Final Report

Project 1: Hydraulic Electric Hybrid Vehicle

<u>Xebra X-Gen</u>

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TABLE OF CONTENTS

ABSTRACT…………………………………………………………………………………………I
EXECUTIVE SUMMARY……………………………………………………………………….. II

1. INTRODUCTION……………………………………………………………………….. 1
   1.1 Background…………………..………………..……………………………..…… 1
   1.2 Motivation……………………..………………………………………………….. 1
   1.3 Fall 2008 Goals……………………..…………………………………..………... 1

2. SPECIFICATIONS………………………………………………………………………. 1
   2.1 Customer Requirements…………………………………………………………... 1
   2.2 Engineering Specifications……………………………………………………….. 2
   2.3 Quality Function Deployment…………………………………………………… 3

3. INFORMATION SEARCH……………………………………………………………… 3
   3.1 Technical Benchmarks………………………………………………………………3
   3.2 Patent Search……………………………………………………………………... 3

4. CONCEPT GENERATION AND SELECTION………………………………………… 4
   4.1 Function Analysis………………………………………………………………… 4
   4.2 Design Concept Development…………………………………………………….. 5

5. ENGINEERING DESIGN PARAMETER ANALYSIS………………………………… 7
   5.1 Mechanical……………………………………………………………………….. 7
   5.2 Hydraulic………………………………………………………………………….. 9
   5.3 Design for Manufacturability…………………………………………………… 13
   5.4 Failure/Safety Analysis………………………………………………………….. 14
   5.5 Design for the Environment……………………………………………………... 14

6. FINAL DESIGN DESCRIPTION……………………………………………………. 16
   6.1 Mechanical……………………………………………………………………….. 16
   6.2 Hydraulic………………………………………………………………………... 17
   6.3 Electrical…………………………………………………………………………. 19

7. MANUFACTURING PROCESS……………………………………………………….. 20

8. COMPLETE PROTOTYPE…………………………………………………………….. 22

9. TESTING……………………………………………………………………………….. 25
   9.1 Dynamometer Testing………………………………………………………….... 25
   9.2 Spin Down Test…………………………………………………………………. 25

10. DISCUSSION…………………………………………………………………………. 27
   10.1 Design Strengths……………………………………………………………….. 27
   10.2 Design Weaknesses…………………………………………………………….... 27
   10.3 Potential Changes……………………………………………………………….. 27

11. RECOMMENDATIONS…….. 28
11.1 Necessary Next Steps…… 28
11.2 Future Suggestions…… 29

12. CONCLUSIONS…... 30

13. ACKNOWLEDGEMENTS….. 31

14. REFERENCES….. 32

APPENDIX A: QFD…… 33
APPENDIX B: ENGINEERING CHANGE NOTICE (ECN)….... 34
APPENDIX C: ENGINEERING DRAWINGS….. 35
APPENDIX D: FAILURE MODE AND EFFECT ANALYSIS (FMEA)…… 36
APPENDIX E: BILL OF MAERIALS…... 37
APPENDIX F: EPA CITY CYCLE…… 38
APPENDIX G: CLUTCH INSTALLATION SPACING…... 39
APPENDIX H: ANTI-ROTATION BRACKET….… 40
APPENDIX I: PRELIMINARY CALCULATIONS…..... 41
APPENDIX J: GANTT CHART….... 42
APPENDIX K: LOGAN HYDRAULIC CLUTCH…… 43
APPENDIX L: TEAM MEMBER BIOGRAPHIES….... 45

ABSTRACT

The Environmental Protection Agency is interested in improving the efficiency of electric vehicles by integrating hydraulic launch and braking systems with the electric system. Electric vehicles have an efficiency drop from nearly 90% to less than 60% when comparing optimal cruising-speed energy usage to acceleration energy usage. Adding a hydraulic launch system reduces the batteries' burden, increasing the vehicle's efficiency and range. In addition, a hydraulic regenerative braking system provides the same benefit during deceleration, and can capture the normally wasted braking energy and store it for use in later accelerations. This project has spanned several semesters and will continue in order to convert the electric Xebra vehicle into the world's first hydraulic-electric hybrid! Past teams worked on the initial design, component layout, and hydraulic launch system while this team focused on designing and installing the hydraulic regenerative braking system onto the Xebra.

EXECUTIVE SUMMARY

Motivation. This project was sponsored by the Environmental Protection Agency with the goal of implementing a regenerative braking system onto the Xebra hydraulic-electric hybrid vehicle. A regenerative braking system will capture energy wasted during braking in the form of hydraulic fluid pressure. This stored fluid can then be used to accelerate the vehicle, minimizing the energy draw on the batteries from the electrical motor. The two drive systems are complementary; hydraulic drive is efficient at high power outputs while batteries are very inefficient when drawing a high current. On the other hand, at constant speeds, electric motors are very efficient while drawing a small current from the battery. Therefore, a vehicle that uses hydraulics to accelerate (high energy output) and an electric motor while cruising (low current draw) is most efficient. To assist, a hydraulic regenerative braking system can capture kinetic energy during braking and store it in the form of pressurized fluid that can then be used to accelerate the vehicle.

Requirements. The main requirement of our sponsor was to install the hydraulic regenerative braking system so that braking energy would be recovered while the vehicle reasonably stopped from 35 mph. Other requirements from our QFD included safety, ease of use, and transferability to future semesters. These and the other requirements were parlayed into the engineering specifications of the system.

Concept Generation. Based off our customer requirements and engineering specifications, we generated numerous concepts. Each of these concepts was evaluated and compared to determine the best ones for the Xebra. Ultimately, we used a Morphological chart to combine the concepts into our Alpha design.

Final Design. We completed several calculations to select components to design the final mechanical, hydraulic, and electrical systems. Our final design includes a hydraulic clutch on the rear drive shaft that is attached to a sprocket. When disengaged, the sprocket can free-wheel on several bearings, but during braking, the clutch engages so that the drive shaft and sprocket spin together. This sprocket is connected to a sprocket on the pump shaft with a chain so that the pump spins and pressurizes the hydraulic fluid. The hydraulic system fully controls the motor and pumps, actuates the clutch, and includes numerous valves that ensure system safety. Finally, the electronics control the hydraulic system through the pedals.

Prototype. Following our final design, we created a prototype by manufacturing, assembling, and installing the various mechanical and hydraulic components. Manufacturing processes included milling, cutting, drilling, welding, and lathing. Due to unforeseen and uncontrollable circumstances, we were unable to achieve a fully functioning prototype. Therefore, we completed limited testing so future dynamometer and spin down tests are required to completely quantify the improvements of the prototype.

Discussion. The Design Expo gave us a chance to reflect on our design's strengths and weaknesses. The strengths include the complete hydraulic system and the efficient and compact mechanical system, both of which fit nicely in the rear of the vehicle. On the other hand, weaknesses include excessive chain slack and messy wiring. Potential changes include using only one e-stop valve, installing an electric feeder pump to actuate the clutch, finding a poppet type pressure reducing/relieving valve, utilizing a variable displacement motor and pump, and using only one component as both a motor and pump.

Recommendations. There are several necessary steps that must be completed to achieve a fully functioning prototype. These include minor mechanical adjustments, more advanced electronics work, and the addition of the oil. After that, the vehicle will be ready for coast-down and LA4 cycle dynamometer tests. Furthermore, the current Xebra vehicle has the potential to offer many other challenging ME 450 projects depending on the needs and wants of the EPA. These include but are not limited to vehicle optimization, design for manufacturability, and motor and pump efficiency testing.

1. INTRODUCTION

1.1 Background

The Xebra project is sponsored by the Environmental Protection Agency (EPA), a United States entity whose mission since its inception in the 1970's has been "to protect human health and the environment [1]." The EPA is interested in technologies that reduce motor vehicle emissions and provide environmentally friendly solutions. The ZAP Xebra electric truck is a zero emission vehicle, and an alternative solution to the traditional combustion engine. Despite its advantages, an electric vehicle faces some limitations during operation. The batteries become very inefficient during high-load operation, such as accelerations, significantly reducing the range of the vehicle and the overall life of the batteries. To address this problem, the EPA seeks to convert the Xebra from a fully electric system to a hydraulic-electric hybrid.

1.2 Motivation

Currently the Xebra hybrid-electric vehicle uses stored energy from batteries to accelerate. These batteries get worn down quickly during the acceleration process due to the high-loading. Therefore, we would like to use other sources of energy besides the batteries during the acceleration process. Implementing a regenerative braking system using hydraulics would help. With a regenerative braking system, we could use a hydraulic pump system to store the braking energy into accumulators. The energy stored in the accumulators will be used to accelerate the vehicle, greatly reducing the power draw on the batteries. Therefore, a Xebra car with regenerative braking will have a higher efficiency and life cycle.

