A High Pressure Shut-Off Valve for a Hydraulic Hybrid Accumulator

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1 ABSTRACT

The US EPA is engaged in ongoing research to develop hybrid diesel-hydraulic power systems for delivery trucks. These systems store energy in high pressure hydraulic fluid accumulators, and require a high pressure shut-off valve (HPSOV) to seal them when not in use. Our goal is to design a HPSOV that: a) when "open" allows flow in and out with minimal pressure drop, b) when normally "closed" allows fluid to enter the accumulator but not to exit the accumulator, and c) would equalize the pressure on both sides of the valve when opening. Other safety and reliability concerns will be addressed.

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3 EXECUTIVE SUMMARY

The goal of this Environmental Protection Agency (EPA) sponsored project is to design a multi-functioning valve to be used in new hydraulic hybrid vehicle systems. A hydraulic hybrid vehicle has the ability to store large amounts of energy in the form pressurized hydraulic fluid in an accumulator tank in the vehicle. Hydraulic hybrid vehicles can use this energy storage method to achieve high fuel economy, but this method requires a safe and reliable shutoff valve. The EPA needs a redesigned shutoff valve with specific functions to be used in hydraulic hybrid systems at the inlet/outlet of the high pressure accumulator. The valve must operate in the following modes:

1. Allow fluid flow into and out of the accumulator with minimal pressure drop. (“open”)
2. Allow no fluid to exit the accumulator (no leaks) while allowing fluid to flow into the accumulator when the system is at higher pressure. This is the normal mode of operation. (“closed”)
3. Close if the pressurized bladder within the accumulator tank is pressing against the valve.
4. Close if the flow rate is too high.
5. If the pressure difference between the inside of the tank and the outside of the tank in the system is large, only allow a small flow of fluid into the system, thus preventing noise from “fluid hammer”.

Taking all of these key functions and specific customer requirements into account, we generated a variety of different valve designs. We then began to narrow down our designs using an extensive concept selection process. We evaluated each of the concepts against functionality in three distinct categories; main valve type, valve actuation type, and pressure equalization. After the best valve type, actuation type, and method of pressure equalization were determined we began designing complete valves around our findings. By evaluating every design according to these specific requirements, we were able to determine which concepts were best overall.

Two different alpha designs were selected after this process. Both alpha designs were poppet style valves with high pressure access through the center of the valve stem. Both valves are actuated by a hydraulic piston, but one uses the line pressure while the other uses accumulator pressure to actuate the piston. The main difference between these valves, though, is that one is a straight through valve while the other has a 90 degree turn built into the housing. Since these designs both completed all necessary functions of the valve very well, it was difficult to choose a final design.

These two valves were once again evaluated against specific requirements using pugh charts and a final design was chosen. Once this design was set, we were able to design the prototype. The prototype internals were made mostly from PVC and Nylon 6/6 plastics, but the housing was made from aluminum to provide maximum strength for the threading interface with the accumulator. For safety and simplicity reasons, the prototype was tested and displayed with water at a maximum pressure of 50psi. The prototype was successful in validating the leak-in, open, and bladder interaction features of the valve design as well as fitment to the current EPA system.

The final valve design is very similar to the first alpha design, with one major alteration. The valve is a poppet style valve with fluid flow straight through. Bladder interaction was accounted for in the final alpha design with the use of a spring. This spring will detach the stem and poppet from the actuating cylinder inside the housing. If the bladder inside the accumulator expands to the point of interacting with the valve, it will simply push against the spring loaded valve stem, closing the valve. The final valve was designed to allow for simplicity of operation and manufacture. The number of parts is significantly reduced from the EPA’s current valve and most of the parts are symmetrical, allowing for simpler manufacturing. These attributes make this final valve design much less complex, while still addressing all of the functionality requirements.

4 INTRODUCTION

We are designing a high pressure shut-off valve (HPSOV) to interface with a high pressure accumulator and hydraulic lines within a hydraulic hybrid vehicle system. The valve itself serves many functions as it allows hydraulic fluid into and out of the accumulator at varying rates, allows the lines and accumulator to be filled with fluid, and provides a safety shutoff function.

4.1 System Background

The United States Environmental Protection Agency (EPA) engages in extensive research to develop and improve hybrid systems for powering automobiles. One such hybrid system is called a series hybrid diesel-hydraulic system. Like electric hybrid systems, the most important premise of a hydraulic hybrid system is to recover and store energy that is usually wasted while the vehicle is braking. In an electric hybrid, about 20-30% of the vehicle's kinetic energy can be recovered and stored in a battery [1, 2]. The battery then can release this stored energy to drive a DC motor that can help the engine accelerate the vehicle, thus improving the fuel economy of the vehicle. In a hydraulic hybrid, up to 70% of the vehicle's energy can be recovered from regenerative braking in the form of pressurized hydraulic fluid [1, 2].

The hydraulic system in a hydraulic hybrid vehicle (HHV) has a high pressure accumulator that stores energy in the form of high pressure hydraulic fluid. The high pressure accumulator uses a nitrogen gas-filled bladder to maintain high pressure in the tank. The energy is stored by using the turning of the vehicle's wheels to pump the hydraulic fluid to the accumulator, an action which also slows down the vehicle (see Figure 4.1). This regenerated high pressure fluid is stored in the accumulator, where it can be used later. When the energy needs to be released from the accumulator, the pump reverses direction and the high pressure fluid drives the pump/motor to accelerate the vehicle (see Figure 4.2). The low pressure fluid that results is stored in a low pressure reservoir.

Figure 4.1: While braking, hydraulic fluid is pumped to the accumulator to store energy.

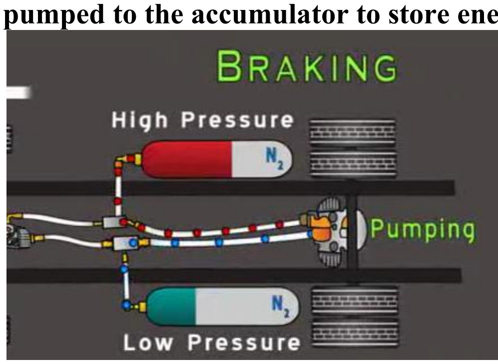
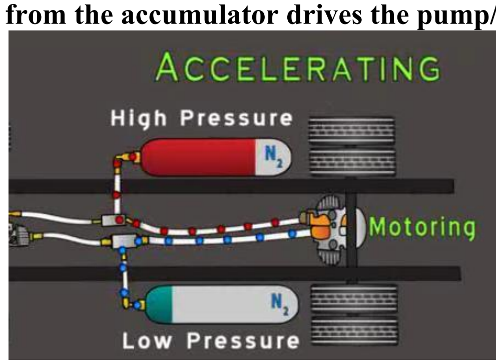


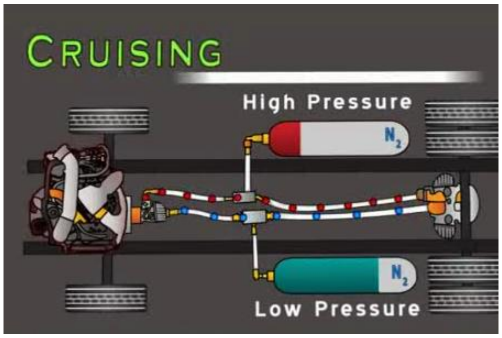
Figure 4.2: While accelerating, hydraulic fluid from the accumulator drives the pump/motor.



The most noticeable feature of a hydraulic hybrid system is that there is no mechanical connection between the engine and the driven wheels of the vehicle. Instead, the engine is coupled with a hydraulic pump, which pumps hydraulic fluid from a low pressure zone to a high pressure zone. This hydraulic fluid then travels to the back of the vehicle, where a hydraulic motor turns the wheels while reducing the pressure of the hydraulic fluid (see Figure 4.3). The addition of the regenerative braking ability for accumulator tank system results in an overall boost in fuel economy (40 – 50% increase) [3].

The operations described above can greatly increase the efficiency of a vehicle, especially at low speeds and while stopping and starting a lot in traffic. When such a system is added to a vehicle that stops and starts often, the system can become so efficient that the engine rarely has to run. The engine, in this case, would be used only when the accumulator does not have enough stored energy to power the vehicle. This is the advantage of a series hybrid, in that the engine can be turned off completely, but the system can still operate the vehicle (if there is enough energy recovered in the braking of the vehicle, the high pressure accumulator can run the vehicle without help from the engine). The relative simplicity and durability of the components of a hydraulic hybrid compared to that of an electric hybrid allows for significantly lower maintenance costs. In addition, the hydraulic hybrid system does not contain hazardous materials that are harmful to the environment, like the battery in an electric hybrid.

Figure 4.3: While braking, hydraulic fluid is pumped to the reservoir to store energy.



On a hydraulic hybrid vehicle, there is only one hydraulic motor that is coupled with both the engine and the hydraulic accumulator systems. In this way, it can be seen that the hydraulic system is "in series" with the engine (in contrast, an electric hybrid is usually "in parallel" with the engine). There are many benefits to combining these systems in series. First, while the engine is running to meet the demand of the rear wheels, any excess pressure can be stored in the accumulator. This often can happen at cruising speeds, where little acceleration is needed.

Second, while accelerating, the accumulator releases energy to make up for any demand of acceleration that the engine cannot meet. This fact allows for the engine not to be sized for maximum acceleration of the vehicle, but rather for the needed acceleration while at cruising speeds. This can allow for a higher operating efficiency of the engine overall. In the same token, the engine can be designed to have a certain RPM where it is most efficient (instead of a range of RPMs that it is only mildly efficient, like a typical engine). This design has the advantage of being able to be operated only at this RPM, and can cycle on and off in response to need, thus further increasing the overall efficiency of the engine. While the engine is not sized for maximum acceleration, the rear motor/pump must be in order to meet the demand of the driver.

Figure 4.4: One of the first fully assembled hydraulic hybrid delivery vehicles.

Third, if the vehicle is in city traffic or otherwise undergoes many stops and starts, the system rarely reaches cruising speeds, and the engine does not have to operate at all. The accumulator system has the ability to release energy when accelerating and recover that energy when decelerating, and the vehicle can be driven solely by the hydraulic system for an extended time. The accumulator does not recover 100% of the energy released during acceleration, so eventually the engine will have to be used to recharge the accumulator and drive the vehicle. It is because of this third advantage to the hydraulic hybrid system that the EPA has partnered with UPS to improve the fuel economy of delivery trucks, which stop and start often (see Figure 4.4). It is estimated that the cost of the technology for this system on a single UPS delivery truck can be recuperated in less than three years for a heavily used vehicle [4].

4.2 Problem Description

The hydraulic system inside a hydraulic hybrid requires safe and reliable high pressure valves to be an acceptable alternative to conventional or hybrid electric vehicles. The hydraulic fluid in the high pressure zone of the system can reach pressures of up to 7000 psi, which can be dangerous if not handled properly. The shut-off valve that needs to be integrated into the high pressure accumulator is not a commercially available product. Most products currently on the market do not have the ability to perform all of the required functions that the valve must perform. The EPA has created a valve that does accomplish all of the required functions, but the design process of this valve was step-by-step. The valve was created to accomplish certain functions but new functional requirements were identified while testing the valve. The designers had to modify the valve to account for the new requirements, but this step-by-step modification process created a valve that is complicated, expensive and not durable. It is with this in mind that the EPA requested that a high pressure shut-off valve (HPSOV) be designed that can satisfy the requirements for the system, while keeping complexity and cost down, and while increasing durability.

The valve that the EPA has requested has multiple functions that are all necessary to enable the efficient and safe operation of the system. The primary function of the valve is to be a normally closed shut-off valve that operates at a range of pressures between 2000 and 7000 psi. In this “closed” position, the valve must not allow any fluid to exit the accumulator (no leaks), but must be able to let fluid into the valve if the pressure in the line is higher than the pressure in the tank. The “open” position of the valve must allow fluid to enter and exit the valve with minimal pressure drop. This “open” position should be activated with an applied 10-12 V signal.

Another important function of the valve is included to prevent excessive noise that is created be large pressure gradients within the line (also known as “fluid hammer”). This function is accomplished by incorporating a “leak-only” mode when the pressure inside the line is much less than the pressure in the accumulator. A challenge of this design is to incorporate a “leak-only” mode into the valve that can safely pressurize the line without causing a long delay in the operation of the vehicle. Ideally, the orifices that allow pressure equalization will be tunable to account for varying sizes of the lines that need to be pressurized.

The valve must also have two other safety functions. First, if the flow rate is too high, it indicates an irregularity in the system, and the valve needs to be closed. This means that the valve would need to incorporate a “velocity fuse” to address this need. Secondly, there is a danger that the pressurized Nitrogen gas-filled bladder may rupture if it presses against the valve opening. The designed valve must have the ability to close if the bladder presses against the valve, however, fluid must be allowed back into the accumulator if the pressure in the line increases. In both of these safety situations, no fluid must be allowed to exit the accumulator.

Another important consideration that we need to include in our design is high reliability. First, the valve must be made so that it resists corrosive conditions of the road (such as salt and water). Second, as a road-going valve, it must be able to survive impacts due to small stones. Third, the valve must be not fail due to many cycles of operation. Fourth, the valve must be reliable for a range of operating temperatures. As an additional consideration, the valve must be compatible with Mobil 1 Automatic Transmission Fluid (ATF) at a range of low and high temperatures. In addition, the inclusion of an accessory port is a desirable outcome, and can be used to do an initial fill of the hydraulic system.

The most important outcome of this project is to develop a design for a HPSOV that satisfies all of the technical and functional requirements that the EPA needs it to satisfy. A fully functional and tested prototype may not be possible due to budget constraints and high pressure requirements. The EPA may also find the ideas that were not selected for development as valuable insight into the system and the valve that may be developed by them in the future. Consequently, any important or interesting ideas that we create will be documented and saved for presentation to the EPA at a later date. As with any design, a lighter, smaller, and more cost effective design will be preferable and will be considered in the design process. As an additional challenge, our design will be designed with considerations for ease of manufacturability, maintenance, and overall simplicity.

5 INFORMATION SOURCES

The performance of hydraulic hybrid vehicles has been documented by various developers. A large number of data that we have on HHV performance are from projects conducted at the EPA. For example, it was the EPA who stated that a 30-70% increase in fuel economy has been achieved in current HHVs [4].

Specific information in regards to the goals and deliverables of our project was obtained through meetings with our two EPA contacts, Andrew Moskalik, and Jim Bryson [5]. At these meetings, a detailed overview of hydraulic hybrid vehicles and the current state of the art was given to us. The main progress up to this point was discussed along with motivation for the HPSOV project itself. It was explained how the current HPSOV lacked refinement and was in need of a redesign that could more smoothly integrate all features that have been implemented to date in a safe, reliable package.

5.1 Valve Performance Sources

Nesbitt [6] gives a detailed overview of different styles of valves that are currently available on the market, however little design information is presented. On the other hand, Smith and Zappe [7] and Skousen [8] give useful procedures and information for the correct sizing of existing valves for various applications. Multitudes of valve styles are discussed including globe valves, parallel gate valves, plug valves, ball valves, and butterfly valves. Check valves and pressure release designs are also discussed. The sizing information presented will be useful when selecting the best configuration of our HPSOV. A thorough presentation of valve design including fluid mechanics, valve geometry selection, design of mechanical components, and design analysis is given by Lyons [9]. While not exclusive to valve applications, specific information regarding both static and dynamic seals is provided by Brown [10]. In addition to discussions on the design of various sealing joints, a seal selection guide is also provided which includes considerations ranging from maximum operating pressures to working fluid compatibility.

When trying to predict the performance of our HPSOV concept before testing, a helpful resource will be the extensive information on valve coefficients available in the literature. Valve coefficients make it possible to predict pressure drops based on the required flow rates, and the valve geometry. Many different flow configurations and valve styles have been studied and their quantitative design parameters have been documented. Using a combination of this information, it should be possible to predict the performance our prototype will achieve. Lyons [9] describes a procedure whereby the performance of a complex valve body can be estimated based on coefficients for the smaller individual sections that make up the valve. Using the coefficients and design data that are available, along with appropriate dimensional analysis techniques, full scale performance can be predicted from prototype testing.

Due to testing conditions that may differ from the real operating conditions, it will be necessary to scale all test data in such a way that full scale performance parameters can be accurately predicted. Lyons [9] gives a description of various dimensional analysis techniques that can be used for this purpose. Typical fluid dynamics textbooks also cover dimensional analysis and may be used as well.

A factor that will serve as a useful comparison between our HPSOV and other valves is the pressure drop across the valve for the various operating conditions. A high pressure drop is undesirable because it results in a decrease in overall efficiency of the HHV system. The higher the pressure drop, the more energy will be lost in any fluid transfer to or from the high pressure accumulator. Additionally, a large pressure drop will prevent the other system components from operating at their design conditions. The benchmarking of our HPSOV will be done against conventional valve designs, as well as the current EPA design. However, since limited data is available for the current EPA HPSOV, comparisons may need to be more qualitative such as how well the concept performs the required functions versus the complexity of the valve.

5.2 Common Valve Types

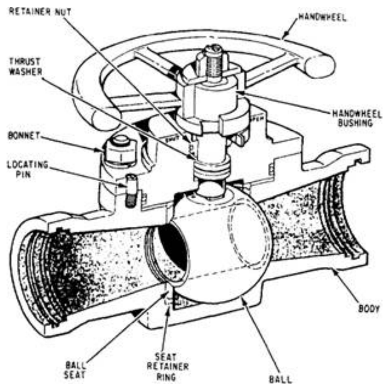
Before beginning the design process of our own valve it was necessary to gain additional understanding of valves themselves, as well as what kinds are currently on the market. Due to the specific functions of the HPSOV, it is unlikely that an off-the-shelf valve can be purchased from a manufacturer and integrated directly into the hybrid vehicles. Nevertheless, some valves have many desirable performance characteristics that the HPSOV could benefit from. Since the HPSOV will be largely based off of existing valve styles, a careful analysis of the performance of existing valves is needed to ensure the highest performance of the HPSOV. A review of currently available literature on the subject of valve design provided many good resources that will be used in our own detailed design process later in the project.

In studying the performance of existing valves, we tried to find and identify which valves had advantages that could be incorporated into our HPSOV. The majority of the fluid will pass through the HPSOV through one main passageway. The valve that is part of this passageway should be compatible with the operating pressures and flow rates required for the HHV application.

5.2.1 Ball Valve

It was found that ball valves have been successfully used in high pressure applications. In addition, many designs feature a flow passage that maintains constant cross sectional area through the entire valve. The result is minimal pressure drop, even for very high flow rates. These characteristics make the ball valve very attractive for our application. On the other hand, the opening/closing motion of the ball requires 90° of rotation about a perpendicular axis to the flow passage. This is a disadvantage for two main reasons. First, the large rotation would require a large sweep if a linear actuator was to be used for position control. Second, the large travel will be detrimental to valve response time. Considering the required response time for the HHV application, the ball valve may be difficult to implement. See Figure 5.1 for a typical cutaway view of a ball valve.

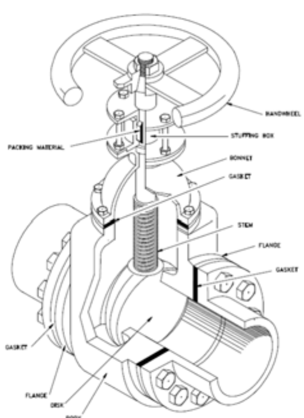
Figure 5.1: Cutaway representation of a ball valve (copied from [11])



5.2.2 Gate Valve

Gate valves offer a low restriction to flow similar to ball valves. In this design, a gate is linearly moved into the flow path of the valve to completely close the passage. These valves also have a relatively large sweep from the open/closed positions which could be detrimental to response time. In addition, these valves also tend to be bulky due to the space required by the gate when the valve is in the open position. Because of their size, use of these valves has typically been limited to stationary applications. See Figure 5.2 for a typical cutaway view of a gate valve.

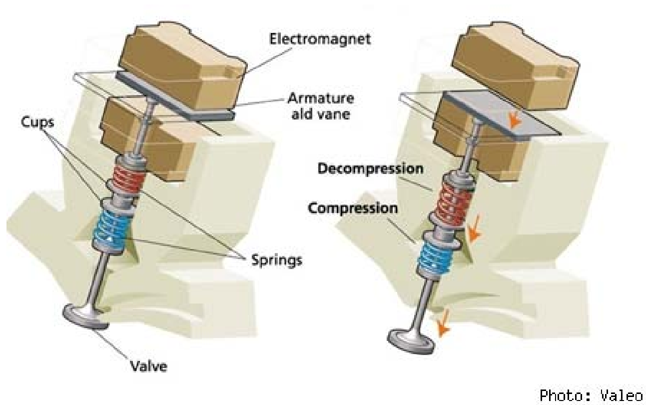
Figure 5.2: Cutaway representation of a gate valve (copied from [12])



5.2.3 Poppet Valve

The poppet style valve is used in the current HPSOV design for the EPA. With proper design, poppet valves can be used in high pressure applications. The relatively short actuation allows fast response times while keeping pressure drops reasonable. As demonstrated in the current EPA design, the linear motion of the valve can be adapted well to hydraulic actuation. The position of the main valve can be controlled by a smaller electronically actuated servo valve. This will direct high or low pressure fluid to the correct side of a positioning piston, depending on which main valve position is desired. The positioning piston is attached to the poppet valve shaft which will cause the valve seat to move into either the open or closed position. A similar operating principle can be applied to most linear motion valves, however the corresponding operating pressures will still be limited based on valve style and sealing capability. See Figure 5.3 for a cutaway view of a poppet valve as used in a cam less piston application for an internal combustion engine.

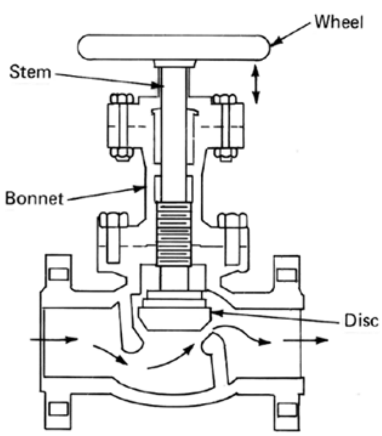
Figure 5.3: Cutaway representation of a poppet valve as used in a cam less piston application (copied from [13])



5.2.4 Globe Valve

Globe valves are another common style of valve. The valve and valve seat are similar to the poppet style, except the entire assembly is enclosed within a housing. The housing is typically designed to accommodate either flange or threaded connections on both the inlet and outlet. Pressure drop is typically higher across this valve as a result of the more complicated turning required by the fluid flow. See Figure 5.4 for a typical cross sectional representation of a globe valve.

Figure 5.4: Cross sectional representation of a globe valve (copied from [14])



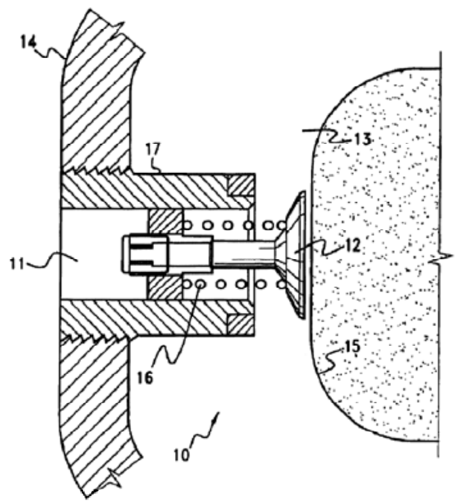
5.3 Patent Search

An extensive and thorough patent search was conducted to explore possible valve designs for this application. While searching for patents that related to the design of a hydraulic accumulator shutoff valve, many specific function valves were found. Other patents that were looked at in our search were either not fully functional from the HPSOV standpoint, or were for different applications than what we are designing for. Those patents that were found will not be included in this paper. Two patents, however, told us a lot about how we should be going about our project, and they are discussed below. The documenting of these patents will allow us to avoid patent infringement while creating our alpha designs.

5.3.1 EPA Patent (6,619,325)

One significant patent that was found was done directly for the EPA. This valve patent focuses on bladder interaction that can be provided by a spring loaded poppet valve, and included in this patent are many applications of this interaction function into full valve designs. Figure 5.5 shows the spring loaded poppet valve configuration. The spring is held between the head of the poppet valve and a poppet valve holder. The spring loaded poppet valve described in this patent allows for both bladder interaction and velocity fuse functionality. When the bladder presses on the head of the poppet valve, it will begin to extrude, and an extrusion force will be applied to the assembly. The force balance due to pressure is equal to zero, so this bladder force needs to overcome only a spring to close the valve, thus preventing harm to the bladder. In addition, during periods of high flow, the pressure on the left side of the poppet head will decrease, and the pressure difference that arises will act against the spring to close the valve, thus creating a velocity fuse. The use of a spring to accomplish both bladder interaction and velocity fuse functions is an idea that could be used in our final design.

Figure 5.5: Spring loaded poppet valve [15]



The many valve configurations that this patent shows were also considered in designing fully functional valves during our concept generation phase. Figures 5.6, 5.7, and 5.8 below show three configurations that the inventor designed that were considered during concept generation. Figure 5.6 shows the valve in conjunction with an actuation cylinder. This valve configuration allows an actuation mechanism to attach to the poppet valve, thus allowing the valve to be actuated closed without bladder interaction or high velocities. A possible disadvantage of this design may be the need for thicker sections than shown on Figure 5.6 and thus a reduced flow area.

Figure 5.7 shows the valve in conjunction with a 90 degree bend, with actuation behind the bend. This valve configuration also allows the valve to be actuated closed. A possible advantage of this design is the removal of the actuation method to beyond the bend, which could reduce pressure drop within the valve. In addition, the inclusion of a 90 degree bend is desirable from the EPA’s standpoint for use in the hydraulic hybrid vehicles. A possible disadvantage of this design is the need to machine much complex geometry which is needed to reduce pressure drop in not only the valve stem holder but also the actuation stem holder and the piece that attaches them together.

Figure 5.6: First valve configuration in EPA patent [15]

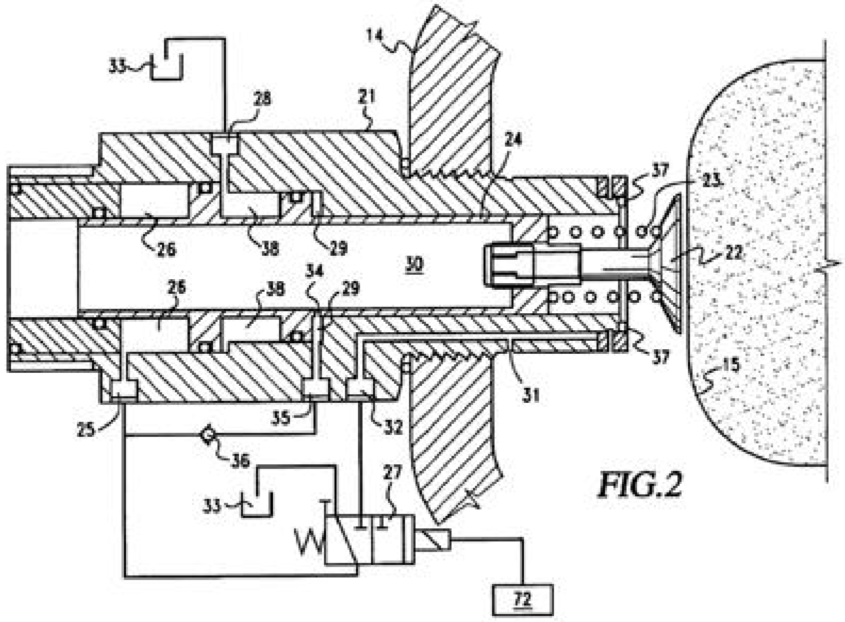


FIG.2

Figure 5.7: Second valve configuration in EPA patent [15]

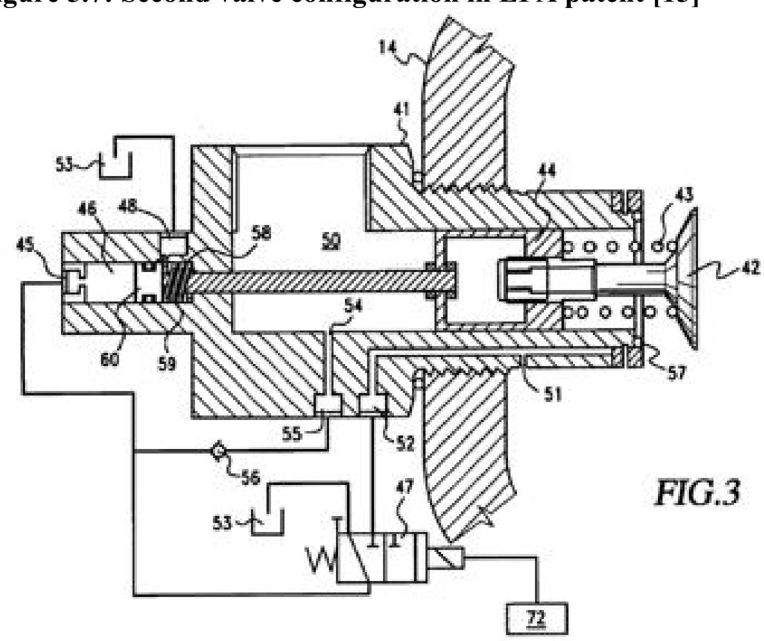
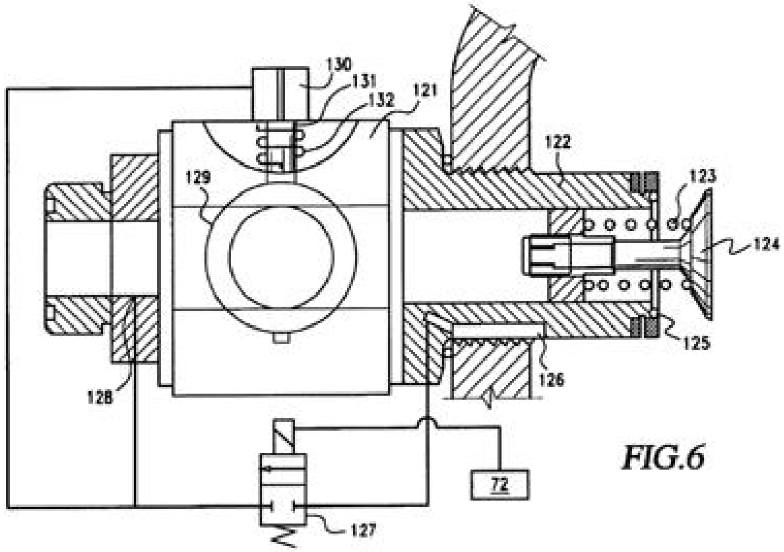


FIG.3

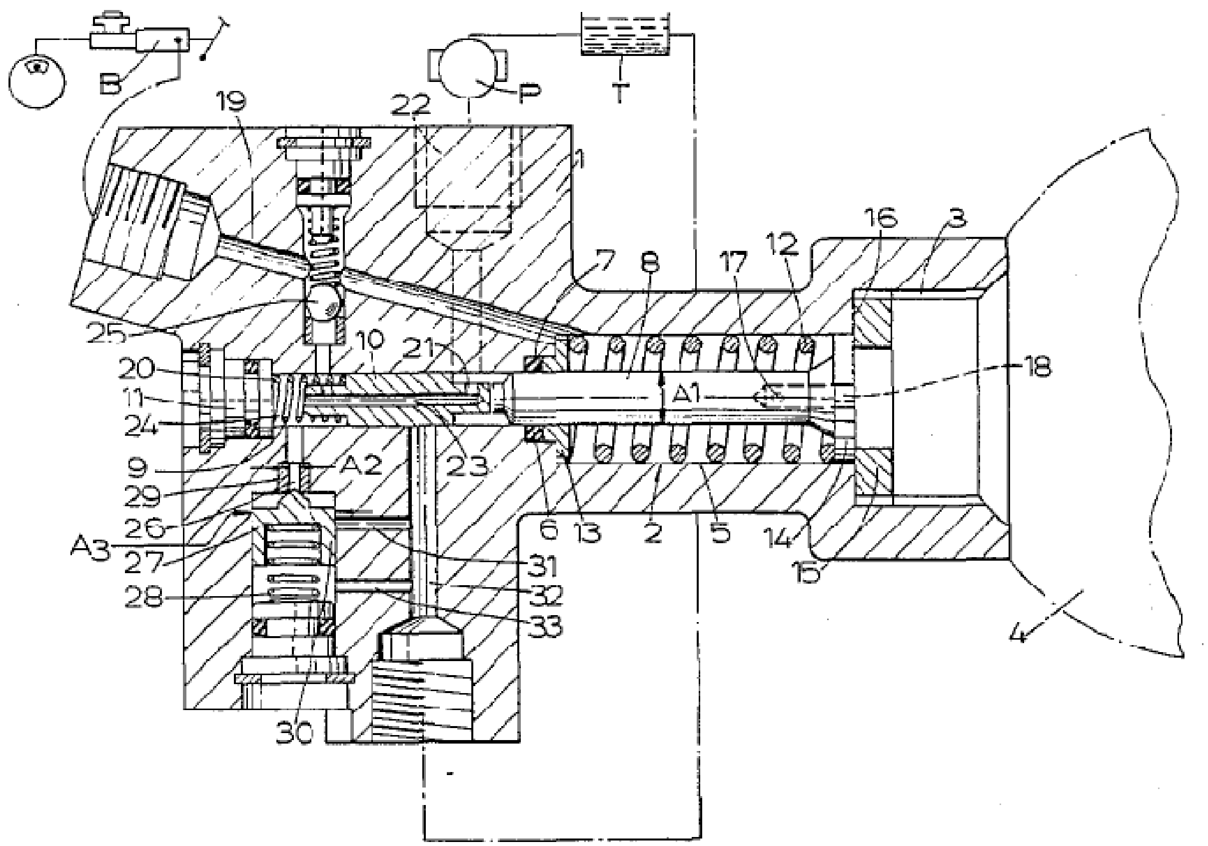
Figure 5.8 shows the valve in conjunction with a ball valve. This valve configuration shows that the valve could be actuated closed in a different position, while leaving the bladder interaction and velocity fuse functions to be accomplished separately. This design is essentially two valves in series, with one valve being used for opening and closing the valve. In addition, the external system that bypasses the ball valve acts to equalize pressure in the line to that in the tank while opening the large ball valve (acting like a valve parallel to the ball valve). Disadvantages of this design include the large activation time that is typical of a ball valve and the large size and weight of a system of three separate valves. An advantage of this design is the low pressure drop through the ball valve, relative to the other designs.

Figure 5.8: Third valve configuration in EPA patent [15]



5.3.2 Patent 4,543,976

Figure 5.9: Complicated shut-off valve design patent [16]



Another patent was found that also accomplishes all of the functions that the EPA requires from their valve. Bladder interaction and velocity fuse are accomplished in its poppet-like head. Multiple orifices are shown in the housing, which are required to be at strange angles in order to accomplish the other functions. The valve is able to be actuated behind the flow if the line flow is used as both an input and an output to the valve. This amount of complexity will result in a large pressure drop. The valve system that we are designing for is attached to only the accumulator, and not the line that heads to the pump and the motor in the hydraulic hybrid system. This valve was designed to interact with the entire system. This valve is more complicated and more expensive to manufacture than the valve the EPA is currently using, so a full system integrated valve will not be considered in this project. A sectional view of this design is shown in Figure 5.69.

6 ENGINEERING SPECIFICATIONS

Since this valve has been previously designed and built and must fit into an existing system, most of the project requirements are actually engineering specifications themselves. When we first met with Andrew Moskalik and James Bryson from the EPA, we were provided with a one page summary of the most important functions and quantitative specifications of the HPSOV (See Appendix A). This sheet was created by engineers at the EPA during a design review of their current valve. This sheet lists many of the operating conditions, necessary functions, and quantitative requirements for the valve. The main requirements that the valve must meet involve functionality, safety, reliability, and cost. Collating all of these specifications gives us an extensive list of qualitative and quantitative requirements to be met. Because the system has already been designed and built and we are essentially redesigning and improving a current part, the valve we design must be completely compatible with the current system. This means meeting size, shape, fitment, and electrical requirements that are already set in place.

6.1 EPA Requirements

The three basic positions of the valve are “open”, “closed”, and “leak only”. When open, the valve must provide flow into and out of the HP accumulator. It must also be able to open quickly; within 150 to 300 milliseconds. The valve must provide one way flow into the accumulator when closed and pressure in the line exceeds that of the accumulator. In the event of a system stoppage or failure, the valve must default to the closed position. It must also close within 150 to 300 ms to both avoid high impacts upon closing, as well as improve efficiency by decreasing energy lost after “close” signal is sent. The valve must also include a pressure equalization function to decrease the sharp pressure rise and noise when opening the high pressure accumulator. The valve must accept a 12V power source. It must also function on a 10V power source in case of a power loss. During the pressure equalization function, some of the high pressure fluid from the accumulator will “leak” into the hydraulic lines on the other side of the valve and equalize the pressures just before the valve is actuated to the open position. This action will prevent loud noise and fluid hammer as the valve is opened. This equalization must happen between 50 and 100 milliseconds.

The high pressure accumulator will be working at pressures ranging from 2000 to 7000 psi meaning that the valve must be able to handle the Mobil 1 Automatic Transmission Fluid at or slightly above these pressures as well as operate successfully at flow rates of 80 GPM nominal and 140 GPM maximum. The valve must be able to operate at a range of pressures from 2000 to 7000 psi in the high pressure accumulator side, 80 to 250 psi in the low pressure tank, and 80 to 7000 psi in the line. The valve will need to operate in temperatures from -40F to 140F and the hydraulic fluid in the lines and valve will be at temperatures of -40F to 180F, so we must make sure the materials used are compatible.

Another set of specifications that relate specifically to the geometry are the constraints of fitting the existing system. Our valve design needs to be able to screw into the HYDAC high pressure accumulators that the EPA is currently using on one end, and it needs to have a 90 degree bend from the tank inlet with a 4 bolt 62 flange attachment on the end attached to the system hydraulic line.

As with any other mechanical system, especially motor vehicles, safety is a top priority and the valve must not leak, break, or allow a rapid release of pressure. Because safety is a big issue, the safety features and checks of the valve are some of the most important components. If the nitrogen bladder inside the high pressure accumulator becomes over-pressurized and reaches the valve itself, there is a possibility for rupture if the bag is pinched in the valve. To prevent this, the valve must be able to close upon contact with the bladder, but must also be able to allow flow of high pressure fluid in the line back into the accumulator. If the flow rates exceed 185 GPM the valve must close as this would indicate a large leak somewhere in the system.

This specific valve and system is designed for extended use on actual road going delivery trucks, reliability and the ability to withstand road hazards are very important issues. Corrosion from salt, impacts of stones, and overall wear and tear must be accounted for in the design of the valve and the choice of materials. The EPA and UPS estimate that throughout the delivery vehicles lifetime, the valve will complete around 1,000,000 cycles. This equates to about 40,000 cycles per year and 200 cycles per day. Designing the valve for this kind of continuous use is a top priority.

Currently, the valve that the EPA has designed and built costs nearly $15,000 to produce, therefore the reduction of cost is an important aspect of our design. This high cost is a major deterrent in making hydraulic hybrid systems feasible for use in vehicles. Our goal is to bring this cost down with a streamlined design and reduction of the number of parts, and the complexity of machined parts.

6.2 Quality, Function, Deployment (QFD) Diagram

When brainstorming, it is best to have a good idea of what parts of the current design are in most need of modification, or what parts of the design are most important. Knowing this information will allow us to focus our brainstorming to better create ideas that will benefit our customer (the EPA). In order to better understand which design parameters were most important to the completion of our project, we created a QFD diagram that gives us the opportunity to compare the customer requirements and design parameters to understand which design parameters are most important. Below is a list of the customer requirements and their customer weights (as supplied through our EPA contacts, Table 4) as well as a list of the design parameters that we chose that represent the valve well (Table 6.1). The customer weights (Table 6.2) and the correlation matrix (Table 6.3) scales are also shown below. According to our EPA contacts, most of the customer requirements are of great importance, so our customer weights scale reflects this fact. The correlation matrix scale was applied to compare the customer requirements and engineering specifications for direct and indirect influences. The QFD Diagram is in Appendix B.

Table 6.1: Design Parameters as chosen by the members of the group to best describe the final valve design.

	Design Parameter:	Target:	Unit:
1	Overall cost	<15,000	$
2	Number of parts	<20	
3	Overall weight	<20	lbs
4	Tank side outer pipe diameter	2	in.
5	Tank side inner pipe diameter	TBD	in.
6	System side outer pipe diameter	TBD	in.
7	System side inner pipe diameter	1.255	in.
8	Threading size	TBD	th./in
9	Threading length	TBD	in.
10	Flow out of accumulator when de-energized (leak)	0	gpm
11	Pressure drop when energized	TBD	ft. hd.
12	Works at 2000 psi	yes	
13	Works at 7000 psi	yes	
14	Flow into accumulator when de-energized	TBD	gpm
15	Pressure difference required to allow flow into accumulator	TBD	psi
16	Flow out of accumulator when "bottomed" (leak)	0	gpm
17	Flow into accumulator when "bottomed"	yes	
18	Flow external to valve (leak)	0	gpm
19	Diameter of equalization orifice	TBD	in.
20	Variability in diameter of equalization orifice	TBD	in.
21	Time to pressure equalization	50-100	ms
22	Materials (resistance to road wear)	steel	
23	Materials (resistance to temperature failure)	steel	
24	Materials (resistance to cyclical failure)	steel	
25	Maximum flow rate	130-140	gpm
26	Velocity fuse	185	gpm
27	Reaction time to close	300	ms
28	Reaction time to open	150	ms
29	Activation voltage	10-12	V

Table 6.2: Customer weights scale

Customer Weight Scale:		
5	→	Essential
4	→	Less Essential
3	→	Important
2	→	Less Important
1	→	Secondary

Table 6.3: Correlation matrix scale

Correlation Matrix Scale:		
9	→	Strongly Correlated
3	→	Correlated
1	→	Weakly correlated
0	→	Not at all correlated

Table 6.4: Customer requirements and weights as provided by the EPA contacts

	Customer Requirement:	Weight:
1	Low cost	2
2	Low complexity	1
3	Designed for manufacturability and maintenance	3
4	Low weight	1
5	Small size	1
6	Low noise	4
7	Compatible with accumulator	4
8	Compatible with pipes of system	4
9	Stop flow when de-energized (no-leaks)	5
10	Flow in both directions when energized (low pressure drop)	5
11	Works at operating pressures	5
12	Allows flow into accumulator when de-energized	4
13	Stop flow when "bottomed"	4
14	No external leaks	3
15	"Bottoms" with bladder and piston-in-shell accumulators	4
16	Gradual pressure equalization to prevent fluid hammer	4
17	Not adversely effected by road conditions (-40F to 140F)	3
18	Operational at oil temperatures of -40F to 180F	3
19	Compatible with Mobile-1 ATF	3
20	Shuts off at extremely high flow rates	4
21	Must energize and de-energize quickly	4
22	Does not fail due to 1,000,000 cycles	4
23	Energizes within a range of acceptable voltages	4
24	Normally closed	5

6.3 Functional Decomposition

To produce a high pressure shut off valve design that meets all requirements, the specific functionality of the valve has been broken down. This data is presented in two ways. First, a text based functional decomposition is given below in Table 6.5. This breakdown lists all design requirements and functionality that must be included in the final HPSOV design. Additionally, this information has been adapted to a graphical flowchart and is presented in Appendix C.

This flow chart helped us understand what kind of designs we had to focus on. In order to break it down even further, we created a valve scenario chart (see Appendix D), which describes all of the separate conditions that this valve will undergo, and the desired action the valve must take under each condition. Finally, to better understand which engineering concepts will be needed for the design of the high pressure shut off valve, the necessary valve functions have been broken down into the areas of expertise (engineering fundamentals) that will be required in their design. This breakdown is presented in Appendix E.

Table 6.5: Valve Functions and Requirements

- Provide two way flow with minimal pressure drop when valve is in open position (Governing equations: Fluid dynamic, head loss, k-factor)
  - Speed of valve opening and closing - 150 – 300ms
  - Handle required flow rates - 80 GPM, nominal / 130 – 140 GPM, max
- Correctly operates at all pressures from 2000 – 7000 PSI
- Provide one directional flow into accumulator when valve is closed during line overpressure situation
- Prevent damage or extrusion of gas bag
- Stop flow from exiting accumulator when almost empty
  - Continue to allow flow into accumulator
- Equalize line and accumulator pressure in less than 100ms before main valve opening
  - Incorporate tuning orifices for any necessary modification of equalizing time
- Interface with current vehicle systems (accumulator, pressure lines, etc.)
  - Accept a 12V control signal from vehicle, down to 10V
- Default to closed position
- No leaks, internal or external
- Auto shut off when flow rate >185GPM
- Durability under design conditions
  - Handles 7000 PSI working pressure
  - All materials compatible with Mobil 1 ATF
  - Durability and strength to withstand road hazards (rocks, salt, water, etc.)
  - Works at ambient temperatures of -40°F to 140°F and oil temperatures of -40°F to 180°F
  - Must survive about two hundred cycles a day and a one million cycle lifetime

7 CONCEPT GENERATION

We wanted to come up with as many ways as possible for the valve to complete the required functions. No idea was thrown out for being too outrageous because thinking outside of the box and starting from scratch is a big part of what the EPA wanted. In order to accomplish the task of concept generation, we used our functional decomposition, QFD diagram, and customer specifications to brainstorm ideas on how to solve the various problems that might arise in our design. There are three main kinds of functionality to focus our concept generation on: actuation method, main valve type, and pressure equalization method.

7.1 Actuation Methods

For any method of valve actuation, a suitable energy source must be available. Once an energy source has been selected, a suitable system can then be built to transform that energy into a valve movement. The two energy sources readily available on highway vehicles include DC electricity from the vehicles battery, and torque from the vehicles engine. In hydraulic hybrid vehicles, an additional source of energy is the stored fluid in the vehicles high pressure hydraulic accumulator.

The actuation methods that will be considered fall into two categories: mechanical and electro-mechanical. The mechanical actuation method that was considered is hydraulic actuation. High pressure hydraulic fluid is readily available, and the EPA uses this fluid to actuate the valve open and closed. High pressure hydraulic fluid, appropriately applied to correctly dimensioned activation areas, can actuate the valve either parallel or perpendicular to the accumulator entrance.

Electro-mechanical actuation includes the use of servomotors, stepper-motors, and solenoid actuators. Servomotors have the ability to turn the crank of a valve (such as a ball valve, gate valve, etc.), and can only be used perpendicular to the accumulator entrance. Stepper-motors require less electrical power to do the same thing, and provide much more precision control of how much the valve crank can turn. Solenoid actuators can use a solenoid coil to force a pin foreword, and therefore can be used in parallel with the accumulator entrance.

7.2 Common Valve Types

While there were a number of different geometries and layouts of main valves that were conceived during concept generation, the main styles could be reduced to one of three main types: gate, ball, and poppet. The main valve types that were considered in this concept generation phase were ball, gate, and poppet type designs. More detailed descriptions of these main valve types can be found in section 5.2. Appendix G shows that poppet, gate, and ball valves were primarily thought of during the concept generation phase.

7.3 Pressure Equalization

To eliminate or minimize the effects of fluid hammer, a system was necessary that would equalize pressure between the relatively low pressure hydraulic line and the high pressure accumulator before main valve opening. Since the current EPA design accomplishes this through the use of intricate, difficult to manufacture components, a simpler, more reliable method was required for the alpha design. It should be noted that pressure equalization is not necessary when line pressure is higher than accumulator pressure. Since this condition allows fluid to flow into the accumulator, there is no chance for a damaging pressure gradient to develop. The EPA mentioned that the pressure equalization function could be necessary every few minutes of vehicle operation, so this function must be durable and fast.

A fairly large number of designs were considered for how to equalize the pressure in the line. These designs included ideas for how pressure could be integrated into the valve or integrated somewhere into another part of the system. Other methods for pressure equalization that are not integrated into the valve will not be used as part of a final design for this project. To most benefit the EPA, a complete, inclusive valve design is desired, but the other methods of accounting for pressure equalization (or simply the effects of fluid hammer) are presented here to be of value to the EPA. The EPA requested that we keep track of and report to them our other ideas regarding this system and the issues they have been facing with this valve.

7.3.1 Pressure Equalization Integrated Into the Valve

It was proposed by the team that the pressure equalization could be done through the use of specialized valve seat geometry. The idea behind this is that the opening of the valve could create a specific pathway for the hydraulic fluid that would equalize the line pressure before the valve reached the fully open position.

The most universal pressure equalization technique conceived was to use a separate, dedicated solenoid valve to connect line and tank cavities immediately before main valve opening. The connection would be established through electrical control of the solenoid valve, timing the actuation to be just before valve opening actuation. The exact flow area can be regulated through an orifice to eliminate excessively large pressure spikes.

Another design, the “fixed orifice” pressure equalization method, uses a passage in the HPSOV to connect and equalize line and tank pressures during normal valve opening sequences (note that this only works for hydraulically actuated valves). When the main valve control solenoid commands an open position, the high pressure can be tapped off from the actuator cavity and sent to the hydraulic line in parallel with the actuating flow. If the main valve is designed such that it cannot be opened unless the line and tank pressures are equal (through the use of carefully designed geometries), then pressure equalization will be ensured for every main valve opening sequence.

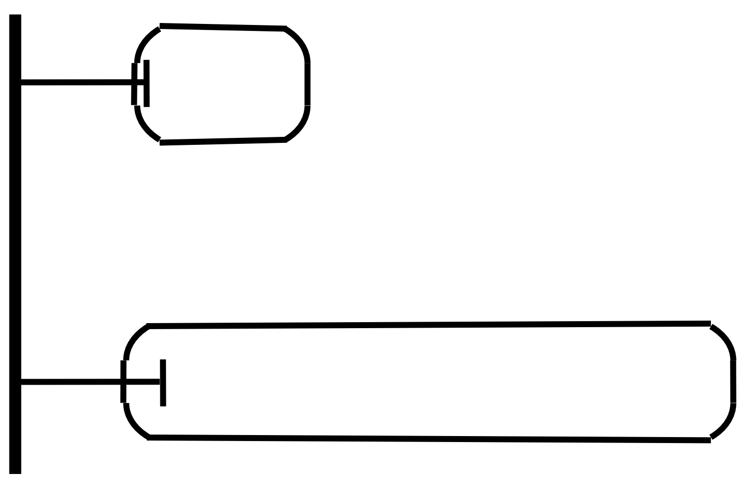
Appendix E shows some ideas for integrating the pressure equalization into the valve that were generated with the help of the rest of the ME 450 class during an open session on concept generation.

7.3.2 Other Methods for Pressure Equalization

In an effort to bypass the pressure equalization problem altogether, the use of a hydraulic pulsation damper was considered. By including such a damper, the noise that is created by opening the valve would be significantly reduced. This device would accomplish this by eliminating the fluid hammer that is traveling through the line, thus eliminating the need for a specific pressure equalization device within the HPSOV. A schematic of a damper is shown in Appendix F, number 5.

One of the first methods of pressure equalization proposed by the team was the use of an auxiliary accumulator connected to the high pressure accumulator. Whenever the line pressure deviated from the high pressure accumulators, and the main accumulator wanted to actuate open, fluid would be drawn from the auxiliary accumulator to equalize the pressure. Two methods of implementing this configuration were thought of. The first method of implementing this device would be a constant one way connection from the accumulator to the hydraulic line. Another method would be if the auxiliary accumulator would be fitted with a poppet valve that would only open a small way, so the flow through it would never be enough to cause loud fluid hammer. In the second configuration, the secondary tank would function much like the primary tank, but would be fully actuated 50-100 ms before the primary tank and would only have the function of equalizing the line. The secondary tank idea can be seen in Figure 7.1 below.

Figure 7.1: Secondary pressure equalization tank schematic



Another method of pressure equalization considered was using a separate pump for hydraulic line pressurization. This could be either an electric pump plumbed into the line specifically for this purpose, or running the vehicle's engine whenever line pressurization was needed.

7.4 "Complete" Valve Designs

After brainstorming for specific valve functions, we then decided to focus to see if we could brainstorm fully functional valve designs. Collating these ideas was a big step forward in coming up with a design that would complete all the requirements successfully and efficiently. By combining the best parts of a few different ideas, we were able to effectively produce a collection of overall valve concepts.

Figure 7.2 shows a sectional view of the valve design that eventually became the "first alpha design". The focus of this design was the use of a static orifice for pressure equalization. This valve is poppet style, and has a low number of parts. No springs are present, and the valve relies on balancing areas that hydraulic fluid is applied to in order to control its actuation. The pressure in the line is also used as part of the valve actuation method, so that the line pressure decides whether the valve is open or closed. This allows for one way flow functionality. The pressure in the tank is sampled through the middle of the stem, which allows most of the cross sectional areas to be circular.

Figure 7.2: Sectional view of design that became the "first alpha design"

Figure 7.3 shows a sectional view of the valve design that eventually became the "second alpha design." The focus of this design was to create a ninety degree turn before the pressure equalization volumes, as well to use a check valve in conjunction with a solenoid actuator to accomplish pressure equalization. Unlike the design of Figure 7.2, this design does not use line pressure, but instead uses tank pressure and low pressure sampled at the accumulator to actuate the valve open or closed. One way flow functionality is accomplished by having a force balance that allows the valve to be forced open when the pressure in the line is high enough.

Figure 7.4 shows a sectional view of a design that is similar to the second alpha design, but has more complex parts and uses springs to allow for bladder interaction easier. The design is exactly the same as the design in Figure 7.3, but the springs allow a much smaller bladder force to close the valve. This valve design is presented as a possible replacement for the design in Figure 7.3, in case the force required by the bladder before closing is very small.

Figure 7.3: Sectional view of design that became the "second alpha design"

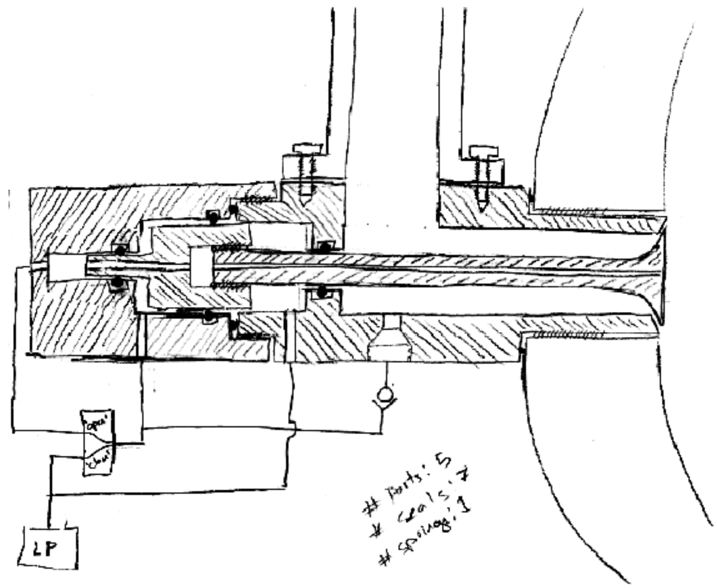


Figure 7.4: Sectional view of a similar design to Figure 7.3

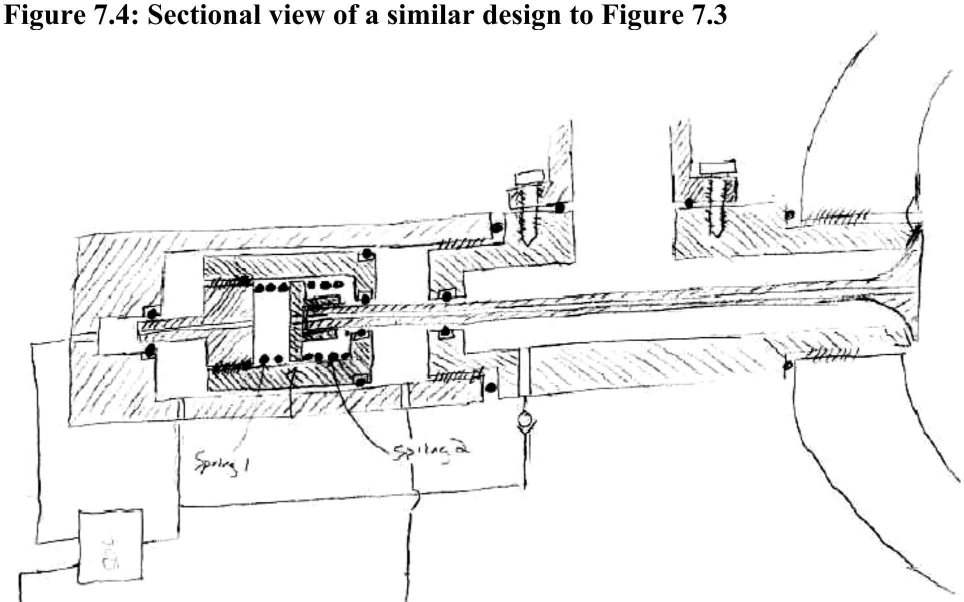
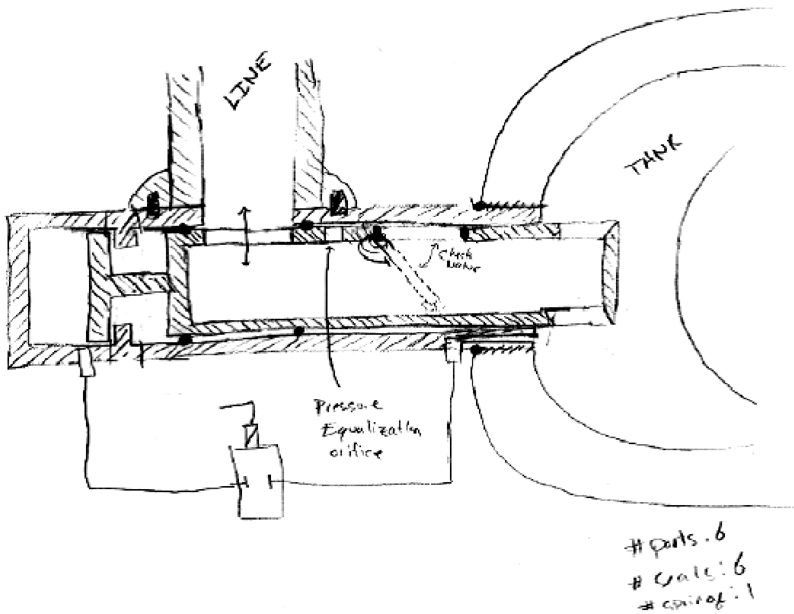


Figure 7.5 is an "interference design" valve, in which a piston slides between orifices in the sides of the inner cylinder, exposing them to the line. Each orifice has a different function. The left-most orifice is the fully open orifice that allows for low pressure drop and a ninety degree turn for flow out of the accumulator. The middle orifice is small and allows for pressure equalization before the large orifice is exposed to the line. The right-most orifice is more complicated. In the wall of the inner cylinder must include a small check valve that is closed when the line pressure is less than the tank pressure and open when the line pressure is greater than the tank pressure. This check valve function is accomplished by a

small spring and flap of material. The inner cylinder extends into the accumulator to allow interaction with the bladder.

