Thermostatic Hydraulic Valve

For use in series hydraulic vehicles

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Abstract

The Environmental Protection Agency has developed viable series hydraulic hybrid technology, which they wish to implement in delivery trucks for the United Parcel Service (UPS). In order to commercialize this technology, the cost of many components needs to be reduced. Efficiency losses in the hydraulic system cause the temperature of the working fluid to rise, which can cause undesired effects throughout the system. A thermostatic control valve is used to keep the hydraulic fluid within a narrow temperature range by routing a percentage of the fluid to a cooling radiator if the fluid is hot, and if the fluid is cool, diverting it back to the system. This valve requires design due to its specific nature and need for a very low production cost.

Executive Summary

The purpose of this design project was to provide the Environmental Protection Agency (EPA) with a thermostatic control valve compatible with their hydraulic hybrid delivery truck. Our main focus was to keep the valve production cost low (under ten dollars) to help drive down production cost of their prototype trucks. The motivation for this challenge is the EPA's desire to take their hydraulic hybrid delivery trucks from prototype phase to a marketable product.

The thermostatic control valve has to route at least 95% of the fluid passing through the housing to a cooling radiator when temperatures reach 180°F and above and divert 95% back into the system when under 160°F. The response time must be on the order of minutes. Pressure loss across the valve is undesirable and should be kept to less than 10 psi. Our system must be capable of long lasting operation in the system conditions, (flow rate of 35 gal/min, maximum pressure of 200 psi, and maximum temperature of 180° F).

We generated many concepts that utilize novel systems to redirect flow in the valve. Through several iterations of pugh chart analysis and critical discussion we were able to generate a final design that fulfilled our sponsor's requirements in the most efficient manner.

This design uses a linear thermal actuator attached to a plunger that seals the outgoing recirculation line when heated. Attached to this plunger is a gate that simultaneously unseals the outgoing radiator line. This valve member is seated in a 3-way junction housing. The valve member is held in place by a spoked retaining disk. The simplicity of our design will help reduce manufacturing costs. The design is as spatially unobtrusive as possible and all flow obstructions are optimized (based on stress analysis) to have as little effect as possible on flow characteristics.

Fabrication of this initial prototype was done in-house and most problems were encountered in creating the radial geometries on the retaining disk and plug and gate. These were resolved through using the waterjet in the ERC/RMS at the University of Michigan and fabricating lathe fixtures to position the pieces and keep all aspects of these geometries on-center.

We performed flow and pressure loss testing with air in an open system; these results were then scaled to match system conditions. These results proved invalid as there were likely local compressibility effects. The valve was then tested in a recirculating system at temperature, pressure, and flow rate. These results showed our pressure drop to be higher than originally anticipated by our tests. These results were comparable to other industry valves.

During testing it seemed that there was an issue with the valve member achieving full actuation. There are several reasons for this, mainly the location and nature of the spring and the degree to which the valve member assembly components are co-axial. The high pressure loss can be lowered by increasing the dimensions of the valve to enlarge flow constriction points. In addition to this the components of the valve member could be assembled in a jig to reduce misalignment. Also the spring's location can be modified to achieve better actuation.

This first run-prototype was successful at allowing us to make recommendations to improve future prototypes and work by the EPA in this field.

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1. Introduction

The U.S. Environmental Protection Agency (EPA) has been conducting extensive research and development of hydraulic hybrid systems for delivery truck vehicles. The purpose of a hydraulic hybrid system is to reduce fuel consumption of a vehicle by over 50%. To achieve the desired efficiency, the temperature of the system's hydraulic fluid must remain in the $170^{o}F \pm 10^{o}F$ range.

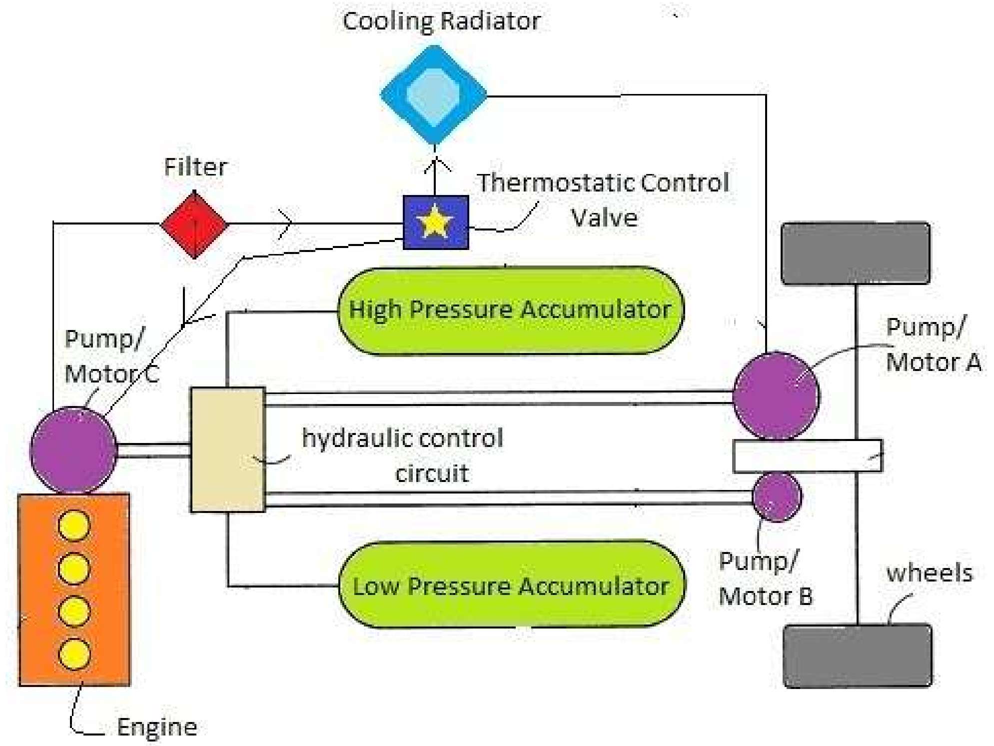
As the hybrid vehicle is in use, the hydraulic fluid heats up. To prevent the fluid from becoming too hot, a certain amount of it is routed to a radiator, where it is cooled. A thermostatic valve is required to route hydraulic fluid based on its temperature. Such thermostatic valves exist; however, they have not performed as anticipated in the EPA's hydraulic hybrid system. Problems with current valves include: high price, poor temperature control, and overall inefficiency in properly routing the hydraulic fluid. To correct these problems and maintain efficiency in both cost and operation, the EPA needs us to design a thermostatic valve which will work with their current hybrid system.

A successful valve design will keep the hydraulic fluid within its optimum temperature range, while minimally affecting pressure and flow of the fluid. Importantly, the valve must be inexpensive to produce on a commercial scale. The desired outcome of the research and development of the EPA's hybrid system is to commercialize the system at a price low enough that fuel savings will offset its cost after only a few years. To this end, our valve design must uphold technical specifications of the hybrid system and the overall energy and cost savings of the EPA's hydraulic hybrid system.

Series Hydraulic Hybrids

Although we are only designing one component of the hydraulic hybrid system it is important to understand the underlying technology behind the process. Hydraulic hybrid technology converts energy from braking that is normally dissipated as thermal energy in a process known as regenerative braking. There are currently two set-ups in place, parallel and series hybrids. In a parallel configuration the hybrid components are connected to a traditional transmission and driveshaft. This configuration uses the engine to power the vehicle when the hydraulic system is not in use, when the hydraulic system is in use the engine is not utilized. In a series configuration there is no driveshaft and it is the energy stored in the hybrid system that directly powers the wheels. The engine is used to run a pump in the hydraulic system and the hydraulic motor is always utilized to power the vehicle. Because the EPA is using the latter system, we will discuss the series configuration in greater detail.

Figure 1: Schematic of Series Hydraulic Hybrid System [5]



The main components of the system are shown in Figure 1, with two accumulators, one for high pressure and one for low pressure. The main principal behind regenerative braking is that when the vehicle slows or stops, the rotational energy of the wheels is used to pump the hydraulic fluid from a low pressure accumulator to a high pressure accumulator. The high pressure accumulator is able to store this energy by compressing nitrogen gas as the hydraulic fluid is pumped in. There are three pumps necessary in this design, the rear drive motor and pump (B) that acts as a motor and is responsible for converting the pressurized hydraulic fluid into rotating power for the wheels. There is another rear drive pump and motor (A) that acts as a pump (versus the other that acts as a motor), this is the pump responsible for pumping hydraulic fluid into the high pressure accumulator during the braking process. The final pump and motor (C) is directly connected to the engine and is able to create more high pressure fluid in the event that there has not been sufficient braking to generate the amount to pressurized fluid needed to drive the vehicle [9].

We will be creating a thermostatic control valve that will automatically respond to the temperature of the hydraulic fluid coming from the filter and route a certain portion to the cooling radiator if necessary, while allowing the rest of the fluid to return to pump/motor C.

2. Project Specifications

2.1 Customer Requirements and Engineering Specifications

Our customers for this design are Dr. Moskalik and the Environmental Protection Agency; as a result, our customer's requirements in many cases were actual hard engineering specifications. When we met with our customer parameters were strictly defined for the function of our final product. We broke down these requirements into nine categories which are listed and ranked in order of importance in Table 1 below.

Safety is a completely non-negotiable requirement; thus, a safety factor of 4 on all components in the hydraulic system is required. This industry standard for pressure rating is very rigid, and is an important consideration for testing our prototype.

Cost effectiveness of our final design is imperative as our valve is meant for mass production. One of the main purposes of this design is to help bring down the overall cost of the hydraulic system in order to make hydraulic hybrid delivery trucks price competitive with conventional trucks. Currently valves of this type cost hundreds of dollars, so ideally we will create an actuation mechanism that costs under $10.

The valve must withstand the lifetime of the hydraulic system and as a result will go through many cycles per day as the vehicle heats up and cools down. This means that our valve design will have to withstand at least 100,000 cycles.

In the pressurized hydraulic system, large pressure drops are unacceptable as they will substantially decrease the efficiency of energy transmission. Our sponsor requires that the pressure drop in the radiator dominate the pressure losses of the system. At temperatures and flow rates (~30 gpm) the radiator can have pressure losses of up to 45 psi. However, valves that are currently on the market that we are benchmarking our project against have far lower pressure than this (in the single digits). We will stay competitive with these products by using the specification of a pressure drop of less than 10 psi.

Table 1: Ranking of customer requirements and engineering specifications [8]

Rank	Customer Requirement	Engineering Specification
1	Safety	Safety Factor of 4 on all parts
2	Cost	≤$10 (mass production)
3	Durability	>10^5 cycles
4	Low Pressure Drop	<10 psi
5	Operating Temperature Range	170^{o}F ± 10^{o}F (strict bounds)
6	Material Compatibility	Materials cannot react with hydraulic operating fluid
	Housing Compatibility	1” inner diameter lines
7	Valve Seal Efficiency	< 5% leakage between lines
	Housing Seal Efficiency	0% Leakage
8	Response Time	<10 minutes
9	Adjustable Temperature Range	-60% adjustability down from maximum temperature range -Not necessary for production design

The operating temperature range that our customer requires has very strict upper and lower bounds. The upper bound is 180^{o}F due to the negative thermal effects (system components begin to melt) on the system when the operating fluid is above this temperature. The lower bound of 160^{o}F was determined as the system's efficiency depends on the operating fluid being maintained above this temperature.

Our valve mechanism has to be compatible with the current operating fluid in the system (Mobile 1 Synthetic ATF), as there were concerns that the thermally expansive wax in a previously tried mechanism may have reacted with the hydraulic fluid causing undesired behavior. Also, the hydraulic lines that are currently used in the hydraulic system have a 1" inner diameter, so in order to avoid any unwanted effects due to an expansion or contraction within the valve we will have to design our channels within our valve casing with a 1" inner diameter. This issue was particularly brought to light after creating a physical model of our alpha design in which the housing florist foam and the inner components were made of modeling clay. When the modeling clay dried it actually contracted and did not fit into place as we had originally anticipated. Therefore we have been particularly rigorous in our material selection of the final design so as to avoid this problem.

Another customer requirement is that the seal on the casing have absolutely 0% leakage. In other words there must be a perfect seal in the casing with respect to the outside. However, internally the valve seal does not have to be 100% efficient, as small amounts of leakage are tolerable and will not have a noticeable negative effect on the efficiency of the system. Our sponsor has asked for the leakage to be kept under 5% between fluid streams for all flow rates when fully open or closed. This means that when the valve is diverting all flow to the radiator, a 5% leakage is allowed to the line routing fluid back into the system and when the valve is diverting all flow to the system 5% leakage back into the radiator line is acceptable.

Since the valve has to respond to temperature fluctuations in the hydraulic fluid, the response time is important. However, due to the relatively long time response of the operating fluid's temperature, our sponsor has asked us to design the temporal response time of the valve to be less than 10 minutes.

We have discussed adding an adjustable temperature range as a feature on our deliverable prototype to aid in our testing as well as an aid for the EPA's testing after completion of the project. This is not a necessary feature as it will ultimately be taken out of the final design for mass production, but it would be appreciated by our sponsor. As the project progressed we determined that this feature would not be necessary for testing, therefore we omitted it from our design.

2.2 QFD

The Quality Function Deployment (QFD) is a helpful tool in relating customer requirements and engineering specifications and determining how they affect one another. It also compares the products being used as benchmarks to the important design considerations. Our QFD can be found in Appendix F.

The left side of the QFD lists the customer requirements, and weight of importance is assigned to each. We determined the importance of each requirement and ranked them based upon our initial meeting with our sponsor Dr. Moskalik. We then showed Dr. Moskalik our rankings and asked for further input to verify that we were providing the customer with what is most important. An in-depth discussion of each requirement is located in Section 2.1. Since our customer is an engineer, and the ultimate goal of the valve is for it operate automatically, based on temperature of the hydraulic fluid, many of the customer requirements are similar to the engineering specifications.

The Quality Characteristics section of the QFD diagram lists specific functional requirements of our valve. These characteristics were also determined based upon our meeting with Dr. Moskalik. He was very clear in specifying which functions the valve must perform. The quantitative values and respective units for each function are listed at the bottom of the QFD diagram. Dr. Moskalik provided us with these target ranges since they are heavily influenced by the operating specifications of the hydraulic hybrid vehicle (HHV) system. More detailed information about each engineering specification and its quantification is located in section 2.1.

The center portion of the QFD diagram specifies the strength of relationship between each customer requirement and engineering specification. We determined these relationships by considering the effect of each customer requirement on the function of the valve. The relationships are determined to be either strong, weak or moderate. Visualizing the relationships between customer requirements and engineering specifications is particularly important to be sure the function of the valve upholds the priorities of customer.

The triangular matrix at the top of the QFD diagram depicts the correlation each engineering specification has on each of the others. These correlations are designated as strong positive correlation, positive correlation, negative correlation, or strong negative correlation. These correlations are important in considering how changing one functional parameter may affect the rest of the valve's functions. Considering these correlations and our quantified target values for each engineering specification is useful in optimizing the function of the valve.

The right side of the QFD diagram presents the important products, against which we are benchmarking our design. We assigned a value of how well they fulfill each of our customer requirements. This allows us to quickly visualize the strengths and weaknesses of each product, and consider how our design should compare. This is particularly important since the EPA has tried using some of the products in this section, and they were not satisfied with them. We must be sure to improve upon the weaknesses that prompted them have us design a valve for them.

3. Concept Generation

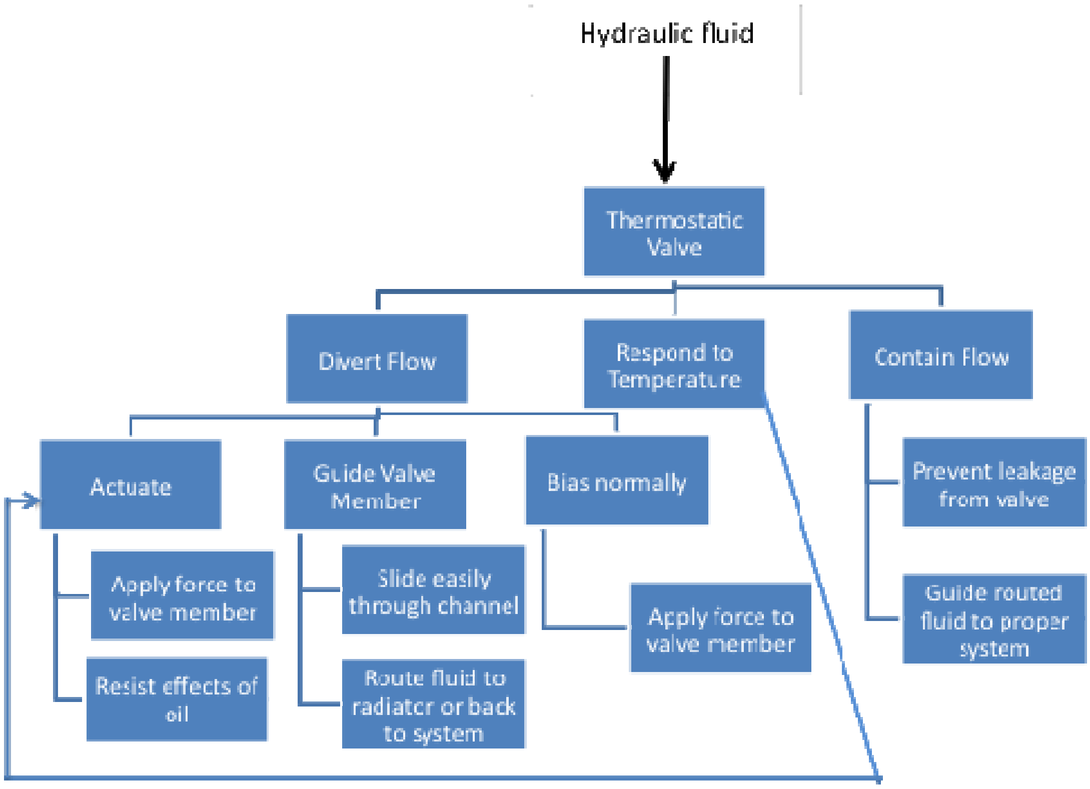
In order to efficiently generate ideas for each concept, we first performed a full functional decomposition. This allowed us to visualize the actual functions that our mechanism will have to perform, as well as how these functions will interact with one another. This further enabled us to ideate different variations on each design, and specific subsystems of the design that are common among our concepts.

The process of functional decomposition results in a breakdown of each specific function the thermostatic valve must perform. This is a useful representation of what specifically must be done, but not what will complete the functions. After determining the functions we organized them in a diagram that shows their progression as hydraulic fluid flows through the valve. This diagram is located in
Figure 2.

The thermostatic valve must divert flow, respond to temperature and contain flow. These are the first functions depicted in the functional decomposition (Figure 2). The input and output of the valve is hydraulic fluid from the HHV system. This hydraulic fluid will arrive at the valve with a temperature resulting from its use in the system and flow through the radiator.

The diverting flow function has three sub-functions, which must be performed to properly divert the flow of the hydraulic fluid. To divert the flow, something must actuate a valve member and influence the direction of the fluid's flow. This sub-function is actuation. Actuation involves the functions of applying a force to the valve member to cause it to move and resisting any harmful effects of the hydraulic fluid. While moving due to actuation, the valve member must be guided to ensure that it moves in the intended direction. Guidance is thus, a sub-function of diverting flow. Proper guidance of the valve member entails sliding easily through the channel (result of lubrication) and routing the fluid either to the radiator or back to the hydraulic system. Finally, once the fluid has been cooled, something must move the valve member back to its original position. This sub-function is considered normally biasing actuation and involves applying a force to the valve member so that it moves to its normal position.

Figure 2: Functional Decomposition- identification of each function of the valve



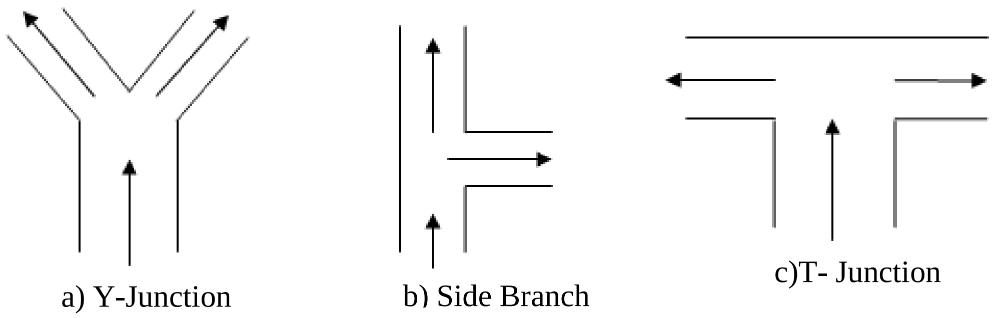
The valve must respond to the temperature of the hydraulic fluid. Although responding to temperature is on its own a function of the valve, it is tied to actuation of the valve. For this reason, on the functional decomposition diagram, we drew a line attaching respond to temperature and actuate. By recognizing the connection of these separate functions, one can easily realize that both functions may be affected by the same component of the valve.

Finally the valve must contain the flow. Containing the flow is important so that when the hydraulic fluid is diverted, it will continue to flow where intended. Containing flow is performed by preventing any leakage of hydraulic fluid from the housing and guiding the fluid that has been routed by the valve member to its intended part of the HHV system. Any leakage of fluid outside the system through our valve is unacceptable, and the hydraulic fluid must be able to flow as easily as possible to the radiator or the rest of the system.

Analyzing each of these functions individually allowed us to begin brainstorming about what could ensure that each function is properly accomplished. This led to our identification of specific subsystems of the valve and generation of concepts for the individual subsystems, as well as the entire valve assembly. We are confident that we have thoroughly analyzed each function required by our valve to fulfill the customer requirements and engineering specifications.

After producing this functional decomposition, we first identified the three most likely junction geometries, which were a Y-configuration, a T-configuration, and a side-branch configuration, (Figure 3). We did come up with several designs outside of these, but these are the most manufacturable types of junctions. After identifying these geometries we moved on to an actual brainstorming session where we could take turns (in a free form manner) to draw and explain either full design concepts, or ideas for separate subsystems. These full designs were then broken down into their subsystems and we "mixed and matched" all of these separate systems that could be used in conjunction with one another into new design ideas. We ended up with many individual designs (detailed in Appendix H) and four subsystems; actuator, normally biasing actuator, valve member and junction geometry. Eventually we were able to rank our designs and subsystem designs based on a Pugh chart breakdown; however that will be discussed further in Section 5.

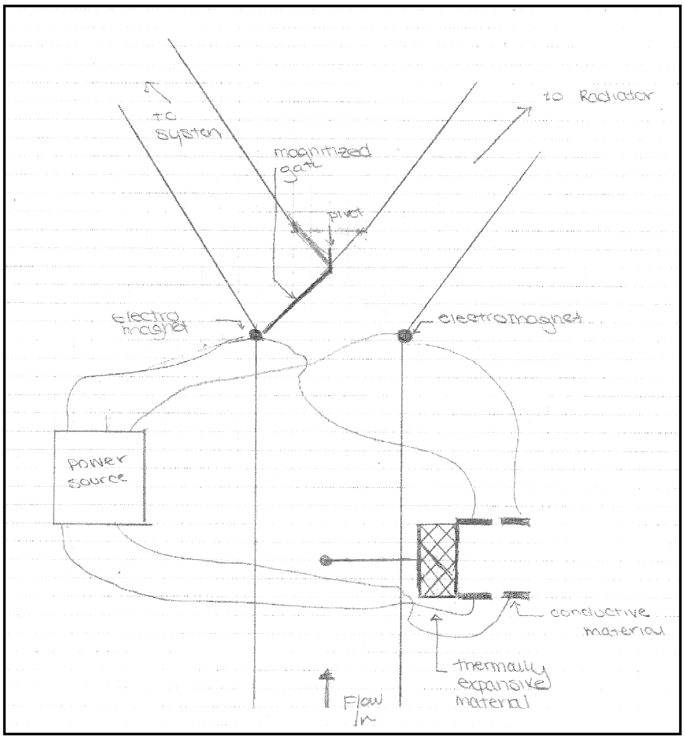
Figure 3: Three main basic junction geometries



We decided to organize our concept designs in terms of blockage type, or rather in terms of what will actually redirect the flow of hydraulic fluid. Therefore, our final categories for our designs are gate, plug, translating channel and inner pipe. Some of our designs actually fall into more than one of these categories; however breaking the designs up in this manner allowed us enough stratification without much confusion. Also, categorizing our designs in this manner lumps together many of the design aspects which have the potential to be problematic. Therefore, the required mechanical troubleshooting for each design in a category will be similar.

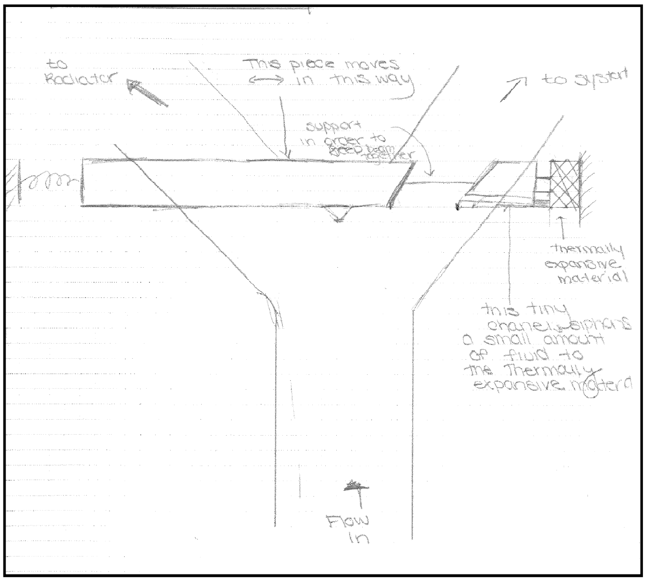
Each flow redirection mechanism had several different types of actuation. One of our gate designs utilized a novel type of electronic actuation mechanism using electromagnets to move the gate from one position to another. In this case we used the Y-valve configuration, with a permanent magnet attached to the free end and an un-activated electromagnet on either side. The electromagnets would be activated via a thermally expansive material that would complete one circuit (activating the first magnet) then, as it expanded, would bridge another circuit activating the other magnet and simultaneously deactivating the original magnet. The gate is free to swing on a hinge and depending on which electromagnet is activated shown in **Error! Reference source not found.**.

Figure 4: Magnetic actuation concept design



One of our concepts consisted of a Y-shaped channel with a translating cylinder that is actuated by an expanding wax thermal actuator. As seen in Figure 5 the cylinder is located in the two out-flows of the Y-split and has a channel machined out of it. There are two places where the channel goes through to the other side of the cylinder, allowing fluid to flow through it. The cylinder is positioned so that when the flow is below 160˚F, one of the openings is blocked by the housing and the other will divert all flow back through the system. As the fluid heats up above 160˚F, the cylinder is pushed by the actuator so that both outlets can pass fluid through them until eventually, by 180˚F, the first opening is blocked, and only the second opening allows fluid to flow. The right side of the cylinder has two small openings that allows the hydraulic fluid to flow around the thermal actuator so that it can accurately respond to the current temperature of the fluid. The position of the actuator and the method of surrounding the actuator with the hydraulic fluid are disadvantages of this concept.

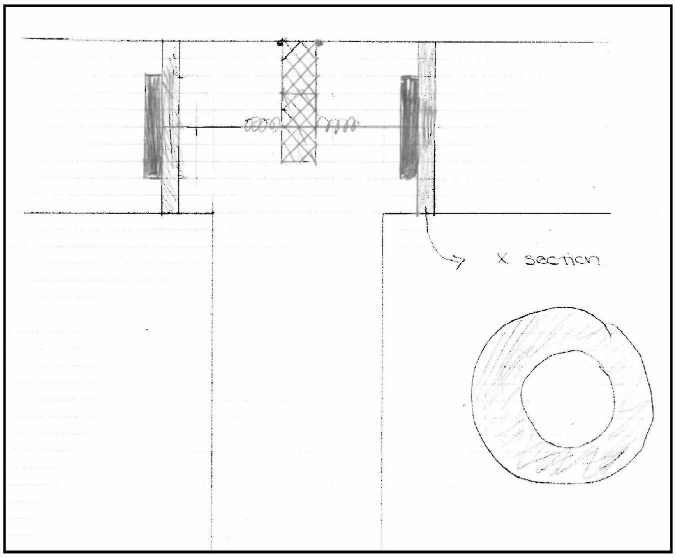
Figure 5: Y-channel concept design



One example of a plug style concept was the “double plug” design which employs two plug stoppage mechanisms, essentially in reverse position in relation to one another. These plugs have separate actuation mechanisms comprised of thermally expansive material. They would both be held in place by a normalizing spring as seen in

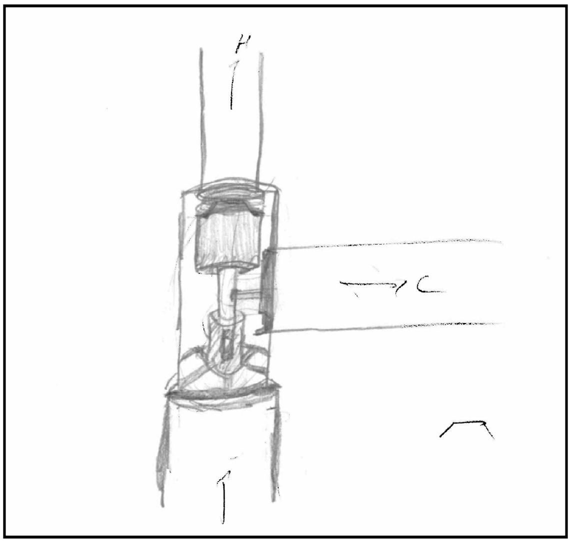
Figure 6.

Figure 6: Double plug concept design



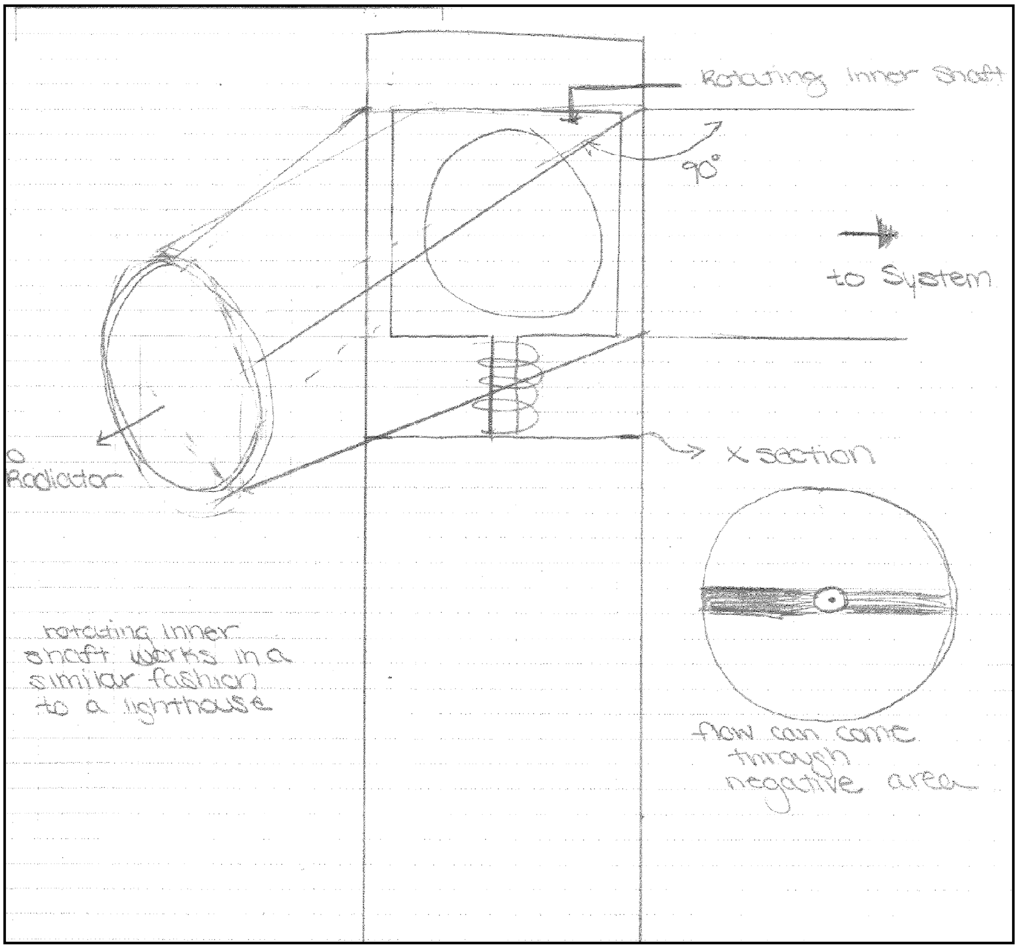
Some concepts that we had actually make use of two different blockage mechanisms, such as our plug and gate design. This design makes use of the plugs described before, but also has a gate mechanism which opens the passage to the radiator as the recirculating opening is closed via plug. This design is discussed in detail in Section 5.

Figure 7: Plug and gate concept design



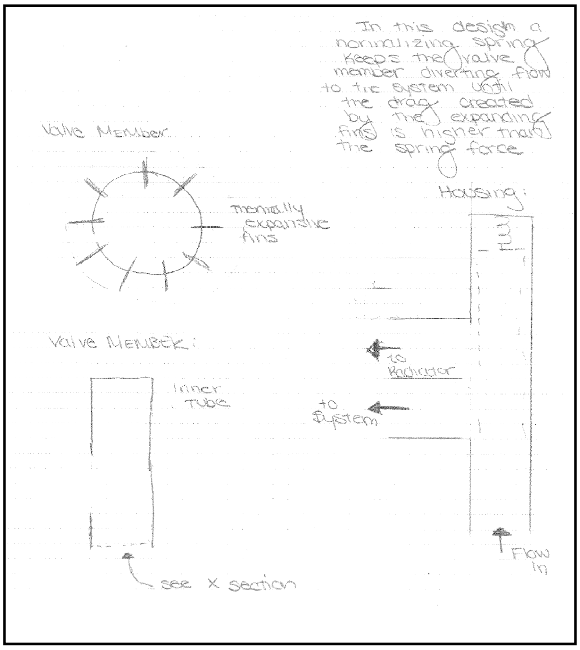
One concept that falls into the inner pipe category is our lighthouse design. This design is comprised of a rotational actuator, normalizing spring and a movable inner pipe as seen in Figure 8. The inner pipe actuates in a rotational fashion changing blockage of the radiator stream to blockage of the recirculating stream, simultaneously opening the port to the radiator stream.

Figure 8: Lighthouse concept design



Other concepts use ideas such as the plug or gate, but have radically unconventional actuators, our drag fins design is one of these. This mechanism uses temperature actuated fins which are pushed into the flow and are moved forward by the linear momentum of the operating fluid. They then push a plug into the path of the flow, thereby redirecting more to the cooler.

Figure 9: Drag fins design



These are only several examples of our full range of concept generation. A full listing of our concept designs, as well as a categorization of these concept designs can be found in Appendix H.

4. Concept Selection Process

In order to determine our best concept design, we first utilized a Pugh chart with all of our concept designs listed with our engineering specifications.

We began Pugh chart analysis immediately following our first brainstorming session in order to facilitate our concept generation. This guided our concept generation in a more constructive direction as it helped to point out design flaws that may not be obvious. As a result of this, our original Pugh Chart analysis contains some concept designs that were realized as unfeasible. Although some of the designs themselves contained inherent flaws, they still had valuable subsystem components that were able to be integrated into new, feasible concepts.

Each design and its subsystems were weighed against our customer requirements in a Pugh Chart analysis in order to assist in our concept generation and ultimately, in choosing our alpha design. We assigned different weights to each customer requirement (1-3) and rating how well each design meets that requirement (-2, -1, 0, 1, 2), each design was given a total value by multiplying each rating with the weight and summing.

Design Criteria	Weight	E.1 Parallel Pipes with Thermal Actuator	E.2 Parallel Pipes with Electromagnet	C.2 Linear Actuator	D.1 Y Valve with Electromag net	K. T Valve with Shape Memory Alloy Gates
Safety	**3**	2	1	1	1	1
Cost	**3**	-1	-2	-2	-1	1
Operating Temperature Range	**3**	2	2	2	2	2
Low Pressure Drop Across	**2**	2	2	2	2	2

Table 2 shows the breakdown of our first 16 designs.

Table 2: Pugh chart analysis for first 16 designs- see Appendix H for images of designs (location of each diagram in Appendix G listed above design name)

Durability	**2**	2	2	2	2	2
Seal Efficiency	**2**	2	2	2	2	1
Response Time	**2**	1	2	2	2	1
Compatibility	**1**	1	1	1	1	1
Adjustability	**1**	1	1	1	1	1
TOTAL		**26**	**22**	**22**	**25**	**27**

Design Criteria	Weight	C.3 Gate with Electromagnet	C.1 Gate with Thermal Actuator	H Lighthouse	J Drag Ring	I.1 Translating Sleeve with Linear Actuator
Cost	**3**	-1	1	1	-1	-1
Low Pressure Drop Across	**2**	2	2	2	2	2
Seal Efficiency	**2**	2	1	2	0	1
Compatibility	**1**	1	1	1	1	1
TOTAL		**25**	**30**	**30**	**17**	**21**

Design Criteria	Weight	L Half Moon	A.1 Double plug with shared actuator	F Capillary	B.1 Plug and Translating Gate	G. Sliding Channel
Safety	**3**	1	2	1	2	1
Cost	**3**	0	1	0	0	-1
Operating \ Temperature Range	**3**	2	2	2	2	2
Low Pressure Drop Across	**2**	2	2	2	2	2
Durability	**2**	1	1	1	2	1
Seal Efficiency	**2**	1	2	1	2	1
Response Time	**2**	1	1	1	1	1
Compatibility	**1**	1	1	1	1	1
Adjustability	**1**	1	1	1	1	1
TOTAL		**22**	**30**	**22**	**29**	**19**

Three designs tied for the most points so no clear alpha design was obtained. Pugh Charts of the individual subsystems incorporated in each design were then created. The subsystems covered by these Pugh Charts were the actuator, normally biasing actuator, valve member, and junction geometry. Appendix G (a-d) show that every idea for a subsystem was weighed against the relevant design criteria and it identifies the best result for each subsystem

Applying Pugh Chart analysis to each subsystem assisted in concept generation. By simply taking parts from each subsystem and putting them together, approximately 20 new concept designs were created. Similarly, integration of the highest scoring subsystems allowed us to generate the most appropriate alpha design. Appendix G (a) shows Pugh Chart analysis of the actuator, from this we determined that a thermally expansive material (particularly wax) would be the most feasible solution. The Pugh Chart of the normally biasing actuator, Appendix G (b), made us heavily consider designs incorporating normally biasing springs. Appendix G (c, d) shows analysis of the valve member and junction geometry, which did not bring conclusive results but allowed us to eliminate a concept from our original pool of concepts. Ultimately, Pugh chart analysis allowed us an effective tool to compare each of our designs against one another in a quantitative manner. This quantification was then used to choose the best subsystems available, integrate them and generate an alpha design.

