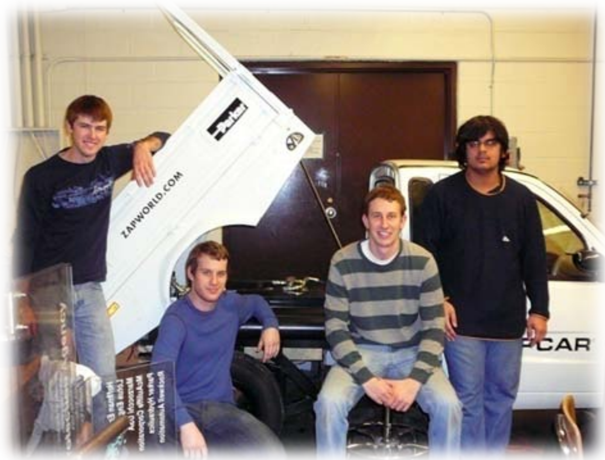
The Xebra

Electric-Hydraulic Hybrid Vehicle



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ME 450
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EXECUTIVE SUMMARY

The Xebra electric-hydraulic hybrid vehicle is an ongoing project at the University of Michigan sponsored by the Environmental Protection Agency (EPA). This vehicle is designed for intercity driving, where frequent start and stop situations exist. For this semester, the EPA proposed a challenge to safely increase the Xebra's current top speed from 35 mph to 45 mph while maintaining its current acceleration, efficiency, range, aesthetics, and reliability. The EPA also preferred that any alterations to the vehicle be reversible and transferable to future semesters.

The currently installed regenerative braking system is designed to recover kinetic energy for future use by storing it in the form of pressurized fluid in high pressure accumulators. The energy that is stored in the high pressure fluid will be used to accelerate the vehicle from stand still. Once the desired speed is reached, the electric motor will be used to maintain the vehicle's speed. This will improve the efficiency of the hybrid system by only using the electric motor in situations where low torque and electrical power are required.

The main design concepts that were generated to increase the top speed included a combination of increasing the hydraulic motor size, increasing the pressure of the hydraulic system, increasing the accumulator size, and altering the gear ratios. After talking to our sponsor and through Pugh chart analysis, we determined the most feasible options, which led to our alpha design. Then, through a rigorous engineering analysis we optimized our alpha design which led us to our final design concept.

Our final design concept consists of two additional accumulators, larger diameter wheels, modified gear ratios, and an additional cooling fan for the electric motor. The additional accumulators will double the Xebra's energy storage capacity, allowing for longer regenerative braking and hydraulic launch cycles. Adding larger wheels and modifying the gear ratios will ensure that the hydraulic pump and motor, as well as the electric motor, will not exceed their RPM limitations when the vehicle reaches 45mph. Finally, the new cooling fan will ensure that the electric motor will not overheat due to the additional load resulting from the higher top speed.

Through a series of cutting, milling, turning, and welding processes, we were able to fabricate and assemble all of the required components into a final design prototype. Upon completion of our prototype, we were able to test our design concept on the dynamometer at the EPA. Due to unforeseen problems with two of the vehicle's systems, we were unable to complete a full validation of our prototype. While testing, we discovered that the electric motor controller was malfunctioning. Also, we determined that the regenerative braking system was not fully functional due to a faulty recirculation valve. These two problems did not allow us to complete a drive cycle and thus, test the vehicle's overall range and efficiency. However, this testing showed that our prototype was still able to achieve a top speed of 43 mph.

After consulting the manufacturer of the electric motor controller, we determined that the unit we are currently using is not suited for even a stock Xebra vehicle and has thus been recalled. Therefore, replacing this controller with a more powerful one is recommended. We also recommend replacing the faulty recirculation valve in the regenerative system. We are confident that replacing these two components will result in a fully functioning prototype capable of sustaining a 45 mph top speed.

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INTRODUCTION

Although electric vehicles are appealing for environmental reasons, the efficiency of these electric vehicles has been known to drop from 90% to 60% [1] during acceleration. The batteries can operate for a longer duration at lower current draw (Appendix E). For this reason, the Environmental Protection Agency (EPA) has invested in the development of electric-hydraulic hybrid vehicles for city driving applications. Previous ME 450 teams in partnership with the EPA have successfully added a hydraulic launch assist to a small electric vehicle. It is our task to increase the top speed of this vehicle from 35 to 45 mph while maintaining the vehicles current safety and efficiency. This added performance will increase the vehicles versatility and ultimately its market appeal.

Problem Description and Background

Electric vehicles have proven to assist in the protection of the environment due to a decrease in emissions when compared to gasoline vehicles, although affordable models have little market appeal due to their lack of performance. In order to increase the market appeal of affordable models, the EPA has sponsored the design and prototyping of an electric-hydraulic hybrid vehicle which consists of the addition of a hydraulic system to an electric Xebra Vehicle. The vehicle uses a hydraulic system during acceleration to increase the vehicles range and efficiency. Despite these improvements the car still has a maximum speed of only 35 mph, which limits its usefulness significantly. This is why Dr. Andrew Moskalik from the EPA National Vehicle Fuel Emissions Laboratory is sponsoring our project to increase the top speed of the Xebra vehicle from 35 mph to 45 mph while maintaining the safety and performance of the current prototype.

The main motivation for the use of a hydraulic motor during acceleration as opposed to an electric motor is the high increase in efficiency. Since the batteries are more efficient running at a low current draw, they are ideal for constant speed applications. During acceleration very large current draw is required, reducing the vehicle's range (Appendix E). The hydraulic system allows the energy from braking to be stored and reused to power the vehicle during acceleration, which increases the range of the vehicle by using the batteries only at low torques where they are most efficient.

BENCHMARKS AND INFORMATION SOURCES

The information we have gathered includes patents, expert knowledge, and past ME 450 reports. The patents, which are listed in the references section, describe previous hydraulic systems used in regenerative breaking, one of which was used on a diesel-hydraulic truck. We have also obtained a lot of information from our sponsor Dr. Andrew Moskalik and his associate David Swain. We met with them and they provided a detailed project description and much of the background information. The past reports from ME 450 on the Xebra vehicle, which are listed in the reference section, were also utilized to obtain background information on exactly how the systems of this vehicle function.

The past reports are being used as technical benchmarks although some work has been done on the Xebra vehicle since the last report by Ben Hagan. It was necessary to use his expertise on the vehicle for our project in order to find out what changes were most adaptable to the current system. Due to the fact that the last report was written in winter '08 and changes have been made since then, it was necessary to test the vehicle to find exactly how the various systems functioned and other information such as the acceleration and top speed. These tests were conducted and are outlined in the safety report found in Appendix AC.

We have compiled specifications from the manufacturers of the Xebra's batteries, accumulators, hydraulic motor, hydraulic pump, electric motor controller, and also the owner's manual for the Xebra. These can be found in Appendixes E, F, G, H, and I respectively. These information sources aided us in determining our specifications and engineering analysis.

CUSTOMER REQUIREMENTS AND ENGINEERING SPECIFICATIONS

The following section provides a justification for how we arrived at the values for each of our engineering specifications, and how those specifications are related to the customer requirements shown in Table 1 below.

Table 1: Project requirements as defined by the EPA

Customer Requirement	Description
Increase top speed	Increase the top speed from 35 to 45 mph
Transferable to future semesters	Layout to be designed keeping future goals in mind (changes should be reversible).
Prevent overheating	Keep internal temperature at a safe operating level.
Maintain previous levels of performance and efficiency	The average acceleration up to 35mph should remain unchanged. Range to remain unchanged.
Reliability	Components should not fail
Safety	The minimum safety factor should be above 2. The car should not rollover.
Easy to service	All components must be accessible
Aesthetics	No changes to the exterior of the car.
Must Drive Comfortably	Use and speed should be similar to normal car.

Top speed from hydraulics: The top speed of 45 mph was specified by our sponsor. Another one of the customer requirements was that the vehicles range should be increased or at least remain unchanged. We determined that the best way to do this was to reach 45 mph using only hydraulic power. This will ensure that the electrical system is not run during high torque, where it is least efficient.

Distance of C.O.M from rear axle: In order to meet the customer requirement of stability we initially decided to set a specification on the steering turns lock to lock and the turning radius. After extensive research we found that the most crucial elements in creating a stable three wheel vehicle similar to the Xebra are the placement of the center of gravity and the distance between the back wheels [15]. Seeing as it was not feasible for us to increase the length of the rear axle due to the extremely high cost and time requirements we decided to set a specification on the placement of the center of gravity. In order to find this placement we assumed the center gravity would be approximately in the center of the car due to the symmetry of heavy components and at a height of 0.684m. This height used is the center of the accumulators, which is very conservative since most of the heavy components lie below this point. Using a force balance, shown in Figure 1 below, we found the max distance the center of mass can be from the rear axle to prevent the car from rolling on a normal turn, shown in Figure 2 below. The information for the velocity and the radius of the turn were taken from the Federal Highway Administration [16].

Figure 1: Free body diagram during rollover

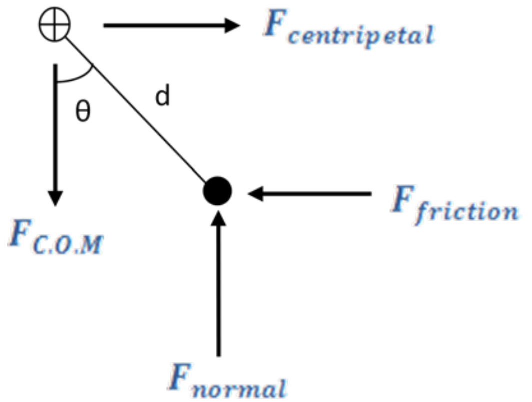
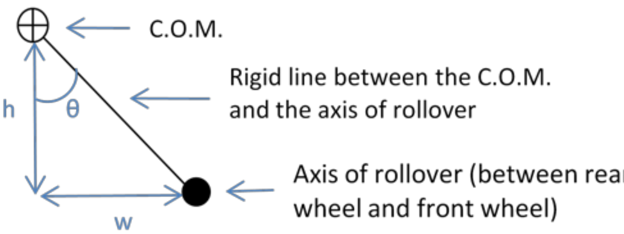


Figure 2: Center of mass position



$$F_{friction} = F_{centripetal} = \frac{mv^2}{r} \quad \text{(Eq.1)}$$

Assuming that the tires do not skid, at the point where the vehicle will begin to roll over, the moment about the axis of rollover due to the centripetal force will be equal to the moment due to the center of gravity as shown in Eq. 2 below. Solving for the angle *θ*, (Eq.3) we can then find the distance from the axis of rollover to the center of mass *(w)* using Eq. 4 below.

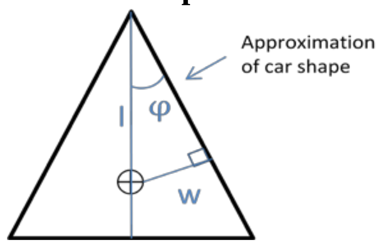
$$\frac{mv^2}{r} cos\theta * \cancel{d} = F_{C.O.M.} cos\left(\frac{\pi}{2} - \theta\right) * \cancel{d} \quad \text{(Eq.2)}$$

$$\theta = \frac{F_{C.O.M.}\pi r}{F_{C.O.M.} r + mv^2} \quad \text{(Eq.3)}$$

$$tan\theta = \frac{w}{h} \quad \text{(Eq.4)}$$

Using the previously calculated distance from the axis of rollover and the dimensions of the car we can then find the distance of the center of mass from the front of the car, shown in Figure 3, p.8. Subtracting this distance from the total car length we get the distance from the rear axle, which is found to be 1.6 m. Since this is at the point when the car will be beginning to roll over we took a safety factor of two giving us a distance of 0.8 m.

Figure 3: Geometric representation of vehicle



$$sin\varphi = \frac{w}{l} \qquad \text{(Eq. 5)}$$

Weight of Vehicle: This specification was determined by the maximum payload that the Xebra vehicle is allowed to carry, which is 500 lbs. This was added to the shipping weight from the manufacturer giving us a maximum total weight of 1,100 kg. This will allow us to install a wide range of accumulators with some additional components while still maintaining a reasonable weight specification.

Power of electric motor: The batteries in the vehicle are most efficient at 1 kW (Appendix E). Since there are five batteries, we determined that to maximize the range of the vehicle, the electric motor should not draw more than 5 kW of power (1 kW per battery).

Hydraulic system pressure: The specification for accumulator pressure was determined from the maximum rated continuous pressure of the hydraulic system. The limiting components are the hydraulic motor and pump, which are rated at 25 MPa, shown in Appendix G.

Initial acceleration: After talking with our sponsor, we agreed that it would be unreasonable to try and keep the same average acceleration of the current vehicle all the way up to 45 mph. In other words, at higher speeds, it is okay if the acceleration decreases. In order to meet the customer requirement that the vehicles current performance is unchanged, we determined that the initial acceleration of the vehicle be unchanged up to 35 mph after which a slight drop would be acceptable.

Hydraulic pump gear ratio: In order to determine the gear ratio for the hydraulic pump we first needed to determine the RPM of the drive axle at 45mph (20.12 m/s). This was done using Eq.6 below.

$$\left(\frac{1}{1.93}\frac{rev}{m}\right) \cdot \left(20.12\ \frac{m}{s}\right) \cdot \left(60\ \frac{s}{min}\right) = 625.8\ RPM \qquad \text{(Eq.6)}$$

Here, 1.93 m is the circumference of the wheel. We then determined the maximum RPM of the hydraulic pump as reported by the manufacturer (Appendix G), which was determined to be 3100 RPM. We then used Eq.7 below to calculate the required gear ratio to reach 45 mph.

$$GR_{pump} = \frac{3100\ RPM}{625.8\ RPM} = 4.95 \qquad \text{(Eq.7)}$$

Hydraulic motor gear ratio: The gear ratio for the hydraulic motor was determined in the same manner as the hydraulic pump. The maximum RPM of the hydraulic motor was determined to be

3300 RPM (Appendix G). We then used Eq.8 below to determine the required gear ratio to reach 45 mph.

$$GR_{motor} = \frac{3300\ RPM}{625.8\ RPM} = 5.29 \qquad \text{(Eq.8)}$$

Layout volume constraint: This value was determined by calculating the maximum possible volume available beneath the truck bed.

Temperature of electric motor: The temperature of the electric motor was specified in the owner's manual for the stock Xebra vehicle (Appendix I). This value should also be measured while it is on the dynamometer to ensure that this value is commensurate with the vehicles current performance.

Range: This value was specified as a result of the vehicles baseline testing in the 2007 Xebra report. [2]

Accumulator volume: The minimum accumulator volume was determined using our mathematical model. We simply changed the gear ratios as specified above in order to reach 45 mph. We then kept all of the vehicles current specifications the same and iteratively increased the accumulator volume until the vehicle could reach the ultimate goal of 45 mph. This value for the accumulator volume was then used as a minimum for our engineering specification.

Table 2: Summary of the engineering specifications derived from the customer requirements

Specification	Value	Units
Top speed from hydraulics	20.1 (45)	m/s (mph)
Distance of C.O.M.* from rear axle	≤ 0.8	m
Weight of vehicle	≤ 1100 (2,425)	kg (lbs)
Power of electric motor	< 5	kW
Hydraulic system pressure	≤ 25	MPa
Initial acceleration	≥ 1.72	m/s^2
Hydraulic pump gear ratio (to wheel)	≤ 4.95	Rev_{pump} / Rev_{wheel}
Hydraulic motor gear ratio (to wheel)	≤ 5.29	Rev_{motor} / Rev_{wheel}
Flow rate of slow fill pump	0.07	L/s
Layout volume constraint	< 1.22	m^3
Temperature of electric motor	< 135	°C
Range	≥ 25.75 (16)	m (mi)
Accumulator volume	< 40	L

*C.O.M. = Center of Mass

Based on the provided customer requirements along with their respective engineering specifications, we created a QFD, which can be found in Appendix D. First, we weighed the customer requirements against each other and rated them on a scale of 1-10. Then we correlated the technical specifications and the customer requirements using values of 1, 3 or 9 in increasing significance, within the correlation matrix. We then filled in the “roof” of the QFD by comparing the engineering specification and ranking the significance of their correlation a 1, 3 or 9. For the customer requirements, the three items that were the most important were increasing top speed to 45 mph, preventing overheating and transferability to future semesters. The three engineering specifications ranked the most important were the gear ratios, the temperature of the electrical systems, and the volume of the accumulators. Special consideration for each of these specifications was taken in our concept selection process.

CONCEPT GENERATION

Before beginning the concept generation we made sure we fully understood the components of the design problem by completing a functional decomposition of the design problem, shown in Appendix J.

Functional Decomposition

Level 1 of the “design problem” functional decomposition list the primary functions of the Xebra. These functions were selected using our customer requirements and engineering specifications. The function of top speed was defined by our customer. The range of 16 miles represents the customer requirement to maintain the current levels of performance. The function of stability fulfills the customer requirement of safety. These were then further divided into sub-functions in level 2 and in level 3 we listed out the inputs and outputs for each sub-function.

Brainstorming

Using the functional decomposition to break down the design problem, we began brain storming ideas that met at least one of the customer requirements. We decided to accept all the concepts no matter how ludicrous they were to make sure we fully explored the design space. The ideas were divided into two categories: “Utilizing the Hydraulic System” and “Not Utilizing the Hydraulic System.” Appendix L lists the different concepts that were generated in our brainstorming sessions. The concepts in the category “Not Utilizing the Hydraulic System” included using human power, using a sail and harnessing wind energy. Concepts that we came up with to improve the hydraulic system included using a larger hydraulic motor, changing the gear ratio and increasing the volume of accumulators. Disregarding the obviously infeasible concepts and those that do not accomplish any of the customer requirements, we were left with six viable concepts that are described and analyzed in detail below.

Increase Maximum Pressure of Accumulators: By increasing the maximum pressure of the accumulators we can store more hydraulic energy, as work done by a gas is given by Eq. 9. If the current volume of hydraulic fluid is maintained, we found out that the fluid would need to be pressurized to 8702 psi from the model described in the Engineering Analysis section on p. 17. This increase in energy would enable the Xebra to reach 45mph although there are many problems associated with this concept.

$$w = \int p \ dv \qquad \text{(Eq. 9)}$$

Changing Gear ratio: Currently the Gear ration of the electric motor is 4.5:1. Since electric motor is rated at 2800 rpm this gives us a maximum velocity of 35 mph. In order to increase the maximum velocity to 45 mph, the gear ratio must be decreased from 4.5:1 to 3.74:1.

Increase volume of accumulators: By increasing the volume of the accumulators we can store more hydraulic energy and allow more work to be done by the gas as shown in Eq.9 on p. 10. Using the mathematical model we found out that if the pressure is kept constant the volume of accumulator required to reach 45 mph would be 40 liters and the volume of the hydraulic fluid will be 20 liters.

Using solar panels: The utilization of solar panels will allow us to increase the range of the vehicle because the panels will charge the batteries and increase the total power available for use. To achieve top speed of 45 mph the electric motor will have to run at a higher speed. Hence, efficiency of the electric motor would drop from 92% to 69%, based on the Simulink model, resulting in an increase in the amount of energy used by the motor. To overcome the increase in energy consumption we would employ the use of solar panels that would provide 1.15 kw (Appendix M).

Torsion Spring: Another concept we considered was the use of a torsion spring to store the deceleration energy. The amount of energy that can be stored by decelerating from 45mph to 0 mph is approximately 250 kJ (Appendix M). To store this vast amount of energy we need a torsion spring with a 90 degree deflection with a spring constant of about 202.64 kJ/rad^2 or we need a torsion spring with a 180 degree deflection with a spring constant of about 50.66 kJ/rad^2 (Appendix M).

Flywheel: The addition of a flywheel would allow us to store vast amounts of energy and utilize the energy with a high efficiency. The flywheel would need to be rotating in the range of 20,000 to over 50,000 rpm [9]. These flywheels are currently used in power plants and Grand Prix cars only, so there are no benchmarks to perform an engineering analysis and to calculate the cost of the system as these are trade secrets.

Functional decompositions were also done for the most feasible concepts that we generated. These functional decompositions can be found in Appendix K. These charts were used as a tool to gain a better understanding of how the subsystems of the concepts fit together.

CONCEPT SELECTION

In order to find which concept best met the customer requirements we utilized a series of Pugh Charts, which rated each generated concept based on the customer requirements, selection criteria and subsequently provided us with a ranking of the concepts. A weight was assigned to each of the selection criteria based on their importance to the sponsor, which was determined from previous discussions with our sponsor. Each concept was rated on how it would affect the individual selection criteria, assigning a one for a positive effect, a zero for no effect, and a negative one for a negative effect.

Initial Pugh Chart

We initially analyzed the six concepts, taken from our brainstorming, that were most feasible and best met the customer requirements. After summing the scores we arrived at the rankings shown in the Pugh Chart, (Table 3) on p. 12.

Table 3: Ranking of initial concepts using Pugh Chart

		Concept A Increase Pressure		Concept B Change Gearing		Concept C Increase Volume		Concept D Solar Panels		Concept E Torsion Spring		Concept F Rotating Disc	
Selection Criteria	**Weight**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**
Increase top speed	**1**	1	1	1	1	1	1	0	0	1	1	1	1
Transferable to future semesters	**0.9**	1	0.9	0	0	-1	-0.9	1	0.9	-1	-0.9	-1	-0.9
Prevent over heating	**1**	0	0	0	0	1	1	-1	-1	1	1	-1	-1
Maintain previous levels of	**0.6**	1	0.6	-1	-0.6	1	0.6	1	0.6	1	0.6	1	0.6
Reliability	**0.5**	-1	-0.5	-1	-0.5	0	0	0	0	-1	-0.5	1	0.5
Easy to service	**0.3**	0	0	0	0	0	0	-1	-0.3	-1	-0.3	-1	-0.3
Aesthetics	**0.3**	1	0.3	1	0.3	-1	-0.3	-1	-0.3	0	0	1	0.3
Must Drive Comfortably	**0.2**	-1	-0.2	0	0	0	0	0	0	-1	-0.2	1	0.2
Safety	**1**	-1	-1	0	0	0	0	0	0	0	0	-1	-1
Score			0.1		-0.8		0.4		-0.1		-0.3		-1.6
Ranking			**2**		**5**		**1**		**3**		**4**		**6**

Concept A: Increasing the pressure of the accumulators was ranked first, although there are many problems associated with this concept. We would need to replace all the components of the hydraulic system due to the high pressure of about 8700 psi, determined from the model described in the Engineering Analysis section on pg. 17. We would need to buy a new hydraulic motor, pump and accumulators which would be rated for such a high pressure. The current accumulators are rated at 4000 psi [1], and most of the hoses and fittings are rated at 5000 psi [1]. Some of these components would need to be custom made since most are not sold currently. Safety will also be compromised while using such high pressures in the vehicle.

Concept B: Changing the gear ratio, will allow the Xebra to reach 45 mph with minimal change. This concept is not feasible as decreasing the gear ratio would not allow the car to start from a standstill at a 6 degree slope (Appendix P). The acceleration of the Xebra will be greatly reduced and it will not be able to maintain previous levels of performance.

Concept C: Increasing the volume of the accumulators, would require increasing the volume of the hydraulic fluid in the accumulators. The main problem with this concept is that it will not adhere to the space constraint under the truck bed. We calculated the required amount of fluid, to achieve a top speed of 45mph, to be 42 liters. The space constraint limits the maximum volume of accumulator to 40 liters. Another problem we face with this concept is the weight of the system, as by increasing the volume of the accumulators we increase the weight of accumulators and the weight of the hydraulic fluid. Hydraulic fluid has a density of 0.85 kg/m^3 and if this concept is introduced the weight of the fluid will be 17kg [10].

Concept D: By our calculations, it will cost us $3300 to buy enough solar panels to make up for the lost electric energy. The solar panels will cover a large surface area of the cab and truck bed and will decrease the transferability of the projects to future semesters. Another problem associated with this design is that the solar panels can only generate electricity when there is bright sunlight. This reduces the practicality of the Xebra as it may not operate at its full potential during the night or cloudy days.

Concept E: Using a torsion spring is shown to be infeasible and does not fit within the constraints of our sponsor. To store this vast amount of energy we need a torsion spring with a 90 degree deflection with a spring constant of about 202.64 kJ/rad^2 or we need a torsion spring with a 180 degree deflection with a spring constant of about 50.66 kJ/rad^2 (Appendix M). A spring with this large of a coefficient would need to be custom made.

Concept F: Using a flywheel, does not fit into the constraints set by our sponsor, who wanted us to use only the hydraulic system. There are many safety concerns with using the flywheel to store energy, as there is a heavy disc that is rotating at high speeds (20,000 to over 50,000 rpm) [9]. This develops a lot of stress in the joints which increases the chance of failure substantially.

Concept A, B and C were then the only possible concepts once concept D, E and F were discarded, although it was also found that none of them would satisfy the customer requirements alone. Solely increasing the accumulator pressure makes the car unsafe and is too costly; solely increasing the accumulator volume will prevent the accumulators from fitting in the volume constraint; changing the gear ratio will decrease the low end torque substantially so the car will not even be able to accelerate from a stop on a six degree slope (Appendix P). Therefore we decided to create new concepts that combined the concepts A, B, C from our initial Pugh Chart.

Combined Pugh Chart

In analyzing our concepts from the initial Pugh Chart we found that it is necessary to modify the gear ratios of the motors no matter what our concept, as shown in the Engineering Analysis section. The new concepts that were generated consisted of combinations of the three feasible concepts from the initial Pugh Chart and increasing the motor capacity, taking into account that all of them will change the gear ratios. The reason for adding the increase in motor capacity is that it will increase the acceleration to meet that specification. All of the concepts that we generated were then analyzed in a combined Pugh chart, shown in Table 4 below. This chart was utilized in the same manner as the initial Pugh Chart to give us the optimal concept based on our revised customer requirements.

Table 4: Ranking of revised concepts* using Pugh Chart

		Concept A Increase Pressure		Concept B Increase Pressure and Motor Capacity		Concept C Increase Volume		Concept D Increase Volume and Motor Capacity		Concept E Increase Pressure and Volume		Concept F Increase Pressure, Volume, and Motor Capacity		Concept G Just Gear Ratio	
Selection Criteria	**Weight**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**
Increase top speed	**1**	1	1	1	1	1	1	1	1	1	1	1	1	1	1
Prevent overheating	**1**	0	0	0	0	1	1	0	0	1	1	1	1	-1	-1
Safety	**1**	-1	-1	-1	-1	0	0	0	0	-1	-1	-1	-1	0	0
Transferable to future semesters	**0.9**	1	0.9	-1	-0.9	-1	-0.9	-1	-0.9	-1	-0.9	-1	-0.9	0	0
Maintain range	**0.9**	-1	-0.9	1	0.9	1	0.9	1	0.9	1	0.9	1	0.9	0	0
Maintain current acceleration	**0.9**	1	0.9	1	0.9	0	0	1	0.9	1	0.9	1	0.9	-1	-0.9
Increase energy recovery	**0.9**	-1	-0.9	1	0.9	0	0	1	0.9	-1	-0.9	1	0.9	1	0.9
Reliability	**0.5**	-1	-0.5	-1	-0.5	0	0	0	0	-1	-0.5	-1	-0.5	0	0
Aesthetics	**0.3**	1	0.3	1	0.3	-1	-0.3	-1	-0.3	-1	-0.3	-1	-0.3	1	0.3
Cost	**0.3**	-1	-0.3	-1	-0.3	-1	-0.3	-1	-0.3	-1	-0.3	-1	-0.3	1	0.3
Easy to service	**0.2**	0	0	-1	-0.2	0	0	0	0	-1	-0.2	-1	-0.2	0	0
Must drive comfortably	**0.2**	-1	-0.2	-1	-0.2	0	0	-1	-0.2	0	0	-1	-0.2	-1	-0.2
Score			-1.5		0.1		0.4		1.2		-1.3		0.5		-0.4
Ranking			**7**		**4**		**2**		**1**		**6**		**3**		**5**

**All concepts change the gear ratio*

Accumulator Selection

Utilizing a Pugh chart and a positives/negatives chart, we were able to select the accumulators that best met our specifications. The concepts that we came up with were all very similar except for the configuration of the high side accumulators and the replacement of the hydraulic motor. In order to find the optimal configuration we utilized a Pugh chart, as shown in Table 5 on p. 14. The different configurations were ranked on a set of criteria that was generated from our specifications specifically for rating the different accumulator configurations. Some criteria are left out because all of the configurations meet them equally.

Table 5: Pugh Chart of different configurations of high pressure accumulators

Selection Criteria	Weight	Concept A: Large Bladder accumulator — Rating	Concept A — Weighted	Concept B: ACP Accumulators(4 x 8 liters) — Rating	Concept B — Weighted	Concept C: 2 x 4 Gallon Accumulator — Rating	Concept C — Weighted	Concept D: ACP Accumulators(2x 8; 2x6) — Rating	Concept D — Weighted
Assembly Time (high means less time)	**1**	6	6	3	3	6	6	6	6
Risk (higher better)	**1**	9	9	9	9	6	6	9	9
Ease of assembly (higher better)	**0.8**	3	2.4	3	2.4	6	4.8	6	4.8
Reversibility (higher better)	**0.6**	3	1.8	9	5.4	3	1.8	9	5.4
Weight/stability (higher better)	**0.5**	6	3	6	3	1	0.5	9	4.5
Size constraints (higher better)	**0.9**	1	0.9	6	5.4	6	5.4	9	8.1
Performance (higher better)	**0.9**	9	8.1	9	8.1	9	8.1	1	0.9
Score			25.2		33.3		26.6		32.7
Ranking			**4**		**1**		**3**		**2**

Concept B, the configuration that adds on two more 8 liter ACP accumulators, ranked first in the Pugh chart, meaning it will be the optimal configuration. Therefore, Concept B was chosen.

Hydraulic Motor Selection

We then needed to decide whether or not to replace the 23cc hydraulic motor with a 33cc motor. This was not added into the Pugh chart because we decided a simple positives/negatives chart is more effective in this situation. The positives were listed on one side and negatives on the other as shown in Table 6, below. After the decision was deliberated by the group, taking into account the effect each positive or negative will have on the specifications and customer requirements, we decided the negatives outweighed the positives.

Table 6: Positives vs. negatives of replacing the hydraulic motor with a larger one

Positives	**Negatives**
Top Speed Increase 15.7 → 16.4m/s	Assembly Time
Initial Acceleration 1.08 → 1.49 m/s^2	Ease of Assembly
Reversibility	Cost

Although, the benefit from the increasing the motor size is minimal for increasing the top speed, since it will only increase 5%, there is a significant benefit in the initial acceleration of 38%. This is a substantial benefit but it is not worth the added risk of failure, since the hydraulic motor assembly is currently functioning. Initial acceleration is not as important as having a working vehicle at the end of the semester and replacing the motor will not only add risk, but will take valuable time away from correcting the current problems with the vehicle and making our other modifications. Although we are not replacing the motor, it will be suggested that future semesters look into this modification more closely.

Another consideration that was discovered during the disassembly, was that the one way bearing attached to the hydraulic motor could not with stand the previous torque. An increase in the motor size will increase the torque on the one way bearing further. After looking for other one way bearing, we could not find a one way that can take the torque applied by a larger motor.

Accumulator Mounting Selection

Due to the current configuration of the Xebra's hydraulic system and to packaging constraints, we decided to mount the accumulators on the outside of the frame. If the accumulators were mounted inside of the frame, the slow fill pump and the low side accumulator would have needed to be moved and the whole hydraulic plumbing system would have needed to be rerouted, adding addition cost and time. In order to find the optimal design to hold the accumulators we came up with three designs. The first was to weld steel beams onto the frame and bolt the accumulator mounts into these beams. The second is to use bolts to secure angled steel to the side of the frame and bolt the accumulator mounts to the angled steel. The third is to design a mechanism that would be welded to steel beams that hold the accumulator mounts and would clamp onto the frame.

These three concepts were analyzed in a Pugh chart, show in Table 7 below, with selection criteria that directly correlated to our specifications. The welding method was ranked first mostly due to its high strength, making it the safest method. See Appendix R for analysis.

Table 7: Pugh chart for accumulator mounting methods

		Concept A Welding		Concept B Fasteners		Concept C Clamping	
Selection Criteria	**Weight**	**Rating**	**Weighted**	**Rating**	**Weighted**	**Rating**	**Weighted**
Strength	**1**	1	1	0	0	-1	-1
Reversibility	**0.8**	-1	-0.8	0	0	1	0.8
Ease of Manufacturing	**1**	1	1	1	1	-1	-1
Cost	**0.3**	1	0.3	0	0	-1	-0.3
Effect on surrounding Parts	**0.6**	0	0	-1	-0.6	1	0.6
Score			1.5		0.4		-0.9
Ranking			**1**		**2**		**3**

ALPHA DESIGN

Our Alpha design consisted of adding a 5 gal (19 L) bladder hydraulic accumulator manufactured by Parker Hannifin Corp. (Appendix F) to the existing vehicle, increasing the hydraulic system pressure, and changing the gear ratio of the hydraulic pump, hydraulic motor and electric motor. The new accumulator position can be seen in Figure 4, p. 16, and the gears/sprockets that will be changed are shown in Figure 5, p. 16. The hydraulic pump and motor use a chain and sprocket drive train, so changing the gear ratio would entail changing the sizes of the sprockets. The electric motor gear box would need to be disassembled to change the gears to meet our specified gear ratio. These alterations that made up our alpha design were determined using our mathematical model, explained in the Engineering Analysis section on pg. 17, along with meeting other criteria, explained in the Concept Selection section on pg. 11. The important performance values of our alpha design and the current Xebra vehicle along with the relevant mathematical model parameters are shown in Table 8, p.16 and Figure 6, p. 16. A functional decomposition diagram showing the subsystems of the alpha design interact is shown in Appendix F.

Figure 4: New accumulator for Alpha design

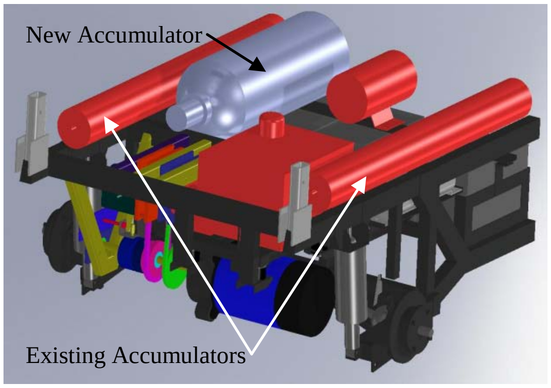


Figure 5: Gear Ratios for Alpha design

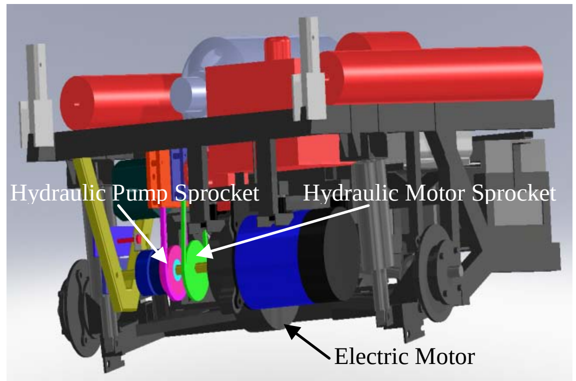


Figure 6: The model predicts that our alpha design will reach 45 MPH (20 m/s) in 21 sec. and will need a total volume of 31 L with 85% motor efficiency

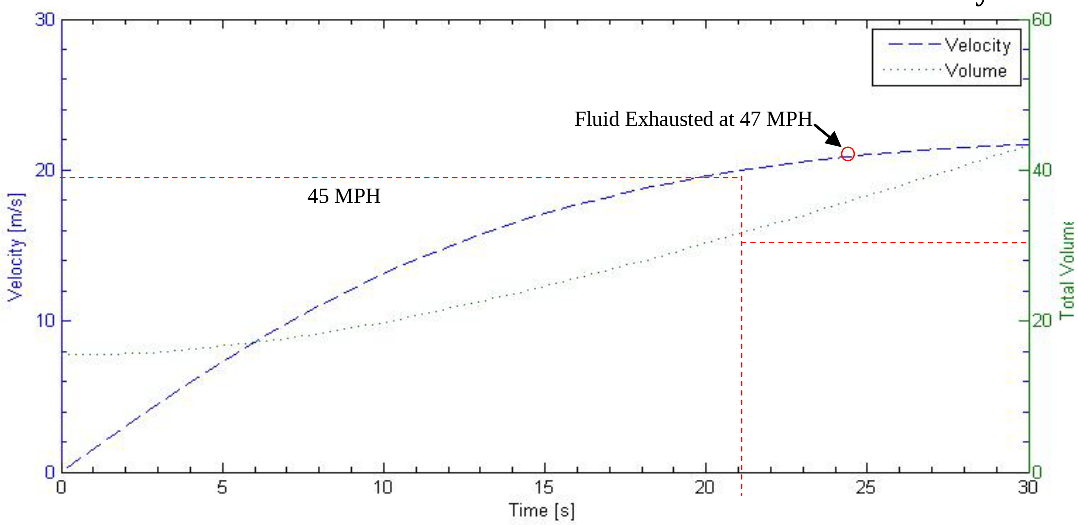


Table 8: Based on mathematical model, the alpha design improves the current vehicle and meets all engineering specifications except for the initial acceleration

Performance Values/Parameters	Current	Alpha Design	Specification
Top Speed From Hydraulics [m/s]	7.6*	20.1	20.1
Initial Acceleration [m/s^2]	1.72	1.53	≥ 1.72
Hydraulic Pump Gear Ratio [rev_{pump}/rev_{wheel}]	5	4.73	≤4.95
Electric Motor Gear Ratio [rev_{motor}/rev_{wheel}]	4.5	3.73	≤ 4.5
Motor Displacement, D_{motor} [cm^3/rev]	23**	23**	N/A
Hydraulic Motor Gear Ratio, N [rev_{motor}/rev_{wheel}]	7	4.29	≤ 5.29
Maximum/Initial Pressure, P_{max} [MPa]	24**	34**	≤ 34
Initial Volume of Nitrogen, V_i [L]	12.6	15.5	N/A
Volume of Hydraulic Fluid Displaced, ΔV [L]	3.4	15.5	N/A
Total Volume needed to reach 45 MPH, $V_i + \Delta V$,	N/A	31	N/A
Total Accumulator Volume in Designs, V_{total} [L]	16**	35**	< 40

*Determined during testing

**Determined using manufacture's specifications

Initial Acceleration Specification

Because we decreased the gear ratio for the hydraulic motor so that the motor doesn't exceed its maximum rated RPM value of 3300 RPM, shown in Appendix G, the motor torque also decreases (Eq. 13, p.19). Because the motor torque decreases, the initial acceleration of the alpha design is decreased. The only way to increase the motor torque and therefore increase the initial acceleration based on Eq. 13, would be to increase the maximum/initial pressure in the accumulators and/or increase the displacement of the hydraulic motor. These other two options were deemed not feasible in the Concept Selection section on p. 11.

Top Speed/Fluid Control

Since our alpha design includes 35 L total accumulator volume but only 31 L is needed to reach 45 MPH, the alpha design could reach 47 MPH (see Fig. 6, p. 16). Because of this, the gear ratios of the hydraulic pump and motor have been lowered to ensure that they are not revved more than what they are rated. Also, a controller should be designed to ensure that the vehicle does not exceed 47 MPH when going downhill so that the motor and pump are not damaged. The controller should also be used to ensure that some pressurized fluid remains in the accumulators during operation to engage the hydraulic clutch can engage the regenerative braking system.

ENGINEERING ANALYSIS

The progression of our design from the alpha design was aided greatly by the analysis of our specifications and the components that would allow us to meet our specifications. Although many stand alone calculations have been done, our mathematical model was the ultimate tool used in optimizing our design. Other tools that were very useful were FEA and our CAD model. Using those tools we arrived at to the optimal design that best met the engineering specifications.

Mathematical Model

In order to simulate the motion of the vehicle under a hydraulic launch and determine the optimal vehicle parameters, the equations of motion (Eqns. 10 & 11, p. 18) were derived based on the free body diagrams shown in Figures 7 & 8, p. 18. The resulting governing differential equation (Eq. 18, p. 19) was solved using Matlab. The Matlab code is in Appendix O. All variables are defined in Table 10, p. 18. The optimization of the alpha design was accomplished by minimizing the total volume of accumulators needed to reach 45 MPH. We wanted to minimize the total volume because we wanted to fit an accumulator with the size constraint as well as modify the vehicles current layout the least. The optimization was done by analyzing the model and determining the initial pre-charge volume of nitrogen gas in the accumulators and the hydraulic motor gear ratio that corresponded to minimum total volume. A spreadsheet of these values can be found in Appendix Q, where an initial pre-charge volume of nitrogen of approximately 15.5 and a gear ratio of 4.3 corresponded to a minimized the total volume of the accumulators of 31 L.

Table 9: Definition of mathematical model parameters and variables

Parameters/Variables	Value
Mass of Vehicle, m $[kg]$	1200
Radius of Drive Tire, R_{Tire} $[m]$	0.254
Optimum Hydraulic Motor Gear Ratio, N $[rev_{motor}/rev_{wheel}]$	4.3
Motor Efficiency, η	0.85 [1]
Torque Constant, k $[1/rev]$	$1.7 \cdot 10^{-3}$ (Appendix G)
Hydraulic Motor Displacement, D_{Motor} $[cm^3/rev]$	23
Initial Pressure, P_i $[MPa]$	34
Optimum Initial Volume of Nitrogen, V_i [L]	15.5 (Appendix Q)
Nitrogen Polytropic Exponent, α	1.3 [8]
Volume of Hydraulic Fluid Through Motor, ΔV	15.5
Drag Coefficient, C_1 $[N \cdot s^2/m^2]$	0.4621
Drag Coefficient, C_2 $[N \cdot s/m]$	2.994
Rolling Resistance Force, F_{RR} [N]	135.1

Table 10: Definition of mathematical variables

Variable Description	Variable Symbol
Angular Acceleration of the System, [rad/s^2]	$\ddot{\theta}$
Vehicle Distance Traveled, [m]	x
Vehicle Velocity, [m/s]	$\dot{x}$
Vehicle Acceleration, [m/s^2]	$\ddot{x}$
Force applied by road on the vehicle, [N]	F_{Road}
Force of air drag on vehicle, [N]	F_{Drag}
Force of rolling resistance on the vehicle, [N]	F_{RR}
Moment of Inertial of the system, [kg·m^2]	J
Road incline/decline, [rad]	β

Figure 7: Free Body Diagram of Vehicle

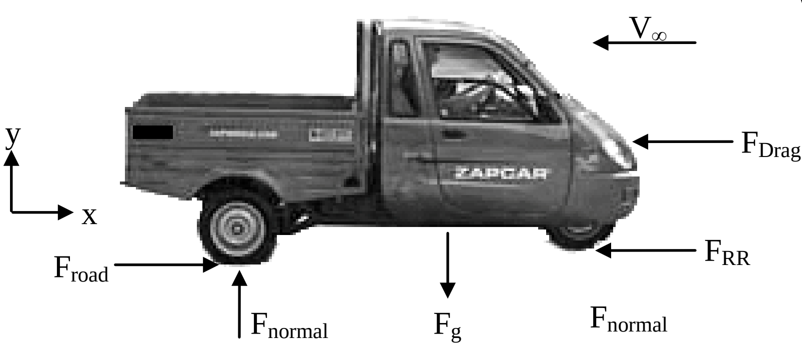
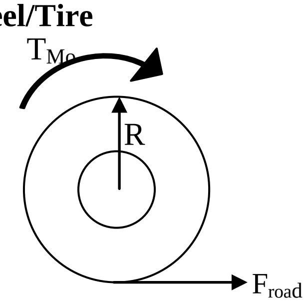


Figure 8: Free Body Diagram of Rear Wheel/Tire



$$\Sigma F = m \cdot \ddot{\boldsymbol{x}} = F_{Road} - F_{Drag} - F_{RR} - F_g \qquad \text{(Eq. 10)}$$

$$\Sigma M = J \cdot \ddot{\theta} = T_{Motor} - \frac{R_{Tire} \cdot F_{Road}}{N} \qquad \text{(Eq. 11)}$$

Assuming that the moment of inertia of the system is negligible compared to the mass of the car:

$$F_{Road} = \frac{N \cdot T_{Motor}}{R_{Tire}} \qquad \text{(Eq. 12)}$$

Using Hannifin Parker Corp. Equation, shown in Appendix E for the torque of the motor:

$$T_{Motor} = \eta \cdot k \cdot D_{Motor} \cdot P \quad \text{(Eq. 13)}$$

Assuming adiabatic expansion of nitrogen in accumulators:

$$P = \frac{P_i \cdot V_i^{\alpha}}{(V_i + \Delta V)^{\alpha}} \quad \text{(Eq. 14)}$$

$$\Delta V = \frac{N \cdot D_{Motor}}{2 \cdot \pi \cdot R_{Tire}} \cdot \boldsymbol{x} \quad \text{(Eq. 15)}$$

$$F_{Drag} = C_1 \cdot \dot{\boldsymbol{x}}^2 + C_2 \cdot \dot{\boldsymbol{x}} \quad \text{(Eq. 16)}$$

Assuming that the vehicle is on level ground:

$$F_g = m \cdot g \cdot \cos\beta = 0 \quad \text{(Eq. 17)}$$

Using Eqns. 10-17, a differential equation results:

$$m \cdot \ddot{\boldsymbol{x}} = \left[\frac{N \cdot \eta \cdot k \cdot D_{Motor} \cdot \left(\frac{P_i \cdot V_i^{\alpha}}{\left(V_i + \frac{N \cdot D_{Motor}}{2 \cdot \pi \cdot R_{Tire}} \cdot \boldsymbol{x} \right)^{\alpha}} \right)}{R_{Tire}} \right] - [C_1 \cdot \dot{\boldsymbol{x}}^2 + C_2 \cdot \dot{\boldsymbol{x}}] - [C_3] \quad \text{(Eq. 18)}$$

Accuracy of Mathematical Model

The current hydraulic system on the vehicle at the start of the semester was predicted to reach 16.7 km/hr, during a hydraulic launch by the mathematical model. During experimental testing a speed of vehicle a speed of 17 km/hr was read off the vehicle's speedometer. This result justified the assumptions made in the model. It also justified using this model to determine parameters for our alpha design. Further testing on a dynamometer was conducted to further verify the precision and accuracy of the model. The results of validation testing are discussed further on p. 33.

Accumulator Mounting Beams

Each of the mounting beams that were welded to the frame can support a downward force of 11,000 lbs before failure by yield. This was determined through the static analysis shown in Appendix R. Since each accumulator is supported by two beams, the force on each beam is the weight of an accumulator divided by two, which is approximately 37.5 lbs (see Appendix F for specifications on the accumulators). This resulted in a safety factor of 293. This safety factor is sufficient to estimate that the beams will not fail under vehicle operation in either static or dynamic loading.

Gear Ratio

In order to achieve the top speed of 45MPH, the new tires with a 0.307m radius would be required to rotate at an angular velocity of 625.1 RPM. Keeping this angular speed in mind we found the maximum gear ratios that can be obtained for the existing hydraulic motor and pump.

The gear ratios we calculated would allow us to reach 45MPH without over-revving the hydraulic motor or hydraulic pump at top speed. Table 11 below lists the displacement of the motor and pump, the maximum rated RPM and the gear ratio required to reach 45MPH.

Table 11: Required gear ratios for pumps and motors

Component	Displacement	Max Rated RPM	Gear Ratio	Max Continuous Pressure	Max Torque
Motor	23cc	3300RPM	5.28	25 MPa	483.19Nm
Pump	33cc	3000RPM	4.8	25 MPa	630.25Nm

FEA

To determine if the components of our design would withstand the forces and torques applied to them, a finite element analysis was conducted using SolidWorks Simulation software. The results, which can be seen in Appendix S, determined that the minimum Von Mises factor of safety on three sprockets and the one-way bearing shaft collar is 5. The components and the factors of safety can be seen below in Table 12.

Table 12: Safety factors for components analyzed

Component Description	Material	Minimum Von Mises Factor of Safety
Sprocket on Hydraulic Motor Shaft	AISI 1040 Steel	5
Sprocket on Hydraulic Pump Shaft	AISI 1040 Steel	7.25
Sprocket on Drive Shaft	AISI 1040 Steel	19.65
One way Bearing Shaft Collar	AISI 1020 Steel	6.72

The displacements, υ, of the hydraulic pump and motor can be found in Table 11 above. In order to conservatively determine if the components would fail, the maximum torque applied by the hydraulic pump and motor was calculated assuming an accumulator pressure, P, of 3800 psi. The maximum accumulator pressure that our design specifies is 3600 psi. Also, a 20° pressure angle, ϕ, on the chain/sprocket interface was assumed. The pressure angle accounts for the increase in the magnitude of the force acting on each tooth face. The radius of the sprockets, r, were determined using the CAD model downloaded from the McMaster-Carr website.

To determine the number of teeth on each sprocket that will experience a force, N_{Teeth}, the sprocket positions were laid out in SolidWorks and a representation of a chain was drawn. From these figures, the number of teeth experiencing a force on each sprocket was determined. The CAD figures can be seen in Appendix U. Based on this, the force on each sprocket tooth, F_{Tooth}, was calculated using Eq. 19 below. This force was specified in SolidWorks, along with the torque.

$$F_{Tooth} = \frac{P \cdot \upsilon}{2 \cdot \pi \cdot r \cdot N_{Teeth} \cdot \cos\phi} \qquad \text{(Eq. 19)}$$

The center of the sprockets and shaft collar were constrained using a bearing fixture in SolidWorks. This fixture ensures that the sprockets and shaft collar can rotate with 1 degree of freedom. The edge of the keyways on the sprockets and shaft collar in contact with the key were also fixed with 0 degrees of freedom. This most closely models the point when the sprockets and collar have the maximum stress applied to them.

Top Speed & Initial Acceleration

The mathematical model section is described on p. 17 and was used to calculate the top speed and initial acceleration of the Xebra using only the hydraulics. The variables that needed to be inputted to the model were the initial pressure, which is the max continuous pressure of the motor or pump, the displacement of the motor or pump, and the gear ratio required for that motor or pump. The model gave us the maximum acceleration for the hydraulic motor and the maximum deceleration for the hydraulic pump.