1.3 Fall 2008 Goals

Our objective was to design and implement a regenerative braking system. Our goal was to recapture some of the braking energy of the vehicle and use it to charge the high-pressure accumulators, thereby reducing the amount of power being drawn from the batteries. This will lead to increases in the performance of the batteries and the range of the vehicle. The plan is to charge the high pressure accumulators with the energy recovered during braking. Initial calculations suggest that we can recapture approximately 65% of the energy in one braking event, which will in turn extend the range of the vehicle.

2. SPECIFICATIONS

2.1 Customer Requirements

The customer requirements were determined by analyzing previous group's work on the Xebra project and by collaborating with the EPA. Through our collaboration with the EPA, we determined that our new customer requirements were to install the regenerative braking system, improve the plumbing layout, and quantify vehicle improvements through testing. See Table 1 on page 2 for a list of customer requirements.

Table 1: Customer Requirements with corresponding descriptions

Customer Requirement	Description
Transferrable to future semesters	Project designed and built with future goals in mind for ease of continuation
Comfortable feel during braking	Reasonable deceleration rate; no jolting
Sufficient braking until stop	Deceleration that stops fast enough to be safe but also remains smooth
Easy to service	Plumbing is designed to allow for easy access for maintenance
Maintains vehicle function	All parts of the vehicle work after completion of project so that the vehicle functions as well or better than the beginning of project
Easy to use	Foot pedal engages braking system so user does minimal work to engage brakes
Aesthetics	Professional and clean look for presentation
Safety	Vehicle stops within a safe distance and time from a speed of 35mph
Reliability	Vehicle lasts through many uses
Improve plumbing layout	Keep hose lengths to a minimum to reduce losses in the system
Clutch improvements	Reduce the noise and improve transmission

2.2 Engineering Specifications

The engineering specifications were determined by analyzing previous groups' work on the Xebra project as well as collaborating with the EPA to determine applicable specifications for this semester's project. There are many new engineering specifications for this project due to the scope of the work that is needed. The new engineering specifications include pump size, valve sizes, and location of the pump and valves. See Tale 2 on for a complete list of engineering specifications.

Table 2: The Engineering Specifications and the Target Value for each

Engineering Specifications	Target Value	Units
Accumulator Pressure	4000	Psi
Pump Size	33	Cc
Valve Sizes	40	gal/m
Braking Type (Variable or Fixed)	6	Sec
Corrosion Resistant	100	%
Losses due to heat/friction of brakes	0	W
Location of pump	n/a	n/a
Location of valves	n/a	n/a
Maximum flow-rate of fluid into high accumulator	1.5	L/s

We have created a quality function deployment (QFD) based on the customer requirements and have translated them into appropriate engineering specifications that relate. According to our QFD analysis, the most important customer requirements are that it is transferrable to future semesters while maintaining the vehicle's function, safety, and reliability. In order for our project to be transferrable to future semesters, we considered the engineering specifications of the braking type that we implemented (variable or fixed), the clutch improvements, and the locations that we plumb in the pump and valves. When our hydraulic regenerative braking system is implemented, it is very important that the vehicle functions are maintained. Therefore, we considered the engineering specifications of the flow rate of the pump, accumulator pressure, pump size, valve sizes, losses in braking due to heat/friction, and the flow rate of hydraulic fluid. Safety is the final most important customer requirement and this translates into the engineering specifications of the flow-rate of the pump, accumulator pressure, pump size, valve sizes, what type of braking is used, and the flow-rate of the hydraulic fluid into the high accumulator. Without careful consideration of all of these engineering specifications, safety would have been compromised.

2.3 Quality Function Deployment

The QFD was developed by taking last semester's QFD and applying our customer requirements and appropriate engineering specifications to it. Many of the items are the same from previous semesters; however, we have applied some new customer requirements and engineering specifications to our QFD (see Appendix A).

The QFD has a list of customer requirements on the left with a ranking of their importance under the "weightage" column. This ranking was gathered through collaboration with the project group and our sponsor. The engineering specifications are aligned across the top of the page. Under the engineering specifications is a corresponding ranking of how each engineering specification relates to each customer requirement. The ranking can be a 1, no relation, a 3, weak relation, or a 9, strong relation. These relations were determined by collaboration with the project team and the sponsor. Finally, the total weighted customer requirement number is calculated by adding all of the engineering specification ratings for one customer requirement and then multiplying by that customer requirement weightage. The importance rating on the right side of the QFD is then determined from the weighted CR based on the percentage of each customer requirement to the total. Similarly, the total weighted engineering specifications are determined by multiplying each number under an engineering specification by its corresponding customer requirement weightage then summing all of them. The importance rating on the bottom of the QFD is determined by the percentage of each engineering specification to the total.

3. INFORMATION SEARCH

Since this is the fourth stage of the Xebra hydraulic-electric vehicle project, most of the background information is available in the reports of previous teams who have worked on the project. We were not able to find a hydraulic-electric hybrid for proper comparison.

3.1 Technical Benchmarks

Currently there are no hydraulic-electric vehicles with regenerative hydraulic braking and hydraulic launching in production. Thus, we will benchmark our design against the initial state of the vehicle prior to the hybridization. With the addition of a hydraulic acceleration, the vehicle can now reach up to 27 mph on a full hydraulic charge. After the hydraulic launch, the high-pressure accumulators are re-charged using a slow-fill pump powered by the batteries, a process that takes two minutes. Our goal was to directly charge the high-pressure accumulators with the energy recaptured during braking, thus minimizing the power drawn from the battery and cutting down on the wait time for a fully charged hydraulic launch.

3.2 Patent Search

In the past, other ME 450 teams have designed and built a bike with regenerative braking and a hydraulic launch assist. The university has filed a patent for the regenerative braking technology (Patent # 20070126284). The EPA in conjunction with industry partners unveiled the first ever fully hybrid hydraulic vehicle in United Postal Service (UPS) delivery trucks [2]. The system was developed primarily with Eaton and the technology is under patent (Patent # 7,272,987). This is similar to our system since it has hydraulic propulsion, but the UPS trucks are series hydraulic hybrids with internal combustion engines while the Xebra is a parallel electric-hydraulic system.

4. CONCEPT GENERATION AND SELECTION

4.1 Function Analysis

It was important that our alpha concept design meets all of the important customer requirements laid out in the quality function deployment diagram. Therefore, we developed a FAST (Function Analysis System Technique) diagram. This diagram was used to prioritize the objectives of the final product. The FAST diagram takes the main objective of the project and breaks it down into sub-sections that describe the functions needed to obtain the main objective.

We have decided that the over-lying objective of implementing regenerative braking is to improve the performance of the Xebra vehicle. Therefore, this is the main objective in the FAST diagram. Branching off of this main objective are secondary objectives that detail the things that are essential to the performance of the task as well as the things that fulfill the basic needs of the user. Also, included in the secondary objectives are things that detail the wants of the user. Finally, on the far right side of the FAST diagram are the solutions to the overlying objective.

Since the main objective of the FAST diagram is to improve performance, we have decided on six secondary categories the include capture braking energy, assure safety, increase life of the batteries, assure reliability, assure convenience, and enhance products. Investigating further into the reliability category, one will find that having compatible materials is essential to the performance of the function. Breaking off from this, one will find the solution to assuring reliability is having optimum valves, correctly sized hoses, and a pump with optimum displacement. The rest of the FAST diagram can be analyzed in the same fashion. See Figure 1 for the complete fast diagram.

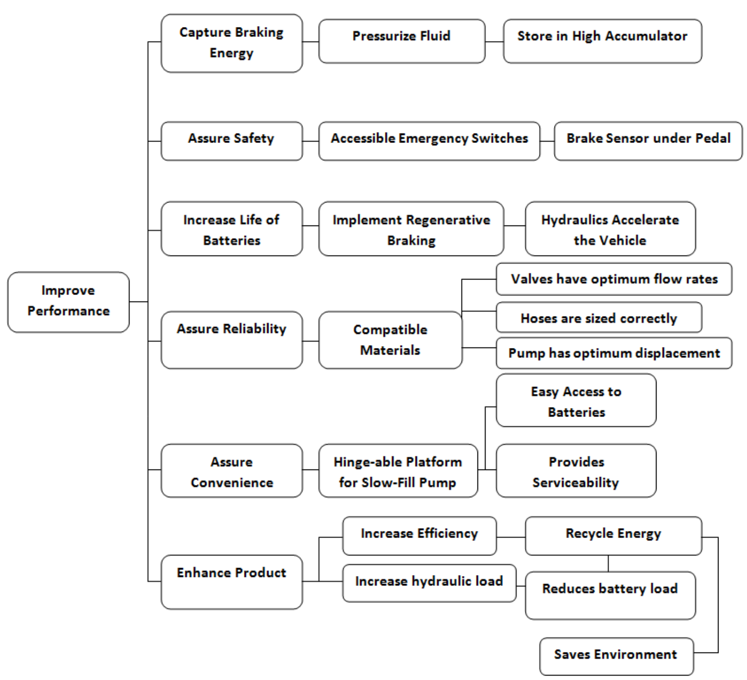


Figure 1: FAST Diagram

4.2 Design Concept Development

Since our design incorporated many elements, we separated the design into its components to generate an initial concept. We began our selection by creating a morphology chart outlying our possible choices.

