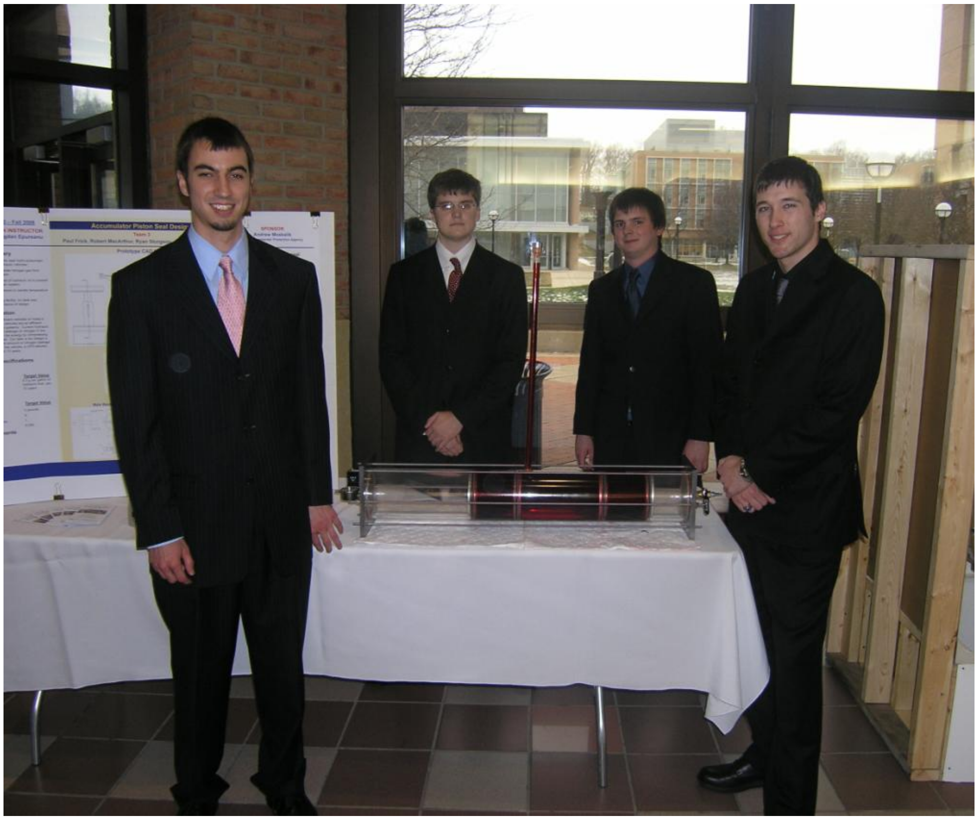
ME 450 Fall 08 Final Engineering Design Report

Accumulator Piston Seal Design and Test Fixture for Hydraulic Hybrid Vehicle



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12/9/08

Executive Summary

Our team was tasked to design an accumulator mechanism that will separate nitrogen and hydraulic fluid, with a maximum leakage of 5.3 grams of nitrogen per gallon of hydraulic fluid in the system for a 10 year period. Since it has been shown in the past that accumulator designs cannot be proved purely by theory, creating a physical prototype as proof of concept was imperative. Our ultimate design should prevent nitrogen from leaking into the hydraulic fluid, have a long lifetime, a low complexity, and as low a cost as we can manage.

We generated concepts which can be broken down into four main categories: elastic bladders, impermeable non elastic membranes, metal bellows, and piston accumulators. After doing further research and interviewing our sponsor about previous designs that the EPA had tried, we discovered inescapable weaknesses with all but the piston accumulator designs. However, the current piston accumulator used by EPA exhibits far more leakage than desired. We narrowed our piston accumulator designs down to three and then chose our top design. Our top design centers around having a chamber inside the piston which will be filled with hydraulic fluid that will be allowed to saturate if nitrogen leaks past the first seal. This will prevent nitrogen from reaching the clean hydraulic fluid in the system.

To complete the prototype for our top design we selected 6061 aluminum as the construction material and performed a stress analysis on many features of the design. This allowed us to verify that the dimensions selected allow our design to withstand the worst scenario possible for the system without failure. We then created a manufacturing plan detailing how our top design will be prototyped. The prototype was formed by welding a rod to each end of our piston, then using mill and lathe processes to remove material until the final size was reached.

To test this prototype, a clear acrylic accumulator was machined and filled with two of our prototype pistons. The test fixture was to be cycled by flowing shop air into the accumulator then venting the built up pressure to atmosphere.

Our project plan required us to complete many tasks. We performed concept generation and concept selection, and from this selected our final design. We then finalized dimensions through stress analysis and chose appropriate materials and parts. A plan for manufacturing was created to facilitate production of the prototype and test fixtures, followed by the actual manufacturing phase. The entire test setup was to be tested over several days to measure leakage rates of the seals. The accumulator watch glass accidently leaked during the initial pressurization which resulted in the closure of testing, so we are unable to have any testing data at this time. We do however have recommendations of how the fixture and prototype can be improved upon for future testing.

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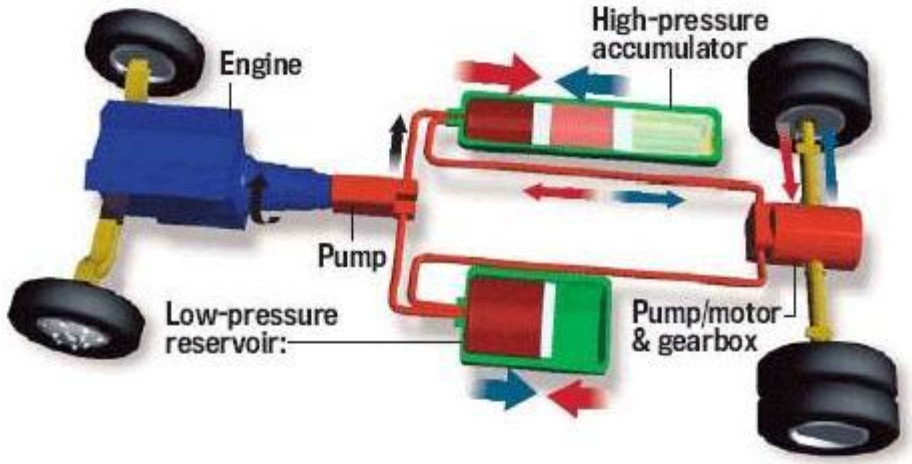
1.0 Introduction

Our project is to design and test a hydro-pneumatic accumulator for operation on hydraulic hybrid vehicles. A hydro-pneumatic accumulator is a device that stores energy in a pressurized tank by filling the tank with hydraulic fluid and compressing a volume of nitrogen stored inside. The accumulator is used as part of a system to store both energy generated by diesel engine and a hydraulic braking system. Figure 1 below shows a series hydraulic hybrid system.

Figure 1 – Series Hydraulic Hybrid System [1]. Black arrow represents the input from the engine, red arrows represent input from regenerative braking, and blue arrows represent flow during acceleration



The EPA (Environmental Protection Agency) is sponsoring our project, with Andrew Moskalik acting as our liaison and contact. The EPA has been working on hydraulic hybrid vehicles for almost ten years, with the ultimate goal of promoting the use of cleaner vehicles. Their designs have come a long way, but still have faults in the accumulator system, such that nitrogen gas is leaking into the hydraulic fluid causing damage to the hydraulic pumps. This is causing life expectancies for their hydraulic systems to be measured in weeks and months, instead of years. If the leakage can be stopped, or reduced significantly enough, then hybrid hydraulic vehicles would be one step closer to wide spread implementation in the coming years. Figure 2 on page 6 shows one of the EPA's hydraulic hybrid UPS trucks.

Figure 2 – Hydraulic Hybrid UPS truck created by the EPA [2]

Our goal for the project is to design an accumulator which will not leak a significant amount of nitrogen into the hydraulic fluid, calculated as 5.3 grams of nitrogen per gallon of hydraulic fluid in the system, during the desired 10 year lifespan of the hydraulic system. Our sponsor also desired that we test our design in an accumulator to show proof of the concept, unfortunately due to an accident in the lab this was not possible.

2.0 Customer Requirements and Engineering Specifications

During a meeting with our sponsor, we identified that our design must achieve a low leakage rate of nitrogen over a ten year lifespan as its primary requirement. This would ideally be 1000 times better than the existing EPA designs. The target specification for leakage rate was: the amount of nitrogen that can be dissolved in 40 gallons of hydraulic oil at 50 degrees C and 80 psi gauge minus the amount of nitrogen that can be dissolved in 40 gallons of hydraulic oil at 50 degrees C and 0 psi gauge. We approximated the hydraulic oil to be n-Hexane [3] and used a reference of experimental data [4] to calculate the leakage spec to be 5.3 grams of nitrogen per gallon of hydraulic fluid in the system. This calculation can be seen in Appendix A. We should stay under this leakage spec for the ten year target lifetime of the system.

Because we only had one significant customer requirement we decided against using a QFD to rank importance of our specifications. We added secondary requirements in order to sort our concepts in the event that multiple designs met the primary requirements. These include:

- Low complexity
- Low cost
- Light weight
- Ease of manufacture

We chose these requirements to ease the design and manufacturing of our prototype since we were highly limited in our budget and schedule. Low complexity and ease of manufacture enable the team to build the concept within our timeframe with available skills and machinery, while a low cost enables us to build a fully working prototype within our budget. Light weight was chosen only as a point to compare competing designs against, the lighter concept would likely be easier to build and implement. We weighted these secondary requirements using a pair-wise table (more important is "1"), and determined the weights as a percentage of the maximum, as shown in Table 1 below.

Table 1 – Comparison of Secondary Requirements

Requirement							**Total**	**Weight**
Low Cost	0	0	0				0	0
Low Complexity	1			0		1	2	0.666
Low Weight		1		1	1		3	1
Ease of Manufacture			1		0	0	1	0.333

For each of these secondary requirements, we set a target engineering specification based on our design and manufacturing capabilities. These are enumerated in Table 2 below, as well as the primary engineering specifications as described above.

Table 2 – Target Values for Concepts

Customer Requirement	**Requirement Type**	**Engineering Specification**	**Target Value**
Low Leakage Rate	Primary	Grams per Gal. per 10 years	5.3
Low Cost	Secondary	Cost ($)	250
Low Complexity	Secondary	Number of moving parts (#)	1
Light Weight	Secondary	Weight (lbs)	5
Ease of Manufacture	Secondary	Number of parts (#)	4

3.0 Concept Generation

Our design concepts were formed from initial information given by our sponsor and through our patent research. Our sponsor informed us about the advantages and disadvantages of elastic bladders and pistons; the EPA had already done previous testing and had gained some valuable insight on the properties of these two concepts, and our sponsor passed much of this on to us. Impermeable bags, and metal bellows were discovered when we performed patent research on hydro-pneumatic accumulators. These designs all appeared in multiple patents, which initially hinted at their feasibility, inspiring us to do further research on them.

After we had gathered these four basic concepts we began to study various permutations of these four basic concepts. The complete listing of our ideas is shown in Appendix B.

3.1 Elastic Bladder

Elastic bladders were mentioned to us during our initial conversation with our sponsor. These bladders are made from an elastomer, such as rubber, and inhibit the mixing of the gas and liquid inside the accumulator by effectively sealing the gas inside the bladder. The bladder accommodates pressure deviations by expanding and contracting like a balloon inside the accumulator; decompressing or compressing the gas inside the bladder to match the pressure outside the bladder. It normally contains a seal or port which allows for the initial inflation, which is then held shut during use. A picture of a basic bladder design is shown in Figure 3 below.

Figure 3 – Basic elastic bladder design, with nitrogen inside the bladder on the left and hydraulic fluid outside on the right.

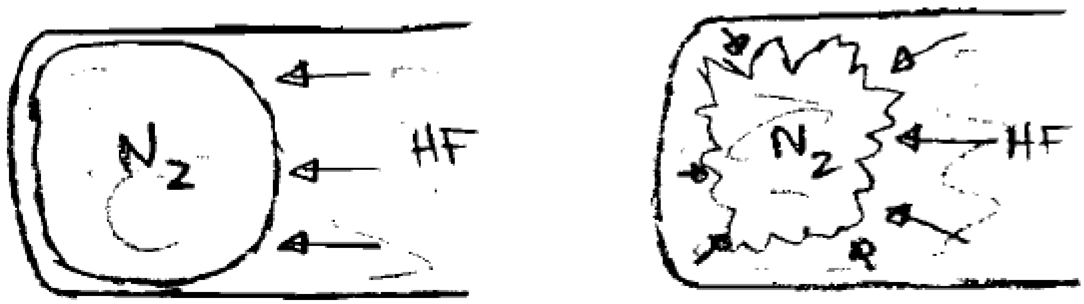
An accumulator with an elastic bladder is advantageous for many reasons. First it is very inexpensive; the elastomer wall itself is often formed out of a simple rubber compound and the only additional component is a seal or port which allows for the addition or removal of gas during maintenance. This leads directly to its second strength, simplicity. The bladder normally contains only two parts, the elastic wall and a port, which are easy to design, manufacture, and maintain. The bladder also possessed an inherently high durability. The elastic wall, the bladders only moving part, is extremely resilient in resisting fatigue, due to the simple elastic nature of its expansion and contraction. It also experiences almost no friction with the walls of the accumulator or the exterior liquid. Finally the elastic bladder can be easily filled with low density compressible foam, along with gas. This foam increases the efficiency of the bladder by reducing losses due to heat flow.

An elastic bladder has one inherent weaknesses which has not yet been overcome; high leakage rate. This is a weakness in the long molecular chains which allow an elastomer to flex and expand without permanent damage. Unlike materials with a crystalline structure, gas molecules are able to squeeze through the gaps between these chains in elastomers. Several actions can be taken to minimize this leakage, such as increasing the thickness of the bladder walls beyond that necessary merely to withstand applied stresses. This extra thickness can provide improvements in leakage rates at the cost of flexibility. Another method is to coat the inner walls of the bladder with various films designed to be impermeable to gases. This method can provide significant improvements in the short term, but the coating will eventually separate and flake off the bladder walls, negating its use.

3.2 Impermeable Bag

Similar to elastic bladders, impermeable bags contain the gas of an accumulator and float freely in the liquid. However impermeable bags are non-elastic and are formed out of materials much more resistant to leakage, such as thermoplastics or metal films. They react to pressure changes by crumpling like a plastic bag instead of simply compressing. They can often be formed from layers of materials, at least one of which is essentially impermeable to gas leakage. A drawing of an impermeable bag is show in Figure 4 below.

Figure 4 – Impermeable bag accumulator design. The state on the left is an expanded bag prior to compression, and the state on the right shows a partly compressed bag.



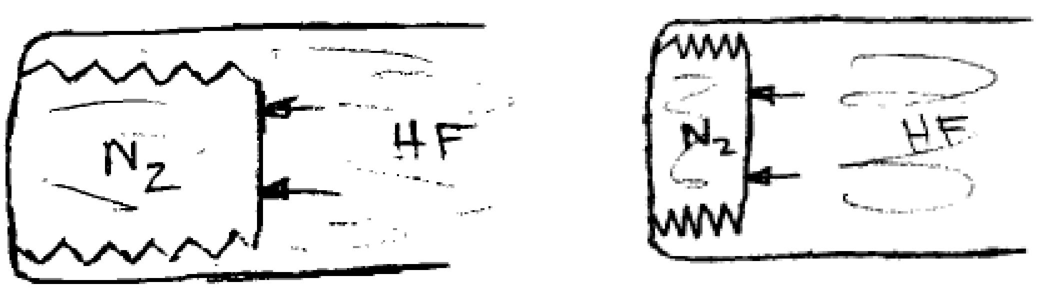
Bags made of plastics are cheap to make and simple to assemble and manufacture, similarly to elastic bladders. They can also be filled with foam to increase heat retention and limit energy loss. However their resistance to gas transmission through the walls of the bag is directly related to the thickness of the walls. Some plastics can be 100 to 1000 times more resistant to gas permeation than rubber compounds. However to allow such plastics to be flexible enough to crumple the walls must be made much thinner, often 100 times thinner than a rubber bladder. This change in thickness virtually negates the improved performance of the material, resulting in little net gain in transmission rates. Also, impermeable bags are not as durable as elastic bladders; after several thousand folds (which can be experienced in months in our application) the material begins to fold consistently in the same location, forming a crease. This crease will expand and generate high leakage rates at the location, ultimately forming a rip in the bag causing catastrophic failure of the system.