The model predicted that the initial instantaneous acceleration of the Xebra car using the 23cc motor would be 1.12m/s^2. It also predicted that the Xebra would achieve a top speed of 16.29m/s (36.4MPH). See Table 13 below for optimal accumulator parameters. The model predicts that the maximum deceleration that can be achieved is -1.485m/s^2.

Table 13: Performance values with total accumulator volume of 32 L & max pressure of 24.8 MPa

Volume of Nitrogen [L]	15	**15.7**	16	17	18	19	20	21	22
Pre-Charge Pressure [MPa]	9.29	**9.84**	10.37	10.94	12.00	12.70	13.81	14.53	15.68
Top Speed [m/s]	16.27	**16.29**	16.26	16.22	15.97	15.89	15.45	15.25	14.64
Time to Top Speed [s]	21.89	**21.26**	20.64	20.01	18.76	18.14	16.89	16.26	15.01
Average Acceleration [m/s^2]	0.743	**0.766**	0.787	0.811	0.851	0.876	0.915	0.938	0.975

Torque

The torque of a hydraulic motor/pump is given by Eq. 20 below.

$$Torque = \frac{(Gear\ \ Ratio\,)\times(Displacement\ \ \ in\ m^3)\times(MaxPressure\ \ \ in\ Pa)}{2\times\pi} \qquad \text{(Eq. 20)}$$

To calculate the maximum torque we used the rated continuous pressure of the motor or pump as the max pressure in Pascals. This was obtained from the motor specifications sheet found in Appendix G. Table 11, p. 20 shows the max torque for a motor or pump of given displacement.

Weight of Components

The maximum weight of the hydraulic components was limited to 500 lbs. This weight is the maximum cargo weight. According to the specifications, the weight of all the additions made by past teams and all our additions should be limited to 500 lbs. The weights of the individual aftermarket components were added to determine the total weight of the prototype, which was found to be 465.38 lbs as show in the Table 14 below.

Table 14: Itemized weights of aftermarket components

Component	Weight	Quantity	Total Weight
Hydraulic Motor	19.2 lbs	1	19.2 lbs
Hydraulic Pump	19.8 lbs	1	19.8 lbs
High Pressure Accumulator	75 lbs	4	300 lbs
Low Pressure Accumulator	13 lbs	1	13 lbs
Slow fill Pump	33 lbs	1	33 lbs
Hydraulic Fluid	1.96 lbs/L	15.5 L	30.38 lbs
Steel Mounting Beams	50 lbs	13 ft.	50 lbs
Total			465.38

Center of Mass

In order to maintain stability the center of mass must be maintained within 0.8 m from the rear axle, as described in the Distance of C.O.M. from rear axle section on p.6. The addition of the two new accumulators posed the highest concern out of all of our design aspects, so we made sure to get accumulators that do not sit too high or too far from the axle. In order to make sure our design met this specification we plugged in the mass and the material of the components into our CAD model and then had SolidWorks calculate the center of mass. The distance from the rear axle was found to be 0.677 m at height of 0.552 m. This height is was than the estimated height of 0.684 m, giving us a safe estimate of the placement of the center of mass.

The CAD model did not include every part of the vehicle, namely the cab and the driver. This was added as a point mass of 300lbs at the approximate place of the center of mass of the cab and driver combination. Although this is a rough approximation, it was the most practical option that we had available at the time and it was done with conservative estimations.

Electric Motor Temperature

It was determined from the owner's manual of the Xebra (Appendix I) that the electric motor temperature should not exceed 135^{o} C. A cooling fan was attached to the electric motor in order to facilitate its cooling. The new cooling fan has a flow rate of 105cfm. Since the original cooling fan was removed and no information about the original cooling fan could be found, the flow rate of the original fan is unknown. As the cooling fan will be actuated when the temperature of the electric motor exceeds 60^{o} C, we calculated the heat transfer rate due to convection. This calculation can be found in Appendix T. We determined that the heat transfer rate at 60^{o} C is approximately 6kW, which is 8 times greater than the heat generated by the electric motor which was approximated as 750W.

Material Manufacturing Process Selection

Utilizing the Cambridge Engineering Selector (CES) software, we were able to determine the ideal materials and optimum manufacturing processes for mass manufacturing the accumulator mounting beams and the drive train sprockets used in our design. We determined that the mounting beams and the sprockets should be made from AISI 1020 and AISI 1040 steel respectively. Due to the relatively small volume of mounting beams being produce, we determined that the most cost effective production method is ceramic mold casting. For similar reasons, powder metal forging is recommended for producing the sprockets used in our design. A more detailed description of these processes and our justifications for choosing them can be found in Appendix C.

Design for Environmental Sustainability

It is important to take into account the overall effect of our design on the environment, including the environmental impact of the materials in our design. Using SimaPro, we compared the environmental impact of two materials that could be used for our design changes, steel and aluminum. The results show that aluminum has a much greater negative environmental impact and uses more water, air, and raw material in production. These results are explained further in Appendix C. We therefore recommend that the use of steel be implemented in place of aluminum where ever possible in the future in order to make the Xebra more environmentally friendly.

Design for Safety

Throughout the design and testing of our project, maintaining safety was our top priority. First, we had to ensure that any component that we designed would result in an overall safe project. Safety factors of at least two were applied to every component that we designed for our project. Second, to ensure safety while manufacturing components for our project, we took the time to plan all of our machining procedures, including feeds and speeds, before we entered the shop. Finally, to ensure safety while testing the Xebra, we carefully outlined all of our validation methods and procedures before testing, allowing us to address potential dangers before they were encountered in the lab. The safety report found in Appendix AC gives detailed descriptions of the safety precautions taken in this project.

PROTOTYPE DESCRIPTION

Although there is a final design and a prototype, our prototype is very similar to our final design and will prove all of the important elements of our design. The main difference between the final design and the prototype is that the prototype has two hydraulic motors, one of which acts as a pump. The reason for this is because our sponsor preferred that we do not replace these components with one motor that would act as a motor and a pump.

The prototype is the Xebra electric hydraulic hybrid vehicle with the specifications shown in Table 15 below. We were not able to fully meet all of the specifications we initially set, namely the top speed from hydraulics. The top speed will reach 45 mph with the assistance of the electric motor, which will ultimately satisfy the customer requirement. The acceleration specification is met although this is with the use of the electric and hydraulic motor. The limiting factors are the energy stored in the accumulators, which relies on the maximum pressure and the volume, and the size of the hydraulic motor, which were all constrained due to the design of the current hydraulic system.

Table 15: Performance specification of current design

Parameter	Alpha Design	Final Design
Top Speed (mph)	45*	45 (35.6*)
Motor Displacement, [cm^3/rev]	23	23
Electric Motor Gear Ratio	3.73	4.5
Hydraulic Pump Gear Ratio (to wheels)	4.1	4.73
Hydraulic Motor Gear Ratio (to wheels)	4.3	5.29
Maximum Pressure, [MPa]	34	25.5
Initial Volume of Nitrogen, [L]	15.5	16
Volume of Hydraulic Fluid Displaced, [L]	15.5	16
Total Accumulator Volume, [L]	31	32
Initial Acceleration [m/s^2]	1.53*	1.97 (1.02*)

*Using Only Hydraulics

Additional Accumulators

The CAD model shown in Appendix U is a representation of the prototype design. The largest change is the additional of two 8 L, 4000 psi accumulators and their mounting assemblies. The mounting beams are made of AISI 1020 steel and are welded onto the frame with an additional beam on the outside of the mounting beams to protect the accumulators in case of a side impact. The dimensions of the accumulators are shown in Appendix F. In order to accommodate these accumulators it was necessary to cut out the wheel wells of the two rear wheels, since they would interfere with the accumulator placing.

Increased Diameter of Tires

As we needed to change the gear ratio of the electric motor to achieve the top speed, we decided that increasing the diameter of the tires would be a cheaper and a more efficient design. So, we decided to increase the diameter of the tires from 20 in to 24.2 in. This increase in the tire size allows the Xebra to achieve the required top speed, while not making any changes to the gear box.

Hydraulic Motor and Pump Gear Ratio

In order for the hydraulic pump and motor to operate at the top speed, the gear ratios were modified. As show in the engineering analysis section, the gear ratios for the hydraulic pump and motor were determined to be 4.8 and 5.29 to the wheels respectively, which ensures the RPM and the torque will not exceed the maximum rated values. The sprockets were connected using the same chains that were on the vehicle when we began the project. The motor and pump use ANSI 40 and ANSI 60 chains respectively.

Electric Motor Cooling System

To prevent overheating of the electric motor our prototype includes the same cooling fan as the final design. This will maintain the motor temperature below 60°C which is below the specification of 135°C. The fan was mounted to the frame as opposed to directly on the motor, due to packaging constraints.

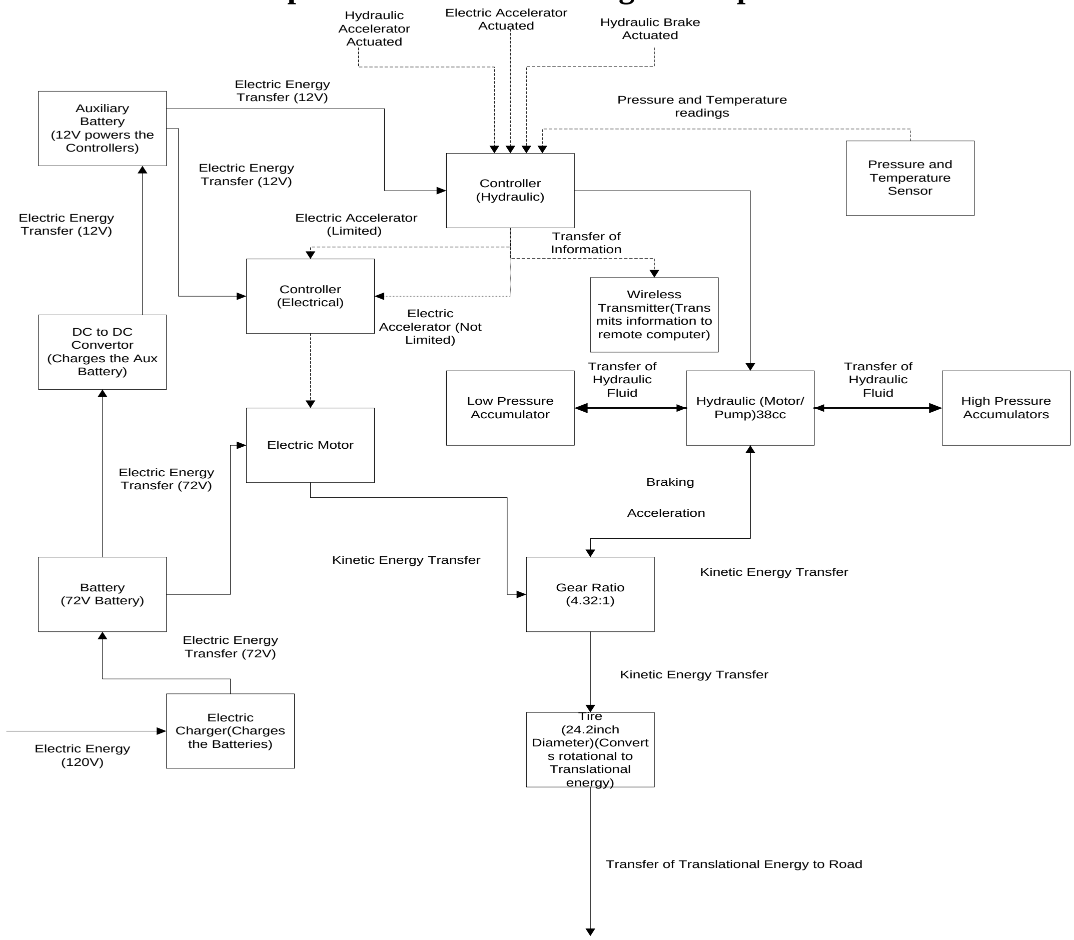
Motor Bearing Replacement

The previously installed one-way bearing that keeps the motor from turning while not engaged was not rated for the applied torque and was therefore replaced with a new bearing in our prototype. This bearing is the same as the one in the final design and was also moved to the electric motor drive shaft. This stopped the chain from rotating while the while the motor is disengaged, reducing friction and wear. In order to accommodate the new bearing, a sleeve was fitted around the electric motor shaft as shown in Appendix U. This sleeve essentially increased the shaft diameter, to ensure the proper fit for the one-way bearing.

FINAL DESIGN DESCRIPTION

This section describes working of the final design and includes our recommendations for work that could be done by future semesters. The prototype design that we have come up with allows the vehicle to reach 45 mph using both the hydraulic and electric motors, however we would have liked to reach the desired top speed by only utilizing the hydraulic motor. Currently the Xebra has a 23cc hydraulic motor and a 33cc hydraulic pump (see Appendix G for manufacturer's specifications). If given the opportunity to completely dissemble and replace parts of the Xebra, we would use only one hydraulic motor, which would also be used as a pump. The final design would also incorporate an additional controller that would change the configuration of the hydraulic system to run as a pump while breaking and as a motor while accelerating. This controller would also limit the power output of the electric motor to 5 kW while the hydraulic system is being used to accelerate the Xebra. The new hydraulic controller will also measure and record the temperature and pressure readings across the hydraulic and the electrical system. The controller would also transmit the recorded values via a wireless connecter to a computer for analysis. Since the final design incorporates a change of the hydraulic motor and pump we would also need to change the gear ratios from the prototype design. The final design would not change the dimensions of the accumulators and the tires since these are sufficient as of now. A functional decomposition of our final design can be seen in Figure 9 below.

Figure 9: Functional decomposition of our final design concept



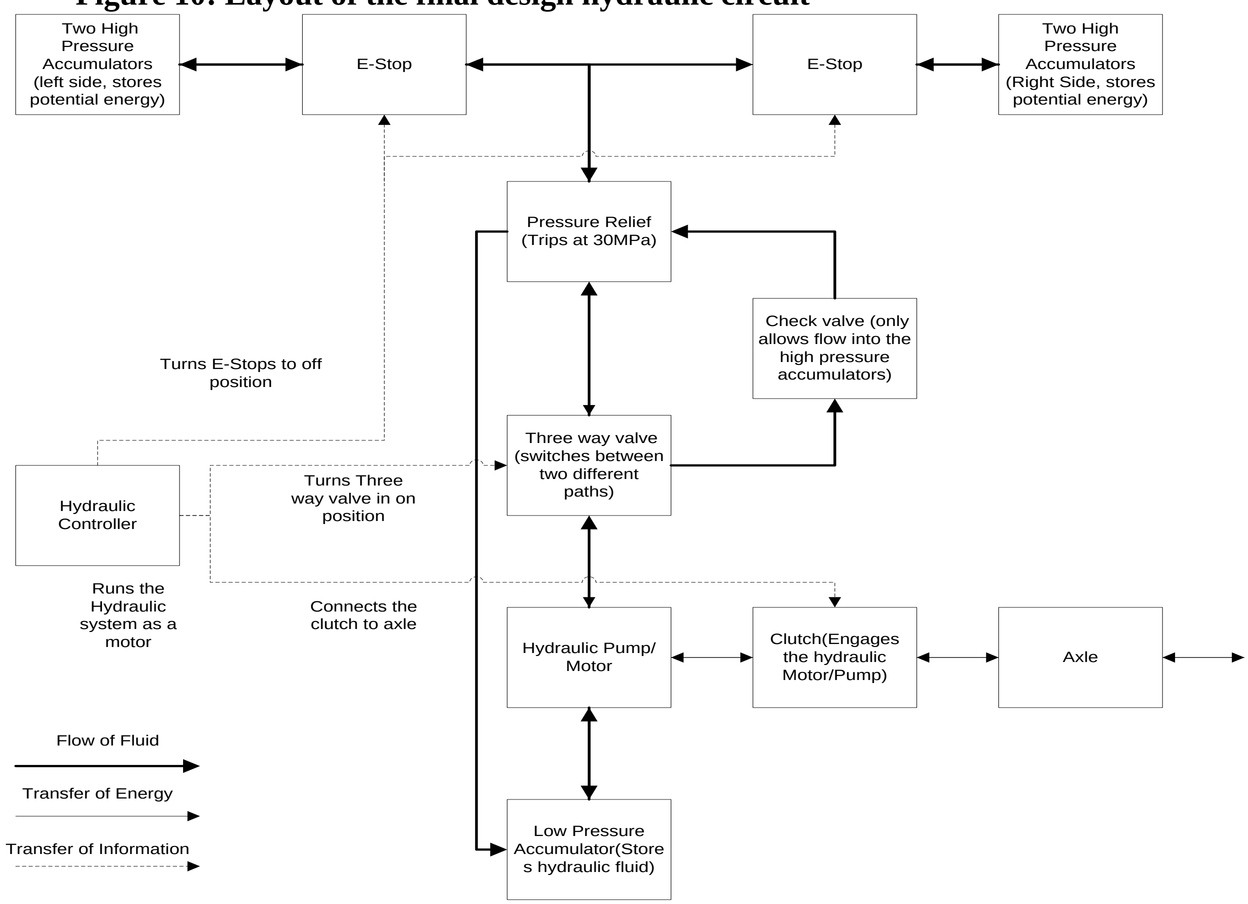
Electrical System

The accelerator works by applying a voltage across a MOSFET potentiometer in the controller, which is a stock vehicle component. In the final design, when the person driving the Xebra is using the hydraulics to accelerate, an electrical switch is actuated and the voltage from the accelerator is applied across the hydraulic controller, which will then apply a voltage across the electrical controller but it will limit the voltage applied and thereby ensuring that the electric controller does not draw more than 5kW of power from the batteries. The batteries are the most efficient at 1kW (Appendix E), and for this reason the power draw from the batteries is limited to that value. When the hydraulic motor is not in use or if the hydraulic fluid runs out, the switch is then deactivated so the voltage applied by the accelerator is directly connected to the electric controller and the power draw is not limited to 5kW. The new hydraulic controller is also connected to a series of pressure, temperature and flow sensors, which are currently installed on the Xebra by a previous team. The controller will take measurements from these sensors and will cut off the hydraulic system before the pressure exceeds 30 MPa and/or before the temperature of the hydraulic fluid exceeds 216^{o}C, which is the flash point of the hydraulic fluid (see Appendix V).

Hydraulic System

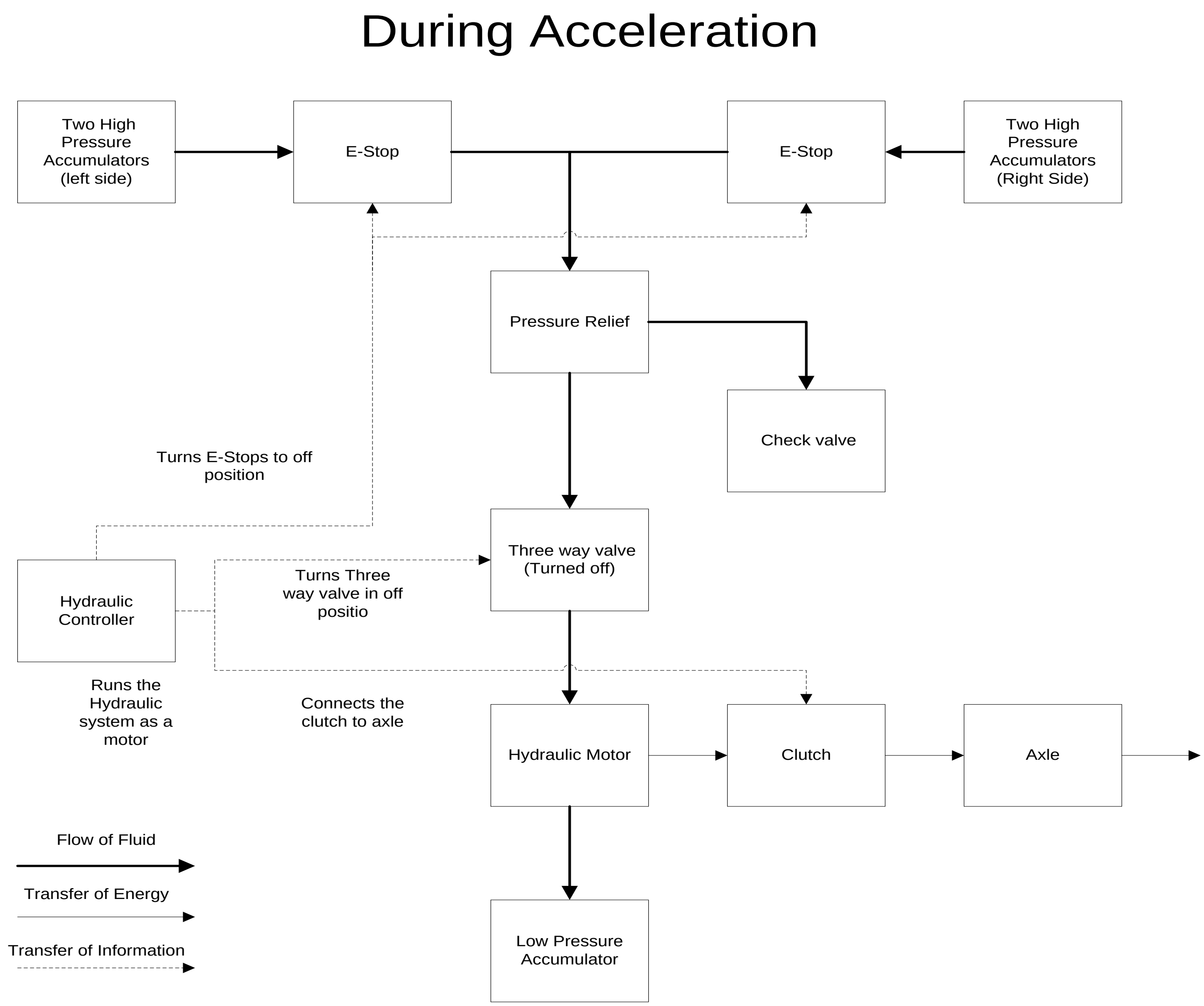
The hydraulic system will use only one 38cc motor/pump with a gear ratio of 4.8:1. This motor/pump was decided upon after calculating the acceleration and torque values for different pumps/motors located in Appendix W. This motor/pump will give us a maximum torque of 725.75Nm compared to 522 Nm of the 23cc motor and 630 Nm of the 33cc pump. It will also give us a maximum acceleration of 1.73 m/s^2 compared to 1.12m/s^2 by the 23cc motor on the prototype. This new motor/pump will allow the Xebra to reach the desired top speed of 45Mph in approximately 23 sec. The hydraulic system would make use of a hydraulic clutch which would allow the transfer of kinetic energy between the hydraulic system and the motor shaft. The clutch would be engaged when the user chooses to use the hydraulic pump or motor; the clutch disengages when only the electric motor is in use. The hydraulic system also incorporates the use of a check valve which would only allow fluid to flow into the high pressure accumulator. A three way valve would be used to direct the flow of the hydraulic fluid into or out off the hydraulic motor/pump. A pressure relief valve is also present in the hydraulic system. This relief valve will be tripped once the pressure in the accumulators exceeds 30 MPa, once tripped the relief valve will dump all the hydraulic fluid from the high pressure accumulators to the low pressure accumulators. A schematic of this system can be seen in Figure 10 below.

Figure 10: Layout of the final design hydraulic circuit



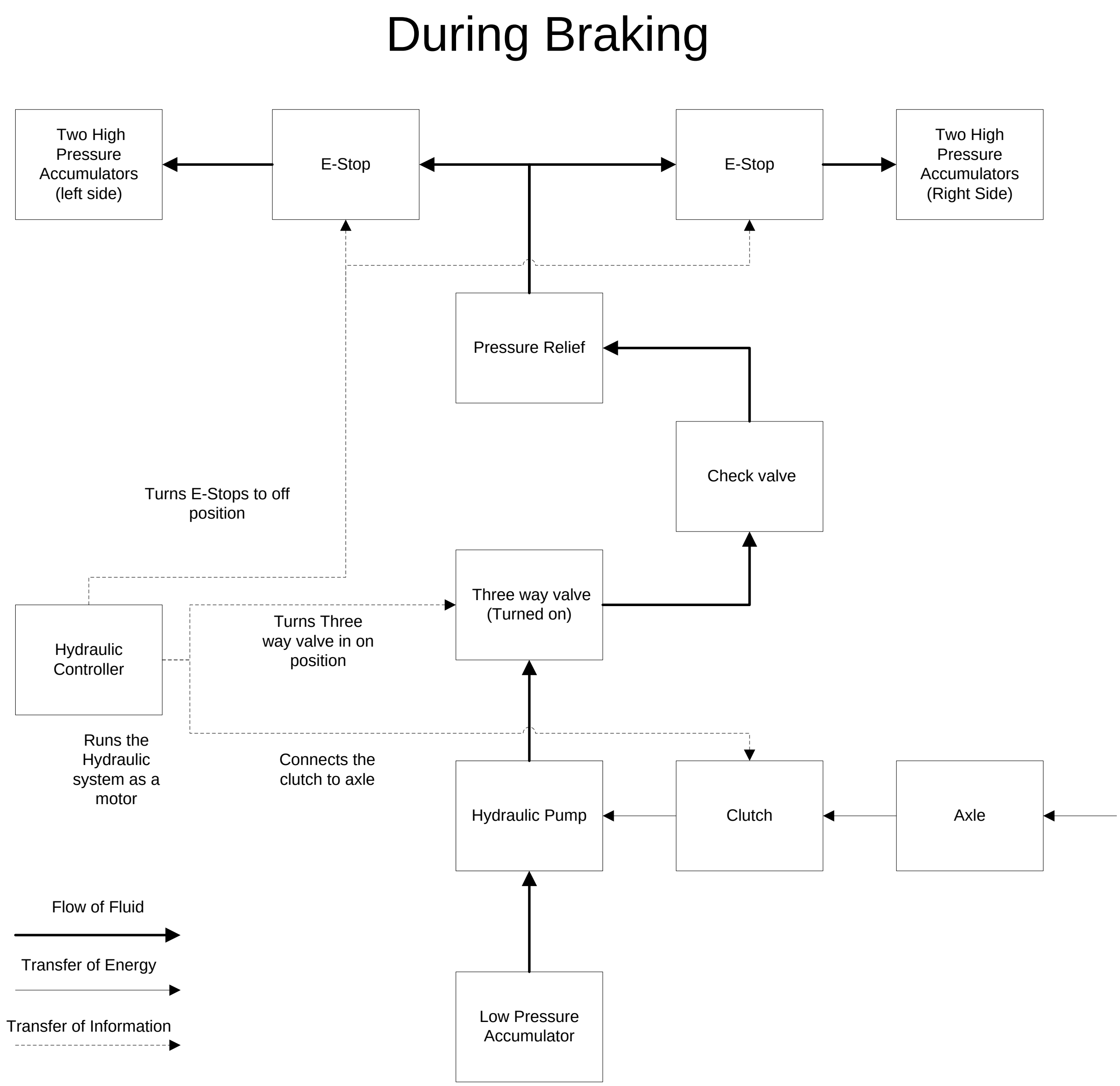
Acceleration: During acceleration the hydraulic controller actuates the clutch and allows the flow of energy from the motor to the axle. The controller also switches the three way valve to the off position thereby not allowing flow into or from the hose that is connected to the check valve. The controller also switches the E-Stop valves to the off position thereby allowing flow out of the high pressure accumulator and into the hydraulic motor. The hydraulic motor would convert fluid potential energy to kinetic energy. This kinetic energy would then be transmitted to the wheel via the clutch and a 4.8:1 gear ratio. This process is shown in Figure 11 below.

Figure 11: Flow of hydraulic fluid during acceleration.



Braking: During braking the hydraulic controller actuates the clutch and allows the flow of energy from the axle to the hydraulic pump. The controller also switches the three way valve to the on position thereby only allowing flow into the hose that is connected to the check valve. The check valve allows flow into the high pressure accumulator and does not allow flow out of the high pressure accumulator. The controller also switches the E-Stop valves to the off position thereby allowing flow into of the high pressure accumulator and out of the hydraulic pump. The hydraulic pump would convert kinetic energy to fluid potential energy. This fluid potential energy would be stored in the high pressure accumulator. This process is illustrated in Figure 12 below.

Figure 12: Flow of hydraulic fluid during braking



During an emergency: Depending upon the type of failure of the system, different steps would be taken to ensure the safety of the people in and around the Xebra and to limit the amount of damage to the hydraulic system. If the pressure in the high pressure accumulator is above 30 MPa the pressure relief valve will get tripped and all the hydraulic fluid will get dumped into the low pressure accumulator as shown on the diagram in Appendix X. If there is a leak in the system or possibility of an accident the driver can hit the E-Stop button which will turn the E-Stop valves to the on position and not allow the flow of fluid out of the high pressure accumulators.

Mechanical System

The mechanical system of the Final Design will be very similar to prototype. The main change will be the removal of the gears and assembly of the 23cc Hydraulic motor installed in the prototype. The final design will use the same sprockets, chain and clutch as the 33cc hydraulic pump installed in the prototype, which is an ANSI 60 chain, a 19 teeth sprocket and a 20 teeth sprocket.

FABRICATION PLAN

Although our project is comprised of complex mechanical and electrical systems, our overall machining and fabrication was relatively minimal. This is because we kept many of the existing systems in tact while making only slight modifications. Also, we are relied on a number of purchased components to be used in our final prototype. The majority of our fabrication was focused on modifying the vehicles gearing and also on mounting the additional accumulators. The following sub-sections describe each component that was fabricated and the steps and details associated with each process. For detailed engineering drawings of the components that were fabricated, see Appendix Y.

Hydraulic pump and motor sprockets: In order to achieve the proper gear ratios, we replaced the sprockets that connect directly to the hydraulic pump and the hydraulic motor. Since the shafts on these components use a non-standard keyway we were unable to purchase pre-finished sprockets. Therefore, we have ordered two machinable sprockets from McMaster-Carr that come with a standard bore size of 0.625 inches. The shafts on both the pump and motor are identical, so the machining of both sprockets was exactly the same. First, the sprockets were bored out on a lathe to a size of 0.875 ± 0.001 inches in order to accommodate the shafts on the hydraulic pump and motor. Then, using a special keyway tool in the ME 450 machine shop, we cut a keyway that will accommodate a square key of dimensions 0.25 x 0.25 ± 0.001 inches. Details for the tooling and speed of these operations can be seen in Tables 16 & 17 below. It is important to carefully match the specified tolerances for these operations. This will ensure that the sprocket-key assemblies will mate properly with the motor and pump shafts.

Table 16: Machining processes for the hydraulic pump sprocket.

Raw Material Stock:	Steel Machinable-Bore Sprocket for #60 Chain, 3/4" Pitch, 19 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck
3	Bore out inner diameter to tolerance (0.875" ± 0.001)	lathe	800	Boring bar	4 Jaw Chuck
4	Cut keyway for 0.249 x 0.249" ± 0.001 key	Press	-	keyway tool	Vice

Table 17: Machining processes for the hydraulic motor sprocket.

Raw Material Stock:	Steel Machinable-Bore Sprocket for #40 Chain, 1/2" Pitch, 17 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck
3	Bore out inner diameter to tolerance (0.875" ± 0.001)	lathe	800	Boring bar	4 Jaw Chuck
4	Cut keyway for 0.249 x 0.249" ± 0.001 key	Press	-	keyway tool	Vice

Hydraulic motor sprocket (drive shaft): Apart from replacing the sprocket that connects directly to the hydraulic motor, we also replaced the sprocket that links the hydraulic motor to the drive shaft. This fabrication process was slightly different from the other sprockets because the bore of this sprocket mates with a one-way bearing instead of a keyed shaft. Since the outer diameter of the bearing is not a standard size and requires a precise press fit, the bore of this sprocket also needed to be machined. Therefore, we ordered a machinable sprocket from McMaster-Carr that comes with a standard bore size of 0.625 inches. The sprocket was bored out in a lathe to a bore size of 1.654 + 0.000, – 0.001 inches. This tolerance is very important because the bearing requires a specific press fit and will not function properly unless this tolerance is met. Table 18 below, shows a detailed breakdown of the speeds and tooling needed to fabricate this part.

Table 18: Machining processes for the hydraulic motor sprocket (drive shaft).

Raw Material Stock:	Steel Machinable-Bore Sprocket for #40 Chain, 1/2" Pitch, 20 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck
3	Bore out inner diameter to tolerance (1.654" + 0.000, - 0.001)	lathe	800	Boring bar	4 Jaw Chuck

One-way bearing spacer: Because the inner diameter of the one-way bearing is larger than the drive shaft, a spacer needed to be fabricated that essentially increases the drive shaft diameter to fit the bearing. To create this part, we started with round AISI 1020 steel stock and turned the outer diameter to 1.378 ± 0.001 inches on a lathe. We then drilled the center of the shaft to an inner diameter of 0.750 inches. Then, we bored out the inner diameter to a dimension of 0.875 + 0.001, -0.000 inches to accommodate the drive shaft. The tolerances on both the inner and outer diameters of this part are very important to ensure that the one-way bearing will function properly. After ensuring that the tolerances were met, the part was cut to a length of 1.5 ± 0.050 inches using a cut-off tool. The final step in creating this part was to create a keyway that will accommodate a square key of dimensions 0.25 x 0.25 ± 0.001 inches. This was done using a special keyway tool in the ME 450 machine shop. Table 19 and 20, p. 31, shows a detailed breakdown of the speeds and tooling needed to fabricate this part.

Table 19: Machining processes for the one-way bearing spacer.

Raw Material Stock:	1020 Steel Round Stock, 1.5” Diameter, 3.0” Length				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	3 Jaw Chuck
2	Face off end of stock	lathe	900	Turning Tool	3 Jaw Chuck
3	Turn down outer diameter to tolerance (1.378 ± 0.001)	lathe	900	Turning Tool	3 Jaw Chuck
4	Center Drill end of part	lathe	1600	Center Drill	3 Jaw Chuck
5	Drill inner diameter undersized	lathe	1600	Ø 0.75” Drill	3 Jaw Chuck
6	Bore out inner diameter to tolerance (0.875” + 0.001, -0.000)	lathe	800	Boring bar	3 Jaw Chuck
7	Cut part to length (1.5” ± 0.050)	lathe	500	Cut-off tool	3 Jaw Chuck
8	Cut keyway for 0.249 x 0.249" ± 0.001 key	press	-	keyway tool	Vice

Accumulator mounting beams: In order to mount the new accumulators, we needed to create new mounting bars that will be welded to the frame of the vehicle. These bars required very little fabrication. We started with 3 feet of stock which was cut into four different pieces at a length of 6.5”. These pieces were cut to length using the stock cut-off saw. This saw is essentially a horizontal band saw that is used to cut stock to length. Any rough edges were then cleaned up with sand paper to ensure the pieces meet up correctly before welding them to the frame. The tolerances for these beams are not that crucial. The length for each beam should be within 1/16th of the specified length.

Table 20: Machining processes for the accumulator mounting beams.

Raw Material Stock:	1020 Steel Extrusion; 1.5” Width, 2.5” Height, 13’ Length, 0.125” Wall Thickness				
No.	**Process Description**	**Machine**	**Feed (ft/min)**	**Tool**	**Fixtures**
1	Clamp part at specified length (four at 6” and two at 38.5”) ± 0.0625	Cut-off saw	-	Cut-off saw	clamp
2	Cut part to length, remove, repeat	Cut-off saw	300	Cut-off saw	clamp
3	Clean up rough edges	-	-	Sand Paper	-

Welding mounting beams to frame: After the accumulator mounting beams were cut, they were welded onto the frame. The first step in this process was to remove the old accumulators and their mounting brackets to ensure they were not damaged in the welding process. The next step in this process was to wire-brush the paint off of the frame of the vehicle. This ensured that the beams were welded properly to the frame and no paint was burnt in the process. We then had Bob Coury assist us with TIG welding the beams to the frame. After the welds were finished and cooled, we ground away any excess weld material to make sure that all of the surfaces are flat.

Drilling holes for accumulator brackets: Once the accumulator mounting beams were welded to the frame, we began to install the accumulator mounting brackets. We first put the accumulators inside the brackets and placed the brackets on the mounting beams. We then installed the fittings between the two accumulators while someone held the un-mounted accumulator securely. This allowed us to determine the exact spacing between the accumulators so that the hydraulic fittings would work properly. We then made marks on the mounting beams where the holes needed to be drilled for the new accumulator brackets. Finally, we used a hand drill and a 0.375” diameter drill bit to make the proper mounting holes.

DESCRIPTION OF VALIDATION APPROACH

Primary Method Validation

Table 21, p.33 lists the specifications along with the validation method used. Our sponsor, the EPA, wanted our design to be tested on a LA4 drive cycle, shown in Appendix Z, on their dynamometer, but due to problems with the electric controller as well as the regenerative braking system during validation testing, the vehicle was unable to run a LA4 drive cycle. Because a portion of the drive cycle requires the vehicle to exceed the 45 mph top speed specification, the vehicle was expected be at its top speed for this duration. The top speed specification would have been verified over this section of the drive cycle. During the dynamometer test, the vehicle’s acceleration, velocity, range, electric motor temperature, and electrical power draw was monitored and recorded. The vehicle was expected to run consecutive LA4 cycles until the batteries ran out, therefore validating the maximum range. A separate test on the dynamometer recorded the speed reached when launching the vehicle using only hydraulics.

The power supplied to the electric motor can be limited by the electric motor controller. A description of the controller is shown in Appendix AA. We had planned to set the controller to limit the power sent to the electric motor to 5kW, so that the electric motor only operates at an efficient power draw from the batteries, but since the vehicle’s top speed on only electric drive was lower than anticipated, the power was not limited to 5kW. See Appendix E for battery specifications. A voltmeter and an ammeter were used to monitor the vehicle’s power draw during dynamometer testing. The monitoring equipment was provided and installed by the EPA.

To verify that the vehicle is stable, a vehicle weight scale located at the EPA was used. The current vehicle, without any of our design alterations, was weighed earlier at the EPA. By weighing the vehicle a second time, with our design alterations, we can determine how much weight our design has added and also the new total weight of the vehicle. This was used to verify the stability customer requirement along with the weight specification.

Also to verify that the vehicle is stable, we used the same weight scale at the EPA, but measure the weight at the rear axle and the weight at the front tire independently. From these measurements, the position of the center of mass was determined. This verified the stability customer requirement along with the distance from rear axle specification.

Table 21: Primary and secondary methods that will be used to validate the specifications

Engineering Specification	Primary Validation Method	Secondary Validation Method
Overall Top Speed of Vehicle	Dynamometer Testing	Outdoor Testing/RPM Sensor
Max. Speed using only Hydraulics	Dynamometer Testing	Outdoor Testing/RPM Sensor
Initial Acceleration	Dynamometer Testing	Outdoor Testing/RPM Sensor
Range	Dynamometer Testing	Outdoor Testing/GPS Unit
Temperature of Electric Motor	Dynamometer Testing	Thermocouple Sensor
Power of electric motor	Dynamometer Testing	Controller Software
Weight of vehicle	Weight Scale at EPA	Sum Weights of Added Components
Distance of C.O.M.* From Rear Axle	Weight Scale at EPA	Portable Scale
Accumulator Pressure	Pressure Gauge	Wireless Pressure Sensor

*C.O.M.= Center of Mass

Secondary Method Validation

None of the contingency methods were necessary to validate our specifications. Contingency methods can be seen in Table 21 above. These secondary tests will be performed if problems or issues arise using the primary testing methods.

Instead of using a dynamometer, outdoor testing on a road or in a parking lot may be substituted. During the outdoor test, the RPM sensor data will be used to determine the speeds as well as the accelerations of the vehicle during testing. A GPS unit will be used to record the distance the vehicle travels during testing. The thermocouple on the electric motor will be used to ensure that the motor does not overheat. The controller software, shown in Appendix AA, will be used to monitor the power supplied to the electric motor during testing.

If the scale at the EPA cannot be used, the weight of the components that our design adds to the vehicle will be added to the previous recorded weight. Also a portable weight scale will be used to measure the weight on the front tire with the vehicle inclined at varying angles. From the weight and angle measurements, the position of the center of mass can be determined. The wireless pressure sensor on the vehicle can be used instead of using the pressure gauge.

VALIDATION RESULTS

Before validation tests were conducted, all the hydraulic fittings were tightened to the manufacturer's specifications using a torque wrench. This was done with supervision from personnel at the EPA.

A list of engineering specifications and the validation method used is shown in Table 22, p.34.

Table 22: Methods used to validate our prototype's specifications along with the measured results from dynamometer testing

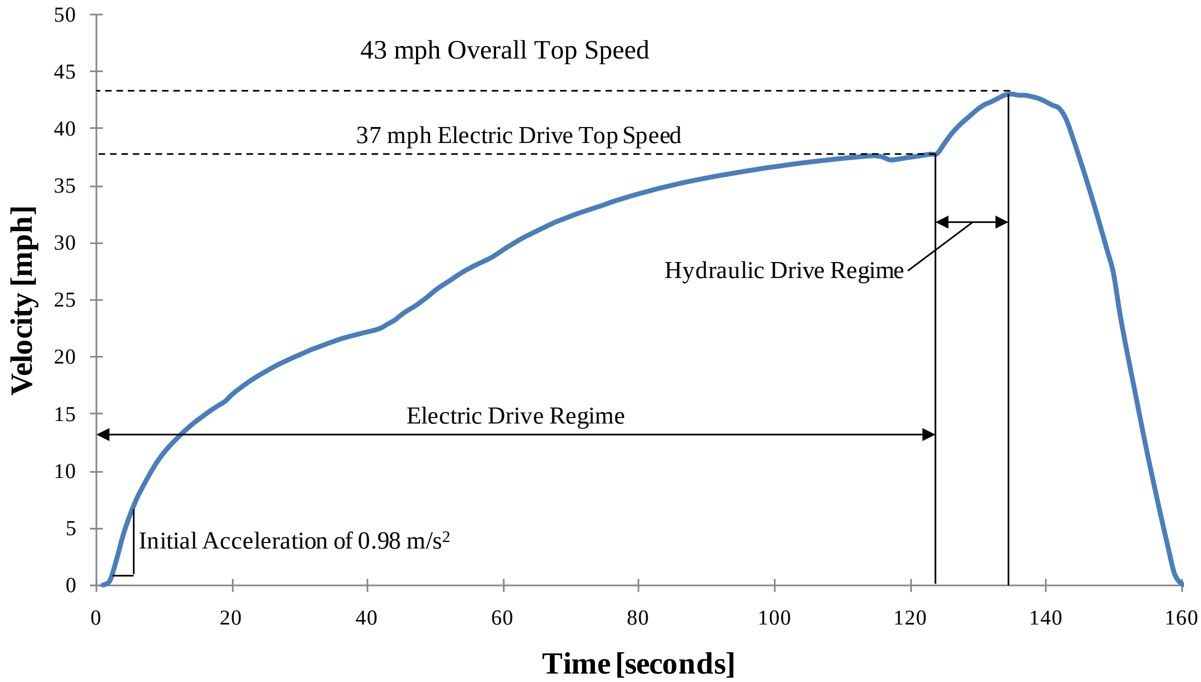
Specification Description	Method	Specification	Measured Result	Unit
Overall Top Speed of Vehicle	Dynamometer	20.1 (45)	19.1 (42.7)	m/s (mph)
Max. Speed Using Hydraulics	Dynamometer	20.1 (45)	4.47 (10)	m/s (mph)
Initial Instantaneous Acceleration	Dynamometer	≥ 1.72	0.98	m/s^2
Range	Dynamometer	25.75 (16)	—	km (mi)
Temperature of Electric Motor	Dynamometer	<135	—	°C
Max. Power of electric motor	Dynamometer	<5	8.2	kW
Mass of vehicle	Weight Scale	≤1100	1153	kg
Distance of C.O.M.* From Rear Axle	Weight Scale	≤0.8	0.841	m
Accumulator Pressure	Transducer	≤ 25	24.8	MPa

*C.O.M.= Center of Mass

Overall Top Speed of Vehicle: A plot of the vehicle's velocity with time, which was recorded by the dynamometer, can be seen Figure 13, p. 35. The top speed of the vehicle was determined by engaging the vehicle's electric and hydraulic systems independently while the vehicle was constrained on a dynamometer. Before the vehicle ran on the dynamometer, the hydraulic system was pressurized to 24.8 MPa using the slow fill pump. The vehicle then accelerated using only the electric drive until the electric drive reached its peak velocity of 37 mph. Once the electric drive reached its peak velocity, the hydraulic system was engaged, accelerating the vehicle to an overall top speed of 43 mph. Due to time constraints, this test was conducted once. Further testing should to be conducted to determine if the results can be replicated.

The electric drive was predicted to reach 45 mph without using the hydraulic system because of the increased diameter of the new tires. During the dynamometer testing, it was determined that the electric drive was capable of reaching 37 mph. This discrepancy in the predicted and actual speeds was determined to be a result of a faulty DC electrical motor controller (Appendix AA) that was installed by the manufacturer. This was determined because the vehicle's electric drive would malfunction at speeds below 15 mph and was generally unpredictable during testing. This problem has been addressed on subsequent Xebra models by replacing the motor controller with a more robust controller.

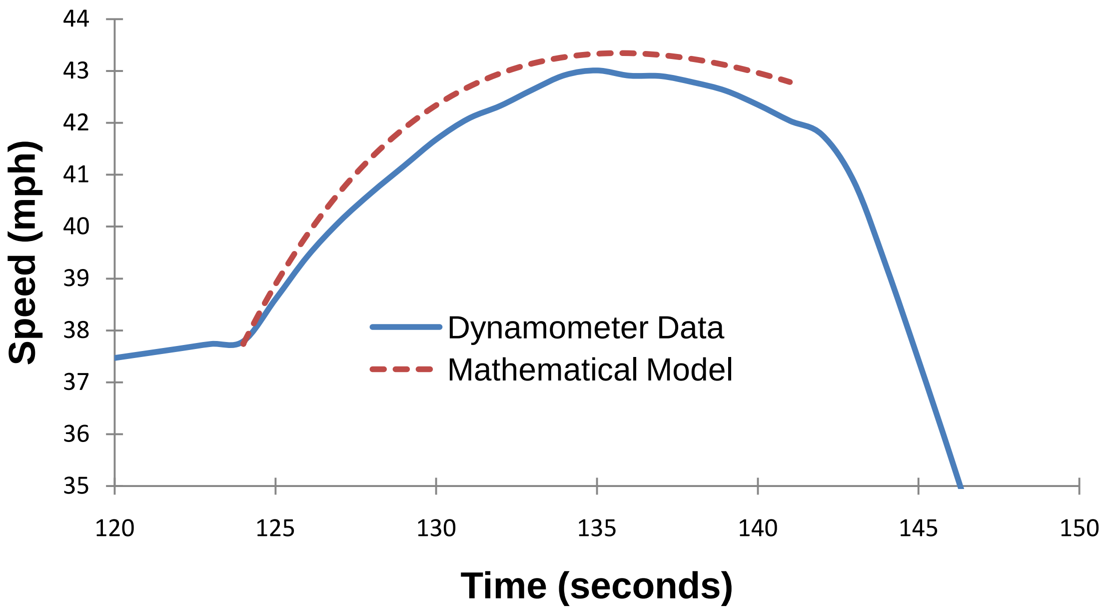
Figure 13: Velocity curve from dynamometer testing at the EPA



Max. Speed Using Hydraulics: A test of a purely hydraulic launch with the system pressurized to 24.8 MPa was conducted on the dynamometer, which resulted in a speed of 10 mph. It was determined that during this test, energy was being transferred from the hydraulic system to the electrical system by running the electric motor as an electric generator. For this test to be valid, the electric system should have been disengaged. A subsequent test with the electric system disengaged could not be conducted due to time constraints.

From our mathematical model, we predicted that the top speed using only hydraulics would be 35.6 mph. To accurately validate our mathematical model, we ran a simulation that would model the procedure used to test the overall top speed of the vehicle which was described in the previous section. To do this we applied an initial velocity to our model of 37 mph. We then ran the model, simulating the hydraulic boost that was experienced while testing on the dynamometer. A comparison of our mathematical model to the dynamometer data is shown in Figure 14, p. 36. As you can see, the mathematical model very closely models the behavior of the actual Xebra. Based on this result, we believe that retesting the hydraulic launch with the electrical system completely disengaged would result in a top speed of 35.6 mph.

Figure 14: Comparison of the mathematical model to the dynamometer data



Initial Instantaneous Acceleration: During testing on the dynamometer, the vehicle's electric drive malfunctioned when the vehicle's electric throttle was 100% (accelerator pedal fully engaged). This malfunction was determined to be the result of the faulty electric motor controller. Therefore the vehicle's maximum initial instantaneous acceleration could not be determined. With the electrical system fully functional and the hydraulic system pressurized to 24.8 MPa, the vehicle was predicted to have an initial instantaneous acceleration of 1.72 m/s^2 with both electric and hydraulic drives engaged simultaneously. The data shown in the velocity plot in Figure 13, p.35 was recorded with the user gradually increasing the throttle in order to prevent the electrical system malfunctioning.

Range: Because the hydraulic regenerative braking system and the electric motor controller were malfunctioning during testing, a range test could not be conducted. The EPA conducted dynamometer testing on an unmodified stock Xebra vehicle with an upgraded electric controller, and determined that its range was 16 miles by running consecutive LA4 drive cycles, which is shown in Appendix Z. With the hydraulic regenerative braking system fully functional, the modified vehicle was predicted to exceed this range because the friction brakes would be used less during deceleration.

Temperature of Electric Motor: Monitoring the temperature of the electric motor was not done because a temperature probe that could connect to the motor was not available. A large floor fan directed air flow toward the motor during testing to ensure that the motor was being cooled.

Max. Power of Electric Motor: The voltage across the batteries and current through the batteries were monitored using equipment installed by EPA certified personnel. From the voltage and current, the power drawn by the electric motor can be determined. During testing a maximum power of 8.2 kW was recorded. Since our priority was to meet the overall top speed specification, the electric controller was set to allow its maximum rated current of 300A to be applied to the electric motor, therefore increasing the vehicle's speed. This resulted in a higher power draw than what was specified in the engineering specifications. The power draw can be limited through the electrical controller software, which is explained in Appendix AA.

