MEE 482 Project Report

Pressure Accumulator For a Hydraulic Motor

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Abstract

Given the changing economic climate of today's world, the typical construction job site has changed as well. As the world is acutely aware of, both the commercial and private housing markets are dismal. Because of the present economic conditions, a strain has been placed on the entire construction community to produce higher quality houses at a much lower cost. This demand is directly related to electrical contractors, and companies hiring these contractors are determined to reap any savings possible. In April 2012, Greenlee Textron, a manufacturer of premium electrician's tools, launched a mobile bending table that can be transported with ease throughout a job site and can form conduit bends up to four inches in diameter. However, the high pressure pump that is used for the bender performs bends on an average of ninety seconds. Given the amount of bends performed throughout the job, time adds up quickly as contractors pay electricians to perform pipe bends. With the addition of the accumulator, the expectation is to reduce this bending time significantly to less than a minute per bend. The reduction of just thirty seconds over a two week project can save over 300 dollars of machine time and reduce the amount of idle stand by time.

Skyler Ripplinger, the Director of Engineering at Greenlee, suggested using a pressure accumulator as a possible solution to create faster bends. Our senior design project includes the research and development of an accumulator that can meet Greenlee's needs. An accumulator has been designed to withstand up to 10,000 psi, the pressure requirement for the motor. It also performs work on bends which require up to 6,200 psi and can be mounted on the table to maintain mobility of the unit itself. For the prototype, Parker Hannifin, a manufacturer of accumulators, requires a 4 to 1 safety factor in order to maintain the safety of the product.

The current pump delivers an average of 56 cubic inches per minute to the accumulator which results in the 90 second bend times currently observed for the production unit. With an accumulator, we can essentially control the flow in order to regulate the bending time. We selected a piston

accumulator for this application because piston accumulators can be designed to withstand pressures up to 30,000 psi, whereas bladder accumulators can only hold up to 7,500 psi. Piston accumulators can also deliver up to 220 gallons per minute of hydraulic fluid, which is more than acceptable to fill the bending ram, which falls around a gallon and a half capacity.

While the accumulator would be able to deliver the time saving results desired for the application, the cost of selling an accumulator on a retail basis remains very high and would likely exceed the cost of the bending unit itself. A high performance pressure accumulator, while fulfilling its intended purpose for this application, would be a very expensive and impractical solution for Greenlee, given its high development cost currently at $6,200 and dry weight of 627 lbs. However, due to the success and quality of the research conducted during the project, Greenlee has elected to continue pursuing hydraulic accumulator technologies for other applications that would be more economically feasible.

Chapter 1: Introduction

Greenlee Textron, a company based out of Rockford, IL, specializes in designing and building equipment to meet the needs of people in the electrical industry. One major focus is on equipment used for installing electrical wire and cable. One specific process that Greenlee assists electricians with is in bending conduit for installation in large facilities. One machine, the 881 Bending Table [Figure 1], is currently used to bend conduit up to 4 inches in diameter.

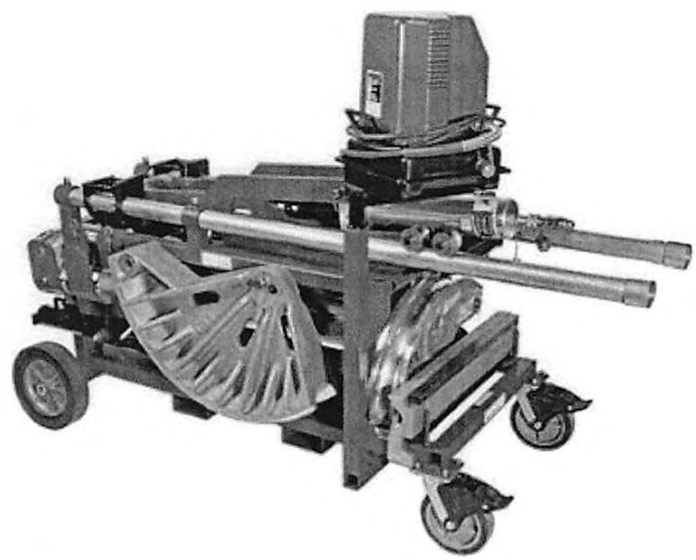

Figure 1: Current 881 Bending Table

The bender currently takes approximately between 76 and 92 seconds to a minute to complete a bend with the current motor and design, depending on the pipe diameter and material. The design scope, as presented by Greenlee, will be to adapt a hydraulic pressure accumulator into the system in order to reduce bending time down to approximately 15-20 seconds. The time savings, in labor, will then translate into cost savings for electrical contractors. Average contracting prices that an electrician may makeare around $60 an hour. If bend times can be reduced by even half, savings in labor can accumulate quickly on the jobsite. This is considering that, in large installation processes, two electricians will be needed to accurately bend the conduit. The savings will not only help out the electrical contractors, but they will help keep Greenlee at the top of their industry by continuing to be on the cutting edge of advanced electrical equipment. Exploration into new technologies and the pursuit of new ideas, no matter how abstract they may be, is the only way for a company at the top to stay there. The investigation of technologies such as the one posed in this project is what true engineers for true industry leaders pride themselves on.

In Greenlee's test lab facility, bends for each pipe size were timed; the pressure used to measure each bend was also recorded by a pressure gauge from an out-sourced pump. The technicians had observed that the pressure demand had remained relatively constant throughout each bend cycle; this was a key assumption made when we designed our accumulator. The technicians were able to record the following times and pressures for each conduit size, shown below in Table 1:

Conduit size	Greenlee 980 (66.7 pounds with oil)	PSI (average during bend)
4" EMT	82 seconds	6200 psi
3-1/2" rigid	92 seconds	5200 psi
3" rigid	89 seconds	4150 psi
2-1/2" rigid	76 seconds	2300 psi

Table 1: Recorded values of time and pressure required to perform conduit bends

The idea of using a pressure accumulator to store energy was presented by Skyler Ripplinger, the Director of Engineering for Greenlee Textron at that time. The basis of the idea was that, while electricians were setting up each pipe and bending shoe in between bends, that the pump could be used to store energy in a pressure accumulator. Skyler had presented this idea to our group at our first Senior Design team meeting, and from this we formed the basis of the accumulator project.

A pressure accumulator has the unique ability to store a high amount of pressure by using compressed nitrogen; it can also deliver that energy, through the use of hydraulic fluid, upon demand and in a quick fashion. Given the amount of time that can elapse in between bends, which can be up to 4-5 minutes, the thought was for the pump to charge the accumulator as its idling, and when the bend is ready to be performed, to release the stored energy in the accumulator upon demand. The accumulator, in theory, will be able to produce much faster bends than Greenlee's two-stage motor.

Some initial thoughts our team had about accumulators were that, not only could the accumulator itself be quite costly, the component routing may prove to be very complicated as well.

Other potential problems included the weight of the accumulator being too heavy, which would render the mobility of the current platform useless. The accumulator, as mounted on the bending table, may not be able to fit through a 36" doorway depending upon its mounting position.

Other concerns included the liabilities involved with allowing end users to manually control and operate an accumulator that can hold up to 10,000 psi. In addition, having to ship an accumulator that can be pre-charged up to 10,000 psi presents liability challenges as well to shipping companies. We also had to ensure that the wall thickness on the accumulator is enough in order to maintain a proper factor of safety for the application so that the safety concerns for the end user are reduced.

Figure 2: Current Two Stage Motor for the 881

The demand for faster bending comes from the fact that the bends observed from the current motor are too slow and time consuming. A main cause for this issue is that the power source is limited to your typical 12 Volt outlet installed into building structures [Figure 2]. As a result, the motor's power capabilities to displace hydraulic fluid are reduced to its source. An accumulator is unique in that it does not require an outside power source to release energy and displace hydraulic fluid; it just needs an input (the motor) to store the energy in the first place.

Given the rapidly changing and competitive economy that we have in the world today, the competition for jobs available to electrical contractors has rapidly grown and the budgets available to contractors have rapidly decreased. As a result, contractors are looking to cut down on overhead costs wherever possible; a major source of overhead for contractors is the electrician's labor. With this application, we will be able to help contractors save almost 80% in machine time on jobsites with the limited power source available.

As an alternative solution to pressure accumulators, Greenlee explored the possibility of out-sourcing a more powerful, efficient pump. One solution they looked at was a pump that SPX offered with an infinite amount of stages. The problem was that this pump would cost Greenlee around $4,500 per unit; by the time the end user was to buy it from Greenlee through a distributor, it could cost them up to $7500-8000. Also, after test trials were run using the pump, it was determined that the pump only reduced bending times by 20% from Greenlee's pump. Patent restrictions on the pump itself restricted Greenlee from being able to reverse engineer the design themselves. As a result, Greenlee concluded that this pump would not be of value for the application, and that they would have to seek out an alternative solution.

Potential savings for electrical contractors will vary; the main factor will be whether electricians are union or non-union. Union electricians tend to cost significantly more to contractors than non-union electricians; this may be a main factor as to which contractors are potential customers for a time saving product. Also, pay rates for electrician labor vary on a geographical basis; a contractor based out of Los Angeles or New York may be more interested in the product than a contractor based out of a less populated area.

The total time savings are also dependent upon the job size itself. Contractors mainly dealing with large job sites; such as skyscrapers or large residential units, may be more interested in a time saving product than a contractor involved with the building of small elementary schools and small retail strips.

For the scope of this project, roles for each team member were determined. Alec LeHouillier was selected as the design lead and would be responsible for the final accumulator design, the calculations used to justify its performance, the design schematic used for the accumulator routing, the FEA analysis used to justify the design, and any prototype testing that may follow as a result of design.

Dan Lapp was chosen to create the economic justification for the project and business end of it. In doing so, he is responsible for justifying the added cost of an accumulator to end users. He also has the responsibility for determining the potential dollar savings through the amount of time saved by an accumulator. Dan is also responsible for determining which type of contractor would be key for marketing the product.

Mark Klein was chosen to determine the manufacturability of the design. As part of this, Mark is responsible for determining a potential in-sourcing strategy for Greenlee if they choose to manufacture their own accumulators. In addition, Mark is also responsible for determining any costs associated with the in-house manufacturing of an accumulator, which may include tooling, labor, warehouse space, and liabilities associated with accumulator production. Mark is also responsible for designing the accumulator for the ease of manufacturability along with the processes involved for making an accumulator in-house.

Chapter 2: Design Specifications, Concept Generation, and Evaluation

For the design specifications, our group originally intended to design an accumulator that would weigh less than 80 pounds total. In order to maintain the mobility of the table, it would need to be less than 36" in width at any point so that it could fit through a standard doorway. It was also desirable to keep the cost of the accumulator as low as possible so that Greenlee could maximize its profits from the accumulator; our original intention was to keep the cost under 20% of what the table cost on a retail basis, which was around $2,500.

The objective for the project is as follows: To design and develop a working prototype of a mobile bending table that combines the mobility of the Greenlee 881 bending table with a pressure accumulator that will reduce the machine time of the original design.

For our house of quality, we generated relationships between important design factors of the project. These can be seen in Figure 3. The house of quality shows a strong relationship between external dimensions vs. ease of mobility, power supply vs. efficiency, and power output vs. efficiency.

