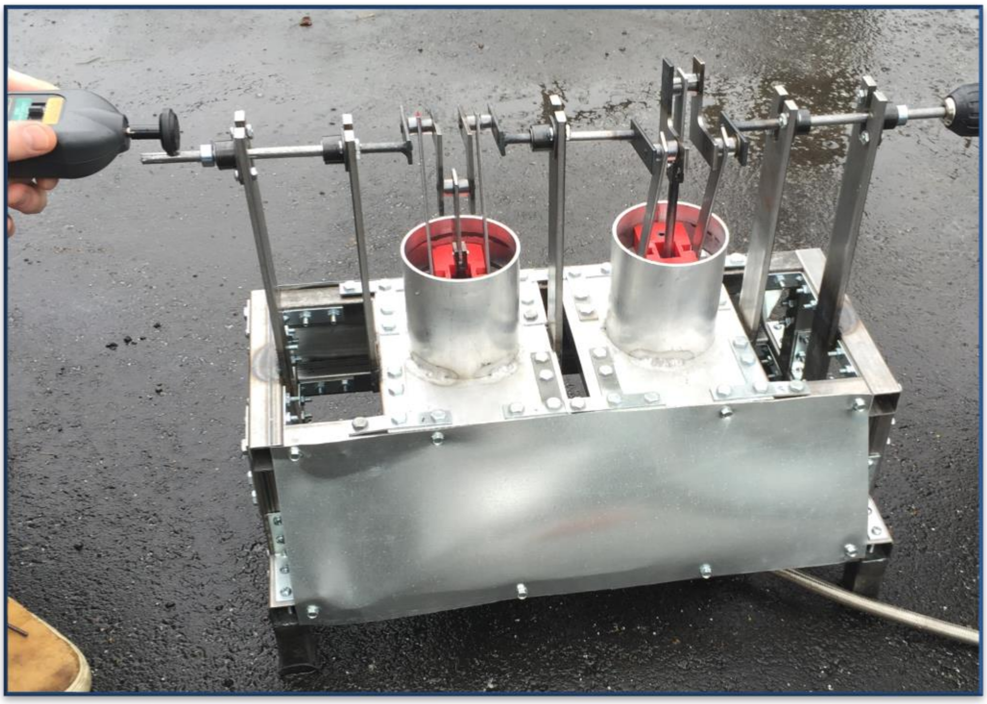


*Figure 47: Rotation Test*

## 5.2 Thermal Test

Temperature was measured using an infrared thermometer and a thermocouple, as seen in figure 48. The maximum temperature was 220°C at the bottom of the cylinder, 100°C in the middle of the cylinder, and 62°C at the top of the cylinder. This shows the large temperature differential across the length of the cylinder, which is necessary for the Stirling cycle.

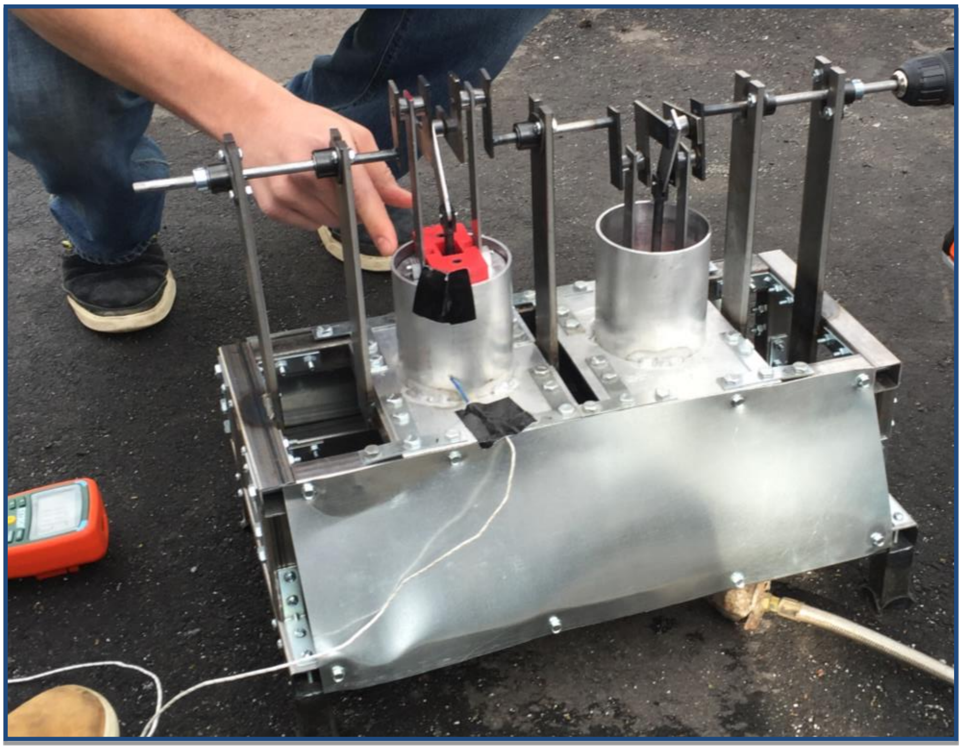


*Figure 48: Thermal Test*

## 6.0 Results & Recommendations

Testing with the heat source resulted in a 27% increase in rotation, thus the internal pressure of the system converted the heat energy into mechanical work. The power calculated from this test was 30 Watts. While this is not as high as the initial goal of 150 Watts, we were pleased with this proof of concept that energy is being transferred within the system. With some improvements, this design could achieve the initial goal.

The following recommendations to improve this design are proposed:

- Manufacture a single-stock crankshaft to facilitate axis alignment
- Design and manufacture a crankshaft with removable pushrods
- Counterbalance piston mass by strategically adding weight on the crank
- Incorporate alignment guides for both the piston and diffuser along its stroke

Thermal testing showed a temperature differential of 158°C, demonstrating the temperature change throughout the system. This temperature change is essential to the Stirling cycle, and our team was successful at creating this temperature differential due to our heat source orientation, heat shield, and diffuser design. To increase the heat differential even more the cooling system should be constructed as outlined in the design methodology section.

This major qualifying project provided challenges that led to lessons learned in design and manufacturing, combining theory and practice in a way that was never before experienced during our academic careers. Again, we would like to thank all of the WPI faculty and staff who helped the team in this project.