Table 3 on page 6 shows the results of our concept development process. Our first choice was whether to use a fixed or variable displacement hydraulic pump. We elected to use a fixed displacement pump for two key reasons. Integrating a fixed displacement pump requires much less control logic. Secondly, this hydraulic-electric hybrid vehicle may be marketed to developing countries, so reducing cost and complexity is important. The downside of this pump choice is the inability to vary the braking force relative to the driver's brake pedal force and the lack of a smooth deceleration.

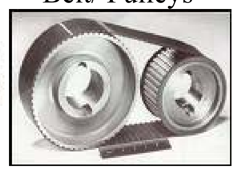
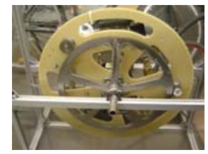
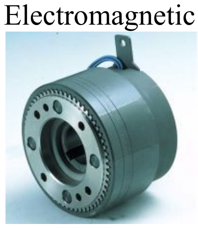
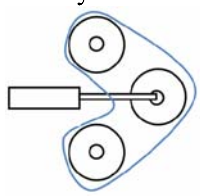
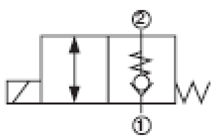
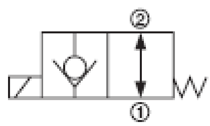
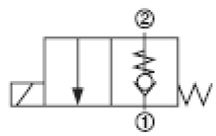
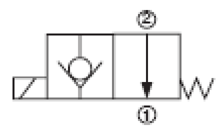
Two mechanical fixed displacement pump designs were considered in addition to the displacement type: gear pumps and piston pumps. Gear pumps pull hydraulic fluid through a set of gears when the input shaft is rotated. By contrast, piston pumps allow fluid to enter and exit a cylinder block. Since we are using relatively low flow rates, a gear pump will be more efficient. When choosing a gear pump, it is important that no net external force on the shaft can disrupt the meshing of the internal gears of a gear pump. Therefore, using a gear pump requires careful analysis, precise mounting, and effective coupling in order to avoid causing an external force on the shaft.

To provide a reduction between the hydraulic pump and the transmission shaft, we considered four possibilities: gears, a chain and sprocket, a belt and pulley system, or a reduction system similar to that of the ME 450 team's bike hub wheel. The gear would work by mating a gear on the transmission shaft to a gear mounted directly on the pump. This would effectively and efficiently transmit the torques, but it creates difficulties regarding mounting and packaging space. A belt and pulley system would allow for a unique clutch design, but would have limitations on torque. A bike hub wheel would be another possibility. However, this design would have to be added to and move with the front wheel, adding complexity and cost to the project. A chain and sprocket system was selected since it can handle the high torque requirements and can be easily integrated into the current system. The chain and sprocket system is also a proven concept since Winter 2008 successfully implemented it in their design.

A clutch is needed to engage the hydraulic pump while the vehicle is in motion in order to provide the regenerative braking. Four types of clutches were considered: an electromagnetic clutch, a piston and pulley system, a mechanical friction disc clutch, and a one-way clutch bearing. An electromagnetic clutch couples two shafts when provided a current. A piston and pulley system would provide tension to a belt when braking was required, and loosen and slip when not. A mechanical clutch couples two shafts by using friction when given a mechanical force. Lastly, a one way bearing allows for the driving shaft to drive the other shaft while also allowing the other shaft to rotate independently of the driving shaft. We determined that the clutch bearing will not work in our application because there is no way to control the engagement of the pump with it. An electromagnetic clutch was initially chosen because it provides the easiest switching and engagement method, allows for larger torques, transmits torques efficiently, and is smaller in size than the other options.

However, we were unable to find an electromagnetic clutch with small size, low mass, high rotational rates, reasonable cost, small lead time, and high torques. Instead, we found a hydraulic clutch that met all of our requirements. The hydraulic clutch is smaller, lighter, stronger, and faster than an electromagnetic clutch. A hydraulic clutch works by using hydraulic pressure to trip a spring that couples multiple friction discs, which transmit the torque.

Table 3: Morphology Chart

Design Parameters	Concept 1	Concept 2	Concept 3	Concept 4
Pump Displacement Type	Fixed ☑	Variable		
Pump Type	Gear ☑	Piston		
Reduction Type	Gears 	Chain/ Sprocket  ☑	Belt/ Pulleys 	Bike Hub Wheel 
Clutch	Electromagnetic  ☑	Piston/ Pulley System 	Friction Disc 	One Way Bearing 
Valve Type	Normally Closed Bidirectional  ☑	Normally Open Bidirectional 	Normally Closed Unidirectional 	Normally Open Unidirectional 
Valve Size	29 gal/min	40 gal/min ☑		
Location of Hydraulic Pump	Front of Vehicle	☑ Rear of Vehicle		

☑ *Indicates chosen concept.*

As a safety measure, emergency stop valves were mounted to each of the high-pressure accumulators. If a hydraulic line blows, power can be cut off so that the valves stop any high-pressure fluid from leaving the accumulators. It is important that these valves be able to meet the maximum possible flow needs, minimize pressure drops, allow bi-directional free flow when energized, and act as a check valve when de-energized (prevent flow from accumulators). We decided that a 40 gal/min, normally closed, bi-directional solenoid poppet valve meets all of the discussed requirements.

We considered coupling the hydraulic pump to the front wheel and rear axle. We originally thought that a front pump location would create extra head pressure due to the fluid's inertial forces in braking to help avoid cavitation, but calculations showed that this potential advantage would be negated by fluid losses in the longer required piping. The complete cavitation analysis is discussed in the Parameter Analysis section below. A front wheel pump location also presents the difficulty of requiring the pump and coupling mechanisms to rotate with the wheel and the associated increased cost to overcome this difficulty. The rear axle has more space for pump mounting and the previous term already installed a transmission shaft with which we can couple our pump. A potential disadvantage is its close proximity to the hydraulic reservoir, implying more sharp bends with losses may be required. We carefully planned in order to minimize the number of bends.

These concept selections make up our Alpha design as seen below in Figure 2.

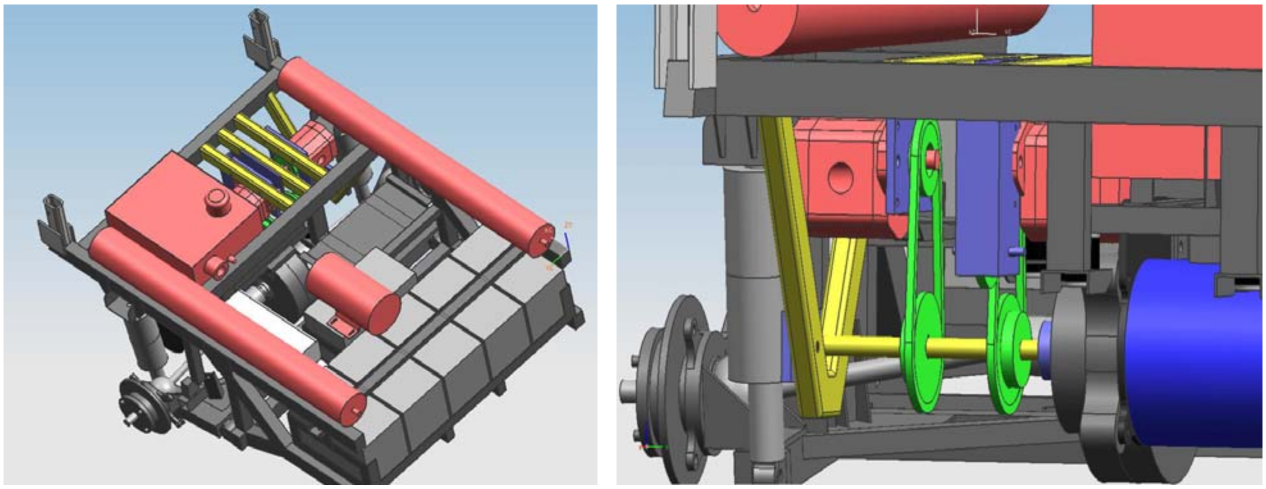


Figure 2: CAD Model of Alpha Design

5. ENGINEERING DESIGN PARAMETER ANALYSIS

5.1 Mechanical

Clutch Torque Requirements. Using Equation 1, the maximum torque at the hydraulic pump is 144.8 Nm during braking.

$$\tau = \frac{EngineDisplacement * Pressure}{2\pi} \qquad \text{(Eq. 1)}$$

The torque at the shaft and clutch is then this maximum torque divided by the speed reduction, 18:20, between the pump and the shaft. Solving, the maximum torque at the clutch and shaft is 161Nm. The clutch used in our design, Logan clutch P-Series Model 350, has a maximum allowable torque of 438Nm, providing a safety factor of 2.7.