Mass of Vehicle: Our modifications added 88 kg to the vehicle, which resulted in the overall mass exceeding the engineering specification by 53 kg. The vehicle was placed on the scale at the EPA before we made any modifications to the vehicle and it had a mass of 1065 kg. The vehicle was then placed on the same scale after our modifications and it had a mass of 1153 kg. Since the majority of the weight from our modifications came from the addition of two 34 kg accumulators, the improved performance that the additional accumulators provided was determined to be a higher priority than the weight specification (as shown in the QFD in Appendix D).

Distance of Center of Mass from Rear Axle: The distance was determined by determining the normal force at each of the three tires using the scale at the EPA. The ratio of normal force on the front tire to the combined normal force on the two rear tires is equal to the ratio of the distance of the center of mass from the rear axle to the total distance between the rear and front tires. The height of the center of mass was assumed to be 0.684 m, although it was found to be 0.661 m during the validation at the EPA. This was found by weighing the front tire when it was flat on the ground then when it was raised a specific height. Using the actual height of the center of mass to find the maximum distance from the rear axle, with a safety factor of two, we get 0.82 m. The center of mass is 0.672 m from the left side of the vehicle.

Accumulator Pressure: The accumulator pressure was monitored using a pressure transducer. The maximum recorded accumulator pressure was 24.8 MPa. The location of the transducer on the hydraulic circuit can be seen in Figure 15, p. 38.

Experimental Testing

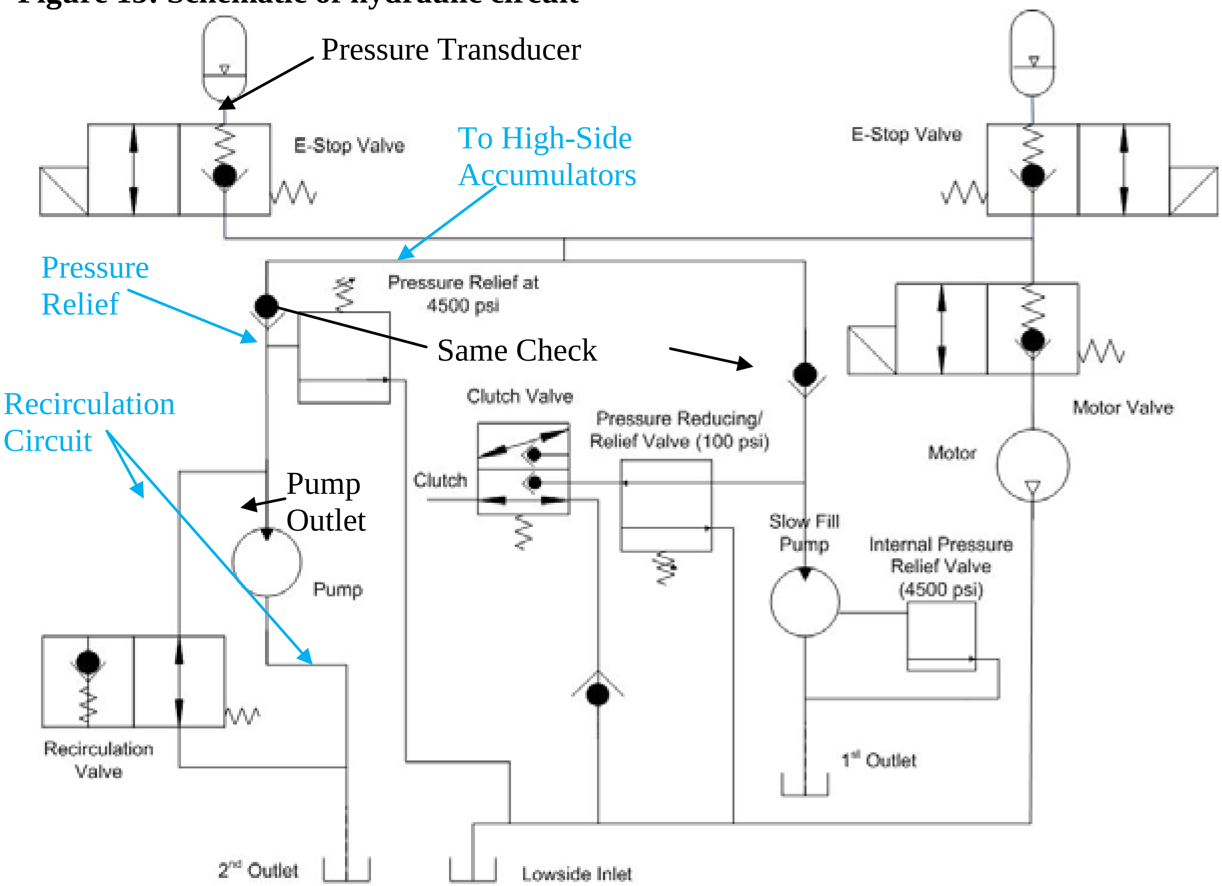
During the validation testing, it was discovered that the DC electric motor controller, shown in Appendix AA, and the recirculation valve, shown in Figure 15, p.38 were faulty.

DC Electric Motor Controller: The motor controller was monitored using the software provided by the controller manufacturer. A 04 error code, explained in Appendix AA, was recorded during multiple tests. This malfunction shut down the electric drive during testing. Also the vehicle's electric drive would start and stop at a constant frequency when the user had the throttle held constant at 100%. The electric drive starting and stopping malfunction was visually confirmed as the vehicle accelerated and decelerated at a constant frequency during several dynamometer tests.

Recirculation Valve: It was determined there was a malfunction in the regenerative braking system because when the regenerative braking was engaged, pressure did not increase in the accumulators. A camera placed behind the vehicle to monitor the clutch and chain drive that engaged the pump, showed that there were no faults in the clutch and chain drive. It was therefore determined that there was a fault in the hydraulic circuit, shown in Figure 15, p. 38. In order to isolate the component in the circuit that was malfunctioning, the dynamometer was used to rotate the tires at a constant 139 RPM (10 mph), with the hydraulic clutch engaged. This rotated the pump at a constant 658 RPM (4.74 gear ratio from tires to pump). With a pump displacement of 33 cm^3/rev, as shown in Appendix G, the flow rate across the recirculation valve was 21.7 L/min (5.73 gal/min). After the clutch was engaged for 1 second (there is a time delay so that the clutch does not have any torque applied to it while it engages), the recirculation valve was engaged to apply torque to the pump and send hydraulic fluid into the high-side accumulators. The voltage and current to the recirculation valve was monitored to ensure that power was being applied to the valve. This flow rate should have pressurized the accumulators to 24.8 MPa (3600 psi) in 48 sec, but the pressure did not substantially increase during this test.

After running the test for 1 min, the hydraulic pipe fittings were touched to feel how much the temperature of the fittings had risen. The temperature of the recirculation pipe fittings were at a higher temperature than the fittings leading to the high-side accumulators. Also, the temperature of the pressure relief had not increased during the test. Since the fluid has three paths (high-side accumulators, pressure relief, recirculation circuit) from the outlet of the pump and the temperature of the recirculation circuit was at a higher temperature than the other paths, the fluid was determined to be flowing primarily through the recirculation circuit. From this evidence, it can be concluded that the recirculation valve is malfunctioning and not allowing fluid to flow into the high-side accumulators.

Figure 15: Schematic of hydraulic circuit



DISCUSSION

Design Strengths

The most beneficial aspect of our design is the increased speed predicted by our mathematical model, while still maintaining the efficiency and acceleration of the vehicle. This increases the application of the vehicle immensely, allowing it to travel on all urban and suburban roadways. Another improvement of our design is the improved optimization of the system, namely the pre-charge pressure and the amount of hydraulic fluid in the system to pressurize the high pressure accumulators. We were allowed to do this because of our extensive mathematical calculations and our resulting computer model.

Due to the testing that we were able to perform initially, we were able to track down the problems with the hydraulic and electric systems in order to progress them further toward working condition. Other than the electric motor controller and the recirculation valve, the system is in fully functioning condition and will perform as we predicted. In addition to this, the system is fully reversible if the changes we made are found to be unnecessary in future semesters due to a substantial change in the system. Another strength to our design is its aesthetically pleasing design, which was reassured during the design exposition by engineers who had worked on the system in the past and their positive comments on the increased aesthetic appeal.

Design Weaknesses

Despite the numerous improvements to the vehicle, our design is not perfect and lacks many aspects that were either not added due to time, reversibility of the systems, or increased complexity of the systems. The main weakness is that the system still has two main issues that were not fully fixed. The overheating of the electrical system was assumed to be solely related to the missing fan on the motor, although it is now apparent that the electric motor controller for the vehicle is not sufficient and will overheat as well. The other major issue is the inoperability of the regenerative braking, which was addressed by replacing a faulty pressure relief valve. Despite this there is another problem with the regenerative braking, found during final testing, which is a faulty recirculation valve.

Both the overheating and the inoperable regenerative braking problems were both addressed and solutions were found, although the discovered problems were not the only issues. We initially considered that the controller was overheating, but the motor was a much more likely cause. We also looked for error codes on the controller and did not find any. The regenerative braking issue was addressed by replacing the leaking pressure relief valve. Since that valve was leaking, and the recirculation valve was receiving a voltage as found during our testing, we decided that the recirculation valve was unlikely to be malfunctioning, although there was no way to confirm this.

Another weakness of our design is the electrical wiring, since it was not addressed fully in our design. The wiring is confusing and not ideal since it could be done much better with color coded wires designed for vehicle electronics. The rewiring of the whole system would not only be much safer, but much easier to understand and allow quicker troubleshooting of the system. In addition, the wireless controller was rebuilt although time constraints impeded us from installing it. The wireless controller would provided more detailed information of the system and may make it much easier to locate problems.

Although we were not able to change the motor or pump, we would have ideally implemented a single variable displacement motor that would allow for a higher displacement during braking and acceleration. Due to time constraints and difficulty of the task, we were not able to work on the control systems and programming the vehicle. Another issue we had was with the addition of protection beams of the high pressure accumulators, as they may have interfered with the tire clearance.

Potential Changes

To correct the issues we had with fixing the existing problems with the vehicle, we would have continued troubleshooting the problems even after we found a problem incase multiple problems existed. We would have also worked on correcting the problems sooner and assembling the vehicle sooner in order to test it while there was still time to do work afterwards. Rewiring the electrical system and wiring in the wireless controller is a simple task that was not completed due to time constraints and should be completed by future semesters.

Implementing the single variable displacement pump would allow the motor to run at its most efficient state continuously while improving the smoothness of the ride. This would be a lengthy and complicated task, and it is recommended for any future projects similar to this one that are starting. By modifying the controller the efficiency could be increased and the vehicle operation simplified and optimized. Finally, the necessity of high pressure accumulator protection beams is recommended to be further explored and implemented if necessary.

RECOMMENDATIONS

In order for the Xebra to complete the LA4 drive cycle on the dynamometer at the EPA and therefore validate the range and the increase in efficiency of the hybrid Xebra two changes need to be made. The insufficient electronic controller needs to be upgraded and the malfunctioning recirculation valve needs to be replaced. In order to achieve and maintain a top speed of 45MPH and to obtain a fully operational prototype a few necessary changes are necessary as listed below:

Recirculation Valve: The three way recirculation valve currently installed on the Xebra is malfunctioning, allowing fluid to flow when it should not. Further analysis is needed in order to determine the type and specifications of this recirculation valve in order to replace it.

Electric Controller: The electric controller currently installed on the Xebra is rate for a max current output of 300 amps. This max current output is insufficient to provide adequate power to the electric motor in order to accelerate and maintain a speed of 45mph. This electric controller needs to be replaced with a controller that would provide a max current output of 450 amps. Such a controller is currently installed on the newer generation Xebra cars and is produced by Alltrax Inc. the model number is AXE -7245P. This controller with the current gear ratio should allow the Xebra to maintain a top speed of 45mph. Increasing to the 450 amp controller may affect the efficiency and range of the vehicle since more power will be used from the batteries.

Rewire Electrical System: The electric system added by previous semesters, needs to be rewired in order to make it easier to understand the function of each relay and the overall function of the electrical system. Due to time constraints we were not able to do so. Following teams should rewire the electric system to make it look professional and to ensure that all the valves and sensors are provided with adequate power. The current design of the electric system only charges the auxiliary battery when the Xebra is turned on, the rewiring should be done in a way that would allow the auxiliary battery to charge while the Xebra is being charged. There is also a current draw somewhere in the system that we could not track down, which is most likely caused by faulty wiring.

Hydraulic Clutch: A shaft collar needs to be installed to ensure that the hydraulic clutch does not translate on the motor shaft.

Accelerator and Brake Sensors: The position of the accelerator and brake sensors in the Xebra cab should be moved from their current location on the pedals and to a location which would be more convenient for the driver to use only the hydraulic or electric systems of the vehicle.

Pressure sensor: The Xebra has a wireless transmitter that allows the user to obtain real time data on the pressure and flow readings. A new pressure sensor was added to the hydraulic system and it needs to be calibrated and installed to the wireless transmitter.

Hydraulic fittings: The hydraulic fittings that are connected the output terminal of the hydraulic pump need to be redesigned and replaced. These fittings are currently very difficult to reach and assembling and disassembling these fittings take a long time. In order to reduce the time for maintenance and upkeep these fittings should be designed in a more efficient manner.

Low pressure accumulator: The low pressure accumulator contains a hydraulic fitting that is welded on. The weld beads on the low pressure accumulator cause interference with the frame of the Xebra. This interference makes it difficult to remove and to put back the low pressure accumulator in place. The following semester team should work on reducing this interference.

Major System Changes

Future semesters must work on replacing the malfunctioning recirculation valve and the electric control at the very least. The following ideas can be combined or modified to achieve a project with suitable scope for ME 450.

Utilizing one hydraulic motor/pump: The following semesters can implement the idea of using one hydraulic motor/pump which would be used as a motor for acceleration and as a pump for regenerative braking. An example of such an idea can be found in the Final Design section of this report on page 24.

Increase system pressure: The system currently can only obtain a maximum pressure of 3600 psi. The system can be redesigned so that it can reach a maximum pressure of 5000 or 6000 psi which would allow for more effective energy storage with less weight. Also, less fluid would be needed to store the same energy as energy.

Component layout: Currently it is a very tedious task to service, replace and reach the batteries of the Xebra. The following semester team can work on repositioning the slow-fill pump and other components so as to allow easier access the batteries of the Xebra.

Advanced controls: Another project for a future semester could be to design a single controller that would control both the electric and hydraulic systems of the Xebra. Currently the two systems are controlled independently and the driver must decide when to use either system. The new controller should be designed in such a way that it would utilize each system only in their respective efficient ranges.

Charging in reverse: Currently, when the Xebra moves in reverse, the hydraulic motor acts as a pump and pressurizes the accumulators. Future semesters could fix this problem.

Design for manufacturing: Once the Xebra is in working order and it is validated that the use of hydraulics does increases the range and efficiency of the Xebra, a project to design the hydraulic system of the Xebra for large scale manufacturability is also a feasible option.

New separately excited motor: The electric motor currently installed on the Xebra is produced by Zibo Boshan Super Motor Co. This company also offers a new electric motor and controller; this new electric motor and controller can vary both current and voltage, which allows for greater efficiency and top speed. As the manufacturer did not supply us with adequate information regarding the new motor, hence we were not able to analyze its potential use. So, a future semester can work on analyzing the feasibility of using this new electric motor and implement it is if found to increase efficiency and top speed of the Xebra.

PROJECT PLAN

The Gantt chart was used as a time line for our project in Appendix AB. For the most part we followed our time line on the Gantt chart. We missed our deadline we set for the completion of the installation and assembly of the hardware. This in turn delayed the date we conducted our tests at the EPA by one day. If there was no decrease in testing time we could have ran the Xebra on the dynamometer more times and conduct another top speed test as we believe that a driver error could have decreased our top speed.

Future semesters should first should order and install a new recirculation valve and a new electric controller. They should then conduct a test to ensure that new controller is in working order and it enables the Xebra to attain a top speed of 45 MPH using only the electrical system. This test can be conducted in a parking lot and it should be done before the team starts working on their changes to the Xebra. If possible future teams should finish installation at least a 7 days before the Design Expo, as this would allow sufficient time for testing and transportation to and from the dynamometer at the EPA.

CONCLUSIONS

In order to increase the functionality of the Xebra electric hydraulic vehicle, we have designed and implemented changes to increase the overall speed of the vehicle to 45 mph, while still maintaining the performance and efficiency. In addition to this, it was necessary to troubleshoot a number of problems with the electrical and hydraulic system that were found during initial testing. Despite our design changes to fix these systems, two new problems were found during validation, namely an insufficient electric motor controller and a faulty recirculation valve. These problems prevented us from completing a full validation of all of our specifications and decreased the performance of the system so we could not meet the specifications we did attempt to validate. None the less, we were able to achieve 43 mph and were able to validate that our gear ratios work correctly.

We have specified the faulty components and suggest the changes that need to be made. With these changes in place, we are confident that the Xebra will function as we predicted and maintain 45 mph. Supporting this claim is our mathematical model of the system, which was validated during initial testing as well as during our final validation. We have also specified any other changes that we would recommend for future semesters as well as anything we would have done differently.

ACKNOWLEDGEMENTS

Our design team would like to thank those who went above and beyond what was necessary in order to further the success of our project. Those people are as follows:

We would like to thank Bob Coury and Marv Cressey for all their invaluable assistance, guidance, and expertise with our manufacturing processes in the machine shop.

David Swain, from the Environmental Protection Agency, thank you for being a source that we can learn from and someone we can turn to with any questions. Also, thank you for the extra guidance you provided while our sponsor was out of town.

James Bryson, from the Environmental Protection Agency, thank you for your assistance in finding the proper hydraulic fittings for our project and for assisting in changing the pre-charge pressure in the hydraulic accumulators. Your expertise in hydraulic systems was a valuable asset to our team.

Ben Hagen, thank you for assisting us in disassembly of the vehicle and for your expertise.

Thank you, Dan Johnson for your help in the lab and with printing our poster. Also, thank you for all of your assistance with testing the Xebra throughout the semester.

Professor Gordon Krauss, we cannot thank you enough for your guidance and help with our project and the work that you put in on behalf of our team to make this a successful project and semester.

And lastly, a very special thanks to our sponsor, Dr. Andrew Moskalik of the U.S. Environmental Protection Agency, for being a constant resource for any questions that arose and any problems that the team encountered. Thank you very much.

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APPENDICES

Appendix A: Bill of Materials

Bill of materials for items purchased during Fall '09 semester

Item	Quantity	Source	Catalog Number	Unit Cost	Total Cost	Contact	Notes
24.2" Wheel Set	2	Discount Tire	83-4414	$128.41	$256.82	Discount Tire Dealership	
High Pressure Accumulator	2	Exotic Automation	ACP 10AA800 E1KTE	$746.00	$1,492.00	Exotic automation dealership	Produced By Parker-Hannifin Corporation
Accumulator Mounting Brackets	4	Exotic Automation	PHZ8700110476	$54.28	$217.12	Exotic automation dealership	Made for the accumulators used
Electrical Motor Cooling Fan	1	McMaster Carr	2059K12	$55.05	$55.05	Mcmastercarr.com	Flow rate of 105cfm
Hydraulic Pump Sprocket	1	McMaster Carr	6793K195	$29.86	$29.86	Mcmastercarr.com	19 Teet, machinable
Hydraulic Motor Sprocket	1	McMaster Carr	6793K152	$18.60	$18.60	Mcmastercarr.com	For electric motor shaft 20 teeth machinable
Hydraulic Motor Shaft Sprocket	1	McMaster Carr	6793K148	$13.39	$13.39	Mcmastercarr.com	17 Teeth machinable
Pressure Relief Valve	1	Sun Hydraulic	RPGS-CWN-CAKS	$115.50	$115.50	Sunhydraulic.com	Trips at 4500 Psi.
Steel Beams	3 feet	Alro Metals Plus	-	$2.8	$8.4		Steel (2.5" x 1.5" x 11) 6A Rect Stl Tube
One Way Bearing (Motor Shaft)	1	McMaster Carr	6392K52	$35.00	$35.00	Mcmastercarr.com	Locking torque of 89.245 ft-lbs.
Hydraulic Motor and Pump Ball Bearings	4	McMaster Carr	60355K508	$8.14	$32.56	Mcmastercarr.com	
Transmission Fluid	17 quarts	Mobil One	10W30	$7.99	$135.83		Donated By Sponsor.
Elbow SAE to JIC	2	Tompkins Industries Inc.	6801-16-16	$29.98	59.96	Tompkinsind.com	Male to male Rated at 5000Psi
Union JIC (F-F)	2	Tompkins Industries Inc.	6510-16-16	$90.50	$181.00	Tompkinsind.com	Female to female Rated at 5000Psi
Extender JIC	3	Tompkins Industries Inc.	6504-16-16	$19.17	$57.51	Tompkinsind.com	Female to male Rated at 5000Psi
Tee SAE-JIC-JIC	2	Tompkins Industries Inc.	FG6804-16-16-16	$33.80	$67.60	Tompkinsind.com	Male to male to male Rated at 5000Psi
Elbow JIC	2	Tompkins	FG6500-16-16	$20.74	$41.48	Tompkinsind.com	Female to male

		Industries Inc.		47			Rated at 5000Psi

Total = $ 2762.63

Bill of materials for items purchased during previous semester

Item	Quantity	Source	Catalog Number	Unit Cost	Total Cost	Contact
High Pressure Accumulator	2	Exotic Automation	ACP 10AA800 E1KTE	$746.00	$1,492.00	Exotic automation dealership
Hydraulic Motor	1	Exotic Automation	PGM517MA 0230 BM1H3ND6 D6B1B1B1	$555.00	$555.00	Exotic automation dealership
Hydraulic Pump	1	Exotic Automation	PGM517MA 0330 BM1H3ND6 D6B1B1B1	$570.00	$570.00	Exotic automation dealership
Low Pressure Accumulator	1	Grainger	4Z980	$76.32	$76.32	Grainger.com
Accumulator Mounting Brackets	4	Exotic Automation	PHZ87001104 76	$54.28	$217.12	Exotic automation dealership
E-Stop Valves	2	Hydac USA	WS 16ZR-01-M-SS 16-N-12-DS	$176.6	$353.2	Morrellinc.com
Recirculation Valve	1	Hydac USA	WS 16ZR-01-M-SS 16-N-12-DS	$176.6	$176.6	Morrellinc.com
Motor Valve	1	Hydac USA	WS 16ZR-01-M-SS 16-N-12-DS	$176.6	$176.6	Morrellinc.com
3- Way valves	1	Sun Hydraulics	DWDA-MAN512-ECI/S	$209.18	$209.18	RHM Fluid Power
Hydraulic Clutch	1	Logan Clutch Corp.	P35-0003 Industrial Clutch	$1,085.0 0	$1,085.00	Loganclutch.com
Time Delay Relay	1	Newark	29K8891	$99.69	$99.69	Amperite.com
Mounting Plates	2	Alro Metals Plus	7”x7.25”x0.5” Al	$15.55	$31.1	Alro Metals Ann Arbor
Bars for Spacers	1	Alro Metals Plus	0.75” RD. 12”long Al	$6.22	$6.22	Alro Metals Ann Arbor
Cross Bar	1	Alro Metals Plus	2”x1.25”x36”, 14Ga	$14.90	$14.90	Alro Metals Ann Arbor
ANSI 60 Chain	1	McMaster Carr	6261K473	$15.81	$15.81	Mcmastercarr.com
ANSI 40 Chain	1	McMaster Carr	6261K444	$12.28	$12.28	Mcmastercarr.com

14 AWG Elec. Wire	50’	McMaster Carr	7587K975	$14.80	$14.80	Mcmastercarr.com
60 Amp fuse	1	Advanced Auto Parts	BPAGU60GP	$3.97	$3.97	
Pic Programmer	1	Microchip Technology	PGI64120-ND	$34.99	$34.99	Digi-Key
28-PIN Board Pic	1	Microchip Technology	DM164120-3-ND	$24.99	$24.99	Digi-Key
Slow-fill pump	1	Provided By Environmental Protection Agency	N/A	$828.00	$828.00	EPA
Oil-Filter	1	McMaster Carr	44185K66	$64.02	$64.02	Mcmastercarr.com
Pressure Gauges	2	McMaster Carr	N/A	$75.00	$150.00	Mcmastercarr.com
Pedal Sensors	2	McMaster Carr	7692K4	$17.41	$34.82	Mcmastercarr.com
Various Hydraulic Fittings	□73	Donated by The Environmental Protection Agency	N/A	□$27.63	□$2016.99	David Swain (EPA)

Total :$8228.78

Overall Total : $10991.41

Many of the hydraulic fittings that are currently installed on the Xebra have not been documented. This is because work was done on the hydraulic system of the Xebra, when it was not utilized by ME450 teams as a project. This loss of documentation makes it difficult to obtain accurate data on the type and quantity of the hydraulic fittings. In order to provide our sponsor an estimate cost of reproducing the current configuration of the Xebra, we had to calculate an approximate cost of wall the hydraulic fittings. Using a previous team’s Bill of Materials (Winter 06) we calculated the average cost of all the fittings they used. We then counted the number of fittings that are currently installed in the hydraulic system. These two values are used to calculate the total cost of the hydraulic fittings that were installed in the Xebra by previous semesters. The average cost of each fitting is $27.63 and we counted that there are 73 hydraulic fitting of various sizes installed on the Xebra by previous semesters.

Appendix B: Description of Engineering Changes Since Design Review #3

Changes: The crossbeam shown in Figure 1 that is fixed between the two cantilevered accumulator mounting beams will no longer be included. This change is illustrated in Figure 2.

Figure B.1: Original Final Design for accumulator mounting beams

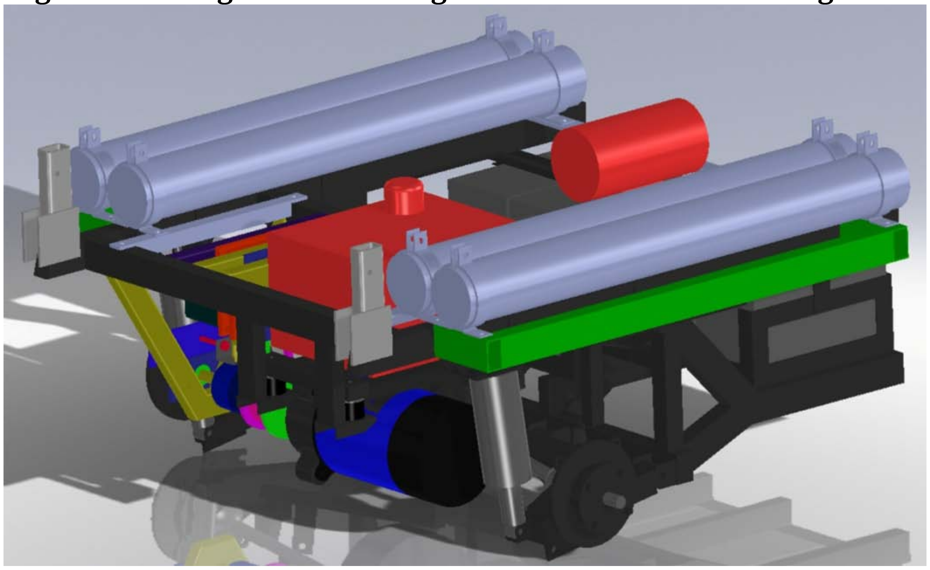
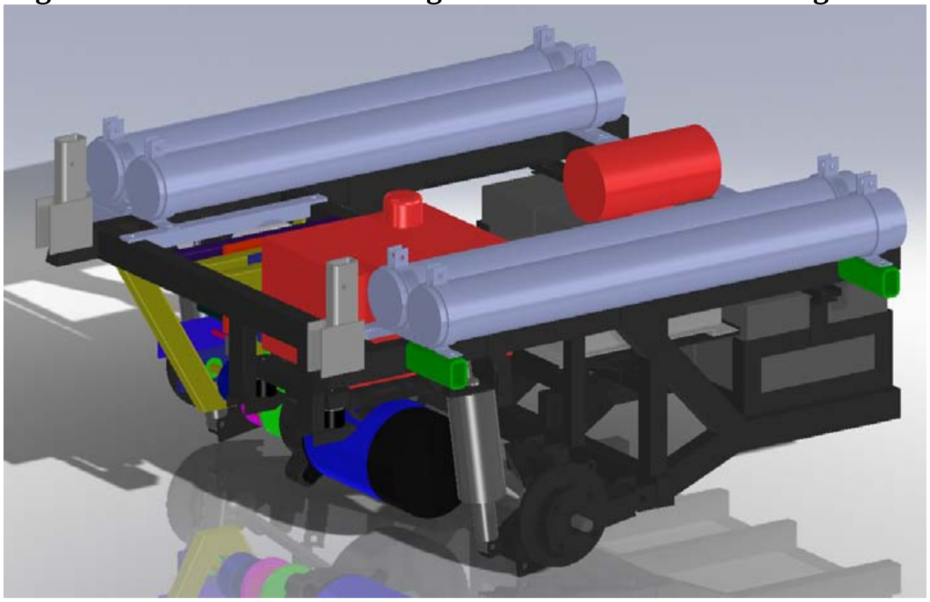


Figure B.2: Modified final design for accumulator mounting beams



Justification: Initially we measured the clearance for the new wheels while the Xebra was on jack stands. This allowed for the suspension to be extended further than it normally would be when the vehicle is sitting on the ground. After further investigation, we noticed this mistake and determined that when the vehicle hits a bump, the new tires would interfere with this crossbeam. For this reason we have decided to remove this crossbeam from our final design.

After noticing this problem, we looked for other solutions where we could keep the crossbeams and still avoid interference with the tires. The only options that we were able to come up with would not be able to fit under the truck bed of the vehicle. Therefore, we decided the best option is to not include the crossbeams.

After removing these crossbeams, we have ensured that the wheels will not hit the accumulators if the suspension is compressed. Figure 3 shows that there is a clearance between the top of the new tire and the bottom of the accumulator of 3 inches. The vehicle's suspension is equipped with a hard-stop, shown in Figure 4, which allows the suspension to travel a maximum of 2.5 inches. This gives a clearance of 0.5 inches between the tire and the bottom of the accumulator when the suspension is fully compressed.

Effects: These crossbeams were included in the original final design to shield the accumulators in the event of a side impact. Although the accumulators will no longer be protected by these crossbeams, we decided that it is more likely for the vehicle to go over a small bump than to incur a side impact, as the vehicle will not be a daily driver.

Figure B.3: Position of new wheel in relation to the frame

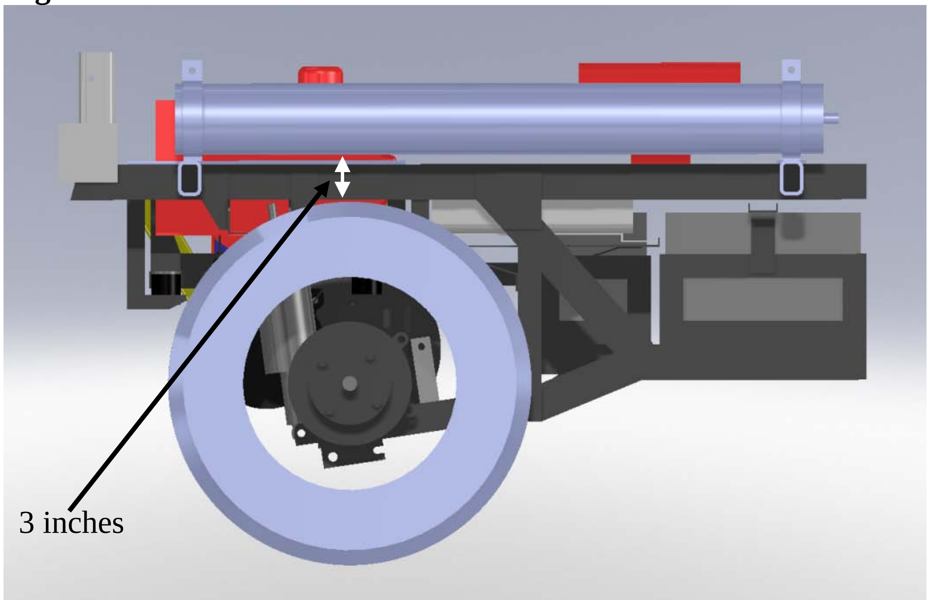
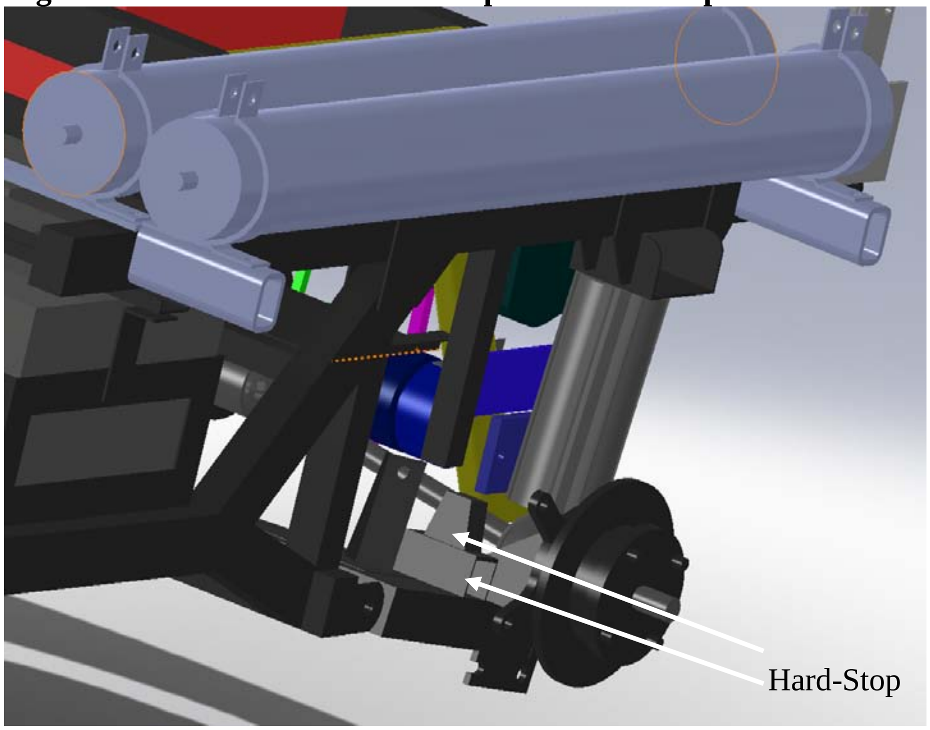


Figure B.4: Illustration of hard-stop for vehicle suspension



Appendix C: Design Analysis Assignment from Lecture

Material Selection Assignment (Functional Performance)

Accumulator Mounting Beams

1. The function of the beams is to support the load of the 75 lb. accumulator by transferring the load to the frame of the vehicle.
2. The beam should have a yield strength greater than 174 psi with a safety factor of 1.
3. Top five material choices generated from CES software:
    a. AISI 1020 Steel
    b. AISI 1095 Steel
    c. Syndiotactic Polystyrene (10% carbon fiber)
    d. Aluminum, 7075, wrought, T6
    e. Stainless Steel, martensitic, AISI 410S, wrought, annealed
4. AISI 1020 steel was chosen because it is the cheapest material listed and because it needed to be welded to the vehicle's steel frame. Since the frame is made of steel, steel needed to be choosen as a material.

Hydraulic Pump Sprocket

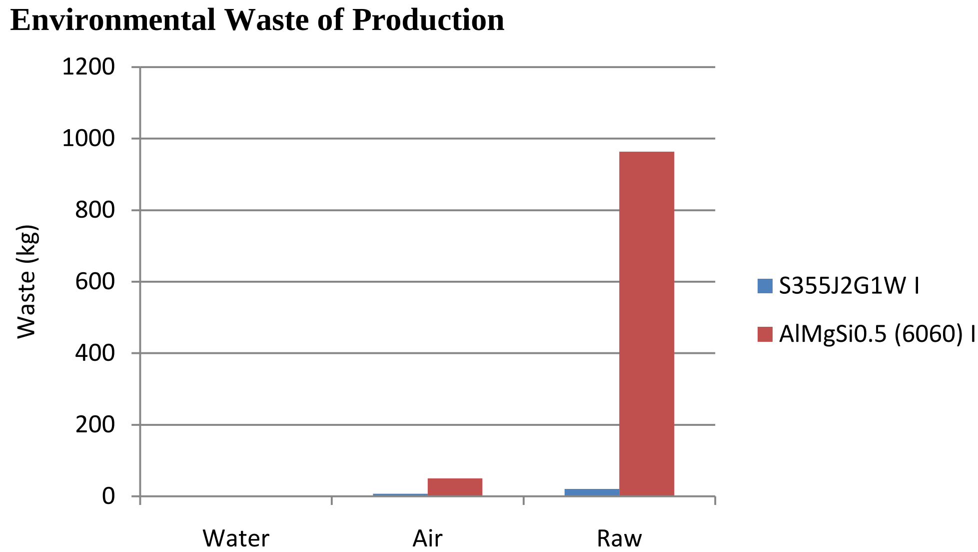
1. The function of the sprocket is to transmit torque from the drive axle to the hydraulic pump.
2. The sprocket should have a yield strength greater than 10,000 psi with a safety factor of 1.
3. Top five material choices generated from CES software:
    a. AISI 1040 Steel
    b. AISI 1095 Steel
    c. Aluminum 7075, wrought, T6
    d. Titanium, alpha alloy, Ti-02Pd (grade 11)
    e. Stainless Steel, martensitic, AISI 410S, wrought, annealed
4. AISI 1040 steel was chosen because it is the cheapest material listed and it is used in the manufacturing of machinable sprockets through a sintering process. The 1095 steel and the stainless steel would be much more difficult to machine and would also cost more.

Material Selection Assignment (Environmental Performance)

Design for Environmental Sustainability

The overall goal of integrating the regenerative braking system into a vehicle is to reduce energy consumption and emmisions by saving usually wasted energy. Although this is obviously a positive effect to the envirinment, there are other effects to the environment that must be taken into account, namely the environemental cost of production and the environmental effects in the whole lifetime of the vehicle. Two materials that could be used in the production of the added accumulator mounting beams were analysed on an environmental impact level, 1020 steel and 6060 aluminum. Although the addition of the accumulator mounting beams is a small aspect of the vehicle, this analysis can be applied to the entire vehicle and all design changes made.

Using SimaPro, the environmental impacts of 7.3 kg of steel and 5.1 kg aluminum were compared. The 7.3 kg of steel is the mass of steel that was actually used in our design. The 5.1 kg of alluminum is the resulting mass if alluminum was used in our design instead of steel. The resulting waste due to production of the materials is shown below, where it can be seen there is substantial raw and air waste from the aluminum, approximately 6.5 times as much as steel for air waste and 46 times as much for raw material watse.



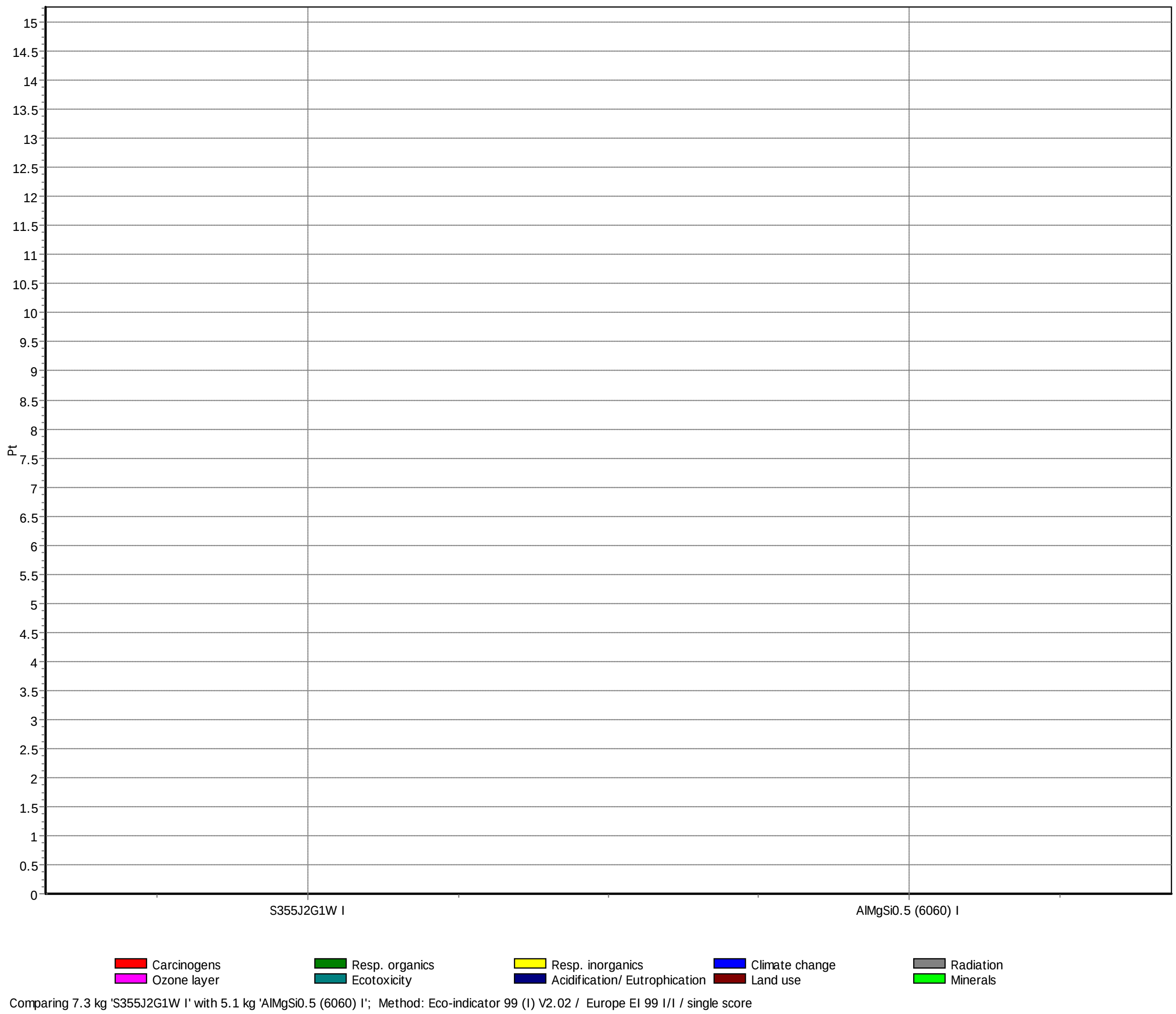
Relative Impacts on the Environment

The impact on the environment is much higher for the aluminum in every aspect when compared to the steel. The steel has substantial resp. organics and eco-toxicity, although even these are approximately only half of that of aluminum. The negative effects of aluminum on human health, ecosystem quality, and resources are all at least triple that of steel. Using a single score comparison, it is seen that aluminum has over ten times the negative environmental impact as steel. These results are all shown in the charts obtained using SimaPro, shown on the next page.

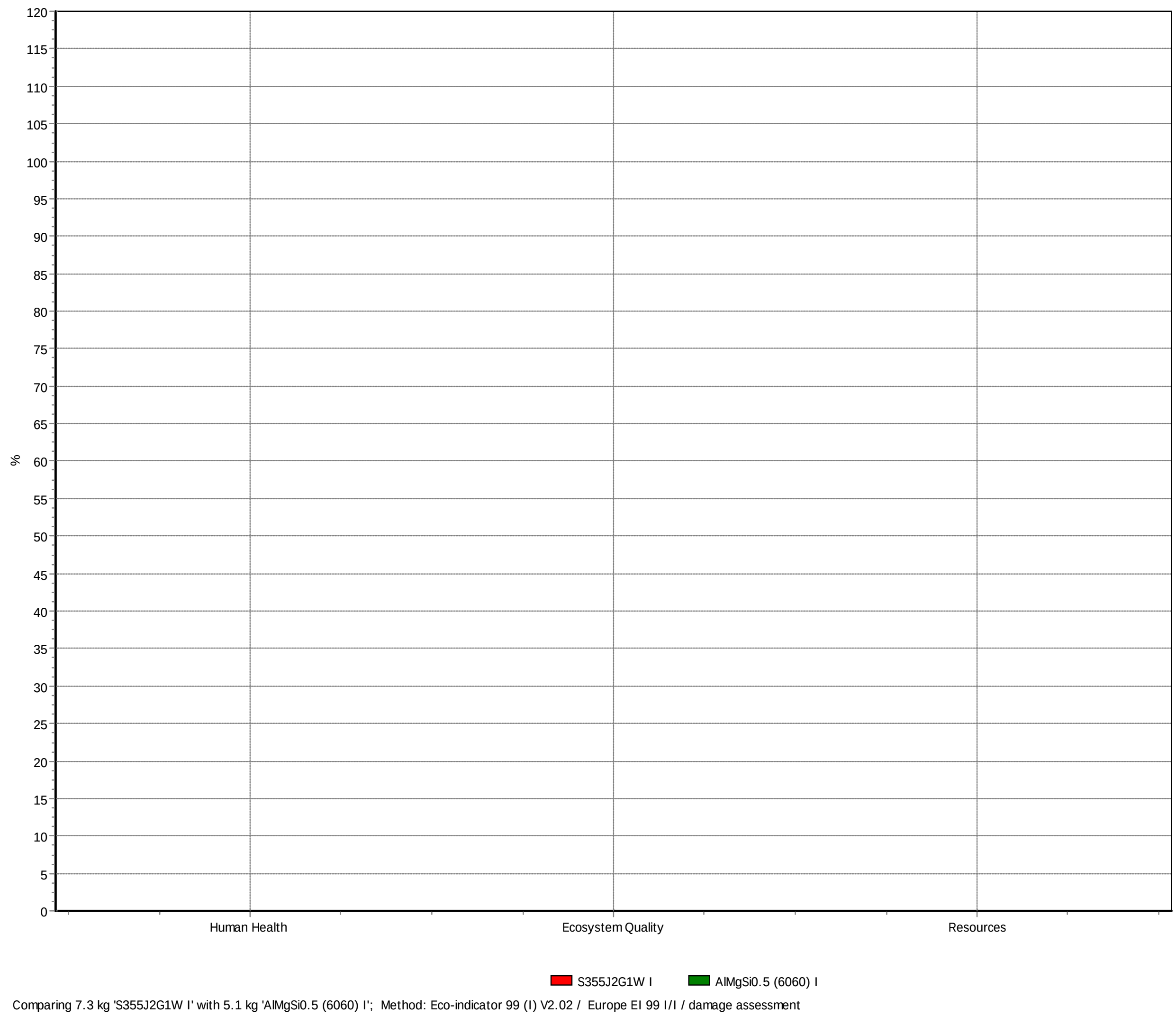
Relative Impacts on the Environment

Comparing 7.3 kg 'S355J2G1W I' with 5.1 kg 'AlMgSi0.5 (6060) I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/I / characterization

Overall Emissions Comparison



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



Manufacturing Process Selection Assignment

The current Xebra electric-hydraulic prototype is designed as a proof of concept, to show that integrating hydraulic regenerative braking and launch systems into electric vehicles will in fact increase their range and efficiency. Due to unforeseen problems with the electric motor controller and a recirculation valve in the regenerative braking system, we were unable to test the overall range of our prototype. Since we are uncertain if our design was able to increase the range of the vehicle, it is unclear if mass producing this concept makes sense. Assuming that our design shows an increase in the range after correcting the problems with the electric motor controller and the regenerative system, a cost-benefit analysis should be conducted to determine if we should proceed with mass production.

Assuming that mass production of our design is logical, it would make sense to equip every electric Xebra vehicle with a hydraulic launch and regenerative braking system. Based on the 1,000 Xebra vehicles that have been produced over the last three years we have estimated that 330 electric-hydraulic hybrid Xebra vehicles should be produced per year. [17] If we assume that this production will continue for three years until a new hybrid model is released, a total of 1000 vehicles should be produced.

As stated in the material selection assignment, AISI 1020 steel was chosen for manufacturing the accumulator mounting beams because of its low cost, and the fact that the beams are to be welded to a steel frame. After determining the appropriate material, we used CES software to determine the optimal manufacturing processes. We determined that the optimal process will be to use ceramic mold casting. In this process, molten metal is poured into a ceramic mold. After the metal cools, the part is then removed from the mold, and the process is repeated. This process is ideal for these accumulator mounting beams because of the batch size. According to the CES software, this process is appropriate for a batch size ranging from 50-5,000 units, which matches well with the 4,000 mounting beams we would need to make 1,000 Xebra hybrid vehicles. This process also is ideal because it allows you to make complex, hollow, 3D geometries. Therefore, we could complete the entire component in one step as opposed to other processes which would require further machining after the initial geometry is created.

If we were working in cooperation with ZAP, the manufacturer of the Xebra, it would make more sense to design a completely different frame for the Xebra rather than welding accumulator mounting beams to the existing frame. If this were the case, we would most likely us roll forming to create the steel tubes for the frame. In this process, sheet metal is fed continuously through a series of shaped rollers until the desired geometry is achieved. This would result in an unclosed seam on one side of the tube. This seam would then be welded closed. The beams would then be cut to length and welded together to create the frame. This method would be much more cost effective than separately manufacturing accumulator mounting beams.

As a result of the material selection assignment, we also chose to explore the manufacturing processes associated with drive train sprockets. Using CES software, we determined that these sprockets should be made from AISI 1040 steel. Based on this material selection we again used CES software and determined that the ideal manufacturing processes for these sprockets is powder metal forging. This process is ideal because it allows for very complex geometries with high tolerances. This process is more expensive than casting. However, with this process, it is possible to eliminate the need for any finishing machining operations, which would be necessary if a casting process were used. For this reason, powder metal forging is the optimal choice for manufacturing the drive train sprockets.