Appendix G shows a few other valve designs that were created.

Figure 7.5: Sectional view of an "interference" type piston design



8 FIRST CONCEPT SELECTION

After all of these different concepts were generated, we went through them individually marking what would work, what wouldn't, and how/if we could fix them. One note to make is that no ball valve or gate valve full designs were further pursued due to the inherent disadvantages of this design (as discussed in section 5.3.1).

In order to establish which valve components will suit the hydraulic hybrid application the best, a comparison was done for the various HPSOV functions. It was determined that, regardless of the specific design, there are three main areas that will classify any HPSOV concept. These are: main valve type, method of main valve actuation, and method of pressure equalization. For each of these categories, a list was compiled of our own concepts, as well as all existing components that perform the necessary function. The alpha design would then try to incorporate the components or methods from each of these categories that would fit the hydraulic hybrid application the best. These selection charts are presented in Appendices I and J.

8.1 Actuation Methods

Due to the high pressures of the HPSOV application, all valve actuation requires large forces acting over relatively short distances. This motion can be either linear or rotary, depending on the valve style. If an electric motor was used to actuate a linear valve, some sort of gear train or rack and pinion system would be necessary to transform the rotary motion of the motor into the linear motion required by the valve. Because this would increase the cost of the valve, this was seen as a negative. Ball valves are actuated in a rotary manner and thus an electric motor could potentially be used, although a gearbox may be necessary for torque and speed matching purposes.

Solenoid actuation was one possibility for linear valve actuation. Solenoids typically have fast response times. Because of the large valve positioning forces involved, a solenoid would need to be driven by a large electrical current. With prolonged operation under engine-off conditions, this could cause an excessive strain on the vehicles electrical system. To minimize the actuation forces necessary, hydraulic balancing of the valve could be done. In this case, much of the complexity of the hydraulic actuation style will be necessary to supplement the solenoid.

The benefits of hydraulic actuation include the fact that the high pressure hydraulic fluid is provided in plentiful supply and in close proximity to the HPSOV. Additionally, when properly applied to the correct areas, large actuating forces can easily be developed. The response time can be acceptable for the hydraulic hybrid application, proven by the success of the current EPA style of actuation. Negatives of hydraulic actuation include the necessity to contain high pressure fluid under all operating conditions, as well as under storage conditions. If this is not done properly, fluid pressure can be lost through seals, resulting in an overall decrease in fuel efficiency of the vehicle, as well as potential leakage. Also, in order for the actuation to properly function, a careful analysis of valve and actuator geometry is necessary to ensure correct force balancing during all operating modes.

Due to the availability of the energy source, the quick response, and the driving force available without gear reduction, a hydraulic system was decided to be the best means of actuating the main valve. A complete selection chart is presented in Appendix I.

8.2 Common Valve Types

In general, ball valves offer the lowest pressure drop of the available valve styles. However, they typically have geometries that are difficult to manufacture. The supporting bearing structure required for the rotary actuation inherent to ball valves would also add complexity to the design, as well as increase the cost of the valve. This rotary actuation is more complicated when compared to the linear motion of the poppet valve. Additionally, the large number of cycles the valve is required to perform through its lifetime would lead to excessive wear of the sealing components of the ball valve. While there are designs that address and overcome this issue, they again add complexity and cost to the design.

Unlike ball valves, gate valves also have a linear actuation. However, gate valves tend to take up more space than poppet or ball valve styles (as explained in the Information Sources section). Gate valves require large forces for adequate sealing, or alternatively, large actuation times. Additionally, they are unable to interact with the bladder or allow one way flow without a secondary line. Like ball vales, gate valves offer low pressure drop.

When coupled with hydraulic actuation, poppet valves can have actuation times that are acceptable for the hydraulic hybrid application. A demonstration of this can be seen in the acceptable response times of the current EPA HPSOV design, which uses a poppet valve. Another reason for their selection is the ability for reasonably simple one way flow capability. Both ball valves and gate valves are actuated by motion that is not in the direction of fluid motion. Because of this fact, they will hold up to a high pressure gradient while continuing to prevent flow through them. Poppet valves however lend themselves reasonably well to one way flow capability, as they inherently provide a surface for the high pressure to act on that is parallel to the motion of valve travel. While motion parallel to the fluid flow is an advantage for one way capabilities, it is a disadvantage in other respects. For example, this parallel motion makes this valve style ultra sensitive to surface areas and pressures acting upon the valve. If not properly balanced, the valve could possibly open or close without being commanded to do so.

Considering the speed of actuation, ability to include one way flow capability, ease of manufacture, and the ability to properly interact with the high pressure gas bag inside the hydraulic accumulator, poppet valves stood out compared to other valve styles. In summary, while there are a number of valves that provide features that would be desirable for this HPSOV application, the need for a highly reliable, low complexity valve, combined with the positive attributes that suit the poppet valve well to this application has resulted in it being selected for the main valve of the alpha design. Appendix I presents a complete selection chart for the main valve types presented here.

8.3 Pressure Equalization

A selection chart for these pressure equalization techniques is presented in Appendix J. The pressure equalization techniques are presented in section 7.3 and their advantages and disadvantages are presented below.

Using the selection chart, the "fixed orifice" method was chosen for the alpha designs. The solenoid which controls main valve opening is also connected to the hydraulic line. When the main valve is commanded open, high pressure is allowed to flow into and equalize the main hydraulic line. Some other pressure equalization methods ranked higher because of easier integration or less moving parts, however not all of them were designs where pressure equalization was done within the HPSOV itself. Although not the top ranked option, this design was chosen for its low complexity, and ease of integration into the HPSOV. The options that ranked higher than this one are discussed in sections 7.3.2 and 8.3.2, so that these ideas are presented to the EPA for consideration.

8.3.1 Pressure Equalization Integrated Into the Valve

The advantage of the valve seat geometry idea is the inherent simplicity, requiring no additional moving parts. The drawbacks are the need for fine machining and small tolerances to ensure reliable equalization. Also to ensure a limited rate of pressure rise, the travel of the valve would need to be relatively large to prevent a rapid area increase and the resulting pressure surge.

The drawbacks of a dedicated solenoid valve include the accurate timing and separate electrical circuitry required. Secondly, the required volume is relatively large and the design is more of an "add on" fix than a smoothly integrated system.

The most promising method for pressure equalization is the use of a "fixed orifice". Drawbacks include the fact that this design will only work with hydraulically actuated valves. Additionally, this design is one where pressure equalization is done through the natural operation of the main valve. Because of this, system volume is increased and it is possible that specific passageways will need to be provided in the valve body.

8.3.2 Other Methods for Pressure Equalization

The method of using a hydraulic pulsation damper would be applicable provided the HPSOV could operate correctly over the maximum line/accumulator pressure gradient observed during vehicle operation. However, the alpha design is intended to be a direct replacement for the current EPA HPSOV, so such functionality will be included in the alpha design.

Using an auxiliary accumulator would eliminate the need for pressure equalization capability to be built into the HPSOV, but other problems still occur. The first method of implementing this device would be a constant one way connection from the accumulator to the hydraulic line. While this would ensure equal pressures, the fact that the hydraulic line would be under pressure while the vehicle was not operating would mean increased fluid leakage during storage. Alternatively, some valving configuration could be designed for the auxiliary accumulator to circumvent this problem, but this route would mean development of a separate valve. Since this would only relocate, not eliminate a pressure equalization valve design, this concept was not chosen.

The separate pump method for pressure equalization was eliminated due to the relatively quick decay (a matter of minutes) of the line pressure while it is in an idle state. This quick decay time would necessitate frequent startup of the vehicle's engine, hurting efficiency and increasing wear and tear on the vehicle. The electric pump option remains viable and will is a suggested option. However, since the current EPA HPSOV includes pressure equalization functionality, the alpha design should include this functionality to be an eligible competitor. Therefore, the pump method for pressure equalization was not chosen for the present time. A separate project could be undertaken to install a pump into the vehicle and report on the performance.

8.4 "Complete" Valve Designs

A selection charts for the full design concepts is presented in Appendix H. The overriding objective is the creation of a HPSOV that delivers proper functionality under all operating modes. It must do so in a simple, easy to manufacture and reliable package with a cost that is significantly reduced from that of the current EPA design. The selection criteria are weighted highly on these objectives. Things such as number of moving parts, number of seals, and difficulty of individual component manufacturing are included and weighted accordingly (negative weighting). Two of these designs were selected, and are further investigated in section 9.

9 FIRST SELECTED CONCEPTS

After completing the Pugh chart for "fully-functional" designs, the two highest scoring designs were chosen as our first selected concepts. Based on the scoring system used, these two concepts were essentially just as strong as each other. These concepts were chosen to be later compared and contrasted in more detail, and one of them will become the final design as their strengths and weaknesses become more apparent. This section describes these two designs and how they function in detail. Both concepts are activated by use of a solenoid switch, which would be integrated into the housing of each valve. This solenoid is not shown integrated into the valve housing of either selected design because the valve housings were not an important part of these designs. Instead of focusing on detailed placement of the actuation method, these designs focus on how the actuation method is used to achieve functionality.

9.1 Selected Concept 1

The use of the line pressure to help determine how the valve functioned was the original basis of this design. By using the pressure of the line, we could easily set up the forcing profile such that the valve would never open unless the pressure in the line was near to or greater than the pressure in the tank. In order to get the pressure in the line higher, high pressure would be applied to a small orifice in the actuator piston (near the rear of the piston, as seen in figure 9.1), which would equalize the pressure in the line to that in the tank. This valve design includes a poppet stem, an actuator piston, a stem holder, and two-part valve housing (one part that simply is an end cap to keep the actuator assembly in place). The poppet stem, stem holder, and actuator piston are all threaded together, and the end cap is threaded to the valve housing.

9.1.1 Closed Position and One-Way Function

Figure 9.1 shows the valve in its closed position. In the closed position, high pressure from the accumulator (between 2000 and 7000 psi) is applied to the small volume on the right side of the actuator piston, through the valve housing (a small green arrow shows where this application occurs). Darker red areas show volumes that are at a higher pressure than lighter red areas. In this configuration, when the line pressure is lower than the accumulator pressure, the force balance is such that the valve is positively sealed.

Figure 9.1: Sectional view of the first selected design in the closed position

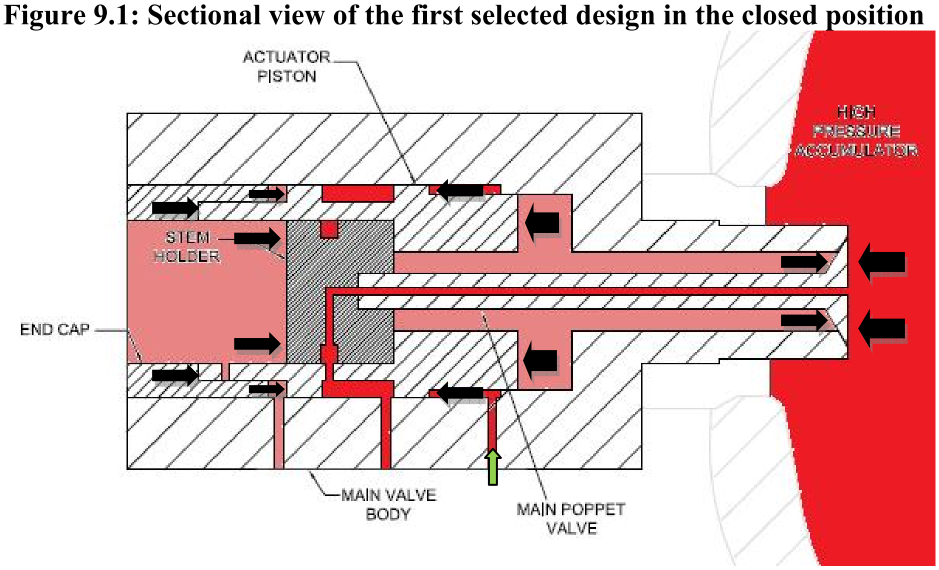


Figure 9.2: Sectional view of the first selected design in the one-way position

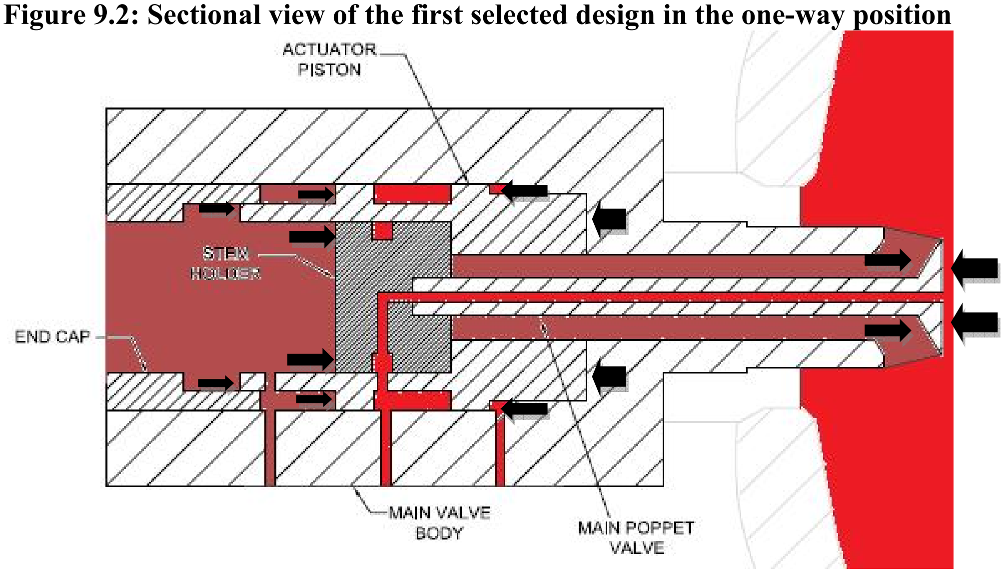


Figure 9.2 shows how the one-way functionality of the valve works. When the line pressure becomes higher than the tank pressure, it overcomes the force imbalance that existed previously. By overcoming

this force imbalance, the higher pressure in the line then can leak into the accumulator by opening the valve, as required by the EPA. The areas that the respective pressures activate on are balanced such that, when the pressure in the line is equal to the pressure in the tank, the total force balance on the actuator piston is zero. Therefore, the line pressure need only overcome friction to open the valve.

9.1.2 Open Position and Pressure Equalization Function

Figure 9.3: Sectional view of the first selected design in the open position

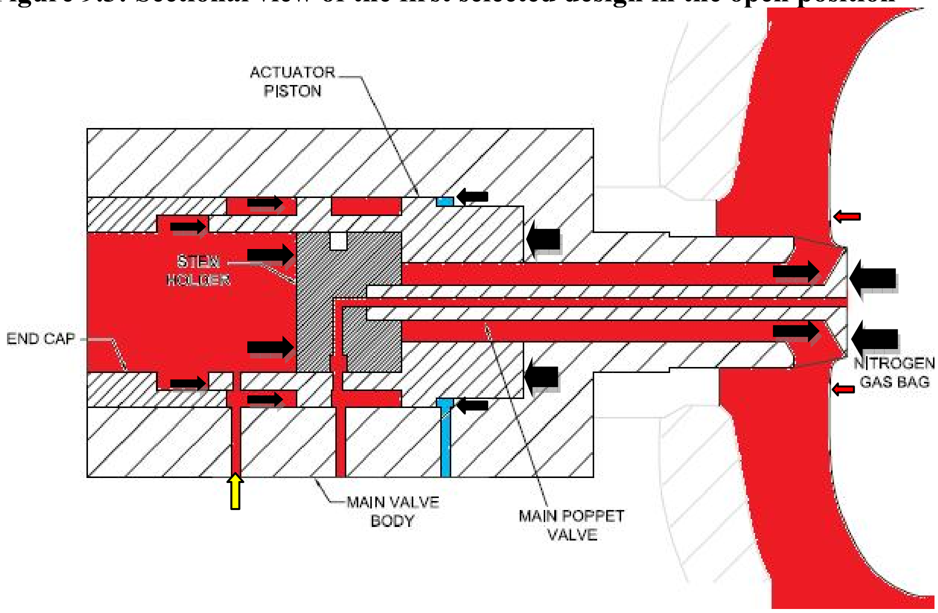


Figure 9.3 shows the valve in its open position. In the open position, the solenoid is actuated differently, and low pressure is applied to the small area in the power piston assembly. The blue areas show where low pressure is applied and the red areas show where the line pressure exists. Before the valve can open, however, the pressure in the line must still overcome a force imbalance. This is accomplished by equalizing the pressure through the orifice in the piston by using the solenoid switch to apply high pressure through the valve housing (a small yellow arrow shows where this is accomplished). In this configuration, a total force imbalance towards the right is created, which opens the valve. In addition, as can be seen in the figure, the bladder can interact with the valve head when in the open position and start to extrude around it. While extruding, a force is applied by the bladder onto the valve head (red arrows). This force, if great enough, will overcome the force imbalance that is keeping the valve open to force the valve closed.

9.1.3 Advantages

The use of the line pressure in the activation of the piston allows the line pressure to determine whether this valve can open. This has an inherent advantage due to the fact that, using an area balance, the valve cannot open unless the pressure in the line is the same. This fact effectively eliminates the problem of fluid hammer in the line. This is better than the existing EPA design because the fluid does not have to force the valve open against a spring causing greater pressure drop than that of this valve given all other conditions are the same.

There are a low number of parts, relative to the current EPA design. The lack of any springs needed for valve function makes this design much easier to assemble and maintain than the current EPA design. Another advantage of this design is the need for only a few dynamic seals, due to the fact that the system includes only one moving assembly. The valve is dimensioned to conform to the accumulator entrance, and the end cap can be made to include a 90 degree bend as well as a 4 bolt, 62 flange connection to the line, which is a necessary part of the connection to the rest of the system. By accessing the pressure in the accumulator through the center of the stem, many of the cross sections internally are circular. Circular cross sections throughout would make the valve easier to machine.

9.1.4 Disadvantages

Two parts of the valve are complex machining jobs. The housing is a complex machining job because it may have circular interior cross sections and square exterior cross sections (for ports and the inclusion of a solenoid). The other complex machining job is for the stem holder, which must have holes of some sort cut through it in order to allow fluid flow. Minimum pressure drop must be achieved through this piece, so the machining of this part must be done with a mill.

This design does not include support for the stem near the accumulator entrance, which may not be ideal in the final design. This fact may increase the possibility of fracture in the stem due to undesirable moments that can occur in the opening and closing movement of the valve. These moments could be created by misalignment between the head of the valve and its seat in the housing.

The lack of any springs needed for valve function is an advantage of this design, but this design needs to have a rather large bladder extrusion force, which may not be possible.

The solenoid that is required for use in this design is not the same solenoid that is used in the current EPA design. In order to activate correctly, low pressure must be applied to one area while high pressure is simultaneously applied to another area (and the line). The current solenoid switch does not have this capability. If a solenoid cannot be found that has this capability, then two solenoids must be used in this design, which would result in major cost and size disadvantages.

9.2 Selected Concept 2

The EPA sponsors mentioned that it was desirable to include a 90 degree elbow integrated into the final design of our valve. With this in mind, a design that incorporated this 90 degree turn was created. The valve includes a poppet stem, an actuator piston, a retaining nut, and two-part valve housing. The housing is also shown with an integral check valve, which is a necessary part of its function.

9.2.1 Closed Position and One-Way Function

Figure 9.4 shows the valve in its closed position. In the closed position, low pressure from the reservoir (between 80 and 250 psi) is applied to both the left and right sides of the actuator piston. In this configuration, when the line pressure is lower than the accumulator pressure, the force balance is such that the valve is positively sealed. The blue areas show where low pressure is applied, and the red areas show where high pressure is applied. Darker red areas show areas that are at a higher pressure than lighter red areas.

Figure 9.5 shows how the one-way functionality of the valve works. When the line pressure becomes higher than the tank pressure, it overcomes the force imbalance that is created by the tank pressure on the front and back ends of the piston. By overcoming this force imbalance, the higher pressure in the line then can leak into the accumulator by opening the valve, as required by the EPA. While in the two modes described above, the same pressure that is applied to both sides of the actuator piston is applied to the check valve. The check valve is needed to prevent the low pressure source from being connected to the higher pressure in the line.

Figure 9.4: Sectional view of the second selected design in the closed position

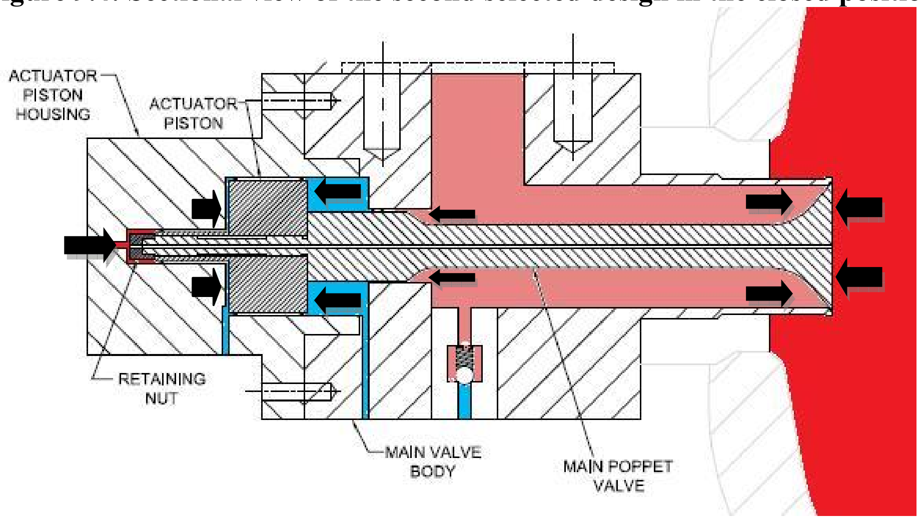
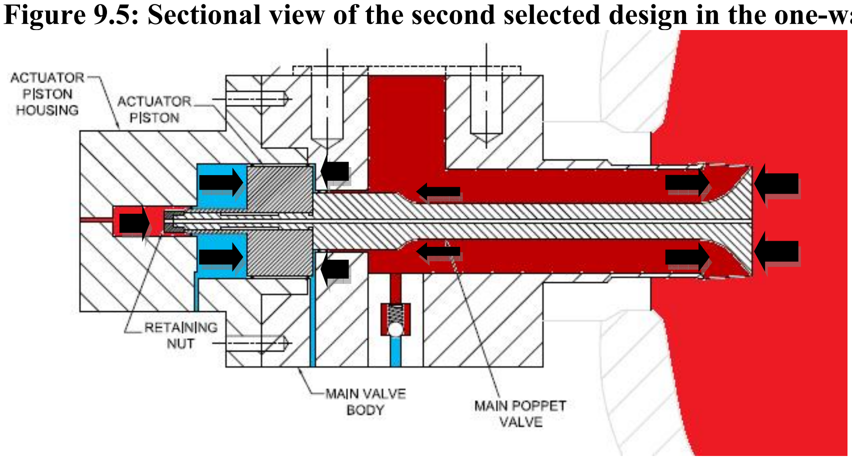


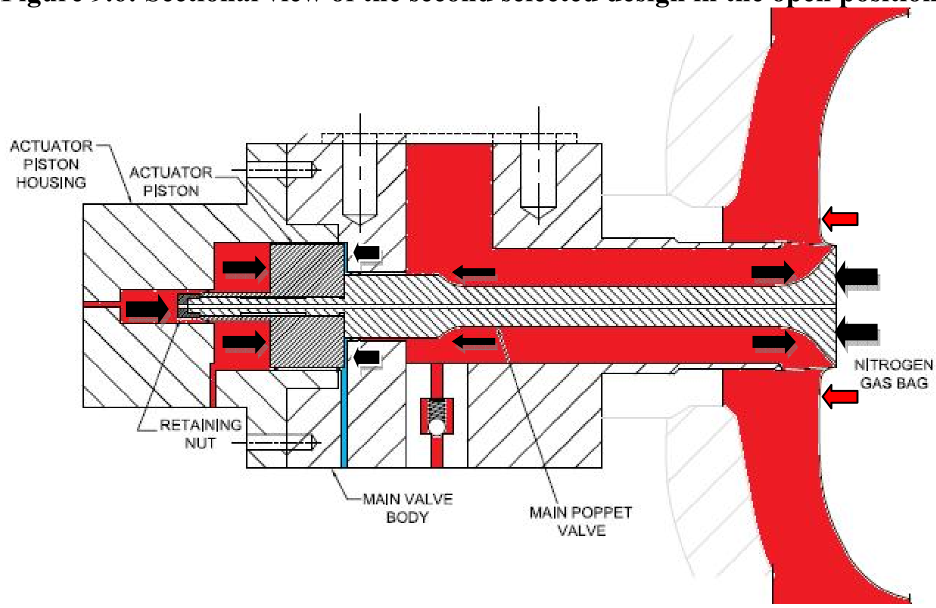
Figure 9.5: Sectional view of the second selected design in the one-way position



9.2.2 Open Position and Pressure Equalization Function

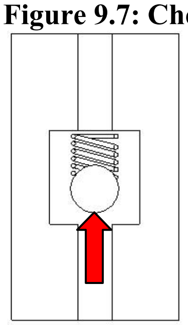
Figure 9.6 shows the valve in its open position. In the open position, the solenoid is actuated and high pressure from the accumulator is applied to the left side of the actuator piston. The right side of the actuator piston is still at low pressure, so a force imbalance towards the right is created, thus forcing the valve fully open. In addition, as can be seen in the figure, the bladder can interact with the valve head when in the open position and start to extrude around it. While extruding, a force is applied by the bladder onto the valve head (red arrows). This force, if great enough, will overcome the force imbalance that is keeping the valve open to force the valve closed.

Figure 9.6: Sectional view of the second selected design in the open position



Before the valve is opened, however, the pressure in the line must be equal to the pressure in the accumulator, in order to prevent loud fluid hammer. When the high pressure is applied to the left side of the actuator piston, high pressure is also applied to the check valve in the valve housing, opening it (as in figure 9.7), and thus applying high pressure to the line. This allows the line pressure, if significantly lower than the tank pressure, to equalize before the valve can open.

Figure 9.7: Check valve in the open position



9.2.3 Advantages

The incorporation of this 90 degree turn was placed immediately after the opening of the valve into the bladder, having an immediate advantage of bringing all of the actuation parts out of the flow volume. Despite the sharp corner that arises, the overall pressure drop that is created by this valve may be less than the current design due to the removal of complex and large geometries from the flow stream.

There are a low number of parts, relative to the current EPA design, and there are no complex cross sections in the interior assembly (they are all circular). This fact make the part easy to machine, easy to assemble, and easy to maintain. The only part that is a complex machining job is the housing, which may have circular interior cross sections and square exterior cross sections (for ports and the inclusion of a solenoid). The lack of any springs needed for valve function makes this design much easier to assemble and maintain than the current EPA design.

The valve is dimensioned to conform to the accumulator entrance as well as include a 4 bolt code 62 flange connection to the line, which is a necessary part of the connection to the rest of the system. The bolt on assembly housing can be easily designed to hold 7000 psi pressure, and is easy to manufacture and simple to use for maintenance. In addition, only one place in the valve housing is holding high pressure at all times that it is closed, which can be a big advantage in the sealing aspects of the valve. In addition, the solenoid actuator valve, as well as the integrated check valve, that are used in the current EPA design can be incorporated into this design.

9.2.4 Disadvantages

One main disadvantage of this design is the rather large final size of the design due to the use of a 90 degree bend in the middle of the valve. The nature of this design may result in a heavy, bulky, and especially long design due to the movement of the entire valve actuation assembly to the back side of the valve. There are a large number of dynamic seals that are a part of this design, which may be a disadvantage due to the increased wear and thus the maintenance costs. The need for a check valve incorporated into the valve housing is also a disadvantage.

In addition, the stem in this design could be longer then in design #1, thus making it harder to machine and assemble, and may increase the possibility of fracture in the stem due to undesirable moments that can occur in the opening and closing movement of the valve. These moments could be created by misalignment between the head of the valve and its seat in the housing. This problem could be compounded because, by the nature of the way it is activated, the opening and closing forces are always large.

The lack of any springs needed for valve function is an advantage of this design, but this design needs to have a rather large bladder extrusion force, which may not be possible.

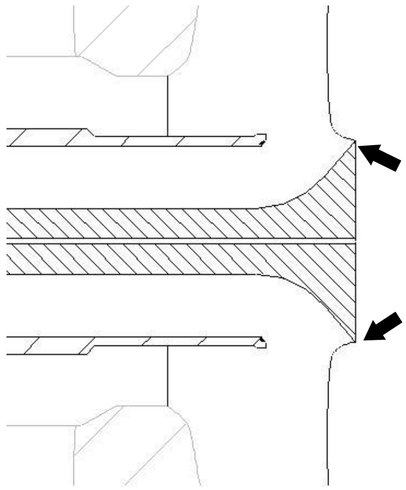
10 SECOND CONCEPT SELECTION

Before continuing the design process, it was first necessary to narrow our design selection to one final design. By re-investigating the functionality of the two selected designs in more detail, it was thought that one of the two designs would clearly be a better design. If this strategy did not work, then a new Pugh chart would be necessary to rank these designs in more detail. After presenting these two designs to our EPA sponsors, the idea of combining the two selected designs into one design that combines the best parts of each design was mentioned. This third design is also presented in this section.

10.1 Bladder Interaction Problem

As expected, when the EPA requirements were looked at more critically, the team realized that the bladder interaction functionality that is required of the final design might not be fully met by either of the selected designs presented in the previous section. The team was not sure whether the use of a poppet style valve was enough to guarantee that the bladder inside the high pressure accumulator could close the valve. The force that the bladder can apply to the valve head without damaging the bladder material is limited. The bladder would apply this force by extruding slightly around the valve head, which stretches the bladder material a small amount and applies a force through the surface tension of the material. The location of this force and a schematic of bladder interaction with a poppet style head are shown in Figure 10.1.

Figure 10.1: Bladder interaction schematic and location of bladder force



With a call to the accumulator manufacturer (HYDAC International), it was found that the maximum pressure difference across the bladder material that the bladder can handle before becoming damaged is about 6 psi. HYDAC also recommended that, if we are designing for bladder interaction, using 5 psi would be recommended. With a valve head size of 1.63 in^2, which is the size used in the current EPA design as well as in the two selected designs, the maximum bladder force is 8.14 lbs.

10.2 Failure of Selected Design 2

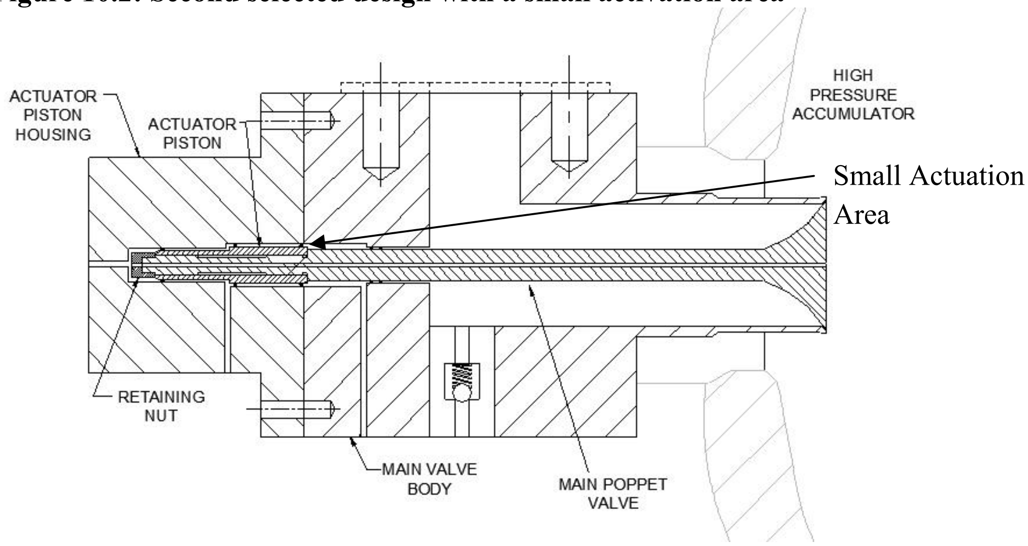
The second selected design does not work with this small bladder force. Figure 9.6 in section 9.2.2 shows the second design interacting with the bladder. In this figure, the high pressure on the left side of the actuating piston opposes the low pressure in the right side of the actuating piston. The high pressure can be as high as 7000 psi (P_{HI}), while the low pressure can be as low as 80 psi (P_{LO}). The area on the right side of the piston (which is the area on which these pressures oppose each other to create a force to the right) is 1.23 in^2 (A_{FORCE}). Using Equation 10.1 below, the total opening force (F_{OPEN}) is 8511 lbs. This force is much greater than 8.16 lbs, so this design cannot function correctly.

$$F_{OPEN} = (P_{HI} - P_{LO})A_{FORCE} \qquad \text{[Eq. 10.1]}$$

10.2.1 Area Modification

As shown by the above calculation, by reducing A_{FORCE}, the opening force can be reduced. This is the only way that the opening force can be reduced without completely changing the activation method that this design relies on, due to the fact that P_{HI} and P_{LO} are unchangeable. Figure 10.2 below shows this solution applied to the second selected design. Using equation 1 above, and setting the opening force to 8.14 lbs, A_{FORCE} must be equal to 0.0012 in^2. Clearly, this small area lies in the range of machineable tolerance, so such a small area would not be tolerated as a solution the bladder interaction problem.

Figure 10.2: Second selected design with a small activation area



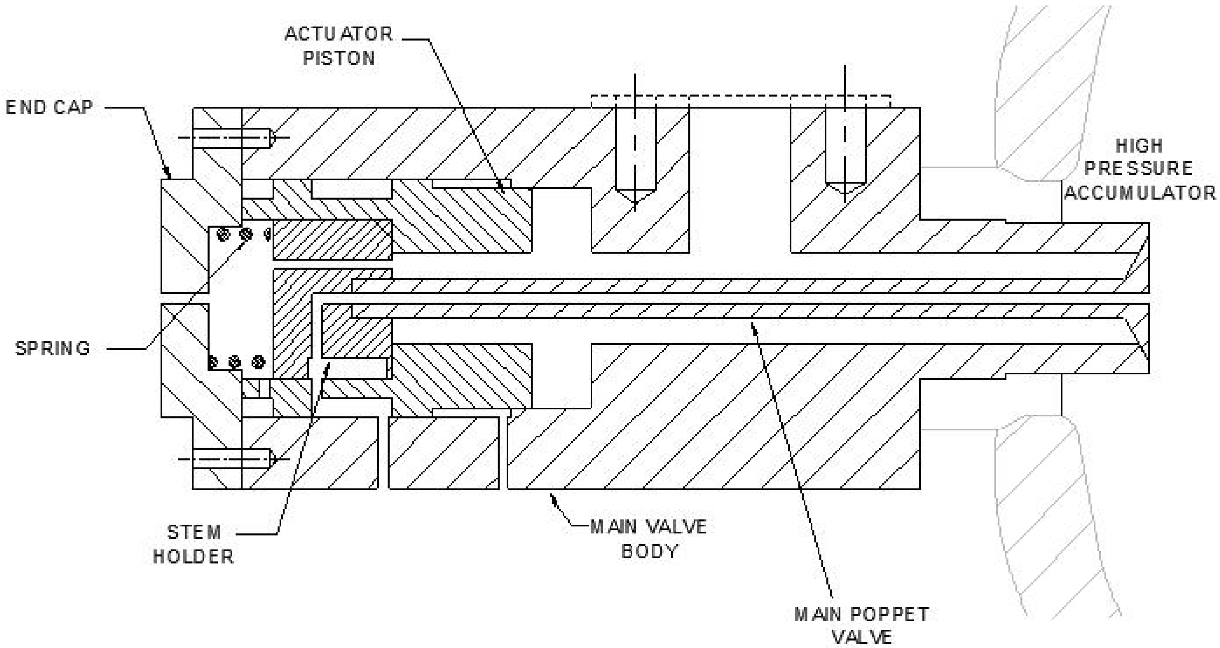
10.2.2 Addition of springs

Another way to design for bladder interaction is to design around this equation. In another attempt to redesign selected concept 2 to function for bladder interaction, one could remove the solid attachment between the stem and the actuating piston. By coupling these two parts with weak springs (no more than 16 lbs per inch, if the opening length is ½"), the bladder can press on the valve head and push closed against a weak spring. By decoupling these parts, a second spring is needed to oppose the first spring in order to allow one-way flow into the accumulator (the valve is also closed hard in the second selected design). A conceptual drawing of a design that incorporates these springs can be seen in Figure 7.4 in section 7.4. This concept was thought of originally without knowing about the size of the bladder force. This concept can make the second selected design fully functional regarding the bladder, but the added complexity that can be seen in Figure 7.4 is not ideal. The added complexity is on the same scale as the complexity of the current EPA design, which makes this design a failure in regards to our original design intent (lower complexity, lower cost).

10.3 Failure of Combination Design

As suggested by our EPA sponsors, a combination design was created that combines the best attributes of the two selected designs. By examining the differences between the first and second selected designs, as presented in sections 9.1 and 9.2, it was determined that the best attribute of the second selected design is its 90 degree integrated turn. Moving the actuation geometry out of the way of the flow of fluid to the line is advantageous. The best attribute of the first selected design, as demonstrated by the previous section, is the activation method. By using the line pressure the way that it does, it is able to function correctly under bladder interaction circumstances without creating much more complicated geometry. Figure 10.4 below shows a cross sectional view of what this combination design might look like. The actuator piston and stem holder are essentially the same. The stem needs to be quite a bit longer, and a stem backer is not needed. The backing of the spring is integrated into the end cap, through which pressure equalization is applied (same orifice as the left most orifice through the housing wall on Figure 10.3).

Figure 10.3: Combination design, cross-sectional view



This design has the potential to be better than the first selected design because of the addition of the 90 degree elbow. Another advantage would be the fact that a spring backer would not need to be manufactured and attached to the piston. However, one disadvantage of doing this design is the long stem that results. This stem needs to be supported in order to assure that no unwanted bending results. This could require an additional part to be manufactured, which would reside in the flow area. In addition, there are practical limitations to pursuing this design. The dimensions of the selected designs are set, and a lot of thought and calculation would need to be undertaken in order to understand fully how this combined valve would function. This project is under a limited timeline, so this design will not be pursued.

10.4 Success of Selected Design 1

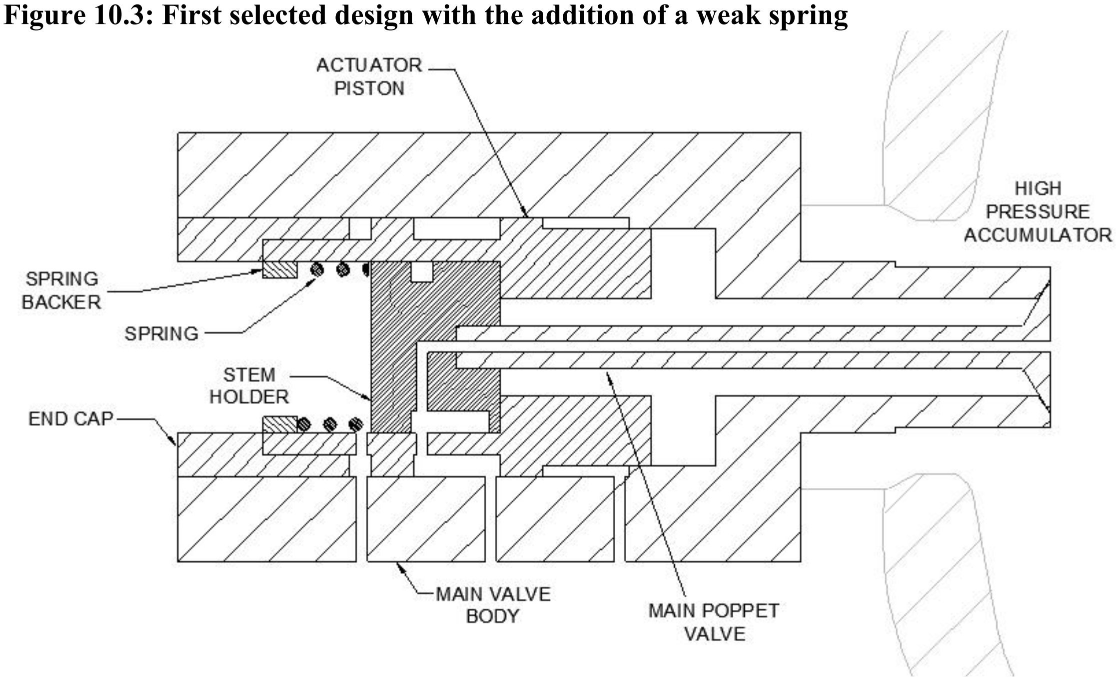
10.4.1 Initial Failure

The first selected design (unmodified) also does not work with this small bladder force. Figure 9.3 in section 9.1.2 shows the first design interacting with the bladder. In this design, the forcing of the piston is more complicated. This design applies the pressure in the line to the left side of the piston to move the piston left and right. When the valve is actuated open, the pressure in the line can be as high as 7000 psi, and opposes low pressure that is applied as seen in Figure 9.3 (80 psi at the lowest). Using equation 1 in section 10.2, the area on which the low pressure opposes the line pressure must be no bigger than 0.0012 in^2. Again, as described in section 10.2.1, this area is impossible to machine and use correctly. The original first selected design does not have an area this small, so it does not accomplish bladder interaction.

10.4.2 Addition of spring

In order to accomplish bladder interaction with the first selected design, the use of springs, as applied to the second selected design in section 10.2.2, is applied to the first selected design. The valve stem and stem holder assembly is detached from the piston. A new piece, called the spring backer, is attached to the piston, and a weak spring is placed between the spring backer and the valve stem holder. A cross sectional view of this design is shown in Figure 10.3 below.

Figure 10.3: First selected design with the addition of a weak spring



This design is like the spring-modified second design, but this design does not need to have two springs opposing each other in order to function. When the valve is in the open position, the bladder can press on the head of the valve and oppose the force supplied by this weak spring, thus pushing the valve closed. Because the line pressure is applied to the piston, the one way function of this valve functions without the need for an opposing a weak spring. When the pressure in the line is higher than that in the tank, and the valve is actuated closed (see Figure 9.2 for a schematic of where the pressures are applied), the pressure in the line is great enough to force open the piston, which transfers force through the spring, thus opening the valve head. If the spring force is not great enough to open the valve head, then a force imbalance on the valve stem assembly will open to valve. The higher line pressure, when opposing the lower tank pressure, can force open the valve.

Due to the success of the first alpha design, as modified with a spring backer and a weak spring, this design will be pursued as our final design selection. As a modification of this design, the spring backer and end cap, with a 90 degree elbow and a 4 bolt code 62 flange connection were combined. This modification simplifies the design and allows for a more durable valve.

11 FINAL DESIGN

11.1 Design Description

In addition to meeting the durability requirements of a valve that will be operating at road conditions (accomplished through material selection), the final design is capable of meeting the following customer specifications:

1) Pressure equalization
2) Opening
3) Closing
4) Interacting with the bladder or incorporating a velocity fuse
5) Performing the leak in (one way flow) function
6) Fitting into the current EPA system
7) Ability to withstand pressure and forces required by customer
8) Use of a solenoid switch (meets 10-14 Volt signal requirement)
9) Defaults to the closed position during power loss
10) Acceptable pressure drop (should be very close to existing valve due to similar geometry)

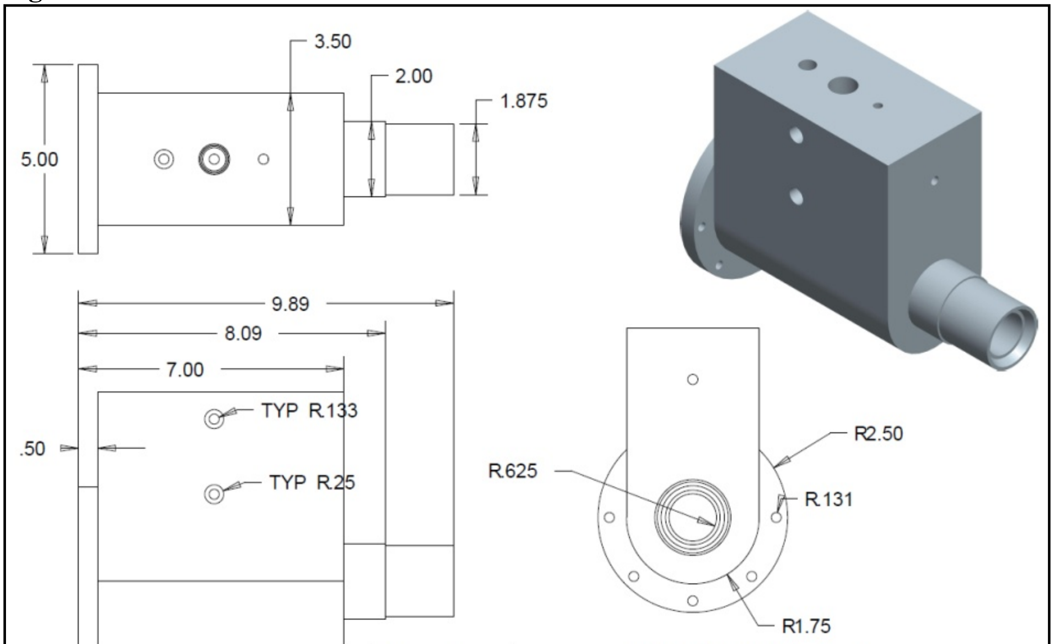
We do not know the actuating times involved in opening, closing, or pressure equalization. Because these times are unknown, we cannot confirm or deny that our final design meets these specifications. The actual actuation times will have to be determined by the customer if or when they build a beta prototype to the specifications of the final design set forth in our final report and test it under real world conditions. Because we cannot test our prototype valve at pressure comparable to real world conditions, we cannot accurately determine from our prototype what the actuating times will be.

11.2 Engineering Drawings

11.2.1 Valve Housing

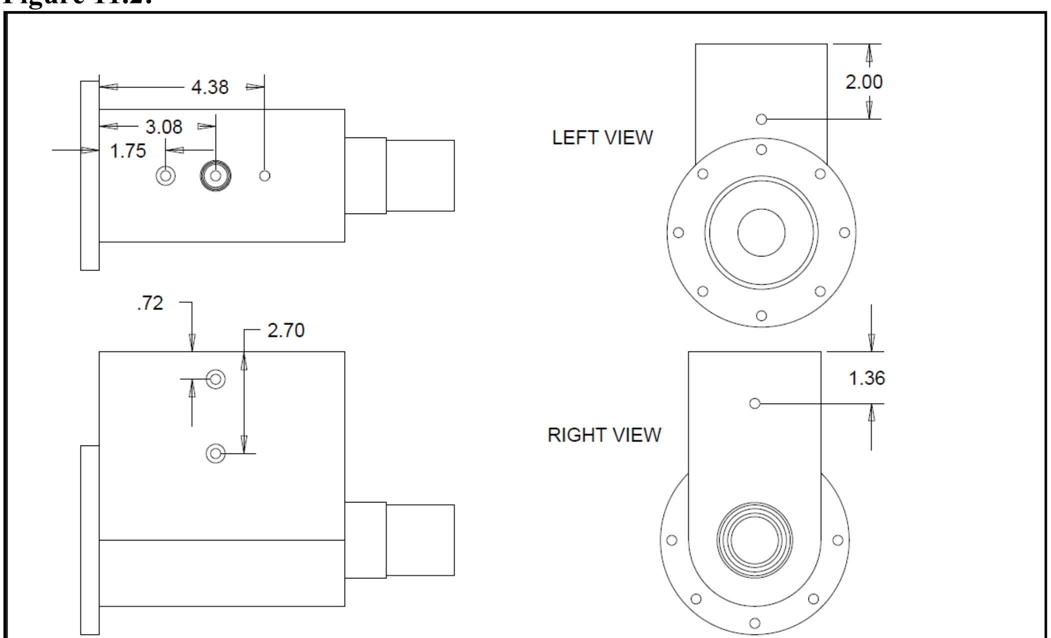
The valve housing is by far the most complex part and will also be made out of stainless steel. The opening into the accumulator will have to be precision machined in order to ensure a positive seal against the valve head. The 2 inner diameters on which the power piston slides will also have to be precision machined in order to ensure good sealing is achieved. We recommend that the EPA use their current valve geometry for the neck of the valve housing at the end of the neck where the housing contacts the back of the valve head. We also recommend looking into the possibility of making the housing out of 2 pieces, one that contains the pistons and housing walls, and another that contains the solenoid controller and accessory ports. Figures 11.1, 11.2, and 11.3 are engineering drawings of the valve housing.

Figure 11.1:



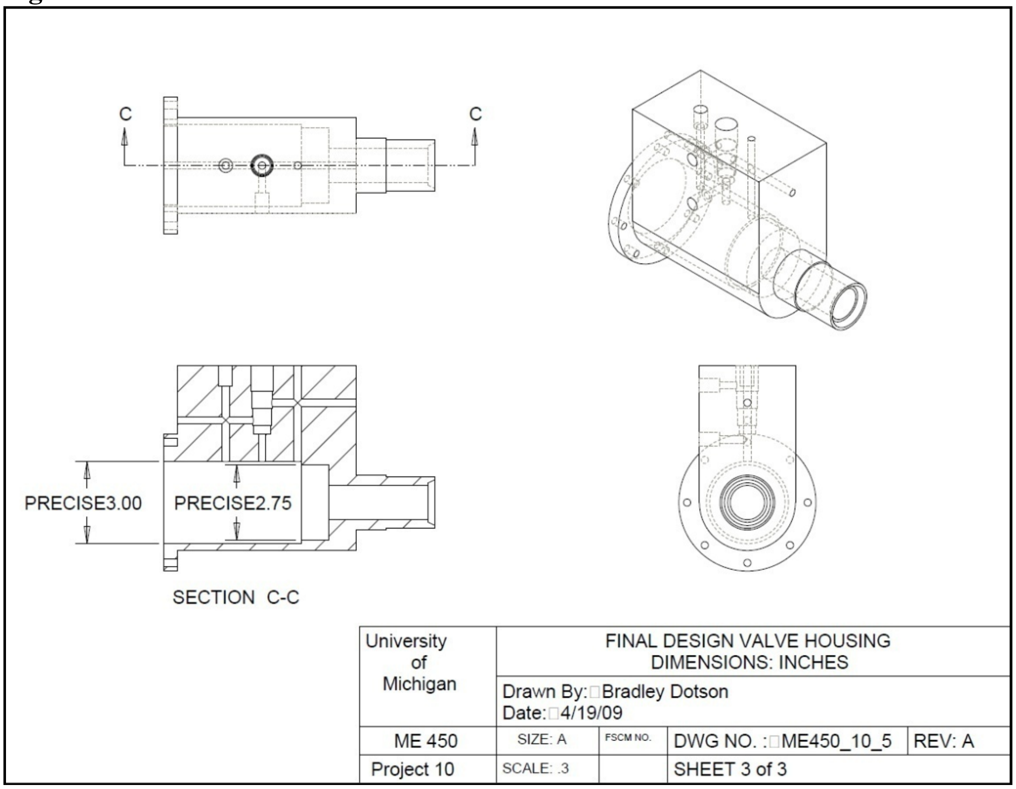
University of Michigan	FINAL DESIGN Valve Housing DIMENSIONS: INCHES			
	Drawn By: Bradley Dotson Date: 4/19/09			
ME 450	SIZE: A	FSCM NO.	DWG NO. : ME450_10_3	REV: A
Project 10	SCALE: .4		SHEET 1 of 3	

Figure 11.2:



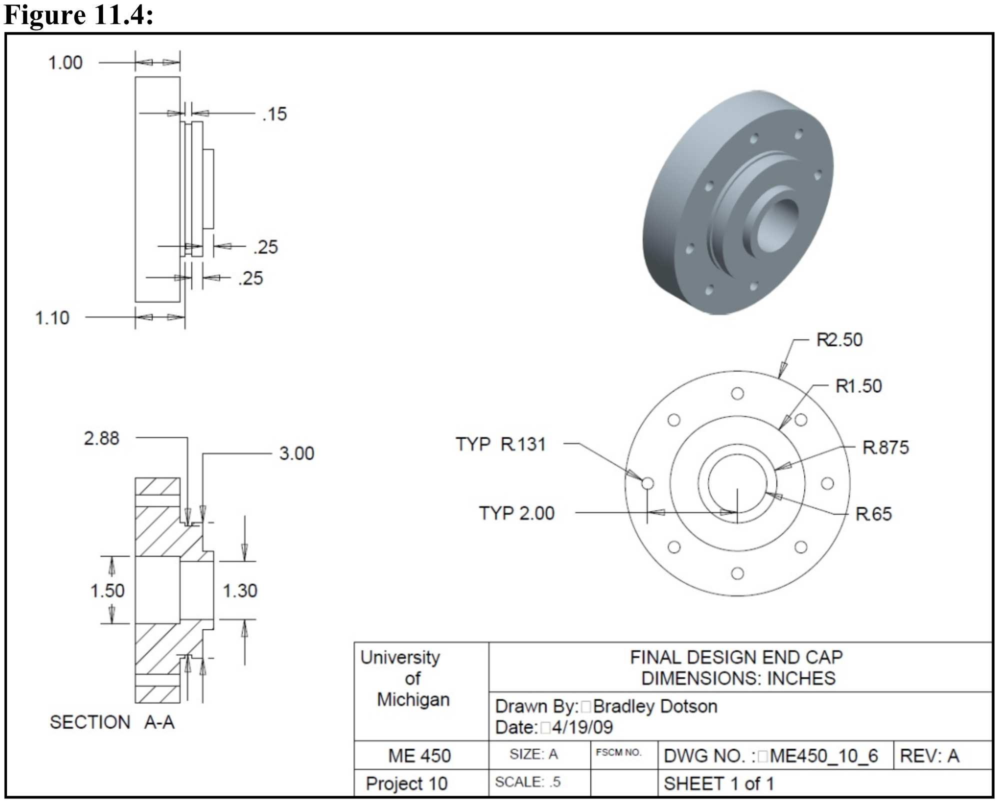
University of Michigan	FINAL DESIGN VALVE HOUSING DIMENSIONS: INCHES			
	Drawn By: Bradley Dotson Date: 4/19/09			
ME 450	SIZE: A	FSCM NO.	DWG NO. : ME450_10_4	REV: A
Project 10	SCALE: .4		SHEET 2 of 3	

Figure 11.3:



11.2.2 End Cap

Figure 11.4:

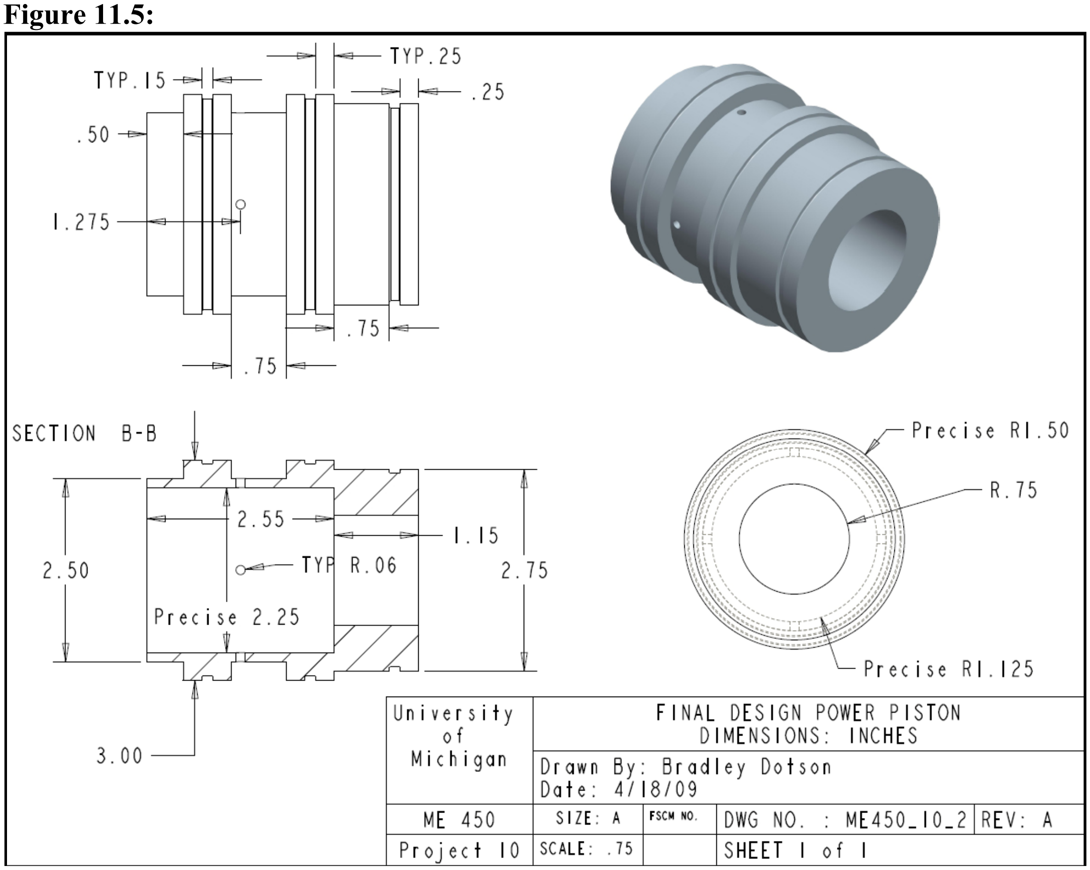


Once again the material of choice is stainless steel. This piece is designed to cap off the end of the valve housing as well as provide a connection to the line. This design of the end cap allows for a connection to a 90 degree elbow that is male threaded on one end (steel 1 5/8 – 12 UN) and has a 4 bolt 62 flange connection on the other end (size dash 20, nominal ID 1.25”). Figure 11.4 shows the end cap.

