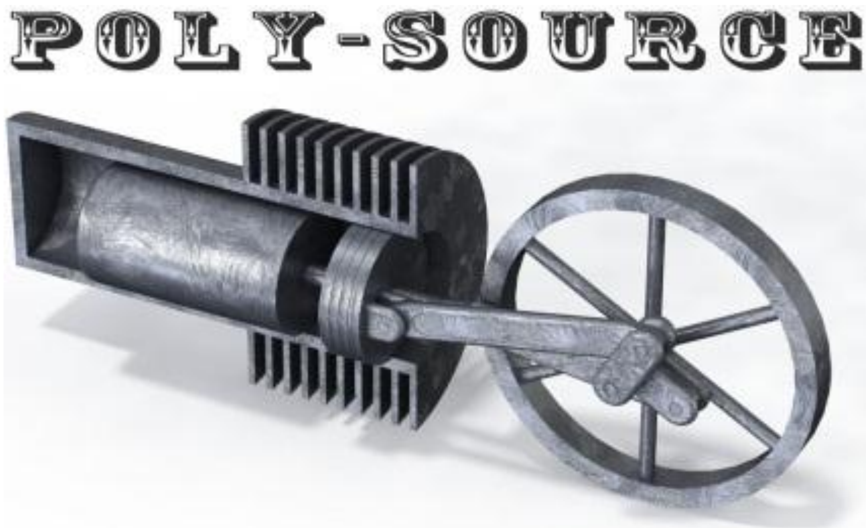

Stirling Engine – Bringing Electricity to Remote Locations

A Senior Project presented to:

The Faculty of the Mechanical Engineering Department

California Polytechnic State University, San Luis Obispo

In Partial Fulfillment

of the Requirements for the Degree

Bachelor of Science

By:

Joshua Dulin

Matthew Hove

Jonathan D. Lilley

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Introduction

The purpose of this project has been to develop a Stirling engine capable of operating on variety of heat sources, specifically on solar heat and traditional fuel combustion. Such an engine would have applications generating power in remote locations, using solar heat to offset fuel consumption. This project developed a prototype engine that can be analyzed to develop a model for use in designing a full scale engine. This project, proposed and funded by student sponsor Jonathan D. Lilley, focuses primarily on the engine itself, rather than the heat sources or the electricity generation system. The Poly Source team, consisting of Joshua Dulin, Matthew Hove, and Jonathan D. Lilley, students of mechanical engineering at California Polytechnic State University was advised by Professor Sarah Harding of the Mechanical Engineering Department with Professor Melinda Keller of the Mechanical Engineering Department acting as faculty sponsor.

Through this senior project, the poly source team has gained experience in designing and manufacturing an engine system that combines thermodynamic and kinetic cycles. The results of this project are available for future developers of Stirling engines, including any students of Cal Poly who wish to take on a similar project.

Background

The Stirling engine is a closed air engine capable or running on a heat differential produced by any heat source, provided that the heat differential is large enough for the designed power output. The engine utilizes the fact that a gas, when heated, increases in pressure and volume, and when cooled decreases in pressure and volume. The engine makes use of this by heating an internal gas and using the gas's expansion to drive a power cylinder. The gas is then cooled, allowing for it to be easily compressed so that it may again be heated. This process is controlled by two pistons. The power piston reacts to and draws power form the pressure changes in the engine. The displacer piston moves the gas back and forth from the expansion space and the compression space, where the gas is heated and cooled, respectively. If an engine were to match this process precisely, it would have the highest thermal efficiency theoretically possible by any heat engine. While mechanical inefficiencies prevent engines from reaching this efficiency, Stirling engines are still among the most efficient heat engines produced.

The great benefit of Stirling engines, over traditional steam or internal combustion engines, is that Stirling engines can hypothetically run on any heat source, regardless of what produced the heat. This allows Stirling engines to operate on a wide variety of energy sources, such as the waste heat of industrial processes, solar thermal heat, or heat from traditional combustion fuels. Because of this capability, many heat sources that are unusable by other engine types are usable by Stirling engines. Furthermore, a Stirling engine can theoretically operate on more than one heat source, gaining the advantages of each. This project aims to design a Stirling engine that can run on both solar thermal heat and on combustion heat. This will combine the advantage of solar power, namely a renewable

energy source with no fuel cost, and the advantage of traditional fuel combustion, to provide on demand power.

Robert Stirling patented the closed-cycle, regenerative Stirling engine in 1816. Stirling was a minister at the Church of Scotland, who developed the engine as a safer alternative to steam engines. While the Stirling engines developed by Robert Stirling and his successors were efficient and safer that steam engines, they fell out of favor for most of the nineteenth century. This was primarily due to the main drawback of Stirling engines, their relatively low specific power (i.e. Stirling engines produce less power than other types of engines of the same physical size). However, Stirling engines gained new interest in the twentieth century when fuel efficiency became a greater concern in industry.

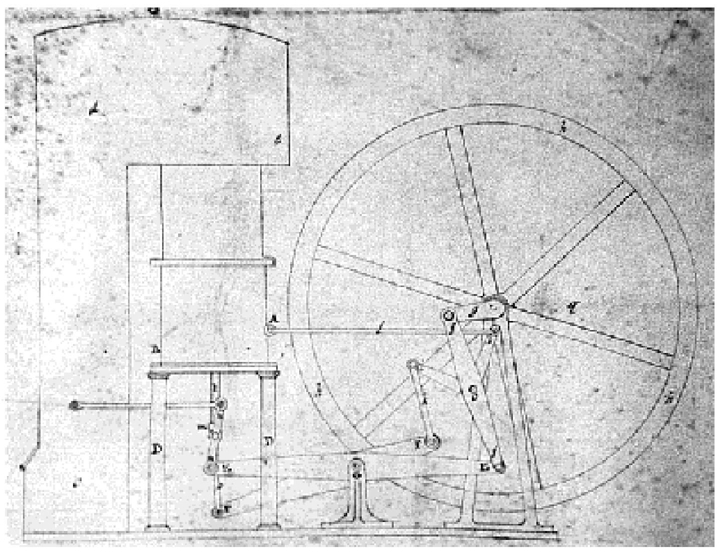


Figure 1: Stirling's Original Patent

There have been three main configurations of the closed cycle Stirling engine. These configurations differ mainly on the relative locations of their pistons. The first configuration is the alpha type, which has two separate cylinders. One is for the hot side (expansion space), the other is for the cold side (compression space), as well as a connection allowing gas to flow between the two cylinders, which may include a regenerator. The second configuration is known as the beta type. It only consists of one cylinder in which both power piston and displacer reside. For this configuration the regenerator can be the displacer piston itself, and the hot and cold air will flow around the thin space between displacer piston and cylinder. The Third configuration is the gamma type. The gammatype configuration is very similar to the beta type; however, the power piston resides in a separate cylinder from the displacer. The gamma type is mainly used for situations with very low heat differentials.

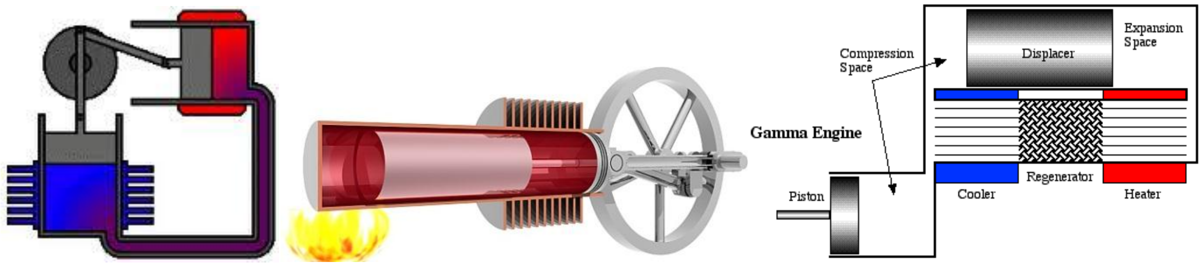


Figure 2: Alpha, Beta and Gamma Configurations (shown left to right)

Early Stirling engines were kinematic engines; the piston motions were controlled by mechanical linkages, such as a crankshaft and flywheel. An alternative design, known as the free-piston Stirling engine, was developed by W.T Beale in the 1960s. The free-piston engine is essentially a beta type configuration, except that the piston motions are controlled by springs rather than a crank and flywheel mechanism. The springs are designed to move the pistons at a natural frequency equal to the engine's operating frequency, while electricity is produced by magnetic linear alternator incorporated in the engine, rather than a separate generator. These innovations allowed free-piston Stirling engines to be smaller and have a much longer life than traditional kinematic Stirling engines. This allowed Stirling engines to be used in more portable, high reliability applications, such as spacecraft and military power generation. Most modern commercially developed Stirling engines are free-piston.

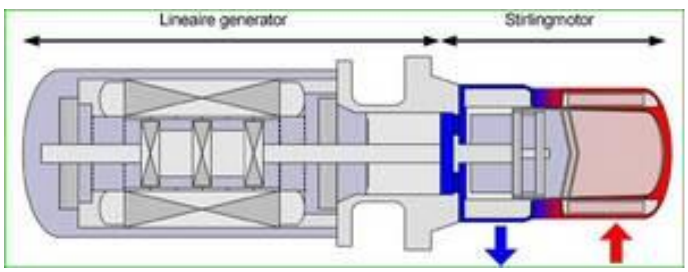


Figure 3: Free-Piston Configuration

The companies that currently develop Stirling engine technology appear to be focused on three main applications: industrial waste heat conversion, grid-based solar power, and remote location power generation. For the application of this project's engine, remote power generation, there are three main competitors: Cool Energy Inc, SunpowerInc, and Infinia Corporation. Cool Energy currently produces a Stirling engine that supplements diesel generators by running off of the exhaust heat of an internal combustion engine. Their engine costs around $10,000, and produces 3 kW of energy. Sunpower and Infinia both produce high efficiency fuel-burning Stirling engines for remote power generation. Infina has also been developing solar power generation by mounting the Stirling engine in a solar tracking and solar concentrating dish.

This project will distinguish its final design from those of existing commercially available engines by its ability to operate on multiple heat sources.

Objectives

The aim of this project has been to develop the design of a Stirling engine capable of running off of both solar thermal energy and the heat from fuel combustion. The theoretical application of this engine is the generation of power in remote areas, off the electricity grid, where current power generation systems are costly. The engine would be able to run on solar heat so that it can produce power with no fuel consumption, and the engine would be able to run on fuel combustion so that it can continue to produce power even if the sun is unavailable, such as during night or times of poor weather.

This project has progressed development to a final design by designing, building, and testing a prototype. The prototype is of the same configuration that would be chosen for a final design. As such, lessons learned from the prototype can be applied to a final design. Specifically speaking, through testing the prototype, it can be determined which parameters (such as spring constants, temperature differential, piston masses, etc.) most affect performance. Furthermore, should a model be developed to predict engine behavior, it can use the prototype's behavior as a baseline.

In order to be economical, the full scale engine will have to be able to produce enough power to be useful. Since a common application of power in remote areas is well pumping, the power required to operate a 1hp well pump will be used as a goal for this engine. A survey of the market of well pumps has shown that 3 kW will be sufficient to run nearly all 1 hp pumps. So the power generation goal of the full scale engine will be 3 kW.

This project will not provide an actual heat source for the engine. Instead, heat sources have been simulated to match the temperature and quality of heat available from solar thermal collectors and fuel combustion. Based on research of various solar thermal collectors, it was determined that the temperature used to simulate a solar thermal heat source should be 300 degrees Celsius. Since the temperature of fuel combustion furnaces can be adjusted by altering the combustion rate, we will assume that the fuel combustion source being simulated has been adjusted to produce heat at the same temperature as the solar thermal collector. We will assume that the theoretical heat source being simulated can provide enough energy to maintain the temperature at 300 degrees.

Since remote areas often go unmanned for long periods of time, the full scale engine should be able to operate for 6 months (or 4400 hours) of continuous or intermittent use without the need for regular maintenance. Furthermore, the engine should be designed for a lifetime of 5 years (or 36000 hours) continuous or intermittent operation. For intermittent use, we will assume that the engine will be turned on and turned off once per day. That presents the requirement of 180 on/off cycles without regular maintenance, and 1830 on/off cycles over its lifetime. The prototype developed by this project does not need to meet these fatigue and lifetime requirements, however, the configuration of the prototype has been designed to correlate to a full scale design which would intend to meet these requirements (i.e. the prototype configuration has been chosen with maintenance and lifetime in mind).

The full scale engine, excluding the heat source and electrical power system, must be able to fit in a space 4 feet wide, 5 feet long and 5 feet high. We assume this size will be sufficient because it is a similar size to the 3 kW engine currently produced by Cool Energy Inc., which has similar heat input and power output requirements. Furthermore, this size is small enough to be transported in the truck bed of most production pickup trucks.

Table 1: Compliance Matrix

Spec #	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Power	3 kW	± 200 W	H	T, A
2	Heat Requirement	300°C	± 20°C	M	T, I, A
3	Time Between Maintenance	4400 Hours	± 100 hours	L	A, S
4	Service Lifetime	36000 Hours	± 500 hours	L	A, S
5	Size	4 x 5 x 5 Feet	MAX	L	I

The target values shown in Table 1 are the ideal parameters that a full scale engine would meet. Each target has a corresponding tolerance to provide an acceptable range. There are three levels of risk, (H) High, (M) Medium, (L) Low for how difficult it will be to achieve the target for that parameter. In addition there is a Compliance section which specifies how each parameter could be verified; (A) is for Analysis, (T) is for Testing, (I) is for Inspection, and (S) is for Similarity to existing designs.

As shown in Table 1, our highest risk parameter is to achieve 3kW power output. This is due to the associated sheer size and material cost involved as well as limited manufacturing time and access. This is the primary reason the project has developed only a prototype, rather than a full scale engine.

Method of Approach

We began our design process by researching modeling techniques of the Stirling cycle and Stirling engines. Our intention was to adapt these techniques into a model that could be used to estimate engine performance based on design parameters. During the project, our model development proved troublesome, that will be discussed later in the Model Generation section.

Out prototype was designed to be partly adjustable so that we can experiment with adjusting few of the critical design parameters and see how these affect performance. Furthermore since the prototype can be adjusted to operate with some differing parameters, it can provide multiple baselines from which model performance can be based or evaluated.

Our team ran out of time before a working performance model could be developed. In our original production plan, our performance model was to be reevaluated to more accurately reflect prototype behavior. Once the model reflected the prototype’s behavior, we were to use the model to find a configuration of critical design parameters that meet our performance goals. Using these parameters, we would iterate prototype designs, modifying the existing prototype or developing new prototypes

that more closely match the parameters, and testing their performance against model performance. Once the performance model had been further verified, we were to design the final engine to meet design performance specifications based on the model's predicted critical design parameters.

A flowchart of our overall production plan is shown in Figure 4.