Appendix D: QFD

System QFD

Project:	Electric-Hydraulic Hybrid Performance Enhancement
Date:	date
	Input areas are in yellow

#	Technical Requirements	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
1	Turning Radius (-)																				
2	Weight of Vehicle (-)																				
3	Power of Electric Motor (-)		3																		
4	Torque of Hydraulic Motor (+)		3																		
5	Volume of Tanks (+)		-3																		
6	Time Taken to Fill Tanks (-)					-9															
7	Max Tank Pressure (-)				-9	-9	3														
8	Time to Reach Top Speed (-)		3	3	-9	3		-9													
9	Gear Ratio			9	3				9												
10	Initial Tank Pressure (+)				9	-3	-3	-9	9												
11	Flow Rate of Slow Fill Pump (+)						9	1			-1										
12	Flow Rate of Accumulators (+)				9		9		9		9										
13	Temperature of Electrical System (-)			9						-9											
14	Layout of Components		1			-9	1		1	-3				3							
15																					
16																					
17																					
18																					
19																					
20																					

Survey Legend

A	Current product
B	Competitor B
C	Competitor C

Technical Requirements / **Customer Opinion Survey**

#	Customer Needs	Customer Weights	Kano Type	Turning Radius (-)	Weight of Vehicle (-)	Power of Electric Motor (-)	Torque of Hydraulic Motor (+)	Volume of Tanks (+)	Time Taken to Fill Tanks (-)	Max Tank Pressure (-)	Time to Reach Top Speed (-)	Gear Ratio	Initial Tank Pressure (+)	Flow Rate of Slow Fill Pump (+)	Flow Rate of Accumulators (+)	Temperature of Electrical System (-)	Layout of Components							1 Poor	2	3 Acceptable	4	5 Excellent
1	Increase Top Speed to 45mph	10		1	3	9	9	9		9	3	9	9	3	9	3									A	B		C
2	Prevent Overheating	10			1	9					3	3				9	3								C	AB		
3	Transferable to Future Semesters	9		1	1	1	1	9	1			9		3	1	3	9								C	B	A	
4	Stability	9		9	9		3				3	3					3									etc.		
5	Maintain Previous Acceleration	7			9	3	9	9	1	9	9	9	9	3	9	3												
6	Safety Factor ≥ 3	7		3	1		3	3	3	9		3	9	1	3	9	3											
7	Maintain Current Range	7			9	9	3	9	3	3	3	9	3	9	9	3												
8	Reliability	5				3	3			9	1	3	9	1	9	9	3											
9	Battery Output Must not Exceed 1kW	4			1	9					3	9		3		9												
10	Aesthetics	3						3									9											
11	Must Drive Comfortably	2		9	1		9		3		9	9					3											
12																												
13																												
14																												
15																												
		Raw score		139	269	324	264	327	64	282	206	444	282	165	291	333	207											
		Scaled		0.31	0.61	0.73	0.59	0.74	0.14	0.64	0.46	1	0.64	0.37	0.66	0.75	0.47											
		Relative Weight		4%	7%	9%	7%	9%	2%	8%	6%	12%	8%	5%	8%	9%	6%											
		Rank		13	8	4	9	3	14	6	11	1	6	12	5	2	10											

Appendix E: Battery Specification Sheet

Deka DOMINATOR®

GEL MARINE/RV BATTERIES

NON-SPILLABLE by DOT (Department of Transportation), ICAO (International Commercial Airline Organization) and IATA (International Airline Transport Association) definitions.

The Dominator vs. Conventional Liquid Acid Batteries

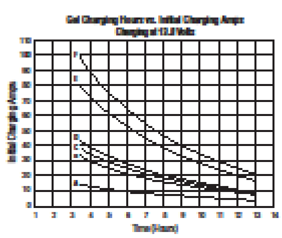
Deka introduces a marine battery second to none. The Dominator contains electrolyte that is permanently locked in a thixotropic gel, instead of conventional and messy liquid acid. Our gelled electrolyte batteries offer many significant advantages over conventional liquid acid batteries. Because there is no liquid electrolyte to leak or spill out of the battery, the Dominator can be easily installed in hard-to-reach locations. Because it has gelled electrolyte, there is no need to check fluid levels. The Dominator's gelled construction also eliminates vibration damage to plates. The Dominator even works under water.

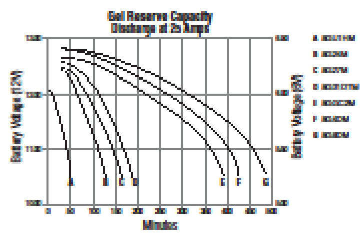
Potential Applications for the Deka Dominator Marine Battery

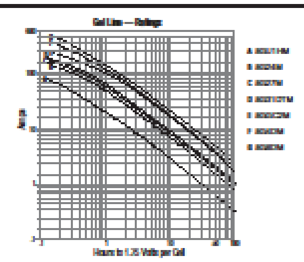
Buoy Lighting	Engine Starting	Safety Lighting
Communication Equipment	House Power	Security Systems
Control Equipment	Marine Equipment	Ships' Batteries
Emergency Lighting	Navigation Beacons	Trolling Motors
Emergency Power Supply Systems	Radar Stations	Other Deep-Discharge Applications
	Radio Relay Stations	

Features and Benefits

100% maintenance-free…	No need to check electrolyte levels. Install it and forget it.
Ideal for engine starting and/or trolling…	Great multi-purpose battery.
Faster recharging…	For quicker turnaround time.
Longer life…	Resists damaging over-discharges.
Longer operating/trolling time…	Runs considerably longer than comparable wet batteries.
Compu-cast grids and computer-controlled oxide…	For maximum durability, power and life.
Sealed, recombinant construction with gelled electrolyte…	Eliminates dangerous spills, gassing and terminal corrosion (unless severely overcharged).
Self-discharge rate of less than 2% per month (at 68°F)…	Recharges to full power, even if left discharged for months.
Thick consistency of gelled electrolyte and tight-pack construction…	Eliminates the damaging effects of vibration.
Easy to install…	Convenient for hard-to-reach connection and hook-up.
Over 250 quality control checks…	Ensure highest quality, performance and reliability.
Broad line of sizes and types…	For all your marine power needs.
Made in U.S.A…	Your assurance of quality.







Type No.	Foot Notes	Volts	CCA @ 0°F	Res. Capacity @ 80°F	Ref. CA @ 32°F	Minutes Discharged at* 75 Amps	50 Amps	25 Amps	15 Amps	8 Amps	5 Amps	Ampere Hour Capacity* 20 Hr. Rate	6 Hr. Rate	3 Hr. Rate	1 Hr. Rate	Approx. Wt. Lbs. (Kgs.)	Dimensions In (mm) L	W	H
8GU1HM	HZ	12	175	44	250	5	12	47	90	190	325	31.6	28.0	25.5	21.0	23.4 (10.6)	8 5/16 (211)	5¼ (133)	7⅛ (181)
8G24M	HQ	12	410	132	575	33	57	140	245	500	845	73.6	64.5	59.3	48.5	52.5 (23.8)	10⅞ (276)	6¾ (171)	9⅞ (251)
8G27M	HQ	12	505	160	700	40	70	170	300	605	1000	86.4	76.0	69.9	57.0	63.2 (28.7)	12¾ (324)	6¾ (171)	9⅞ (251)
8G31DTM	HSX	12	550	180	780	51	84	190	335	690	1160	97.6	85.2	78.6	64.5	70.0 (31.7)	12 15/16 (329)	6¾ (171)	9⅜ (238)
8GGC2M	G	6	585	345	850	92	155	375	680	1360	2200	180.0	155.0	136.0	99.0	68.4 (31.0)	10¼ (260)	7⅛ (181)	10⅞ (276)
8G4DM	HS	12	970	375	1245	105	175	395	685	1385	2300	183.0	160.2	147.6	122.0	127.0 (57.5)	20¾ (527)	8½ (216)	10 (254)
8G8DM	HS	12	1150	475	1470	135	220	500	890	1750	3000	225.0	198.0	181.8	150.0	157.0 (71.1)	20¾ (527)	11 (279)	10 (254)

* NOMINAL

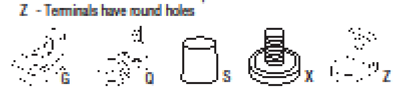
ALL RATINGS ARE AFTER 15 CYCLES AND CONFORM TO B.C.I. SPECIFICATIONS.

All batteries manufactured in polypropylene cases and covers except where noted.

IMPORTANT CHARGING INSTRUCTIONS: WARRANTY VOID IF OPENED OR IMPROPERLY CHARGED. Do not install in a sealed container. Constant under or overcharging will damage any battery and shorten its life! Use a good constant potential, voltage-regulated charger.
For 12-volt batteries, charge to at least 13.8 volts but no more than 14.1 volts at 68°F (20°C).
For 6-volt batteries, charge to at least 6.9 volts but no more than 7.05 volts at 68°F (20°C).
The **open circuit voltage** of a fully charged 12-volt battery is 12.8V at 68°F (20°C). However, as the battery charges, the building internal pressure (voltage) causes resistance to the charge. Therefore, the **on-charge voltage** must be higher (at least 13.8V) to overcome this internal pressure (voltage) during charging.

FOOTNOTES:
- G - Offset post w/ horizontal hole, stainless steel 5/16" bolt & hex nut
- H - Includes handles
- Q - Dual terminals with SAE post and 3/8" POS., 5/16" NEG. stainless steel studs & wing nuts
- S - SAE "automotive type" post
- X - 3/8" x 16 stainless steel stud posts
- Z - Terminals have round holes

UL Recognized Component MH17215



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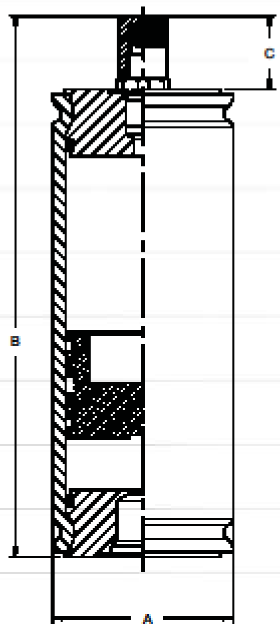
E.P.M. Form No. 0909 Rev. 2/06

Appendix F: Parker Hannifan Accumulator Specifications

ACP SERIES ACCUMULATOR DIMENSIONS

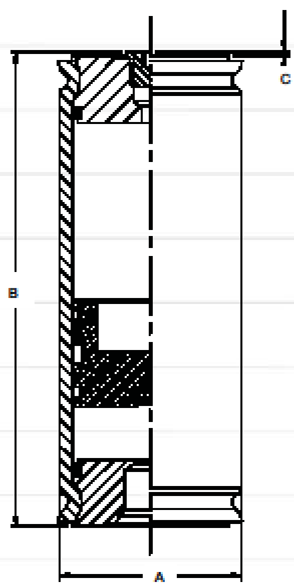
Piston Accumulator with Gas Valve

Model No.	Gas Volume Liters	Gas Volume Cu. In.	A (In)	B (In)	C (In)	Hydraulic Port
ACP04AA008R1KTB	.08	4.88		6.09		SAE #6
ACP04AA016R1KTB	.16	9.76	1.75	8.85	1	9/16-8 THD
ACP04AA032R1KTB	.32	19.52		14.37		
ACP05AA016E1KTC	.16	9.76		6.97		
ACP05AA032E1KTC	.32	19.52		10.00		SAE #8
ACP05AA050E1KTC	.50	30.51	2.38	13.41	1	3/4-16 THD
ACP05AA075E1KTC	.75	45.76		18.15		
ACP05AA100E1KTC	.95	57.97		21.94		
ACP08AA032E1KTI	.32	19.52		7.74		
ACP08AA050E1KTI	.50	30.51		9.30		
ACP08AA075E1KTI	.75	45.76		11.46		SAE #10
ACP08AA100E1KTI	.95	57.97	3.56	13.18	1	7/8-14 THD
ACP08AA150E1KTI	1.50	91.53		17.93		
ACP08AA200E1KTI	2.00	122.04		22.25		
ACP08AA300E1KTI	3.00	183.07		30.89		
ACP10AA075E1KTD	.75	45.76		9.47		
ACP10AA100E1KTD	.95	57.97		10.43		
ACP10AA150E1KTD	1.50	91.53		13.06		SAE #12
ACP10AA200E1KTD	2.00	122.04	4.75	15.45	1	1 1/16-12 THD
ACP10AA300E1KTD	3.00	183.07		20.24		
ACP10AA400E1KTD	4.00	244.09		25.02		
ACP10AA600E1KTD	6.00	366.14		34.59		
ACP10AA800E1KTD	8.00	488.19		44.16		



Piston Accumulator without Gas Valve

Model No.	Gas Volume Liters	Gas Volume Cu. In.	A (In)	B (In)	C (In)	Hydraulic Port
ACP04AD008R1KTB	.08	4.88		5.17		SAE #6
ACP04AD016R1KTB	.16	9.76	1.75	7.93	.08	9/16-8 THD
ACP04AD032R1KTB	.32	19.52		13.45		
ACP05AD016E1KTC	.16	9.76		6.05		
ACP05AD032E1KTC	.32	19.52		9.08		SAE #8
ACP05AD050E1KTC	.50	30.51	2.38	12.49	.08	3/4-16 THD
ACP05AD075E1KTC	.75	45.76		17.23		
ACP05AD100E1KTC	.95	57.97		21.02		
ACP08AD032E1KTI	.32	19.52		6.82		
ACP08AD050E1KTI	.50	30.51		8.38		
ACP08AD075E1KTI	.75	45.76		10.54		SAE #10
ACP08AD100E1KTI	.95	57.97	3.56	12.26	.08	7/8-14 THD
ACP08AD150E1KTI	1.50	91.53		17.01		
ACP08AD200E1KTI	2.00	122.04		21.33		
ACP08AD300E1KTI	3.00	183.07		29.97		
ACP10AD075E1KTD	.75	45.76		8.55		
ACP10AD100E1KTD	.95	57.97		9.51		
ACP10AD150E1KTD	1.50	91.53		12.14		SAE #12
ACP10AD200E1KTD	2.00	122.04	4.75	14.53	.08	1 1/16-12 THD
ACP10AD300E1KTD	3.00	183.07		19.32		
ACP10AD400E1KTD	4.00	244.09		24.10		
ACP10AD600E1KTD	6.00	366.14		33.67		
ACP10AD800E1KTD	8.00	488.19		43.24		



ACP SERIES ACCUMULATOR ORDER INFORMATION

How to Order Series ACP Series Piston Accumulators

Piston accumulators can be specified by using the symbols in the chart below to develop a model number. Select only those symbols that represent the features desired, and place them in the sequence indicated by the example at the top of the chart.

AC Series	Piston Type	Model	Type of Construction	Options	Capacity	Working Pressure	Design Number	Seal Compound	Port Size
AC	P	05	A	A	075	E	1	K	TC

Model	
04	40mm (1.5") Bore
05	50mm (2") Bore
08	80mm (3") Bore
10	100mm (4") Bore

Type of Construction	
A	Standard

Options	
A	Gas Valve (Standard)
D	No Gas Valve

Model/Capacity			
008		0.08 liter	5 cu. in.
016	04	0.16 liter	10 cu. in.
032		0.32 liter	20 cu. in.
016		0.16 liter	10 cu. in.
032		0.32 liter	20 cu. in.
050	05	0.50 liter	31 cu. in.
075		0.75 liter	46 cu. in.
100		0.95 liter	58 cu. in.
032		0.32 liter	20 cu. in.
050		0.50 liter	31 cu. in.
075		0.75 liter	46 cu. in.
100	08	0.95 liter	58 cu. in.
150		1.50 liter	92 cu. in.
200		2.00 liter	122 cu. in.
300		3.00 liter	183 cu. in.
075		0.75 liter	46 cu. in.
100		0.95 liter	58 cu. in.
150		1.50 liter	92 cu. in.
200	10	2.00 liter	122 cu. in.
300		3.00 liter	183 cu. in.
400		4.00 liter	244 cu. in.
600		6.00 liter	366 cu. in.
800		8.00 liter	488 cu. in.

Design Number	
1	Standard

Working Pressure	
E	275 Bar (4000 PSI)
R	260 Bar (3800 PSI) 40mm only

Seal Compounds	
K	Nitrile (Std)
H	Hydrogenated Nitrile
E	Fluoroelastomer
D	EPR
S	Special

Ports (see Availability Table)	
TB	Female SAE #6
TC	Female SAE #8
TI	Female SAE #10
TD	Female SAE #12
TE	Female SAE #16
AD	Male SAE #12
AE	Male SAE #16
RA	Female BSPP 3/8-19
RB	Female BSPP 1/2-14
RC	Female BSPP 3/4-14
RD	Female BSPP 1-11
LC	Male BSPP 3/4-14
LD	Male BSPP 1-11

Port Availability

Model	Female SAE					Male SAE		Female BSPP				Male BSPP	
	6	8	10	12	16	12	16	3/8	1/2	3/4	1	3/4	1
04	●					●		●	★			●	
05	★	●	★			★	●	★	★	●		★	●
08	★	★	●	★	★	★	●	★	★	★	●	★	●
10	★	★	★	●	★	★	●	★	★	★	●	★	●

● = Standard
★ = Optional

⚠ WARNING

FAILURE OR IMPROPER SELECTION OR IMPROPER USE OF THE PRODUCTS AND/OR SYSTEMS DESCRIBED HEREIN OR RELATED ITEMS CAN CAUSE DEATH, PERSONAL INJURY AND PROPERTY DAMAGE.

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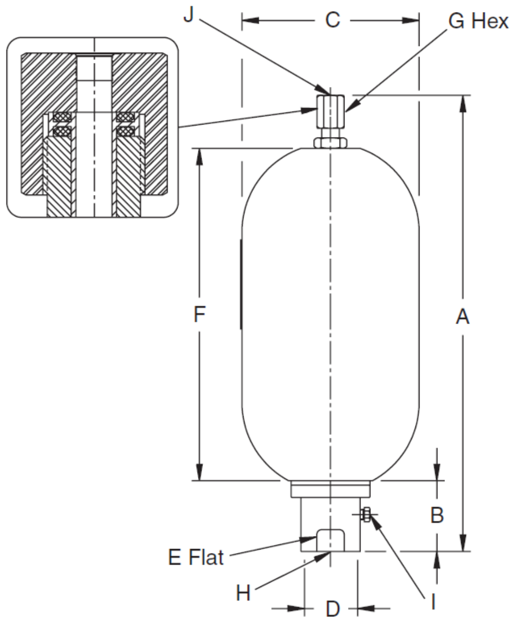
Bulletin No. 1640-ACP
8M 4/01 SL

Bladder Type Accumulators



5000 PSI (345 Bar)[2]

Models	Nominal Size	Gas Volume	Dimensions, in (mm)							Hydraulic/Gas Ports			Weight lbs. (Kg.)
Oil Service Water Service	Gallon (Liters)	cu in (Liters)	A	B	C	D	E	F	G	H (Thread)	I (Thread)	J (Thread)	
BT02B5TT01A1 BT02B5TT01WA1	2.5 (9.45)	556 (9.11)	22.55 (573)	3.62 (92)	9.63 (245)	3.00 (76)	2.88 (73)	16.12 (409)	2.00 (51)	SAE #24 (1-7/8 - 12)	SAE #6 (9/16 - 18)	SAE #12 (1-1/16 - 12)	120 (55)
BT05B5TT01A1 BT05B5TT01WA1	5 (18.9)	1124 (18.42)	34.80 (884)	3.62 (92)	9.63 (245)	3.00 (76)	2.88 (73)	28.36 (720)	2.00 (51)	SAE #24 (1-7/8 - 12)	SAE #6 (9/16 - 18)	SAE #12 (1-1/16 - 12)	200 (91)
BT10B5TT01A1 BT10B5TT01WA1	10 (37.8)	2097 (34.36)	55.30 (1405)	3.62 (92)	9.63 (245)	3.00 (76)	2.88 (73)	48.88 (1242)	2.00 (51)	SAE #24 (1-7/8 - 12)	SAE #6 (9/16 - 18)	SAE #12 (1-1/16 - 12)	335 (152)
BT15B5TT01A1 BT15B5TT01WA1	15 (56.7)	3267 (53.54)	76.80 (1951)	3.62 (92)	9.63 (245)	3.00 (76)	2.88 (73)	70.38 (1788)	2.00 (51)	SAE #24 (1-7/8 - 12)	SAE #6 (9/16 - 18)	SAE #12 (1-1/16 - 12)	485 (220)

2) Note: Available with 6600 PSI (455 Bar) Appendix 22

Appendix G: Hydraulic Motor and Pump Specifications sheet

PGP/PGM 517 Specifications

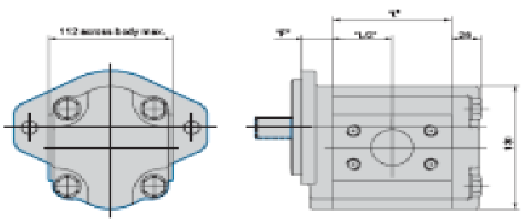
Description	Code	0140	0160	0190	0230	0250	0280	0330	0360	0380	0440	0520
Displacements	cm^3/rev in^3/rev	14 0.85	16 0.98	19 1.16	23 1.40	25 1.53	28 1.71	33 2.01	36 2.20	38 2.32	44 2.68	52 3.17
Continuous Pressure	bar psi	250 3625	250 3625	250 3625	250 3625	250 3625	250 3625	250 3625	250 3625	250 3625	220 3190	200 2900
Intermittent Pressure	bar psi	275 3988	275 3988	275 3988	275 3988	275 3988	275 3988	275 3988	275 3988	255 3698	220 3190	215 3118
Minimum Speed @Max. Outlet Pressure	rpm	500	500	500	500	500	500	500	500	500	500	500
Maximum Speed @ 0 Inlet & Max. Outlet Pressure	rpm	3400	3400	3300	3300	3100	3100	3100	3000	3000	2800	2600
Pump Input Power @ Max. Pressure and 1500 rpm	kW HP	9.6 12.87	11 14.75	13.1 17.57	15.8 21.19	17.2 23.07	19.3 25.88	22.7 30.44	24.6 32.99	26.1 35.00	27 36.21	28.6 38.35
Dimension "L"	mm in	68.3 2.69	70.3 2.77	73.3 2.89	77.4 3.05	79.4 3.13	82.4 3.24	87.5 3.44	90.5 3.56	92.5 3.64	98.6 3.88	106.7 4.20
Approximate Weight*	kg LB	7.92 17.50	8 17.68	8.12 17.95	8.29 18.32	8.37 18.50	8.5 18.79	8.7 19.23	8.83 19.51	8.91 19.69	9.16 20.24	9.49 20.97

*Single pump with Shaft End Cover H3 and non ported Port End Cover

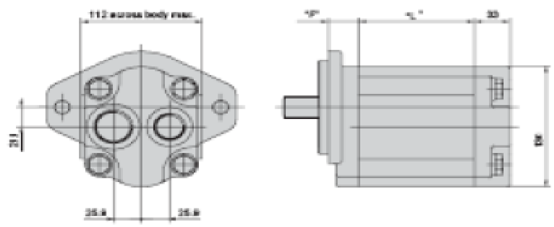
Motor Pump

PGP/PGM 517 Dimensions

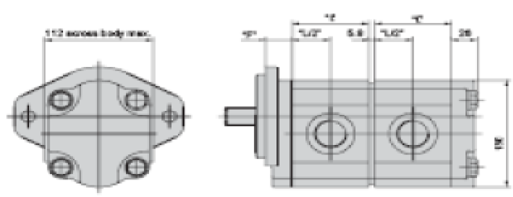
Single Unit PGP/PGM 517



Single Unit PGP/PGM 517 with rear ports



Tandem Unit PGP/PGM 517



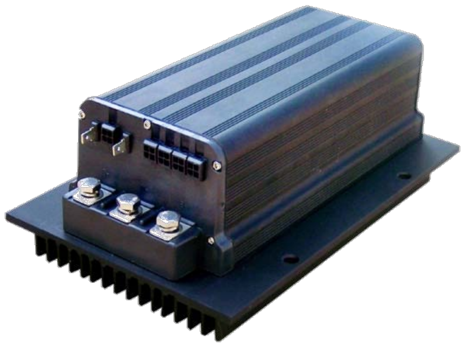
NOTE:
Dimension "F" see shaft end covers on page 21
Dimension "L" see table above

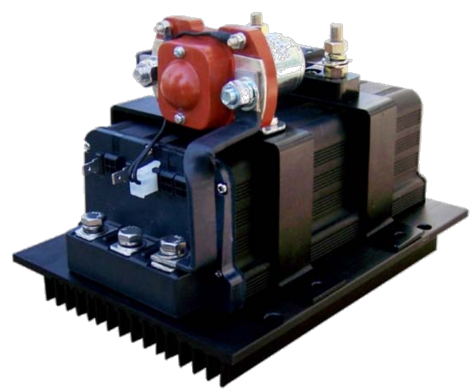
Notes:
1. Dimensions are in millimeters.
2. Dimensions are nominal except where noted.
3. Subscript and/or superscript numbers are tolerances.
4. To convert from millimeters to inches, divide millimeters by 25.4.

$$\text{Torque [Nm]} = \frac{\text{Displacement [cm}^3\text{/rev] x Pressure [bar]}}{57.2}$$

Appendix H: Specification of the Separately Excited Motor

Produced by Zibo Boshan Super Motor co.





Description

Sep series programmable controllers provide smooth and seamless regenerative control of separately excited motors. An advanced MOSFET power section, combined with a sophisticated microprocessor provides very high efficiency, quiet operation and reduces motor and battery losses.

Sep series controllers are designed for widely application in golf cars, sightseeing vehicles, hunting buggies, electric vehicles, heavy-duty trucks, electric yachts and other kinds of utility vehicles.

Feature Overview

- ✓ Power MOSFET technology provides smooth, quiet, efficient, and cost-effective operation.
- ✓ No direction contactors are required. Fully electronic direction is adopted, which provides quiet and low error rate operation.
- ✓ Easy to install and maintain. The pre-charge resistor and diode for contactor are built in.
- ✓ Regenerative brake system efficiently shortens the brake distances, increases usable battery energy, and reduces motor heating temperature.
- ✓ “Ramp Restraint” feature provides automatic electronic brake that restricts vehicle movement while in neutral
- ✓ Anti-rollback function provides improved control when brake pedal is released on slopes.
- ✓ Optional over-charge protection function efficiently prevents the battery overcharge and increases its life span.
- ✓ Fully programmable. Adjustable parameters enable custom optimization of speed, torque, and brake control, etc.
- ✓ Optimized program specification enables a convenient and practical vehicle performance adjustment.
- ✓ Mult le optimized systems are set in programmer, which makes vehicle development period much shorter.
- ✓ Extensive fault detection and diagnostic reports are fulfilled by a programmer or buzzer in controller.
- ✓ Extensive system monitors capabilities are well performed by using a programmer.
- ✓ Optional electromagnetic brake output.
- ✓ Sealed package standard: 66.

Type Description

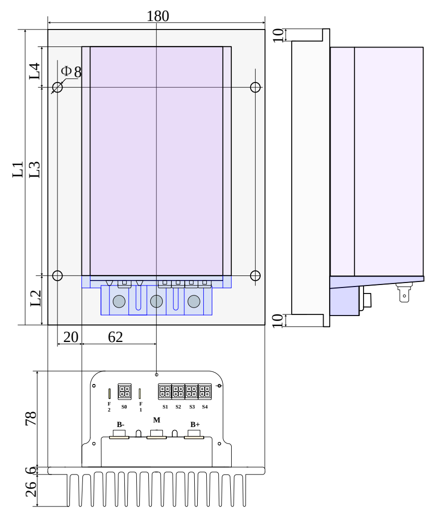
MCSep - 7240

- Maxim current   40 mean 400A
- Voltage grade   72 mean 72V
- Type   Separately excited Motor Controller

Type Table

Type	Battery Voltage	Current	Type
Sep-4840	48V	400A	I
Sep-4865	48V	650A	II
Sep-7230	72V	300A	I
Sep-7240	72V	400A	II
Sep-7265	72V	650A	II
Sep-9630	96V	300A	II
Sep-9640	96V	400A	II
Sep-10830	108V	300A	II
Sep-12030	120V	300A	II
Sep-14420	144V	200A	II
Sep-14430	144V	300A	II

Installation Size



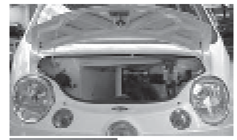
Remarks: According to the difference of the maximum output efficiency, there are two types of dimensions, I and II

Type	L1	L2	L3	L4
I	240	40	153	33
II	280	70	140	59

Appendix I: Excerpt from owner's manual

FRONT & REAR DECK LIDS

The Xebra Sedan is a <u>hatchback</u> with storage capacity in the rear but no storage space inside the front deck lid. Do not attempt to stow, create storage or strap anything in this area. It allows no safe place for foreign objects. It's very important not to this open while the Xebra is turned on. This could result in unexpected injuries.



To open the front deck lid, pull the release handle located on the front passenger's side. Pull gently and the lid latch will release. There is a bar that is intended to hold the deck lid open. Do not force the deck lid upward, this will result in damage to the windshield and put stress on the hinges.

Inside the front deck lid you will find the brake fluid reservoir, the horn, and the windshield fluid tank. In the center you will see the access to the front tire and suspension. It is not recommended that you make any adjustments to braking or suspension systems aside from checking brake and windshield fluids regularly.

To close the front deck lid, remove the support bar and place it into its clip. Close and press the lid down (towards the back) gently until it clicks. Do not slam the front deck lid. Slamming could damage the deck lid clamp. Check to ensure the deck lid closed firmly and properly. If not, release the lid as before and close it again. Pressing down hard on it with a hand or knee may leave an indentation in the deck lid.

To open the hatchback press the release button in the rear of the vehicle and open upwards. Beneath the rear storage area you will find a rear deck lid. To open, you must push the rear seats forward and lift the lid (Fig 1). Inside you will find the motor, the contactors, the battery charger and the motor controller. These are often very hot, even during charging. DO NOT TOUCH them. If you see loose or frayed wires or damage, contact your local service center or dealership. Do not operate the Xebra if any such damage has occurred.

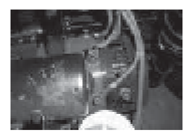
To close the hatchback, lower, close and gently press at center until it clicks. Do not slam it. Rough handling can cause damage to the latch and/or body.

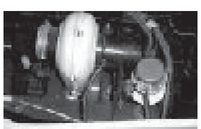
7

The following parts and components are Not intended to be serviced by the user, only a ZAP-trained technician. Any repairs or alterations made to the vehicle **WILL VOID THE WARRANTY!**

DRIVE MOTOR

The Xebra's rear wheels are powered by a 72VDC brushed series wound drive motor rated at 5 KW. Located inside the rear deck lid, this motor has a continuous horsepower output rating of about 5 HP with a peak power somewhat higher. This is sufficient to propel the vehicle at up to 40 MPH on level terrain.



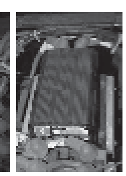


COOLING FAN

The motor is equipped with a temperature-activated cooling fan, located inside the rear deck lid and above the motor. A temperature sensor on the motor turns the fan on at 60 degrees C. Operating the vehicle without the cooling fan working can cause the motor to overheat and could lead to serious damage. If the motor gets too hot (135 degrees C), a light on the dashboard voltage meter will illuminate, alerting you to the problem. Stop the car and allow the motor to cool before continuing. Do NOT use water to cool the motor! It is an electrical component and can be damaged by moisture.

MOTOR CONTROLLER

Located in the rear deck lid and inside the High Voltage Electrical Components Box, large cables connect the battery pack to the motor controller through the power contactors. Its function is to take the input from the accelerator and adjust the speed of the motor accordingly. It is programmed by ZAP with a general-purpose power curve, and is not user serviceable.



BATTERY CHARGER

Located inside the rear deck lid, the battery charger regulates the power going into the traction battery pack (72 volt) from the 110VAC charging cord. It will shut off automatically when charging is complete.

<u>To determine proper function of charger, refer to the **Checking Battery Voltage** section.</u>

8

Appendix J: Functional Decomposition of Design Problem

Design Problem: To increase the top speed of the Xebra vehicle from 35 to 45 mph safely while also maintaining the vehicles current range and efficiency.

List Form Functional Decomposition:
Level 1:
1. Top Speed of 45 mph
2. Range of 15 Miles
3. Stability
4. Operation of electric system

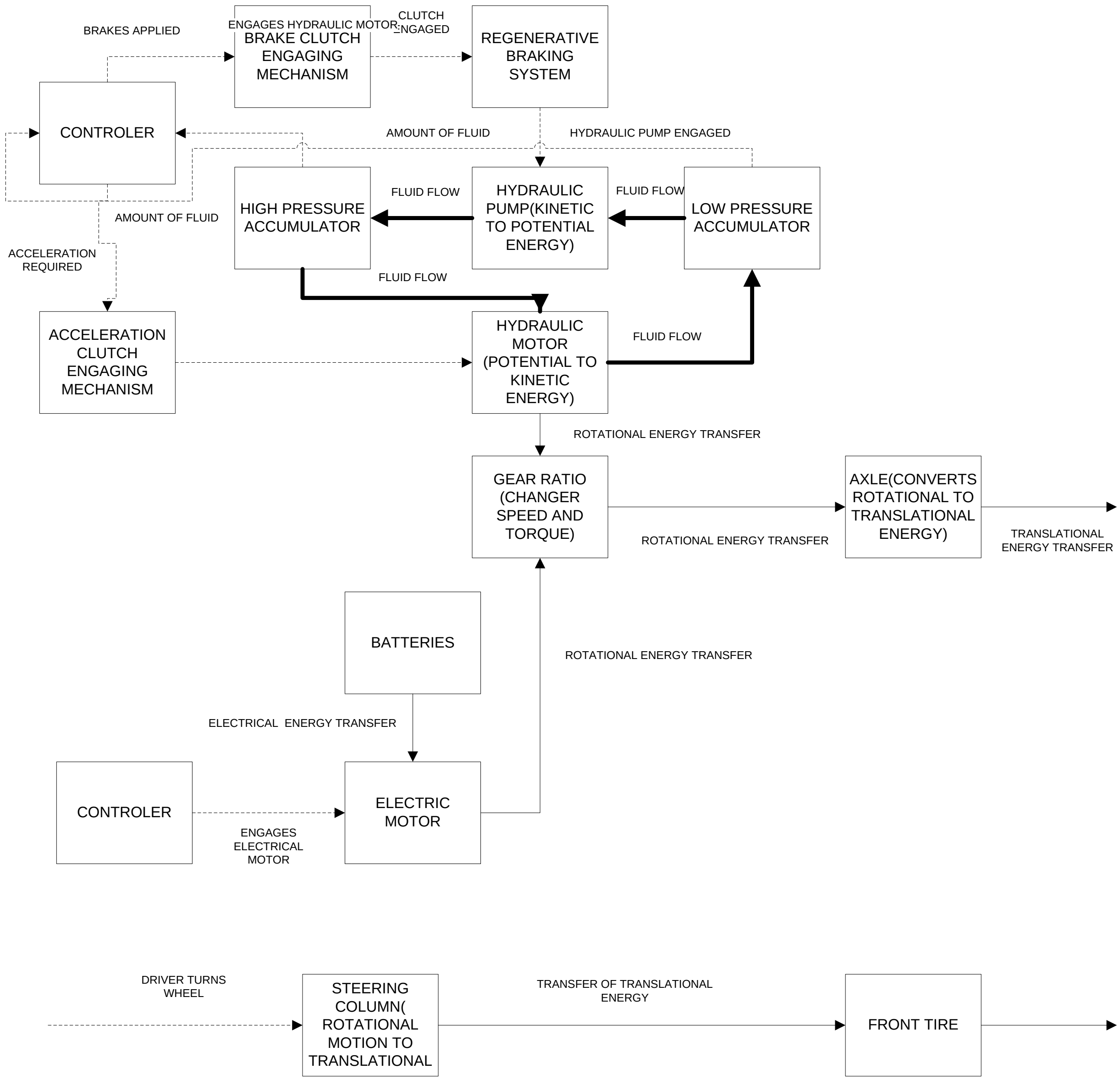
Level 2:
1. Top speed of 45 mph
    - 1.1. Regenerative Breaking
    - 1.2. Launch Assist
    - 1.3. Gear Ratios
    - 1.4. Electric Motor
2. Range of 15 miles
    - 2.1. Launch Assist
3. Stability
    - 3.1. User input (Turning the wheel)
    - 3.2. Motion transferred from steering column to wheels
4. Operation of electrical System
    - 4.1. Battery
    - 4.2. Safe operating temperature
    - 4.3. Electric Motor

Level 3:
1. Top speed of 45 mph
    - 1.1. Regenerative Breaking
        - 1.1.1. Application of brakes
        - 1.1.2. Hydraulic Clutch engaged
        - 1.1.3. Kinetic energy input from axle to motor
        - 1.1.4. Fluid is pumped from low to high pressure accumulator
        - 1.1.5. Fluid is stored for use at a later time
    - 1.2. Launch Assist
        - 1.2.1. Depression of accelerator pedal
        - 1.2.2. Hydraulic Clutch engaged
        - 1.2.3. Fluid is pumped from high to low pressure accumulator
        - 1.2.4. Generation of kinetic energy by the Hydraulic motor
        - 1.2.5. Kinetic energy generated by motor transferred to axle via chain
    - 1.3. Gear Ratios
        - 1.3.1. Rotational energy transferred back and forth between the axle and motors
        - 1.3.2. Increase/Decrease rotational speed and torque that is transferred
    - 1.4. Electric Motor
        - 1.4.1. Actuated when cruising speed is achieved
        - 1.4.2. Electrical energy is supplied by the batteries

        1.4.3. Converts electric potential to mechanical work
        1.4.4. Transfers kinetic energy to axle via the gears

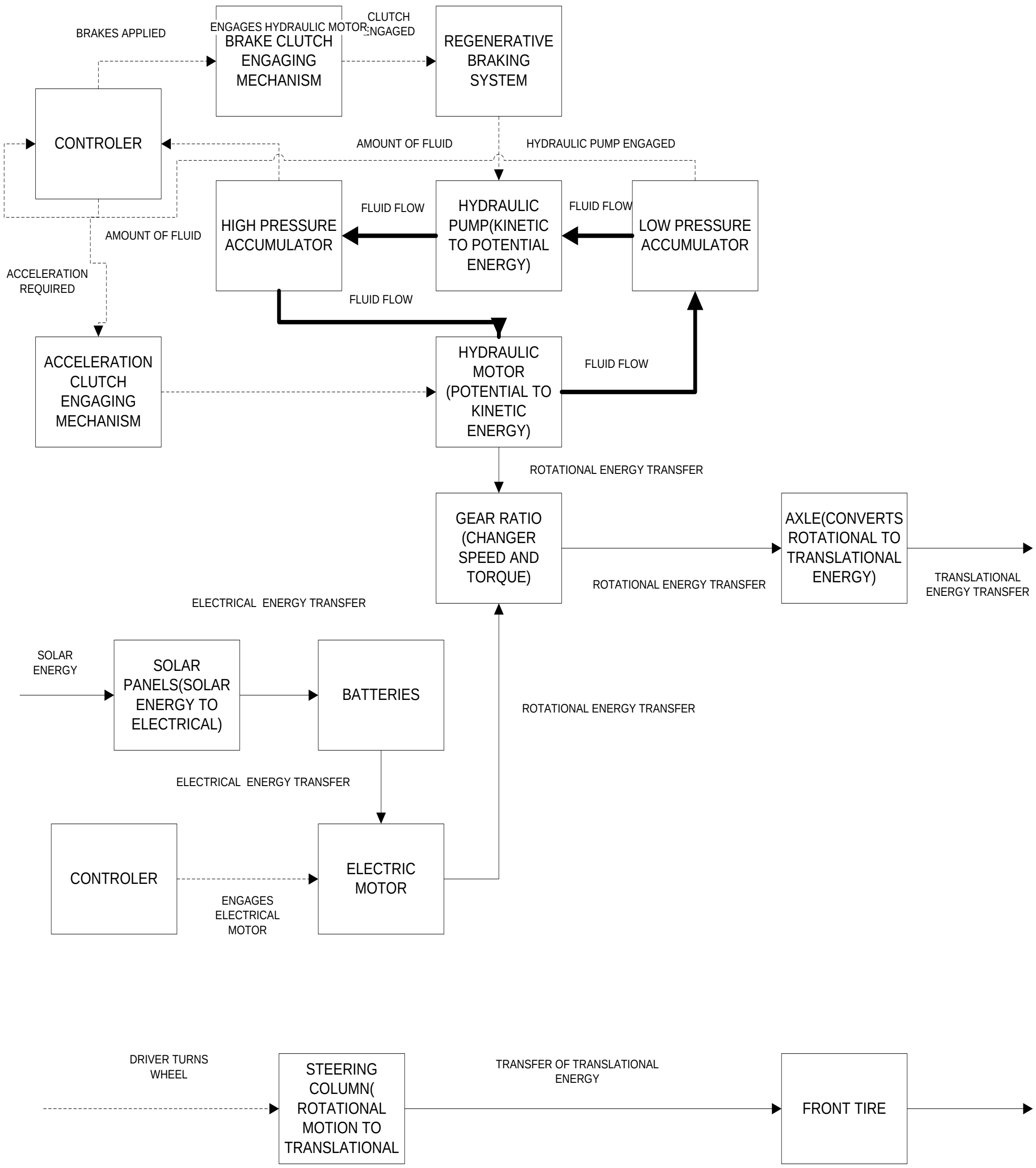
2. Range of 15 miles
    2.1. Launch Assist
        2.1.1. Depression of accelerator pedal
        2.1.2. Hydraulic Clutch engaged
        2.1.3. Fluid is pumped from high to low pressure accumulator
        2.1.4. Generation of kinetic energy by the Hydraulic motor
        2.1.5. Kinetic energy generated by motor transferred to axle via chain
3. Stability
    3.1. Steering Column
        3.1.1. Input from driver
        3.1.2. Steering column rotates
        3.1.3. Rotational motion converted to transverse motion
4. Operation of electrical System
    4.1. Battery
        4.1.1. Electric Energy input while Xebra is charging
        4.1.2. Input from controller
        4.1.3. Storage of electric energy
        4.1.4. Transfer stored energy to electric motor
    4.2. Safe operating temperature
        4.2.1. Temperature measured by thermocouple
        4.2.2. Information transferred to controller
        4.2.3. Shuts down/ Controls current flowing through system
    4.3. Electric Motor
        4.3.1. Actuated when cruising speed is achieved
        4.3.2. Electrical energy is supplied by the batteries
        4.3.3. Converts electric potential to mechanical work
        4.3.4. Transfers kinetic energy to axle via the gears
        4.3.5. Thermocouple on motor transfers information to controller

Flow Form Functional Decomposition:

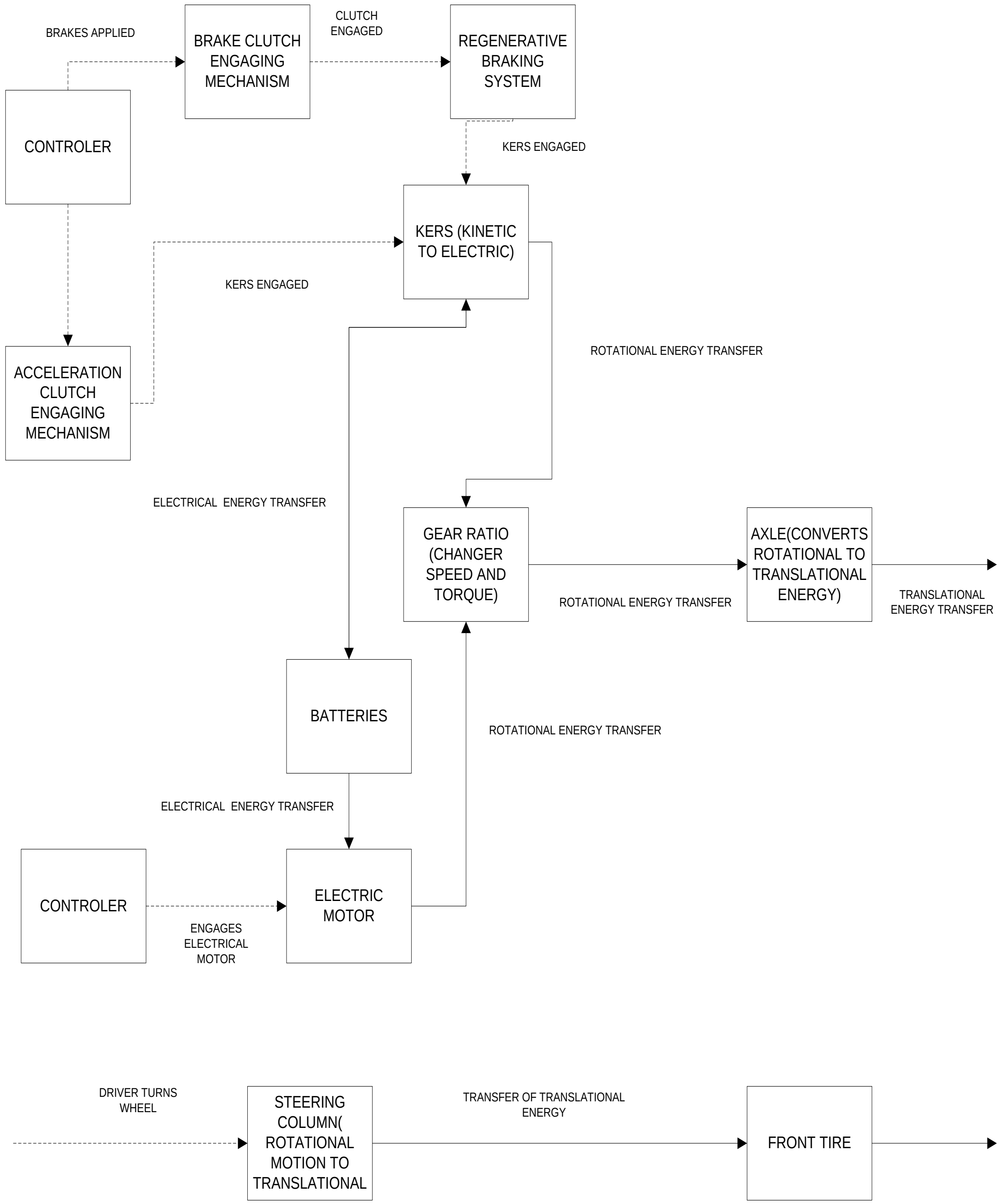


Appendix K: Functional Decomposition of Different concepts

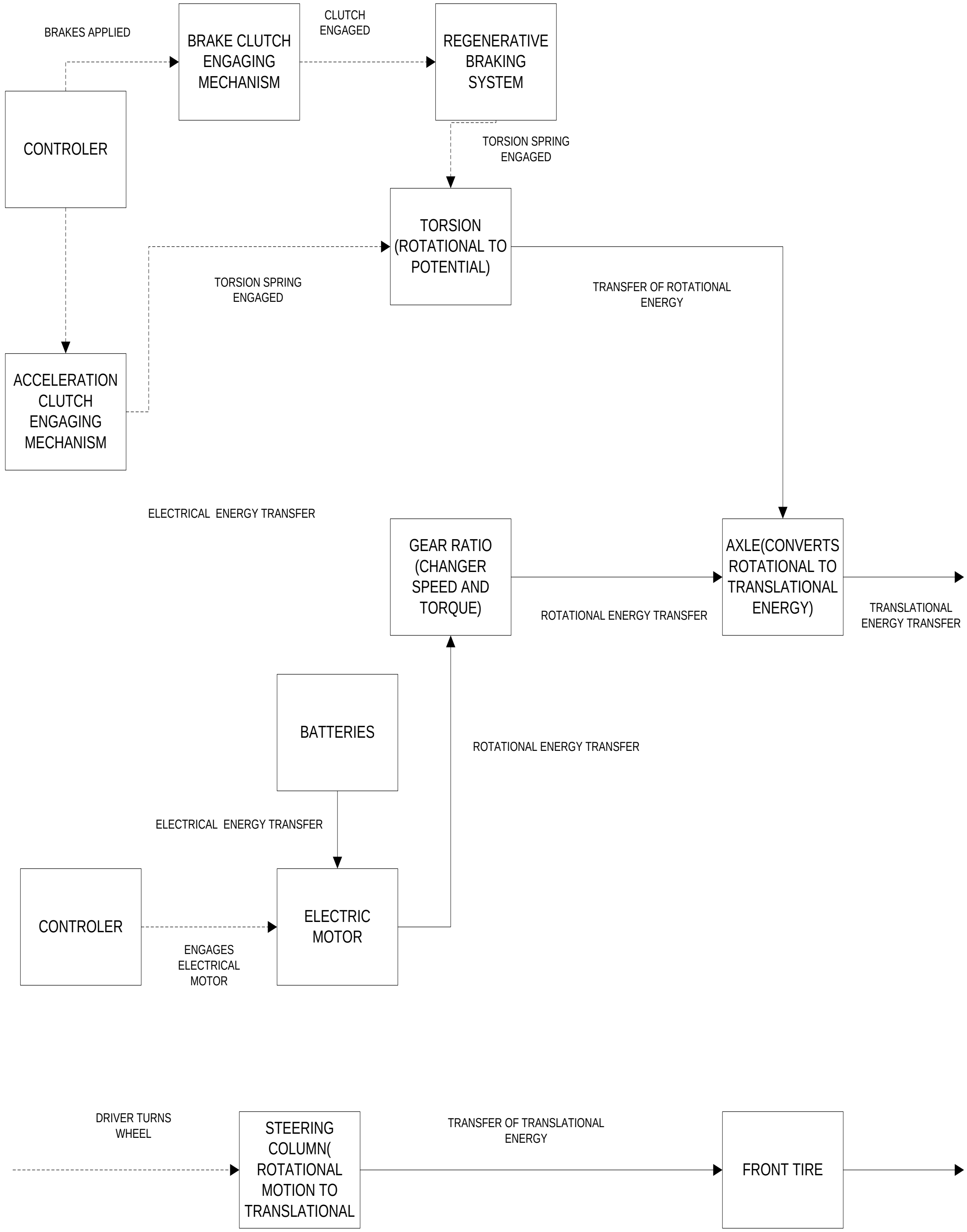
1. Solar Panels :



2. Flywheel:


BRAKES APPLIED
BRAKE CLUTCH ENGAGING MECHANISM
CLUTCH ENGAGED
REGENERATIVE BRAKING SYSTEM
CONTROLER
KERS ENGAGED
KERS (KINETIC TO ELECTRIC)
KERS ENGAGED
ROTATIONAL ENERGY TRANSFER
ACCELERATION CLUTCH ENGAGING MECHANISM
ELECTRICAL ENERGY TRANSFER
GEAR RATIO (CHANGER SPEED AND TORQUE)
ROTATIONAL ENERGY TRANSFER
AXLE(CONVERTS ROTATIONAL TO TRANSLATIONAL ENERGY)
TRANSLATIONAL ENERGY TRANSFER
BATTERIES
ROTATIONAL ENERGY TRANSFER
ELECTRICAL ENERGY TRANSFER
CONTROLER
ENGAGES ELECTRICAL MOTOR
ELECTRIC MOTOR
DRIVER TURNS WHEEL
STEERING COLUMN(ROTATIONAL MOTION TO TRANSLATIONAL
TRANSFER OF TRANSLATIONAL ENERGY
FRONT TIRE

3. Torsional Spring:



Appendix L: Concept generation

Concept	Classification	Description
Increase accumulators	Using Hydraulics	More accumulators would allow us to store more hydraulic energy
Increase wheel size	Using Hydraulics	This would increase circumference and hence increase the translational speed of the car keeping the rotational speed constant.
Larger hydraulic motor	Using Hydraulics	Would produce more power and torque for the same amount of fluid.
Larger slow-fill pump	Using Hydraulics	Would result in quicker filling of the accumulators leading to more hydraulic energy
Solar Panels	Alternative sources	Would charge the batteries using solar energy
Change working fluid	Using Hydraulics	Might increase the amount of energy sotred per unit volume.
Change gear ratios	Using Hydraulics	Would increase the rotational velocity of wheels, while keeping rotational speed of motor constant
Add electrical cooling system	Using Hydraulics	Mitigate heating problems there by allowing us to overclock the motor
Add hydraulic heating system	Using Hydraulics	Increase energy stored in fluid
Larger hydraulic motor	Using Hydraulics	More fluid stored so more energy is stored
Run electrical motor and hydraulic motor while accelerating	Using Hydraulics	
Increase pressure in accumulators	Using Hydraulics	Increases energy stored
Add combustion engine	Alternative sources	Increases power output and hence speed
CVT	Alternative sources	Variable transmission help in acceleration and in increasing speed
Variable displacement motors	Alternative sources	More efficient use of hydraulic fluid.
More or better batteries	Alternative sources	More electrical energy to reach top speed.
Wind up spring	Alternative sources	Stores kinetic energy as potential energy
Inertial storage	Alternative sources	Stores kinetic energy as potential energy
Streamlining	Alternative sources	Reduce drag, there by increases the attainable top speed
Use sails	Alternative sources	Harness wind energy
Pedal powered	Alternative sources	Harness human power
Rockets	Alternative sources	Use of rockets to accelerate to desired speed

Methanol	Alternative sources	Use of combustible gas in an internal combustion engine to increase top speed
Gasification reactor	Alternative sources	
Steam power	Alternative sources	Use of heat energy to produce kinetic energy
Fuel Cells	Alternative sources	Can be used to power the electric motor or to charge up the batteries
Decrease weight	Alternative sources	Lighter car would need less force to accelerate to a faster speed
Optimize the operation times of motors	Alternative sources	Runs each system only at their optimal time to increase range and acceleration

Appendix M: Calculations for Torsion Spring and Solar Panel

Solar Panel Spcifications

Model	Max Power	Current at Max Power	Voltage at Max Power	Maximum System Voltage	Open Circuit Voltage	Short Circuit Current	Series Fuse Rating	Dimensions (LxWxD)	Weight
KD 210 GX LPU	210 Watts	7.9 Amps	26.6Volts	600V	33.2V	8.58 Amps	15 Amps	59.1"x39"x1.8"	40 lbs

Solar Panel:
Efficiency falls from 92% to 69%
Electric Motor Power = 5000 watts
So loss of power is = 5000 x (0.92-0.69)= 5000 x 0.23= 1150 W
So number of solar panels required = $\frac{1150}{210}$ = 5.321
So number required approximately = 5.