11.2.3 Power Piston

The power piston should be made from stainless steel to ensure strength as well as corrosion resistance. The largest inner diameter of the power piston (2.25” diameter) and the outer diameters that contact the valve housing must be precision manufactured to ensure good seals at high pressure. We recommend that the EPA generate their own O-ring groove designs and clearances in order to ensure positive sealing around the piston as well as ensure that the piston will still slide within the valve housing. Figure 11.5 shows the power piston in an engineering drawing format.

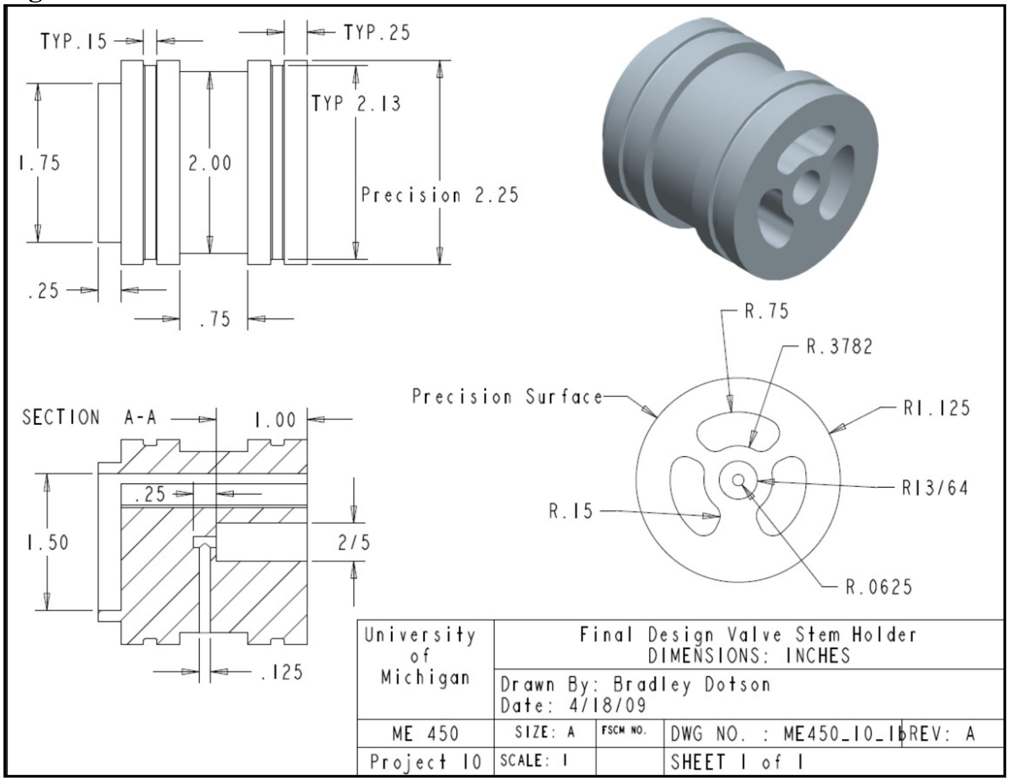
Figure 11.5:



11.2.4 Stem Holder

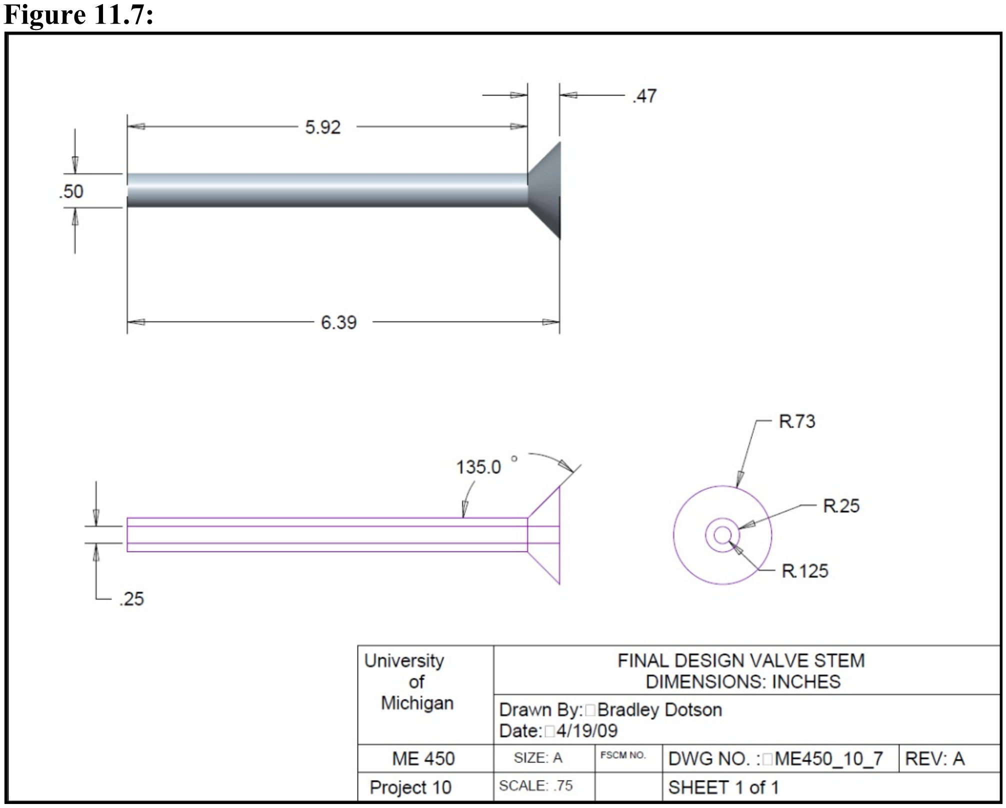
The valve stem holding piston will be made from stainless steel. The outer diameter is a precision geometry and will require precise manufacturing. All other surfaces have a much greater tolerance (do not require as precise manufacturing). We recommend that the EPA re-design the sealing geometry to ensure a positive seal, and still allow movement of the part. Figure 11.6 shows the stem holder in an engineering drawing format.

Figure 11.6:



11.2.5 Stem

Figure 11.7:



The valve stem will also have to be made from stainless steel as it will have to hold a great deal of force when the valve is in the closed position and the line is allowed to depressurize. We recommend that the EPA use their own valve head design to ensure a positive seal when the valve is closed. We believe the current EPA valve head design will suffice. This also goes for the opening of the valve housing into the accumulator. We believe the current designs are sufficient to ensure positive sealing, however, we also believe the wall thickness can be reduced so a slight redesign should be achievable easily to enlarge this opening and reduce the pressure drop across the valve.

11.3 Comparing to Current Design

The current design is the one currently implemented by the EPA. A full cost analysis was not done to compare the approximately $15,000 cost of the current EPA valve to one of our final design valves. Instead, the number of parts and the number of high precision surfaces are compared in this section. A smaller number of parts as well as a smaller number of high precision surfaces will translate to lower costs of the valve.

11.3.1 Number of Parts (not including seals)

The final design that we are proposing has a 8 major parts, 5 of which have to be custom manufactured, two are purchased parts, and 1 can be custom ordered. The EPA's current design, from the design plans we have been given has 23 parts (not including seals and backup rings). Of these 23 parts, we believe 8 are custom made, and the rest are purchased stock or purchased from a custom manufacturer. See the table below for a list of these parts.

Table 11.1: Breakdown of Part Count

Final Design	Custom/Purchased	EPA Design	Custom/Purchased
Valve Stem	Custom	Valve Body	Custom
Valve Stem Holder	Custom	Piston	Custom
Power Piston	Custom	End Cap	Custom
Valve Housing	Custom	Tulip Valve	Custom
End Cap	Custom	Seal Retainer	Purchased
4 Way Valve	Purchased	Tulip Valve Spring	Custom Purchased
Spring	Custom Purchased	Tulip Valve Nut	Purchased
90 Degree Elbow	Purchased	Poppet Piston	Custom
		Spring, PE Poppet	Purchased
		Seal Outer Support	Purchased
		Seal Support Retainer	Purchased
		Poppet Stem	Custom
		3 Way Valve	Purchased
		Retainer Nut Spring	Purchased
		Spring Follower	???
		Spring Keeper	Custom
		Socket Cap Screw	???
		Spring Follower	???
		Retention Sleeve	???
		Socket Cap Screw	???
		Retainer Ring	???
		Stroke Limiter	Custom
		Poppet Spring	Purchased

11.3.2 Number of Sealing Surfaces (or precision surfaces)

Another comparison that shows our final design should be less expensive than the current design is the number of surfaces that require precision machining. Our design has 8 surfaces that will require precision machining, excluding the fitting for the spool type actuator. The EPA's current design, as best we can tell, has 11. A simplified breakdown of where these surfaces are is in the table below. The EPA design is not showed here because the status of the drawings are not known. We cannot be sure whether the drawings provided by our EPA sponsors can be posted publicly.

Table 11.2: Breakdown of Precision Surfaces

Final Design	EPA Design
Outer Diameter of Power Piston 1	Valve Stem w/ Piston
Outer Diameter of Power Piston 2	Piston Outer Diameter 1
Inner Diameter of Power Piston	Piston Outer Diameter 2
Outer Diameter of Stem Holder	Piston Outer Diameter 3
Inner Diameter of Housing 1	Piston Inner Diameter
Inner Diameter of Housing 2	Backside of Valve Head
Housing Contact Point with Valve Head	Housing Inner Diameter 1
Backside of Valve Head	Housing Inner Diameter 2
	Housing Contact Point with Valve Head
	Inner Diameter of Valve Stem (with tulip valve)
	Outer Diameter of Tulip Valve

11.4 Material Selection for Functional Performance

According to our analysis of the functionality requirements of the valve, using CES material selector [17], the entire final design valve will be made from stainless steel. By investigating the two most critical components of the valve design (the housing and the stem), we were able to show that stainless steel was the best option for both parts from strength, cost, and corrosion standpoints. Other benefits of using stainless steel are the fracture toughness and stiffness of the material, which make it even more ideal for our design. See Appendix K for a more detailed report.

11.5 Material Selection for Environmental Performance

Three different materials were analyzed using SimaPro and the EcoIndicator99 analysis pack [20]. The materials were Stainless Steel 420 (X20Cr13), High Silicon Cast Iron (GGG-NiSiCr I), and Glass reinforced Polypropylene (GF30 I). Generally these three materials have similar eco-indicator value ranges; Stainless Steel with 16-18, Cast Iron with 3-10, and Polypropylene with 3.2-3.4. These materials were analyzed for our specific application with specific mass values. The following provides our estimated environmental impact as well as our material recommendations based on this information.

11.5.1 Valve Housing

The top two materials chosen for the valve housing were Stainless Steel 420 (X20Cr13) and Polypropylene (GF30 I). The volume of the valve housing is about 114 cubic inches, equating to 31.9 lbs of stainless steel or 5.03 lbs of PP. According to the SimaPro EcoIndicator99 analysis, the stainless steel would have a much greater impact on the environment than the polypropylene. This is largely due to the fact that the stainless steel is much denser than the PP and its larger mass would simply require more raw materials to manufacture.

In all categories, save for respiratory organic impact, the stainless steel would have a much greater impact. The single score analysis also recommends using the PP for reduced environmental impact. The PP has a total single score of about 0.2 while the stainless steel has a score of nearly 16.5. See the appropriate section in Appendix L for figures relating to this analysis.

Using the EcoIndicator99 score analysis, it is clear to see the much larger impact the stainless steel would have on human health, ecotoxicity, and resources. Unfortunately, stainless steel was chosen as the final recommendation for the valve housing because of its superior strength and corrosion resistance properties. We feel the PP could simply not perform successfully under the desired operating conditions. The third best material choice for the valve housing was High Silicon Cast Iron (GGG-NiSiCr I). This material would have improved strength over the PP, but based off of the valve stem environmental analysis, would have a larger negative impact on the environment than the stainless steel. Economically, metals are generally less costly than glass reinforced polymers, so market forces indicate that stainless steel would be a better choice.

11.5.2 Valve Stem

After completing material analysis using CES software, the two top material choices for the valve stem were determined to be Stainless Steel 420 (X20Cr13) and High Silicon Cast Iron (GGG-NiSiCr I). The valve stem was determined to have a volume of about 5 cubic inches, this a mass of 1.27 lbs for cast iron and 1.4 lbs for stainless steel. Environmental impact was then examined using SimaPro software with EcoIndicator 99.

According to SimaPro, the stainless steel would require slightly more mass of raw materials to produce than the cast iron. Stainless steel, though, provides less impact on the air and less solid waste than the cast iron. Water impact is small for both materials and can be considered practically negligible. SimaPro also determined that the cast iron would generally have a much greater effect on the environment in terms of resources and human health concerns. Stainless Steel has a higher ecotoxicity value, but looks to have a much smaller impact overall. The final single score for cast iron is about 3.4, as compared to about 0.7 for stainless. See the appropriate section in Appendix L for figures relating to this analysis.

After examining all of the data, it can be determined that cast iron would have a much larger impact on the environment during production and life time use. This makes Stainless Steel 420 the recommended choice for reduced environmental impact. Because stainless steel was the top choice during CES material analysis as well, our final recommendation for the valve stem is Stainless Steel.

11.6 Manufacturing Processes

By using CES manufacturing process selector [19], we were able to come up with the best machining processes for both the housing and the valve stem. The housing should be formed using high pressure die casting, with finished high tolerance machining on the required surfaces after. The valve stem should be formed using cold closed die forging. For a more complete report, see Appendix M.

12 FINAL DESIGN ENGINERING ANALYSIS

With the final design selected, the next step in the design process is to thoroughly analyze the design and set all of the engineering parameters. The following section describes in detail what engineering calculations were done in order to set all of the engineering parameters of the design.

12.1 Design Forces

By running a MATLAB simulation of all of the forcing modes that the final design will see, the maximum force that the valve will undergo in each component was found. Because all of the internal components are connected in series, each component will see the same force. The simulation was completed for the entire range of applicable pressures, from 0 to 7000 psi. Figures 12.1, 12.2, 12.3 and 12.4 show the resultant forcing profiles. These forcing profiles relate to a situation when the stem holder and power piston are attached and moving together only. If the stem holder and power piston are not coupled together, the forcing profiles shown below do not describe the force seen on any part of the valve exactly. Any negative values for the force mean that the full valve assembly is being forced away from the accumulator entrance. Any positive values mean that the assembly is being forced into the accumulator entrance.

Figure 12.1: Maximum force that the series assembly will see when forced open

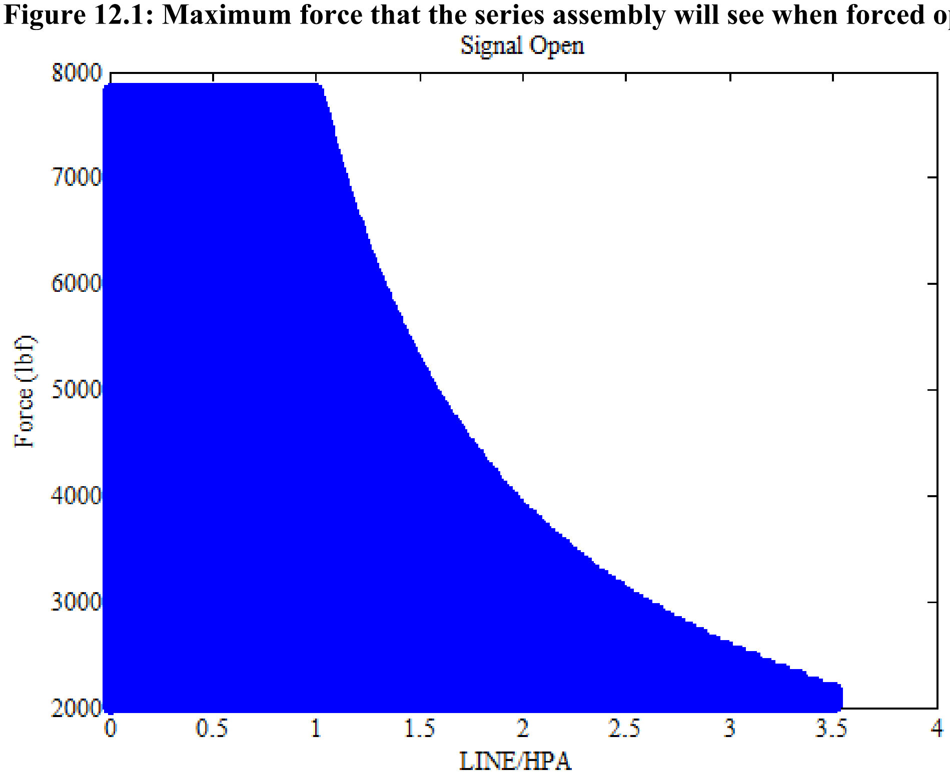


Figure 12.2: Maximum force that the series assembly will see when forced open

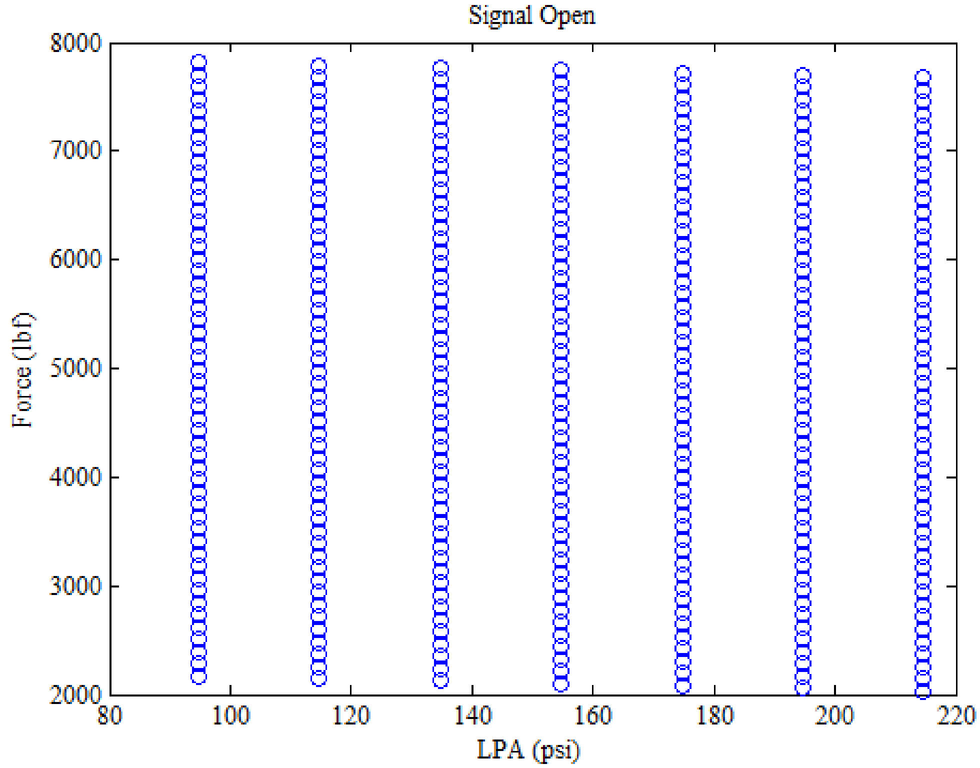


Figure 12.3: Maximum force that the series assembly will see when forced closed

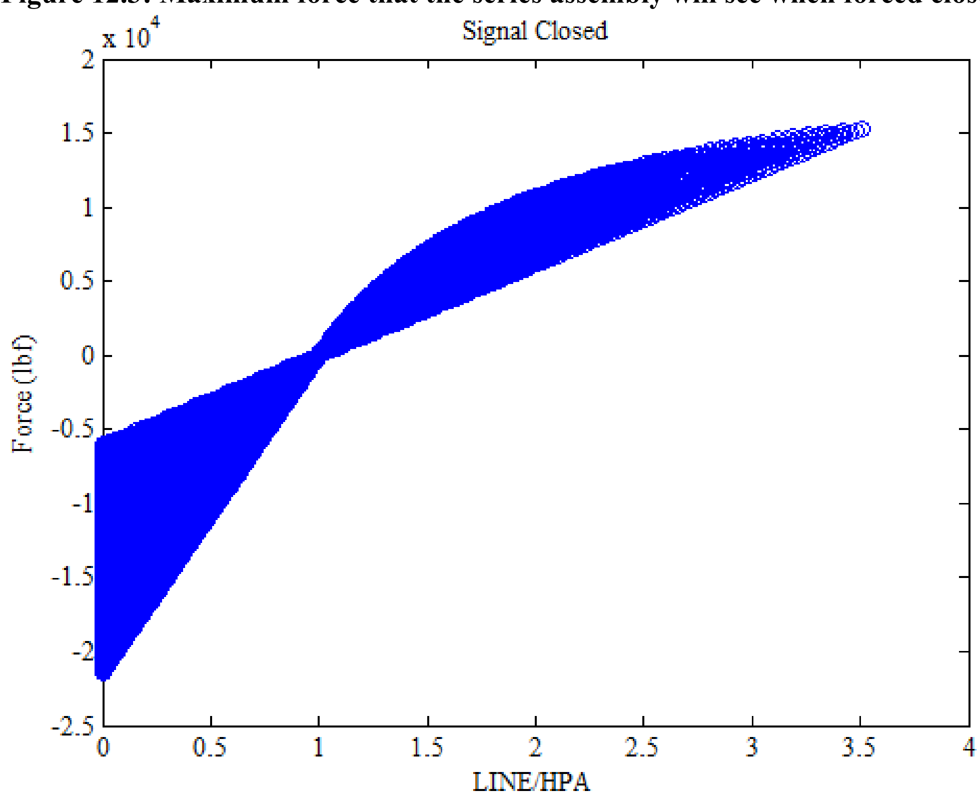
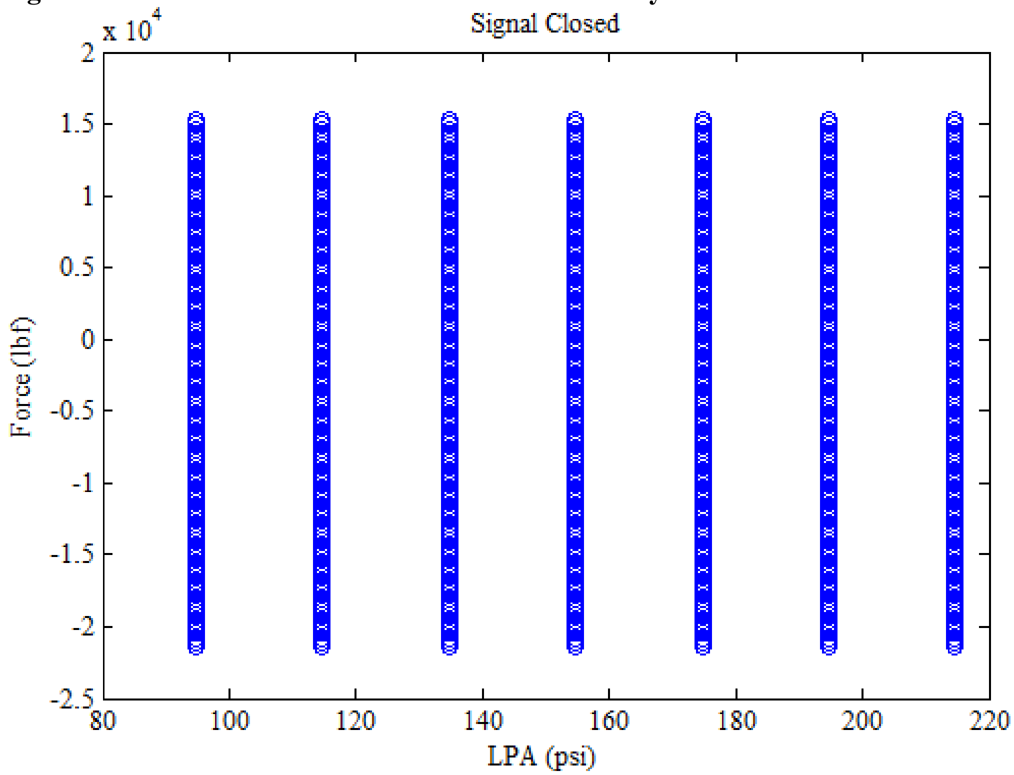


Figure 12.3: Maximum force that the series assembly will see when forced closed



12.2 Bladder Interaction Force

First, the bladder extrusion force that will be applied to the head of the poppet valve must be known to ensure that the bladder will be protected from rupture. HYDAC, International is a manufacturer of high pressure accumulators and the bladders inside of them. A call to this company resulted in useful information that allowed us to finalize the size of the spring in the final design. A representative stated that the maximum pressure difference that the bladder can see before dangerously extruding is 6 psi. Any pressure difference greater than this amount would cause damage to the bladder material. The representative also stated that for a safe valve we should design our system to prevent pressure differences greater than 5 psi. Equation 12.1 shows how the maximum bladder force is calculated using this information. In this equation, $F_{BLADDER}$ is the force that the bladder can supply to the valve head; ΔP is the maximum bladder pressure difference; A_{HEAD} is the area of the poppet head; and D is the diameter of the poppet valve head (1.44 inches).

$$F_{BLADDER} = \Delta P * A_{HEAD} = \Delta P * \left(\frac{1}{4}\right)\pi D^2 = 8.14\ lbs \qquad \text{[Eq. 12.1]}$$

The diameter of the valve head is equal to the diameter of the valve head of the current EPA design. The current EPA design opens half of an inch in the fully open position. Upon our own investigation using simplified pressure drop equations we found that the minimum pressure drop is achieved through a poppet style valve of our dimensions when opened is when the valve head has moved half of an inch into the accumulator with diminishing returns after half of an inch of extrusion. These two reasons support our decision for the valve actuation length to be half an inch. Because the valve opens half of an inch, a spring constant of 15 lbs/in was chosen. This means that, at a force of 7.5 pounds on the valve head, the bladder will be able to close the valve completely, thus preventing rupture of the bladder.

In the final design, however, a spring that is designed for velocity fuse functionality will be used in place of this spring. Instead of a spring, when the pressure in the tank becomes lower than 2200 psi, a pressure sensor should send a signal to close the valve. In this way, the bladder will be protected, despite a stronger valve spring.

12.3 Velocity Fuse

The EPA's requirements call for a velocity fuse function to be integrated into the valve. The current EPA valve accomplishes this with a spring, and so the spring in our design could also accomplish the velocity fuse function. This valve should not allow operation at flow rates above 185 GPM. The Bernoulli equation could be used to calculate the force that the valve head will see due to a large flow rate. This force could then be used to calculate the necessary spring constant to prevent this excessive flow rate in the same way the bladder force was used to determine the spring rate necessary for bladder interaction. However since the current EPA design incorporates a spring for velocity fuse with a spring rate determined experimentally, we believe that a spring with the same spring constant as the current velocity fuse spring could be used in our final design in place of the spring currently employed for bladder interaction. This spring has a spring rate of 75 lb/in.

12.4 Failure of Each Component

In the following section, the engineering analysis that went into the design of each component will be presented. The main failure modes and their prevention will be described in detail.

12.4.1 Valve Housing

Housing Failure modes:

1. Shear through threading in bolts
2. Shear through threading in accumulator
3. Rupture due to internal pressure
4. Tensile yield through area

12.4.1.1 Shear Through Bolt Threading

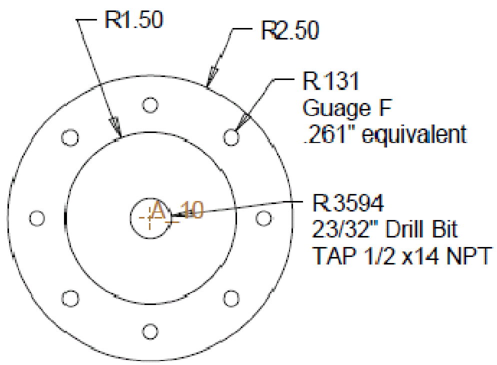
There are 8 bolts that hold the end cap of the housing onto the rest of the housing. The total force that the bolt threading sees that these bolts connect into is 12,315 lbf. This is the force due to the maximum pressure inside the housing, times the area of the end cap face within the housing. This force will be seen entirely by the eight Gauge F 0.261" equivalent bolts located as shown on Figure 12.4 (this figure shows the housing end cap geometry). These bolts will be made of steel and threaded into the steel housing for a full ½ inch engagement length. Table 12.1 shows the parameters that will be plugged into Equation 12.2 [17] in order to determine the shear area of each of these bolts (.2695 in^2). The total shear area is equal to 8 times that area, or 2.1559 in^2. Applying the total force to this shear area results in a shear stress of 5,715 psi. The yield strength of AISI 420 stainless steel is 193,000 psi, so these threads will not fail. The factor of safety against this type of failure is 33.7.

$$A_n = \pi n L_e D_{s-min}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}(D_{s-min} - E_{n-max})\right] \qquad \text{[Eq. 12.2]}$$

Table 12.1: Parameters used to find shear area of the end cap bolting

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	0.5 in
$D_s$min	Minimum major diameter of external thread	0.2408 in
$E_n$max	Maximum pitch diameter of internal thread	0.2224 in

Figure 12.4:



12.4.1.2 Shear Through Threading in Accumulator

The accumulator threading (2-12UN-2A) will see the maximum force on the Valve Head. This force is equal to 19,350 lbf. Table 12.2 shows the parameters of the threading into the accumulator that are used to find the shear area. Using equation 12.3 [17], the shear area is equal to 3.6995 in^2. Using this shear area and a force of 19,350 lbf, the total shear stress in the threading will be 5,300 psi. The yield strength of AISI 420 stainless Steel is 193,000 psi, which gives a factor of safety of 36.4.

$$A_n = \pi n L_e K_{n-max}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}(E_{s-min} - K_{n-max})\right] \qquad \text{[Eq. 12.3]}$$

Table 12.2: Parameters used to find shear area of the accumulator thread

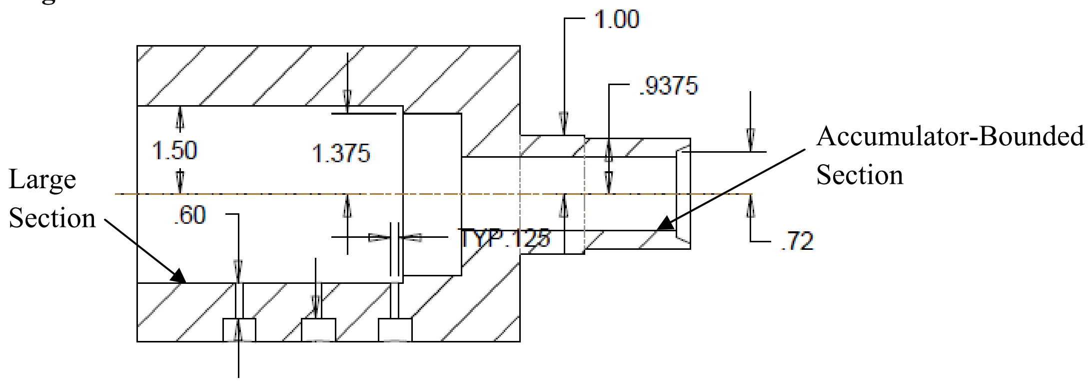
Parameter	Physical Meaning	Value
n	Threads per inch	12
L_e	Length of engagement	1.09 in
$K_n$max	Maximum minor diameter of internal thread	1.7848 in
$E_s$min	Minimum pitch diameter of external thread	1.8000 in

12.4.1.3 Rupture Due to Internal Pressure

Using equation 12.4, we can estimate the stress on the internal wall of two sections of the housing that are critical: the large section and the accumulator bounded section. This equation is taken from “Structural Mechanics of Buried Pipes” [18]. In this equation, P is the pressure inside the vessel, r_o is the outer radius of the vessel, r_i is the inner radius of the vessel, and S_I is the stress in the internal part of the vessel, which is the location of the maximum stress. The two sections that this equation applies to are shown in Figure 12.5. In the large section, the outer diameter is 3.5 inches and the inner diameter is 3 inches. The maximum pressure inside the valve is 7000 psi. The tensile stress due to this pressure in the large section is about 18,700 psi. In the accumulator-bounded section, the inner diameter is 1.25 inches and the outer diameter is 1.875 inches. The tensile stress due to the pressure in the accumulator-bounded section is 2,300 psi. The factors of safety for each of these sections will be discussed in section 12.4.1.4.

$$S_Y \geq S_I = \frac{P(r_o^2 - r_i^2)}{(r_o^2 + r_i^2)} \qquad \text{[Eq. 12.4]}$$

Figure 12.5:



12.4.1.4 Tensile Yield Through Area

In addition to the tensile stress due to pressure on each area of the valve housing, a force of 12,315 lbf is acting on these two sections from the pressure trying to force the end cap out. In the large section of the valve housing, the tensile area is 2.55 in^2, and the tensile stress is equal to 4,850 psi. In combination with the tensile stress in the previous section, the total tensile stress in the large section is 23,550 psi. The factor of safety on this section for total tensile loading is 8.1.

In the accumulator bounded section, the tensile area is 1.53 in2, and the tensile stress is equal to 8,050 psi. In combination with the tensile stress in the previous section, the total tensile stress in the accumulator-bounded section is 10,350 psi. The factor of safety on this section for total tensile loading is 18.6.

12.4.2 Valve Housing End Cap

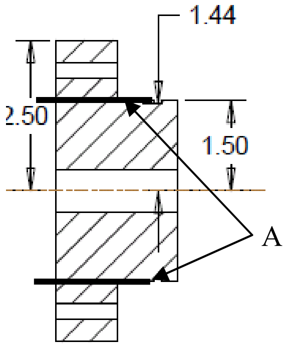
End Cap Failure modes:

1. Shear through end cap thickness

12.4.2.1 Shear Through End Cap Thickness

Equation 12.5 can be used to determine the maximum stress that the end cap will see due to shear in the section. The maximum shear force that the end cap will see is 12,350 lbf (V_{MAX}). The shear area can be taken from the Figure 12.6 (see the cut lines labeled A). The shear area in this geometry is equal to about 9.42 in^2. The maximum shear stress that this part will see is about 1,350 psi. The yield strength of the aluminum material (AISI 420 stainless steel) that this part is made of is 193,000 psi. This allows for a factor of safety of about 142 against this mode of failure.

Figure 12.6:



$$S_Y \geq \tau = \frac{V_{MAX}}{A_{SHEAR}} \qquad \text{[Eq. 12.5]}$$

12.4.3 Power Piston

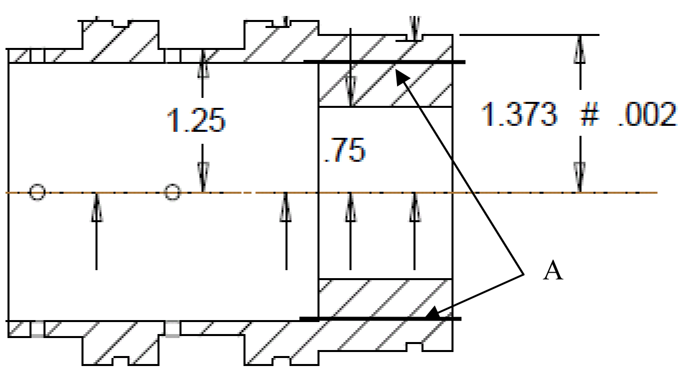
Power Piston failure modes:

1. Shear through section
2. Shear at sealing geometries
3. Tension at sealing geometry due to moments

12.4.3.1 Shear Through Section

Equation 12.5 in section 12.4.2.1 can be used to determine the maximum stress that the power piston will see due to shear in the middle section. The maximum shear force that the piston will see through the middle is 8,000 lbf (V_{MAX}). The shear area can be taken from the CAD drawing shown below (see the cut lines labeled A in Figure 12.7). The shear area in this geometry is equal to about 8.13 in^2. The maximum shear stress that this part will see is about 1,000 psi. The yield strength of the AISI stainless steel material that this part is made of is 193,000 psi (according to CES Edupack 2008 [19]). This allows for a factor of safety of about 193 for this mode of failure.

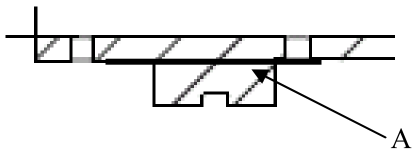
Figure 12.7:



12.4.3.2 Shear at Sealing Geometries

The sealing geometry can undergo large shear stresses when high pressure is applied to one side and low pressure is applied to the other side. The sealing geometry is shown in Figure 12.8, with the location of the shear section labeled by line A. The maximum shear stress would occur when the pressure in the middle of the geometry is 7000 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 2.16 in^2. The maximum shear force is equal to the shear area multiplied by 7000 psi, or 15,200 lbf. Equation 12.5 in section 12.4.2.1 can be used to find the total shear stress, with a shear area equal to 5.11 in^2 in this geometry. The maximum shear stress that this part will see is 3,000 psi, and will act through line A in Figure 4.5. This allows for a factor of safety of about 64.8 for this mode of failure.

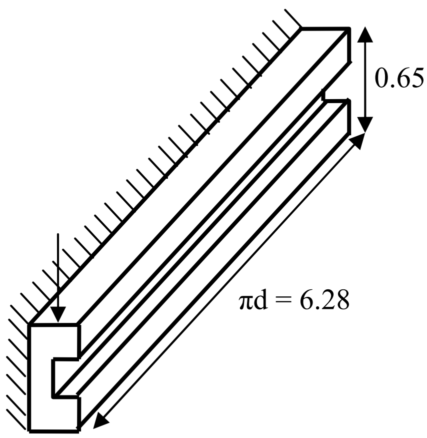
Figure 12.8:



12.4.3.3 Tension at Sealing Geometries Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This will act as a cantilevered beam, as in figure 12.9. The force in this section is 15,150 and is located halfway out on this cantilever beam (0.125 inches). The moment that is seen at the wall is then equal to 1900 lb-in (*M*). The moment of inertia of this "beam" is equal to 0.1797 in^4 (*I*). In equation 12.6, y is equal to half of the height, or 0.325 in. The maximum stress in this beam is therefore 700 psi. With AISI 420 stainless steel being the material, the factor of safety against this kind of failure is about 275.

Figure 12.9:



$$S_Y \geq \frac{My}{I} \qquad \text{[Eq. 12.6]}$$

12.4.4 Valve Stem Holder

Stem holder failure modes:

1. Shear through threading
2. Shear through middle
3. Shear at sealing geometries
4. Tension at sealing geometry due to moments

12.4.4.1 Shear Through Threading

Analysis was done to ensure that the internal threads of the stem holder would not fail by shear. The threading will be the ½-20UNF form and will be used to fasten the external threads of the valve stem. The fine thread profile was chosen to maximize the available tensile stress area of the valve stem. To ensure safe operation, the calculation of thread shear area was done assuming the worst case geometry. Namely, this assumption was the minimum possible major diameter of the internal thread. Using the formula for shear area of an internal thread (equation 12.3 in section 12.4.1.2), the shear area, A_n, was found to be 1.08 in^2. The parameters that entered into equation 12.3 are given in Table 12.3.

Table 12.3: Parameters used to find shear area of the internal stem holder thread

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	1 in
$D_s$min	Minimum major diameter of external thread	0.4906 in
$E_n$max	Maximum pitch diameter of internal thread	0.4731 in

The length of engagement was chosen based on the allowable space in the stem holder. This length was verified by considering the varying strengths of the stem holder and the valve stem. For internal and external threads made of the same material, the minimum length of engagement can be found from equation 12.7 [17], where A_t is the tensile stress area of the externally threaded member, and $K_{n\text{-}max}$ and $E_{s\text{-}min}$ are the maximum minor diameter of internal thread, and the minimum pitch diameter of external thread, respectively.

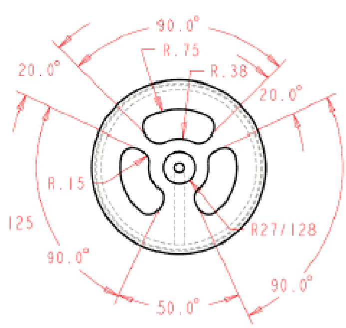
$$L_e = \frac{2A_t}{\pi K_{n-max}\left[\frac{1}{2}+\frac{1}{\sqrt{3}}n(E_{s-min}-K_{n-max})\right]} \qquad \text{[Eq. 12.7]}$$

For this case, L_e was found to be 0.222 inch. Assuming a maximum axial load on the stem holder threads of 8,000 lbf, the resulting shear stress on the threads is 7,500 psi. The yield strength for AISI 420 stainless steel was found to be 193,000 psi from the materials database [19], so the factor of safety for this kind of failure is about 25.7.

12.4.4.2 Shear Through Middle Section

Equation 12.5 in section 12.4.2.1 can be used to determine the maximum stress that the stem holder will see due to shear (in the middle section). The maximum shear force that the stem will see through the middle is 8,000 lbs (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 12.10). The shear area is equal to the total arc length on the inside cylinder (where the three side posts meet the inner cylinder) multiplied by the depth of the piece. This total shear area is equal to 1.22 in^2. The maximum shear stress that this part will see is 6,600 psi. The yield strength of the stainless steel material that this part is made of is at least 193,000 psi, which allows for a factor of safety of about 29 for this mode of failure.

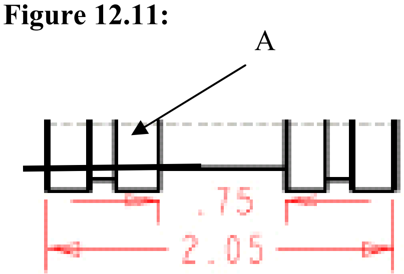
Figure 12.10:



12.4.4.3 Shear Through Sealing Geometry

The same equation can be used to determine the maximum shear stress that the sealing geometry in the stem holder will see. This sealing geometry is shown in the CAD drawing below. The maximum shear stress would occur when the pressure in the middle of the geometry is 7,000 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 0.8345 in^2. The maximum shear force is equal to 8,400 lbf. The total shear area is equal to 4.08 in^2. The maximum shear stress that this part will see is 2,100 psi, and will act through line A in Figure 12.11. This allows for a factor of safety of about 91.9 for this mode of failure.

Figure 12.11:



12.4.4.4 Tension at Sealing Geometry Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This geometry can be modeled as a cantilever beam. The force in this section is 8,400 lbs, and is located halfway out on this cantilever beam (0.0625 inches). The moment that is seen at the wall is then equal to 525 lb-in (M). The moment of inertia of this beam is equal to 0.1437 in^4 (I). In equation 12.6, y is equal to half of the height, or 0.0325 in. The maximum stress in this beam is therefore 120 psi. With stainless steel being the material, the factor of safety against this kind of failure is 965.

12.4.5 Valve Stem

Valve stem failure modes:

1. Shear through threading
2. Tensile yield through stem area
3. Rupture due to internal pressure

12.4.5.1 Shear Through Threading

The analysis of shear failure on the threads of the valve stem was performed in the same manner as that for the stem holder. Here, the equation for the shear area of an external thread was used (equation 12.3 in section 12.4.1.2). The shear area, A_s, was found to be 0.799 in^2 using the parameters in Table 12.4. Again, using a load of 8,000 lbf as in the case of the stem holder (since the load will be transferred through the valve stem), the shear stress was found to be 10,050 psi. The yield strength of AISI 420 stainless steel was given in the materials database [19] as 193,000 psi, which results in a factor of safety of 19 for the valve stem threads.

Table 12.4: Parameters used to find shear area of the external valve stem thread

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	1 in
$K_n$max	Maximum minor diameter of internal thread	0.457 in
$E_s$min	Minimum pitch diameter of external thread	0.4619 in

12.4.5.2 Tensile Yield Through Stem Area

Equation 12.9 can be used to determine the maximum stress that the stem will see in tension. The maximum tensile force is 8,000 lbs (F_{MAX}). The outer and inner diameters are taken at the thinnest point of the stem, which will be where the threading is created on the outside of the stock material. The stock material has an outer diameter (D_O) of ½ inch, but where the threading is formed, the outer diameter is 0.419 inches. This diameter is assuming worst case geometry (minimum material condition). The stock material has an inner diameter (D_I) of ¼ inch to give a total tensile stress area of 0.0885 in^2. The maximum tensile stress that this part will see is 60,750 psi. The yield strength of the AISI 420 stainless steel material that will be used for the stem is 193,000 psi (found from the materials database [19]). This provides the valve stem with a factor of safety of 3.1.

$$S_Y \geq \frac{F_{MAX}}{A_{FORCE}} = \frac{F_{MAX}}{\left(\frac{1}{4}\right)\pi\left(D_O^2 - D_I^2\right)} \qquad \text{[Eq. 12.9]}$$

12.4.5.3 Rupture Due to Internal Pressure

Equation 12.4 in section 12.4.1.3 can be used to determine the maximum stress that will be seen in the wall of the stem if 7,000 psi is internal to the stem and 0 psi is external to the stem. According to this equation, the maximum stress is equal to 4,200 psi. Adding this stress to the stress found in the previous section results in a stress of 64,950 psi maximum. This allows for a factor of safety of 2.97 for this mode of failure.

12.4.6 Valve Head

Stem head failure modes:

1. Shear through middle
2. Tension through section due to moments
3. Tensile yield through area

12.4.6.1 Shear Through Section

Equation 12.4 in section 12.4.2.1 can be used to determine the maximum stress that the stem head will see due to shear in the middle section. The maximum shear force that the stem will see through the middle is 8,000 lbs + 19,350 (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 12.12, which is a simplified model of our valve head). The shear area in this geometry is equal to about 0.83 in^2. The maximum shear stress that this part will see is 33,000 psi. The yield strength of the stainless steel that this section of the valve stem is made of is at 193,000 psi (according to CES Edupack 2008 [19]). This allows for a factor of safety of about 5.8 against this mode of failure.

Figure 12.12:

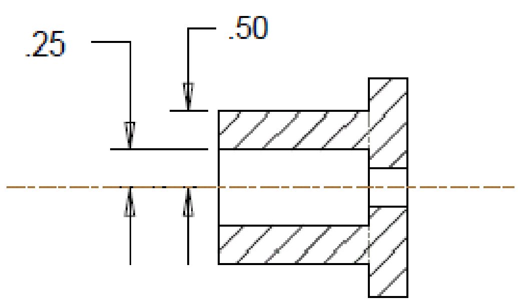
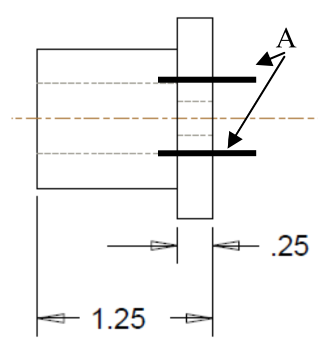


Figure 12.13:



12.4.6.2 Tension Through Section Due to Moments

Equation 12.6 in section 12.4.3.3 can be used to find the maximum tension on the valve stem head due to a moment. The force of 8,000 + 19,350 lbs is applied through the stem head. The reaction at the base of the valve head can be found by modeling this as a cantilever beam. The force in this section is 27,350 lbs, and is located on the end of this cantilever beam (0.22 inches). The moment that is seen at the wall is then equal to 6,050 lb-in (*M*). The moment of inertia of this beam is equal to 0.2618 in^4 (*I*). In the equation, y is equal to half of the height, or 0.125 in. The maximum stress in this beam is therefore 3,000 psi.

12.4.6.3 Tension Yield Through Area

The smallest area at which tensile yield would occur is 0.147 in^2. Similarly to section 12.4.5.2, with a maximum tensile force of 8,000 lbs, the stress in this section is equal to 54,500 psi. Adding this stress to that found in section 12.4.6.2, the total tensile stress seen in this part is equal to 57,300 psi. With AISI 420 stainless steel as the material of choice (yield strength of 193,000 psi), the factor of safety against this kind of failure is about 3.3.

12.5 Internal "Plumbing"

In order to actuate the valve hydraulically, ports have to be drilled through the housing walls to access the correct volumes and apply the correct pressurized fluid. In the final design, these orifice sizes will be critical in ensuring that the pressure equalization occurs in 50-100 milliseconds. Unfortunately, all we could do to ensure that this size will be adjustable by our sponsors (The EPA). All of the current sizes are 1/8", and there is space located around each port to allow for adjustment. In the final design, a solenoid actuator is built into the housing wall, which has a large square section to accommodate it, and the rest of the housing wall has plumbing inside it to connect the solenoid to the appropriate areas.

Also included in this extra square section are ports for low pressure access and an accessory port for filling of the system.

12.6 Connections and Plug-ins

12.6.1 Solenoid Actuator

A solenoid actuator is required to convert a 12 volt electric signal from the vehicle to a hydraulic signal that actuates the valve open. When the 12 volt signal is not applied, the valve must be actuated closed. This actuator is integrated into the housing. The chosen solenoid also has the properties that are required for correct actuation functionality of the valve (sending the correct pressures to the correct areas). The chosen solenoid is not rated for 7000 psi, but rather 5000 psi. It is suggested that a 7000 psi rated solenoid switch with the same switching capabilities as the WK10N-01 by HYDAC is used in this design.

12.6.2 System Fitment

On the final design, the end cap must have a 90 degree elbow with threading on one side to connect to the valve and a 4 bolt 62 flange connection to the line on the other side. This part is included in the current EPA design, and a similar part is used in our final design. The flange connection is a standard that can handle 6000 psi. A different standard connection may need to be used to be useful in a 7000 psi application. The connection suggested and sized for out design is the 6892-20-20 JIC to Flange Elbow (90°) by Thompkins Industries, Inc.

13 PROTOTYPE DESIGN

13.1 Prototype Functions

Our prototype will clearly demonstrate that the final design can perform the following functions: closing when commanded closed, equalizing pressure in the line with the pressure in the accumulator and opening when commanded open, leaking pressure into the accumulator when commanded closed and pressure in the line exceeds the pressure in the accumulator, closing when commanded open but the bladder reaches the valve opening, being able to fit into the current EPA hydraulic system, and being of a less complicated design than the existing EPA high pressure accumulator valve. To show that our final design valve will function, our prototype will be built to the same interior dimensions (piston sizing and acting areas) as the final design. Although the working pressures themselves will be different, they will act on the same surfaces. It can be assumed that if the valve actuates at these low pressures, it should only actuate better at the higher pressures used in the EPA system.

13.2 Design Differences

The prototype will have several differences from the final design, however we still plan to demonstrate and prove the main functional requirements of the final design. The main differences between the prototype and the final design recommendation are in the material selection, the exact build of the prototype, and the working pressures that the prototype will be exposed to. The materials differ due to the need to see the inside of the prototype during functioning, and the much smaller strength required by the much lower working pressure when compared to the final design requirements. There are several simplifications in the final design, and these are mainly to aid in our manufacturing process and to reduce the cost of building the prototype.

13.2.1 Materials

The materials of the prototype are almost all completely different than those used in the final design. The final design will be made almost entirely out of stainless steel for its strength and corrosion resistance properties. The prototype design will have a housing made from aluminum, valve stem made from nylon with a PVC valve stem head, a valve end cap made from aluminum, and a power piston and valve stem holder made out of the same PVC as the valve stem head. The PVC was selected for its cost and ease of manufacturing. Strength of the PVC was considered and we decided that the PVC would be strong enough for the applications we will be using in the prototype. The nylon for the valve stem was selected because it came in hollow rod form and it was shown that we do not have the capabilities to manufacture a through hole in any material that is one eighth inch diameter and 7 inches in length without cutting it into sections first and then reconnecting the sections. The nylon also fulfilled our low cost requirement.

13.3 Simplifications

The following are the simplifications we have made to the prototype design to either save cost, or reduce complexity so that the prototype can be manufactured within the time and technological constraints of this class: end cap design, valve stem through hole dimensions, valve actuation.

13.3.1 End Cap Design Change

The end cap design was modified from the final design plans so that instead of meeting the standard required by the EPA the port to the line would be able to accept a male 1/2 x 18 NPT plastic fixture. We also decided that incorporating a 90 degree exit at the back of the valve housing was not needed for the prototype design because it did not help us demonstrate any of the major functions we are aiming to demonstrate.

13.3.2 Valve Stem Dimension Modifications

The valve stem dimensions, particularly the diameter of the hole in the center, have been altered due to manufacturing constraints. A discussion with GSI Dan revealed that we would not be capable of drilling a hole to a depth of 7 inches with a diameter of 1/8 inch. Instead of finding a way around drilling the hole, we searched for any sort of tubes that had the same outer diameter (1/2 inch) and had the smallest inner diameter we could find. Our search for this yielded inexpensive nylon tube, however the minimum inner diameter was 1/4 inches. This yielded a change in our prototype design. Because a simple tube was used for most of the stem, a separate valve head (with a simplified geometry) was created.

13.3.3 Valve Actuation Change

The change in valve actuation leads to several changes in the prototype design. First and foremost, the decision to change from an automated solenoid type switching mechanism to a manual ball valve system was made in order to decrease the cost of the prototype, make the valve housing manufacture less complicated. The effects of this change are probably the most significant of those listed in this section.

1) There will be three ports along the side of the valve housing instead of one.
2) A ‘controller’ will be constructed of 3 ball valves plus tubing and fittings
3) The holes that contain the pressure signals will be holes straight through the valve housing to the outer ports instead of directed through the housing to the main solenoid switch.
4) There will be no solenoid switch.

13.3.4 Working Pressure

The prototype is designed for a maximum pressure of 150 psi. We do not anticipate testing at this pressure due to general concerns of safety. For instance, water shooting out of a small leakage at this pressure and spraying someone in the eye, or spraying onto anything nearby that may be not be water resistant, is a safety concern. Even at lower pressures, safety glasses will be worn to protect the eyes of those in the area. This is an incredible difference from the final design and is the main enabling factor allowing all of our material selection changes. The working pressures of the final design are anticipated to be at a maximum of 7000 psi which is much, much higher than we will use for demonstration purposes. This does limit the provability of the prototype in that we will be unable to prove by physical demonstration that our final design is capable of operating under all of the circumstances necessary for safe operation. Instead to prove this most important function of the final design we will have to rely on engineering analysis and mathematical calculations which fall short of actual modeling, however through careful application should give an adequate representation of the final design capabilities for recommendation to the EPA.

13.4 Description

The prototype we are building is designed to demonstrate the main functionality of the final design. It will show that our final design will open when signaled to open, and close when it is signaled to close. The prototype will also demonstrate that the final design will fit into the existing hydraulic system at the accumulator connection. This will be shown by using an accumulator in our experimental setup with identical fittings to the EPA’s current hydraulic system. We also wish to demonstrate that the final design is capable of allowing one-way flow into the accumulator when the pressure in the line exceeds the accumulator pressure and the valve is signaled closed. We also believe that with this prototype we can show that our final design incorporates the ability to interact with the bladder and prevent harmful bladder extrusion from occurring. To summarize we plan to prove that the final design can open and close correctly, as well as complete the customer one-way-flow function, bladder interaction function, and ability to be installed directly into the existing system.

Our prototype will be able to demonstrate all of these things through pressure monitoring in a simulated high pressure accumulator, simulated line, and simulated low pressure accumulator, as well as monitoring flow into and out of the 'high pressure' accumulator, and lastly by creating a system for measuring the position of the valve stem holder during operation.

13.5 Engineering Drawings

Below are the engineering drawings for the various prototype components we plan to manufacture ourselves as they are designed thus far. The drawings are presented in Figures 13.1 to 13.6.