Our top five concepts based on Pugh Chart Analysis, in no particular order of rank, were the double plug with shared actuator (Appendix H (A.1)), shape memory alloy gates (K), the lighthouse (H), the gate with thermal actuator (C.1) and the plug and translating gate (B.1). The double plug with shared actuator is a great idea while on paper; however, in reality, the idea simply wouldn't work. While it would meet all of the engineering requirements, after further analysis and modeling, the design is not capable of performing the functions we desire without excessive machining of the housing which would lead to high costs and make it difficult to mass produce cost effectively. Furthermore, the idea itself seemed troublesome as finding the placement for a normally biasing actuator that would effectively return the valve to its normal position seemed impossible. While it was a good design on paper as it would be safe, effectively work within the operating temperature range, maintain leakage to be under 5% between streams as well as responding promptly and have a low pressure drop across the valve; the possibly flawed functionality as well as high machining costs made us stray from choosing it as our alpha design.

The shape memory alloy gates (Appendix H (K)) are great if cost and safety weren't our main concerns. This shape memory alloy is an expensive piece of metal which responds to temperature changes and it would be difficult to create a final design that would ultimately cost less than \$10. Similarly, since it is a relatively new idea and there isn't expansive information on it, the safety and durability come into question. While it is metal, we don't absolutely know its ability to withstand the pressures and forces within the system. Also the durability comes into question as there is no recorded tests involving this shape memory alloy being immersed in hydraulic fluid and lasting millions of cycles; thus, the overall cost and concerns with safety and durability made us sway from this design. There are positives to this design though if budget wasn't a main concern such as, if it was to function as shape memory alloy should, it would effectively reroute flow at the operating temperature with proper response time and seal efficiency while maintaining a low pressure drop across the valve.

The lighthouse concept (Appendix H (H)) made use of a rotating gate which would actuate at the lower bound of our temperature range, rotating and blocking off the passage back to the system and opening the line to the radiator. While this would prove to be a safe design, as well as one that would work under the operating temperature range, have a low pressure drop across the valve, and have less than 5% leakage in between streams, the overall cost and durability is concerning. Finding an actuator that would effectively rotate these gates at the correct temperature and a normally biasing actuator to return them to their normal position would prove to be costly and ultimately prevent us from creating a valve that will cost less than \$10 during mass production. Also the chance that said actuator and normally biasing actuator will work for 10^5 cycles while immersed in hydraulic oil and undergoing fluctuating pressures and forces isn't 100%, which it needs to be in order for us to make it our alpha design; thus, due to durability and cost issues we will not move forward with this design.

The gate with thermal actuator concept (Appendix H (C.1)) meets almost every engineering and customer requirement. This design would prove to be safe, durable, operate within the effective temperature range, and have a low pressure drop across it. However, this design also makes use of a thermally actuated wax that must be set outside of the housing. Any of our concepts without completely internal parts would prove to be too costly as parts can't be as easily cast or machined. The valve member that moves into position and redirects flow would also be difficult to machine and prove too costly. Therefore, poor seal efficiency caused by having parts outside of the housing, as well as the fact that the spring and thermally expansive material must be outside of the housing would rule this design out as alpha design.

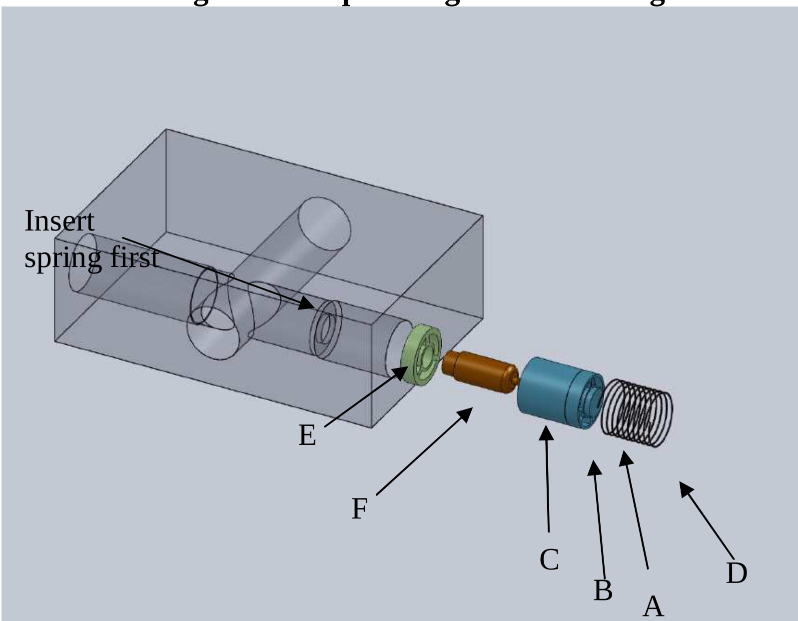
After consideration of every concept, the plug and translating gate (Appendix H (B.1)), proved to best match our engineering and customer requirements. This concept proves to meet safety standards, will be able to be mass produced for under $10, will work within the operating temperature range, have a low pressure drop across the valve, maintain less than 5% leakage between lines, respond quickly, and be durable for 10^5 cycles. Low cost will be achieved by casting some of the parts as well as using a thermal actuator and normally biasing spring which when purchased in mass quantity would cost next to nothing. Also, since we know the exact temperature range in which the thermally expansive material we choose will actuate we know it will effectively work within the operating temperature range of the system. The design has nearly no flaws except if the actuator were to fail and the line to the radiator was left blocked. In this situation the vehicle would overheat; however, since we created a cartridge-style design, the malfunctioning part alone could be easily replaced for minimal cost and effort. Contrary to that however, if the normally biasing actuator were to fail, the system would not over-heat as all fluid would simply be diverted to the radiator at all times. Also, since this thermally expansive actuator and normally biasing spring have been tested in numerous conditions similar to our operating conditions, and the materials we are choosing are for the housing and other parts are robust materials, there would be no problems with durability. Since this concept is the only one which doesn't have a noticeable disadvantage we determined that it was the best candidate for our alpha design.

The results of the Pugh chart were utilized to narrow down our concept choices to several concepts that would best satisfy our customer requirements. We had to further narrow down the choices after the Pugh chart, in order to do this; we discussed each design in more detail. This open forum discussion allowed us to identify the feasibility of each design, as satisfaction of our customer's requirements is not the only consideration in our design process. Availability of commercial actuators was an important part of this analysis. We also had the option of manufacturing our own actuation mechanism, however we believe that our resources would be better allocated by making use of a commercially available actuator as opposed to designing our own thermally expansive material and implementing it into an actuator. Although this removed some flexibility from our overall design space, it will reduce our cost and time spent in this area due to the ready availability of commercial linear thermal actuation mechanisms.

5. Alpha Design

Our chosen design is hybrid of the plug and gate classifications as seen below in Figure 10. This mechanism is a hybrid of two categories of concept having a plug that serves both as a plug and a gate (A, B) rigidly connected to a thermally expansive wax actuator (C). The normalizing spring element (D) is positioned between the top of the plug and the seat which the plug will fill when fully actuated (E), which is a conventional spring designed to return the mechanism to its original position as the mechanism cools down and the thermal actuator contracts. The entire mechanism is attached via female threading in the plug/gate itself, as well as in the inner hole of the spoked retaining washer (F). Male threading will be present on the connecting rod as well as the outside of the retaining washer. Therefore, when completely assembled, the retaining washer can be threaded into the housing to hold the mechanism in place. As can be in Figure 10, one major advantage of this design is that the plug and gate are machined out of one piece allowing flow between the two via its spoked design.

Figure 10: Alpha Plug and Gate design



The custom thermal actuator that we are currently planning on using has a maximum stroke of 1inch which will allow the valve to completely actuate out of the way of the outgoing radiator line. If this actuator is not available however, one with a smaller stroke could be used in which case the radiator line will be partially blocked when at full stroke, however, this should only induce a localized velocity variation and pressure loss, not severely affecting the overall pressure loss of the valve. We have contacted a supplier that can custom fabricate an actuator with a 1inch stroke, however we are currently waiting on a price quote from them.

These actuators can be calibrated to actuate at any temperature range (30^{o} F – 300^{o} F), we can opt for a slow opening response, or a more discrete, step-function-style response. will allow us to choose our mechanism response time, so satisfaction of the requirement actuation happen over the course of several minutes can easily be met. This means that valve member is in its initial unheated position, the gate will fully eclipse the radiator line directing all flow to the re-circulating line as seen in Error! Reference source not found.**a. When the mechanism is fully actuated, the plug will ideally eclipse the re-circulation line and the gate will fully withdraw from the radiator line as seen in**

Figure 11b.

Figure 11: Flow routes in alpha design in:

a) Normal Position **b) Actuated Position**

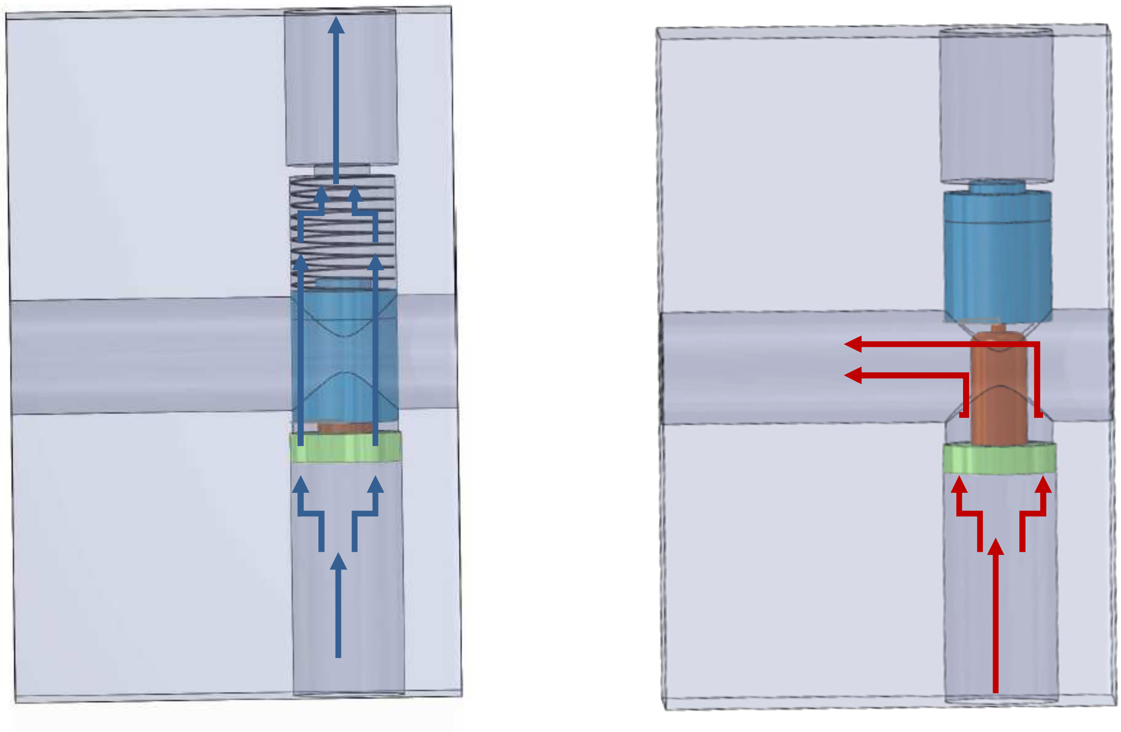
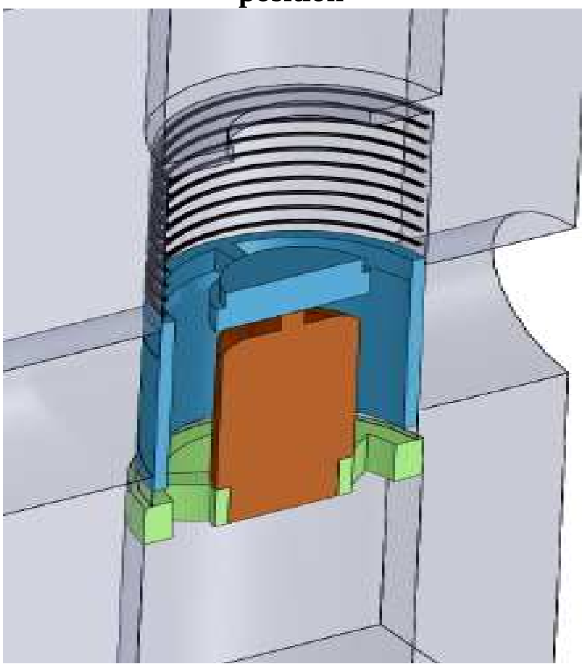


Figure 12 shows a close up cross section of the valve in normal position. Once assembled the part is actually one rigid piece (apart from the movement from actuation) meaning that, when commercially available, the lifetime of the housing is not dependent on that of the valve member and actuation system, as they may be easily replaced and the housing can be utilized for a new member and actuation system. This "cartridge" style design makes assembly and replacement far simpler than a design that would require the componential assembly of a member and actuation system. This will increase the lifetime of the system part, thereby reducing the cost of the design.

Since the plug and gate are rigidly joined, there is no way that they could possible block both passages at the same time. We created a physical model to test this feature, and found that it worked appropriately. This safety feature is an important consideration for us, as our first and foremost customer requirement was safety of the product. Many of our original concepts did not have both blockage attachments moved by the same actuator; as a result we deemed that both actuators could not be assumed to behave in the exact same manner, therefore allowing for the possibility that both passages could be closed at the same time causing catastrophic failure of the part.

Figure 12: Close up cross section of valve member and actuation system in normal position



Tight tolerances on the plug and gate will be required in order to minimize leakage between streams when in the fully open and closed positions. Therefore we plan to machine the plug/gate to 0.001inch tolerances on the outside edge and plug head. This will keep leakage between the two outgoing streams to a minimum and well within the 5% cross-stream leakage customer requirement. The housing will be designed with standard threaded 1inch (inner diameter) line ports (the same as those used by our sponsor), which will be the only openings within the housing, thereby fulfilling the 0% out of housing leakage required by our sponsor.

Material compatibility will not be an issue with this concept, as the design of the thermal actuator will not allow the operating fluid to penetrate into the wax; therefore no reaction between the two can occur. Also, all other components are to be made from materials commonly used in hydraulic applications (steel, aluminum, brass). So, no reactive compatibility is expected from these materials.

The spoked design of the plug/gate, as well as that of the retaining washer will cause some pressure loss. Also the redirection of flow at a right angle will most likely cause some pressure loss, especially if a custom actuator is not available to us, only allowing the radiator line to open half-way. However, we anticipate that these losses will not be too great, since we are already operating in a turbulent flow regime.

We are looking in to the possibility of modifying this design to have an adjustable temperature range. We will only attempt to include adjustability if our testing processes require this though. An adjustable temperature range would only be useful for testing and will ultimately be eliminated for the final manufacturing design plans if incorporated in our prototype.

As has been mentioned before, we created a mock-up prototype of our alpha design from florist foam and modeling clay. This was intended to demonstrate the function of our alpha design and to illuminate and issues that had been before unseen. As a result of our mock-up we were able to determine that our valve could re-direct flow as desired as well as return to a normal position. However, due to contractions in the clay during curing, all parts did not fit together as desired. Therefore we will be sure to take the thermal expansion coefficients of each material used into account. Also, static friction was foreseen to be an issue and will be discussed later. Our mockup showed has provided a basis for justification of our design.

Therefore our plug and gate style alpha design is what we have determined will best suit the needs of our sponsors. It is has built-in safety features, will be low cost, is easy to verify, and should successfully redirect operating fluid flow as desired.

6. Parameter Analysis

The field that is most applicable to our design in terms of engineering analysis is fluid mechanics, since we are designing a part for using in a hydraulic system. We also applied advanced knowledge of materials science to optimize the geometry of our valve. Detailed knowledge of manufacturing processes was also required as our final design requires cheap, simple manufacturability and assembly. We are confident that our level of analysis is adequate to specify all of our parameters and optimize our design to the utmost degree.

6.1 Fluid Mechanics Analysis

In order to gain an understanding of the order of magnitude head loss that our valve would experience in a worst case scenario during operation we modeled each obstruction to fluid flow as an abrupt expansion or contraction. This Matlab model can be found in Appendix S. This is not entirely valid due to the vena contracta experienced at each contraction which will change flow characteristics. These changes did not allow for flow eddy currents and other flow characteristics experienced downstream of the interface, which requires velocity readings for a fully developed flow field after the obstruction. We found our absolute worst case scenario to be on the order of 10s of psi, which is an unrealistically high pressure loss through our valve for the geometry present in the flow path. Current valves on the market with significantly lower head loss than this require at least one or two 90^{o} changes in fluid direction, as well as significant contractions and change in flow path geometry. Therefore we can reasonably assume that our head loss through the valve will be below 10 psi, we also plan to rigorously test this head loss using methods described in Section 11.

Major losses due to friction have been neglected do to the relatively short length of our valve in relation to flow diameter. Minor losses due to flow constriction and expansion will have a dominating effect in relation to our valve. The energy required to accelerate or decelerate a fluid at these interfaces will be much larger that the energy lost due to friction.

Since anything short of a full blown computational fluid dynamics analysis will not garner valid results we have designed our valve using engineering logic. In order to reduce our head loss coefficient, we have operated on the principal that the smaller the change in cross sectional area with respect to flow direction, the smaller the pressure loss across the valve. In order to minimize flow obstruction, we have performed rigorous stress analysis on each part of the valve member to reduce them to their minimum dimensions still within our safety factor as described below.

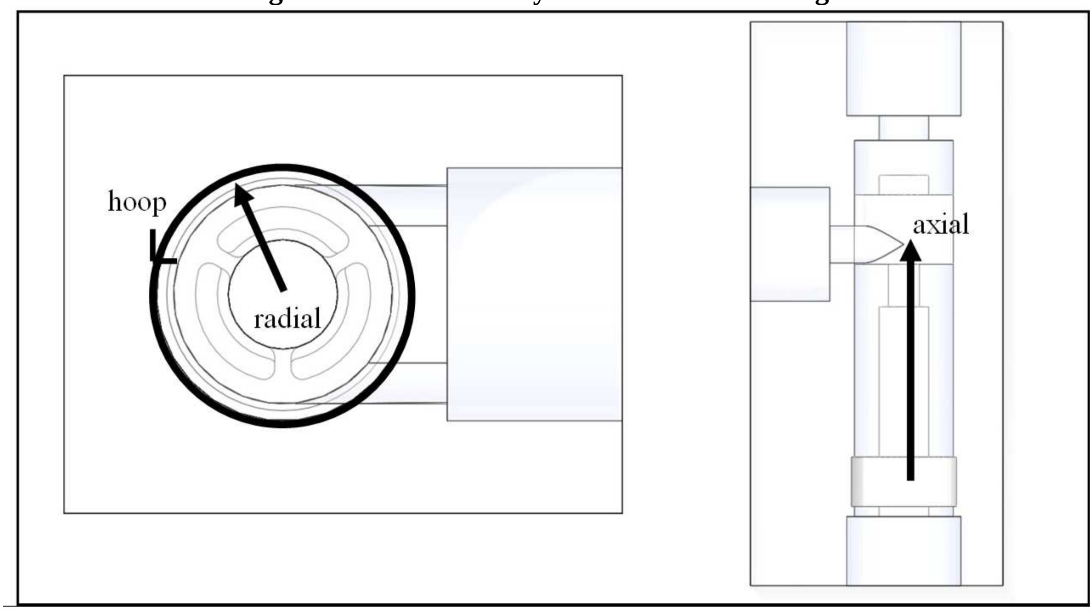
6.2 Sizing and Stress Analysis

When optimizing the sizing of our parts it was important to consider the sources of stress in our system and make sure not to exceed the yield strength of our selected material (with a safety factor of 4). Once all of the stresses were analyzed, we were able to use the information in tandem with machining limitations to size our parts. Each part will be discussed in detail, including the sources of stress, machining techniques and various other limiting factors.

The Housing

The housing was modeled as a pressure vessel in order to determine the minimum wall thickness possible. The inner diameter was set at a maximum of 1-5/32inches, in order to be compatible with the 1inch inner diameter lines that we will be using for testing (as well as what the EPA will be using). A customer requirement we have is that the system must be pressure rated to 200 psi, meaning that failure must not occur below 800 psi. Using Matlab, we modeled our system as a pressure vessel using equations 1, 2, and 3 [19]. The coordinate system that we utilized is shown in Figure 13.

Figure 13: Coordinate system shown on housing



$$\sigma_h = \frac{r_i^2 P}{r_o^2 - r_i^2}\left(1 + \frac{r_o^2}{r^2}\right) \quad (1$$

$$\sigma_r = \frac{r_i^2 P}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right) \quad (2$$

$$\sigma_a = \frac{r_i^2 P}{r_o^2 - r_i^2} \quad (3$$

Where, r_i is inner radius of pressure vessel (set at 37/64inches), r_o is the outer radius and r is the radial variable. P is the gauge pressure within the system and σ_h is hoop stress, σ_r is radial stress and σ_a is axial stress. These principal stresses were combined using Von Mises yield criterion [19] in order to determine the yield stress (σ_{yield}) shown in$\boldsymbol{\sigma yield}$=$\sqrt{\frac{1}{2}((\sigma_h - \sigma_r)^2 + (\sigma_h - \sigma_a)^2 + (\sigma_a - \sigma_r)^2)}$ (4.

$$\sigma_{yield} = \sqrt{\frac{1}{2}((\sigma_h - \sigma_r)^2 + (\sigma_h - \sigma_a)^2 + (\sigma_a - \sigma_r)^2)} \quad (4$$

A plot was generated showing stress versus outer radius of the housing, this can be found in Appendix I and the code is in Appendix L. In an effort to keep our costs low we would like to use as little material as possible and for this reason we were able to determine that our outer radius will be set at 0.75inches. At this radius, if pressure were to reach 800 psi, the maximum stress present within the walls of the housing would be 2.5 ksi, well under the yield strength of our 6061 aluminum (45 ksi). The stress is variable within the thickness of the wall and a plot of this can be found in Appendix K for reference.

The Spring

The spring did not require a stress analysis, but it was important to find a spring that would output a force no less than 10 lbs in the normal position and a force no greater than 40 lbs in the actuated position. These are the requirements of our prefabricated actuator. We were also interested in finding a spring that could produce these forces at very small lengths so that we could keep our overall design compact. Taking all of these factors into account, as well as the very important issue of low cost, we were able to find a cut to length spring with the specifications shown in Table 2.

Table 3: Specifications for McMaster Car part 9637K25

Cut Length	0.75 [in]	Spring Constant (k)	51	[lb/in]
Outer Diameter	0.96 [in]	Wire Diameter	0.08	[in]
Length in Normal Position	0.55 [in]	Force in Normal Position	10.2	[lb]
Length in Actuated Position	0.15 [in]	Force in Actuated Position	30.8	[lb]

Retaining Ring and Plug and Gate Forces

There are two forces acting on the plug and gate, as well as the retaining ring, the force from the fluid momentum as well as the force that the actuator and spring produce. $\boldsymbol{Ffm}$=$\boldsymbol{Q\rho(u_1 - u_2)}$ (5 [18] provides our method for determining the force that the fluid will exert on both the Plug and Gate and the retaining ring.

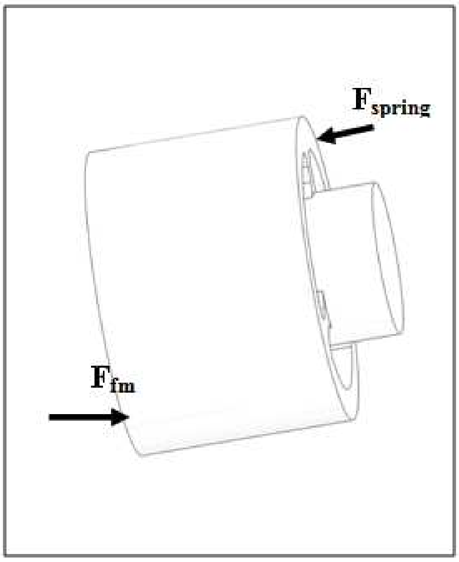
$$\boldsymbol{F_{fm} = Q\rho(u_1 - u_2)} \qquad \textbf{(5}$$

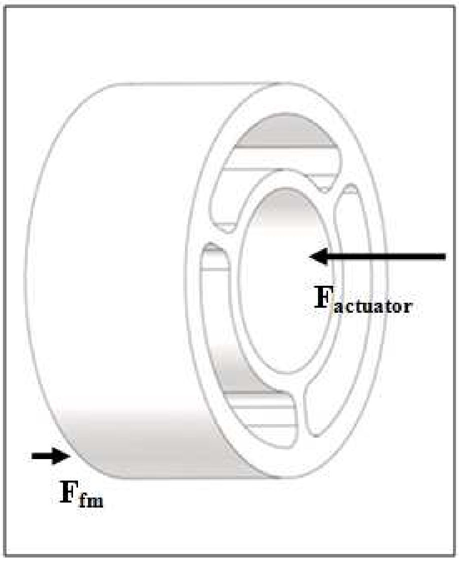
Where Q is the volumetric flow rate (a maximum of 35 gallons per minute in our system), ρ is the fluid density, the EPA uses Mobil 1 Synthetic Hydraulic Fluid which has a density of 9.66e-4 slugs per cubic inch at 15 °C [23]. Because density of fluids decrease with temperature (if at all) it is safe to assume that our working fluid density will not exceed this value. The variables u_i and u_2 refer to the starting and ending velocities of the fluid respectively. For our purposes the starting velocity is 171.6 in/sec and the ending velocity is zero. The results of our calculations show that the maximum force provided by the changing fluid momentum will be 1.8 pounds.

The second force pair that we examined was the force from the spring and actuator. The actuator piston will produce very large amounts of force during phase change, however so long as the plug and gate has the ability to translate, the only force that will result is the normal force from the spring. This can be seen in **Error! Reference source not found.**a, applying a resistive force to the plug and gate as it would be trying to move forward. **Error! Reference source not found.**b shows a free body diagram (FBD) for the retaining disk. The actuator force here is the result from the spring pushing back on the plug and gate which is rigidly connected to the piston of the actuator. If the retaining disk was not producing this force, the actuator would translate in the negative axial direction. This force dominates the system as it is much greater than the fluid momentum force. The maximum force that the spring will output is 30.8 lbs, however we chose to model all of our stress analysis using a total force of 60 lbs on each of the bodies in order to ensure that we will not have yielding failure as well as to allow for some future flexibility in the spring design.

Figure 14: Free Body Diagram of:

a) Plug and Gate **b) Retaining Disk**

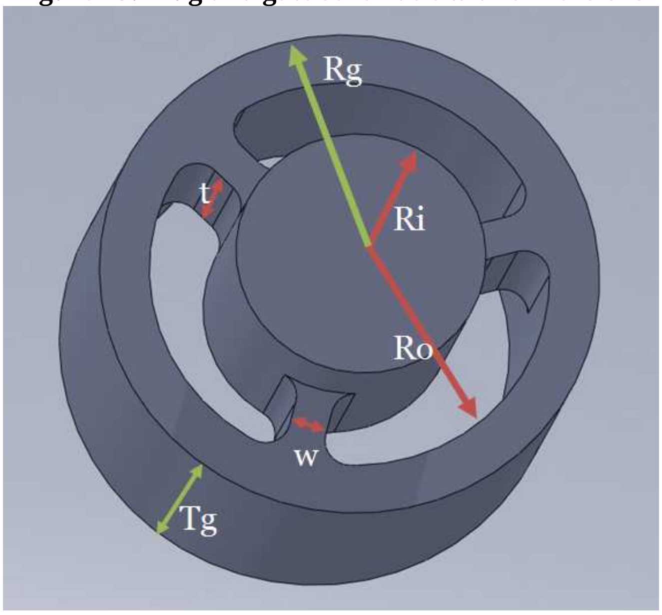




Plug and Gate Optimization

In order to optimize the dimensions of the Plug and Gate we first set those parameters which were dictated by outside factors and then modeled stresses, considered machining and flow volume to finalize the remaining dimensions. The outside diameter (R_g in Figure **15**) of this piece was predetermined to be 1inch based on our desire for compatibility with the EPA's current system. The length of the gate (T_g) was determined by the geometry of the housing and the length of the actuator stroke. Figure 15 shows these two set factors in green and the dimensions that we had freedom to set in red. These include: the radius of the plug (Ri), the thickness of the spokes (t), the outer radius of the spokes (Ro) and the width of the spokes (w). We decided to design with three spokes in order to eliminate the possibility for a moment, as would be the case with one or two, but also in an effort to keep machining steps to a minimum we wanted to keep the number of spokes as low as possible.

Figure 15: Plug and gate schematic with dimensions



The flow faces a contraction as it moves through arced cutouts but later in the flow path it faces another contraction as it moves through the seat in the housing into which the plug fits. For this reason a limiting factor we considered was trying to get the area of the arced slots as close as possible to the area of the plug in an effort to avoid having one be unnecessarily small. We utilized Matlab to model the stress in the thin circumference between Ro and Rg; this is also the depth of the gate. A plot of stress versus Ro can be found in Appendix M. We also considered keeping machining tolerances relatively low (to reduce cost in mass manufacture) as well as keeping the gate wall deep enough to allow for handling without possibility to internal collapse (in the manufacture and assembly stages). The result of these considerations was to set Ro at 0.4inches.

We next considered the inner radius (Ri). As mentioned previously, we wanted to ensure that neither the arced slot area nor the plug area was smaller than necessary but we had to also consider available tooling. We plan to machine the seat in the housing using through a simple drilling procedure. These factors led us to set Ri at 0.25inches.

The final two dimensions, width and thickness of the spokes, were modeled together in Matlab. There will be an axial stress on the face of the spokes due to the axial forces shown in Figure 14a and there will be a radial stress due to a bending moment. Appendix M contains a plot of stress versus thickness at multiple widths. This information was used to determine these final two dimensions. The Matlab code used to do perform this analysis can be found in. Table 4 contains the final dimensions of the Plug and Gate.

Table 4: Optimized dimensions of the Plug and Gate

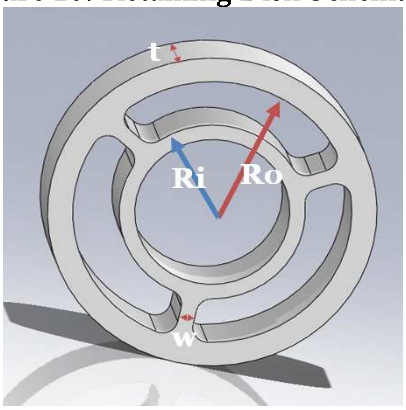
Radii		Spokes	
Ri (plug)	0.25 ± 0.005 [in]	Width	0.075 ± 0.005 [in]
Ro (arced slots)	0.40 ± 0.005 [in]	Thickness	0.375 ± 0.005 [in]

After finalizing these dimensions we created an engineering drawing of our part and began to more deeply consider our methods of manufacture. We had originally intended to mill out the arced slots, but then realized that the shaft of the end mill would make contact with the plug. We considered using wire EDM to cut them out; however the wire EDM in the Wu Manufacturing Center is currently unavailable. At this point we had to consider drilling holes instead of milling the slots, which would cause us to lose flow area. We want to avoid this whenever possible so we have decided to use the water jet in the Wu Manufacturing Center. We will finish the piece on the lathe. Manufacturing plans are provided in Appendix D and discussed in section 9.

Retaining Disk Optimization

Figure 16 shows a Retaining Disk schematic with the predetermined dimensions in blue and the dimensions that we could set in red. The size and thread pattern of the actuator that we are purchasing dictated the inner most hole radius as well as the inner radius of the arced slots (Ri). The outer radius of the piece (R_R) was also predetermined by the housing size and the thread pattern that we plan to use. The dimensions left for optimization were the width and thickness of the spokes and the outer radius of the arced slots. The optimization of this piece was much simpler than that of the Plug and Gate because there was only one main focus, enlarging the arced spoked area as much as possible without compromising the integrity of the piece.

Figure 16: Retaining Disk Schematic



We analyzed the stresses in this part using Matlab, then using our results as well as considerations for fabrication we were able to set the three dimensions. The plot used to set the outer radius of the arced slots can be found in Appendix P. We determined that the maximum outer radius possible to ensure good machining and low stress is 0.45”, and because we were trying only to maximize the arced slots’ area we moved forward with this as our Ro. We then modeled the stresses in the spokes to determine the minimum thickness and width. The plot to stress versus thickness for various widths can be found in Appendix P and the code used to produce this plot is in Appendix R.

We used the results of our stress analysis along with our machining plans to determine the spoke dimensions. One very important factor that we didn’t want to overlook is the fact that we will be threading the outside of this part and then having to screw it into place during assembly. We needed a thickness great enough to allow for the threading to fully engage as well as spokes stable enough to handle the torque that we will need to put on them when screwing the Retaining Disk in place. We were not able to model the torque that our hands will put on the piece, so picked a very conservative thickness and then an appropriate corresponding width. Table 5 shows the results of our optimization.

Table 5: Optimized dimensions of the Retaining Ring

Radii		Spokes	
Ri	0.30 ± 0.005 [in]	Width	0.075± 0.005 [in]
Ro	0.45 ± 0.005 [in]	Thickness	6.50 ± 0.005 [in]

6.3 Analysis using course software

Using CES EduPack 2010 software, SimaPro 7.2 software, and DesignSafe 5 software we were able to learn a great deal about the material and manufacturing processes, environmental effects, and safety concerns involved in our project. A more detailed version of this can be seen in Appendix C.

The CES EduPack 2010 software first aided in the material selection of our parts by allowing us to input certain specifications such as strength, temperature, hardness, etc. that the materials would have to withstand during operation or testing. This program was extremely helpful because while it ultimately led to numerous materials we could use for our components, the software came with a detailed list of other variables such as cost or machinability that aided in our selection. Overall we learnt exactly what materials would work and then were able to narrow it down based off of other specifications. After narrowing down our selection to aluminum alloys we stayed in the CES EduPack 2010 software and were able to look at different manufacturing processes that were applicable to aluminum. Also, the software gave us pertinent information on mass manufacturing as well by providing information on tolerances, thicknesses, batch sizes, possible geometries that can be constructed, and among other things. We quickly learnt which processes would be applicable to our parts depending on what specifications we would need. For example we decided the Retaining Disk would ultimately be extruded or stamped, but since the stamping process could only go up to 197 thou and the Retaining Disk is larger than that, we couldn't go with that option. Also, by looking at tolerances and knowing that we ultimately wanted a press fit, we could choose the process with the smallest tolerances which in this case is also the extruding process. Overall CES EduPack 2010 assisted in choosing what materials we would use for our project and what manufacturing process would best suit those materials in a mass production situation.

The SimaPro 7.2 software helped to show the environmental effects that using the materials selected from the CES EduPack 2010 software. We compared the difference between 6061 aluminum which we used to make our prototype and cast aluminum which we would recommend to be used for the mass production of our project. We looked at the full life cycle of the materials as opposed to simply the materials themselves. This way, we were able to easily see and compare the effects these materials would have on the environment broken down into 10 sub categories and 3 meta-categories. These main categories were human health, ecotoxicity, and resources. We learnt that both human health and ecotoxicity are affected similarly in the full life cycle of these two materials; however, 6061 aluminum uses almost twice as much resources as cast aluminum. We were also able to analyze the exact mass of raw materials, air emissions, water emissions, and solid waste between 1.9272 lbs of 6061 aluminum and cast aluminum. This amount of aluminum represents the mass of our housing. Oddly enough we learnt that while there is a larger mass of raw materials needed to create this amount of cast aluminum than there is to create 6061 aluminum, the resources taken from the environment were much larger in the creation of 6061 aluminum which can be explained because it solely relates to the mass of the raw material and doesn't easily break down the specific materials going into the creation of each material. SimaPro 7.2 was very informative in the environmental effects that go into the full life cycle of different materials while opening our eyes about just how many things are affected and damaged due to the multiple processes that are involved in the material's creation.

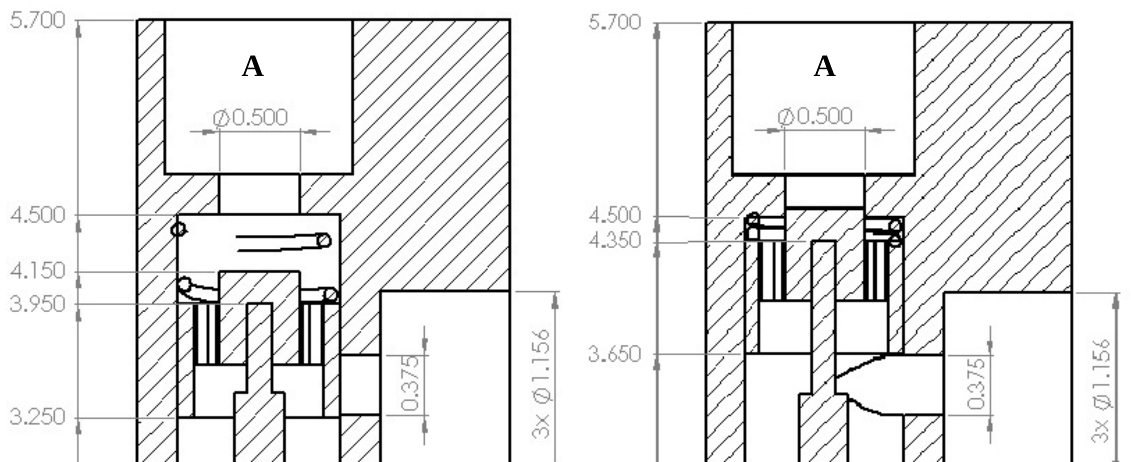
Our design for safety was produced using DesignSafe 5 software. This was very helpful in analyzing everything and anything that could go wrong with our project. By first breaking down every possible problem into categories we could choose the ones most relevant to our project. After that we looked at how the hazard may occur, what would happen if it did occur, the severity of the problem, how probable said problem would be, and then how to reduce the risk of the problem. This was a great tool because if done correctly we were able to identify and minimize any problems before they could occur. The DesignSafe 5 software also expressed the level of risk involved in each task exemplifying how cautious we should be. Since most all of our risks came out to be negligible/low, we knew going in that as long as we followed the steps we laid out to reduce any possible risk then there should be virtually nothing that would go wrong during the operation of our valve.

7. Initial Prototype Description

Our prototype is unique in that we are able to produce a viable part with only several cosmetic differences from our final design. These differences only serve to accommodate our testing facilities as well as the available actuation mechanism. Therefore our prototype can be viewed as a scale model of our final design.

An exploded view of our prototype in its normal and actuated positions as well as a fully dimensioned cross sectional view of our assembly in both positions can be seen below in **Figure 22**. Engineering drawings of each individual part in the full assembly can be found in **Error! Reference source not found.**.

Figure 17: Cross sectional view of the assembled prototype in the normal position (left) and fully actuated position (right). Letters A-D represents different thread patterns (see below). All tolerances are set to ±0.005inches.