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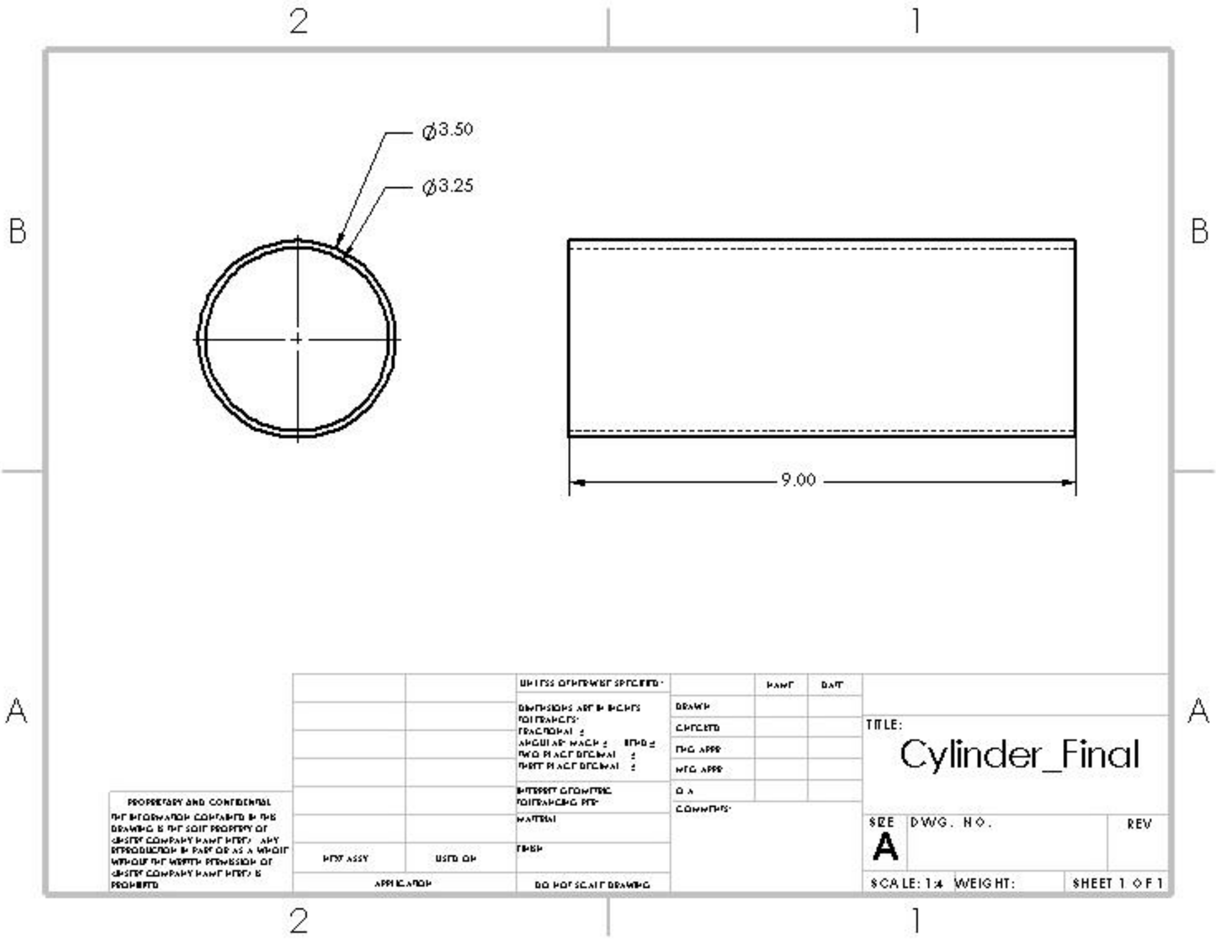
# Appendix A – Angular Displacement of the Crankshaft

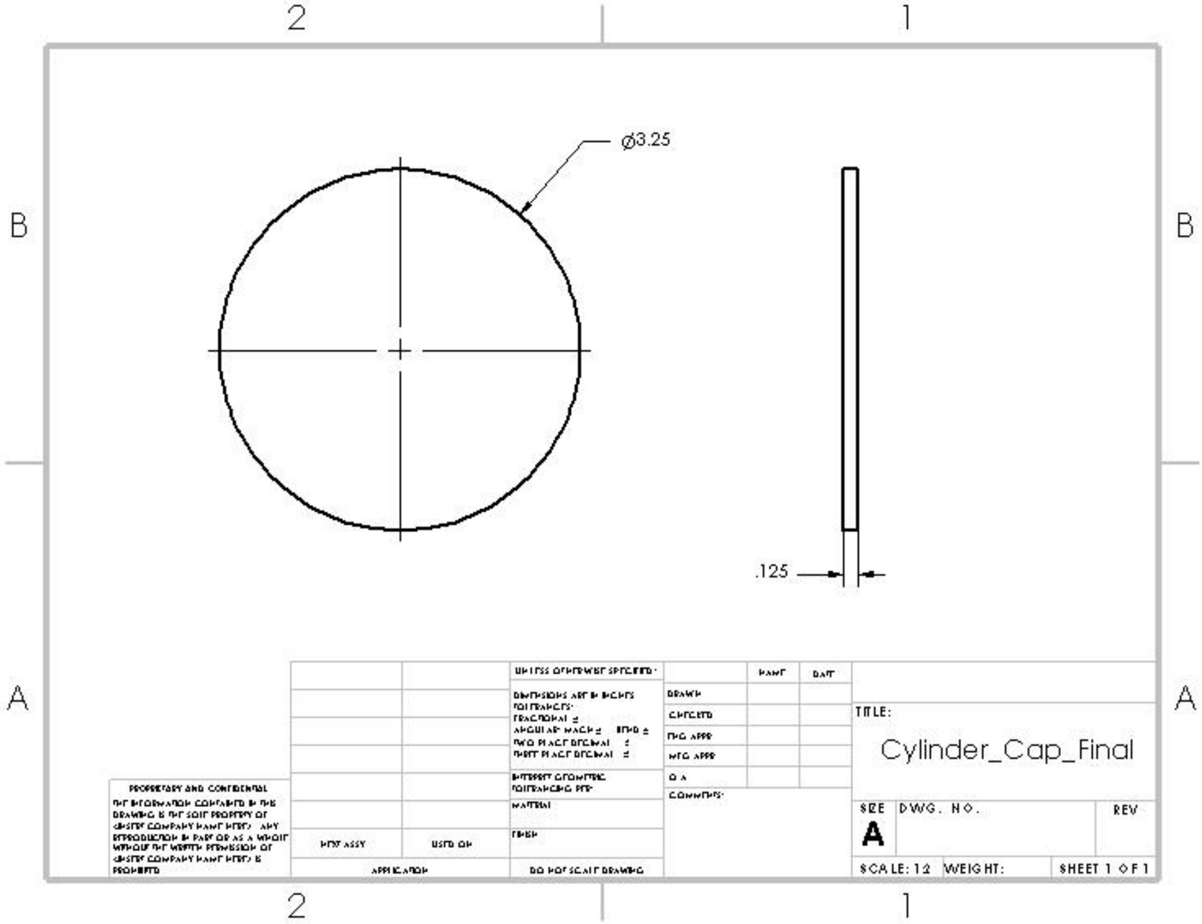
| Theta/Pi | H | Change in h | L | B | h' | Change in h' | L' | B' | hBottom | Dist Pistons | h'Bottom |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 0 | -5.1197 | -4.807 | 5 | 1.375 | -9.44 | -3.875 | 2.5 | 1.375 | -5.62 | 3.82028 | -12.44 |
| 0.0625 | -4.859 | -4.546 | 5 | 1.375 | -9.399 | -3.834 | 2.5 | 1.375 | -5.359 | 4.0402 | -12.3991 |
| 0.125 | -4.6222 | -4.31 | 5 | 1.375 | -9.279 | -3.714 | 2.5 | 1.375 | -5.122 | 4.15709 | -12.2793 |
| 0.1875 | -4.4161 | -4.104 | 5 | 1.375 | -9.089 | -3.524 | 2.5 | 1.375 | -4.916 | 4.17257 | -12.0887 |
| 0.25 | -4.2448 | -3.932 | 5 | 1.375 | -8.84 | -3.275 | 2.5 | 1.375 | -4.745 | 4.09568 | -11.8405 |
| 0.3125 | -4.1105 | -3.798 | 5 | 1.375 | -8.552 | -2.987 | 2.5 | 1.375 | -4.611 | 3.94165 | -11.5522 |
| 0.375 | -4.0144 | -3.702 | 5 | 1.375 | -8.244 | -2.679 | 2.5 | 1.375 | -4.514 | 3.72998 | -11.2444 |
| 0.4375 | -3.9567 | -3.644 | 5 | 1.375 | -7.938 | -2.373 | 2.5 | 1.375 | -4.457 | 3.4816 | -10.9383 |
| 0.5 | -3.9375 | -3.625 | 5 | 1.375 | -7.653 | -2.088 | 2.5 | 1.375 | -4.438 | 3.21541 | -10.6529 |
| 0.5625 | -3.9567 | -3.644 | 5 | 1.375 | -7.402 | -1.837 | 2.5 | 1.375 | -4.457 | 2.9451 | -10.4018 |
| 0.625 | -4.0144 | -3.702 | 5 | 1.375 | -7.192 | -1.627 | 2.5 | 1.375 | -4.514 | 2.6776 | -10.192 |
| 0.6875 | -4.1105 | -3.798 | 5 | 1.375 | -7.024 | -1.459 | 2.5 | 1.375 | -4.611 | 2.41383 | -10.0244 |
| 0.75 | -4.2448 | -3.932 | 5 | 1.375 | -6.896 | -1.331 | 2.5 | 1.375 | -4.745 | 2.15113 | -9.89592 |
| 0.8125 | -4.4161 | -4.104 | 5 | 1.375 | -6.802 | -1.237 | 2.5 | 1.375 | -4.916 | 1.88603 | -9.80216 |
| 0.875 | -4.6222 | -4.31 | 5 | 1.375 | -6.739 | -1.174 | 2.5 | 1.375 | -5.122 | 1.61642 | -9.73866 |