Bags made with metal foil are more expensive to manufacture and assemble. However they are impermeable to gas transmission through the metal foil. Very small leaks can occur at seams or where layers of material meet or overlap; metal foil bags are too thin to be welded together, especially if they are layered with other materials, and are simply over lapped at joints and sealed together. This seal is not perfectly impervious to leakage, causing non-zero leakage for the system as a whole. However foil bags also do not have the resilience to withstand hundreds of thousands of expansion and compression cycles demanded of them for use in automotive accumulators. They are susceptible to creases and cracks and other failures. Metal foils may be layered with other compounds and plastics to compensate for this, but this does not cure failure by creasing or cracking. The layering of foil with other compounds introduces the problem of delaminating; gas may permeate through the layered material but stop at the foil, ultimately resulting in bubbles forming along the bag's walls. This ultimately results in the separation of the foil from other layers, negating all benefits of applying the layering.

3.3 Metal Bellows

Metal bellows are a series of thin metal plates welded together in such a way that as the bellows expand the plates bend and unfold, similar to an accordion, allowing the inner volume of the bellows to increase. A bellows of sufficient size for our use may contain several hundred plates welded together. These welds act similar to hinges for the two plates they link. These plates have a metal cap on the end of the cylindrical bellows sealing the inner volume from the outer. The bellows would be filled with nitrogen, with hydraulic fluid existing between the bellows and the accumulator. A bellows system may also feature rails or other objects along the walls of the accumulator to guide its expansion and contraction. A drawing of a bellows is shown in Figure 5 below.

Figure 5 – A bellows type accumulator, with nitrogen inside the bellows



The primary advantage of a bellows is that it is entirely made of metal. Thus it is initially impermeable to gas transmission, perfectly eliminating the saturation of our fluid with nitrogen gas.

However a bellows of sufficient size would be extremely large and complex. It would contain several hundred small metal rings and welds, resulting in large manufacturing costs. Our sponsor has estimated the cost in the tens of thousands of dollars. It would also be extremely vulnerable to fatigue, if only a small portion of any of these welds fails during any of the hundreds of thousands of expansions and compressions it is expected to experience during its lifetime, it would result in a catastrophic failure of the system.

3.4 Piston Accumulator

The final major type of accumulator we found is a piston accumulator. Piston accumulators function using a metal piston which slides back and forth, compressing the gas. The piston is snug against the sides of the accumulator, with a seal, or seals, preventing leakage of gas past the piston and into the liquid. The piston often has guide rings along its sides to maintain alignment within the accumulator and may also have scrapers along its walls to prevent debris from damaging the seals. A drawing of a piston accumulator is shown below in Figure 6.

Figure 6 – A basic piston accumulator, with a single o-ring seal.

Piston accumulators are inexpensive to manufacture. They require only one machined part, which is the piston, and an off the shelf part, the seal. Pistons are also almost perfectly impermeable to gas leakage. The only way for gas to leak in a piston is for the gas to pass through the thin gap between the piston and the accumulator wall, and past the seal(s) along the piston's edge. These seals themselves are manufactured from materials with very low permeation rates and are very resistant to gas flow.

There are ways for the gas to make it past the seal and into the hydraulic fluid. First, the surface finish on the accumulator walls must be extremely precise, any minute cracks or crevices can allow gas to slowly leak past a seal when the piston cycles back and forth. The piston and accumulator must also be extremely round; any deviations cause irregularities in the pressure of the piston seal against the accumulator wall. Low pressure against the wall can allow gas to seep past. High pressure against the wall causes increased friction between the wall and the piston, increasing pressure imbalances between gas and liquid which accelerates wear. The piston seals generate a significant amount of pressure against the accumulator wall, causing friction, and will eventually wear out. This wear will initially occur as small abrasions along their surface, causing small leakage, but will ultimately result in a catastrophic failure of the system due to fracture of the seal.

4.0 Concept Selection

Since we only had one requirement, to keep nitrogen out of hydraulic fluid, we could not use a scoring matrix. To select out best design concept we had to use another method. Since the EPA has been working on this project for the past ten years we removed the ideas that the EPA has already tried from consideration. This removed the single seal piston, elastic bladder, metal bellows and the impermeable membranes. The single seal piston design showed great results in testing, but when put into the actual system yielded poor results. The elastic bladder designs have been tried with many different kinds of coatings, but still do not yield satisfactory results. The impermeable membranes work very well for the first cycle, but after a few cycles of the system they break down from the constant folding and unfolding. The metals bellows designs are far too expensive to be practical.

Once we removed all the ideas that the EPA had already tried we removed the ideas that would not perform any better than the current designs. The inflatable seal would create a problem with pressure differences between the seal and the two fluids. This would actually perform worse than the current single seal piston design. Also the flexible diaphragm designs were removed because they could not sweep out a large enough area to create enough change from high and low pressure. The spring piston idea was also removed because of the large pressure difference over the seal.

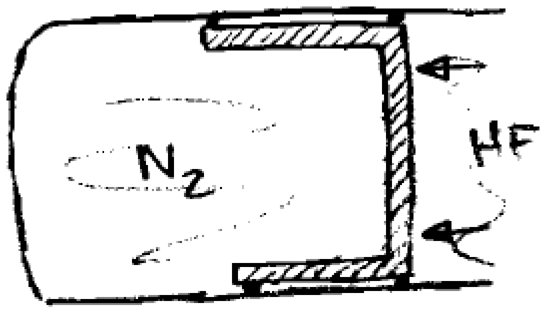
With the poor performing designs removed we then removed the ideas that we could not make in the timeframe of the class. This removed the metal and plastic bellows designs. They may be able to keep the nitrogen from getting into the hydraulic fluid, but are too complicated of a design. With all the interlocking pieces involved in these designs if one of the connections is not correctly made the whole system will fail. The team decided it was not worth the risk to build one of these designs.

The designs remaining were piston designs. We selected three of these designs with the least complexity, with the idea that fewer moving parts will lead to less of a chance of failure.

4.1 Elongated Piston Dual X-Ring Seal

The first of the top designs is the elongated, dual x-ring sealed piston shown in Figure 7 on page 13. This design uses a piston with two seals that are far enough apart that neither seal will pass over the same part of the cylinder wall. This will allow for the cylinder wall that is in contact with the hydraulic fluid to never be exposed to nitrogen. Any residue of the hydraulic fluid left on the cylinder wall will not be saturated by the nitrogen. A disadvantage to this design is that the length of both the piston and the accumulator will have to increase. Another disadvantage to this design is that a small pocket of air may be created between the two seals that would not change pressure with the system. This would cause a large pressure difference over the seals

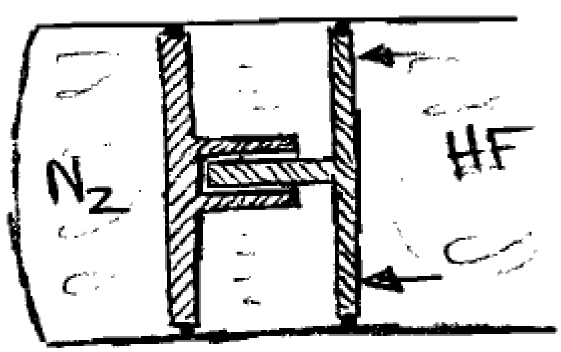
Figure 7 – Elongated Dual X-Ring Sealed Piston



4.2 Dual X-Ring Seal with a Fluid Filled Pocket

The second top design is the dual x-ring sealed piston with a fluid filled pocket shown in Figure 8 below. The pocket is filled with hydraulic fluid and the two parts of the piston are allowed to move together and apart to allow for pressure changes. This way the pressure difference over either seal should remain under approximately 2 psi. It is easier for the nitrogen to pass the seal, but the hydraulic fluid in the pocket should not be able to pass the seal into the hydraulic fluid of the system. The design allows for the hydraulic fluid in the pocket to become saturated, but not the hydraulic fluid in the system, so that there is no damage to the pumps. The disadvantage to this design is that if the saturated hydraulic fluid in the pocket becomes so saturated that nitrogen gas is in the pocket, then the second seal could still leak the nitrogen into the hydraulic fluid of the system.

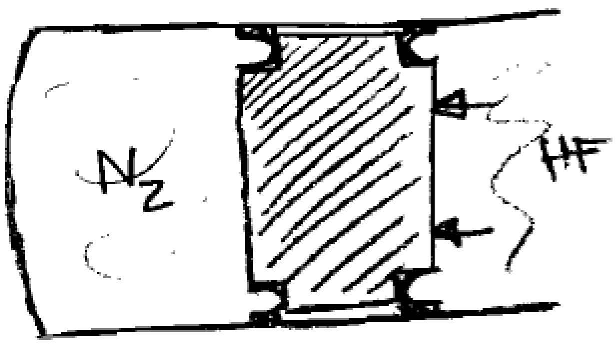
Figure 8 – Dual X-Ring Sealed Piston with a Fluid Filled Pocket



4.3 Dual U-Cup Sealed Piston

The last of the top three designs is a dual U-cup sealed piston design shown in Figure 9 below. This design uses the U-cup seal that is mounted with the open end of the U-cup facing the nitrogen on the nitrogen side of the piston and facing the hydraulic fluid on the hydraulic fluid end of the piston. This allows both the nitrogen and the hydraulic fluid to flow into the U-cup and push the seal against the cylinder wall and the piston. Using the pressure of the system to create a large pressure on the U-cup, this will seal against the cylinder wall and piston. The disadvantage to this design is if the nitrogen leaks into the small area between the two seals and increase the pressure behind the seal this could make the seals no longer work.

Figure 9 – Dual U-Cup Sealed Piston



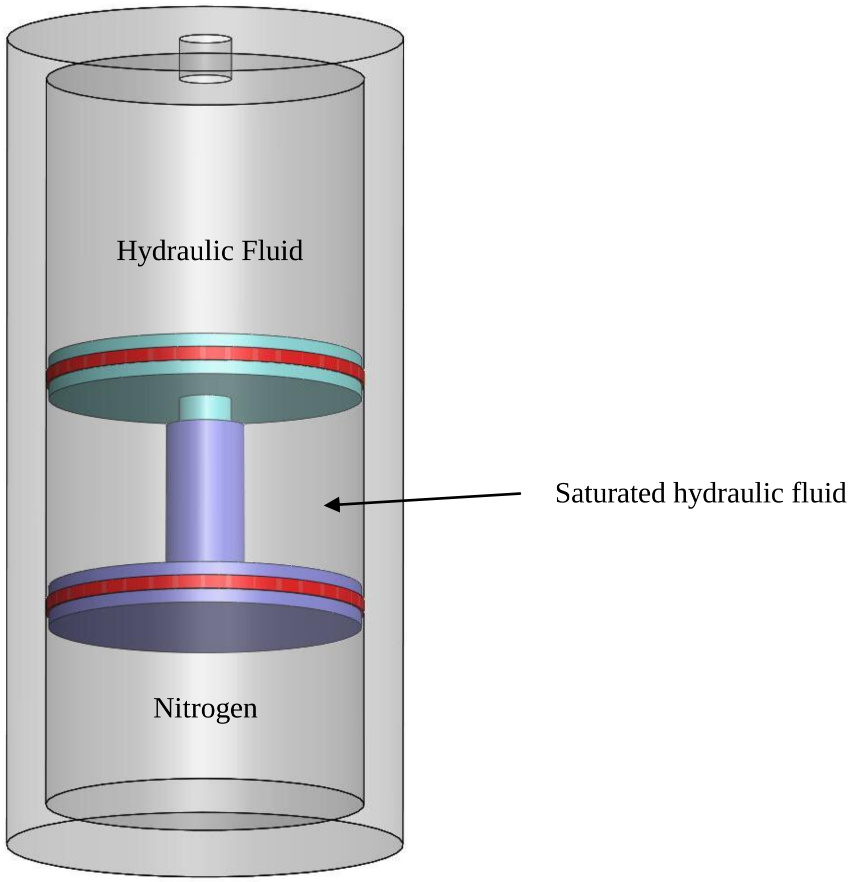
4.4 Alpha Design Selection

To decide on an alpha design the group discussed the top three designs. The elongated, x-ring sealed piston and the u-cup sealed piston could have an issue with pressure difference over the seals. That is why the alpha design that was selected is the dual x-ring sealed piston with fluid filled pocket. This has a possibility of becoming over saturated and leaking nitrogen past the second seal, but still holds the most promise.

5.0 Selected Concept Description

Our selected concept is the adjustable pocket piston design which can be seen in Figure 10 on page 15.

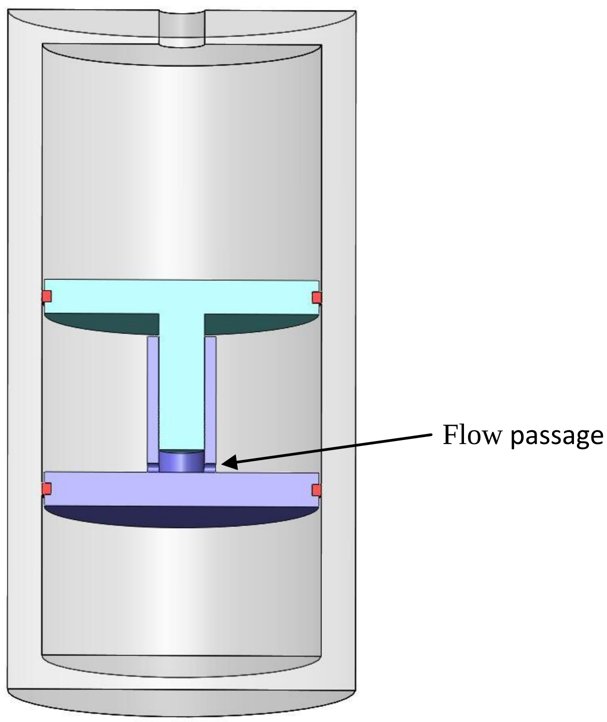
Figure 10 – Adjustable Pocket Piston Design



The piston is composed of two pieces, a top half and a bottom half which are allowed to slide relative to each other and form a chamber in between them. Hydraulic fluid is placed in the cavity, nitrogen is below the piston and the hydraulic fluid is above the piston. As the pressure of the nitrogen and hydraulic fluid changes, the two halves of the piston expand and contract to equalize the pressure between the three compartments. There is still a slight pressure differences (approximately 1-2 psi) over the seals due to the friction between the seals and the sides of the cylinder. There is a flow passage at the base of the bottom piston as seen in Figure 11 on page 16, to allow the hydraulic fluid to flow in and out of the bottom piston as the piston halves expand and contract.

The logic for this design is that it is harder for saturated hydraulic fluid to pass by a seal than it is for nitrogen to pass by a seal. So if nitrogen leaks past the first seal then it will be absorbed into the hydraulic fluid and not pass into the clean hydraulic fluid of the system. Also, once the hydraulic fluid in the chamber becomes saturated, then the leakage of nitrogen into the chamber would slow because the chamber would be full of saturated hydraulic fluid. If there were a film of hydraulic fluid left on the walls of the cylinder below the piston then no more nitrogen would be able to be dissolved into this film of hydraulic fluid because it would already be saturated. Then if the seal pulls the hydraulic fluid back into the chamber, there would be no more nitrogen being pulled back with it.

Figure 11 – Cross Section of Adjustable Pocket Piston Design



6.0 Engineering Design Parameter Analysis

When determining the design parameters for our prototype, we selected the needed materials first and developed dimensions and tolerances with that information. To ease manufacturing of the prototype, we decided to use one material for the piston with the exception of seals. The resulting dimensions were set to be easy to machine and scale, while providing considerable safety factors to allow for machining errors and material variances. We based our safety factor calculations on a single worst case scenario, which is that of the piston seal becoming caught on one point of the cylinder wall, experiencing five pounds per square inch (psi) of differential pressure between the sides of the piston before it is dislodged. The forces caused by this scenario are detailed in Appendix C.

6.1 Material Selection

The material for the piston prototype was chosen before dimensions, as it was determined to be the most important factor in gas transmission. This means our material selection was limited to metals, woods, plastics, and ceramics which are not permeable to gas transmission. Our choice of metal was further limited to common steels and aluminums due to cost and availability considerations. Of these two metals, aluminum was chosen for its weight and machinability, as the cost differences between the two materials are negligible. Of the aluminums available, 6061 has the widest availability and lowest cost; therefore we decided to machine our prototype out of 6061 aluminum.

6.2 Piston Diameter

The first parameter chosen was the diameter of the piston, and the rest of the piston was based around this dimension. The actual accumulator the EPA uses in their vehicles, and for which our piston is designed for, is 10 inches in diameter. After a brief review of the costs associated with building a piston of this diameter, we decided a full scale model would be far too expensive for our budget. Rough estimates of the cost for a 5 inch and 3 inch diameter model and test fixture were $800 and $350 respectively. Our sponsor decided that extra funding would be supplied for us to make a 5 inch diameter piston and test fixture. Using an accumulator diameter of 5 inches, the piston diameter was then dictated as 4.995 inches, allowing for the five thousandths of an inch between piston and accumulator wall specified by the seal manufacturers.