D A

C B

A

D A

C B

A

Figure 18: Exploded View of Prototype in Normal Position

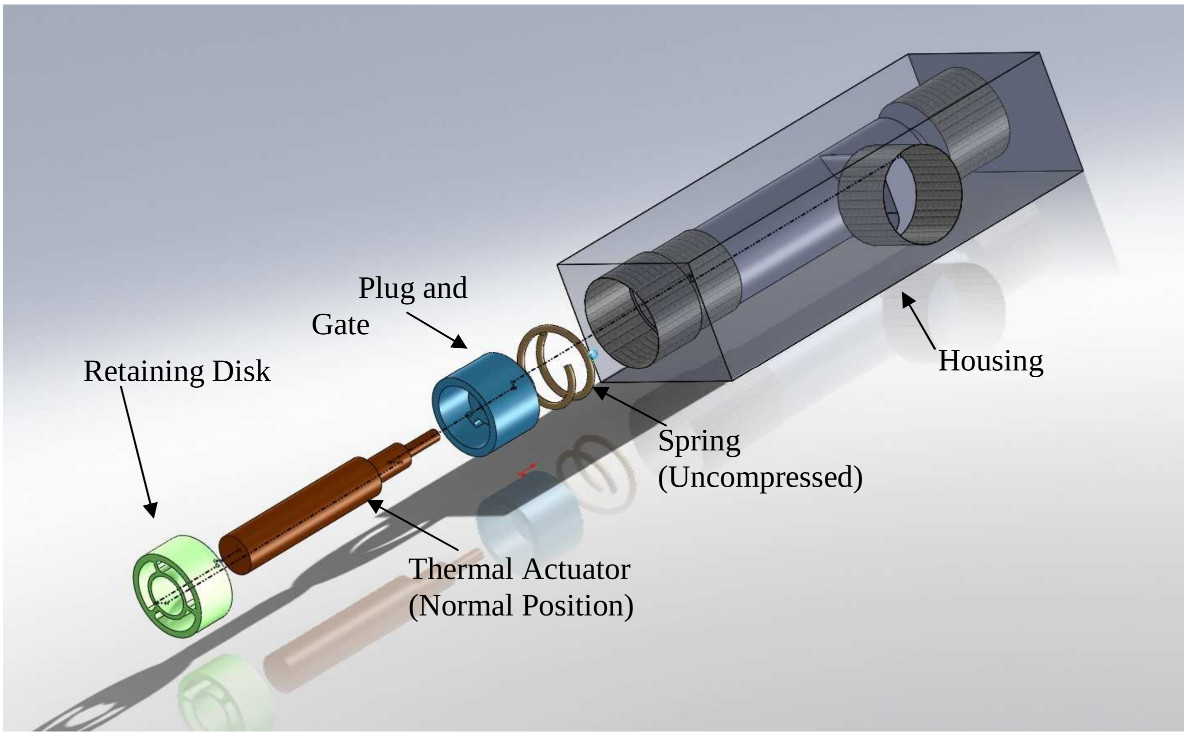
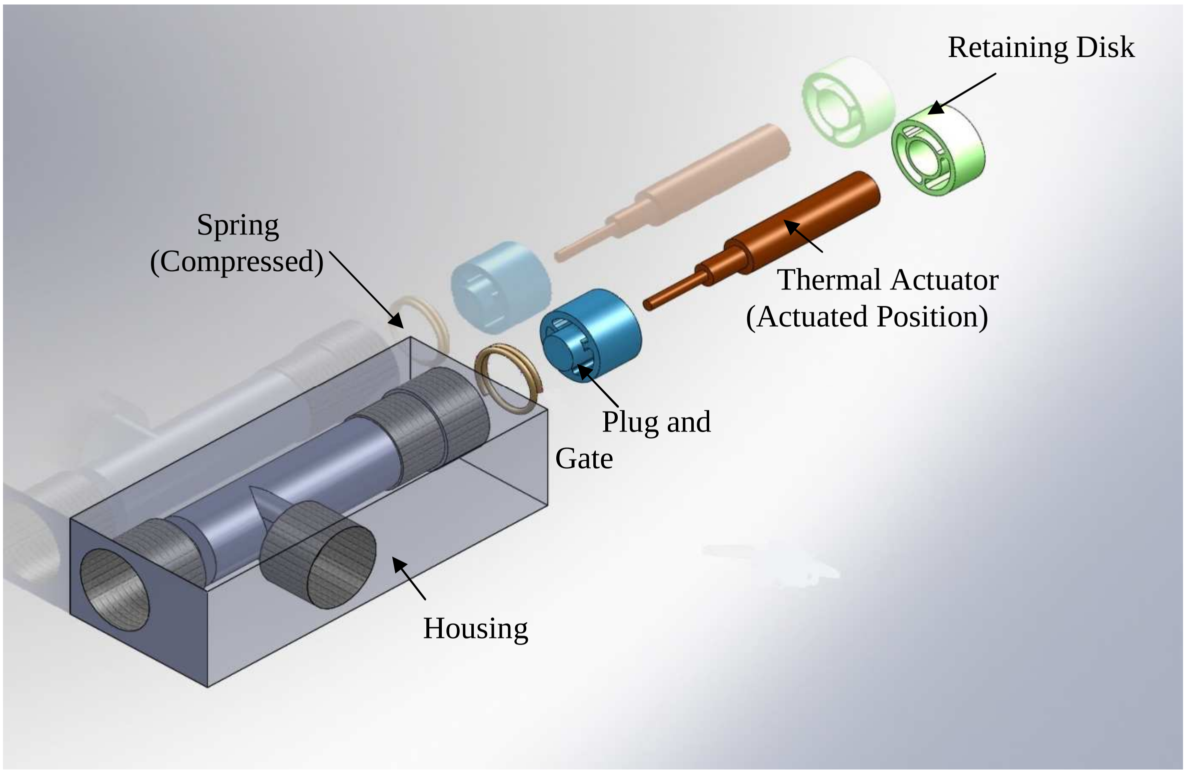


Figure 19: Exploded View of Prototype in Actuated Position



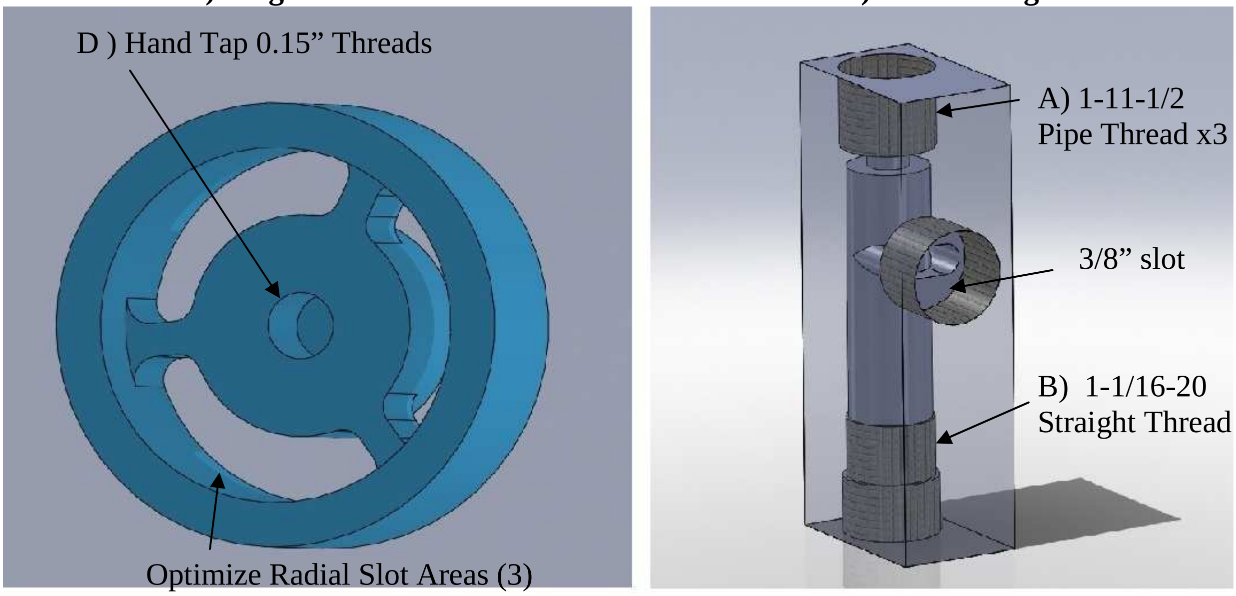
In **Figure 17**, each letter A-D indicates a different thread pattern which we have determined most suitable for our needs. The lines which will be screwed into the housing for our testing (A) will have a 1-11-1/2 pipe thread pattern. These threads will be created in the housing using a tap and will attach to our PVC test apparatus which will be threaded using single point threading on the lathe. This is not consistent with the EPAs thread pattern because each fitting used to attach the housing to a pipe costs roughly $100.00 which is not financially feasible in the scope of this project. Thread pattern B will be a 1-1/16inch-20 straight thread which will be created using a hand tap. The outside threads on the retaining disk which will be thread into thread pattern B will be created using single point lathe cutting. Our custom actuator created by Therm Omega Tech, Inc. will come pre-threaded along the bottom half, thread pattern C, and along the piston, thread pattern D. These are selected by the actuator company; thus, we will use a ½-20inch hand tap to screw the retaining disk into threading C and we will use hand tap #6 on the underside of the plug and gate to match the thread pattern on the actuator. Threading has been selected as the method for joining our components in the prototype due to the ease of assembly and disassembly while still allowing for a solid connection for testing.

The main difference between our prototype and alpha design is the custom actuator. As stated, it will be pre-threaded by our actuator company as opposed to us threading both sides manually. While this will cost more it will save time and we can assure that the hand taps for both our parts will be in shop when we need them since most part diameters do not have a common hand tap associated with them; thus, we will save time allowing us to focus on our testing. Another huge difference in our prototype's actuator is the actuation distance. For our desired specifications Thermo Omega Tech can solely supply us with an actuator with a 3/8" stroke. Since we originally created our alpha design based off of a 1inch stroke we had to change dimensions of the housing and plug and gate. As can be seen from the isometric view of the housing below (Figure 20) we created a 3/8inch slot which will maximize the volume of flow into the radiator when fully actuated and keep the pressure drop at a minimum. After setting the dimension of the slot we worked on the plug and gates parameters. First since the actuator's piston can be threaded for us, instead of rubbing flush against the bottom of the plug, we decided to hand tap a hole which the piston can simply screw into and actuate against. We then cut down the length of the gate so that it would be sitting just 1/80inch below the stream to the radiator when in the normal position; thus, the opening to the radiator will be fully open when the prototype is fully actuated. These changes can be seen in the isometric view from the underside of the plug and gate below in Figure 20a.

Apart from choosing proper thread patterns and adjusting to a much shorter stroke of actuation, our prototype differs from our alpha design in that we've used Matlab to perform a stress analysis as well as our knowledge of fluid mechanics in order to optimize the dimensions of our plug and gate and retaining washer to allow for the most flow through the rounded slots as well as taking machinability into account. These optimized dimensions can be seen on the engineering drawings in **Error! Reference source not found.** and an isometric view of the newly dimensioned retaining disk can be seen in Figure 21a below.

Figure 20: Prototype changes of the:

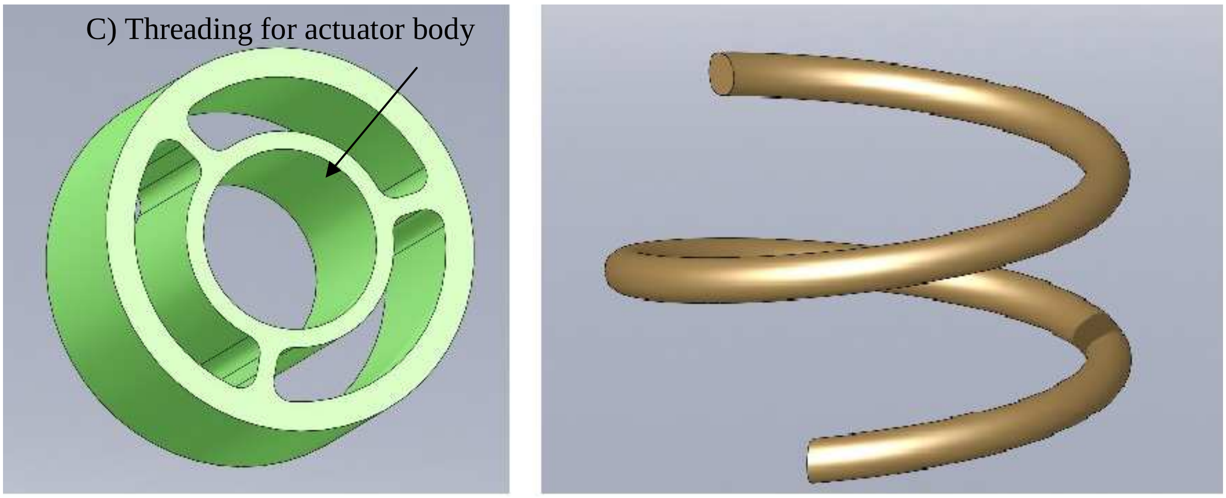
a) Plug and Gate **b) the Housing**



The normally biasing spring also has been changed for our prototype. In our alpha design we arbitrarily decided on a spring that we thought would work, but after thorough analysis of forces, keeping our customer requirements in mind, and equipped with the knowledge of the actuation force of the thermal actuator we were able to find a cut to length spring with adequate force to return the actuator to the normal position once the system is properly cooled (<160ºF). Figure 21b below shows this spring in its uncompressed position. Since we predetermined the length of the spring, we solely have to cut the spring at that length; however, since this length correlates to 1.6 coils there won’t be a flat surface on both sides; thus, we will grind both surfaces to create flatness.

We have not yet performed a full cyclic loading stress analysis on our current geometries in order to estimate a full life cycle of our part. This is due to the fact that we have anticipated that due to our rigorous testing procedures, modifications to optimize our design may have to be made. However, a full blown cyclic loading analysis will be performed once we have undergone testing.

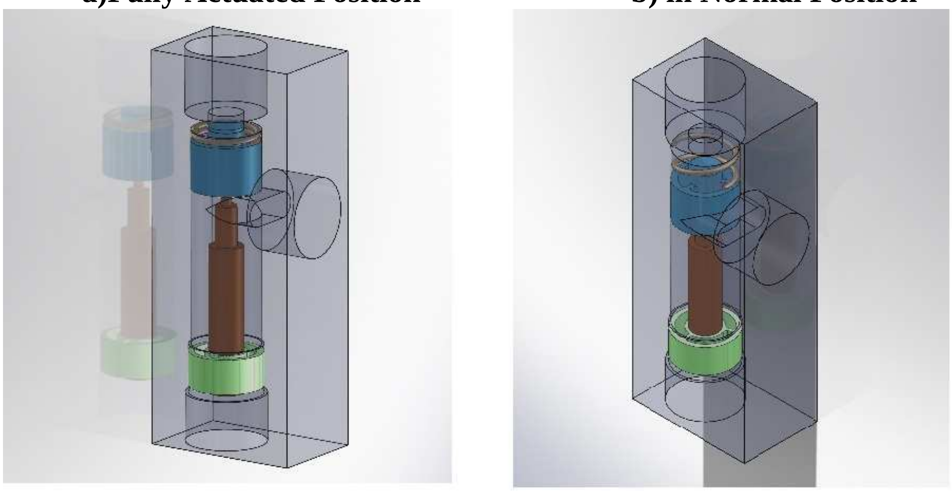
Figure 21: Prototype Changes of the:
a) Retaining Disk **b) Spring**



After all the changes to the alpha design above have been made the result is our assembled prototype which can be seen below in both the normal and actuated positions in

Figure **22**.

Figure 22: Assembly of Prototype
a)Fully Actuated Position **b) in Normal Position**



8. Final Design Description

There are key differences between our prototype and the final design which will be mass produced by the EPA. This process was thoroughly thought out in the development of each part of our prototype so that with a few slight alterations in the machining processes cost can be greatly reduced. By casting the retaining disk and plug and gate pieces, we can reduce material costs from approximately $90.00 to under $1.00. Also since the EPA plans to manufacture roughly 10,000-25,000 of these valves to be implemented into their HHVs, perfectly dimensioned springs will only cost a few cents each.

Unlike our prototype, the mass manufactured actuator will actually have a 1inch stroke so instead of the extra machining process needed to create the 3/8inch slot in our prototype, we can simply cast the 1inch channel leading to the radiator. Also, our prototyped actuator will be created from stainless steel and be threaded, but seeing how this greatly improves cost, the mass produced actuator will be made of a cheaper material and not be threaded.

The final design will not have any threading at all because the cost of that machining process could outweigh the cost of every part and other machining process combined. Therefore, we suggest press fitting the assembly together as opposed to any threading. The entire design should fit together perfectly, similar to our prototype, so that the valve can be simply press fit into the housing via the retaining disk and be fully operational, thereby reducing production costs.

Our final design represents an optimized version of our alpha design, all parameters have been optimized using rigorous engineering analysis. Therefore the function of our valve remains the same as is described in Section 7. It will be able to actuate at the correct temperature range and by doing so, will re-direct the flow of operating fluid from the re-circulation outlet to the radiator line. This re-direction will be accomplished by utilizing a plug style blockage to the re-circulating stream and simultaneously withdrawing a gate from the radiator outlet. Leakage between streams has been minimized through tight tolerancing and all parts are pressure rated to 200 psi. Also, as mentioned above, cost of manufacturing has been thought through in every step along the way. Pressure loss through the final valve has been minimized via geometry optimization. The housing channels and thread patterns at outlets/inlet have been designed to the specifications of the EPA system to have full compatibility, as well as to disallow leakage outside the housing. We believe that this final design will fully comply with our customer requirements to the best of our abilities.

9. Fabrication Plan

Our manufacturing involved some troubleshooting to finally create the radial geometries of the plug and gate and the retaining disk. We initially had several interviews with Bob Coury and John Mears to investigate the processes and tooling that would be necessary to complete our prototype. Also, we have had an interview with Dr. Albert Shih to verify that our most complicated component can be manufactured on a mass scale and will not require a full redesign when production is taken into consideration. Below is a summary of our bill of materials and a description of our machining processes, a full set of machining instructions can be found in Appendix D. Also, a full bill of materials including prototype components as well as testing apparatus is included in Appendix A.

In order to calculate machining speeds and feed rates we used the Machinery's Handbook [20] to find the proper surface speeds and the feed rates (inches per tooth) for our materials and tooling for each operation (drilling, reaming, boring, etc.). These speeds and feeds were then utilized along with Equations 6 and 7.

$$N = \frac{12s}{\pi D}$$ **(6**

$$\boldsymbol{f = n_t f_t N}$$ (7

Where *N* is spindle speed in RPM, *s* is surface speed in feet per minute, *D* is tool diameter (or part diameter for lathing operations), *f* is feed rate in inches per minute, n_t is number of teeth, f_t is inches per tooth. The results of these calculations are found in the detailed manufacturing plan in Appendix D

9.1 Manufactured Components and Materials

The housing, spoked retaining disk and plug and gate (parts 1,2,3) were all manufactured from stock obtained from an outside vendor. They will be made in the undergraduate machine shop. Below are summaries of the material, stock, cost and source of all of our manufactured parts and the processes we used for manufacture.

1. Housing	*Material*	6061 Aluminum
	Stock	1.5” X 6” X 3”
	Cost	$13.25
	Source	Alro Metals Plus

The housing is necessary to our design to contain the hydraulic fluid. It also serves as a functional portion of the valve system, creating areas that are designed to work with the valve member to divert the flow of the operating fluid. The housing holds the valve member in place via the spoked retaining disk and valve member geometry. It also has threaded ports to hold incoming and outgoing lines. Manufacturing this part involves drilling three holes, tapping four areas, reaming one hole and performing one end milling operation.

2. Retaining Disk	*Material*	6061 Aluminum
	Stock	1” Flat Stock
	Cost	-
	Source	Donated Scrap

The retaining disk fixes the actuator to the housing via the threads on the inside and outside of the washer. The spokes will allow the operating fluid to flow through the disk and around the actuator. The blank for the retaining disk was cut from flat stock on the waterjet, the center hole of the blank was then drilled (the drill followed the piercing to give a larger sized hole from which to center off of) this was then fixed to the lathe using a special fixture along with tail stock pressure and turned down to size. This piece was then fixtured using a four jaw lathe chuck for the boring operation. After boring, the outer circumference was threaded using the lathe and finally parted off.

3. Plug and Gate	*Material*	6061 Aluminum
	Stock	1” Flat Stock
	Cost	-
	Source	Donated Scrap

The plug and gate will actually divert the fluid flow between outlet ports. It will be fixed to the actuator’s piston via a press fit. This component will block flow through the re-circulation port via a plug action, and will slide with a gate motion to divert flow to and from the radiator outlet port. The blank for this piece was also initially cut out using the waterjet and turned down to size using the same process as the retaining disk. After turned down to size, the end of the plug was turned down while still fixtured to the lathe with tail stock pressure. Then the piece was fixed in a four jaw chuck and the center cavity was bored out. Finally the center hole was reamed to size.

9.2 Purchased Components

The linear thermal actuator and spring (parts 4 and 5) will be purchased from vendors and modified in the undergraduate machine shop. Below are summaries of the material, stock, cost and source of all of our purchased parts and any modifications we made to them.

4. Linear Thermal Actuator	*Material*	Stainless steel, thermally expansive wax
	Part No.	TOT-04L
	Cost	$80.00
	Source	Therm Omega Tech, Inc.

The linear thermal actuator has the function of moving the plug and gate to redirect flow through the housing. This actuator was ordered such that it would operate in a certain temperature range and had a small cross sectional profile with respect to the flow. No modifications to the actuator were made.

5. Spring	*Material*	Tempered Steel
	Part No.	9637K25
	Cost	$10.59
	Source	McMaster Carr

The (compression) spring returns the valve member to a normal position when the actuator is cooled/cooling by applying a force. The spring was cut to length such that it will compress to a length that will not interfere with the performance of the plug. The ends of the spring were ground so that it would sit flat against the step in the housing and the leading edge of the plug and gate.

The initial fabrication plan is very different from our final design, in that our prototype will have to be made completely from scratch and all cavities will be machined. Our final design will be made using mass manufacturing processes. The housing will be cast, as well as the plug and gate. The spoked retaining disk will be extruded and the threading process will be omitted. These initial form processes will eliminate a great deal of machining and cost. These differences can be visualized in Appendix D, where all machining processes that will be replaced in mass manufacture are shown with their step number highlighted in orange. Steps that won't be replaced during mass manufacturing will have their step number highlighted in green. All tapping and threading operations (except those for inlet/outlet ports) will be replaced with machining processes consistent with press fitting these parts (drilling and reaming).

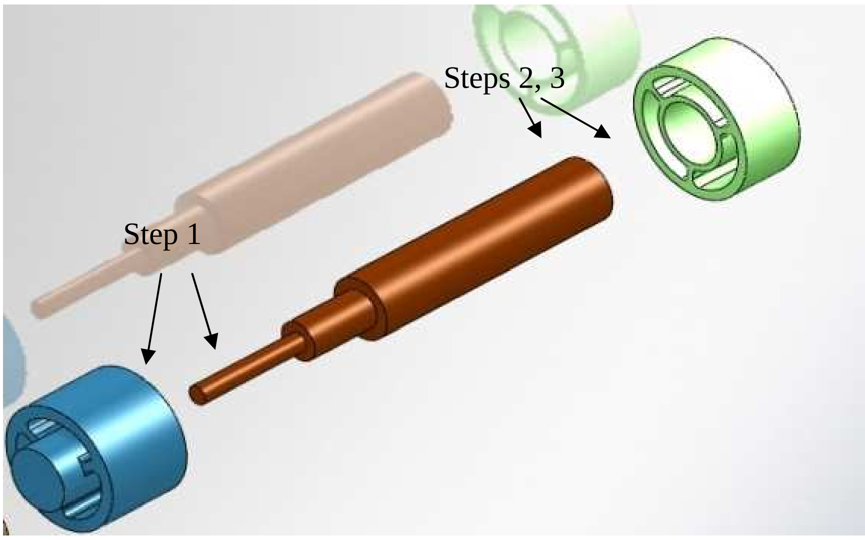
9.3 Valve Member Assembly Instructions

The assembly of our prototype is very simple, as we intended it to be viable for mass production with only a slight modification to aid in removing. We also took some extra precautions to ensure that we did not have to recreate any parts of the assembly. These processes are summarized in Figure 23 and Table 6.

Table 6: Valve member assembly procedure

Step	Process
1	Press plug and gate onto actuator arm using an arbor press
2	Place retaining disk on a piece of stock, heat stock from bottom so heat is evenly distributed throughout retaining washer. Washer will expand enough to place actuator in without pressing.
3	Once hot enough, place the actuator into center hole of retaining disk. There should be no resistance. Apply pressure to the actuator as the piston arm will actuate at this temperature.
4	Apply pressure to the top of the plug and gate until fully cooled to ensure actuator is returned completely to proper position. Use an air nozzle to cool.

Figure 23: Plug and gate is pressed on to piston arm, retaining disk is fitted to base of actuator



The plug and gate was able to be pressed on to the piston since the actuator could be flush to the table of the arbor press and the pressure distributed to this base. We chose to use a heat expansion method to fit the disk on to the actuator to avoid pressing on the plug and risking damage. It is very important that pressure be applied to the top of the plug during the heat fitting process. Otherwise the actuator could over-actuate and push the piston entirely out.

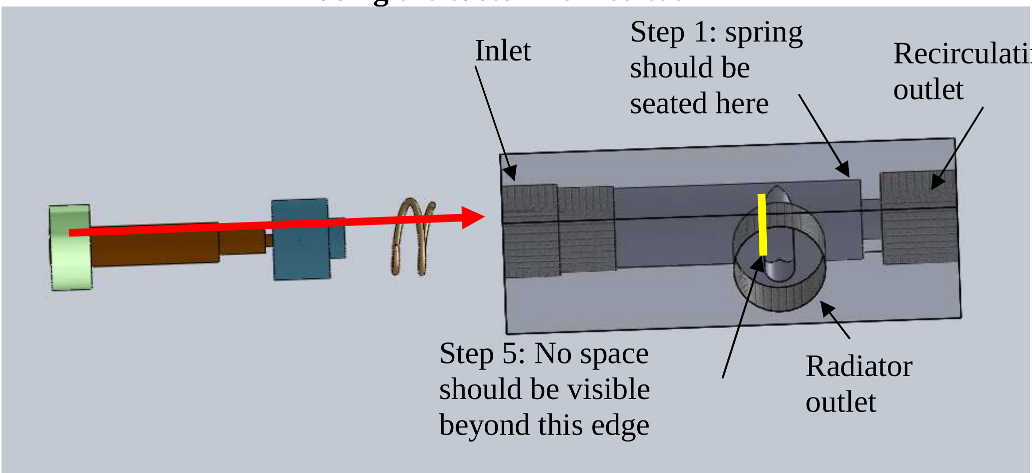
10. Assembly of Prototype

Assembly of the delivered prototype can be achieved through several easy steps summarized in Figure 24 and Table 7Table 7.

Table 7: Step-by-step prototype assembly instructions

Step	Process
1	Drop spring into main passage from inlet of valve, make sure it is seated properly
2	Insert front end of valve member as shown
3	Make sure plug and gate portion of the valve member is in fully un-actuated position. To ensure this is the case, see that plug is seated flush with threaded portion of the actuator (no part of the piston arm should be showing)
4	Use custom forked tool to screw valve member into housing
5	Peer in to radiator outlet (side branch). Screw valve member into place until bottom edge of gate is flush with bottom edge of outlet. Unscrew ~1/4 turn this approximates the necessary 1/80” coverage.
6	Check recirculating outlet to make sure spring has not flipped onto its side
7	Screw 1” NPT fittings into inlets and outlets

Figure 24: Spring is inserted, followed by the valve member. The valve member is screwed in using the custom forked tool



11. Validation

Since we identified the testing of our valve to be one of our major challenges, we have been consulting with Dr. Ceccio, at the University of Michigan, about how to overcome the challenge. We have identified three possible methods of testing the fluid dynamic characteristics of our valve. These methods are: a recirculation system, an open system with large barrels of testing fluid, and an open system that uses air. These testing methods make use of a variety of testing fluids; however, each method has distinct advantages and disadvantages. We have decided to test the actuation as a function of temperature separately from the pressure drop and leakage between streams, since heating a working fluid would require a lot of energy, time and expense.

11.1 Testing Methods Considered

A recirculation system consists of a closed system with a pump to pump the test fluid through the valve at the desired flow rate and pressure. The working fluid could be oil or water. If water were used, we could scale our results based off of Reynolds number of the flow and dynamic viscosity of the testing fluid. The system would also have the necessary gauges to measure pressure drop, and reservoirs to measure leakage between streams. With this recirculation system we could recreate the actual flow characteristics of the valve when it is in service in the HHV and verify that the valve can actuate against the forces imparted by the working fluid and pressure differentials. This type of testing method is in accordance with ISO 4411:2008(E) [13]; however, unfortunately, the equipment needed to construct this system would cost more than $4000.00 brand new. In addition to the high cost, safety and time are major issues. Since the system would operate at 200 psi, all fittings and components would need to be meticulously designed and constructed to ensure the safety of the system and its operators. Building this system for testing would require more time than we have once the valve is fabricated.

The second testing method is an open system that has a large reservoir of testing fluid. The reservoir can be pressurized to a necessary pressure that would allow proper flow out of the reservoir and into the valve. Pressure drop across the valve can be measured with a pressure gauge. The flows out of the valve are into two separate reservoirs so leakage between streams can be measured. This system could be used to test flow with water, and we would scale the results based on Reynolds number and kinematic viscosity to obtain pressure drop and leakage between streams, using oil at the service specifications of the valve. This system would require large reservoirs and testing area, which most likely require more time than we will have to properly collect and assemble.

The third and preferred method of testing the dynamic fluid characteristics of the valve uses air and a method of scaling the results to find the pressure drop over the valve and the leakage between streams. The air used for testing would initially be stored in a compressed air tank. Since we would scale the flow characteristics based on Reynolds number and kinematic viscosity, we could use a pressure low enough that PVC piping would be adequately strong between the system components. Pressure can be measured with manometers and flow rates (needed for leakage between streams) can be measured with orifice flow meters. This testing method is safe, cheap, and can be constructed in the amount of time we will have to test the valve.

The above testing methods all require an analysis of the flow through the valve at operating specifications so the test can be designed to properly scale the results. We calculated Reynolds number for our valve, using oil at 160˚F and 68˚F (approximately room temperature). We then calculated Reynolds numbers our valve for the same temperatures using water and air. We used the equation $\mathrm{Re} = \frac{DV}{\nu}$,where *Re* is Reynolds number, *D* is diameter of the channel, *V* is the fluid velocity, and is the kinematic viscosity of the fluid. These values are in Table 8. If Reynolds number of the test is the same as the operating conditions of the valve, then pressure drop and leakage between streams of the valve at operating conditions of the HHV system can be extrapolated from the test results. Since Reynolds number is dependent on velocity, which is found from the flow rate, changing the flow rate (optimally lowering it), can produce a test that uses a flow rate more manageable than 35 gpm and matches the Reynolds number needed to scale the results.

Table 8: Reynolds Number for oil, water and air at operating temperature and room temperature for 35 gpm flow rate.

	Temperature ˚C	Kinematic Viscosity (m^2/s)	Reynolds Number
SAE 10W	20	1.00E-04	1.10E+03
SAE 10W	75	1.00E-05	1.10E+04
Water	20	1.00E-06	1.10E+05
Water	75	4.00E-07	2.76E+05
Air	20	1.80E-05	6.12E+03
Air	75	2.05E-05	5.38E+03

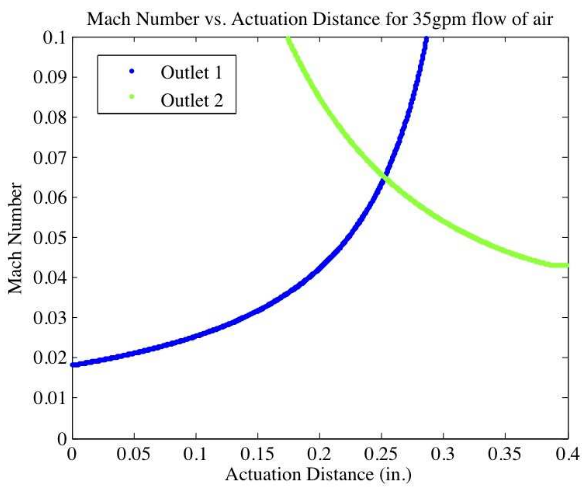
For a test that uses a compressible fluid such as air, in addition to matching Reynolds number, Mach number must also be considered. If the Mach number is lower than 0.1, then the compressibility effects can be ignored and the results of can be used to find valid pressure drop and leakage between streams for the valve. We have begun calculating the Mach number within our valve at various actuated positions. So far the Mach numbers we have found are less than 0.1. We anticipate that after more calculation, the test method that utilizes air will be a valid test, and after verifying this completely, we will use this testing method to validate the operation of our valve. Although this testing method is not the preferred method of ISO 4411, Dr. Ceccio has offered his professional opinion by recommending this test to us.

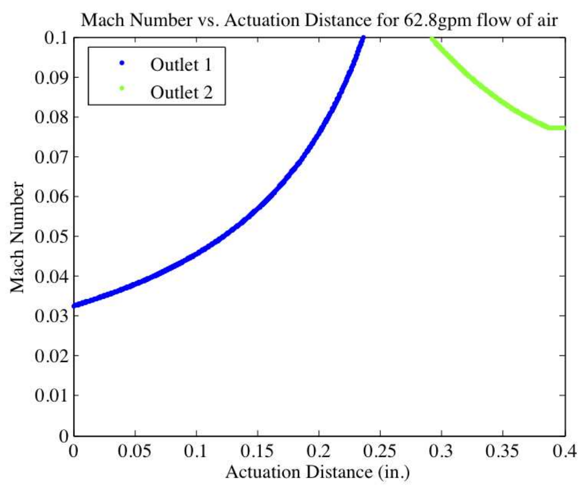
11.2 Validity of Tests with Air

We calculated the Mach number at each outlet of the valve as a function of actuation distance. This calculation was completed for both 35gpm flow and 62.8gpm flow of air. Since the fraction of the flow through each outlet is impossible to calculate and must be measured [17], the Mach number calculations were made with the worst-case assumption that all of the volumetric flow was traveling through the outlet (no flow diversion whatsoever for any actuation distance), for which Mach number was being calculated. Mach numbers were calculated Matlab (see Appendix T). Figure 30 shows the results of these calculations.

According to the worst-case analysis we performed, our test is valid for 62.8gpm flow for the first 0.235 inch of actuation for the outlet back to the system and valid after the first 0.290 inch of actuation for the outlet to the radiator. For 35gpm flow, it is valid for the first 0.286 inch of actuation for the outlet back to the system and after the first 0.174 inch of actuation for the outlet to the radiator. Since we will run at a flow volumetric flow rate in this range, our analysis confirms a large enough range of actuation to obtain meaningful results for both pressure drop and leakage between streams. Since the actual flow rate through each outlet will be less than that of the worst-case assumption, the range of actuation for which our test is valid will be larger. We will use our test results to calculate the actual Mach number at each outlet during each test performed to ensure its validity with actual data.

Figure 25: Plotting Mach number vs. actuation distance for each outlet shows the valid range of our test for the worst-case assumption of the entire flow through each individual outlet





11.3 Air Test Procedure and Results

Since we tested the fluid characteristics of the valve with air, we cannot replicate the forces imparted by the hydraulic fluid at 200 psi and 35gpm. For this reason we have completed a thorough analysis of the forces and stresses caused by the fluid momentum and pressure (Section 6).

We conducted our air test at room temperature. To simulate the position of the valve member at varying temperatures, we constructed a replica of our actuator. With this "dummy" actuator we could manually vary position of the plug and gate by screwing it on or off of a threaded rod that represented the actuator's piston. Our test rig was powered by a Paxton blower capable of 250CFM. We varied the frequency of the electricity provided to the blower with a motor control. We used lengths of PVC pipe that were at least 40 times the diameter (40 inches) long into and out of the valve to ensure air flow was fully developed. We made flow straighteners out of drinking straws and placed them at the inlet to each pipe to rid the flow of any swirling motion. We put an orifice flow meter on each pipe out of the valve. Pressure taps were placed at the inlet and both outlets of the valve and before and after each flow meter. The flow meters and pressure taps were constructed according to specifications for thin-plate, square-edged orifice meters with D (diameter) and ½ D pressure taps found in *Fluid Meters, Their Theory and Application*[26]. We used a U-tube manometer with water to measure pressure differential over the valve and the flow meters. The resolution of the manometer was 0.05 in. H_2O. A diagram of the test rig can be seen in Figure 26 and the actual test rig can be seen in Figure 27.

Figure 26: Diagram of Test Rig

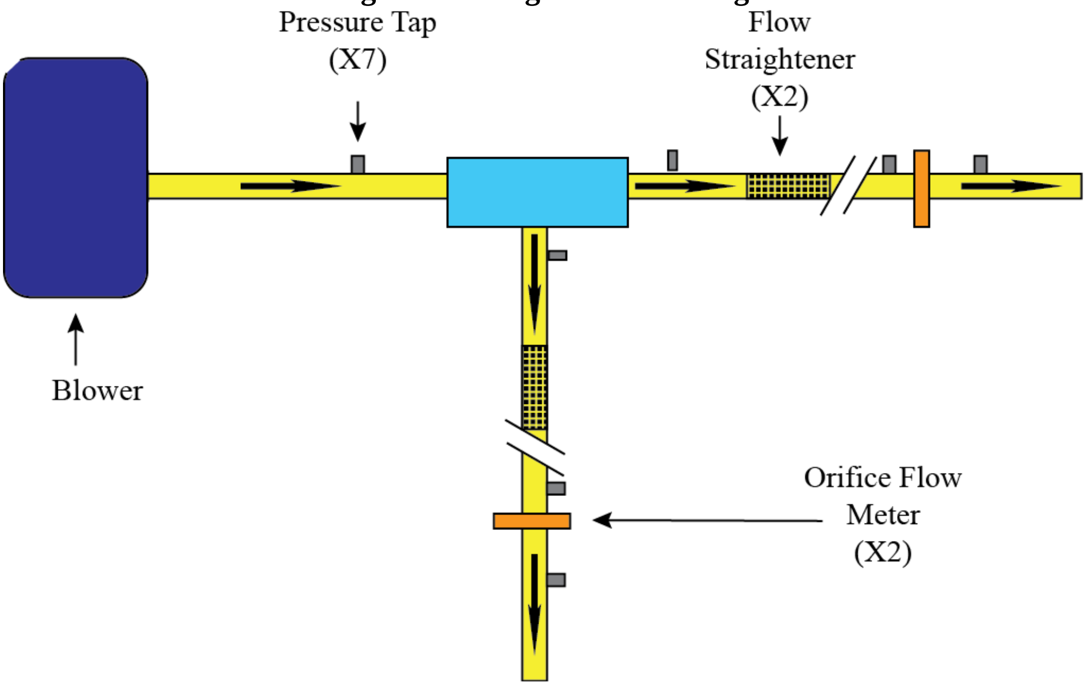


Figure 27: Complete test rig with valve and blower attached

The diameter of the PVC pipe used in the test rig was 1 inch. The orifice in each orifice flow meter was 0.5 inches. We sized the orifice so that Mach number through it would be less than 0.1. Using (8, volumetric flow (Q) can be calculated from the pressure differential over the meter ($p_1 - p_2$), the ratio of orifice diameter to pipe diameter (β), the density of air (ρ), the area of the orifice normal to the flow direction (A_0), and the orifice meter discharge coefficient (C_0) [18].

$$Q = C_0 A_0 \sqrt{\frac{2(p_1 - p_2)}{\rho(1 - \beta^4)}} \quad \text{(8}$$

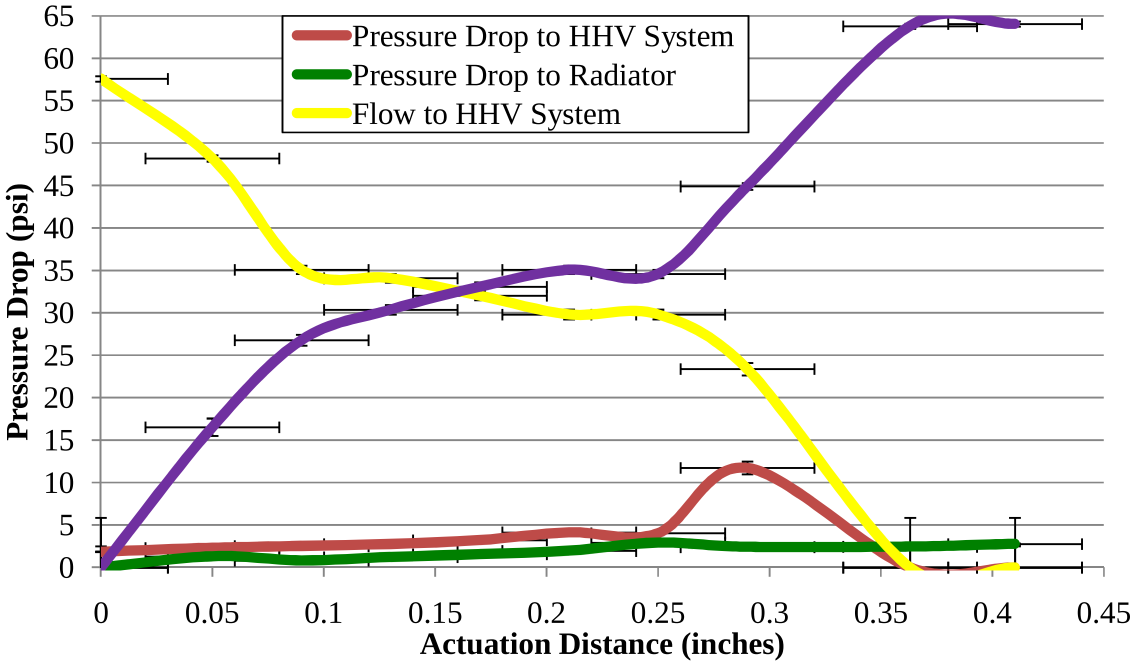
The orifice discharge coefficient used was 0.62, and was determined from figure 8.41 of *Fundamentals of Fluid* Mechanics [18] for a β of 0.5 and a Reynolds number of approximately 11000.