| 0.9375 | -4.859 | -4.546 | 5 | 1.375 | -6.702 | -1.137 | 2.5 | 1.375 | -5.359 | 1.34304 | -9.70199 |
| 1 | -5.1197 | -4.807 | 5 | 1.375 | -6.69 | -1.125 | 2.5 | 1.375 | -5.62 | 1.07028 | -9.69 |
| 1.0625 | -5.3954 | -5.083 | 5 | 1.375 | -6.702 | -1.137 | 2.5 | 1.375 | -5.895 | 0.80654 | -9.70199 |
| 1.125 | -5.6746 | -5.362 | 5 | 1.375 | -6.739 | -1.174 | 2.5 | 1.375 | -6.175 | 0.56404 | -9.73866 |
| 1.1875 | -5.9439 | -5.631 | 5 | 1.375 | -6.802 | -1.237 | 2.5 | 1.375 | -6.444 | 0.35821 | -9.80216 |
| 1.25 | -6.1893 | -5.877 | 5 | 1.375 | -6.896 | -1.331 | 2.5 | 1.375 | -6.689 | 0.20659 | -9.89592 |
| 1.3125 | -6.3971 | -6.085 | 5 | 1.375 | -7.024 | -1.459 | 2.5 | 1.375 | -6.897 | 0.12729 | -10.0244 |
| 1.375 | -6.5551 | -6.243 | 5 | 1.375 | -7.192 | -1.627 | 2.5 | 1.375 | -7.055 | 0.13694 | -10.192 |
| 1.4375 | -6.6539 | -6.341 | 5 | 1.375 | -7.402 | -1.837 | 2.5 | 1.375 | -7.154 | 0.24795 | -10.4018 |
| 1.5 | -6.6875 | -6.375 | 5 | 1.375 | -7.653 | -2.088 | 2.5 | 1.375 | -7.188 | 0.46541 | -10.6529 |
| 1.5625 | -6.6539 | -6.341 | 5 | 1.375 | -7.938 | -2.373 | 2.5 | 1.375 | -7.154 | 0.78444 | -10.9383 |
| 1.625 | -6.5551 | -6.243 | 5 | 1.375 | -8.244 | -2.679 | 2.5 | 1.375 | -7.055 | 1.18932 | -11.2444 |
| 1.6875 | -6.3971 | -6.085 | 5 | 1.375 | -8.552 | -2.987 | 2.5 | 1.375 | -6.897 | 1.65511 | -11.5522 |
| 1.75 | -6.1893 | -5.877 | 5 | 1.375 | -8.84 | -3.275 | 2.5 | 1.375 | -6.689 | 2.15113 | -11.8405 |
| 1.8125 | -5.9439 | -5.631 | 5 | 1.375 | -9.089 | -3.524 | 2.5 | 1.375 | -6.444 | 2.64475 | -12.0887 |
| 1.875 | -5.6746 | -5.362 | 5 | 1.375 | -9.279 | -3.714 | 2.5 | 1.375 | -6.175 | 3.10471 | -12.2793 |
| 1.9375 | -5.3954 | -5.083 | 5 | 1.375 | -9.399 | -3.834 | 2.5 | 1.375 | -5.895 | 3.5037 | -12.3991 |
| 2 | -5.1197 | -4.807 | 5 | 1.375 | -9.44 | -3.875 | 2.5 | 1.375 | -5.62 | 3.82028 | -12.44 |