Torsion Spring:
Approximate Kinetic Energy required to reach top speed:
Mass = 1200 kg
Velocity = 45mph= 20.11m\s

K.E.= $\frac{1}{2}mv^2$

So, K.E. = 242659 J= 242 kJ

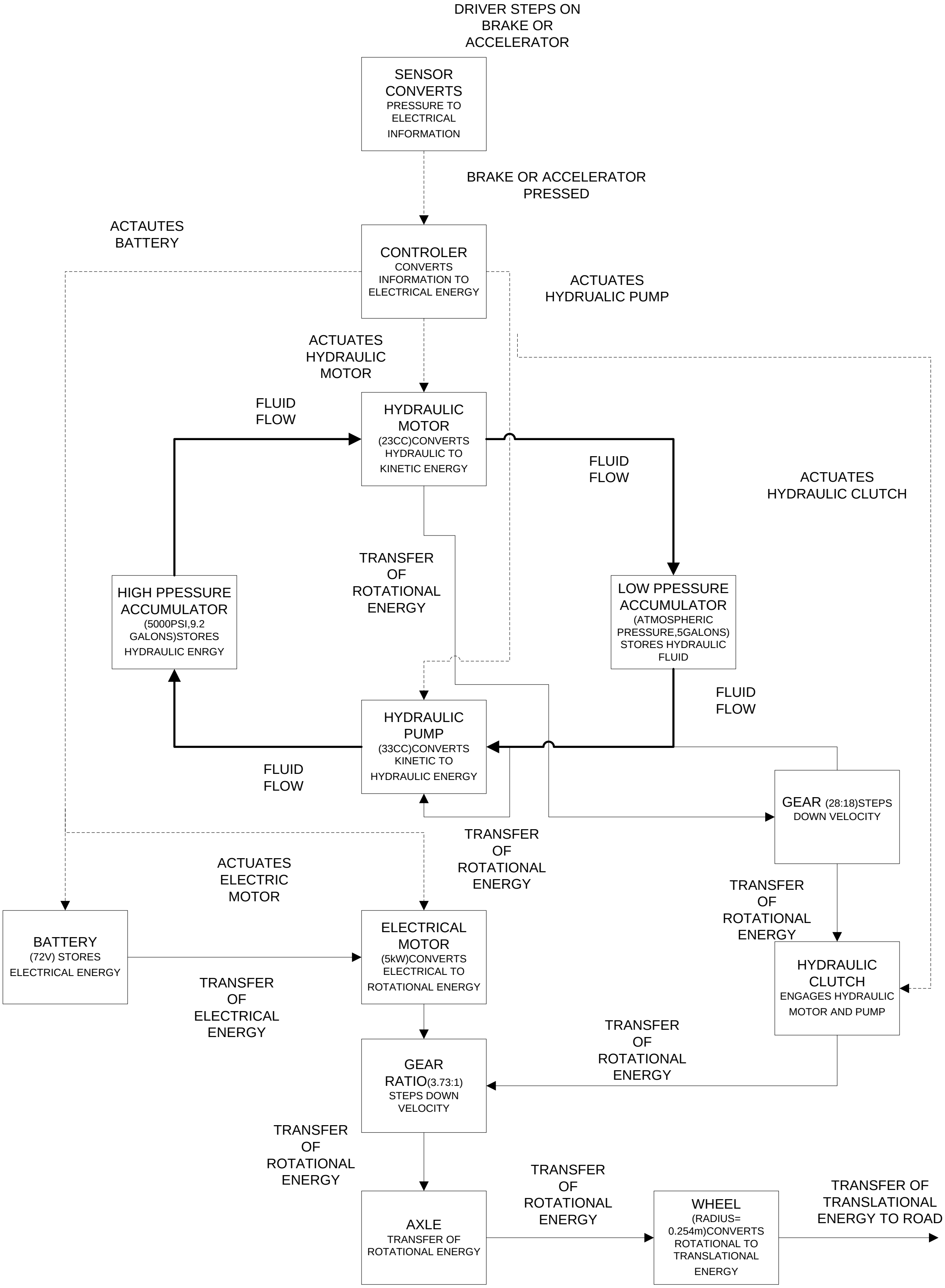
The Kinetic Energy is to be saved in the torsion spring of deflection angle 90^{o} and 180^{o} degrees:

P.E. = $\frac{1}{2}k\theta^2 \quad so, k = \frac{2 \times 242000}{\theta^2}$

So the required spring constant:

For 90^{o} Deflection angle = 202.64 $\frac{kJ}{\theta^2}$
For 180^{o} Deflection angle = 50.66 $\frac{kJ}{\theta^2}$

Appendix N: Alpha Design Functional Decomposition



Appendix O: Matlab code for Mathematical Model

Solver.m (Function number 1)

```
clear all
%clc
close all

%Define System Variables
m=1200;        %Mass of Vehicle (kg)
R=.254;        %Tire Radius (m)
alpha=1.3;     %Adiabatic Exponent for Nitrogen
k=2.3*10^(-5); %Motor Displacement (m^3/rev)
Ef=0.85;       %Motor Efficiency

%Coefficients of Drag
c1=0.462;      %x_dot^2 term (N*s^2/m^2)
c2=2.994;      %x_dot term (N*s/m)
c3=135.1;      %rolling resistance term (N)

%Initial Conditions
P0=34*10^6;    %Inital Nitrogen Pressure (Pa)
Si=0;          %Inital Position of Vehicle (m)
Ui=0;          %Inital Velocity of Vehicle (m/s)
ti=0;          %Inital Time (s)
tf=30;         %Final Time (s)

%Optimization Variables
N=4.3;         %Overall Gear Ratio
Vi=.0155;      %Inital Nitrogen Volume (m^3)

[T,X]=ode45(@StateEqs,[ti tf],[Si,Ui]);

%Volume of fluid through motor
V1=zeros(length(X),1);  %V1 is in L
V2=zeros(length(X),1);  %V2 is in m^3
for i=1:(length(X));
   V1(i,1)=(k*N*1000/(2*pi*R))*X(i,1); %(L)
   V2(i,1)=(k*N/(2*pi*R))*X(i,1);      %(m^3)
end

%RPM of Motor
RPM=zeros(length(X),1);
for i=1:(length(X));
   RPM(i,1)=(60*N/(2*pi*R))*X(i,2);
end

%Accumulator Pressure
P1=zeros(length(X),1);  %P1 is in psi
P2=zeros(length(X),1);  %P2 is in Pa
P3=zeros(length(X),1);  %P3 is in Bar
for i=1:(length(X));
   P1(i,1)=(P0*Vi^(alpha)*(Vi+(V2(i,1)))^(-alpha))/6895;   %Pressure in Accumulator (psi)
   P2(i,1)=(P0*Vi^(alpha)*(Vi+(V2(i,1)))^(-alpha));        %Pressure in Accumulator (Pa)
   P3(i,1)=(P0*Vi^(alpha)*(Vi+(V2(i,1)))^(-alpha))/100000; %Pressure in Accumulator (Bar)
end
```

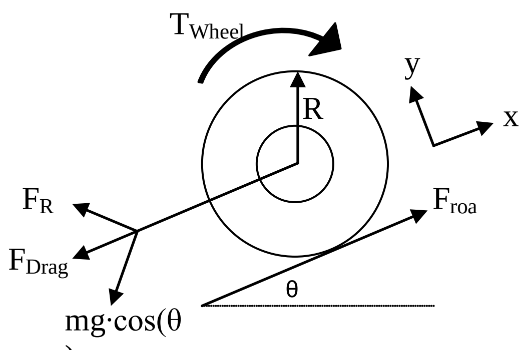
```
%Torque of Motor
Tm=zeros(length(P3),1);
for i=1:(length(X));
   Tm(i,1)=Ef*((k*10^6)/57.2)*P3(i,1); %Parker Equation Torque Motor (N*m)
end

%Find Max Position and Velocities
[C,I]=max(X);   %I(1) is the index value where the position is maximum
                %I(2) is the index value where the velocity is maximum

VelIndex = 0;
for i=1:(length(X))
   if X(i,2) <= 20
      VelIndex = i;
   end
end
disp('==================');
VolReq = Vi*1000+V1(VelIndex,1)
TimeReq = T(VelIndex)
disp('==================');

%Total Energy Stored In Accumulator
W=(P2(I(2),1)*(Vi+V2(I(2),1))+P0*Vi)/(alpha-1); %Work done by Nitrogen Gas (J)

%figure
%plot(T,X(:,2))
%xlabel('time (s)')
%ylabel('Velocity (m/s)');

%figure
%plot(T,RPM(:,1))
%xlabel('time (s)')
%ylabel('Angular Velocity (rev/min)');

%figure
%plot(T,P1(:,1))
%xlabel('time (s)')
%ylabel('Pressure (psi)');

%figure
%plot(T,V1(:,1))
%xlabel('time (s)')
%ylabel('Volume through motor (L)');

%figure
%plot(RPM(:,1),Tm(:,1))
%xlabel('Angular Velocity (rev/min)')
%ylabel('Motor Torque (N*m)');
```

StateEqs.m (Second part of model):

```
function dx=StateEqs(t,x)

%Define System Variables
m=1200;         %Mass of Vehicle (kg)
R=.254;         %Tire Radius (m)
alpha=1.3;      %Adiabatic Exponent for Nitrogen
k=2.3*10^(-5);  %Motor Displacement (m^3/rev)
```

```
Ef=0.85;        %Motor Efficiency

%Coefficients of Drag
c1=0.462;       %x_dot^2 term (N*s^2/m^2)
c2=2.994;       %x_dot term (N*s/m)
c3=135.1;       %rolling resistance term (N)

%Initial Conditions
P0=34*10^6;     %Inital Nitrogen Pressure (Pa)

%Optimization Variables
N=4.3;          %Overall Gear Ratio
Vi=.0155;       %Inital Nitrogen Volume (m^3)


deltaV=(k*N/(2*pi*R))*x(1);                   %Volume through motor (m^3)
P=(P0*Vi^(alpha)*(Vi+deltaV)^(-alpha))/100000;  %Nitrogen Pressure (bar)
Fd=c1*(x(2))^2+c2*x(2)+c3;                    %Drag Force (N)
Tm=Ef*((k*10^6)*P)/57.2;                      %Parker Equation Torque Motor (N*m)
Fm=Tm*N/R;                                    %Motor Force (N)

%State Equations
dx=zeros(2,1);
dx(1)=x(2);
dx(2)=(Fm-Fd)/m;
```

Appendix P: Starting on a hill Calculations



$$\sum F_x : \cancelto{0}{m\ddot{x}} = F_{road} - F_{RR} - \cancelto{0}{F_{Drag}} - mg \cdot \cos(\theta)$$

$$\sum \Gamma : \cancelto{0}{I\alpha} = T_{wheel} - F_{road} \cdot R$$

$$T_{wheel} = N \cdot T_m$$

$$\theta = cos^{-1}\left(\frac{T_m \cdot N}{R \cdot mg} - \frac{F_{RR}}{mg}\right) = 6°$$

$$T_{maximum} = 96\ N \cdot m$$
$$\omega_{wheel} = 750\ RPM$$
$$\omega_{motor\ ,max} = 2800\ RPM$$
$$N = \frac{\omega_{motor\ ,max}}{\omega_{wheel}} = 3.73$$

$$\theta = 6°$$

Appendix Q: Volume Optimization

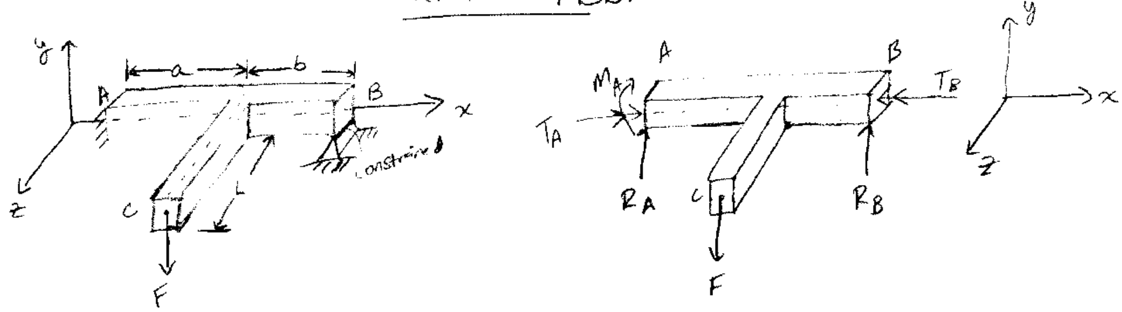
The table below relates the gear ratios and the initial volume of compressed nitrogen at 35MPa. The combination matrix gives us the volume of the accumulators (liters) required to make the Xebra to reach a top speed of 45mph.

Initial Volume of compressed Nitrogen at 35MPa in liters

Gear Ratios	10	12	14	15.5	16	18	20	22	24	26	28	30
2	N/A	N/A	N/A		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
2.1	N/A	N/A	N/A		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
2.2	N/A	N/A	N/A		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
2.3	N/A	N/A	N/A		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
2.4	N/A	N/A	N/A		N/A	N/A	N/A	N/A	N/A	N/A	N/A	N/A
2.5	N/A	N/A	N/A		N/A	N/A	39.5	39.4	39.6	40.9	41.6	43.4
2.6	N/A	N/A	N/A		N/A	40.2	38.2	38.4	38.8	40.4	41.8	42.3
2.7	N/A	N/A	N/A		39.1	38.8	37.2	37.8	38.5	40	41.4	41.3
2.8	N/A	N/A	N/A		41.4	37.2	36.6	37.2	38	39.4	40.9	42
2.9	N/A	N/A	N/A		42.8	36.1	35.8	36.6	37.6	38.7	40.5	40.6
3	N/A	N/A	N/A		37.9	35.3	35	36.3	37.1	38.6	40.6	41.2
3.1	N/A	N/A	N/A		35.6	34.4	34.9	35.7	37.1	38.6	40	41.6
3.2	N/A	N/A	N/A		34.8	34	34.2	35.7	37.1	38.5	39.9	39.7
3.3	N/A	N/A	N/A		34	33.4	34.2	35	36.4	38.5	39.8	40
3.4	N/A	N/A	36.9		33.3	33.4	34.2	34.9	36.2	38.3	39.7	40.3
3.5	N/A	N/A	35		32.6	32.6	33.3	34.7	36	38.2	39.5	40.6
3.6	N/A	N/A	34.1		32.6	32.5	33.2	34.5	35.8	37.9	39.2	41.2
3.7	N/A	N/A	33.3		31.7	32.3	32.9	34.3	35.6	37.7	38.9	41
3.8	N/A	N/A	32.4		31.5	32.1	32.7	34	36.1	37.4	39.4	40.7
3.9	N/A	N/A	31.5		31.3	31.9	33.2	34.5	35.8	37	39	41.1
4	N/A	N/A	31.3		31.1	31.6	32.9	34.1	35.4	37.5	38.7	40.7
4.1	N/A	N/A	31.1		30.8	31.2	32.5	33.7	35.8	37	39.1	40.3
4.2	N/A	N/A	30.8		30.4	30.8	32.1	34.2	35.4	36.6	38.6	40.7
4.3	N/A	N/A	30.5	29.1	30	31.3	32.5	33.7	34.9	37	39	40.2

Appendix R: Beam Failure Analysis

Overall FBD:

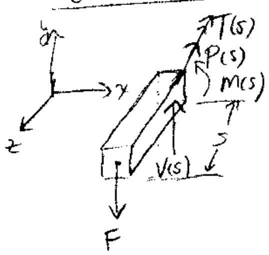


$$\sum F_y = -F + R_A + R_B = 0 \Rightarrow F = R_A + R_B$$

$$\sum M_{z,A} = -M_A - F\cdot a + R_B(a+b) = 0 \Rightarrow M_A = R_B(a+b) - F\cdot a$$

$$\sum T_x = T_A - T_B + F\cdot L \Rightarrow T_B = T_A + F\cdot L \Rightarrow M_A = (F - R_A)(a+b) - F\cdot a$$

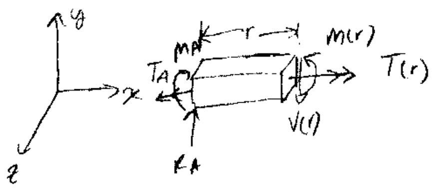
Segmented FBDs: $0 \le s \le L$:



$$P(s) = 0;\ T(s) = 0;\ V(s) = F$$

$$\sum M_x = F\cdot s + m(s) = 0 \Rightarrow m(s) = -F\cdot s$$

$0 \le r \le a$:

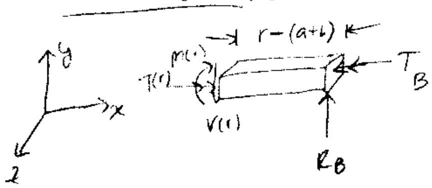


$$V(r) = R_A$$

$$\sum M_z = m(r) - R_A\cdot r - M_A \Rightarrow m(r) = R_A\cdot r + M_A$$

$$\sum T_x = T(r) - T_A = 0 \Rightarrow T(r) = T_A$$

$a < r \le (a+b)$:



$$V(r) = R_B$$

$$\sum M_z = -m(r) + R_B\left[r - (a+b)\right] = 0$$

$$m(r) = R_B\left[r - (a+b)\right]$$

$$\sum T_x = T(r) - T_B = 0 \Rightarrow T(r) = T_B$$

Torsional Deflection at point A, $\phi_A = 0$:

$$\phi_A = \left\{ \int_0^{a+b} \left[\frac{T(r)}{GJ}\cdot\frac{\partial T(r)}{\partial T_A}\right] dr \right\} = \left\{ \int_0^{a} \frac{T_A}{GJ}\, dr + \int_a^{a+b} \frac{(T_A+FL)}{GJ}\, dr \right\} = 0$$

$$\phi_A = \frac{T_A \cdot a}{GJ} + \frac{T_A + F\cdot L}{GJ}(a+b) - \frac{T_A + F\cdot L}{GJ}\cdot a = 0$$

$$\Rightarrow T_A = \frac{-FLb}{a+b} \quad ; \quad T_B = \frac{FLa}{a+b}$$

Deflection at Point A, $\gamma_A = 0$:

$$\gamma_A = \left\{ \int_0^{a+b} \left[\frac{M(r)}{EI}\frac{\partial M(r)}{\partial R_A} + \frac{T(r)}{GJ}\underbrace{\frac{\partial T(r)}{\partial R_A}}_{0}\right] dr \right\} = 0$$

$$\gamma_A = \left\{ \int_0^{a} \frac{\left[R_A\cdot r + (F-R_A)(a+b) - Fa\right]}{EI}\cdot(r-a-b)\, dr + \int_a^{a+b} \frac{(F-R_A)(r-a-b)}{EI}(a+b-r)\, dr \right\}$$

$$R_A = \frac{1}{2}\cdot\frac{F\cdot b(3a^2+6ab+2b^2)}{a^3+3a^2b+3ab^2+b^3} \quad ; \quad R_B = \frac{1}{2}\frac{Fa^2(2a+3b)}{a^3+3a^2b+3ab^2+b^3}$$

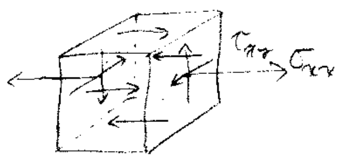
$$M_A = \frac{1}{2}\frac{(2a+3b)Fa^2}{a^2+2ab+b^2}$$

From Vehicle Measurements:

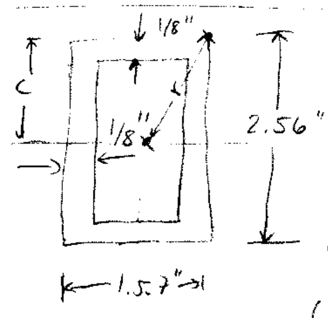
$a = 2.4$ in ; $b = 12.4$ in ; $L = 6.25$ in.

Failure at point A:

$R_A = 0.963 \cdot F$; $R_B = 0.037 \cdot F$; $M_A = 0.552 \cdot F$; $T_A = \frac{F \cdot L b}{a+b}$



Cross Section Area.



$I = 0.456 \text{ in}^4$

$J = 4.146 \text{ in}^4$

$c = 1.28 \text{ in.}$

$r = 1.50 \text{ in.}$

$\sigma_y = 51{,}000 \text{ lb/in}^2$

(AISI 1020 Steel)

$\sigma_{xx} = \frac{M_A \cdot c}{I}$; $\tau_{xy} = \frac{T_A \cdot r}{J}$

Principle Stresses:

$\sigma_1 = 3.596 \cdot F$; $\sigma_2 = -0.498 \cdot F$

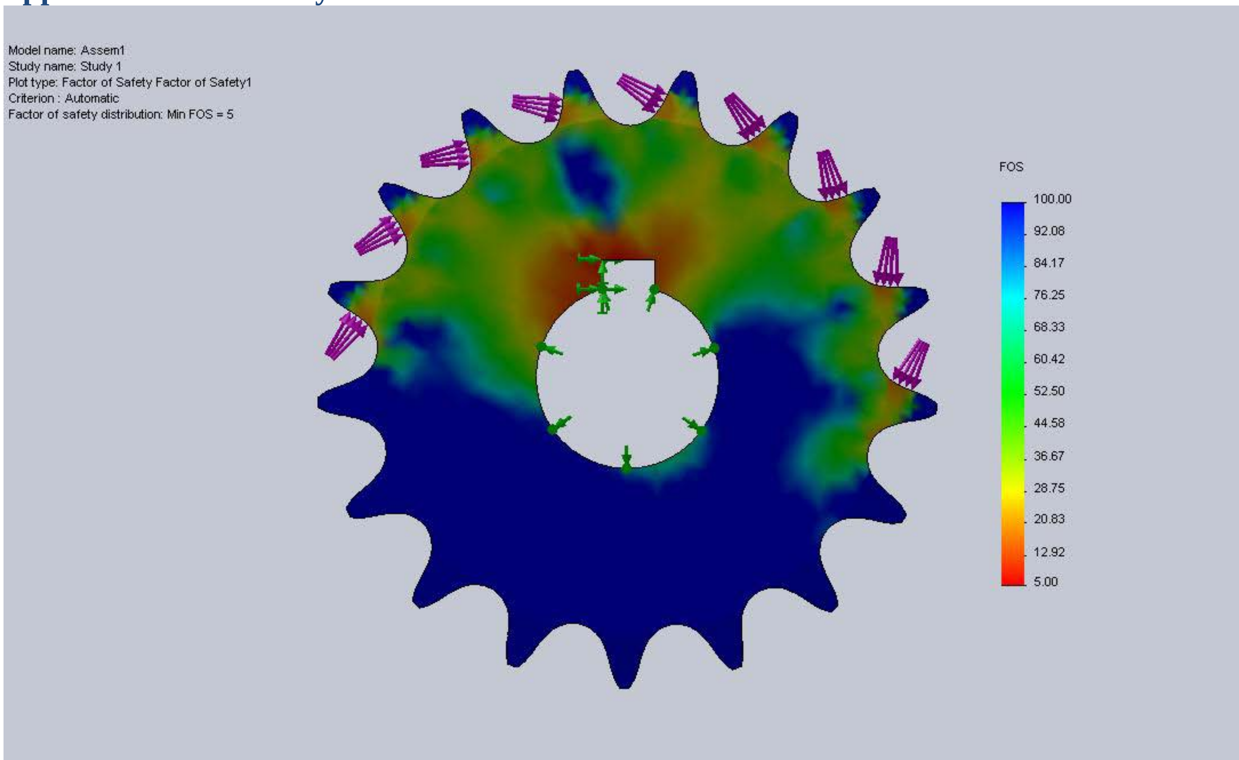
Von mises:

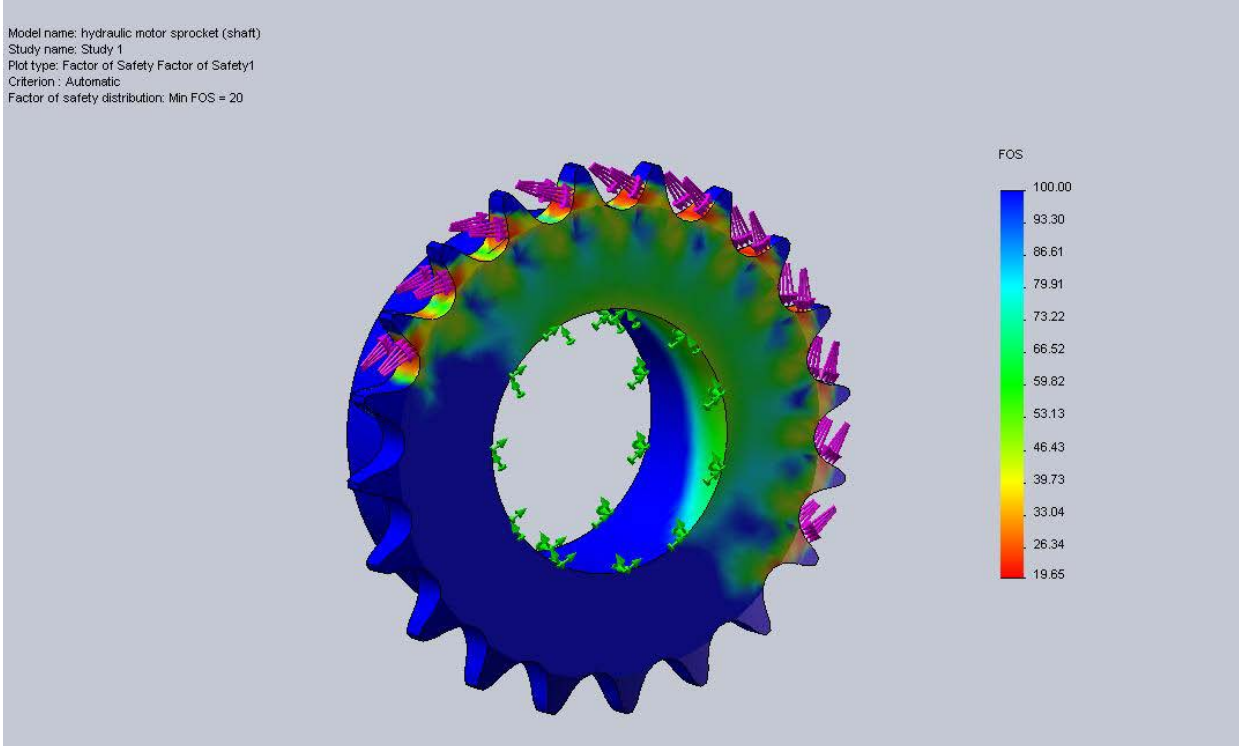
$\bar{\sigma}_M = 4.643 \cdot F \Rightarrow$ Yeild when $\bar{\sigma}_M \geq \sigma_y$

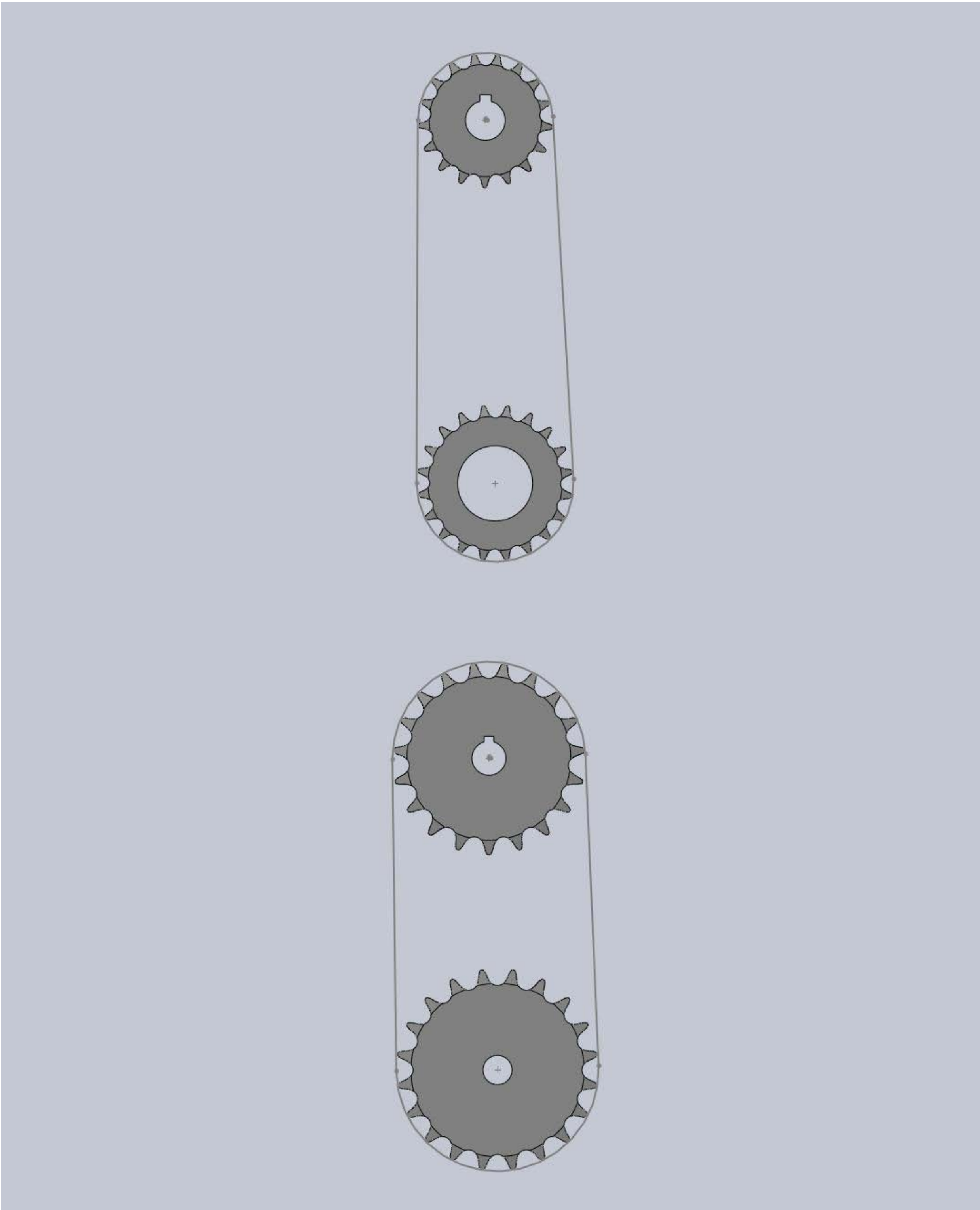
$F < \frac{\sigma_y}{4.643} \approx \boxed{11{,}000 \text{ lbs}}$

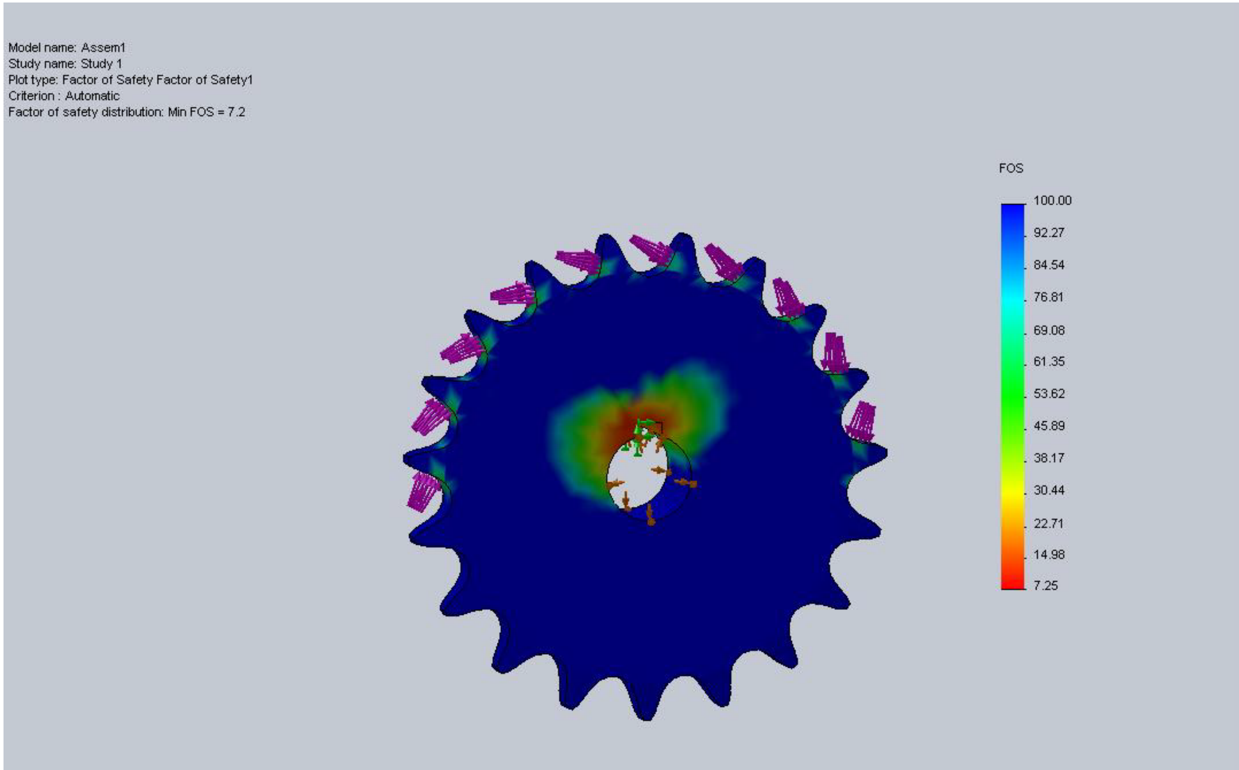
The beam can support a force up to 11,000 lbs

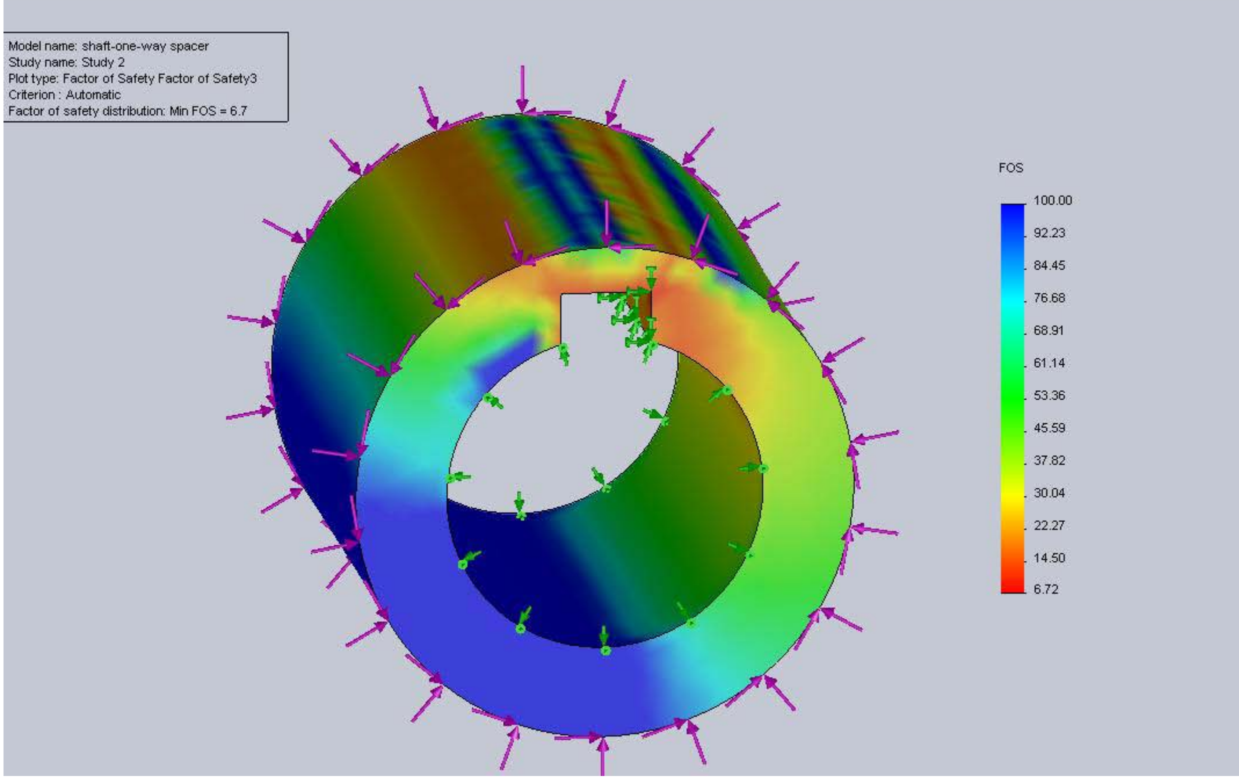
Appendix S: FEA Analysis







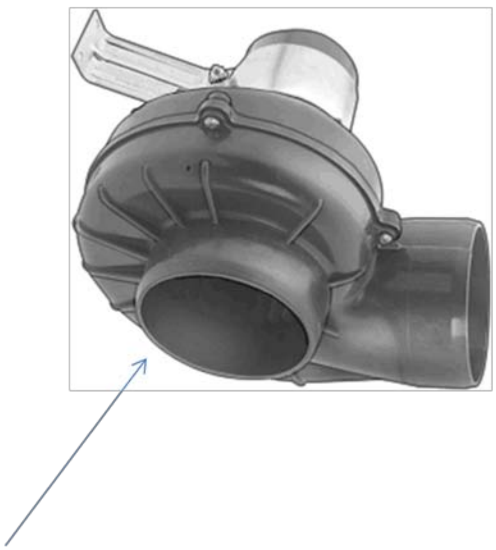

Model name: Assem1
Study name: Study 1
Plot type: Factor of Safety Factor of Safety1
Criterion : Automatic
Factor of safety distribution: Min FOS = 7.2
FOS
100.00
92.27
84.54
76.81
69.08
61.35
53.62
45.89
38.17
30.44
22.71
14.98
7.25

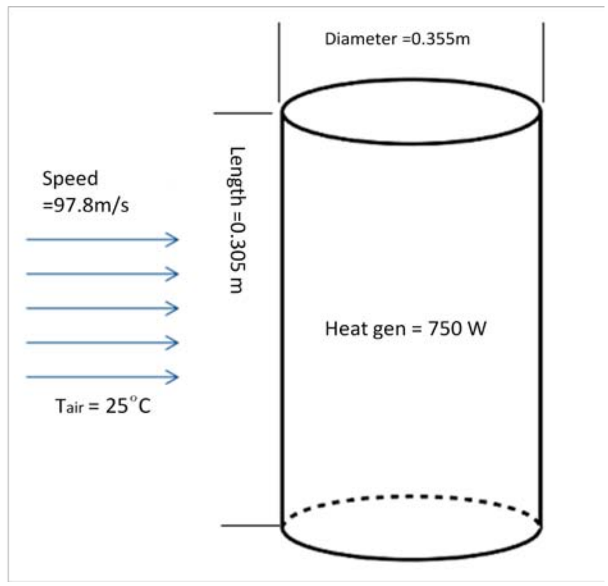

Model name: shaft-one-way spacer
Study name: Study 2
Plot type: Factor of Safety Factor of Safety3
Criterion : Automatic
Factor of safety distribution: Min FOS = 6.7
FOS
100.00
92.23
84.45
76.68
68.91
61.14
53.36
45.59
37.82
30.04
22.27
14.50
6.72

Appendix T: Calculations for Electric Motor cooling fan

Assumptions made:

1. Flow is modeled as flow over a cylindrical surface
2. Temperature of air at infinity T_∞=25°C.
3. Minimal static pressure build up
4. Negligible internal heat conduction
5. Total efficiency loss of the electric motor is converted to heat energy.
6. Air speed constant at 97.8m/s so the Reynolds number is 2188651, so flow is laminar





$$\dot{Q}_{gen} = loss\% \times 5kW$$
$$\dot{Q}_{gen} = 750W$$

To find the heat loss by convection at 60°C, as that is the temperature at which the cooling fan is actuated. So,

$$A \times \bar{h}(T_s - T_\infty) = \dot{Q}_{gen}$$
$$T_s = 60^o C \text{ }, h = convection\ coefficient.$$

To find h, we use the simplified flow for flow over a cylinder

H = $\frac{k}{D} NU_D$;

And NU_D = $C \times Re^m \times \Pr^n \times (\frac{Pr}{\Pr_s})^{1/4}$ and

$$\dot{Q}_{gen} = C \times Re^m \times \Pr^n \times 2k(\frac{Pr}{\Pr_s})^{1/4} \times (T_s - T_\infty) \times \frac{1}{D}$$

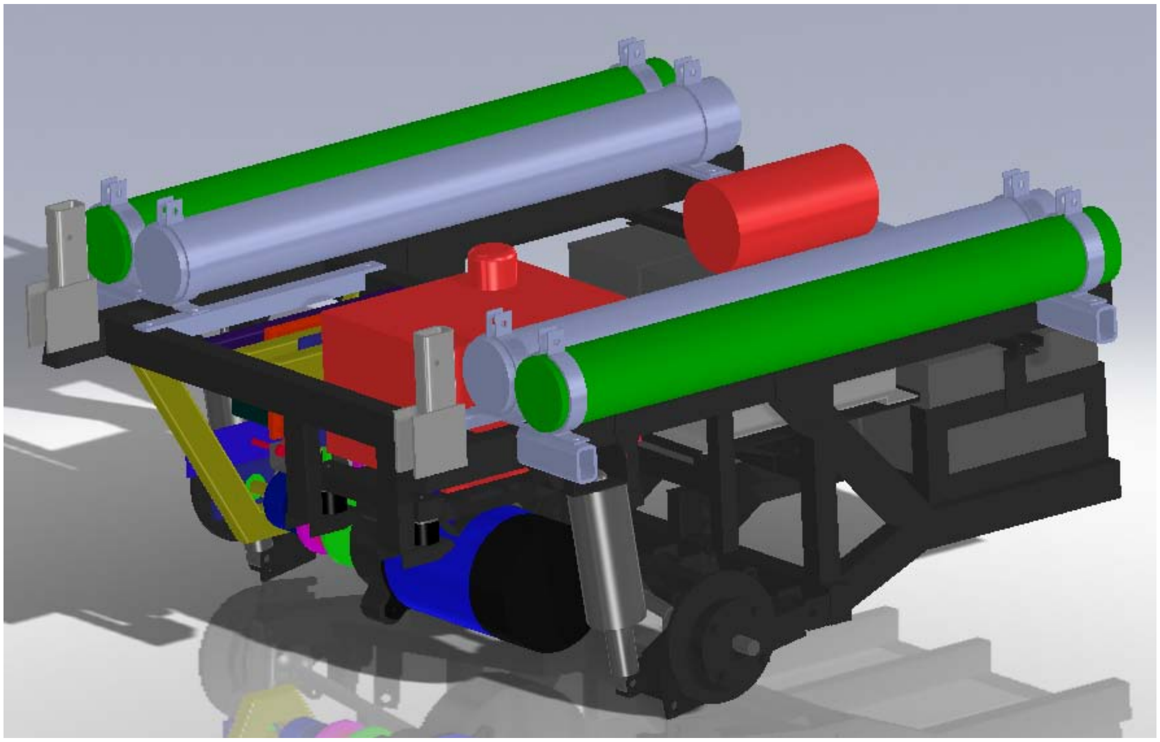
So using the values of Re=2188651, C= 0.0266, m=0.805, n =0.37, Pr air at T=25°C = 0.713, Pr_s (T=60°C)= 0.709 and k = 0.0263.

So at 60°C the heat transfer rate by convection is 6005.885 J/s.