For a determined flow rate through the valve, we calculated the loss coefficient (K_L) through each outlet with pressure drop from inlet to outlet (ΔP), density of air (ρ), and velocity of air (V), using equation (9 [18].

$$\Delta P = K_L \frac{1}{2} \rho V^2 \quad \text{(9}$$

Since the Reynolds numbers for the air test were matched to that of in service conditions, once loss coefficient was calculated for air, it could be used with the density and velocity of oil at 170°F and 35gpm flow to predict pressure drop through each outlet of the valve. These results are shown in Figure 28.

Figure 28: Scaled Air Testing Results

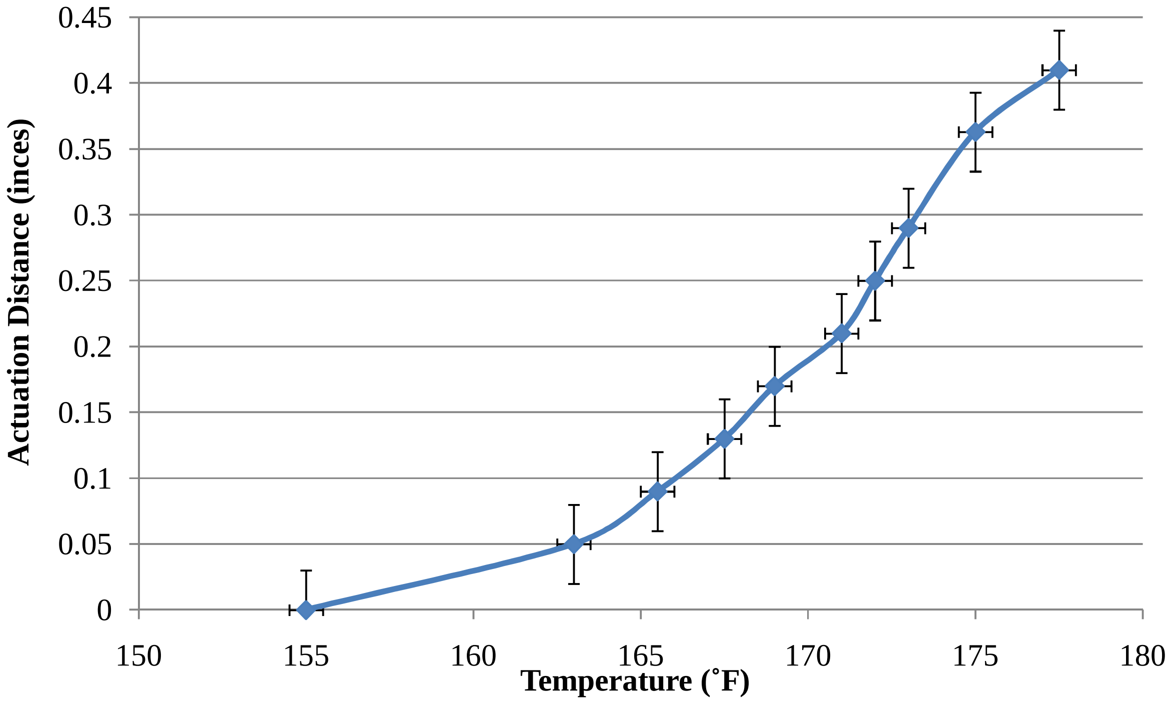


Due to the measurement increments on the manometer, we could not calculate a firm value for leakage between streams for both the normal and actuated positions. In both positions the flow through the orifice flow meter of the blocked outlet produced a pressure differential smaller than the resolution of the manometer. This means the flow was lower than 5.85gpm, which corresponds to a manometer reading of 0.05 in. H_2O. We could conclude that leakage between streams was less than 9% of the total flow in the unactuated position and less than 10% of the total flow in the actuated position.

11.4 Temperature Response of Actuator

Therm Omega Tech, Inc. tested the actuator we used before they delivered it to us. They sent us a curve of the actuation distance versus temperature. Between 155°F and 177.5°F the actuator's piston moves 0.41 inches, and follows the curve presented in
Figure 29.

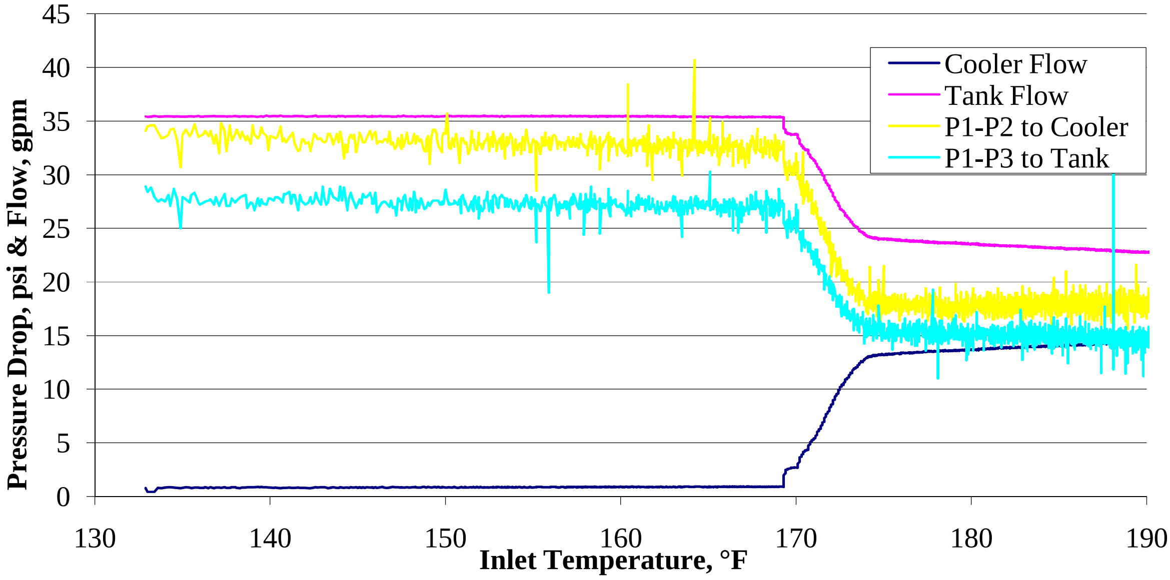
Figure 29: Actuation Response to Temperature



11.5 EPA Test Results

The EPA conducted tests of our valve with a recirculating test stand that is similar to the set up we recommended from ISO 4411:2008(E). This test completely replicated in service flow and temperature of Mobil 1 ATF. During their test, a problem occurred that caused the flow to the radiator to never go above 15 gpm. Furthermore, they measured pressure drop to be much higher than we predicted with our scaled results of the air test. The EPA provided us with their results in Figure 30.

Figure 30: EPA's Test Results



Before the valve was in the actuation temperature range, the EPA measured the pressure drop to the radiator to be around 33psi and to the HHV system to be around 27psi. In this range, one would expect pressure drop the radiator to be high since flow to the radiator should be blocked, and the only flow through this outlet is leakage. The high pressure drop to the HHV system is likely a result of the high volumetric flow through the outlet. As shown in Equation(9, pressure drop is dependent on the square of velocity, and velocity of the fluid increases with volumetric flow through a given area. The actuator responded within the proper temperature range, and as it actuated, pressure drop through each outlet became lower. This makes sense since the outlet to the radiator was no longer blocked and its area increased. Additionally, since some of the flow was diverted to the radiator, the velocity through the outlet to the HHV system was also lower.

The valve failed before the EPA's test before was complete. The retaining disk screwed further into the valve. Also, when the valve was disassembled the spring was turned sideways, this could have happened during testing or when the valve was taken apart. This made it impossible for the valve member to reach its fully actuated position and seal the outlet to the HHV system. The flow through the radiator for this test only reached about 14gpm, so leakage between streams could only be calculated for the unactuated position. At this position, about 2% of the total flow exits to the radiator. This is within the 5% leakage between streams that was customer requirement.

11.6 Comparison of Results

Our scaled results for pressure drop from the air test varied quite a bit from the values measured by the EPA. The EPA's test stand replicates all flow, temperature and fluid characteristics of the valve in service and does not rely on scaling. The differences in results are likely due to some invalidities in our air testing procedure. In hindsight, it would have been better to have an orifice flow meter at the entrance to the valve. This would have allowed us to measure total flow out of the blower rather than add the total flow out of each outlet of the valve. Using the orifice flow meters at each outlet meant that the Reynolds number of the air through the meter changed with actuation distance; however the orifice discharge coefficient was chosen for the total Reynolds number. Local effects of compressibility could have also caused our test to be invalid. We sized all of our components to avoid this, however, small defects in both the valve and test rig could have caused flow of air to compress, in which case the results of the test would not have scaled properly.

Although the values we determined may not be valid, we were still able to observe the trends of the air test's results. From these trends it is clear that flow does get redirected between the outlets as actuation distance increases. Except for one outlying point in the results of pressure drop to the system, the pressure drop through each outlet of our valve does roughly follow the same change. This trend was also clearly evident in the results of the EPA's test.

The EPA was not able to test our valve for the entire actuation distance since it failed before the test was complete. The results measured before the failure are valid and prove that our prototype has some problems that need to be resolved, but more importantly these results validate that or design is one that could easily be improved upon without too many changes. The EPA informed us that pressure drop measured for our valve was similar to a larger commercial valve they tested. By making our valve larger, pressure drop can be reduced. Although it had some failures, we are satisfied with the performance of our prototype, and we are confident that the validation suggests our valve concept can be made successful.

12. Discussion

Initial Prototyping Difficulties
Our parts were difficult to manufacture in a first batch prototype scope. We realize that this is always the case. To combat this we made a very detailed manufacturing plan, but still underestimated the difficulty of our processes and as a result spent a considerable amount of time manufacturing our prototype. The major aspects that were problems were involved with the complicated radial geometry of the plug and gate, as well as the retaining disk. We should have evaluated our machining skills before designing parts of this complexity.

Strengths
Our design fulfilled our requirement of reducing cost for large scale manufacture as it had as few components and manufacturing processes as possible. We had to consider the trade-offs associated with reducing the number of parts and our overall valve performance.

The seal performance was a particular cost-performance trade-off as we did not require a full seal at either outlet. We decided not to use o-rings to achieve a seal since these would require extra manufacturing and assembly processes. Most of the assemblies are achieved with press fits in our prototype except for the threaded retaining disk. This threading proved to be a difficulty in the prototype manufacture phase. The threaded portion of the disk could be changed to a press fit in a future design, especially one for larger scale production. We used two press fits in our prototype, and these did not fail in the EPA's test rig. Press fits are a connection type often uses in industry that also reduce manufacturing costs.

The pressure drop across our valve was not in the desired range although it was comparable with some other industry valves. However, during testing the linear actuator did activat within the correct temperature range.

Weaknesses

After testing in the recirculating system several issues were encountered with the spring and the valve member assembly. One possibility for the high leakage at the upper end of our temperature boundary in testing is that the valve member was not able to actuate the intended distance. In addition to this there was a much greater head loss than anticipated, this was most likely not due to any mechanical difficulty, but due rather to the small scale of valve geometry that we used.

One limitation of actuation distance may have stemmed from the threading used to install the valve member into the housing. This threaded connection was able to move slightly when testing at operational temperature, pressure and flow. The threading issue may have stemmed from any of these factors, as well as movement when coupling to the test stand.

When disassembled after testing, the spring had turned to the side, which could have caused binding of the valve when the actuator was trying to move. This reduced performance of the valve in that full actuation could not be achieved. It is possible that the spring turned during disassembly, but we cannot verify this as a disassembly issue as the valve must be disassembled in order to observe the position of the spring. The EPA tested the valve without the spring at our request with washers in its place. This produced worse results than before.

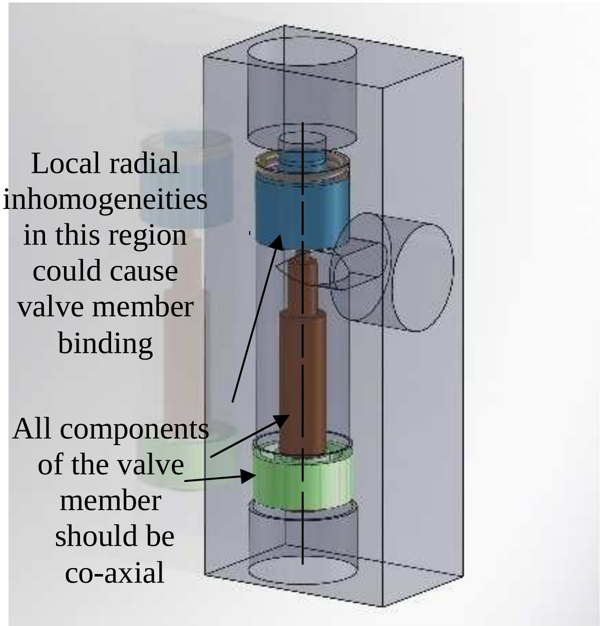
In addition to the spring there is another possible explanation of the valve member's inability to actuate. During assembly the components of the valve member may have had some tilt due to a small run-out on the lathe which would cause the member components to not be fully co-axial in final assembly (

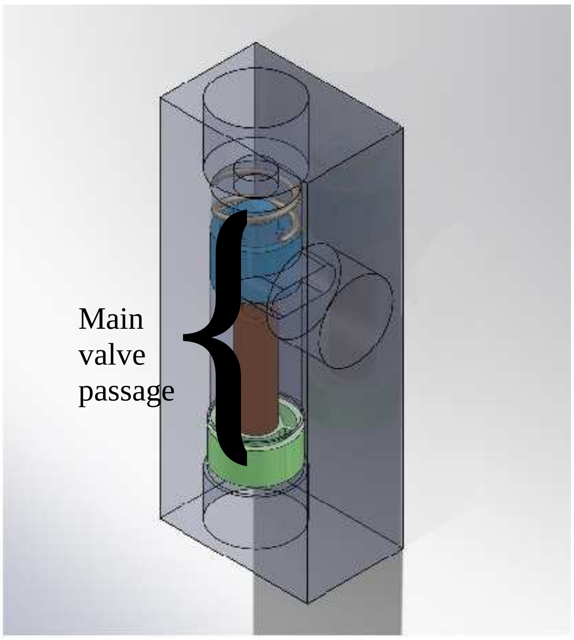
Figure 31). If this were the case, during actuation the plug and gate would run onto the side of the valve causing binding of the member.

Binding of the valve member may have also been due to the limitations in our machine shop experience, as well as available tooling. In order to create the main valve passage (

Figure 31) in the valve housing we had to use a drill, then an end mill to remove the drill taper. The inability ream or bore this passage due to its length caused some localized inhomogeneities in the width of the main passage. We were able to test that the plug could freely move down the main passage when disconnected from the valve member, however were unable to test the ability of the full mechanism under load before testing on the EPA test stand.

Figure 31: Reproduced for reference from Figure 22





An additional consideration that may have caused valve binding was the radial moment caused by the spring on the valve member. We attempted to combat first using a heated block to deform the ends of the spring to closed rings which would better distribute the spring load, however the addition of a full coil at each end of the spring blocked the full actuation of the valve so this avenue was abandoned. The next option was to grind the ends of the spring so they would sit flush with the top edge of the gate. This appeared to work, however full contact was not achieved around the entire upper edge of the gate, so a moment was still created. This moment was more distributed around the circumference of the gate as opposed the point contact achieved through the un-ground cut-to-length spring. Grinding the spring did heat it considerably, so this may have also caused a change in its temper.

Currently the most expensive part incorporated in our design is the actuator. The actuator that we chose to use was based primarily on its performance and availability. Since the actuation distance that was desired was not available to us readily, we modified our design to accommodate the actuator. In large batches this particular model is more expensive than we are able to utilize. We needed to use this model as it was available within our budget and time-frame, but the casing was made of stainless steel and it is one of the manufacturer's best models.

13. Recommendations

The valve pressure drop and leakage were the main performance issues encountered in testing at pressure and temperature. The possible causes of these performance weaknesses concerning the components are addressed above in the design critique. Table 9 shows the application of these recommendations for use in production of a second phase prototype, incorporation in a limited run manifold, and mass manufacture to the components of our design.

Actuation of the valve member is important to our valve achieving its primary function. To reduce sticking we recommend that the main passage be reamed or bored out to ensure a good surface finish and to eliminate localized widening/constriction. This will help to ensure that the sides of the gate will be able to pass smoothly through the passage. It will also decrease the high aluminum-on-aluminum frictional effects. To keep the components of the valve member in-line with one another during assembly, a specialized jig should be utilized.

In addition to this it should be ensured that the entire valve member is co-axial after assembly as this could cause problems in the installation of the valve member. In addition to installation issues it could also cause the outer wall of the gate to run into the wall of the main passage. Therefore great care must be taken in the assembly of the disk, actuator, and plug and gate. To reduce this, a guiding mechanism could also be employed to keep the member in line during operation. We chose to omit this to reduce cost.

The spring was a major issue, therefore it will require modification. The spring should be mechanically attached to the valve member. Mechanical attachment is necessary as any addition of heat, such as welding, could change the temper of the spring. Ideally the spring could be attached to the leading end of the retaining disk and the back end of the gate. In this way, it could reduce the overall length of the valve. Also, placing the spring below the plug and gate will allow the dimensions of the housing to be changed such that the leading edge of the gate will be able to be actuated to a flush position with respect to the constriction the plug fits into. This will increase seal performance when fully actuated.

The overall dimensions of the valve should be increased as the localized constriction points in the plug and gate, retaining disk, and entryways to the outlet of the valve cause too much restriction. Ideally, the dimensions of the valve should be be increased until these cross-sectional areas match the inner diameter of the incoming line. In this way, pressure loss could be greatly reduced, but this will increase the material cost of the valve as a whole. Reducing couplings will have to be used at the inlets and exits of the valve to maintain compatibility with the system. However this may not be possible due to spatial constraints within the vehicle as well as cost considerations (both monetary and weight costs to the entire system).

We were constricted to the use of a semi-custom elongated actuator design due to the small internal geometry of our valve. This was done to reduce flow constriction. A side benefit of the valve enlargement is that the actuator could then be changed to a larger, more standardized model. This actuator change could possibly counteract the material cost increase in mass production. Along with this we recommend that the actuator be selected such that it has a longer length of actuation such that the entryway to the radiator outlet can be widened and a lower pressure drop be achieved.

For another prototype to be manufactured we recommend that the prototype dimensions (housing, plug and gate, and retaining disk) be enlarged to the point where the constrictional cross-sectional flow areas match the cross-sectional area of the incoming line. This will reduce head loss due to fluid constriction. We also recommend that the threaded connection from disk to housing be maintained so the member may be removed for inspection/servicing. However, to keep the retaining disk in place we recommend that locking disks with threading on their outer diameter be used to keep the disk in place. The spring should be mechanically attached to the valve member between the bottom edge of the gate and the leading edge of the retaining disk to reduce the valve length as well as ensure that the spring cannot flip edgewise. Due to the larger dimensions of the second round prototype, a more standard actuator with longer actuation distance could be used as the cross-sectional area with respect to the incoming flow would not have to be reduced beyond non-standard sizing. Also a small tool is very helpful in assembly and disassembly of the threaded pieces (we used a bent 1/8” rod).

For integration into limited run manifold we would recommend the same changes as mentioned above with the exception of the housing dimensions as only the internal dimensions will have to be maximized (the outer dimensions of the housing are incorporated into the manifold). The threading should be kept the same such that threading installation into a manifold will be simpler than pressing. Also, since the valve is incorporated into a manifold this threading will allow installation of a new valve member if any issues arise (also for inspection purposes) such that a new manifold will not be required in the case of valve failure.

In mass manufacture the prototype should be streamlined. The excess material created through the use of a solid block of material should be eliminated. This will be simple since the housing is intended for casting. The elimination of this material will reduce final production cost, as well as the spatial obtrusiveness of the valve when incorporated into the system. In addition to this elimination, the internal threading of the housing for use in valve member installation will be eliminated. The threading on the outside of the retaining disk should also be eliminated. We recommend that the retaining disk instead be pressed into the housing as this is a more cost-effective method of installing the valve member.

Table 9: Summary of recommended design changes for prototype, manifold and mass manufacture production runs

	Prototype	Manifold	Mass Manufacture

Housing	Increase dimensions to allow less head loss (internal and external) and accommodate larger valve member,	Increase internal geometry dimensions	Increase internal dimensions, eliminate extra material outside of valve geometry, eliminate retaining disk threading
Spring	Mechanical attachment to valve member (between gate and retaining disk). Scale to fit valve member	Mechanical attachment to valve member (between gate and retaining disk). Scale to fit valve member	Mechanical attachment to valve member (between gate and retaining disk). Scale to fit valve member)
Plug and gate	Increase dimensions so flow areas match inlet size	Increase dimensions so flow areas match inlet size	Increase dimensions so flow areas match inlet size
Actuator	Longer actuation distance, larger more standardized design	Longer actuation distance, larger more standardized design	Longer actuation distance, larger more standardized design
Retaining Disk	Keep outside threading, use locking washers to hold in place	Keep outside threading, use locking washers to hold in place	Use tolerance fit for installation into housing
Inlet/Outlet Plumbing	Reducing couplings to accommodate larger valve geometry	Reducing couplings to accommodate larger valve geometry	Reducing couplings to accommodate larger valve geometry

For mass manufacture we recommend that the plug and gate be cast initially. Then turning the piece to achieve a good surface finish on the outer surface will be necessary to reduce frictional impact on the performance. The center hole of the plug will also have to be reamed to size such that the actuator piston arm may be pressed into place. The valve housing may also be cast and will require drilling and tapping processes at each inlet to allow coupling to the system. The main passage will have to be either reamed or bored to ensure a good surface finish as well as to reduce inhomogenieties inherent with drilling. This will also allow for a good tolerance fit with the retaining disk. In addition to this the hole at the re-circulating line restriction should also be reamed to ensure a good fit with the end of the plug. The retaining disk can be either stamped or extruded and may require another process to ensure a good press fit with the housing as well as a reaming operation on the center hole such that the actuator may be pressed into place.

14. Conclusions

The EPA requested that we design and build a prototype of a thermostatic valve for use in their Hydraulic Hybrid Vehicles (HHV). The valve must route hydraulic fluid to a cooling radiator or back to the HHV system to keep the fluid in the optimal temperature range (between 160°F and 180°F). In addition to proper flow diversion, it was important that the valve be safe, and be cheap to produce on a large scale.

In order to generate concept designs, we first held an extended brainstorming session. We then identified all of the functions of our design using a functional decomposition. This was followed by categorization of the subsystems involved in each of our preliminary designs. These subsystems were then quantified in terms of customer requirement satisfaction using a Pugh Chart analysis in order to determine the best possible subsystems to be integrated into more effective designs. These new designs were then rated with another Pugh Chart analysis in order to obtain the most successful final alpha design.

Our alpha design mechanism employed a gate design in tandem with a plug, and linearly actuated using thermally expansive material. This afforded us the required stream blockage when in fully actuated and normal positions of the valve member. Most components of the valve will be able to be cast for mass production, with some additional machining required, which allows for cost to be kept low.

From this alpha design we were able to define parameters that were available for optimization. Through thorough analysis of these parameters using materials science and fluid mechanics we were able to create an alpha design that has perfected performance. Our final design retained the form and function of our alpha design, apart from the optimized geometries that were before only chosen arbitrarily. When creating the final design we kept in mind the ultimate goal of our project, to produce a product for mass manufacture.

We machined our prototype as true to our optimized parameters as possible with the manual machines available to us. Due to limitations in machining capabilities, as well as availability of an optimal actuator on a short time-scale we had to change several small geometric components of our prototype. Our design project is unique in that we fabricated a full scale model of our final design. The main differences between the two lie in their actuation mechanism and methods of joining all components together. The prototype actuator will has a shorter stroke, and utilizes threading as opposed to press fitting the components together (this facilitates easy assembly and disassembly).

After researching the feasibility of many different validation methods, we decided to test the flow characteristics of our valve by measuring pressure drop and leakage between streams of air through the valve and using non dimensional scaling to predict the results for oil at the in service flow conditions. The EPA also conducted a test of the valve with their test stand that completely replicates the in service conditions. The air test resulted in results with values are likely invalid, but we were able to observe their trends. During the EPA's test, the valve failed, but many useful results were still obtained. The tests proved that the actuator functions in the required temperature range and that in the normal position, there is only about 2% leakage between streams. The measured pressure drop for the valve was higher than the 10psi goal, but was similar to other commercial valves.

The results obtained for testing allow us to offer recommendations to improve upon our design. By increasing the size of the valve and reducing areas for flow to stagnate, the pressure drop can be lowered. An actuator with a longer stroke should be obtained for a final production version. Replacing threaded interfaces with press fits will also reduce the cost of the valve in mass production. Although the valve failed part of the way through the EPA's test, we believe we created a successful prototype. The results of our validation prove that our design concept can be made successful with some optimization in size and more proficient machining.

15. Acknowledgements

Dr. Andrew Moskalik
Dr. Grant Kruger
Dr. Steven Ceccio
Bob Coury
John Mears
Todd Wilber
Dan Johnson

16. Information Sources

16.1 Benchmarks

Thermostatic valves are often used in industry for applications such as cooling automobile engines, temperature control of faucets, as well as home thermostat control. We have researched different valves already in production and have used this information to benchmark the operation and price of our valve.

Most thermostatic valves make use of a thermally expansive wax as means of actuation, while others are electronically actuated. Thermally expansive waxes are calibrated at the plant where they are refined. Many manufacturers stand behind their calibration and operating temperature ranges. Electronically actuated valves precisely control the valve based on constant temperature feedback from a thermocouple. Typically, electronically actuated valves are expensive since sensors, circuitry and computing power are needed to control flow as a function of temperature. The EPA is currently using a manually operated ball valve in their prototype vehicles, which requires the operator to monitor the hydraulic fluid temperature and manually divert the flow one way or the other. Not only is this an inefficient method because of the manual operation, the ball valves themselves are extremely costly.

The EPA has tried using an expanding wax actuated valve, a manual ball valve, and an electronically actuated valve. They were completely dissatisfied with the performance and price of these valves. The wax actuated valve did not operate at the required temperatures, and when all of the flow was supposed to be diverted to the cooler, approximately 50% of the total flow would leak back into the system rather than being diverted. The leakage could be attributed to bad seals within the valve. It is unknown whether fluid leaked into the sleeve containing wax and reacted changing the properties of the wax. The manually operated ball valve is not an automatic thermostatic valve and is inefficient and ultimately is an unacceptable option because a person must constantly monitor the temperature and physically adjust the valve. The ball valves themselves are extremely costly, making them, as well as the electrically actuated valves, impractical for use on a commercial version of the hydraulic hybrid system.

All manufacturers stand behind the claims of their products, and thermostatic valves are successfully used for many applications. Two manufacturers whose valves particularly stood out based upon product specification sheets and their specified use with oil were Fluid Power Energy, Inc. (FPE) and ThermoStasis. Both valves are reported to have a pressure drop of less than 10 psi across the valve with oil flow rates of up to 35gpm, which is within the required range for the EPA's specifications. Despite the wide use of thermostatic valves, we can improve upon the following important aspects: adhering to the required temperature range, completely diverting flow with minimal leakage, and reducing commercial cost.

Electronically actuated valves can cost more than $500 while most wax-actuated thermostatic valves retail between $100 and $200, yet thermally expansive wax thermostats themselves, excluding the valve's housing, retail for $5 to $40. Considering these retail prices, one can conclude that the valve housing is a very large portion of the valve's price. While designing our valve we will keep low production price as a priority, so that a commercial version of our valve will cost significantly less than those already on the market.

16.2 Patents

Thermostatic valves have been in use for a long time and as a result many different variations of this idea are patented. We found that most patented valves use the common idea of a conventional spring that applies a normal bias to the valve member at normal temperatures and a thermally reactive material that acts against this spring as temperature fluctuates.

Although improving on an existing technology will give us preliminary design ideas, it also serves to limit the design space in which we are operating as many variations of this type of valve already are patented.

US Patent 6,719,080 B1

This patent contains the most current version of information regarding hydraulic hybrid vehicles and was very helpful its emphasis on the underlying principles motivating our project. The assignee on the patent is the Environmental Protection Agency, making this information even more relevant to our work. Particularly this patent contained very helpful schematics of the series hydraulic hybrid system and detailed information on how all of the components work together [5].

US Patent 6,834,737 B2

This patent contains more information on how energy is harnessed and stored in hydraulic hybrid vehicles. It was helpful to review this information as it served to reinforce our understanding of the hybrid hydraulic technology that we are working with [1].

U.S. Patent 5,261,597

This patent describes a three-way valve that automatically responds to temperature fluctuations in the operating fluid. In this application the valve is operated by two springs, one conventional spring that biases the valve member to a position allowing flow through one stream, and one shaped memory effect spring (a spring made of an alloy that will change into a predetermined shape at a certain temperature) that will apply a force to the member opposite that of the conventional spring as temperature fluctuates. Therefore at higher temperatures the valve will redirect more flow to the other opening [10].

U.S. Patent 3,237, 862

This patent is for a temperature actuated valve that operates in-line with its system, designed particularly for use with a gas (in this case air) in delivering a specified amount of thermal energy. The operating fluid and ultimate purpose of this valve are separate from our design problem, but the principles remain the same. This patent also addresses issues involved with making the valve mechanism aerodynamically balanced (in our case it will have to be hydrodynamically balanced) so as to eliminate the need for adaptation to pressure variation about the valve member. This mechanism also incorporates a conventional spring that biases the valve member to a certain position at normal temperatures. The temperature actuating mechanism in this particular valve is based on a stack of dish shaped bimetallic discs that flatten when their temperature is raised thereby allowing the spring to actuate the valve member [4].

U.S. Patent 4,285,467

This patent is for a three way temperature actuated valve design and it provided us with another actuation mechanism. This valve is implemented in automobiles and is intended for the diversion of exhaust gas within an engine. The thermal actuator used in this case is a thermally expansive material such as wax refined with metallic (copper) flakes. This mixture allows for actuation in a relatively narrow predetermined temperature band [6].

16.3 Articles

During our literature review we also found several articles explaining the underlying concepts of series hydraulic hybrid technology and its application to delivery style trucks. These articles also allowed us to define the scope of our project on a mass production scale and the impact that it will be able to have when implemented in these systems.

Energy Conservation with Thermostatic Control Valves

This journal article from 1979 demonstrated that new applications of thermostatic control valves are not an emerging technology. This article weighed the cost to benefit of retrofitting home heating systems with thermostatic control valves. We were provided with a parallel engineering study of quality versus cost [7].

Manufacturing Climate Solutions

This article discusses the mechanisms of hybrid technology applied to delivery style trucks (both electric and hydraulic hybrids). This article was specific with actual commercial delivery truck fleet sizes (up to 70,000 vehicles for some companies). It was extremely helpful in defining the actual scope that our valve design will have to encompass with regard to its intention for mass production [3].

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18. Team Biographies

Eric Haapaniemi was born and raised in Farmington, Michigan, a suburb of Detroit. He has always had an interest in science and math, particularly physics which is what motivated him to pursue a degree in mechanical engineering. He has had a long interest in environmentalism sparked by his early affiliation with the Boy Scouts and has worked on several sustainable projects. These include the design and implementation of a rainwater collection system to help a community gardening center get off the grid, as well as a design to put piezo-electric crystals in shoes to harvest energy from walking. He has a love for the outdoors and enjoys camping, canoeing and hiking in every season, and as a result hopes to someday hike the Appalachian Trail and complete the Hunter's Island Loop in Quetico Provincial Park. Eric has been growing his hair out since high school with the intent of donating it to an organization that makes wigs for children who have experienced a loss of hair due to many different reasons. He plans to pursue a higher degree, but is not sure what field this will be in, as of yet.

Sarah Markey was born and raised in Ann Arbor, MI. She had always enjoyed the sciences, especially physics which led her to follow in father's footsteps and pursue a degree in mechanical engineering. She is a member of Engineering Global Leadership and will be continuing at Michigan in the fall of 2011 in pursuit of her masters in mechanical engineering. Upon completion, Sarah is considering applying for the Peace Corps before she tries to find a job in industry. This year she is living in Luther House, a cooperative on campus with fellow team member Eric. She spent this last summer in Luneburg Germany, improving her German language skills and learning more about the culture. Sarah's parents met as students at U of M and now she and her younger sister, Annie both attend. Sarah and Annie are trying to convince their other siblings Kate and Myles that Michigan is the university for them. She is very excited about working with the EPA and learning more about green technologies.

Andrew Jessop is a senior in mechanical engineering, with a minor in German. He has lived in Sylvania, Ohio for his whole life and attended high school at St. Francis de Sales High School, in Toledo, Ohio. He always enjoyed math, science and taking apart everything he could get his hands on, so he chose to do engineering at the University of Michigan. Reading about high performance sports cars and traveling are some of his favorite activities. He completed an internship with BMW in Regensburg, Germany, where he worked in the quality department of the body shop. Most of his time was spent on projects for the doors of the Z4 Roadster. He plans on either finding a job for after graduation or completing a Master's in mechanical engineering at the University of Michigan.

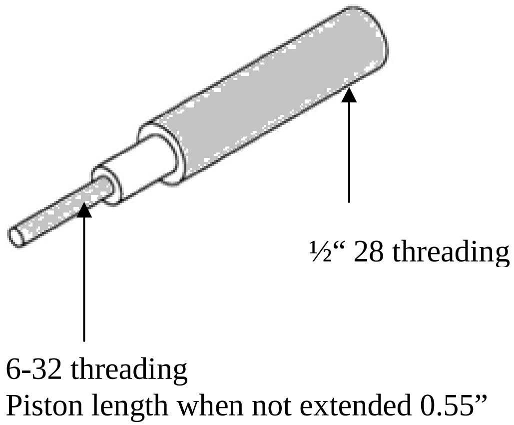
William Blumberg was born on June 3, 1989 in Hollywood, Florida, and moved only 2 years later to the neighboring town of Plantation, Florida and lived there until he moved to Ann Arbor for his freshman year. His father graduated from the University of Michigan in 1970 with a similar Mechanical Engineering degree, which started him out on the Wolverine trail at an early age. His dad clearly tried to mold him into an engineer from the start, teaching him math and always giving him problems to solve, whether he was out to dinner or just sitting in his room, from as long as he can remember back to. He never had any specific interests in mechanical engineering growing up, but knew he was good at math, not very good at English, and that engineering was the most profitable path to follow at the time. Even after divulging into the world of engineering for a few years, his future is still as hazy as his decision to go into engineering. Realistically his future will involve a Masters degree in engineering, a few years of field work, and then a return to school to get a MBA. The reason for the business degree stems from his belief that there is no reason to be paid less or ranked lower than the person in charge of all the engineers if that person isn't even qualified to do what the engineers are doing; thus, he plans to continue school until the job market is on the rise and unless he finds an engineering job he can see himself doing forever, he'll go to business school to become a boss.