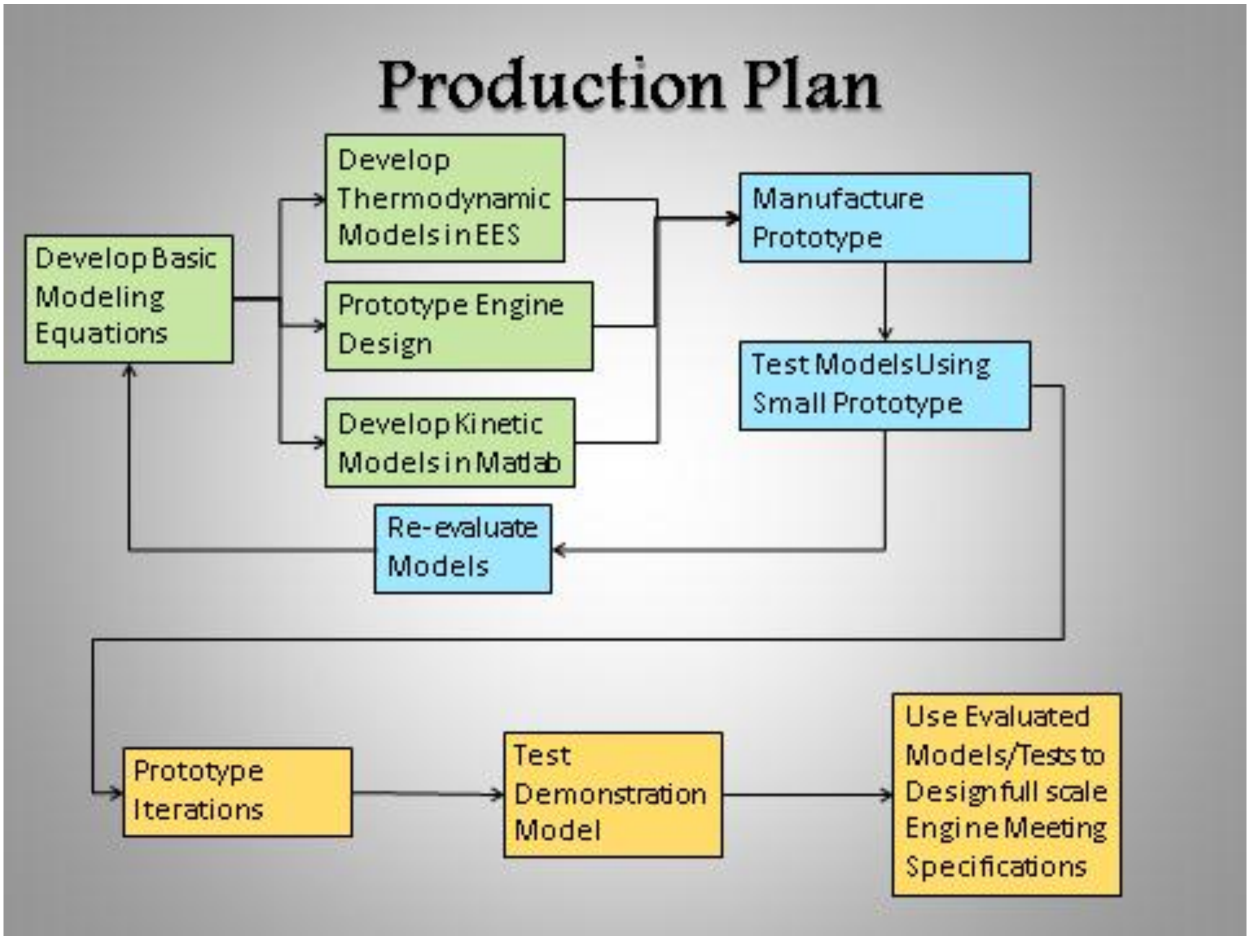


Figure 4: Production plan, color coordinated breakdown per quarter

Idea Generation and Selection: Configuration

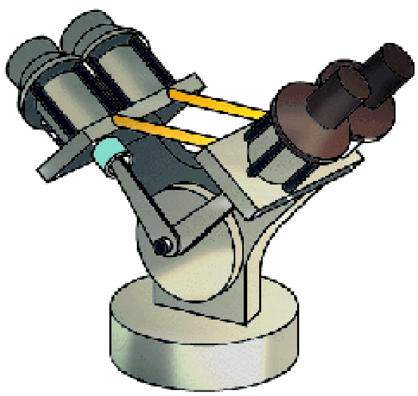
Concept Generation

For our initial concept generation we decided to focus on the configuration of Stirling engine that would be sufficient in order to achieve our design specifications. We researched various types of Stirling engine configurations, which have succeeded or failed in industry. Once we had found a number of previously used configurations, we used our house of quality to find the configuration that best met our goals. There are four main types of configurations that we considered: Alpha configuration, Beta configuration, gamma configuration, and free-piston beta configuration. The first three of these configurations, Alpha, Beta, and Gamma, all produce power and control piston kinematics by connecting the pistons to drive shafts or other drive systems via mechanical linkages.

These three configurations differ in where their pistons are located relative to each other. The fourth configuration, the free-piston beta configuration, does not have mechanical linkages. Instead, power is removed by a linear alternator, and the piston kinematics are controlled by springs and pressure differences in the engine's compartments.

Idea Selection

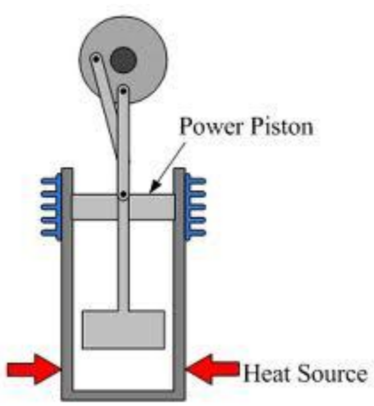
Off the configurations which use mechanical linkages to control the pistons and remove power, the first type described is the alpha configuration. The alpha configuration locates the piston and the displacer in separate cylinders, and attaches the heater and the cold sink to either cylinder. Alpha engines have good reliability and efficiency. However, they would also require more time, money to manufacture, and take up more space than other configurations.



Pros	Cons
High Specific Power	Higher Cost
High Efficiency	More Difficult to Transport
Large temperature Differentials	Startup Time and Torque
	System Life Time
	Large in Size Comparably
	Seal on Hot Side

Figure 5. Two cylinder alpha style Stirling engine

The second type is the beta, and it consists of only one cylinder with the piston and displacer both inside it. The beta engine has less efficiency than alpha configuration. However, it gained strong consideration due to its benefits of small volume, and ease of manufacturability. A beta-type engine would be fairly portable, aiding in our requirement of the engine being able to fit in the bed of a pickup truck. The manufacturing cost will also likely be lower, compared to the other configuration types, due to its single cylinder style.



Pros	Cons
Small in Size	Maintenance
System Life Time	Startup Time and Torque
Large temperature Differentials	System Life Time
Decent Specific Power	
Portability	
Seals are Less Crucial	

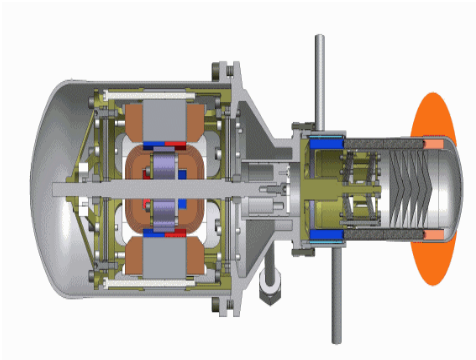
Figure 6: Beta style Stirling Engine

The gamma-type configuration is mainly used for low temperature differential Stirling engines. The configuration is similar to the beta, however the power piston is in a separate smaller cylinder than the displacer. Due to this arrangement some of the expansion of the displacer must take place in the compression space. This generally means there will be a reduction in specific power, leading to a bulky overall engine, and that is why gamma style engines are almost exclusively used to run on low temperature differentials. For our objectives we will need a high specific power ratio, in order to ensure that we get the desired four kilowatts of electricity. Therefore the gamma style engine was ruled out quickly in our idea selection.

Pros	Cons
Low Temperature Differential	Low Specific Power System Life Time Low Efficiency

Figure 7: Gamma style Stirling Engine

The last configuration selection was the free-piston beta-type Stirling engine. Unlike the previous three configurations, the free-piston contains no mechanical linkages connecting the power piston to the power generating device. In most cases the free piston type engine will power a linear alternator, that would generate electricity using magnets attached to the reciprocating power piston. Due to the fact that no mechanical linkages are needed, and no rotating equipment is necessary, the problem of lubricating and pressure sealing the engine becomes fairly easy. The style of the organization is similar to the beta style engine, with both the power piston and the displacer in the same cylinder, however the pistons will be controlled not by mechanical linkages, but instead by springs and pressure differences within the engine. Since we can fit both the piston and displacer in to one cylinder, this configuration is both cheap and portable.



Pros	Cons
Small in Size	Heat Efficiency
	Harder to Design and Manufacture
System Life Time	
Decent Specific Power	
Slim to No Lubrication	
Portability	
Ease of Starting	
Maintenance	

Figure 8: Free piston Stirling Engine

Model Generation

In generating models for our Stirling engine, we had two main goals:

1. Estimating the power output for a giving input and load.
2. Estimating kinematic behavior of the engine, to ensure that our design will behave as expected.

For our first mode of analysis the Schmidt model could be used to determine an approximate power output of the engine. The Schmidt model simplifies the equations to relate volumetric changes to kinematic changes, and by integrating the P-V curve the work out per cycle can be solved. We intended to find the volumetric and kinematic behavior of a design by developing a kinetic model to predict the engine kinematics, which correlated with our second modeling goal.

Modeling the kinematic behavior of the engine is important to ensure that the displacer piston and the power piston do not physically travel beyond the spaces where they were designed to operate. Without properly designing for kinematics, it is possible for the pistons to strike each other or the end of the engine cylinder, reducing performance and possibly damaging the engine. Since our design is a free-piston engine, the kinematic motions of the pistons are not mechanically constrained. Therefore, the motion of the pistons must be estimated by considering the forces applied to each piston by the various pressures within the engine, and the springs and loads designed for the engine.

The project team attempted to develop a kinetic model to determine engine kinematic behavior. However, the model proved to be very mathematically difficult, and the model failed to produce useful results. Due to time constraints, we decided to focus more energy on the development of the prototype, and leave the model to be completed later, when the prototype was functional. The kinetic model was developed as follows:

Dynamic equations were applied to each piston (the displacer and the power piston). The forces applied to the pistons were caused by the internal pressure of each engine chamber, by the springs, and by frictional damping. Since the pressures in the engine chambers relates to the position and velocity of the pistons, and the acceleration (and thus the velocity and position) of each piston relates

to the pressures, the complete system ends up being described by non-linear differential equations. Since no closed form solution to these equations is available, we decided to use numerical integration to solve for the piston kinematics.

Simulink was used as a numerical integration program. The piston dynamic equations were converted into block diagram form using the integration backbone technique. In the integration backbone technique, the forces are summed on the left and divided by the mass to get the piston acceleration. This acceleration is then integrated to obtain velocity, and the velocity integrated to obtain position. Thus, the kinematic variables are solved for, and can be drawn off of the block diagram to be used in the calculation of forces. Figure 9 shows the integration backbone of the power piston.

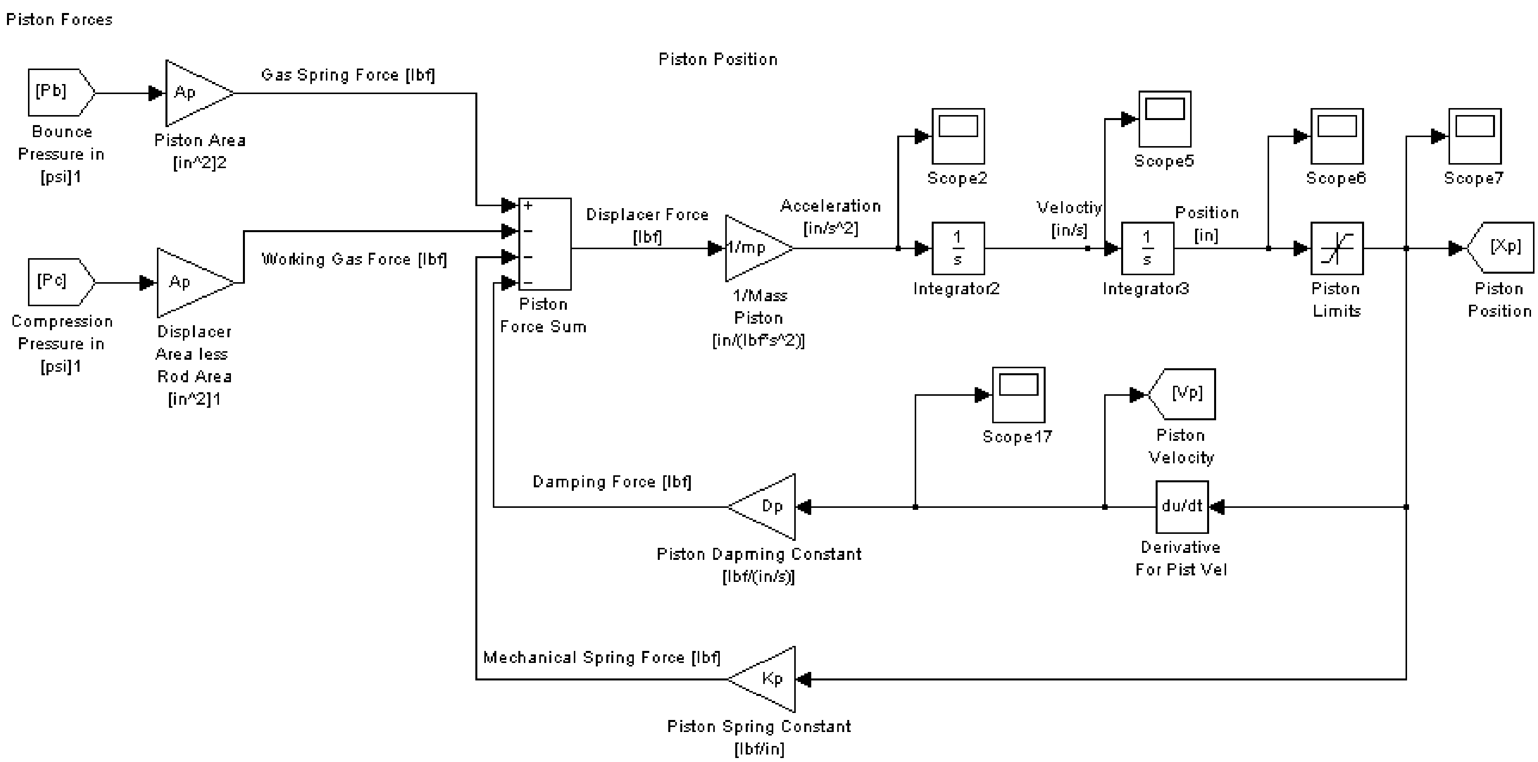


Figure 9: Integration Backbone of the Power Piston

The pressure of the working fluid in the piston chambers was calculated using a modified ideal gas law, assuming that the pressure difference between the chambers is small. The mass in each chamber (the hot chamber and the cold chamber) could be determined via the equation $= \frac{PV}{RT}$. Since the system is closed, the total mass of working fluid is constant, and can be expressed as the sum of the mass of working fluid in each respective chamber: $m_t = \frac{P}{R}\left(\frac{V_c}{T_c} + \frac{V_h}{T_h}\right)$. From this, the pressure can be solved for: $P = m_t R / \left(\frac{V_c}{T_c} + \frac{V_h}{T_h}\right)$. This was taken to be the pressure in the cold chamber. The pressure in the

hot chamber was assumed to differ according to the pressure loss across the displacer passage caused by fluid flow. This loss was solved for using Darcy-Weisbach equation

While the model appeared to be dynamically and thermodynamically correct, it often produced very strange and unhelpful results. This is attributed to oddities that arose due to the numerical integration modeling technique, and the mathematical phenomena are not entirely understood.

Due to time constraints, the team was not able to perform an in depth study of the modeling techniques to fix the bugs in the model. As such, the team decided to focus on developing and testing the prototype. Completion of the model could be completed as a future project.

Idea Generation and Selection: Prototype

To develop a baseline for the prototype we paid close attention to our past research; especially that from Gary Kunkels's "Free-Piston Stirling-Cycle Engine", as well as guidelines discovered through the experimental work of Dr. William Beal. Dr. Beal's guidelines include:

1.) To ensure that proper phase relationship between the displacer and the piston, the piston should be appreciably greater in mass, approximately a ration of ten to one.
2.) The cross-sectional area of the displacer rod should be approximately one-quarter of the cross-sectional area of the working cylinder.
3.) The expansion space and the compression space should be separated by a sufficiently long annulus region, approximately three times the displacer diameter. (Walker 1980, pg. 151)

These guidelines are not rigid but provided us a good starting place.

Aiming at these guidelines we began a great deal of material and tool research online, in the Cal Poly machine shops and local small-engine shops as well. A large challenge Stirling Engines face is the high cost and difficult manufacturability. Since this is a prototype, we were interested in avoiding the need to purchase custom tooling and many hours of machining in order to expedite the testing and production of our design. After many hours and multiple attempts to machine an accurate cylinder surface with sufficient length using the available lathe tooling on campus, it was decided that we would purchase a machined cylinder. The prototype is small, designed around a 1" OD cylinder. To keep cost of materials down, the design and material selection is focused on minimizing the manufacturing time and assembly while allowing adjustability to expand our testing and tuning capabilities. The following image is a solid modeling overview of our assembled Stirling prototype and its main components.