6.3 Piston Length

The length of the piston was chosen by studying the importance of aspect ratio in piston design. A rod with greater length than diameter was found to have very minimal frictional problems, as it was unable to rotate and bend enough within the confining cylinder to lock in place from the generated frictional forces. As our piston diameter was chosen to be 5 inches, the length would need to be greater than 5 inches. We therefore chose the minimum length of the piston in its compacted state to be 6 inches, with a fully expanded state of 7.5 inches. Expansion allows the piston to accommodate temperature and pressure variations, and small amounts of gas leakage past the first seal into the pocket.

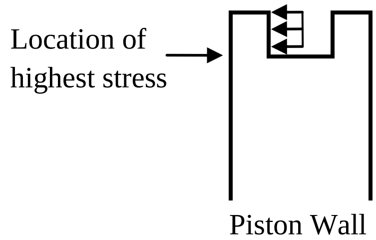
6.4 Piston Wall Dimensions

The piston wall thickness was partially determined by our piston diameter. The seal we selected for our 5 inch diameter accumulator has a width of 0.484 inches. We therefore chose to add an additional 0.25 inches on each side of the seal to support and prevent movement of the seal. This gives the piston wall a total thickness of 0.984 inches. The wall also has a groove 0.352 inches deep and 0.484 inches wide along its circumference in the middle of its length to contain the seal for the piston. The piston wall was designed to not be thin or carry a complex shape in an effort to ease manufacturing.

6.5 Piston Wall Thickness Stress Analysis

To determine whether the piston wall is sturdy enough to support itself during the worst case scenario, a stress analysis was performed on it. A sketch of this can be seen in Figure 12 below. We determined that the greatest stress experienced would be on the 0.25 inch lip containing the seal, stretched over a half inch wide area. This would cause a shear stress of 783 psi locally. The shear strength of 6061 aluminum is 30,000 psi, giving the part a safety factor of 38.3 at this location. The calculations used for this analysis are shown in Appendix D.

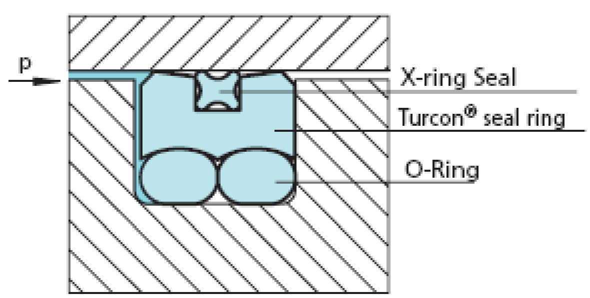
Figure 12 - Piston wall stress analysis set-up



6.6 Seal Selection

For our piston, we chose a Trelleborg AQ 5 series seal. This seal utilizes two o-rings that act as energizers to force the seal ring and the X-ring against the cylinder wall as seen in Figure 13 below.

Figure 13 - Cross section of Trelleborg AQ-Seal 5 series from Trelleborg seal catalog [5]



We chose this seal based on a phone conversation with an engineer at Trelleborg Seals and the advantages and specs listed in the Trelleborg seal catalog [5]. This seal is recommended for sealing around a piston with pressure on both sides. Some of the advantages listed in the Trelleborg Seals catalog were high sealing effect in applications requiring media separation, low gas permeation rate, and outstanding sliding properties.

6.7 Piston Inner Rod Dimensions

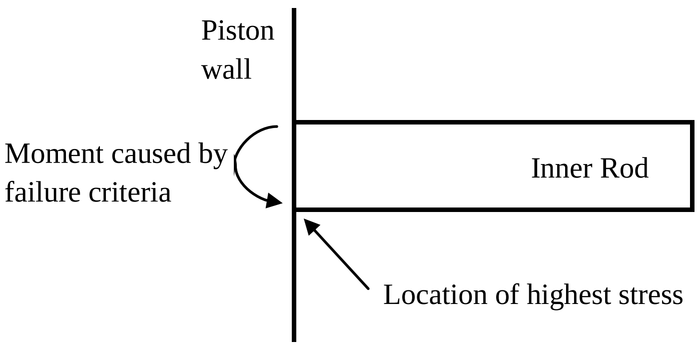
The male rod connecting the two piston walls has two important dimensions: diameter and length. We chose to have a diameter of 1 inch to allow for sufficient strength during welding. Another reason we chose a 1 inch diameter is because of the high availability of 1 inch tools.

The length of the rod is 4 inches, allowing the piston to measure 6 inches when fully compacted, as mentioned in Section 6.3 on page 17.

6.8 Piston Inner Rod Stress Analysis

To determine if the inner rod was thick enough to support the bending moment produced by the failure scenario, we performed a stress analysis to check for failure in bending. The most stress would be experienced at the joint between the rod and the piston wall, on the exterior surface, as detailed in Figure 14 below. This point receives a stress of 4,999 psi during the failure criteria. The yield strength of 6061 aluminum is 40,000 psi, thus our part has a safety factor of 8.0 at this location. The calculations used for this analysis are shown in Appendix E.

Figure 14 - Piston inner rod stress analysis set-up



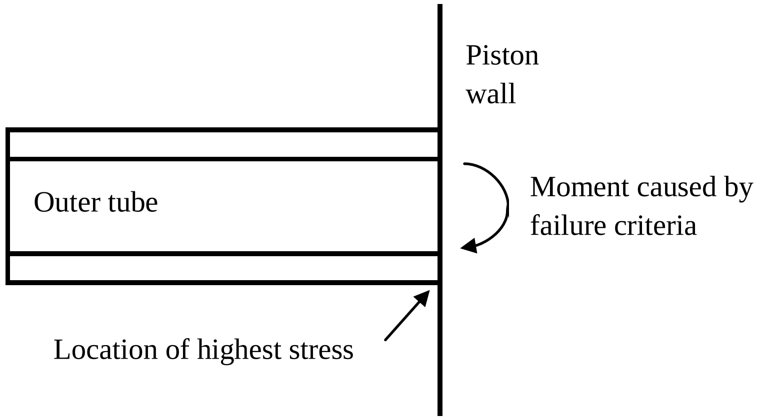
6.9 Piston Outer Tube Dimensions

The female tube connecting the two piston walls has three important dimensions: inner diameter, outer diameter, and length. The inner diameter was determined to be a few thousands larger than 1 inch to match the inner rod; this allows the rods to slide together while preventing any significant tilt angle between them. The outer diameter was designed to be 1.5 inches, giving the tube a wall thickness of ¼ of an inch. The length of this outer tube was set to be 3.9 inches. This prevents the tube from contacting the far piston wall.

6.10 Piston Outer Tube Stress Analysis

To determine if our tube was thick enough to support the bending moment produced by the failure scenario loading, we performed a stress analysis to check for failure in bending. The most stress would be experienced at the joint between the tube and the piston wall, on the exterior surface. This is displayed in Figure 15 on page 20. This point receives a stress of 1,060 psi during the failure criteria. The yield strength of 6061 aluminum is 40,000 psi, thus our part has a safety factor of 37.7 at this location. The calculations used for this analysis are shown in Appendix F.

Figure 15 - Piston outer tube stress analysis set-up



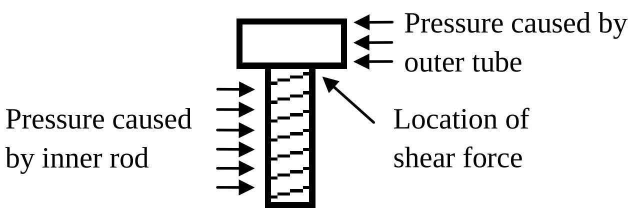
6.11 Piston Pin Dimensions

The pin used to hold the piston together during assembly is a 10-32 x 3/8 inch set screw. This set screw size was chosen due to an appropriate length and a 3/8 inch diameter; this diameter is the smallest common size that has an appropriate safety factor.

6.12 Piston Pin Stress Analysis

To determine if the pin could withstand the failure criteria when the cylinder is already fully extended, we performed a stress analysis. The maximum force it will experience is a shear created between the inner rod and the outer tube, which is illustrated in Figure 16 below. The pin is only transmitting force when it is forced against the end of the slot due to leakage past the seals. The pin transmits a total force of 97.97 pounds at its worst case scenario, which would cause a shear stress of 2661 psi on the pin; however a yield strength of 21,000 psi gives the pin a safety factor of 7.89. The calculations used for this analysis are shown in Appendix G.

Figure 16 - Piston outer tube stress analysis set-up



6.13 Piston Slot Dimensions

The slot in which the piston slides is 1.875 inches long, which gives the pin a travel distance of 1.5 inches. This is the same distance mentioned in our overall piston length dimensions, the pin travels the full length of its slot as the piston expands and contracts fully. The slot is also 0.375 inches wide, allowing the pin to rest inside the slot without a significant tilt angle.

6.14 Manufacturability and Assembly

When designing our prototype we decided to build it at half scale to reduce material costs while still maintaining a reasonable size for testing. The prototype design itself was determined by the tools and skills we had for machining, and the materials we could reasonably obtain. We designed each piston half to consist of two parts welded together; this decision was made because we had no access to metal casting, and machining the piston from a single block of material would be too costly and time consuming. We left about an inch of material sticking out past the welded end plate to allow for the piece to be held in a four-jaw chuck on a lathe, so that we could machine all surfaces at once and not need to worry about a lack of concentricity. To ease the assembly of the piston and seal, we machined a chamfer into the inner edge of each piston half. During assembly we found that this chamfer was not sufficient for expanding the seal into the seal groove, so a separate tool was made for that purpose. As the pistons were intended to be one-of-a-kind parts, we were able to adjust machining as needed for fit and assembly.

Our final design is simply a scaled up model of our prototype; however, we expect different manufacturing techniques to be used to minimize cost and labor. Each piston half can be formed by casting: this would eliminate the welding and assembly steps present in our prototype design. The cast part would still need to be ground down to a smooth finish, but the cast piece would be made closer to the final size so that the amount of finishing machining would be minimal. Additionally, with a large enough run of the parts, standardized tools could be made, which would allow for a larger amount of automation of the part machining and assembly.

Materials for the piston were selected based on availability, and this selection was verified using the Cambridge Engineering Material Selector. We required the pistons to be made from metal as the gas transmission properties were ideal. We specifically selected aluminum because of machinability and cost. Aluminum is far easier to machine than steel, and when calculating the stresses in the system, we determined that aluminum would easily handle expected loading in our prototype test fixture. Additionally we analyzed cost and found a local supplier providing aluminum at a much cheaper price than elsewhere. The available stock determined some of the specifications of the piston, specifically the diameter of the connecting rod. Seals were chosen based on correspondence with Trelleborg. The particular seals we went with are specified as being used in applications requiring separation of a gas from a liquid. The thickness of this seal was what determined the thickness of the end plate as well as the depth of the seal groove.

Materials readily available were Aluminum 6061 and Steel 1018. Both of these materials were found to fit well within our stress and strain requirements by looking at their entries in CES. Both materials display favorable characteristics with regards to reactivity with hydraulic fluid and nitrogen, but the aluminum was cheaper and easier to machine.

6.15 Design for the environmental

The automatic transmission fluid that is being used as hydraulic fluid in the test fixture is known to be toxic, and has regulations regarding its disposal. To meet environmental qualifications with our fixture, it must not leak hydraulic fluid into the surrounding environment.

The full scale design will be in an enclosed accumulator during its lifetime, so we do not anticipate any difficulties with the piston system directly contaminating the environment. Care must be taken when disposing of the hydraulic fluid in the system, but during operation there should be no leakage of the fluid to the outside of the accumulator. The full scale piston is to be made from steel, which is easier and cleaner to produce than aluminum, and can be recycled.

6.16 FMEA

To determine the risk factors inherent in our system, we used the DesignSafe FMEA software to analyze risk and develop ways to mitigate the risk. Our test fixture presents safety concerns in that it is a pressurized vessel that could catastrophically fail if any of the components holding the pressurized fluid in were to fail. Additionally, during testing, the pistons will cycle through the fixture, creating constant changing pressure, increasing the possibility of a fatigue failure. We expect the test fixture to last a long time; no specification was given for the exact lifetime of the fixture, but with safety factors in excess of 7 we had hoped it would at least last through testing.

The full size piston and accumulator present similar concerns as far as a pressurized containment vessel is concerned. The accumulator in use by the EPA is rated for much higher pressures than our test fixture, and has been in use for some time now, so the risks of an unforeseen catastrophic failure of the accumulator is minimal. The piston itself is a scaled model of the prototype, so we expect any failure modes present for the prototype to also be present on the full scale model. According to the customer specifications, the piston seal system needs to last for at least ten years before replacement, so long term failure modes need to be taken into account when weighing risk factors.

As shown in this section, we calculated safety factors for all components of the piston, and have a minimum factor of 7.8. This number is sufficient to minimize risk of a catastrophic failure due to material failure when operating within bounds. We determined using the software that we needed to watch for failure of the test fixture during operation, wiring hazards, and hydraulic fluid leakage. The first and the last are related, in that if the fixture ruptures, hydraulic fluid will leak. Otherwise, even if a small non-critical leak is present, under high pressure a large amount of hydraulic fluid will be forced out, contaminating the environment. We discovered this during testing, as detailed in Section 11 on page 34 – during initial pressurization, a fill cap was found to not have been glued properly due to contamination with hydraulic fluid. The cap came off, resulting in a large amount of hydraulic fluid leaving the test fixture and posing a health hazard. The failure of the cap due to this mode was not considered during initial preparation of the

DesignSafe chart, as it was not perceived to be a threat at the time. Recommendations for the redesign of the system with regard to this component are in Section 13 on page 35.

We found wiring hazards to have a high risk factor as none of the team members are certified in electrical work. The system contains a solenoid that draws approximately 0.38 amps at 12 volts, which is a health hazard if improperly handled during operation. Extra precaution was taken during wiring, and will be taken during any future testing.

Our DesignSafe chart is located in Appendix I.

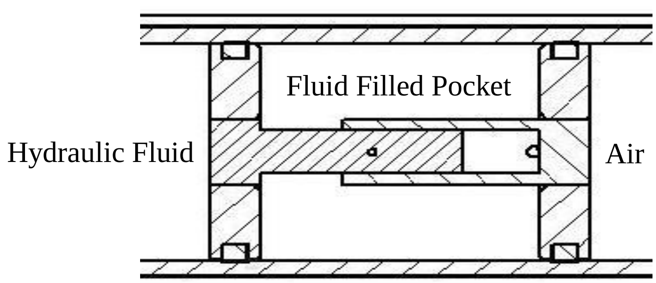
7.0 Final Design Description

The final design is a modification of the fluid-filled pocket concept on page 15. This concept has been altered slightly, with the addition of an extension limiting device to prevent separation of the two piston halves, as detailed in the text below.

7.1 Design description

The design, as shown in Figure 17 below, consists of two pistons joined with concentric connecting rods, and a fluid-filled pocket filling the intermediate space. The piston is to be made from aluminum, as it has good strength characteristics and has a low environmental impact.

Figure 17 - Cross Sectional View of Final Piston Design



This pocket is filled with hydraulic fluid, and can change volume to allow for temperature and pressure changes. Assuming a perfect seal, the volume change is minimal, at around 3 psi pressure difference due to friction of the seal against the cylinder wall, the fluids on both sides of the pistons will be at very similar pressures. As the seals will not be ideal, some nitrogen will leak past into the pocket and be absorbed into the hydraulic fluid. As nitrogen leaks into the pocket, the compressibility of the fluid in the pocket increases. The volume in the pocket is allowed to increase as the system cycles from high and low pressure. As the piston system cycles in the accumulator, heat generated due to friction will affect the saturated mixture differently from the unsaturated fluid, resulting in increased pressure under fixed volume conditions. This design prevents leakage by allowing the pocket to expand and contract to keep pressure inside and out at an equilibrium.

The leakage of nitrogen past the first seal is inevitable over the ten year lifespan of the piston due to imperfect seal design and permeability to gases. The seals are less permeable to liquids, due to an increased particle size and lower kinetic energies; a saturated gas-liquid mixture will leak through the second seal slower than nitrogen gas leaks through the first seal. This will help to increase the lifespan, as the gas will need to leak past the first seal, saturate the hydraulic fluid in the pocket, leak in mixture form past the second seal and a reduced rate, and saturate the working hydraulic fluid. These extra steps will reduce the rate at which nitrogen can leak into the system.