Appendix A Bill of Materials

Item Description	Vendor	Product #	Unit Price	No. needed	Total Price
Thermal Actuator	Therm Omega Tech	TOT-O4L	$80.00	1	$80.00
Aluminum Block	Alro Metals Plus	n/a	$8.25	1	$8.25
Cut Alu. Block	Alro Metals Plus	n/a	$5.00	1	$5.00
Aluminum Round Stock	Alro Metals Plus	n/a	$11.00	1	$11.00
11" Cut-to-Length Compression Spring (5pk)	McMaster Carr	9637K25	$10.59	1	$10.59
6-32 Threaded Rod (12")	Stadium Hardware	n/a	$0.59	2	$1.18
1"Schedule 40 PVC (10')	Home Depot	611942066643	$2.71	2	$5.42
1" PVC Male Adapter	Home Depot	012871626050	$0.53	3	$1.59
1" PVC Union	Home Depot	032888646353	$4.86	2	$9.72
PVC Redcucer Fitting (2" to 1.5")	Home Depot	012871559488	$1.18	1	$1.18
PVC Reducer Bushing (1.5" to 1")	Home Depot	012871626630	$0.97	1	$0.97
PVC Primer and Glue Pack	Home Depot	038753302485	$6.96	1	$6.96
U-Bolt	Home Depot	030699095063	$1.24	6	$7.44
2" Schedule 40 PVC (2')	Home Depot	611942109463	$3.36	1	$3.36
Stainless Steel Hose Clamps	Stadium Hardware	n/a	$0.79	4	$3.16
1" Flexible Hose (price/foot)	Stadium Hardware	n/a	$2.49	3	$7.47
1" Barbed Poly. Fitting	Stadium Hardware	n/a	$0.69	2	$1.38
1" PVC Female Adapter	Stadium Hardware	n/a	$0.59	2	$1.18
Plumber's Tape	Stadium Hardware	n/a	$1.49	1	$1.49
Water Jet Usage (price/hour)	ERC/RMS	n/a	$88.03	3.833	$337
Subtotal					$504.76
Tax					$4.61
Shipping					$4.36
TOTAL					**$513.73**

Engineering Change Notice

WAS:



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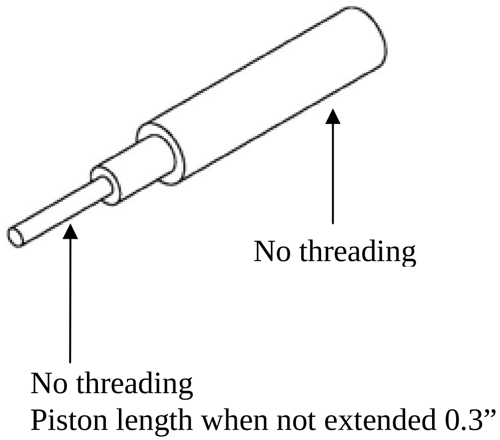
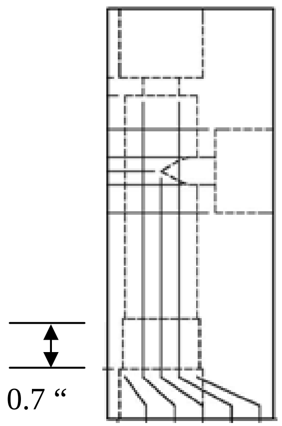


Notes: Custom actuator was initially miss quoted and out of our budget. Off the shelf actuator had to be used

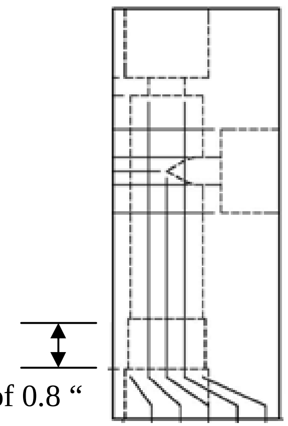
TEAM 18	
PROJ: THEMOSTATIC VALVE	
REF DRAWING : ACTUATOR	
Eng: Andrew Jessop	11/29/2010

WAS:



Thread depth of 0.7 “

IS:



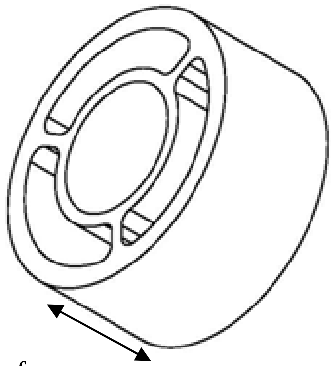
Thread depth of 0.8 “

Notes: Due to changes in the actuator design we increased the depth of our threads to allow for more variability in the assembly.

TEAM 18	
PROJ: THEMOSTATIC VALVE	
REF DRAWING : HOUSING	
Eng: Sarah Markey	11/30/2010

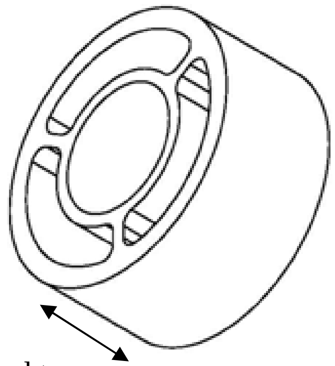
Engineering Change Notice

WAS:



Width of
0.500”

IS:



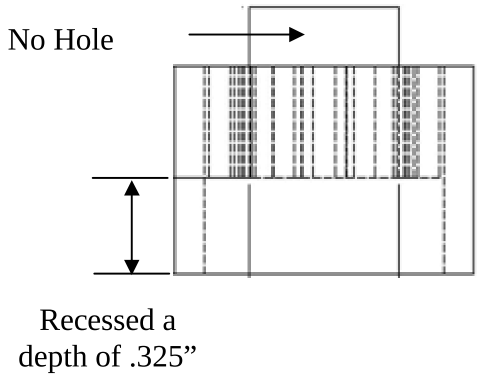
Width reduced to
0.420”

Notes: Due to changes in our manufacturing plan we were unable to expose enough length of our blank on the lathe. The retaining disk had to be shortened in order to allow the 1/8” parting tool room to move and not hit the lathe chuck.

TEAM 18	
PROJ: THEMOSTATIC VALVE	
REF DRAWING : RETAINING DISK	
Eng: William Blumberg	12/6/2010

Engineering Change Notice

WAS:



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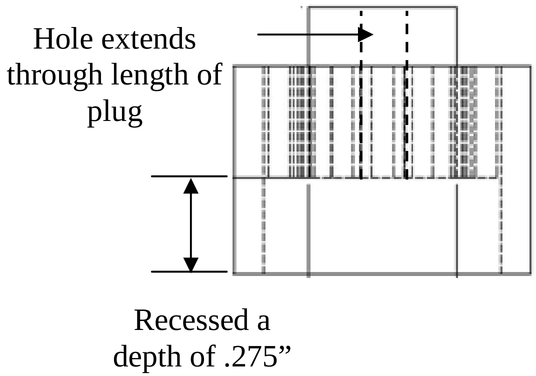
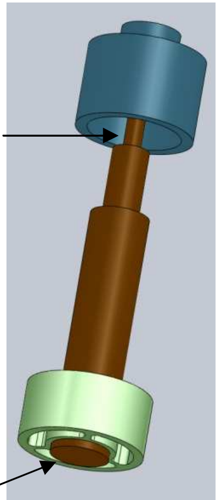


Notes: Due to shortening of actuator piston, space between Plug and Gate and actuator body eliminated. See Engineering Change Notice: Assembly. A thru-hole need to be made due to manufacturing limitations

TEAM 18	
PROJ: THEMOSTATIC VALVE	
REF DRAWING : PLUG AND GATE	
Eng: William Blumberg	11/30/2010

WAS:

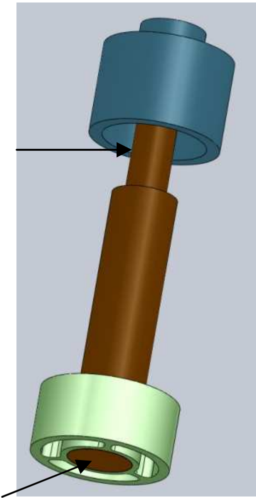
0.05” clearance between actuator body and plug recess



Actuator sticks out the back of the retaining disk 0.1”

IS:

No clearance between actuator body and plug recess



Actuator bottom is flush with retaining disk

Notes: Due to shortening of actuator piston the plug had to be pressed onto the piston as far as possible. Because we had to press fit the actuator body into the retaining disk the two parts had to be flush.

TEAM 18	
PROJ: THEMOSTATIC VALVE	
REF DRAWING : PLUG AND GATE	
Eng: William Blumberg	12/2/2010

Appendix C Design Analysis

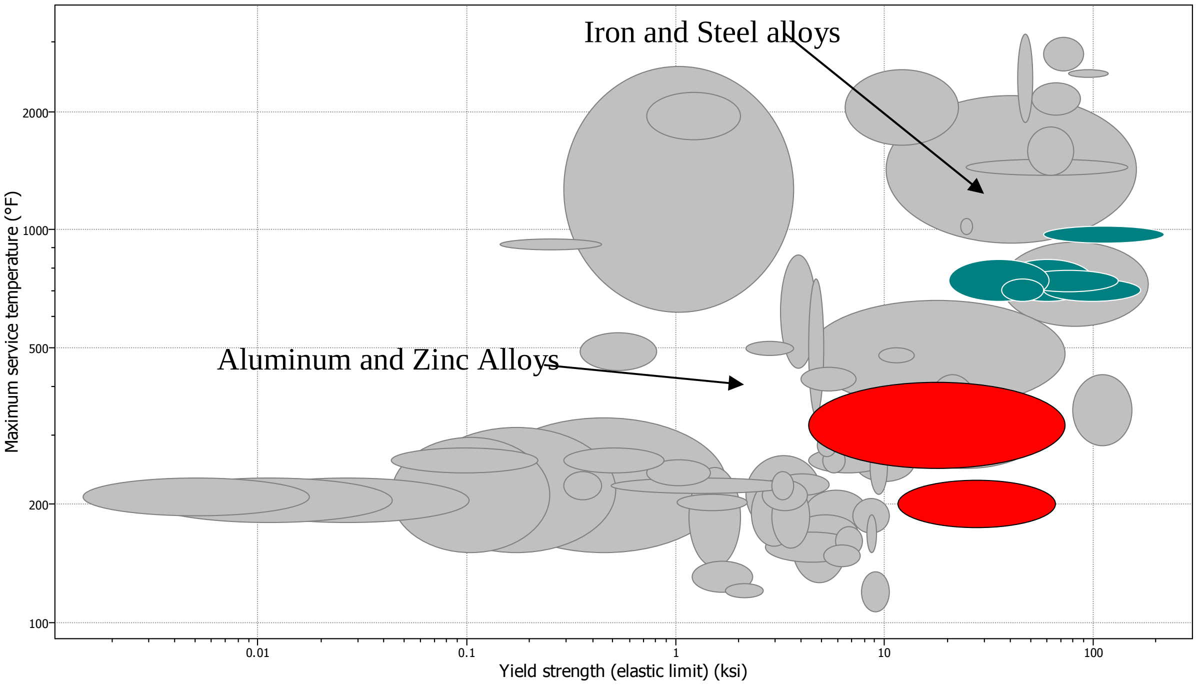
Material Selection- Functional Performance

We have access to the Ashby's Material Selection Database through software titled CES EduPack 2010 [25]. This program, as well as outside research and knowledge, greatly assisted in our material selection process.

This CES software assisted in the material selection of two major components of our final design, the Housing and the Retaining Disk. The two main functions of our housing is to properly hold our actuator in position as well as be used for testing validation. The first function provided little to no constraints in our material selection process apart from machinability of the material. Testing validation, however, would require that the Housing be exposed to high temperatures, pressures, and forces; thus, based off of the limits one can choose from, which refine the material selection process in the CES software, we used yield strength and maximum service temperature. The temperature of the hydraulic fluid passing through the housing can reach 180□F during operation so we had to ensure the material wouldn't be warped or affected through 10^5 cycles. Similarly since the system is pressurized and large forces will be pushing against this housing at all times we had to ensure the material's strength would be able to withstand the forces that will ensue throughout operation. As a result the CES software showed that aluminum, iron, steel, and zinc alloys as the top results. Since any of these options would suffice we had to somehow narrow these choices down to a final one. Considering material price, manufacturability, and ease of obtaining the material, and since the CES software gives prices per pound of the material as well as the hardness and strengths which go into the machinability, we ultimately chose aluminum as our final material selection for the housing. Figure 32 below shows the results from the CES software showing yield strength vs. maximum service temperature.

We then used this CES software in order to obtain proper materials for our Retaining Disk. The main functions of our retaining disk is to hold the actuator in place via a press fit, screw properly into the housing so the Plug and Gate properly was aligned before instillation, and to block the least amount of flow. All three of these functions relate back to manufacturability since all functions can be obtained if the part is properly bored, threaded, and cut via water jet. Since this part will also be experiencing the same high operational temperature, pressure, and forces we used the same limits for our initial material refinement as with the housing, maximum service temperature and yield strength. Therefore, the results we obtained were the same and can be represented by Figure 32. Also similar to the housing, we then narrowed these choices down to a final material by considering mainly the machinability and price of the materials and in the end decided to use aluminum just as we did with the housing. In the end, aluminum was the material chosen for every part we would manufacture

Figure 32: Material Selection Graph for Testing Housing, Plug and Gate, and Retaining Disk



Environmental Performance

In order to determine the mass of the material needed in our final design we referred to Solidworks 2010. With the density of our aluminum alloy 6061 known we solely used Solidworks to calculate the volume of our machined parts and multiplied by the density, leaving the mass. The masses and volumes of the material needed for our final design of each of the parts can be seen below in Table 10.

Table 10: Volumes and masses required for machining

Part	Density (lb/in^3)	Volume Before Machining (in^3)	Volume After Machining (in^3)	Mass Before Machining (lb)	Mass After Machining (lb)
Housing	0.098	19.665	13.87	1.9272	1.3592
Retaining Disk	0.098	0.441	0.18	0.0432	0.0176

Using SimaPro 7.2 [27] we were able to determine the environmental effects the materials used in our final design will have. The only parts we actually manufactured, however, were formed from aluminum; thus, we decided to compare the material used for our final prototype's housing to the materials that would be used for the mass manufactured version of our housing. Therefore using the possible categories given by the Professional Database of SimaPro and the EcoIndicator 99 associated with the program, we compared the environmental effects of 1.9272 lbs of 6061 aluminum against 1.9272 lbs of cast aluminum. Using the Professional Database allowed us to see the full life cycle of our materials including the production, transportation, disposal and other parts of the process which ultimately results in creating the two different materials. Results in the form of four bar graphs can be seen in Figure 33 through Figure 36 below.

Figure 33:Total mass of raw materials, air emissions, water emissions and solid waste created

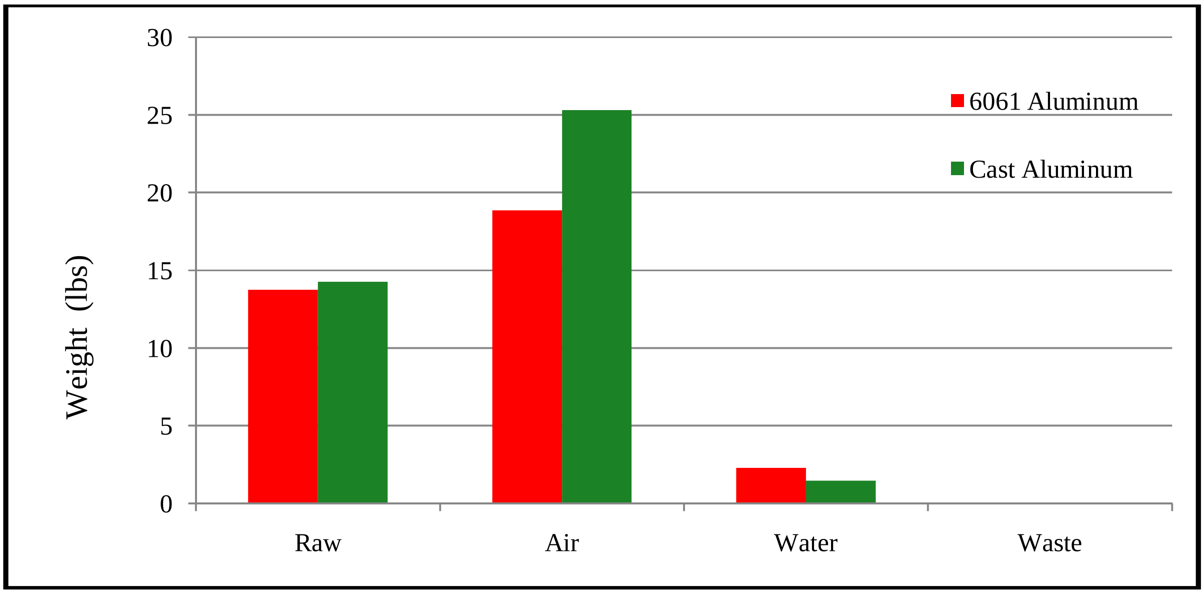


Figure 33 shows the total mass of raw materials, air emissions, water emissions, and solid waste created in order to form 1.9272 lbs of either material as created using the EcoIndicator 99. From this graph one can see that there is almost as much mass of raw materials used in the creation of either product; thus, the environmental effects caused by using raw materials will be approximately the same. The amount of air emissions created during the production of cast aluminum can be seen as approximately 25% greater than that of the 6061 aluminum, clearly demonstrating cast aluminum having a larger impact on the environment. However, water emissions, while both low values (< 2.5lbs), are about 40% larger for 6061 aluminum. Finally, while cast aluminum had absolutely 0 lbs of waste, 6061 aluminum solely had 0.0113 lbs of waste; therefore, the environmental impact can be almost seen as irrelevant. Based off of this figure cast aluminum would be slightly more hazardous for the environment; however, if water emissions is much more important than air emissions, which would require more research, it is possible that they have an equal impact, or even possibly that 6061 aluminum has a larger impact.

Figure 34: Damage classifications and their impact on the environment

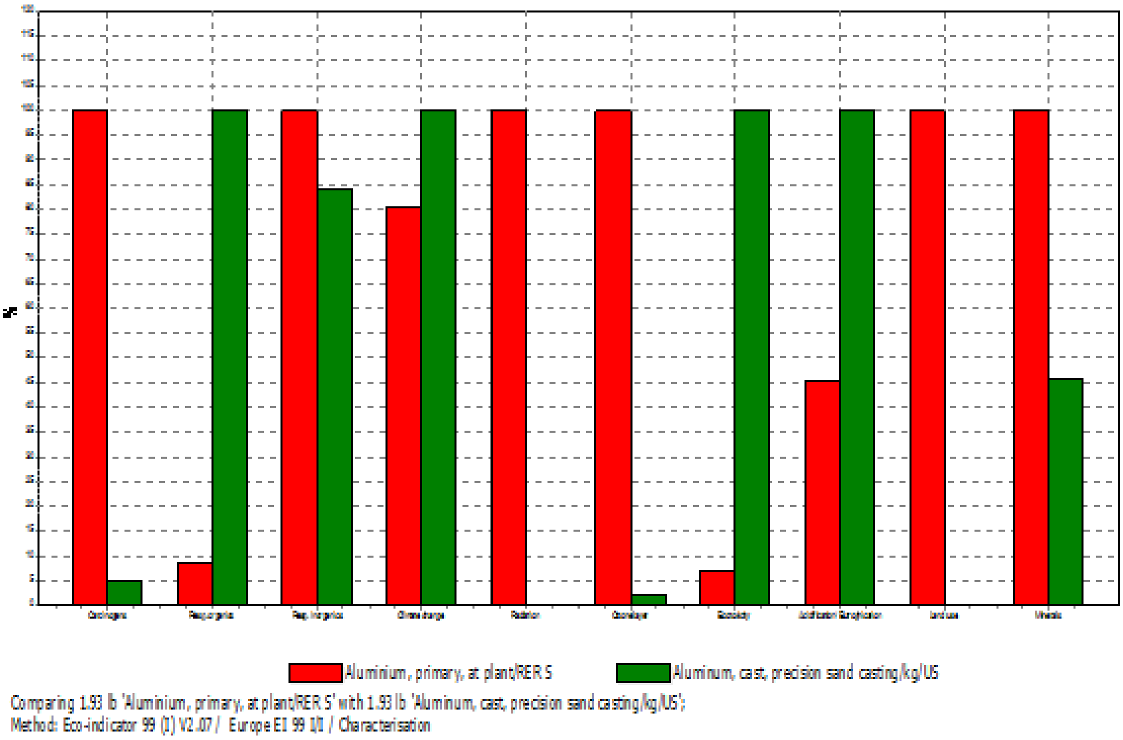
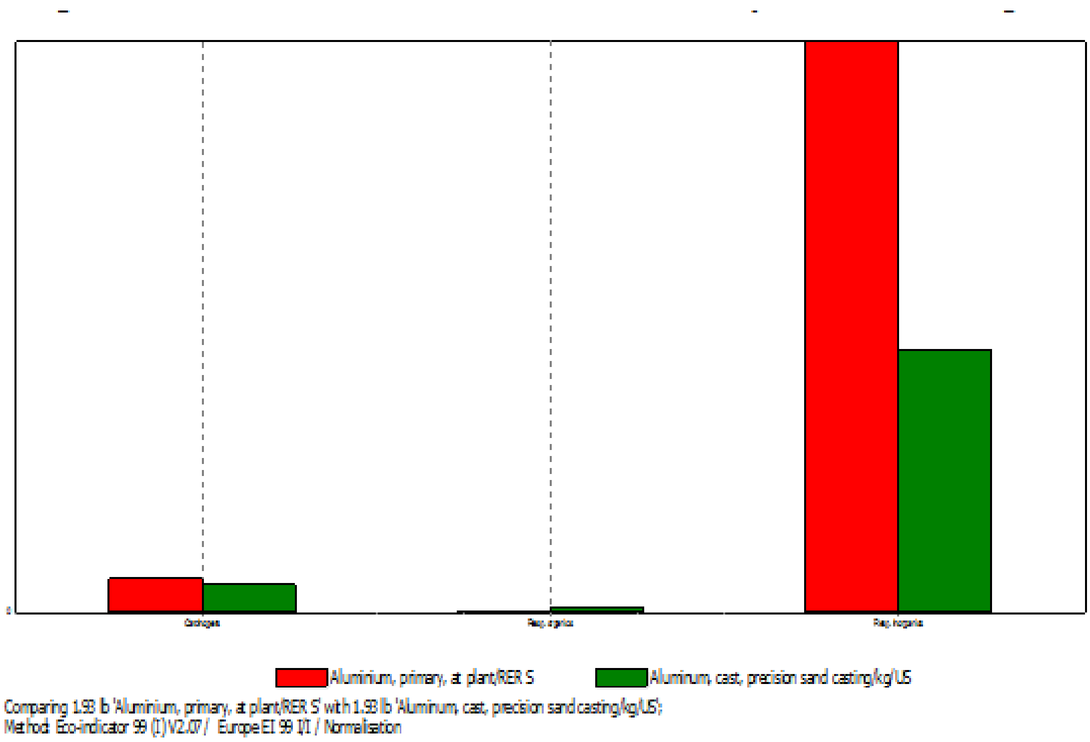


Figure 34 shows each of the EcoIndicator 99 damage classifications and their impact on the environment. Of the 10 categories the EcoIndicator 99 looks into during the full life cycle of our two materials, 6 are greater in the production of 6061 aluminum while 4 are greater in cast aluminum production. 6061 aluminum has a bigger impact on the environment in terms of carcinogens, respiratory inorganics, radiation, ozone layer, land use, and minerals. Cast aluminum more greatly impacts the environment in terms of respiratory organics, climate change, ecotoxicity, and acidification/eutrophication. Based on this damage assessment, 6061 aluminum will seem to have a greater impact on the environment during its entire life cycle.

Figure 35: Normalized score in human health, eco-toxicity, and resource categories



Based off of the 10 EcoIndicator 99 damage assessment categories, SimaPro breaks these into three damage meta-categories-human health, ecotoxicity, and resources. By looking at the weighting section in SimaPro, one can clearly see by Figure 35, the normalization of the damages, that resources will be the largest concern, followed by human health, and then barely any ecosystem quality problems. One can also see this by looking at the point values associated with the weighted values in SimaPro. For 6061 aluminum, there is a point value of 3.6 for resources, 0.6 for human health, and 0.01 for ecosystem quality. On the other hand, for cast aluminum there is a point value of 1.65 for resources, 0.5 for human health, and 0.05 for ecosystem quality. While ecosystem quality is slightly higher, the much larger use of resources and the slightly larger human health concerns created during the full life cycle of 6061 aluminum would cause us to believe it to be the more environmentally impacting material. This contradicts what we would have believed based off of looking at Figure 33because that graph solely shows the mass of the emissions and materials used; thus, since these would be different in the creation of 6061 aluminum and cast aluminum Figure 35 better represents the environmental impact of the entire life cycle of each material.

Figure 36: Single score comparison in "points"

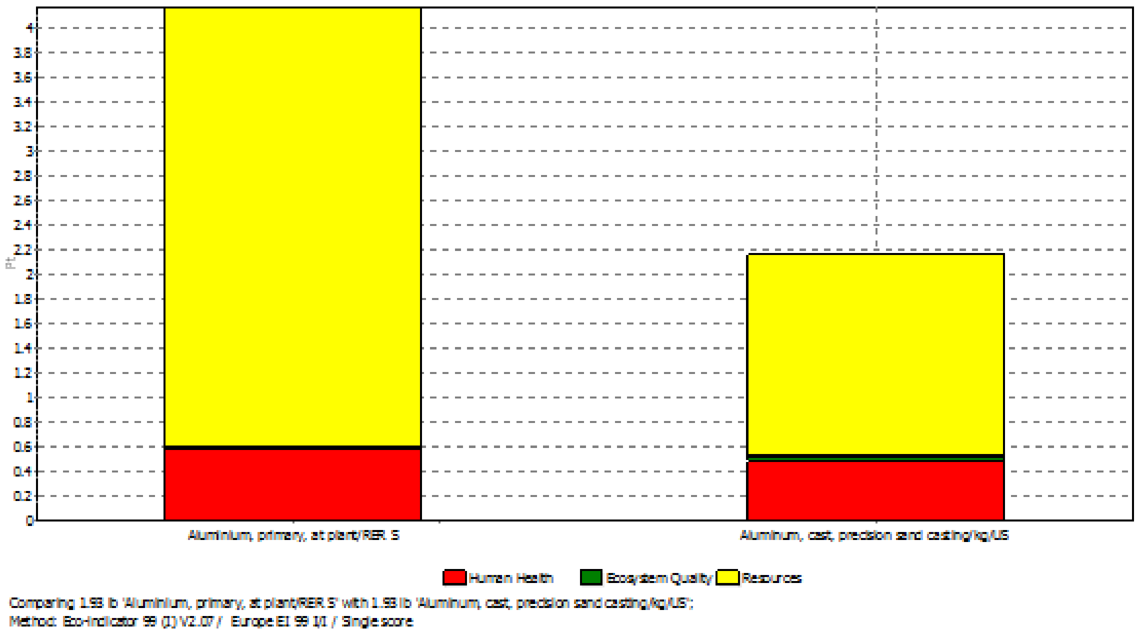


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Figure 36 further develops the idea that 6061 aluminum will have a greater environmental impact during its full life cycle then cast aluminum. This graph takes the point values in the weighted tab of SimaPro and stacks them atop one another. Once again it would seem that human health and ecosystem quality are roughly the same (<0.1 pts.) but the resources used during the life cycle of 6061 aluminum is almost double that of cast aluminum which would be enough to express that SimaPro with the use of EcoIndicator 99 will have a much bigger impact when the life cycle of the whole product is considered.

Manufacturing Process Selection Assignment

Our thermostatic valve is intended to be used in new hydraulic hybrid vehicles. This will include UPS delivery trucks, garbage trucks, and many other types of large vehicles that regenerative breaking would make much more fuel efficient. Therefore, our production volume will be 10,000-25,000. In order for the EPA to be convinced that they should invest in that magnitude of our thermostatic valve, we have to prove it is cost efficient in mass production.

In the Material Selection Section above, we used the CES EduPack 2010 software to deduce that aluminum was the best material for both the Housing and the Retaining Disk. Once this had been selected we were able to use the CES Manufacturing process selector by following the Process Universe link at the bottom of the Aluminum alloy section. From here we were able to look at different manufacturing processes that would greatly reduce cost at a production value on the magnitude of 10^4 or larger.

For our mass manufactured housing we wouldn't need the tolerances or the surface finish to be that great because if there was a process that would easily create a hollow 3-D geometry we could simply drill holes later to better make the tolerances, if it was even necessary. For this we looked at both die casting and sand casting since both could do hollow 3-D geometries. Die casting had smaller tolerances, 5.91-19.7 thou vs. 31.5-118 thou, but since tolerances weren't our biggest issue we continued to compare the techniques. Both of these could produce the batch size of the magnitude we wanted so there wasn't any solution there. The tooling cost and equipment cost for die casting was much higher than for sand casting; however, the labor costs were reversed with sand casting being much higher than die casting. Therefore we took a look at the possible thicknesses that each of these processes could perform. Die casting could create a section thickness of 19.7-472 thou while sand casting could go from 118-3.93e4 thou. Since the housing is going to be larger than 472 thou, we deduced that sand casting would be our best method for mass manufacturing of this part and if the tolerances are too extreme we would simply drill the holes to the proper diameter.

Mass manufacturing the Retaining Disk would require a manufacturing process that had very small tolerances and could create a circular part that can be press fit into the housing. After looking through the possible options the CES Manufacturing process selector had to offer, extrusion and stamping stood out for the mass production of this part. Either of these techniques could properly create the geometry of the retaining disk so other attributes were examined. Stamping had a slightly lower tooling and equipment cost than extrusion while labor costs were very similar, but this wasn't enough to choose stamping over extrusion. The tolerances during stamping range from 3.94-31.5 thou while tolerances from extrusion are from 0.984-19.7 thou. Since we ultimately plan to have this press fit in, it would make more sense to go with extrusion since if we can get the part to be under a thou as is there could be no extra machining involved such as turning in order to get the disk to size. This alone could save more money than the slightly higher tooling and equipment costs combined. Also, the section thickness during a stamping process maxes out at 197 thou, which is smaller than our actual part, while one could extrude up to 3.54e4 thou. Being able to extrude this length as well as possibly have perfect tolerances after the process could save an extreme amount of time and money since one could simply extrude the maximum distance and cut the piece to length many times since our part is approximately 500 thou long.

The CES Manufacturing process selector that accompanies the CES EduPack 2010 software greatly assisted in finding new machining processes that could easily create our parts on a mass scale. This would be necessary since our prototype cost around $100 and we were requested that it should be much cheaper. These processes used at this production volume will greatly decrease cost and time and hopefully allow us to create a final mass produced thermostatic valve that actually costs less than the $10 our sponsor and the EPA were hoping for.

Housing

Process: Rough cut correct length from stock with bandsaw, face ends with facing tool, drill two holes, ream three holes, tap three holes

Stock: 6061 Aluminum 1.5" X 6" X 3"

Machine	Fixturing	Machine Setup	Tooling	Feed Rate (ipm)	Cutting Speed (rpm)	Process Description	Time (mins)	Safety Concerns	Additional Notes
Step 1									
Band Saw	N/A	Turn machine on, set speed	N/A	N/A	See band saw chart, choose steel	Measure and score desired length and width on stock. Manually push piece through, cutting larger than marks	10	Use wood block to push piece	
Step 2									
Mill	Normal straight mill vice with parallels	Turn Machine on, set speed	¾" 2 flute end mill	Manual feed (slowly)	2300	Face mill 3 edges so working surfaces are all parallel	15	Normal Mill Safety	

Step 3									
Mill	Normal straight mill vice, use parallels	Turn machine on, set speed	Edge finder, center drill, 15/32" drill bit (2 flute)	Manual feed (slowly)	2300 (center drill) 1400 (15/32")	Drilling hole: Use edge finder to center over hole, center drill, drill	30	Normal Mill Safety	Operation for Main Branch Outlet, Peck drill to remove chips. Drill a little more than ½ way through
Step 4									
Mill	Normal straight mill vice, parallels	Turn machine on, set speed	1/2" Ream	Manual feed (slowly)	100	Reaming hole: Center over hole, ream	15	Normal Mill Safety	Operation for Main Branch Outlet
Step 5									
Mill	Normal straight mill vice, parallels	Turn machine on, set speed	1" drill bit (2 flute)	Manual feed (slowly)	650	Drilling hole: Center over hole, drill to depth of 1.2"	20	Normal Mill Safety	Operation for Main Branch Outlet, Peck drill to remove chips
Step 6									
Mill	Normal straight mill vice, parallels	Turn Machine on, set speed	Boring bar	auto feed	490	bore to 1 5/32"	30	Normal Mill Safety	Increase cuts by .01"

Step 7									
Mill	FLIP block over to parallel side Normal straight mill vice with parallels	Turn Machine on, set speed	¾” 2 flute end mill	Manual feed (slowly)	2400	Face to length	15	Normal Mill Safety	
Step 8									
Mill	Normal straight mill vice, use parallels	Turn machine on, set speed	Edge finder, center drill, 1" drill bit (2 flute)	Manual feed (slowly)	2300 (center drill) 1400 (1”)	Drilling hole: Use edge finder to center over hole, center drill, drill	30	Normal Mill Safety	Operation for Main Branch Outlet, Peck drill to remove chips. Drill to a depth of 4"
Step 9									
Mill	Normal straight mill vice	Turn machine on, set speed	1" 4 flute end mill	Manual feed (slowly)	80	End milling: Center over hole, mill	20	Normal Mill Safety	
Step 10									
Mill	Normal straight mill vice, parallels	Turn Machine on, set speed	Boring bar	auto feed	490	bore to 1 5/32"	25	Normal Mill Safety	Increase cuts by .01"

Step 11									
Mill	Normal straight mill vice, flip stock 90 degrees to machine side branch	Turn machine on, set speed	Edge finder, center drill,1" drill bit (2 flute)	Manual feed (slowly)	650	Drilling holes: Use edge finder to center over hole, center drill, drill	20	Normal Mill Safety	DO NOT pierce sidewall of main branch. Leave block in position for milling operation
Step 12									
Mill	Normal straight mill vice	Turn machine on, set speed	1" 4 flute end mill	Manual feed (slowly)	80	End milling: Center over hole, mill	20	Normal Mill Safety	
Step 13									
Mill	Normal straight mill vice, parallels	Turn Machine on, set speed	Boring bar	auto feed	90	bore to 1 5/32"	45	Normal Mill Safety	Increase cuts by .01"
Step 14									
Mill	Normal straight mill vice, parallels	Turn machine on, set speed	9/32" end mill (4 flute) LONG	Manual feed (slowly)	1000	Milling: mill slot at end of side branch	45	Normal Mill Safety	increase cut depth by .01"

Step 15									
Mill	Tap guide	N/A	Edge finder,1-1/16"-20 Tap	N/A	N/A	Use edge finder to center over hole. Insert tape guide into chuck. Begin tapping, back tap out until burr is removed continue tapping, repeat	40	N/A	Operation for retaining disk threading. Keep tap very well oiled. Use wrench to turn tap
Step 16									
Tap	Tap guide, angle plate and C-clamps	N/A	Edge finder, 1-11-1/2 Tap	N/A	N/A	Use edge finder to center over hole. Insert tape guide into chuck. Begin tapping, back tap out until burr is removed continue tapping, repeat	90	N/A	Operation for inlets and outlets. Keep tap very well oiled. Grind Tap if necessary to get thread engagement

Retaining Disk

Process: Cut blank with water jet, drill and tap center hole, thread outside

Stock: 6061 Aluminum 1” X 12” X 12"

Machine	Fixturing	Machine Setup	Tooling	Feed Rate (ipm)	Cutting Speed (rpm)	Process Description	Expected Time	Safety Concerns	Additional Notes
Step 1									
Water Jet	Bar clamps used for fixturing against square in water jet cuttuing bed	Create .ord file for water jet toolpath	N/A	Set by software	Set by software	Run toolpath on water jet, computer control	10 min.	Hands clear of jet	Outer Diameter larger than needed, center hole smaller than needed
Step 2									
Drill Press	C-Clamp to vice with spacers	Set speed according to tooling and stock material	G Drill Bit (2 flute)	Manual feed (slowly)	750	Drill hole through center hole, the part should self center	5 min.	Normal drill press safety, make sure part is secure (more difficult since it is circular)	

Step 3									
Lathe	Collet with fixture that fits into the hole in retaining disk on one end and into collet. Use tail stock pressure, Dial Indicator	Turn on machine, set speed	Single Point Cutting Tool	Manual, slow	500	Turn down diameter	30	run slowly to avoid breaking tool	
Step 4									
Lathe	3 Jaw chuck, dial indicator	Turn on machine, set speed	Boring bar	Manual, slow	300	Bore out center hole to size	20		Make cuts of 0.01” off diameter
Step 5									
Lathe	3 Jaw Chuck, dial indicator	Set machine to 20 threads per inch	Threading tool	Manual, slow	150	Thread using automatic threading function on lathe	45		Make sure that threads are deep enough (should come to peaks)

Step 6									
Lathe	3 Jaw Chuck	Turn machine on	1/8” parting tool	Manual, slow	150	Cut disk to length	7	Go very slowly when cutting off spokes to avoid toque	

Plug and Gate

Process: Cut blank with water jet, drill and ream center hole, bore cavity

Stock: 6061 Aluminum 1” X 12” X 12"

Machine	Fixturing	Machine Setup	Tooling	Feed Rate (ipm)	Cutting Speed (rpm)	Process Description	Expected Time	Safety Concerns	Additional Notes
Step 1									
Water Jet	Bar clamps used for fixturing against square in water jet cutting bed	Create .ord file for water jet toolpath	N/A	Set by software	Set by software	Run toolpath on water jet, computer control	10 min.	Hands clear of jet	Outer Diameter larger than needed, center hole smaller than needed
Step 2									
Drill press	Vice	turn machine on, set speed	No. 25 drill bit	Hand	1150	Drill through hole at center piercing	1 min	Normal drill press safety	

Step 3									
Drill press	Vice	turn machine on, set speed	Center drill	Hand	1150	Drill through hole at center piercing	1 min	Normal drill press safety	This creates a chamfer for the tail stock to seat in
Step 4									
Lathe	Use custom fixture in 3-jaw chuck, use tail stock pressure to hold part in place	turn machine on, set speed	Turning/facing tool	Autofeed	450	Turn down outer diameter of plug to 1”	31 min	Make small cuts because of tail stock pressure	
Step 5									
Lathe	Use custom fixture in 3-jaw chuck, use tail stock pressure to hold part in place	turn machine on, set speed	Turning/facing tool	Autofeed	450	Turn down end of plug to ½”	31 min	Make small cuts, especially when cutting spokes	Do not remove from previous step to keep everything on-center

Step 6									
Lathe	4-jaw chuck, small end facing in	turn machine on, set speed	Boring bar	Autofeed	300	Bore center cavity of plug and gate	25 min	Make small cuts, especially when boring spokes	
Step 7									
Lathe	4-jaw chuck, small end facing in	turn machine on, set speed	No. 23 ream	Hand feed	700	Ream center hole	½ min	Normal lathe safety	Quickly make cut and remove, keep same fixture as before

Custom Lathe Fixture

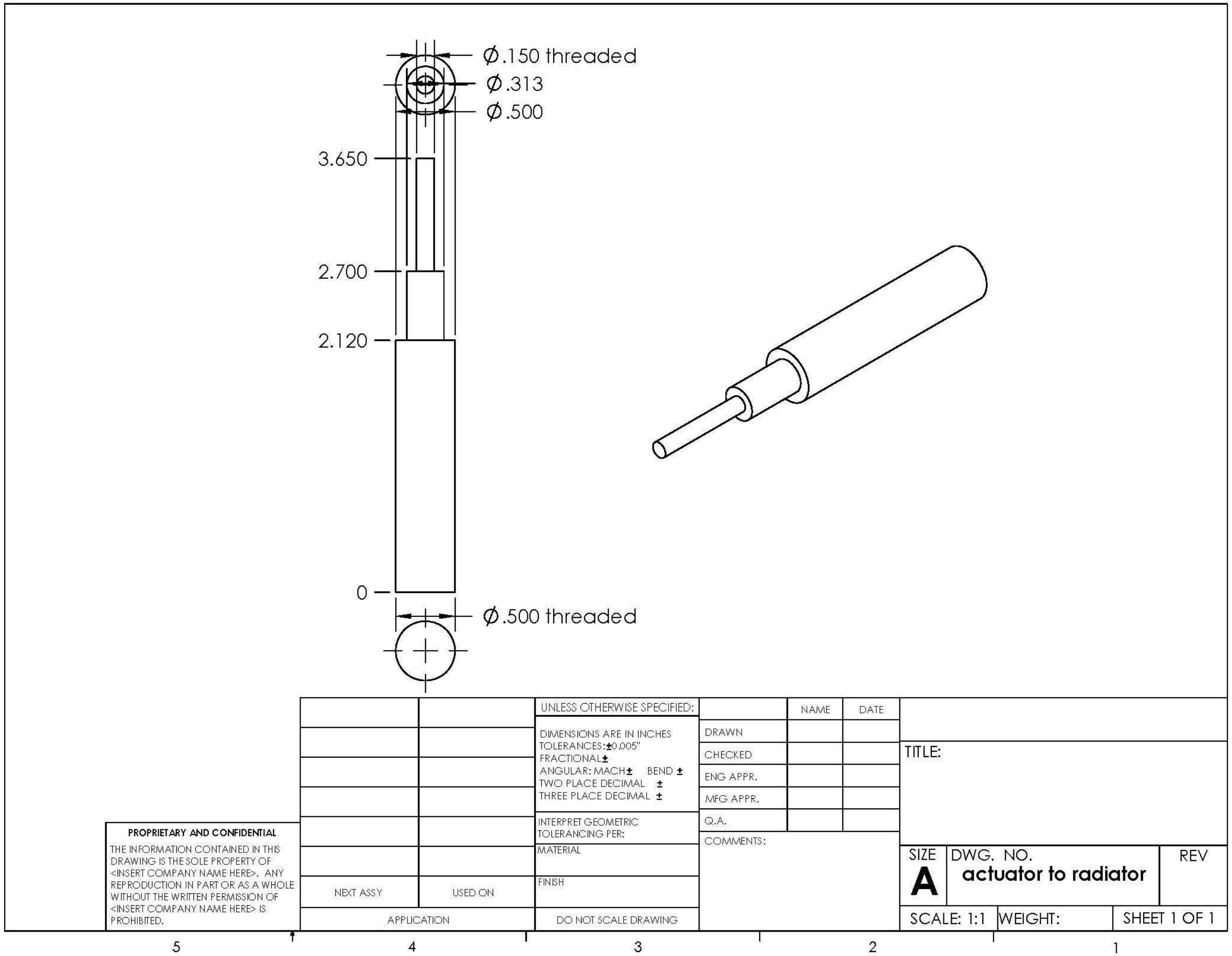
Process: Cut blank with water jet, drill and ream center hole, bore cavity