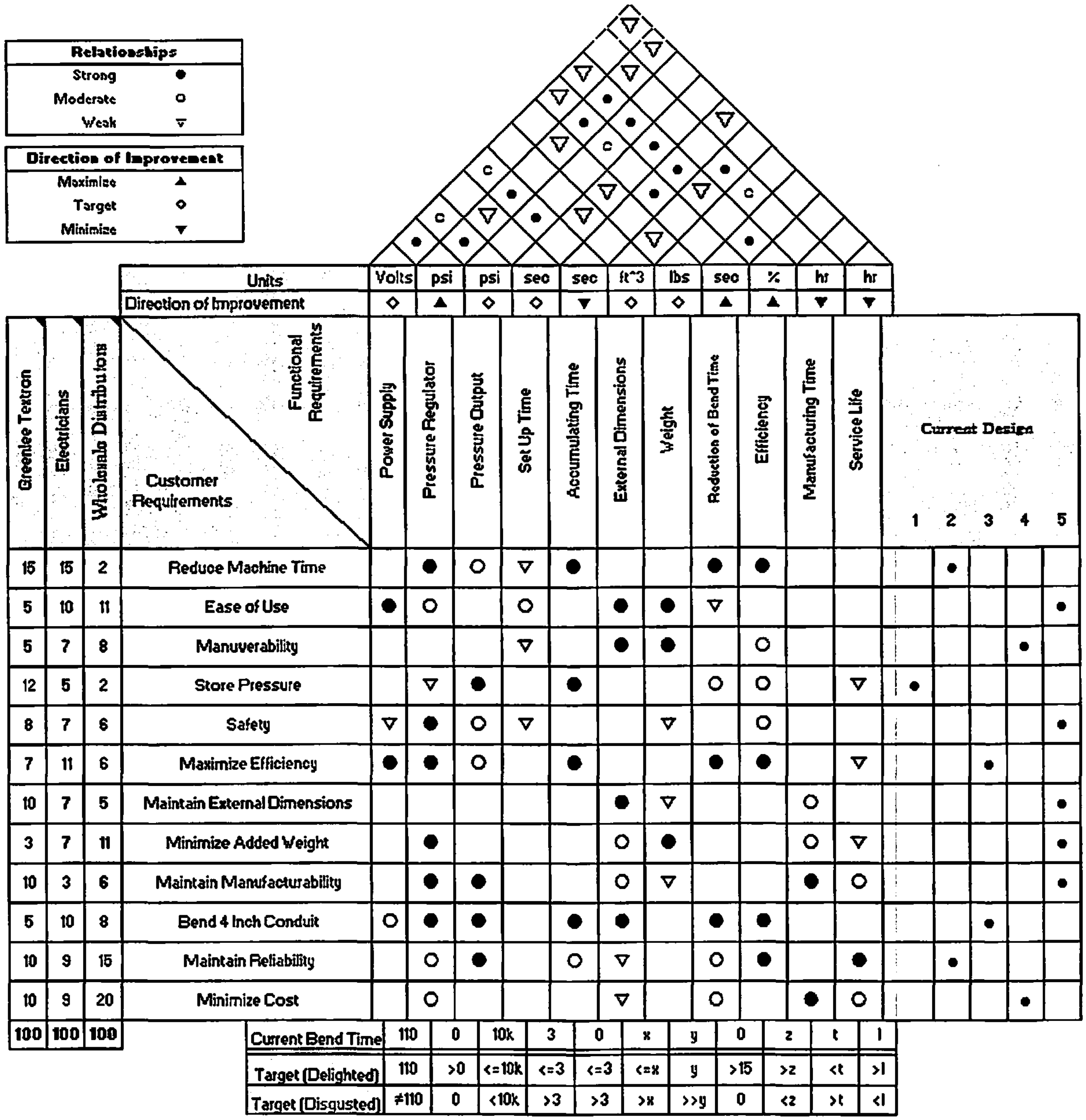

Figure 3: House of Quality

For the initial accumulator concept, our team had researched, compared and contrasted hydro-pneumatic accumulators, bladder accumulators, and piston accumulators. The advantages of hydro-pneumatic accumulators include how they are low-cost and can be easily sized to the application at hand. Bladder accumulators are also low-cost due to the fact that material costs with these accumulators are less due to the use of a bladder. The decision matrix on which design to choose is given in Figure 4:

Issue: Design a Pressure Accumulator that can hold up to 10,000 psi		Baseline	Hydro-Pneumatic	Bladder	Piston	Weight-Loaded	Spring
Weight	15	Datum	1	1	0	-1	0
Cost	25		1	0	0	0	0
Manufacturability	20		1	1	0	0	0
Safety	25		0	0	1	1	1
Efficiency	15		1	-1	-1	0	-1
	Total		4	1	0	0	0
	Weighted Total		75	20	10	10	10

Figure 4: Decision Matrix for Determining Accumulator Type

However, at the time we did not know that neither type accumulator is able to hold and deliver the desired 10,000 psi for the application. As a result, we selected the more costly but very powerful piston accumulator. Piston accumulators are able to be sized up to 30,000 psi and can deliver flow rates as high as 220 GPM. Piston accumulators are manufactured by their inner diameter: 2", 3", 4", or 6". They also have a very high fatigue limit due to the fact the piston is aluminum and can handle repeated cycles; bladder and hydro-pneumatic accumulators are more likely to fail from the membrane.

Chapter 3: Cost/Market Analysis and Patentability

Cost represents the driving force behind every design consideration in the engineering field. While engineers can understand the mechanics of a system or part, often times the product is not sold on its abilities alone. The universal language of money is the most understood and quantifiable source for feasibility and marketing. For this purpose, the economic value and feasibility was analyzed and presented in this section.

As will later be discussed, the design considerations for this project were not only up to the team, but rather up to the Greenlee management staff as well. Essentially, through conversations with Greenlee and Parker Hannifin, the final decision was to have a 10,000 psi accumulator. The 10,000 psi accumulator would have the capability to bend even the largest pipe that the original motor was designed to endure. With this accumulator, and as will later be described, the contractor working with this upgraded table will experience an average of about 44 seconds when the accumulator assisted with the entire bend.

With regards the contractors that will be benefiting from this system, it was initially proposed to us that the cost savings should be calculated. Through research of contractor rates in the area, in Chicago where much of Greenlee's business is located, and in general, a simple truth was found for the hourly rate that contractors charge; there is no set rate. Hourly rates range from as low as $24 per hour for a small independent contractor, to as much as $120 per hour for a union contractor from Chicago.

When deciphering cost savings relative to time, a wide range of rates poses a problem. There was also another problem when determining the amount of time necessary to complete a job. The managers that employ contractors hire them for specific jobs. This job may be a house, a school, a mall, a hospital, or even a sky scraper. As you may imagine, the diverse range of possible project takes a different amount of time for each, as well as a different quote from each contractor on how long it will take them to complete their assignment. A house, for example, may take one contractor one week to install all of the necessary equipment, while another contractor may only require 3 days in the same house.

Such a wide range of possibilities made it difficult to truly say how much money is saved with the addition of an accumulator. For this reason, a program was developed in order to justify cost for the team, as well as help employers select the best possible candidate for their job when an accumulator is used. In order to justify basic costs, some variables will be held constant. Table 2: Cost Analysis Program shows a screenshot of a contractor's scenario.

	Average
$59.50	/hr pay rate
80.325	% improvement (time)
8	hr/day
20	Days to Complete Job
40	bends/day
60	seconds/bend
$0.02	/sec pay rate
5	bends/hour
300	bending seconds/hour
$4.96	bending pay per hour
	Improvement
11.805	seconds/bend
59.025	bending seconds/hour
$0.98	bending pay per hour

Money Spent Bending	**Days**	**Original Design**	**With Accumulator**	**Difference [Money Saved]**
$/hour	0.04	$4.96	$0.98	$3.98
$/day	1	$39.67	$7.80	$31.86
$/1 week	5	$198.33	$39.02	$159.31
$/1.5 weeks	7	$277.67	$54.63	$223.04
$/2 weeks	10	$396.67	$78.04	$318.62
$/3 weeks	15	$595.00	$117.07	$477.93
$/1 month	20	$793.33	$156.09	$637.25
$/1.5 months	30	$1,190.00	$234.13	$955.87
$/2 months	40	$1,586.67	$312.18	$1,274.49
$/4 months	80	$3,173.33	$624.35	$2,548.98
$/6 months	120	$4,760.00	$936.53	$3,823.47
$/10 months	200	$7,933.33	$1,560.88	$6,372.45
$/1 year	240	$9,520.00	$1,873.06	$7,646.94

Savings w/ Accumulator	**$637.25**

Table 2: Cost Analysis Program

As you can see, the larger the job, or rather the longer it takes to complete the job, the more cost savings one may see. For a one month job shown in the table, $637 dollars is saved with the accumulator performing the entire bend. This program is open to different variables. The hourly pay rate, percent improvement, days to complete a job, and number of bends per day are all open for alteration depending on the quote of the contactor. The purpose of such a program is to demonstrate how seconds can add up to dollars quickly for this particular field.

Naturally the draw of the 881 Mobile Bending table would be for large scale contracting jobs such as a sky scraper, school, or hospital. Essentially jobs that take over two months see enormous benefits in savings with the accumulator, whereas the smaller scale projects may not see such outstanding benefits from this design. For this reason, the proposition was to offer the accumulator as an add-on or upgrade. That way, those who would benefit greatly from the design could purchase the upgrade and quickly repay the initial investment of added cost, whereas the smaller scale contractors could purchase the unit without the accumulator as the 881 is a particularly outstanding machine to begin with.

Some of the benefits of the 881 is its ability to fit through a standard doorway, run off a standard wall outlet, and have incredible maneuverability for such a robust machine. The addition of an accumulator posed a problem to the mobility of the table. Though the accumulator was designed to be a tall thin shaft to maintain its original dimensions as much as possible, as will be described in more depth later, the dry weight of the accumulator is over 600 pounds. While the bending table is on wheels, that much additional weight does sacrifice some mobility.

The greatest idea in the world may be shot down because the cost is too high or cannot be justified. It can be highly frustrating for an engineer to go through all of the prescribed engineering steps and follow the engineering process, come up with an amazing design or idea, and then have the project be scrapped due to the costing considerations. As an engineer, having an idea revoked solely on the premise of cost considerations is an extremely unsatisfying endeavor.

Though the time savings were calculated to be abundant, and therefore the cost savings noticeable, the problem that arose in the economics of this project was mostly associated with the initial cost of the accumulator. Having nothing to go on in the accumulator field from Greenlee, our initial specification was to keep the additional cost of the accumulator addition to be under a 20% increase from the base cost of the table, which calculated out to be around $600. We felt this was a reasonable asking price for such an upgrade at the time.

Working with Mark on the cost to manufacture the accumulator in house, or to have it outsourced, it became apparent that neither option was particularly cost effective. The rough calculations disregarding tooling, machining, labor, and the greatest cost; pre-charging, the cost of materials alone totaled almost $1,000. Including all of the necessary production steps, the accumulator would cost close to $2,500. Regardless of the dramatic improvements in bending time that the accumulator would have, there is no possible feasibility or economic justification for adding 100% of the initial cost as an add-on. Refer to Chapter 7: Design for Manufacturability for a more in depth analysis of the cost of components required.

Chapter 4: Potential Time-Savings for Accumulator

Three types of conduit, with varying wall thicknesses and materials, are typically used on the jobsite: Rigid, EMT, and IMC. For each time, Greenlee lists the amount of travel distance the ram must move in order to create bends for each type. The travel distances for varying conduit types, sizes, and bends can be seen below in Table 3: Ram Travel Distances where the highlighted fields are the worst case scenario (IMC):

		TOTAL RAM TRAVEL REQUIRED (ΔT) (in)											
		EMT				IMC				RIGID			
	SIZE	**2.5**	**3**	**3.5**	**4**	**2.5**	**3**	**3.5**	**4**	**2.5**	**3**	**3.5**	**4**
Degree of Bend	**0**	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
	10	0.81	1.06	1.00	0.88	1.13	1.38	1.06	1.19	1.06	1.38	1.00	1.06
	15	1.06	1.44	1.38	1.38	1.56	1.81	1.50	1.81	1.50	1.81	1.50	1.69
	30	2.00	2.44	2.38	2.44	2.88	3.13	3.00	3.19	2.88	3.13	2.88	3.19
	45	2.94	3.44	3.38	3.63	4.31	4.69	4.38	4.81	4.25	4.69	4.25	4.81
	60	3.81	4.50	4.38	4.88	5.81	6.19	5.81	6.44	5.63	6.19	5.75	6.44
	90	5.31	6.50	6.69	7.25	8.13	9.38	8.81	9.69	8.13	9.38	8.75	9.69

Table 3: Ram Travel Distances

Each travel distance corresponds to a fluid displacement required by the ram to perform the required bend. The fluid volume can be calculated using the well-known equation for the volume of a cylinder, shown below [Equation 1]:

Equation 1

$$V = \pi r^2 h$$

V=volume (in^3)
r = cylinder bore radius (in)
h = cylinder height (in)

Using the volume formula for a cylinder, we were able to calculate the fluid displacement requirement for the most common bends: 45 degrees, 60 degrees, and 90 degrees. Given the bore diameter of 3.25", we were able to find the following values shown in Table 4 that shows the fluid displacement for varying bends and conduit types. The highlighted fields indicate the greatest amount of fluid required to perform bends:

		FLUID VOLUME DISPLACEMENT REQUIRED (ΔT) (in^3)											
	SIZE	EMT				IMC				RIGID			
Degree of Bend	45	24.37	28.52	28.00	30.07	35.78	38.89	36.29	39.92	35.26	38.89	35.26	39.92
	60	31.63	37.33	36.29	40.44	48.22	51.33	48.22	53.40	46.66	51.33	47.70	53.40
	90	44.07	53.92	55.48	60.14	67.40	77.77	73.11	80.37	67.40	77.77	72.59	80.37

Table 4: Fluid Displacement for Varying Bends and Conduit Types

Charging times for the accumulator were also determined as a function of the fluid displacement required for the size of the pipe and for the degree of the bend. These values can be seen below in Table 5: Accumulator Charging Time:

Accumulator Charging Times (s)			
Pipe Size	**45°**	**60°**	**90°**
2.5"	38.34	51.66	72.21
3"	41.67	55	83.33
3.5"	41.67	55	83.33
4"	42.77	57.21	86.11

Table 5: Accumulator Charging Time

These values were determined from charts provided by Parker Hannifin, a leading manufacturer of piston and bladder accumulators. These charts took ambient temperature and the average system pressure requirements into account; the values were then interpolated from the charts and used as the baseline charging times for the accumulator. Given that the most amount of time needed to charge the accumulator will be 86.11 seconds, there will be more than enough time for the accumulator to fully charge in between bends.