# Appendix B – Drawings

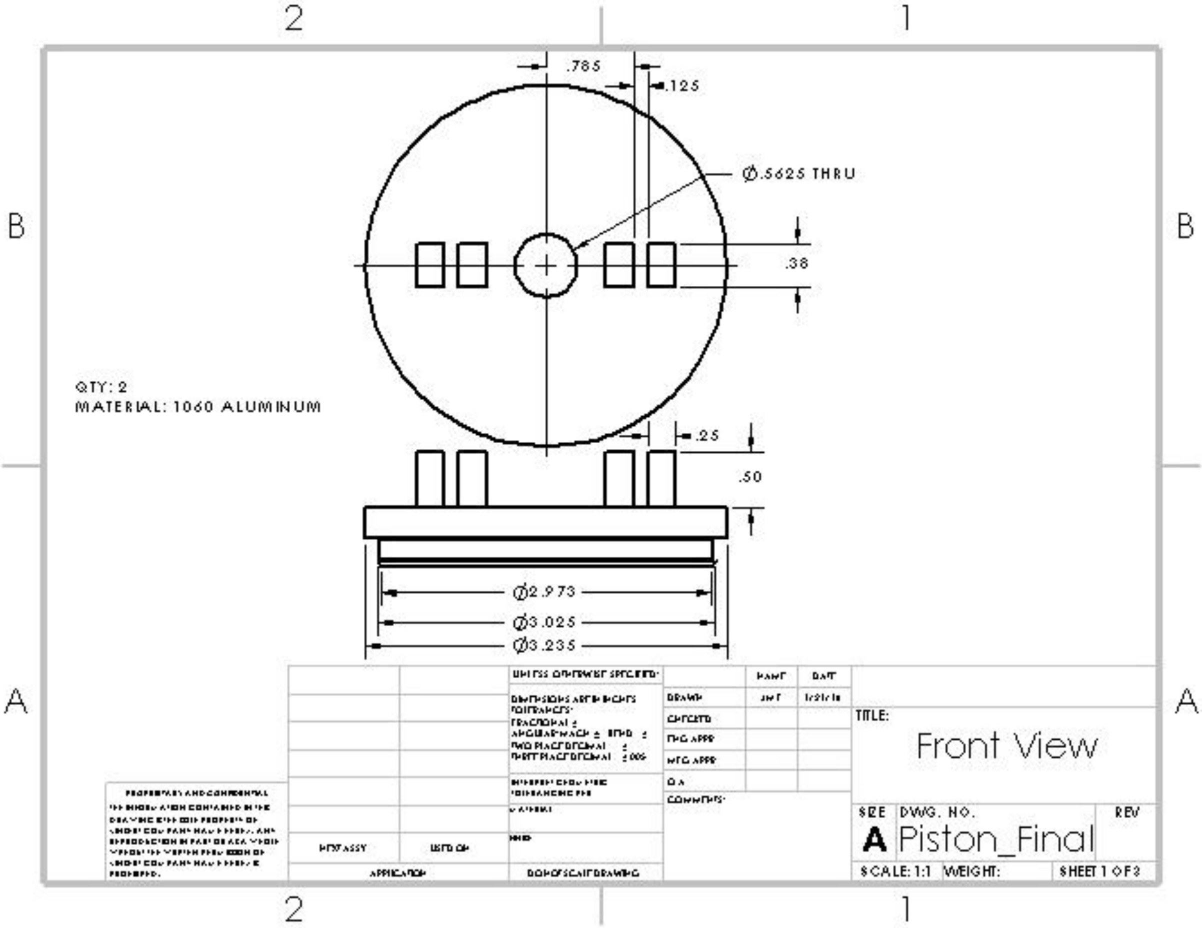
## B1: Cylinder



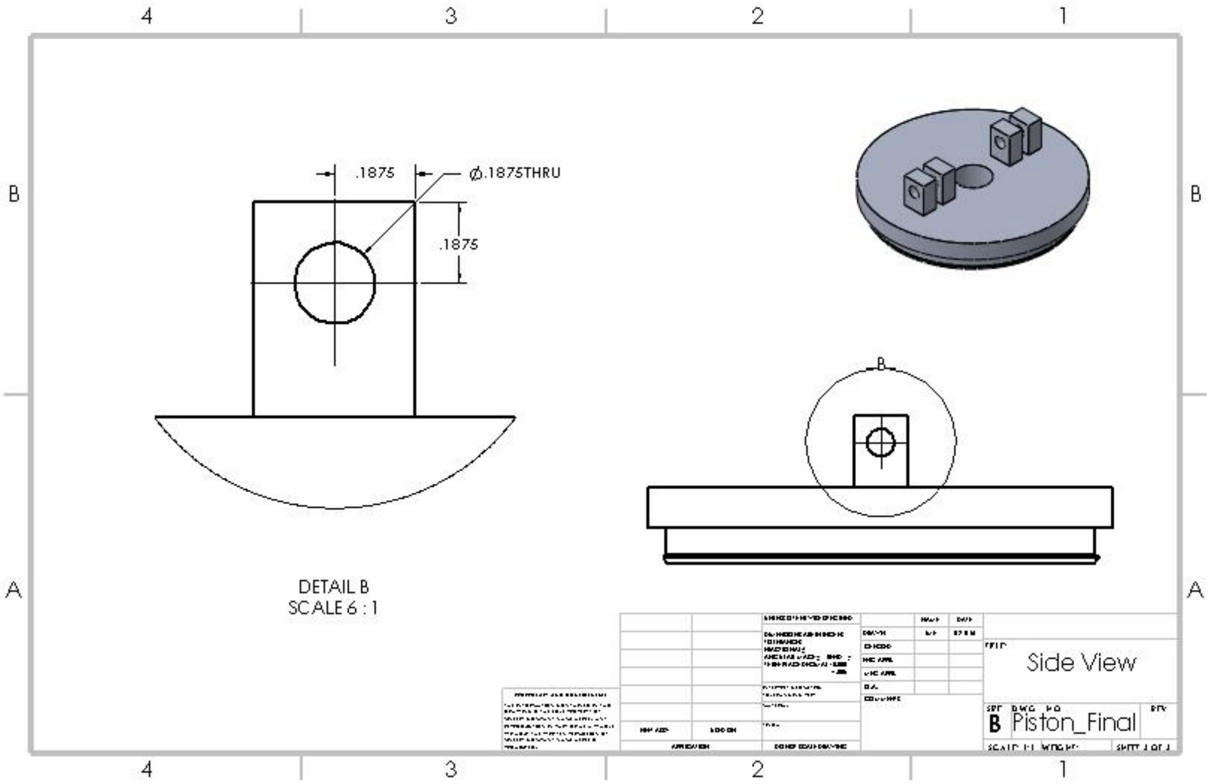
## B2: Cylinder Cap



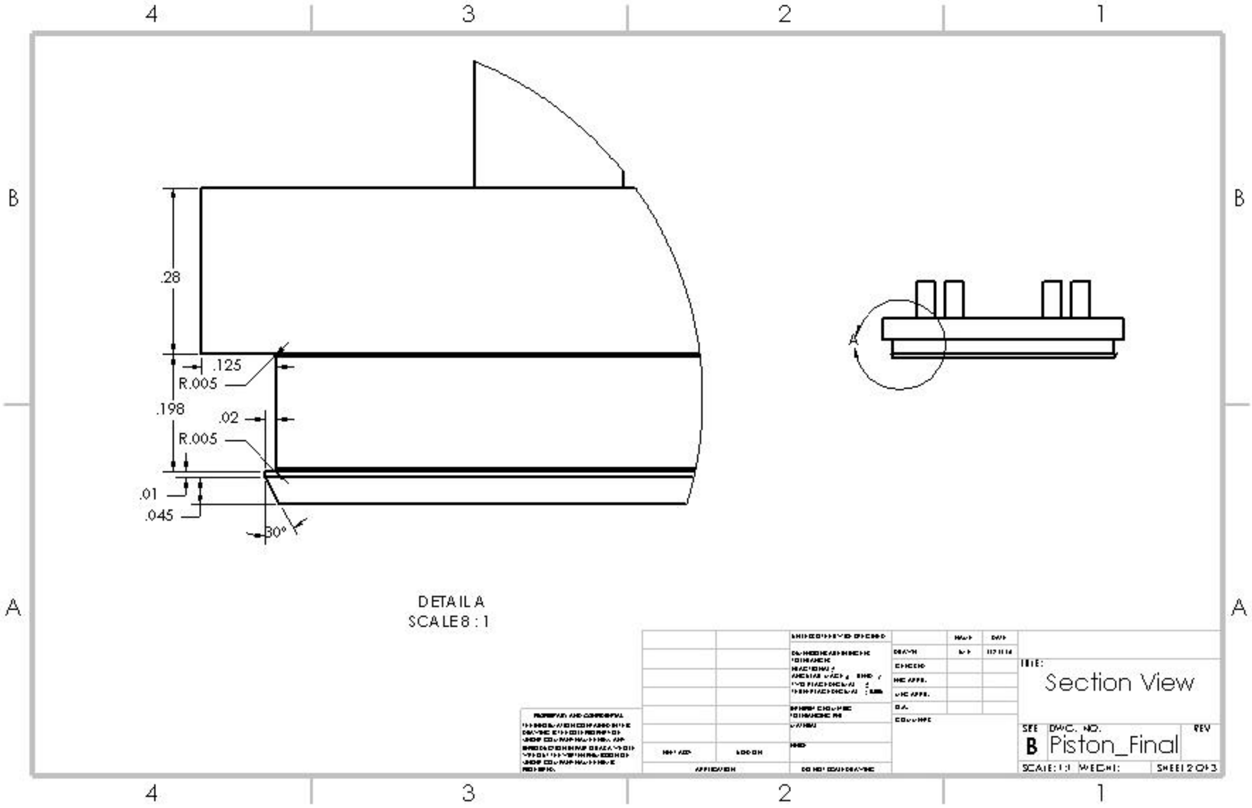
## B3: Piston Front



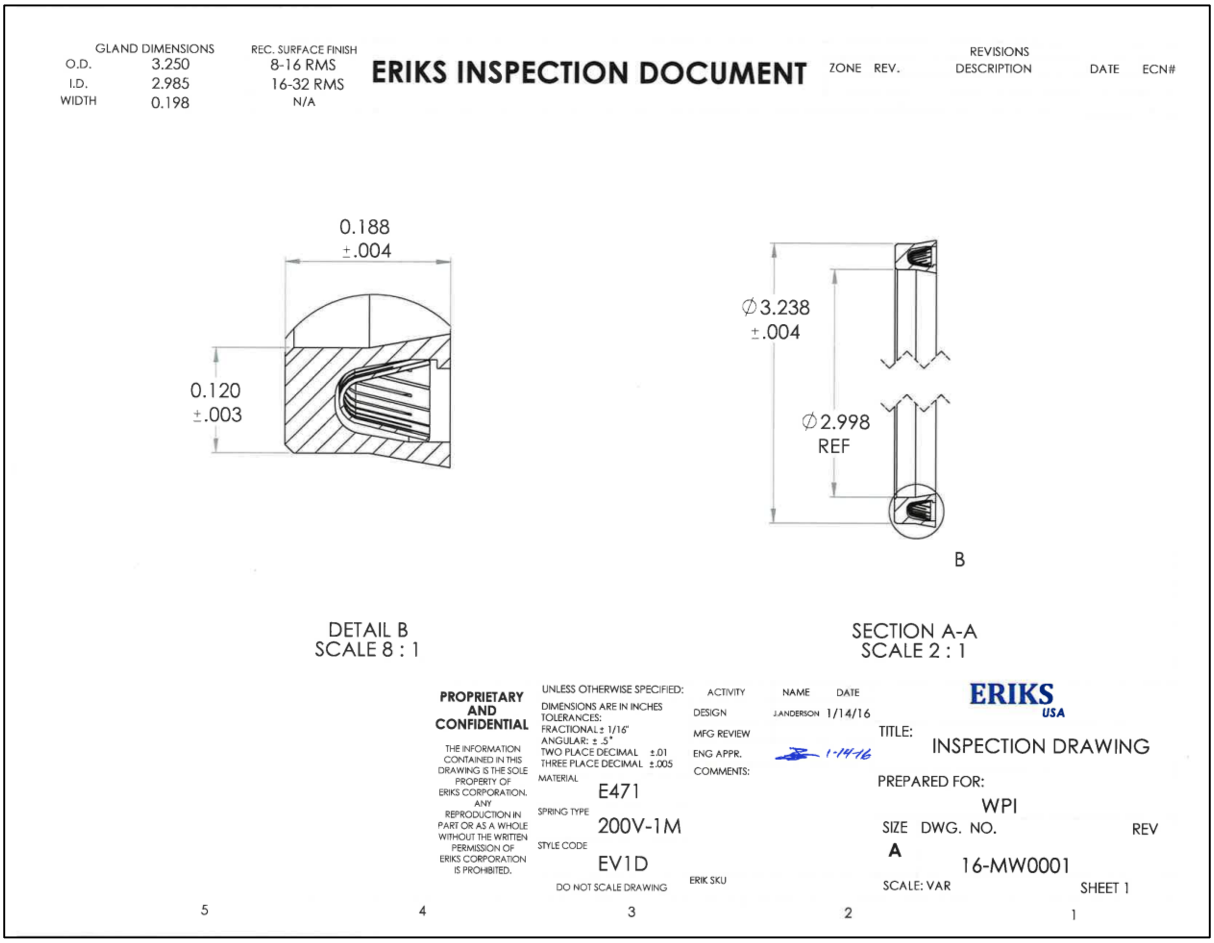