Stock: 6061 Aluminum Bar D0.5” X 3.5 “

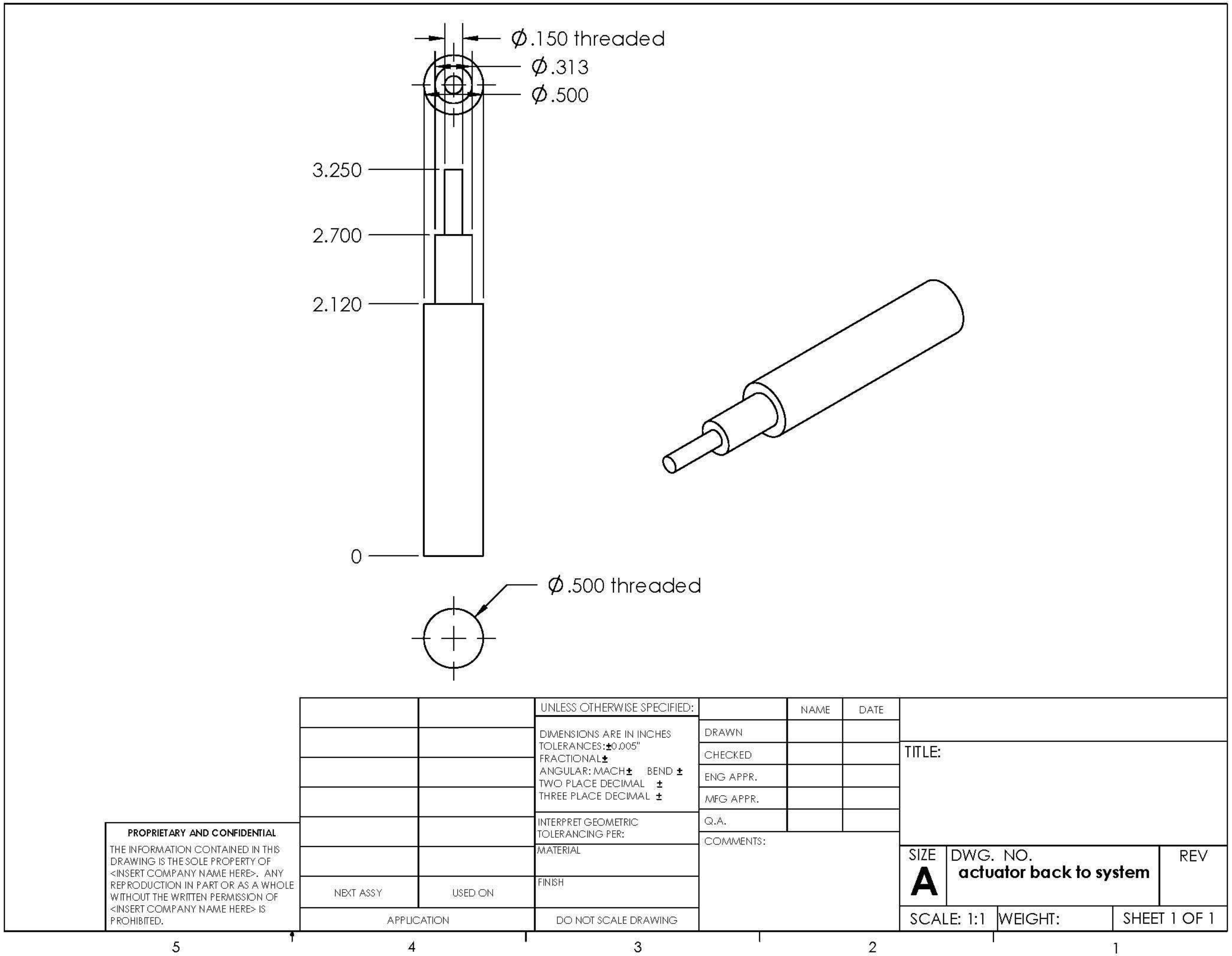
Machine	Fixturing	Machine Setup	Tooling	Feed Rate (ipm)	Cutting Speed (rpm)	Process Description	Time	Safety Concerns	Additional Notes
Step 1									
Lathe	3 jaw chuck	Turn machine on, set speed	Facing/turning tool	Autofeed	500	Turn diameter down for ~0.6” until it barely fits into center hole of plug and gate or retaining disk	~7.5 min	Normal lathe safety	At end of cut, make sure there is a flat step for face of plug and gate or retaining disk to sit on during turning

Appendix E Engineering Drawings

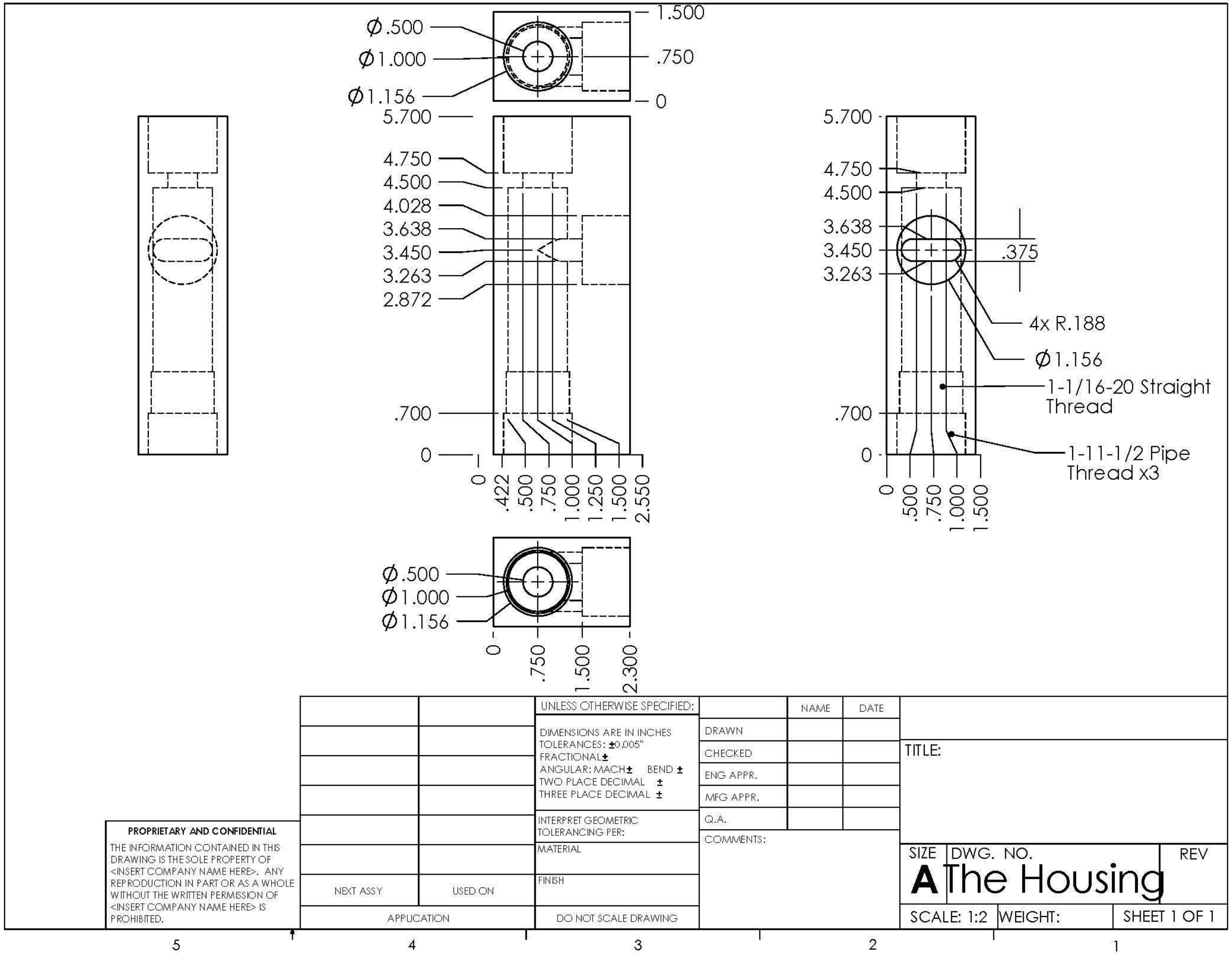
Linear Thermal Actuator (Actuated Position)



Linear Thermal Actuator (Normal Position)



Housing



Ø.500
Ø1.000
Ø1.156
1.500
.750
0
5.700
4.750
4.500
4.028
3.638
3.450
3.263
2.872
.700
0
0
.422
.500
.750
1.000
1.250
1.500
2.550
.375
4x R.188
Ø1.156
1-1/16-20 Straight Thread
1-11-1/2 Pipe Thread x3
.500
.750
1.000
1.500
.750
1.500
2.300
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.005"
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
DWG. NO.
REV
A
The Housing
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.
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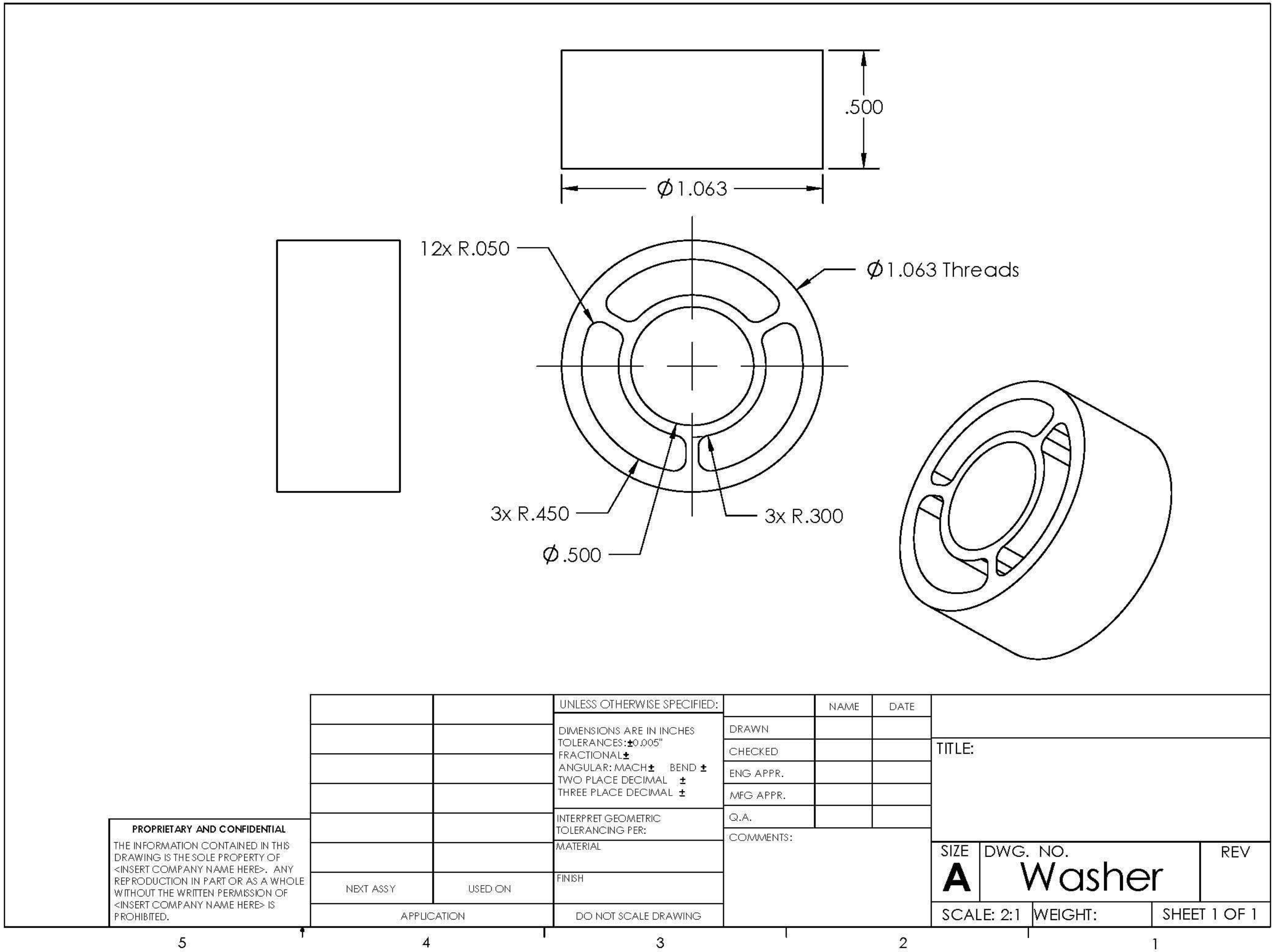
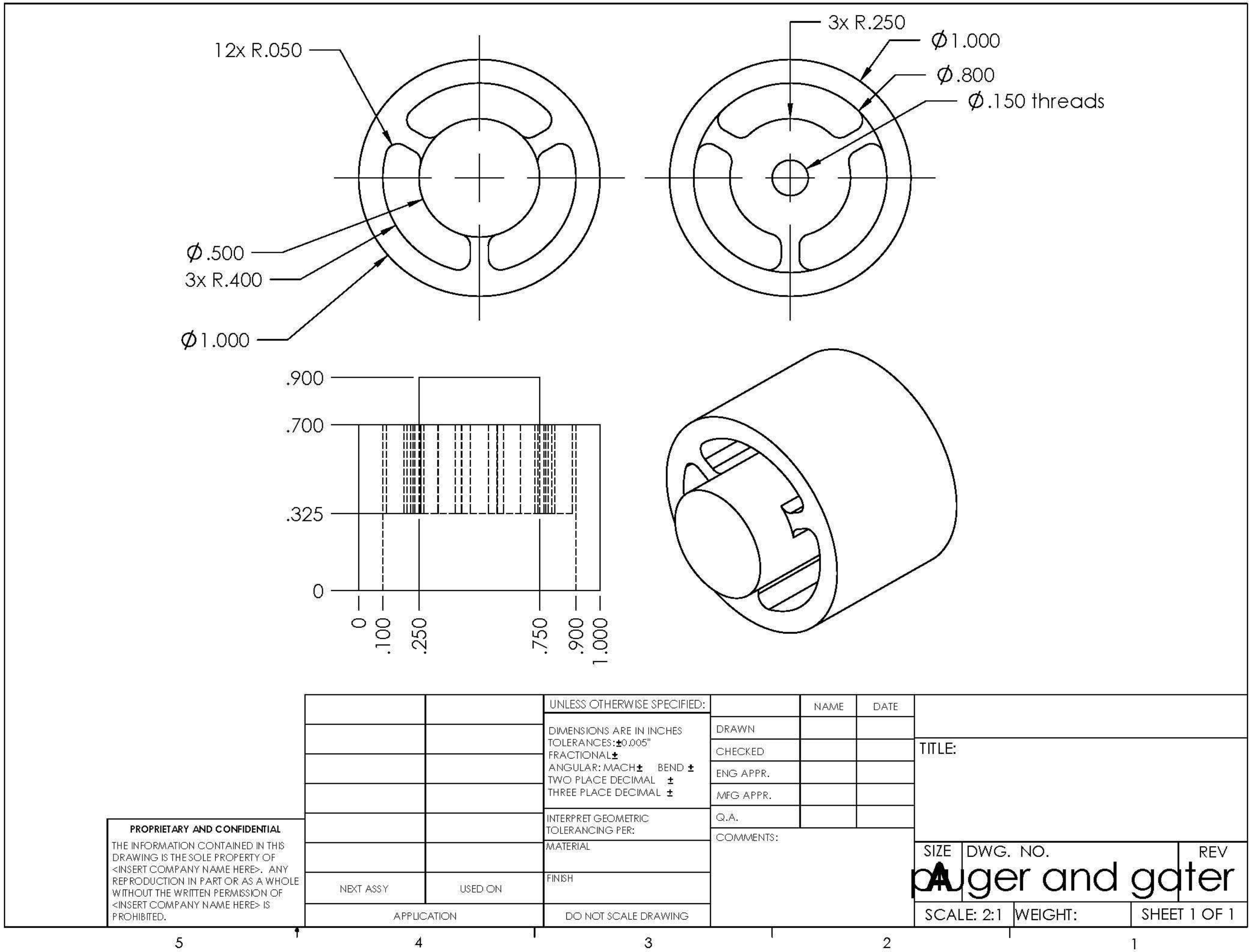
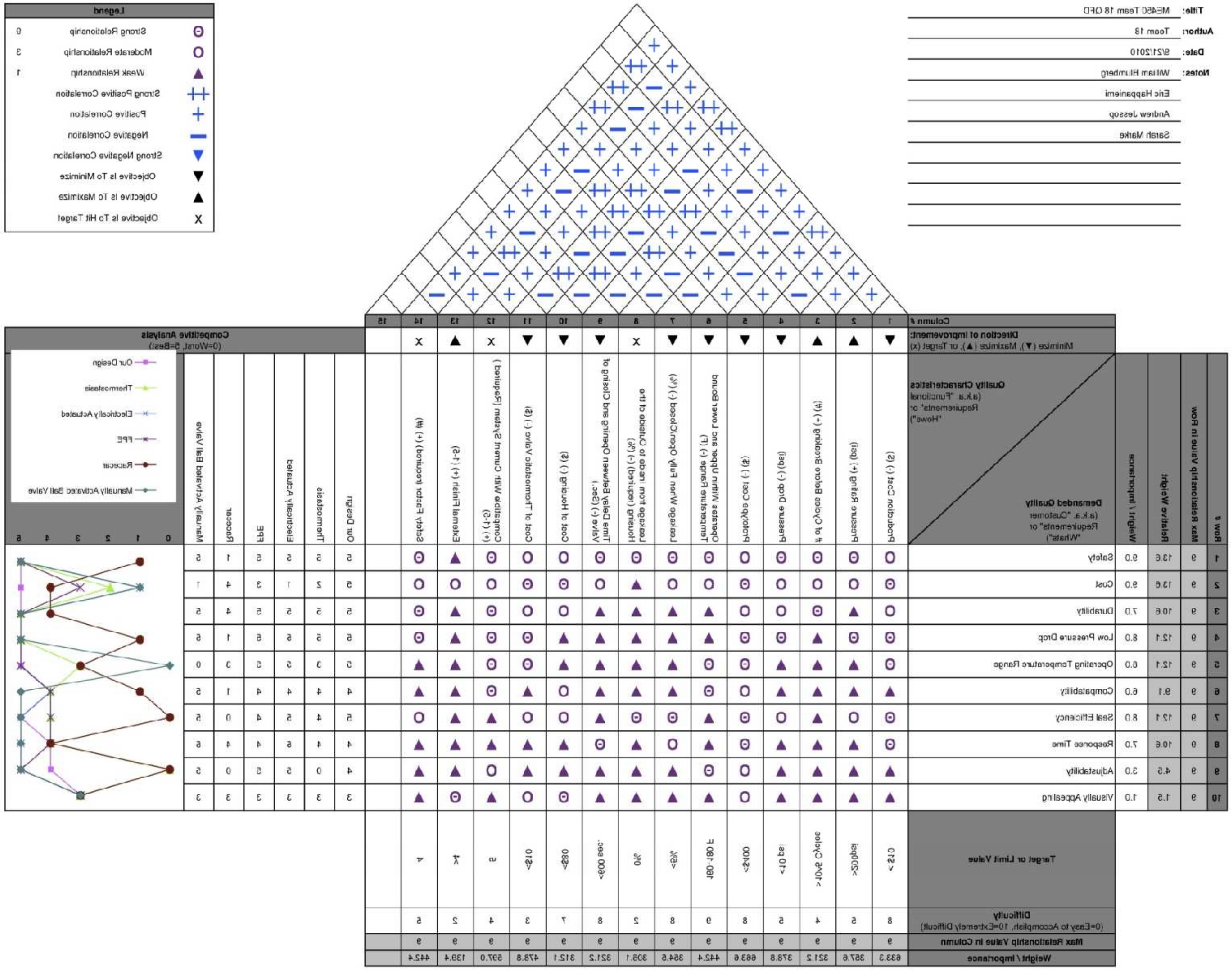
Retaining Disk



Plug and Gate



Appendix F QFD



Appendix G Subsystem Pugh Charts

a. Actuator

Design Criteria	Weight	Thermally Expansive Wax	Electronic	Shape Memory Alloy	Bimetallic	Ethanol	Electromagnet
Safety	**3**	2	2	0	2	-1	2
Cost	**3**	2	-1	-1	-1	2	-1
Operating Temperature Range	**3**	1	2	1	1	2	2
Response Time	**2**	1	2	1	1	2	2
Adjustability	**2**	1	2	0	0	0	2
Durability	**2**	2	1	2	2	0	1
TOTAL		**23**	**19**	**6**	**12**	**13**	**19**

b. Normally Biasing Actuator

Design Criteria	Weight	Spring	Electronic	Electromagnetic
Safety	**3**	2	2	2
Cost	**3**	2	-1	-1
Operating Temperature Range	**3**	2	2	2
Response Time	**2**	2	2	2
Adjustability	**2**	1	2	2
Durability	**2**	2	1	1
TOTAL		**28**	**19**	**19**

c. Valve Member

Design Criteria	Weight	Sliding Gate	Rotating Gate	Plunger	Translating Channel	Plug & Gate	Inner Pipe
Safety	**3**	2	2	2	2	2	2
Cost	**3**	2	2	2	2	2	2
Seal Efficiency	**3**	2	1	2	2	2	1
Minimally impedes flow	**2**	1	2	1	1	1	2
Durability	**2**	2	2	2	2	2	2
Machinability (member and immediate surroundings)	**2**	0	2	2	2	2	2
TOTAL		**24**	**27**	**28**	**28**	**28**	**27**

d. Junction Geometry

Design Criteria	Weight	T -in into 2 directions	T- 1 branch off main	Y
Safety	**3**	2	2	2
Cost	**3**	1	1	1
Low Pressure Drop Across	**2**	1	2	2
Machinability	**1**	2	2	2
Seal Efficiency	**1**	2	2	2
TOTAL		**15**	**17**	**17**

Appendix H Concept Generation- Initial Designs

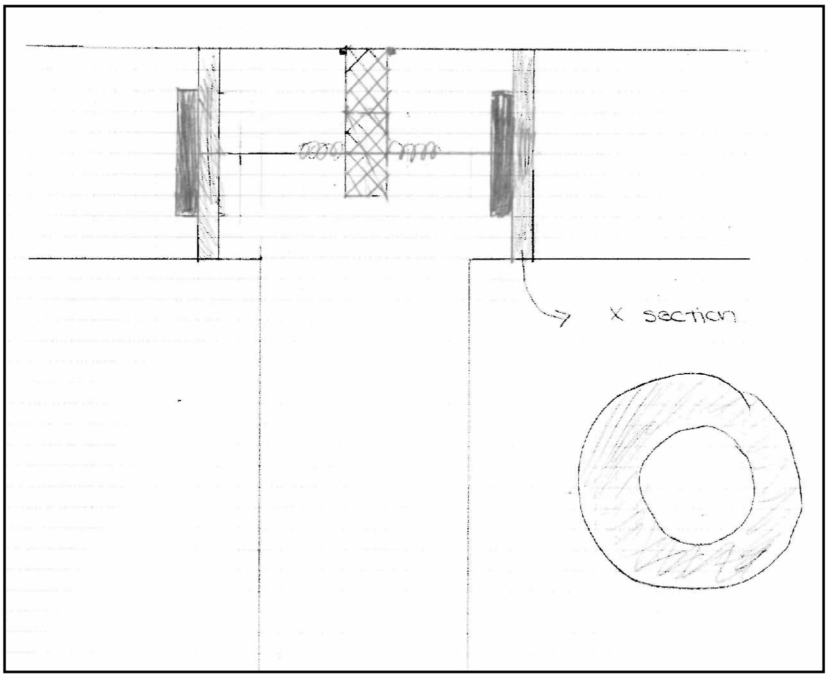
A.

This family of designs involves a T-shaped housing and two linearly translating plugs that move in tandem, with one opening a pipe while the other closes a pipe off. After we put more thought into these designs we realized that this design is not feasible with a standard linear actuator.

A.1 Double plug with shared actuator

Actuator: *Thermally Expansive Material (Thermal Actuator)*
Normally Biasing Actuator: *Spring*
Valve Member: *2 Plugs*
Junction Geometry: *T valve*

Figure 37: Initial Concept r Design Sketch for A.1



A.2 Double plug with individual actuators

In this variation, each plug is connected to its own thermal actuator. A downside to this design is the potential for both plugs to be closed at the same time, should one of the actuators, this blockage could cause serious damage as pressure would build up until failure.

Actuator: *Thermally Expansive Material (2 Actuators)*
Normally Biasing Actuator: *Springs*
Valve Member: *2 Plugs*
Junction Geometry: *T valve*

A.3 Double plug with ethanol

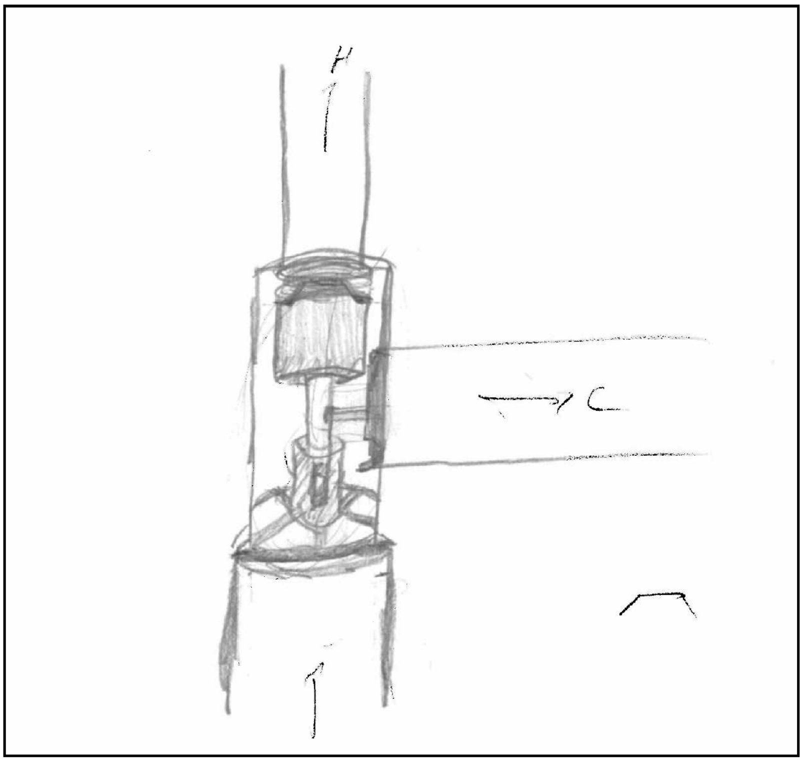
In this variation instead of using a thermal actuator a balloon of ethanol (which boils at 170 F) would be used for actuation. Ethanol is flammable and there is a potential for the balloon to break; thus, for these reasons as well as our concerns over what material this balloon could be made from, we ruled this design out..

Actuator:	*Ethanol balloon*
Normally Biasing Actuator:	*Spring*
Valve Member:	*2 Plugs*
Junction Geometry:	*T valve*

B.

This family of designs involves a piston with a plug at one end and a gate connected orthogonally to the piston.

Figure 38: Initial Concept Sketch for Design B.1



B.1 Plug and Translating Gate

In this design a linear actuator moves a piston axially along a tube, at the end of the piston is a plug and orthogonally connected to the piston is a gate that can block a path that branches from the original pipe at a right angle.

Actuator:	*Thermally Expansive Material (Thermal Actuator)*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Plug and Gate*
Junction Geometry:	*T- One Branch off Main*

B.2 Plug with Translating Gate using Ethanol

In this variation we designed a set up similar to Figure 15 except that we used an ethanol balloon that would need to be connected to the piston, via some rigid plate, as an actuation device.

Actuator: *Ethanol Balloon*
Normally Biasing Actuator: *Electromagnet*
Valve Member: *Plug and Gate*
Junction Geometry: *T- One Branch off Main*

B.3 Plug and Translating Gate using Electromagnets

In this variation instead of using a linear actuator to directly move the piston we would use one to move a piece of conductive material that could complete or break two different circuits (see
Figure 39). When completed, each circuit would turn on a different electromagnet (see Figure 32).

Actuator: *Thermally Expansive Material and Electromagnets*
Normally Biasing Actuator: *Electromagnet*
Valve Member: *Plug and Gate*
Junction Geometry: *T- One Branch off Main*

Figure 39:Electromagnet Actuation

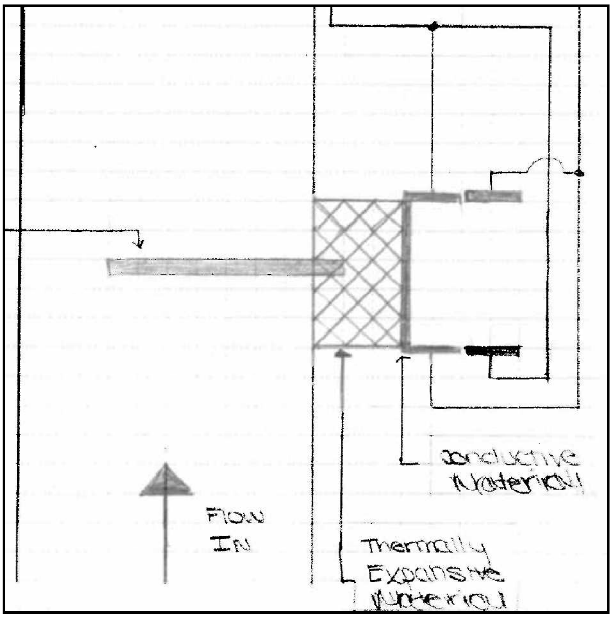
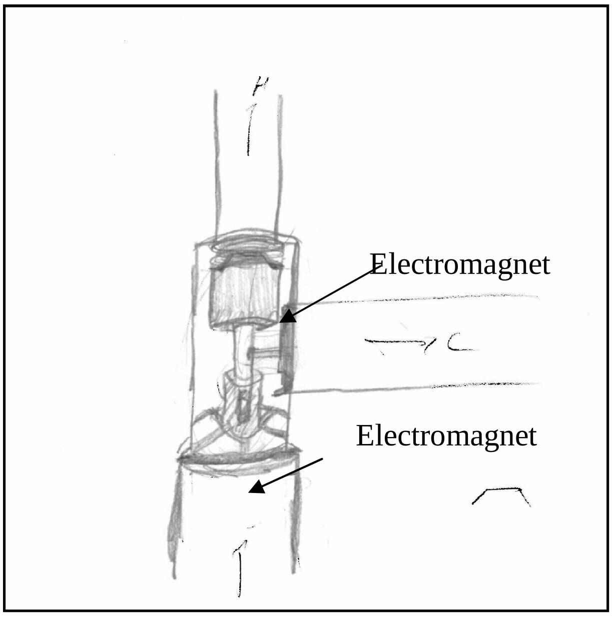


Figure 40: Initial Design of Concept B.3



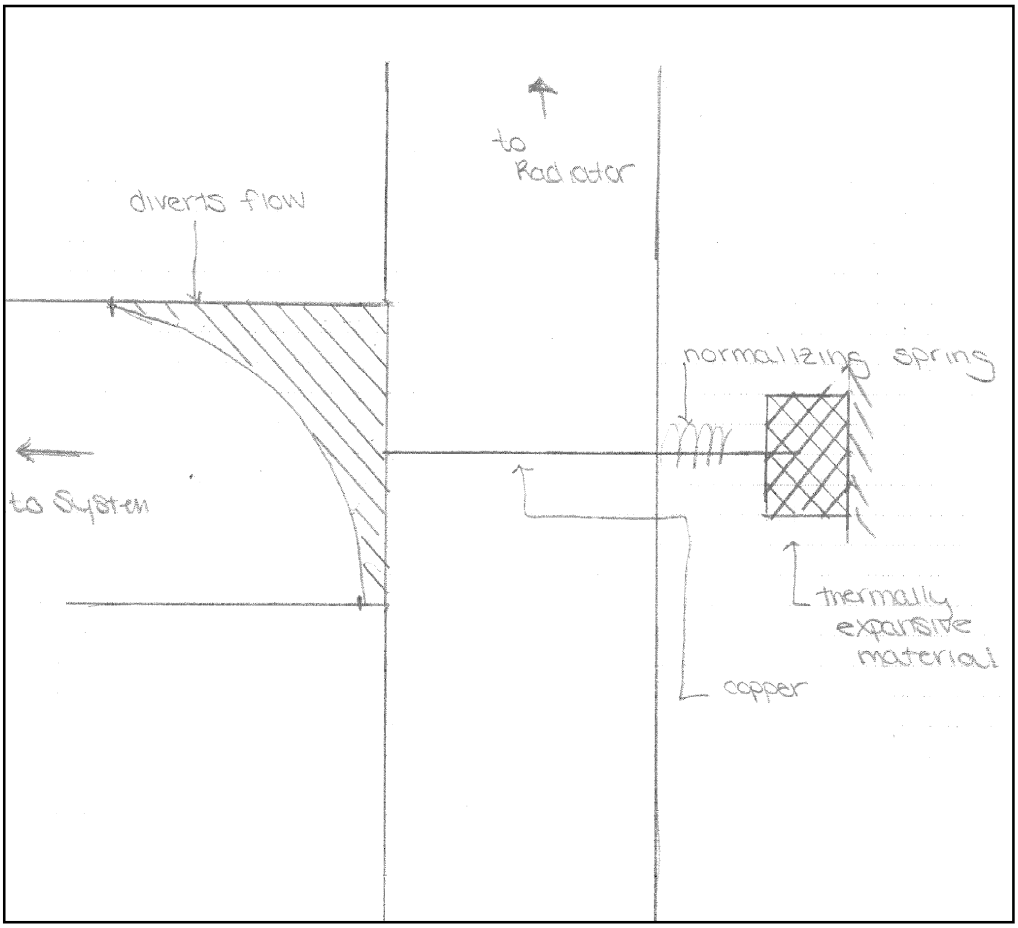
C.

Branch off Main" geometry with a sliding gate actuated in various ways.

C.1 Gate with Thermal Actuator

Actuator: *Thermally Expansive Material (Thermal Actuator)*
Normally Biasing Actuator: *Spring*
Valve Member: *Gate*
Junction Geometry: *T- One Branch off Main*

Figure 41: Initial Design of Concept C.1

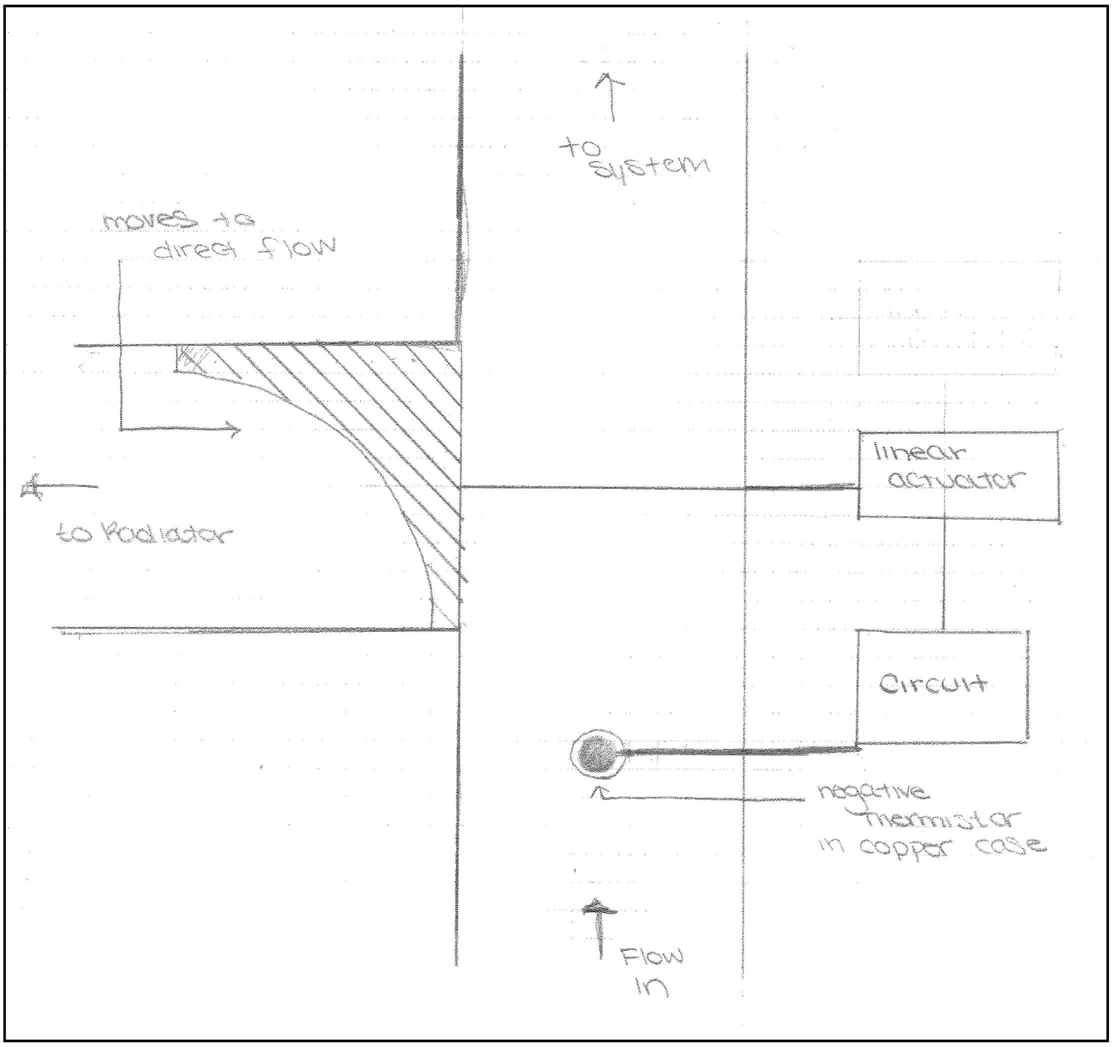


C.2 Gate with Linear Actuator

This variation uses a thermostat that sends temperature information to a linear actuator that uses electrical power.