Figure 13.1: Prototype Housing (Dimensions in Inches and Degrees)

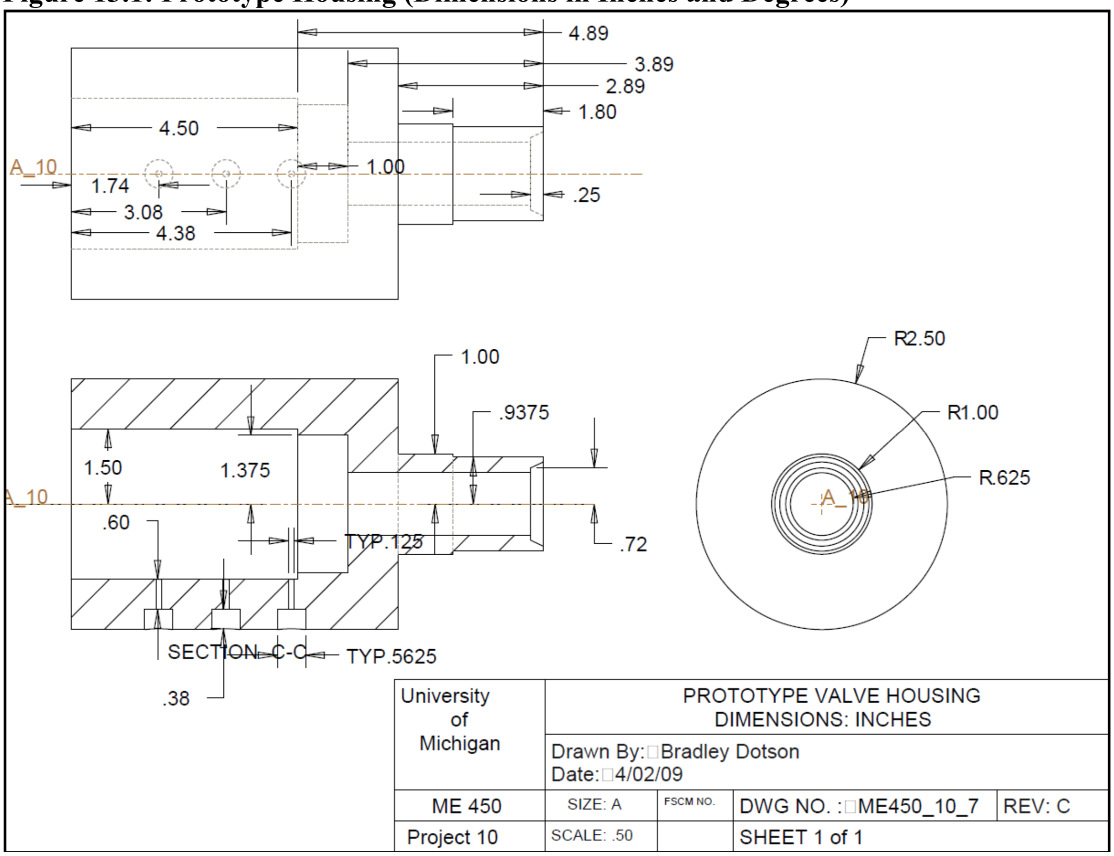


Figure 13.2: Prototype Housing End Cap (Dimensions in Inches and Degrees)

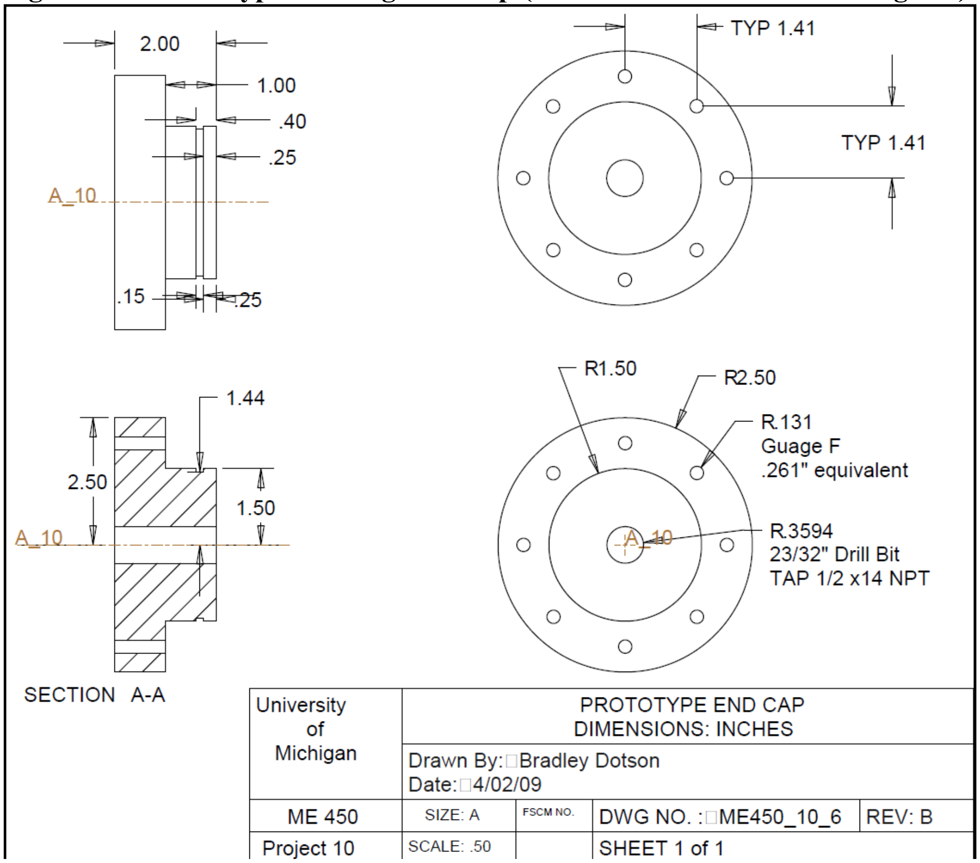


Figure 13.3: Prototype Power Piston (Dimensions in Inches and Degrees)

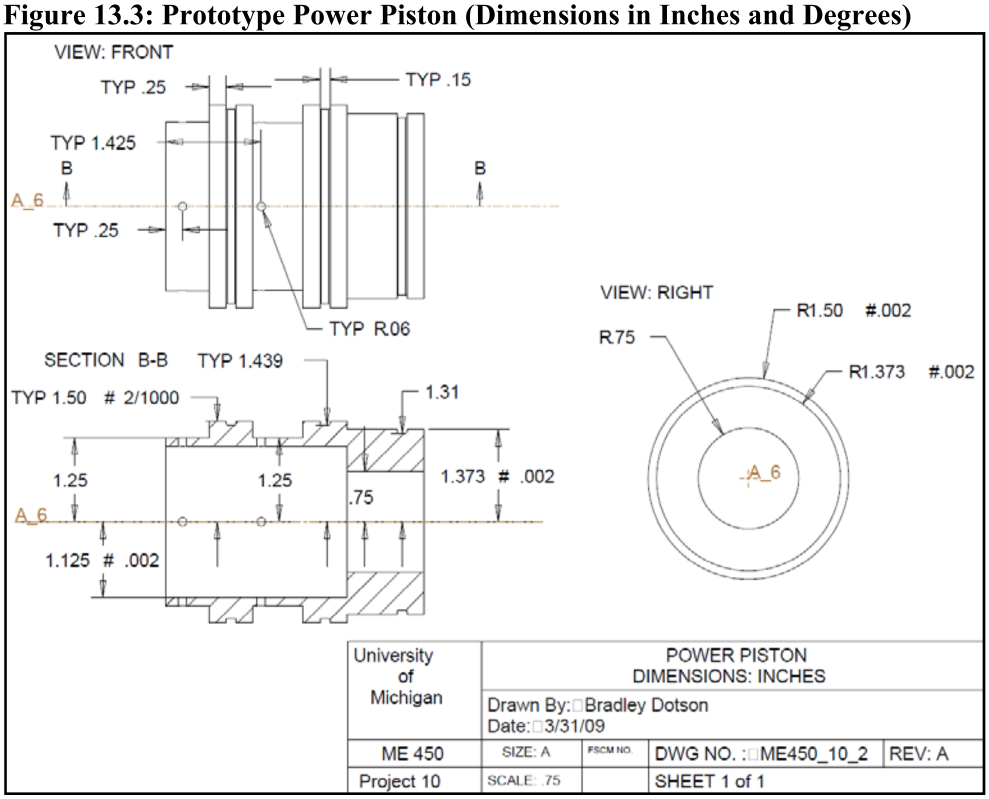


Figure 13.4: Prototype Stem Holder (Dimensions in Inches and Degrees)

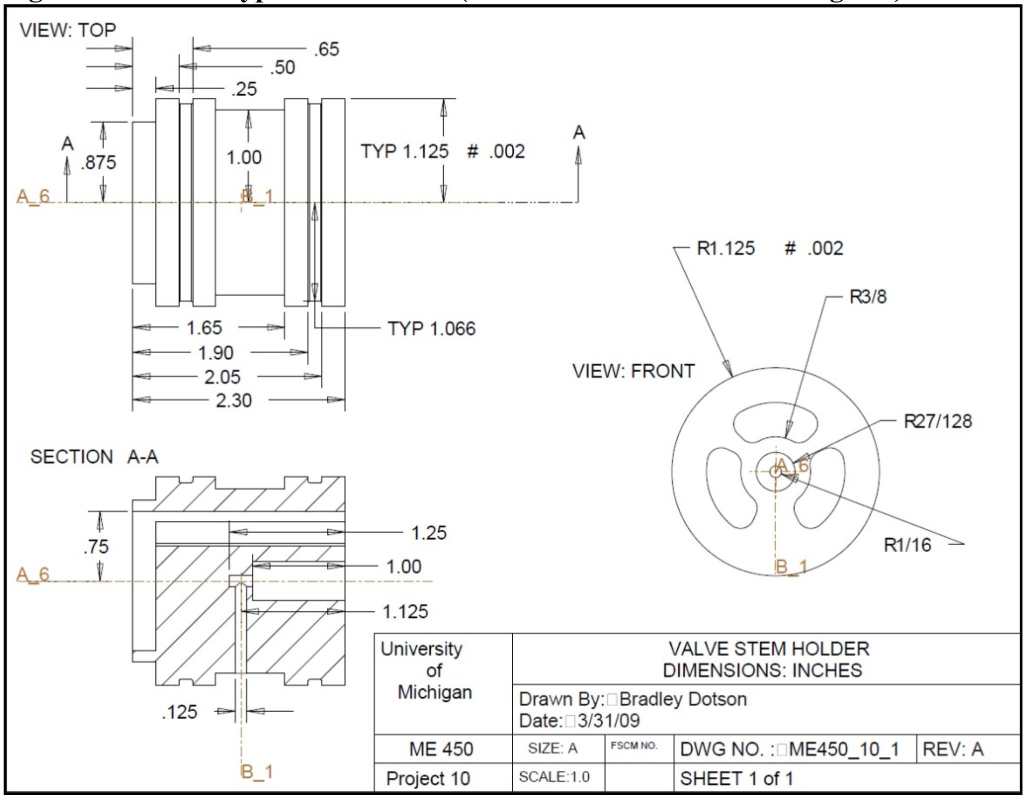


Figure 13.5: Prototype Stem (Dimensions in Inches and Degrees)

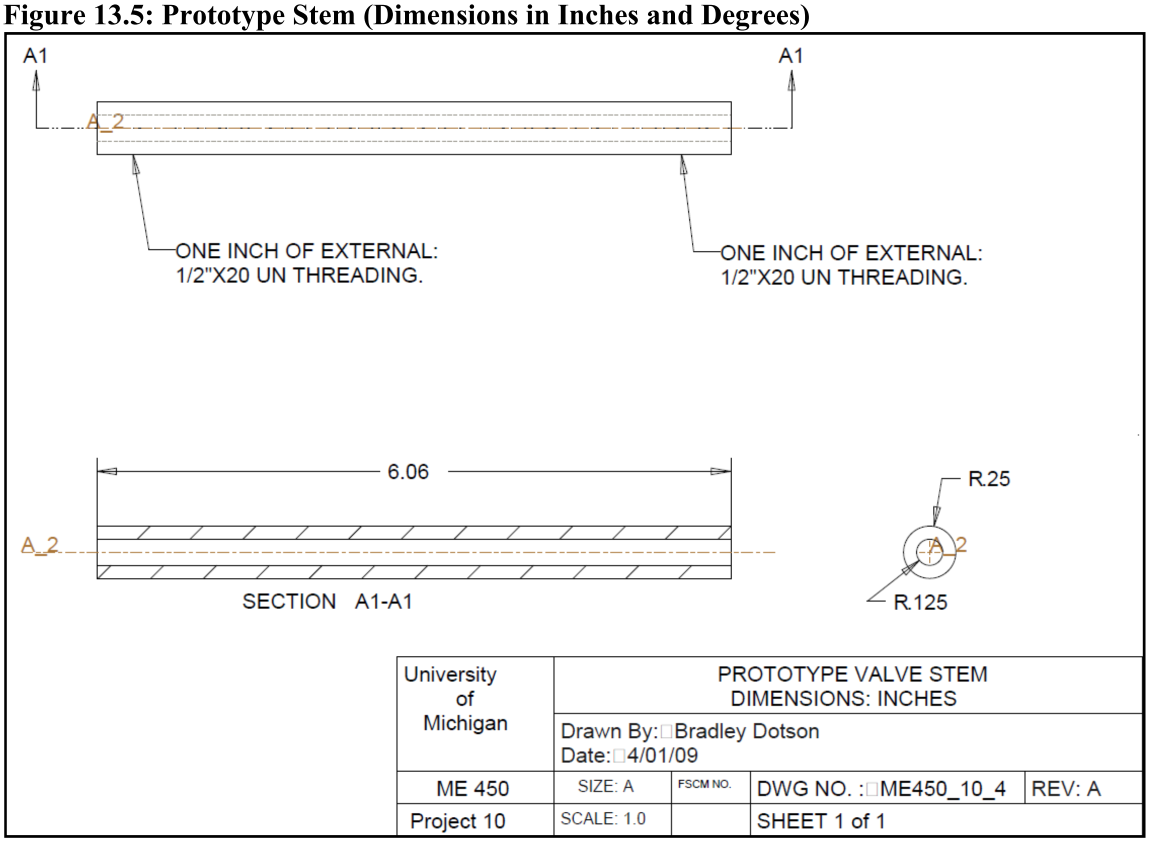
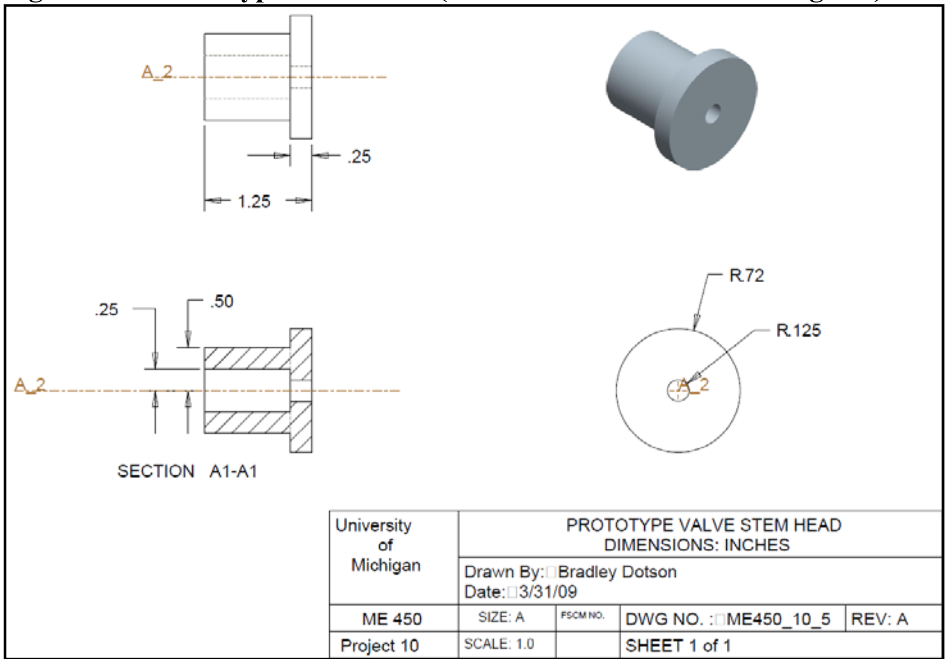


Figure 13.6: Prototype Stem Head (Dimensions in Inches and Degrees)



13.6 Material Selection

In this section we give our reasoning for choosing the materials we chose for the various parts of the valve design shown in the engineering drawings above.

13.6.1 Prototype Valve Housing

For the prototype valve housing we have chosen to use aluminum. Originally we planned on using a clear casting resin or clear acrylic housing because they allowed us to see the movement of the piston and stem holder inside the housing during different conditions and actuation positions.

13.6.2 Prototype End Cap

The prototype end cap will be made out of the same aluminum that the prototype valve housing is constructed from. This is because the two pieces will have the same outermost diameter already, and we will have extra material left over since the aluminum comes in one foot long sections and we only need 9.39 inches for the valve housing. This saves us the expense of ordering another piece of material solely for the end cap construction.

13.6.3 Prototype Power Piston and Valve Stem Holder

For the prototype power piston and valve stem holder we have chosen to use PVC. This was chosen due to the relatively low cost of PVC, ease of manufacture, its resistance to corrosion in water, and that it is strong enough to withstand the forces predicted to be acting on it.

13.6.4 Prototype Valve Stem

For the prototype valve stem we are going to use nylon and PVC. Upon finding out that the only way we would be able to machine a 1/8” hole all the way through the center would be to cut the valve stem into segments, drill the hole through each segment, and then reassemble the valve stem using epoxy, or glue, we decided to find a tube with the center hole already cut. The only tube of this nature that we found with the outer diameter we desired is a one half inch diameter nylon tube with a one quarter inch diameter center hole. The nylon tube is still thick enough to add threads to both ends, and has a high enough tensile strength to resist the predicted thread shearing forces. For the valve head itself we will be using leftover PVC from the generation of the power piston and valve stem holder. This valve head will be attached to the valve stem with threading.

14 PROTOTYPE DESIGN ENGINEERING ANALYSIS

Like with the final design, the prototype design needed to be analyzed so that the design was assured not to fail under testing. 150 psi was chosen as the maximum pressure set for our prototype for all strength calculations, but actual testing is not expected to exceed 50 psi. The following section describes in detail what engineering calculations were done in order to set all of the engineering parameters of the prototype.

14.1 Design Forces

By running a MATLAB simulation of all of the forcing modes that the prototype will see, the maximum force that the prototype will undergo in each component was found. Because all of the internal components are connected in series, each component will see the same force. The simulation was completed for the entire range of applicable pressures, from 0 to 150 psi. Figures 14.1, 14.2, 14.3, and 14.4 show the resultant forcing profiles. It can be seen from these profiles that the maximum force that the internal components of the prototype will see under the most adverse loading is 403 lbs. This force will be used throughout this section to test the various failure modes of each internal component. Any negative values for the force mean that the assembly is being forced away from the accumulator entrance. Any positive values mean that the assembly is being forced into the accumulator entrance.

Figure 14.1: Maximum force that the series assembly will see when forced open

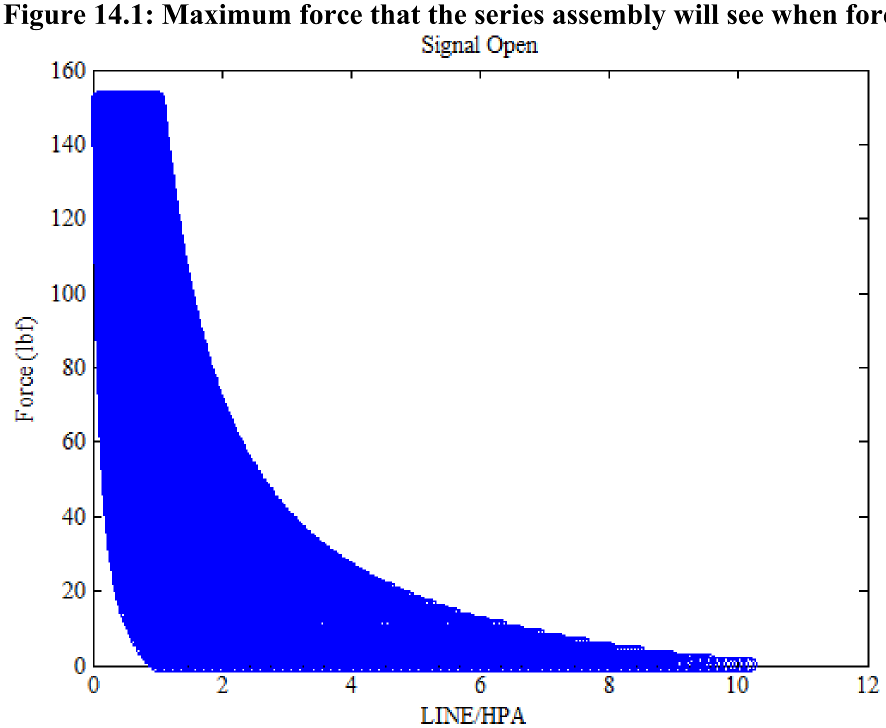


Figure 14.2: Maximum force that the series assembly will see when forced open

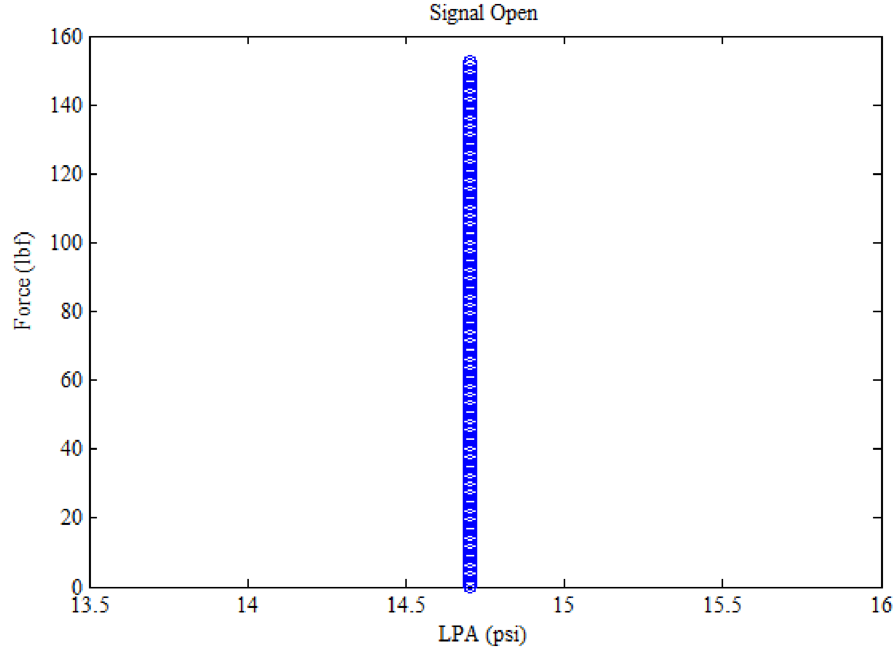


Figure 14.3: Maximum force that the series assembly will see when forced closed

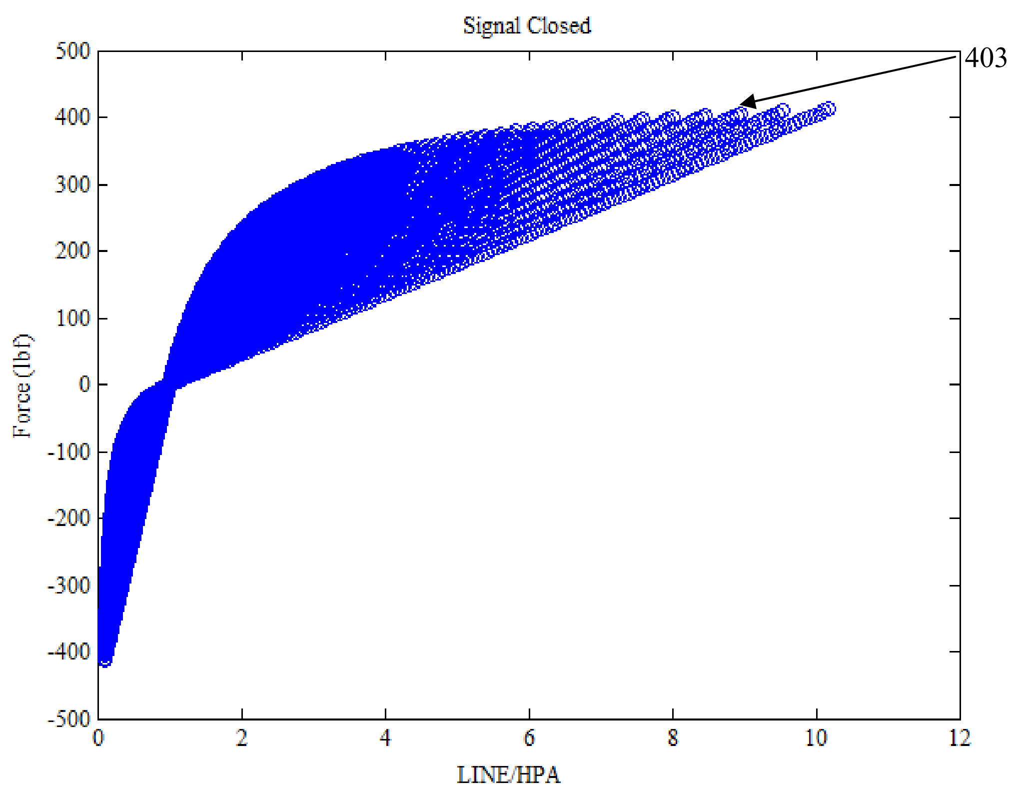
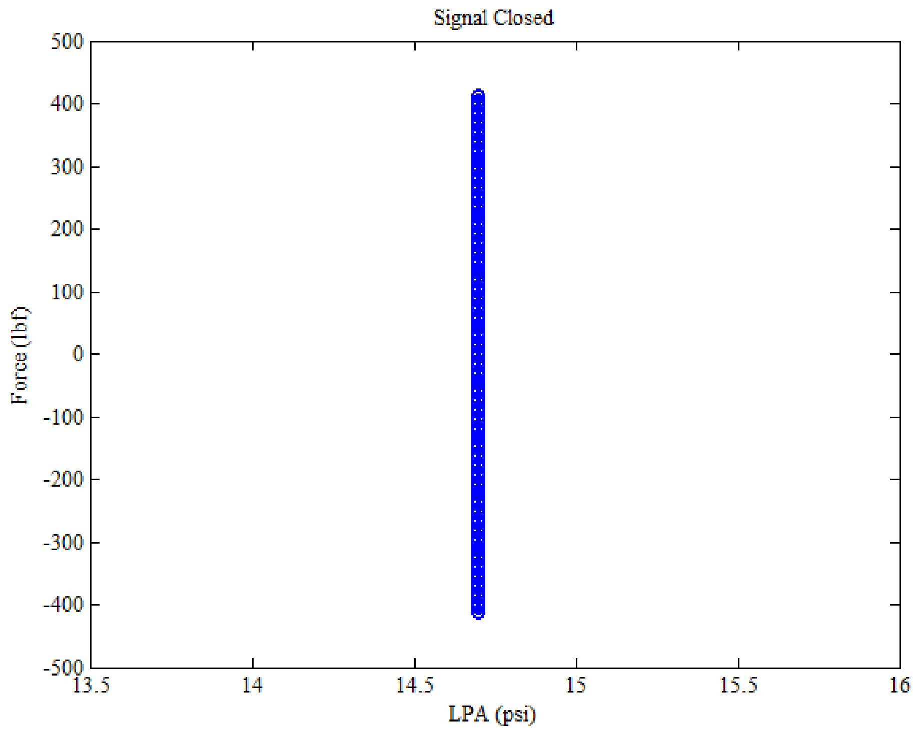


Figure 14.4: Maximum force that the series assembly will see when forced closed



14.2 Materials

For the prototype, material was chosen for what was easily available in stock, and was ensured to not fail under all of the conditions listed in the following sections before it was ordered.

14.3 Bladder Interaction Force

As in the final design, the maximum bladder extrusion force allowable is 8.14 lbs. A spring with a spring rate of 15 lb/in was chosen for this design as well. Such a spring rate is necessary to protect the bladder from harm during testing of this valve.

14.4 Velocity Fuse

The spring that is used in the prototype design will be for bladder interaction only. The prototype design will demonstrate the ability of the valve to meet only some of the customer requirements. The velocity fuse functionality of the valve will not be tested by the prototype.

14.5 Failure of Each Component

The areas on the final design that cause movement of the piston are designed using the full scale pressures. Although our prototype will be operating at much lower pressures, the prototype needs to only demonstrate the correct movement that the piston will undergo under the applied signal. It was for this reason that the areas in the prototype were designed for the full scale pressures. At lower pressures, the valve will not open as fast, but as long as the pressures are the right magnitude relative to one another, the valve should still move correctly. In the following section, the engineering analysis that went into the design of each component will be presented. The main failure modes and their prevention will be described in detail.

14.5.1 Valve Housing

Housing Failure modes:

5. Shear through threading in bolts
6. Shear through threading in accumulator
7. Rupture due to internal pressure
8. Tensile yield through area

14.5.1.1 Shear Through Bolt Threading

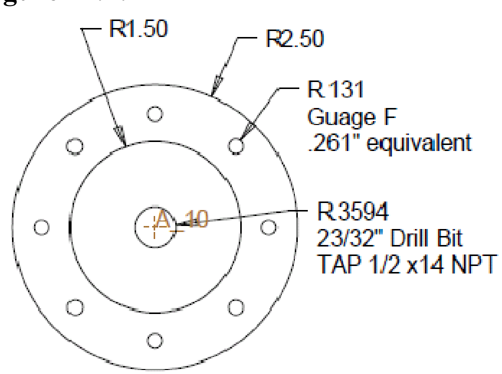
There are 8 bolts that hold the end cap of the housing onto the rest of the housing. The total force that the bolt threading sees that these bolts connect into is 1060.29 lbf. See section 14.2.1 for details on how this force is found. This force will be seen entirely by the eight Gauge F 0.261" equivalent bolts located as shown on Figure 14.1 (this figure shows the housing end cap geometry). These bolts will be made of steel and threaded into the aluminum housing for a full ½ inch engagement length. Table 14.1 shows the parameters that will be plugged into Equation 4.1 [17] in order to determine the shear area of each of these bolts (.2695 in^2). The total shear area is equal to 8 times that area, or 2.1559 in^2. By applying the total force to this shear area, the result is a shear stress of 491.8 psi. The yield strength of 6061 aluminum is 35,000 psi, so these threads will not fail. The factor of safety against this type of failure is 71.2.

$$A_n = \pi n L_e D_{s-min}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}\left(D_{s-min} - E_{n-max}\right)\right] \qquad \text{[Eq. 14.1]}$$

Table 14.1: Parameters used to find shear area of the end cap bolting

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	0.5 in
$D_s$min	Minimum major diameter of external thread	0.2408 in
$E_n$max	Maximum pitch diameter of internal thread	0.2224 in

Figure 14.1:



14.5.1.2 Shear Through Threading in Accumulator

The accumulator threading (2-12UN-2A), as well as the whole of the valve housing, will see the maximum closing force as well as the force on the end cap. This force is equal to 1463 lbf, where the closing force is equal to the force that is applied to the in-series assembly, which has a maximum of 403 lbf. Table 14.2 shows the parameters of the threading into the accumulator that are used to find the shear area. Using equation 14.2 [17], the shear area is equal to 3.6995 in^2. Using this shear area and a force of

1463 lbf, the total shear stress in the threading will be 395.5 psi. The yield strength of 6061 aluminum is 35000 psi, which gives a factor of safety of 88.5.

$$A_n = \pi n L_e K_{n-max}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}\left(E_{s-min} - K_{n-max}\right)\right] \qquad \text{[Eq. 14.2]}$$

Table 14.2: Parameters used to find shear area of the accumulator thread

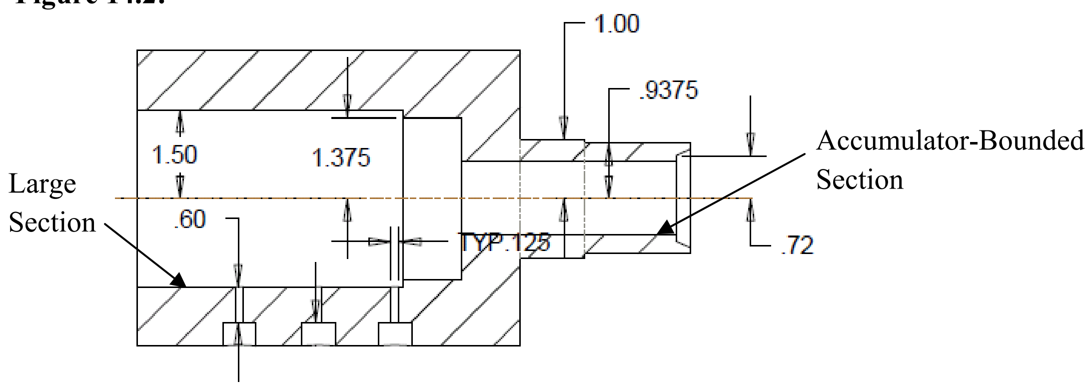
Parameter	Physical Meaning	Value
n	Threads per inch	12
L_e	Length of engagement	1.09 in
$K_n$max	Maximum minor diameter of internal thread	1.7848 in
$E_s$min	Minimum pitch diameter of external thread	1.8000 in

14.5.1.3 Rupture Due to Internal Pressure

Using equation 14.3, we can estimate the stress on the internal wall of two sections of the housing that are critical: the large section and the accumulator bounded section. This equation is taken from "Structural Mechanics of Buried Pipes" [18]. In this equation, P is the pressure inside the vessel, r_o is the outer radius of the vessel, r_i is the inner radius of the vessel, and S_I is the stress in the internal part of the vessel, which is the location of the maximum stress. The two sections that this equation applies to are shown in Figure 14.2. In the large section, the outer diameter is 5 inches and the inner diameter is 3 inches. The maximum pressure inside the valve is 150 lbf. The tensile stress due to this pressure in the large section is about 70.59 psi. In the accumulator-bounded section, the inner diameter is 1.25 inches and the outer diameter is 1.875 inches. The tensile stress due to the pressure in the accumulator-bounded section is 57.69 psi. The factors of safety for each of these sections will be discussed in section 4.1.4.

$$S_Y \geq S_I = \frac{P\left(r_o^2 - r_i^2\right)}{\left(r_o^2 + r_i^2\right)} \qquad \text{[Eq. 14.3]}$$

Figure 14.2:



14.5.1.4 Tensile Yield Through Area

In addition to the tensile stress due to pressure on each area of the valve housing, a force of 1463 lbf is acting on these two sections. In the large section of the valve housing, the tensile area is 12.57 in^2, and the tensile stress is equal to 116.4 psi. In combination with the tensile stress in the previous section, the total tensile stress in the large section is 187 psi. The factor of safety on this section for total tensile loading is 187.

In the accumulator bounded section, the tensile area is 1.53 in2, and the tensile stress is equal to 953.7 psi. In combination with the tensile stress in the previous section, the total tensile stress in the accumulator-bounded section is 1011.4 psi. The factor of safety on this section for total tensile loading is 34.6.

14.5.2 Valve Housing End Cap

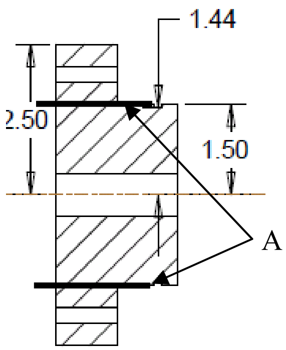
End Cap Failure modes:

2. Shear through end cap thickness

14.5.2.1 Shear Through End Cap Thickness

Equation 14.4 can be used to determine the maximum stress that the end cap will see due to shear in the section. The maximum shear force that the end cap will see through the middle is 1060.29 lbf (V_{MAX}). The shear area can be taken from the Figure 14.3 (see the cut lines labeled A). The shear area in this geometry is equal to about 9.42 in^2. The maximum shear stress that this part will see is about 112.5 psi. The yield strength of the aluminum material (6061 alloy) that this part is made of is 35,000 psi. This allows for a factor of safety of about 311 for this mode of failure.

Figure 14.3:



$$S_Y \geq \tau = \frac{V_{MAX}}{A_{SHEAR}} \qquad \text{[Eq. 14.4]}$$

14.5.3 Power Piston

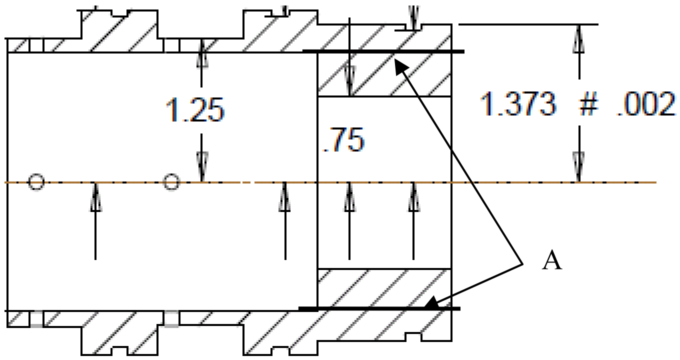
Power Piston failure modes:

4. Shear through section
5. Shear at sealing geometries
6. Tension at sealing geometry due to moments

14.5.3.1 Shear Through Section

Equation 14.4 in section 14.5.2.1 can be used to determine the maximum stress that the power piston will see due to shear in the middle section. The maximum shear force that the piston will see through the middle is 403 lbf (V_{MAX}). The shear area can be taken from the CAD drawing shown below (see the cut lines labeled A in Figure 14.4). The shear area in this geometry is equal to about 8.13 in^2. The maximum shear stress that this part will see is about 49.58 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi (according to CES Edupack 2008 [19]). This allows for a factor of safety of about 121 for this mode of failure.

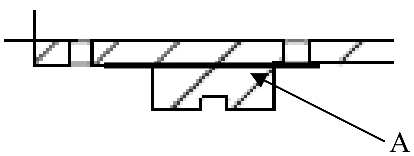
Figure 14.4:



14.5.3.2 Shear at Sealing Geometries

The sealing geometry can undergo large shear stresses when high pressure is applied to one side and low pressure is applied to the other side. The sealing geometry is shown in Figure 14.5, with the location of the shear section labeled by line A. The maximum shear stress would occur when the pressure in the middle of the geometry is 150 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 2.16 in^2. The maximum shear force is equal to the shear area multiplied by 150 psi, or 323.98 lbf. Equation 14.4 in section 14.5.2.1 can be used to find the total shear stress; with a shear area is equal to 5.11 in^2 in this geometry. The maximum shear stress that this part will see is 63.46 psi, and will act through line A in Figure 14.5. This allows for a factor of safety of about 94.5 for this mode of failure.

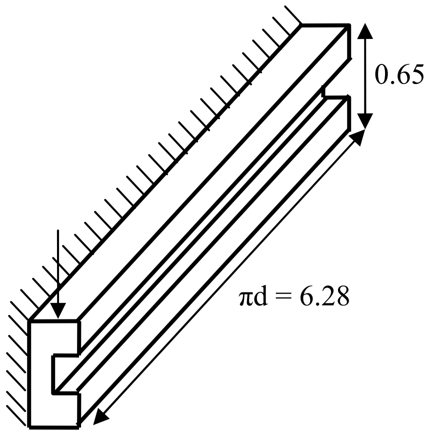
Figure 14.5:



14.5.3.3 Tension at Sealing Geometries Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This will act as a cantilevered beam, as in figure 14.6. The force in this section is 323.98 lbf, and is located halfway out on this cantilever beam (0.125 inches). The moment that is seen at the wall is then equal to 40.5 lb-in (M). The moment of inertia of this "beam" is equal to 0.1797 in^4 (I). In equation 14.5, y is equal to half of the height, or 0.325 in. The maximum stress in this beam is therefore 73.22 psi. With PVC being the material, the factor of safety against this kind of failure is about 82.

Figure 14.6:



$$S_Y \geq \frac{My}{I} \qquad \text{[Eq. 14.5]}$$

14.5.4 Valve Stem Holder

Stem holder failure modes:

5. Shear through threading
6. Shear through middle
7. Shear at sealing geometries
8. Tension at sealing geometry due to moments

14.5.4.1 Shear Through Threading

Analysis was done to ensure that the internal threads of the stem holder would not fail by shear. The threading will be the ½-20 UNF form and will be used to fasten the external threads of the valve stem. The fine thread profile was chosen to maximize the available tensile stress area of the valve stem. While the internal threads will be made of a weaker material than the external threads (a generally undesirable condition for the fine thread form), it will be shown that this is not a problem for this situation. To ensure safe operation, the calculation of thread shear area was done assuming the worst case geometry. Namely, this assumption was the minimum possible major diameter of the internal thread. Using the formula for shear area of an internal thread (equation 14.2 in section 14.5.1.2), the shear area, A_n, was found to be 1.08 in^2. The parameters that entered into equation 14.2 are given in Table 14.3.

Table 14.3: Parameters used to find shear area of the internal stem holder thread

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	1 in
$D_s$min	Minimum major diameter of external thread	0.4906 in
$E_n$max	Maximum pitch diameter of internal thread	0.4731 in

The length of engagement was chosen based on the allowable space in the stem holder. This length was verified by considering the varying strengths of the PVC stem holder and the nylon valve stem. For internal and external threads made of the same material, the minimum length of engagement can be found from equation 14.6 [17], where A_t is the tensile stress area of the externally threaded member, and $K_{n\text{-}max}$

and $E_{s\text{-}min}$ are the maximum minor diameter of internal thread, and the minimum pitch diameter of external thread, respectively.

$$L_e = \frac{2A_t}{\pi K_{n-max}\left[\frac{1}{2}+\frac{1}{\sqrt{3}}n(E_{s-min}-K_{n-max})\right]} \qquad \text{[Eq. 14.6]}$$

For this case, L_e was found to be 0.222 inch. However, because nylon and PVC have different yield strengths, an engagement length adjustment factor was calculated. This would indicate whether the thread length needed to be increased due because of the reduced strength of the PVC stem holder. The adjustment factor was found to be 0.988 from equation 14.7.

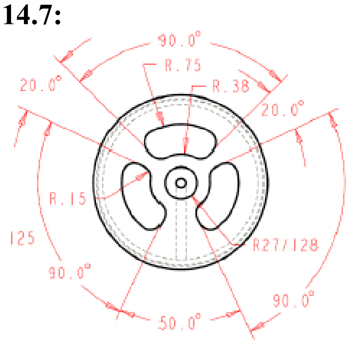
$$J = \frac{A_s\sigma_{y,ET}}{A_n\sigma_{y,IT}} \qquad \text{[Eq. 14.7]}$$

Because J was less than 1, no adjustment in engagement length was necessary, thus it was kept at 1 inch (a length that greatly exceeds the minimum requirement of equation 14.6). Assuming a maximum axial load on the stem holder threads of 403 lbf, the resulting shear stress on the threads is 373 psi. The lower bound yield strength for PVC Type 1 was found to be 6000 psi from the materials database [19], so the factor of safety for this kind of failure is about16.

14.5.4.2 Shear Through Middle Section

Equation 14.4 in section 14.5.2.1 can be used to determine the maximum stress that the stem holder will see due to shear (in the middle section). The maximum shear force that the stem will see through the middle is 403 lbs (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 14.7). The shear area is equal to the total arc length on the inside cylinder (where the three side posts meet the inner cylinder) multiplied by the depth of the piece. This total shear area is equal to 1.22 in^2. The maximum shear stress that this part will see is 330.33 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi, which allows for a factor of safety of about 18 for this mode of failure.

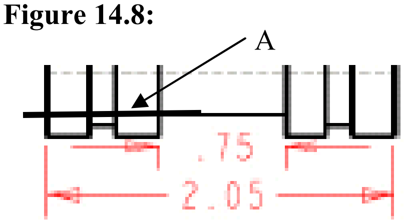
Figure 14.7:



14.5.4.3 Shear Through Sealing Geometry

The same equation can be used to determine the maximum shear stress that the sealing geometry in the stem holder will see. This sealing geometry is shown in the CAD drawing below. The maximum shear stress would occur when the pressure in the middle of the geometry is 150 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 0.8345 in^2. The maximum shear force is equal to 125.2 lbf. The total shear area is equal to 4.08 in^2. The maximum shear stress that this part will see is 30.66 psi, and will act through line A in Figure 14.8. This allows for a factor of safety of about 195 for this mode of failure.

Figure 14.8:



14.5.4.4 Tension at Sealing Geometry Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This geometry can be modeled as a cantilever beam. The force in this section is 125.2 lbs, and is located halfway out on this cantilever beam (0.0625 inches). The moment that is seen at the wall is then equal to 7.825 lb-in (M). The moment of inertia of this beam is equal to 0.1437 in^4 (I). In equation 14.5, y is equal to half of the height, or 0.325 in. The maximum stress in this beam is therefore 17.7 psi. With PVC being the material, the factor of safety against this kind of failure is 339.

14.5.5 Poppet Valve Assembly

14.5.5.1 Valve Stem

Valve stem failure modes:

4. Shear through threading
5. Tensile yield through stem area
6. Rupture due to internal pressure

14.5.5.1.1 Shear Through Threading

The analysis of shear failure on the threads of the valve stem was performed in the same manner as that for the stem holder. Here, the equation for the shear area of an external thread was used (equation 14.2 in section 14.5.1.2). The shear area, A_s, was found to be 0.799 in^2 using the parameters in Table 14.4. Again, using a load of 403 lbf as in the case of the stem holder (since the load will be transferred through the valve stem), the shear stress was found to be 504psi. The lower bound yield strength of Nylon 6/6 was given in the materials database [19] as 8010 psi, which results in a factor of safety of 15.88 for the valve stem.

Table 14.4: Parameters used to find shear area of the external valve stem thread

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	1 in
$K_n$max	Maximum minor diameter of internal thread	0.457 in
$E_s$min	Minimum pitch diameter of external thread	0.4619 in

14.5.5.1.2 Tensile Yield Through Stem Area

Equation 14.8 can be used to determine the maximum stress that the stem will see in tension. The maximum tensile force is 403 lbs (F_{MAX}). The outer and inner diameters are taken at the thinnest point of the stem, which will be where the threading is created on the outside of the stock material. The stock material has an outer diameter (D_O) of ½ inch, but where the threading is formed, the outer diameter is 0.419 inch. This diameter is assuming worst case geometry (minimum material condition). The stock material has an inner diameter (D_I) of ¼ inch to give a total tensile stress area of 0.0885 in^2. The maximum tensile stress that this part will see is 4552 psi. The lower bound yield strength of the Nylon 6/6 material that will be used for the stem is 8010 psi (found from the materials database [19]).

$$S_Y \geq \frac{F_{MAX}}{A_{FORCE}} = \frac{F_{MAX}}{\left(\frac{1}{4}\right)\pi\left(D_O^2 - D_I^2\right)} \qquad \text{[Eq. 14.8]}$$

14.5.5.1.3 Rupture Due to Internal Pressure

Equation 14.3 in section 14.5.1.3 can be used to determine the maximum stress that will be seen in the wall of the stem if 150 psi is internal to the stem and 0 psi is external to the stem. According to this equation, the maximum stress is equal to 71.24 psi. Adding this stress to the stress found in the previous section results in a stress of 4623 psi maximum. This allows for a factor of safety of 1.73 for this mode of failure. It is worth mentioning that the 403 lbf loading is for a working fluid pressure of 150psi. There is no plan for running this high of pressure during any experimental testing of the valve.

14.5.5.2 Valve Head

Stem head failure modes:

4. Shear through threading
5. Shear through middle
6. Tension through section due to moments
7. Tensile yield through area

14.5.5.2.1 Shear Through Threading

Analysis was done to ensure that the internal threads of the stem head would not fail by shear. The threading will be the ½-20UNF form and will be used to fasten the external threads of the valve stem. The reasons for this thread are the same as described in the stem holder section. The threading profile and engagement length are the same as in the stem holder, so the failure calculations are the same. Assuming a maximum axial load on the stem holder threads of 403 lbf, the resulting shear stress on the threads is 373 psi. The lower bound yield strength for PVC Type 1 was found to be 6000 psi from the materials database [19], so the factor of safety for this kind of failure is about 16.

14.5.5.2.2 Shear Through Section

Equation 14.3 in section 14.5.2.1 can be used to determine the maximum stress that the stem head will see due to shear in the middle section. The maximum shear force that the stem will see through the middle is 403 lbs (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 14.9). The shear area in this geometry is equal to about 0.83 in^2. The maximum shear stress that this part will see is 486.275 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi (according to CES Edupack 2008). This allows for a factor of safety of about 12 for this mode of failure.

Figure 14.9:

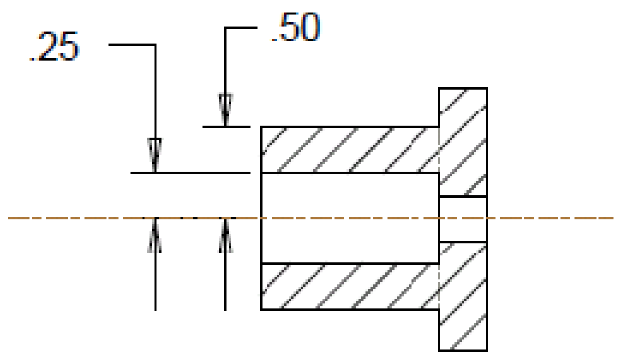
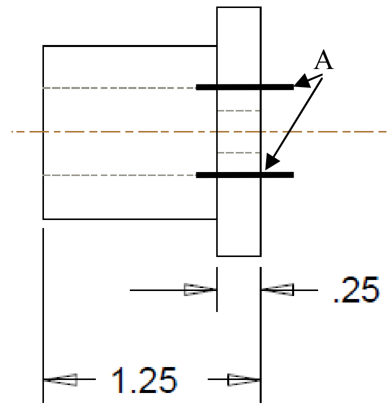


Figure 14.10:



14.5.5.2.3 Tension Through Section Due to Moments

Equation 14.5 in section 14.5.3.3 can be used to find the maximum tension on the valve stem head due to a moment. The force of 403 lbs is applied through the stem head at the outside, which means that a tensile reaction will be created on the inside of the threaded area. This reaction can be found by modeling this as a cantilever beam. The force in this section is 403 lbs, and is located on the end of this cantilever beam (0.22 inches). The moment that is seen at the wall is then equal to 88.66 lb-in (*M*). The moment of inertia of this beam is equal to 0.2618 in^4 (*I*). In the equation, y is equal to half of the height, or 0.125 in. The maximum stress in this beam is therefore 42.3 psi.

14.5.5.2.4 Tension Yield Through Area

The smallest area at which tensile yield would occur is 0.589 in^2. Similarly to section 14.5.5.1.2, with a maximum tensile force of 403 lbs, the stress in this section is equal to 684.2 psi. Adding this stress to that found in section 6.3.3, the total tensile stress seen in this part is equal to 726.5 psi. With PVC being the material, the factor of safety against this kind of failure is about 8.3.

14.6 Internal "Plumbing"

In order to actuate the valve hydraulically, ports have to be drilled through the housing walls to access the correct volumes and apply the correct pressurized fluid. In the final design, these orifice sizes will be critical in ensuring that the pressure equalization occurs in 50-100 milliseconds, but in the prototype, the timing is not critical. In the prototype, unlike the final design, extensive plumbing will not be necessary. Instead, the housing is thick enough to allow for ports to be drilled in order to access the areas on which pressures need to be applied. The prototype description section will describe this in more detail.

14.7 Connections and Plug-ins

14.7.1 System Fitment

A 4-bolt code 62 flange connection does not need to be included in the prototype. Instead of this kind of cap, a different style end cap is designed. This end cap must have standard threaded connections to connect to the design expo setup. See the validation setup for more information. The items used in the design expo setup are rated for 150 psi at minimum.

14.8 Recent Design Changes

Since our last report, there have been few design changes to our prototype. The most important design change is that, in our original plans, the valve housing was to be made entirely of acrylic. The choice of acrylic was to allow for direct viewing of the movement of the internal parts of the valve during testing. In a conversation with our design project sponsor, Andrew Moskalik, an issue with threading the acrylic into the accumulator during testing arose. Acrylic was found to be too brittle of a material to thread and torque into a steel accumulator entrance. In response, the housing was changed to aluminum, which corrected this issue.

In addition to this issue, in the last report, the valve head was to be made out of one piece of material. After investigating our material choices further, we came to the conclusion that a one piece valve head was impossible for us to complete due to the long hole through the center of the stem. Instead of one piece, the valve stem was split into two parts, which are detailed above. This separation allowed for purchase of a pre-formed through hole tubing that could be used for the main part of the valve stem. The head of the valve would then be able to be manufactured from a separate piece of PVC and attached with threading.

15 PROTOTYPE FABRICATION PLAN

15.1 Comparison with Final Design

The main difference between the prototype and the final valve is the use of vastly different materials. The prototype will be testing at a fraction of the maximum pressure of the final design, thus the final valve will be made from much stronger materials than the prototype. Specifically, the prototype will be testing with water at an absolute maximum pressure of 150 psi (starting as low as 30 psi) and will be made of a variety of different plastics. The prototype's valve housing will be made from aluminum while the power piston, stem holder, and valve head will be made from PVC type 1. The valve stem will be Nylon 6/6 tubing.

The prototype and final design also possess slightly different geometries. For one, the prototype is not designed to integrate with a 4-bolt code 62 flange, like the final design is. Instead, the prototype will interface with a hose connected to a threaded nylon fitting. Also, because of manufacturing and purchasing restrictions, the prototype's valve stem has a larger inner diameter. The larger inner diameter will still allow for functional fluid flow through the stem as well as successful bladder interaction.

15.2 Manufacturing Concerns

The accuracy and surface roughness of all finishing cuts are a concern (from a manufacturability standpoint). Correct operation necessitates small tolerances on all geometries that constrain any valve movement. The ability to produce parts that meet this requirement may require multiple iterations of a manufacturing step. This is something that cannot be afforded (both monetarily and from a time constraint), and attempts will be made to avoid this.

Threading operations on the valve housing, valve stem, stem holder, and stem head are all concerns from a manufacturing standpoint.

15.3 Bill of Materials for Prototype Valve

The bill of materials for the prototype design is shown in Appendix N.

15.4 450 Shop

Manufacturing considerations have a close relationship to overall valve cost. The more complicated manufacturing operations that are required, the higher the final cost of the valve will be. Because of this, the HPSOV was designed from the start to incorporate overall simplicity, as well as manufacturing simplicity. It was determined that any manufacturing operation that could not be performed with resources provided by the ME450 shops would be too costly for production. Therefore, it is possible for the prototype valve geometry to be manufactured using ME450 resources.

In the final design, we anticipate that the valve housing and valve stem may not be able to be manufactured easily. The housing may be complicated to machine because of the multiple ports that have to be included in the housing. These ports include a fill port, a port for low pressure access, and a port for the solenoid valve.

15.5 Tolerance considerations

To ensure long valve life, there are a number of critical tolerances on the valve that must be maintained. Particularly, these include all sliding interfaces throughout the valve. If geometrical tolerances are too excessive, wear rates will increase dramatically and long life will not be achieved. Therefore, the final valve design drawings include specific tolerances on the necessary components. For the prototype, however, two factors reduce the importance of accurate tolerance. These are the relatively short component lifetimes required, and the relatively low operating pressures.

15.6 Prototype Manufacturing Instructions

This section includes detailed instruction on how to manufacture all of the parts we plan to machine in-house in order to build our prototype valve. All valve components will be machined from stock materials. Since they are all axi-symmetric pieces, all machining will be done on a manual lathe. Some cutting with a band saw will also be needed, as well as drilling with a drill press or mill.

15.6.1 Prototype Valve Housing

The valve housing will be created from the same stock aluminum cylinder as the end cap below. The end cap will be created first, and then cut away from the remainder of the stock using an automated band saw operated by either Marve or Bob in the ME 450 machine shop. After mounting on the lathe, we will use a 1” diameter drill bit to drill a center hole to a depth of 5.6” through the stock. Then we will use a boring tool to create the internal geometries shown in the drawings above from the end-cap end of the housing, up to where the inside diameter drops from 2.75” to 1.25”. None of the 1.25” internal geometry will be created from this end. Next we will take off a thin outer layer of aluminum all along the stock almost up to the chuck. This will make the housing both shiny, and help us re-align the stock when we insert it into the chuck in the other direction.

We will remove the work piece, and insert what is now the end-cap end of the valve housing work piece into the chuck, and re-mount it. We will drill the 1.25” center hole in the work piece, and then lathe down the exterior to create the neck of the housing. We will then have Bob or Marve help us create the necessary threads on the valve housing neck. The remaining holes will be drilled and tapped using a drill press and hand taps according to the engineering drawings above. All of the speeds and feeds used for machining the valve housing geometry will be determined by consulting either Bob or Marv in the ME450 machine shop before beginning machining (see Table 15.1 below for this information).

Table 15.1:

Process	Material	Machine	Cutting Speed (RPM)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Turning Tools
Internal Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Boring Bar
Separating Material	Aluminum	Band Saw	-	-	-	Band Saw
Center Hole	Aluminum	Drill Press	165-240		.004-.005	1” Bit, 1.25” Bit

15.6.2 Prototype End Cap

The valve housing end cap will be created from a 5” diameter by 2’ long aluminum cylinder. The external geometry will be generated using a turning tool on the lathe. The hole through the center will be drilled on the lathe as well. The bolt holes will be drilled as clearance holes on the drill press and the back end will be tapped for pipe fittings by hand using a tap and accompanying handle. The dimensions, drill bit sizes, and hole sizes can be found on the engineering drawing. The speeds and feeds are shown below in Table 15.2.

Table 15.2:

Process	Material	Machine	Cutting Speed (RPM)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Turning Tools
Internal Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Boring Bar
Separating Material	Aluminum	Band Saw	-	-	-	Band Saw
Center Hole	Aluminum	Drill Press	165-240		.004-.005	1/2” Bit

15.6.3 Prototype Valve Power Piston

The power piston is to be created largely on a lathe from PVC stock. The PVC is the remainder of a cylindrical 3.25” diameter by 2’ length rod purchased from McMaster Carr that will already be used to create the valve stem head and valve stem holder pieces of our prototype valve design. The external geometry will be machined using the tooling in conjunction with the speed and feed information found in Table 5.3 below. The equations below Table 15.3 will be used to find the RPM settings for each process. These are listed in Table 15.4. After the external geometry is created, the internal geometry will be removed using a boring tool. After both of these steps are completed, a drill press will be used to generate the eight 1/8” diameter holes throughout the piston.

Table 15.3:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	Turning Tools
Internal Geometry	PVC	Lathe	650-1600	.6	.002-.020	Boring Bar
Separating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8” Bit

Lathe RPM equation:

$$RPM = \frac{Cutting\ Speed \cdot 4}{Diameter\ of\ Workpiece}$$

Drill Press RPM equation:

$$RPM = \frac{Cutting\ Speed\ \cdot 4}{Diameter\ of\ Drill\ Bit}$$

Table 15.4:

Process	Material	Machine	Cutting Speed (RPM)	Feed (ft/min)	Tool
External Geometry	PVC	Lathe	850-2100	.002-.020	Turning Tools
Internal Geometry	PVC	Lathe	650-1600	.002-.020	Boring Bar
Separating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8” Bit

15.6.4 Prototype Valve Stem Holder

The valve stem holder will be made in the ME Machine Shop. The valve stem holder will be created from a 3.25” diameter, 2’ long piece of stock PVC rod that has been purchased from McMaster Carr. Only a small section of the rod will be used for the valve stem holder, the remainder will be used to make other pieces of our prototype. The stock PVC will be mounted on a lathe. The external geometry will be created using a turning tool. We plan to ask Bob Curry for advice on tool selection. Next we will use a band saw to cut the newly created piece from the remaining PVC rod. A drill press will be used to create the center holes and side hole using appropriate drill bits for each hole. The through holes in the valve stem holder will either be created on a mill, or altered to be 3 round holes and created on a drill press. The speeds, feeds, and tooling for these processes will be listed in the table below. Cutting speed and feed rates will be looked up in the Machinery Handbook in the machine shop before any machining takes place. The equations below Table 15.5 will be used to find the RPM settings for each process and fill in Table 15.6. Once all of these operations are completed, the large center hole will be tapped with a 1/2” x 13 TPI tap. This operation will be done by hand.