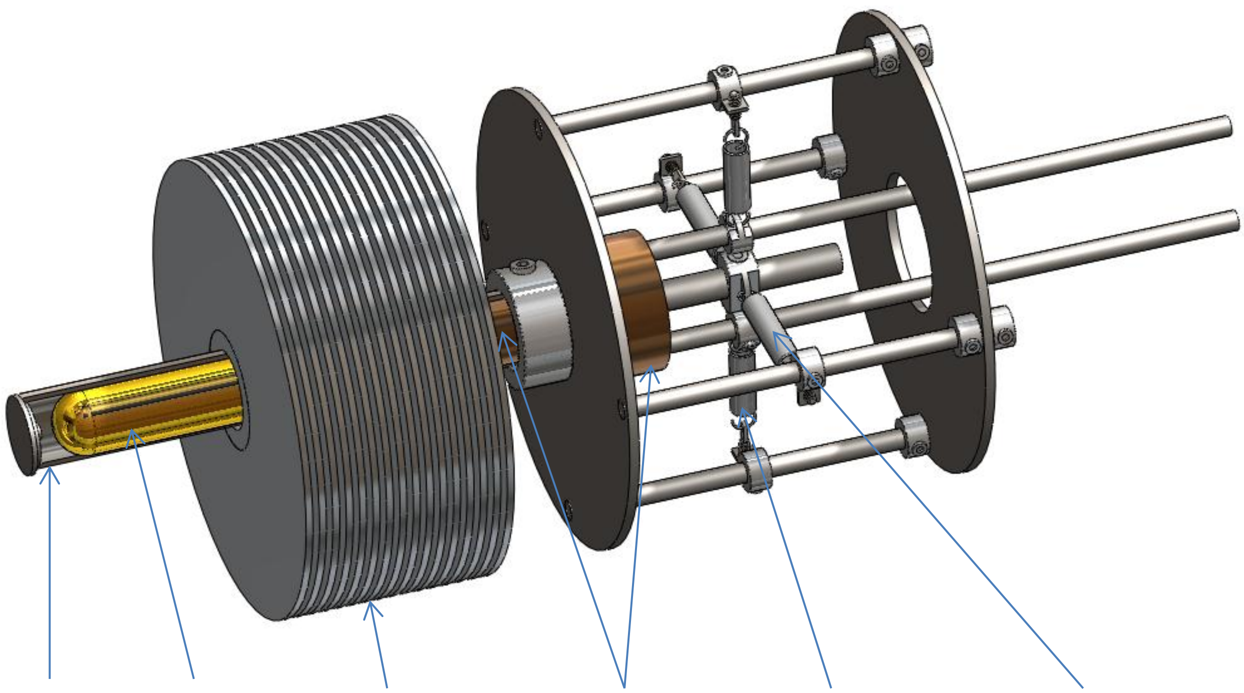
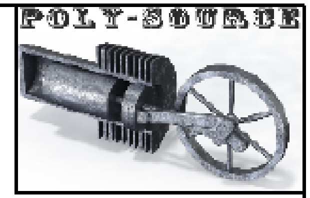


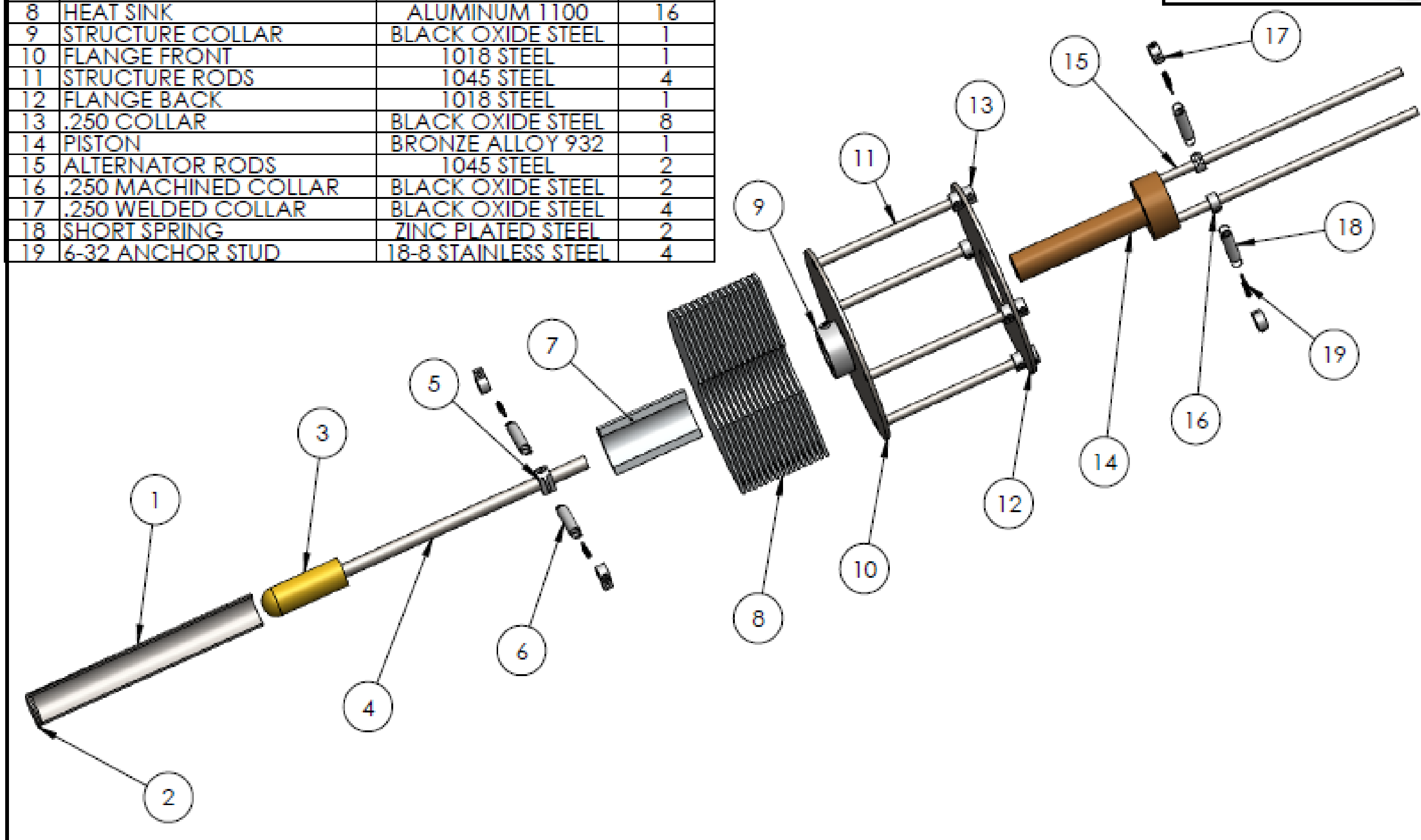
Figure 10: Prototype component overview

Shown in Figure 10, the Cylinder is relieved in half to view the Displacer and Power Piston. The Rods connected to the Power Piston and extending out the back of the spring structure are for the linear alternator power generation.

Figure 11 on the next page is an exploded overview of the engine components and the materials to be used. The material choices for the Power Piston, the Cylinder and the Displacer Rod are the most crucial due to thermal expansion coefficients; with different thermal expansion coefficients the engine could be susceptible to leaking instead of creating a seal or possibly seizing up. These materials also need a high melting temperature to avoid melting and creeping – especially that of the cylinder and displacer the two components closest to the heat input. The maximum temperatures the prototype materials are designed to withstand are nearly 600^{o}C (1112^{o}F).

NO.	DESCRIPTION	MATERIAL	QTY.
1	CYLINDER	STEEL 316/316L STEEL	1
2	CYLINDER CAP	STEEL 316/316L STEEL	1
3	DISPLACER	BRASS ALLOY 360	1
4	DISPLACER ROD	STAINLESS STEEL	1
5	.375 MACHINED COLLAR	BLACK OXIDE STEEL	1
6	LONG SPRING	ZINC PLATED STEEL	2
7	FIN BASE	ALUMINUM 6061	1
8	HEAT SINK	ALUMINUM 1100	16
9	STRUCTURE COLLAR	BLACK OXIDE STEEL	1
10	FLANGE FRONT	1018 STEEL	1
11	STRUCTURE RODS	1045 STEEL	4
12	FLANGE BACK	1018 STEEL	1
13	.250 COLLAR	BLACK OXIDE STEEL	8
14	PISTON	BRONZE ALLOY 932	1
15	ALTERNATOR RODS	1045 STEEL	2
16	.250 MACHINED COLLAR	BLACK OXIDE STEEL	2
17	.250 WELDED COLLAR	BLACK OXIDE STEEL	4
18	SHORT SPRING	ZINC PLATED STEEL	2
19	6-32 ANCHOR STUD	18-8 STAINLESS STEEL	4





California Polytechnic State University	Title: CONCEPTUAL PROTOTYPE EXPLODED COMPONENT VIEW			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #:	Nxt Asb:	Date: 2/14/13	Scale: 1:4	Chkd. By: JOSHUA DULIN

Figure 11. Prototype Exploded View

The selection of 316/316L Steel for the cylinder was chosen, on a large part, due to the availability of the smooth machined interior surface. Additionally, the strength of stainless steel is desirable- allowing a thin cylinder wall for better heat transfer into and out of the engine. The thermal expansion for this stainless steel in our temperature range of (68 – 1112^{o}F)is between 9 and 10 in/in/^{o}F.

This coefficient of thermal expansion helped to select bearing grade Bronze-alloy 932 for the power piston due to its coefficient of about 10in/in/^{o}F as well as its melting point range of 1570-1790^{o}F. Bronze 932 possesses good anti-friction properties, adequate ductility and excellent machinability and is commonly used as bearings and bushings; thus it is desirable to allow our engine to operate without lubrication.

The Displacer Rod material can be made from steel alloy 4130 due to its melting temperature of of 2610^{o}F and average thermal expansion of 7 in/in/^{o}F .The Thermal expansion of the Displacer is less vital since it has clearance within the cylinder allowing the working gas to move around it. In future thoughts, the displacer rod should be made out of stainless steel in order to prevent rusting that occurred with ordinary steel. This has caused uneven surfaces and unreliable surface conditions on a needed precise fit.

Brass alloy 360 was chosen for the displacer due to its corrosion resistance and melting point range of 1630-1650oF and thermal expansion coefficient of 11.4in/in/^{o}F. Aluminum displacers were also manufactured.

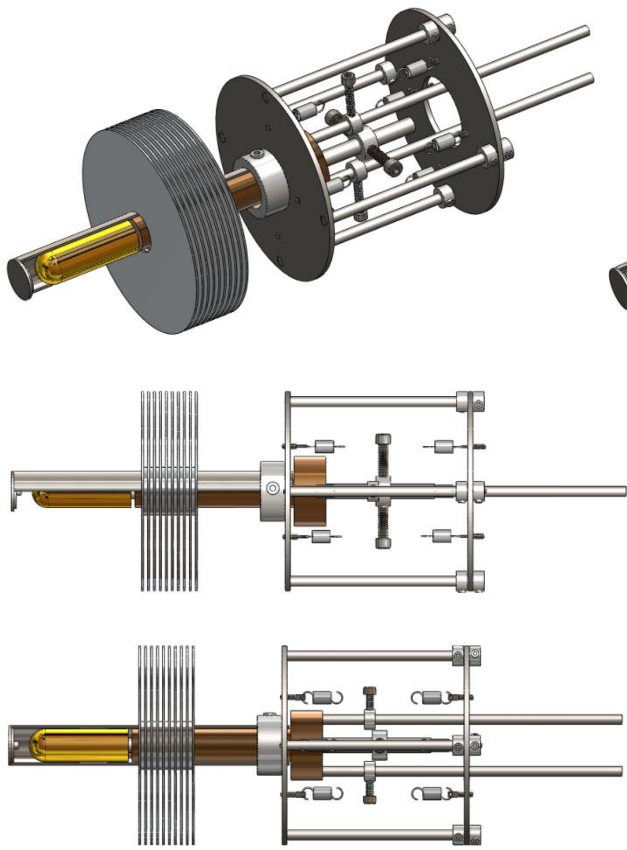
The shaft collars around the cylinder, spring structure and displacer rod offer quick assembly and disassembly if we wish to get inside the engine during testing and adjustability of the range of motion for both the piston and displacer. The shaft collar turned out to pinch our cylinder and hence we decided to do much of the clamping force with hose clamps instead. This allowed us to take off friction that was being created between the piston and cylinder by the single collar set screw.

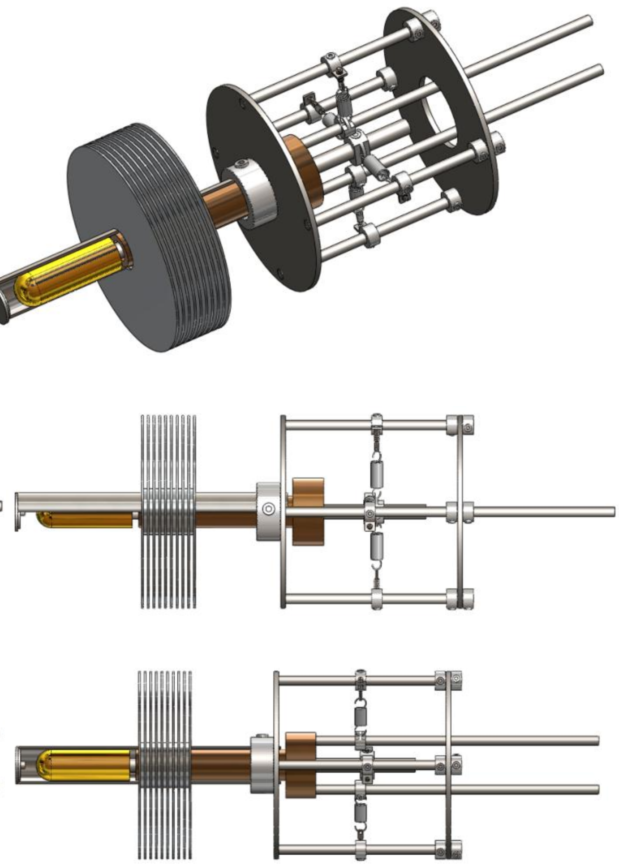
The anchor studs allow easy connections for the springs, to aid testing an assortment of springs, as well as adjustability to the spring constant applied to the displacer and piston individually.

The material selected for fins on the heat sink is Aluminum alloy 1100 due to its excellent conduction of heat and melting point range of 1190-1215^{o}F. The fin base was selected to be aluminum alloy 6061 due to material availability, good strength, corrosion resistance, and machinability as well as well as a melting point range of 1080-1205^{o}F.

For the design of the heat sink a heat transfer spread sheet was developed to calculate the thermal resistance of the finned array, fin base and spreading resistance. This spread sheet was then altered within the available material supply and suitable dimensions to minimize the thermal resistance. The thickness determined for the fin material was calculated to be .090” thick and the fin base thickness was calculated to be .25” thick.

To limit and tune the range and motion of the Displacer and Piston springs will be used. The top two ideas for spring configurations are shown in Figure 12 and Figure 13.





Spring Configuration 1 (Inline)	
Pros	Cons
High adjustability	More Hardware required
More Fine Tuning	More Machining required
	More Springs to purchase and adjust
	Possible pistion interference
	Undesired wieght on displacer/rod
	Tight Clearance
	Torque on Piston alighment
	Individual adj. of displacer and piston-spring length

Spring Configuration 2 (Radial)	
Pros	Cons
High adjustability	Tougher Geometry
Less Hardware	Tight Clearance
Can balance torque on pistion alignment	
Quick adjustability to aid multiple tests	

Figure 12: Spring Configuration 1 (Inline):

Figure 13: Spring Configuration 2 (Radial):

The radial configuration #2 succeeds the Inline configuration #1 mostly due to less hardware and machining required. However, along with less hardware, configuration #2 offers quick adjustments to the spring constant and enables us to take greater advantage of the spring variety sold through McMaster Carr; thus the cost – especially due to the springs will decrease. Additionally the motion of the Piston and Displacer may be closer to uniform due to a more constant spring rate in both directions of travel.

We learned the springs were very crucial in order to have the engine start. Due to all the very precise fits of the cylinder with the piston, and the piston with the displacer rod, the springs had to be balanced in order to have no friction with the precision fits. The set pins holding the springs also seemed to want to turn in the threads and hence the spring balance would eventually become un-balanced as time progressed. In the future using different types of springs such as flexures, or have it entirely run using magnets would make running the engine much simpler, and effective.

Cost

A breakdown of the cost of materials for this project is shown in the bill of materials in Table 2.