Drive Cup Attachment. The clutch drive cup needs to be attached to the sprocket on the drive shaft. The fasteners need to have enough shear strength to accommodate the maximum torque at the clutch. The attachment was made 1 5/16" from the center with six ¼" countersunk bolts. To calculate the force at each bolt location, we divided the torque by the radius to the hole center. The shear at the bolt is then calculated below in Equation 2.

$$\tau = \frac{F}{\pi r^2} \qquad \text{(Eq. 2)}$$

The shear at each bolt is then 38.14 MPa. Each bolt has an ultimate shear strength of 596 MPa providing a safety factor of 15.63.

Gear Ratio and Torque. For the vehicle to use regenerative braking at its top speed of 35mph with a 33cc hydraulic pump, a 5:1 total gear ratio is desired. With a 20 inch (0.508 meters) wheel diameter, the vehicle travels 1.6 meters per revolution of the wheel. Therefore, when the vehicle is traveling at its top speed, the wheels are rotating at 454 rpm (Equation 3).

$$1.6m * 585rpm = 15.6 \ {}^{m}/_{s} = 35\ mph \qquad \text{(Eq. 3)}$$

The maximum rotational speed of the 33cc hydraulic pump is 3100rpm. Given this limit, we calculate a gear ratio of 5:1 from the hydraulic pump to the wheel rotational rate. The gear reduction between the motor (where the drive shaft is coupled to) and the wheels is 4.5. So, a sprocket reduction of 20:18 was selected to achieve the total 5:1 reduction required. Additionally, the clutch needs to accommodate the maximum rotational rate of the drive shaft at 2790 rpm. The clutch selected has a maximum rotational rate of 3600rpm.

Mounting. The pump was mounted next to the motor under the truck bed in the back of the Xebra. It was mounted using the same technique as the motor and with the same materials. To mount our pump, a 15.5 inch piece of 14 GA 1117 steel rectangular tubing was welded next to the current steel rectangular tubing in the rear of the vehicle. This steel was chosen to match the material of the truck for welding ease. A 7"x7.25"x0.5" piece of 6061 aluminum was then bolted to the steel tubing and the motor mounted on the aluminum plate. The aluminum was machined to allow for a secure and flush fit of the motor surface. Engineering drawings with dimensions for the aluminum plates are in Appendices B and C. The mounted pump is included in the model in Figure 9 on page 16. The design is validated by its successful prior use in mounting the hydraulic motor which is of similar size and weight with the pump.

Modeling. Michael Woon, a Mechanical Engineering Graduate Student, is developing a Simulink model of our Xebra vehicle. This model will be developed fully to replicate our final design. One extremely useful feature of this model is its ability to "drive" the vehicle through dynamometer test data. The LA4 test cycle previously performed on the Xebra was driven through this model (Figure 3). Hand calculations of torque and rotational rates of the hydraulic pump were confirmed using this Simulink model (Figure 4).