Mill RPM equation:

$$RPM = \frac{4 \cdot Cutting\ Speed}{Diameter\ of\ Mill\ Bit}$$

Mill feed rate equation

$$Feed = (ChipLoad) \cdot (numer\ of\ flutes) \cdot (Spindle\ Speed)$$

Table 15.5:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	
Separating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8” Bit
Center Large Hole	PVC	Drill Press	650-1600			27/64” Bit
Side Port Hole	PVC	Drill Press	650-1600			1/8” Bit
a) Through Holes	PVC	Mill	650-1600			3/8” End Mill
b) Through Holes	PVC	Drill Press	650-1600			3/8” Bit

Table 15.6:

Process	Material	Machine	Cutting Speed (rpm)	Feed	Tool
External Geometry	PVC	Lathe	850-2100		
Separating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8” Bit
Center Large Hole	PVC	Drill Press	6100-15000		27/64” Bit
Side Port Hole	PVC	Drill Press	20800-51200		1/8” Bit
a) Through Holes	PVC	Mill	6900-17000		3/8” End Mill
b) Through Holes	PVC	Drill Press	6900-17000		3/8” Bit

15.6.5 Poppet Valve Assembly

15.6.5.1 Valve Stem

The valve stem will be made in the ME Machine Shop. The valve stem is to be created from a section of 5’ nylon 6/6 tube (1/2” OD by 1/4” ID) purchased from McMaster Carr. Only a small section of the tube will be used for the valve stem head, the remainder will be set aside in case we make a mistake and need to start over. The stock nylon tube will first have to be cut to a length of 6.5” using the band saw. Once that is done, both ends of the tube will be tapped to create external threads using a 1/2” x 20 UNF tap. These operations will be done by hand. When assembling the valve, a portion of the valve stem will need to be cut off to make it the correct length.

15.6.5.2 Valve Head

The valve stem head will be made in the ME Machine Shop. The valve stem head will be created from what remains of the 3” diameter, 2’ long piece of stock PVC rod that was purchased from McMaster Carr and used to make the valve stem holder. Only a small section of the rod will be used for the valve stem head, the remainder will be used to make the power piston. The stock PVC will be mounted on a lathe. The external geometry will be created using a turning tool. Next we will use a band saw to cut the newly created piece from the remaining PVC rod. A drill press will be used to create the center hole using the appropriate 1/8” diameter drill bit, then a larger hole using a 29/64” drill bit that is appropriate for the threading. The speeds, feeds, and tooling for these processes will be listed in the table below. Cutting speed and feed rates will be looked up in the Machinery Handbook in the machine shop to verify before any machining takes place. The equations below Table 5.5.1 will be used to find the RPM settings for each process and fill in Table 5.5.2. Once all of these operations are completed, the protrusion will be tapped to create internal threads using a 1/2” x 20 UNF tap. This operation will be done by hand.

Table 5.7:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	
Separating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8” Bit, 29/64” Bit

Table 5.8:

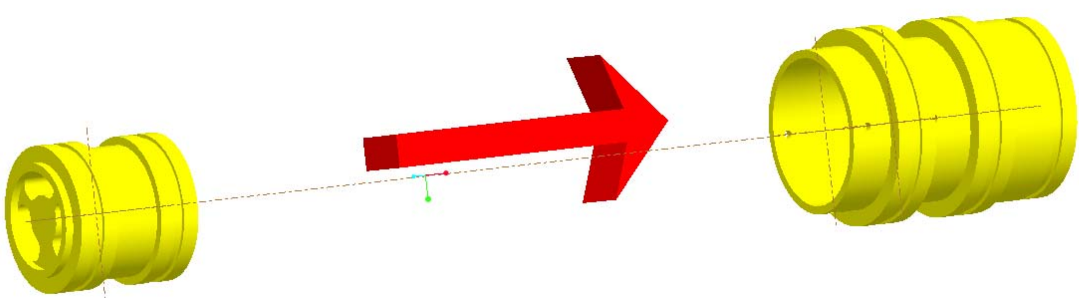
Process	Material	Machine	Cutting Speed (rpm)	Feed	Tool
External Geometry	PVC	Lathe	850-2100		
Separating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8” Bit

15.7 Valve Assembly Instructions

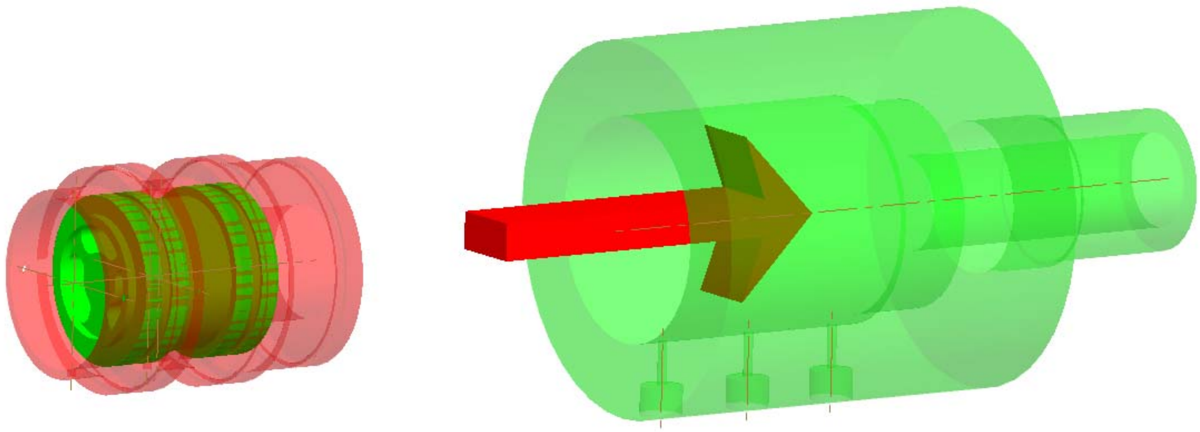
Following is a list of simple instructions on how to assemble the prototype valve after all of the parts have been manufactured following the instructions above. You should have before you a valve housing, valve stem, valve stem holder, power piston, end cap, and spring as defined above.

Step 1: Place the appropriate o-ring into each of the 3 o-ring grooves on the power piston, two on the valve stem retaining piston, one on the end plug, and the remaining o-ring on the outside of the valve housing at the base of the neck so that it will seal the housing against the accumulator when the valve housing is screwed into the accumulator.

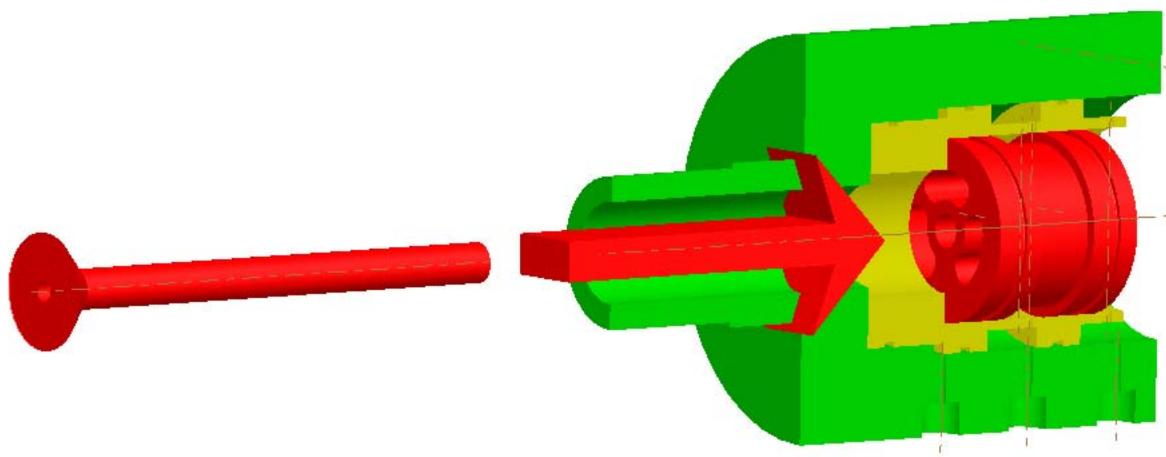
Step 2: Grease the inside of the valve housing and power piston to reduce friction and ensure proper valve operation at low pressures.
Step 3: Insert valve stem retaining cylinder into back of power piston as shown below:



Step 4: Insert Power Piston into Valve housing show below:



Step 5: Wrap the threads of the valve stem with Teflon tape to ensure a water tight seal.
Step 6: Thread the valve stem into the valve stem retaining piston as shown below:



Step 7: Hold the valve assembly vertically so that the head of the valve stem is facing the ground. Next, insert the spring into what is now the top of the valve housing. Make sure that the spring is centered on the innermost piston (valve stem holder piston) so that the cylindrical protrusion on the piston adds lateral support to the spring.

Step 8: Attach the end cap onto the back of the valve housing with 8 ¼-20 bolts and appropriate washers.

16 VALIDATION PLAN

In order to validate that our final design meets the customer specifications, validation testing will be conducted on a prototype. This will be done using a test apparatus built to simulate (at a lower pressure) the various pressure conditions that will be experienced on the vehicle. The test apparatus will include two accumulators, as well as an unpressurized fluid reservoir. This setup is shown in figure 16.1. One reservoir will simulate the high pressure accumulator, and another reservoir will simulate the line. Each accumulator will be pressurized using a manually operated air pump connected to the bladders of the accumulators. The unpressurized fluid reservoir will simulate the low pressure accumulator, and will be used to catch any discharged water from the system. For all testing, the working fluid will be water.

To simulate a specific function (for example the leak-in function), pressures in the simulated “Line” and high pressure accumulators will be set according to what would be seen in the vehicle under the same situation (at a lower magnitude). In this case, the “Line” accumulator would be pressurized higher than the high pressure accumulator. Normally, an electronic signal from the vehicle would be sent to the solenoid valve. The solenoid valve would the adjust pressures to the actuator piston to control the position of the main poppet. For our testing, this function will be entirely replaced by manual control. Instead of the solenoid controlling where pressure is directed, ball valves will be used to direct pressure to the correct areas. This will be done to reduce the cost and complexity of the test apparatus, as the control method will not affect the primary functionality of the valve. Detail of the control valve plumbing is shown in figure 16.2. There is a unique positioning of each ball valve depending on which function is being tested. Table 16.1 gives a listing of all the ball valve positions and the respective function. For more detailed instructions on how the apparatus will be operated and monitored to prove final design function, see Appendix O.

Figure 16.1: Test Apparatus

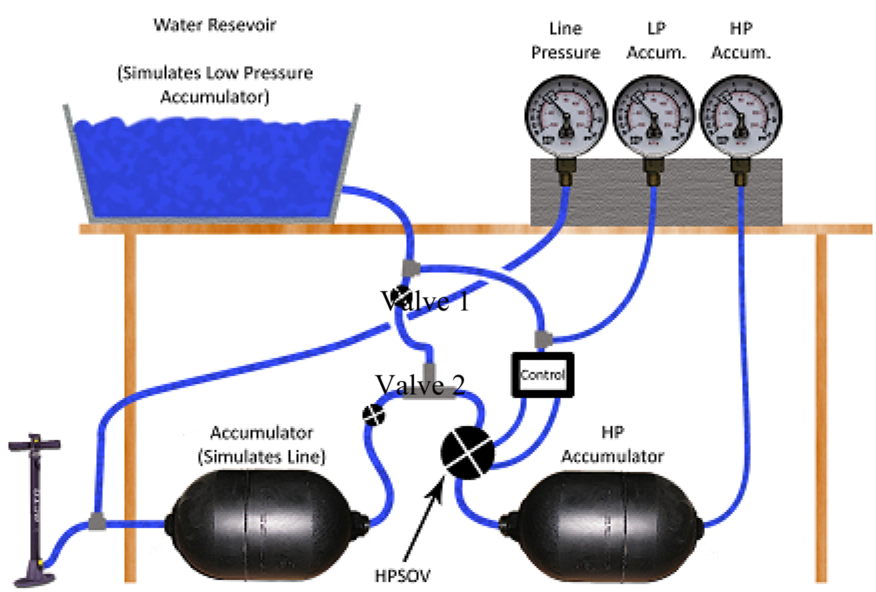


Figure 16.2: Manual control assembly

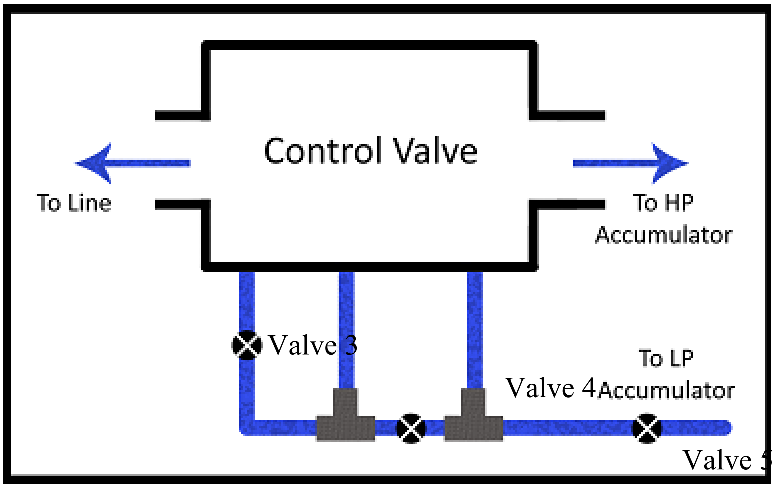


Table 16.1: Positioning of valves and the highest pressure accumulator for all validation tests (valve numbers are indicated on figures 16.1 and 16.2)

	Valve 1	Valve 2	Valve 3	Valve 4	Valve 5	Highest Pressure
Command Close	Open	Close	Close	Open	Close	HPA
Command Open - Into HPA	Close	Open	Open	Close	Open	Line
Command Open - Out of HPA	Open	Close	Open	Close	Open	HPA
Leak-in test	Close	Open	Close	Open	Close	Line
Bladder interaction test	Open	Close	Open	Close	Open	HPA

As mentioned above, the purpose of the test apparatus is to specifically validate several of the customer requirements. These are listed in Table 16.2, along with a number of requirements that will not be validated during our validation testing. While some requirements will not be specifically tested, the testing seeks to prove out the general valve concept. The level of testing that will be performed is such that the major issues concerning whether or not the valve can work in the application are illustrated. The goal is that if a problem is found with the remainder of the issues, the level of these operations is such that solutions can be designed without affecting the remainder of the valve operation or overall layout.

Table 16.2: Prototype validation will address specific customer requirements

Prototype Will Validate	Prototype Will NOT Validate
Open Function	Velocity Fuse
Closed Function	Open/Close Reaction Times
One Way Flow Function	Corrosion/Impact Resistance
Pressure Equalization	Oil/Ambient Temperature Conditions
Bladder Interaction	Lifetime Cycling
Size and Seal Integration	Pressure drop

In order to further validate the correct features of the valve operations, the positioning of the stem holder, and thus the stem head (open, closed) must be seen. In order to accomplish this, a steel rod is connected to the back of the stem holder inside a hole with epoxy. This rod extends through the hole in the end cap and into the clear tubing that is used throughout the system. The position of the end of this rod will therefore be useful in determining the position of the valve. In addition, a flow meter will be attached to the back end of the valve to show that the flow direction that is desired is achieved in each situation.

Customer testing can be done to validate the features that were not proven in the testing phase. These features will now be discussed.

The valve was designed to have an integrated velocity fuse. The velocity fuse will function using the same spring that controls the bladder interaction. When the fluid is flowing through the poppet valve, there will be a pressure drop on the back side of the valve head due to the increased fluid velocity. This will tend to close the valve, only to be reacted by the bladder interaction control spring. If this control spring is properly sized and the flow rate is exceeded, this spring will allow the valve to move to the closed position, thus cutting off the fluid flow and accomplishing the fuse function. This velocity fuse can meet all design requirements, as demonstrated by the current EPA HPSOV design which uses the same concept.

Included in the customer requirements were open/close times for the valve movements. The specifications for these times will not be validated in prototype testing. However, there are two reasons for having high confidence in the ability of the final design to meet these specifications. First, the opening process only occurs when the line equals the pressure of the high pressure accumulator. Without this, the force imbalance cannot be overcome. The pressure equalization function accomplishes this pressure balance. Furthermore, the rate of pressure equalization can be controlled by passage diameters within the valve housing. If it is found that opening time is too great, the orifice diameters can be increased to allow the pressure equalization, and thus the valve opening, to occur faster. The opposite can be done if the opening time is too quick. The second reason for having high confidence in the prototype design is the fact that the current EPA design also operates with hydraulic actuation and the time requirements have been met. Since the same type of actuation is used, the actuation times will likely be comparable.

Special testing procedures and equipment will be needed for corrosion and impact resistance, and thus will not be validated in our testing. Alternatively, these requirements may be demonstrated from on-highway operation of the valve, followed by subsequent inspection after a specified operating time.

A requirement of the HPSOV is that it operates successfully throughout the ambient temperature and oil temperature ranges that will be seen in service. Because the prototype will be made out of materials that differ from the final design, the prototype will not be tested to see if it meets these requirements. Should the design be adopted, these requirements can be tested later and changes can be made to meet them that will not affect the overall operation of the valve (e.g. small clearance or geometry changes that will keep stresses from thermal expansion to acceptable levels).

Because the prototype is made from different materials than the final valve, and because experimental testing is only intended to prove functionality of the general valve design, no lifetime testing will be conducted. Again, this can be done in final valve qualifications prior to production.

Certain pressures will be measured during valve testing. These will primarily be to aid the accumulator pressurization process, and will not be fine enough to accurately measure pressure drop through various positions in the fluid flow path through the valve. Because of this, no attempt will be made to characterize the pressure drop of the prototype valve in comparison to the EPA design. Additionally, the prototype geometry is slightly modified from that of the final design in order to facilitate manufacturing. Correlating pressure drop for this geometry would be of no use for comparisons with the current EPA valve, as it would not be representative of the full scale conditions.

17 VALIADTION RESULTS

The prototype HPSOV was validated using experimental testing designed to simulate each of the varying operating modes that the valve would see in the full scale application. In total, six validation tests were to be conducted on the prototype (for a listing of these, see “Prototype Validation Plan”). Having passed all these, the valve actuation concept used in the HPSOV prototype could be adopted with confidence for future production.

The valve successfully completed three of the six experimental tests. These included the one-way flow function, bladder interaction, and size and seal integration. Of the remaining three, two were unable to be tested and one was hindered by valve head sealing issues.

One-way flow functionality was achieved by pressurizing the line above the high pressure accumulator. Once the pressure difference was established, a control valve was opened and fluid was observed to flow into the high pressure accumulator even though the valve was being signaled to close. When fluid was set up to flow out of the high pressure accumulator, the valve position was observed to move into the closed position toward the end of fluid flow. This was a result of the bladder pressing against the valve head, thus verifying the bladder interaction function.

Both size and seal integration were also successfully demonstrated. The prototype valve was designed and built to full scale and was tested with an accumulator having a full size outlet port. The prototype attached to the accumulator and there were no fitment issues that arose. Because none of the geometries would change in the full scale product, all components are geometrically compatible with the current equipment.

Seal integration was verified by the successful external sealing of the valve housing. No external leaks from the valve housing were experienced during any of the experimental testing. The internal seals of both the piston and valve stem holder performed well as designed, but were later modified in an attempt to reduce the force necessary to move these two components (to promote movement by the low test pressures). The modification reduced the seal integrity, but this was an expected consequence. In the full scale design the fluid pressures available would likely eliminate the necessity for such a modification and the seals would be allowed to perform to their full potential.

The remaining three tests were all affected by valve head seal performance. The valve head seal forms the main fluid seal between the high pressure accumulator and the rest of the hydraulic system. In the prototype, this seal did not perform as well as necessary. As a result, the command close operation could not be verified to correctly function since even in the closed position the valve was observed to leak fluid into the line. Upon commanding the valve closed and relieving line pressure, flow would continue to escape from the high pressure accumulator. It is expected that the leak is due to the faulty valve head seal, however it is possible the leak could be due to incorrect movement of the piston (see “Discussion” for a test procedure designed to verify this is not the case).

Since the system could not be held in the closed position with zero flow, the necessary initial conditions for valve opening could not be maintained. Therefore the valve opening commands could not be tested for correct operation. Valve opening depends on achieving pressure equalization, however since valve opening could not be tested the pressure equalization function was also unable to be tested. These tests were the last of those planned for the HPSOV prototype validation.

18 DISCUSSION

Although only two of the five tests were completed successfully, the valve concept could still be proven with only a minor investment of time and design modifications. The only change necessary to continue validation would be to improving the valve head seal. Improving the seal would eliminate the uncertainty of which component is causing the leak under the closed command. If the leak remains after seal integrity has been verified, the validation testing can move on to determine any other causes. Should there still be issues, the next modifications should be to ensure the piston and valve stem holder seals are functioning properly while allowing reasonable actuating forces. It should be verified that both the piston and valve stem are moving when being commanded to do so. These simple checks and modifications should be done to ensure the necessary functionality of these components which would then allow all validation testing to continue in order to prove the valve concept.

It should be noted that the prototype was designed and built to full scale, even though the operating pressures were of lower magnitude. This means that the actuating force available to move the piston and stem holder are far greater in the full scale application than what was available in experimental testing. Because of this, any difficulty moving the piston and stem holder is likely due to off-design operating conditions and not necessarily to inadequate design that requires modification. Friction forces from seals that cannot be overcome with the testing pressures would likely be easily overcome at full scale pressure.

With regard to proper sealing, should the full scale product exhibit problems at full pressure, this is an area that can be addressed and solved independently of the valve actuation method or valve concept in general. Sealing is an area that can draw upon the expertise of the design firm contracted. The valve allows for seal modification and seal groove modification without necessitating changes to the general valve configuration.

19 RECOMMENDATIONS

At the advent of this project, a list of design requirements was given by the sponsor. This included specific and quantitative requirements as well as more functional and qualitative ones. This original list is given in Appendix A. Not all of these requirements were validated by the prototype design and testing. The untested requirements were addressed in the unproduced final design.

19.1 Continuation of current design

We believe, that with correct manufacturing and planning, our final design satisfies the given customer requirements. If research and implementation of this design were to continue, we are recommending a few areas that should be examined closely.

19.1.1 Sealing Geometries

When examining production processes for this valve design, special consideration needs to be given to sealing geometries. During the one-off production of our valve prototype, manual machining techniques were used and subsequently created tolerance issues. Specifically, we believed the sealing surfaces between the valve stem head and the valve housing neck inside of the accumulator were not good enough to create a positive seal. This was caused by threading error between the valve stem and valve stem holder as well as an inefficient sealing geometry between the head and housing neck. For simplicity's sake, the prototype was designed and produced with a flat sealing surface as opposed to a more conical surface. The current EPA designed valve has a succesful poppet style valve head with positive sealing and we believe this design can be used in our final design.

We also believe that in the prototype there were sealing issues and leaks between the sealing surfaces on the inner diameter of the valve housing. This was the result of trial and error. During testing it became apparent that the friction from the seals was too great for our pressure differences to overcome. To reduce friction we increased the clearances between the pistons and their housings. Unfortunately this negatively affected our ability to seal the pistons. We recommend that the sealing geometries and tolerance be inspected and altered in order to produce positive seals and simultaneously ensure that the pistons will still be able to move.

19.1.2 Velocity fuse and flow rates

The prototype valve was not manufactured to work at full operating conditions and, as such, was not designed to validate the velocity fuse or flow rate requirements. These features are important, though, and are included in or could be worked into our final valve design. The current valve in use by the EPA accomplishes the velocity fuse feature with a spring that limits the flow rate by compressing due to force from the flowing fluid. The spring rate was determined experimentally, so we believe that a spring with the same spring rate as the current velocity fuse spring could be used in our final design in place of the spring currently employed for bladder interaction. This spring has a spring rate of 75 lb/in.

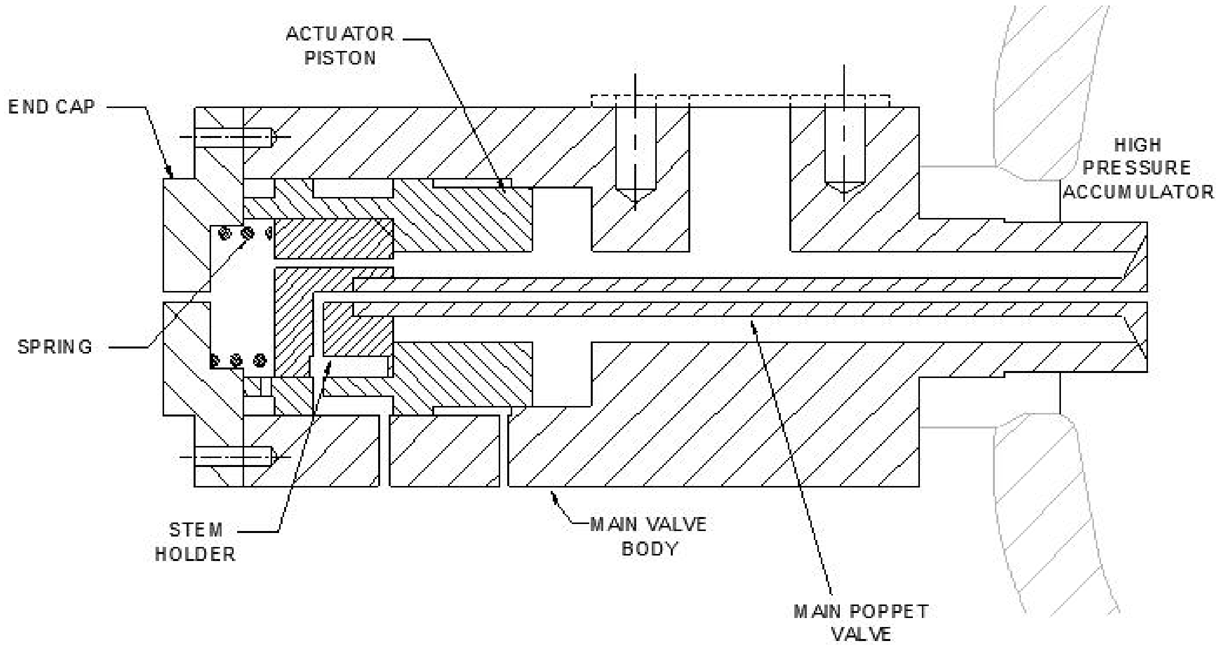
19.2 Evaluation of previous/other designs

The design for this valve is open ended, creating a number of different designs that could satisfy all of the customer requirements. During the concept generation process many designs were created with successful functions or features. We believe a few of these designs are worthy of further examination and research. These designs were not pursued by our team because of time constraints, as the semester is only so long.

19.2.1 Combination design

During the process of narrowing down our two alpha designs to one final alpha design, a suggestion was made to combine the best parts and functions of each of the alpha designs into one compilation design. This design was explored, but it was determined that in order for this design to be successful, a full evaluation would need to be started from scratch. The length of the semester and required due dates simply would not allow time for this and further analysis of the combination design was abandoned. The advantages and disadvantages of this design are discussed in Section 10.3. While this design was not adopted by our team, we recommend further analysis as we believe it has the potential to provide better system integration along with simplicity of manufacture and full functionality.

Figure 19.1: Combination design, cross-sectional view

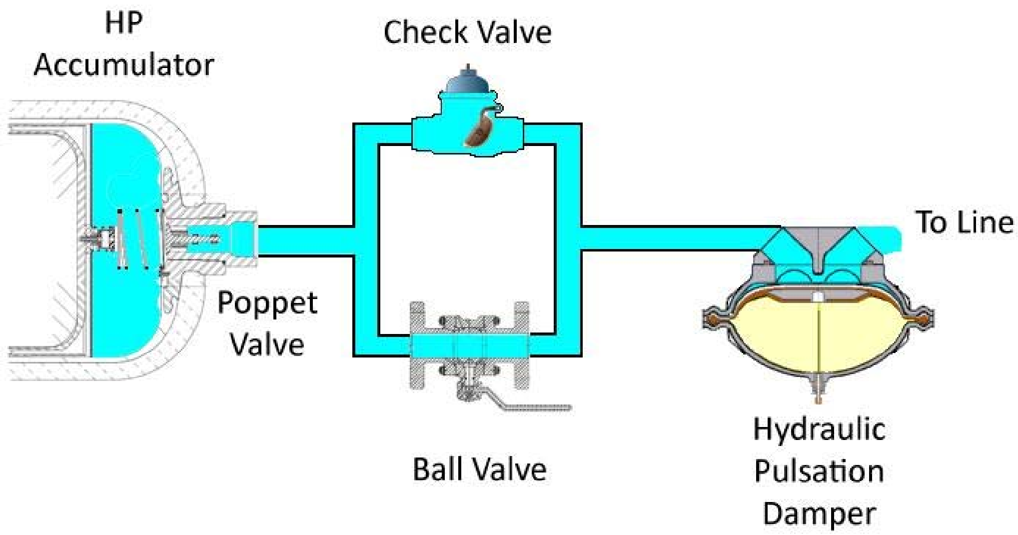


19.2.2 "Off the shelf" design

Another design that we believe would satisfy all of the customer requirements involves a number of different "off the shelf" valves and parts connected in series and in parallel. This design was not pursued because we felt it would be beneficial, for maintenance and complexity issues, to create a single element that would satisfy all requirements. In addition, a single valve solution would be more cost effective in the long term. In the near term, however, a multiple valve solution may exist, if 7000 psi rated parts were obtained for each component.

In order to accomplish open and closed functionalities, a ball valve with electronic actuation can be used. Using the high pressure in the tank to actuate this ball valve could be beneficial. In parallel with this ball valve would be a one-way flow valve that would allow leak in functionality of the valve. Attached to the accumulator would be a simple poppet valve (like the one that is included with the accumulators). This spring in this poppet valve would have to be replaced with one that performed the velocity fuse function under the required flow rates (should be about 75 lb/in.). In order to accomplish bladder safety, an emergency electronic shutoff should be provided when the pressure in the tank becomes less than 2200 psi. Finally, downstream of the parallel connection would be a hydraulic pulsation damper, which would allow the system to remain relatively quiet during fluid hammer situations. Figure 19.2 shows a schematic of what this kind of system would look like.

Figure 19.2: Schematic of "Off-the-Shelf' design



20 CONCLUSIONS

The EPA requires a high pressure shutoff valve for use on the high pressure accumulator of the hydraulic hybrid vehicles they are developing. Due to the complexity and high cost of the current EPA valve, a new valve design is required that integrates all the existing features into a safe, low cost valve of reduced complexity. The required valve must be compatible with the current system and must satisfy a variety of functional, safety, reliability, and cost requirements. Concept generation and selection processes have been implemented, and two "alpha" designs were selected. These designs were then analyzed, screened, and combined in an effort to create one final design. The resulting design was a slight modification on one of the alpha designs, but performs better than either. The final design is a straight through valve design with hydraulic actuation and spring assisted bladder interaction. After the final valve was designed and analyzed using engineering methods and simple computer simulations, a prototype was developed. The prototype was modeled using CAD programs and all important geometries were defined. Material analysis was then performed using simple strength calculations and CES [19]. This yielded a variety of plastics that would work with our prototype. The final materials were chosen and work began on designing the prototype's testing conditions and validation plan. The prototype was made to demonstrate the four basic functions of the HPSOV; open, closed, leak-only, and pressure equalization. The resulting test data was analyzed, and some additional modifications to the prototype can be undertaken to finish validation testing. Further engineering study was be completed for the final valve design and its properties. In addition to our final design, we recommended that EPA look at the results of our concept generation and selection processes, as we believe that other successful designs can be researched from our ideas. It is believed that hydraulic hybrid vehicles have large potential in the automotive industry and the team is excited to have worked on this project with its relevance to the success of hydraulic hybrid technology.

21 PROJECT TIMELINE

Most of the time we have spent working on this project has been dedicated to developing and refining multiple designs for the HPSOV. We began with a handful of designs and, using an itemized list of necessary functions as well as group discussion, we narrowed our choices down to two different alpha designs. These two designs both scored very well when put through our scoring system. After narrowing down to these designs, we focused on other factors that might limit or improve the functions of one design over the other. Engineering analysis was performed on each of the designs with specific emphasis placed on force balancing, fluid flow, strength and durability, and ease of manufacture. Because our prototype will be built with different materials and will perform under different conditions than the final valve, we had to take into account both the prototype and final valve when comparing both of the alpha designs. After comparing and contrasting the pros and cons of each alpha design, we needed to choose a final design. We looked into the possibility of combining the best features of both valve designs, but because of complexity issues and the inherent difference in actuation for the two designs, this would have required a near complete redesign. Thus, after discussion and analysis, a final valve design was chosen. The valve chosen as our final alpha design is very similar to our alpha design 1, with one small addition of a spring to improve bladder interaction inside the high-pressure accumulator.

Once the final alpha design was chosen, we began research to decide on the best materials and production method for our prototype. As we have stated, since testing the prototype at actual working conditions of 7000 psi with Mobil 1 transmission fluid is neither safe nor feasible, we tested the prototype at much lower pressure with water. A strength analysis was performed using dimensions from our final CAD model. Material selection exercises were undertaken to assess the functional performance and environmental performance of the final design material.

The complete project plan is outlined in a GANTT chart format in Appendix P. Throughout this entire process, all ideas, analysis, designs, recommendations, and failures were recorded in this report, as presenting these to the EPA is important in helping them get up to speed with our work and successfully continue with it in the future.

22 ACKNOWLEDGEMENTS

Team 10 would like to acknowledge a number of people for making this project a success. First, we would like to thank Mr. Jim Bryson and Mr. Andrew Moskalik, our EPA sponsors. They were able to describe in great detail the purpose and requirements of this project. Without their help, understanding the problem and finding the correct solutions that would most benefit the EPA would have been impossible. We would also like to thank Professor Hulbert for his constant support and encouragement throughout this project. His weekly comments were useful in steering us in the correct direction throughout this project, as well as focusing our efforts to complete the required goals of the ME 450 class.

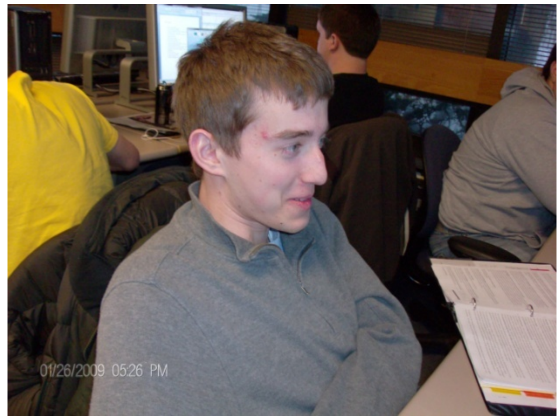
We would like to thank Professor Skerlos and the course GSI, Dan Johnson, for making ME 450 run as smoothly as possible, as well as for delivering entertaining and informative lectures. Professor Sienko was helpful in giving our team an outsider's view of the status of our project, and helped us prepare for presenting our project at the Design Expo.

We would also like to acknowledge Bob Coury and Marv Cressey for allowing us access to their knowledge of machining processes during our prototyping phase. We spent long hours in the shop, especially at the lathe, and their advice helped keep us safe and productive in the shop. We would also like to thank Tracie Straub for making reimbursements more enjoyable, and Mr. Russ Pitts for helping us improve our presentation skills.

23 BIOS

Kurt Cunningham was born in Southfield, MI and has lived in Redford, MI all of his life. He went to Thurston High School in Redford and was admitted into the University of Michigan at Ann Arbor for the fall 2005 semester. Kurt has always wanted to be an engineer, and the choice to become a Mechanical Engineer grew out of the desire to design and build many kinds of products and processes. Mechanical Engineers are uniquely equipped to have the ability both to design and to handle many engineering problems that can arise in any design. A Mechanical Engineer is useful in many industries, and it this reason also attracted Kurt to Mechanical Engineering, because he was not sure what kind of industry to pursue. Now that Kurt is close to graduating, he has decided to pursue a career in a field that focuses on sustainable practices. Kurt has two years of experience in Architectural Engineering of sustainable building systems with SmithGroup, Inc., which is headquartered in Detroit, MI. While at SmithGroup, Kurt contributed to the design of the mechanical systems for multiple LEED accredited building projects. Kurt is interested and is actively pursuing projects and work experience relating to alternatively powered automobiles. Kurt will be traveling to Berlin, Germany this summer to work on an alternative energy research project for automobiles with IAV, Inc. that will be continued as a Master's thesis project next year at the University of Michigan with Professor Volker Sick.

Bradley Dotson (goes by Brad) is from Jenison Michigan, and a graduate from Jenison Public High School. Brad was considering three colleges: University of Michigan, Grand Valley State University, and Kettering University as a senior in high school and chose Michigan for its opportunities to pursue swimming. As a freshmen Brad was a member of the varsity men's swimming and diving team and yes, he swam with Michael Phelps daily and did get to know him some. Brad loves engineering, particularly because he loves figuring things out. He often finds himself pondering how things work as he walks around campus, or sees something on television. You could call it an obsession, but who could blame him? After graduation from the University of Michigan Brad plans on getting a job. He would like to work on something like boats, motorcycles, ATVs, or the like. Something he can either play with at the end of the day, or if not, something interesting, such as tanks or other specialty military equipment.



Paul Juska is originally from Troy, MI, but moved to Shelby Township, MI half way through fifth grade. A graduate from Eisenhower High School, Paul always knew from early on that he wanted to study mechanical engineering in college. Having always been attracted to things with moving parts, there was never really a question of what kind of career he wanted to have. From very early in his life he'd always been attracted to engines of all kinds. An early fascination with them combined with complete ignorance as to their inner workings led him to spending most of his time learning about them. He knew next to nothing when he was 13 and by the time he was 16 he had pulled and was rebuilding the engine out of his first car about four months into owning it. The rest of high school was spent doing similar activities. Early in college he realized that his real attraction was to jet propulsion and has since focused on becoming (or trying to become!) an expert in that area. After graduation he plans to work developing propulsion engines for aircraft or spacecraft.

Paul has two older sisters, Jennifer, and Claudette. Both are married and Jennifer has two little boys, Andrew, and Alex. Jennifer lives in Novi, MI and Claudette works in Washington D.C. and lives just outside of the city.

Theodore Ross Ligibel graduated from Northview High school in Sylvania, Ohio with honors in 2005. He has always simply gone by the name Ross. During high school he was very interested in math, science, computer programming, and architecture. When it came time to apply for acceptance into an undergraduate program, he looked primarily at three universities. The University of Michigan was the final choice, within the college of engineering. Ross grew up with a very tit-nit and loving extended family; all based in the Toledo, Ohio area. Most of his cousins, aunts, and uncles are graduates and avid fans of the Ohio state university. Strangely, Ross has always been a Michigan fan. Every since he was a toddler he was attracted to the colors maize and blue and could be found any day wearing Michigan gear. As long as he can remember he has been watching Michigan football games and yearning to attend the school. When his particular skill set and academic skills developed he knew that U of M's engineering school would be perfect for him. Since first attending the University, Ross' interest in science and engineering has continued to grow. He has held three engineering internships during his time here and continues to hone his particular career interests. Because he was so interested in architecture as well as engineering, his first two internships were with companies that plan and design the construction and operation of large facilities like sports stadiums or automotive factories. He plans to continue in this direction, hopefully in the state of Colorado. In his spare time, Ross enjoys a variety of activities. Mostly, he is interested in snowboarding, skiing, and mountain biking. These hobbies are a big part of the reason he wishes to move out west.

24 REFERENCES

[1] NextEnergy.org, "Hybrid Hydraulics", Retrieved January 2009, <http://www.nextenergy.org/industryservices/Hybrid__Hydraulics.asp>

[2] U.S. Environmental Protection Agency, "EPA Announces Partnership to Demonstrate World's First Full Hydraulic Hybrid Urban Delivery Vehicle", February 2005, <http://www.epa.gov/otaq/technology/420f05006.htm>

[3] Skydel, Seth. "Ready for prime time: hydraulic hybrids. (Light- & Medium-duty). ." Fleet Equipment. 34.12 (Dec 2008): 4(1). General OneFile. Gale. University of Michigan - Ann Arbor. 28 Jan. 2009

[4] Brooks, David. "EPA shows off hydraulic hybrid technology. (United States. Environmental Protection Agency). ." New Hampshire Business Review. 28.16 (July 21, 2006): 21(1). General OneFile. Gale. University of Michigan - Ann Arbor. 28 Jan. 2009

[5] Andrew Moskalik, Ph.D., P.E., Jim Bryson. Mechanical Engineers, US Environmental Protection Agency, National Vehicle and Fuel Emissions Laboratory, Advanced Technology Division.

[6] Nesbitt, B., 2007, *Handbook of Valves and Actuators: Valves Manual International*, Butterworth-Heinemann, Oxford

[7] Smith, P., and Zappe, R.W., 2004, *Valve Selection Handbook,* 5th Edition, Gulf Professional Publishing, Burlington, MA

[8] Skousen, P.L., 2004, *Valve Handbook*, 2nd Edition, McGraw-Hill Companies, Inc., New York, NY

[9] Lyons, J.L., 1982, *Lyons' Valve Designer's Handbook*, Van Nostrand Reinhold Company, New York, NY

[10] Brown, M., 1995, *Seals and Sealing Handbook*, 4th Edition, Elsevier Science Publishers LTD, Kidlington, Oxford

[11] "Types of Valves - Ball Valve", Retrieved 12 March.2009, <http://profmaster.blogspot.com/2008/10/type-of-valves-ball-valve.html>

[12] "Knowledge Publications", Retrieved 12 March.2009, <http://knowledgepublications.com/doe/doe_mechanical_science_web_educational_textbook_solar_hydrogen_fuel_cells.htm>

[13] Autoblog.com, Retrieved 12 March.2009, <http://www.autoblog.com/2006/12/22/behold-the-cam-less-future/>

[14] Answers.com, Retrieved 12 March, 2009, <http://www.answers.com/topic/globe-valve>

[15] Gray, J., *Hydraulic Hybrid Accumulator Shut -off Valve*, in *uspto.gov*, USPTO, Editor. 2003, US EPA: United States.

[16] Farr, G.P.R., *Control Valves for Vehicle Hydraulic Systems*, in *uspto.gov*, USPTO, Editor. 1985, Lucas Industries Public Limited Company: United States.

[17] Jones, F., Ryffel H., Oberg E., McCauley C., Heald R., 2004, *Machinery's Handbook, 27th Edition*, Industrial Press, Inc.

[18] Watkins, R. K., Anderson, L. R. (2000). Structural Mechanics of Buried Pipes, CRC Press.

[19] *CES Selector Version 4.8.0*, 2008, Cambridge, UK, Granta Design Limited

[20] *SimaPro Version 7.1.4*, PRé Consultants, 2007

25 APPENDIX A: EPA Valve Requirements

Integrated HP Accumulator Valve Functions

- Stop flow from exiting accumulator when de-energized (positive seal – no leak) (Normally closed)
- Allow flow both directions with minimal pressure drop when energized
- Always allow flow into the accumulator (can be relatively large pressure drop when off)
- Stop flow from exiting accumulator when almost empty “bottom”*
- Allow flow back into accumulator when “bottomed”*
- No external leaks
- Rated for 7000psi (48MPa) operation
- Interface with both piston-in-shell and bladder accumulator barriers*
- Gradual pressure equalization to prevent sharp pressure rise/noise when opening
- Not adversely affected by road salt, water, occasional small stone impact
- Ambient temperatures of -40F/C to 140F/60C
- Oil temperatures of -40F/C to 180F/82C

Other Information

- Velocity fuse/safety shutoff ~<185gpm
- Fast reaction time to close 150 – 300ms
- Fast to open (but not too fast) 150 – 300ms
- Cycles ~200/day, 40,000/year, 1,000,000/lifetime
- 4-bolt code 62 flange
- Actuating valve should operator down to 10V or less

*features unique to integrated accumulator valve

26 APPENDIX B: Quality, Function, Deployment Diagram

Customer Weight Scale:	
5	→ Essential
4	→ Less Essential
3	→ Important
2	→ Less Important
1	→ Secondary

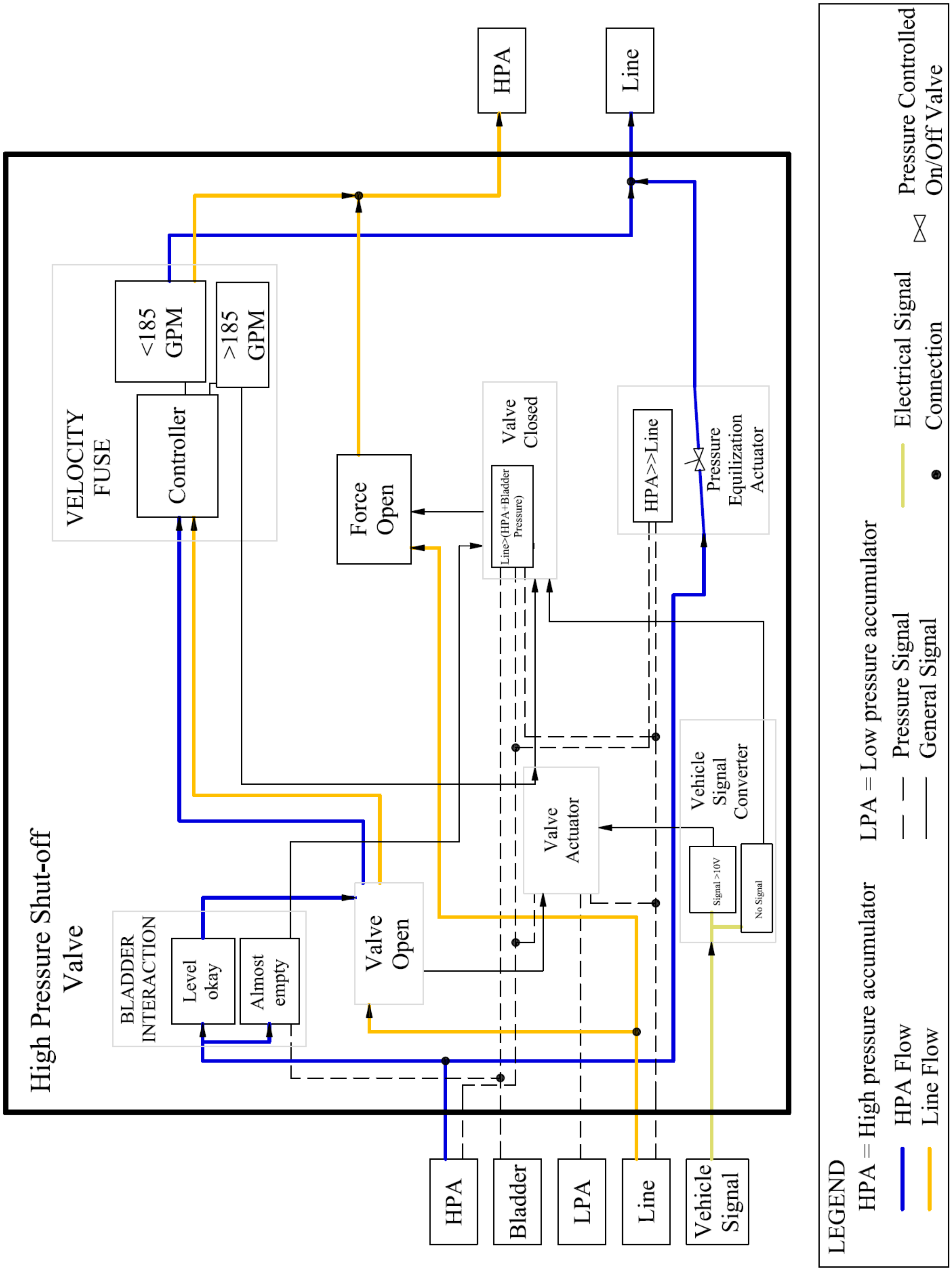
Correlation Matrix Scale:	
9	→ Strongly Correllated
3	→ Correlated
1	→ Weakly correllated
0	→ Not at all correllated

Compatability Matrix Scale:	
++	→ Strongly Positive
+	→ Positive
-	→ Negative
--	→ Strongly Negative

		Overall cost	Number of parts	Overall weight	Tank side outer pipe diameter	Tank side inner pipe diameter	System side outer pipe diameter	System side inner pipe diameter	Threading size	Threading length	Flow out of accumulator when de-energized (leak)	Pressure drop when energized	Works at 2000 psi	Works at 7000 psi	Flow into accumulator when de-energized	Pressure difference required to allow flow into accumulator	Flow out of accumulator when "bottomed" (leak)	Flow into accumulator when "bottomed"	Flow external to valve (leak)	Diameter of equalization orifice	Variability in diameter of equalization orifice	Time to pressure equalization	Materials (resistance to road wear)	Materials (resistance to temperature failure)	Materials (resistance to cyclical failure)	Maximum flow rate	Velocity fuse	Reaction time to close	Reaction time to open
Overall cost	Low																												
Number of parts	Low	++																											
Overall weight	Low	+	+																										
Tank side outer pipe diameter	Fits																												
Tank side inner pipe diameter	Low	-	-	-																									
System side outer pipe diameter	Fits																												
System side inner pipe diameter	Low	-	-	-																									
Threading size	Fits																												
Threading length	Short	+																											
Flow out of accumulator when de-energized (leak)	None																												
Pressure drop when energized	Low					--		--																					
Works at 2000 psi	Yes										+	+																	
Works at 7000 psi	Yes		--	--		++		++			+	+																	
Flow into accumulator when de-energized	Low													+															
Pressure difference required to allow flow into accumulator	Low	-	+								-		+	+	-														
Flow out of accumulator when "bottomed" (leak)	None	+	-																										
Flow into accumulator when "bottomed"	Low															+													
Flow external to valve (leak)	None				+		+		+	-																			
Diameter of equalization orifice	Low	-	-	-																									
Variability in diameter of equalization orifice	High	--	--	--																									
Time to pressure equalization	Low	-	-											-						--	++								
Materials (resistance to road wear)	High	--	--	--		+		+						+					+										
Materials (resistance to temperature failure)	High	--	--	--		+		+						+					+				++						
Materials (resistance to cyclical failure)	High	--	--	--		++		++	+	-	+			+					+		+		++	++					
Maximum flow rate	High					-		-											-										
Velocity fuse	High																									++			
Reaction time to close	Low	--									+														+		++		
Reaction time to open	Low	--										+								--	+	-			+			+	
Activation voltage	Low	-									+	+																	

	←Customer Weight / Correlation Value→	Overall cost	Number of parts	Overall weight	Tank side outer pipe diameter	Tank side inner pipe diameter	System side outer pipe diameter	System side inner pipe diameter	Threading size	Threading length	Flow out of accumulator when de-energized (leak)	Pressure drop when energized	Works at 2000 psi	Works at 7000 psi	Flow into accumulator when de-energized	Pressure difference required to allow flow into accumulato	Flow out of accumulator when "bottomed" (leak)	Flow into accumulator when "bottomed"	Flow external to valve (leak)	Diameter of equalization orifice	Variability in diameter of equalization orifice	Time to pressure equalization	Materials (resistance to road wear)	Materials (resistance to temperature failure)	Materials (resistance to cyclical failure)	Maximum flow rate	Velocity fuse	Reaction time to close	Reaction time to open	Activation voltage
Low cost	2	9	3	3																			1	1	1					
Low complexity	1		9																		1									
Designed for manufacturability and maintenance	3		9																		3		3	3	3	1	1			1
Low weight	1		3	9		1		1						3							3		3	3	3					
Small size	1		3	1										1							1		1	1	1					
Low noise	4		1			1		1			3	3			1					9	9	9			1	1	1			
Compatible with accumulator	4				9	9			9	9			3	3			3	3	3									1	1	
Compatible with pipes of system	4						9	9	9	9				3																
Stop flow when de-energized (no-leaks)	5				3	3					9		3	3									1	1	1			1	1	9
Flow in both directions when energized (low pressure drop)	5					3		3				9	3	3		9	1						1	1	3	3	3	3	3	3
Works at operating pressures	5				3	3	3	3	1	1	3		9	9	3	9	3	3					3	3	3			3	3	
Allows flow into accumulator when de-energized	4												3	3	9	9														
Stop flow when "bottomed"	4												3	3		9	9	9												
No external leaks	3				3		3		3	3									9				3	3	3					
"Bottoms" with bladder and piston-in-shell accumulators	4												3	3		3														
Gradual pressure equalization to prevent fluid hammer	4												3	3		9			1	9	9	9								
Not adversely effected by road conditions (-40F to 140F)	3																						9	9	9					
Operational at oil temperatures of -40F to 180F	3																							9						
Compatible with Mobile-1 ATF	3										3	3	3	3					3	3				3		3	3			
Shuts off at extremely high flow rates	4												3	3												3	3			
Must energize and de-energize quickly	4												3	3						1	1	3						9	9	3
Does not fail due to 1,000,000 cycles	4					3		3	3	3			1	1									1	1	9			1	1	3
Energizes within a range of acceptable voltages	4												3	3																9
Normally closed	5										9		3	3									1	1	1				3	1
Total:		18	52	16	75	98	60	83	98	98	126	66	199	215	55	210	68	63	52	85	90	84	85	121	131	43	43	79	94	128
Normalized:		0.0068	0.0197	0.0061	0.0285	0.0372	0.0228	0.0315	0.0372	0.0372	0.0478	0.0250	0.0755	0.0816	0.0209	0.0797	0.0258	0.0239	0.0197	0.0323	0.0342	0.0319	0.0323	0.0459	0.0497	0.0163	0.0163	0.0300	0.0357	0.0486
Importance Ranking:		28	T24	29	18	T8	22	16	T8	T8	6	20	3	1	23	2	19	21	T24	T13	12	15	T13	7	4	T26	T26	17	11	5
Measurement Unit:		$	0	lbs	in.	in.	in.	in.	th./in	in.	gpm	ft. hd.	0	0	gpm	psi	gpm	0	gpm	in.	in.	ms	0	0	0	gpm	gpm	ms	ms	V
Target Value:		<15,000	<20	<20	2	TBD	TBD	1.255	TBD	TBD	0	TBD	yes	yes	TBD	TBD	0	yes	0	TBD	TBD	50-100	steel	steel	steel	130-140	185	300	150	10-12
EPA Valve Design		15,000	45		2	1.15		1.225			0		yes	yes	0	0	0	yes	0			50-100	steel	steel	steel	130-140	185	300	150	10-12

27 APPENDIX C: Functional Decomposition



28 APPENDIX D: Valve Scenario Chart

Valve Scenario Chart

Static (No Flow in either direction)

Initial Valve Position	Control Signal	Pressure in Tank	Pressure in line	Desired Valve Function
Closed	Close	High	Low	Remain Closed

Flow into Tank

Initial Valve Position	Control Signal	Pressure in Tank	Pressure in line	Desired Valve Function
Closed	Close (default)	Low	High	Leak Fluid into Tank
Closed	Close	High	Approaching High	Remain Closed until PL > PT
Closed	Close	High	Equal to PT	Remain Closed until PL > PT
Closed	Close	High	Higher	Leak Fluid into Tank
Open	Open	Low	High	Full Flow into Tank

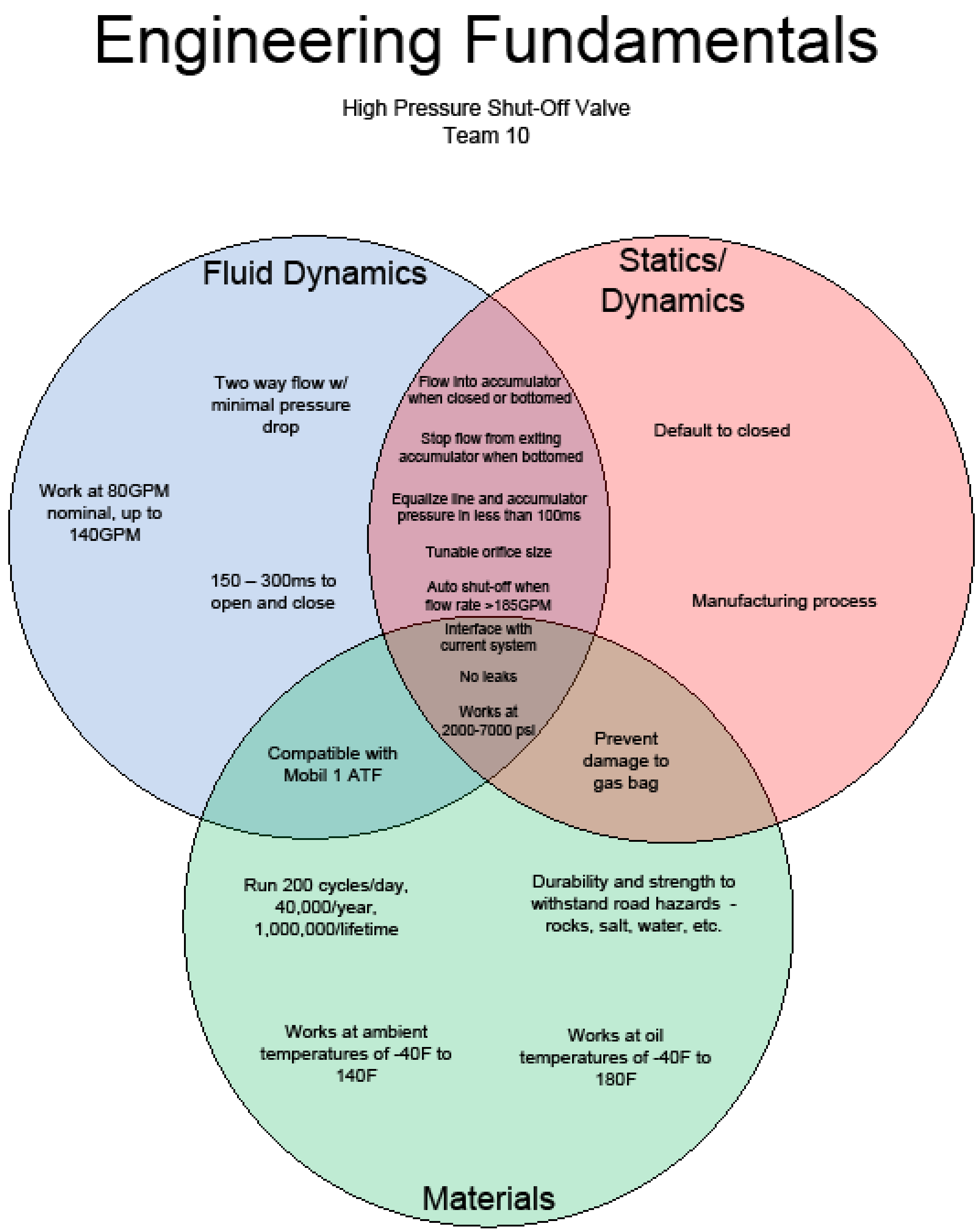
Flow Out of Tank

Initial Valve Position	Control Signal	Pressure in Tank	Pressure in line	Desired Valve Function
Closed	Close	High	Low	Seel Tank, (no leaks)
Open	Open	High	Equal to PT	Open (no flow, PT = PL)
Open	Open	High	Low	Free Flow, Low pressure Drop
Open	Open	High	Low	Shut off if GPM>185???