Table 2. Bill of Materials

Line	Quantity	Product		Ships	Unit Price	Total Price
1	1	3334K161	Ultra-High-Polish Type 316/316L SS Tubing Smooth, 1" (	today	$27.94	$27.94
2	1	89325K242	Super Corrosion Resistant SS (Type 316/316L) 1" Diame	today	$21.93	$21.93
3	1	8953K971	Ultra Machinable Brass (Alloy 360) 7/8" Diameter, 1' Le	today	$27.68	$27.68
4	1	89955K44	Easy-to-Weld 4130 Alloy Steel Round Tube .375" OD, .0	today	$25.06	$25.06
5	1	8914K776	Bearing Grade Bronze (Alloy 932) Oversize Rod, 1-1/2"	today	$51.76	$51.76
6	1	8924K31	High-Strength 1045 Medium Carbon Rod 1/4" Diamete	today	$4.40	$4.40
7	2	1388K39	Low-Carbon Steel Sheet 1/8" Thick, 6" X 6", Ground Fin	today	$23.35	$46.70
8	2	9414T19	Black-Oxide Steel Set Screw Shaft Collar 1" Bore, 1-1/2	today	$1.56	$3.12
9	3	9414T8	Black-Oxide Steel Set Screw Shaft Collar 3/8" Bore, 3/4	today	$0.72	$2.16
10	16	9414T6	Black-Oxide Steel Set Screw Shaft Collar 1/4" Bore, 1/2	today	$0.63	$10.08
11	1	9056K271	Multipurpose Aluminum (Alloy 6061) Tube 1-1/2" OD,	today	$18.33	$18.33
12	1	88685K79	Basic Aluminum (Alloy 1100) .090" Thick, 24" X 24"	today	$34.82	$34.82
13	1	9370K11	Zinc-Plated Music Wire Spring Assortment 120 Extensi	today	$43.46	$43.46
14	1	95907A44(	SS Extension Spring Anchor Stud Standard, 6-32 Thread, 5/8" Overall Length		$5.80	$23.20
15	1	3334K252	Ultra-High-Polish Type 316/316L SS Super Smooth, 1" OD, .87" ID, .065" Wall, 1'L		$177.90	$177.90
16	1		Omega Resistive Heat Rope		$38.00	$38.00
				Merchandise Total		$556.54
				Shipping		$10.00
				Total Cost		$566.54

Manufacturing

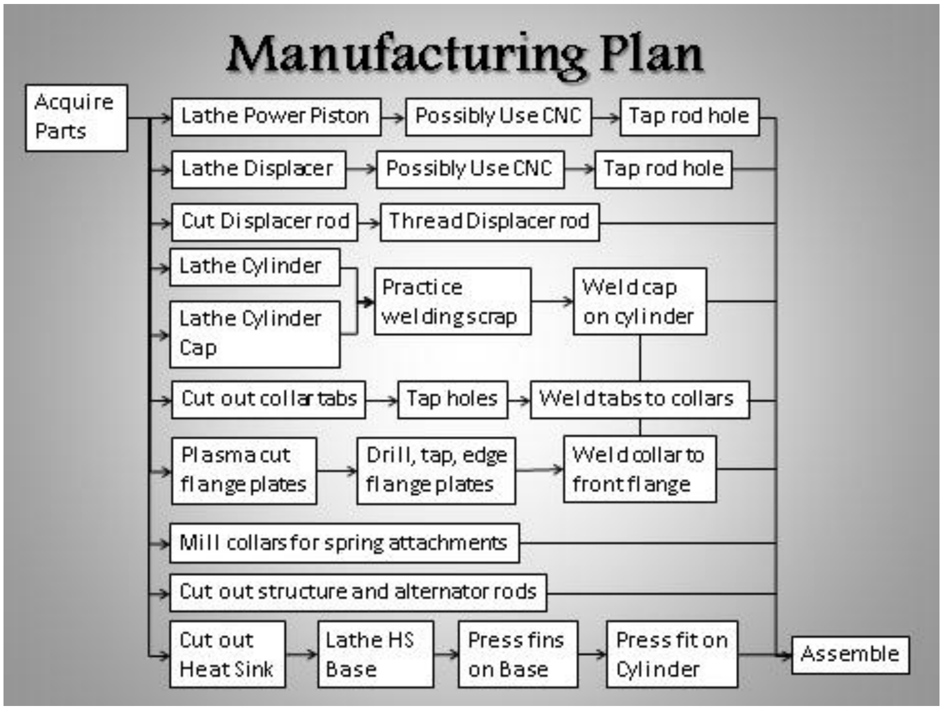


Figure 14: Manufacturing Plan Flowchart

Figure 14 shows the general manufacturing plan flowchart. The most challenging parts to manufacture are those that need CNC mill or CNC lathe work, specifically, the displacer requires CNC lathe work. Fortunately, Matthew Hove, our manufacturing lead worked as a shop tech in the IME department and received special access to the VF2 mill.

A snag was encountered when developing the flange parts. In hopes of avoiding the need for a CNC on the flange parts, it was intended to use the plasma cutter, however, practicing on scrap material taught us that this will not be a good approach, due to warping of thin material and a lack of precision. As an alternative, the flanges were cut out with the VF2 CNC, shown in Figure 15. To expedite the manufacture of the heat sinks, their design was changed slightly. Instead of using the plasma cutter to make circular fins, they were cut into octagonal shapes using the shear. The assembled heat sink is shown in Figure 16.

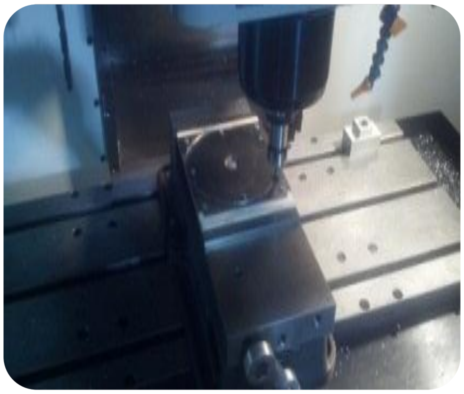


Figure 15: Flange Part CNC Manufacturing

Figure 16: The Assembled Heat Sink

The collars for spring attachments were drilled and milled using a knee-mill. The collar tabs for the anchor mount spring attachments were cut out of scrap metal from the flange plates using a vertical band saw and welded onto collars using TIG process. The power piston as well as the heat sink base, cylinder cap and cylinder were cut to shape and size using a manual lathe. Besides the TIG welding on the front flange plate to the 1" shaft collar and the cylinder to the cylinder cap, the remaining assembly is straight forward with simple Allen-key hand tools.

We decided to manufacture 7 bull nose displacers. 3 of them were made out of aluminum, while the other 4 were made out of brass. There was a significant difference in the weight of the two different materials, and therefore would have a different response inside of the engine. The different displacers were also made of different diameters and lengths to allow for a variety of testing air flow in the engine. To manufacture the displacer, code was created for the CNC lathe to cut bar stock of each material. The especially difficult part in the code was figuring out how to best cut the bull nose shape we had for our displacer. Excel sheets were created to find the best angle, and distance to make sure we did not take off too much material in one pass which is dangerous in this cantilever beam configuration due to chattering and possible material failure in the stock or tooling. Besides safety concerns, taking too deep of a cut creates a vibration in the cantilever and produces a chattered surface finish. Displacer manufacturing is shown in Figure 17.

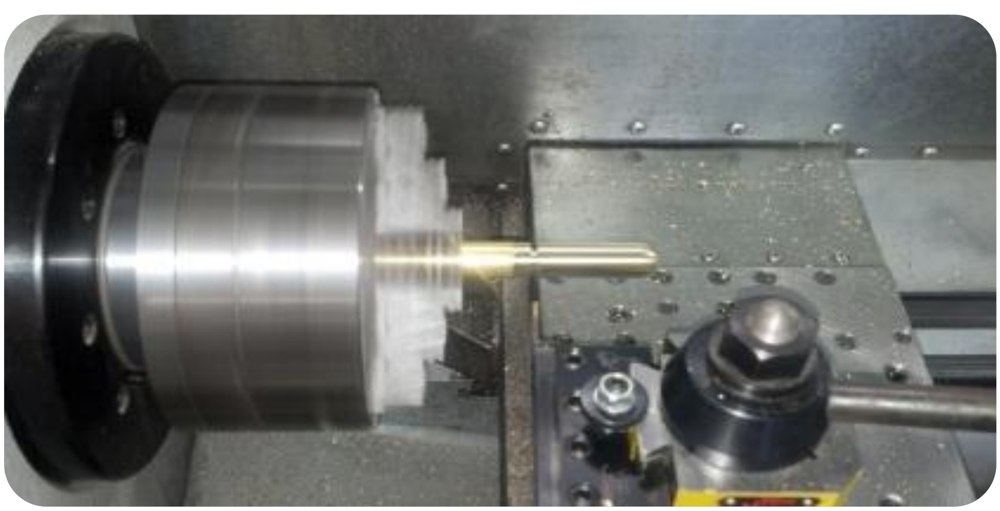


Figure 17: Lathe Manufacturing of the Displacers

The power piston was machined on a hand lathe, and has the highest tolerances. 6” X 1.75” stock was turned on a manual lathe all in one chuck setup to create concentric surfaces, namely the displacer rod bore through the piston with the outer diameter of the piston. These surfaces are meant to be sealed and trap the air within the cylinder, creating a closed system. A center-dill was used to locate a live tailstock to support the cantilever while turning the outer diameter of the piston. A short 3/8” drill bit began the bore to reduce flex in the drill bit and essentially create a bushing for the first inch of the bore. Then a 6” drill bit, also with a 3/8” diameter, continued the bore.

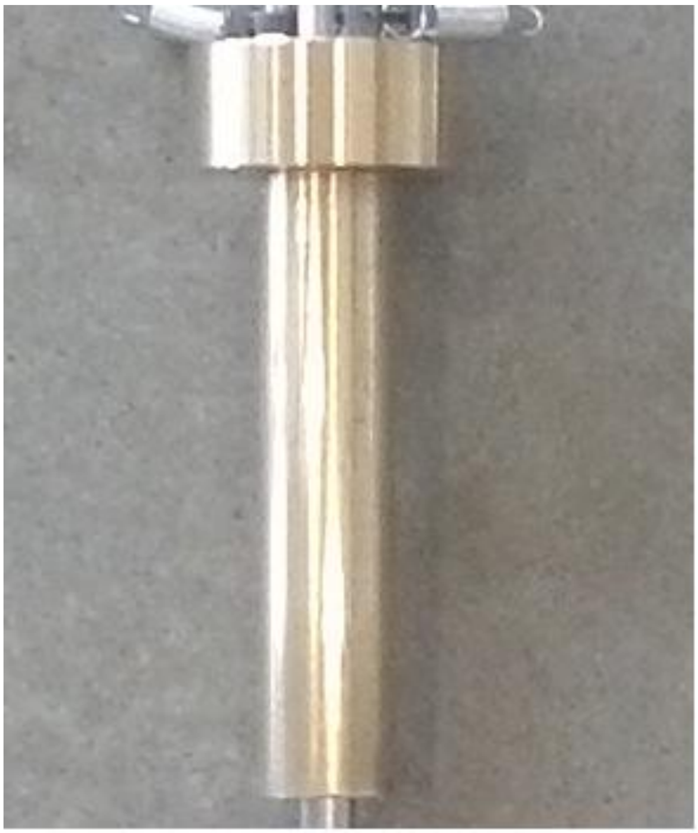


Figure 18: The Manufactured Power Piston

Cylinder caps were also machined from bar stock using a manual lathe, then welded on to the end of the polished cylinder. The cylinder cap manufacturing is shown in Figure 19.

Figure 19: Manufacturing of the Cylinder Caps

Prototype Operation and Test Results

Our testing plan was altered due to the fact that after we assembled the engine, it took considerable time trying to get the engine to start for the first time. We started off using propane as our main source of heat. We quickly found out that without a controlled flame the surfaces on the engine and around it would start to get very warm. We also discovered that it is very challenging to create a repeatable heat input with the propane torch to produce good output test results. This led us to invest in a secondary heat source, which was a resistive heat rope. The heat rope was wrapped around the end of the cylinder and could be controlled with a Variac to adjust the temperature in a controlled manner. The heat rope is also able to be plugged directly into a standard 120V outlet; however, this just causes a max temperature in the rope. The heat rope also provided for a much more consolidated heat input and cooler operation away from the heated end. This in turn, aids the cool end of the system and benefits the duration of the cycling.

At first our tests were mainly aimed at finding the duration for which the engine would last, and this was important in the beginning because the engine would die out quickly. However, experimenting

with the spring tuning helped us to develop trials that would not die out even after 45min. At this point we could end the testing and our testing strategy changed.

Next, our testing worked toward measuring the power output of the engine, and we wished to display the capability of using a linear alternator to draw electricity. We purchased shake-flashlights which have a magnet that one shakes back and forth through a coil to charge the batteries and power the light bulb. This magnet easily mounted onto the steel alternator rods and when the coil was placed around the magnet the engine would light up the bulb. Measurements from this miniscule linear alternator only produced microwatts, however, it was being maxed out and yet still not even placing a load on the engine. Therefore, with a developed linear alternator much more power could be drawn from the engine.

In our recorded testing we had four main different spring configurations, which can be seen in the different colored sections of table 3 below. During these configurations we changed the specific displacer used and tested how long the engine took to start, and how long the engine lasted before dying out. We had multiple sizes of displacers which varied in length, diameter, and material choice.

Other testing involved magnetic suspension. Since the mechanical springs proved to be the weakest link, due to the importance of precise tuning, it is desired to eliminate them and introduce either flexures or magnets. Since magnets were easier to obtain quickly, we experimented with them. By placing magnets on the bottom of the pistons 'shoulder' and on the flange plate, oriented in opposition, we were able to suspend the pistons initial dynamic weight. The engine would cycle and the piston would bounce off due to the magnetic field, but only for a short while. Then the magnets, which were not physically mounted to the flange plate, would be pressed away, sliding and flipping over on the flange plate causing the piston to stick down. With a few calculations and modifications the magnets could be mounted in place and allow the removal of the springs. Furthermore, the magnets would eliminate the physical collisions in the system and create a much quieter operation.

Going forward, we intend to take quality video, possibly high speed video, to accurately measure the frequency and displacement to calculate the potential load that could be given to the engine. Otherwise, accelerometers could be used to gather the displacement, accelerations and frequency, and likewise calculate the potential output. With this testing we could easily compare the various component configurations and glean even more understanding of how to benefit performance. We have now reached one componentry configuration that would call for this testing; prior we had not reached a steady operation worthy of this testing effort. More experimenting is also left to be carried out, because we believe that a longer cylinder length and displacer rod length could allow for a larger displacement of both displacer and power piston, thus, providing more potential power.

With further prototype testing and model development, we could produce a design to reach our initial full scale specifications.

Table 3. Testing Results

	Run	Cyl Length (in)	Rel Cyl Length (in)	Disp Rod	Disp Size-Mat'l	Time to Start	Duration	H.S.	Graf Oil	Shield	Power Disp (in)	Duration till Knock	Ave Voltage (AC)	Notes/Comments
Displacer: 3+11/32" Piston: 3"	1	8	31/32	s	M-AL	2:33	10:40					Entire time/No unthreading		Hit entire time but did not unthread
	2	8	31/32	s	M-AL	1:33	12:22	Y				Fell Apart		Hit entire time and loosened
	3	8	26/32	s	M-AL	1:20	15:00	Y	Y		1.5	Never hit/ loosened		
	4	8	26/32	s	M-AL	2:40	20:00	Y			1.25	Never hit/ loosened		
	5	8	26/32	s	L-AL	1:00	8:00				1.25	Had to Clamp Down		
	6	8	26/32	s	S-AL	NEVER	--	--	--	--	--	Had to Clamp Down		
	7	8	26/32	s	M-AL	Warm Up	5:00		Y		1.125	--		Restarted
Displacer: 4+10/32" Piston: 3+30/32"	8	8	26/32	s	M-AL	1:35	32:48		Y		1.25	Never hit		
	9	8	26/32	s	M-AL	2:00	32:55		Y		1.25	Never hit		
	10	8	26/32	s	M-AL	1:30	32:24		Y	y	1.25	Never hit		
	11	8	26/32	s	L-BR	1:30	2:00 Heat Off				1.75	Hard Knocking and Slamming		
	12	8	26/32	s	M-BR	1:00	2:00 Heat Off				1.75	Hard Knocking and Slamming		
	13	8	26/32	s	S-BR	1:00	17:00				1.75	Less Heat, Hard Knocking		
	14	8	26/32	s	M,SH-BR	1:00	4:00 Heat Off				1.75	Less Heat, Hard Knocking		
	15	8	26/32	s	M,SH-BR	1:35	10:36				2	lamming piston into Cylinder		
D: 5+4/32" P: 4+24/32"	16	8	13/32	s	M,SH-BR	NEVER	--				--	--	--	I believe that there was simply too much friction due to cylinder set screw and with some work, this could allow finer testing on cylinder length
	17	8	26/32	s	M,SH-BR	NEVER	--				--	--	--	
	18	8	1	s	M,SH-BR	NEVER	--				--	--	--	
	19	8	1+14/32	s	M,SH-BR	NEVER	--				--	--	--	
D: 4+11/32" P: 3+31/32"	20	8	15/32	s	M-BR								2.5	
	21	8	20/32	s	M,SH-BR								2.4	
	22	8	15/32	s	L-BR							Slamming piston into Cylinde	2.7	Pounded to 9/32"
	23	8	29/32	s	S-BR								2	
	24	8	26/32	s	M-AL								1.5	

Conclusion

This project provided a great opportunity to gain experience with mechanical design, manufacturing, and the fundamentals of Stirling engines. We did not anticipate how challenging it was to find appropriate materials to withstand the heat with similar expansion coefficients, manufacture the components with campus resources and tolerances tight enough to create a pressurized air seal while keeping low friction, and produce a consistent operating free-piston Stirling engine. However, with more than 20 hours of testing we have gained a great deal of insight into the astonishing cycle of Stirling engines. Our current system configuration has run at a steady-state reliably for over 4 hours, and we have gained iteration ideas to improve the performance and reliability.

We aim to test the longer stoke length configuration, already manufactured, as well as magnetic suspension. Once satisfied with the system performance, design and produce a linear alternator capable of loading the engine. It would be great at this point to mount it on a solar concentrator and draw electricity from the sun even from remote locations.