7.2 Method of Operation

A hydraulic hybrid vehicle uses a hydraulic loop and accumulators to store energy from a power source (usually a diesel motor) when it is not otherwise in use by the drive system. When this excess energy is present in the system, the hydraulic pump pushes hydraulic fluid into the accumulator, pushing against a piston, and compressing nitrogen gas. This gas acts as a spring, such that energy can be extracted as the hydraulic fluid pressure is decreased. The gas must be kept separate from the fluid, as oversaturated hydraulic fluid in the system leads to cavitation in the pump and motor, resulting in damage. The final design resists the flow of nitrogen gas into the working fluid as described above: a saturated mixture within the pocket is less able to leak through a seal than pure gas, so the rate of transmission is reduced. The pocket is allowed to change volume as the system pressure inside the accumulator changes, preventing a large difference in pressure over the seals that would result in leakage. Extreme expansion is limited by a set screw attached to the inner cylinder and projecting through a slot in the outer cylinder, as shown in Figure 19 on page 26, to keep the pistons from separating fully. Such separation would likely result in non-axial motion of the piston disks, allowing fluid to leak through spaces between the seal and accumulator wall.

This design limits the leakage of nitrogen over other existing designs by providing an enclosed area for the nitrogen to first saturate, and keeping this saturated volume sealed against the unsaturated hydraulic fluid elsewhere in the system. The premise of the concept is that a saturated fluid will have a more difficult time leaking through a seal than a gas. In other existing single seal designs, the gas will eventually leak past the seal and saturate the hydraulic fluid in the system. This design takes this knowledge and adds a buffer to increase the time for the nitrogen to make its way through the seal and out into the main hydraulic loop.

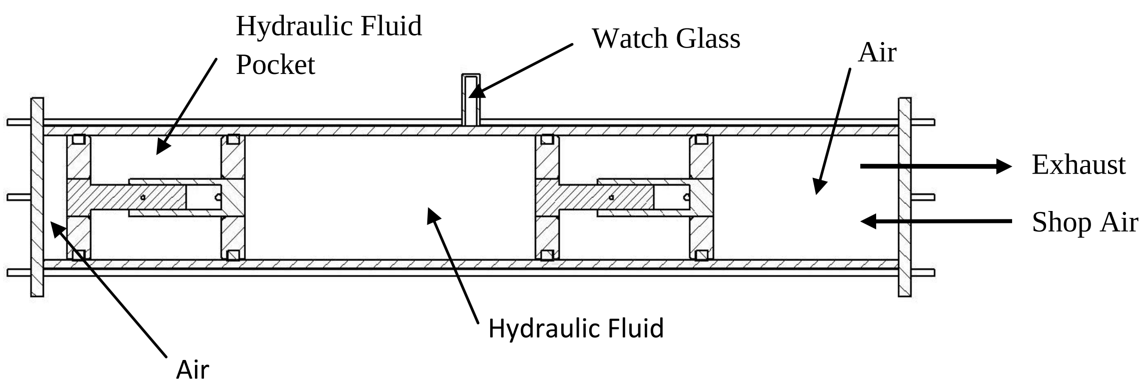
The team did not have the funds to perform a test of the full scale design. Our scaled down prototype was to be tested as a proof of concept. If our test had shown our design could produce the desired results, the EPA would perform a full scale test of the design.

8.0 Prototype Description

Due to the limited budget and timeframe for this project, certain aspects of the final design must be altered when manufacturing a prototype for testing. The design functionality is simple enough to be replicated fully in the prototype, though due to material costs and machining time available, the model must be reduced in scale. As shown in Section 6.2 on page 17, the prototype will be scaled such that every dimension is halved, which will reduce material usage to an eighth of that required for the full scale system. As mentioned in Section 6, we will be using 6061 aluminum and Trelleborg AQ-Seal 5 series seals for the prototype.

Functionally the prototype will be identical in every aspect to the full scale model, and will be tested in a scaled down accumulator based on the full size system. This test fixture accumulator, pictured in Figure 18 below, will position two pistons in a single pipe. The use of this fixture is detailed more fully in Section 10 on page 31.

Figure 18 - Cross Sectional View of Test Fixture with Labels



The prototype would validate the final design by showing expansion and contraction according to pressure variations, and by not allowing leakage of air during testing. If the data obtained from testing the prototype in our test fixture scaled to within the desired specifications, then we could assume that the full scale design would have meet customer requirements.

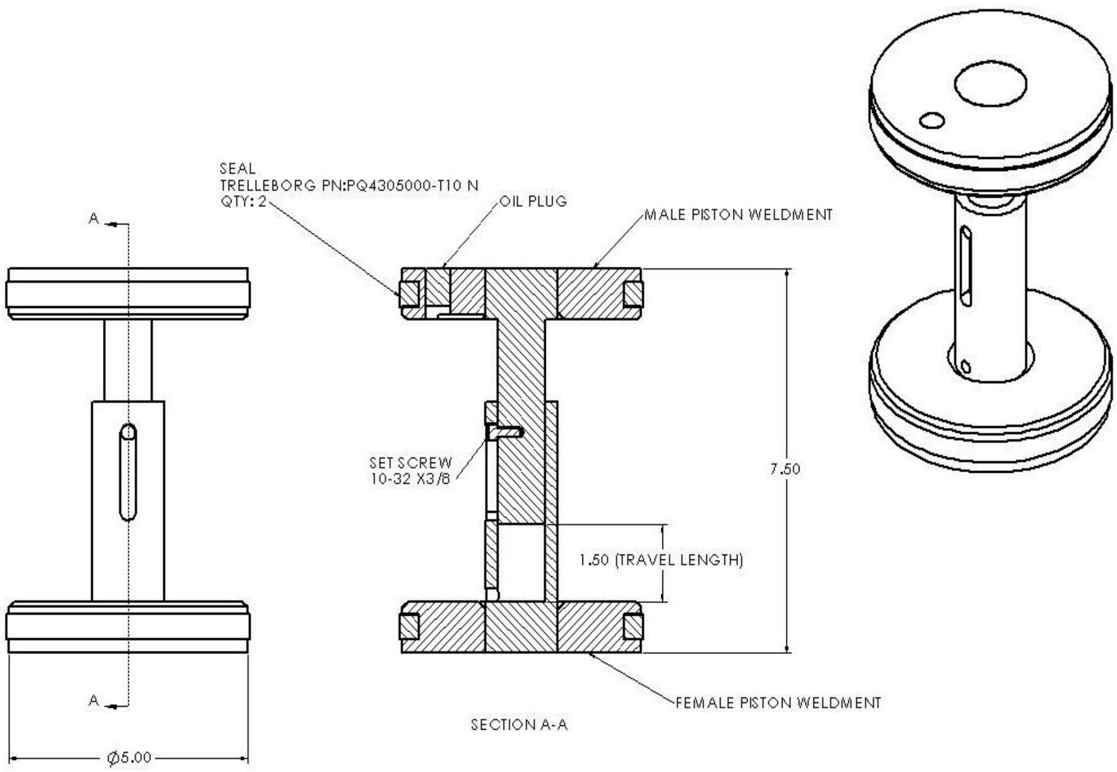
9.0 Initial Manufacturing Plan

We created a prototype to show proof of concept and predict the performance of a full scale model. The bill of materials for our prototype can be seen in Appendix H.

9.1 Prototyping of the Pistons

An assembly drawing of our piston prototype can be seen in Figure 19 below. The piston is composed of two halves, a male piston weldment and a female piston weldment. The piston assembly also includes a set screw, oil plug and two seals.

Figure 19 - Piston Prototype



To create the male and female piston weldments which can be seen in Figure 20 on page 27 and Figure 21 on page 28 respectively, we will used 6061 Aluminum round stock of 5 ½" and 1 ½" diameters. We used a band saw, lathe, and mill to machine the two halves of the piston. The 5 ½" and 1 ½" diameter round stock pieces were welded together after a 1½" diameter hole was bored through the 5 ½" diameter piece. We used the 1" overhang length as material for the lathe chuck to hold on to. This allowed us to machine the seal groove and the inner diameter (for female weldment) or outer diameter (for male weldment) in one setup to ensure the best concentricity of the piston end and seal groove to the male or female part of the piston. This was done at a lathe speed of 200 RPM for the seal groove and 500 RPM for the inner diameter (for female weldment) or outer diameter (for male weldment). The concentricity ensured that the seals slide concentric to the inside diameter of the cylinder to create the best performance. To limit the travel of each piston half we machined a slot in the female piece and a tapped hole for a set screw in the male piece using a mill. The slot was machined at a tool speed of 1000 RPM and the tapped hole was drilled at 800 RPM. In order to fill the piston cavity with hydraulic fluid, we machined a 0.25" tapered tapped hole in the male piston weldment using a mill at 1000 RPM and a ¼" NPT tapered pipe tap.

Figure 20 – Male Piston Weldment

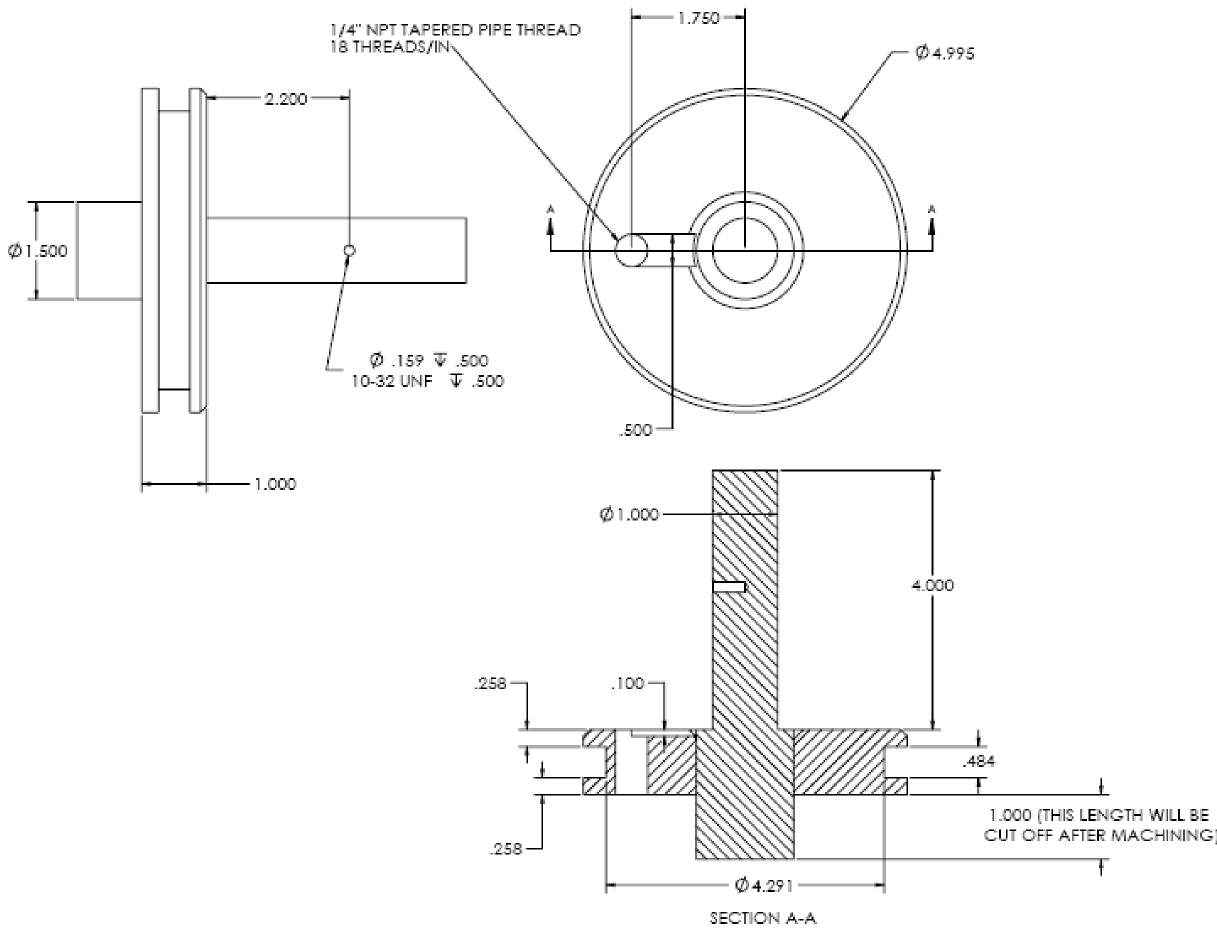
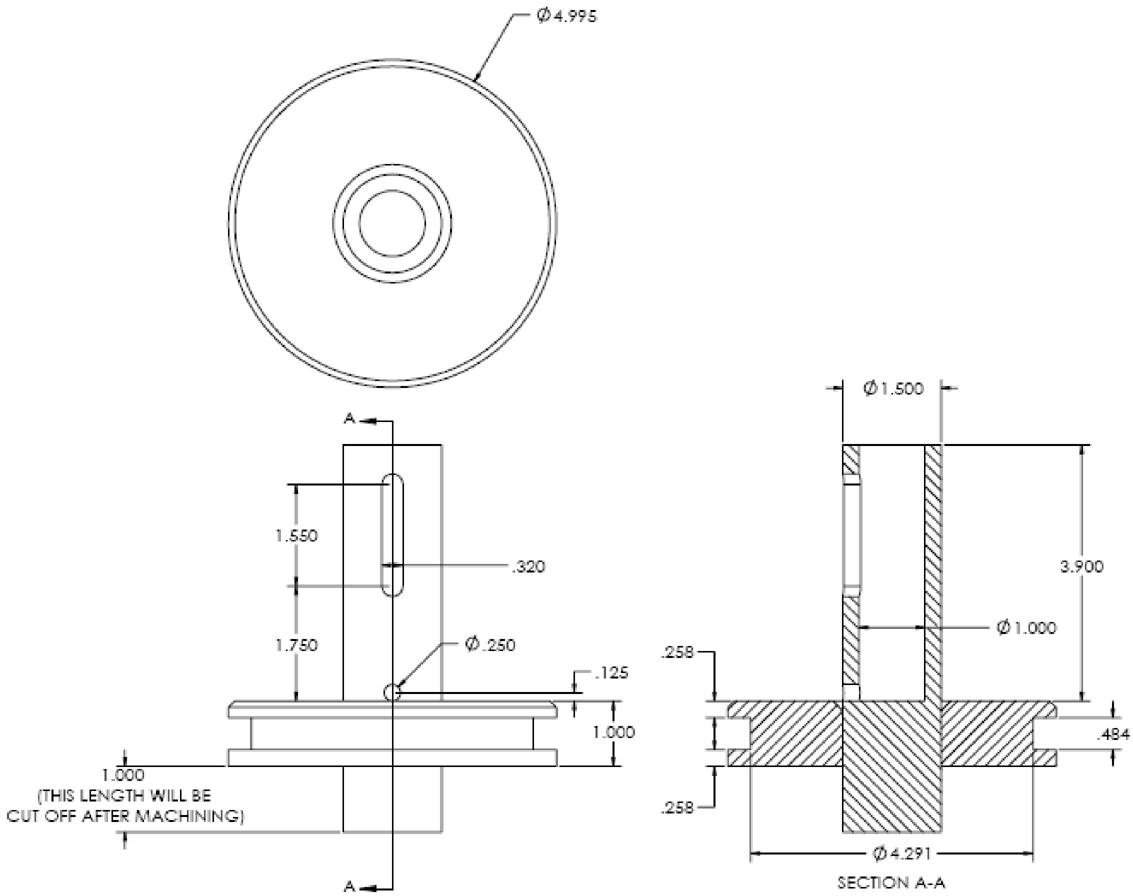