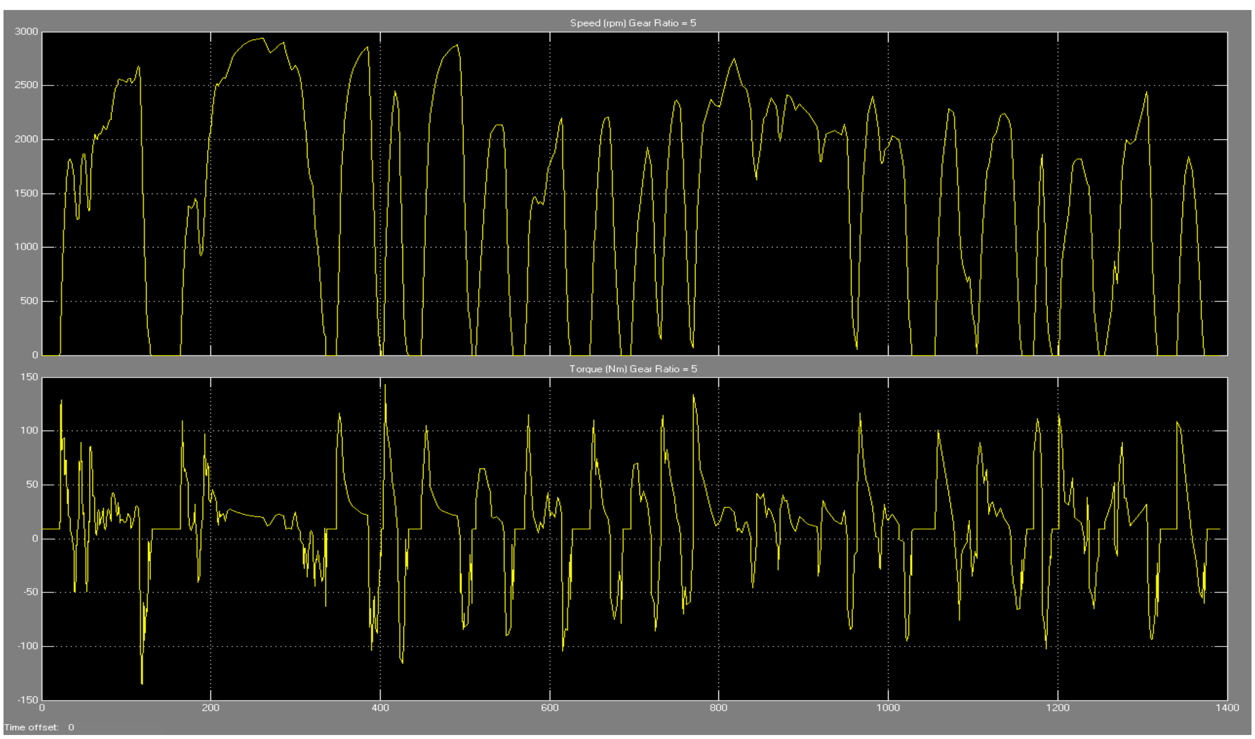


Figure 3: Rotational Rate and Torque through complete LA4 Test Cycle

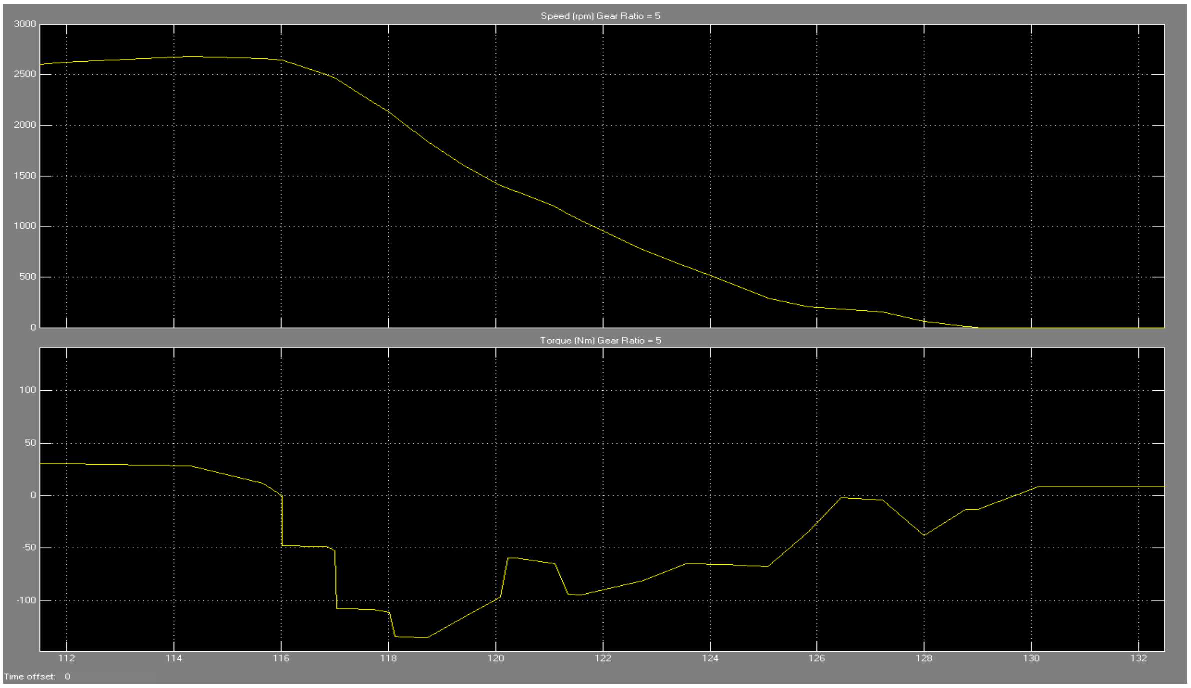


Figure 4: Rotational Rate and Torque at Hydraulic Pump during LA4 Braking Event

5.2 Hydraulic

Cavitation Analysis. In order to determine if placing the hydraulic pump caused an issue with cavitation, we performed an analysis. Cavitation is an important factor to consider when designing hydraulic systems containing pumps. When the hydraulic fluid pressure is less than its vapor pressure, cavities can arise due to local vaporization of the fluid. As these cavities meet higher pressures (like in a pump), they can collapse and impinge on the system's surfaces. This process creates vibrations, noise, and physical damage [3]. Another cause occurs when the fluid is not supplied fast enough to the rotating pump, forming a vacuum between the pump inlet and the fluid. This causes a loss in capacity and efficiency [4]. We completed several calculations to ensure that the pump's inlet pressure is above oil's vapor pressure (~0 kPa) when it is initially activated [5].

The first concept involved coupling the pump to the front wheel to take advantage of fluid momentum and inertial forces during braking that would push the fluid towards the front pump and increase the pump's inlet pressure. In order to evaluate this idea, we calculated a pressure advantage due to these phenomena. The pressure gain from the inertial forces at the beginning of deceleration equals 1.44 psi and results from Equation 4, which states that pressure equals force (fluid mass in the hose multiplied by acceleration) divided by pipe area; where P is the pressure in the hose, A is the cross-sectional area of the hose, a is the fluid acceleration, and m_f is the mass of the fluid. However, viscous losses in the long hose would cause a pressure drop of 1.76 psi using Equation 5 [6]. In Equation 5, the maximum flow rate, Q_{max} results from the pump's displacement, initial vehicle velocity, gear ratio, and tire diameter and a represents the hose radius. This shows that the pressure drop in the long length of hose required completely negates any advantages from momentum or inertial forces and causes a net pressure loss, discrediting the main idea behind the front wheel pump concept. Note that we assumed a distance of 2 m between the low pressure reservoir and the front wheel as well as a hose diameter of one inch.

$$P = \frac{m_f * a}{A} \qquad \text{(Eq. 4)}$$

$$\frac{dp}{dx} = \frac{-8\mu Q_{max}}{\pi a^4} \qquad \text{(Eq. 5)}$$

We also evaluated the second option, placing the pump in the rear of the vehicle, to ensure that the pump's inlet pressure remained above oil's vapor pressure. This idea does not use inertial forces or a long length of hose, but would probably have several tight bends due to the close proximity of the low pressure reservoir. Bends cause pressure losses that are greatest at the highest fluid flow rates and velocities, which occur when the pump is initially engaged. If one assumes a straight hose length of one meter, a maximum pressure drop of 0.88 psi would occur, as seen in Equation 5. Equation 6 quantifies pressure losses across bends and fittings with v as the fluid velocity, ρ as the fluid density, and K_L as the loss coefficient, which depends on the geometry [7]. If one assumes the maximum allowable pressure loss is the difference between atmospheric pressure and the hose pressure drop, a maximum loss coefficient of 10.94 is found. As long as the sum of all the loss coefficients is less than 10.94, the pump's inlet pressure will be above oil's vapor pressure and vacuum pressure. Since a system with one 180º bend, one valve, one tee branch, one check valve, and two 90º bends has a loss coefficient less than 10.94 and contains more bends and fittings than should exist between our system's hydraulic reservoir and pump inlet, it is safe to assume a rear mounted pump will maintain an inlet pressure great enough to avoid cavitation.

$$\Delta P = \frac{1}{2}\rho K_L v^2 \qquad \text{(Eq. 6)}$$

We also incorporated several measures to ensure that the fluid supply is able to adequately supply the pump at the highest volumetric flow rate, when the pump is initially engaged. First, we used a relatively large hose diameter of 0.75 inch that can provide a large flow rate. We carefully planned to minimize the number of bends and fittings impeding flow between the hydraulic reservoir and the pump's inlet. Finally, if it is found that these passive methods are unable to ensure a sufficient fluid supply to avoid vacuum creation, an electric feeder pump can be incorporated to supply a steady flow of fluid to the regenerative braking pumping when it is initially engaged.

Valves. All the valves that connect to the high side accumulators were sized to operate at 4000 psi of pressure and to allow for a flow rate of 40 gpm. The highest expected flow rate is 26 gpm so a flow exceeding valve specifications is really a nonissue. Furthermore, the high flow rates of the valves minimize pressure drops, increasing system efficiency. We also chose to use poppet-type valves because the leakage rate is significantly less than most other types. With the exception of the recirculation valve, which is in a series configuration with the pump, all the valves are normally closed during operation (see Figure 10 on page 17). The selected valve configurations allow them to be controlled correctly by the electronics and open at the desired time.

Pump. The pump was supplied by the EPA at the start of the project. It is a 33cc fixed-displacement gear pump from Parker-Hannifin made of aluminum. The pump operates at 3100 maximum rpm with a continuous pressure of about 4000 psi. Previous teams sized the pump so that it can adequately meet the maximum deceleration rate or braking needs of the LA4 cycle. A fixed displacement pump will not stop the vehicle as smoothly as a variable displacement pump, but the design of the controls is much simpler. Also, a gear pump has better efficiencies at the flow rates and range we are expecting.

Deceleration. The average deceleration of the Xebra vehicle with the regenerative braking system will be -2.94 m/s^2, implying that the vehicle will stop from 35 mph in 5.3 seconds and within 136 feet. During the Fall 2007 term, the Xebra team found that the maximum deceleration in the "LA4" test, an EPA standard test that simulates city driving, was -2.146 m/s^2. Even though the predicted average deceleration is only an estimate, the fact that it exceeds the greatest required deceleration in the LA4 test proves the adequacy of the purchased pump and validates the hydraulic braking system concept.

We calculated an average volumetric flow rate (Q_{avg}) of 43.68 Lpm using Equation 7 with V as the pump displacement (cm^3), n_{avg} as the average pump revolutions per minute, and η_{vol} as the volumetric efficiency

(.9) [8]. Note that n_{avg} was found using the average vehicle speed (17.5 mph), the tire diameter of 20 inches, and an assumed gear ratio of 5.

$$Q_{avg} = \frac{V * n_{avg} * \eta_{vol}}{10^3} \qquad \text{(Eq. 7)}$$

$$P_{avg} = \frac{Q_{avg}}{600 * \eta_{pump}} \Delta p_{avg} \qquad \text{(Eq. 8)}$$

During deceleration, the pump absorbs an average power (P_{avg}) of 22.55 kW. Equation 8 yielded this value with η_{pump} as the pump's overall efficiency, Δp_{avg} as the average pressure difference across the pump in bar, and previously calculated average flow rate [8]. We assumed a pump efficiency of 72 percent and a pressure difference of 3250 psi (224.1 bar) to atmosphere (1 bar). 3250 psi represents a middle value of the accumulator pressure since it is pre-charged at 2500 psi and pumped up to 4000 psi.

$$t = \frac{\frac{1}{2}m * v_i^2 \; - \; t * P_d}{P_{avg}} \qquad \text{(Eq. 9)}$$

$$t = \frac{\frac{1}{2}m * v_i^2}{P_{avg} + \; P_d} \qquad \text{(Eq. 10)}$$

We calculated a deceleration time (t) of 5.32s using Equation 10, a rearranged form of Equation 9. Equation 9 assumes that the energy difference between the kinetic and drag energy of the vehicle is absorbed by the pump at the calculated average power in order to stop the Xebra. The kinetic energy is half of the vehicle's mass (m) multiplied by the initial velocity (v_i) squared. Using Bosch's *Automotive Handbook*, we found the average drag power (P_d) to be 1.4575 kW assuming a frontal area of 2.07 m^2 and half the initial velocity [9]. The drag energy is the drag power multiplied by time and it is negative because it slows the Xebra.

$$a_{avg} = \frac{\Delta v}{t} \qquad \text{(Eq. 11)}$$

$$d = t * v_i \; + \frac{t^2 * a_{avg}}{2} \qquad \text{(Eq. 12)}$$

The average deceleration (a_{avg}) of -2.94 m/s^2 will stop the Xebra within a distance (d) of 136 feet (41.60 m). Equation 11 states the average acceleration simply as the change in velocity (Δv) divided by time. Furthermore, basic physics relates distance, time, initial velocity, and acceleration as Equation 12.