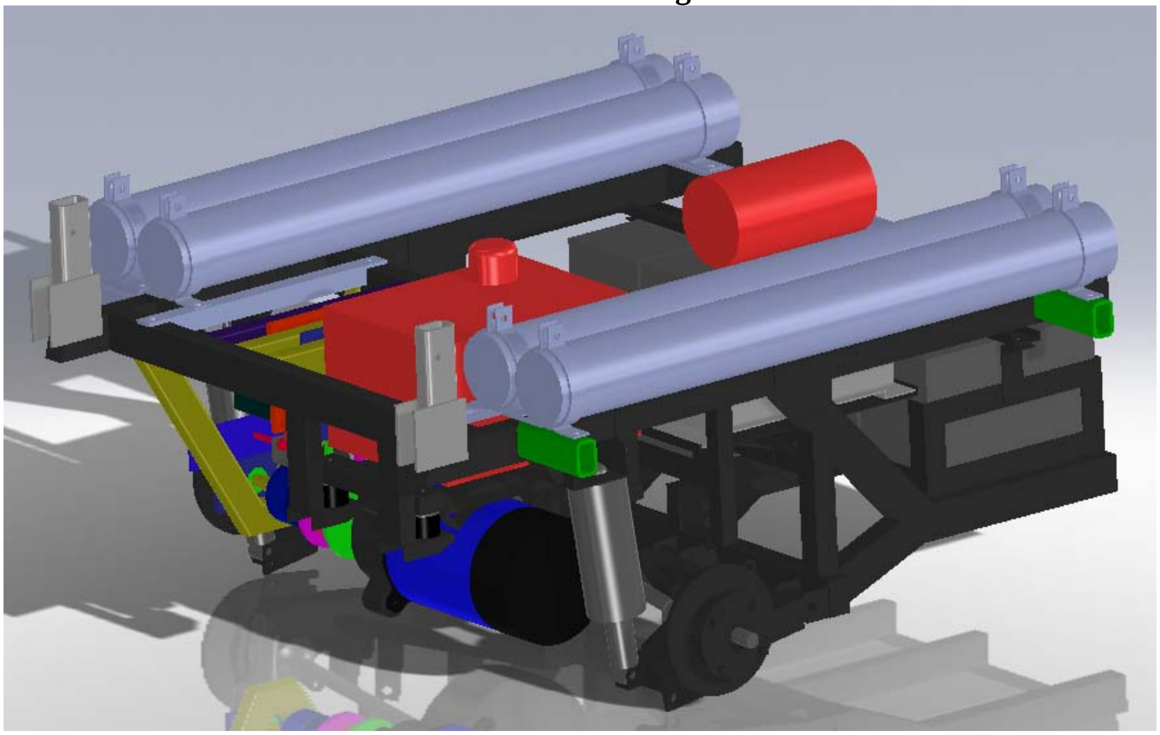
Which is much higher then $\dot{Q}_{gen}$ 750W

Appendix U: CAD Model Screen Shots

Additional Accumulators

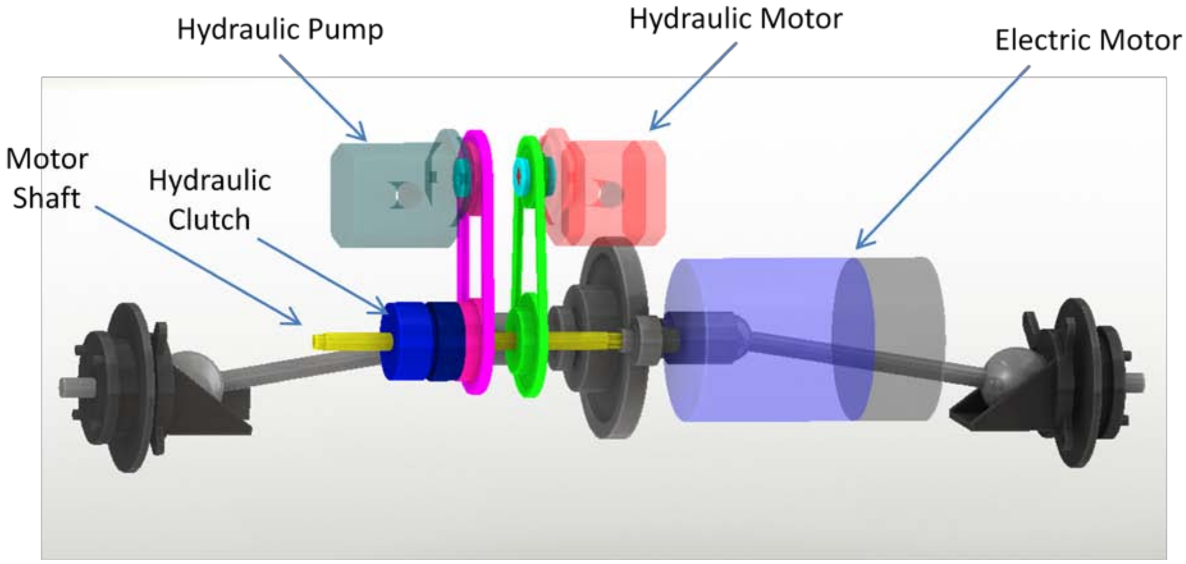


Accumulator Mounting Beams

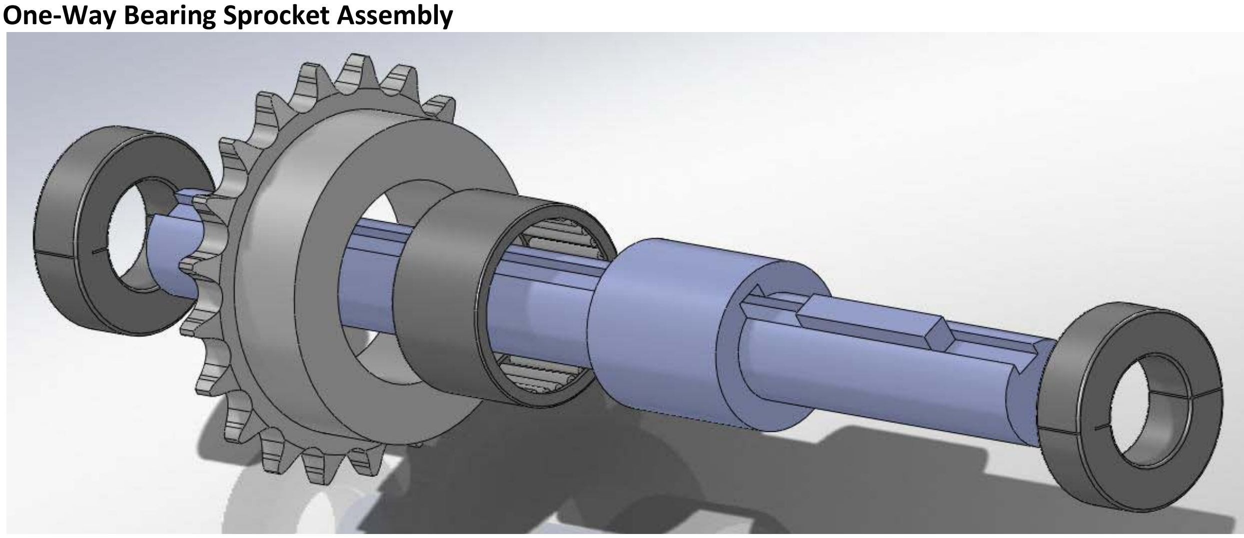


*Green components indicate additions made for our prototype design.

Drive Train Assembly



One-Way Bearing Sprocket Assembly



Appendix V: Mobil One Synthetic Transmission Oil MSDS

MOBIL OIL CORP -- AUTOMATIC TRANSMISSION FLUID 210 -- 9150-00B130099
===================== Product Identification =====================

Product ID:AUTOMATIC TRANSMISSION FLUID 210
MSDS Date:12/30/1992
FSC:9150
NIIN:00B130099
MSDS Number: BQDFF
=== Responsible Party ===
Company Name:MOBIL OIL CORP
Address:3225 GALLOWS ROAD
City:FAIRFAX
State:VA
ZIP:22037-0001
Country:US
Info Phone Num:800-424-9300
Emergency Phone Num:609-737-4411
CAGE:3U728
=== Contractor Identification ===
Company Name:MOBIL OIL CORP, NORTH AMERICAS MARKETING AND REFINING
Address:3225 GALLOWS ROAD
Box:City:FAIRFAX
State:VA
ZIP:22037
Country:US
Phone:800-662-4525/ 856-224-4644
CAGE:3U728

============= Composition/Information on Ingredients =============

Ingred Name:ZINC (SARA III)
CAS:7440-66-6
RTECS #:ZG8600000
Fraction by Wt: .04%
Other REC Limits:NONE SPECIFIED
OSHA PEL:10 MG ZNO/M3
ACGIH TLV:10 MG ZNO/M3; 9192
EPA Rpt Qty:1000 LBS
DOT Rpt Qty:1000 LBS

Ingred Name:ZINC DIALKYL DITHIOPHOSPHATE
CAS:68457-79-4
Fraction by Wt: .61%
Other REC Limits:NONE SPECIFIED

Ingred Name:MINERAL OIL, PETROLEUM DISTILLATES, SOLVENT-DEWAXED HEAVY
PARAFFINIC
CAS:64742-65-0
RTECS #:PY8038500
Fraction by Wt: 95.0%
Other REC Limits:NONE SPECIFIED
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:REFINED HEAVY PARAFFINIC DISTILLATES
CAS:64741-88-4
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:REFINED HEAVY PARAFFINIC DISTILLATES
CAS:64741-88-4
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:DISTILLATES, SOLVENT-REFINED LIGHT NAPHTHENIC
CAS:64741-97-5
RTECS #:PY8041000
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:300 PPM
ACGIH TLV:300 PPM

Ingred Name:2-PROPENOIC ACID, 2-METHYL-, BUTYL ESTER, POLYMER WITH N-(3-DIMETHYLAMINO PROPYL)-2-METHYL-2-PROPENAMIDE
CAS:50867-55-5
Fraction by Wt: <2.0%
Other REC Limits:NONE SPECIFIED

Ingred Name:MINERAL OIL, PETROLEUM DISTILLATES, HYDROTREATED LIGHT NAPHTHENIC
CAS:64742-53-6
RTECS #:PY8036000
Fraction by Wt: 0.39%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

==================== Hazards Identification ====================

LD50 LC50 Mixture:TLV: 5.00 MG/M3
Routes of Entry: Inhalation:NO Skin:NO Ingestion:NO
Reports of Carcinogenicity:NTP:NO IARC:NO OSHA:NO
Health Hazards Acute and Chronic:IRRITATING TO EYES & SKIN.
Explanation of Carcinogenicity:THIS COMPOUND CONTAINS NO INGREDIENTS AT
CONCENTRATIONS OF 0.1% OR GREATER THAT ARE CARCINOGENS OR
SUSPECT
CARCINOGENS.
Effects of Overexposure:NONE.
Medical Cond Aggravated by Exposure:PERSONS WITH A HISTORY OF AILMENTS
OR WITH A PRE-EXISTING DISEASE INVOLVING THE SKIN MAY BE AT
INCREASED RISK FROM EXPOSURE.

====================== First Aid Measures ======================

First Aid:EYES: FLUSH WITH RUNNING WATER FOR 15 MINUTES WHILE HOLDING
EYELIDS OPEN. SKIN: WASH WITH SOAP AND WATER. REMOVE
CONTAMINATED
CLOTHING. INGESTION: DO NOT INDUCE VOMITING. RINSE MOUTH & DRINK
LARGE AMO UNTS OF WATER. GET MEDICAL ATTENTION.

==================== Fire Fighting Measures ====================

Flash Point Method:COC
Flash Point:350F,177C
Lower Limits:.6%
Upper Limits:7.0%
Extinguishing Media:USE WATER FOG, CARBON DIOXIDE, FOAM, OR DRY
CHEMICAL.
Fire Fighting Procedures:WEAR FIRE FIGHTING PROTECTIVE EQUIPMENT AND A
FULL FACED SELF CONTAINED BREATHING APPARATUS. COOL FIRE EXPOSED
CONTAINERS WITH WATER SPRAY.
Unusual Fire/Explosion Hazard:NONE

================== Accidental Release Measures ==================

Spill Release Procedures:ABSORB ON FIRE RETARDANT TREATED SAWDUST,
DIATOMACEOUS EARTH, ETC. SHOVEL UP AND DISPOSE OF AT AN
APPROPRIATE
WASTE DISPOSAL FACILITY.

===================== Handling and Storage =====================

Handling and Storage Precautions:STORE IN A COOL, DRY PLACE. KEEP
CONTAINERS CLOSED WHEN MATERIAL IS NOT IN USE. KEEP AWAY FROM
IGNITION SOURCES. AVOID PROLONGED OR REPEATED CONTACT.
Other Precautions:NONE

============= Exposure Controls/Personal Protection =============

Respiratory Protection:NONE
Ventilation:LOCAL AND MECHANICAL(GENERAL) EXHAUST TO PROVIDE ADEQUATE
VENTILATION.
Protective Gloves:NONE
Eye Protection:SAFETY GLASSES - CHEMICAL SPLASH GOGGLES
Other Protective Equipment:NONE
Work Hygienic Practices:WASH THOROUGHLY AFTER HANDLING AND BEFORE
EATING, DRINKING OR SMOKING. LAUNDER CONTAMINATED CLOTHING BEFORE
REUSE.
Supplemental Safety and Health
AVOID PROLONGED OR REPEATED EXPOSURE. DO NOT GET ON SKIN OR IN EYES. DO
NOT INGEST. READ PRECAUTIONS ON LABEL BEFORE USE.

================== Physical/Chemical Properties ==================

HCC:V6
Boiling Pt:B.P. Text:>600F,>316C
Melt/Freeze Pt:M.P/F.P Text:NA
Vapor Pres:<.1
Vapor Density:0.868
Solubility in Water:NEGLIGIBLE
Appearance and Odor:RED LIQUID, MILD ODOR
Corrosion Rate:N/KNOWN

================= Stability and Reactivity Data =================

Stability Indicator/Materials to Avoid:YES
STRONG OXIDIZING AGENTS
Stability Condition to Avoid:HIGH HEAT, OPEN FLAMES AND OTHER SOURCES
OF IGNITION
Hazardous Decomposition Products:CARBON MONOXIDE AND CARBON DIOXIDE

==================== Disposal Considerations ====================

Waste Disposal Methods:DISPOSAL OF WASTE AND HAZARDOUS MATERIAL SHALL
COMPLY WITH LOCAL, STATE AND FEDERAL ENVIRONMENTAL PROTECTION
AGENCY REGULATIONS. SEND WASTE MATERIAL TO AN APPROVED RECYCLING
FACILITY IF FEASIBLE. CITY , STATE AND FEDERAL REGULATIONS MUST
BEFOLLOWED.

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Appendix W: Calculations of max torque and gear ratio

The maximum torque was calculated using the maximum continuous pressure the motor/pump are rated for and its displacement.
The gear ratios were calculated using the formula:

$$GearRatio = \frac{Max\ RPM\ of\ Motor/Pump}{Wheel\ RPM\ at\ 45MPH}$$

where wheel RPM at 45MPH = 625.1 RPM

Using the formula:

$$Torque = \frac{(Gear\ Ratio) \times (Displacement\ in\ m^3) \times (MaxPressure\ in\ Pa)}{2 \times \pi}$$

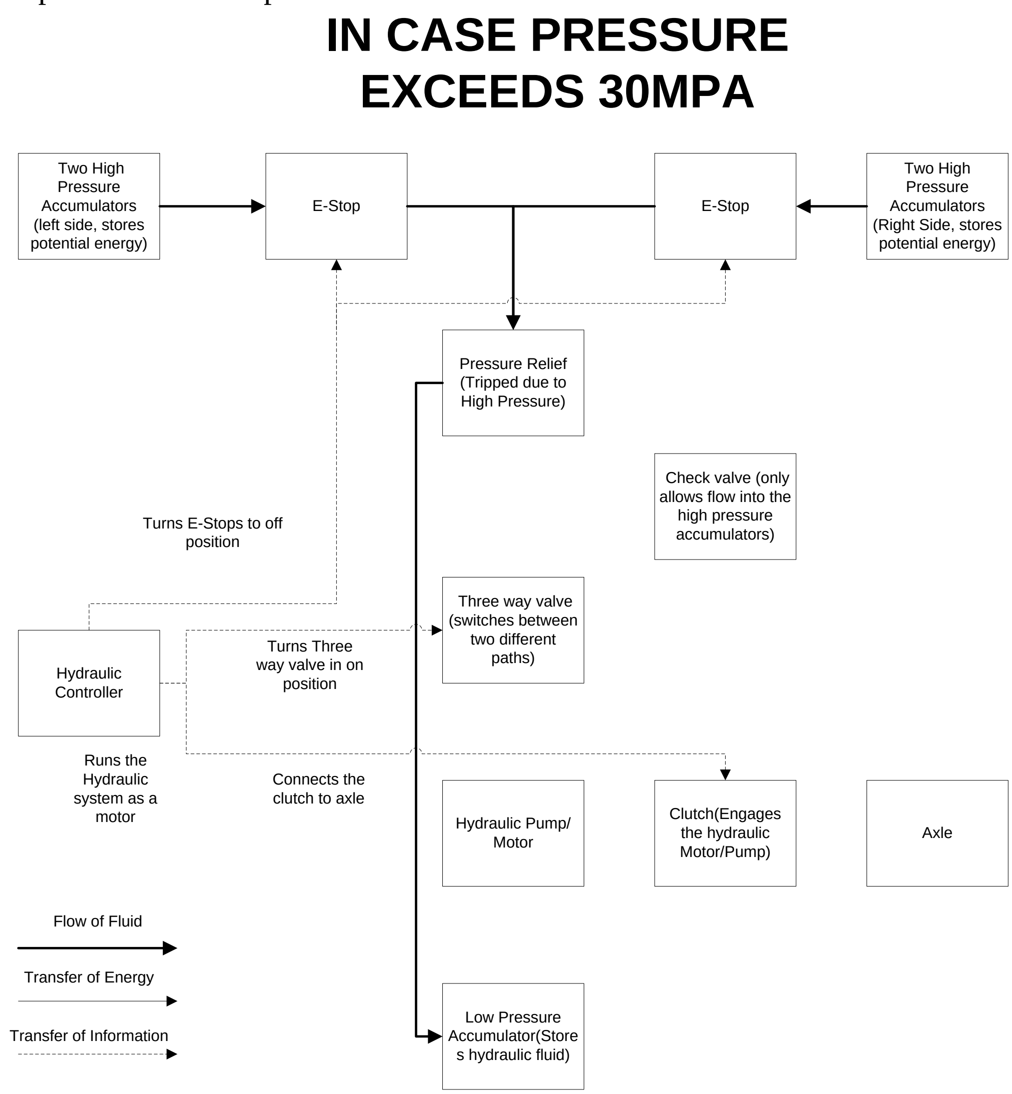
The table below lists the different motors/pumps and gives us the max RPM, gear ratio, rated continuous pressure and max torque. These values were obtained from the motor specifications sheet in appendix ___ and the formulae listed above.

Table #: Torques and speeds for different motor displacements

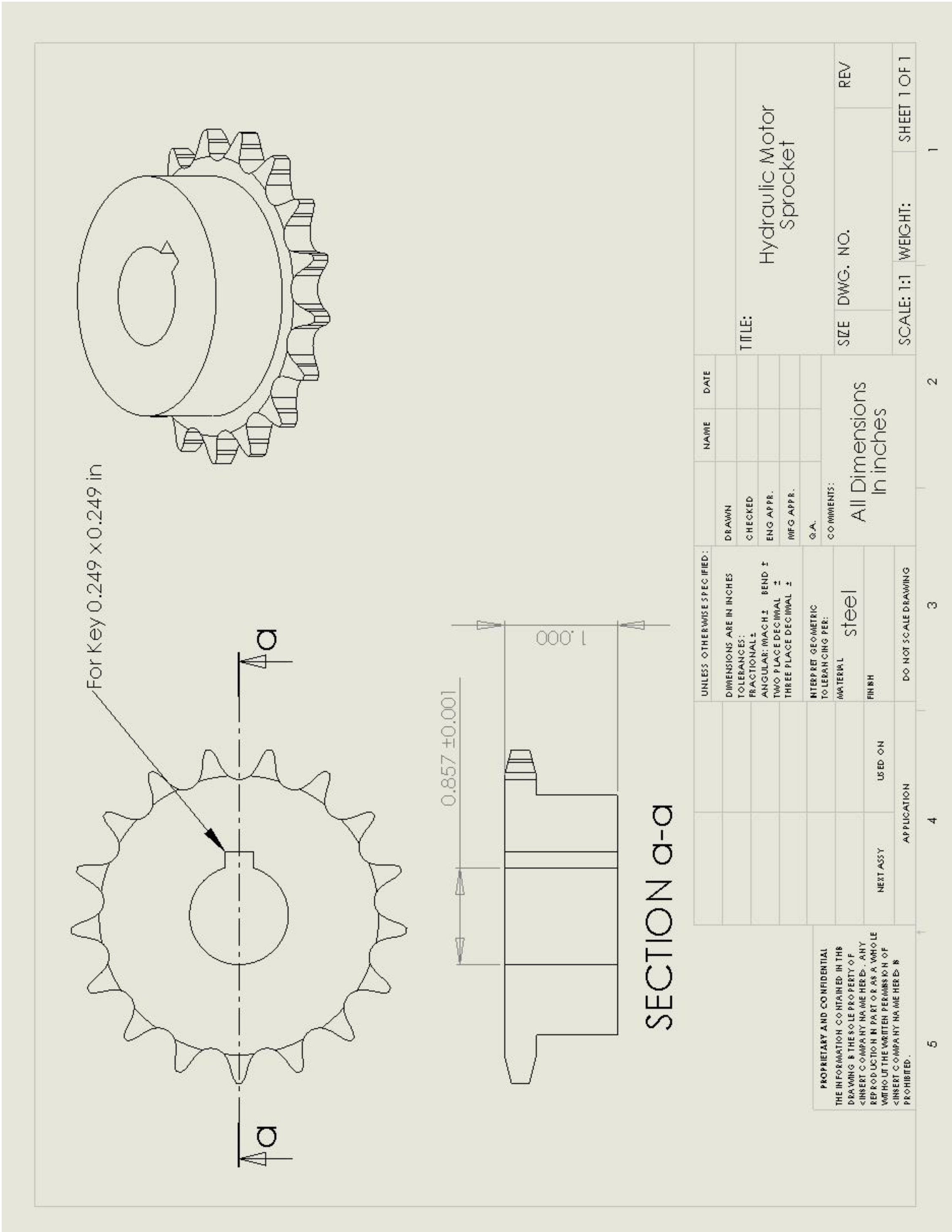
Displacement	Max RPM	Max Continuous Pressure	Gear Ratio Required	Max Torque
33	3000	25MPa	4.8:1	630.25Nm
38	3000	25MPa	4.8:1	725.75Nm
44	2800	22MPa	4.48:1	690.2Nm
52	2700	20MPa	4.32:1	715.05Nm
70	2400	16MPa	3.84:1	684.5Nm

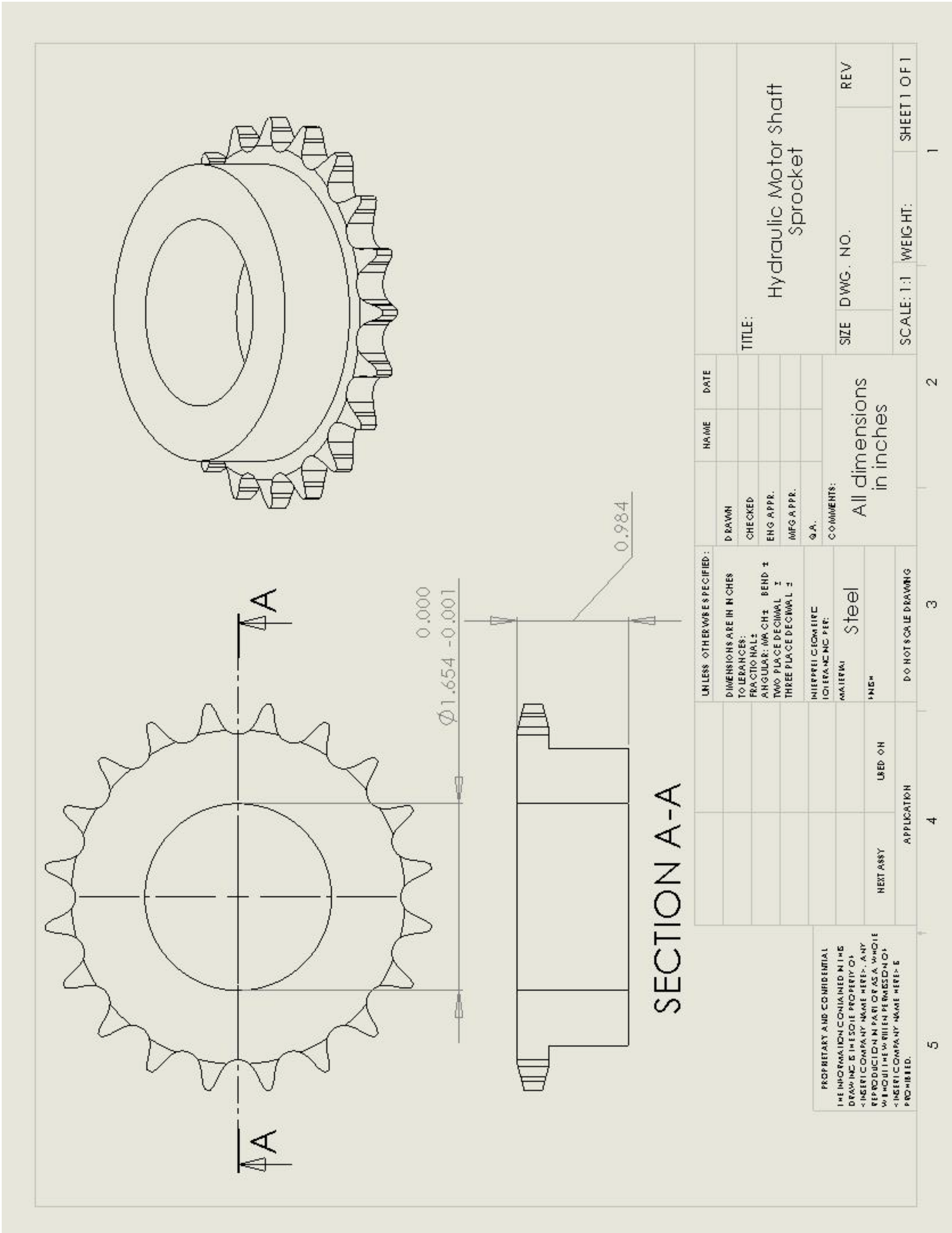
Appendix X: Flow of fluid if pressure exceeds 30MPa

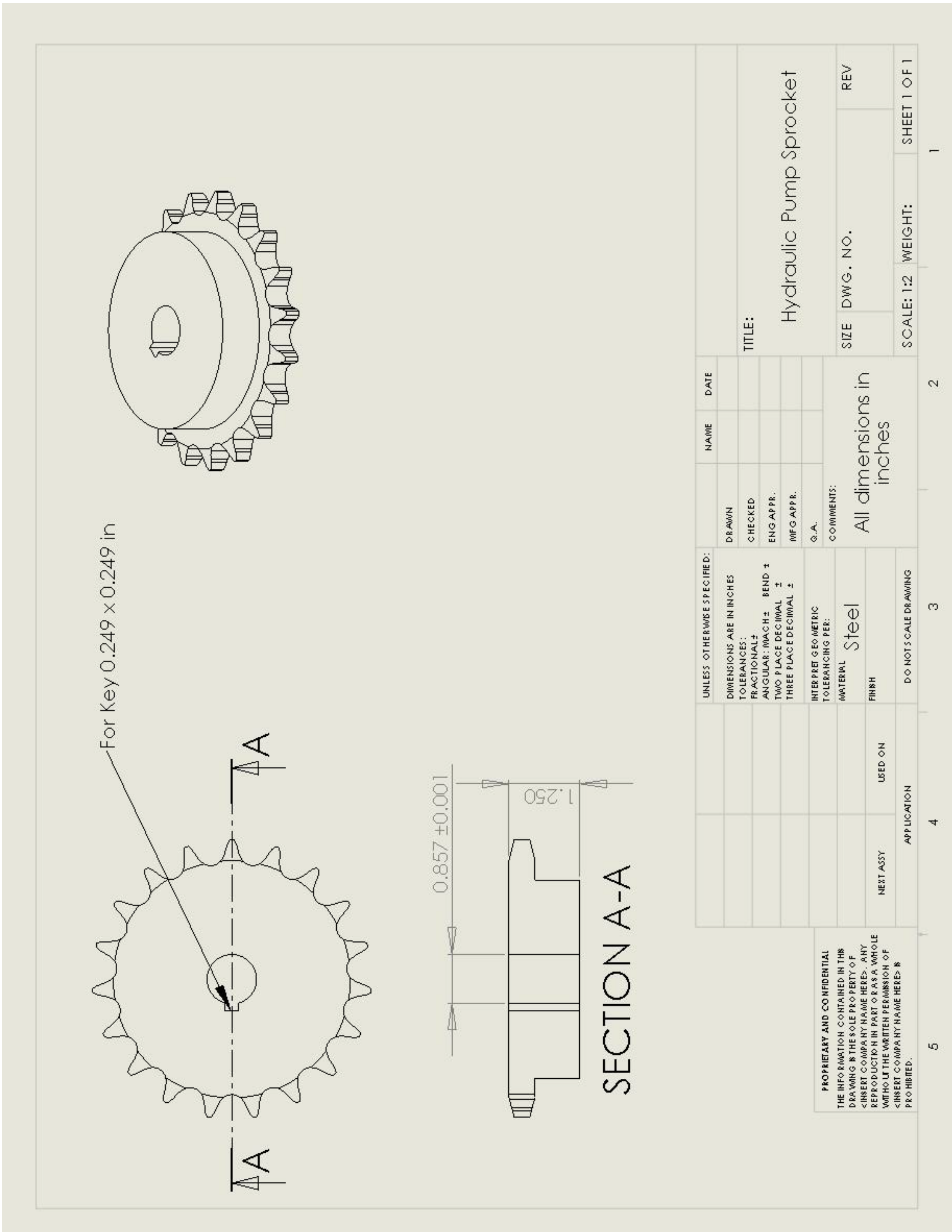
The diagram below shows the flow of hydraulic fluid once the pressure relief valve has been tripped. The fluid flow bypasses the three way valve, check valve and the motor. The fluid is directly dumped into the Low pressure accumulator.

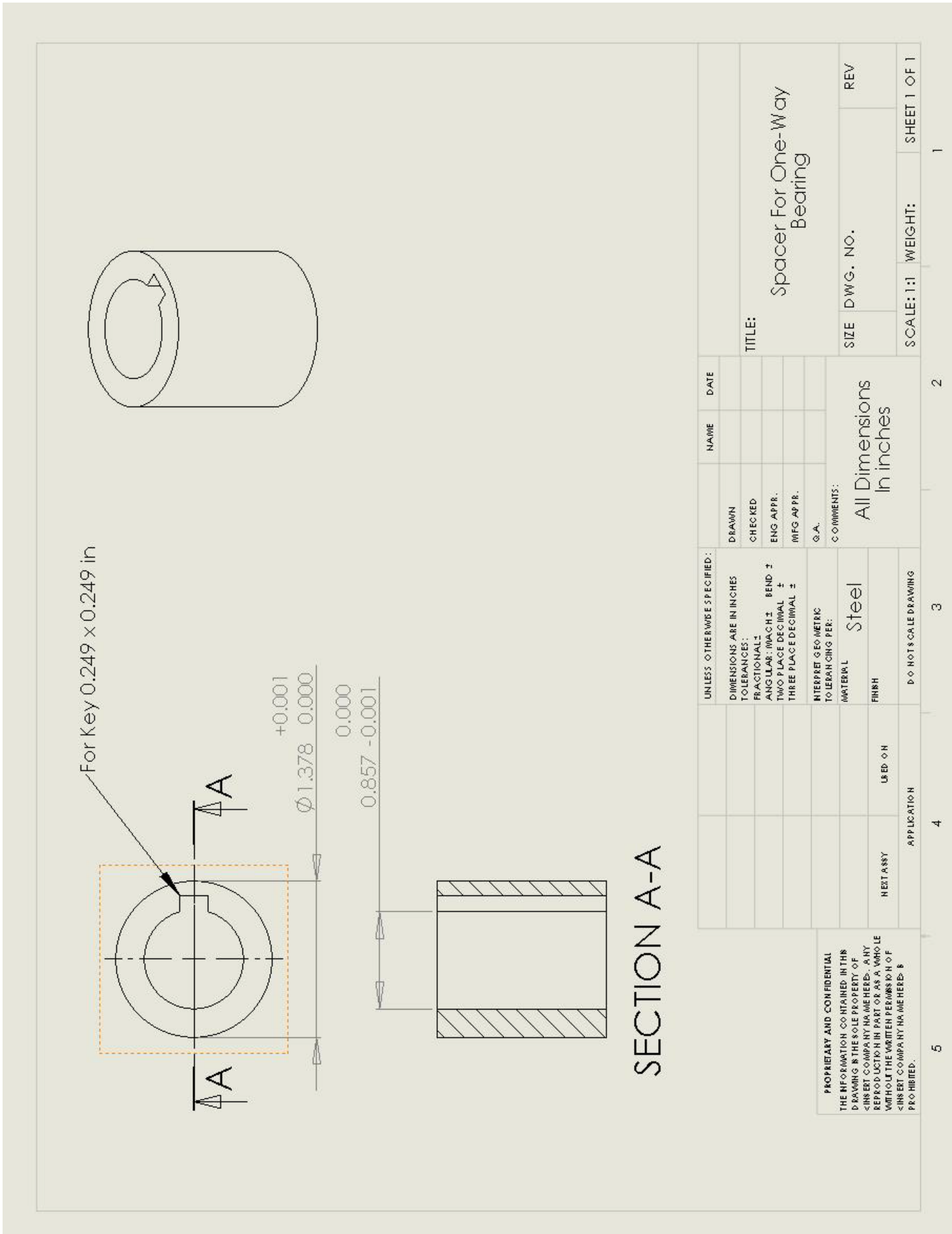


Appendix Y: Engineering Drawings



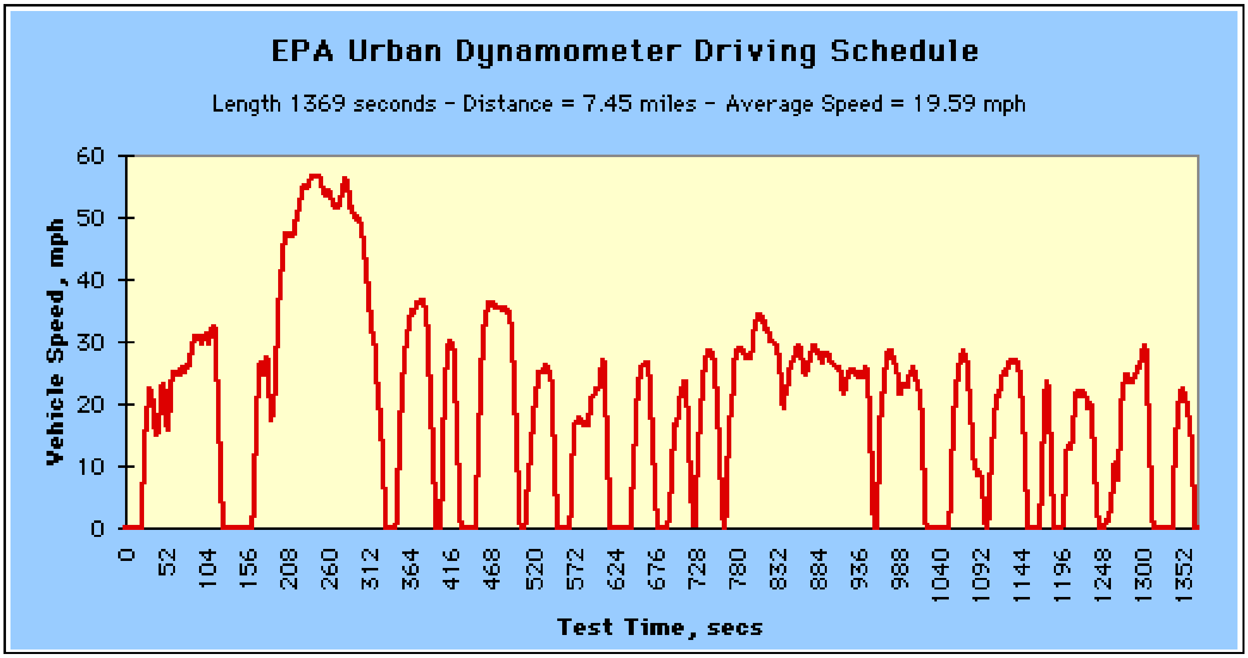

SECTION A-A
A
A
Ø1.654 -0.001
0.000
0.984
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
MATERIAL
Steel
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
PROPRIETARY AND CONFIDENTIAL
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
NAME
DATE
All dimensions in inches
TITLE:
Hydraulic Motor Shaft Sprocket
SIZE
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
5
4
3
2
1


For Key 0.249 x 0.249 in
A
A
0.857 ±0.001
1.250
SECTION A-A
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Steel
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
PROPRIETARY AND CONFIDENTIAL
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
All dimensions in inches
TITLE:
Hydraulic Pump Sprocket
SIZE
DWG. NO.
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1



For Key 0.249 x 0.249 in
A
A
Ø1.378 +0.001 0.000
0.857 0.000 -0.001
SECTION A-A
PROPRIETARY AND CONFIDENTIAL
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
MATERIAL
Steel
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
All Dimensions
In inches
TITLE:
Spacer For One-Way
Bearing
SIZE
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1

Appendix Z: EPA LA4 Drive Cycle



During dynamometer testing, the goal will be for the Xebra to follow the red LA4 curve shown above as closely as possible. As you can see, the maximum speed of this cycle is roughly 57 mph. Since the Xebra cannot go that fast, it will simply achieve its maximum top speed (45 mph) and then begin to follow the curve again when the cycle decelerates from the top speed.

Appendix AA: Electronic Controller Setting from Manual

6.1.2 SETTINGS:

Maximum Output Current
This slider adjusts the maximum output current that the controller can provide to the motor. Output current is adjusted as a percent of the maximum rating of the controller. For example, an AXE4844 will provide a maximum of 400A to the motor when this slider is set to 100%. A 75% setting on Maximum Output Current will limit the controller to 300A, 50% will limit the max output current to 200A and so forth.

Under Voltage
This slider sets the under voltage shutdown of the controller, in units of 1/10ths Volt. Generally speaking, it is undesirable to pull the terminal voltage of a 6V lead-acid battery below 4.0V, for example 24V on a 36V system.

Over Voltage
This slider sets the maximum operating voltage of the controller. If the voltage present across the B- to B+ bus bars exceeds this setting, the controller will not produce output, given that DC voltage is below the absolute ratings of the controller.

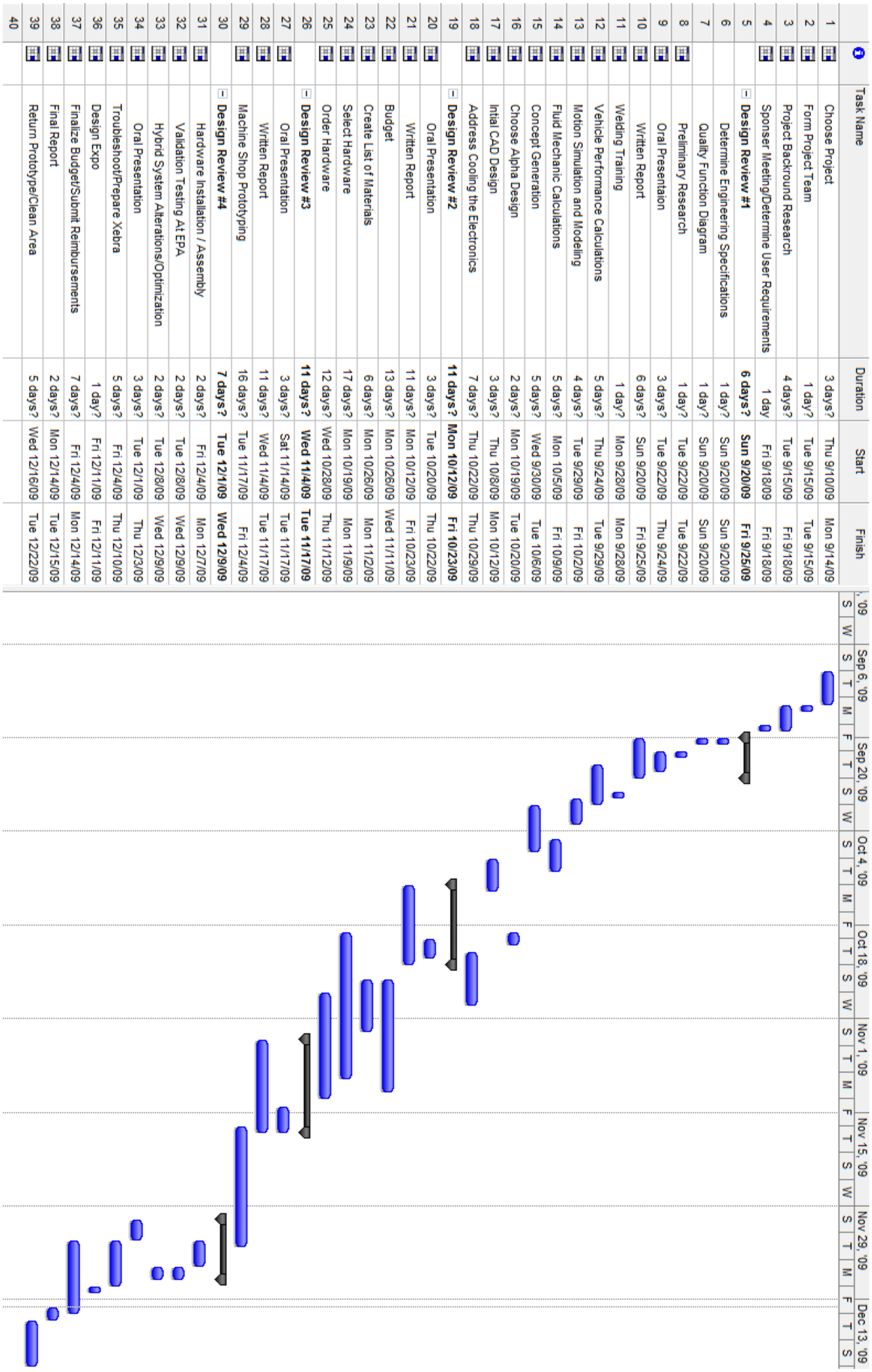
Throttle Up Rate
This slider adjusts the rate at which the controller increases it’s output current in response to an increase in throttle position. 1 is the slowest, 15 the fastest.

Throttle Down Rate
This slider adjusts the rate at which the controller reduces it’s output current in response to a decrease in throttle position. 0 is the slowest, 15 the fastest. It is recommended that this parameter typically be set to twice the value of the throttle up rate, when throttle up rate is less than 7. Lower values of Throttle Down Rate can result in the vehicle feeling as if their were a large flywheel connected to the motor.

Brake Current
On those models equipped with a plug brake (suffix “P” in the model number), this slider adjusts the amount of brake current as a percent of maximum available brake current. Refer to AXE specifications for maximum available brake current depending on the model of controller.

Top Speed
Top speed of the controller can be limited from 100% down to 50%. This is a helpful tool for elderly or handicapped drivers.

Appendix AB: Updated GANTT Chart



	Task Name	Duration	Start	Finish
1	Choose Project	3 days?	Thu 9/10/09	Mon 9/14/09
2	Form Project Team	1 day?	Tue 9/15/09	Tue 9/15/09
3	Project Backround Research	4 days?	Tue 9/15/09	Fri 9/18/09
4	Sponser Meeting/Determine User Requirements	1 day	Fri 9/18/09	Fri 9/18/09
5	**Design Review #1**	**6 days?**	**Sun 9/20/09**	**Fri 9/25/09**
6	Determine Engineering Specifications	1 day?	Sun 9/20/09	Sun 9/20/09
7	Quality Function Diagram	1 day?	Sun 9/20/09	Sun 9/20/09
8	Preliminary Research	1 day?	Tue 9/22/09	Tue 9/22/09
9	Oral Presentaion	3 days?	Tue 9/22/09	Thu 9/24/09
10	Written Report	6 days?	Sun 9/20/09	Fri 9/25/09
11	Welding Training	1 day?	Mon 9/28/09	Mon 9/28/09
12	Vehicle Performance Calculations	5 days?	Thu 9/24/09	Tue 9/29/09
13	Motion Simulation and Modeling	4 days?	Tue 9/29/09	Fri 10/2/09
14	Fluid Mechanic Calculations	5 days?	Mon 10/5/09	Fri 10/9/09
15	Concept Generation	5 days?	Wed 9/30/09	Tue 10/6/09
16	Choose Alpha Design	2 days?	Mon 10/19/09	Tue 10/20/09
17	Intial CAD Design	3 days?	Thu 10/8/09	Mon 10/12/09
18	Address Cooling the Electronics	7 days?	Thu 10/22/09	Thu 10/29/09
19	**Design Review #2**	**11 days?**	**Mon 10/12/09**	**Fri 10/23/09**
20	Oral Presentation	3 days?	Tue 10/20/09	Thu 10/22/09
21	Written Report	11 days?	Mon 10/12/09	Fri 10/23/09
22	Budget	13 days?	Mon 10/26/09	Wed 11/11/09
23	Create List of Materials	6 days?	Mon 10/26/09	Mon 11/2/09
24	Select Hardware	17 days?	Mon 10/19/09	Mon 11/9/09
25	Order Hardware	12 days?	Wed 10/28/09	Thu 11/12/09
26	**Design Review #3**	**11 days?**	**Wed 11/4/09**	**Tue 11/17/09**
27	Oral Presentation	3 days?	Sat 11/14/09	Tue 11/17/09
28	Written Report	11 days?	Wed 11/4/09	Tue 11/17/09
29	Machine Shop Prototyping	16 days?	Tue 11/17/09	Fri 12/4/09
30	**Design Review #4**	**7 days?**	**Tue 12/1/09**	**Wed 12/9/09**
31	Hardware Installation / Assembly	2 days?	Fri 12/4/09	Mon 12/7/09
32	Validation Testing At EPA	2 days?	Tue 12/8/09	Wed 12/9/09
33	Hybrid System Alterations/Optimization	2 days?	Tue 12/8/09	Wed 12/9/09
34	Oral Presentation	3 days?	Tue 12/1/09	Thu 12/3/09
35	Troubleshoot/Prepare Xebra	5 days?	Fri 12/4/09	Thu 12/10/09
36	Design Expo	1 day?	Fri 12/11/09	Fri 12/11/09
37	Finalize Budget/Submit Reimbursements	7 days?	Fri 12/4/09	Mon 12/14/09
38	Final Report	2 days?	Mon 12/14/09	Tue 12/15/09
39	Return Prototype/Clean Area	5 days?	Wed 12/16/09	Tue 12/22/09
40				

Appendix AC: Safety Report

EXECUTIVE SUMMARY

The Xebra is an experimental electric vehicle that is outfitted with a hydraulic regenerative braking and launch assist system. Working with this vehicle presents some special safety concerns because it operates under high pressures and at relatively high speeds. The purpose of this report is to identify the parts of our project which may be hazardous and detail how we plan to safely address those hazards. For simplicity, we have addressed these areas in the following sub-sections.

Initial experimentation: In order to benchmark the initial performance of the Xebra at the beginning of the semester, we conducted a few experiments in a parking lot. Before conducting these experiments, we addressed all possible safety concerns in a small informal safety report which can be found in Appendix A. After conducting these experiments, we determined that there were some problems with some of the systems on the Xebra. To determine the exact cause of these problems, we needed to conduct several other small experiments. Informal safety reports for these experiments can also be found in Appendix B.

Design: In order to make sure that we have addressed all the possible failure modes associated with our design, we have performed a DesignSafe analysis. This analysis showed that the main dangers associated with our design are the high operating speed and the high pressures stored within the hydraulic system. We have completed a rigorous engineering analysis on all of the components of our design to ensure that our design will be safe and robust. Through this analysis we have shown that all the systems on the Xebra vehicle will operate within a safety factor of at least three.

Manufacturing: The specific processes involved in our manufacturing are turning, milling, cutting, and drilling. The hazards associated with these processes are sharp tools and edges, fast spinning machines, and flying debris. To ensure safety during our manufacturing processes, all team members have completed the ME 450 machine shop training, which covers all of the necessary processes for our manufacturing. Safety glasses will be worn at all times while manufacturing and all processes will be conducted in the ME 450 machine shop under proper supervision.

Assembly: The assembly of the vehicles systems is a relatively low risk procedure. However, special care needs to be taken when assembling the hydraulic circuit to ensure that no lines or fittings will leak or rupture. To ensure that this circuit is assembled properly, we will connect all parts of the hydraulic system under the supervision of experts, and we will have them sign off on our work. The rest of our assembly will mainly consist of press-fitting bearings and sprockets together as well as mounting two new accumulators. Although these procedures are very low risk, every precaution will be taken, and safety glasses will be worn at all times.

Testing/ validation: We plan to validate our design by testing the vehicle on the dynamometer at the EPA. To do this, we will have to drive the vehicle over to the EPA which poses several safety concerns. These concerns have been taken into consideration, and are addressed in Appendix A. The rest of the testing and validation is relatively low risk. While on the dynamometer, all of the equipment including the vehicle will be operated by professionals at the EPA. After completion of the testing, the vehicle will be brought back to the university following the same procedure outlined in Appendix A.

EXPERIMENTATION PLANS PRIOR TO DESIGN COMPLETION

Several experiments were conducted prior to our design completion. First, we needed to test the performance of the Xebra at the beginning of the semester to get a baseline against which we could measure our improvements. A separate safety report was written for this procedure. This report can be found in Appendix A.

To get more accurate results, we also tested the vehicle on the dynamometer at the EPA. This involved driving the vehicle to the EPA. Due to the safety concerns associated with this procedure, a separate safety report was written which detailed the rout to be taken and any extra safety precautions that needed to be taken. This report can be found in Appendix A.

While conducting the tests at the EPA and in the parking lot tests we found that the regenerative braking system was malfunctioning. For this reason we designed an experiment to troubleshoot this system. To address the safety concerns with this experiment, a separate safety report was written. This report can be found in Appendix B.

PURCHASED COMPONENT AND MATERIAL INVENTORY

A number of components will be purchased to complete the final prototype. Only one piece of raw material will be required to make the accumulator mounting beams for the final prototype shown on the CAD drawing on page 10 of the prototype. The new components that are purchased will be used in combination with existing components that are currently installed on the Xebra.

Raw Material Inventory: We purchased one piece of raw material that is rectangular steel tubing. This tubing will be used to mount the high pressure accumulators.
Accumulator Mounting Beams:
Material: Steel (6A)
Shape and Dimension: Rectangular Steel Tube (2.5" x 1.5" x 11')
Source: Alro Metals Plus

Description: The steel stock material will be cut into six parts. Four of those parts will be used as the mounting beams and have the dimensions 2.5" x 1.5" x 6". The other two sections of the rectangular tube will be used for protection against side impacts. The dimensions of these two beams will be 2.5"x1.5"x38.5". These tube sections will be welded onto the frame of the Xebra using TIG welder.

Purchased Component Inventory: The system that we design deals with high pressures up to 4000Psi, we had to purchase the high pressure accumulators, the hydraulic fittings and the pressure relief valve from reputed manufacturers. We decided that the new accumulators should be purchased from the manufactures of the already installed accumulators that is Parker-Hannifin Corporation. All the hydraulic fittings will be order from the same supplier so as to maintain consistency in quality, that is from Tompkins Industries Inc. We also needed to change the existing gear ratio, so we decided to purchase the required sprockets and bearings from McMaster-Carr.

24.2" Wheel Set
Quantity: 2
Supplier: Discount Tire

Part Number: 83-4414

Description: Two 24.2” inch diameter wheels will be purchased. This will allow us to obtain the desired speed without altering the gear ratio within the electric motor gear box. This wheel is designed for the bolt pattern present on the Xebra so no extra machining will be required.

High Pressure Accumulator
Quantity: 2
Supplier: Parker-Hannifin Corporation (Exotic Automation)
Part Number: ACP 10AA800 E1KTE

Description: The new high pressure accumulators are the same model as the ones already installed in the Xebra. These accumulators are rated for a pressure of 4000Psi; the system pressure will not exceed 3850Psi. The high pressure accumulators have a safety factor of 4. The new accumulators will allow us to store enough hydraulic potential energy to achieve a speed of 35MPH.

Accumulator Mounting Brackets
Quantity: 2
Supplier: Exotic Automation
Part Number: PHZ 8700110476

Description: The accumulator mounting brackets that have been purchased they are designed specifically by the manufacture for the accumulators that we will be using. These brackets have also been used by previous teams to mount the currently installed accumulators.

Electric Motor Cooling Fan
Quantity: 1
Supplier: McMaster Carr
Part Number: 2059K12

Description: The electric motor currently heats up cause the electrical system to shut off. The electric motor did have a cooling fan which was removed by a previous team. The cooling fan has a flow rate of 105cfm and draws 4.1 amps of current at 12V DC.

Hydraulic Pump Sprocket
Quantity: 1
Supplier: McMaster Carr
Part Number: 6793K195

Description: The sprocket has 19 teeth, a bore size of ¾” and an outside diameter of 4.95inch. It is a machinable sprocket. It is made of steel and uses a ANSI # 60 chain. This sprocket will allow us to obtain the desired gear ratio for the hydraulic pump, 4.8:1.