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Appendix A

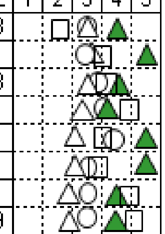
Table 2: House of Quality

▲ Larger is Better
◉ Nominal is Best
▼ Smaller is Better

Customer Desciption:
1 = Mello and his cabin
2 =
3 = Manufacturer

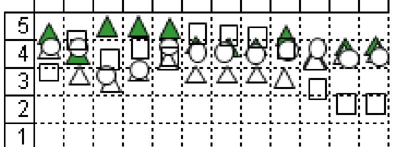
Customer Requirements (Whats)	Importance	A Power Output ▲	B Cost ▼	C Startup Time ▼	D Startup Torque ▼	E Life Cycle ▲	F Length ▼	G Width ▼	H Height ▼	I Weight ▼	J Internal Pressure ◉	K Heat Source Efficiency ▲	L Heat Differential ▼
Constant Power Output	5	9				1					1	3	3
Ease of Starting	5			3	3								
Manufacturability	5		9				1	1	1	1	3		3
Low Cost	5		9			1	1	1	1	1	3	3	
System life time	4		3			9							
Maintenance	4	1	3			3					3		
Portablility	3		1				9	9	9	9		3	
Small Size	3						9	9	9	3		1	9

Customer Ratings (1 Bad – 5 Good)



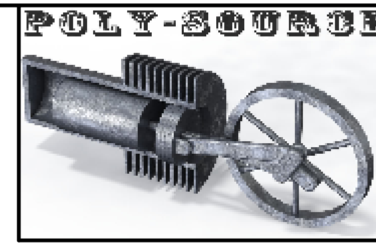
△ Alpha
○ Beta
□ Gamma
▲ Free Piston

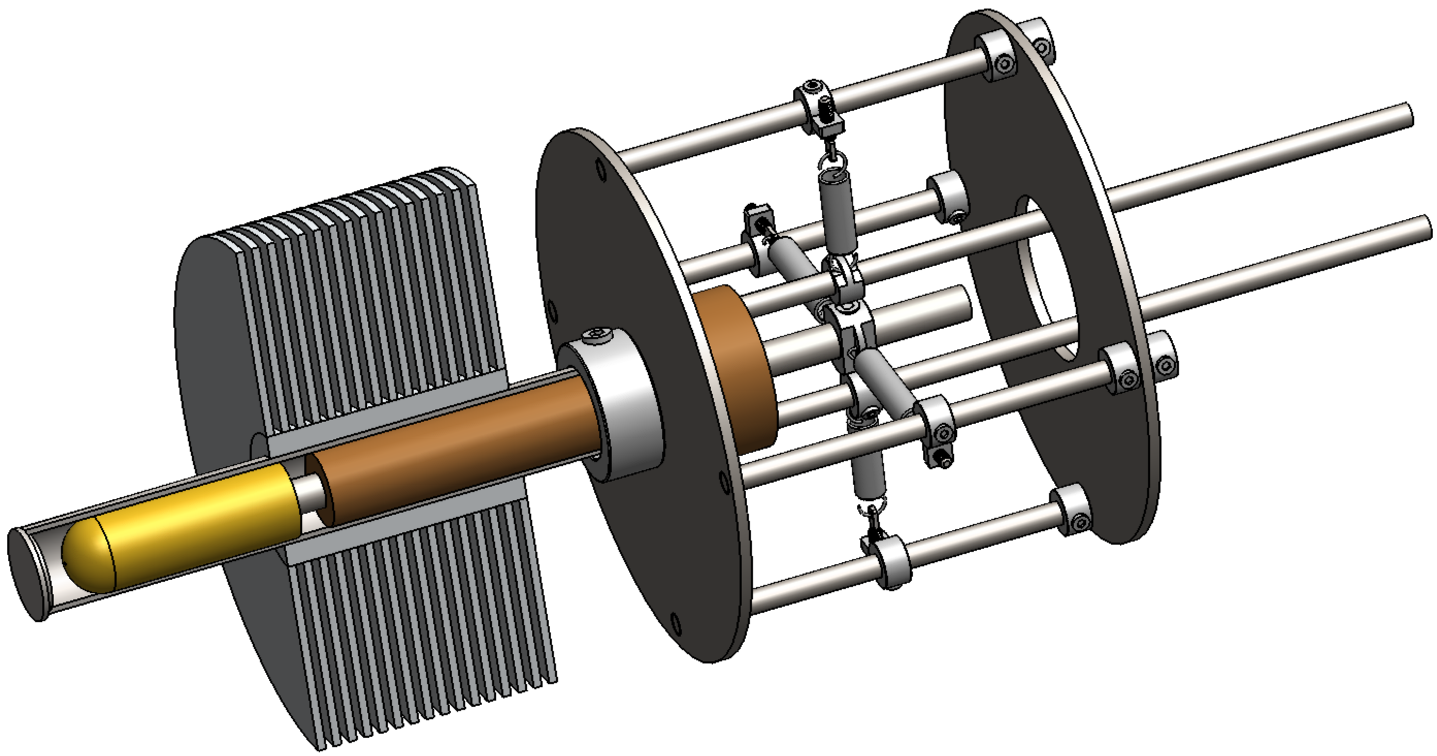
Strong - 9 ◎
Medium-3 ○
Weak - 1 △
Relationship Strength



	A	B	C	D	E	F	G	H	I	J	K	L
Targets	3 Kw	Less than 4,000 dollars	Less than 5 minutes	Less than xxx ft-lbs	At least 5 years	Less than 4 ft	Less than 5 ft	Less than 5 ft	Less than 1,500 lbs	atmospheric to xxx psi	xxx % or more	Between 300 and 20 Degrees celsius
Weighted Importance	9	9	7	5	9	6	6	6	6	5	7	8
% Importance	10.0	10.0	8.4	6.0	10.0	7.2	7.2	7.2	7.2	6.0	8.4	9.6

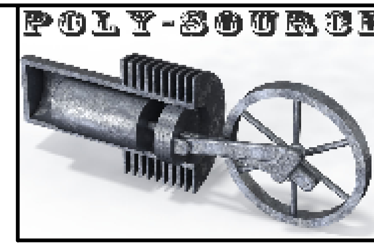
Appendix B (Detail Drawings)

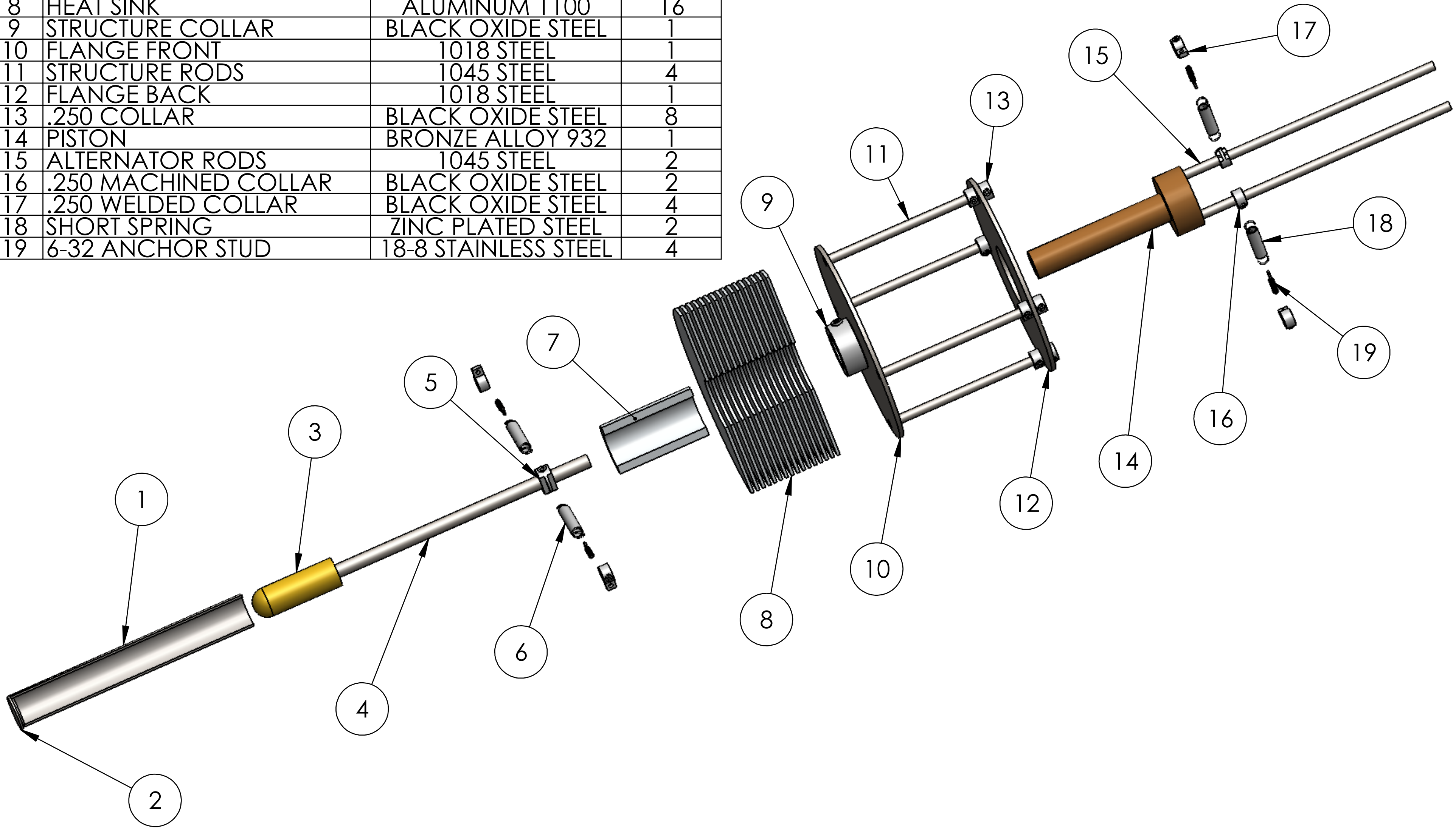




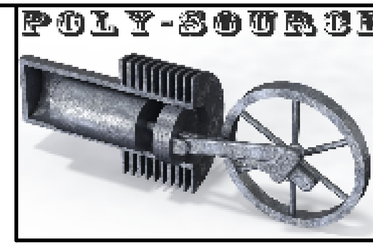
California Polytechnic State University	Title: ASSEMBLED PROTOTYPE			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #:	Nxt Asb:	Date: 2/15/13	Scale: 1:2	Chkd. By : JOSHUA DULIN

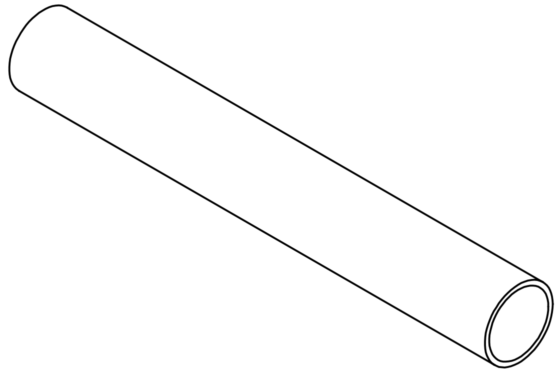
NO.	DESCRIPTION	MATERIAL	QTY.
1	CYLINDER	STEEL 316/316L STEEL	1
2	CYLINDER CAP	STEEL 316/316L STEEL	1
3	DISPLACER	BRASS ALLOY 360	1
4	DISPLACER ROD	STAINLESS STEEL	1
5	.375 MACHINED COLLAR	BLACK OXIDE STEEL	1
6	LONG SPRING	ZINC PLATED STEEL	2
7	FIN BASE	ALUMINUM 6061	1
8	HEAT SINK	ALUMINUM 1100	16
9	STRUCTURE COLLAR	BLACK OXIDE STEEL	1
10	FLANGE FRONT	1018 STEEL	1
11	STRUCTURE RODS	1045 STEEL	4
12	FLANGE BACK	1018 STEEL	1
13	.250 COLLAR	BLACK OXIDE STEEL	8
14	PISTON	BRONZE ALLOY 932	1
15	ALTERNATOR RODS	1045 STEEL	2
16	.250 MACHINED COLLAR	BLACK OXIDE STEEL	2
17	.250 WELDED COLLAR	BLACK OXIDE STEEL	4
18	SHORT SPRING	ZINC PLATED STEEL	2
19	6-32 ANCHOR STUD	18-8 STAINLESS STEEL	4

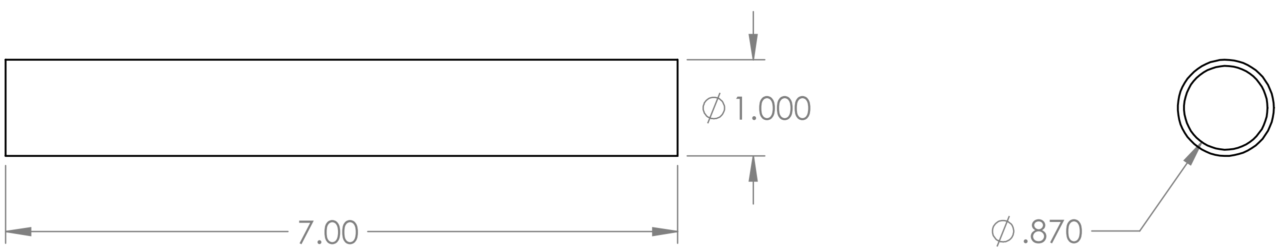




California Polytechnic State University	Title: CONCEPTUAL PROTOTYPE EXPLODED COMPONENT VIEW			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #:	Nxt Asb:	Date: 2/14/13	Scale: 1:4	Chkd. By: JOSHUA DULIN

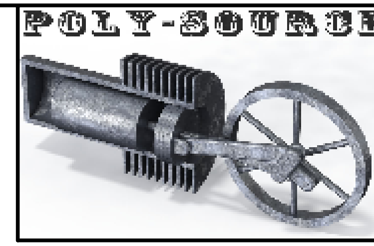


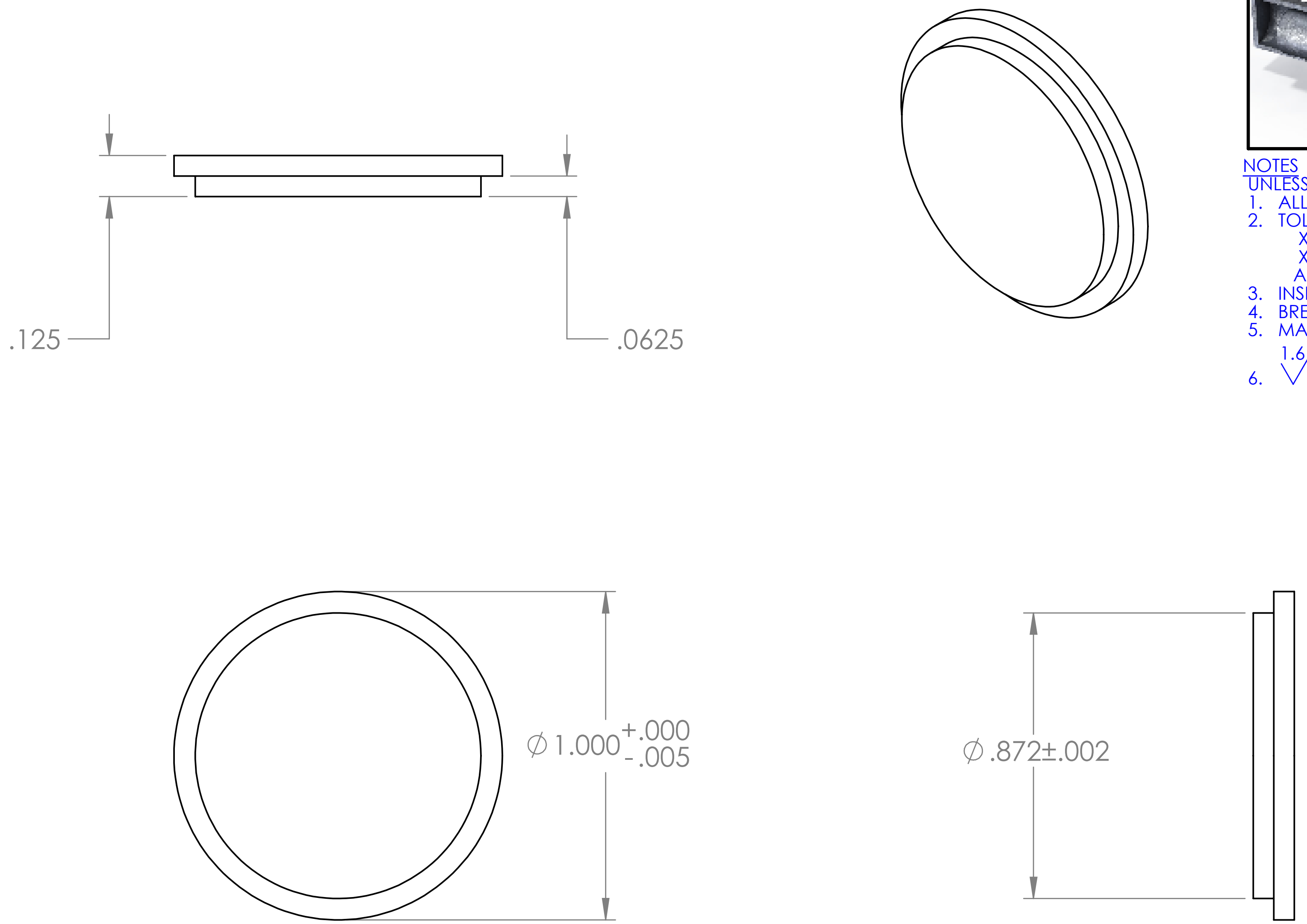




NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX= ±.01
   X.XXX= ±.005
   ANGLES= ±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 316/316L STAINLESS
6. 1.6 √ FAO

California Polytechnic State University Mechanical Engineering	Title: CYLINDER				Drwn. By: MATTHEW HOVE
	Dwg. #: 1	Nxt Asb:	Date: 2/7/13	Scale:1:2	Chkd. By: JOSHUA DULIN