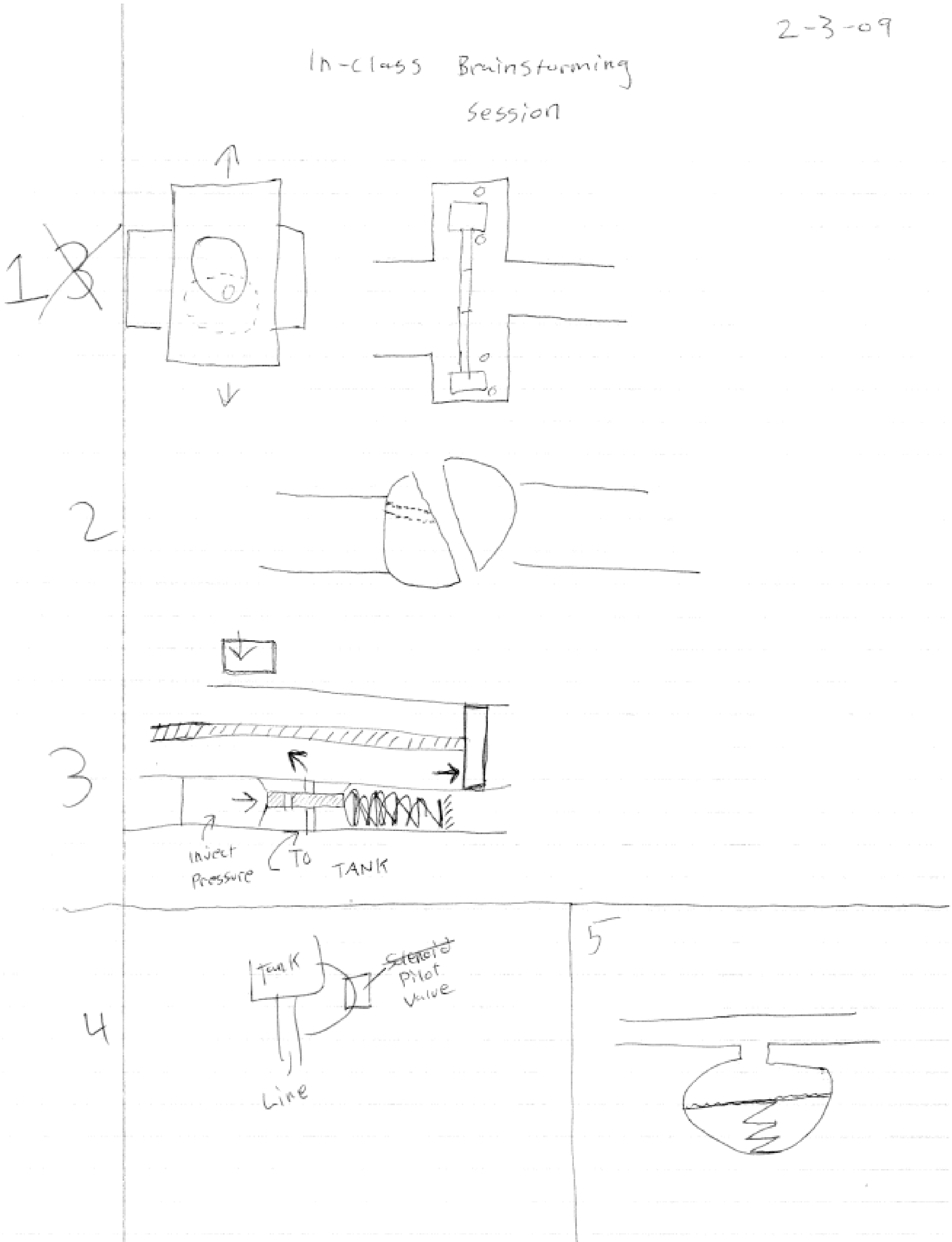
Dynamic Signal (Opening/Closing)

Initial Valve Position	Control Signal	Pressure in Tank	Pressure in line	Desired Valve Function
Closed	Open	High	Low	Equalize Pressure, Then Open
Closed	Open	High	High	Allow Flow
Closed	Open	Low	High	Allow Flow into tank
Open	Open	Low	Lower	Bladder presses on Valve- closes Valve
Open	Close	High	Low	Valve Closes (Dampers to reduce slamming?)

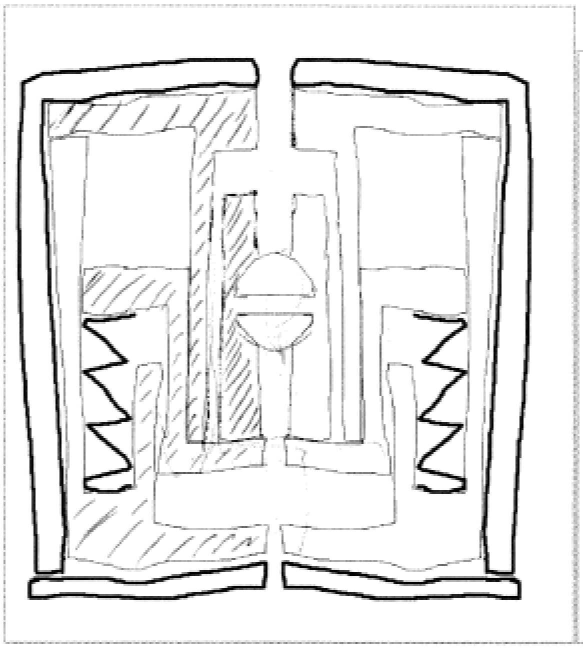
29 APPENDIX E: Breakdown of Engineering Fundamentals Required in the Design

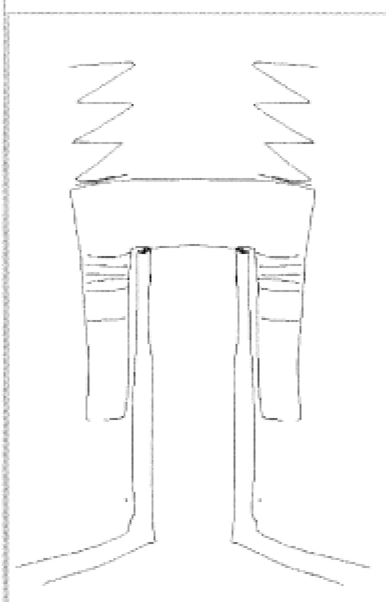


30 APPENDIX F: Pressure Equalization Concepts

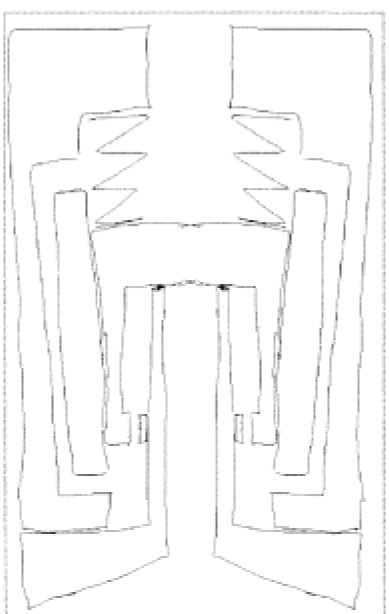


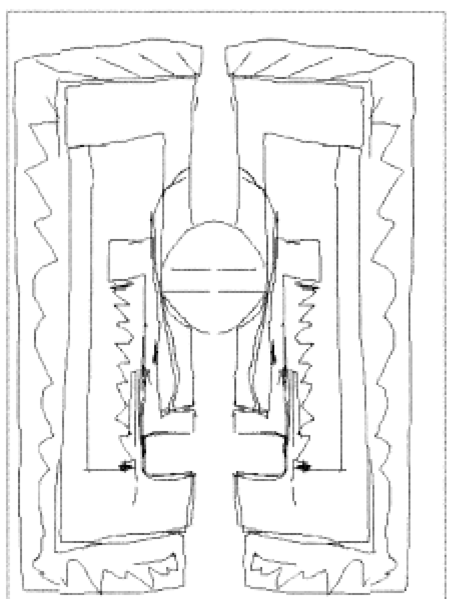
31 APPENDIX G: First Design Concepts – Main valve and individual components

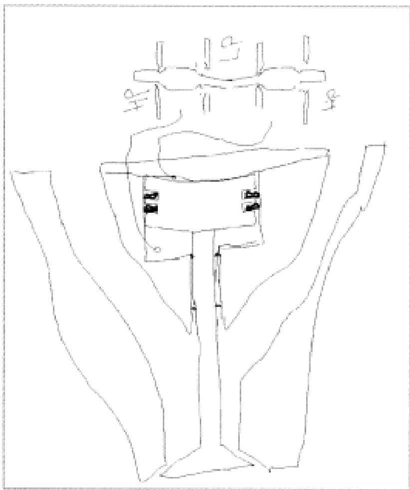


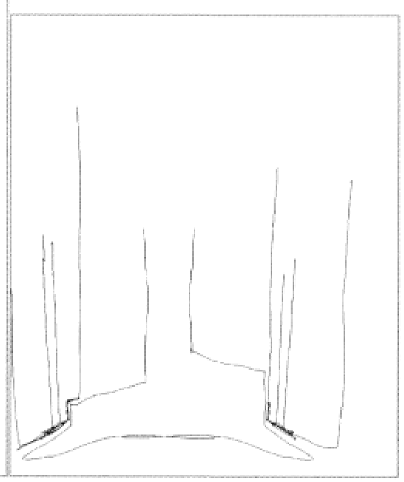


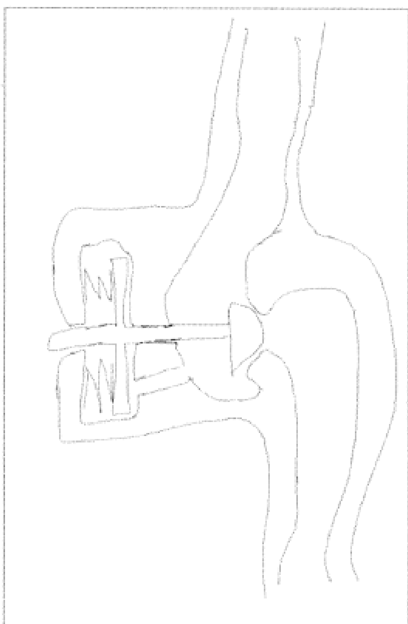
TANK

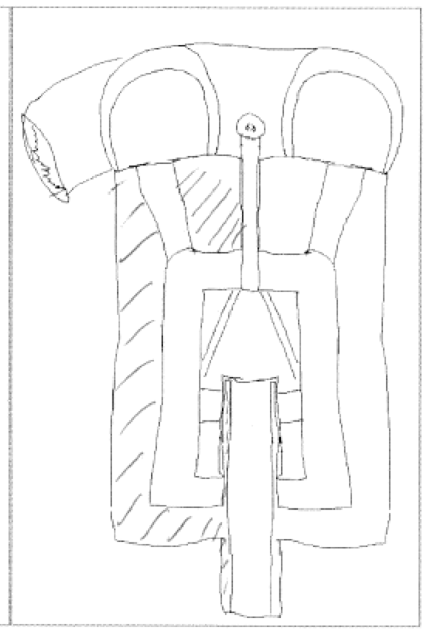


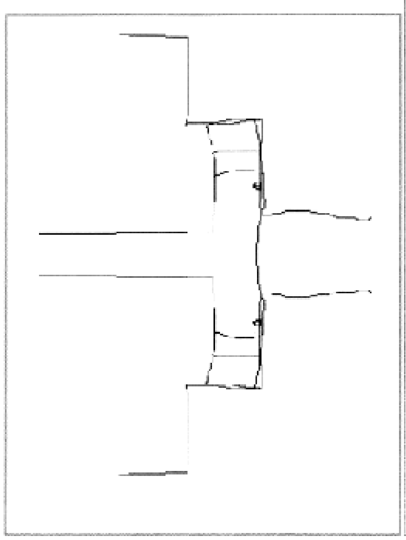


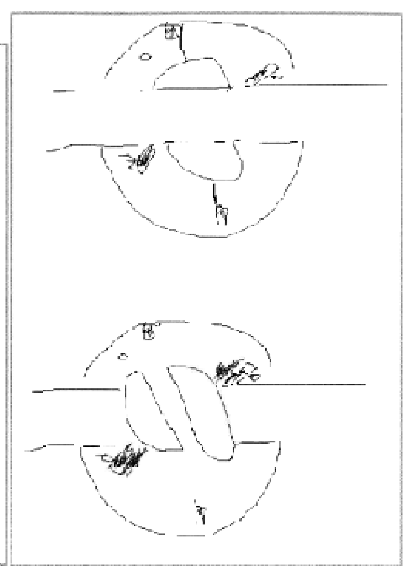












32 APPENDIX H: Concept Selection Chart – Complete Designs

Legend of Ratings:
0 N/a
1 Bad
3 Poor
6 Good
9 Excellent

		CONCEPTS							
		Current EPA Design		**Poppet valve 1**		**Poppet valve 2 (90 degree)**		**Poppet valve 3 (springs)**	
Selection Criteria	**Weight**	Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Bladder interaction	9	9	1.23	6	0.82	6	0.82	9	1.23
Appropriate pressure difference for flow into accumulator	6	6	0.55	9	0.82	6	0.55	9	0.82
Ease of manufacture	9	1	0.14	6	0.82	6	0.82	3	0.41
Ease of maintenance	9	3	0.41	9	1.23	9	1.23	6	0.82
How smoothly fill function is integrated into valve	6	6	0.55	6	0.55	6	0.55	6	0.55
How smoothly pressure equilization is integrated into valve	6	3	0.27	9	0.82	6	0.55	6	0.55
Incorporates velocity fuse	3	0	0.00	0	0.00	0	0.00	0	0.00
Low complexity	9	3	0.41	9	1.23	9	1.23	6	0.82
Low pressure drop at operating conditions and flowrates	6	6	0.55	6	0.55	9	0.82	9	0.82
Low weight	3	6	0.27	6	0.27	6	0.27	6	0.27
Resistance to hazardous road conditions	6	6	0.55	6	0.55	6	0.55	6	0.55
Small size	3	6	0.27	6	0.27	6	0.27	6	0.27
Compatibility with post valve connections	6	6	0.55	6	0.55	9	0.82	9	0.82
Actuation source compatibility	9	9	1.23	9	1.23	9	1.23	9	1.23
Number of seals	-6	8	-0.73	8	-0.73	7	-0.64	8	-0.73
Number of high pressure seals when de-energized	-9	3	-0.41	6	-0.82	5	-0.68	5	-0.68
Number of springs	-3	3	-0.14	1	-0.05	1	-0.05	3	-0.14
Number of static seals		2		4		3		3	
Number of dynamic seals	-6	6	-0.55	4	-0.36	4	-0.36	5	-0.45
Number of parts		45		5		6		10	
Total Score			5.1		7.7		8.0		7.1
Rank			**4**		**2**		**1**		**3**

33 APPENDIX I: Concept Selection Charts – Valve actuation & Main Valve Style

VALVE ACTUATION	Electric motor	Steppermotor	Hydraulic Acutation	Solenoid actuation
Availability of source energy	2	2	3	2
Ease of integration	3	3	4	3
Speed of actuation	3	2	4	4
Durability	3	3	3	3
Available driving force (relative to other t	1	1	4	2
Position control	2	4	2	2
Total Score	14	15	20	16
Rank	**4**	**3**	**1**	**2**

MAIN VALVE	Ball	Gate	Poppet
Pressure drop at design flowrate	4	4	3
Speed of actuation (with similar actuator)	2	2	4
One way flow	0	0	3
Potential for pressure equilization	3	1	2
Ease of manufacture	1	3	3
Force required to open	3	2	2
Bladder interaction	1	1	4
Normally closed	2	2	3
Can handle high pressures	4	2	3
Total Score	20	17	27
Rank	**2**	**3**	**1**

Legend of Ratings:

0 None
1 Bad
2 Not so good
3 Good
4 Excellent

34 APPENDIX J: Concept Selection Chart – Pressure Equalization

PRESSURE EQUILIZATION	Weight	Start engine Rating	Start engine Weighted	Electric motor Rating	Electric motor Weighted	Separate tank Rating	Separate tank Weighted
No energy requirement	0.2	1	0.2	2	0.4	3	0.6
In series with main valve	0.05	4	0.2	4	0.2	4	0.2
Speed	0.15	1	0.15	2	0.3	4	0.6
Complexity	0.3	4	1.2	2	0.6	2	0.6
Ease of incorporation into system/valve	0.3	4	1.2	1	0.3	1	0.3
Total Score			2.95		1.8		2.3
Rank					**6**		**5**

Hydraulic damper Rating	Hydraulic damper Weighted	Valve seat geometry Rating	Valve seat geometry Weighted	Solenoid valve Rating	Solenoid valve Weighted	Fixed orfice Rating	Fixed orfice Weighted
4	0.8	3	0.6	3	0.6	3	0.[illegible]
4	0.2	4	0.2	1	0.05	1	0.0[illegible]
4	0.6	3	0.45	3	0.45	3	0.4[illegible]
4	1.2	3	0.9	2	0.6	3	0.[illegible]
1	0.3	3	0.9	4	1.2	4	1.[illegible]
	3.1		3.05		2.9		3.[illegible]
	2		**3**		**4**		[illegible]

Legend of Ratings:
1 Bad
2 Not so good
3 Good
4 Excellent

35 APPENDIX K: Material Selection (Functional Performance)

The two components to be investigated for material selection are the main valve housing and the valve stem. Each component has specific criteria for which the material must provide good performance. In particular, the valve housing must provide protection from the environment to all other valve components. Some hazards include salt and fresh water, as well as stone impacts. The valve stem must carry an extremely large tensile load, therefore the criteria for a strong tensile tie will be applied. Because the main goal of the project is to reduce cost, all material indices will minimize cost for each objective.

The material indices were plotted on log-log axes. In general, the numerator and denominator of the material index were separated allowing each to be plotted on separate axes. A coupling line could then be formed which would allow the best material to be easily recognized (for most cases, it would be toward the top left of the graph). Materials that fell on the same coupling line would perform equally well when used for the same component. Because more than one material index was compared for each valve component, the optimum material could either show the best combination between the two, or be biased toward one performance objective. Here, materials were usually biased toward higher strength as opposed to other properties.

35.1 Valve Housing

Function: Contain fluid pressure; provide precision surfaces for component translation; protect internal components and fluid from environment

Objective: Minimum cost strong pressure vessel, corrosion resistance

Constraints: No yield, Leak-before-break

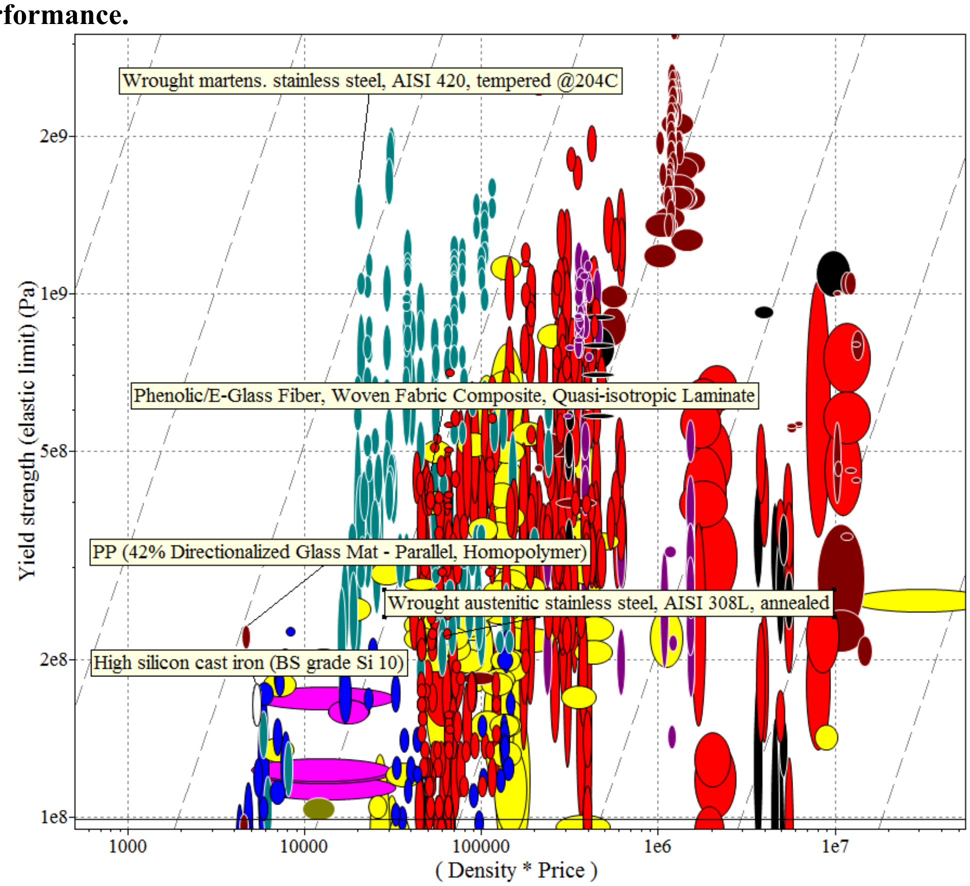
The specific material selection criteria for the valve housing are presented above. Two performance indices were used for the material selection: the index for a strong, cheap pressure vessel, and the index for maximum leak-before-break performance. These were chosen to avoid the failure by yield, as well as by fracture. The pressure vessel material index was chosen since the valve housing will closely represent a pressure vessel in form and function. The resulting plot of the strong, cheap cylindrical pressure vessel is presented in Fig. 35.1. In addition, a plot of materials with the greatest leak-before break performance (essentially highest fracture toughness for the range of materials chosen) was made, but it of the same style as Figure 35.1 and is therefore not presented here. Instead, the results from both plots are presented in Table 35.1.

The top five material choices are shown in Table 35.1. These were chosen as the best overall performing materials considering the combination of both performance indices shown. The best performance was found by drawing a coupling line of materials with constant performance (these lines can be seen in Fig 35.1) and choosing the material furthest to the top left, thus maximizing the material index. The materials with the best combination of performance indices are those listed in Table A.

Table 35.1: Material choices for the valve housing. Materials are ranked from 1 to 5 for each performance attribute (1 being the best, 5 being the worst).

Material Index	**Strong, cheap, pressure vessel** $\sigma_y / c_m \rho$	**Leak-before-break** K_{IC}^2 / σ_y
Wrought martensitic Stainless steel, AISI 420, tempered @ 204°C	1	5
PP (42% Directionalized Glass Mat – Parallel, Homopolymer)	2	4
High silicon cast iron (BS grade Si 10) (roughly equivalent to High Cr white cast iron (BS grade 3A))	3	3
Phenoloic/E-Glass Fiber, Woven Fabric Composite, Quasi-isotropic Laminate	4	2
Wrought austenitic stainless steel, AISI 308L, annealed	5	1

Figure 35.1: Log-log plot of performance index: Cheap, strong pressure vessel. The best materials are found toward the upper left of the plot; materials on the same line have equal performance.



The top material was selected to be AISI 420 stainless steel tempered at 204°C. This material was chosen mainly for its strength in a low cost pressure vessel application. In addition, it has very good corrosion resistance to fresh water, salt water, and organic compounds, all of which will be experience by the valve housing. It can be seen in Table 35.1 that 420 stainless steel has the lowest leak-before-break performance of the materials chosen, but this is only relative to the other materials in the table (who all have excellent performance in this area) and the actual performance is suitable for the valve housing application.

Note that aluminum was eliminated from the above choices because of lower resistance to salt water. All the materials listed in the final selection have a maximum service temperature above the customer specification for the HPSOV of 180°F.

35.2 Valve Stem

Function: Contain fluid pressure; carry large tensile load

Objective: Minimum cost strong pressure vessel (+strong cheap tensile tie), cheapest stiff tie, corrosion resistance

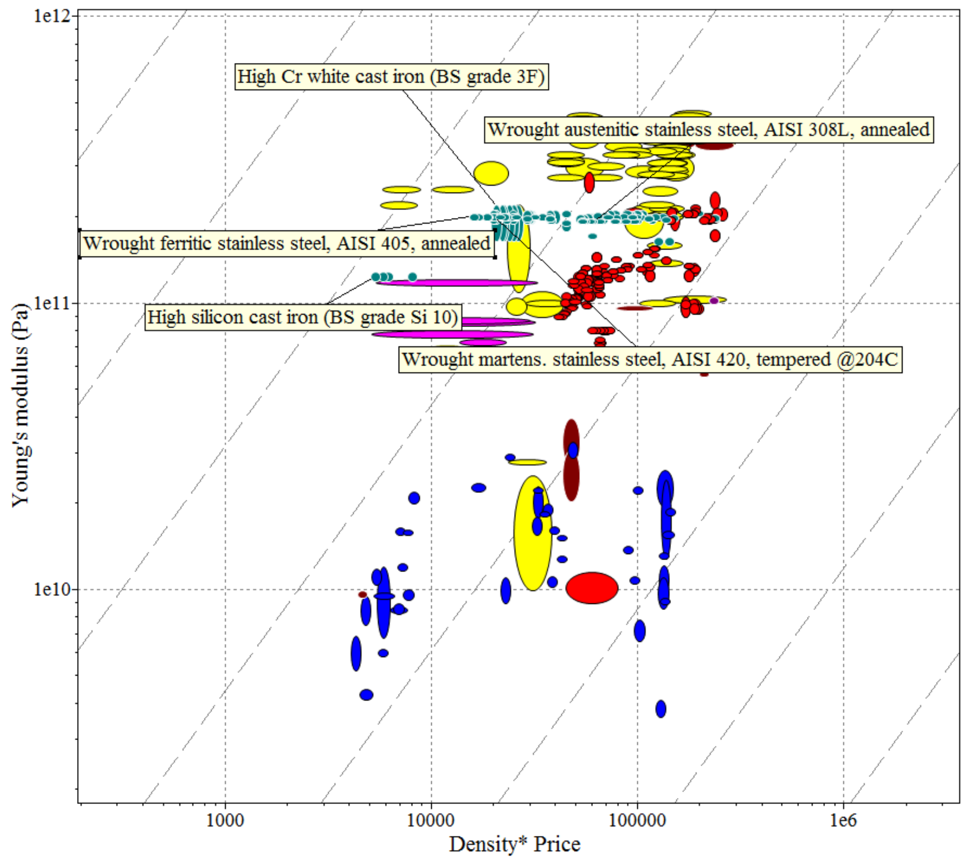
Constraints: No yield

As for the valve housing, the criteria for the materials selection of the valve stem are presented here. The valve stem has somewhat similar functionality to the valve housing in that it must contain the high pressure fluid flowing through its center. In addition, however, it must also support a strong tensile load at the same time. This load comes from the force on the actuator piston that is transferred through the valve stem, passes through the valve seat, and is grounded on the valve housing. Incidentally, the material index for a strong pressure vessel is the same as that for a strong tensile tie, so Figure 35.1 was also used for this material selection. Maximizing this performance index would provide good results for both types of loading the valve stem will see. The plot was combined with that in Figure 35.2 (cheap stiff tensile tie) to find the best overall materials for the valve stem. The stiff tensile tie performance index was used so that the deflection of the valve stem would be kept to a minimum when it is in the open position. Finally, strong corrosion resistance from organic fluids was necessary and only materials with this capability were considered. After considering all these aspects, the top five materials were chosen and are listed in Table 35.2.

Table 35.2: Material choices for the valve stem. Materials are ranked from 1 to 5 for each performance attribute (1 being the best, 5 being the worst).

	Strong, cheap, tie (& pressure vessel)	**Stiff, cheap tie**	**Fracture toughness**
Material Index	$\sigma_y / c_m \rho$	$E / c_m \rho$	K_{IC}
Wrought marens. Stainless steel, AISI 420, tempered @ 204C (great combination of stiffness and strength)	1	4	3
High silicon cast iron (BS grade Si 10) (best stiffness)	2	1	5
High Cr white cast iron (BS grade 3F) (favors stiffness)	3	3	4
Wrought ferritic stainless steel, AISI 405, annealed (good stiffness)	4	2	2
Wrought austenitic stainless steel, AISI 308L, annealed (worst overall combination)	5	5	1

Figure 35.2: Log-log plot of performance index: Cheap, stiff tensile tie. The best materials are found toward the upper left of the plot; materials on the same line have equal performance.

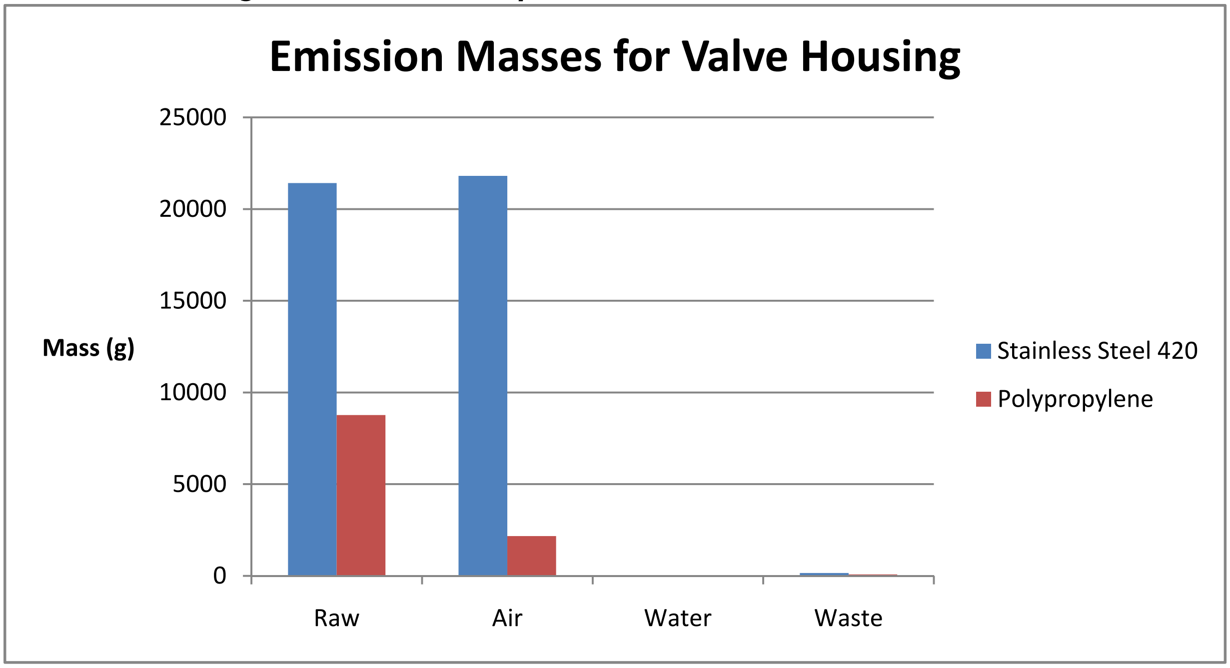


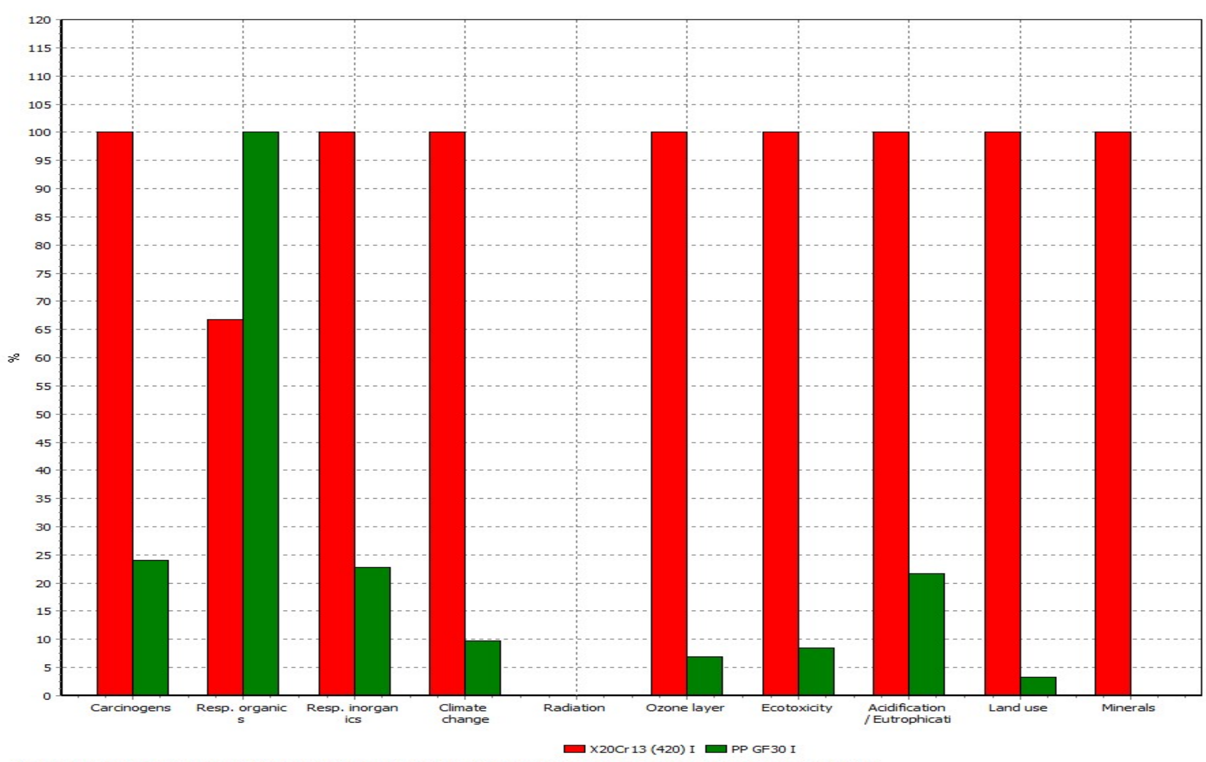
The best overall material for the valve stem was chosen to be AISI 420 stainless steel tempered at 204°C. This is the same material that was selected for the valve housing. Although the material has excellent properties for the application in question, it has the added benefit of being similar to what is used for the valve housing. Using the same material for both components may provide an additional cost benefit during production.

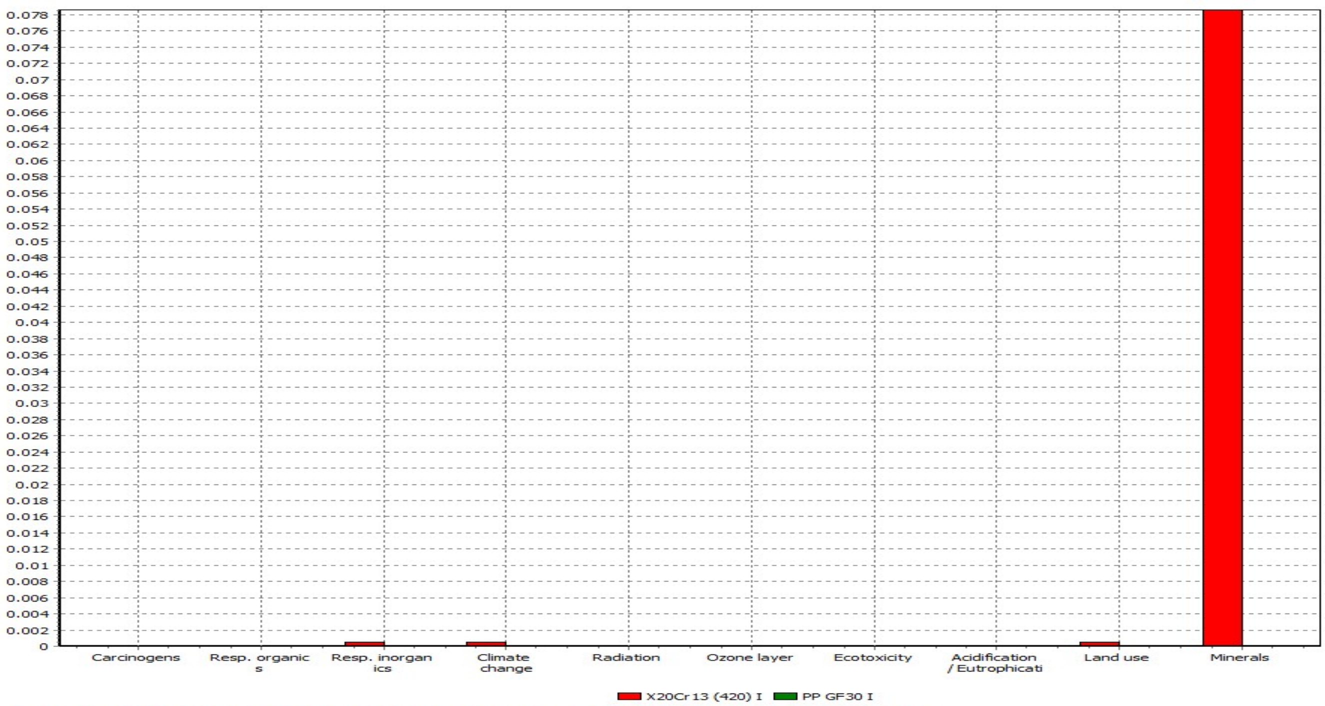
Note that aluminum was eliminated from the above choices because of lower resistance to salt water. All the materials listed in the final selection have a maximum service temperature above the customer specification for the HPSOV of 180°F.

36 APPENDIX L: Material Selection (Environmental Performance)

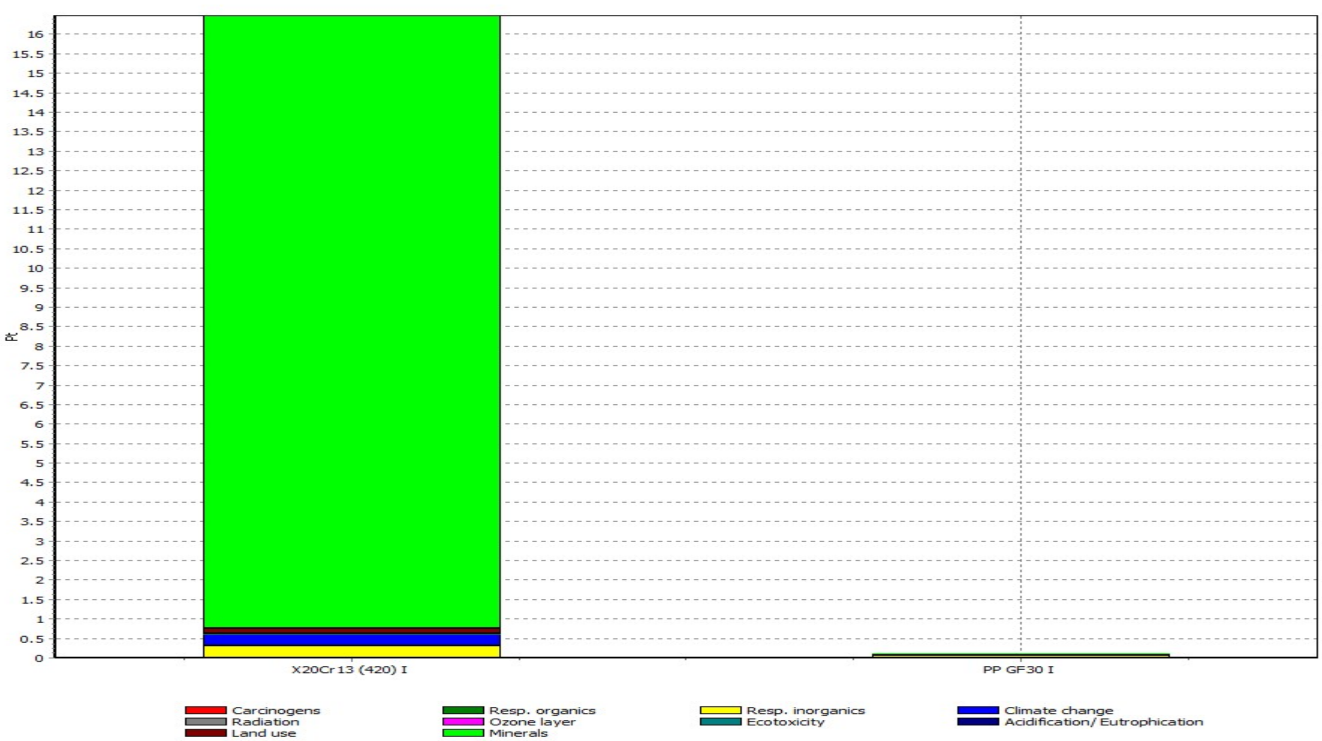
36.1 Valve Housing Environmental Performance





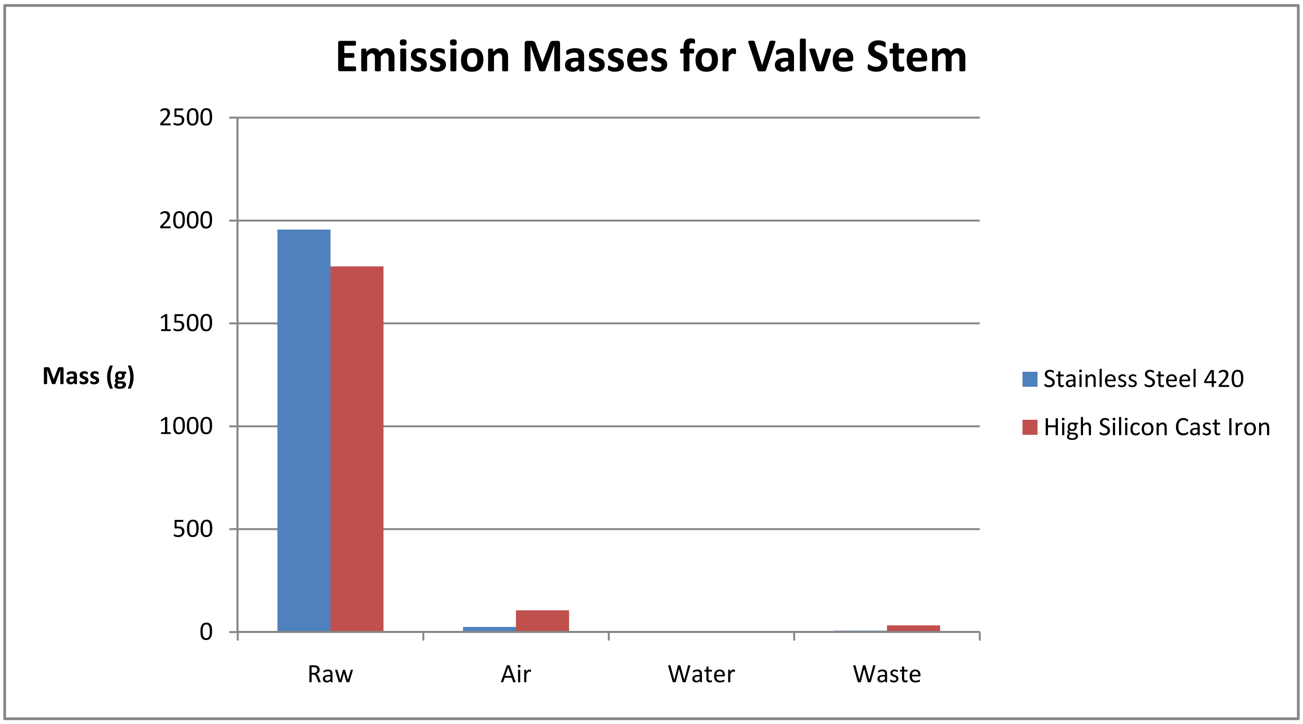


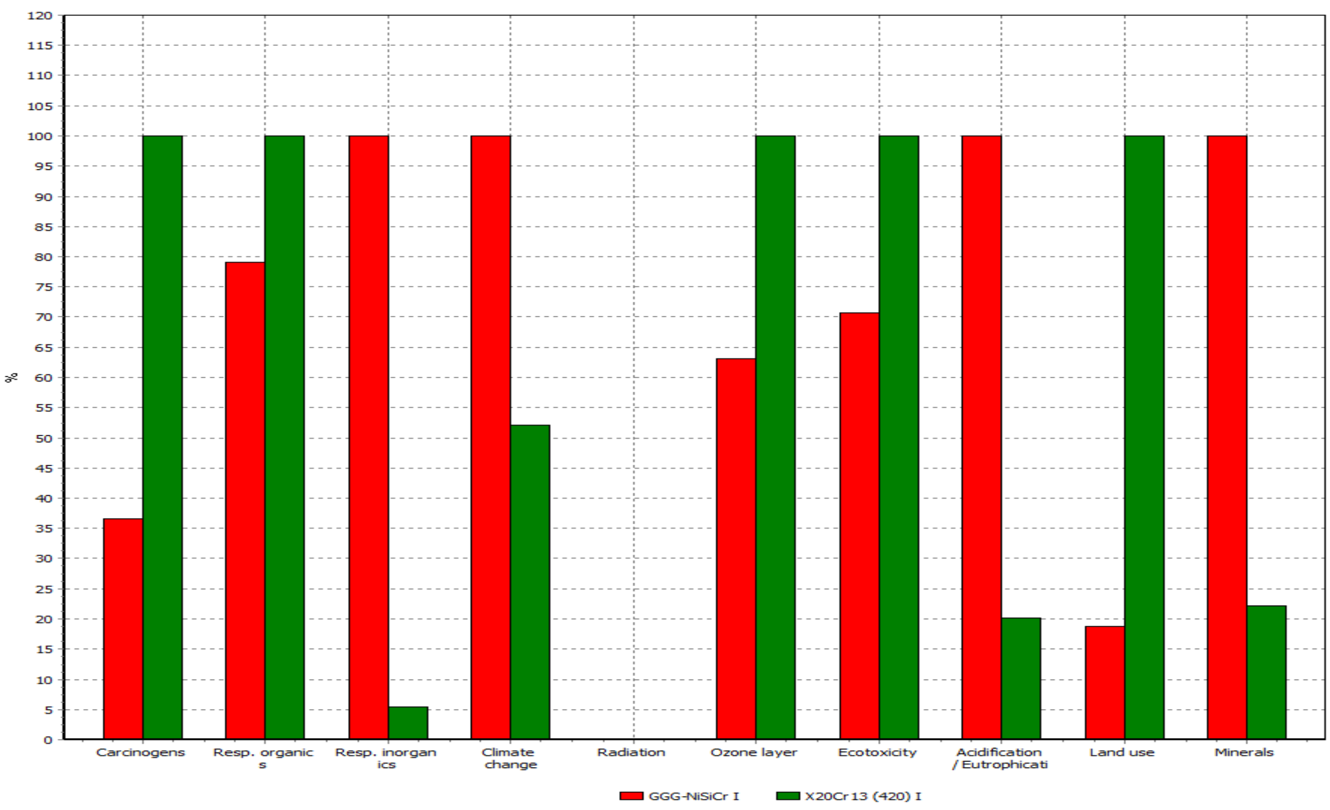
Comparing 31.9 lb 'X20Cr13 (420) I' with 5.03 lb 'PP GF30 I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/I / normalization

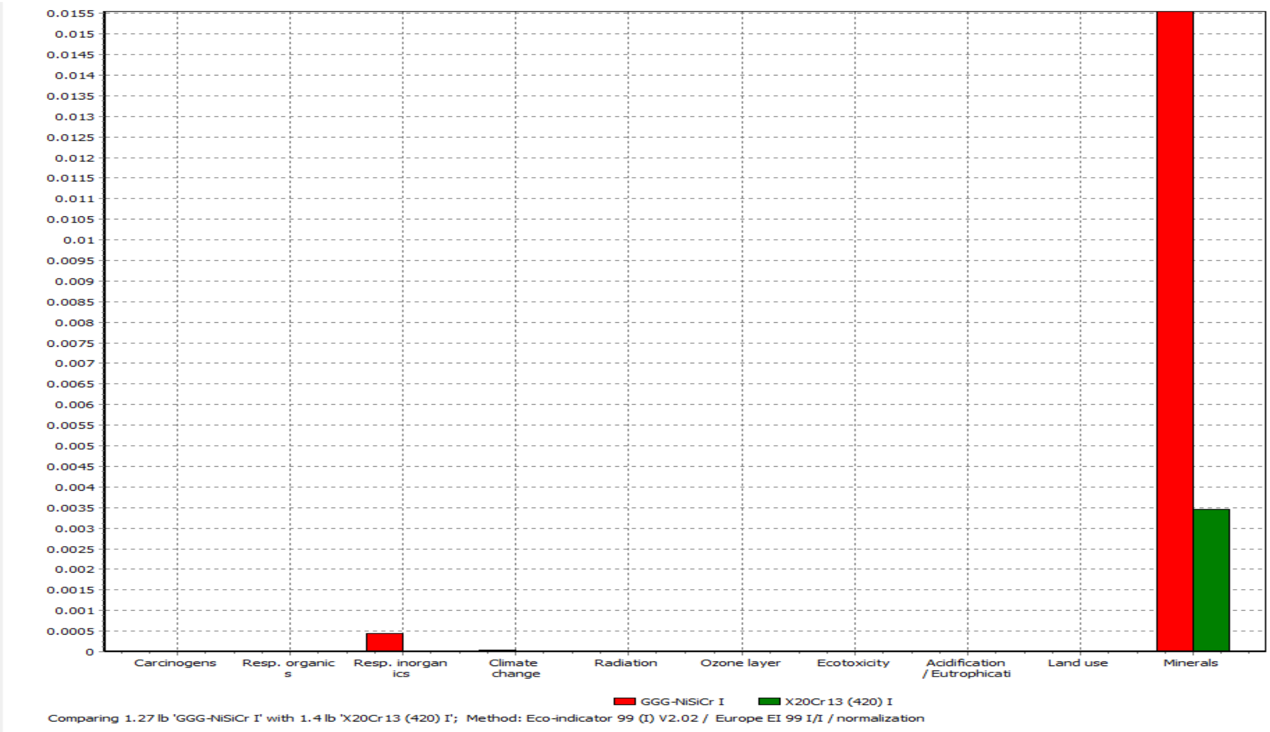


Comparing 31.9 lb 'X20Cr13 (420) I' with 5.03 lb 'PP GF30 I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/I / single score

36.2 Valve Stem Environmental Performance






0.0155
0.015
0.0145
0.014
0.0135
0.013
0.0125
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0.0035
0.003
0.0025
0.002
0.0015
0.001
0.0005
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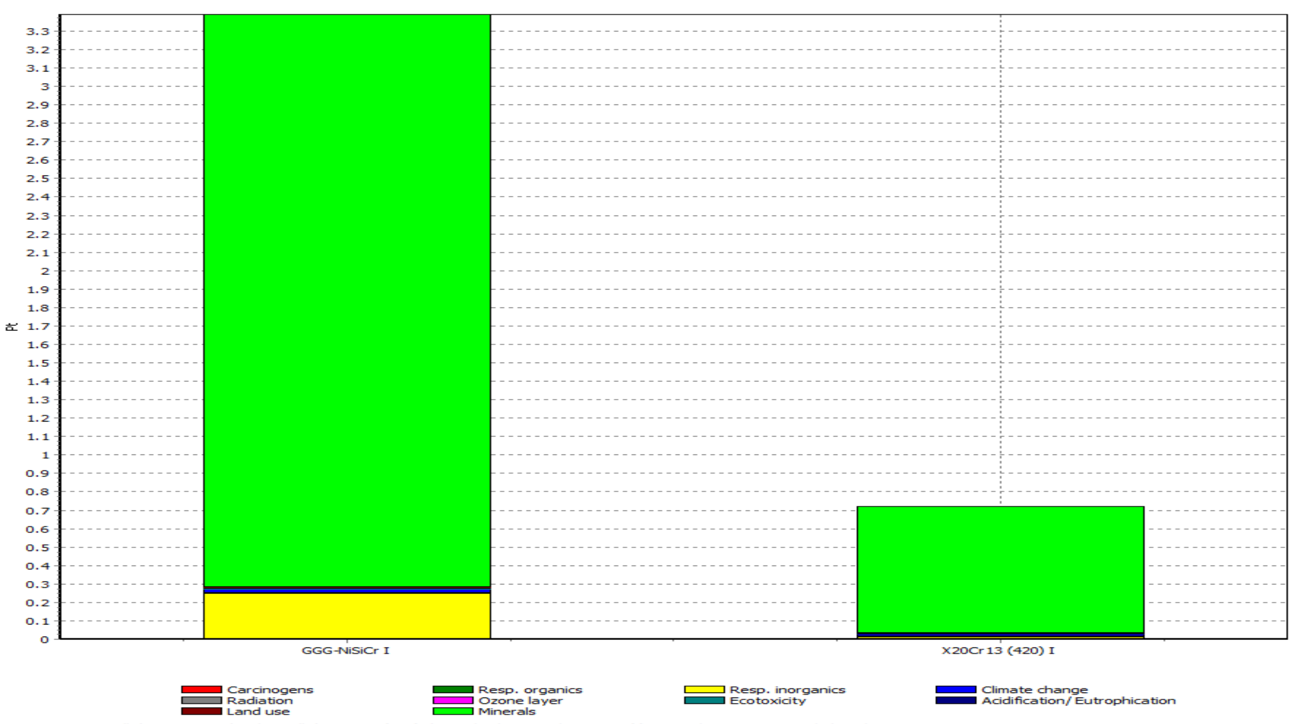
Carcinogens
Resp. organics
Resp. inorganics
Climate change
Radiation
Ozone layer
Ecotoxicity
Acidification / Eutrophicati
Land use
Minerals
GGG-NiSiCr I
X20Cr13 (420) I
Comparing 1.27 lb 'GGG-NiSiCr I' with 1.4 lb 'X20Cr13 (420) I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/I / normalization


Pt
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3
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GGG-NiSiCr I
X20Cr13 (420) I
Carcinogens
Resp. organics
Resp. inorganics
Climate change
Radiation
Ozone layer
Ecotoxicity
Acidification/ Eutrophication
Land use
Minerals
Comparing 1.27 lb 'GGG-NiSiCr I' with 1.4 lb 'X20Cr13 (420) I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/I / single score

37 APPENDIX M: Manufacturing Process Selection

The material selection for the HPSOV has been discussed previously (see "Material Selection Assignment"). Knowing which materials will be used, one can immediately narrow down the processes available for the manufacture of components made from these materials. There are other factors that must be considered in addition to the type of material. These include the production run, necessary production rate, the number of manufacturing operations required, etc. Since most of the selection will be based on it, perhaps the most reasonable factor to consider first is the number of units to be produced.

Should the HPSOV design be adopted, its immediate use will be in UPS delivery vehicles. It is this application for which the valve has been developed. With this in mind, one can make a rough estimate of the number of units that might be produced. According to the UPS website, there are approximately 94,000 delivery vehicles in operation. We shall assume that UPS decided to implement hybrid hydraulic technology into a majority of their delivery vehicles. We shall also assume that some other companies decided to explore the same technology, and that there is a small demand for the HPSOV in markets other than delivery vehicles. Therefore, the process selection will be determined based on a production run of 100,000 units. This should supply all necessary customers, as well as leave room for expansion if hybrid hydraulic vehicles further increase in popularity.

37.1 Valve Housing

With the production quantity determined, the next task was to begin filtering down the manufacturing processes available in the CES database [13]. Several things were input into a process filter in addition to the production quantity. First, the valve was assumed to be a circular prismatic, hollow 3-D shape which would first be manufactured by a primary shaping processes (finish machining on close tolerance surfaces will be considered later). Although the final design is not strictly circular prismatic, it will be assumed that it is compatible with this definition from a manufacturing standpoint. Next, the mass of the valve housing was used to eliminate manufacturing processes that would not be compatible with this size component. After these filters, 7 suitable processes remained in the database (see Fig. 37.1).

The primary manufacturing process was chosen to be high pressure die casting. As shown in Fig. 37.1, high pressure die casting is one of the lowest cost manufacturing processes that will work for the housing, while still providing the best tolerances possible from the alternatives.

While high pressure die casting has very good tolerances (down to 0.006 inch), finish machining is still required all surfaces which will locate moving components. A turning operation will be used to do all finish machining on the valve housing. The tolerances required for the locating surfaces (~±0.001 inch) are well within the capability of the turning process (0.0005-0.015 inch, see Fig. 37.2).

It is reasonable to assume that the stainless steel material will be purchased in the correct heat treat, therefore the manufacturer will not be responsible for performing this process and it will not be investigated further. The cost of this process is included in the material price, which is something that has already been optimized during material selection.