Recovered Energy and Volume. The proposed hydraulic regenerative braking system can recover 86.34 kJ of energy during braking from 35 mph to zero, which represents 67.6 percent of the initial kinetic energy. Equation 13 shows that the pump's inefficiencies cause only 72 percent of the pump's absorbed energy, average absorbed power multiplied by time, to be actually stored in the system. Equation 14 states that the regenerative braking system's efficiency equals the stored energy (E_{regen}) divided by the initial kinetic energy at 35 mph.

$$E_{regen} = \eta_{pump} * t * P_{avg} \quad \text{(Eq. 13)}$$

$$\eta_{regen} = \frac{E_{regen}}{\frac{1}{2} * m * v_i^2} \quad \text{(Eq. 14)}$$

During the deceleration, the hydraulic regenerative braking system stores 3.87 L of hydraulic fluid in the accumulators. The stored fluid in the accumulators (V_{acc}) equals the average volumetric flow rate multiplied by time (Equation 15).

$$V_{acc} = t * Q_{acc} \quad \text{(Eq. 15)}$$

In addition to the 33 cc pump, the Xebra also currently has a slow-fill pump that will be used to "top off" the accumulators while waiting at a stop light. In order to obtain the 4.57 L that the winter 2008 team calculated as necessary for a launch [10], the slow-fill pump will need to operate for 67s at an average flow rate of 0.0315 L/s and an average pressure of 3750 psi [11]. Overall, the hydraulic regenerative braking system and the slow-fill pump should provide all the energy necessary for acceleration up to 27 mph.

Hosing and Connectors. The hoses were sized to meet the requirements of different parts of the system and range from 6/16th to 1 inch in diameter (JIC 6 to JIC 16). Each connector was individually chosen to fit the hoses, the pump, the accumulators, the valves, and the motor.

JIC-16 size fittings and one inch diameter hoses were chosen for the regenerative braking system's high pressure lines and valves. The energy of a hydraulic regenerative braking system is stored in the pressure of the fluid so minimizing pressure losses throughout the system is paramount. As Figure 5 shows, pressure losses are minimized at larger diameters. Pressure losses in fittings and bends are also reduced at lower fluid velocities, which are lower for larger hose diameters at a given volumetric flow rate. Finally, JIC-16 fittings have reasonable costs, good availability, and do not compromise the flow rate.

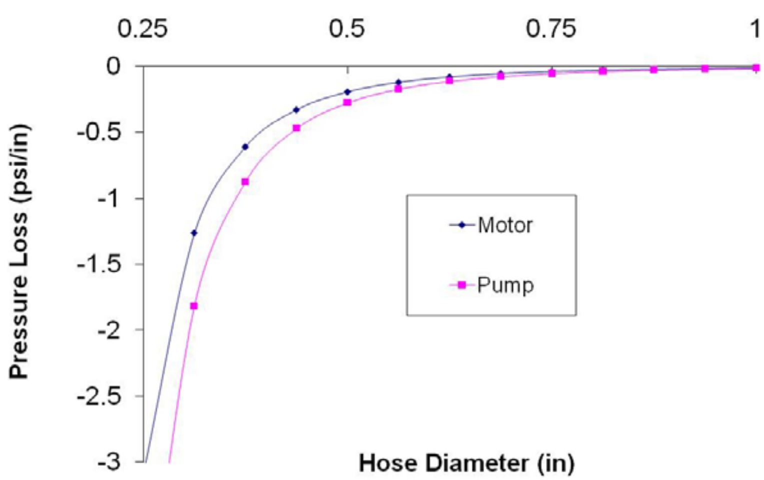


Figure 5. Pressure losses for maximum motor and pump flow rates decrease

In order to calculate the pressure losses in the hoses, we modeled the hoses as straight pipes and assumed steady, laminar flow. This situation yields Equation 16 with Q_{max} as the maximum volumetric flow rate, a as the pipe diameter, μ as the dynamic viscosity, and dp/dx as the change in pressure over length [6].

$$\frac{dp}{dx} = \frac{-8\mu Q_{max}}{\pi a^4} \quad \text{(Eq. 16)}$$

The maximum flow rate (Q_{max}) was calculated based on a maximum vehicle speed (v_{max}) of 27 mph (12.07 m/s), the tire diameter (D), the displacement of the pump (V), and an assumed gear ratio (GR) of 6. Equation 17 puts this relationship in equation form.

$$Q_{max} = \frac{V * GR * v_{max}}{\pi * D} \quad \text{(Eq. 17)}$$

For a given volumetric flow rate, fluid velocity decreases with increasing area, implying that fitting pressure losses decrease. Equation 18 gives the relationship for pressure losses (Δp) across fittings at a given fluid velocity (v). Note that γ is the specific gravity of the hydraulic fluid, g is the acceleration due gravity, and K_L is the loss coefficient. The loss coefficient depends on the geometry of the fitting: K_L equals 1.5 for 90 degree bends, 2 for tee branches, 0.9 for tee lines, and 2 for check valves [12].

$$\Delta p = \gamma * K_L \frac{v^2}{2g} \quad \text{(Eq. 18)}$$

Low Side Accumulator. The existing outlet from the low-side accumulator does not allow for hosing access to the pump; therefore, we added an outlet on the accumulator's bottom side. Tapping a new hole into the bottom of the accumulator allowed for a more direct and out of the way plumbing line to the pump which is mounted behind the accumulator. After consulting with our sponsor at the EPA and with Bob Coury at the University of Michigan, we decided to tap a hole on the bottom side of the accumulator 6" from the back and 4" in from the outer corner of the accumulator. We cleaned out all of the oil with soapy water and welded a round stock with 0.75" female pipe thread. To do this, we also removed the low-side accumulator from the Xebra to bring it to a welding location.

5.3 Design for Manufacturability

The Xebra hydraulic-electric vehicle is currently a unique design and is intended as a prototype at this stage. It is meant as a forum to explore the applicability and feasibility of a hydraulic-electric hybrid vehicle. The prototype has potential for future mass manufacturing, but this depends on the overall efficiency increase at the final completion of the project as well as the outcome of a cost-benefit analysis. Our main goal was to achieve a working prototype, which can then be streamlined for manufacturing.

The goal is to generate a working prototype with marked increases in performance when compared to the baseline electric vehicle. Thus, designing for mass manufacturing was not a priority. However, one of the main customer/engineering requirements was transferability to future ME 450 teams that will continue work on the project. To fulfill that requirement, we designed for ease of assembly and disassembly.

The criteria in our search for mounting plate material were the material yield strength (Young's Modulus), maximum cost per unit mass of material, and material density. We calculated that we needed material yield strength of at least 2 kPa with a low cost and medium density. Based on the Cambridge Engineering Selector (CES) software, possible materials were wood, high-strength foam, graphite, and other metals such as tin-lead alloys and aluminum. We selected aluminum 6061 because it met or exceeded all the criteria, has a high resistance to corrosion, is easy to machine, and is readily available.

5.4 Failure/Safety Analysis

For our prototype, the most hazardous risk would be hydraulic oil leaking onto the vehicle's components. To address this, hydraulic fittings will be meticulously tightened and safety mats will be placed on top of the electrical components to protect them. These safety mats can absorb the hydraulic oil. Moreover, we have also installed e-stop valves that prevent high pressure fluid from flowing throughout the system during leaks.

Another slightly less hazardous issue present in the vehicle is the vibration and movement of components as the vehicle moves. To address this issue and prevent failure, we have created special fixtures for our components to reduce vibration. For example, we have keyed our clutch to our drive shaft so that it locks into place. We have used keys to lock our components into place as well as nuts and bolts.

In our calculations throughout the project, we used a safety factor between 5 and 20 when selecting our components. This will ensure that the vehicle's hydraulic and mechanical components have as long a life as the batteries. Due to lack of testing, the exact lifetime is currently unknown. Appendix D contains the complete Failure Mode and Effect Analysis (FMEA) using the DesignSafe software.