Before running any performance calculations for the accumulator itself, our team decided to calculate the potential time savings of an accumulator based on an arbitrary discharge time of 15 seconds for each case. From the motor specifications, we know that a gerotor (positive displacement) pump will deliver up to 300 cubic inches of fluid per minute with no pressure load down to 65 cubic inches of fluid per minute at 300 psi. At 300 psi, the pump goes into its second stage, which is a high pressure capacity piston pump that delivers fluid at relatively low flow rates. The piston pump will deliver up to 65 cubic inches of fluid per minute at 300 psi down to 54 cubic inches per minute of fluid per minute at 10,000 psi. Given the linear relationship of the values, our team determined flow rate equations for the two stages. These can be seen below in Figure 5 and Figure 6:

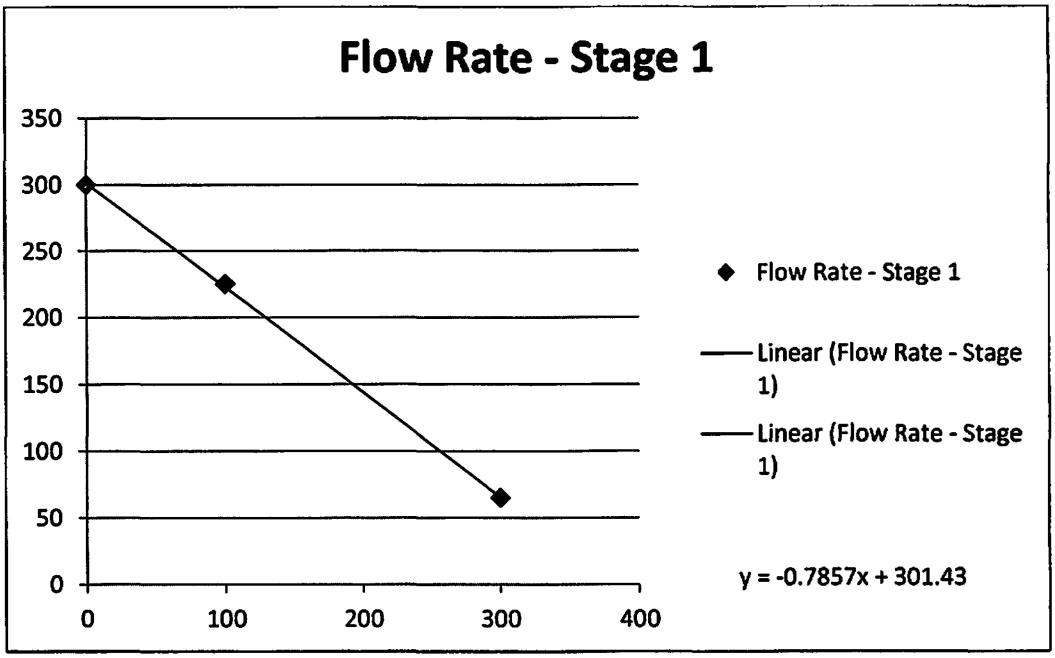

Figure 5: Flow Rate for the First Stage of the Pump (Gerotor)

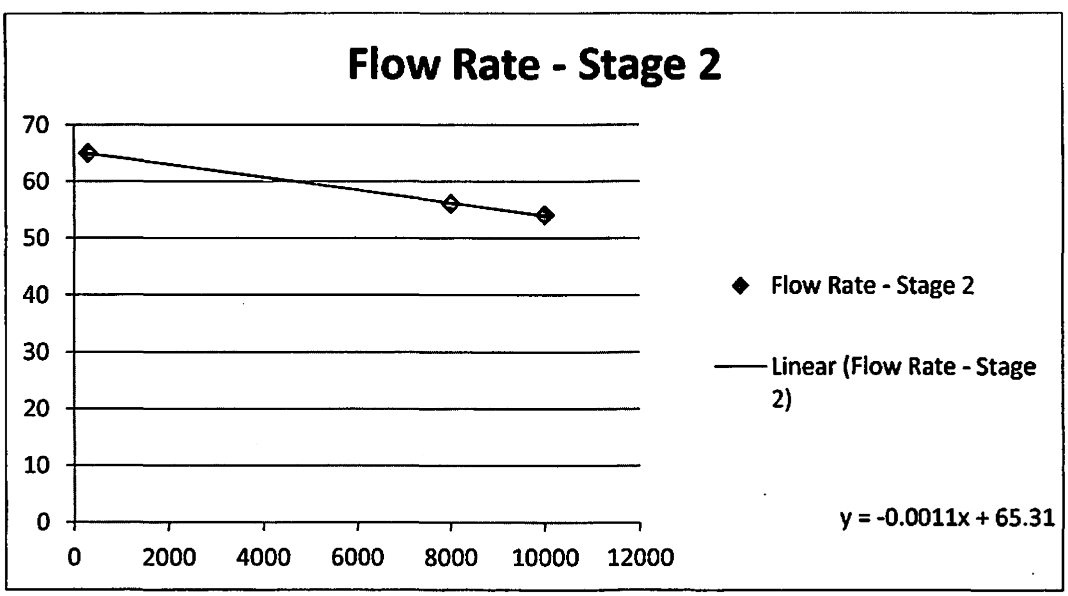

Figure 6: Flow Rate for the Second Stage of the Pump (Piston Pump)

Since we know the pressure requirements for bending with each pipe size and can estimate the flow rates of the current motor at any pressure between 0 psi and 10,000 psi, our team was able to make initial estimates as to how much time we could save with a pressure accumulator. We broke the results down into three categories: making a full 90 degree bend with the pressure accumulator, making a 60 degree bend with the accumulator followed by 30 degrees of bending from the motor, and then by splitting the work in half (45 degrees accumulator, 45 degrees motor).

At the time of these estimates, we did not know exactly how the accumulator would perform given that we did not have a final design. However, we picked an arbitrary performance value of 15 seconds of bending time, since we viewed that as our initial goal. Since we knew the hydraulic fluid flow rates of the motor at any given pressure, we calculated averages of time that could be saved in each scenario along with an expected improvement from the current design for each pipe size. Table 6 Shows the average time savings and improvement over the current setup for a full 90 degree bend performed by the accumulator, Table 7 shows the average time savings and improvement over the current setup for a 60 degree bend performed by the accumulator followed by 30 degrees of bending from the motor, and Table 8 shows the average time savings and improvement over the current setup for a 45 degree bend performed by the accumulator followed by 45 degrees of bending from the motor.

90 degree bends (Accumulator)		
Bend Size	Avg Time Savings (s)	Avg Improvement (%)
2-1/2	52.04	76.7
3	66.13	81.0
3-1/2	64.44	80.8
4	73.86	82.8

Table 6: 90 Degree Bend by the Accumulator

60 degree bends (Accumulator) + 30 (Motor)		
Bend Size	Avg Time Savings (s)	Avg Improvement (%)
2-1/2	35.36	51.9
3	43.26	53.0
3-1/2	41.29	51.6
4	48.68	54.5

Table 7: 60 Degree Accumulator Bend; 30 Degree Motor Bend

45 degree bends (Accumulator) + 45 (Motor)		
Bend Size	Avg Time Savings (s)	Avg Improvement (%)
2-1/2	25.45	37.2
3	32.17	39.3
3-1/2	30.33	37.9
4	35.91	40.1

Table 8:45 Degree Accumulator Bend; 45 Degree Motor Bend

From these results, we then assumed to save about 75-80% of the time with the full volume accumulator, 50-55% of the time with the partial volume accumulator, and 35-40% of the time with the half volume accumulator. Given the high performance from the full volume accumulator, we initially decided to pursue this design.

Chapter 5: Design Considerations

Our team then proceeded to get in touch with Bryan McGehee, an applications engineer at Parker Hannifin, specializing in hydraulic accumulators. Bryan was able to size an initial design and provide an estimated production unit cost. An accumulator that could deliver the full fluid volume from 10,000 psi to a minimum working pressure of 6,200 psi was 30 inches in length and approximately $5,500; the sizing was dependent upon the fact that the accumulator would be pre-charged to 6,200 psi.

After meeting Bryan and Jeff Sage, a salesman for accumulators at Parker Hannifin, they informed us that their distributor, Barker Rockford, only had pre-charge capabilities up to a maximum 3,000 psi. Given that Greenlee does not manufacture accumulators, the company would be dependent on Barker to pre-charge the prototype, so the 3,000 psi of pre-charge would become a limiting factor.

An additional design suggestion presented by our team was to design a 5,000 psi accumulator that would bend only the two lower pipe sizes. Since 5,000 psi accumulators are more common for Parker Hannifin to manufacture and are listed in their catalog, we felt that pursuing this route could help Greenlee save a considerable amount of cost towards the product; the obvious drawback to this design would be that the accumulator would be unable to perform any work on the two largest pipe sizes. Nonetheless, we asked Bryan to quote a 5,000 psi accumulator that would be pre-charged at 2,300 psi and would deliver the full fluid volume required to bend 2.5 and 3 inch conduit.

Bryan was able to give us the following specs: it would be 18 inches long, 128 pounds of dry weight, and would cost approximately $1,539 on a production basis. This resulted in a cost savings of $3,961 difference between the two designs, or 69.22% of cost savings. As a result, our team thought it would be worth it to pursue this route, especially for the purposes of prototyping and design validation. This new design was presented to Jerry Tully, the hydraulics engineer at Greenlee Textron.

After some discussion between Jerry Tully and Skyler Ripplinger, they had decided that the value of the product would come from the fact that it could bend all pipe sizes that could be used on a jobsite, and decided that the 10,000 psi unit would be the only unit worth pursuing. As a result, we decided to have Bryan quote us an accumulator that could deliver ¾ and ½ of the full working fluid volume to save on accumulator costs. Given a pre-charge requirement of 3,000 psi for both, Bryan specified the two accumulators as follows [Figure 7]: the half volume accumulator would be 2.5 gallons of size, 33.75 inches long, would have a dry weight of 427 pounds, and the three quarter volume accumulator would be 3.5 gallons of size, 42.5" long, and would have 520 pounds of dry weight.