Hydraulic Motor Sprocket (electric motor shaft)
Quantity: 1
Supplier: McMaster Carr
Part Number: 6793K152

Description: The sprocket has 20 teeth, a bore size of 5/8” and an outside diameter of 3.46inch. It is a machinable sprocket. It is made of steel and uses a ANSI # 40 chain. This sprocket will allow us to obtain the desired gear ratio for the hydraulic pump, 5.2:1.

Hydraulic Motor Sprocket (Hydraulic motor shaft)
Quantity: 1
Supplier: McMaster Carr
Part Number: 6793K148

Description: The sprocket has 17 teeth, a bore size of 5/8” and an outside diameter of 2.98inch. It is a machinable sprocket. It is made of steel and uses a ANSI # 40 chain. This sprocket will allow us to obtain the desired gear ratio for the hydraulic pump, 5.2:1.

One way Bearing (Hydraulic Motor Shaft)
Quantity: 1
Supplier: McMaster Carr
Part Number: 6392K52

Description: The one way bearing will stop the hydraulic motor running when it is not in use. The one way bearing that is currently installed is very badly damaged and it was not strong enough to withstand the max applied torque. This one way bearing has a locking torque of 89.245 ft-lbs and a max rpm of 3900 both of which are greater than the applied max by the hydraulic motor.

Hydraulic Motor and Pump Ball Bearings
Quantity: 4
Supplier: McMaster Carr
Part Number: 60355K508

Description: These ball bearings are connected allow the shafts to rotate in the hydraulic pump and motor to the mounting. These are made of stainless-steel. The inner diameter is 7/8” and the outside diameter is 1-7/8”. The max RPM is 9300, which is much greater than the max RPM of the system (3300RPM).

Pressure Relief Valve (4500Psi)
Quantity: 1
Supplier: Sun Hydraulics
Part Number: RPGS-CWN-CAKS

Description: The pressure relief valve trips if the pressure in the system is above 4500Psi. The pressure relief valve once tripped dumps all the hydraulic fluid from the high pressure accumulators to the low pressure accumulator.

Synthetic Transmissions Fluid
Quantity: 17 Quarts
Supplier: Mobil One
Part Number: 10W30

Description: The transmission fluid is used as a means to store hydraulic potential energy. It is pumped from the low side accumulator to the high pressure accumulator during braking and it flows from the high pressure accumulator to the low pressure accumulator during acceleration. You can find the MSDS for this product in Appendix I.

Elbow SAE to JIC
Quantity: 2
Supplier: Tompkins Industries Inc.
Part Number: 6801-16-16
Description: This fitting is made of forged steel. It is used to connect the accumulators to the rest of the fittings. It is a male to male fitting. The SAE side connects to the accumulators and the JIC connects to the rest of the fittings. These fittings are rated for 4000Psi.

Union JIC
Quantity: 2
Supplier: Tompkins Industries Inc.
Part Number: 6501-16-16

Description: This fitting is made of forged steel. It is a female to female joint. These fittings are rated for 4000Psi. These fittings are used to connect to the already installed fittings.

Extender JIC
Quantity: 2
Supplier: Tompkins Industries Inc.
Part Number: 6801-16-16

Tee SAE-JIC- JIC
Quantity: 2
Supplier: Tompkins Industries Inc.
Part Number: FG6804-16-16-16

Description: This fitting is made of forged steel. It is a three way valve of male to male to male..These fittings are rated for 4000Psi. These fittings are used to connect the old accumulators to the new accumulators.

Elbow JIC
Quantity: 2
Supplier: Tompkins Industries Inc.
Part Number: FG6500-16-16

Description: This fitting is made of forged steel. It is a male to female joint. These fittings are rated for 4000Psi. These fittings are used to connect to the already installed fittings.

FMEA Analysis of purchased components

A FMEA analysis was conducted on the purchased components that would either have the most severe impact if they fail or have the highest probability of failure. These components included the high pressure accumulator, the electric motor cooling fan, the pressure relief valve, the one way bearing on the hydraulic motor and all of the hydraulic fittings (Appendix C). A complete failure of the hydraulic fittings and the high pressure accumulators would be very severe as there would be debris flying around and spraying of high pressure fluid. A complete failure is very unlikely to occur as the fittings and the accumulators are rated for 4000Psi. A less serve failure can also be possible if the sealings are not tightened to the correct torque rating. The consequence of the electric motor cooling fan is not that severe as the electric motor would heat up but the motor has a safety feature that would not allow the temperature to exceed 135^{o}C. This failure could occur if there is current spike or if the cooling fan is not assembled properly.

DESIGNSAFE SUMMARY FOR DESIGNED PARTS

To ensure that our design would function safely we used DesignSafe, which not only helps you document your safety concerns and precautions, but helps you to fully explore the safe operation of your design. Since we used FEA and static analysis on the components that we manufactured and the inherent safety concerns associated with the high pressure hydraulic system, we decided to do a DesignSafe on the high pressure hydraulic system as well as the whole vehicle. The risk of explosion or rapid depressurization and the following expulsion of the hydraulic fluid were the major safety concerns. In order to design against these we made sure that the hydraulic system is assembled correctly with a safety factor of four on the crucial components. Additionally the hydraulic system will have a cover, which is the truck bed.

While using DesignSafe for the entire vehicle, we left out the major concerns already addressed in the DesignSafe of the high pressure hydraulic system, although any concerns that stem from that system and were not formerly covered were included. The most significant risk involved with the Xebra vehicle is its high speed use on public roads. To design against this, the braking system includes the traditional disk brakes in addition to the hydraulic braking system to ensure proper braking. There is no way to completely design against every danger you will encounter on the roads and this is up to the user, although we will provide a thorough operation manual which will help the vehicle be used safely and help prevent the user from entering an unsafe situation. Other major dangers included the electrical system and the following risk of electric shock and fire if exposed to hydraulic fluid. For this reason we are going to put the hydraulic electrical components into a box and we will tighten and visually inspect the electrical system connections.

Printouts from the DesignSafe analysis can be found in Appendix D.

MANUFACTURING

Although our project is comprised of complex mechanical and electrical systems, our overall machining and fabrication will be relatively minimal. This is because we are keeping many of the current systems in tact while making only slight modifications. The majority of our fabrication will be focused on modifying the vehicles gearing and also on mounting the additional accumulators. The following sub-sections describe each component that will be fabricated and the steps and details associated with each process. All of the machining and fabrication will be completed in the ME 450 machine shop. For detailed engineering drawings of the components being fabricated, see Appendix E.

Hydraulic pump and motor sprockets: In order to achieve the proper gear ratios, we are replacing the sprockets that connect directly to the hydraulic pump and the hydraulic motor. Since the shafts on these components use a non-standard keyway we were unable to purchase pre-finished sprockets. Therefore, we have ordered two machinable sprockets from McMaster-Carr that come with a standard bore size of 0.625 inches. The shafts on both the pump and motor are identical, so the machining of both sprockets will be exactly the same. First, the sprockets will be bored out on a lathe to a size of 0.875 ± 0.001 inches in order to accommodate the shafts on the hydraulic pump and motor. Then, using a special keyway tool in the ME 450 machine shop, we will cut a keyway that will accommodate a square key of dimensions 0.25 x 0.25 ± 0.001 inches. Details for the tooling and speed of these operations can be seen in Tables 1 & 2 below. It is important to carefully match the specified tolerances for these operations. This will ensure that the sprocket-key assemblies will mate properly with the motor and pump shafts.

Table 1: Machining processes for the hydraulic pump sprocket.

Raw Material Stock:	Steel Machinable-Bore Sprocket for #60 Chain, 3/4" Pitch, 19 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck
3	Bore out inner diameter to tolerance (0.875" ± 0.001)	lathe	800	Boring bar	4 Jaw Chuck
4	Cut keyway for 0.249 x 0.249" ± 0.001 key	Press	-	keyway tool	Vice

Table 2: Machining processes for the hydraulic motor sprocket.

Raw Material Stock:	Steel Machinable-Bore Sprocket for #40 Chain, 1/2" Pitch, 17 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck
3	Bore out inner diameter to tolerance (0.875" ± 0.001)	lathe	800	Boring bar	4 Jaw Chuck
4	Cut keyway for 0.249 x 0.249" ± 0.001 key	Press	-	keyway tool	Vice

Hydraulic motor sprocket (drive shaft): Apart from replacing the sprocket that connects directly to the hydraulic motor, we are also replacing the sprocket that links the hydraulic motor to the drive shaft. This fabrication process will be slightly different from the other sprockets because the bore of this sprocket will mate with a one-way bearing instead of a keyed shaft. Since the outer diameter of the bearing is not a standard size and requires a precise press fit, this sprocket will also need to be machined. Therefore, we have ordered a machinable sprocket from McMaster-Carr that comes with a standard bore size of 0.625 inches. The sprocket will be bored out in a lathe to a bore size of 1.653 + 0.000, – 0.002 inches. This tolerance is very important because the bearing requires a specific press fit and will not function properly unless this tolerance is met. Table 3 below, shows a detailed breakdown of the speeds and tooling needed to fabricate this part.

Table 3: Machining processes for the hydraulic motor sprocket (drive shaft).

Raw Material Stock:	Steel Machinable-Bore Sprocket for #40 Chain, 1/2" Pitch, 20 Teeth, 5/8" min Bore				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	4 Jaw Chuck
2	Locate center of part	lathe	-	Dial Indicator	4 Jaw Chuck

3	Bore out inner diameter to tolerance (1.654" + 0.000, - 0.001)	lathe	800	Boring bar	4 Jaw Chuck

One-way bearing spacer: Because the inner diameter of the one-way bearing is larger than the drive shaft, a spacer needs to be fabricated that will essentially increase the drive shaft diameter to fit the bearing. To create this part, we will start with round AISI 1020 steel stock and turn the outer diameter to 1.378 + 0.000, -0.0005 inches on a lathe. We will then drill the center of the shaft to an inner diameter of 0.750 inches. We will then bore out the inner diameter to a dimension of 0.875 + 0.001, -0.000 inches to accommodate the drive shaft. The tolerances on both the inner and outer diameters of this part are very important to ensure that the one-way bearing will function properly. After ensuring that the tolerances have been met, the part will be cut to a length of 1.5 ± 0.050 inches using a cut-off tool. The final step in creating this part is to create a keyway that will accommodate a square key of dimensions 0.1875 x 0.1875 ± 0.001 inches. This will be done using a special keyway tool in the ME 450 machine shop. Table 4 below shows a detailed breakdown of the speeds and tooling needed to fabricate this part.

Table 4: Machining processes for the one-way bearing spacer.

Raw Material Stock:	1020 Steel Round Stock, 1.5” Diameter, 3.0” Length				
No.	**Process Description**	**Machine**	**Speed (rpm)**	**Tool**	**Fixtures**
1	Hold part in chuck	lathe	-	-	3 Jaw Chuck
2	Face off end of stock	lathe	900	Turning Tool	3 Jaw Chuck
3	Turn down outer diameter to tolerance (1.378 ± 0.001)	lathe	900	Turning Tool	3 Jaw Chuck
4	Center Drill end of part	lathe	1600	Center Drill	3 Jaw Chuck
5	Drill inner diameter undersized	lathe	1600	Ø 0.75” Drill	3 Jaw Chuck
6	Bore out inner diameter to tolerance (0.875” + 0.001, -0.000)	lathe	800	Boring bar	3 Jaw Chuck
7	Cut part to length (1.5” ± 0.050)	lathe	500	Cut-off tool	3 Jaw Chuck
8	Cut keyway for 0.249 x 0.249" ± 0.001 key	press	-	keyway tool	Vice

Accumulator mounting beams: In order to mount the new accumulators, we need to create new mounting bars that will be welded to the frame of the vehicle. These bars will require very little fabrication. We will start with 13 feet of stock which needs to be cut into six different pieces, two at a length of 38.5” and 4 at a length of 6.25”. These pieces will be cut to length using the stock cut-off saw. This saw is essentially a horizontal band saw that is used to cut stock to length. Any rough edges will then be cleaned up with sand paper to ensure the pieces meet up correctly before they are welded to the frame. The tolerances for these beams are not that crucial. The length for each beam should be within 1/16th of the specified length.

Table 5: Machining processes for the accumulator mounting beams.

Raw Material Stock:	1020 Steel Extrusion; 1.5” Width, 2.5” Height, 13’ Length, 0.125” Wall Thickness				
No.	**Process Description**	**Machine**	**Feed (ft/min)**	**Tool**	**Fixtures**

1	Clamp part at specified length (four at 6" and two at 38.5") ± 0.0625	Cut-off saw	-	Cut-off saw	clamp
2	Cut part to length, remove, repeat	Cut-off saw	300	Cut-off saw	clamp
3	Clean up rough edges	-	-	Sand Paper	-

Welding mounting beams to frame: After the accumulator mounting beams have been cut, they will be welded onto the frame. The first step in this process will be to wire-brush the paint off of the frame of the vehicle. This will ensure that the beams will be welded properly to the frame and no paint will be burnt in the process. We will then have Bob Coury assist us with TIG welding the beams to the frame. After the welds are finished and cooled, we will need to grind away any excess weld material to make sure that all of the surfaces are flat.

Drilling holes for accumulator brackets: Once the accumulator mounting beams are welded to the frame, we can install the accumulator mounting brackets. We will first put the accumulators inside the brackets and place the brackets on the mounting beams. We will then install the fittings between the two accumulators while someone holds the un-mounted accumulator securely. This will allow us to determine the exact spacing between the accumulators so that the hydraulic fittings work properly. We will then make marks on the mounting beams where the holes will need to be drilled for the new accumulator brackets. Finally, we will use a hand drill and a 0.5" diameter drill bit to make the proper mounting holes.

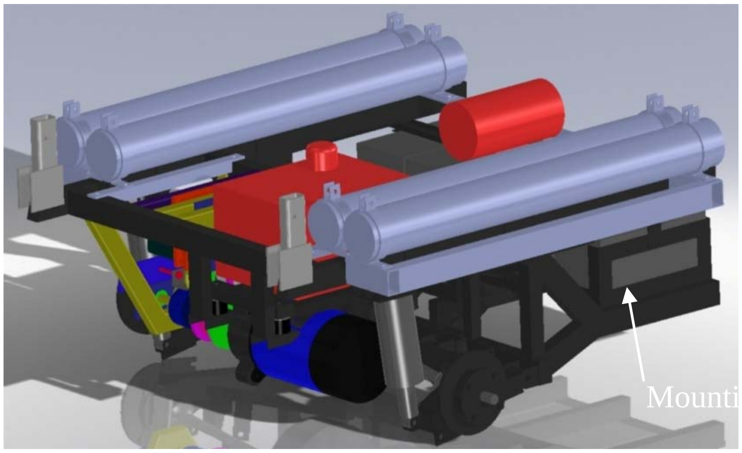
ASSEMBLY

Whenever parts are being assembled on the Xebra vehicle, the electrical system will be turned off and unplugged, and the hydraulic system will be depressurized. All assembly and disassembly will take place in the X50 lab or the ME 450 Machine shop.

Disassembly: In order to reach all of the components in the system that need to be modified, we had to disassemble some of the Xebra's systems. The procedure for disassembly can be found Appendix F.

Accumulator Mounting: The new accumulators will be mounted onto 2 ½"X 1 ½" 11 gauge AISI 1020 hot rolled steel beams of 6 ¼" lengths. These mounting beams will be TIG welded to the existing frame beam shown in Figure 1. The accumulator mounting brackets will be bolted to the mounting beams using the ordered brackets/bolts. Another beam of the same cross sectional dimensions will be TIG welded between the mounting beams to protect the accumulators. This assembly will take place in the X50 lab and in the machine shop. The beam analysis shown in Appendix G determined that the mounting beams have a safety factor above 200. The welding will be done with the supervision of Bob Coury to ensure that the beams do not fail during assembly.

Figure 1: Overall vehicle assembly including new accumulators and mounting



Sprocket Gearing: Sprockets on the hydraulic pump and motor will use a 3/16" X 3/16" key and a 10-32 set screw to constrain them to the shafts. The sprocket on the shaft that is attached to the electric motor shaft, shown in Figure 2, will have a one way bearing press fit into the center bore of the sprocket. A shaft collar with a 3/16" key on the inside bore will be placed inside the one way bearing. This keyed shaft collar will be placed on the shaft that is attached to the electric motor and will be constrained using a 3/16" X 3/16" key and two other two piece clamp on shaft collars with 10-32 set screws. This assembly will take place in the X50 lab and in the machine shop. FEA was done on the sprockets and the shaft collar (Appendix H) and a minimum safety factor of 5 was determined. The torque on the one way bearing was determined to be below the manufacture's maximum rated torque. The one way bearing manufacture's tolerances will be used in machining the keyed shaft collar and the sprocket to ensure it does not fail during assembly.

Figure 2: Assembled view of one-way locking sprocket

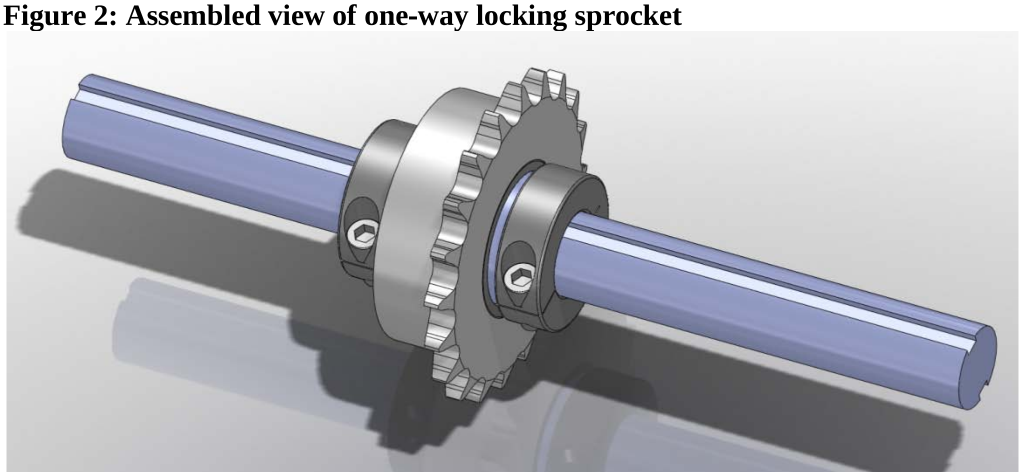
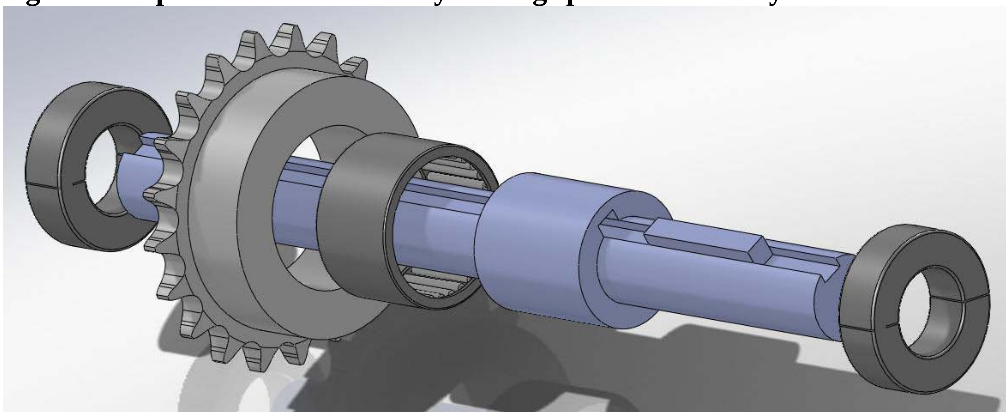
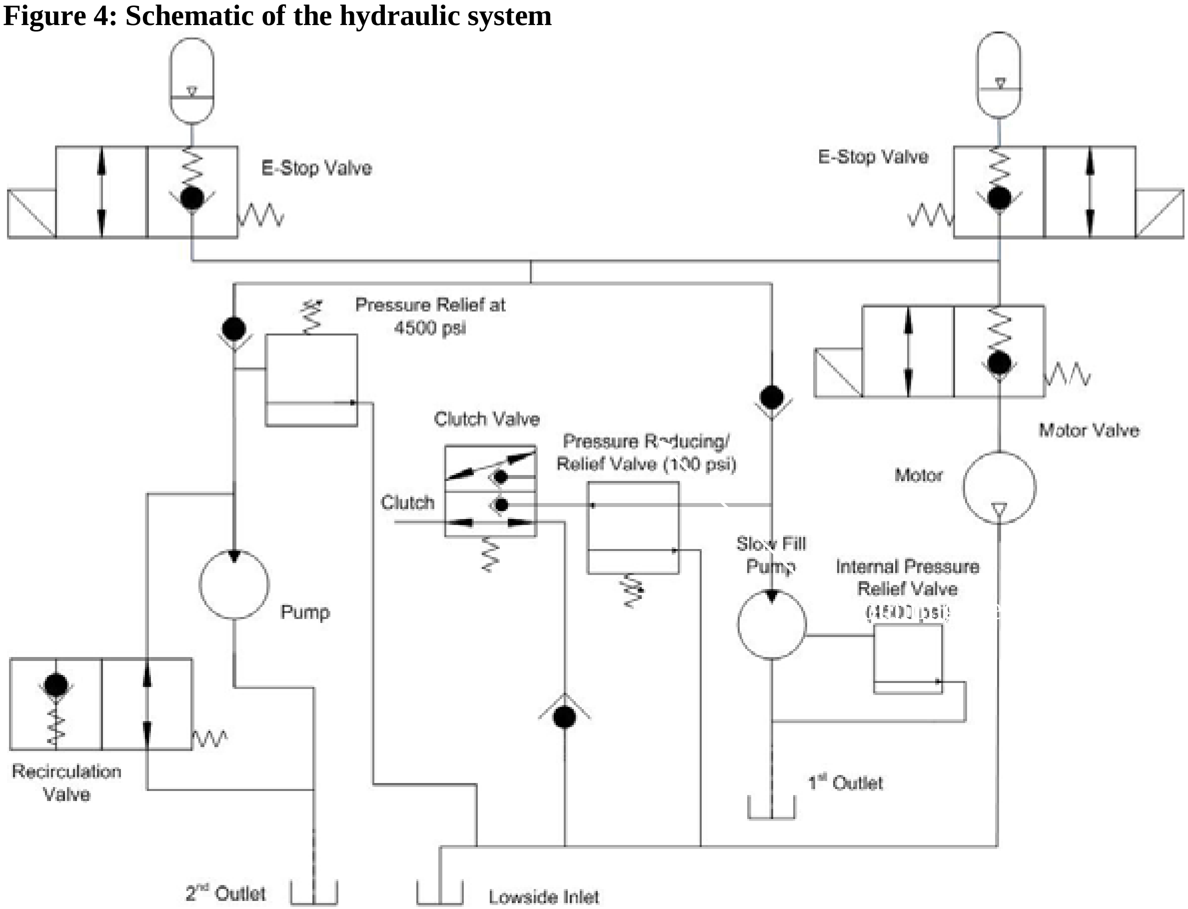


Figure 3: Exploded view of one-way locking sprocket assembly



Hydraulic System: A schematic of the hydraulic system can be seen below in Figure 4. The hydraulic system was drained of hydraulic fluid and disassembled in order to get access to components that needed to be replaced or modified for our prototype. Since the hydraulic fluid has been fully drained from the system, there will not be as much risk during assembly as there was during disassembly. The hydraulic system will be assembled in the X50 lab. All o-rings will be replaced and silicon tape will be used on the threads of the hydraulic fittings to prevent hydraulic fluid leaks during use. This assembly will be verified by certified specialists at the EPA to ensure that the hydraulic system will not fail during use at high pressures.

Figure 4: Schematic of the hydraulic system



DESIGN TESTING AND VALIDATION

The following describes our plan for validating our final design. This validation will entail moving the Xebra to the EPA which poses several safety concerns in itself. Since we have successfully moved the Xebra to the EPA in the past, we plan to follow the same procedure for moving the Xebra for validation. This procedure is detailed in Appendix A.

Primary Method Validation

Our sponsor, the EPA, wants our design to be tested on a LA4 drive cycle, shown in Appendix AB, on their dynamometer. Because a portion of the drive cycle requires the vehicle to exceed the 45 mph top speed specification, the vehicle will be at its top speed for this duration. When this portion of the drive cycle is reached, the top speed specification will be verified. During the dynamometer test, the vehicle's acceleration, velocity, range, electric motor temperature, and electrical power draw will be monitored and recorded. The vehicle will run consecutive LA4 cycles until the batteries run out, therefore validating the maximum range. A separate test on the dynamometer will record the speed reached when launching the vehicle using only hydraulics. The results from dynamometer tests will validate that our design has met the engineering specifications shown in Table 6 below.

The power supplied to the electric motor will be limited by the electric motor controller. We will set the controller to limit the power sent to the electric motor to 5kW, so that the electric motor only operates at an efficient power draw from the batteries. See Appendix M for battery specifications. A voltmeter and an ammeter will be used to monitor the vehicle's power draw during dynamometer testing. The monitoring equipment will be provided by and installed by the EPA.

To verify that the vehicle is stable, a vehicle weight scale located at the EPA will be used. The current vehicle, without any of our design alterations, was weighed earlier at the EPA. By weighing the vehicle a second time, with our design alterations, we can determine how much weight our design has added and also the new total weight of the vehicle. This will verify the stability customer requirement along with the weight specification.

Also to verify that the vehicle is stable, we plan on using the same weight scale at the EPA, but measure the weight at the rear axle and the weight at the front tire separately. From these measurements, the position of the center of mass can be determined. This will verify the stability customer requirement along with the distance from rear axle specification.

Table 6: Primary and secondary methods that will be used to validate the specifications

Engineering Specification	Primary Validation Method	Secondary Validation Method
Overall Top Speed of Vehicle	Dynamometer Testing	Outdoor Testing/RPM Sensor
Max. Speed using only Hydraulics	Dynamometer Testing	Outdoor Testing/RPM Sensor
Initial Acceleration	Dynamometer Testing	Outdoor Testing/RPM Sensor
Range	Dynamometer Testing	Outdoor Testing/GPS Unit
Temperature of Electric Motor	Dynamometer Testing	Thermocouple Sensor
Power of electric motor	Dynamometer Testing	Controller Software
Weight of vehicle	Weight Scale at EPA	Sum Weights of Added Components
Distance of C.O.M.* From Rear Axle	Weight Scale at EPA	Portable Scale
Accumulator Pressure	Pressure Gauge	Wireless Pressure Sensor

*C.O.M.= Center of Mass

Secondary Method Validation

Contingency methods can be seen in Table 6 above. These secondary tests will be performed if problems or issues arise using the primary testing methods.

Instead of using a dynamometer, outdoor testing on a road or in a parking lot may be substituted. During the outdoor test, the RPM sensor data will be used to determine the speeds as well as the accelerations of the vehicle during testing. A GPS unit will be used to record the distance the vehicle travels during testing. The thermocouple on the electric motor will be used to ensure that the motor does not overheat. The electric motor controller software will be used to monitor the power supplied to the electric motor during testing.

If the scale at the EPA cannot be used, the weight of the components that our design adds to the vehicle will be added to the previous recorded weight. Also a portable weight scale will be used to measure the weight on the front tire with the vehicle inclined at varying angles. From the weight and angle measurements, the position of the center of mass can be determined. The wireless pressure sensor on the vehicle can be used instead of using the pressure gauge.

Since this secondary method of validation is very similar to the parking lot testing that we have already completed safely, we will follow the same safety procedures again. These procedures are detailed in Appendix A.

SAFETY REPORT APPENDICES

Appendix A: Initial Xebra Testing

Before proceeding with any concept selection we will need to complete some tests to measure the performance of the Xebra vehicle in its current state. The data obtained in these tests will serve two specific purposes:

The data will be used to validate the current mathematical model we have derived for our system.

The data will give us a good baseline, against which, we can compare our final design's performance.

The test will show us if there are any leaks in the hydraulic system. If there are any leaks, this test will allow us to see where the leaks are.

The following will provide a summary of the current testing we have planned for the Xebra vehicle. This summary will address the safety concerns associated with this testing and how we plan to address those concerns.

Slow Speed Parking Lot Tests

We currently believe all of the systems on the Xebra to be fully operational. However, to ensure there will be no embarrassing surprises when we go to do the coast-down testing, we would like to conduct an initial slow speed test in one of the parking lots on north campus. During this test, we do not plan to exceed normal parking lot speeds, but to ensure the safety of those around us, we could conduct this test over the weekend. This would minimize the amount of traffic in the parking lot. Also, for this test we will be accompanied by Ben Hagan, a graduate student who has been working on this vehicle for quite some time, and also by David Swain from the EPA. If necessary, we could also try to reserve a parking lot for a period of time but this could substantially delay this testing. Keeping our timeline in mind, we would like to avoid that option if at all possible. The test will be conducted on the 20th of October at 1:00 p.m. We have decided to conduct multiple tests in order to ensure that we can charge the hydraulic system to max pressure, while ensuring that we stay under the speed limit of the parking lots.

Test Number 1: We will accelerate the Xebra up to 20mph using only the electrical system. This is done to pressurize the high side accumulators to approximately 2700 psi. In this test we will check if there are any internal or external leaks in the system. Using a stop watch and the pressure gauge that is already installed in the car, we will see if there is any drop in pressure against time. If there is no drop in pressure we can infer that there are no internal leaks. We will also look for any dripping of hydraulic fluid from the system, if there is dripping; we know there are external leaks. If there are any external leaks, we will first address and fix the external leakages if it is possible to do so and then proceed to the next test.

Test Number 2: We will accelerate the Xebra using only the electrical system up to 20 mph and decelerate using the hydraulic system; this will be repeated once more. This procedure will allow us store more energy into the hydraulic system without exceeding the speed limit of the parking lot. Then we will measure the pressure of the hydraulic fluid in the high pressure accumulator, this is done to collect data on the empirical working of the Xebra and to compare these values to the ones obtained from the Matlab model, to verify the accuracy of this model. Then we will launch the Xebra using only the stored hydraulic energy and measure the speed obtained and the time taken to reach the speed, this will again allow us to verify the mathematical models.

Test Number 3: The Xebra will be put through an acceleration and deceleration cycle four times to increase the amount of stored energy in the hydraulic system and to increase the pressure of the hydraulic fluid in the high pressure accumulator to about 3700 psi. Then the Xebra will be launched using only the hydraulic motor and by doing so we can see the maximum speed that we can get to using only the hydraulic motor in its current configuration. This test will also allow us to check if there are any leaks at high pressures.

Safety Consideration: All members of the team will be required to wear eye protection gear, while inspecting the Xebra for any leaks. The first test will be conducted so as to ensure that there are no leaks. If we find that there are external leaks, we will first fix them on the spot if it is possible. If the external leaks cannot be fixed on the spot, we will continue the tests on a later date. If there are internal leaks we can still continue the tests. Make sure that there are no pedestrians near the vehicle while it is pressurized.

E.P.A. Dynamometer Testing

The Xebra vehicle is scheduled to be tested at the EPA on October 28th. We plan to move the Xebra from the X50 lab on October 26th to ensure the vehicle will be ready for testing as soon as the dynamometer is available. The dyno testing itself poses no real safety concerns; however, the same cannot be said for transporting the vehicle to the EPA. Originally we had planned to rent a flatbed truck to transport the vehicle until we learned that in the past, the vehicle had simply been driven over to the EPA. We have chosen to go with this option for the following reasons:

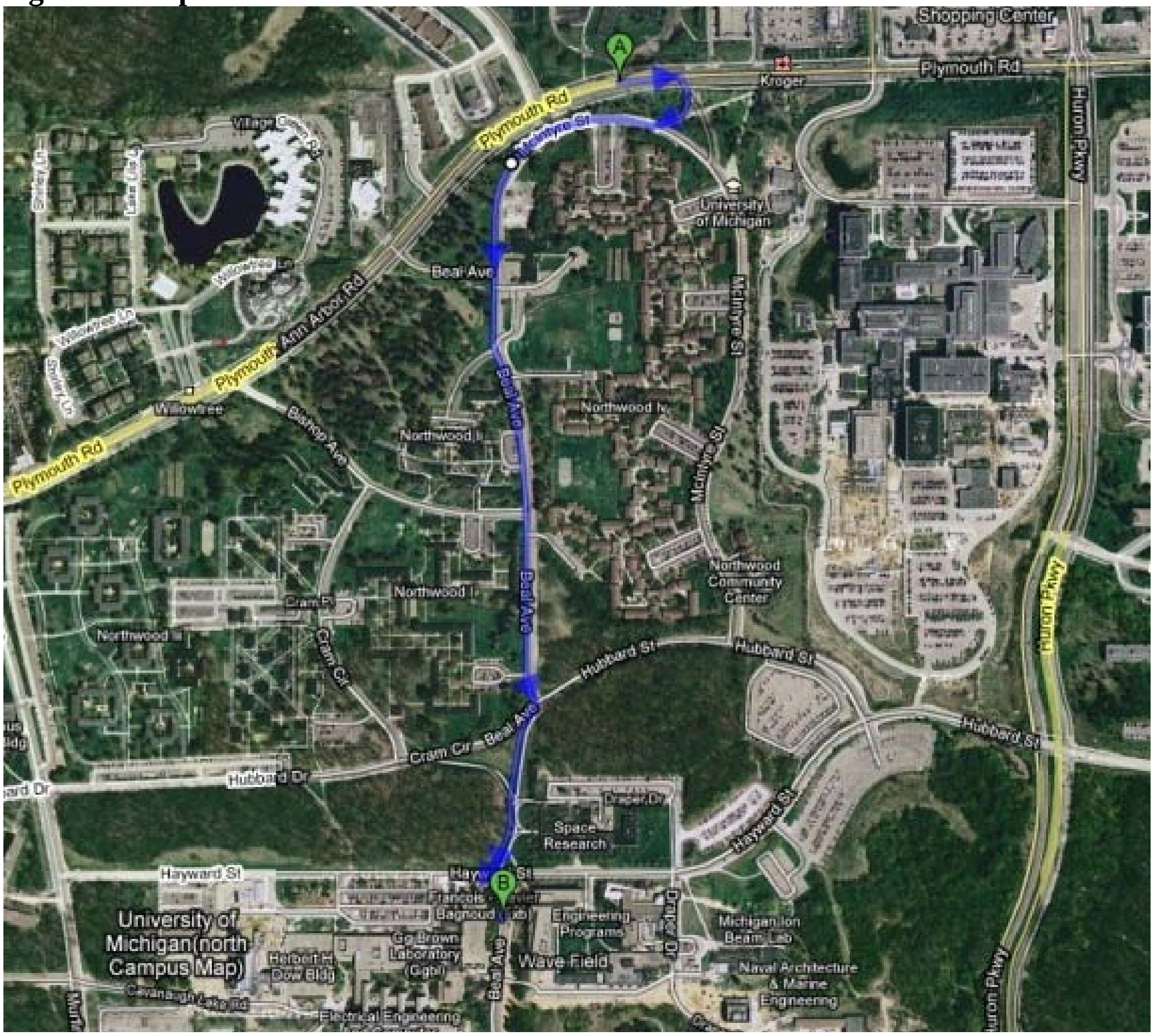
- Loading the vehicle onto a flatbed is a risky procedure which could result in damage to the vehicle.
- Renting a flatbed could cost as much as $200 which seems excessive for transporting the vehicle a mere 1.2 miles.
- This procedure has been conducted safely in the past with no complications.

Obviously because this is an experimental vehicle there is some safety concerns associated with driving it on public streets. The main issue that we have to address is making sure that no hydraulic energy is accidentally released in the process. We also need to ensure that the vehicle will not undergo any impacts. And finally, we need to ensure that the vehicle will not be subject to driving conditions which could result in rollover. In order to avoid any of these potential safety issues we plan to take the following actions:
To ensure that no hydraulic energy is accidentally released during transport, we will deactivate the hydraulic system and rely only on electric power to drive to the EPA.
To avoid impacts with other vehicles, we will ensure that all of the brake and turning lights are fully operational and we will drive with the emergency flashers on so other vehicles know to avoid our vehicle if possible. We will also try to avoid major roads as much as possible.
To avoid rollover we will choose a route which will minimize the driving speed and will allow us to execute all turns under a speed of 5 mph.

David Swain from the EPA has volunteered to drive the vehicle over for us because he has done it in the past and is comfortable with the procedure. He has also made us aware that he is covered by AAA insurance with a policy under which he is covered in any vehicle that he operates. This extra precaution will help eliminate any liability in the unlikely event that something does go wrong. To ensure that more people than necessary are put at risk, we will ensure that no other passengers accompany David on this trip.

In order to minimize the speed of the Xebra during transport and to ensure that there are no collisions with other vehicles, we are proposing the vehicle be driven along public walkways and low traffic streets. A proposed route is detailed in Figure 1 below. The vehicle will depart from GG Brown and head north up the public walkway towards Beal Ave. From there, the vehicle will follow Beal Ave. until it turns into McIntyre St. The vehicle will proceed down McIntyre St until it reaches another public walkway. The vehicle will then follow the walkway and cross Plymouth Ave. where it will arrive at the EPA. During this process we will make sure the vehicle does not exceed 10 mph on the public walkways and 20 mph on the public roads. Also, we will walk along with the vehicle to ensure that all pedestrians are out of harm's way.

Figure 1: Proposed route to and from the EPA.



Appendix B: Troubleshooting Regenerative Braking System

Summary

To diagnose the problem with the regenerative braking system, we want to run a few tests to determine if the pressure relief or check valve is malfunctioning. Safety precautions and a procedure for these tests are found below. Based on previous testing, we have determined that the hydraulic clutch that engages the pump, the recirculation valve that allows hydraulic fluid to flow through the pump, and the pump are currently functional. The components that we plan to isolate during testing to determine which component is malfunctioning are the pressure relief valve and the check valve. The failure of these components would cause the symptoms observed.

Testing Overview

To visually observe fluid flow into the low side accumulator, we would like to use a clear acrylic piece to cover the top of the low side accumulator. This would enable us to see if fluid is being displaced by the pump while it is rotating. When the regenerative braking system is engaged, no fluid should be dumped into the low side accumulator. If fluid is seen being dumped into the low side accumulator, we can determine that the pressure relief valve is releasing fluid at too low of a pressure causing the system to malfunction. If fluid is not being dumped into the low side accumulator, the check valve is not allowing fluid to flow into the high side accumulators.

The pump will be engaged using the clutch with the electric motor also engaged. Since the vehicle is raised off the ground, the electric motor will rotate the pump while the clutch is engaged. This will allow us to test the regenerative braking system without moving the vehicle from its current position.

Safety Precautions/Concerns

Hydraulic Fluid Leak: Absorbent PIG mats will be placed around the low side accumulator and under the vehicle to collect any spills. The first test will be done with PIG mats completely containing the low side accumulator. The acrylic piece on the low side accumulator will not have any pressure exerted on it since the low side accumulator is exposed to atmospheric pressure using a valve. Leaks may occur if the hydraulic fluid penetrates the seals around the acrylic cover when fluid is being pumped into the low side accumulator. Safety glasses will be worn at all times while in the lab. Latex gloves will be worn when touching any hydraulic fluid.

Hydraulic Fluid Spray: A shield will be placed over the hydraulic to contain any hydraulic fluid that may spray.

Electrical System Operation: The vehicle will be turned off with the keys removed when people are around the vehicle with the shielding removed. This will prevent any accidental operation of the electrical motor and/or hydraulic system.

Pressure Buildup: To engage the hydraulic clutch, the slow fill pump needs to be running. This will cause pressure to accumulate in the high side. Someone will monitor the pressure during testing to ensure that it does not exceed 2700 psi. The first and second tests will run for 5 sec. to ensure that the slow fill pump does not have enough time to exceed 2500 psi (2500 psi is the lowest accumulator pressure). If the problem is still not determined after the second test, subsequent test durations will be incrementally increased by five seconds. Subsequent testing duration can be increased until the pressure reaches 2700 psi. Once 2700 psi is reached, subsequent testing duration will be limited and will no longer be incrementally increased. The pressure will be released after each test by running the hydraulic motor with the friction brakes engaged to prevent the wheels from spinning without any load.

Test Procedure

Ensure all safety measures are in place and that everybody is in a safe position.
Engage the slow fill pump valve to ensure pressure to the clutch
Engage the recirculation valve to ensure fluid is not being recalculated by the pump
Turn on the electrical system/place keys in ignition
Engage the electric motor for around 5 sec.
Engage the hydraulic clutch/pump for 5 sec. for the first and second tests (incrementally increased by 5 sec.)
While the pump is spinning, visually determine if fluid is flowing into the low side accumulator
Disengage hydraulic clutch/pump
Disengage electric motor
Apply friction brakes
Engage hydraulic motor to release any pressure
Ensure all pressure is released
Turn off the electrical system/remove keys from ignition
Test may be repeated to replicate results

Appendix C: FMEA for purchased components

Appendix C: FMEA

Project# 24 Date: 11/29/2009
Project Title: Xebra Electric-Hydraulic Hybrid
Team Members: Alexander Douggan, Andrew LaNoue, Mike Makowski, Aazam Vishram

Part Number, Name and Function	Potential Failure Mode	Potential Effects of Failure	Severity (S)	Causes of Failure	Occurrence (O)	Current Design Controls/ Tests	Detection(D)	Recommendation	RPN (SxOxD)
Accumulator Mounting Beams	Beam loosely connected to the frame	Noise and vibrations	2	Improper welding	1	Visual inspection by Bob Courey . Manual Test	1	Ensure welding is performed at a slow speed and high power for max penetration depth.	2
	Fracture of beams	Flying debris and high pressure fluid leak	5	Defects in manufacturing and side impact on the Xebra	1	Built with a safety factor of appx280	2	Ensure welding is performed at a slow speed and high power for max penetration depth.	10
High Pressure Accumulators	Loose fitting and incomplete seal	Leak of hydraulic fluid	4	Manufacture defect	1	The rated pressure is 4000Psi. Where as max system pressure is 3850Psi.	1	Test assembly with slowly increasing pressure. Watch / listen for leaks.	4
	Fracture of accumulator	Flying Debris and leaking high pressure fluid	5	Manufacturing Defect	1	Accumulator designed for leak before rupture	1	Test assembly with slowly increasing pressure. Watch / listen for leaks.	5

Electric Motor Cooling Fan	Malfunction of fan	Overheating electric motor	3	Loose electric connection, manufacturing defect	2	Visual inspection and manual test using thermo couples.	3	While running the Xebra on the dyno check thermocouple for over heating	18
Pressure relief valve	Dumping of fluid during braking	Malfunction of regenerative braking	5	Manufacture defect	2	Test described in report on page__	3	Test regenerative braking to ensure it is functioning	30
One way bearing	Spinning of hydraulic motor when it is not in use	Loss of efficiency	2	Manufacture defect	2	Visual and auditory inspection	4	Check for metal fillings from bearing during dyno test.	16
	Fracture of bearing	Malfunction of hydraulic motor	5	Manufacture defect	2	Visual and auditory inspection	1	Check for metal fillings from bearing during dyno test.	10
Hydraulic fitting	Loose fitting and incomplete seal	Leak of hydraulic fluid	1	Manufacture defect and assembly defect	4	The rated pressure is 4000Psi. Where as max system pressure is 3850Psi	2	Test assembly with slowly increasing pressure. Watch / listen for leaks.	8
	Fracture of fitting	Flying debris and high pressure fluid leak	5	Manufacture defect and assembly defect	1	The rated pressure is 4000Psi. Where as max system pressure is 3850Psi	1	Test assembly with slowly increasing pressure. Watch / listen for leaks.	5

Appendix D: Design Safe printouts

designsafe Report

Application: High Pressure System
Description:
Product Identifier:
Assessment Type: Detailed
Limits:
Sources:

Analyst Name(s):
Company:
Facility Location:

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 : stabbing / puncture The impact from another vehicle could puncture the high pressure accumulator causing rapid discharge of hydraulic fluid.	Serious Occasional Possible	High	A steel bar will be welded to the frame to sheild the accumulator from the direct impact of a side collision. There is also a safety factor of four on the accumulators.	Serious Occasional Unlikely	Moderate	
All Users All Tasks	chemical : failure at key points and trouble spots The failure of any fittings connected to the high pressure system could cause fluid to vaporize or to shoot out of the system.	Serious Remote Possible	Moderate	All fittings will be put on using a torsion wrench so they have the right amount of torque applied. The high pressure system will be covered by the steel truck bed.	Serious Remote Unlikely	Moderate	
All Users All Tasks	chemical : chemical / toxicity effects felt at distant time / place The hydraulic fluid is toxic if it enters the blood stream and the vaporized hydraulic fluid is toxic if breathed.	Serious Occasional Possible	High	The truck bed will be sheilding the high pressure system. There is also a safety factor of four on the accumulators and hoses.	Serious Occasional Unlikely	Moderate	
All Users All Tasks	fluid / pressure : hydraulics rupture If any hoses, fittings, or accumulators rupture hydraulic fluid could vaporize or shoot out of the system. Explosion is also possible if the pressure is exceeded.	Serious Occasional Possible	High	The truck bed will be sheilding the high pressure system. There is also a safety factor of four on the accumulators and the hoses. The high pressure hydraulic system has a safety release valve which will release pressure at 4500 PSI.	Serious Occasional Unlikely	Moderate	

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 : explosion / implosion The pressure in the high side accumulators could exceed the rating and cause the accumulators to explode.	Catastrophic Occasional Possible	High	The truck bed will be sheilding the high pressure system. There is also a safety factor of four on the accumulators and the hoses. The high pressure hydraulic system has a safety release valve which will release pressure at 4500 PSI.	Serious Occasional Unlikely	Moderate	
All Users All Tasks	fluid / pressure : fluid leakage / ejection Fluid can leak from any fittings or if any hoses, fittings, or accumulators rupture hydraulic fluid could vaporize or shoot out of the system.	Serious Occasional Possible	High	A torsion wrench will be used so the fittings are not over-tightened. The truck bed will be sheilding the high pressure system.	Slight Occasional Unlikely	Moderate	
All Users All Tasks	fluid / pressure : liquid / vapor hazards The hydraulic fluid is toxic if it enters the blood stream and the vaporized hydraulic fluid is toxic if breathed.	Serious Occasional Possible	High	The truck bed will be sheilding the high pressure system. There is also a safety factor of four on the accumulators and hoses.	Serious Occasional Unlikely	Moderate	

designsafe Report

Application: Xebra Vehicle

Description:

Product Identifier:

Assessment Type: Detailed

Limits:

Sources:

Analyst Name(s):

Company:

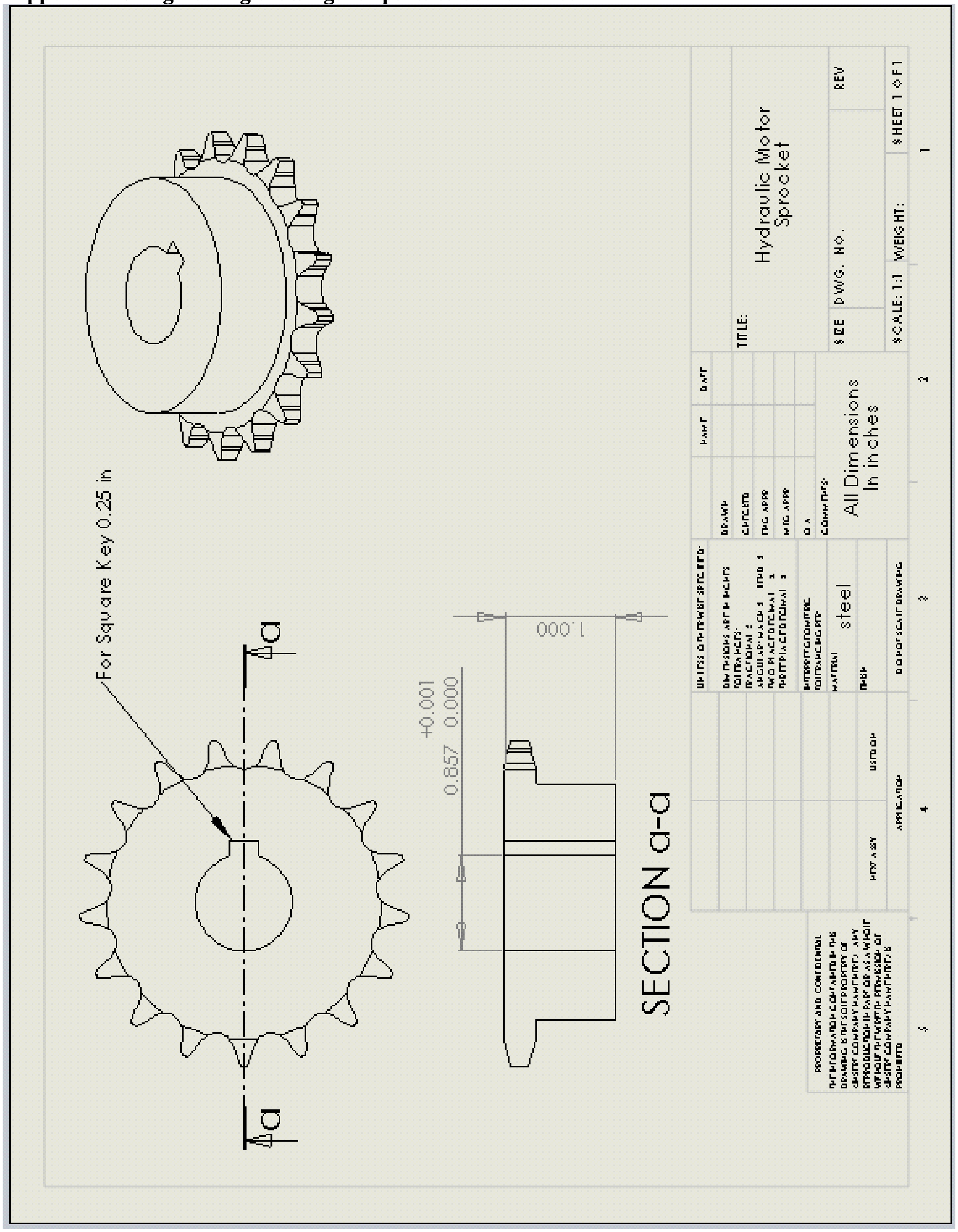
Facility Location:

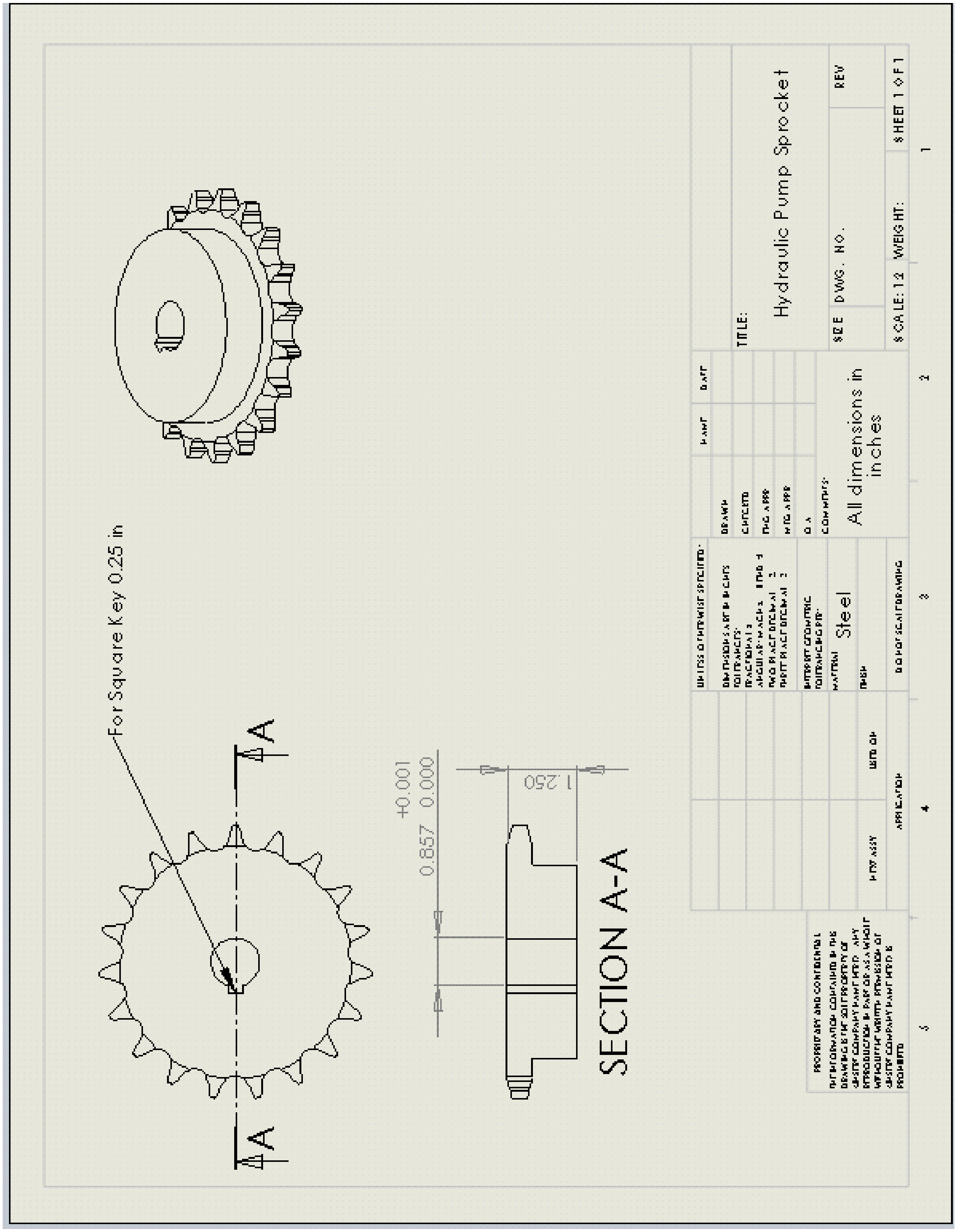
Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Initial Assessment Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Final Assessment Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : machine instability The Xebra can tip onto ists side if a turn is taken too fast or if it is driven improperly.	Serious Occasional Possible	High	The center of mass was examined and analysis on its placement was done. A safety factor of atleast two was used. The user should be cautioned about rollover.	Serious Occasional Unlikely	Moderate	
All Users All Tasks	mechanical : impact Crashing with another vehicle or with a large object could cause serious damage to the user.	Serious Frequent Possible	High	The user will be given instructions on how to operate the vehicle and the braking system properly and will be cautioned to follow all driving laws. If the hydraulic system does not brake quick enough the disk brakes can be engaged.	Serious Occasional Unlikely	Moderate	
All Users All Tasks	electrical / electronic : energized equipment / live parts The electrical system operates at a high voltage and current level, which can cause electricution.	Catastrophic Remote Unlikely	Moderate	A warning that only an experienced engineer or electrician should operate on the electrical system. There is also an electrical shut-off in the cab.	Catastrophic Remote Unlikely	Moderate	
All Users All Tasks	electrical / electronic : shorts / arcing / sparking Spraks or archs can light the hydraulic fluid and cause a fire.	Serious Remote Unlikely	Moderate	All hydraulic fluid is contained in the hydraulic system. The only atmospheric opening is a special cap on the low side accumulator.	Serious Remote Unlikely	Moderate	

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	electrical / electronic : improper wiring The hydraulic system wiring is not all done to satisfactory standard, which could cause parts of the hydraulic system to not work at times.	Slight Remote Unlikely	Low	The main parts of wiring have been tightened and checked.	Minimal Remote Unlikely	Low	
All Users All Tasks	electrical / electronic : water / wet locations The wiring is all open and not contained in a box of any sort.	Slight Occasional Possible	Moderate	The truck bed covers the electrical system and a box will be made to house the major electrical components.	Slight Occasional Unlikely	Moderate	
All Users All Tasks	material handling : motor vehicle movement The Xebra vehicle is extremely heavy and will be traveling at up to 45mph. The causes any normal problems that a moving vehicle would have.	Serious Frequent Possible	High	The user will be given instructions on how to operate the vehicle and the braking system properly and will be cautioned to follow all driving laws. If the hydraulic system does not brake quick enough the disk brakes can be engaged.	Serious Frequent Unlikely	High	

Appendix E: Engineering drawings for parts to be machined

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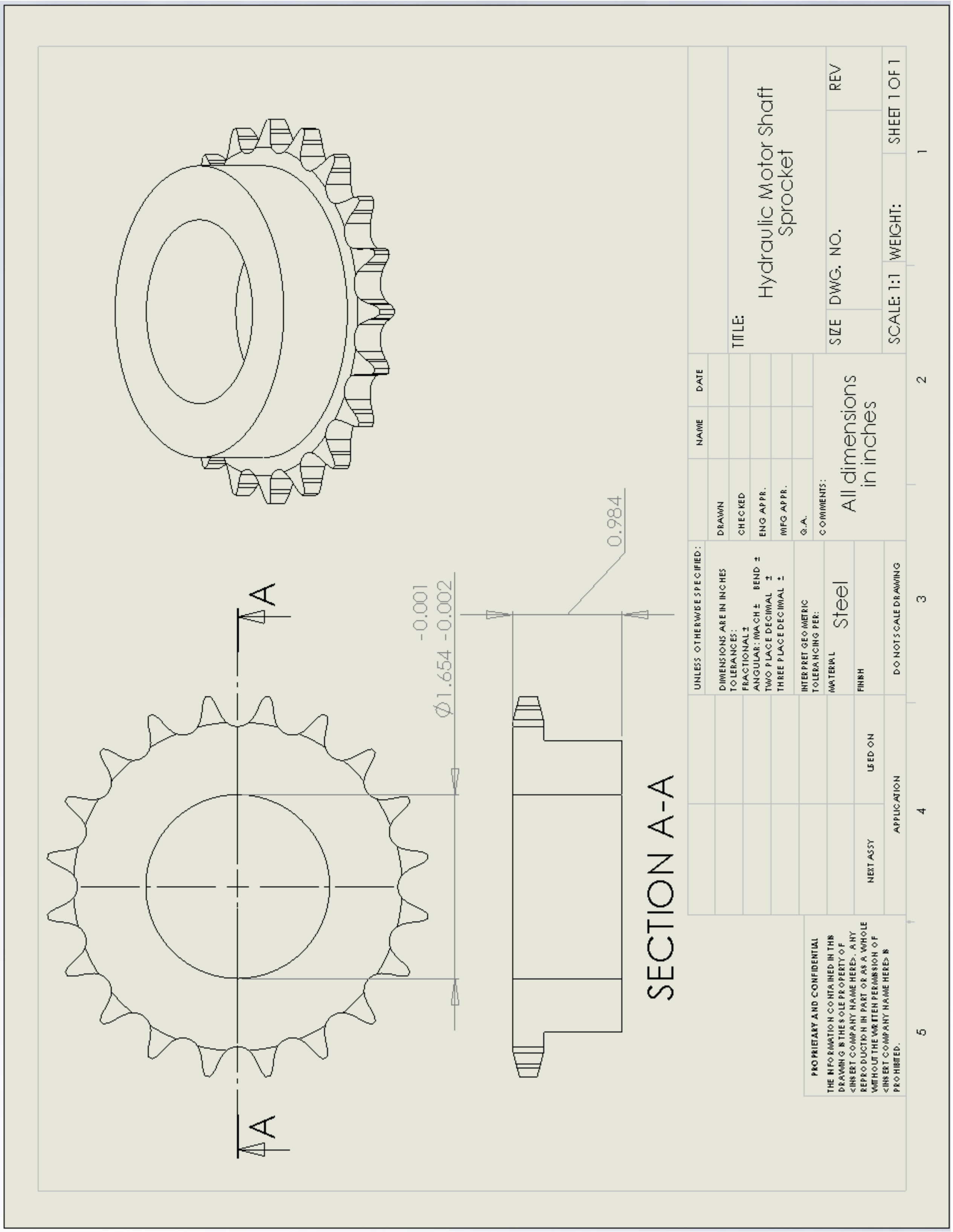
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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
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CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
All dimensions in inches
TITLE:
Hydraulic Motor Shaft Sprocket
SIZE
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
APPLICATION
PROPRIETARY AND CONFIDENTIAL
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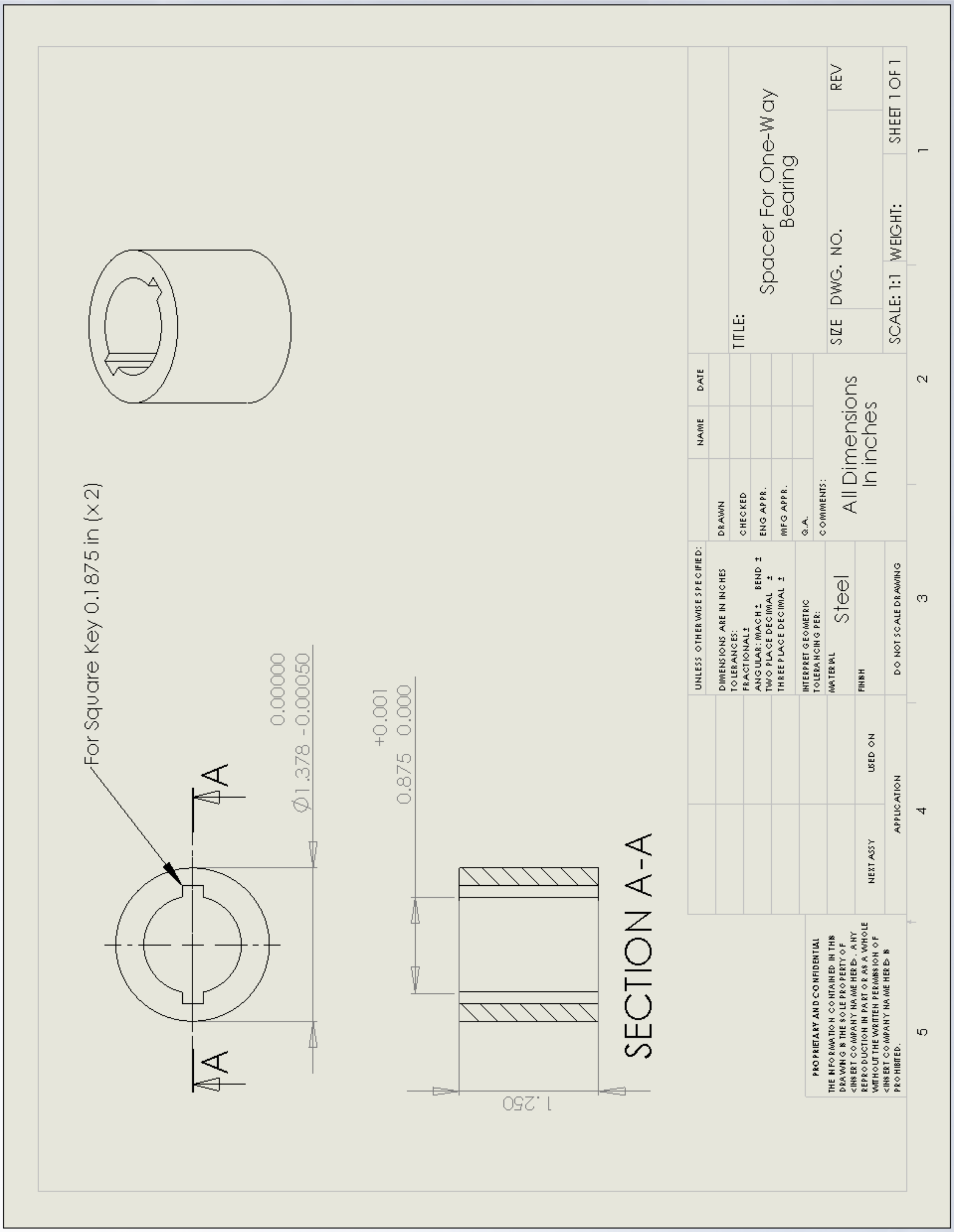
1
2
3
4
5


For Square Key 0.1875 in (x 2)
A
A
Ø1.378 0.00000 -0.00050
0.875 +0.001 0.000
1.250
SECTION A-A
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Steel
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
All Dimensions In inches
TITLE:
Spacer For One-Way Bearing
SIZE
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
NEXT ASSY
USED ON
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5
4
3
2
1

Appendix F: Partial Disassembly of Hydraulics on Xebra Vehicle

Summary:
In order to install the additional high side accumulators, pressure relief valve and the motor and pump sprockets, several hydraulic components must be temporarily disconnected or added. The partial disassembly of the hydraulic system into the accumulators must be done very soon so we can then decide on and order the hydraulic fittings that are needed to connect the additional high side accumulators. This report details the safety concerns, methods to address these safety concerns, and the method to disassemble part of the hydraulic system.

Safety Concerns:
We will be working with a system that can have a hazardous material at high pressures leading to a few major safety issues that need to be addressed.

High Pressure System: The pressure in the system could be released when during disassembly leading to hazardous material being expelled at a high velocity. This could lead to cuts, blinding, and hazardous material being injected into the bloodstream.

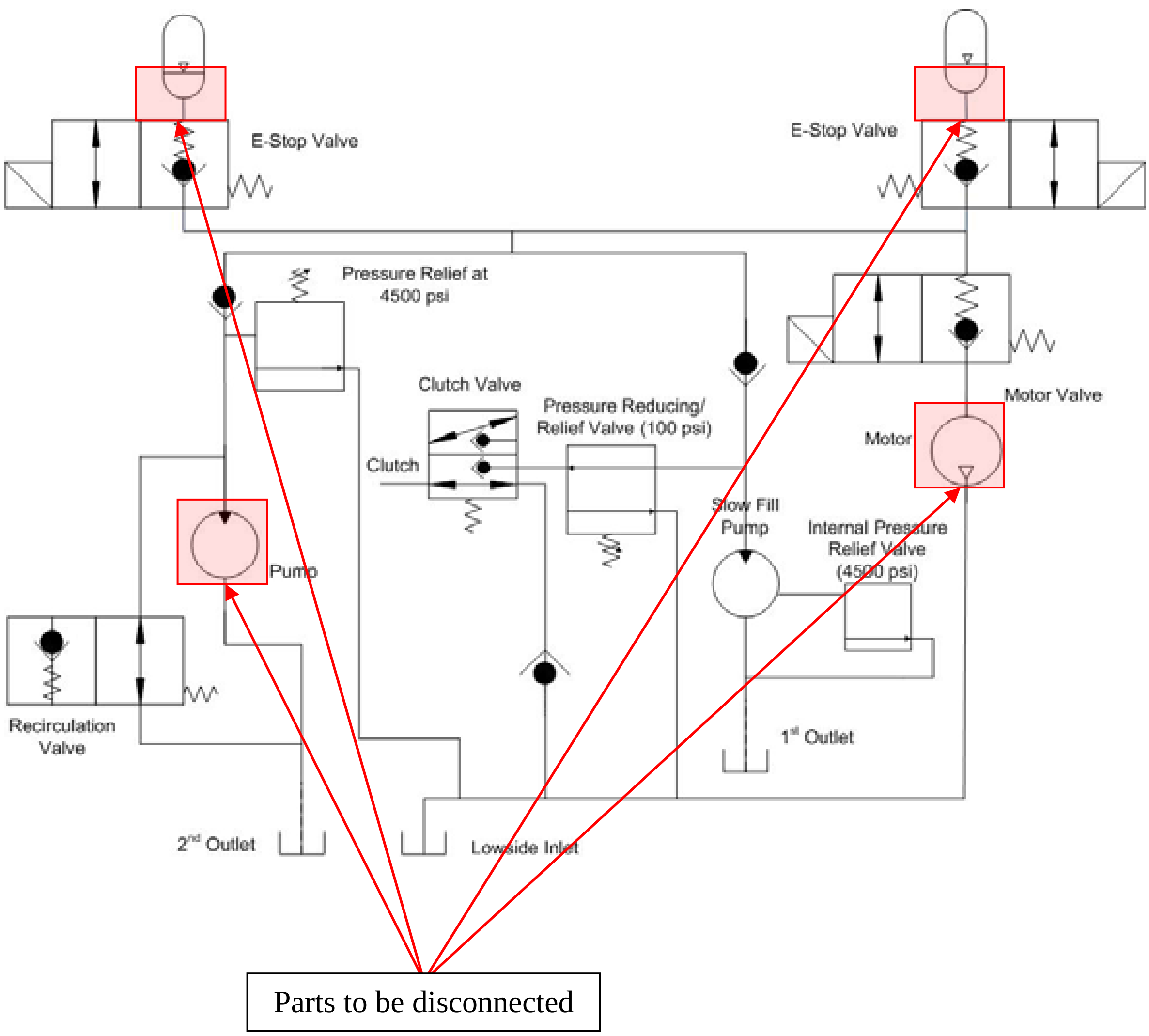
Hazardous Material: The hydraulic fluid is Mobil One synthetic transmission oil and is hazardous when breathed in or introduced to the eyes or the blood stream. When under high pressure it can be expelled, vaporized, and enter into your lungs. Being expelled can also lead to any of the problems outlined in the High Pressure System section. This material is also very slippery and flammable and could lead to slips, falls, burns and other injuries. The flash point is 160 ^{o}C and could be ignited by an electrical arc.

Damaged Parts: Parts that are removed could be damaged. This is a safety concern as the once reassembled the damaged parts could leak.

Methods to address Safety Concerns:
The followings methods will be implemented to address the safety concerns listed above:
We will use gloves and safety goggles while working on the hydraulic system.
The system will be depressurized and all fluid drained into buckets before we work on the disassembling of the hydraulic system.
Pig mats will be placed under the vehicle and used to wipe up any hydraulic fluid that spills
To avoid electrical arc, we will disconnect the charger of the Xebra from the wall socket and ensure that the car is turned off before beginning.
The parts that are removed will be stored in bubble wrapping and stored in the cabinets located in the X50 lab.
We will ask Ben Hagan to be present while the parts are being removed, so as to make sure the parts are removed correctly.
An MSDS will be reviewed thoroughly by all members prior to the disassembly.
Bob Coury will be notified during the time of disassembly.

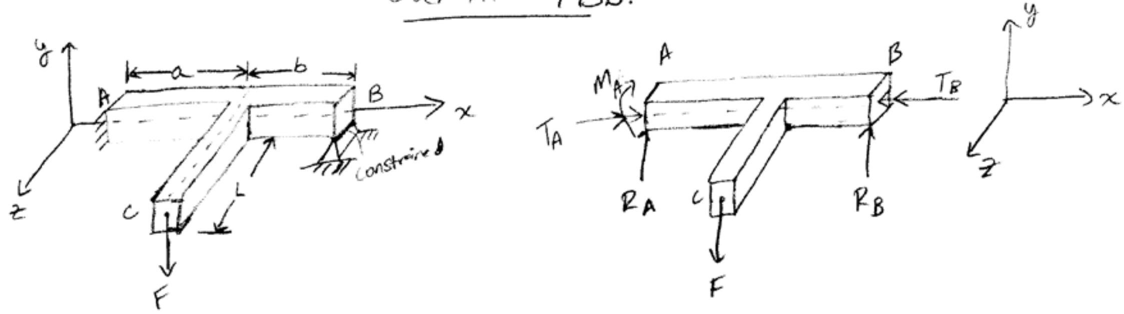
Procedure:
The steps are listed below and the attached picture of the hydraulic system illustrates parts that will be removed.
The electric charger will be disconnected from the car.
Safety glasses and gloves will be utilized for the following steps.

The hydraulic system will be depressurized by running the hydraulic motor while the car is on jacks. There should be almost no pressure so this procedure will produce almost no rotations of the wheels.
The fitting under the low side accumulator will be loosened incrementally to ensure that there is still not pressure in the system and that if there is any remaining pressure, it will be reduced slowly. A bucket will be under the fitting to catch the fluid.
The hydraulic fluid will be drained from the low side accumulator.
The fitting connected to the high side accumulator will be disconnected and any fluid drained into a bucket.
The fittings connected to the motor and pump will be disconnected and any fluid drained into a bucket.
The parts that are removed will be wrapped in bubble paper and stored in the cabinets.
Any fluid will be wiped from the parts and the floor
Reassembly will be done at a later time either by us with the supervision of an expert or by a hose doctor.



Appendix G: Beam analysis for accumulator mounting beams

Overall FBD:

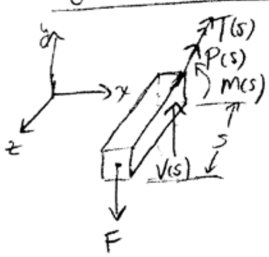


$$\sum F_y = -F + R_A + R_B = 0 \Rightarrow F = R_A + R_B$$

$$\sum M_{z,A} = -M_A - F \cdot a + R_B(a+b) = 0 \Rightarrow M_A = R_B(a+b) - F \cdot a$$

$$\sum T_x = T_A - T_B + F \cdot L \Rightarrow T_B = T_A + F \cdot L \qquad \Rightarrow M_A = (F - R_A)(a+b) - F \cdot a$$

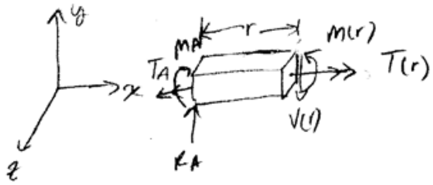
Segmented FBDs: $0 \le s \le L$:



$$P(s) = 0 \,;\; T(s) = 0 \,;\; V(s) = F$$

$$\sum M_x = F \cdot s + m(s) = 0 \Rightarrow m(s) = -F \cdot s$$

$0 \le r \le a$:

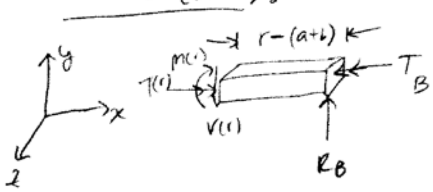


$$V(r) = R_A$$

$$\sum M_z = m(r) - R_A \cdot r - M_A \Rightarrow m(r) = R_A \cdot r + M_A$$

$$\sum T_x = T(r) - T_A = 0 \Rightarrow T(r) = T_A$$

$a < r \le (a+b)$:



$$V(r) = R_B$$

$$\sum M_z = -m(r) + R_B\,[r - (a+b)] = 0$$

$$m(r) = R_B[r - (a+b)]$$

$$\sum T_x = T(r) - T_B = 0 \Rightarrow T(r) = T_B$$

Torsional Deflection at point A, $\phi_A = 0$:

$$\phi_A = \left\{ \int_0^{a+b} \left[\frac{T(r)}{GJ} \cdot \frac{\partial T(r)}{\partial T_A} \right] dr \right\} = \left\{ \int_0^{a} \frac{T_A}{GJ}\, dr + \int_a^{a+b} \frac{(T_A + F L)}{GJ}\, dr \right\} = 0$$

$$\phi_A = \frac{T_A \cdot a}{GJ} + \frac{T_A + F \cdot L}{GJ}(a+b) - \frac{T_A + F \cdot L}{GJ} \cdot a = 0$$

$$\Rightarrow T_A = \frac{-FLb}{a+b} \quad ; \quad T_B = \frac{FLa}{a+b}$$

Deflection at Point A, $\gamma_A = 0$:

$$\gamma_A = \left\{ \int_0^{a+b} \left[\frac{m(r)}{EI} \frac{\partial m(r)}{\partial R_A} + \frac{T(r)}{GJ} \underbrace{\frac{\partial T(r)}{\partial R_A}}_{0} \right] dr \right\} = 0$$

$$\gamma_A = \left\{ \int_0^{a} \frac{\left[R_A \cdot r + (F - R_A)(a+b) - F a \right]}{EI} \cdot (r-a-b)\, dr + \int_a^{a+b} \frac{(F - R_A)(r-a-b)}{EI} (a+b-r)\, dr \right\}$$

$$R_A = \frac{1}{2} \cdot \frac{F \cdot b(3a^2 + 6ab + 2b^2)}{a^3 + 3a^2 b + 3ab^2 + b^3} \quad ; \quad R_B = \frac{1}{2} \frac{F a^2 (2a + 3b)}{a^3 + 3a^2 b + 3ab^2 + b^3}$$

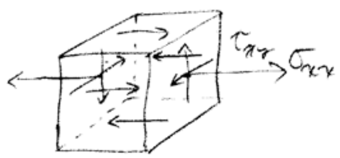
$$M_A = \frac{1}{2} \frac{(2a+3b)\, F a^2}{a^2 + 2ab + b^2}$$

From Vehicle Measurements:

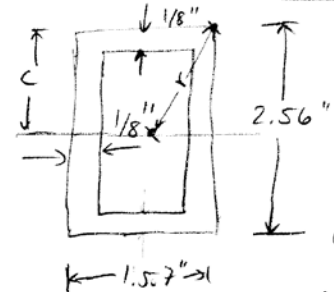
$a = 2.4$ in ; $b = 12.4$ in ; $L = 6.25$ in.

Failure at point A:

$R_A = 0.963 \cdot F$; $R_B = 0.037 \cdot F$; $M_A = 0.552 \cdot F$; $T_A = \frac{F \cdot L b}{a+b}$



Cross Section Area.



$I = 0.456 \text{ in}^4$

$J = 4.146 \text{ in}^4$

$c = 1.28 \text{ in.}$

$r = 1.50 \text{ in.}$

$\sigma_y = 51,000 \text{ lb/in}^2$

(AISI 1020 Steel)

$\sigma_{xx} = \frac{M_A \cdot c}{I}$; $\tau_{xy} = \frac{T_A \cdot r}{J}$

Principle stresses:

$\sigma_1 = 3.596 \cdot F$; $\sigma_2 = -0.498 \cdot F$

Von mises:

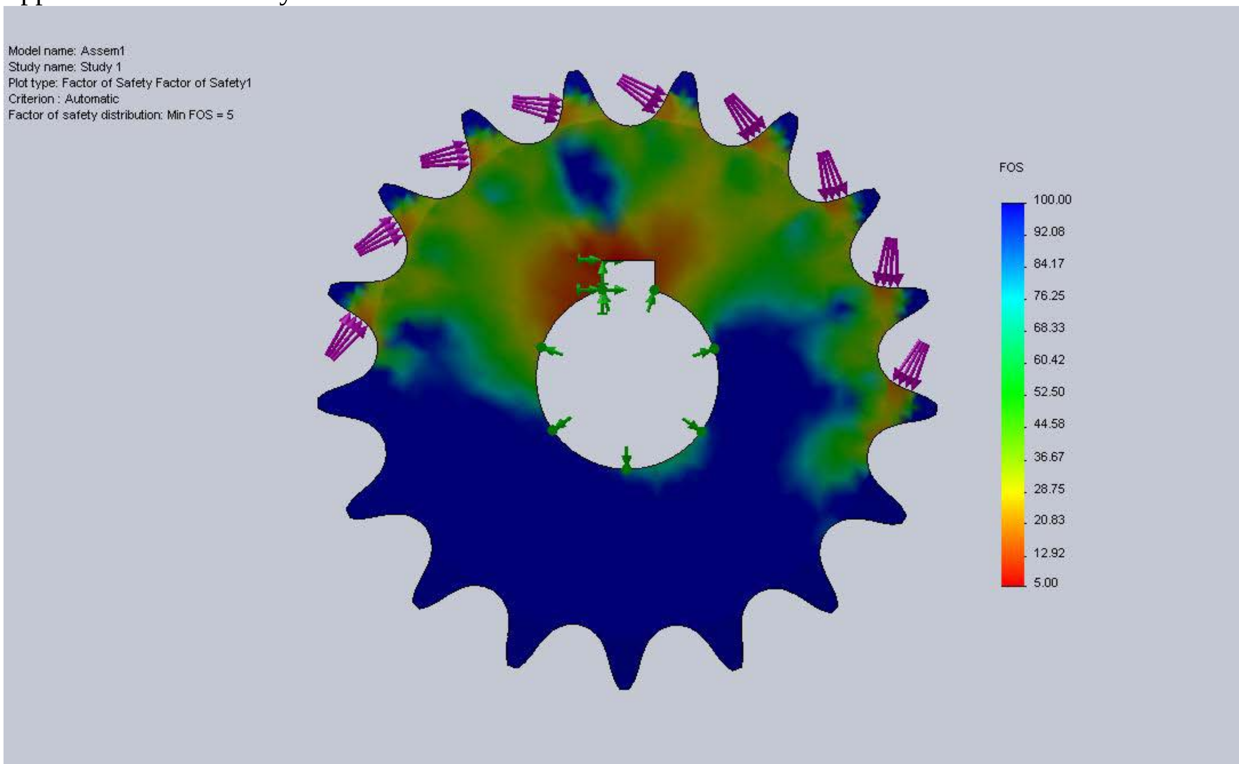
$\overline{\sigma_H} = 4.643 \cdot F \Rightarrow$

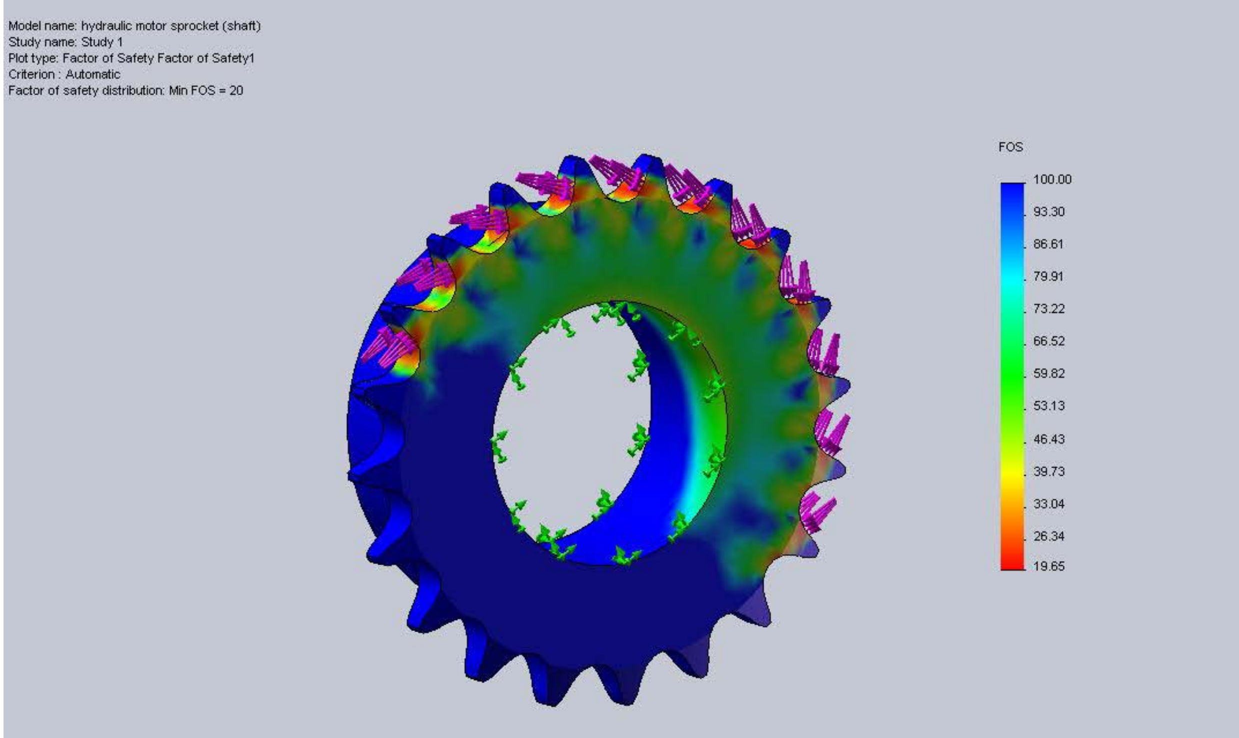
Yeild when $\overline{\sigma_H} \geq \sigma_y$

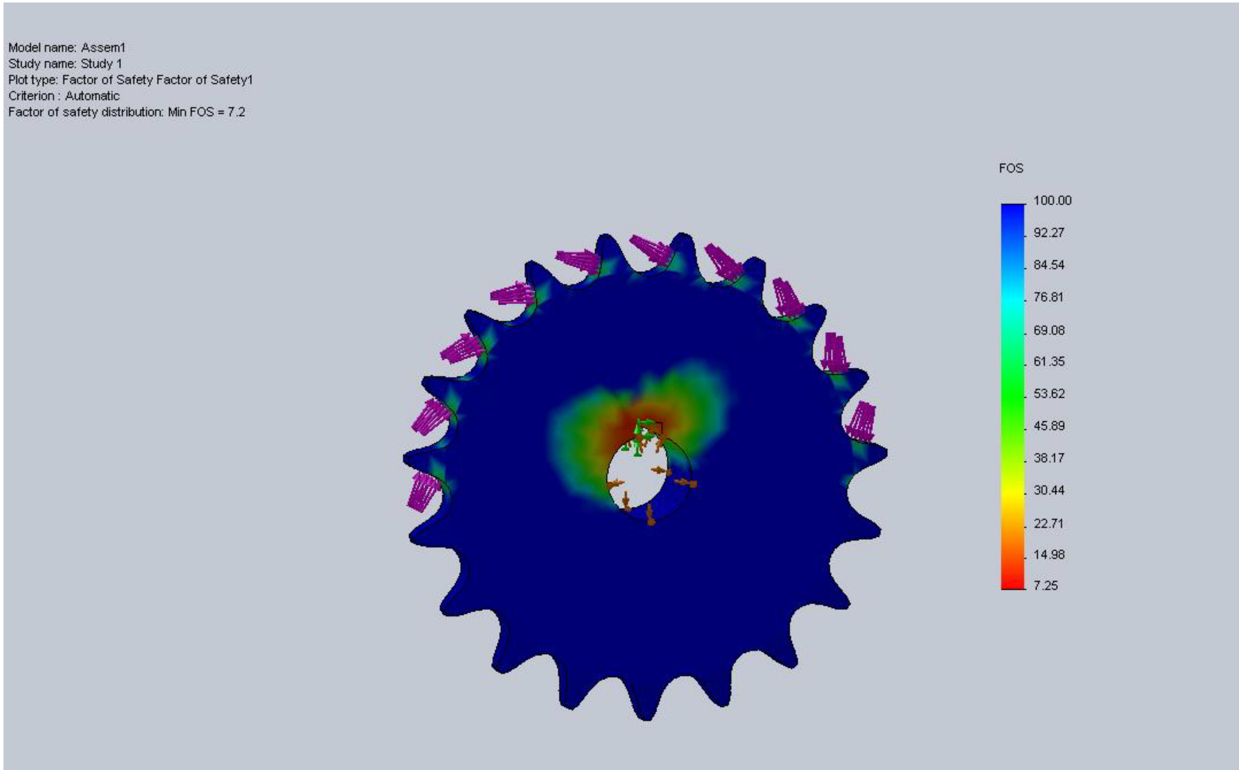
$F < \frac{\sigma_y}{4.643} = \boxed{11,000 \text{ lbs}}$

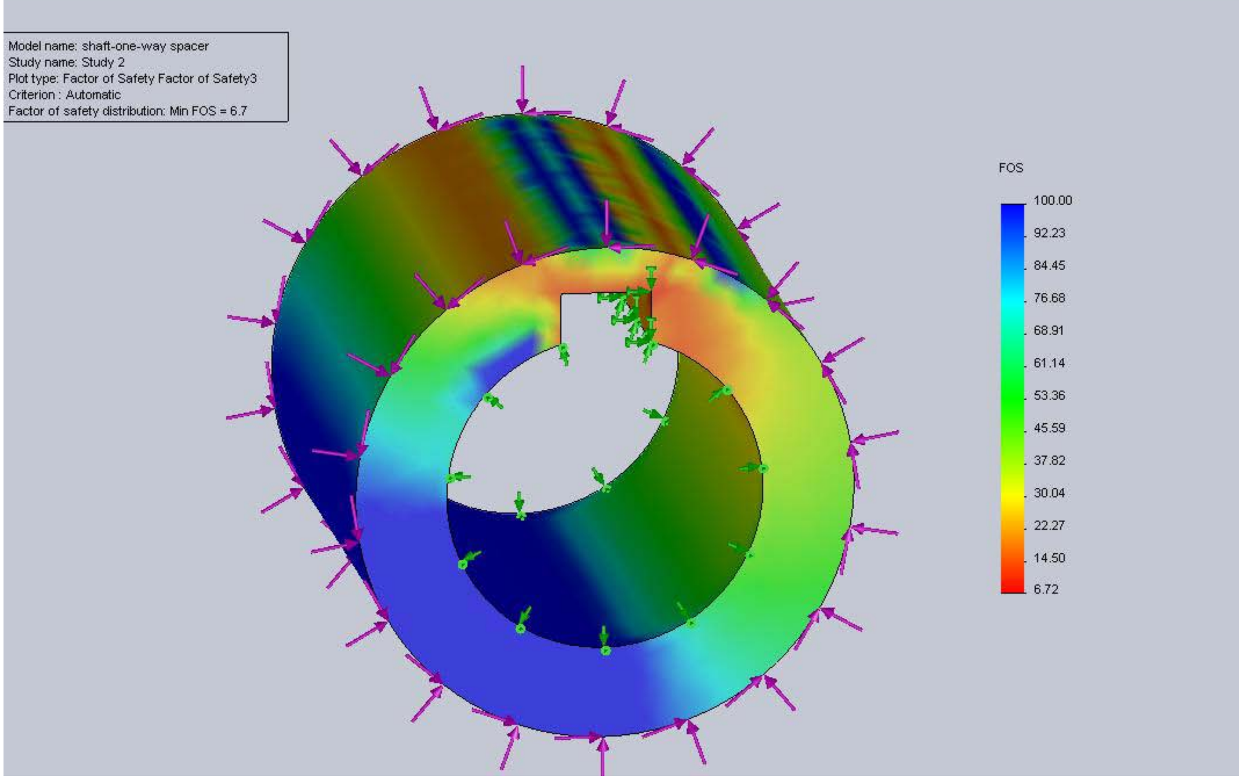
The beam can support a force up to 11,000 lbs

Appendix H: FEA Analysis






Model name: Assem1
Study name: Study 1
Plot type: Factor of Safety Factor of Safety1
Criterion : Automatic
Factor of safety distribution: Min FOS = 7.2
FOS
100.00
92.27
84.54
76.81
69.08
61.35
53.62
45.89
38.17
30.44
22.71
14.98
7.25


Model name: shaft-one-way spacer
Study name: Study 2
Plot type: Factor of Safety Factor of Safety3
Criterion : Automatic
Factor of safety distribution: Min FOS = 6.7
FOS
100.00
92.23
84.45
76.68
68.91
61.14
53.36
45.59
37.82
30.04
22.27
14.50
6.72

Appendix I: Mobil One Synthetic Transmission Oil MSDS

MOBIL OIL CORP -- AUTOMATIC TRANSMISSION FLUID 210 -- 9150-00B130099
==================== Product Identification ====================

Product ID:AUTOMATIC TRANSMISSION FLUID 210
MSDS Date:12/30/1992
FSC:9150
NIIN:00B130099
MSDS Number: BQDFF
=== Responsible Party ===
Company Name:MOBIL OIL CORP
Address:3225 GALLOWS ROAD
City:FAIRFAX
State:VA
ZIP:22037-0001
Country:US
Info Phone Num:800-424-9300
Emergency Phone Num:609-737-4411
CAGE:3U728
=== Contractor Identification ===
Company Name:MOBIL OIL CORP, NORTH AMERICAS MARKETING AND REFINING
Address:3225 GALLOWS ROAD
Box:City:FAIRFAX
State:VA
ZIP:22037
Country:US
Phone:800-662-4525/ 856-224-4644
CAGE:3U728

============= Composition/Information on Ingredients =============

Ingred Name:ZINC (SARA III)
CAS:7440-66-6
RTECS #:ZG8600000
Fraction by Wt: .04%
Other REC Limits:NONE SPECIFIED
OSHA PEL:10 MG ZNO/M3
ACGIH TLV:10 MG ZNO/M3; 9192
EPA Rpt Qty:1000 LBS
DOT Rpt Qty:1000 LBS

Ingred Name:ZINC DIALKYL DITHIOPHOSPHATE
CAS:68457-79-4
Fraction by Wt: .61%
Other REC Limits:NONE SPECIFIED

Ingred Name:MINERAL OIL, PETROLEUM DISTILLATES, SOLVENT-DEWAXED HEAVY
PARAFFINIC
CAS:64742-65-0
RTECS #:PY8038500
Fraction by Wt: 95.0%
Other REC Limits:NONE SPECIFIED
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:REFINED HEAVY PARAFFINIC DISTILLATES
CAS:64741-88-4
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:REFINED HEAVY PARAFFINIC DISTILLATES
CAS:64741-88-4
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

Ingred Name:DISTILLATES, SOLVENT-REFINED LIGHT NAPHTHENIC
CAS:64741-97-5
RTECS #:PY8041000
Fraction by Wt: >95.0%
Other REC Limits:NONE SPECIFIED
OSHA PEL:300 PPM
ACGIH TLV:300 PPM

Ingred Name:2-PROPENOIC ACID, 2-METHYL-, BUTYL ESTER, POLYMER WITH N-
(3-DIMETHYLAMINO PROPYL)-2-METHYL-2-PROPENAMIDE
CAS:50867-55-5
Fraction by Wt: <2.0%
Other REC Limits:NONE SPECIFIED

Ingred Name:MINERAL OIL, PETROLEUM DISTILLATES, HYDROTREATED LIGHT
NAPHTHENIC
CAS:64742-53-6
RTECS #:PY8036000
Fraction by Wt: 0.39%
Other REC Limits:NONE SPECIFIED
OSHA PEL:5 MG/M3 (OIL MIST)
ACGIH TLV:5 MG/M3 (OIL MIST)

==================== Hazards Identification ====================

LD50 LC50 Mixture:TLV: 5.00 MG/M3
Routes of Entry: Inhalation:NO Skin:NO Ingestion:NO
Reports of Carcinogenicity:NTP:NO IARC:NO OSHA:NO
Health Hazards Acute and Chronic:IRRITATING TO EYES & SKIN.
Explanation of Carcinogenicity:THIS COMPOUND CONTAINS NO INGREDIENTS AT CONCENTRATIONS OF 0.1% OR GREATER THAT ARE CARCINOGENS OR SUSPECT CARCINOGENS.
Effects of Overexposure:NONE.
Medical Cond Aggravated by Exposure:PERSONS WITH A HISTORY OF AILMENTS OR WITH A PRE-EXISTING DISEASE INVOLVING THE SKIN MAY BE AT INCREASED RISK FROM EXPOSURE.

======================= First Aid Measures =======================

First Aid:EYES: FLUSH WITH RUNNING WATER FOR 15 MINUTES WHILE HOLDING EYELIDS OPEN. SKIN: WASH WITH SOAP AND WATER. REMOVE CONTAMINATED CLOTHING. INGESTION: DO NOT INDUCE VOMITING. RINSE MOUTH & DRINK LARGE AMO UNTS OF WATER. GET MEDICAL ATTENTION.

===================== Fire Fighting Measures =====================

Flash Point Method:COC
Flash Point:350F,177C
Lower Limits:.6%
Upper Limits:7.0%
Extinguishing Media:USE WATER FOG, CARBON DIOXIDE, FOAM, OR DRY CHEMICAL.
Fire Fighting Procedures:WEAR FIRE FIGHTING PROTECTIVE EQUIPMENT AND A FULL FACED SELF CONTAINED BREATHING APPARATUS. COOL FIRE EXPOSED CONTAINERS WITH WATER SPRAY.
Unusual Fire/Explosion Hazard:NONE

================== Accidental Release Measures ==================

Spill Release Procedures:ABSORB ON FIRE RETARDANT TREATED SAWDUST, DIATOMACEOUS EARTH, ETC. SHOVEL UP AND DISPOSE OF AT AN APPROPRIATE WASTE DISPOSAL FACILITY.

====================== Handling and Storage ======================

Handling and Storage Precautions:STORE IN A COOL, DRY PLACE. KEEP CONTAINERS CLOSED WHEN MATERIAL IS NOT IN USE. KEEP AWAY FROM IGNITION SOURCES. AVOID PROLONGED OR REPEATED CONTACT.
Other Precautions:NONE

============= Exposure Controls/Personal Protection =============

Respiratory Protection:NONE
Ventilation:LOCAL AND MECHANICAL(GENERAL) EXHAUST TO PROVIDE ADEQUATE
VENTILATION.
Protective Gloves:NONE
Eye Protection:SAFETY GLASSES - CHEMICAL SPLASH GOGGLES
Other Protective Equipment:NONE
Work Hygienic Practices:WASH THOROUGHLY AFTER HANDLING AND BEFORE
EATING, DRINKING OR SMOKING. LAUNDER CONTAMINATED CLOTHING BEFORE
REUSE.
Supplemental Safety and Health
AVOID PROLONGED OR REPEATED EXPOSURE. DO NOT GET ON SKIN OR IN EYES. DO
NOT INGEST. READ PRECAUTIONS ON LABEL BEFORE USE.

================== Physical/Chemical Properties ==================

HCC:V6
Boiling Pt:B.P. Text:>600F,>316C
Melt/Freeze Pt:M.P/F.P Text:NA
Vapor Pres:<.1
Vapor Density:0.868
Solubility in Water:NEGLIGIBLE
Appearance and Odor:RED LIQUID, MILD ODOR
Corrosion Rate:N/KNOWN

================= Stability and Reactivity Data =================

Stability Indicator/Materials to Avoid:YES
STRONG OXIDIZING AGENTS
Stability Condition to Avoid:HIGH HEAT, OPEN FLAMES AND OTHER SOURCES
OF IGNITION
Hazardous Decomposition Products:CARBON MONOXIDE AND CARBON DIOXIDE

==================== Disposal Considerations ====================

Waste Disposal Methods:DISPOSAL OF WASTE AND HAZARDOUS MATERIAL SHALL
COMPLY WITH LOCAL, STATE AND FEDERAL ENVIRONMENTAL PROTECTION
AGENCY REGULATIONS. SEND WASTE MATERIAL TO AN APPROVED RECYCLING
FACILITY IF FEASIBLE. CITY , STATE AND FEDERAL REGULATIONS MUST
BEFOLLOWED.

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Appendix AD: Biographies

Michael Makowski

I grew up in Grosse Pointe Woods, MI where my Dad is Director of Public Safety and my Mom works at a Periadontist office. I have one sister who graduated from MSU in Criminal Justice. Many of my family members were U of M alumni that were mechanical engineers so I started to think of studying that when I was very young. I really considered mechanical engineering when I realized that I not only love math and science but I am very good at it. I really want to make a difference in this world as well as have a career that I am interested in. I feel that the future of this world rests on the shoulders of engineers more than most people. Not only is mechanical engineering rewarding but challenging as well and I realized I could not be happy without being able to prove myself.

I have many future hopes and dreams but my plans are not completely set. I am currently a senior, graduating in December ’09. I have many interests and have not decided which career would satisfy those most. I have recently considered graduate studies more than before but would still like to get a job once I graduate then decide if I want to go back to school and what to study. Some interests that I have, other than engineering, are playing racquetball, traveling, backpacking, cycling, rock climbing and playing cards. The picture is from the top of Quandry Peak, which is 14,265 ft, in Colorado where I traveled to this summer after coming back from studying in Berlin.

Alex Duggan

I’m a senior mechanical engineer at the University of Michigan and will graduate after the winter 2009 semester. After which I am still undecided about going into the workforce or going to graduate school. I have a strong interest in solid mechanics as well as dynamics.

Grand Rapids Michigan is my hometown, which is about two hours west of Ann Arbor by car. When I’m not doing homework or working on projects, I enjoy biking around town and playing baseball or softball.

Andrew LaNoue

I am a 5th year senior at the University of Michigan, and will graduate in December 2009 with a Bachelors degree in mechanical engineering. I am a member of the formula SAE race team, and have a fairly strong background in machining and fabrication. I spent the summer of 2009 in Berlin taking engineering classes in sustainability and also working as a research assistant at the Technische Universität Berlin. My main interests in engineering are in mechanical design and systems engineering. I chose this project because it seemed like a great chance to improve my design skills, and also, I love working on any project that deals with making things faster and more efficient.

Aazam Vishram

I am a Junior at the University of Michigan, I will be graduating in May 2010 with a Bachelors in Mechanical Engineering. Last summer I went back home, Mumbai, India, I worked in a company that fabricates parts of cranes, I have also worked in a Naval Shipyard, where I helped in the design and fabrication of air intake systems and cooling systems, for cruisers and submarines. I am very interested in thermodynamics and heat transfer. I enjoy designing and fabrication as well. I plan on continuing my education and I will be applying to graduate school, to obtain a masters degree in Financial Engineering.