## B4: Piston Side View



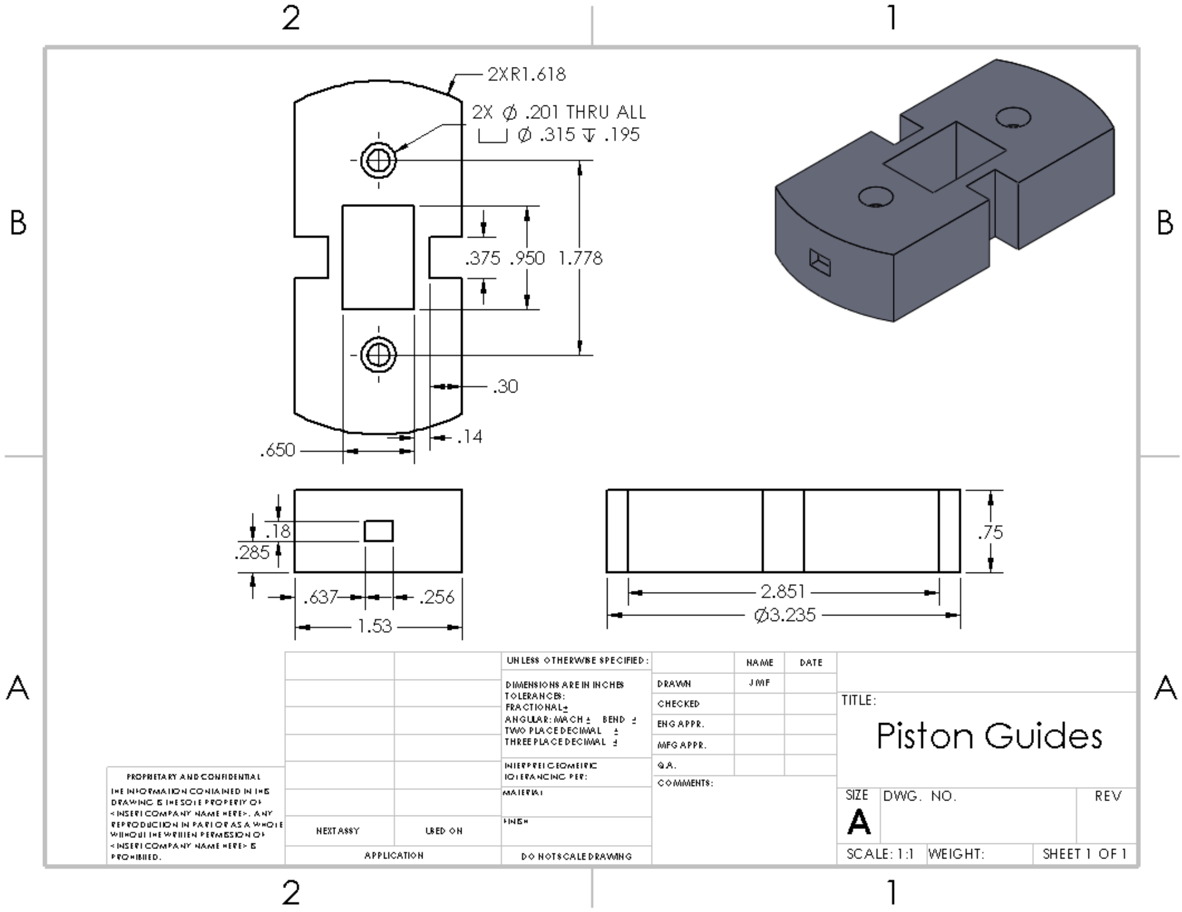
## B5: Piston Section View



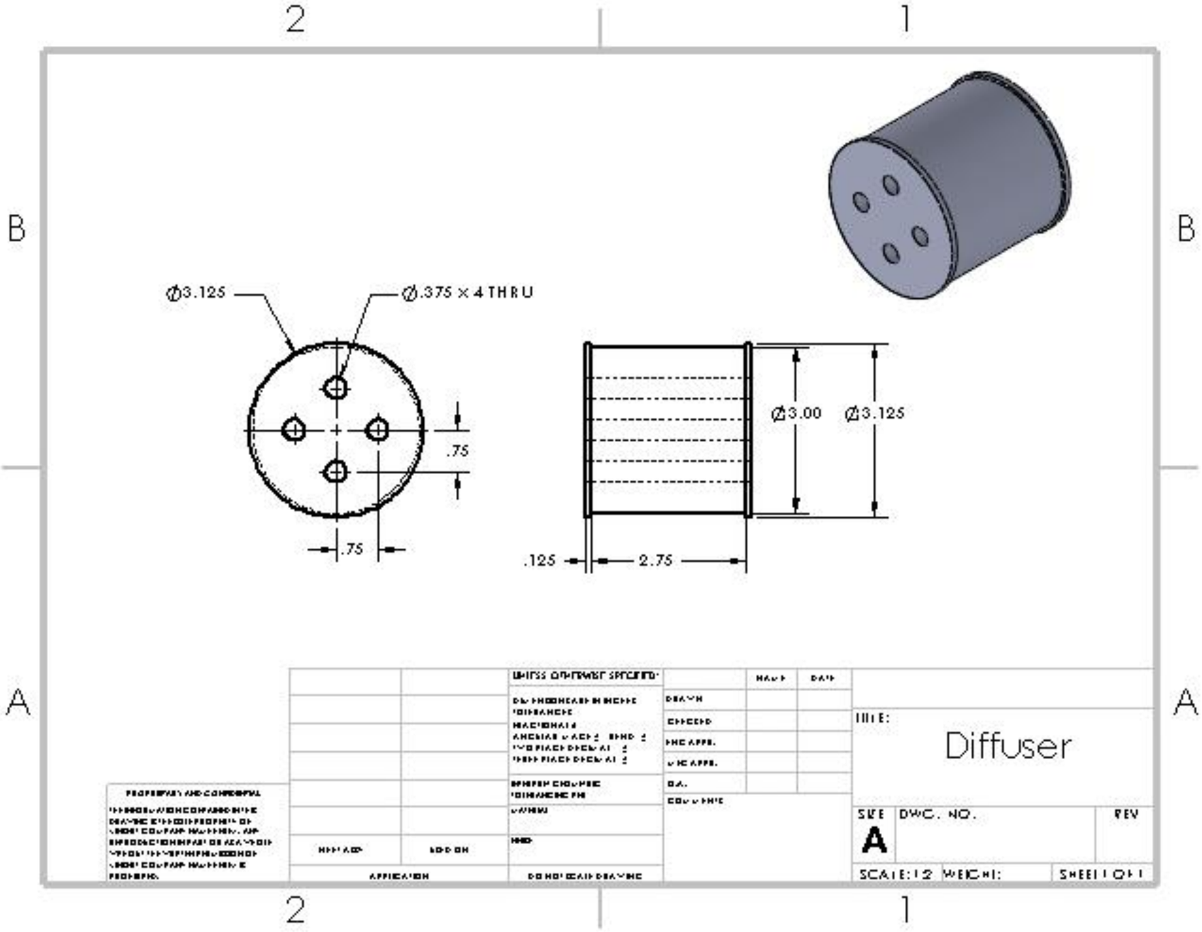
## B6: Piston Ring



# B7: Piston Guides

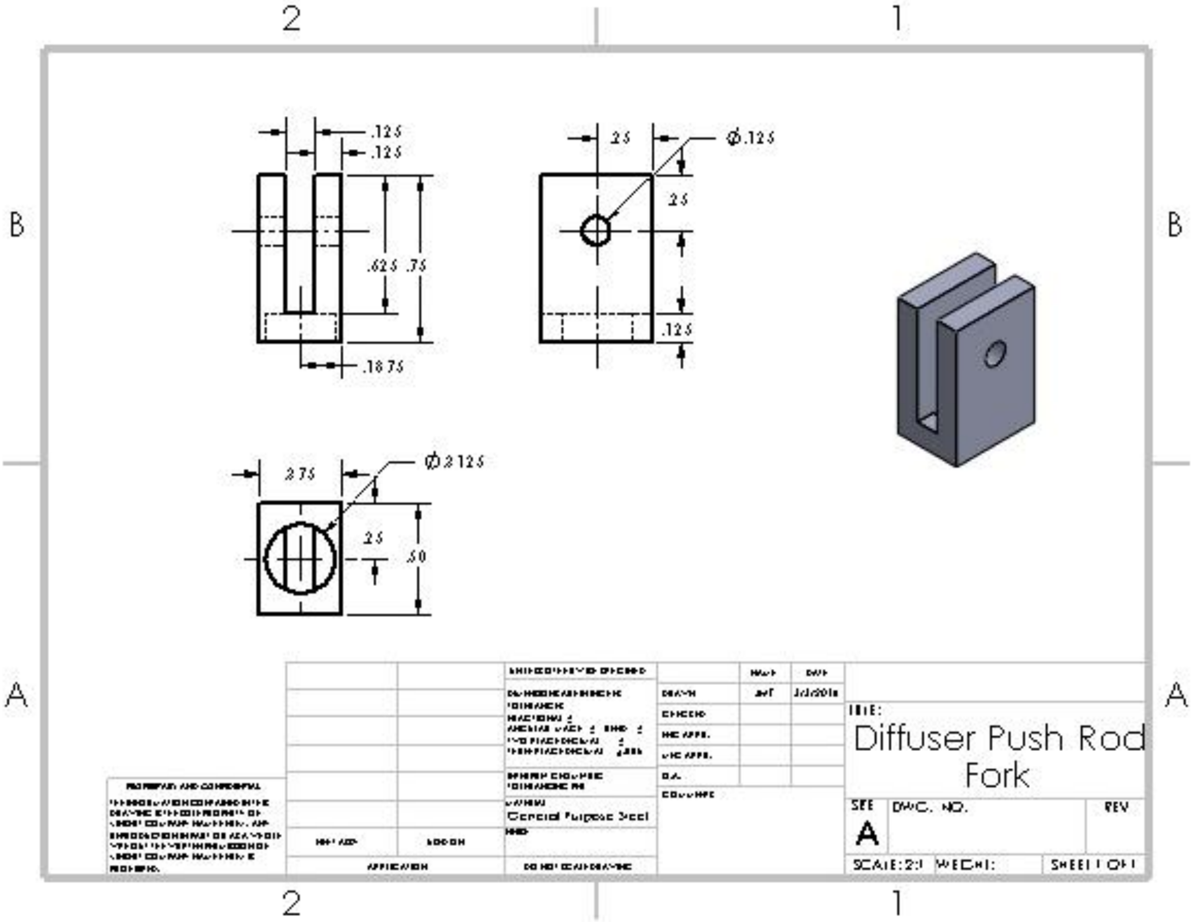