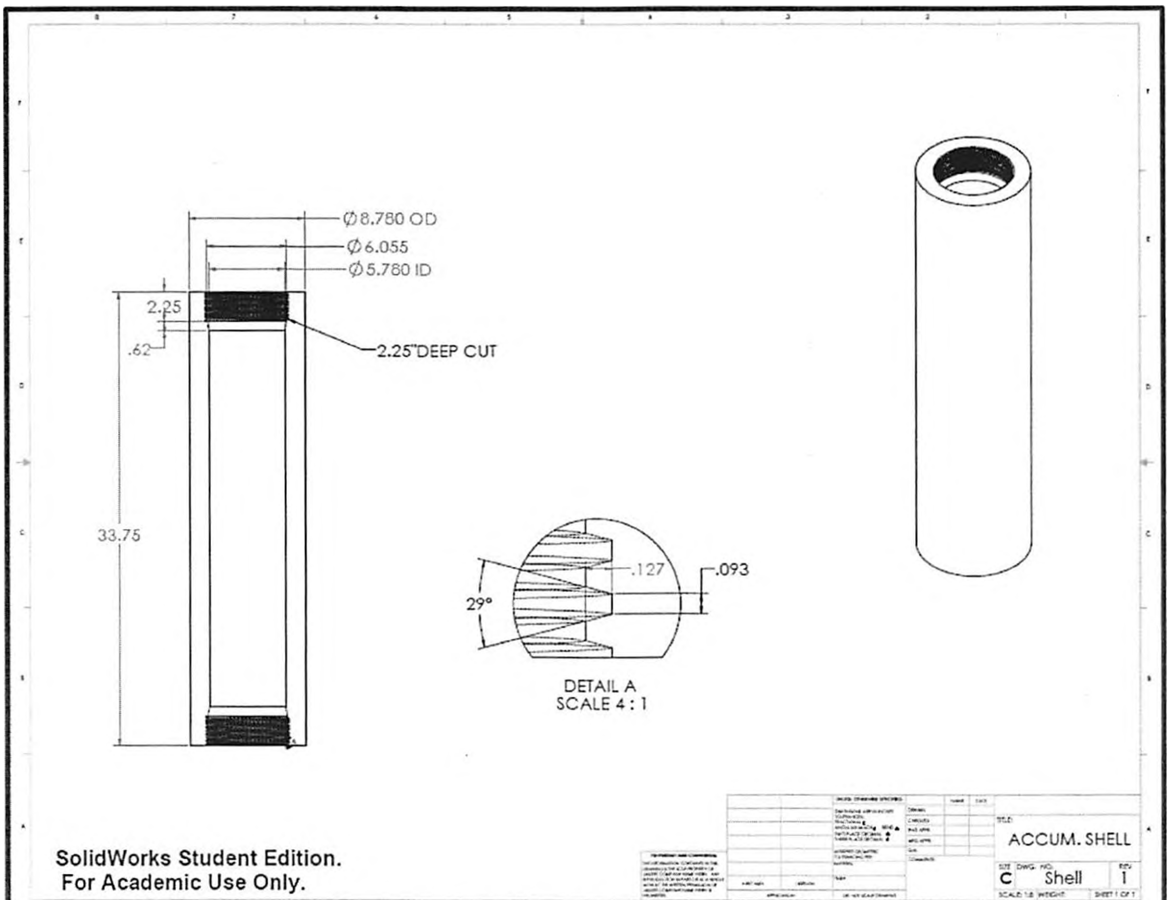

Figure 7: Accumulator Shell Dimensions for Final Design

Given the heavy weights for the two accumulators, we decided to pursue the smaller of the two options, or the half volume accumulator. After consulting with Mark Lancaster, an applications engineer at Barker Rockford, he was able to quote us an exact dollar cost for the accumulator with pre-charging, seals, and the required port diameter: it came to a $6,126.67 total cost. This price does not reflect the components that would be necessary to route the hydraulic fluid to the bending ram and then back to the pump reservoir.

Given the high cost of prototyping, Jerry Tully needed to discuss this aspect of the project with his project manager as to whether they would move forward with the project at this point. Mark had also given us a lead time of 10 weeks for the delivery of the accumulator. At this point, our team knew that regardless of the outcome of discussions, we would be unable to move forward with a physical prototype and testing for the purposes of the senior design course. As a result, we would have to move forward with simulations of the accumulator in order to validate our design.

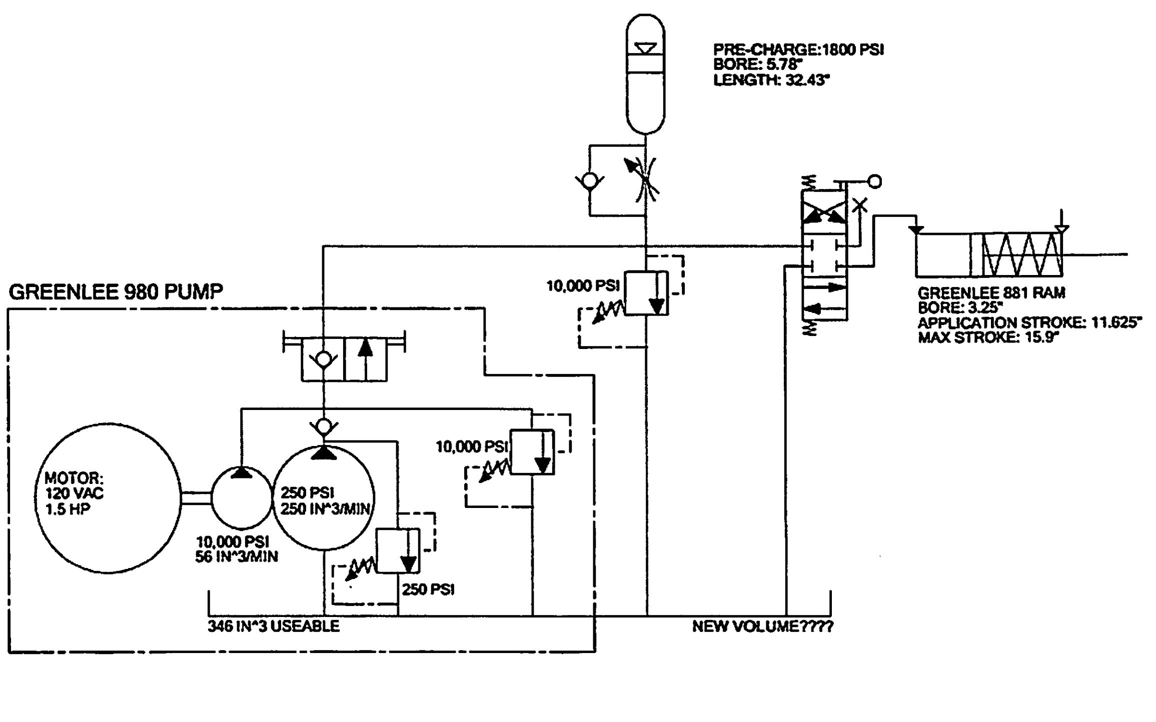

Figure 8: Schematic for Accumulator Setup on Bending Table

In addition to the accumulator final design, we needed to be able to show how the accumulator would be routed in conjunction with the hydraulic motor and the bending ram. The area shown inside the phantom line, in Figure 8 above, shows the existing motor setup. The motor will operate the gerotor, shown on the right, up to 250 psi, which will deliver higher flow rates at low pressures. After 250 psi, the piston pump, shown to the left of the gerotor, will deliver up to 10,000 psi at an average of 56 cubic inches of hydraulic fluid per minute. A check valve is shown at the top of the assembly; this valve ensures that the flow is one directional and restricts its tendency to move back to the pump reservoir. To the right of the pumps is the directional relief valve; this is shown on the pump as a manual handle and will direct the flow back to the reservoir if turned.

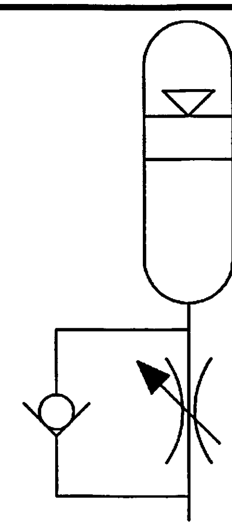

Figure 9: Needle Valve and Check Valve Setup

From the accumulator, we would likely come from the hydraulic port with a 3/8"coupling to a 3/8" hose. All components would need to be able to withstand up to 10,000 psi. In addition, we would direct the flow into a needle valve; the needle valve allows the end user to be able to manually adjust the flow at an infinite number of diameters from 3/8" to 0" [Figure 9]. A check valve is also shown parallel to the needle valve. As the accumulator releases fluid, the check valve closes as to prevent the fluid from flowing out of the largest path and forces it through the restricted needle valve. However, as the pump charges the accumulator to capacity, the valve allows for the free flow of hydraulic fluid back into the accumulator.

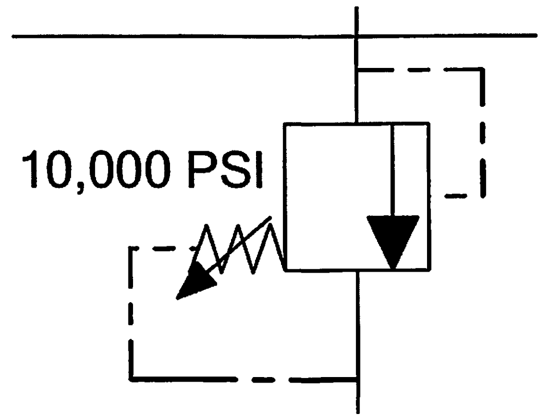

Figure 10: External Relief Valve

Attached from the accumulator is an additional relief valve [Figure 10]; this allows the end user to direct hydraulic fluid from the accumulator system back into the reservoir pump as necessary. To the right of the four way connection, the fluid would travel from the accumulator into a four way manifold [Figure 11].

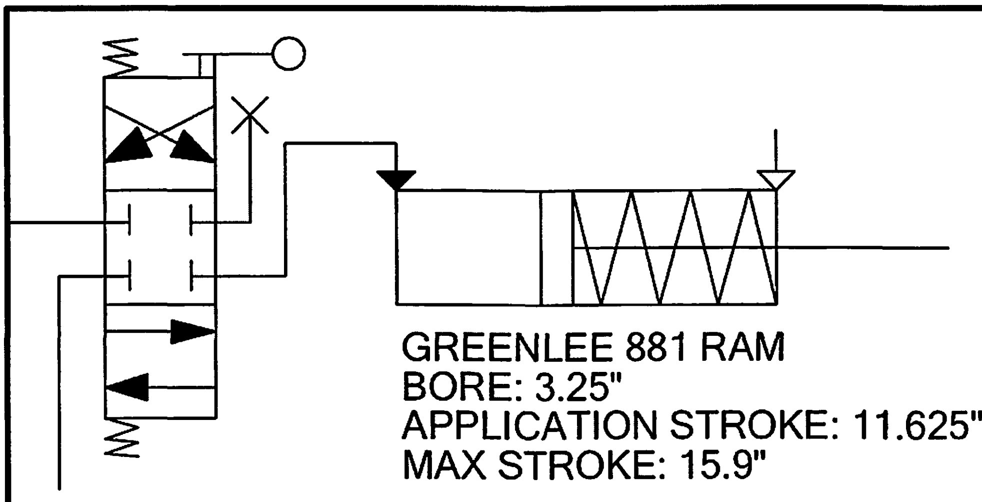

Figure 11: Four Way Manifold In-Line with the Bending Ram

At the top left of the manifold, the hydraulic fluid would flow inside; the directional valve shown would allow the end user to control the routing within the manifold manually. As the fluid flows to the ram, it would then exit the bottom right port of the manifold into the ram. After the bend is performed, the fluid goes back into the pump reservoir.

Chapter 6: Accumulator Performance

Our team had come to the point where we had a final design for the accumulator. It came out to be 5.78"ID x 8.78"OD x 33.75" length as specified by Parker Hannifin. Parker specified the 1-1/2" wall thickness in order to be able to maintain a factor of safety of 4 for the accumulator. The accumulator is designed to be able to deliver 48.25 cubic inches of hydraulic fluid to the bending ram. However, considering that we would be unable to prototype and test the unit within the timeframe of the senior design course, we've attempted other alternatives to test and validate our design.

After coming across the Oilgear Fluid Power Reference Handbook, they showed an equation for the pressure drop (ΔP) for a hydraulic circuit in series, such as our design. This equation is shown below [Equation 2]:

Equation 2

$$\Delta P = \frac{SG * GPM^2}{29.81^2}\left\{\left[\frac{1}{cd_1 * d_1{}^2}\right]^2 + \left[\frac{1}{cd_2 * d_2{}^2}\right]^2 + \left[\frac{1}{cd_3 * d_3{}^2}\right]^2 + \left[\frac{1}{cd_4 * d_4{}^2}\right]^2\right\}$$

SG = Specific Gravity of hydraulic fluid
GPM = Flow Rate of Fluid (Gallons per minute)
cd_n = orifice flow restriction variable
d_n = orifice diameter (in)

For the CD values of each orifice, we referred to a diagram which showed the specific cd value for each orifice type. This is seen below in Figure 12:

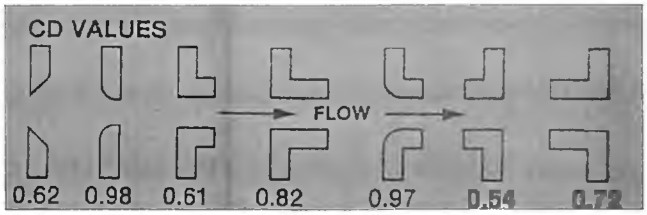

Figure 12: CD Values for Each or Orifice Type

For the purposes of this study, we looked at four main orifice openings in order to show our performance calculations. The needle valve constricting the flow is the most influencing factor in this calculation, given its small diameter. These orifices can be seen in Table 9: Orifice Types, Diameters, and CD Values:

Orifice Type	Diameter (in)	Assumed CD Value
Hydraulic Port	0.375	0.82
Hydraulic Hose	0.375	0.61
Needle Valve	0.0625	0.97
Ram Coupling	0.375	0.72

Table 9: Orifice Types, Diameters, and CD Values

With the needle valve having an infinitely variable diameter, the 1/16" opening is an arbitrary, small flow opening which showed to have the ideal performance after repeated attempts at each calculation. By using [Equation 2], and having each d and CD value, and the initial pressure difference between the maximum accumulator pressure (10,000 psi) and the minimum working pressure (6,200 psi for 4 inch conduit), we are able to back-solve to obtain the instantaneous GPM value at that moment in time. For the purposes of this study, we selected an iterative time step of 0.1 seconds to calculate the performance, now that we have both the GPM and Δt, we are also able to calculate the instantaneous amount of fluid displaced in 0.1s, in cubic inches of fluid.