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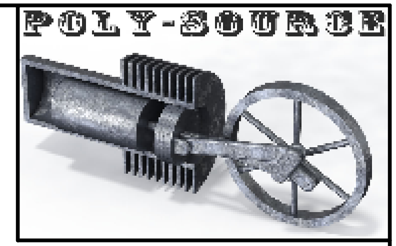
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 316/316L STAINLESS
6. 1.6√ FAO

California Polytechnic State University	Title: CYLINDER CAP			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 2	Nxt Asb:	Date: 2/7/12	Scale: 2:1	Chkd. By: JOSHUA DULIN



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X.XXX= ± .005
ANGLES= ± 1°
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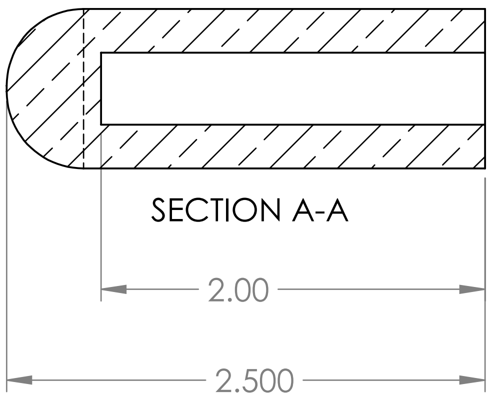
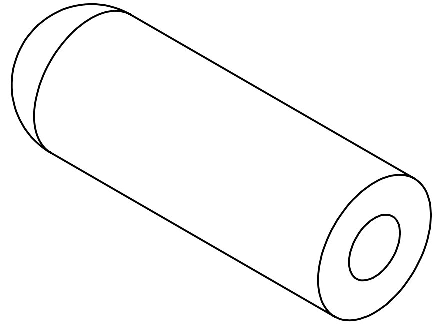
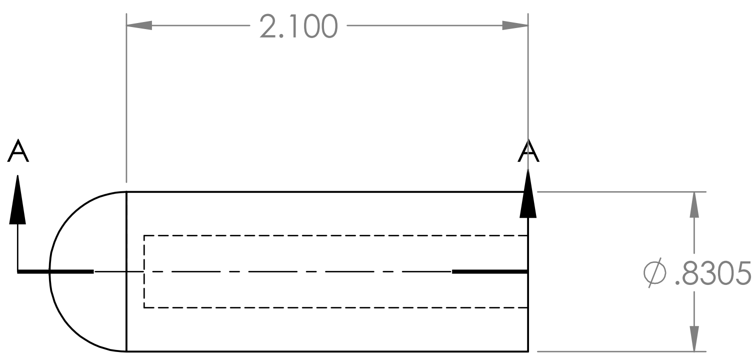
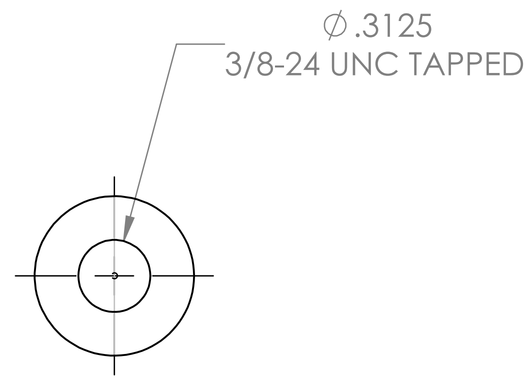
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: BRASS 360 ALLOY
6. 1.6 √ FAO









California Polytechnic State University Mechanical Engineering	Title:DISPLACER			Drwn. By: MATTHEW HOVE
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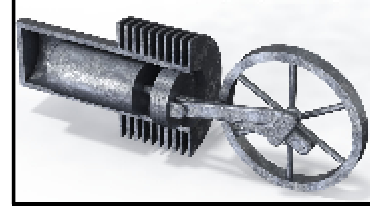
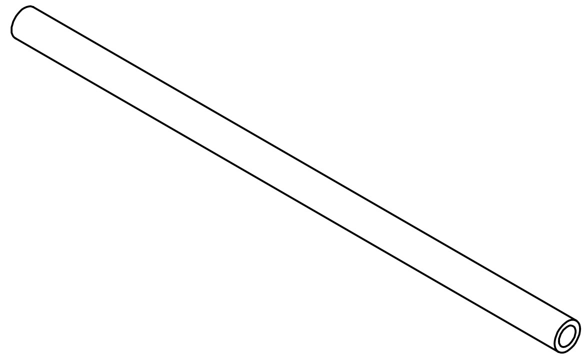
POLY-SOURCE





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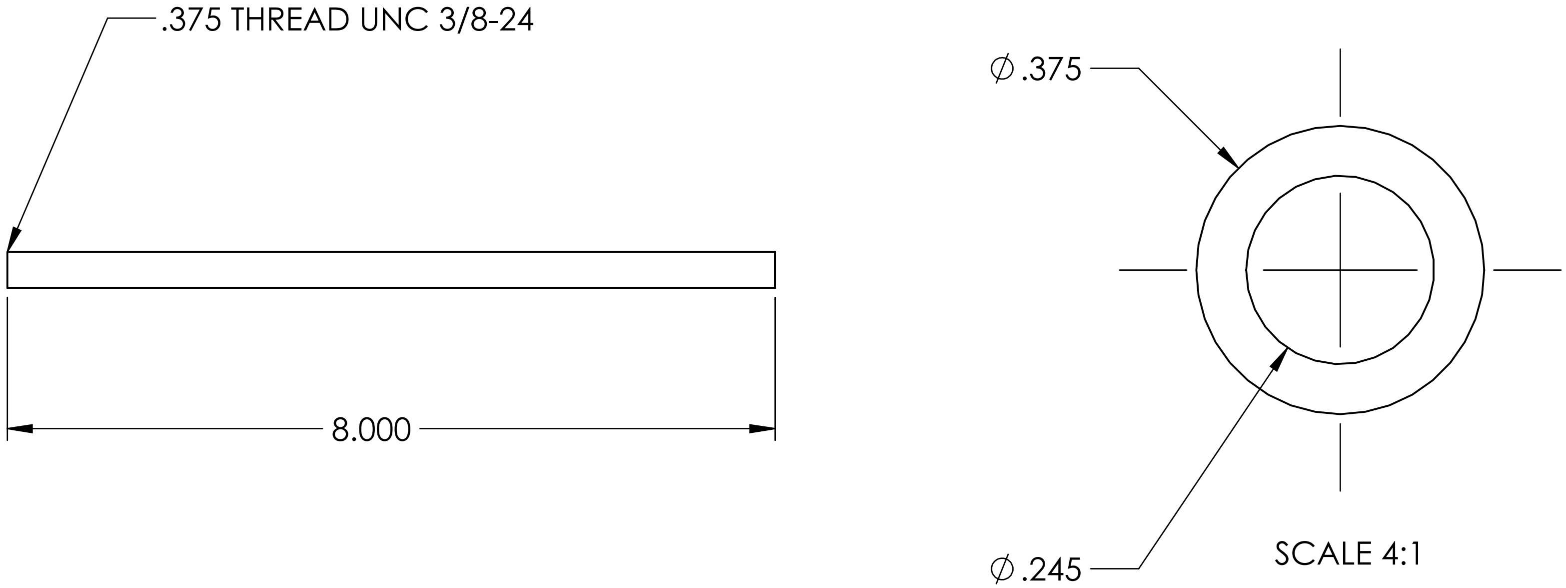
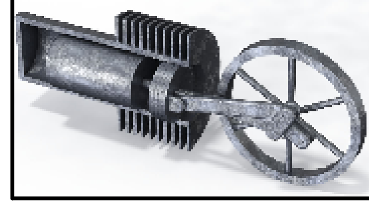
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX= ±.01
   X.XXX= ±.005
   ANGLES= ±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 4130 STEEL
6. 1.6 √ FAO



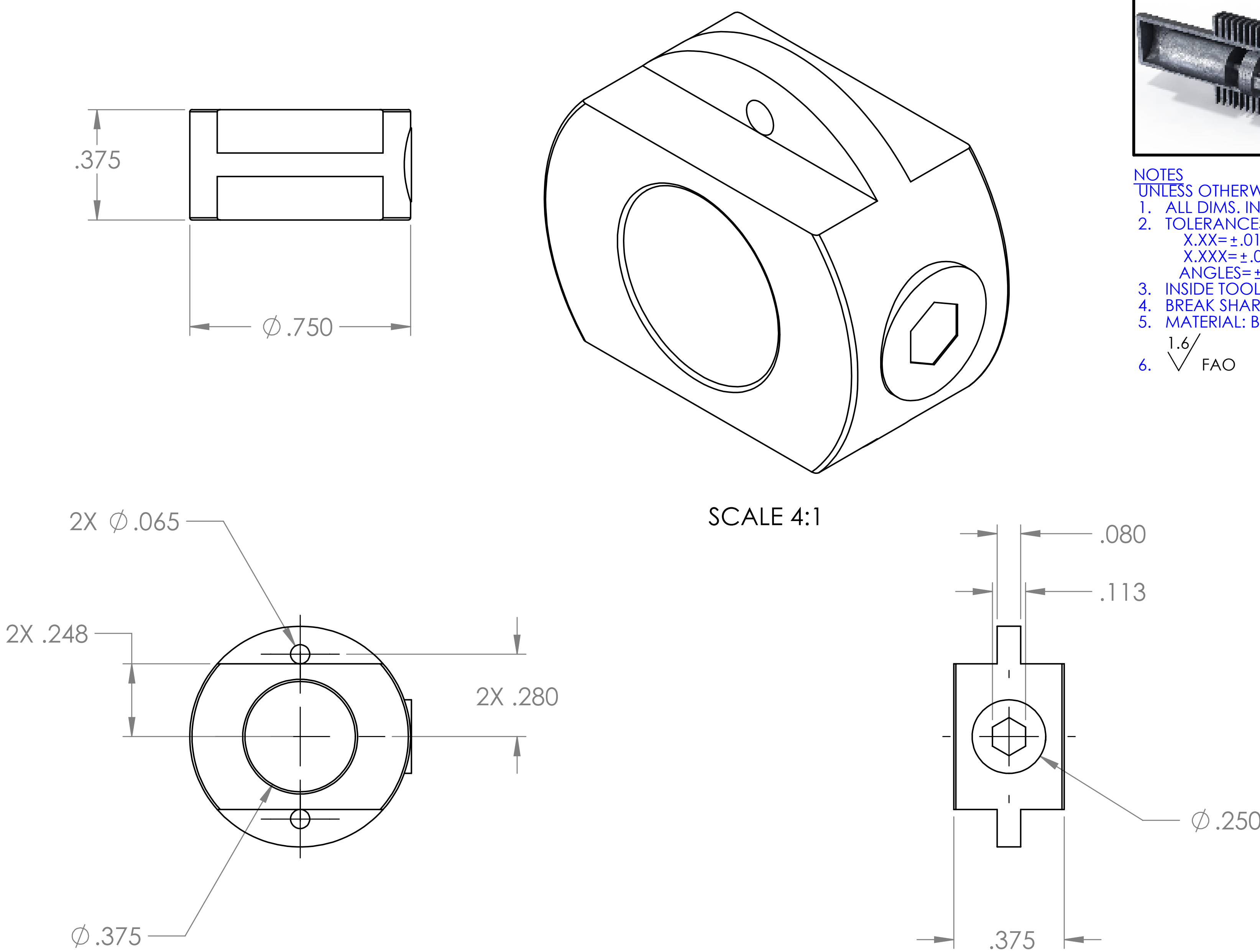
SCALE 4:1

California Polytechnic State University	Title:DISPLACER ROD			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #:4	Nxt Asb:	Date:2/7/2013	Scale: 1:2	Chkd. By: JOSHUA DULIN

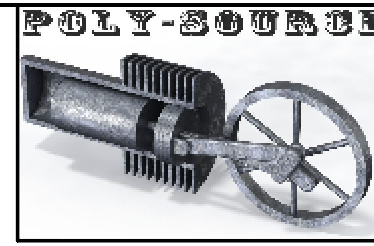
POLY-SOURCE

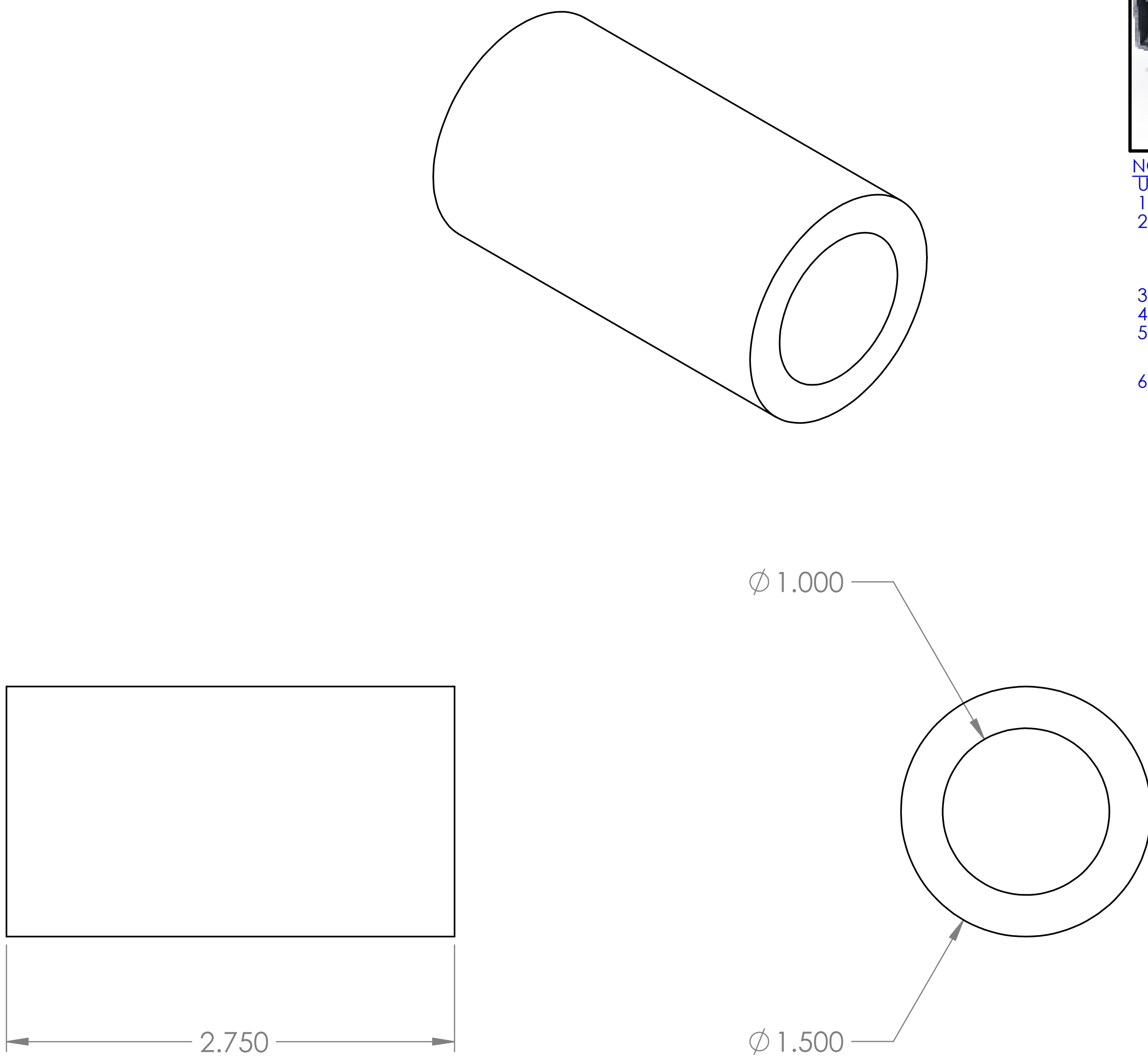


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: BLACK OXIDE STEEL
6. 1.6 √ FAO



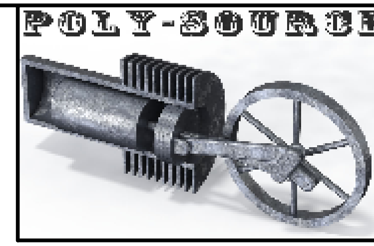
California Polytechnic State University	Title:.375 MACHINED COLLAR			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 5	Nxt Asb:	Date:2/7/2013 Scale:2:1	Chkd. By:JOSHUA DULIN