Actuator: *Linear Actuator (electric power)*
Normally Biasing Actuator: *Linear Actuator*
Valve Member: *Gate*
Junction Geometry: *T- One Branch off Main*

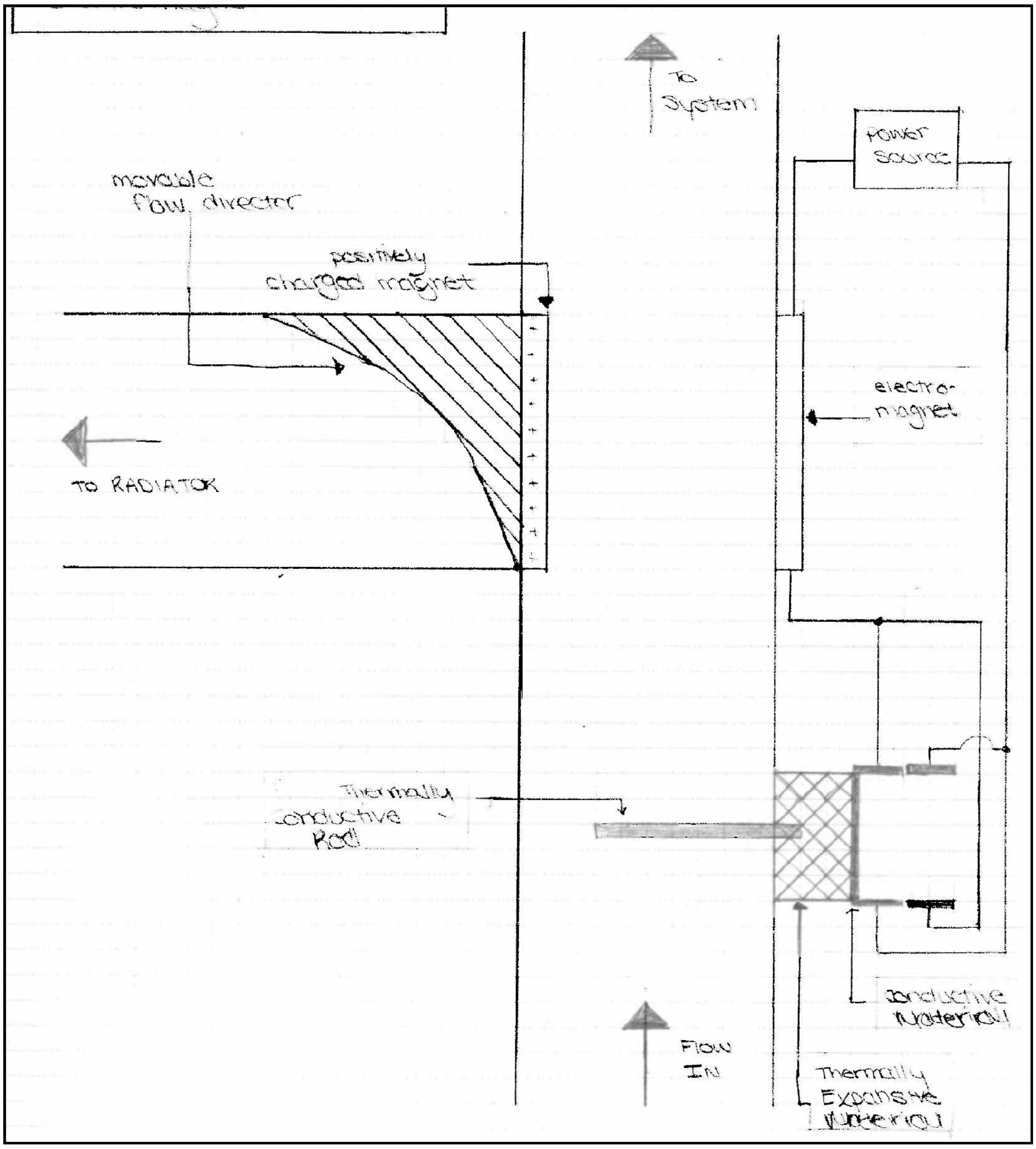
Figure 42: Initial Design of Concept C.2



C.3 Gate with Electromagnet

Actuator:	*Thermal Actuator and Electromagnet*
Normally Biasing Actuator:	*Electromagnet*
Valve Member:	*Gate*
Junction Geometry:	*T- One Branch off Main*

Figure 43: Initial Design of Concept C.3

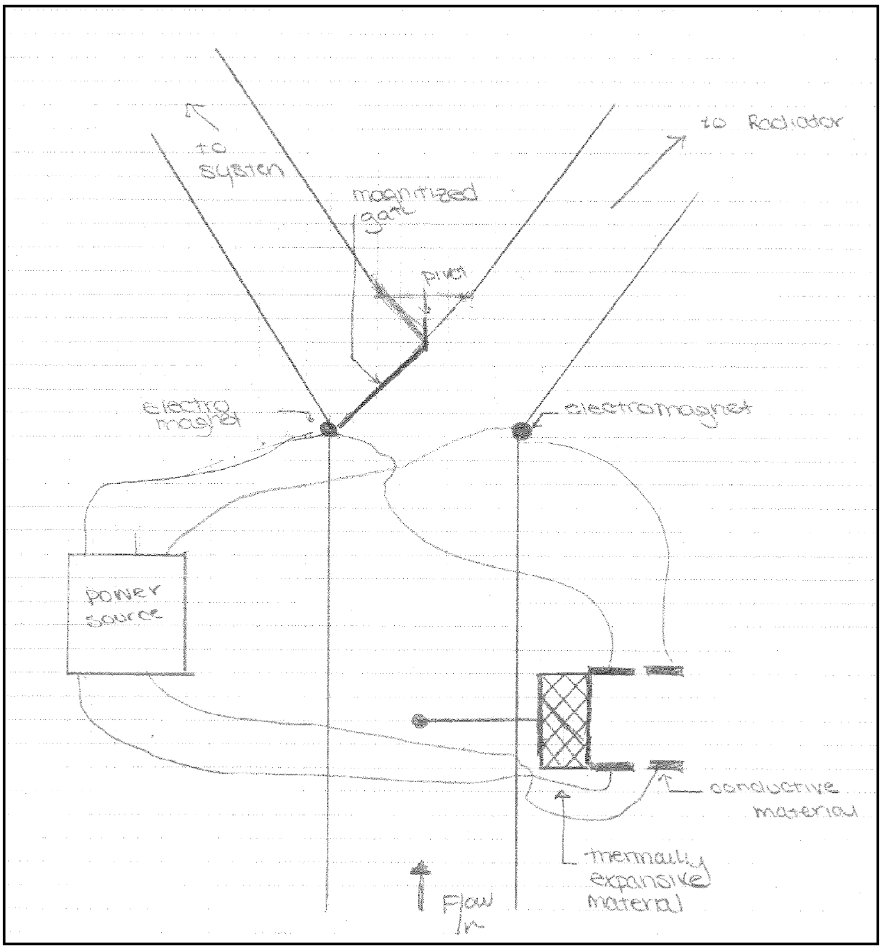


D.
These designs use a Y shaped junction and a swinging gate that is pivoted at the junction.

D.1 Y Valve with Electromagnet

Actuator:	*Thermal Actuator and Electromagnet*
Normally Biasing Actuator:	*Electromagnet*
Valve Member:	*Gate*
Junction Geometry:	*Y*

Figure 44: Initial Design of Concept D.1



D.2 Y Valve with Thermal Actuator

In this variation we considered using a thermal actuator and a normalizing spring, connected outside of the housing to an extended pivot arm, however this would be rather difficult to construct and could cause leaking.

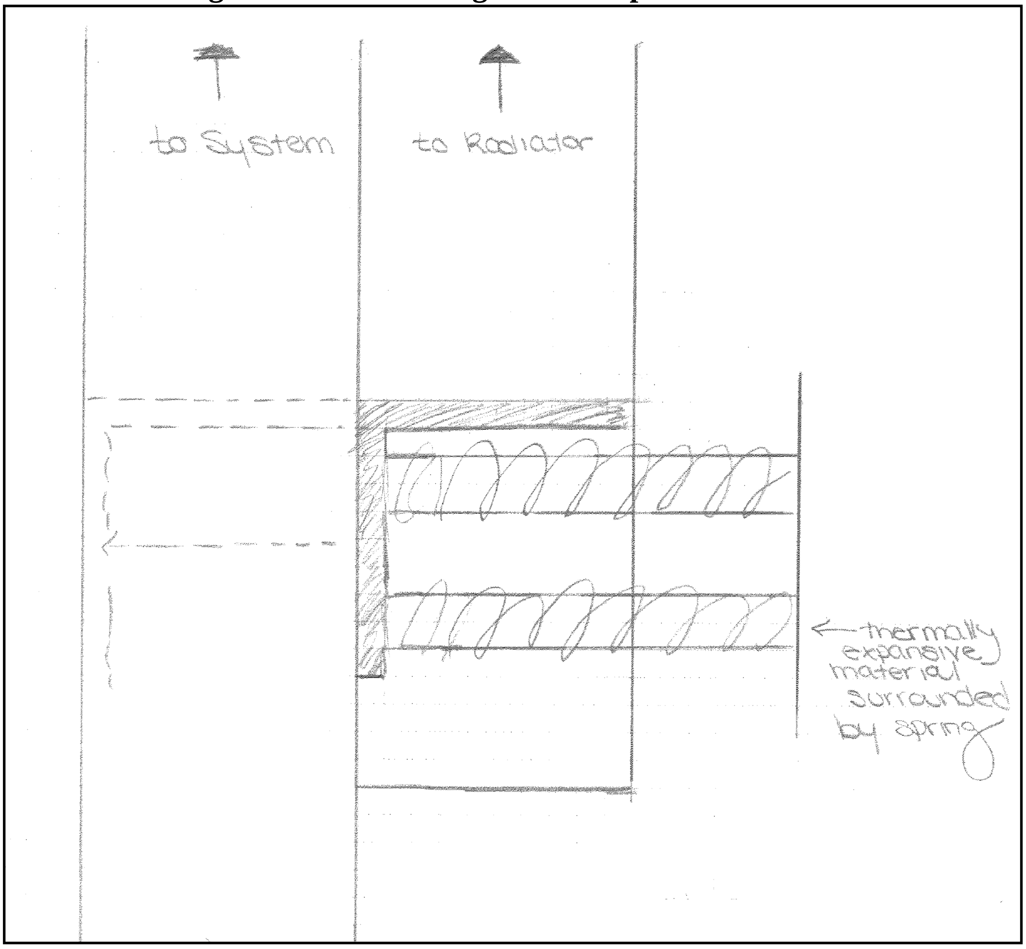
Actuator:	*Thermal Actuator*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Gate*
Junction Geometry:	*Y*

E.
These designs are based on a housing of parallel pipes and a gate translating between them. This design was not one of the top contenders because we realized that machining this into round cavities would be very difficult, and pulling a gate normal to the flow will create more force than in some of our other designs.

E.1 Parallel Pipes with Thermal Actuator

Actuator:	*Thermal Actuator*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Gate*
Junction Geometry:	*Parallel Pipes*

Figure 45: Initial Design of Concept E.1



E.2 Parallel Pipes with Electromagnet

We considered modifying design E.1 to use an electromagnet as the actuator, see Figure 16 and design C.3 for an example of a similar application of electromagnet actuation.

Actuator: *Thermal Actuator and Electromagnet*
Normally Biasing Actuator: *Electromagnet*
Valve Member: *Gate*
Junction Geometry: *Parallel Pipes*

E.3 Parallel Pipes with ethanol

In this variation behind the two pistons there would be an ethanol balloon that would expand when the hydraulic fluid was hot and push on the pistons that would then translate the gate.

Actuator: *Ethanol Balloon*
Normally Biasing Actuator: *Spring*
Valve Member: *Gate*
Junction Geometry: *Parallel Pipes*

F. Capillary

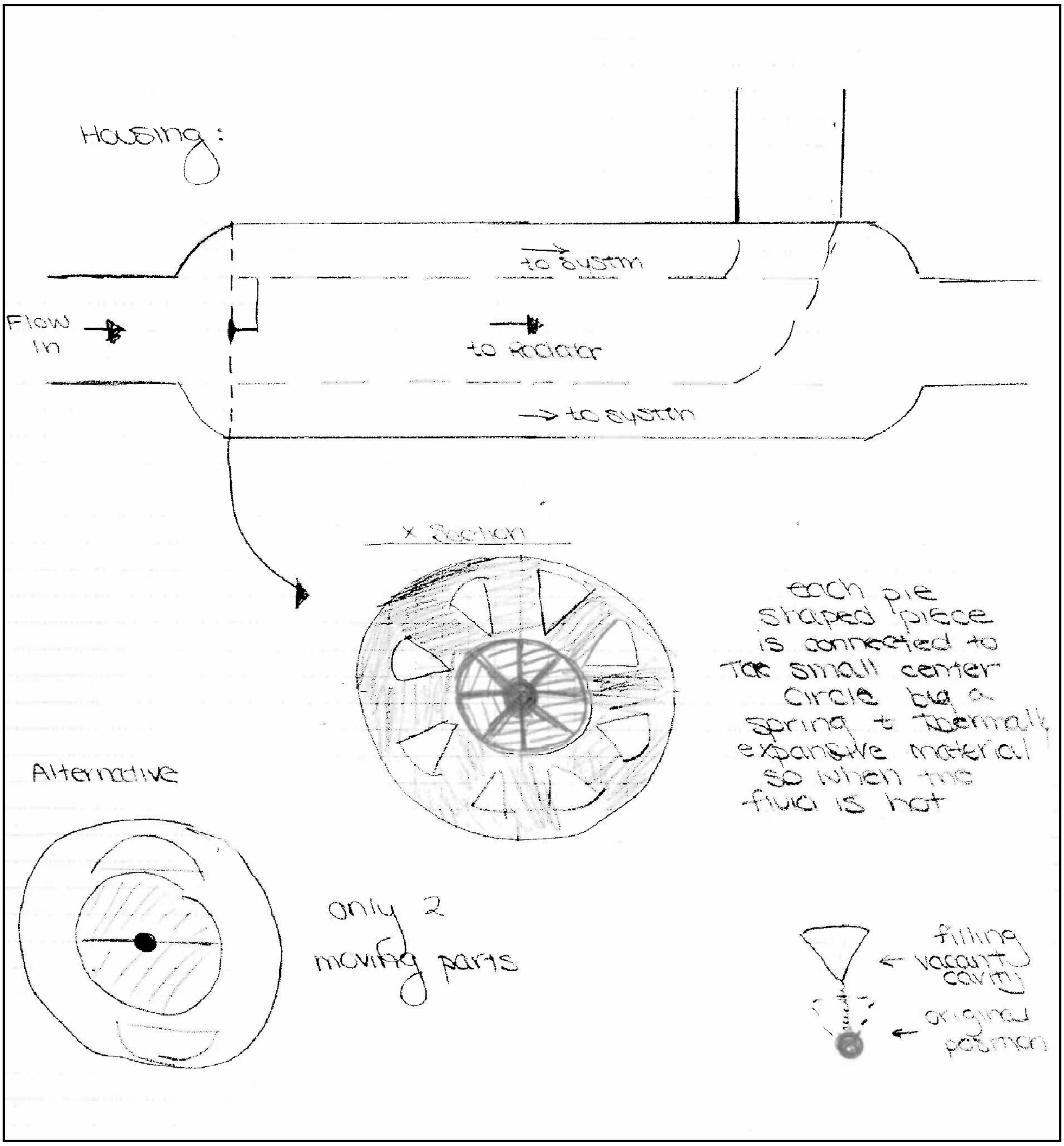
These two designs use a junction and housing of a pipe within another pipe, like a capillary. The inner cavity would be blocked by thermally expansive wedges or semicircles that when hot would extend away from the inner pipe's opening and slide up to block cut outs in a gate over the outer tube. We ruled this design out early on due to challenges in machining, problems with spring placement and how to get the gate pieces to translate where we would need them to go.

Actuator: *Thermally Expansive Material*
Normally Biasing Actuator: *Spring*
Valve Member: *Gates*
Junction Geometry: *Capillary*

F.1 Capillary with wedges

F.2 Capillary with semicircles

Figure 46: Initial Design of Concepts F.1 and F.2

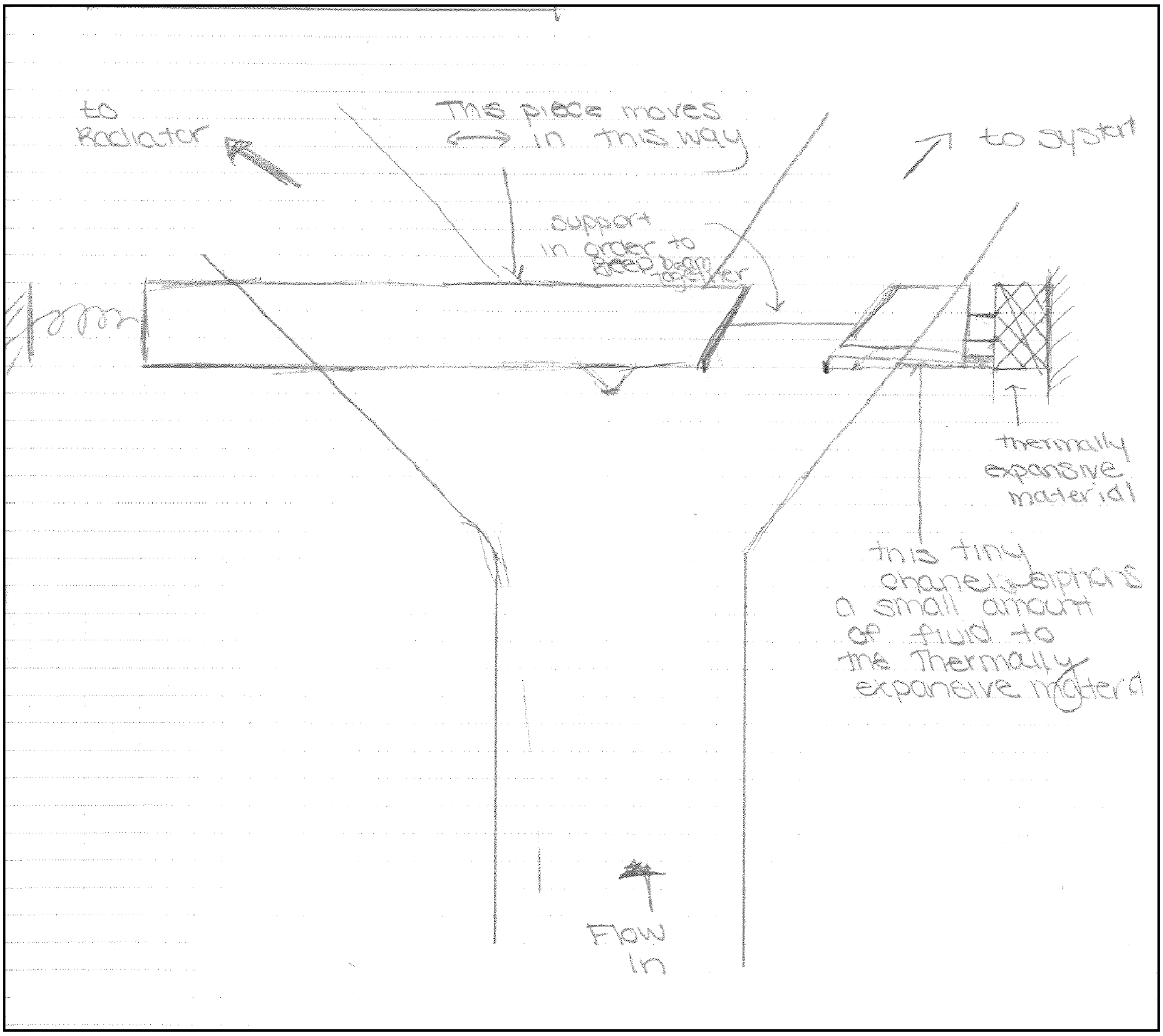


G. Moving Channel

In this design the Y junction is intersected by a moving channel that would translate normally to the flow and create openings for flow to pass through in the stems of the Y.

Actuator: *Thermally Expansive Material (Thermal Actuator)*
Normally Biasing Actuator: *Spring*
Valve Member: *Channel*
Junction Geometry: *Y*

Figure 47: Initial Design of Concept G

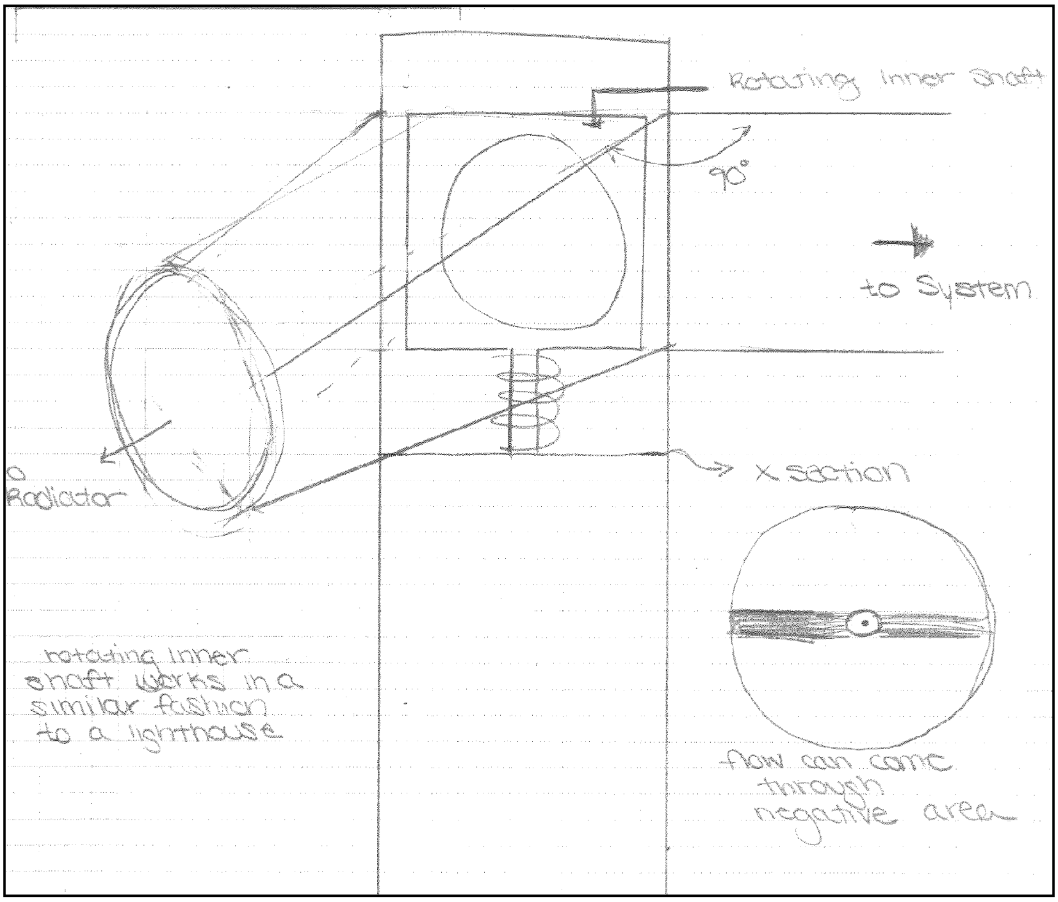


H Lighthouse

In this design a sleeve is inserted into a main pipe, this sleeve would be connected to a shape memory alloy spring that would create a rotational motion, allowing the hole in the sleeve to line up with one of two holes in the housing (one leading to the radiator and one to the system). This design did not make the cut because of the cost of shape memory alloy and our concerns that there could easily be misalignment and contact between the housing and the inner pipe.

Actuator:	*Shape Memory Alloy*
Normally Biasing Actuator:	*Shape Memory Alloy*
Valve Member:	*Sleeve*
Junction Geometry:	*T- One Branch off Main*

Figure 48: Initial Design of Concept H



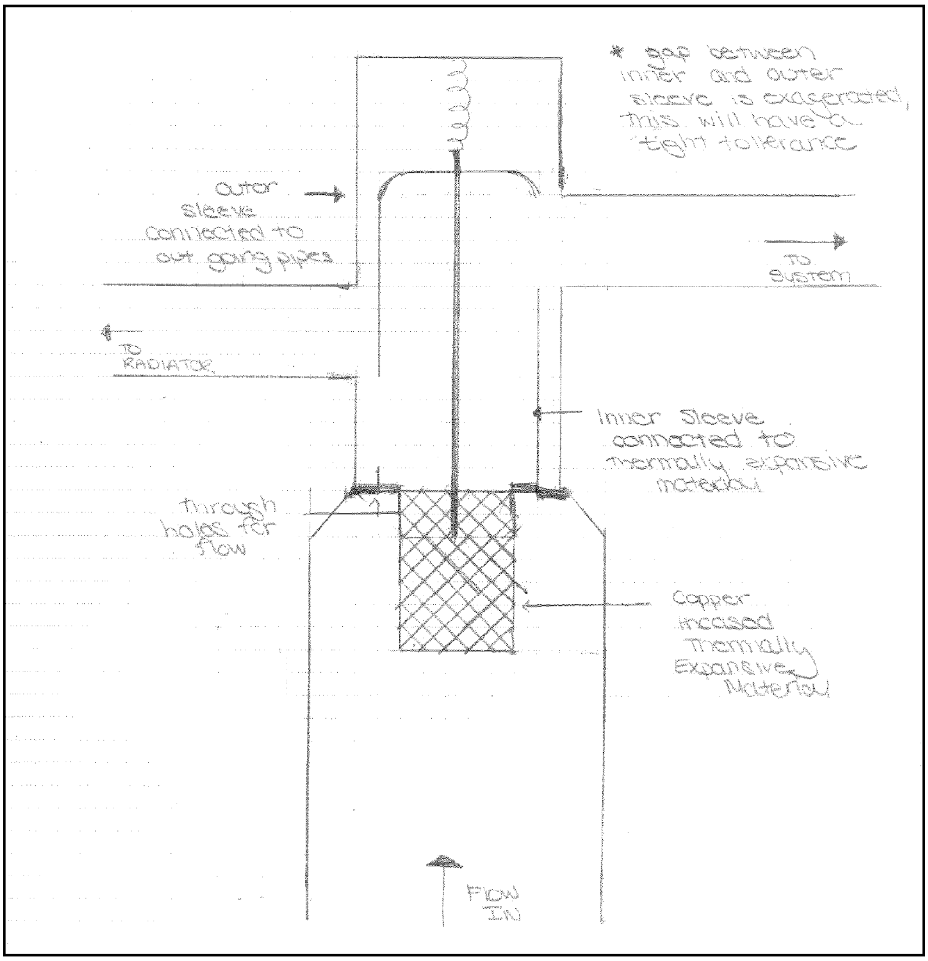
I.

These designs employ a translating sleeve with a thru hole, inside of a modification of a T-shaped housing.

I.1 Translating Sleeve with Thermal Actuator

Actuator:	*Thermally Expansive Material (Thermal Actuator)*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Sleeve*
Junction Geometry:	*T- One Branch off Main*

Figure 49: Initial Design of Concept I.1



I.2 Translating Sleeve with Linear Actuator

In this variation the overall design would be the same, except that a thermistor and an electronic linear actuator would be used to move the sleeve

Actuator:	*Linear Actuator (electronic)*
Normally Biasing Actuator:	*Linear Actuator (electronic)*
Valve Member:	*Sleeve*
Junction Geometry:	*T- One Branch off Main*

I.3 Translating Sleeve with Electromagnets

This variation would use an electromagnet to actuate the motion in a very similar fashion as design B.3.

Actuator:	*Thermal Actuator and Electromagnet*
Normally Biasing Actuator:	*Electromagnet*
Valve Member:	*Sleeve*
Junction Geometry:	*T- One Branch off Main*

I.4 Translating Sleeve with Ethanol Balloon

This design type could also employ an ethanol balloon to push the sleeve up and down (the balloon would need to be a donut shape, constricted to expansion only in the axial direction.

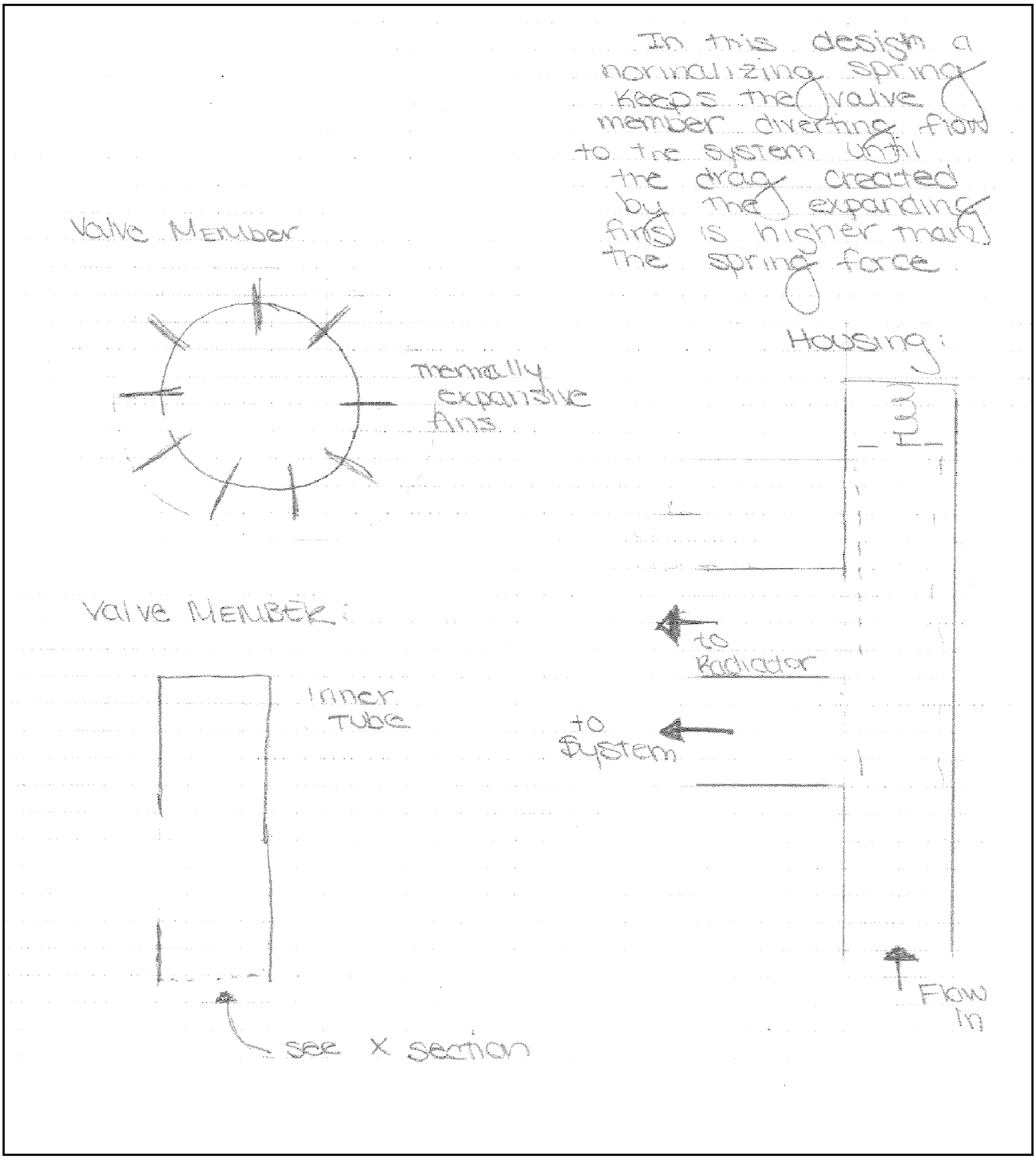
Actuator:	*Ethanol Balloon*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Sleeve*
Junction Geometry:	*T- One Branch off Main*

J. Drag Fins

This design is one of the more creative designs we developed. The actuator is a ring of thermally expansive material connected to the inlet of a sleeve. When the fluid warmed the ring would expand and begin to block more and more of the sleeve inlet. This would create a force on the ring that would push the sleeve axially. A spring would be connected behind the sleeve to counteract this motion when cooling. In our initial sketch we drew the expansive material as fins instead of one continuous ring but then as we talked more about this design decided that one piece would be better because it creates the most surface area.

Actuator:	*Thermally Expansive Ring*
Normally Biasing Actuator:	*Spring*
Valve Member:	*Sleeve*
Junction Geometry:	*Two Branches off Main*

Figure 50: Initial Design of Concept J

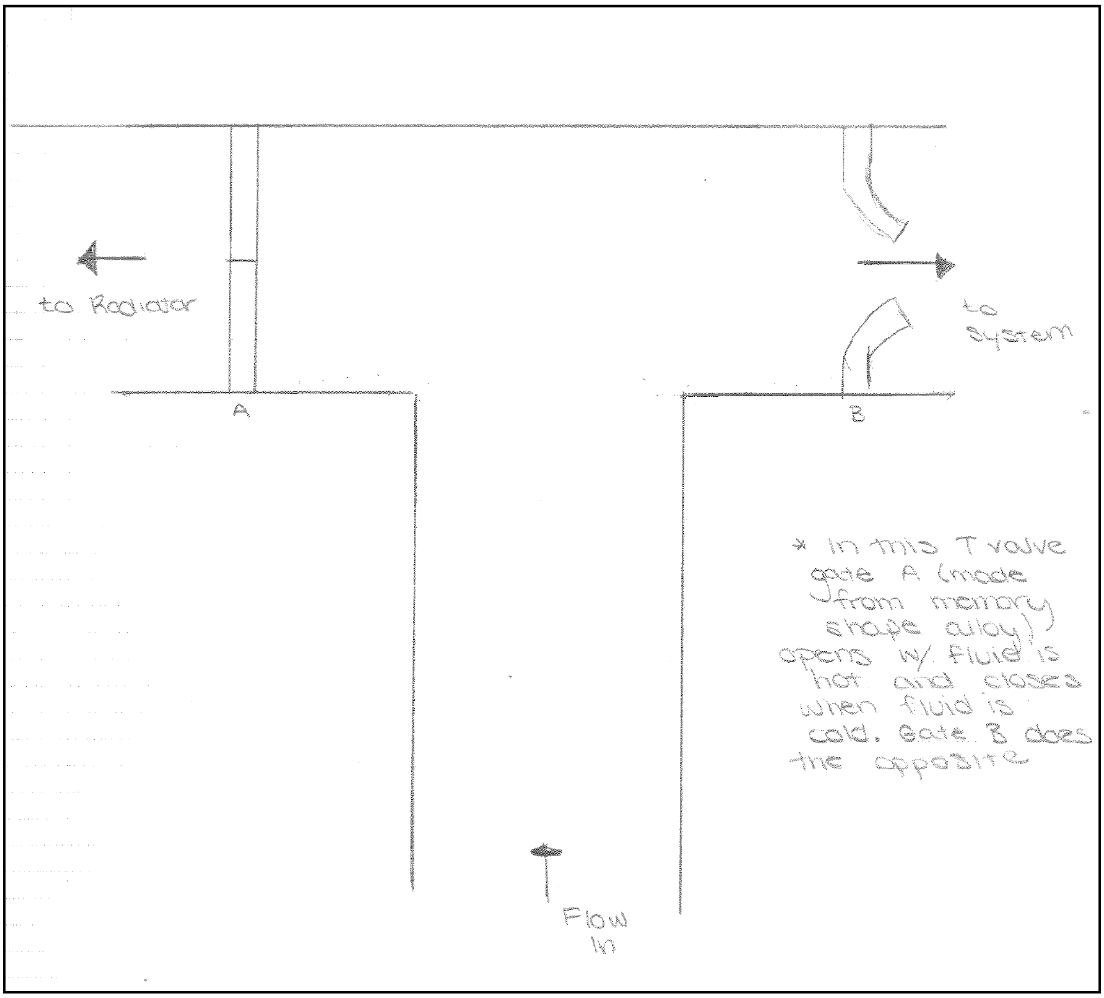


K. T Valve with Shape Memory Alloy Gates

In this design two memory shape alloy gates would be used to open and close in tandem. Downsides to this design include feasibility of construction, expense of shape memory alloy and the potential for failure in one of the gates could cause a pressure build up that could be catastrophic.

Actuator: *Shape Memory Alloy*
Normally Biasing Actuator: *Shape Memory Alloy*
Valve Member: *Gate*
Junction Geometry: *T*

Figure 51: Initial Design of Concept K

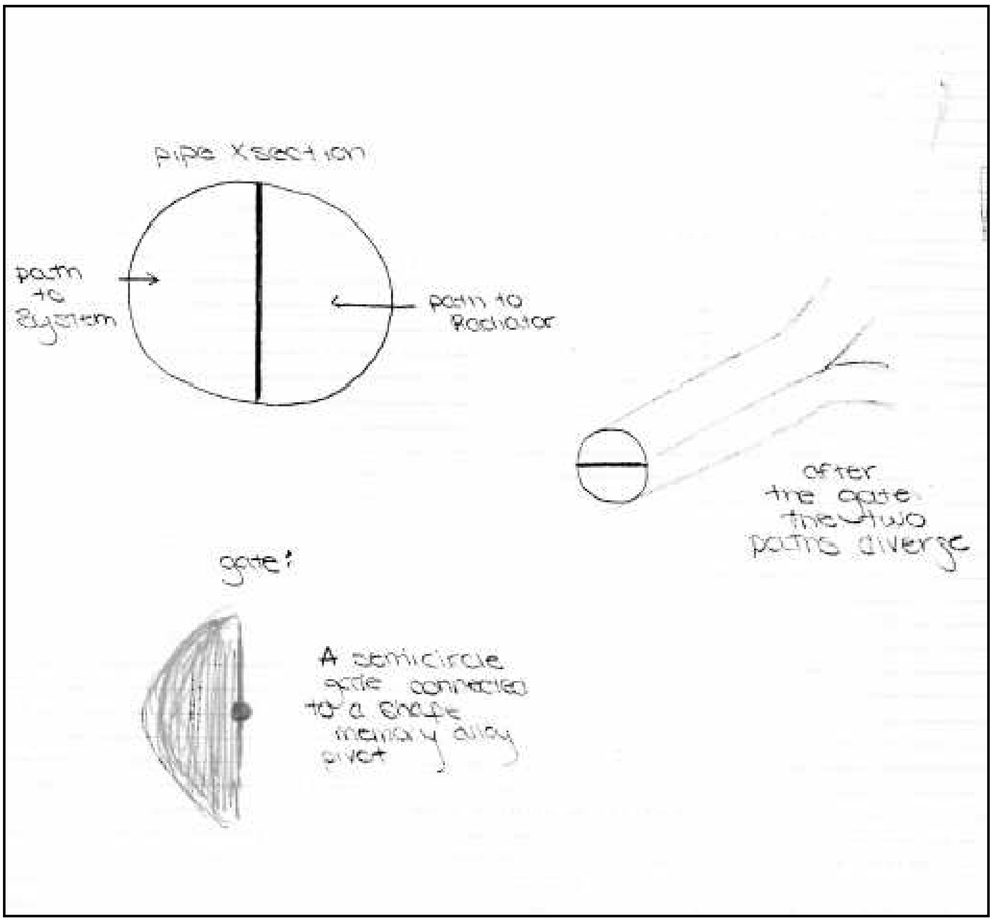


L. Half Moon

In this design a semicircular shape memory alloy would be used and would swing an arc when heated.