Using Parker Hannifin's accumulator sizing selection guide for an accumulator with an auxiliary power source, we used their equation to determine the instantaneous pressure drop from the initial ΔP

Equation 3]

Equation 3

$$V_1 = \frac{V_W\left(\frac{P_3}{P_1}\right)^{\frac{1}{f}}}{\left[1-\left(\frac{P_3}{P_2}\right)^{\frac{1}{n}}\right]}$$

V_1 = Required Accumulator volume (in^3)
V_W = Fluid inside of accumulator (in^3)
P_1 = pre-charge pressure (psi)
P_2 = maximum operating pressure (psi)
P_3 = minimum operating pressure (psi)
f = nitrogen gas-charging constant
n = nitrogen gas-discharging constant

For the purposes of these calculations, we assumed an ambient temperature of 75°F, and with Parker's literature, results in f and n constants of 1.4. Since we know V_W starts at 48.25 cubic inches and then subsequently drops after each GPM calculation, V_1 is a fixed variable at 611 cubic inches of space, P_1 is constant at 3000 psi, P_3 remains constant depending on the size and type of conduit bend being performed, we can back-solve this equation for a new P_2 value, which can then be plugged back into Equation 2 for a new iteration.

Using the capabilities of MathCAD, our team was able to run performance iterations for the accumulator using each pipe size, and we were able to collect the following information at intervals of Δt = 0.1s for each size: pressure drop as a function of time, fluid displacement as a function of time, and flow rate as a function of time.

After performing calculations for 4 inch conduit, we found that the full fluid volume was delivered in 5.4 seconds, assuming a 1/16" needle valve opening. Given the fluid displacement requirements for 4 inch conduit, this will result in a bend that is more or less 60 degrees, depending on the type of conduit being used. To complete the 90 degree bend, the motor must be run after the accumulator has depleted its energy.

Throughout the accumulator's run of 5.4 seconds for the 4 inch conduit, we were able to determine the pressure drop, flow rate, and fluid displacement, all as a function of time. The pressure drop and flow rates as a function of time can be seen in Figure 13: Pressure Drop inside the Accumulator and Figure 134:

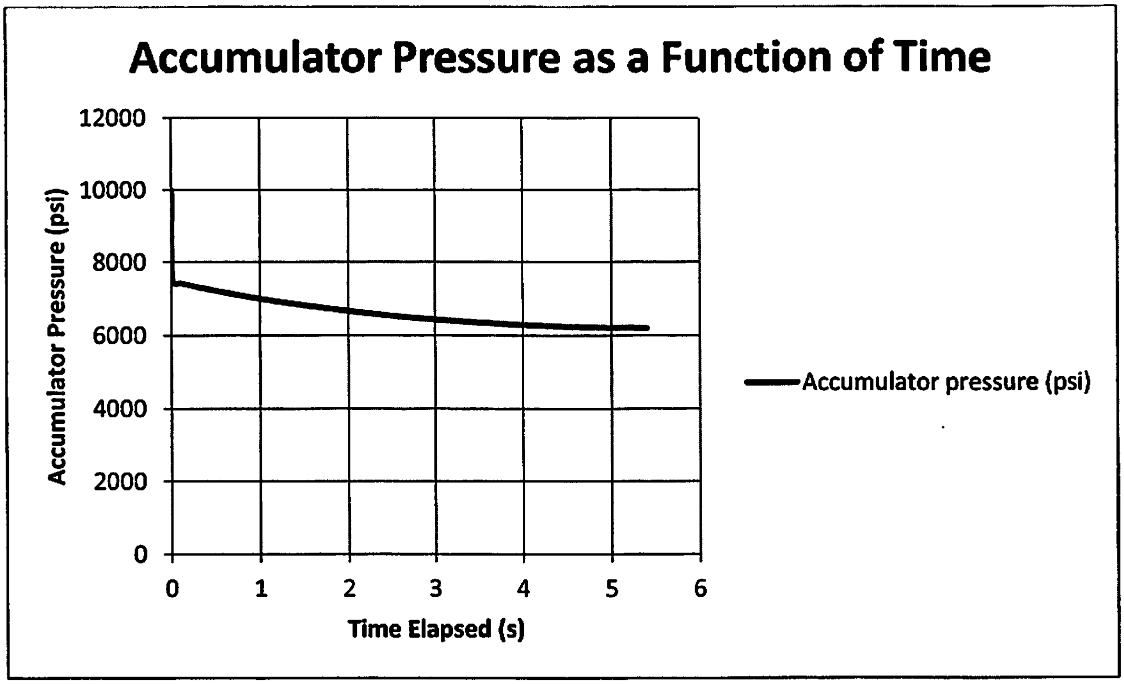

Figure 13: Pressure Drop inside the Accumulator

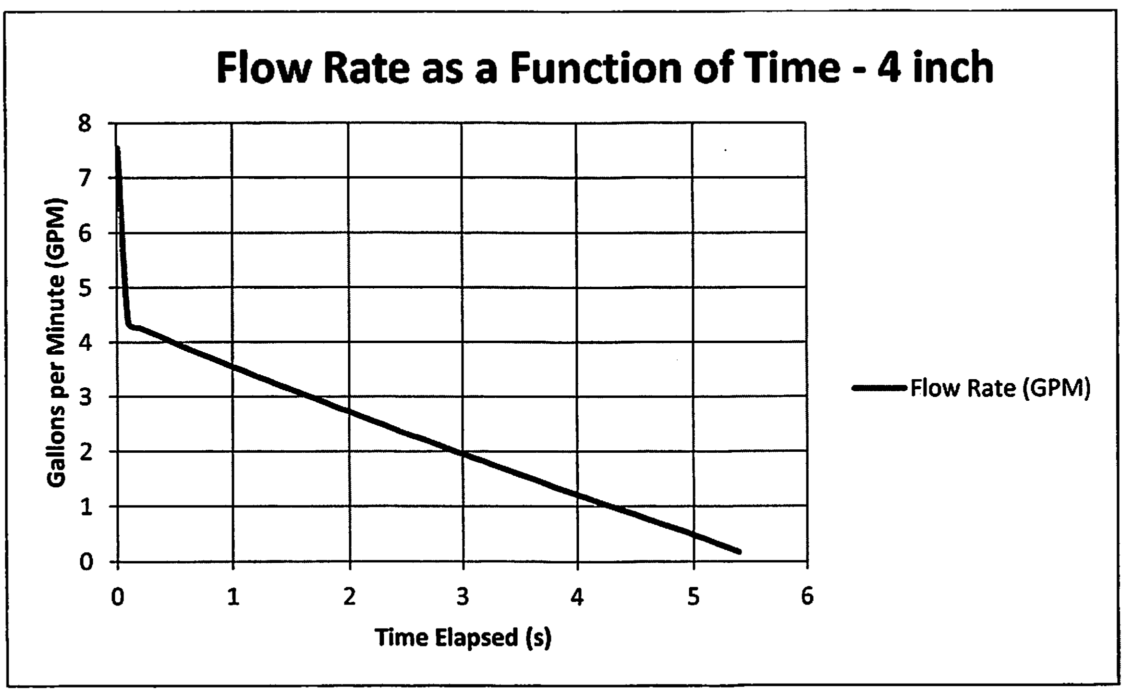

Figure 14: Fluid Flow Rate from the Accumulator

In Figure 13: Pressure Drop inside the Accumulator, we observed that there was a sudden pressure drop of about 2,500 psi within the accumulator, which then leveled off as the rest of the energy was depleted. We researched that this pressure drop is a common phenomenon within pressure accumulators and that this is to be expected which is factored into our calculation. Considering that the accumulator is designed to deliver all of its fluid within the minimum working pressure of 6,200 psi, it makes sense that the flow rate goes to zero as the bend is performed. It can be observed from Figure 14 that, as the accumulator has the sudden drop in pressure, the flow rate drops suddenly, but then steadily decreases in a linear fashion towards zero.

For the 3.5", 3", and 2.5" conduit, the pressure drop as a function of time remained relatively the same as the 4" case. This is likely because the accumulator was designed to deliver the full volume of fluid from 10,000 psi to 6,200 psi, regardless of the working pressure. However, the fluid volume for the 3.5" conduit was delivered in 2.5 seconds, 1.9 seconds for the 3" conduit, and 1.4 seconds for the 2.5" conduit. While the pressure drops remained the same, there was a noticeable difference in flow rate for each size. These flow rates, for each corresponding pipe size from 3.5" to 2.5", can be seen in Figure 15, Figure 16, and Figure 17:

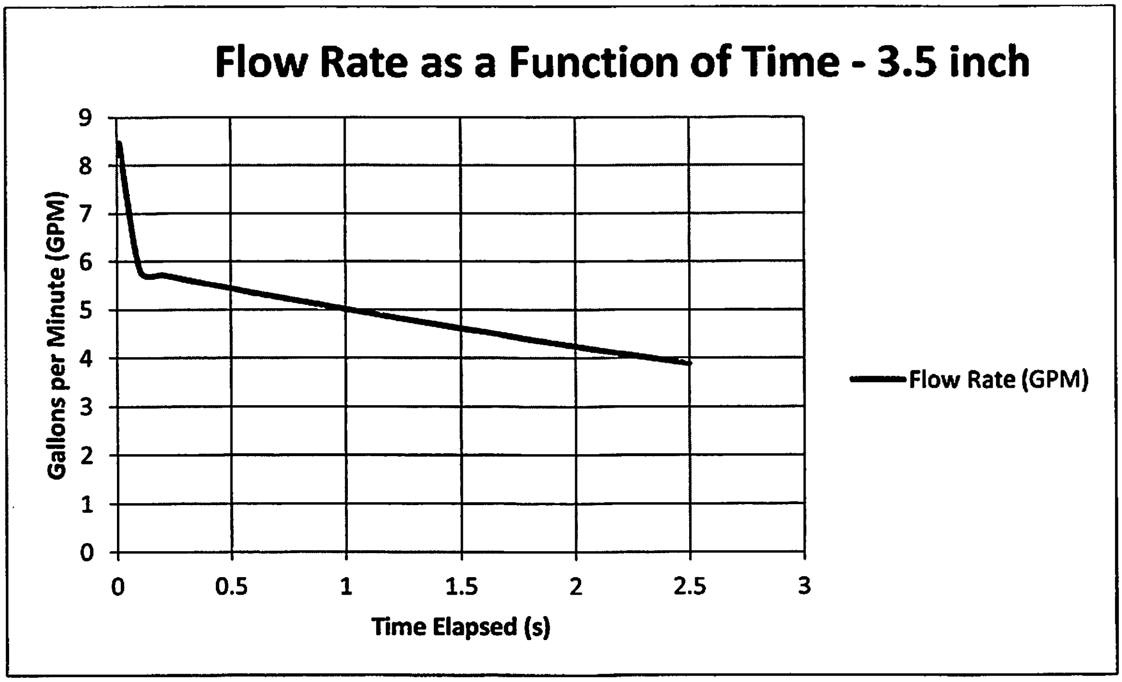

Figure 15: Flow Rate as a Function of Time for 3.5" Conduit

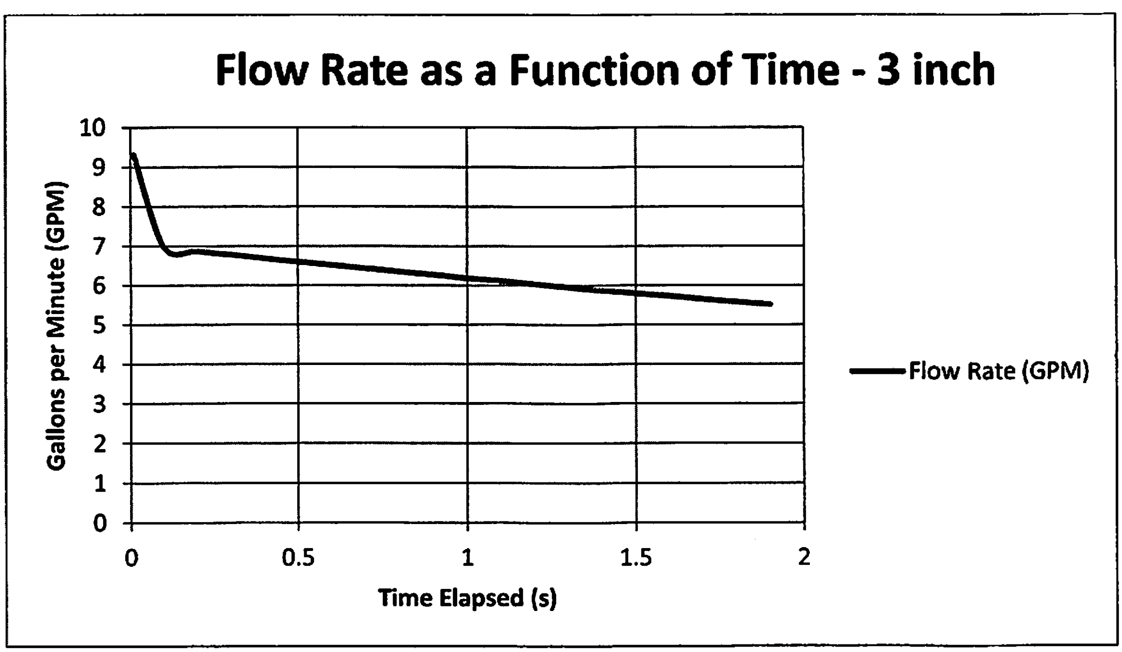

Figure 16: Flow Rate as a Function of Time for 3" Conduit

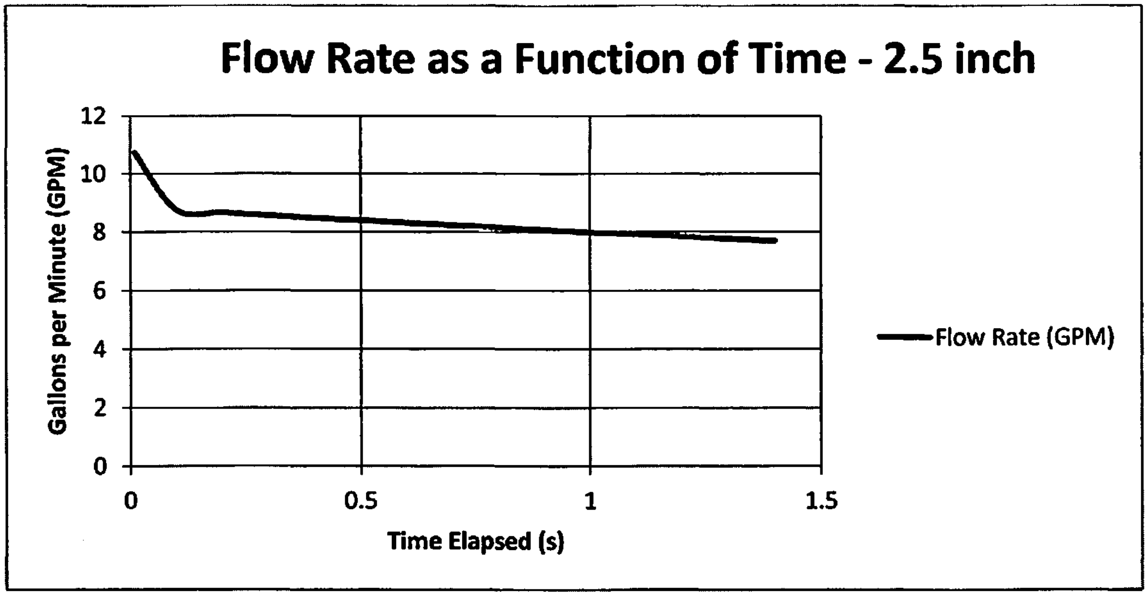

Figure 17: Flow Rate as a Function of Time for 2.5" Conduit

With each decreasing pipe size, the initial flow rate of the accumulator increased; the rate of change of the flow rate then decreases with each decreasing pipe size. This is likely because, as the accumulator has a weaker working pressure with each decreasing pipe size, it has less resistance towards performing work; as a result, the work is performed faster and at greater rates with each decreasing working pressure requirement.

Chapter 7: Design for Manufacturability

A comparison for Greenlee Textron was made to confirm the difference of costs between in-housing and out-sourcing the production of a 10k psi accumulator for use on the 881 bender. A comparison costing report was created below describing both the process and costing associated with each process. With recent sales of the 881 bending table around 300 units per year, it was assumed that approximately 30% of new units would have the upgrade with the accumulator. From this consideration, all costing was configured upon the basis of producing 100 accumulators annually.

Out-Sourcing

Option one for Greenlee Textron to consider is to purchase an accumulator from a wholesale manufacturer. A company out of Rockford, Parker Hannifin, was contacted for outsourcing and prototyping the accumulator. The original prototype would cost approximately $6200 to have completed. The actual production unit would cost approximately $3450 to purchase from Parker's wholesaler Barker. The purchased unit would be an assembly of the piston, end caps, and shell. Upon being delivered to Greenlee, the Schrader high pressure valve and valve bracket would have to be assembled adding an additional cost to the unit. The Table 10: Outsourcing Cost below displays a breakdown of the overall cost of outsourcing the accumulator per part as well as the annual cost of production when considering the production of 100 units per year.

Outsourcing Costing Summary		
Name	Cost/Piece	Annual Cost
Accumulator	3450	345000
798 Schrader High Pressure Valve	25.51	2551
Valve Bracket	8.78	878
5/16-18 Grade 5 Bolts	0.12	24
Assembly	25	2500
	Total	350953
	Cost/unit	3509.53

Table 10: Outsourcing Cost

The total cost to Greenlee per accumulator unit comes out to approximately $3509.53.

In-Housing

Option number two for Greenlee to consider is purchasing the material and equipment to produce an accumulator in-house. Many factors must be considered for this. The first step was in selecting material and pricing out costs accordingly. For the shell, end caps, and piston, 1020 cold drawn steel was chosen. 1020 steel was chosen for its machinability and strength. It can also be heat treated if surface hardening is necessary later on. The valve bracket also is made out of 1020 steel and will be fastened with two 5/16-18 grade 5 bolts. Grade 5 bolts were determined strong enough for use as the bracket serves only as a guard. Schrader International was also contacted and a high pressure valve was selected that can handle the high pressures of the accumulator.

The material chosen for the four different seals was either Buna Nitrile or PTFE depending on the application. Buna Nitrile was chosen for the -431 V-Ring on the piston and the -431 o-ring used on the end caps. Buna Nitrile was chosen for its durometer hardness of A70 and its material properties which allow it to be compatible with oil. PTFE (Polytetrafluoroethylene) was chosen for the glide rings and the backup washers for its properties of being non-reactive and non-stick with a low coefficient of friction making it ideal for use as a solid lubricant reducing friction and wear in the accumulator.

The accumulator components, materials of components, and cost per component may be viewed in Table 11 below.

Bill of Materials				
Quantity	Name	Material	Price/unit	Total Price
1	Shell	1020 DOM Steel	568.01	568.01
2	Endcaps	1020 Steel	85.7	171.4
1	Piston	1020 Steel	85.7	85.7
2	-431 Glide Rings	PTFE	18.57	37.14
1	-431 V-Ring	Buna Nitrile	2.47	2.47
4	-431 Backup Washers	PTFE	3.85	15.4
2	-431 O-Ring	Buna Nitrile	1.22	2.44
1	798 Schrader High Pressure Valve	Stainless Steel	25.51	25.51
1	Valve Bracket	1020 Steel	8.78	8.78
2	5/16-18 Grade 5 Bolts	Zinc Plated Steel	0.12	0.24
		Total Material Price per Unit:		**917.09**
NOTE: -431 relates to a AS568A standard dash number				

Table 11: Bill of Materials

The total material cost to make one accumulator came out to be $917.09.

The next cost to consider was that of machining. A Haas TL-2 CNC was chosen for its size and ability to machine the shell which is 8.75" in OD and 33.75" in length. A onetime setup cost was assumed and a basic annual machine maintenance cost was configured as well. Assuming that the machine would be placed in an existing machine shop, basic drills, end mills, reamers, etc were not factored into the costing. Specialty tooling for the boring, facing, and threading operations were considered and priced out accordingly from Grainger. The cost of the tooling and tool holders may be viewed in the table below.

The next cost to consider was that of labor. Setup time and run time for each operation and part was considered. Labor was based on eight hour days with a labor cost of $15 an hour. The assembly costs are the labor costs associated with transporting the material to a new location in the plant and assembling all of the components together. A step-by-step process will be provided. The labor costs per part can be viewed as well as the annual cost assuming production of 100 pieces in a year.

The last expense considered was that of pre-charging. Equipment such as gas bottles and a charging set would have to be purchased in order to pre-charge the accumulators up to 10,000 psi. The cost of nitrogen was also factored into the pricing.

In-House Annual Costing Summary			
Name	Description	Cost/Piece	Annual Cost
Materials		917.09	91709
labor	Set-up Time (assume 1 day)	0.8	80
	Run Time (assume 30 parts/day for shell)	4	400
	Run Time (assume 100 parts/day for End Caps/Piston)	1.2	360
Machines	Haas TL-2 CNC Lathe	31995	31995
	Machine Set-Up	6500	8000
Specialty Tooling	Boring Bar+Tool holder	395.5	791
	Thread Insert+Tool Holder	240.1	480.2
	Facing Insert+Tool Holder	155.55	311.1
Maintenance	Machine Maintenance		500
Assembly		55	5500
Charging	Equipment	60000	60000
	Nitrogen	500	150000
		Total	350126.3
		Cost/Unit	3501.26

Table 12: In-House Annual Costing Summary

The total cost for Greenlee to produce an accumulator in-house for the first year considering all expense considered in Table 12 comes out to be $3501.26. In the first year of production, the cost of in-housing and outsourcing are very similar. In order to get a better comparison, the first ten years of production were compared against each other. It can be seen that over a ten year span, the average cost of in-housing an accumulator drops down to approximately $2601.31/unit [Table 13]. The cost of outsourcing remains the same at $3509.53/unit. This figure is likely to increase in the future with inflation, so this figure remains to be interpreted as the very best case scenario where outsourcing cost remains constant. There is a significant cost savings to in-housing, however the assumptions made above when developing costing must be considered. Depending on warehouse space, inventory, labor personnel, and other factors, out-sourcing may become the more viable option.

Ten Year Costing Summary		
Year	In-House	Out Source
1.00	350126.30	350953.00
2.00	250131.30	350953.00
3.00	250131.30	350953.00
4.00	250131.30	350953.00
5.00	250131.30	350953.00
6.00	250131.30	350953.00
7.00	250131.30	350953.00
8.00	250131.30	350953.00
9.00	250131.30	350953.00
10.00	250131.30	350953.00
Total	2601308.00	3509530.00
Cost/Unit	2601.31	3509.53

Table 13: Future Costing Summary

A basic manufacturing process was created. The process is set up to allow changes to be made after the initial production run determines what works well and what doesn't for Greenlee. After material is purchased, the first step will be machining.

Machining

1) Initially, the outside casing is cut from DOM (Drawn over Mandrel) Steel. It is cut slightly oversized in length. DOM steel is used because it holds its ID and OD tolerances tighter when being extruded compared to other typical extrusions. The tighter tolerances reduces the amount of additional machining.
2) The DOM tubing is placed in the Haas TL-2 turn center. Both ends must be faced, resulting in the final length of the tube at 33.75".
3) The shell is then run through another operation where the ID is bored out to 5.75". A possible option for Greenlee would be to look into outsourcing this process because the cost of the machine to bore out large accumulators is extremely high. After proper ID sizing, internal threads are added to each end of the shell to supply threads for both endcaps.
4) The end caps are then cut to length from the 1020 solid bar stock. They are then faced and turned down to size using a turn center.
5) External threads are added to the faces of both the end caps to allow for assembly. Holes/ports for the inlet and outlet fittings can then be added by an operation in a mill.
6) The internal slide, piston, is then turned using similar methods to that of the end caps by using a turn center.