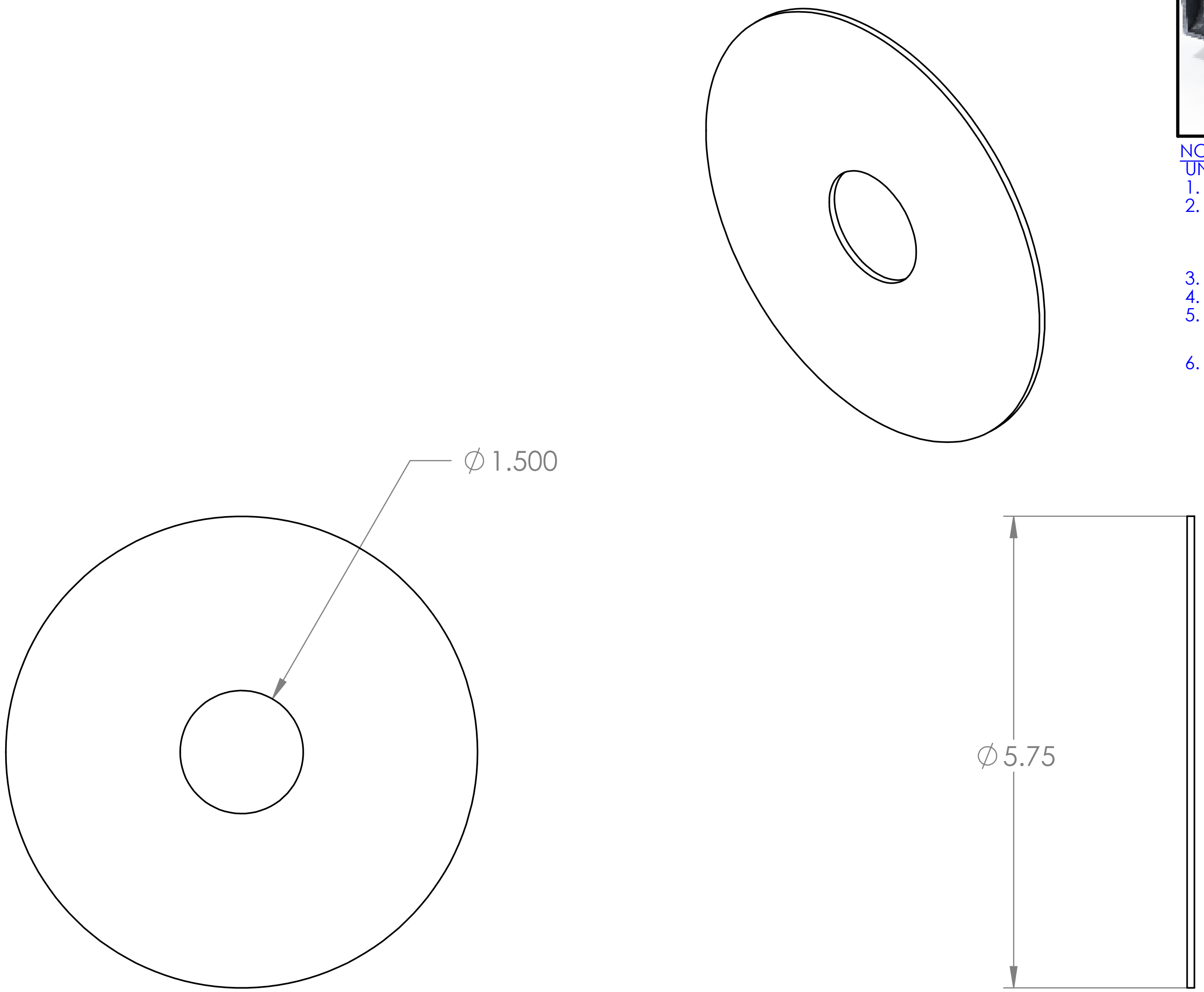


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: AL 6160 ALLOY
6. 1.6√ FAO

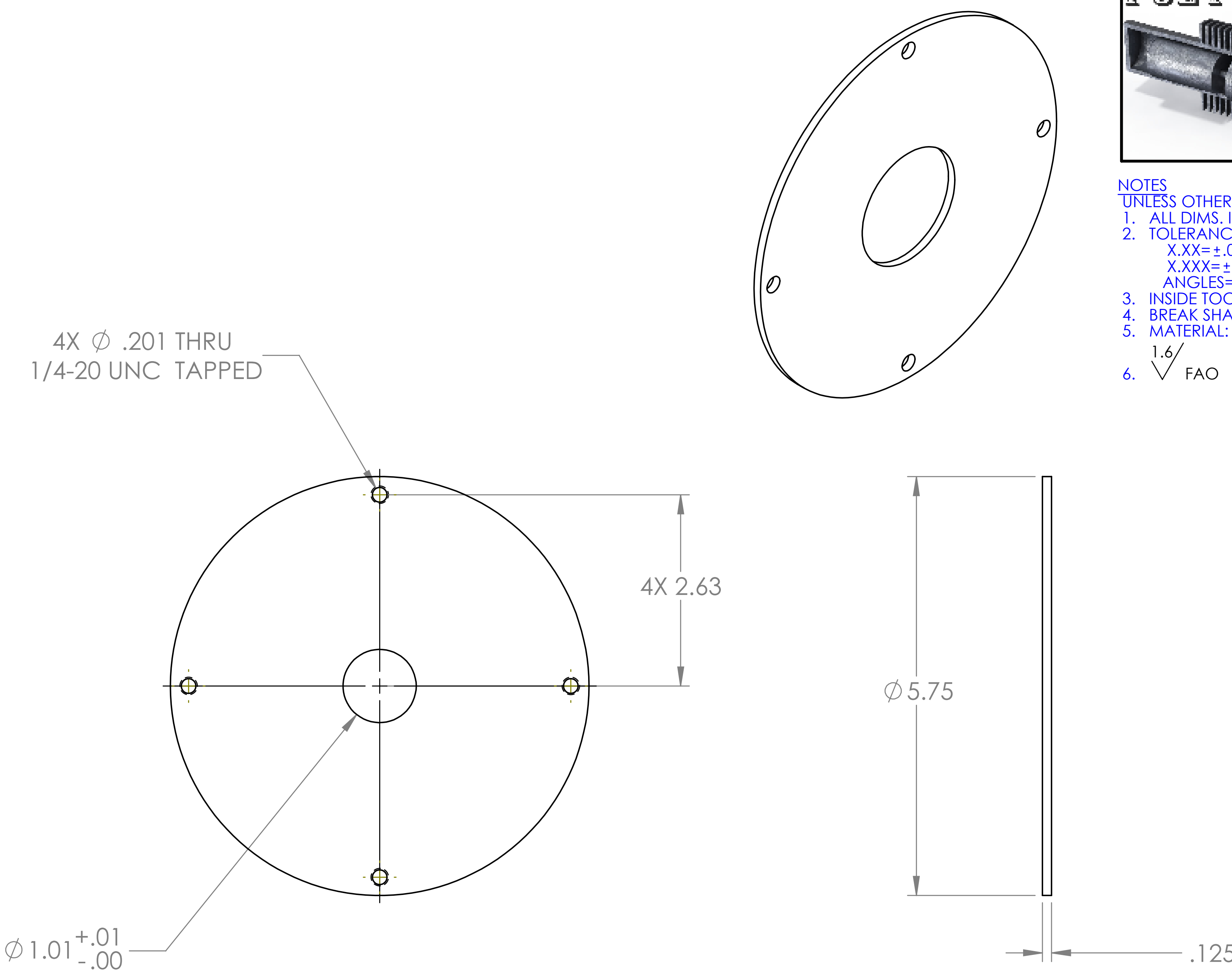
California Polytechnic State University	Title: FIN BASE				Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 7	Nxt Asb:	Date: 2/7/13	Scale: 1:1	Chkd. By: JOSHUA DULIN

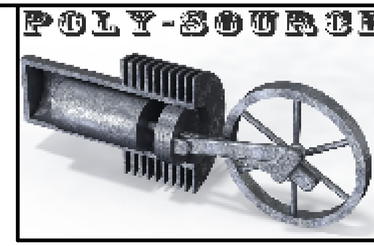


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX= ±.01
   X.XXX= ±.005
   ANGLES= ±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: AL 1100 ALLOY
6. 1.6 √ FAO



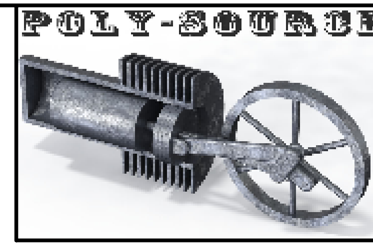
California Polytechnic State University	Title: HEAT SINK			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 8	Nxt Asb:	Date: 2/7/13	Scale: 1:2	Chkd. By: JOSHUA DULIN



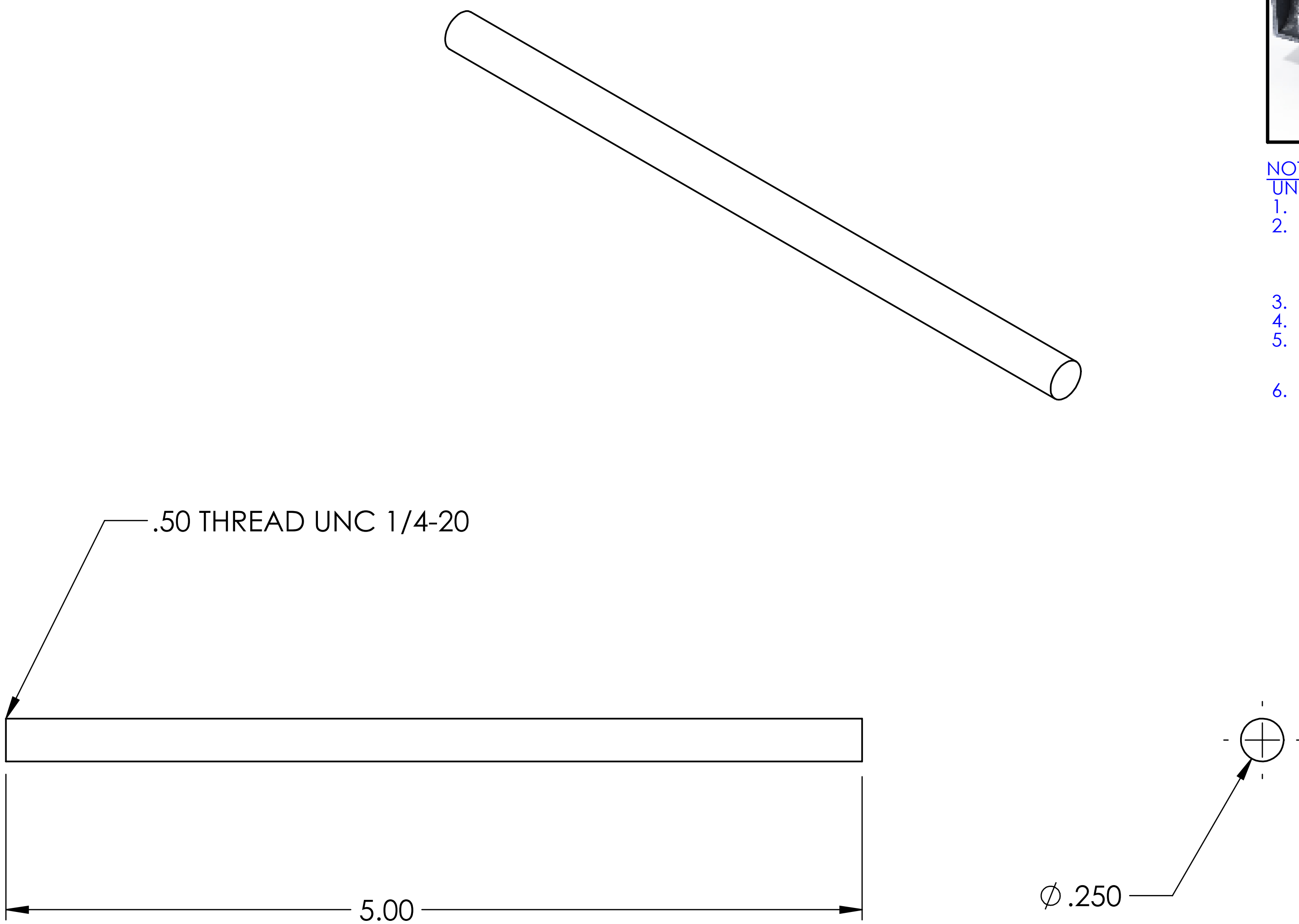


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX= ±.01
   X.XXX= ±.005
   ANGLES= ±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 1018 STEEL
6. 1.6 √ FAO

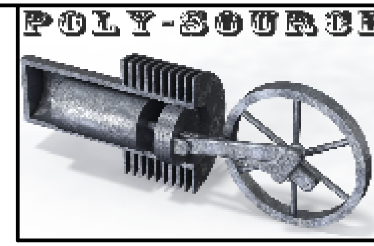
California Polytechnic State University Mechanical Engineering	Title: FRONT FLANGE PLATE				Drwn. By: MATTHEW HOVE
	Dwg. #:10	Nxt Asb:	Date:2/7/2013	Scale:1:2	Chkd. By:JOSHUA DULIN



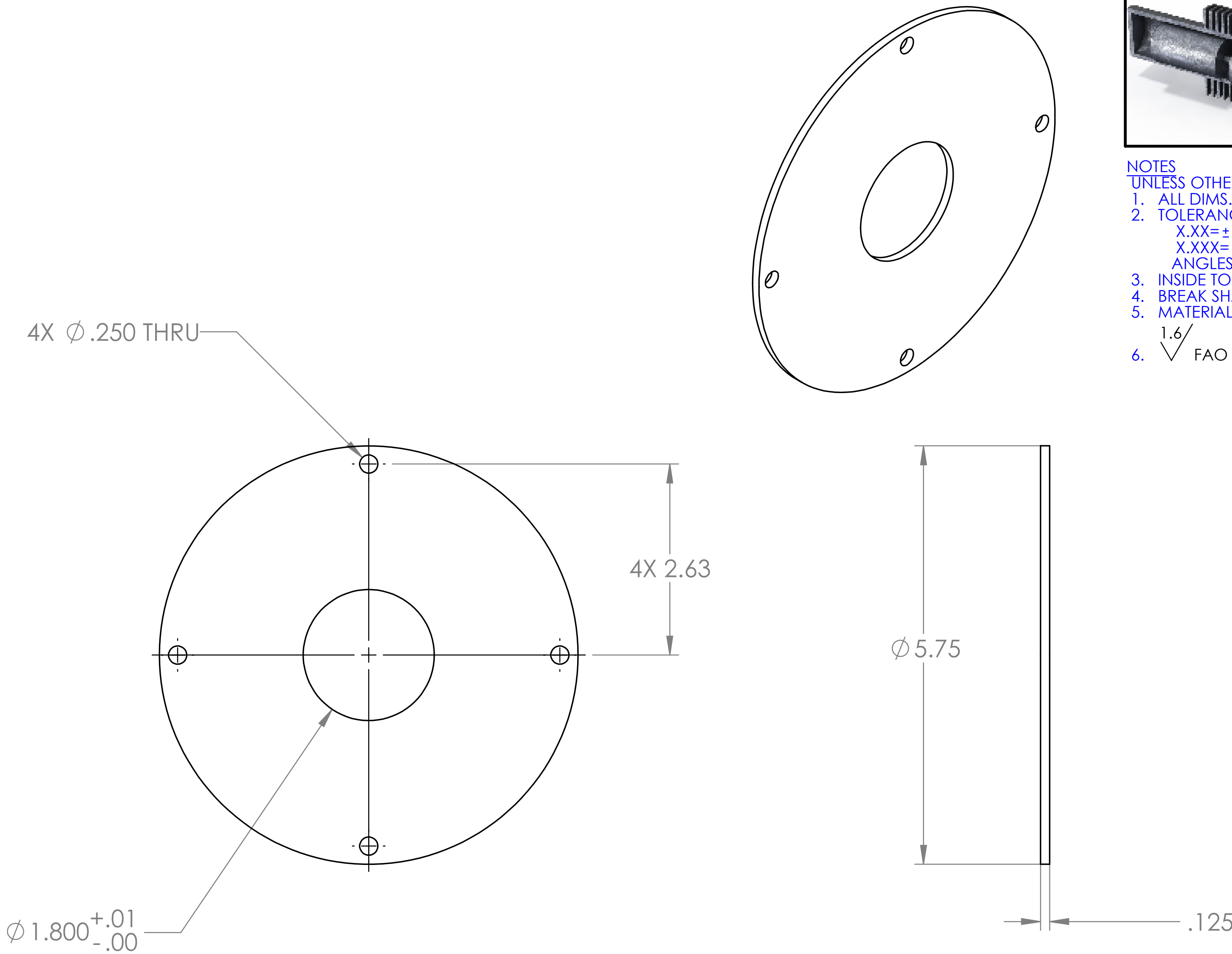
NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 1045 STEEL
6. 1.6 √ FAO



California Polytechnic State University	Title: STRUCTURE RODS			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 11	Nxt Asb:	Date:2/7/2013	Scale:1:1	Chkd. By: JOSHUA DULIN