Actuator: *Shape Memory Alloy*
Normally Biasing Actuator: *Shape Memory Alloy*
Valve Member: *Gate*
Junction Geometry: *Divided Pipes*

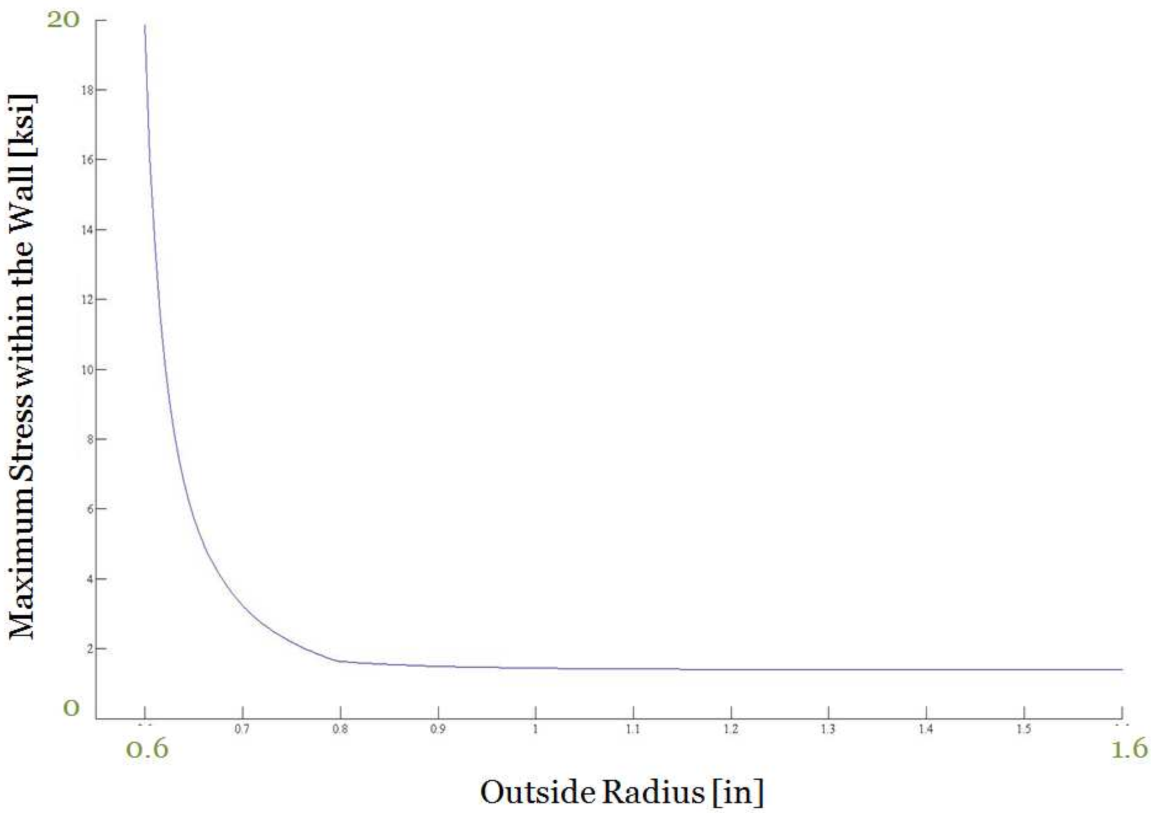
Figure 52: Initial Design of Concept L



Appendix I Housing Sizing

This is a model of the housing as a pressure vessel at 800 psi, in order to be able to pressure rate to 200 psi.

Figure 53: Determining minimum outer radius of housing



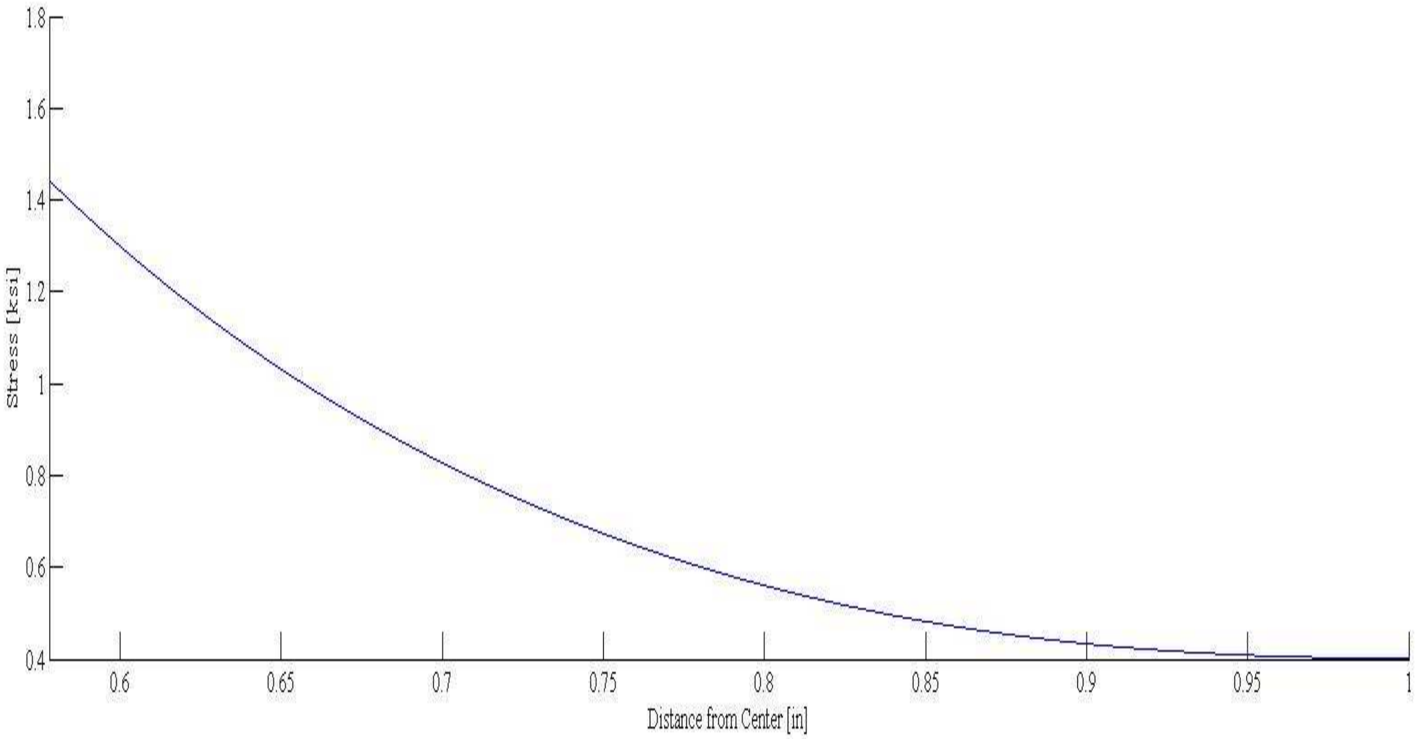
Appendix J Housing Sizing Matlab Code

Matlab code used to determine stress in housing and generate plot in Appendix F.

```
function Stress = housing_stress

%p=  gauge pressure in psi
p=800;
%n is number of steps
n=100000;
%r1= tube radius [in]
r1=(1+5/32)/2;
%R= distance from inner radius
%big=largest possible outer radius
big=2.5;
x=(big-r1)/n;
R=[r1:x:big];
%r2 = minimum distance from inner radius to outside of housing
r2=[.:.005:big];

y=length(r2);
Stress=zeros(1,y);

for j=1:y

    Sh=zeros(1,n);
    Sx=zeros(1,n);
    St=zeros(1,n);
    Sr=zeros(1,n);

    for i =1:n
    %Sx= stress  in axial direction
    %Sr= stress in radial direction
    %St= hoop stress

    Sr(i)= -p*r1^2/(r2(j)^2-r1^2)*(r2(j)^2/R(i)^2-1);
    St(i)= p*r1^2/(r2(j)^2-r1^2)*(r2(j)^2/R(i)^2-1);
    Sx(i)= p*r1^2/(r2(j)^2-r1^2);
    %using VonMises yield criterion Sh
    Sh(i)=sqrt(((Sr(i)-St(i))^2+(St(i)-Sx(i))^2+(Sx(i)-Sr(i))^2)*.5);
    end
Psx(j)=max(Sx);
Pst(j)=max(St);
Psr(j)=max(Sr);
Stress(j)=max(Sh);
end
```

```
plot (r2,Stress/1000);
xlabel('Outside Radius [in]')
ylabel('Max Stress within the Wall [ksi]')


end
```

Appendix K   Stress distribution within the wall of the housing



Appendix L Housing wall stress Matlab code

```
function Stress = hsd

%p=  gauge pressure in psi
p=800;
%n is number of steps
n=100000;
%r1= tube radius [in]
r1=(1+5/32)/2;
%R= distance from inner radius
%big=largest possible outer radius
r2=1;
x=(r2-r1)/n;
R=[r1:x:r2];

%r2 = minimum distance from inner radius to outside of housing

y=length(r2);

   Sh=zeros(1,n);
   Sx=zeros(1,n);
   St=zeros(1,n);
   Sr=zeros(1,n);

   for i =1:n+1
   %Sx= stress  in axial direction
   %Sr= stress in radial direction
   %St= hoop stress

   Sr(i)= -p*r1^2/(r2^2-r1^2)*(r2^2/R(i)^2-1);
   St(i)= p*r1^2/(r2^2-r1^2)*(r2^2/R(i)^2-1);
   Sx(i)= p*r1^2/(r2^2-r1^2);
   %using VonMises yield criterion Sh
   Sh(i)=sqrt(((Sr(i)-St(i))^2+(St(i)-Sx(i))^2+(Sx(i)-Sr(i))^2)*.5);
   end

size(R)
size(Sh)
plot (R,Sh/1000);
xlabel('Outside Radius [in]')
ylabel('Max Stress within the Wall [ksi]')




end
```

Appendix M Stress analysis of Plug and Gate

Stress analysis on circumferential area between outer radius of arced slots and outside radius of the Plug and Gate, essentially the gate. Forces were modeled at 60 lbs.

Region of stress analysis showing underside of Plug and Gate with small taped hole

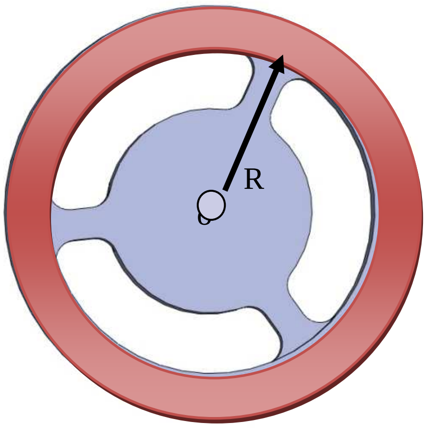
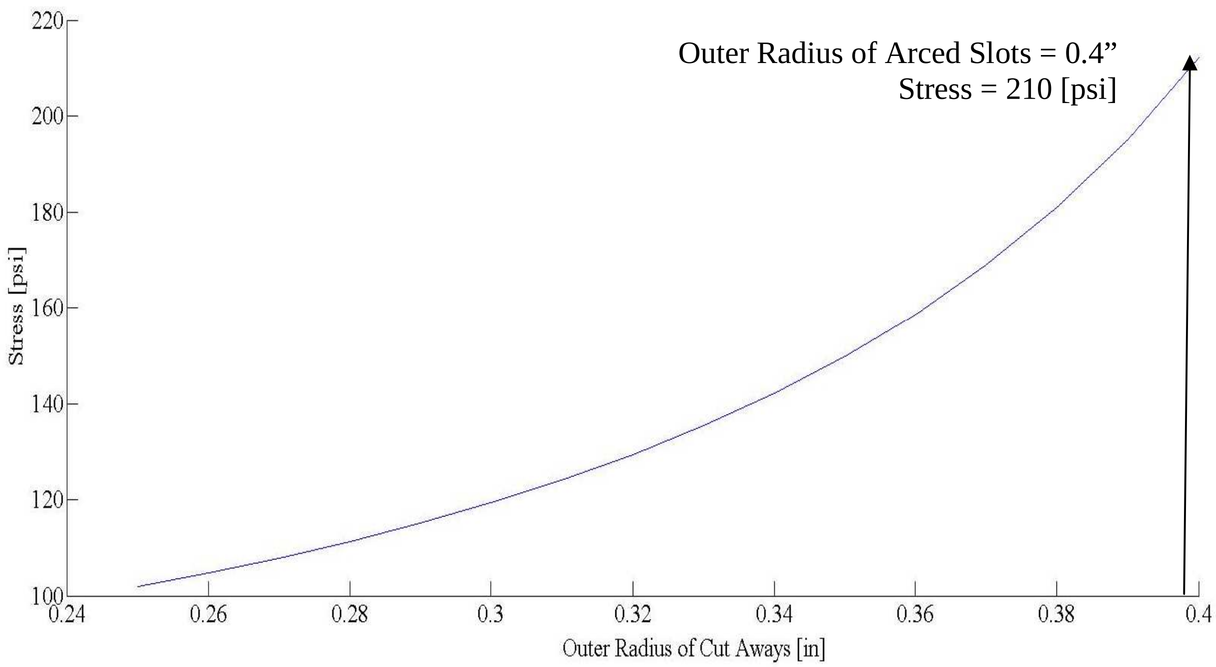
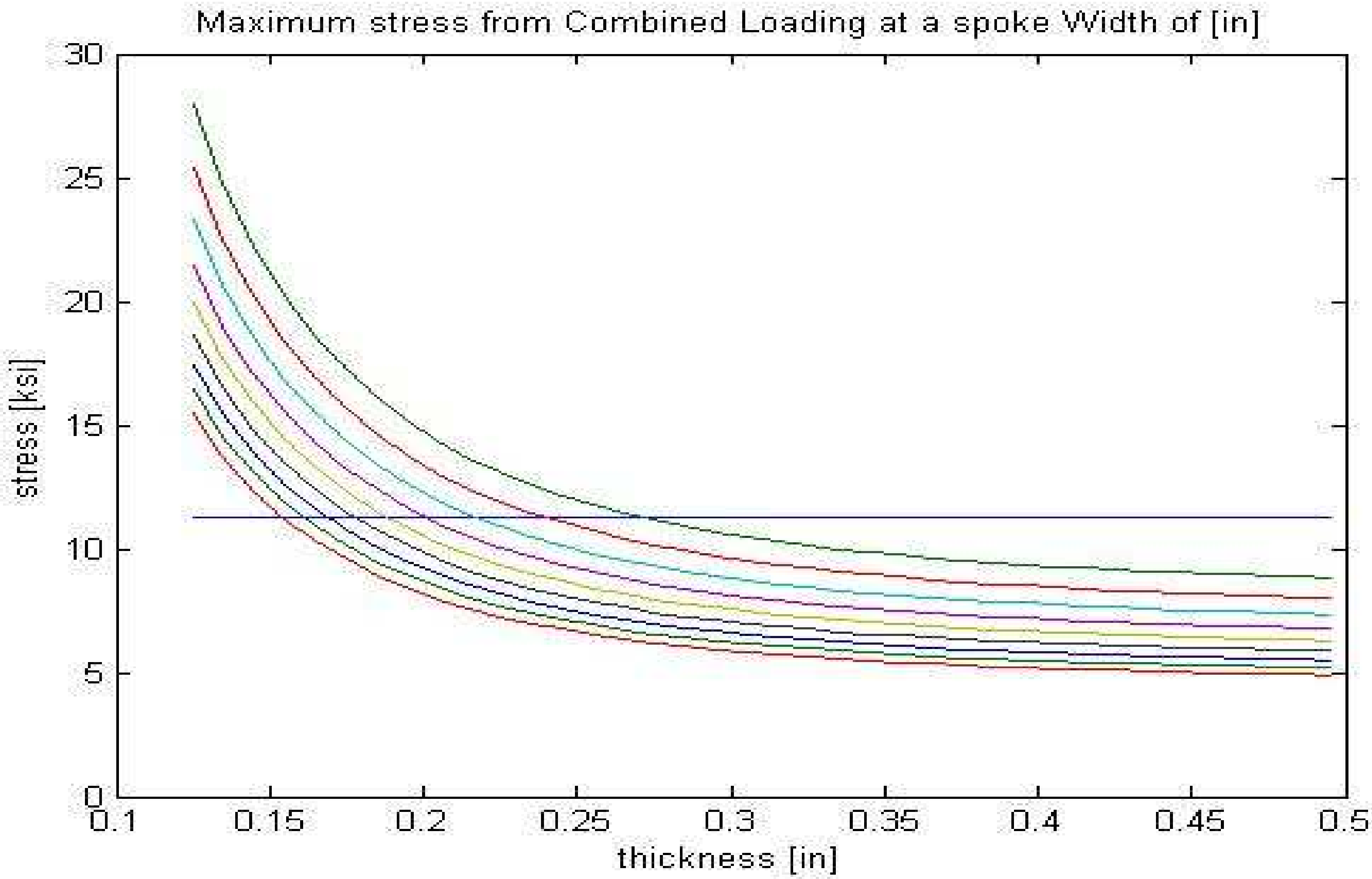


Figure 54: Stress in gate versus outer radius of arced slots




Maximum stress from Combined Loading at a spoke Width of [in]
stress [ksi]
thickness [in]
30
25
20
15
10
5
0
0.1
0.15
0.2
0.25
0.3
0.35
0.4
0.45
0.5

Appendix O   Matlab code for stress analysis of Plug and Gate

```
function main = washer

F=60;
ri=.3;
ro=[.25:.01:.48];
rf=.5;

w=[.05:.005:.1];
thickness=[.125:.01:.5];
yield_strength = 45*ones(1,length(thickness));

stress_inner_ring=F/(pi*(ri^2-.25^2))

for i=1:length(ro);
   l=ro(i)-ri;
   a_shell=pi*(rf^2-ro(i)^2);
   stress_shell(i)=F/a_shell;
end
plot(ro,stress_shell);
title('Stress in Thin Shell')
xlabel('Outer Radius of Cut Aways')
ylabel('Stress [psi]')

ro=.45;
l=ro-ri;
stress=zeros(length(w),length(thickness));
stress_y=zeros(1,length(w));

for j=1:length(w);
   width=w(j);
   a_spoke=l*width;
   stress_y(j)=F/a_spoke;

   stress_x=zeros(1,length(thickness));
   for k=1:length(thickness)
      %one third of the force from the acutator will be applied to each
      %spoke
      Force=F/3;
      %stress in x comes from moment created by this force
      t=thickness(k);
      stress_x(k)=Force*l*6/(width*t^2);
      stress_x(k)
      %using Von Mises
      stress_y(j)
      stress(j,k)=sqrt((stress_x(k))^2+(stress_y(j))*(stress_x(k))+(stress_y(j))^2);
```

```
        end

        %str1=num2str(w(j));

    end
         Figure

plot(thickness,yield_strength/4,thickness,(stress(1,:)/1000),thickness,(stress(2,:)/1000),thickness,
(stress(3,:)/1000),thickness,(stress(4,:)/1000),thickness,(stress(5,:)/1000),thickness,(stress(6,:)/10
00),thickness,(stress(7,:)/1000),thickness,(stress(8,:)/1000),thickness,(stress(9,:)/1000))
        title(['Maximum stress from Combined Loading at a spoke Width of [in]'])
        xlabel('thickness [in]')
        ylabel('stress [ksi]')
    end
```

Appendix P Stress analysis of Retaining Disk

Stress analysis on thin hoop between outer radius of cut-outs and outside radius of the Retaining Disk. Forces were modeled at 60 lbs.

Region of Stress Analysis

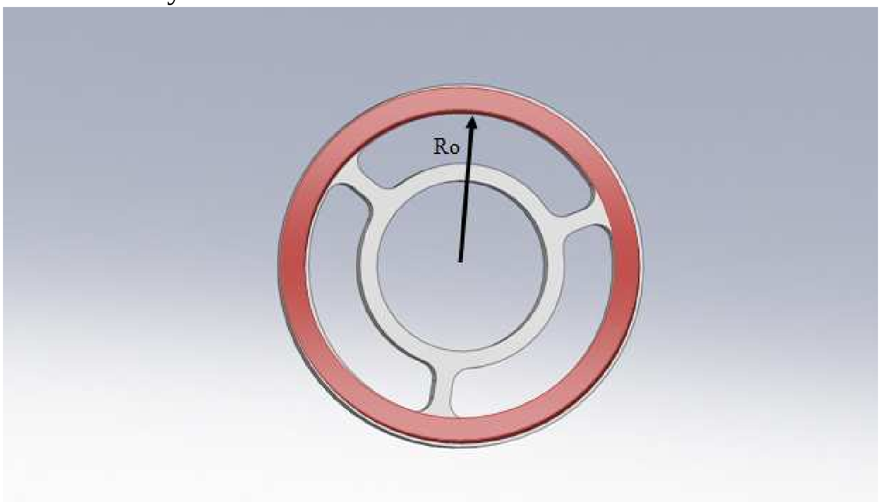
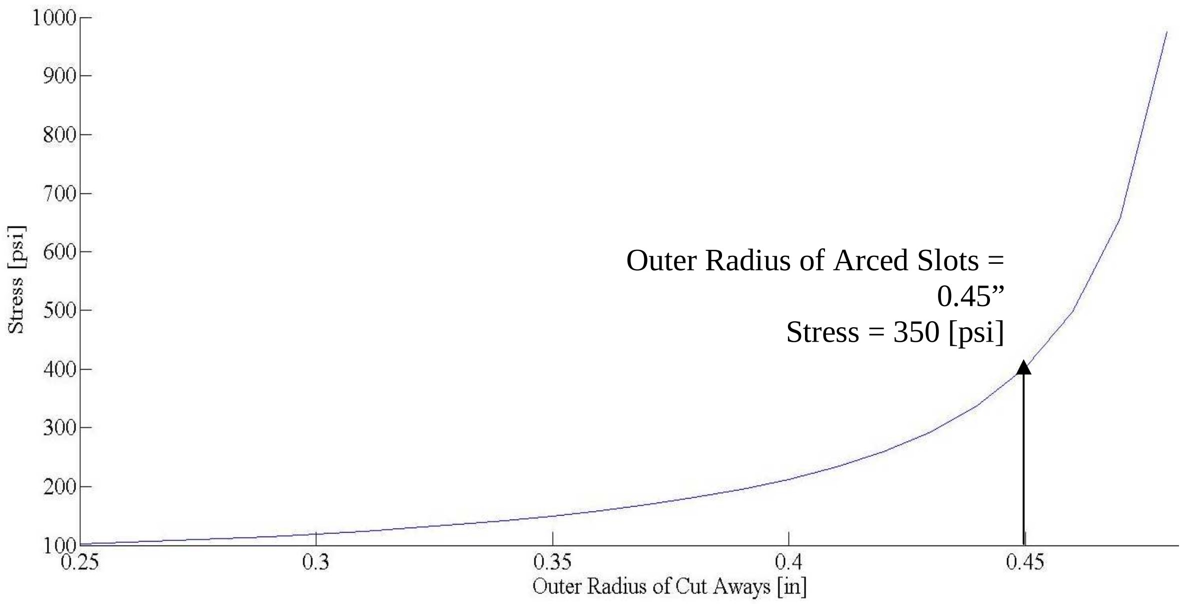


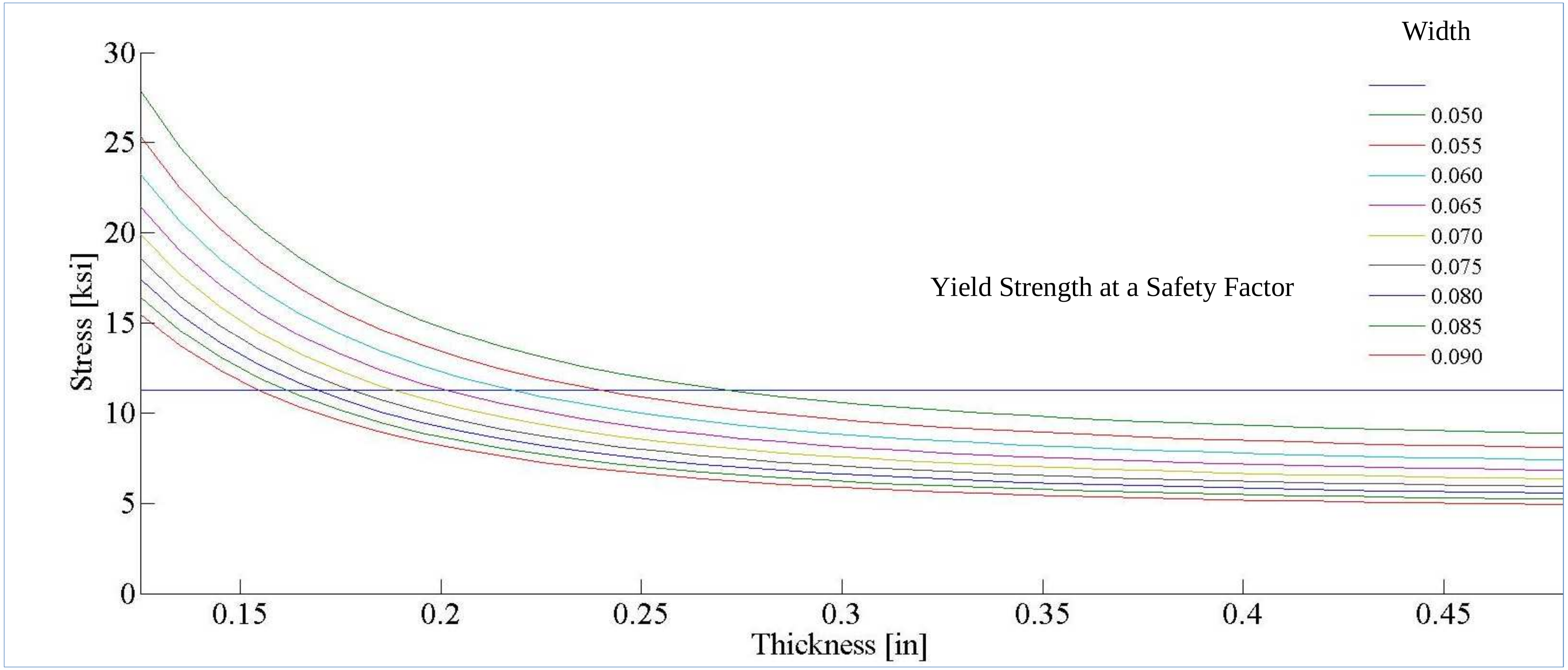
Figure 55: Stress over varying outer radius of cutouts (Ro)



Appendix Q Stress analysis of Retaining Disk spokes

Determines optimal width and thickness of spokes. Modeling force was 60 lbs.

Figure 56: Maximum Stress Determined using Von Mises Criterion at varying thickness and widths



Appendix R MATLAB code for determining stresses in Retaining Ring

generates plots in Appendices M and N.

```
function main = washer

F=60;
ri=.3;
ro=[.25:.01:.48];
rf=.5;

w=[.05:.005:.1];
thickness=[.125:.01:.5];
yield_strength = 45*ones(1,length(thickness));

stress_inner_ring=F/(pi*(ri^2-.25^2))

for i=1:length(ro);
   l=ro(i)-ri;
   a_shell=pi*(rf^2-ro(i)^2);
   stress_shell(i)=F/a_shell;
end
plot(ro,stress_shell);
title('Stress in Thin Shell')
xlabel('Outer Radius of Cut Aways')
ylabel('Stress [psi]')

ro=.45;
l=ro-ri;
stress=zeros(length(w),length(thickness));
stress_y=zeros(1,length(w));

for j=1:length(w);
   width=w(j);
   a_spoke=l*width;
   stress_y(j)=F/a_spoke;

   stress_x=zeros(1,length(thickness));
   for k=1:length(thickness)
      %one third of the force from the acutator will be applied to each
      %spoke
      Force=F/3;
      %stress in x comes from moment created by this force
      t=thickness(k);
      stress_x(k)=Force*l*6/(width*t^2);
      stress_x(k)
      %using Von Mises
```

```
        stress_y(j)
        stress(j,k)=sqrt((stress_x(k))^2+(stress_y(j))*(stress_x(k))+(stress_y(j))^2);
    end

    %str1=num2str(w(j));

end
     Figure

plot(thickness,yield_strength/4,thickness,(stress(1,:)/1000),thickness,(stress(2,:)/1000),thickness,
(stress(3,:)/1000),thickness,(stress(4,:)/1000),thickness,(stress(5,:)/1000),thickness,(stress(6,:)/10
00),thickness,(stress(7,:)/1000),thickness,(stress(8,:)/1000),thickness,(stress(9,:)/1000))
    title(['Maximum stress from Combined Loading at a spoke Width of [in]'])
    xlabel('thickness [in]')
    ylabel('stress [ksi]')
end
```

Appendix S Fluid Analysis Matlab Code

```
%Kinematic Vicosity of oil
nu=1E-5; %m^2/sec
rho=837; %kg/m^3
%gravity (m/s^2)
g=9.8;
%Volume Flowrate
VFR=0.00220815687; %m^3/sec
%Position of Actuator (m)
x=linspace(.001,0.009525,100);
%Diameters (m)
D_Inlet=.0254;
D_Outlet_Recirc=.0254;
D_Outlet_Radiator=.0254;
D_PlugSeat=.0254/2;
D_Actuator = .0254/2;

%R4=D4/2;
%theta4=acos(((D4/2)-x)/(D4/2));

%Areas determined by Diameters
A_Inlet = pi*(D_Inlet/2)^2;                    %Inlet
A_Actuator = ((D_Inlet/2)^2-(D_Actuator/2)^2)*pi; %Actuator
A_AfterPlug = (0.009525-x)*(D_PlugSeat/2)*2*pi;   %Cylinder after Plug
A_PlugSeat = (D_PlugSeat/2)^2*pi;                 %Plug seat
A_Outlet_Recirc = (D_Outlet_Recirc/2)^2*pi;       %Outlet to recirculation

%Areas that aren't determined by Diameters
A_Washer = 0.000225806;  %Washer 0.35 inches^2
A_Plug = 0.000225806;    %Plug 0.35 inches^2
A_Gate = 0.0254 * x;
A_Radiator = 0.0254;


%'----------------------ReCirculating_Line-------------------------------'

%Losses from Expansion and Contraction
KL_Inlet_Washer = ones(1,length(x))*0.3;            %Sudden contraction worst case scenario
KL_Washer_Actuator=(1-A_Washer./A_Actuator).^2 * ones(1,length(x));     %Sudden
Expansion
KL_Actuator_Plug = ones(1,length(x))*0.2;     %Sudden contraction worst case scenario
%KL_Plug_AfterPlug
for a=1:length(x)
   if A_Plug < A_AfterPlug(a)
      KL_Plug_AfterPlug(a)=(1-A_Plug./A_AfterPlug(a)).^2;
   else
```

```
        KL_Plug_AfterPlug(a)=0.5;
     end
  end
  %KL_AfterPlug_PlugSeat
  for b=1:length(x)
     if  A_AfterPlug(b) < A_PlugSeat
        KL_AfterPlug_PlugSeat(b) = (1-A_AfterPlug(b)./A_PlugSeat).^2;
     else
        KL_AfterPlug_PlugSeat(b)=0.5;
     end
  end
  KL_PlugSeat_Outlet_Recirc = (1-A_PlugSeat./A_Outlet_Recirc).^2 .* ones(1,length(x));

  %Total KL
  KL_Recirc = KL_Inlet_Washer + KL_Washer_Actuator + KL_Actuator_Plug +
KL_Plug_AfterPlug + KL_AfterPlug_PlugSeat + KL_PlugSeat_Outlet_Recirc;


  %'----------------------Radiator Line--------------------------------'
  %Losses from Expansion and Contraction

  KL_90Deg_Threaded = ones(1,length(x))*1.5;
  KL_Actuator_Gate = ones(1,length(x))*0.5;
  KL_Gate_Radiator = (1-A_Gate./A_Radiator).^2;

  KL_Radiator = KL_Inlet_Washer + KL_Washer_Actuator + KL_90Deg_Threaded +
KL_Gate_Radiator;


  %---------------------------Velocities (m/s)-----------------------------

  V_Inlet = VFR/A_Inlet;
  V_Actuator = VFR/A_Actuator;
  V_AfterPlug=VFR./A_AfterPlug;
  V_PlugSeat=VFR/A_PlugSeat;
  V_Outlet=VFR/A_Outlet_Recirc;

  V_Washer=VFR/A_Washer;
  V_Plug=VFR/A_Plug;
  V_Gate=VFR./A_Gate;
  V_Radiator=VFR/A_Radiator;

  %-----------------------------Head Losses--------------------------------
  %Contraction => h_L = K_L *(V_2)^2/(2*g)
  %Expansion   => h_L = K_L *(V_1)^2/(2*g)
  %-----------------------------ReCirculating Line-------------------------
```

```
    hL_Inlet_Washer = KL_Inlet_Washer * V_Washer^2/(2*g);           %Sudden contraction
worst case scenario
    hL_Washer_Actuator = KL_Washer_Actuator * V_Washer^2/(2*g);     %Sudden Expansion
    hL_Actuator_Plug = KL_Actuator_Plug * V_Plug^2/(2*g);           %Sudden contraction
worst case scenario
    %hL_Plug_AfterPlug (Expansion or Contraction)
    for c=1:length(x)
      if A_Plug < A_AfterPlug(c)
        hL_Plug_AfterPlug(c) = KL_Plug_AfterPlug(c)*V_Plug^2/(2*g);
      else
        hL_Plug_AfterPlug(c)=KL_Plug_AfterPlug(c)*V_AfterPlug(c).^2/(2*g);
      end
    end
    %hL_AfterPlug_PlugSeat
    for d=1:length(x)
      if  A_AfterPlug(d) < A_PlugSeat
        hL_AfterPlug_PlugSeat(d) = KL_AfterPlug_PlugSeat(d) * V_AfterPlug(d).^2./(2*g);
      else
        hL_AfterPlug_PlugSeat(d) = KL_AfterPlug_PlugSeat(d) * V_PlugSeat^2/(2*g);
      end
    end
    hL_PlugSeat_Outlet_Recirc = KL_PlugSeat_Outlet_Recirc * V_PlugSeat^2/(2*g);
%Sudden Expansion

    %Total =hL
    hL_Recirc = hL_Inlet_Washer + hL_Washer_Actuator + hL_Actuator_Plug +
hL_Plug_AfterPlug + hL_AfterPlug_PlugSeat + hL_PlugSeat_Outlet_Recirc;


    %'----------------------Radiator Line--------------------------------'
    %Losses from Expansion and Contraction

    hL_90Deg_Threaded = KL_90Deg_Threaded * V_Actuator^2/(2*g);
    hL_Actuator_Gate = KL_Actuator_Gate .* V_Gate.^2./(2*g);
    hL_Gate_Radiator = KL_Gate_Radiator .* V_Gate.^2./(2*g);

    hL_Radiator = hL_Inlet_Washer + hL_Washer_Actuator + hL_90Deg_Threaded +
hL_Gate_Radiator;


    %---------------------------------------------------------------------
    %------------------------Delta P----------------------------------------
    %------------------Recirculation Line-----------------------------------

    dP_Recirc = rho * g * hL_Recirc;
```

```
dP_Radiator = rho * g * hL_Radiator;

%Plot of Head Losses
Figure;clf;

subplot(2,1,1),plot(x(1:3/4*length(x)),dP_Recirc(1:3/4*length(x))*0.00014503774,'k','linewidth',2)
title('Recirculation Outlet')
ylabel('\Delta P (psi)')
xlabel('Actuation Distance, x (m)')
grid on

subplot(2,1,2),plot(x(1/4*length(x):length(x)),dP_Radiator((1/4*length(x):length(x)))*0.00014503774,'k','linewidth',2)
title('Radiator Outlet')
ylabel('\Delta P (psi)')
xlabel('Actuation Distance, x (m)')
grid on
```

Appendix T Matlab Code Used to Compute Mach Numbers

```
format long
c=343.3; % speed of sound (m/s)
D=.0254; % Diameter (m) = 1inch
Q=0.00396;%flow rate (m^3/s) = 62.77gpm
%Q=0.002208; %flow rate (m^3/s) = 35gpm

%Actuation Distance
dy=0.000001:.00001:.01016;%actuation distance (m) = 0.4inch

%Area outlet 1 (A1) cylindrical area at plug
y1=.00889; %initial height of A1(m) = 0.35 inch
A1=(D/2)*pi.*(y1-dy);%A1 in terms of actuation (m^2)

%Velocity and Mach number at outlet 1
V1=Q./A1; %velocity at outlet 1 (m/s)
M1=V1/c; %Mach number at outlet 1

%Area outlet 2 (A2) rectangular+circular area at gate

%rectangular (A2R)
w=0.0079375; %width of rectangle (m) = 5/16inch
A2R=w.*dy(33:985); %rectangular portion of A2 (m^2)
%circular
r=0.0047625; %radius (m) = 3/16inch
theta1=acos((r-(dy(33:508)-0.0003175))/r);
A2ca=((r^2).*theta1)-((r-(dy(33:508)-0.0003175)).*(sqrt((r^2)-((r-(dy(33:508)-
0.0003175)).^2)))); %circular A2 for dy<=r
theta2=acos((((dy(509:985)-0.0003175)-r)./r);
A2cb=((pi*(r^2)).*(1-(theta2./pi)))+(((dy(509:985)-0.0003175)-r).*sqrt((r^2)-
(((dy(509:985)-0.0003175)-r).^2)));%circular A2 for dy>r
A2_2=horzcat(A2ca,A2cb);
A2_2=A2_2+A2R; %A2 in terms of actuation (m^2)

A2_1=0.*dy(1:32);
A2_3(1:31)=A2_2(953);

A2tot=horzcat(A2_1,A2_2,A2_3);


%Velocity and Mach number at outlet 2
V2=Q./A2tot; %velocity at outlet 2 (m/s)
M2=V2/c; %Mach number at outlet 2
YMAX1_62=['Outlet 1 test valid for actuation less than ', num2str(dy(600)*39.3700787), '
inches with 62.77gpm flow'];
disp(YMAX1_62)
```

```
YMAX2_62=['Outlet 2 test valid for actuation greater than ' num2str(dy(740)*39.3700787), ' inches with 62.77gpm flow'];
disp(YMAX2_62)

disp(' ')

YMAX1_35=['Outlet 1 test valid for actuation less than ', num2str(dy(728)*39.3700787), ' inches with 35gpm flow'];
disp(YMAX1_35)

YMAX2_35=['Outlet 2 test valid for actuation greater than ' num2str(dy(443)*39.3700787), ' inches with 35gpm flow'];
disp(YMAX2_35)

flow=num2str(Q*15850.3231,3);
plottitle=['Mach Number vs. Actuation Distance for ',flow,'gpm flow of air'];


%plot(dy(255:928).*39.3700787,M1(255:928),dy(255:928).*39.3700787,M2(255:928))
plot(dy.*39.3700787,M1,'b.',dy.*39.3700787,M2,'g.')
axis([0 .4 0 0.1])
title(plottitle)
xlabel('Actuation Distance (in.)')
ylabel('Mach Number')
legend('Outlet 1','Outlet 2')
```