Assembly

7) The V-Ring and glide rings are added to the piston and the o-ring and backup washers are added to the end caps.
8) The piston, shell, and end caps may then all be assembled.
9) The schrader high pressure valve and valve bracket may then be added to finish up the initial assembly.

Pre-Charging and Testing

10) The 6200 psi precharge is then added to the accumulator using the gas bottles and charging manifolds.
11) Leak and pressure testing is then done to complete the production process. The testing is very important as there is a large safety liability associated with bottled up high pressure equipment.

By following the basic plan stated above, a method and process has been set up for Greenlee which can be used to potentially in-house an accumulator design down the road. The process is set up to provide Greenlee with basic figures to determine which route of production best fits their company policies.

Chapter 8: Results

Final design considerations of the shell may be viewed below in Figure 18: Drawing Specifications of the Shell. The final length of the shell was 33.75" with an OD of 8.75" and an ID of 5.78". The ends of the shell contain 6-8 UNC -2B threads with a slight taper after the threads for the end caps to seat in.

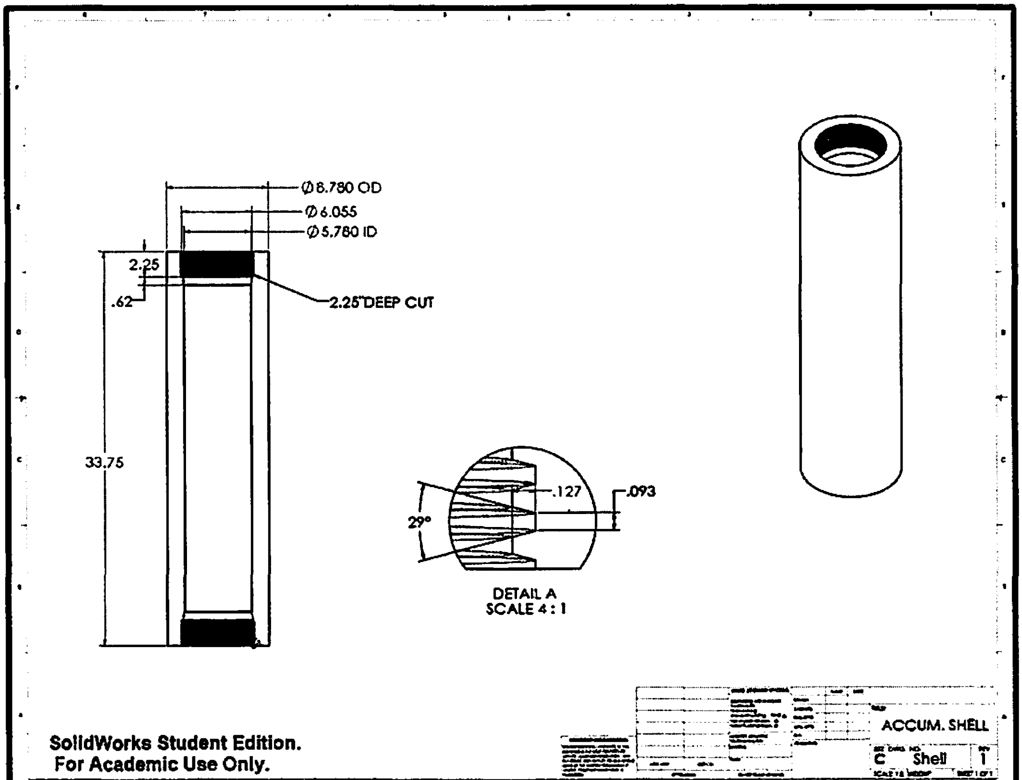

Figure 18: Drawing Specifications of the Shell

A Von-Mises stress FEA analysis was conducted to determine the strength of the shell. The 1020 steel shell had a yield strength of about 50,763 psi. As can be seen below in Figure 19: Von Mises Stress Analysis of the Shell, the largest stress the shell will likely see is about 15,000 psi. This means that the shell is designed to withstand three times the amount of stress that it will see. From this, it was determined that the shell passes the design consideration.

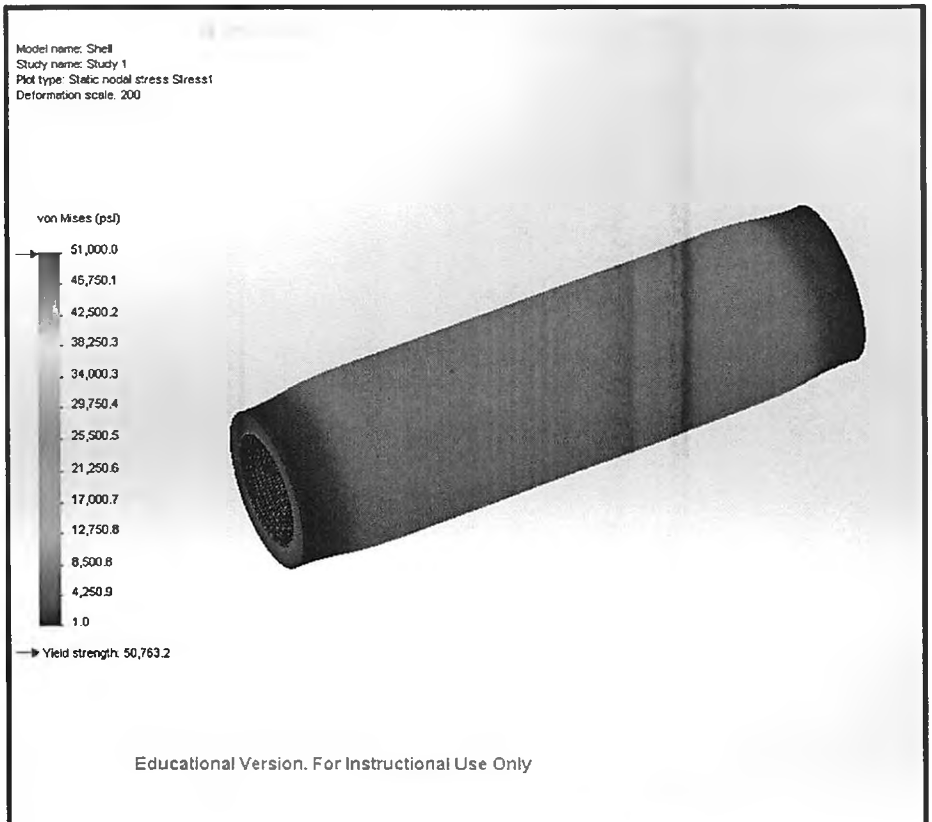

Figure 19: Von Mises Stress Analysis of the Shell

The end caps ended up being 3.5" in length. A 6-8 UNC -2A thread was applied externally to the end caps as well as the taper to seat with the shell. The end caps were identical except that the charging side end cap had three holes, two for a bracket and the other for a Schrader valve, while the outlet end cap had one hole for entering and exiting hydraulic fluid. Drawings for both end caps may be viewed below in Figure 20 and Figure 21 respectively.

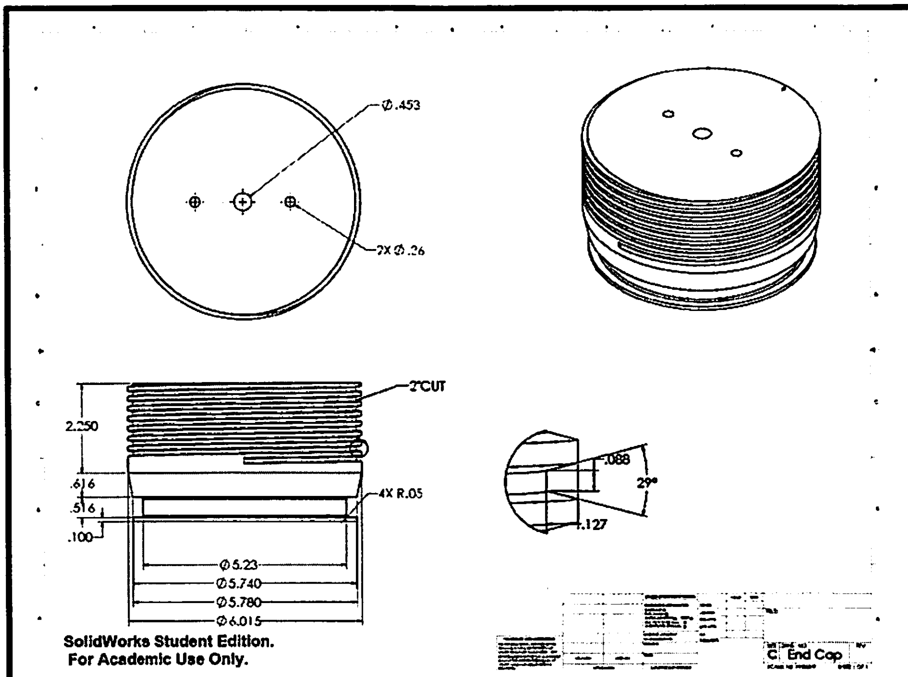

Figure 20: Charging Side End Cap

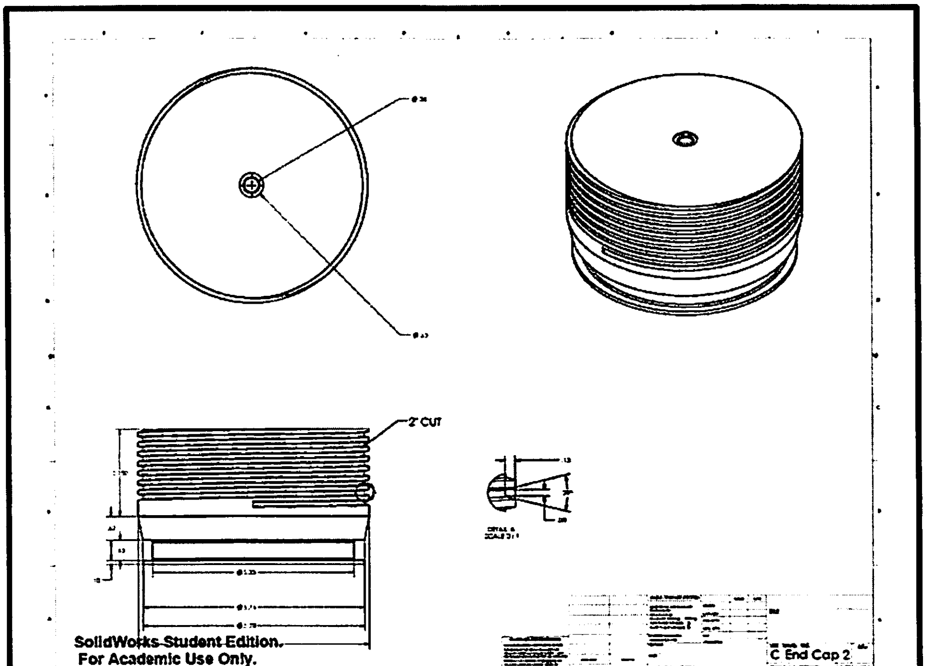

Figure 21: Hydraulic Fluid Exit Side End Cap

Analysis was conducted on the 6-8 UNC threads to confirm that they would support 10,000 psi of pressure. The analysis may be viewed below in Figure 22. It was confirmed that, with a length of 3.5", the threads would have sufficient strength to withhold the expected 10,000 psi of pressure.

Determinining Length of Thread

The thread being considered is the 6-8 UNC thread on the endcaps and shell

$S_u := 68000$ $\quad D := 5.99$ $\quad E_s := 5.928$

$L_e := 3.5$ $\quad n := 8$

E.s=minimum pitch diameter of external thread (in)
A.s=tensile stress area of bolt (in^2)
A.TS=shear area at pitch line of both threads (in^2)
L.e=length of thread engagement (in)
S.u=ultimate shear strength (psi)
F=Force required to shear threads (psi)
D=maximum inner diameter of internal thread (in)

$$A_{TS} := \pi \cdot n \cdot L_e \cdot D \cdot \left[\frac{1}{2 \cdot n} + .57735 (E_s - D) \right]$$

$$F := S_u \cdot A_{TS}$$

$$F = 9.568 \times 10^5$$

The internal force of 10000 psi that the accumulator will experience will not shear the threading on the endcaps

Figure 22: Confirmation of End Cap Thread Strength

The FEA analysis of the threading can be seen below in Figure 23. The 1020 steel used for the end cap had a yield strength of approximately 51,000 psi. The highest stress that the end cap will be likely to see is around 30,000. This means that the end cap can withstand over 1.5 times the stress that it will be expected to see when in operation.