Figure 37.1: High pressure die casting was selected as the primary manufacturing processes for the valve housing

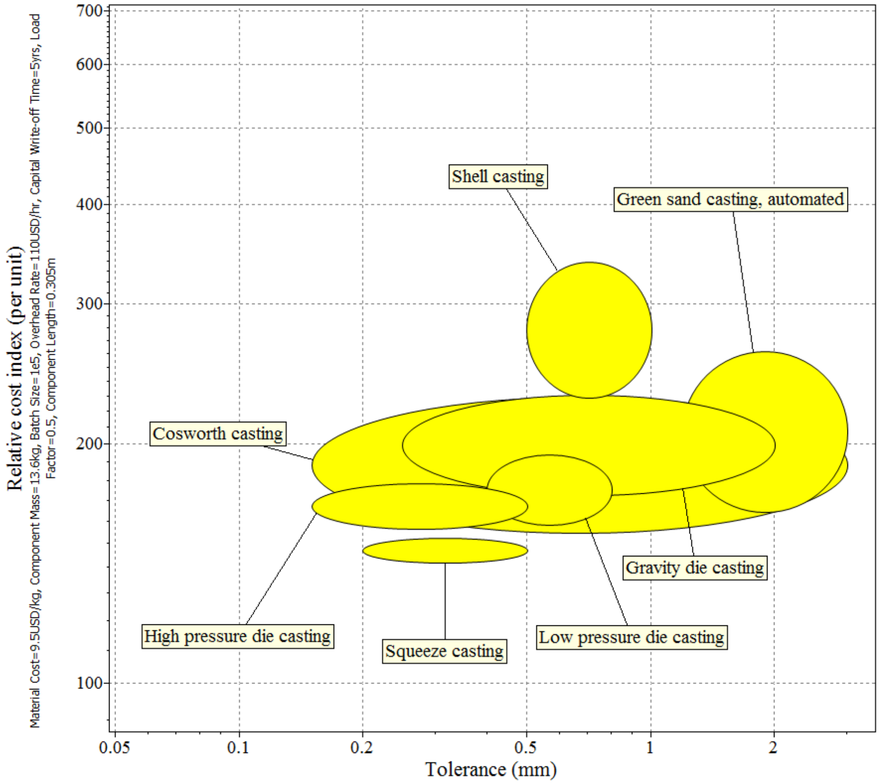
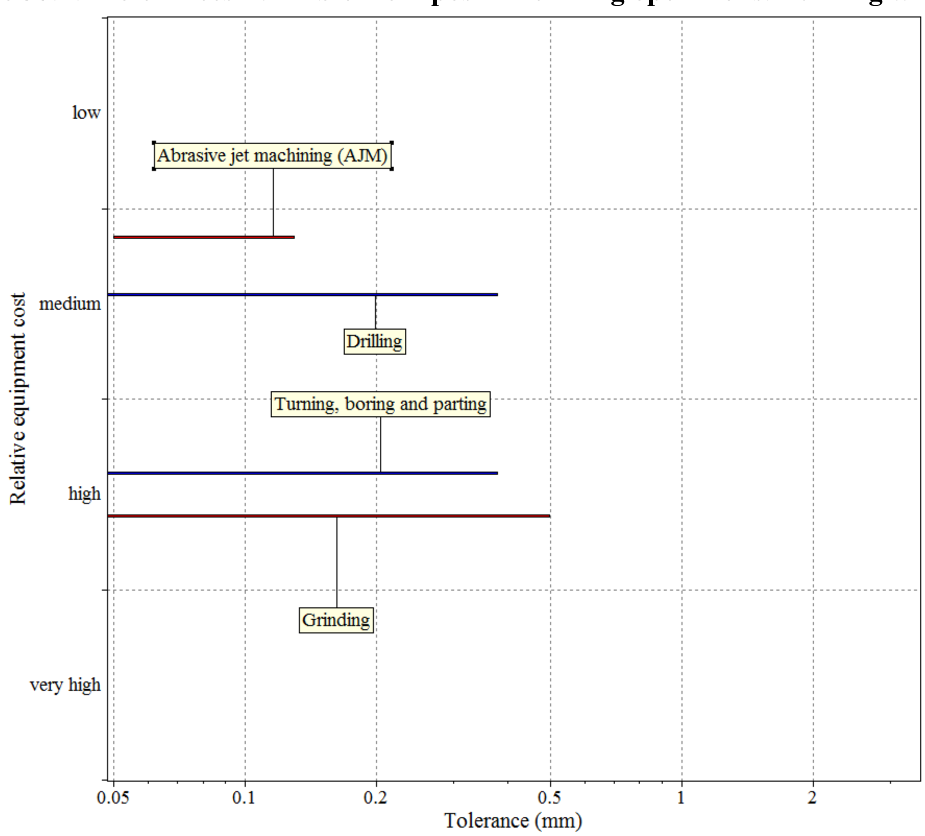


Figure 37.2: Tolerances available from post-machining operations. Turning was selected



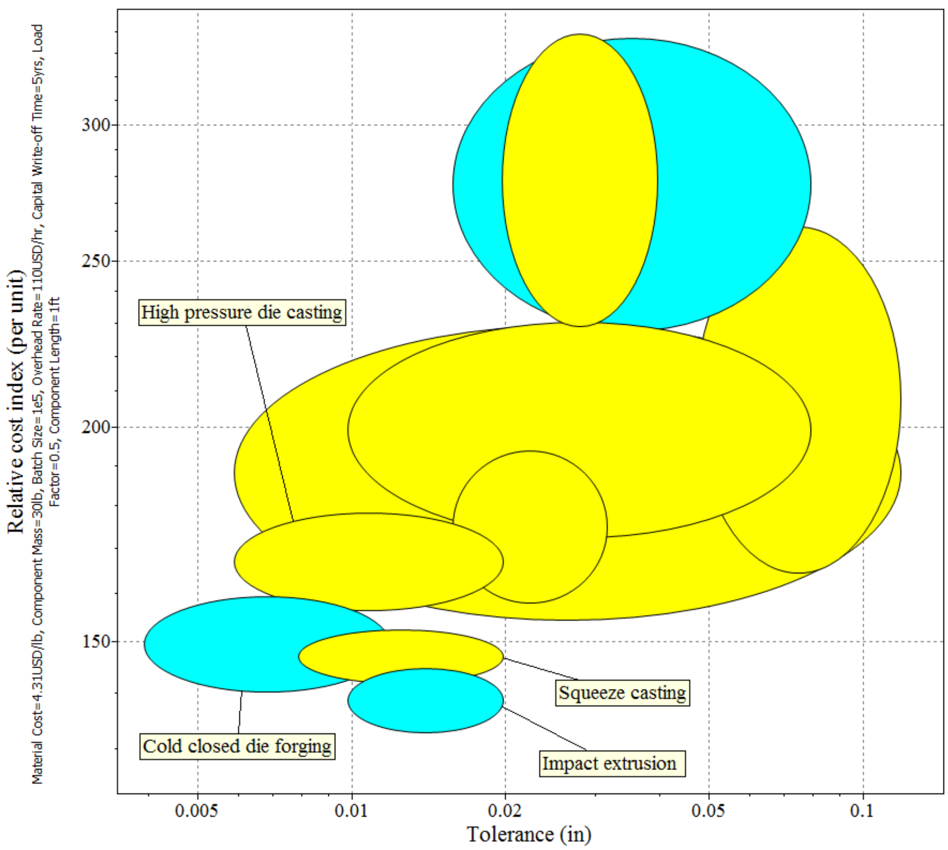
37.2 Valve Stem

There is one valve stem required per HPSOV, therefore the production quantity will be the same. Again, the geometry will be circular prismatic. The valve stem itself has a constant circular cross-section, followed by a valve head of specific geometry on the end. The entire part has a hole through the center which will allow access to the high pressure fluid accumulator during operation. The production quantity was initially used, but later eliminated from the process filter due to elimination of valid processes from the filter results. These processes had no production quantity data, therefore they were eliminated unnecessarily by the software.

Cold closed die forging is the selected manufacturing process for the valve stem. This process provides the lowest cost available for the required quantity and geometry. The other processes of similar cost do not allow the valve head to be an integral part of the valve, therefore they were not chosen (impact extrusion, for example, would require further machining after the primary process). The available processes are plotted in Fig. 37.3. The primary difference between the primary process criteria for the valve housing and the valve stem (Fig. 37.1 and Fig. 37.3) is that the valve stem is not a hollow 3-D section. Without this requirement, several other processes become available.

Once the manufacturing process was selected, the required mass and quantity of the valve stem was checked with the specifications of the cold closed die forging process. The process is compatible with the number and size of valve stems necessary.

Figure 37.3: Cold closed die forging was selected as the primary manufacturing processes for the valve stem



After the primary shaping process, a hole will need to be machined through the entire length of the valve stem. Since the tolerances on the through hole are high, a simple drilling operation would be sufficient. Once the hole has been drilled through the center, both ends of the valve stem must be finish machined to an outer diameter suitable for the ½-20 UNF thread form. The threads can then be cut into the valve stem using a turning process. All these operations can be simultaneously handled on a lathe.

38 APPENDIX N: Prototype bill of materials

Part #	Manufacturer	Price in US dollars	Part Name	Qty
A5-568B-224	Apple Rubber Products Inc.	quote & samples requested	O-ring	1
A5-568B-040	Apple Rubber Products Inc.	quote & samples requested	O-ring	3
A5-568B-039	Apple Rubber Products Inc.	quote & samples requested	O-ring	1
A5-568B-032	Apple Rubber Products Inc.	quote & samples requested	O-ring	2
8628K27	McMaster-Carr	10.36	Valve Stem	1
8745K64	McMaster-Carr	39.30	Stem Holder, Stem Head, & Power Piston	1
8528K48	McMaster-Carr	227.75	Valve Housing	1
5116K89	McMaster-Carr	3.59/ 10 parts	Barbed 1/2" to 1/2" threaded	1
5116K38	McMaster-Carr	4.93/ 10 parts	Barbed 1/2" fitting Tees	1
4796K75	McMaster-Carr	11.30	Ball Valves	5
52375K14	McMaster-Carr	.96/ foot	Tubing	20
NA	NA	NA	Pressure gauge	3
NA	NA	NA	Air Pump w/ Schrader fitting	1
NA	NA	NA	Flow Meter	1
NA	NA	NA	Plastic Tub	1
SD 330 -20A1/172 S - 210	HYDAC	NA	High Pressure Accumulator	2

[Continued on next page]

Material	Color/Finish	Size (inches)	Mass	Manuf. Process	Function
Buna-N (Nitrile) (NBR)	black	ID = 2	(gms)	none	accumulator inlet sealing
Buna-N (Nitrile) (NBR)	black	ID = 2 7/8		none	power piston & end cap sealing
Buna-N (Nitrile) (NBR)	black	ID = 2 3/4		none	power piston sealing
Buna-N (Nitrile) (NBR)	black	ID = 1 7/8		none	valve stem holder sealing
Nylon 6/6	white	OD = 1/2, ID = 1/4		Cut, add threads	valve stem
PVC Type 1	grey	OD = 3.25, L = 24		Lathe, Drill Press	valve components
Aluminum	grey	OD = 5, L = 12		Lathe, Drill Press	valve housing
Nylon	black	.5 to .5 connections		none	attach tubing to housing
Nylon	black	0.5 connections		none	splits flow for experimental setup
Nylon	black	.5 connections		none	controls for expo system
PVC	clear	OD = .75, ID = .5		Cut to Length	controlled flow
NA	NA			none	valve function verification
NA	NA	Schrader Fitting		none	setup manipulation/demonstration
NA	NA	fits .5 tube		none	show flow
NA	NA	5-10 gallons		none	low pressure reservoir
NA	NA	5 gallons, 2-12UN-2A connection		none	borrowed from EPA sponsor

39 APPENDIX O: Test apparatus operating instructions

Our prototype will clearly demonstrate that the final design can perform the following functions: closing when commanded closed, equalizing pressure in the line with the pressure in the accumulator and opening when commanded open, leaking pressure into the accumulator when commanded closed and pressure in the line exceeds the pressure in the accumulator, closing when commanded open but the bladder reaches the valve opening, being able to fit into the current EPA hydraulic system, and being of a less complicated design than the existing EPA high pressure accumulator valve. To show that our final design valve will function, our prototype will be built to the same interior dimensions (piston sizing and acting areas) as the final design. Although the working pressures themselves will be different, they will act on the same surfaces. It can be assumed that if the valve actuates at these low pressures, it should only actuate better at the higher pressures used in the EPA system.

39.1 Closing When Commanded Closed

Setup: the simulated low pressure accumulator will be at atmospheric pressure which will be less than the simulated line pressure which will be less than the simulated high pressure. The valve controls will be set such that high pressure will be routed to the actuating volume, and nothing will be routed to the line pressure. The valve will close, and remain closed. This will be visible through the clear valve housing, and the pressures will be visible via pressure gauges attached to the simulated high pressure accumulator and line pressure. Valve control signals will be easily seen by looking at the position of the simple ball valves used to simulate control signals. To verify that the flow rate out of the accumulator is in fact zero, we will have a simple flow-meter in the line immediately after the valve.

39.2 Opening & Pressure Equalization.

To show that our final design valve will equalize the line and high pressure accumulator pressures before opening and then remain open when commanded to open we will monitor the both the pressure in the line, and in the tank before and during the operation. The operation will consist of switching the ball valves that control the direction of pressure within the valve so that high pressure from the accumulator is directed through a small opening into the line, and low pressure from the simulated low pressure accumulator is directed to the acting volume. The power piston within the valve should act before the valve stem moves, this is because the power valve stem is detached from the power piston, and the spring acting on the valve stem will not allow the valve to open until the pressure difference is less than 7.5 psi. This will be shown by being able to see the valve stem and power piston positions through the clear valve housing. The pressures in the line and high pressure accumulator can be monitor via the pressure gauges and the flow can be seen on the installed flow meter. Once the valve is open, it can be demonstrated that the valve remains open simply by watching the valve stem position.

39.3 One Way Flow

To show this feature of our valve design we will again monitor the valve position through the clear housing. We will add pressure to the line and monitor the gauges of the line and high pressure accumulator. As the pressure in the line begins to exceed the pressure in the high pressure accumulator, the valve should open and allow this excess pressure into the accumulator. This will be shown both the valve position (seen through the valve housing), pressure difference (displayed on the gauges), and fluid flow (shown by reading the flow meter and demonstrating fluid is flowing into the high pressure accumulator).

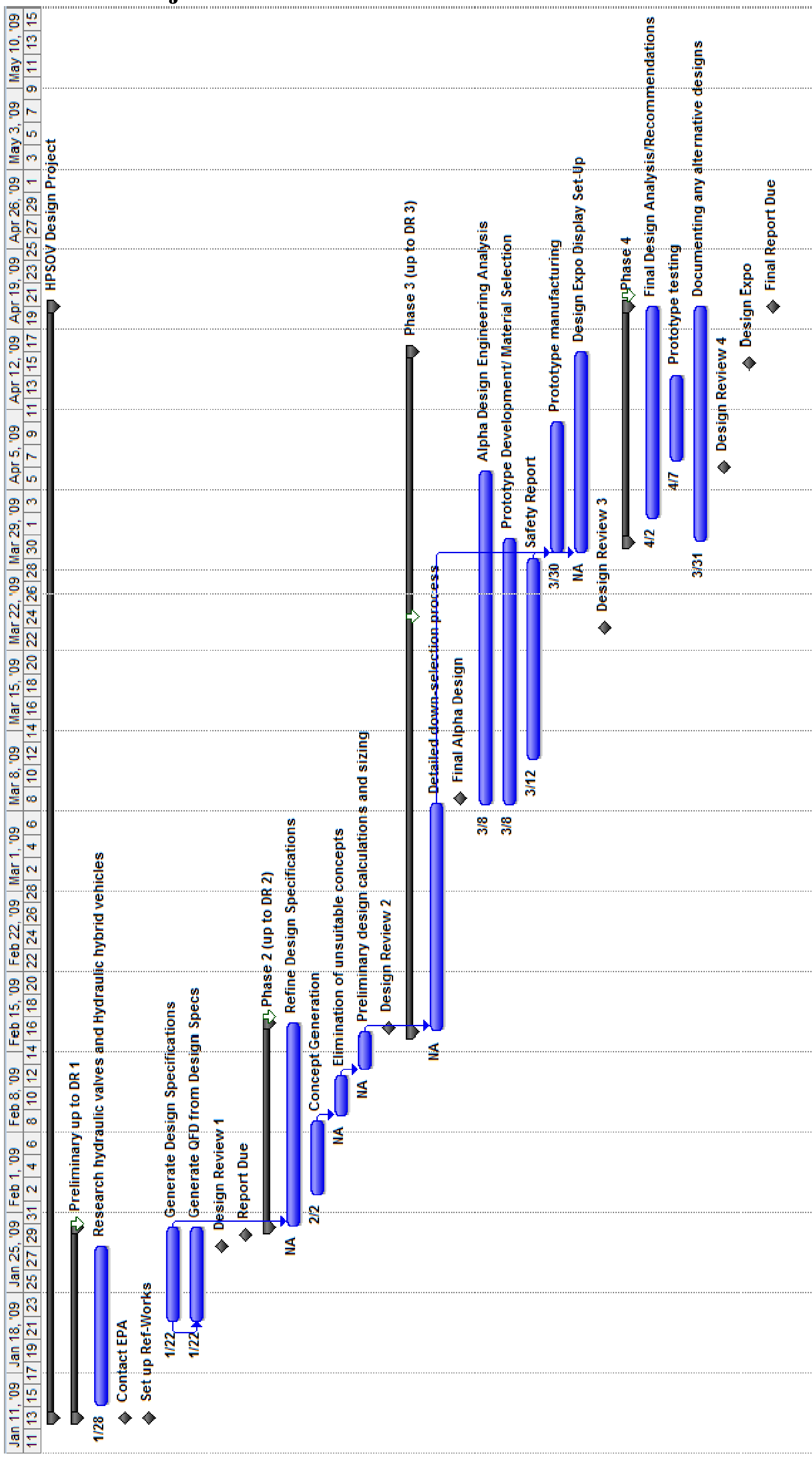
39.4 Bladder Interaction

To demonstrate that our valve will close before allowing the bladder to extrude we will allow all fluid to exit the high pressure accumulator by switching the valve to the open position and draining the fluid into the simulated low pressure accumulator. When the high pressure accumulator is empty, the bladder will try to extrude around the top of the valve stem, which will not be visible since this part of the valve will be contained within the accumulator, which is opaque. Instead, to see the interaction we will have to watch the valve stem position, which will be visible as the valve stem will be painted or marked so that we can see externally what position it is in. By this we will be able to see that the valve stem closes, which will only be the result of the bladder pressing on the valve head.

39.5 Fitment into Current EPA Systems

In order to demonstrate that the final design will fit into the current EPA hydraulic system we will be using the same accumulator fittings for our prototype that they use for their hydraulic hybrid systems. In this way we can demonstrate that what we construct fits the accumulator, and that our final design has the same accumulator fitting designs. The other side of the valve, (the side to which the line is attached) has a set of exact specifications that we have included in our final design, but have not demonstrated with the prototype. We also do not plan on showing the use of a switching mechanism that can operate at a range of 10 to 12 volts and defaults to closed, although we have chosen such a device for our final design. It would be too complicated and expensive to include such a device in our prototype, and does not help us to demonstrate the main functionality that we have chosen to display.

40 APPENDIX P: Project Gantt Chart



41 APPENDIX Q: Complete Safety Report

EXECUTIVE SUMMARY

The following safety report describes the manufacture of components for a high pressure hydraulic shutoff valve. The shutoff valve is designed to perform a number of fluid control functions within a hydraulic system. In total, the valve consists of the valve housing, a power piston, a stem holder, and a valve stem. The valve stem consists of two components: the stem and the valve head.

The valve housing consists of an aluminum shell that contains the power piston, stem holder, and valve stem components of the valve. On one side of the valve housing, the housing end cap is attached, which attaches to the hydraulic line in the hydraulic hybrid engine system. In the prototype, the line consists of an accumulator that allows for changing the line pressure and PVC tubing. On the other side, the housing attaches to a second accumulator, which stores the hydraulic energy in the hydraulic hybrid system. In the prototype, the high pressure accumulator will have a maximum pressure of 150 psi. The hydraulic housing will be made out of 6061 aluminum and will be threaded into the accumulator on one end, and bolted to the housing end cap on the other end..The wall of the valve housing will also contain threaded ports to connect to the control system for the valve. More information on this control system can be found in the validation section of this report. The modes of failure for the valve housing are: shear through end cap bolting, shear through accumulator threading, rupture due to internal pressure, and tensile yield through area. The smallest factor of safety on this housing geometry is about 35.

The valve end cap is also made of 6061 aluminum, and is bolted into the side walls of the valve housing via 8 equally spaced bolts. Also a part of this end cap is a threaded ½ inch connection to the line. The end cap geometry also retains the spring internal to the valve from displacing off of center. The end cap also has a bore type seal in to prevent fluid from escaping the valve housing. The only mode of failure for this part would be due to shear through the end cap thickness, which has a factor of safety of 311 in this design.

The power piston is made from PVC type 1 and is the first piece internal to the valve housing. The power piston has full bore type seals at three different locations. The outer geometry of the power piston is designed for the application of high and low pressure to specific geometries to activate the power piston to the open and closed positions. The three ports in the valve housing leads to the external geometries of the power piston. A series of holes through the power piston lead from the external geometry furthest from the accumulator that lead straight to the line. Another series of holes through the power piston lead from a sealed geometry in the stem holder to the middle housing port. The modes of failure for the power piston are: shear through smallest section, shear through sealing geometries, and tension at sealing geometry due to moments. The smallest factor of safety on this geometry is about 82.

The stem holder is made from PVC type 1 and is designed to work inside of the valve housing. This component will have a threaded end to which the valve stem will be attached. The valve stem's center bore will then be ported to the outside of the stem holder through a hole in the center cavity. The outside of the stem holder will have a sealed geometry that leads to a series of holes in the power piston. The stem holder also has three pathways designed to allow fluid flow through it and into or out of the accumulator. In addition, the stem holder is designed to retain the spring in the center of the valve. The modes of failure for the stem holder are: shear through the stem threading, shear through smallest section, shear through sealing geometries, and tension at sealing geometry due to moments. The smallest factor of safety on this geometry is about 16.
The valve stem is made from Nylon 6/6 tube with ½ inch outer diameter and ¼ inch inner diameter. The center bore in the valve stem leads from the accumulator to the inside of the stem holder, which eventually finds its way out of the housing through the center port. The valve stem is threaded on both

ends to the stem holder and to the valve head. The modes of failure for the stem holder are: shear through threading, tensile yield through the stem area, and rupture due to internal pressure. The smallest factor of safety on this geometry is about 1.7.

The valve head is made from PVC type 1 and is made to travel in the accumulator. The head consists of a neck that has internal threads to connect to the stem and a flat head with a small hole to allow for access between the accumulator and the inner bore of the stem. The flat head is made wide to allow for a large area that the bladder can press against in order to close the valve. The modes of failure for the stem head are: shear through threading, shear through smallest section, tensile yield through the stem area, and tension through section due to moments. The smallest factor of safety on this geometry is about 8.3.

The maximum working fluid pressure was assumed to be 150 psi, which will generate a maximum axial load of 403 lbf that will need to be resisted by the stem holder and valve stem assembly. As the components are assembled in series, each component will see this load. Using this pressure as the design point significantly reduces any risk of failure as our system will actually be tested at a lower pressure using simple manual shut-off valves for control functions. The use of manual valves will simplify experimental testing and reduce the likelihood of failure by eliminating complex control devices. A Designsafe report for the each component can be found in Appendix A.

Manufacturing Hazards:

For all of the parts listed in this report, the most important manufacturing process that will be used is manual turning. All of the basic external and internal geometries of all parts will be made on a lathe. The stock 3 inch diameter PVC rod is turned down to multiple diameters to create the critical external geometries as well as the external sealing geometries on the, power piston, stem holder, and the valve head. The stock 5 inch diameter aluminum rod is also turned down to create the geometry of the end cap and the part of the housing that fits into the accumulator. Also on the lathe will be multiple boring operations. The internal geometries of the power piston and housing will be bored out. In addition to boring, multiple drilling processes are done on the stem holder, valve head, and housing end cap.

The second machine that will be used is a mill. In order to accurately shave off faces of turned pieces as well as accurately drill the through holes in the stem holder, a mill will be used. The 8 bolt holes that exist through the end cap and the housing may be done on the mill.

The next machine that will be used is a drill press. The side holes in the power piston, the stem holder, and the front hole in the valve head will be done on a drill press. The holes through the end cap and off of the holes on the outside of the housing (11 total holes on the housing) may be done on a drill press as well.

Another process that will be used is tapping to create threading. Threading will be used to connect the stem holder to the stem and the stem head to the stem. Another tap will need to be used to make the threading on the outside of the housing that threads into the accumulator. In addition, in both the housing and end cap, ½ inch threading will be used on all of the port holes (4 total).
All of these processes possess the standard safety hazards that will be addressed and reduced through the thorough understanding and implementation of the machine shop training rules and regulations. If any additional assistance is required, the machine shop instructor will be consulted before machining progress resumes.

1. EXPERIMENTATION PLANS PRIOR TO DESIGN COMPLETION

The only experimental testing that is planned will be for the prototype. There will be no experiments run on any component before the prototype is finished and completely assembled. Once the prototype is finished, testing will be done to simulate the operating conditions that will be seen in service and verify that the valve functions correctly under all operating modes. By validating the successful operation of the prototype valve, we simultaneously validate the operation of the final design valve since they will both operate on the same geometries and principles.
The main data that will be collected during experimental testing is the valve position. The valve position will give us enough information to verify correct functionality under all the operating conditions being tested. While this data will not give specific performance information like pressure drop or reaction time, it will valid the valve concept so that other testing can address these aspects, should this valve design be produced.

In addition to valve position, operating pressures of the main valve outlet line, high pressure accumulator, and simulated line pressure will also be monitored. While pressure measurements at different locations could give an indication of valve pressure drop performance, these measurements are not intended for this purpose. Instead, the pressure measurements will be used to ensure that safe operating limits (namely pressures) are not exceeded. Specifically, the pressure of both accumulators will be monitored during filling to ensure they do not exceed the levels the prototype can handle safely, as determined in the engineering analysis section.

There are several safety risks inherent in the experimental testing. Perhaps the largest risk is the rapid release of stored energy from the accumulators. Each accumulator will be pressurized with air to a specific level. If there is a failure of a valve component that maintains sealing of fluid lines, a rapid release of pressure will result. Likely, this will propel water to areas surrounding the test rig. Therefore, to minimize the risk of damage to people or other objects from this type of failure, an enclosure will be placed around critical test rig components, namely the HPSOV. In the event of a failure, the enclosure will prevent water from reaching any of the surroundings. Accumulators were chosen with a minimum volume so that, if a leak does occur, there will be minimal fluid spray before all pressurizing energy is released.

Other risks inherent to the experimental testing are those that are encountered in any type of manufacturing or assembly. Minor assembly tools and equipment will be used to build the experimental setup, and the necessary safety precautions will be taken when using these. The tools and equipment include hand saws for cutting hoses and mounting devices, as well as screwdrivers and wrenches for attaching hose clamps and fittings.

Despite the safety risks, the experimental testing is necessary to prove a functional valve concept and will therefore be undertaken. Particular attention to the risks presented above will be paid during the experiment to ensure a safe test.

2. PURCHASED COMPONENT AND MATERIAL INVENTORY

Part #	Manufacturer	Price in US dollars	Part Name	Qty
A5-568B-224	Apple Rubber Products Inc.	quote & samples requested	O-ring	1
A5-568B-040	Apple Rubber Products Inc.	quote & samples requested	O-ring	3
A5-568B-039	Apple Rubber Products Inc.	quote & samples requested	O-ring	1
A5-568B-032	Apple Rubber Products Inc.	quote & samples requested	O-ring	2
8628K27	McMaster-Carr	10.36	Valve Stem	1
8745K64	McMaster-Carr	39.30	Stem Holder, Stem Head, & Power Piston	1
8528K48	McMaster-Carr	227.75	Valve Housing	1
5116K89	McMaster-Carr	3.59/ 10 parts	Barbed 1/2" to 1/2" threaded	1
5116K38	McMaster-Carr	4.93/ 10 parts	Barbed 1/2" fitting Tees	1
4796K75	McMaster-Carr	11.30	Ball Valves	5
52375K14	McMaster-Carr	.96/ foot	Tubing	20
NA	NA	NA	Pressure gauge	3
NA	NA	NA	Air Pump w/ Schrader fitting	1
NA	NA	NA	Flow Meter	1
NA	NA	NA	Plastic Tub	1
SD 330 -20A1/172 S - 210	HYDAC	NA	High Pressure Accumulator	2

[Continued on next page]

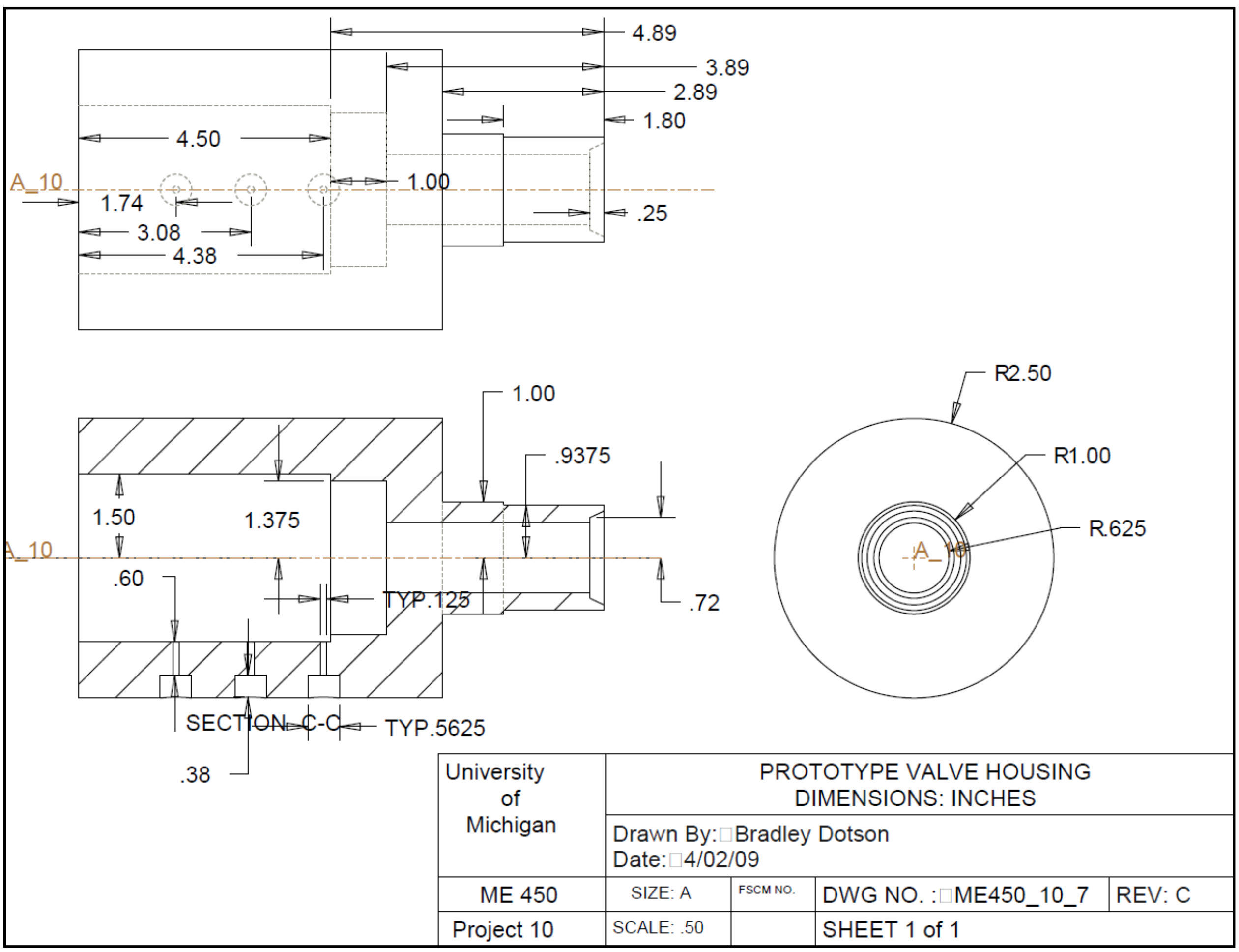
Material	Color/Finish	Size (inches)	Mass	Manuf. Process	Function
Buna-N (Nitrile) (NBR)	black	ID = 2	(gms)	none	accumulator inlet sealing
Buna-N (Nitrile) (NBR)	black	ID = 2 7/8		none	power piston & end cap sealing
Buna-N (Nitrile) (NBR)	black	ID = 2 3/4		none	power piston sealing
Buna-N (Nitrile) (NBR)	black	ID = 1 7/8		none	valve stem holder sealing
Nylon 6/6	white	OD = 1/2, ID = 1/4		Cut, add threads	valve stem
PVC Type 1	grey	OD = 3.25, L = 24		Lathe, Drill Press	valve components
Aluminum	grey	OD = 5, L = 12		Lathe, Drill Press	valve housing
Nylon	black	.5 to .5 connections		none	attach tubing to housing
Nylon	black	0.5 connections		none	splits flow for experimental setup
Nylon	black	.5 connections		none	controls for expo system
PVC	clear	OD = .75, ID = .5		Cut to Length	controlled flow
NA	NA			none	valve function verification setup
NA	NA	Schrader Fitting		none	manipulation/demonstration
NA	NA	fits .5 tube		none	show flow
NA	NA	5-10 gallons		none	low pressure reservoir
NA	NA	5 gallons, 2-12UN-2A connection		none	borrowed from EPA sponsor

3. CAD DRAWINGS AND DESIGNSAFE SUMMARY FOR DESIGNED PARTS

Presented in this section are engineering drawings for all the valve components that will be manufactured by the team. In addition to these drawings, detailed DesignSafe reports were created for each valve component, the fully assembled valve, and the display expo set-up including the valve. These DesignSafe reports can be found in Appendix A.

The manufacturing process as well as testing and demonstration were taken into account when using DesignSafe. The most serious risks found occurred during the machining process. As with any machining process, cutting, turning, milling, and drilling all present inherent safety risks. Specifically, while machining the components there are risks of cutting or severing, entangling, pinching, or loose debris. These risks were analyzed and discussed with DesignSafe and it was determined that following the proper machine shop safety procedures along with consultation from machine shop staff would help to reduce these potential hazards. Determining proper machine and tool speeds and feeds will also help to reduce these hazards.

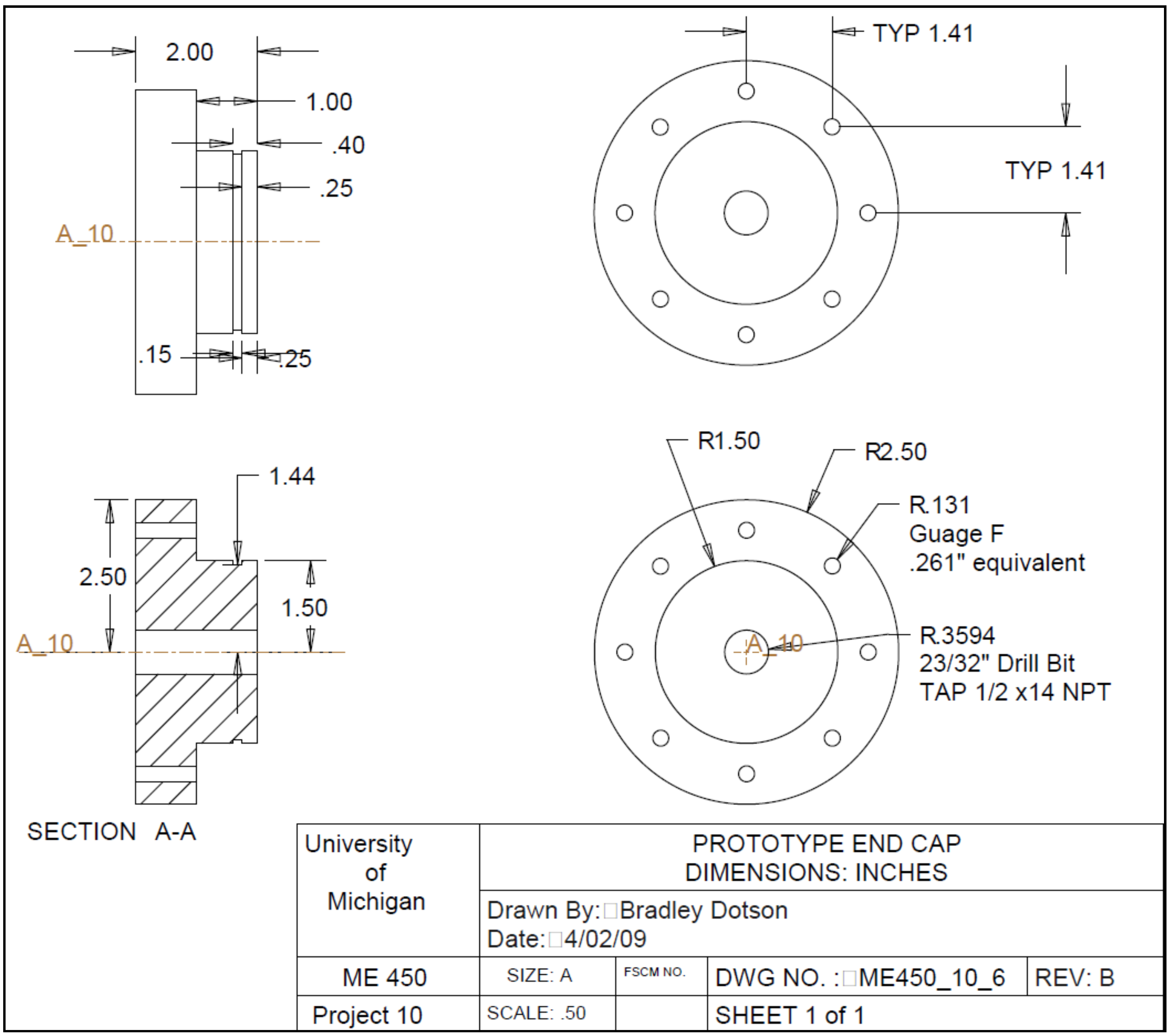
3.1 Valve Housing



The valve housing and end cap require the largest amount of material removal of all the valve components. Because of this, considerable machine time will be required for this component. From the standpoint of safety, this can be an issue due to an excessive accumulation of chips around the workplace, or operator fatigue. To minimize these risks, the valve housing will likely be machined in stages over multiple days. The break time in between machining will help address both these issues.

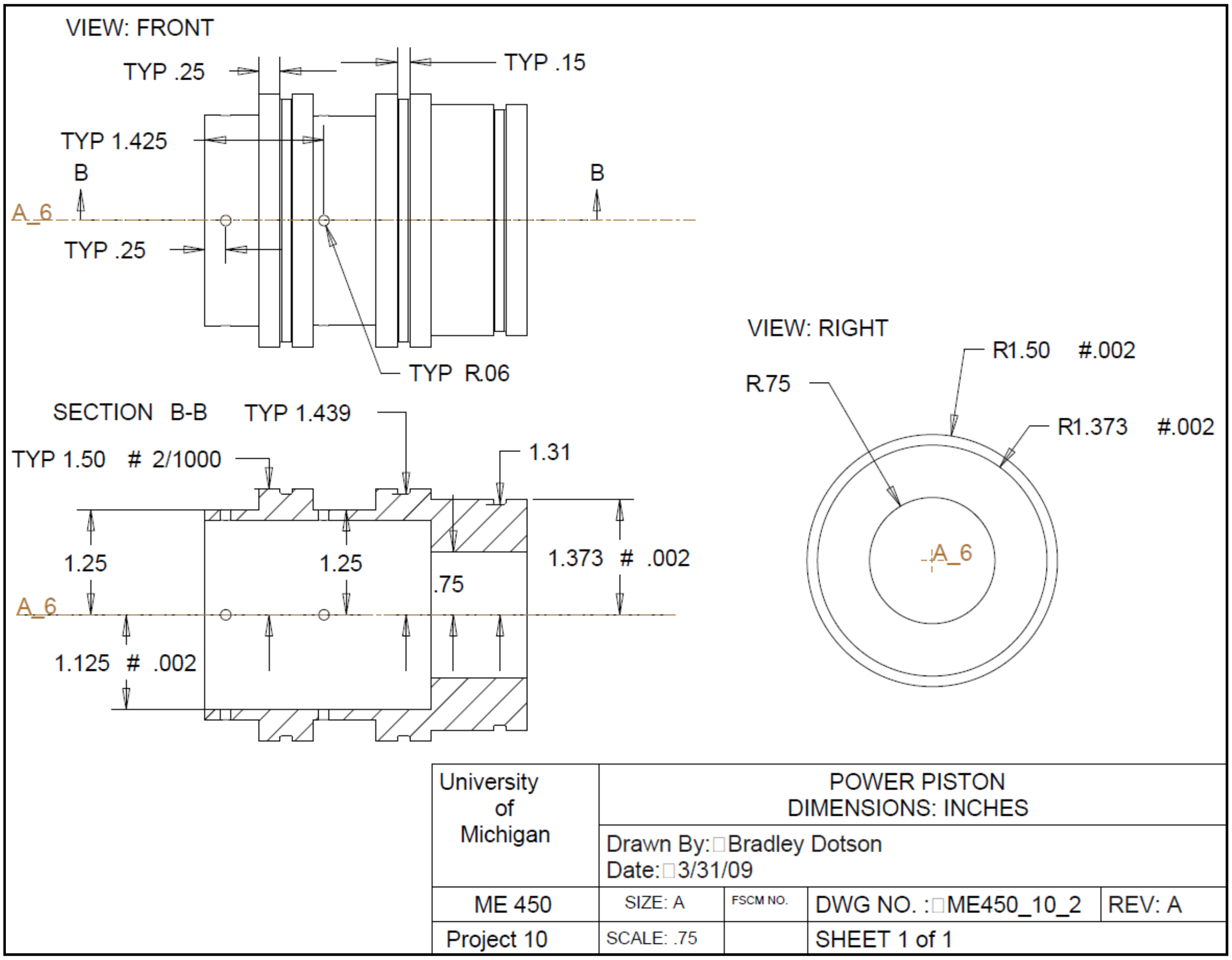
During operation and testing, however, the valve housingand the valve end cap have the largest factor of safety on failure. Considering the large safety factors on all the other components in addition to the low operating pressures, the risk of failure of the valve housing or the end cap will be minimal provided it has been properly manufactured according to specifications.

3.2 Valve Housing End Cap



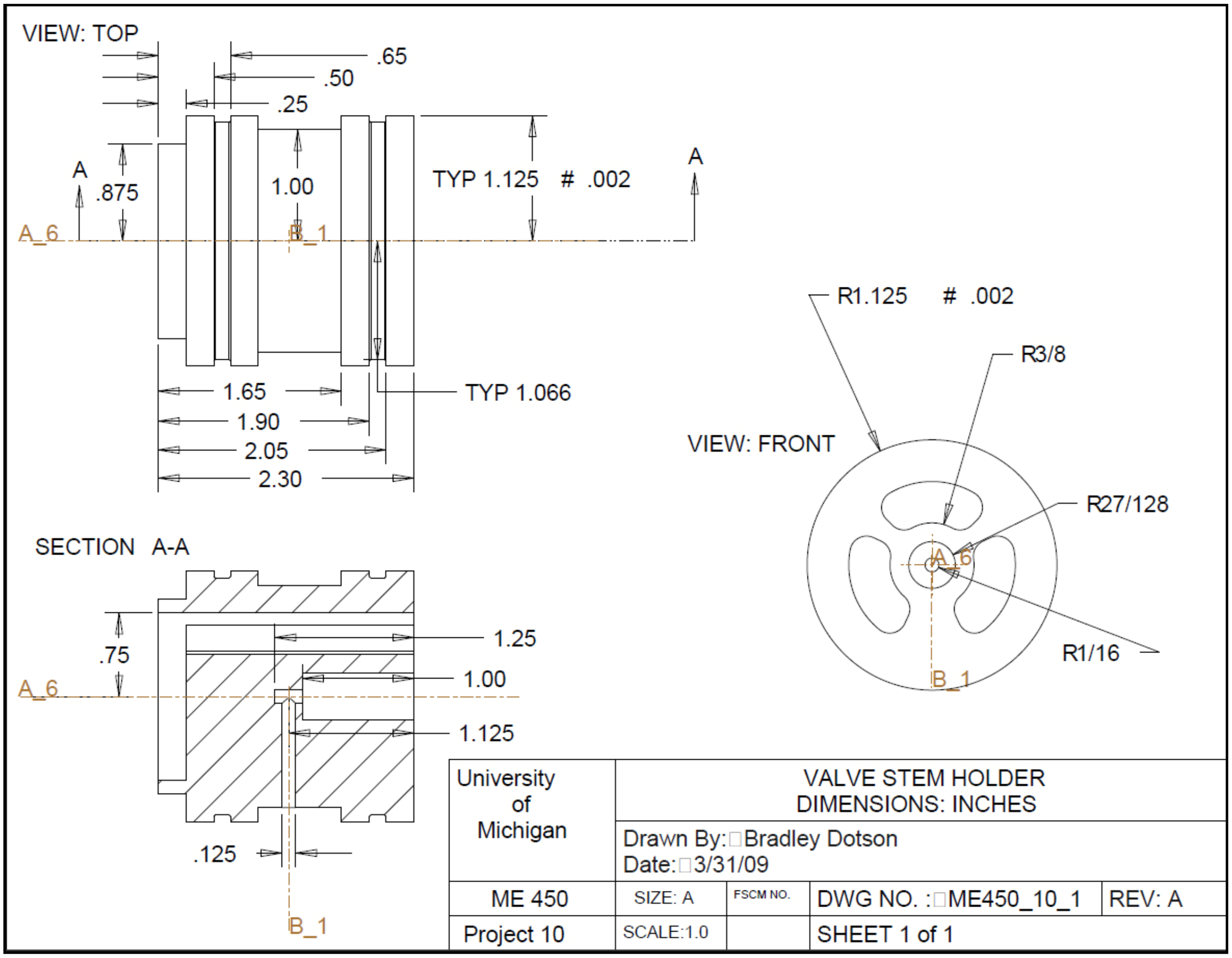
3.3 Power Piston

The power piston was analyzed by DesignSafe for use during testing and during design expo demonstration. Because water will be running through and around this component at pressure, albeit low, the risk of fluid leakage or rupture had to be addressed. If pressure, in fact, did build up and cause a rupture in the system it could potentially release pressurized water and valve fragments into the valve housing or the accumulator. These risks were deemed moderate as the low pressure involved is unlikely to cause a separation of the valve stem head from valve stem, but the risk was still reduced through the use of large safety factors when analyzing material strength. Great care will also be taken to ensure that the valve components and display set-up are assembled precisely and safely. The power piston is a dynamic component, and because of this, risks failure due to general wear and tear, degradation of materials, and friction/interaction with other components. Again, to reduce this risk, strength calculations were done and the components were analyzed with large safety factors included.



3.4 Valve Stem Holder

The valve stem holder is a dynamic component, and because of this it runs the risk of failure due to general wear and tear and degradation of materials. Again, to reduce this risk, strength calculations were done and the components analyzed with large safety factors included.

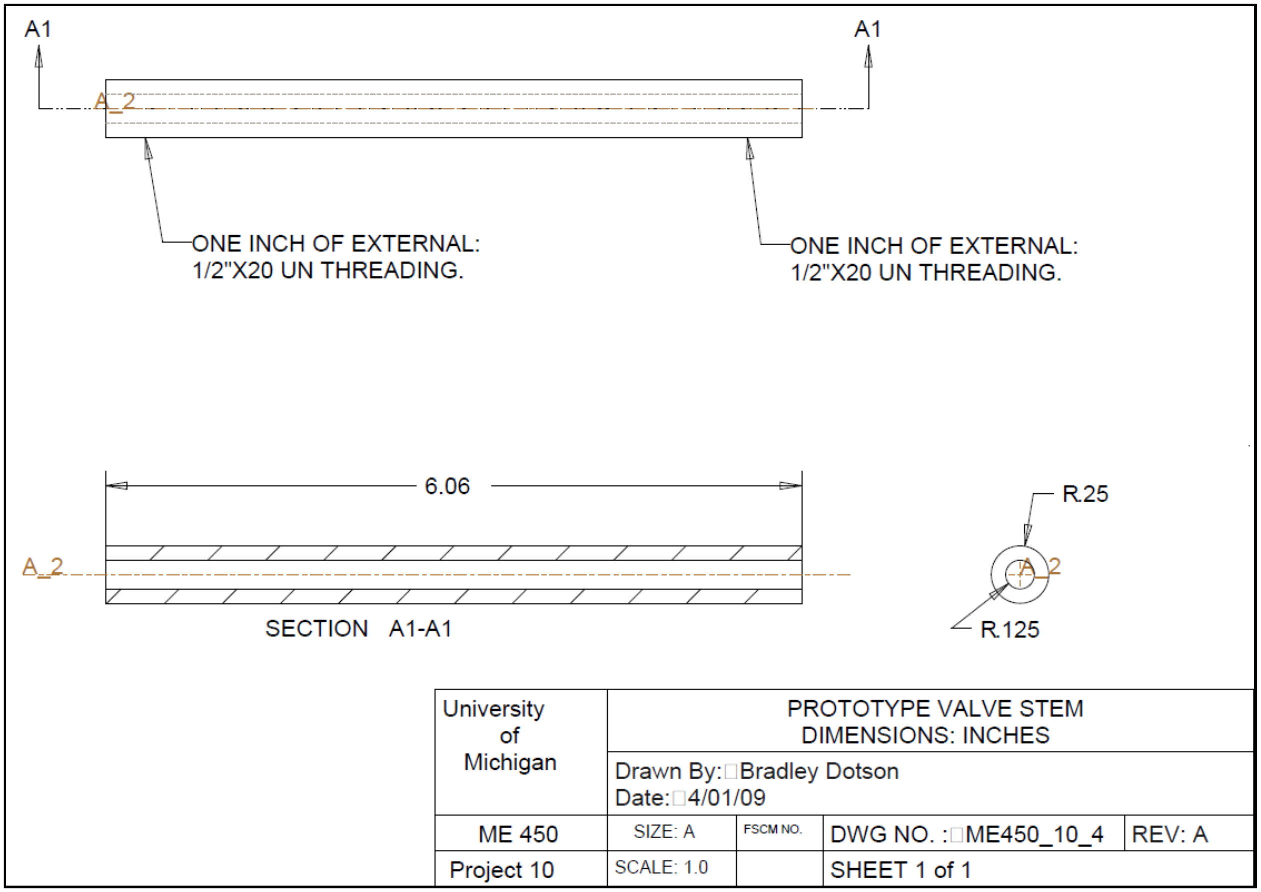


3.5 Poppet Valve Assembly

The poppet valve is made of two components. Each component is discussed below.

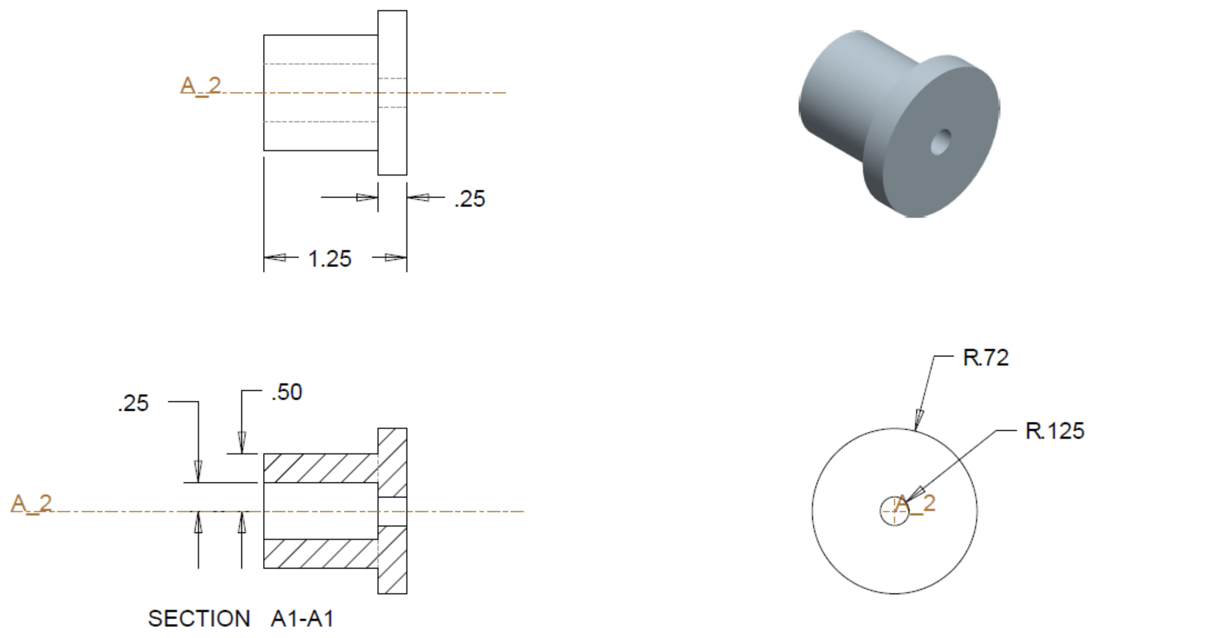
3.5.1 Valve Stem

The valve stem largely consists of stock material. The only modifications to suit the HPSOV are cutting the material to length, and threading each end. Of course, there are safety issues concerning the manufacturing process. Since all manufacturing of the valve stem will be done using hand tools, the main issues include the safe operation of these tools. During operation the valve stem will be inaccessible to the operator, therefore operational safety is dictated entirely by component strength. This component does not expose the operator to any specific hazards during HPSOV operation.



3.5.2 Valve Head

The valve head will be manufactured separate from the valve steam and attached via a threaded connection. The main safety hazard concerning the valve head is the threaded connection, and this is discussed in the analysis section. Like the valve stem, the valve head is inaccessible to the operator during valve operation, therefore operational safety is dictated entirely by component strength.



University of Michigan	PROTOTYPE VALVE STEM HEAD DIMENSIONS: INCHES			
	Drawn By: Bradley Dotson Date: 3/31/09			
ME 450	SIZE: A	FSCM NO.	DWG NO. : ME450_10_5	REV: A
Project 10	SCALE: 1.0		SHEET 1 of 1	

4. ENGINEERING ANALYSIS

In the following section, the engineering analysis that went into the design of each component will be presented. The main failure modes and their prevention will be described in detail.

4.1 Valve Housing

Housing Failure modes:

1. Shear through threading in bolts
2. Shear through threading in accumulator
3. Rupture due to internal pressure
4. Tensile yield through area

4.1.1 Shear Through Bolt Threading

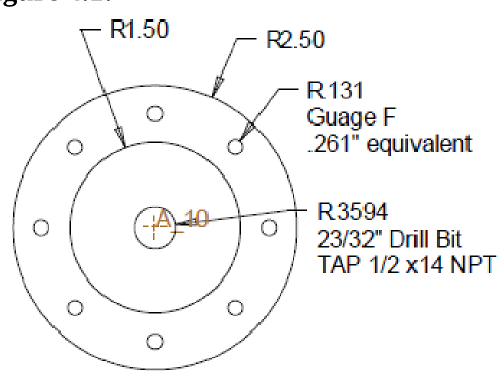
There are 8 bolts that hold the end cap of the housing onto the rest of the housing. The total force that the bolt threading sees that these bolts connect into is 1060.29 lbf. See section 4.2.1 for details on how this force is found. This force will be seen entirely by the eight Gauge F 0.261" equivalent bolts located as shown on Figure 4.1 (this figure shows the housing end cap geometry). These bolts will be made of steel and threaded into the aluminum housing for a full ½ inch engagement length. Table 4.1 shows the parameters that will be plugged into Equation 4.1 [1] in order to determine the shear area of each of these bolts (.2695 in^2). The total shear area is equal to 8 times that area, or 2.1559 in^2. Applying the total force to this shear area results in a shear stress of 491.8 psi. The yield strength of 6061 aluminum is 35,000 psi, so these threads will not fail. The factor of safety against this type of failure is 71.2.

$$A_n = \pi n L_e D_{s-min}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}(D_{s-min} - E_{n-max})\right] \quad \text{[Eq. 4.1]}$$

Table 4.1: Parameters used to find shear area of the end cap bolting

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	0.5 in
$D_s$min	Minimum major diameter of external thread	0.2408 in
$E_n$max	Maximum pitch diameter of internal thread	0.2224 in

Figure 4.1:



4.1.2 *Shear Through Threading in Accumulator*

The accumulator threading (2-12UN-2A), as well as the whole of the valve housing, will see the maximum closing force as well as the force on the end cap. This force is equal to 1463 lbf, where the closing force is equal to the force that is applied to the in-series assembly, which has a maximum of 403 lbf. Table 4.2 shows the parameters of the threading into the accumulator that are used to find the shear area. Using equation 4.2 [1], the shear area is equal to 3.6995 in^2. Using this shear area and a force of 1463 lbf, the total shear stress in the threading will be 395.5 psi. The yield strength of 6061 aluminum is 35000 psi, which gives a factor of safety of 88.5.

$$A_n = \pi n L_e K_{n-max}\left[\frac{1}{2n} + \frac{1}{\sqrt{3}}(E_{s-min} - K_{n-max})\right] \quad \text{[Eq. 4.2]}$$

Table 4.2: Parameters used to find shear area of the accumulator thread

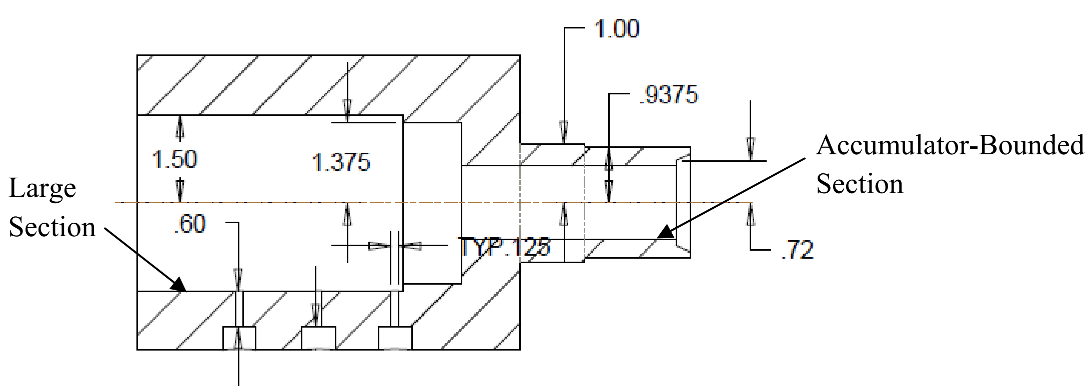
Parameter	Physical Meaning	Value
n	Threads per inch	12
L_e	Length of engagement	1.09 in
$K_n$max	Maximum minor diameter of internal thread	1.7848 in
$E_s$min	Minimum pitch diameter of external thread	1.8000 in

4.1.3 *Rupture Due to Internal Pressure*

Using equation 4.3, we can estimate the stress on the internal wall of two sections of the housing that are critical: the large section and the accumulator bounded section. This equation is taken from "Structural Mechanics of Buried Pipes" [3]. In this equation, P is the pressure inside the vessel, r_o is the outer radius of the vessel, r_i is the inner radius of the vessel, and S_I is the stress in the internal part of the vessel, which is the location of the maximum stress. The two sections that this equation applies to are shown in Figure 4.2. In the large section, the outer diameter is 5 inches and the inner diameter is 3 inches. The maximum pressure inside the valve is 150 lbf. The tensile stress due to this pressure in the large section is about 70.59 psi. In the accumulator-bounded section, the inner diameter is 1.25 inches and the outer diameter is 1.875 inches. The tensile stress due to the pressure in the accumulator-bounded section is 57.69 psi. The factors of safety for each of these sections will be discussed in section 4.1.4.

$$S_Y \geq S_I = \frac{P(r_o^2 - r_i^2)}{(r_o^2 + r_i^2)} \quad \text{[Eq. 4.3]}$$

Figure 4.2:



4.1.4 Tensile Yield Through Area

In addition to the tensile stress due to pressure on each area of the valve housing, a force of 1463 lbf is acting on these two sections. In the large section of the valve housing, the tensile area is 12.57 in^2, and the tensile stress is equal to 116.4 psi. In combination with the tensile stress in the previous section, the total tensile stress in the large section is 187 psi. The factor of safety on this section for total tensile loading is 187.