5.5 Design for the Environment

Due to the scope of the project, it was not possible to get a full accounting of all the parts and material types, nor the mass of each that was installed on the vehicle during this semester. Thus, we focused the environmental impact assessment on parts that were manufactured from metals- this includes the pump, the hydraulic clutch, the hose fittings, the sprockets and chain, the steel cross member, and the mounting plates. The listed parts are made of steel or mostly of steel, so they are approximated as steel for the purposes of this analysis; the mounting plates are made of aluminum. We installed approximately 40 kg of steel and 1 kg of aluminum on the Xebra.

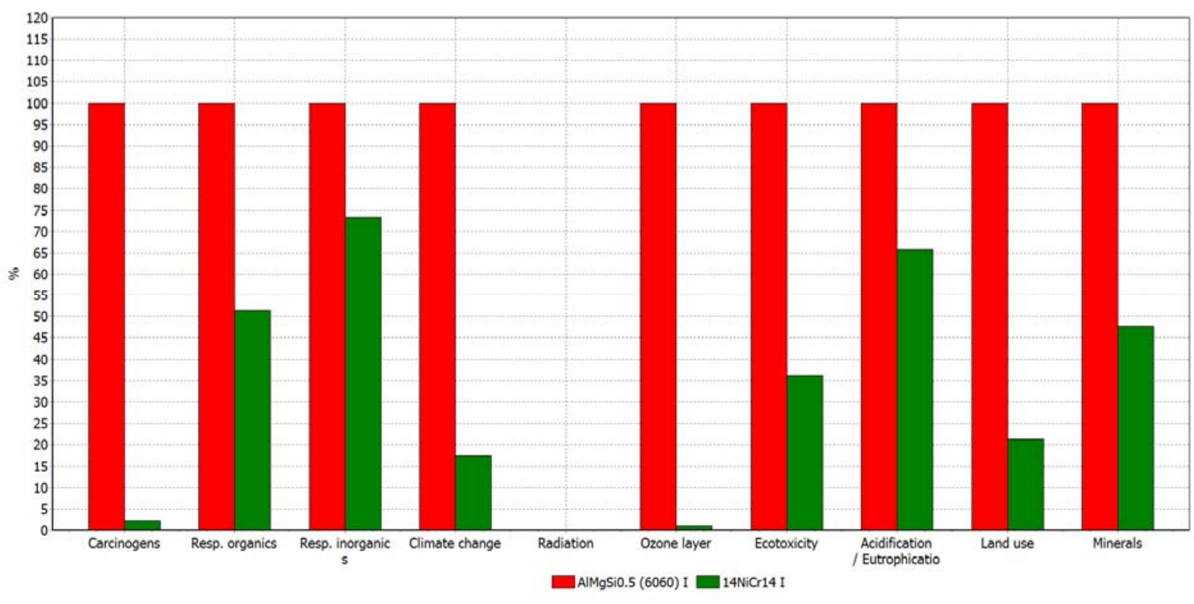


Figure 6: Characterization of Emissions

The environmental impact assessment was conducted using the SimaPro7 program. And, for a better comparison of environmental impact, we evaluated the materials per unit mass. The results are plotted below on Figures 6, 7 and 8. Figures 6 and 7 show the emissions from mining and refining the materials;

Figure 6 characterizes the emissions and Figure 7 summarizes them on a point bases. From the scoring in Figure 7, the aluminum scoring 3 points has more than twice the impact as the steel, which scores about 1.45 points. Figure 8 compares mass quantities of resources that are used or wasted per kilogram mass of each material. Again, the waste from aluminum far outweighs that of the steel.

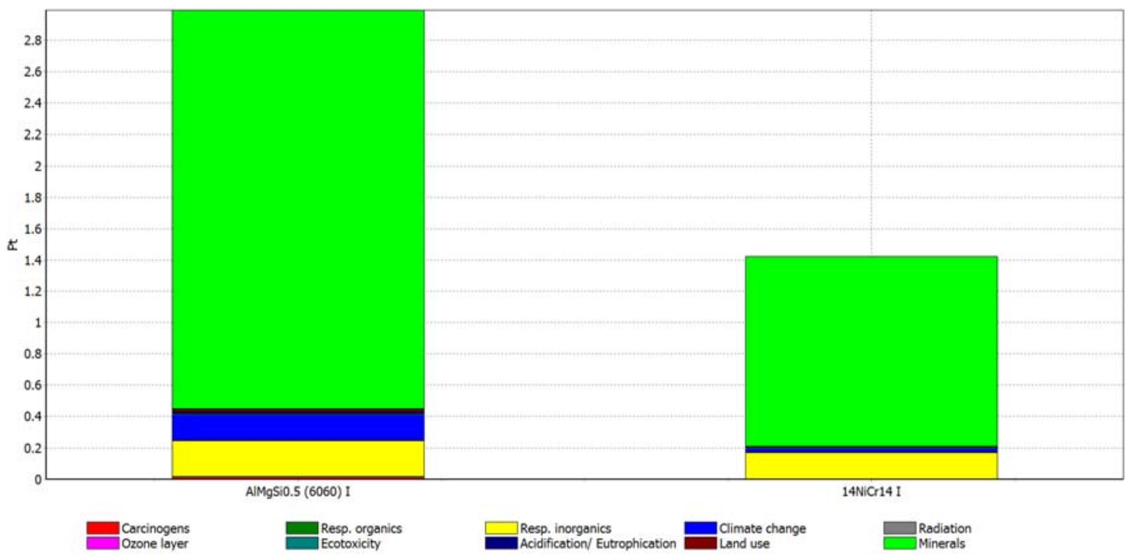


Figure 7: Summary of Emissions

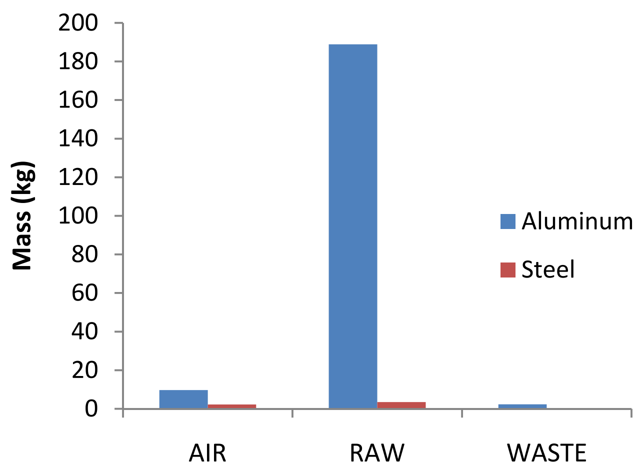


Figure 8: Resources Expended in Material Production

For a more environmentally friendly design, future teams should consider manufacturing new Xebra components from steel. Not only is steel better than the aluminum in environmental impacts, it also is a stronger material so less of it may be needed. The drawbacks of steel components lie in the greater weight since steel has a higher density and the increased machining difficulty due to steel's higher strength.

There are also several fundamental design aspects that make the Xebra vehicle more environmentally friendly. The hydraulic system reduces the impact of the vehicle since there is less battery consumption during accelerations and energy is saved during braking. Furthermore, there are no consumables such as gasoline. Replacing materials will also be minimal since hydraulic fluid has a very long life and the battery life will be increased due to decreased loading.

6. FINAL DESIGN DESCRIPTION

6.1 Mechanical

The final mechanical design layout incorporates many elements from our Alpha design. Several changes were made after later review and component selection. A hydraulic actuated clutch was used in place of the electromagnetic clutch. This was due to our inability to find an electromagnetic clutch with an appropriate torque rating, maximum rotational rate, and packaging space to be used in our design. Additionally, cost and availability factored in to our selection of the Logan P-350 hydraulic actuated clutch. We used the existing slow fill pump to supply pressure to the clutch when engagement is needed. See Figure 9 for the CAD model of the final design.

An anti-rotation bracket was designed to prevent the stationary portion of the clutch from rotating when the through shaft is rotating. Per Logan's specifications, this bracket should allow for some rotational float and not be rigidly attached. A sprocket will be attached to the drive cup via set screws. This sprocket will be mounted on bearings so that the drive shaft can rotate without the sprocket rotating when the clutch is disengaged. When the clutch engages, the drive shaft, drive cup, and sprocket will rotate together, causing the pump shaft to rotate and pressurized fluid to be pumped throughout the system.