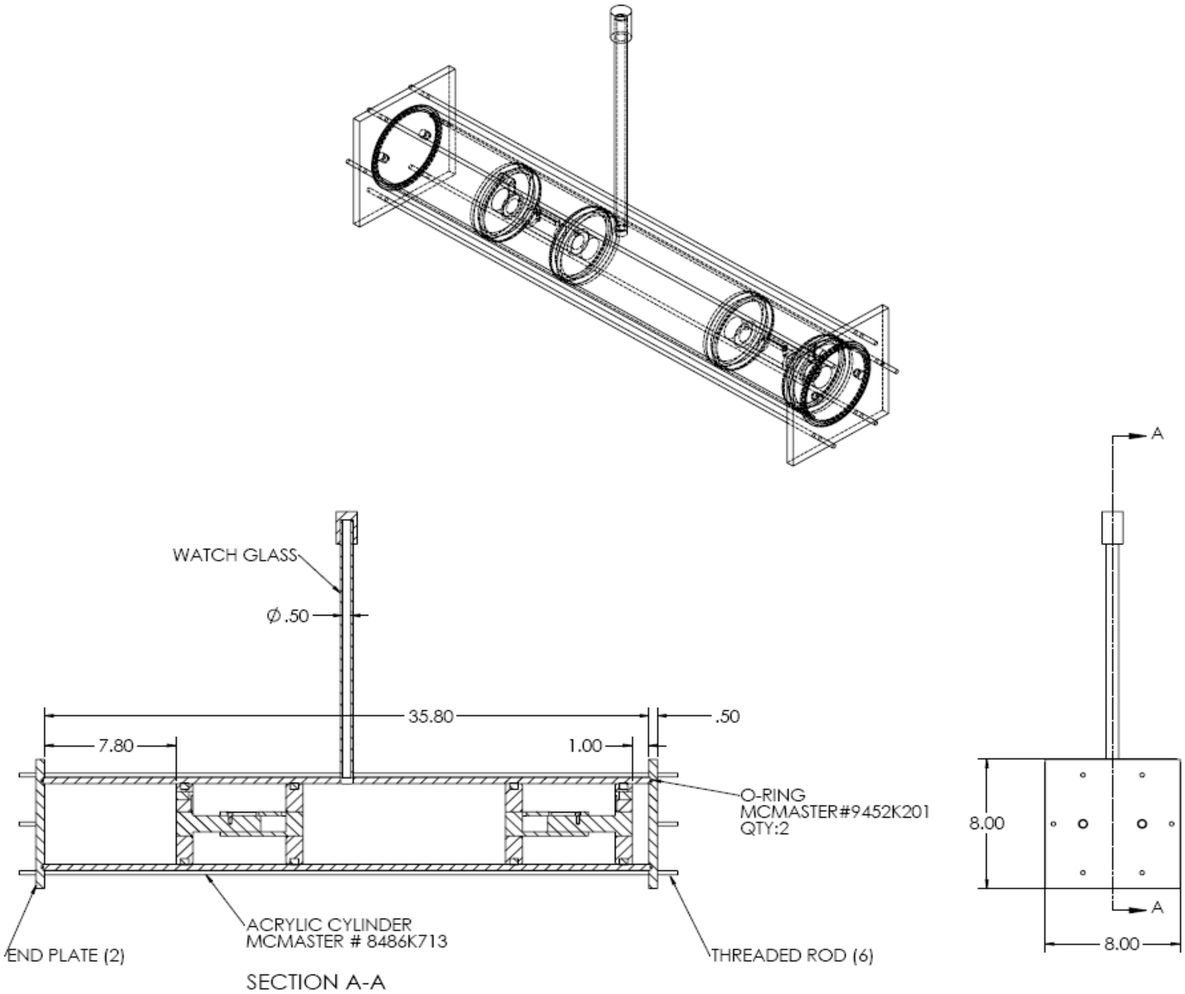
Figure 21 – Female Piston Weldment



9.2 Prototyping of the Test Fixture

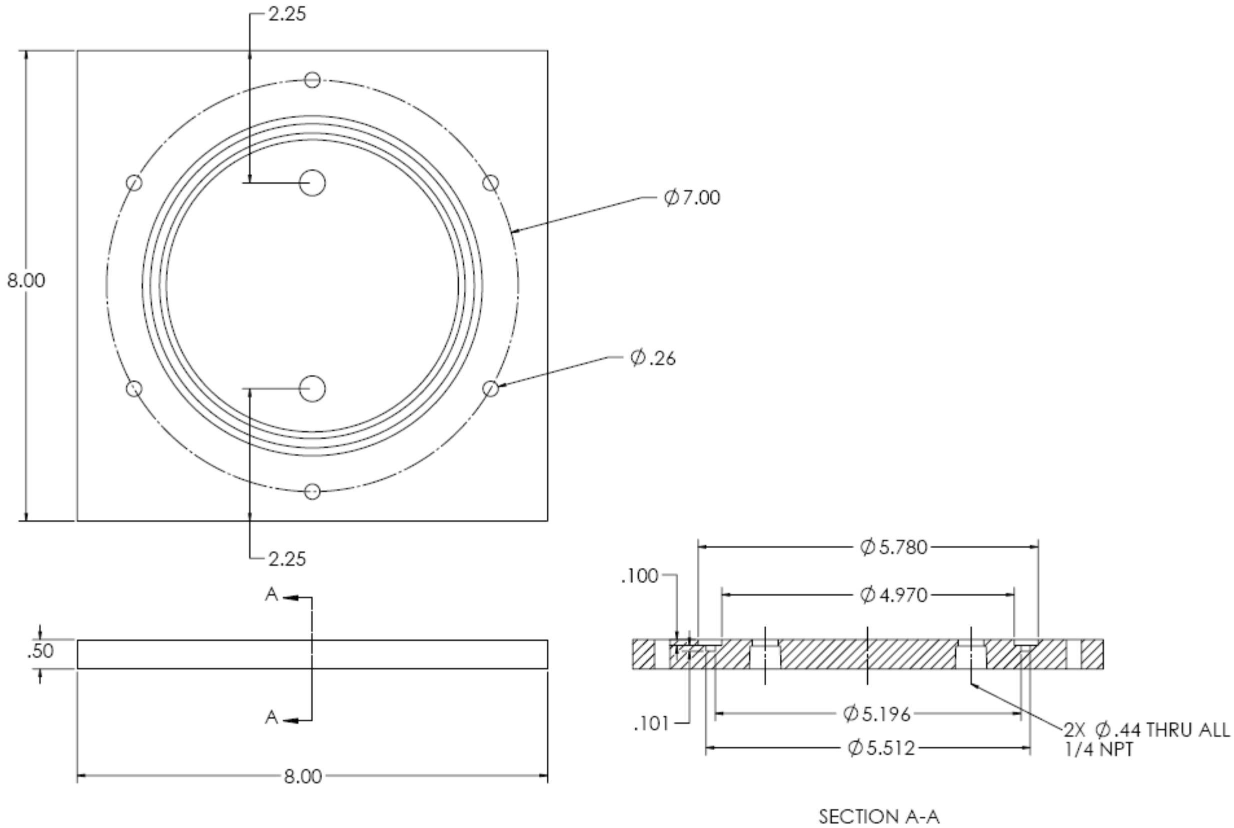
To test our sealing concept with dynamic motion of the piston, we have designed the test fixture seen in Figure 22 on page 29. There is hydraulic fluid between the pistons and in the pocket of each piston. There is air between the pistons and the cylinder ends.

Figure 22 – Test Fixture



The cylinder for our pistons is an acrylic cylinder 36" long with 5" inner diameter and 3/8" wall thickness. We chose to use acrylic because it is clear, which allows us to see where the pistons are, and also allows us to see any air pockets that developed or see any air leaking past a seal. We used two steel plates with o-rings to seal the ends of the cylinder. The end plates, seen in Figure 23 on page 30 were machined with a CNC mill with a tool speed of 1200 RPM to create the circular grooves. We machined the end caps for fittings of air inlet/exit valves and pressure gauge using a mill at 800 RPM and a hand tap. We used 6 .25" threaded rods to hold the end caps around the cylinder. There is a watch glass attached to the top of the cylinder for air to collect in and be measured if any leaked past the seals. The watch glass was cut to length using a band saw, and the cap was cut and drilled using a band saw and mill. To get dynamic motion of the pistons, one end of the cylinder was connected to shop air pressure. The pressure in the cylinder was to be varied from 0 psi gauge to 100 psi gauge, which would cause the pistons to move back and forth. To automate the cycling of the pressure in the cylinder, we used a solenoid valve which was controlled by a signal generator emitting a square wave and a power supply to increase the current.

Figure 23 – Cylinder End Plate



9.3 Cost analysis

The total cost of the prototype is estimated based on the total cost of parts and the number of hours the team worked on the project with appropriate labor and shop costs. From receipts, the cost of two prototype pistons for our test fixture totaled at $390, with the test fixture itself costing $495. We spent a total of about 110 man hours in the machine shop working on the manufacture of the prototype, with an estimated hourly pay of $20 resulting in a labor cost of $2200. Therefore, we estimate the total cost of the prototype to be $2590, with the total overall cost of the prototype and test fixture being $3085.

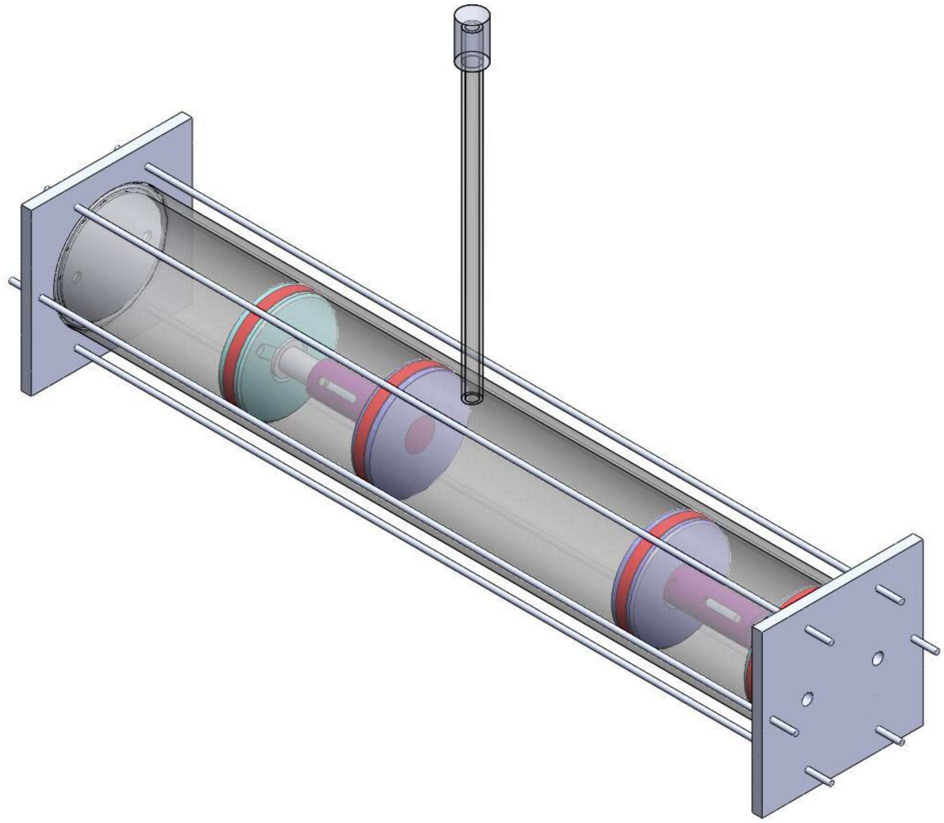
A cost estimate of the full sized system is harder to estimate, as we are not sure of the amount of machining time required for various steps.

10.0 Validation Plan

For this project we had one primary customer requirement and four secondary requirements. The secondary requirements were created by the team to assist in designing a working prototype, but are not requirements from the customer. Low cost, low complexity, light weight and ease of manufacturing are the four secondary requirements. For the requirement of low cost we set the target value at 250 dollars or less for the prototype, our prototype cost 195 dollars. We did meet this requirement, but our test fixture requires two prototypes which cost 390 dollars total. The target value for the requirement of low complexity is one moving part. We did not meet this requirement because there are two moving parts of the design, the male and female ends of the piston, which were necessary to meet our primary requirement. The target value for the requirement on weight was 5 lbs or less, and our piston weighs 4 lbs. The requirement of ease of manufacturing has a target value of four or less parts, and we have twelve. We did not meet this requirement because our seals are complex, multiple part seals, and we had to add two bolts to the original design. We have met 2 out of the 4 secondary design requirements.

The primary requirement for this project was a low leakage rate. The target value for leakage was 5.3 grams, or less, of nitrogen per gallon of hydraulic fluid per 10 years. To prove that our design was capable of providing the desired results we had to create a test fixture and run a test with the prototype. Since we did not have the budget to construct a full size and full pressure system, we had to make a smaller scale prototype for testing. The test fixture can be seen in Figure 24 on page 32.

Figure 24 - Test fixture for prototype



To test the prototype we constructed a test fixture. The test fixture is a three foot long, five inch inner diameter, three eighths inch thick acrylic tube with steel plate end caps. Six long quarter inch diameter rods connect the end caps to hold the cylinder shut. There is a one inch inner diameter quarter inch thick acrylic tube that acts as a watch glass. This tube is inserted perpendicular to the large cylinder and is used to fill the cylinder with hydraulic fluid before it is capped off. There are two prototype pistons in the large tube with hydraulic fluid between them. There is also air between the pistons and the end caps of the cylinder. There is a two way value in one of the end caps that will be used to set the amount of air on that side of the cylinder, and a three way solenoid valve on the other end cap, along with a pressure gauge. The solenoid valve had constant shop air attached to it, and it was connected to a function generator and a power source. This allowed us to set up a square wave such that when the value was positive the valve opened and compressed air pressurized the system. When the value was zero the valve closed off the compressed air source and opened to atmosphere, which depressurized the system.

Once the test fixture was completed and set-up we planned to start testing. One member of the group was required to be present to turn on the power source and start the function generator.

Once they were turned on, the system would start cycling. At least one member of the group would be present to observe the test and see if everything was running correctly. This member would also be watching to see were leakage was coming from, if there was any. We would have run the test for a week and tried to get 10,000 cycles, which was representative of one year in service. Once we were done cycling the system we would bring the system back to atmospheric pressure and let it sit overnight to let any air come out of solution and gather in the watch glass. Then we would have measured the length of the watch glass tube that was filled with air and calculated the volume.

Once we had the volume of air that leaked we would have used the density of air to find how many grams had leaked. Once we reach 10,000 cycles in testing we would have to multiply the amount of leakage by a scaling factor of 10 to see how much leakage would have occur if we had reached 100,000 cycles. The circumference of the piston is the only place where leakage can occur. Since the circumference of our prototype is half the circumference of the full scale design, we would have to multiply the grams of nitrogen by two to get the amount of leakage for the full scale design, but we are using two pistons in our test fixture so this negates the factor of two.

We would have been using compressed air instead of nitrogen in our prototype test. There is compressed air in the X50 room, and since air is mostly nitrogen this is a negligible difference. The full scale design will be run with a low pressure of 2000 psi and a high pressure of 5000 psi, but we would have been running our prototype with atmospheric pressure as the low pressure and shop air pressure, which is about 100-120 psi, as our high pressure. We did not have access to compressed air or nitrogen at the pressure levels of the full scale design, and the materials needed to handle those pressures were well out of our budget.

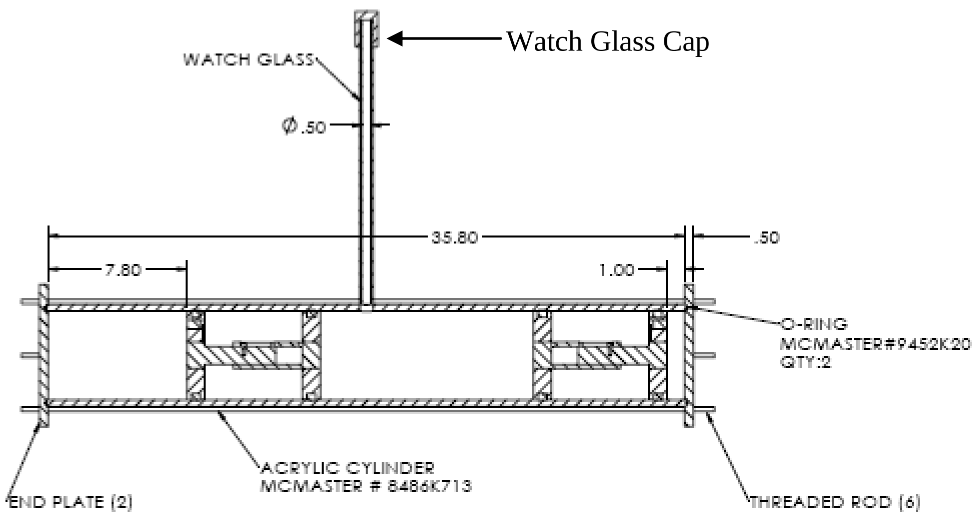
What causes the leakage over the seal in the piston design is the pressure difference over the seal. This pressure difference is estimated to be about two psi. The prototype is designed, like the full scale design, to have an almost equal pressure between the nitrogen, hydraulic fluid in the pocket, and the hydraulic fluid in the system. This means that there should be only a 2 psi difference over the seal for any pressure range, so even though we would not have tested at the full scale pressure, we would have been able to see similar results.

The difference in pressure of our prototype test and the full scale design is quite large. At the pressures in the full scale design the hydraulic fluid can actually compress, which will cause the fluid filled pocket to expand and contract as it cycles through the range of pressure. This would only occur in our test of the prototype if air leaked past the first seal and into the pocket, allowing for the gas to expand and contract in the packet. Section 11.0 on page 34 discusses our test results.

11.0 Test Results

We attempted to validate our design as previously discussed using the test fixture we designed and built. We planned on running our design through 10,000 cycles of 0 to 100 psi gauge and back, which would be completed in approximately 25 hours. Testing would have been completed in an assembly lab provided for our use by the University of Michigan. We assembled the test rig, filled it with the necessary 10 quarts of hydraulic fluid, sealed the system, and installed the function generator and power supply required to automate the process. However, when we first triggered our solenoid to pressurize the system, the cap on the watch glass, shown below in Figure 25, was unable to handle the required pressure and was dislodged. This resulted in a massive leak of hydraulic fluid which contaminated the testing laboratory.