### B8: Diffuser

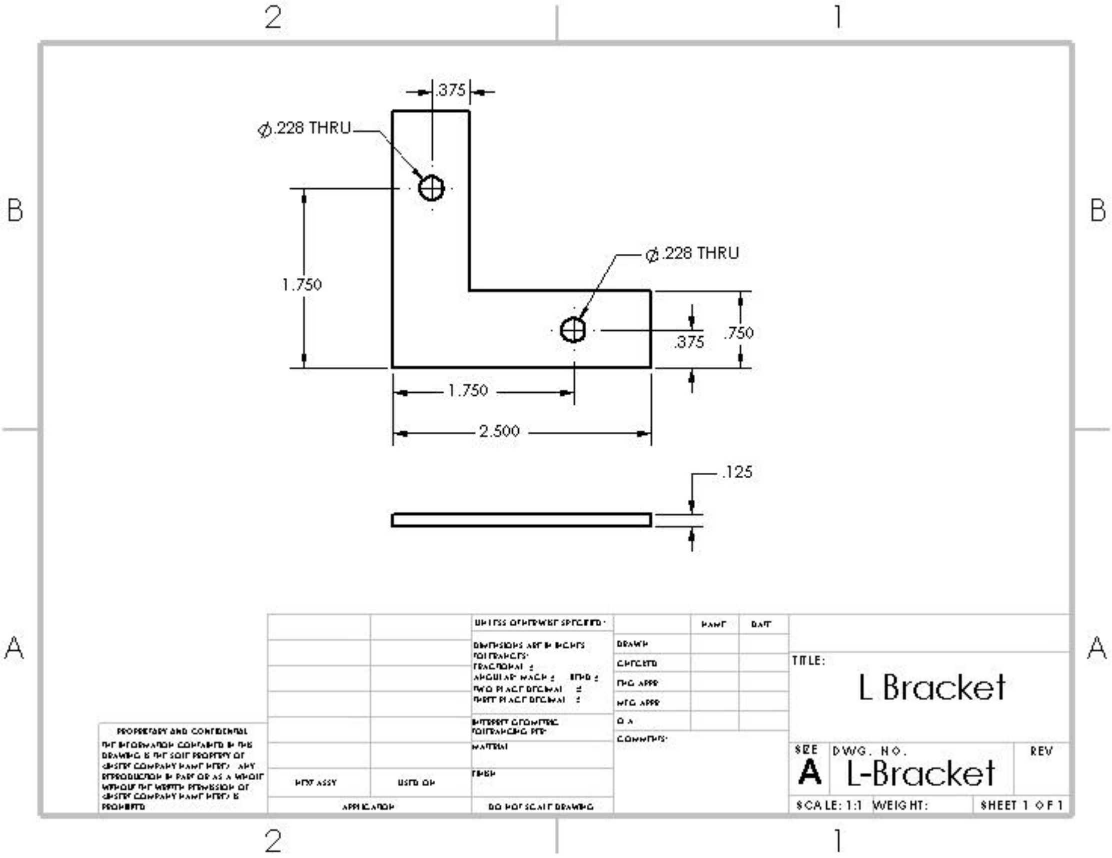


Ø3.125
Ø.375 × 4 THRU
.75
.75
Ø3.00
Ø3.125
.125
2.75
TITLE:
Diffuser
SIZE
A
DWG. NO.
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1

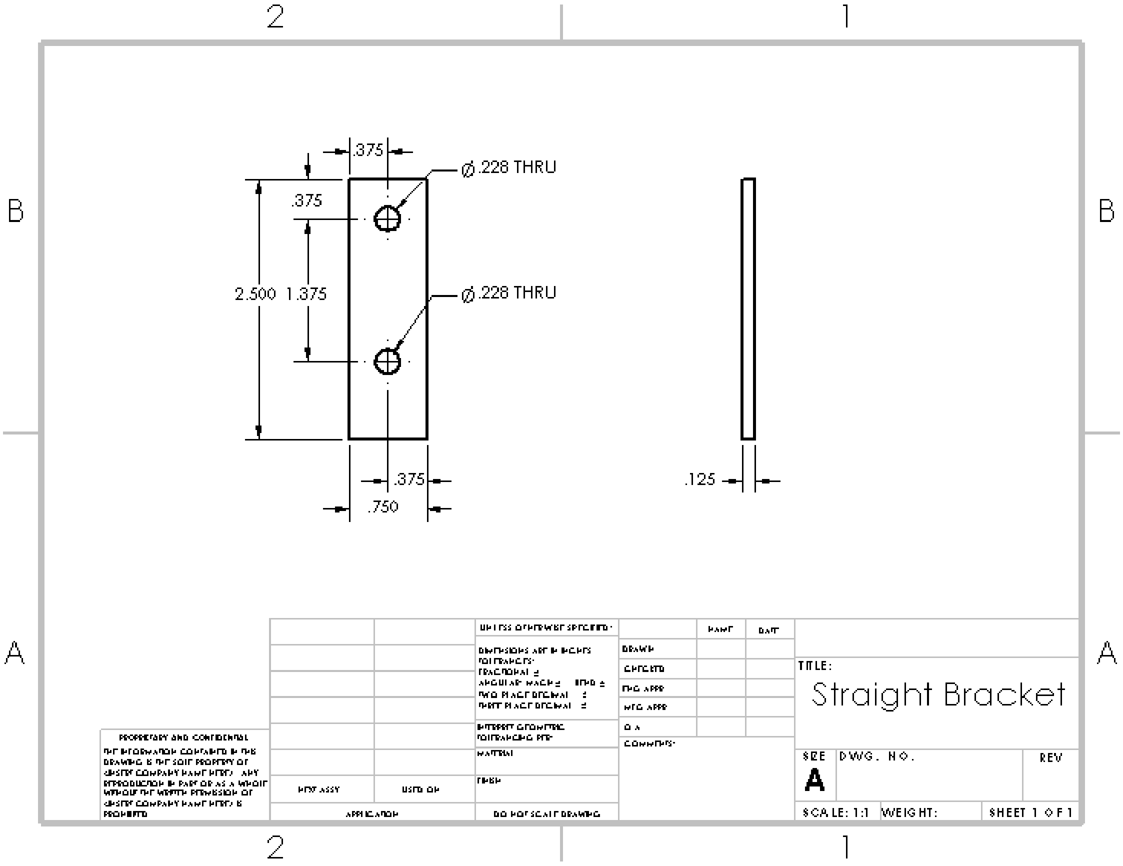