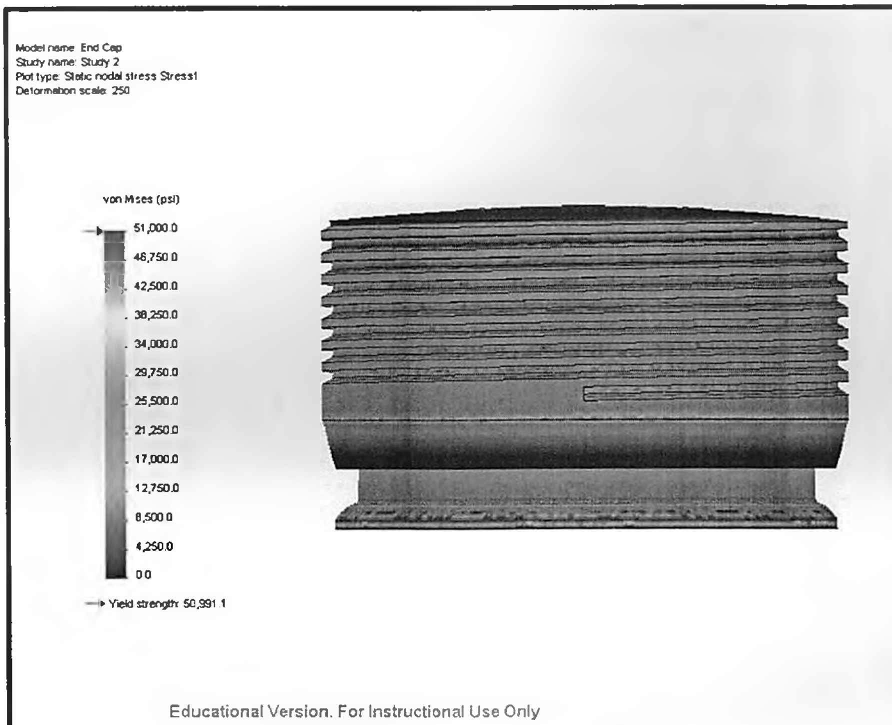

Figure 23: FEA analysis of the End cap

A drawing of the final piston may be viewed below in Figure 24. The length of the piston is 3.5" and the OD was 5.75" to allow for clearance within the shell. Grooves were added to the outside edges of the piston to allow for o-ring and glide ring placement.

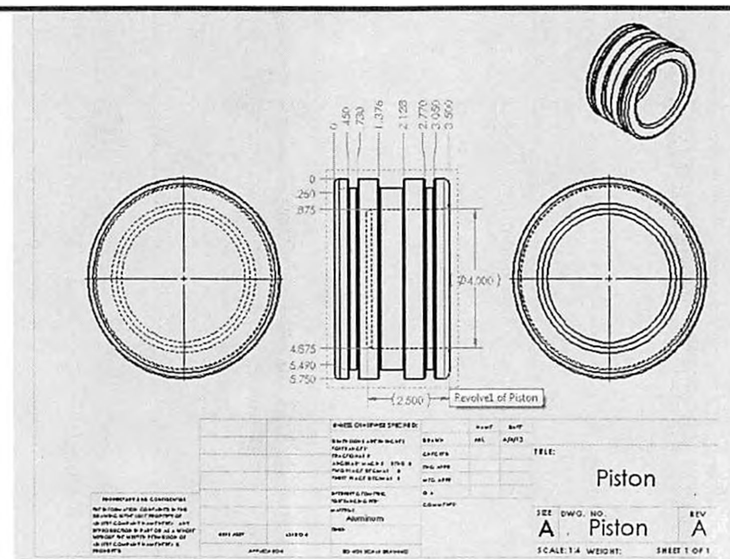

Figure 24: Piston Drawing with Specs

The FEA analysis of the piston below in Figure 25 confirms that the piston was designed sufficiently. Considering that, at most, the piston will have a maximum displacement of 0.04 mm, this deformation can assumed to be negligible for this application. This piston should be able to last the 3,000,000 life cycle limit of your typical piston accumulator.

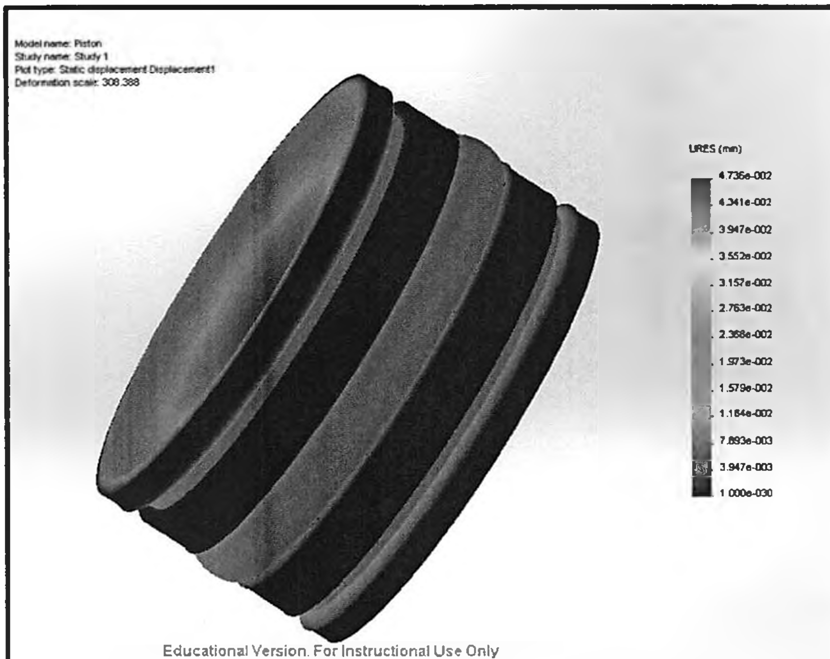

Figure 25: FEA Analysis of the Piston

The Schrader valve chosen for use with pre-charging the accumulator was selected from Schrader International's catalog. The 798 High Pressure Valve was chosen due to its ability of withstanding the high pressures that will be expected. Testing on the valve wasn't necessary as Schrader guaranteed the operation of their valve. The drawing specifications of the valve may be viewed in Figure 26 below.

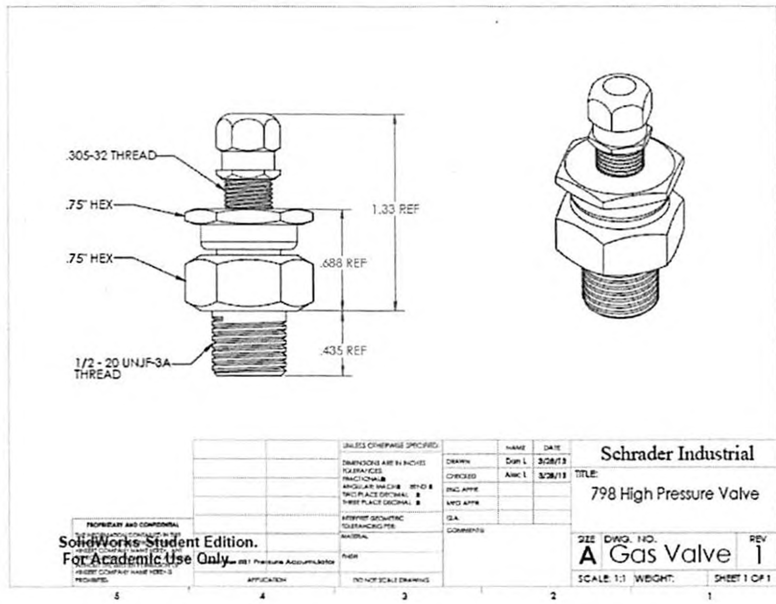

Figure 26: 798 High Pressure Schrader Valve

An assembled piston hydraulic accumulator with all components may be viewed below in Figure 27. The assembly consists of the shell, end caps, piston, Schrader valve, guard bracket, and assorted o-rings.

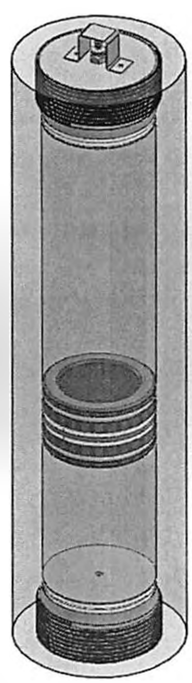

Figure 27: Finalized Hydraulic Piston Accumulator Assembly

Chapter 9: Discussion and Conclusion

Designing an accumulator for use on the 881 conduit bender was found to have huge benefits it terms reducing bend times. Time savings of up to 80% could be expected when using the accumulator to perform the entire bend. With quicker bend times, electrical contractors can complete jobs quicker increasing their job profit. The biggest drawback to the design is the large cost associated with the price of a 10,000 psi accumulator. Not only is the cost of production high, but the added cost of pre-charging and safety liabilities removes the practicality of the design. To this end, other cost effective alternatives for reducing the 881 bend times were discussed and presented to Greenlee Textron.

From this conclusion, Greenlee Textron has been advised not to pursue an accumulator prototype for the 881 bender. Instead, pursuing a larger hydraulic motor or varying the design of the motor would be beneficial. Testing of the SPX pump proved that savings were possible. Cost will still be an issue as the SPX pump ranged around $4500, however, Greenlee may be able to make changes to their existing pump in order to better its performance. Such changes could possibly be tightening up tolerances on parts, increasing the size of the motor, or redirecting flow of the fluid through the pump. Such changes will be up to Greenlee's engineering staff that will continue the project. Research on hydraulic accumulators for use with Greenlee Textron products was not wasted as the technology is believed to have potential on other Greenlee products in the form of small scale accumulators. This research will also be passed on to Greenlee's hydraulic production engineers assigned with the task of improving the product line. Even though the accumulator assisted hydraulic conduit bender failed to prove practical, the knowledge gained through research will still be applied towards product improvements.

References

[1]"Hydraulic Accumulators." Parker Hannifin, n.d. Web. 29 Nov. 2012. <http://www.parker.com/ >.

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<http://www.greenlee.com/products/BENDER>.

[3] Parker Hannifin. *Hydraulic Accumulator Products*. Rockford, IL: Parker Hannifin, 2003. Print.

[4] *Accumulator Formulas*. Digital image. Oilgear Fluid Power Reference Handbook, n.d. Web.

[5] "Sizing and Selection." *Sizing and Selection* (n.d.): 129-40. Web. <http://www.parker.com/literature/Literature%20Files/ACCUMULATOR/CAT/ENGLISH/1630007.pdf>

Joanne Ganshirt - Re: Honors Capstone Approval

From: <cmjones@niu.edu>
To: "Brianno Coller" <bcoller@niu.edu>, <bdcoller@gmail.com>, "Joanne Ganshi...
Date: 5/12/2013 6:17 PM
Subject: Re: Honors Capstone Approval

Thanks, Brianno. Regards, Chris
Sent via BlackBerry by AT&T

From: Brianno Coller <bcoller@niu.edu>
Sender: bdcoller@gmail.com
Date: Sun, 12 May 2013 16:41:06 -0500
To: Christopher Jones<cmjones@niu.edu>
Subject: Re: Honors Capstone Approval

I approve projects for Scott Anderman, Jen Case, and Alec Fisher.

On Thu, May 9, 2013 at 10:33 PM, Christopher Jones <cmjones@niu.edu> wrote:

> Dear Colleague:
> You are receiving this message, because you are the Honors Capstone supervisor of one or more of the following University Honors students.The University Honors Programhas received an Honors Capstone from the listed students without the standard approval page. Thus, it is important to know whether you have reviewed and approved the project. Once your approval has been noted, the student's graduation with Upper Division Honors or full University Honors will be processed by my office.
> Would you please do me the kind favor of e-mailing or calling Joanne Ganshirt (jganshir@niu.edu; 753-9398) about your student on Friday or no later than Monday? We will not trouble you with signing paperwork, but we do need your clear approval in an e-mail or over the telephone.
> Scott Andermann
> Paul Antczar
> Jennifer Case
> Michelle Case
> Alec Fisher
> Joseph Griffey
> GeoffMiller
> Andrew Wegner
> Thank you.
> Cordially,
> Chris
> Christopher M. Jones, Ph.D.
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Capstone Approval Page

Capstone Title (print or type) Pressure Accumulator for a Hydraulic Motor

Student Name (print or type) Alec LeHouillier

Faculty Supervisor (print or type) Dr. Sciammarella

Faculty Approval Signature

Department of (print or type) Mechanical Engineering

Date of Approval (print or type) 5/1/13