Figure 25 – Test Figure with watch glass specified



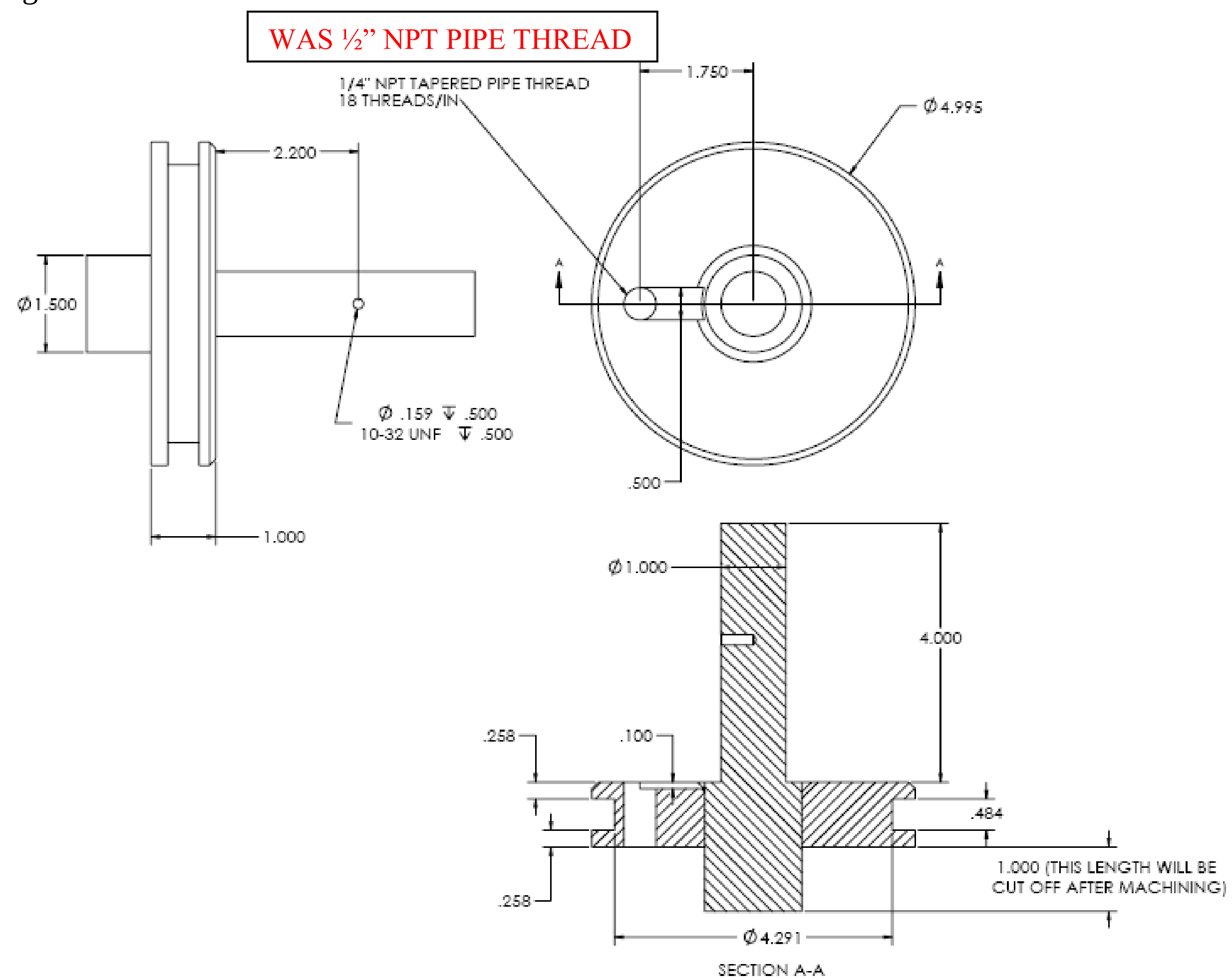
In response to this incident, the University immediately suspended our testing of the accumulator system. We were therefore unable to provide any testing results to validate our design.

After performing a review of the situation, we concluded that our test fixture failed due to contamination by hydraulic fluid of the glue used to fuse the acrylic cap in place. If this test rig were to be re-assembled and tested we recommend that the watch glass be sealed using a steel plate with grooves to position a face contact o-ring and held in place using a second set of steel rods. The steel rods would stretch down vertically to a pair of mounts secured around the center of the accumulator at the base of the watch glass. This setup would be similar to the two endplates which sealed the ends of the accumulator.

12.0 Engineering Change Notice

We made a change to the size of the brass oil plug to be used in the male piston weldment, which can be seen in Figure 26 below. It was changed from ½" NPT pipe thread to ¼" NPT pipe thread because we realized that the ½" NPT pipe thread was larger than necessary to fill the cavity of the piston with hydraulic fluid. There were no other changes to our design from what we had established at Design Review 3.

Figure 26 – ECN



13.0 Discussion

During the initial pressurization of our cylinder, the watch glass end cap came loose and hydraulic fluid sprayed out. We believe that the glue failed because it was contaminated with the hydraulic fluid when in was drying. If we were to redesign the watch glass end cap, we would use a similar design as in the end plates for our main cylinder. The end cap would be composed of an aluminum plate with an o-ring groove, an o-ring for a face contact seal, and threaded rods to hold the plate down. This would provide a mechanical means of holding the end on which would not allow it to come loose.

Before the pistons cycle in the cylinder, the two halves of each piston need to be positioned in such a way so that they are allowed to expand and contract. The halves need to be allowed to expand to accommodate any nitrogen leakage into the cavity and also be allowed to contract to equalize the pressure inside and outside the cavity as the pressure in the accumulator increases. After pressing a piston in the cylinder, the two piston halves were completely compressed (the only relative travel that could occur between the two halves would be in the extension direction). We needed a method of expanding the piston halves about a half an inch to allow the small air bubble that was trapped in the piston cavity to compress as the system was pressurized to 100 psi. In a full scale model, the compressibility of the hydraulic fluid would also become important as the compressibility is about 1% per 1000 psi and the pressure would reach 5000 psi. To expand the piston halves, one option could be to drill and tap a blind hole in the center of the piston, and then use a threaded tool with a handle to pull the piston half back to the desired location. Another option could be to fill the cavity of the piston with hydraulic fluid using a pump, so that the pressure generated by the pump would force the two halves apart.

After filling the cavity of each piston with hydraulic fluid, there was an air bubble left in each piston that we could not get out. This was due to a chamfer that we put on the inside 5" diameter edge of each piston. This chamfer was machined to allow easier installation of the seals, but was found to be not needed as a special tool for installation of the seals was required. The chamfer trapped air between it and the cylinder wall so that when the cavity was filled with hydraulic fluid, a small air bubble remained. The presence of an air bubble is not critical, but since the goal is to keep air out of hydraulic fluid, the least amount of air already inside the cavity possible is desired.

14.0 Recommendations

We recommend that our prototype be tested with shop air pressure varying from 0 to 100 psi. The watch glass end cap should be secured by a mechanical method as explained above. The piston halves should be expanded part way before beginning the test which could be done by a method explained above in the discussion section.

If the design works as expected and has a low leakage rate, then a full scale, full pressure prototype should be created. Part of our designs merit would not be realized with a 0 to 100 psi version. Since the pressure goes to 0 psi gauge at each cycle, there would be no permanently dissolved nitrogen in the hydraulic fluid in the cavity of the piston. The nitrogen would dissolve when the system reached 100 psi but would come out of solution when it reached 0 psi. In an accumulator that cycle between 2000 psi and 5000 psi, nitrogen could be permanently dissolved into the hydraulic fluid in the cavity of the piston.

Some changes will be required for this project to be continues into future semesters. The watch glass and test fixture will need to be redesigned, with a mechanical mechanism for sealing the watch glass as stated in section 13 on page 35. Additionally, the fixture will need to have a valve on the end so that vacuum can be applied to pull all remaining air out of the hydraulic fluid before testing. Two more valves will need to be placed on either side of the watch glass tube so that a vacuum can also be applied to the hydraulic fluid in the packets of the piston. Then thorough testing can be done.

Another thing that a team can do is create another similar piston design. To conserve space in the accumulator, the piston should be mostly hollowed out so that nitrogen can fill in that spot. The next team could work on making a fluid filled pocket design that is hollowed out and has a smaller volume of hydraulic fluid in the pocket.

15.0 Conclusion

Initially we developed an extended list of concepts and variations of possible accumulator designs which could accomplish the requirements set by our sponsor. After careful analysis of this list we trimmed ideas to reach our final design, which is a dual X-ring seal piston, with the seals spaced such that there is a hydraulic fluid filled pocket between them. The piston is expandable which allows the fluid filled pocket to change volume as pressures change. Material selection, engineering analysis, and stress calculations on critical parts of the design have been completed. We designed a test rig to verify our prototype's performance and measure its nitrogen leakage over the course of testing to scale to a full sized model. Unfortunately, during the beginning of our testing, an accident occurred which prevented us from continuing the testing. We recommend that a second ME 450 team perform some modifications to our piston design and demonstrate proof of concept for our design.

16.0 Acknowledgments

We would like to thank:

- Andrew Moskalik and the EPA for providing funding, information, and advice for our project.
- Professor Bogdan Epureanu for all his assistance, advice, and letting us borrow a function generator for our test fixture.
- Tom Bress for allowing us to borrow a power source for our test fixture.
- Bob Coury and Marv Cressey for there assistance in the manufacturing of the prototype and test fixture.

17.0 Information Sources

Below is a list of the relevant patents [6] that were researched by our team to study prior work done on accumulators and seals. These patents gave us several ideas including:

- Metal baffles
- Non-elastic impermeable membranes
- Layered elastic impermeable membranes
- Fluid filled seals
- Double seals with high pressure fluid in between

Further research was done on the properties of several thermoplastics, as several designs require the use of an impermeable membrane.

17.1 Patent research on Hydro-pneumatic Accumulators

We researched patents, looking at hydro-pneumatic accumulator designs, to study basic methods of separating the two fluids from each other. Each design was studied to see what methods were used to improve standard accumulators. One accumulator used helium gas instead of nitrogen, the standard due to its stability, with the idea that helium would have less temperature variation and allow for a larger volume of liquid to be stored at a predetermined pressure (Patent 3,856,048). Another accumulator circumvented the permeability of standard elastic bladder accumulators by using an elastic bladder material that is impermeable to gas (Patent 6,058,976) allowing the bladder to retain more pressure over time. A third accumulator used a specially shaped inner cavity to support the bladder (PCT/FR2000/003633) presumably allowing for much higher expansion rates and pressure differentials between high and low pressures. A forth accumulator used a ribbed diaphragm to separate the fluids as opposed to a bladder (PCT/EP2002/002750). The last accumulator of interest we found used a pair of metal bellows to separate the fluids (PCT/EP2000/002083).

17.2 Patent Research on Seals

Research was also done specifically on seals for piston type accumulators. One piston used a flexible skirt that expands according to pressure, and had a sharp-edged scraper to clean the cylinder walls (Patent 6,539,976). Another piston used an accumulator ring that protected from explosive decompression by filling with fluid and expanding to press against the cylinder walls (Patent 6,948,715). A third piston used a lip seal as well as a bi-directional oil seal to prevent leakage past the piston (Patent 5,974,910). A forth piston used a pair of seals arranged such that gas can enter the space between then, saturating the fluid (Patent 7,284,475). The final piston used high pressure fluid filling the space between a pair of seals, preventing leakage through use of positive pressure (Patent 5,701,797).

17.3 Research on Material Properties of Thermoplastics

We researched material properties to confirm the viability of several of our design ideas, which depended on the availability of an impermeable membrane. Several thermoplastics were chosen for study to see if such a membrane was available. Information shown in Table 3 below is from Matweb.com [7].

Table 3 – Properties of Researched Thermoplastics

Material Name	**Ultimate Tensile Strength (psi)**	**Yield Strength (psi)**	**Oxygen Transmission rate (cc/100in^2/day)**	**Melting point (°F)**
Polypropylene Film Grade	2860 – 22000	3340 – 5830	0.00644 – 4.51	266 – 338
High Density Polyethylene (HDPE), Film Grade	4210 – 6530	3050 – 3980	95.8 – 359	162 – 268
Linear Low Density Polyethylene (LLDPE), Film Grade	1420 – 3800	1414 – 3210	225 – 322	248 – 262
Polyethylene Terephthalate (PET), Unreinforced	3190 – 22500	6829 – 13100	13.0 – 58.4	392 – 491

Of these material properties, the oxygen transmission rate, abbreviated as (OTR) above, is the most important to the design, as it is a measure of how permeable our membrane would be. The other statistics shown above are important only in that they must meet the requirements of the accumulator; they must be strong and resistant to the temperatures encountered. Of the plastics studied, polypropylene stands far above the rest by virtue of its extremely low transmission rate.

18.0 References

[1]www.cleanbreak.ca
Online Newspaper About Innovation In Green Technology

[2]www.epa.gov
United States Environmental Protection Agency

[3] M. Radhakrishnan, 2003, *Hydraulic Fluids A Guide to Selection, Test methods and Use*, The American Society of Mechanical Engineers, New York, NY, pp 8-9.

[4] Journal of Physical and Chemical Reference data, April 1984, volume 13, issue 2, Rubin Battino, Timothy R. Rettich, and Toshihiro Tominaga, *The Solubility of Nitrogen and Air in Liquid,* Department of Chemistry, Wright State University, Dayton Ohio, pp 563-569, 583.

[5] www.tss.trelleborg.com
Trelleborg Seal website

[6] Uspto.gov
The US Patent and Trademark Office website

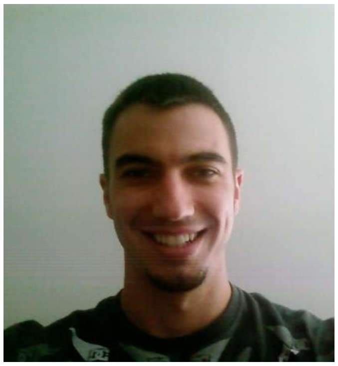
[7] Matweb.com
Materials database website

19.0 Biographies

Paul Frick is a 5th year Mechanical Engineering student interested in spacecraft research and design, but will probably end up working elsewhere right after college. He has had some experience working in a vehicle testing facility run by the US Army, and wants to end up doing something similar. In his free time, Paul likes to read good books, play a variety of computer games, and contemplate the mysteries of life. Not that he has much free time this semester, but he likes to pretend he's not just a student. He is currently trying to be active in the Michigan Rifle Team, and trying to join ASME and remain active in the University of Michigan Student Space Systems Fabrication Laboratory (S3FL), where he previously worked on a design for a space elevator climber.

Robert MacArthur was born in Sault Ste. Marie, Ontario Canada, but never lived there. Soon after he was born his family moved from Sault Ste. Marie, Michigan to Traverse City, Michigan. Robert is a fifth year senior in Mechanical Engineering at the University of Michigan. He has work for International Truck and Engine for the past two summers as a Manufacturing Process Engineering intern and has done one Co-op with Toyota in chassis design. Robert will start his second Co-op with Toyota in chassis durability in January. He is also an active member in the Zeta Psi Fraternity. Robert has grown up around cars and motorcycles and loves to work on them. He has a 300 all wheel hp Eagle Talon, and a drag car he is currently building. He also rides a 07 Suzuki GSX-R 750. Robert hopes to complete his degree in December 2009 and start working for Toyota in chassis design.

Ryan Sturgeon is a Senior Mechanical Engineering student with an interest in ancient history and philosophy. He is from outside Houston, Texas and has come to the University of Michigan as a Legacy student; both of his parents are U of M alumni. Ryan's free time is spent with intramural sports of all types, such as soccer and broom-ball, along with the rest of his housemates who provide the base for his teams. Ryan hopes to find a job up in the north, where he can experience more than one season a year and escape from the oppressive heat of southern Texas.



Jordan Timmerman is a senior in Mechanical Engineering who will be finishing his undergraduate career this December. Jordan hopes to start his professional career in an exciting field related to technical design or research and development early next year. His hobbies are Mountain biking, Windsurfing, Golf, downhill skiing, and playing intramural sports. Nothing seems quite as relaxing yet exhilarating as biking a trail such as Fort Custer in Augusta, MI. Jordan took up Windsurfing this last summer and became quit proficient at it, as he spent a couple of hours almost every weekend out on Gull Lake in Hickory Corners, MI windsurfing with his Dad. Even though windsurfing seems to be an 80’s sport, the feeling of zipping across the lake with a good wind pushing you is quite unlike any other.