## B9: Diffuser Connector



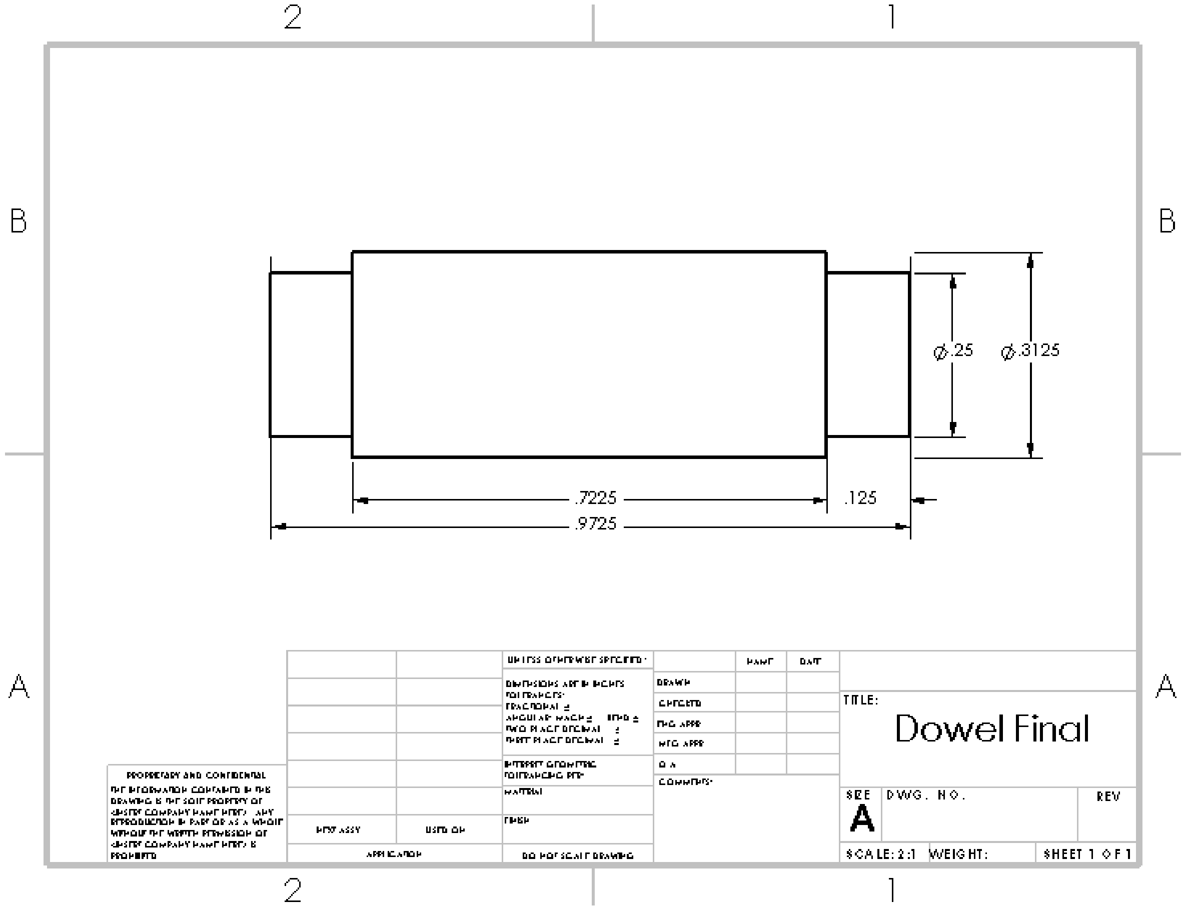
## B10: Crank L Links



## B11: Crank Straight Links



## B12: Crank Rods



# Appendix C: Build Schedule

*excluding trials not included in the final design (ex. 3D printed parts)