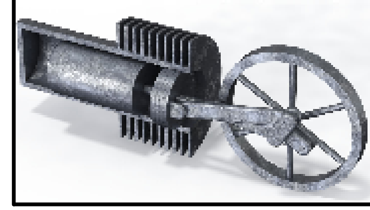


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 1018 STEEL
6. 1.6 √ FAO

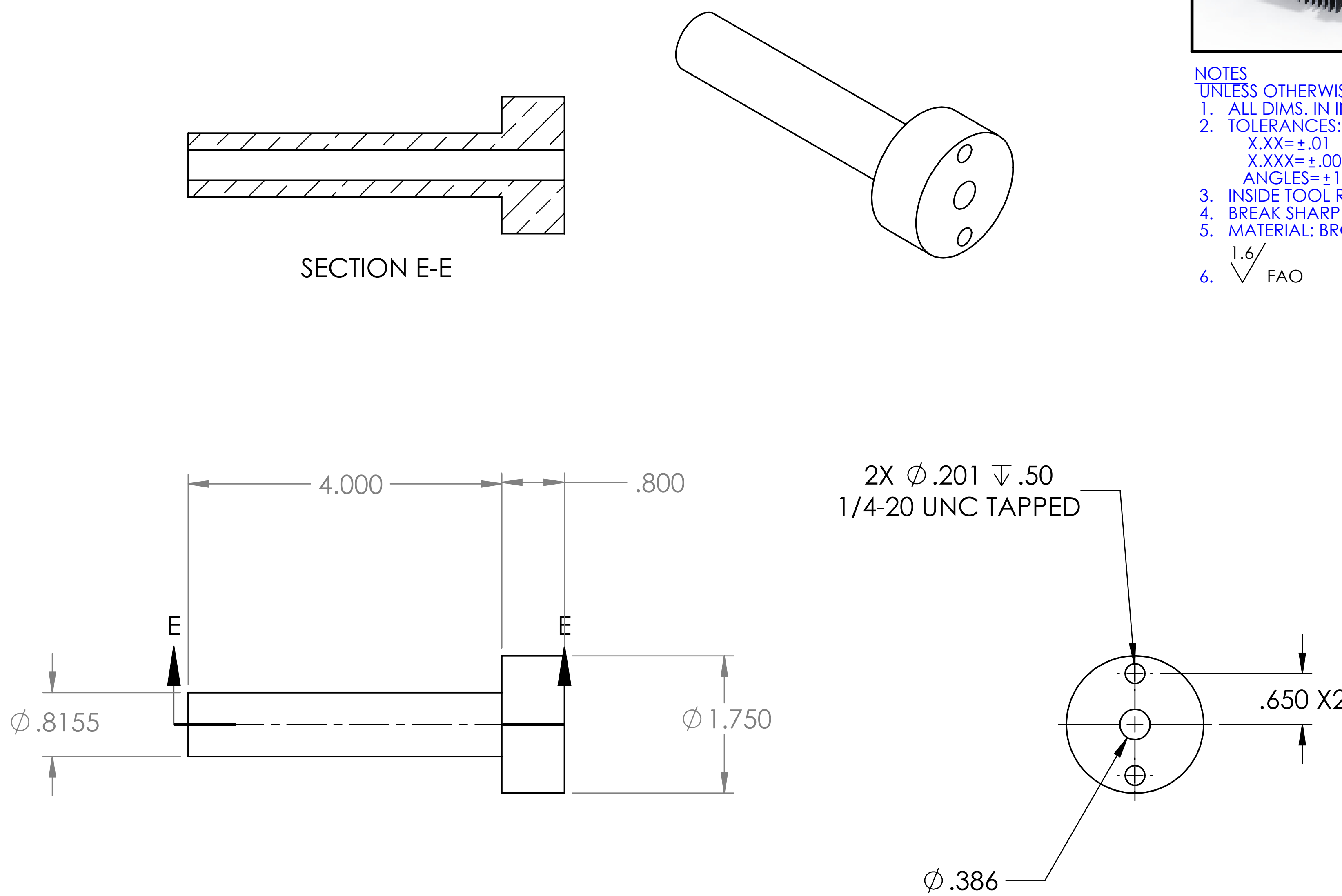


California Polytechnic State University	Title:BACK FLANGE PLATE			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 12	Nxt Asb:	Date:2/7/2013	Scale:1:2
				Chkd. By:JOSHUA DULIN

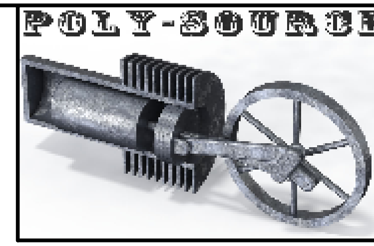
POLY-SOURCE



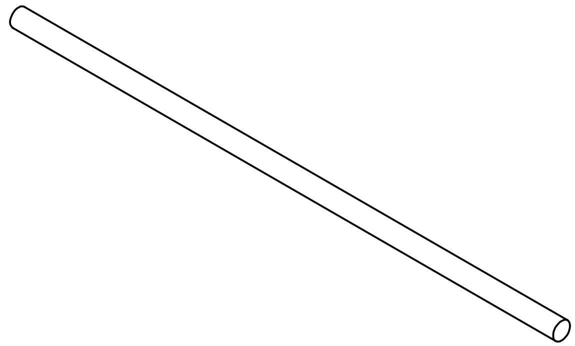
NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: BRONZE 932 ALLOY
6. 1.6 √ FAO

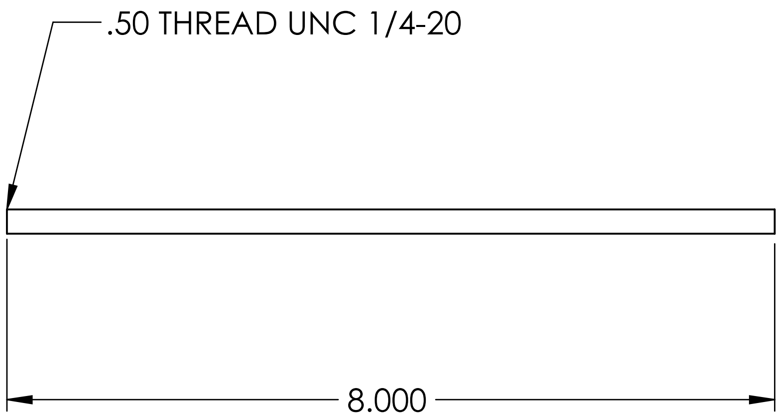


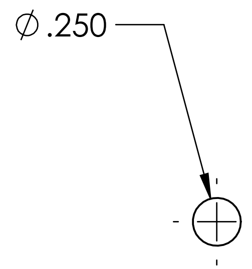
California Polytechnic State University Mechanical Engineering	Title:POWER PISTON			Drwn. By: MATTHEW HOVE
	Dwg. #: 14	Nxt Asb:	Date:2/7/2013	Scale: 1:2	Chkd. By: JOSHUA DULIN



NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: 1045 STEEL
6. 1.6 √ FAO

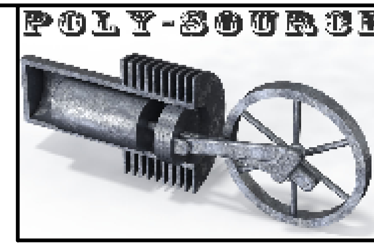


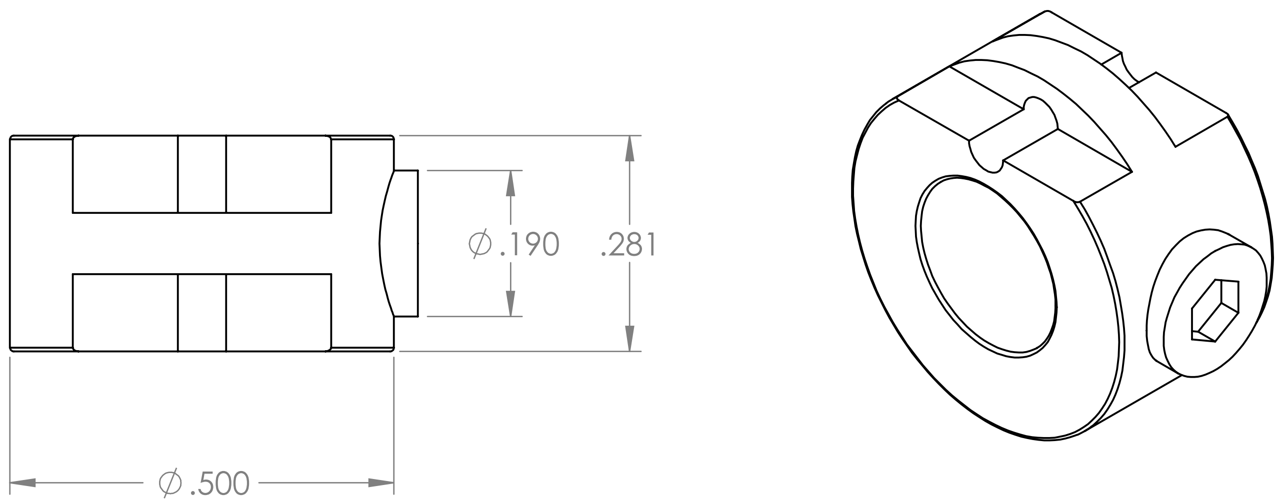




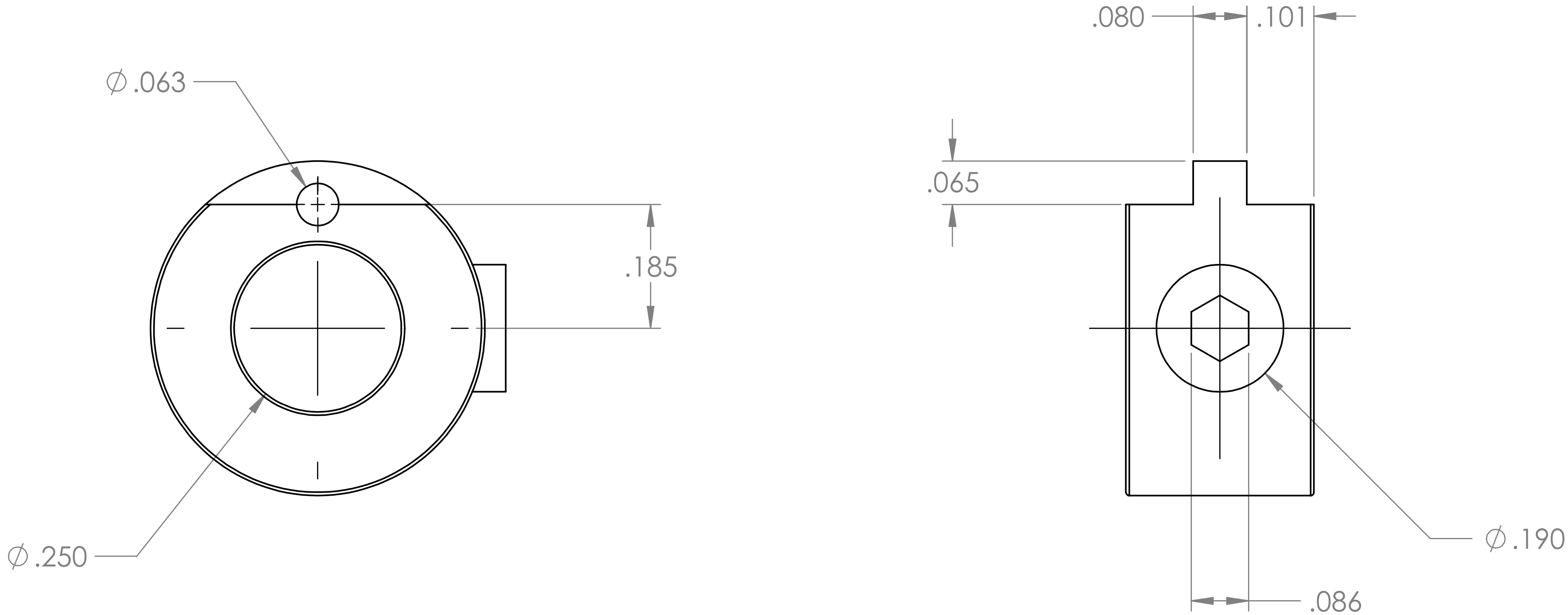
SCALE 1:1

California Polytechnic State University Mechanical Engineering	Title:ALTERNATOR RODS				Drwn. By: MATTHEW HOVE
	Dwg. #: 15	Nxt Asb:	Date:2/7/2013	Scale: 1;2	Chkd. By: JOSHUA DULIN

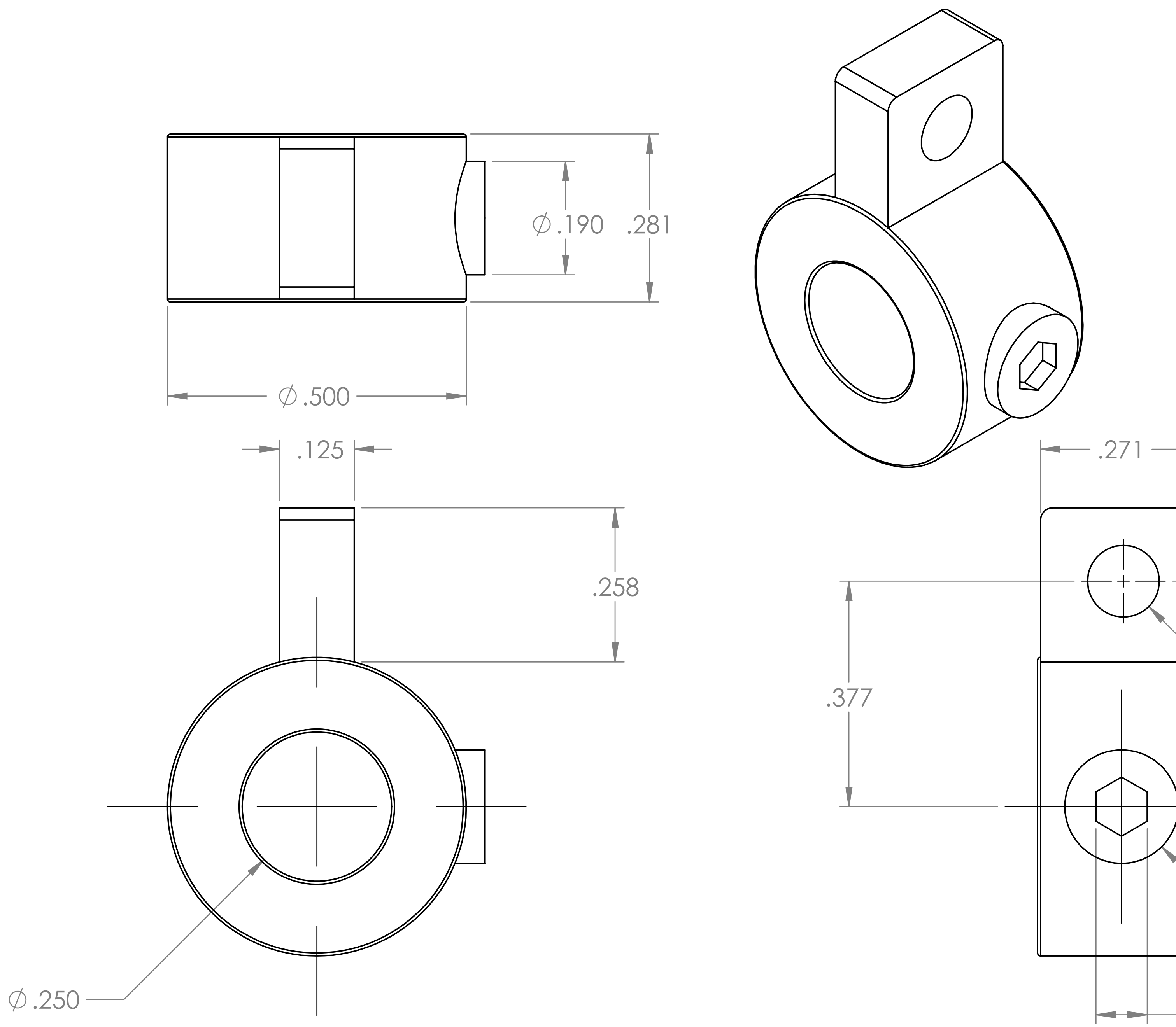


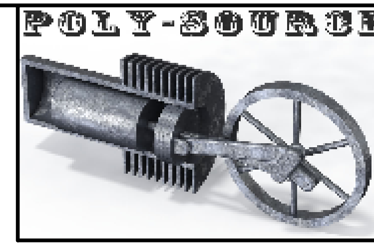


NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
   X.XX=±.01
   X.XXX=±.005
   ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: BLACK OXIDE STEEL
6. 1.6 √ FAO



California Polytechnic State University	Title:.250 Machined Shaft Collar			Drwn. By: MATTHEW HOVE
Mechanical Engineering	Dwg. #: 16	Nxt Asb:	Date:2/7/2013 Scale:4:1	Chkd. By:JOSHUA DULIN





NOTES
UNLESS OTHERWISE SPECIFIED:
1. ALL DIMS. IN INCHES
2. TOLERANCES:
X.XX=±.01
X.XXX=±.005
ANGLES=±1°
3. INSIDE TOOL RADIUS .02 MAX.
4. BREAK SHARP EDGES .02 MAX.
5. MATERIAL: BLACK OXIDE STEEL
6. 1.6 √ FAO

California Polytechnic State University Mechanical Engineering	Title:.250 WELDED COLLAR			Drwn. By: MATTHEW HOVE
	Dwg. #: 17	Nxt Asb:	Date:2/7/2013	Scale:4:1
				Chkd. By:JOSHUA DULIN