20.0 Appendix

A – Solubility Calculation

The number of grams of nitrogen gas that can be dissolved into a gallon of n-hexane can be calculated as follows. [4]

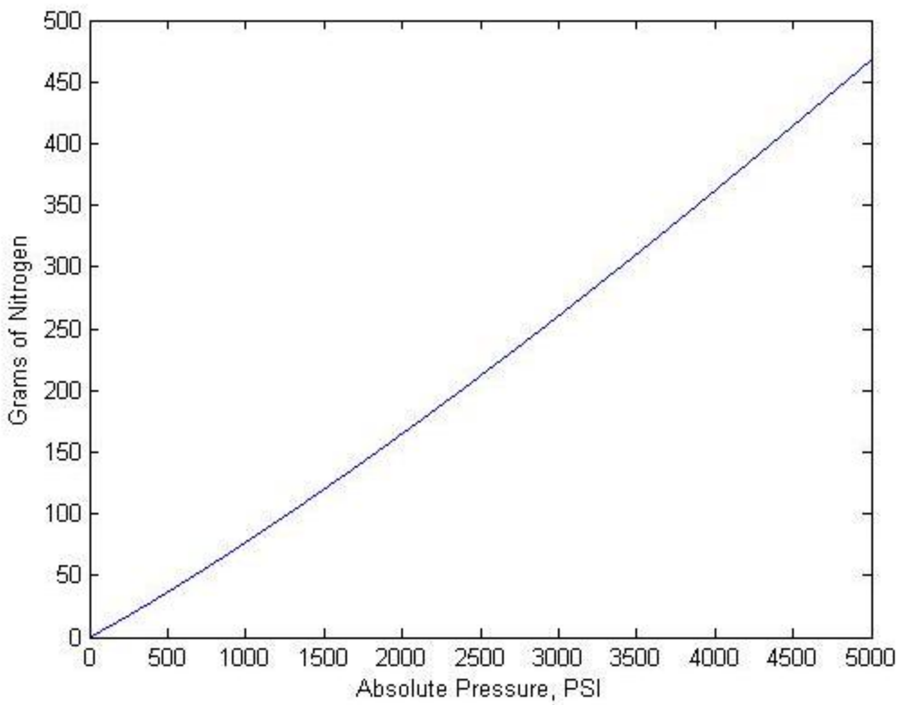
X_1 is the mole fraction of nitrogen dissolved in n-hexane (C_6H_{14}) at equilibrium
P is the pressure of the nitrogen in MPa

$$\tau = °K/100$$

$$x_1 = \exp\left(-12.4959 + \frac{10.7058}{\tau} + 4.0535 * \ln(\tau) - 9.6932\,X10^{-3}\left(\frac{P}{MPa}\right) + 1.0623 * \ln\left(\frac{P}{MPa}\right)\right)$$

$$\frac{x_1 * 28.013\ g\ N_2/mole}{(1 - x_1) * 86.178\ g\ C_6H_{14}/mole} * \frac{654000\ g\ C_6H_{14}}{m^3} * \frac{.00379\ m^3}{gallon} = \frac{g\ N_2}{gallon\ C_6H_{14}}$$

Figure 27 – Grams of nitrogen dissolved in 1 gallon of n-hexane at 50°C vs. pressure



B – Additional Concepts

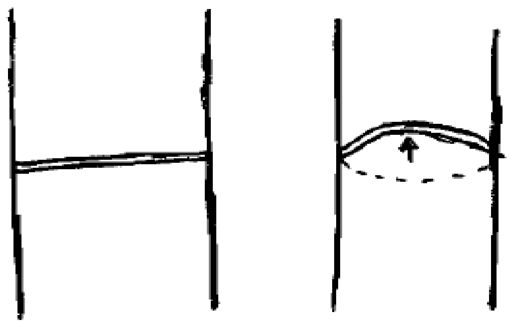
While brainstorming and deep-diving during the concept generation phases of the project, we developed many unusual concepts, some based heavily on existing designs. This appendix catalogs the better design concepts we developed. This goes from diaphragms, to bladders, to metal baffles, and finally to metal diaphragms. The individual designs are referenced as numbers, with supporting figures.

1. The first, most basic, concept design takes the form of a single flexible diaphragm stretched between the walls of the pressure cylinder. As shown in Figure 28 on page 45, the diaphragm would deform into a hemisphere as fluid is pumped in, thereby compressing the nitrogen on the other side. The nitrogen is kept from leaking into the hydraulic fluid by the diaphragm itself, likely made from a flexible plastic membrane covered in rubber or a metal foil. The plastic substrate would prevent stretching and therefore increased permeability of the diaphragm, while the rubber and metal foil would provide additional permeability resistance and thicken the material to resist folding fatigue. The shape of the diaphragm would not allow it to sweep out much volume, preventing it from achieving large amounts of compression or expansion, but the hemispherical shape would prevent folding and thereby tearing of the material. Another concern is the delamination of the layers in the membrane. This could be caused by leakage of gas through one or more layers such that it pools under an impermeable layer. As stresses from compression and expansion manifest, the pooling may stress the bonding between layers, resulting in a catastrophic delamination event. Depending on the location and severity of the event, this may result in catastrophic failure of the diaphragm.

   Pros
   - Forms a solid barrier between gas and liquid
   - No folding or twisting during pressure changes

   Cons
   - Sweeps out the least volume of all the membrane designs
   - Delamination of layers will destroy integrity

Figure 28 – Cross Sectional View of a Simple Diaphragm Accumulator System.

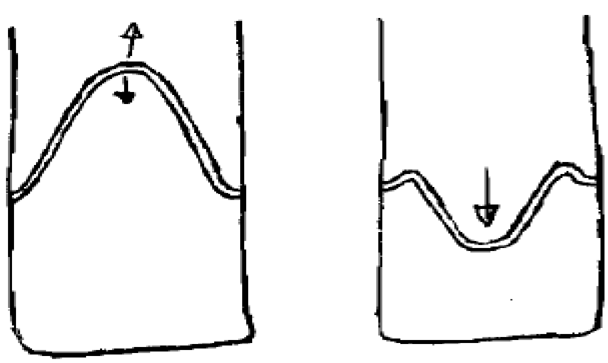


2. A more complex variation of the design 1 presented above, the diaphragm would be constructed in a rounded cone shape to sweep out more volume. This would allow the design to achieve better compression and expansion rates, while minimizing folding and tearing of the membrane when not fully inflated. Figure 29 below, shows a cross section of the system during an expansion (left) and compression (right) event. As with design 1, the layered membrane in this design is vulnerable to delamination, which would possibly result in failure.

   Pros
   - Forms a solid barrier between gas and liquid
   - Resists folding and therefore fatigue stresses

   Cons
   - Difficult to design
   - Delamination of layers will destroy integrity

Figure 29 – Design 2 During Expansion and Compression



3. Another variation on the diaphragm of design 1 combines the system with a piston cylinder approach. This concept forms the diaphragm membrane into a torus (donut) shape and secures one point to the cylinder wall, and the other to a piston, as can be seen in the cross

section of one side of the cylinder in Figure 30 below. The design takes cues from a conveyer belt or more accurately a caterpillar tread system – as the piston moves in the cylinder, the membrane will be carried along with it, while remaining attached to the cylinder wall. This range of motion allows for much greater movement and therefore compression as in design 1, while still keeping a solid seal between the hydraulic fluid and nitrogen. Difficulties in the system include all weaknesses of the membrane diaphragm as covered above, as well as a possible stress on the material when it changes diameter from the inner wall to the outer wall of the torus. This stretching would increase the permeability of the membrane; however, if the membrane was built as to be taut when at the outer wall, the inner wall would instead fold and not fail. Failure modes would differ, stemming from folding stresses and fatigue, limiting the lifespan of the design.

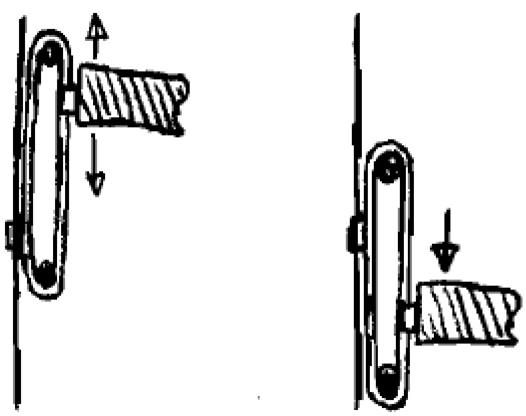
Pros

- Solid barrier between gas and liquid
- Lacks folding and bending inherent in the other membrane designs
- Reduces surface area of membrane, reducing the possible leakage rate

Cons

- Stretching occurs as the membrane changes diameter
- Delamination

Figure 30 – Piston Membrane System



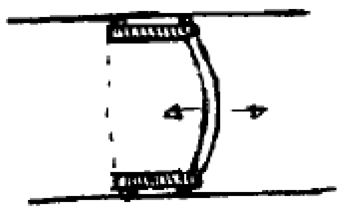
4. A variation on the hybrid membrane piston design presented as design 3 is inverted – a standard piston cylinder design with a flexible diaphragm built into the piston. This design would allow for more compression than a normal piston on a pressure tank, as the end caps tend to be hemispherical and a piston would not fit such a form very well. As shown in Figure 31 on page 47, the diaphragm would deform to fit the hemisphere of the end cap, providing a greater range of motion over the basic piston. This system has a higher

probability of failure, as the use of both a diaphragm and o-ring seal on the piston creates two points of weakness.

Pros
- More compression possible

Cons
- More points of failure due to both a membrane and a seal
- Difficult to design
- No inherent advantages over other similar designs

Figure 31 – Hybrid Piston Membrane Concept.



5. Another variation on the simple diaphragm system in design 1 attempts to solve the difficulties present due to possible delamination, and overall permeability of the membrane. This concept calls for the bioengineering of some sort of organic membrane which would resist the diffusion of nitrogen, while allowing for a great range of motion with some self-repair capabilities to allow it to last the entire ten year lifespan. This concept cannot be built because the technology to create a membrane with these properties does not exist.

   Pros
   - Should effectively resist diffusion of nitrogen

   Cons
   - Very difficult to design and engineer
   - Lengthy design and engineering timescale
   - Uses technology that is only now becoming available

6. Another variation on the hybrid concept presented in designs 3 and 4 includes a piston with extensions that fit into rails in the sides of the pressure cylinder. These rails are rifled, or spiraled, so the piston rotates as it moves through the cylinder. This twisting motion would twist an attached membrane skirt about itself, hopefully in a similar fashion every time, such that we could reinforce the membrane only where these folds are likely to occur. Reasons for not reinforcing the entire membrane include a lack of flexibility of the reinforced sections, and cost. The concept is shown in Figure 32 on page 48, with an untwisted stage on top, and a fully twisted stage on bottom.

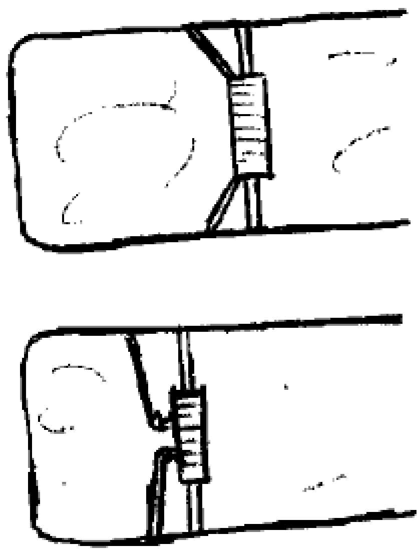
Pros

- Theoretically should fold the same time each time, allowing for reinforcement
- Solid barrier between gas and liquid

Cons

- Uncertainty as to whether folding would indeed occur in certain areas

Figure 32 – Twisting Piston Membrane Design



7. This design uses a piston in cylinder setup with a plastic flexible sheet attached to the piston and the top of the cylinder as shown in Figure 33 on page 49, to separate the nitrogen from the hydraulic fluid. The piston would be designed with slots to keep it from rotating and allow the nitrogen to flow up past the piston. The plastic sheet would stay attached to the piston during its travel and keep the nitrogen and hydraulic fluid separate. The motivation for this design is that there are no sliding seals to create leakage. The friction force between the piston and the cylinder could also be considerably less than a design with a sliding seal on the outside of the piston. The plastic sheet would be designed to "crumple and un-crumple" or fold like an accordion as the piston moves in the cylinder so stretching of the plastic sheet would be eliminated. The main challenge with this design is to find a material that is flexible, strong, will not fatigue and is impermeable to nitrogen and hydraulic fluid. The material should not need to withstand a pressure difference greater than approximately 2 psi.

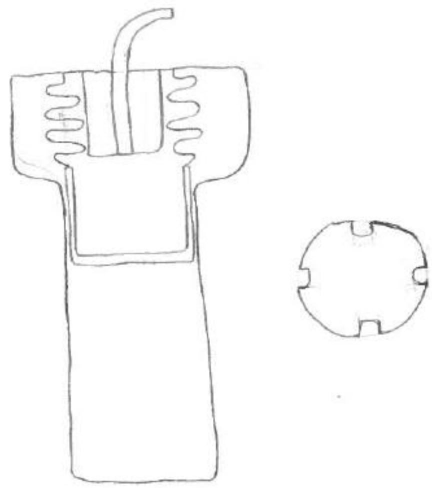
Pros

- No sliding contact seals reduce friction losses
- Guide rails prevent twisting of the sleeve

Cons

- Folding as sleeve compresses may weaken plastic over lifetime

Figure 33 – Plastic Sleeve and Piston Design



8. An alternative to fixed diaphragm designs manifests as a flexible bladder design. Gas filled bladders can be made from materials similar to those in the diaphragms, with the only difficulty being in finding a way to seal the bladder during construction. This design, presented in picture form in Figure 34 on page 50, consists of two bladders placed inside the pressure cylinder. The one on the left is filled with nitrogen, while the one on the right with hydraulic fluid. Both bladders would be built with integrated fill valves, to facilitate filling operations. The space between the bladders in this design would be allowed to fill with hydraulic fluid – the theory is that small amounts of nitrogen gas would eventually leak through the gas filled bladder into the inter-bladder space, to be absorbed into hydraulic fluid. This fluid could freely absorb the gas to saturation, and would not be likely to be able to pass through the fluid filled bag into the hydraulic fluid. Therefore, nitrogen would have little to no possibility of escape into the hydraulic fluid if we keep the leakage rates through the bladders low. Gas in the hydraulic fluid in the intermediate space would eventually saturate the fluid and bubble out as gas, which could then leak again into the fluid filled bag, so the leakage rate out of the gas filled bag should be kept low enough to last the ten year lifespan as such.

Pros

- Provides for eventual leakage of nitrogen through the membrane

Cons

- Bladder might twist and fold while compressed, weakening the structure
- More difficult to design than single membrane solutions

Figure 34 –Two Bladder Accumulator Design.

9. A minor variation on the design presented above as design 7 involves replacing the intermediate hydraulic fluid with a compound with a higher nitrogen capacity. This capacity might be in saturation, or as a result of a reaction with the nitrogen to take it out of solution. Hopefully the reactive product would be less likely to diffuse through the hydraulic fluid bag than the nitrogen gas itself. Difficulties we've experienced with such a design is in finding such a compound that fits within cost and feasibility requirements, while still being relatively non-toxic or corrosive to the materials inside the pressure cylinder.

   Pros
   - Prevents leakage of nitrogen into the hydraulic fluid
   - Provides for leakage of nitrogen out of its bladder

   Cons
   - Bladders might fold during compression
   - Difficult to find a compound with the necessary requirements, and likely costly
   - Very difficult to research and design

10. As our task is to reduce or eliminate leakage of nitrogen into the hydraulic fluid, an obvious concept is to remove the nitrogen entirely from the system. This design uses a different mechanism for providing reactionary force in this design. This idea takes the form of a simple linear spring in place of the nitrogen – this spring would resist compression as the gas would, and also springs back when pressure is relieved, providing the hydraulic fluid with pressure. This design certainly meets the primary requirement for the project, in that no nitrogen would ever leak into the hydraulic fluid as none is present in the system. The system is not without flaws however, in that with no gas present in the spring chamber, the pressure difference over the seal on the piston is much larger. The design, as shown in Figure 35 on page 51 consists of a pressure cylinder with a basic piston and o-ring seal, but with the gas on the left hand side replaced with the spring. As designing a seal to withstand many thousand pounds per square inch is much more difficult than designing one that allows for minimal leakage under a very small pressure difference, this design will not be seriously pursued at this time.

Pros

- No nitrogen in the system, meaning zero leakage into the hydraulic fluid

Cons

- Very large pressure difference over the seal, requiring a very strong seal design

Figure 35 – Spring Replacement for Gas in a Basic Piston Design.