| Components | Parts | Operation | Machine/Tool | Date Start | Date Complete |
|---|---|---|---|---|---|
| Cylinder | • Aluminum Tube Stock<br>• Sheet | 1. Cut tube<br>2. Cut Cap<br>3. Weld Cap<br>4. Cut Flange<br>5. Mill Flange<br>6. Weld Flange | Horizontal Band Saw, Vertical Band Saw, Welding Rig, MiniMill | 11/30/15 | 1/18/16 |
| Pistons | • Aluminum Cylinder Stock<br>• U-Cup Seals<br>• Piston Rings<br>• Piston Guides | 1. Turn down for collet<br>2. Turn Grooves<br>3. Drill Hole<br>4. Mill out piston connectors<br>5. Cut pin holes in connectors<br>6. 3D Printed Guides | Manual Lathe, ST-10 Lathe, MiniMill, VM2, MakerBot | 2/3/16 | 3/29/16 |
| Crank | • Steel Rod Stock<br>• Flat Steel Stock<br>• Plastic spacers | 1. Turn down cylindrical links on the lathe<br>2. Cut straight and L links<br>3. Cut holes in links<br>4. Compression fit pieces<br>5. Weld into place | ST-10, Manual Mill, Arbor Press, Welding Rig | 12/12/15 | 2/19/16 |
| Frame | • Square Tube Stock<br>• T and L Brackets<br>• Hardware<br>• Steel Flats | 1. Cut frame pieces<br>2. Drill holes for bolts<br>3. Assemble with brackets<br>4. Cut feet<br>5. Weld feet<br>6. Cut arms<br>7. Mill slots in arms<br>8. Drill Holes for bearings in arms<br>9. Weld arms in place | Horizontal Band Saw, Manual Mill, Welding Rig | 1/7/16 | 2/16/16 |
| Diffuser | • Aluminum Tube Stock<br>• Cap sheets | 1. Turn down tube<br>2. Cut out caps | Manual Lathe, MiniMill, Welding Rig, | 2/10/16 | 3/2/16 |

| | | | | | |
|---|---|---|---|---|---|
| | • Regeneration Tube<br>• Steel Wool<br>• Aluminum Nut<br>• Steel Push Rod<br>• Steel Connector | 3. Cut holes in caps for regeneration tubes<br>4. Weld aluminum nut on bottom cap<br>5. Weld together the tubes and caps<br>6. Mill connector piece<br>7. Cut rod to length<br>8. Die 1” of the rod<br>9. Weld connector piece to other end of rod<br>10. Assemble rod into welded diffuser | Vertical Band saw | | |

## Appendix D: Bill of Materials

*excluding parts not included in the final design

| Part | Qty | Size | Material | Manufacturer | Part no. | Unit Price | Total Price |
|---|---|---|---|---|---|---|---|
| Cylinder Tubing | 1 | 3 ½" OD<br>3 ¼" ID<br>24" Length | Aluminum | Speedy Metals | t61r3.5x.125-24 | $17.77 | $17.77 |
| Diffuser Tubing | 1 | 3 ¼" OD<br>3" ID<br>12" Length | Aluminum | Speedy Metals | t61r3.25x.375-12 | $22.50 | $22.50 |
| Flange-Mount Needle Roller Bearings | 5 | 5/16" Diameter | Steel | McMaster-Carr | 1434K24 | $9.21 | $46.05 |
| Sleeve Bearings | 2 | 5/16" ID<br>¾" Length | Polymer | Grainger | 2MTE2 | $6.15 | $12.30 |
| O-ring Supported U-cup Seals | 2 | 5/16" ID<br>9/16"OD | Polyurethane | McMaster-Carr | 9505K14 | $2.94 | $5.88 |
| Piston Stock | 2 | 3 ¼" Diameter<br>3" Length | Aluminum | McMaster-Carr | 1610T27 | $19.45 | $38.90 |
| Piston Rings | 3 | 3 ¼" custom to piston | Polytetrafluoroethylene (PTFE) | Eriks Plastics | n/a | $50.00 + ship | $174.08 |
| Heat Shield | 2 | 12" x 18"<br>22 Gauge | Aluminum | Home Depot | 47240 | $7.98 | $15.96 |
| Piping for Cooling | 1 | ¼" OD<br>20" length | Copper | Home Depot | PCLA-250R020 | $17.38 | $17.38 |
| Diffuser Cap | 1 | 12" x 12" | Aluminum | Home Depot | 56032 | $5.98 | $5.98 |
| Crank and Diffuser Rods | 4 | 5/16"<br>3' length | Steel | Home Depot | 48110 | $3.47 | $13.88 |
| Crank "L" Links | 1 | 2 ½" x 1/8"<br>2' length | Steel | McMaster-Carr | 8910K401 | $8.44 +ship | $16.07 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Crank straight Links | 1 | ½" x 1/8"<br>3' length | Steel | Home Depot | 801777 | $3.57 | $3.57 |
| Mount Tubing | 5 | 1" x 36" | Steel | Home Depot | 801257 | $10.82 | $54.10 |
| Arms | 2 | 1 ¼" x 36" | Steel | Home Depot | 801877 | $9.58 | $19.61 |
| T Brackets for Mount | 4 | 3" x 3" | Zinc-Plated | Home Depot | 15169 | $2.78 | $11.12 |
| L Brackets for Mount | 3 | 1 ½" | Zinc-Plated | Home Depot | 15304 | $2.67 | $8.01 |
| Hex Bolts | 1 | ¼" - 20<br>1½" Length | Zinc-Plated | Home Depot | 800600 | $13.77 | $13.77 |
| Nuts for Hex Bolts | 1 | ¼" - 20 | Zinc-Plated | Home Depot | 801730 | $5.37 | $5.37 |
| Long Bolts | 8 | #10-24<br>1 ½" length | Zinc-Plated | Home Depot | 27771 | $1.18 | $9.44 |
| Short Bolts | 1 | #10-24<br>½" length | Stainless steel | Home Depot | 37962 | $6.21 | $6.21 |
| Feet | 1 | 1 ½" | Rubber | Home Depot | 9970 | $2.48 | $2.48 |
| Honing Tool | 2 | Up to 4" | Aluminum Oxide | Harbor Freight | 97164 | $24.99 | $49.98 |
| Honing Tool Coarse Stones | 1 | 4" | Silicon Carbide Abrasive | AutoZone | 2541C | $12.74 | $12.74 |
| PTFE Spray | 1 | n/a | PTFE | Amazon | 17080 | $9.54 | $9.54 |
| DEI High-Temp Silicone Coating | 1 | n/a | Silicone | Amazon | 10301 | $16.77 | $16.77 |
| Diffuser Regeneration Tubes | 1 | ½" OD<br>36" Length | Aluminum | Home Depot | 801247 | $10.67 | $10.67 |
| Diffuser Fork | 1 | ½" x 1"<br>6" length | Steel | McMaster-Carr | 8910K698 | $4.65 | $4.65 |
| Diffuser Nuts | 2 | ¼" - 20 | Aluminum | McMaster-Carr | 90012A106 | $4.83 | $9.66 |
| Clevis Pins | 6 | 3/16"<br>¾" length | Stainless Steel | MSC | 67926600 | $1.47 | $8.82 |

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Cotter Pins | 1 | 3/32”<br>1” length | Steel | MSC | 679796<br>41 | $4.23 | $4.23 |