In the accumulator bounded section, the tensile area is 1.53 in2, and the tensile stress is equal to 953.7 psi. In combination with the tensile stress in the previous section, the total tensile stress in the accumulator-bounded section is 1011.4 psi. The factor of safety on this section for total tensile loading is 34.6.

4.2 Valve Housing End Cap

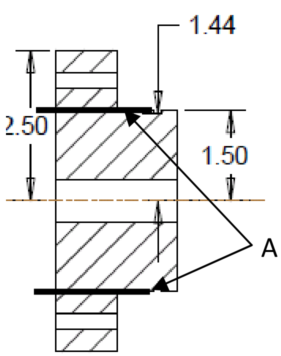
End Cap Failure modes:

1. Shear through end cap thickness

4.2.1 Shear Through End Cap Thickness

Equation 4.4 can be used to determine the maximum stress that the stem head will see due to shear in the section. The maximum shear force that the stem will see through the middle is 1060.29 lbf (V_{MAX}). The shear area can be taken from the Figure 4.3 (see the cut lines labeled A). The shear area in this geometry is equal to about 9.42 in^2. The maximum shear stress that this part will see is about 112.5 psi. The yield strength of the aluminum material (6061 alloy) that this part is made of is 35,000 psi. This allows for a factor of safety of about 311 for this mode of failure.

Figure 4.3:



$$S_Y \geq \tau = \frac{V_{MAX}}{A_{SHEAR}} \qquad \text{[Eq. 4.4]}$$

4.3 Power Piston

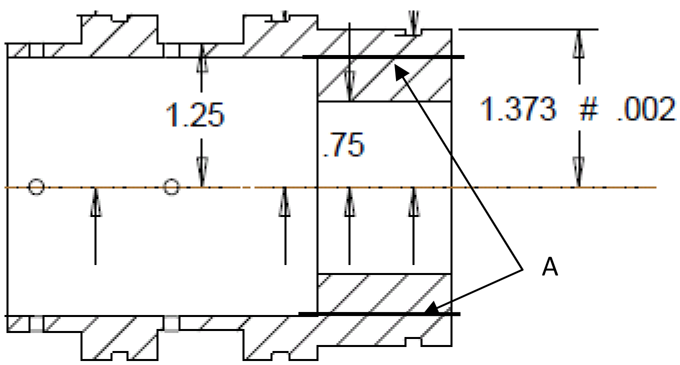
Power Piston failure modes:

1. Shear through section
2. Shear at sealing geometries
3. Tension at sealing geometry due to moments

4.3.1 Shear Through Section

Equation 4.4 in section 4.2.1 can be used to determine the maximum stress that the stem head will see due to shear in the middle section. The maximum shear force that the stem will see through the middle is 403 lbf (V_{MAX}). The shear area can be taken from the CAD drawing shown below (see the cut lines labeled A in Figure 4.4). The shear area in this geometry is equal to about 8.13 in^2. The maximum shear stress that this part will see is about 49.58 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi (according to CES Edupack 2008 [2]). This allows for a factor of safety of about 121 for this mode of failure.

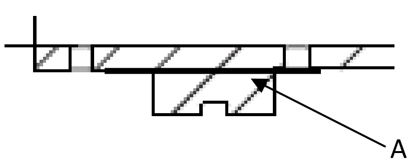
Figure 4.4:



4.3.2 Shear at Sealing Geometries

The sealing geometry can undergo large shear stresses when high pressure is applied to one side and low pressure is applied to the other side. The sealing geometry is shown in Figure 4.5, with the location of the shear section labeled by line A. The maximum shear stress would occur when the pressure in the middle of the geometry is 150 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 2.16 in^2. The maximum shear force is equal to the shear area multiplied by 150 psi, or 323.98 lbf. Equation 4.4 in section 4.2.1 can be used to find the total shear stress, with a shear area is equal to 5.11 in^2 in this geometry. The maximum shear stress that this part will see is 63.46 psi, and will act through line A in Figure 4.5. This allows for a factor of safety of about 94.5 for this mode of failure.

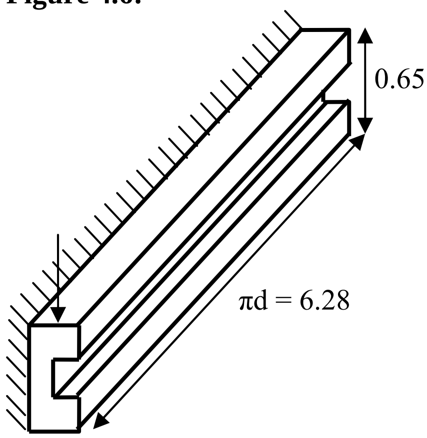
Figure 4.5:



4.3.3 Tension at Sealing Geometries Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This will act as a cantilevered beam, as in figure 4.6. The force in this section is 323.98 lbf, and is located halfway out on this cantilever beam (0.125 inches). The moment that is seen at the wall is then equal to 40.5 lb-in (*M*). The moment of inertia of this "beam" is equal to 0.1797 in^4 (*I*). In equation 4.5, y is equal to half of the height, or 0.325 in. The maximum stress in this beam is therefore 73.22 psi. With PVC being the material, the factor of safety against this kind of failure is about 82.

Figure 4.6:



$$S_Y \geq \frac{My}{I} \qquad \text{[Eq. 4.5]}$$

4.4 Valve Stem Holder

Stem holder failure modes:

1. Shear through threading
2. Shear through middle
3. Shear at sealing geometries
4. Tension at sealing geometry due to moments

4.4.1 Shear Through Threading

Analysis was done to ensure that the internal threads of the stem holder would not fail by shear. The threading will be the ½-20UNF form and will be used to fasten the external threads of the valve stem. The fine thread profile was chosen to maximize the available tensile stress area of the valve stem. While the internal threads will be made of a weaker material than the external threads (a generally undesirable condition for the fine thread form), it will be shown that this is not a problem for this situation. To ensure safe operation, the calculation of thread shear area was done assuming the worst case geometry. Namely, this assumption was the minimum possible major diameter of the internal thread. Using the formula for shear area of an internal thread (equation 4.2 in section 4.1.2), the shear area, A_n, was found to be 1.08 in^2. The parameters that entered into equation 4.2 are given in Table 4.3.

Table 4.3: Parameters used to find shear area of the internal stem holder thread

Parameter	Physical Meaning	Value
n	Threads per inch	20
L_e	Length of engagement	1 in
$D_s$min	Minimum major diameter of external thread	0.4906 in
$E_n$max	Maximum pitch diameter of internal thread	0.4731 in

The length of engagement was chosen based on the allowable space in the stem holder. This length was verified by considering the varying strengths of the PVC stem holder and the nylon valve stem. For internal and external threads made of the same material, the minimum length of engagement can be found from equation 4.6 [1], where A_t is the tensile stress area of the externally threaded member, and $K_{n\text{-}max}$ and

$E_{s\text{-}min}$ are the maximum minor diameter of internal thread, and the minimum pitch diameter of external thread, respectively.

$$L_e = \frac{2A_t}{\pi K_{n-max}\left[\frac{1}{2}+\frac{1}{\sqrt{3}}n(E_{s-min}-K_{n-max})\right]} \qquad \text{[Eq. 4.6]}$$

For this case, L_e was found to be 0.222 inch. However, because nylon and PVC have different yield strengths, an engagement length adjustment factor was calculated. This would indicate whether the thread length needed to be increased due because of the reduced strength of the PVC stem holder. The adjustment factor was found to be 0.988 from equation 4.7.

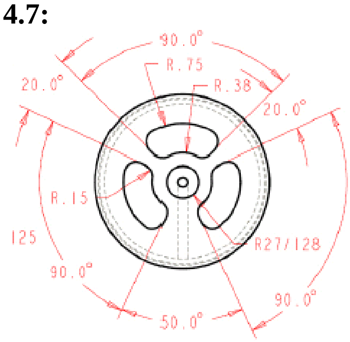
$$J = \frac{A_s\sigma_{y,ET}}{A_n\sigma_{y,IT}} \qquad \text{[Eq. 4.7]}$$

Because J was less than 1, no adjustment in engagement length was necessary, thus it was kept at 1 inch (a length that greatly exceeds the minimum requirement of equation 4.6). Assuming a maximum axial load on the stem holder threads of 403 lbf, the resulting shear stress on the threads is 373 psi. The lower bound yield strength for PVC Type 1 was found to be 6000 psi from the materials database [2], so the factor of safety for this kind of failure is about16.

4.4.2 Shear Through Middle Section

Equation 4.4 in section 4.2.1 can be used to determine the maximum stress that the stem holder will see due to shear (in the middle section). The maximum shear force that the stem will see through the middle is 403 lbs (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 4.7). The shear area is equal to the total arc length on the inside cylinder (where the three side posts meet the inner cylinder) multiplied by the depth of the piece. This total shear area is equal to 1.22 in^2. The maximum shear stress that this part will see is 330.33 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi, which allows for a factor of safety of about 18 for this mode of failure.

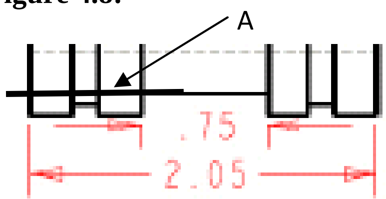
Figure 4.7:



4.4.3 Shear Through Sealing Geometry

The same equation can be used to determine the maximum shear stress that the sealing geometry in the stem holder will see. This sealing geometry is shown in the CAD drawing below. The maximum shear stress would occur when the pressure in the middle of the geometry is 150 psi while the pressure outside of the geometry is at 0 psi. The area of the sealing geometry that sees these pressures is equal to 0.8345 in^2. The maximum shear force is equal to 125.2 lbf. The total shear area is equal to 4.08 in^2. The maximum shear stress that this part will see is 30.66 psi, and will act through line A in Figure 6.2. This allows for a factor of safety of about 195 for this mode of failure.

Figure 4.8:



4.4.4 Tension at Sealing Geometry Due to Moments

The same sealing geometry above will see a moment due to the force on the pressurized area. This geometry can be modeled as a cantilever beam. The force in this section is 125.2 lbs, and is located halfway out on this cantilever beam (0.0625 inches). The moment that is seen at the wall is then equal to 7.825 lb-in (M). The moment of inertia of this beam is equal to 0.1437 in^4 (I). In equation 4.5, y is equal to half of the height, or 0.325 in. The maximum stress in this beam is therefore 17.7 psi. With PVC being the material, the factor of safety against this kind of failure is 339.

4.5 Poppet Valve Assembly

4.5.1 Valve Stem

Valve stem failure modes:

1. Shear through threading
2. Tensile yield through stem area
3. Rupture due to internal pressure

4.5.1.1 Shear Through Threading

The analysis of shear failure on the threads of the valve stem was performed in the same manner as that for the stem holder. Here, the equation for the shear area of an external thread was used (equation 4.2 in section 4.1.2). The shear area, A_s, was found to be 0.799 in^2 using the parameters in Table 4.4. Again, using a load of 403 lbf as in the case of the stem holder (since the load will be transferred through the valve stem), the shear stress was found to be 504psi. The lower bound yield strength of Nylon 6/6 was given in the materials database [2] as 8010 psi, which results in a factor of safety of 15.88 for the valve stem.

Table 4.4: Parameters used to find shear area of the external valve stem thread

Parameter	**Physical Meaning**	**Value**
n	Threads per inch	20
L_e	Length of engagement	1 in
$K_n$max	Maximum minor diameter of internal thread	0.457 in
$E_s$min	Minimum pitch diameter of external thread	0.4619 in

4.5.1.2 Tensile Yield Through Stem Area

Equation 4.8 can be used to determine the maximum stress that the stem will see in tension. The maximum tensile force is 403 lbs (F_{MAX}). The outer and inner diameters are taken at the thinnest point of the stem, which will be where the threading is created on the outside of the stock material. The stock material has an outer diameter (D_O) of ½ inch, but where the threading is formed, the outer diameter is

0.419 inch. This diameter is assuming worst case geometry (minimum material condition). The stock material has an inner diameter (D_I) of ¼ inch to give a total tensile stress area of 0.0885 in^2. The maximum tensile stress that this part will see is 4552 psi. The lower bound yield strength of the Nylon 6/6 material that will be used for the stem is 8010 psi (found from the materials database [2]).

$$S_Y \geq \frac{F_{MAX}}{A_{FORCE}} = \frac{F_{MAX}}{\left(\frac{1}{4}\right)\pi\left(D_O^2 - D_I^2\right)} \qquad \text{[Eq. 4.8]}$$

4.5.1.3 Rupture Due to Internal Pressure

Equation 4.3 in section 4.1.3 can be used to determine the maximum stress that will be seen in the wall of the stem if 150 psi is internal to the stem and 0 psi is external to the stem. According to this equation, the maximum stress is equal to 71.24 psi. Adding this stress to the stress found in the previous section results in a stress of 4623 psi maximum. This allows for a factor of safety of 1.73 for this mode of failure. It is worth mentioning that the 403 lbf loading is for a working fluid pressure of 150psi. There is no plan for running this high of pressure during any experimental testing of the valve.

4.5.2 Valve Head

Stem head failure modes:

1. Shear through threading
2. Shear through middle
3. Tension through section due to moments
4. Tensile yield through area

4.5.2.1 Shear Through Threading

Analysis was done to ensure that the internal threads of the stem head would not fail by shear. The threading will be the ½-20UNF form and will be used to fasten the external threads of the valve stem. The reasons for this thread are the same as described in the stem holder section. The threading profile and engagement length are the same as in the stem holder, so the failure calculations are the same. Assuming a maximum axial load on the stem holder threads of 403 lbf, the resulting shear stress on the threads is 373 psi. The lower bound yield strength for PVC Type 1 was found to be 6000 psi from the materials database [2], so the factor of safety for this kind of failure is about 16.

4.5.2.2 Shear Through Section

Equation 4.3 in section 4.2.1 can be used to determine the maximum stress that the stem head will see due to shear in the middle section. The maximum shear force that the stem will see through the middle is 403 lbs (V_{MAX}). The shear area can be taken from the CAD drawing shown below (Figure 4.9). The shear area in this geometry is equal to about 0.83 in^2. The maximum shear stress that this part will see is 486.275 psi. The yield strength of the PVC material that this part is made of is at least 6000 psi (according to CES Edupack 2008). This allows for a factor of safety of about 12 for this mode of failure.

Figure 4.9:

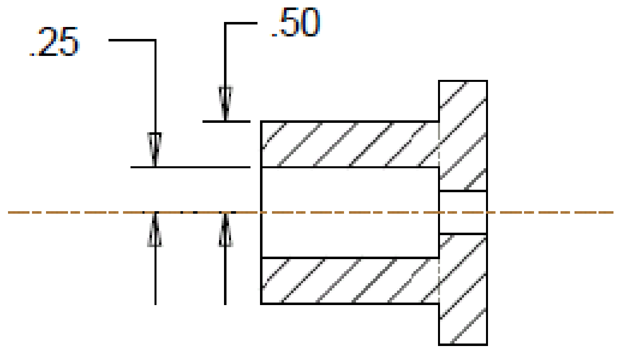
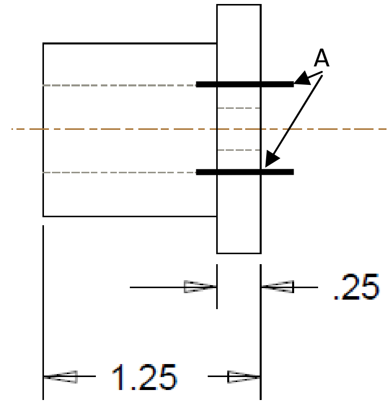


Figure 4.10:



4.5.2.3 Tension Through Section Due to Moments

Equation 4.5 in section 4.3.3 can be used to find the maximum tension on the valve stem head due to a moment. The force of 403 lbs is applied through the stem head at the outside, which means that a tensile reaction will be created on the inside of the threaded area. This reaction can be found by modeling this as a cantilever beam. The force in this section is 403 lbs, and is located on the end of this cantilever beam (0.22 inches). The moment that is seen at the wall is then equal to 88.66 lb-in (*M*). The moment of inertia of this beam is equal to 0.2618 in^4 (*I*). In the equation, y is equal to half of the height, or 0.125 in. The maximum stress in this beam is therefore 42.3 psi.

4.5.2.4 Tension Yield Through Area

The smallest area at which tensile yield would occur is 0.589 in^2. Similarly to section 4.5.1.2, with a maximum tensile force of 403 lbs, the stress in this section is equal to 684.2 psi. Adding this stress to that found in section 6.3.3, the total tensile stress seen in this part is equal to 726.5 psi. With PVC being the material, the factor of safety against this kind of failure is about 8.3.

5. MANUFACTURING

5.1 Valve Housing

The valve housing will be created from the same stock aluminum cylinder as the end cap below. The end cap will be created first, and then cut away from the remainder of the stock using an automated band saw operated by either Marve or Bob in the ME 450 machine shop. After mounting on the lathe, we will use a 1” diameter drill bit to drill a center hole to a depth of 5.6” through the stock. Then we will use a boring tool to create the internal geometries shown in the drawings above from the end-cap end of the housing, up to where the inside diameter drops from 2.75” to 1.25”. None of the 1.25” internal geometry will be created from this end. Next we will take off a thin outer layer of aluminum all along the stock almost up to the chuck. This will make the housing both shiny, and help us re-align the stock when we insert it into the chuck in the other direction.

We will remove the workpiece, and insert what is now the end-cap end of the valve housing workpiece into the chuck, and re-mount it. We will drill the 1.25” center hole in the workpiece, and then lathe down the exterior to create the neck of the housing. We will then have Bob or Marve help us create the necessary threads on the valve housing neck. The remaining holes will be drilled and tapped using a drill press and hand taps according to the engineering drawings above. All of the speeds and feeds used for machining the valve housing geometry will be determined by consulting either Bob or Marv in the ME450 machine shop before beginning machining (see Table 5.1.1 below for this information).

Table 5.1.1:

Process	Material	Machine	Cutting Speed (RPM)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Turning Tools
Internal Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Boring Bar
Seperating Material	Aluminum	Band Saw	-	-	-	Band Saw
Center Hole	Aluminum	Drill Press	165-240		.004-.005	1” Bit, 1.25” Bit

5.2 Valve Housing End Cap

The valve housing end cap will be created from a 5” diameter by 2’ long aluminum cylinder. The external geometry will be generated using a turning tool on the lathe. The hole through the center will be drilled on the lathe as well. The bolt holes will be drilled as clearance holes on the drill press and the back end will be tapped for pipe fittings by hand using a tap and accompanying handle. The dimensions and drill bit sizes and hole sizes can be found on the engineering drawing. The speeds and feeds are shown below in Table 5.2.1.

Table 5.2.1:

Process	Material	Machine	Cutting Speed (RPM)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Turning Tools
Internal Geometry	Aluminum	Lathe	165-240	.04	.004-.005	Boring Bar
Seperating Material	Aluminum	Band Saw	-	-	-	Band Saw
Center Hole	Aluminum	Drill Press	165-240		.004-.005	1/2” Bit

5.3 Power Piston

The power piston is to be created largely on a lathe from PVC stock. The PVC is the remainder of a cylindrical 3.25" diameter by 2' length rod purchased from McMaster Carr that will already be used to create the valve stem head and valve stem holder pieces of our prototype valve design. The external geometry will be machined using the tooling in conjunction with the speed and feed information found in Table 5.3.1 below. The equations below Table 5.3.1 will be used to find the RPM settings for each process. These are listed in Table 5.3.2. After the external geometry is created, the internal geometry will be removed using a boring tool. After both of these steps are completed, a drill press will be used to generate the eight 1/8" diameter holes throughout the piston.

Table 5.3.1:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	Turning Tools
Internal Geometry	PVC	Lathe	650-1600	.6	.002-.020	Boring Bar
Seperating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8" Bit

Lathe RPM equation:

$$RPM = \frac{Cutting\ Speed \cdot 4}{Diameter\ of\ Workpiece}$$

Drill Press RPM equation:

$$RPM = \frac{Cutting\ Speed\ \cdot 4}{Diameter\ of\ Drill\ Bit}$$

Table 5.3.2:

Process	Material	Machine	Cutting Speed (RPM)	Feed (ft/min)	Tool
External Geometry	PVC	Lathe	850-2100	.002-.020	Turning Tools
Internal Geometry	PVC	Lathe	650-1600	.002-.020	Boring Bar
Seperating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8" Bit

5.4 Valve Stem Holder

The valve stem holder will be made in the ME Machine Shop. The valve stem holder will be created from a 3.25" diameter, 2' long piece of stock PVC rod that has been purchased from McMaster Carr. Only a small section of the rod will be used for the valve stem holder, the remainder will be used to make other pieces of our prototype. The stock PVC will be mounted on a lathe. The external geometry will be created using a turning tool. We plan to ask Bob Curry for advice on tool selection. Next we will use a band saw to cut the newly created piece from the remaining PVC rod. A drill press will be used to create the center holes and side hole using appropriate drill bits for each hole. The through holes in the valve stem holder will either be created on a mill, or altered to be 3 round holes and created on a drill press. The speeds, feeds, and tooling for these processes will be listed in the table below. Cutting speed and feed rates will be looked up in the Machinery Handbook in the machine shop before any machining takes place. The equations below Table 5.4.1 will be used to find the RPM settings for each process and fill in Table 5.4.2.

Once all of these operations are completed, the large center hole will be tapped with a 1/2" x 13 TPI tap. This operation will be done by hand.

Mill RPM equation:

$$RPM = \frac{4 \cdot Cutting\ Speed}{Diameter\ of\ Mill\ Bit}$$

Mill feed rate equation

$$Feed = (ChipLoad) \cdot (numer\ of\ flutes) \cdot (Spindle\ Speed)$$

Table 5.4.1:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	
Seperating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8" Bit
Center Large Hole	PVC	Drill Press	650-1600			27/64" Bit
Side Port Hole	PVC	Drill Press	650-1600			1/8" Bit
a) Through Holes	PVC	Mill	650-1600			3/8" End Mill
b) Through Holes	PVC	Drill Press	650-1600			3/8" Bit

Table 5.4.2:

Process	Material	Machine	Cutting Speed (rpm)	Feed	Tool
External Geometry	PVC	Lathe	850-2100		
Seperating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8" Bit
Center Large Hole	PVC	Drill Press	6100-15000		27/64" Bit
Side Port Hole	PVC	Drill Press	20800-51200		1/8" Bit
a) Through Holes	PVC	Mill	6900-17000		3/8" End Mill
b) Through Holes	PVC	Drill Press	6900-17000		3/8" Bit

5.5 Poppet Valve Assembly

5.5.1 Valve Stem

The valve stem will be made in the ME Machine Shop. The valve stem is to be created from a section of 5' nylon 6/6 tube (1/2" OD by 1/4" ID) purchased from Mcmaster Carr. Only a small section of the tube will be used for the valve stem head, the remainder will be set aside in case we make a mistake and need to start over. The stock nylon tube will first have to be cut to a length of 6.5" using the band saw. Once that is done, both ends of the tube will be tapped to create external threads using a 1/2" x 20 UNF tap. These operations will be done by hand. When assembling the valve, a portion of the valve stem will need to be cut off to make it the correct length.

5.5.2 Valve Head

The valve stem head will be made in the ME Machine Shop. The valve stem head will be created from what remains of the 3" diameter, 2' long piece of stock PVC rod that was purchased from Mcmaster Carr and used to make the valve stem holder. Only a small section of the rod will be used for the valve stem head, the remainder will be used to make the power piston. The stock PVC will be mounted on a lathe.

The external geometry will be created using a turning tool. Next we will use a band saw to cut the newly created piece from the remaining PVC rod. A drill press will be used to create the center hole using the appropriate 1/8” diameter drill bit, then a larger hole using a 29/64” drill bit that is appropriate for the threading. The speeds, feeds, and tooling for these processes will be listed in the table below. Cutting speed and feed rates will be looked up in the Machinery Handbook in the machine shop to verify before any machining takes place. The equations below Table 5.5.1 will be used to find the RPM settings for each process and fill in Table 5.5.2. Once all of these operations are completed, the protrusion will be tapped to create internal threads using a 1/2” x 20 UNF tap. This operation will be done by hand.

Table 5.5.1:

Process	Material	Machine	Cutting Speed (sfpm)	Cut Depth (in)	Feed (in./rev)	Tool
External Geometry	PVC	Lathe	650-1600	.6	.002-.020	
Seperating Material	PVC	Band Saw	650-1600			Band Saw
Center Small Hole	PVC	Drill Press	650-1600			1/8” Bit, 29/64” Bit

Table 5.5.2:

Process	Material	Machine	Cutting Speed (rpm)	Feed	Tool
External Geometry	PVC	Lathe	850-2100		
Seperating Material	PVC	Band Saw			Band Saw
Center Small Hole	PVC	Drill Press	20800-51200		1/8” Bit

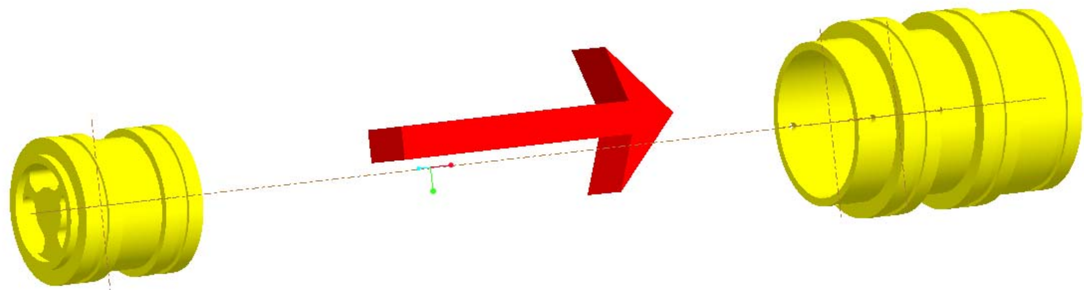
6. ASSEMBLY

Following is a list of simple instructions on how to assemble the prototype valve after all of the parts have been manufactured following the instructions above. You should have before you a valve housing, valve stem, valve stem holder, power piston, end cap, and spring as defined above.

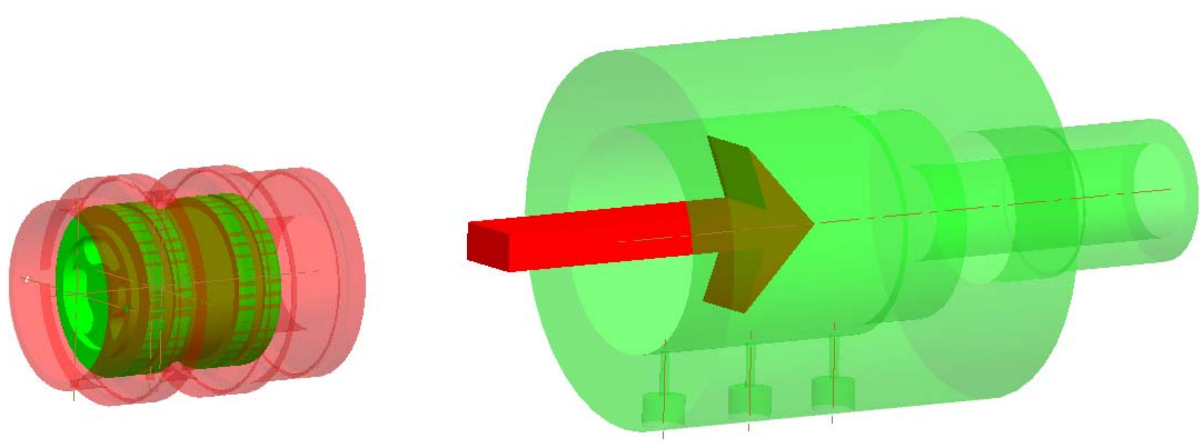
Step 1: Place the appropriate o-ring into each of the 3 o-ring grooves on the power piston, two on the valve stem retaining piston, one on the end plug, and the remaining o-ring on the outside of the valve housing at the base of the neck so that it will seal the housing against the accumulator when the valve housing is screwed into the accumulator.

Step 2: Grease the inside of the valve housing and power piston to reduce friction and ensure proper valve operation at low pressures..

Step 3: Insert valve stem retaining cylinder into back of power piston as shown below:

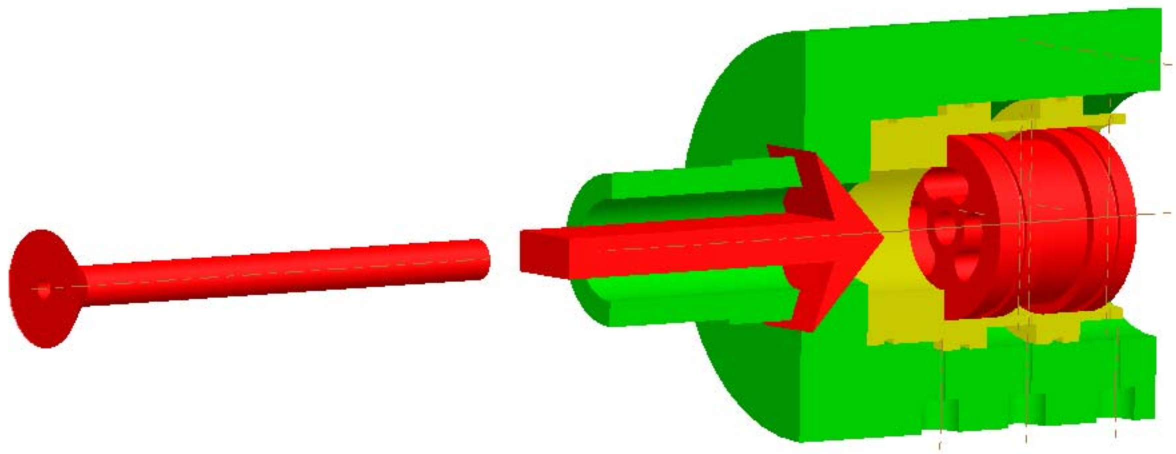


Step 4: Insert Power Piston into Valve housing show below:



Step 5: Wrap the threads of the valve stem with Teflon tape to ensure a water tight seal.

Step 6: Thread the valve stem into the valve stem retaining piston as shown below:



Step 7: Hold the valve assembly vertically so that the head of the valve stem is facing the ground. Next, insert the spring into what is now the top of the valve housing. Make sure that the spring is centered on the innermost piston (valve stem holder piston) so that the cylindrical protrusion on the piston adds lateral support to the spring.

Step 8: Attach the end cap onto the back of the valve housing (further description omitted until completion of redesign).

7. DESIGN TESTING AND VALIDATION

In order to validate that our final design meets the customer specifications, validation testing will be conducted on a prototype. Our prototype will clearly demonstrate that the final design can perform the following functions: closing when commanded closed, equalizing pressure in the line with the pressure in the accumulator and opening when commanded open, leaking pressure into the accumulator when commanded closed and pressure in the line exceeds the pressure in the accumulator, closing when commanded open but the bladder reaches the valve opening, being able to fit into the current EPA hydraulic system, and being of a less complicated design than the existing EPA high pressure accumulator valve. To show that our final design valve will function, our prototype will be built to the same interior dimensions (piston sizing and acting areas) as the final design. The motion of the valve stem will be monitored using a metal rod connected to the valve stem holder. The rod will extend into the outlet tubing allowing visual verification of the valve stem's position (necessary to verify correct functionality). Although the working pressures themselves will be different, they will act on the same surfaces. It can be assumed that if the valve actuates at these low pressures, it should only actuate better at the higher pressures used in the EPA system.

Figure 7.1: Test Apparatus

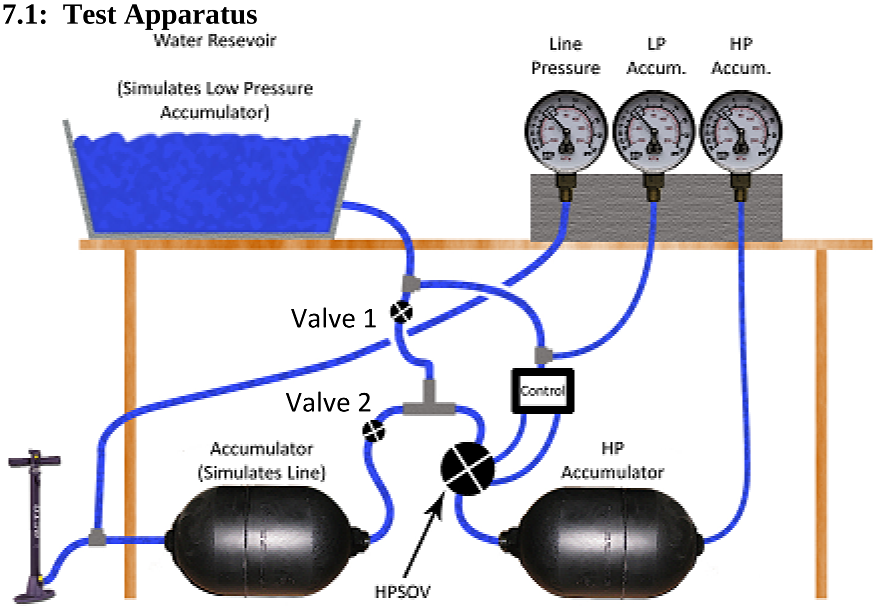
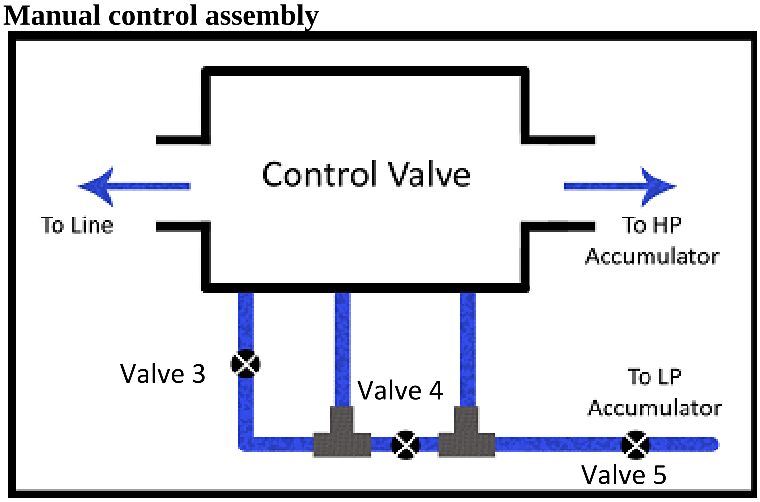


Figure 7.2: Manual control assembly



Prototype testing will be done using a test apparatus built to simulate (at a lower pressure) the various pressure conditions that will be experienced on the vehicle. The test apparatus will include two accumulators, as well as an unpressurized fluid reservoir. This setup is shown in Fig. 7.1. One reservoir will simulate the high pressure accumulator, and another reservoir will simulate the line. Each accumulator will be pressurized using a manually operated air pump connected to the bladders of the accumulators. The unpressurized fluid reservoir will simulate the low pressure accumulator, and will be positioned at a higher elevation than the rest of the setup. The available gravity head of this reservoir will be useful for initial filling, as well as to verify high pressure flow (by moving fluid from a lower elevation to a higher elevation). For all testing, the working fluid will be water.

To simulate a specific function (for example the leak-in function), pressures in the simulated "Line" and high pressure accumulators will be set according to what would be seen in the vehicle under the same situation. In this case, the "Line" accumulator would be pressurized higher than the high pressure accumulator. Normally, an electronic signal from the vehicle would be sent to the solenoid valve. The solenoid valve would the adjust pressures to the actuator piston to control the position of the main poppet. For our testing, this function will be entirely replaced by manual control. Instead of the solenoid controlling where pressure is directed, ball valves will be used to direct pressure to the correct areas. This will be done to reduce the cost and complexity of the test apparatus, as the control method will not affect the primary functionality of the valve. Detail of the control valve plumbing is shown in Fig. 7.2. There is a unique positioning of each ball valve depending on which function is being tested. Table 7.1 gives a listing of all the ball valve positions and the respective function.

Table 7.1: Positioning of valves and the highest pressure accumulator for all validation tests (valve numbers are indicated on figures 35 and 36)

	Valve 1	Valve 2	Valve 3	Valve 4	Valve 5	Highest Pressure
Command Close	Open	Close	Close	Open	Close	HPA
Command Open - Into HPA	Close	Open	Open	Close	Open	Line
Command Open - Out of HPA	Open	Close	Open	Close	Open	HPA
Leak-in test	Close	Open	Close	Open	Close	Line
Bladder interaction test	Open	Close	Open	Close	Open	HPA

More detailed instructions on how the apparatus will be operated and monitored to prove final design function will now be given:

Closing When Commanded Closed
The simulated low pressure accumulator will be at atmospheric pressure which will be less than the simulated line pressure which is less than the simulated high pressure. The valve controls will be set such that high pressure will be routed to the actuating volume, and nothing will be routed to the line pressure. The valve will close, and remain closed. The pressures will be monitored via pressure gauges attached to the simulated high pressure accumulator and line pressure. Valve control signals will be easily seen by looking at the position of the simple ball valves used to simulate control signals. To verify that the flow rate out of the accumulator is in fact zero, we will have a simple flow-meter in the line immediately after the valve.

Opening & Pressure Equalization
To show that our final design valve will equalize the line and high pressure accumulator pressures before opening and then remain open when commanded to open we will monitor both the pressure in the line, and in the tank before and during the operation. The operation will consist of switching the ball valves that control the direction of pressure within the valve so that high pressure from the accumulator is directed through a small opening into the line, and low pressure from the simulated low pressure

accumulator is directed to the acting volume. The power piston within the valve should act before the valve stem moves, this is because the power valve stem is detached from the power piston, and the spring acting on the valve stem will not allow the valve to open until the pressure difference is less than 7.5 psi. The pressures in the line and high pressure accumulator can be monitor via the pressure gauges and the flow can be seen on the installed flow meter. Once the valve is open, it can be demonstrated that the valve remains open simply by watching the valve stem position (seen by the metal rod attached to the stem which extends into the clear outlet tubing).

One Way Flow
To show this feature of our valve design we will monitor the valve position using the metal rod attached to the stem holder. We will add pressure to the line and monitor the pressure gauges of the line and high pressure accumulator. As the pressure in the line begins to exceed the pressure in the high pressure accumulator, the valve should open and allow this excess pressure into the accumulator. This will be shown both the valve position (metal rod position), pressure difference (displayed on the gauges), and fluid flow (shown by reading the flow meter and demonstrating fluid is flowing into the high pressure accumulator).

Bladder Interaction
To demonstrate that our valve will close before allowing the bladder to extrude we will allow all fluid to exit the high pressure accumulator by switching the valve to the open position and draining the fluid into the simulated low pressure accumulator. When the high pressure accumulator is empty, the bladder will try to extrude around the top of the valve stem, which will not be visible since this part of the valve will be contained within the accumulator. Instead, to see the interaction we will have to watch the valve stem position, which will be visible by the metal rod in the outlet tubing. By this we will be able to see that the valve stem closes, which will only be the result of the bladder pressing on the valve head.

Fitment into Current EPA Systems
In order to demonstrate that the final design will fit into the current EPA hydraulic system we will be using the same accumulator fittings for our prototype that they use for their hydraulic hybrid systems. In this way we can demonstrate that what we construct fits the accumulator, and that our final design has the same accumulator fitting designs. The other side of the valve (the side to which the line is attached) has a set of exact dimensions that will be included in our final design, but will not be manufactured on the prototype.

8. FMEA ANALYSIS

FMEA Analysis was performed on all the components of the experimental setup that were not manufactured by the team. These components included the high pressure accumulators, ball valves, hydraulic fittings, hose, hose clamps, and pressure gauges. A list of the components with their corresponding failure modes and FMEA Analysis is given in SR Appendix B.

Because the experimental setup includes water under pressure, the most severe failure mode would be a leak or break big enough to cause near instant release of the stored energy. This particular mode is unlikely because all components are rated for pressures far greater than those that will be used during testing. In addition, simple visual inspections of each component will be performed before testing. Any flaw great enough to cause a catastrophic failure would likely be noticed since the components have such design margins at these low pressures. A more likely failure would be a leak from either of the fittings or connections. These, however, would easily be remedied by relieving the pressure from the system, repairing the faulty connection, and repressurizing the system and resuming testing. Care will be taken when installing all fittings, as well as the HPSOV itself to minimize the likelihood of any of these leaks.

REFERENCES

[1] Jones, F., Ryffel H., Oberg E., McCauley C., Heald R., 2004, *Machinery's Handbook, 27th Edition*, Industrial Press, Inc.

[2] *CES Selector Version 4.8.0*, 2008, Cambridge, UK, Granta Design Limite

[3] Watkins, R. K., Anderson, L. R. (2000). Structural Mechanics of Buried Pipes, CRC Press.

SR APPENDIX A: DesignSafe Analysis
A1. Valve Housing

designsafe Report

Application:	Valve Housing	Analyst Name(s):	
Description:	Aluminum housing that contains all valve components, threads into accumulator, and attaches to end cap/hydraulic line.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, drill press, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : fatigue Valve housing is made from aluminum and could lose stength and integrity during operation due to wear and tear from internal components or from fluid flow. This could lead to stoppage of the system, leakage, or rupture of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the aluminum pressure vessel. Cracks could occur or part could fracture.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	slips / trips / falls : debris During the manufacturing process, debris from the component or machine could come loose and injure an operator.	Serious Occasional Unlikely	Moderate	Proper machine shop safety procedures will be followed. Proper attire will be worn, including approved safety glasses.	Slight Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	ergonomics / human factors : deviations from safe work practices Assembly work will be done mostly in machine shop, where any deviation from safety practices could lead to injury.	Serious Remote Unlikely	Moderate	Remember to adhere to safety procedures. Team members will assist and remind each other.	Serious Remote Unlikely	Moderate	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the housing, fluid or component pieces could leak or eject from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]
All Users All Tasks	fluid / pressure : fluid leakage / ejection If pressure overcomes strength of the housing, fluid could leak or eject forcefully from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor.	Serious Remote Negligible	Low	Complete [3/31/2009]

A2. Valve Housing End Cap

designsafe Report

Application:	End Cap	Analyst Name(s):	
Description:	Aluminum cap that bolts and seals to end of aluminum valve housing and connects tohydraulic line with fittings.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, drill press, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : fatigue End cap is made from aluminum and could lose stength and integrity during operation due to wear and tear from internal component interaction or from fluid flow. This could lead to stoppage of the system, leakage, or rupture of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the end cap or the seals between end cap and valve housing. Cracks could occur or part could fracture.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	slips / trips / falls : debris During the manufacturing process, debris from the component or machine could come loose and injure an operator.	Serious Occasional Unlikely	Moderate	Proper machine shop safety procedures will be followed. Proper attire will be worn, including approved safety glasses.	Slight Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	ergonomics / human factors : deviations from safe work practices Assembly work will be done mostly in machine shop, where any deviation from safety practices could lead to injury.	Serious Remote Unlikely	Moderate	Remember to adhere to safety procedures. Team members will assist and remind each other.	Serious Remote Unlikely	Moderate	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the end cap, seals, or fittings, fluid or component pieces could leak or eject from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]

A3. Power Piston

designsafe Report

Application:	Power Piston	Analyst Name(s):	
Description:	PVC component inside valve housing with porting that allows for valve actuation.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, drill press, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	mechanical : fatigue Power piston is made of PVC and could lose stength and integrity during operation due to wear and tear. This could lead to stoppage of the system, or leakage of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the PVC part. Cracks could occur or part could fracture.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the power piston, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]
All Users All Tasks	fluid / pressure : fluid leakage / ejection If pressure overcomes strength of the power piston, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor.	Serious Remote Negligible	Low	Complete [3/31/2009]

A4. Valve Stem Holder

designsafe Report

Application:	Stem Holder	Analyst Name(s):	
Description:	PVC component inside valve housing that the stem threads into.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	mechanical : fatigue Stem holder is made of PVC and could lose stength and integrity during operation. This could lead to stoppage of the system, or leakage of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the PVC part. Cracks could occur or part could fracture.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the stem holder, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]
All Users All Tasks	fluid / pressure : fluid leakage / ejection If pressure overcomes strength of the stem holder, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor.	Serious Remote Negligible	Low	Complete [3/31/2009]

A5. Poppet Valve Assembly

A5.1. Valve Stem

designsafe Report

Application:	Valve Stem	Analyst Name(s):	
Description:	Nylon component inside valve housing, through neck, and into accumulator. Threads into stem holder. Valve stem head interacts with bladder inside accumulator.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	mechanical : fatigue Stem is made of Nylon 6/6 and could lose stength and integrity during operation. This could lead to stoppage of the system, or leakage of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the Nylon part. Cracks could occur or part could fracture.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	slips / trips / falls : debris During machining, loose pieces could be ejected from the machine.	Serious Remote Unlikely	Moderate	Proper machine shop safety procedures will be followed. Proper attire will be worn, including approved safety glasses.	Slight Remote Unlikely	Low	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the stem holder, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]
All Users All Tasks	fluid / pressure : fluid leakage / ejection If pressure overcomes strength of the stem holder, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor.	Serious Remote Negligible	Low	Complete [3/31/2009]

A5.2. *Valve Head*

designsafe Report

Application:	Valve Head	Analyst Name(s):	
Description:	Conical shaped PVC component that will be attached to end of valve stem, creating the valve seal. Will also interact with bladder inside accumulator.	Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : cutting / severing This part will need to be machined using a mechanical lathe, band saw, drill press, and a mechanical mill. During the machining process the component as well as the machines may possess sharp edges that present a safety hazard.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : drawing-in / trapping / entanglement A lathe, mill, drill press, or saw could present a hazard by entagling clothing of users into the machine potentially causing serious injury.	Catastrophic Occasional Unlikely	High	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Catastrophic Occasional Unlikely	High	On-going [Daily]
All Users All Tasks	mechanical : pinch point While machining parts, a component could fall or be compressed together, creating a pinch point and potentially injuring an operator.	Serious Occasional Unlikely	Moderate	All required safety procedures will be followed exactly while working in the machine shop. Consultation or instruction will be sought from the lab instructor if needed.	Serious Occasional Unlikely	Moderate	On-going [Daily]
All Users All Tasks	mechanical : fatigue Valve stem head is made of PVC and could lose stength and integrity during operation due to wear and tear. This could lead to stoppage of the system, or leakage of water.	Serious Occasional Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Occasional Negligible	Moderate	On-going [Daily]

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : break up during operation Pressure could overwhelm strength of the PVC part. Bladder interaction could break head from rest of stem. Cracks could occur or part could fracture. Could be ejected into accumulator.	Serious Remote Unlikely	Moderate	Strength and material interactions are being analyzed. Large safety factors are included to reduce risk.	Serious Remote Negligible	Low	On-going [Daily]
All Users All Tasks	fluid / pressure : hydraulics rupture If pressure overcomes strength of the valve stem head, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor. Installation will be precise and thorough.	Serious Remote Negligible	Low	Complete [3/31/2009]
All Users All Tasks	fluid / pressure : fluid leakage / ejection If pressure overcomes strength of the valve stem head, fluid could leak or spray from valve housing.	Serious Remote Unlikely	Moderate	Material strength was determined to allow for expected pressures with a large safety factor.	Serious Remote Negligible	Low	Complete [3/31/2009]

SR APPENDIX B: FMEA Analysis

Failure Mode and Effects Analysis Worksheet (Adapted from Cincinnati Machine PFMEA)

<table>
<tr><td rowspan="4">Description of system and mode of operation

High pressure shut-off valve for hybrid hydraulic vehicles:
High-pressure shut-off valve will be tested for proper functionality using hydraulic accumulators. Accumulators will be filled with water and pressurized to varying levels</td><td>Key Contact / Phone
Paul Juska / (248) 495-3909</td><td>Date of Initial FMEA
04/07/09</td></tr>
<tr><td rowspan="2">Core Team:
ME450
Team # 10</td><td>Date of Initial System Demonstration
04/15/09</td></tr>
<tr><td>Review Board Approval / Date
04/15/09</td></tr>
<tr><td>Location:
Testing done next to GG Brown Blue Lounge</td><td></td></tr>
</table>

Potential Failure Modes and Hazard Identification Discussion: Identify all potential failures and safety hazards for this system in the applicable mode of operation. Complete a FMEA rating form for each significant item.

Accumulator contains stored energy in the form of pressurized fluid. Main hazards are fluid leakage.
The severity of the hazard varies directly with fluid leakage rate. Rapid release of energy can cause injury and damage to surrounding people and objects.

Individual component failure may occur. Again, this will result in fluid leakage at varying rates.

Accumulators are relatively massive, therefore injury to persons handling the equipment could occur if objects are lifted incorrectly or if they fall.

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and mode of operation	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion Date	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
High pressure accumulators	Bad seal / incorrect connection	Fluid leak	2	Low to medium	2	Inspection of connection after small pressurization	2	8	Carefully install valve to accumulator. Check to make sure seal is installed properly						
	Breakage	Explosion / High fluid leakage rate	10	Minimal	1	Visual inspection of accumulator before pressurization	1	10	Accumulator rated for higher pressure than required. Perform visual inspection to verify accumulator integrity						

1. Discuss root cause of the failure mode (based on the 5 whys)
Cause of first failure mode can be improper installation of valve onto the accumulator. The likelihood can be reduced if care is taken to thread the valve into the accumulator properly.

Breakage of the accumulator is highly unlikely due to the low test pressures compared to the rated pressure of the accumulator

2. Discuss/justify the severity rating (SEV)
SEV is very low and is not a concern for either failure mode

3. Discuss/justify the rating for probability of occurrence (OCC)
Probability of occurrence for both failure modes is low. Improper installation likelihood is higher than breakage, but is unlikely due to the manual installation that will be performed

4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET)
Visual inspections will be sufficient to prevent accumulator breakage (due to their high pressure rating, a large physical defect would be necessary to cause failure).
Detection for the first failure mode can be done safely by pressurizing the accumulator to modest levels to check for leaks before main testing begins.

5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered.

Carefully install valve onto accumulator, and visually inspect accumulator before pressurization. If leaks are noticed on the valve/accumulator connection, repair the seal before proceeding. If leaks are noticed on the accumulator itself, do not proceed with testing until a functional accumulator has been acquired.

6. Notes on Actions taken:

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and mode of operation	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion Date	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
Ball valves	Bad seal / incorrect connection	Fluid leak	2	Medium for small fluid leak (NPT fittings)	4	Inspection of connection after small pressurization	2	16	Take care when making line connections. Check seal integrity with small pressure level						
	Breakage	Explosion / High fluid leakage rate	9	Minimal	1	Visual inspection of valve before pressurization	1	9	Ball valves rated for higher pressure than required. Perform visual inspection to verify valve integrity						

1. Discuss root cause of the failure mode (based on the 5 whys)
Cause of first failure mode can be improper connection of lines and fittings. Care should be taken when these connections are made

Breakage of the ball valves is highly unlikely due to the low test pressures compared to their rated pressures

2. Discuss/justify the severity rating (SEV)
SEV is very low and is not a concern for either failure mode

3. Discuss/justify the rating for probability of occurrence (OCC)

Probability of occurrence for both failure modes is low. Improper installation likelihood is higher than breakage, but the severity is low and the connections can be easily repaired upon depressurization

4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET)

Visual inspections will be sufficient to minimize risk of valve breakage (due to their high pressure rating, a large physical defect would be necessary to cause failure).

Detection for the first failure mode can be done safely by pressurizing the accumulator to modest levels to check for leaks in the system and at the connections before main testing begins.

5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered.

Carefully make all line connections and visually inspect ball valves before main pressurization. If leaks are noticed on either connection, depressurize the system and repair the connection before proceeding. If leaks are noticed on the ball valve itself, do not proceed with testing until the valve is replaced.

6. Notes on Actions taken:

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and mode of operation	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion Date	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
Hydraulic fittings	Bad seal / incorrect connection	Fluid leak	2	Medium for small fluid leak (NPT fittings)	4	Inspection of connection after small pressurization	2	16	Take care when making line connections. Check seal integrity with small pressure level						

	Breakage	Explosion / High fluid leakage rate	9	Low	2	Visual inspection of fitting for cracks and proper geometry	2	36	The fittings are rated for higher pressures than will be tested. Perform visual inspections before installation and replace fitting if any problem is found.						

1. Discuss root cause of the failure mode (based on the 5 whys)
Cause of first failure mode can be improper connection of lines and fittings. Care should be taken when these connections are made

Breakage of the fittings is unlikely due to the low test pressures compared to their rated pressures. Higher risk is for breakage during installation (from over tightening)

2. Discuss/justify the severity rating (SEV)
SEV is very low and is not a concern for either failure mode

3. Discuss/justify the rating for probability of occurrence (OCC)
Probability of occurrence for both failure modes is low. Improper installation likelihood is higher than breakage, but the severity is low and the connections can be easily repaired upon depressurization

4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET)
Detection for the first failure mode can be done safely by pressurizing the accumulator to modest levels to check for leaks in the system and at the connections before main testing begins.
Careful inspection of the fittings can maximize the probability of detecting failures

5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered.
Carefully make all line connections and visually inspect the fittings before main pressurization. If leaks are noticed on either connection, depressurize the system and repair the connection before proceeding. If the fitting is found to be damaged, replace the fitting before proceeding with testing.

6. Notes on Actions taken:

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and mode of operation	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion Date	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
Hose Clamps	Shear Screw Threads. Exccessive Pressures, overtightening.	Water ejected from system into surroundings	1	Low	1	visual inspection, feeling when screw strips during tightening.	1	1	Replace hose clamp						

1. Discuss root cause of the failure mode (based on the 5 whys). The root causes of failing are improper installation, if the installer tightens th

2. Discuss/justify the severity rating (SEV): We believe that the severity of hose clamp failure is small. This is because if the hose clamp is des

3. Discuss/justify the rating for probability of occurrence (OCC): We rated the probability of occurrence very low. It is highly unlikely that the ho

4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET). This we rated low as well. It is easy to detect a failed hose clamp (stripped threads) during installation as the screw will tighten indefinately without producing resistance. Unless the clamp fails during installation, it is unlikely to fail during testing. if it does begin, it is likely to leak before bursting, at which point we can use a ball valve to empty fluid into a bucket before failure under pressure occurs.

5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered: if

a failure is detected, we recommend immediate replacement of the hose clamp.

6. Notes on Actions taken: Always De-Pressurize the system before making adjustments to system setup.

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and **mode of operation**	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion **Date**	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
Pressure Gauges	explosion. Overpressurizatio n, and getting hit by something.	water ejection from system	2	Low	1	visual inspection, and careful placement	1	2	replace pressure gauge	installer.					

1. Discuss root cause of the failure mode (based on the 5 whys): Root causes of failure are striking the clear face of the pressure gauge with a

2. Discuss/justify the severity rating (SEV). The severity rating is 2, because although water ejection is not extremely severe, due to the nature

3. Discuss/justify the rating for probability of occurrence (OCC): the probability of occurance is almost never. This is because after intallation, n

4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET). We will be unable to detect cracking due to pressure, however it is easy to see if the face of the pressur gauge is cracked, and this will be inspected before pressurizing the system.

5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered. If a crack is detected, we recommend replacing the pressure gauge before pressurizing the system.

6. Notes on Actions taken: Always make sure the system is depressurized before making adjustements to the setup.

FMEA rating form for a single Failure / Hazard

Categorize: Identify susbsystem and mode of operation	Potential Failure Mode and 5 Whys [1]	Potential Effect of Failure [2]	SEV	Probability of Occurrence of Failure [3]	OCC	Current Controls for Detection / Prevention [4]	DET	RPN	Recommended Action [5]	Person Responsible & Completion Date	Action Results: Action Taken [6]	SEV	OCC	DET	RPN
Hose	Being Cut, exploding. Cut by a sharp object, or continual friction, exploding due to internal pressure	Water squirting from system	2	Low	1	visual inspection	1	2							

1. Discuss root cause of the failure mode (based on the 5 whys). Root causes are a sharp object cutting the hose, overpressurization, or conti

2. Discuss/justify the severity rating (SEV): The severity of any of these failure modes is 2 because it will again cause water to be ejected from
3. Discuss/justify the rating for probability of occurrence (OCC). The probability of cutting the hose is low, no knives will be in the vicinity of the
4. Discuss/justify the rating for the probability of detecting a "failure imminent" condition and avoiding the failure (DET). Detection will occur when the hose leaks, a hole of this nature is likely to leak only slightly at first, leaving us enough time to identify the problem and depressurize the system before a major leak occurs.
5. Recommended actions: Make specific recommendations for action and include some discussion of the alternatives that were considered. We recommend that the section of hose that is damaged be replaced.
6. Notes on Actions taken: Always De-Pressurize the system before making adjustments to the setup.