11. Similar in outcome to design 10, this design uses a piston in cylinder setup where part of the piston would always protrude outside the cylinder. There would be two seals, one on the outside of the piston to prevent the gas leaking to ambient, and one on the inside of the piston to prevent the fluid from leaking to ambient as can be seen in Figure 36 below. In this case there would be no instance of nitrogen and hydraulic fluid mixing, but there would be the issues of leakage of both to ambient with a possible pressure drop as large as 5000 psi.

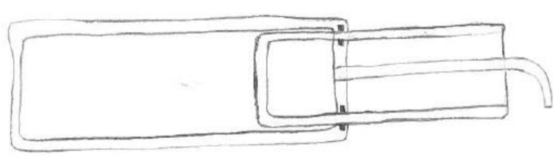
    Pros

    - Nitrogen leaks to the ambient, instead of into the hydraulic fluid

    Cons

    - Pressure drop over seals as great as 5000 psi
    - Seal design in this case very difficult as seal needs to be strong

Figure 36 – External Cylinder Accumulator Design



12. The other variation on the standard piston cylinder design is to replace the nitrogen with another gas less likely to leak through an o-ring seal as shown in Figure 37 on page 52. This design would be ideal, as a gas in the chamber would be pressurized to near that of the hydraulic fluid, resulting in only a small pressure drop over the seal. Difficulties have cropped up in the search for such as gas – at present we have found no good readily available replacement that has similar characteristics to nitrogen but is less likely to leak through a material. Therefore, this design will not be pursued until research into gases finds a good replacement.

Pros

- Uses existing proven design
- Limits leakage of gas by reducing diffusion rates

Cons

- Difficult to find gas that meets requirements
- Costly

Figure 37 – Cross section of a standard piston cylinder accumulator, for reference in design 10.

13. Another direction of variations on the basic piston accumulator is in redesign of the piston and its seals. This concept presented in Figure 38 below extends the cylinder in length while keeping it hollow to reduce displaced volume, and places multiple seals along its length. The multiple seals would reduce the rate of leakage through into the hydraulic fluid by increasing the resistance to diffusion. The idea behind this is that while nitrogen might leak through one seal, it will take longer to leak through two seals, and so perhaps will last the required lifetime if a sufficient number of seals impede the nitrogen leakage rates to what is required by our engineering specifications.

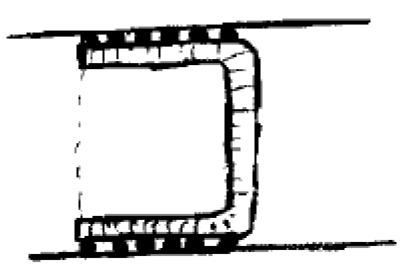
Pros

- More seals better resist flow of nitrogen
- Redundant design in case of seal failure

Cons

- Increased frictional resistance on piston when moving through cylinder.

Figure 38 – Multi-Seal Piston.

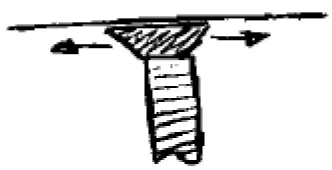


14. Similar in function to a basic o-ring seal present in a piston accumulator design, the dual-scraper seal as shown in Figure 39 below is shaped as a trapezoid, allowing it to not only compress to present a good seal, but also to scrape away at any buildups along the cylinder wall, preventing some leakage.

    Pros
    - Basic design
    - Prevents buildups of material on the cylinder walls

    Cons
    - Same disadvantages of an o-ring seal

    Figure 39 –Trapezoidal Dual-Scraper Seal.

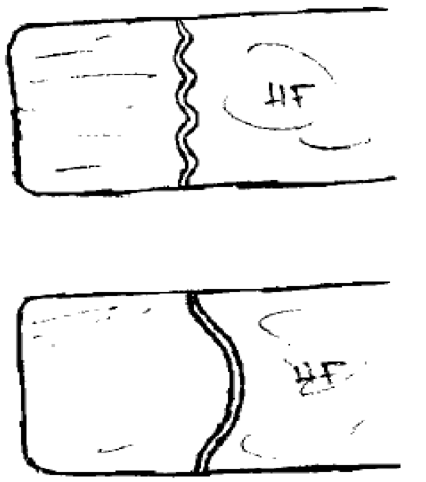
    

15. Using metal in place of a flexible plastic membrane offers a much higher resistance to nitrogen leakage. Design of a metal diaphragm is possible, and has been demonstrated in industrial applications. Therefore, this design takes the diaphragm concept of design 1 and replaced the membrane with a flexible corrugated metal sheet as shown in Figure 40 on page 54. The corrugations in the sheet should help it to flex more easily, and reduce stress that may cause a fatigue fracture. This design is flawed in that similar to design 1; it would not sweep out a large volume, and therefore would be of limited utility compared with other designs.

    Pros
    - Very good seal against leakage
    - Simple design

    Cons
    - Costly
    - Sweeps out less area

Figure 40 –Corrugated Metal Diaphragm Accumulator



16. Metal baffles are flexible metal sleeves capable of expansion and contraction similar to an accordion mechanism. Placement of a single long sealed metal baffle in half the pressure chamber as shown in Figure 41 below would allow for contraction and expansion of the nitrogen gas during operation, but would be unlikely to leak significantly until a fatigue fracture occurs. Their usage is similar to bladders, though they do not have quite the range of motion in both compression and expansion, but they are less likely to leak. The baffle design however does hold well over the diaphragm concept, as the range of motion is much greater. Preliminary research into suppliers for metal baffles has led us to determine that they are very costly solutions, and should be looked at as a last resort.

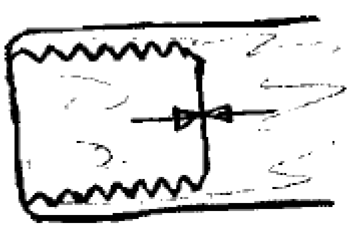
Pros

- Very good seal against leakage
- Allows for greater pressure changes than design 13

Cons

- Very costly
- Fatigue could cause failure
- Limited mobility when compared to a piston

Figure 41 – Metal Baffle Accumulator



17. As the metal baffle design expands and contracts over its lifetime, fatigue stresses build on the flexing joints until failure. To combat this, a hybrid metal-plastic baffle design was designed, shown in part in Figure 42 below. In this design, metal rings making up the baffle are connected with thin sheets of flexible plastic, allowing the system to compress without stressing metal joints. In addition, because the plastic can flex much further than metal, the hybrid baffle design is able to compress much further than an all-metal design. The design is still limited by the permeability of the plastic sheets, but the surface area of them is to be kept to a minimum.

Pros
- Greater range of motion over an all-metal design
- Decreased area of permeable material

Cons
- Very costly
- Must be custom made
- Leakage possible through plastic joints

Figure 42 – Metal Baffle Design with Plastic Joints

18. As another variation on the metal diaphragm concept, this design attempts to solve the problem of limited mobility without resorting to baffles. As shown in Figure 43 on page 56, this design takes the metal diaphragms from design 13 and places two parallel along the long axis of the pressure chamber. The nitrogen is contained between the diaphragms, while the hydraulic fluid filling the outer chambers is able to freely compress the inner chamber. This design takes all the advantages of design 13 and adds much better compression and expansion.

Pros
- Very good seal against leakage
- Allows for greater expansion and compression over design 13

Cons
- Costly

Figure 43 –Two Parallel Metal Diaphragms.

19. A modification of the basic piston accumulator concept alters the design of the piston to have two seals, with a space between, as seen in Figure 44 below. This space will tend to fill with nitrogen due to imperfect seals, mixing with hydraulic fluid already present in the pocket. Ideally the saturated hydraulic fluid will not be able to leak out into the pure hydraulic fluid due to the seal resisting movement of liquid better than gas. This particular design has a weakness in its solid design – as the piston cycles over its lifetime, the fluid in the space will heat up, and expand differently than the gas and liquid in the rest of the cylinder, creating pressure on the inner edge of the seals. This will likely result in more leakage of nitrogen into the hydraulic fluid.

Pros
- Dual seal design limits leakage of nitrogen at equilibrium

Cons
- Thermodynamic heating during operation will cause leakage due to expansion

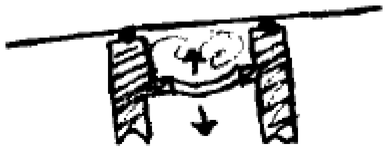
Figure 44 –Dual Seal System

20. A further modification of design 18 adds a flexible inner wall to the space between the seals as shown in Figure 45 on page 57, to allow for expansion of the contents during thermodynamic heating. This design requires that the piston be hollow, to allow for equalizing pressure on both sides of the wall membrane. This membrane would likely be made from a metal diaphragm, as it would resist further contamination of the pocket, and would not need to sweep out large volumes for its normal operation.

Pros
- Dual seal design limits leakage of nitrogen at equilibrium
- Flexible wall allows for thermodynamic heating

Cons
- Saturated fluid might leak into hydraulic fluid

Figure 45 – Dual Seal System Design With Flexible Inner Wall.

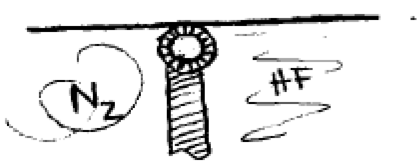


21. Pressure applied to the cylinder wall by an o-ring seal depends on the tolerances of the piston and the surface finish of both. If the piston fits loosely, the seal won't have as much sealing force and will fail, and if the surface finishes are not very good, the seal will be unable to prevent leakage of material through these gaps. An inflated seal would solve both these problems – the seal as shown in Figure 46 below would have an internal pressure greater than that of the surrounding fluids, causing it to deform to fill the space between the piston and cylinder wall, and fit into any imperfections in the wall. The internal pressure would depend on the maximum pressure the fluids in the cylinder will experience, as it must be greater than this value.

Pros
- Compensates for poor tolerances in piston and cylinder
- Simple in function

Cons
- pressure requires strong seal material

Figure 46 – Inflatable Seal System Design



C – Worst Case Scenario Calculations
Tensile Force:
A = area of piston end
D = diameter of piston
P = pressure on piston end
F = tensile force

$$A = \frac{D^2 * \pi}{4}$$

$$F = A * P = \frac{4.995^2 * \pi}{4} * 5 = 97.97 \ pounds$$

Bending Moment:
M = moment produced at neutral axis
F = force
R = distance from neutral axis

$$M = F * R = 97.97 * 5 = 489.89 \ pound \ inches$$

D – Piston Wall Thickness Calculations
Shear Stress:
σ = shear stress on wall
F = force applied
A = area of applied force
l = length of area
w = width of area

$$\sigma = \frac{F}{A} = \frac{F}{l * w} = \frac{97.97}{0.5 * 0.25} = 783 \ psi$$

E – Piston Rod Calculations
Bending Stress:
I = moment of inertia
d = diameter of rod
σ = bending stress
M = moment produced at neutral axis
y = distance from neutral axis

$$I = \frac{\pi * d^4}{64}$$

$$\sigma = \frac{M}{I} * y = \frac{M}{\frac{\pi * d^4}{64}} * y = \frac{489.89}{\frac{\pi * 1^4}{64}} * 0.5 = 4999\ psi$$

F – Piston Tube Calculations

Bending Stress:

I = moment of inertia

d_o = diameter of tube

d_i = diameter of rod

σ = bending stress

M = moment produced at neutral axis

y = distance from neutral axis

$$I = \frac{\pi * {d_o}^4}{64} - \frac{\pi * {d_i}^4}{64}$$

$$\sigma = \frac{M}{I} * y = \frac{M}{\frac{\pi * {d_o}^4}{64} - \frac{\pi * {d_i}^4}{64}} * y = \frac{489.89}{\frac{\pi * 1.5^4}{64} - \frac{\pi * 1^4}{64}} * 0.75 = 1060\ psi$$

G – Pin Calculations

Shear Stress:

σ = shear stress on wall

F = force applied

A = area of applied force

D = diameter of pin

$$\sigma = \frac{F}{A} = \frac{F}{\frac{D^2 * \pi}{4}} = \frac{97.97}{\frac{0.375^2 * \pi}{4}} = 2661\ psi$$

H – Parts List Summary

Description	Part Number	Supplier	Cost Each	Qty.	Total Cost
5-/2" round stock 6061 Aluminum, 4 1" sections	21413805	Alro Metals Plus	$84.44	1	$84.44
1-1/2" round stock 6061 Aluminum, 36" long	21411805	Alro Metals Plus	$45.57	1	$45.57
¼-20 threaded rod	19100550	Alro Metals Plus	$1.20	6	$7.20
3-way solenoid valve	4HN46	www.grainger.com	$59.75	1	$59.75
Hex Nipple, Size ¼ in, Hex Size 5/8	1DGB3	www.grainger.com	$4.33	2	$8.66
Ball Valve,1/4"	6GD11	www.grainger.com	$6.30	1	$6.30
Steel plate 8” by 8” square, 1/2” thick 1006-1020	6544K35	McMaster-Carr	$51.95	2	$103.90
O-ring seals for end plates	9452K201	McMaster-Carr	$5.70	1	$5.70
Acrylic 36” cylinder, 5” ID, 3/8” wall	8486K713	McMaster-Carr	$249.99	1	$249.99
acrylic cylinder for watch glass	8532K15	McMaster-Carr	$7.26	1	$7.26
watch glass end cap	8528K53	McMaster-Carr	$7.43	1	$7.43
Oil Plug	9171K252	McMaster-Carr	$2.93	2	$5.86
Piston seals	PQ4305000-T46-N	Trelleborg Seals	$62.50	4	$250.00
¼-20 nuts	N.A.	Jack’s Hardware	$0.17	24	$4.08
Set Screw	N.A	Jack’s Hardware	$0.30	2	$0.60
Case of Automatic Transmission Fluid	N.A.	Murry’s Auto Parts	$38.03	1	$38.03
				Total	$884.77

I – Design Safe Risk Analysis

On Next Page

designsafe Report

Application:	Accumulator Piston Prototype	Analyst Name(s):	Team 3
Description:		Company:	Me450 Team 3
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 : fatigue Unforeseen structural weakness, cracking	Serious Occasional Unlikely	Moderate	Raise safety factor, do FEA analysis	Serious Remote Unlikely	Moderate	
All Users All Tasks	mechanical : break up during operation Unforeseen structural weakness	Catastrophic Frequent Possible	High	Raise Safety factor, FEA and stress calculations	Serious Frequent Unlikely	High	
All Users All Tasks	mechanical : impact Lack of buffer during actuation	Slight Occasional Unlikely	Moderate	Watch system during testing	Minimal Remote Negligible	Low	
All Users All Tasks	electrical / electronic : insulation failure Improper insulation	Serious Frequent Possible	High	Provide insulation on wiring	Minimal Remote Negligible	Low	
All Users All Tasks	electrical / electronic : improper wiring Improper wiring	Serious Frequent Possible	High	Check wiring guide before supplying power	Minimal Remote Negligible	Low	
All Users All Tasks	electrical / electronic : electrical noise Noise in signal from function generator	Slight Frequent Possible	High	Calibrate signal generator, apply filter	Minimal Remote Unlikely	Low	
All Users All Tasks	electrical / electronic : power supply interruption Failure of power supply or mains supply	Slight Occasional Unlikely	Moderate	Monitor system during operation, use surge protection	Slight Occasional Unlikely	Moderate	
All Users All Tasks	material handling : excessive weight Weight of system when testing	Minimal Frequent Negligible	Low	Secure system to immovable object	Minimal None Negligible	Low	
All Users All Tasks	chemicals and gases : oxygen Air used in testing contains oxygen	Minimal Remote Unlikely	Low	Don't ignite air during testing	Minimal None Negligible	Low	

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	fluid / pressure : high pressure air Shop air, ~100psi	Slight Frequent Possible	High	Aim air away from face, wear safety equipment	Minimal Remote Negligible	Low	
All Users All Tasks	fluid / pressure : hydraulics rupture Hydraulic fluid in use during testing	Serious Frequent Unlikely	High	Wear safety equipment, know hydraulics operational safety	Serious Remote Negligible	Low	
All Users All Tasks	fluid / pressure : vacuum Moving system might create low pressure spot if jammed	Minimal Occasional Unlikely	Low	Monitor system during testing	Minimal Remote Negligible	Low	
All Users All Tasks	fluid / pressure : surges / sloshing Jolting motion if function generator provides improper signal	Slight Occasional Possible	Moderate	Monitor system during testing	Minimal Remote Negligible	Low	
All Users All Tasks	fluid / pressure : fluid leakage / ejection Failure of containment vessel	Catastrophic Frequent Unlikely	High	FEA analysis and calculations on test fixture	Serious Occasional Unlikely	Moderate	
All Users All Tasks	fluid / pressure : liquid / vapor hazards General leakage of fluids	Catastrophic Frequent Unlikely	High	Use safety equipment, ensure leaking fluids cannot damage electronics	Slight Occasional Unlikely	Moderate	