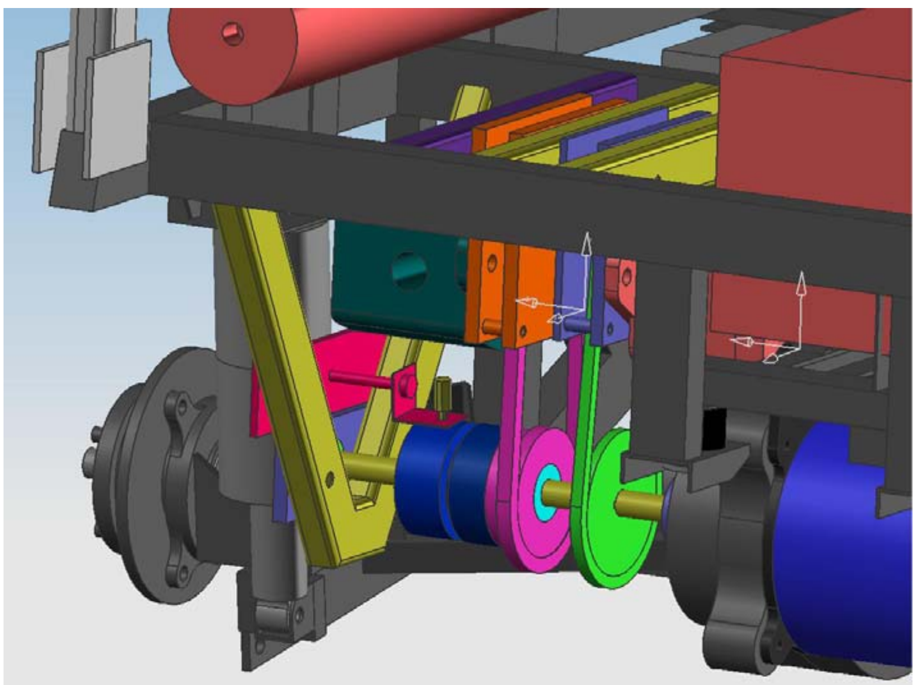


Figure 9: Final Mechanical Design CAD Layout

Thought was put into the spacing of the two sprockets to accommodate the roller chain. A tight fit is desired to provide good efficiency while still allowing a slight slack in the chain. Engineers at Martin Sprocket & Gear, our sprocket supplier, communicated that roller chains typically operate between four and six percent slack. We determined the length of the ANSI 60 chain to be 2.625 feet and consist of 42 chain links allowing for 4.1% slack between the 7.932 inch center to center distance.

The gear reduction between the hydraulic pump and shaft was chosen to accommodate the maximum allowable rotation at a vehicle speed of 35 mph. The reduction between the pump and shaft (18:20) and the gear reduction in the differential (4.5:1) allow for a total reduction of 5:1.

6.2 Hydraulic

The designed hydraulic layout can be seen in Figure 10. This hydraulic system will allow for the acceleration and braking of the Xebra vehicle. The normally-closed E-stop valves outside the accumulators prevent pressurized fluid from leaving the accumulators if a line blows. A two position, 3 way valve allows fluid to flow to actuate the clutch and also drains the clutch to disengage it.

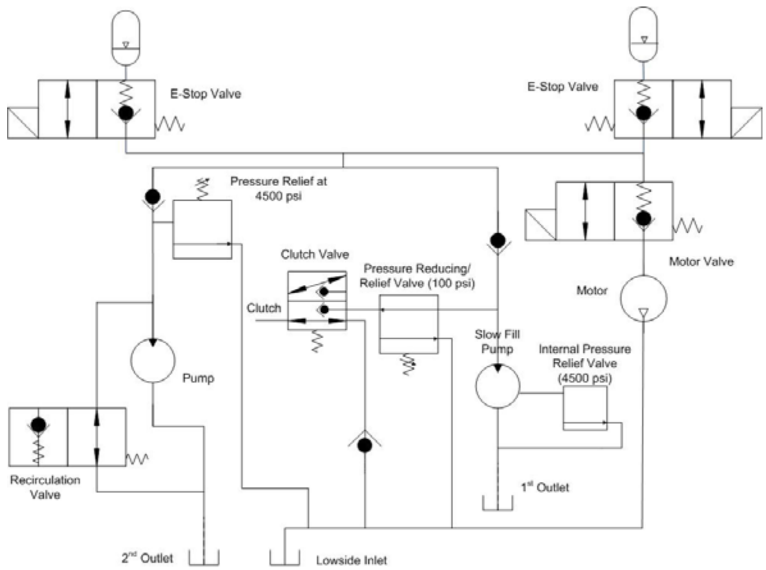


Figure 10: Designed Hydraulic Layout

When the driver presses his or her foot on the accelerator pedal touch pad, the motor and e-stop valves open allowing high pressure fluid to pass through the hydraulic motor and drive the vehicle as shown with blue lines in Figure 11. The dotted line denotes the clutch releasing pressure back to the low pressure reservior after braking to disengage the clutch. Check valves prevent the high pressure fluid from flowing across the slow-fill pump and the braking pump, which could actually cause the vehicle to go backwards.

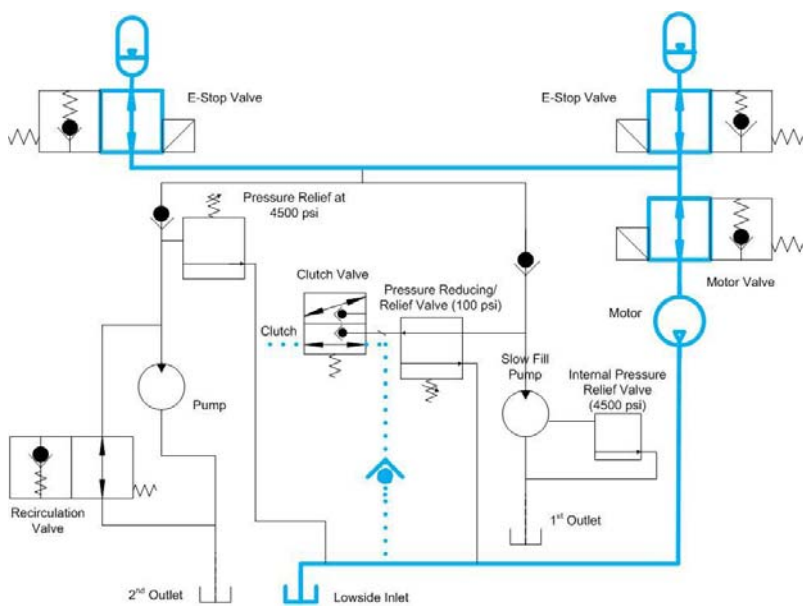


Figure 11: Acceleration Event

When the driver presses his or her foot on the brake pedal touch pad, the slow fill pump is used to engage the clutch. A pressure reducing/relieving valve reduces the slow-fill's pump pressure to the necessary 100 psi. Plus, any pressure spikes can be relieved back down to the low side reservoir. Fluid is re-circulated around the pump initially with the normally open recirculation valve to allow the clutch to engage before loading in order to reduce wear on the clutch. See Figure 12 for the initial braking fluid flow.

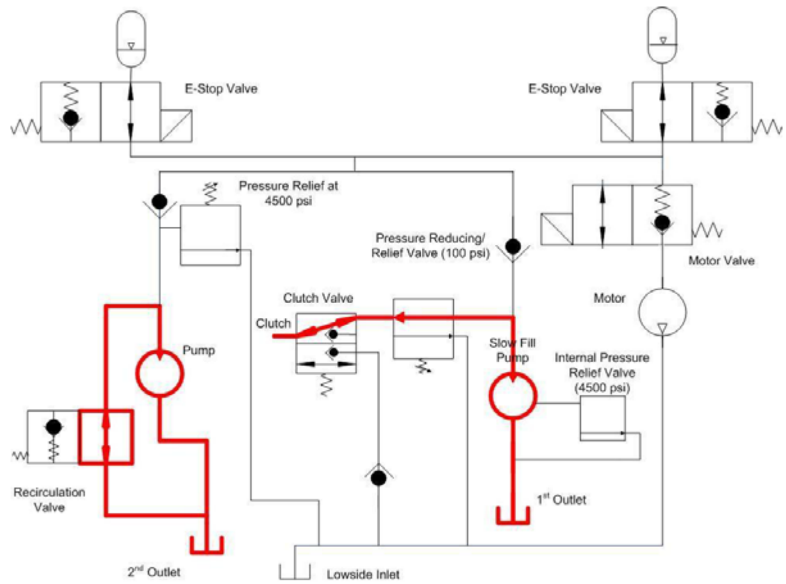


Figure 12: Initial Braking Event

After the prescribed time, the recirculation valve closes and the hydraulic pump pumps fluid up to the high pressure accumulators and the vehicle decelerates as shown in Figure 13. Once the clutch actuation pressure of 100 psi is reached, the pressure reducing/relieving valve shuts so that the slow-fill pump also pumps pressurized fluid up to the high pressure accumulators. Finally, we included a pressure relief valve that blows at 4500 psi to prevent pressures from reaching dangerous levels.

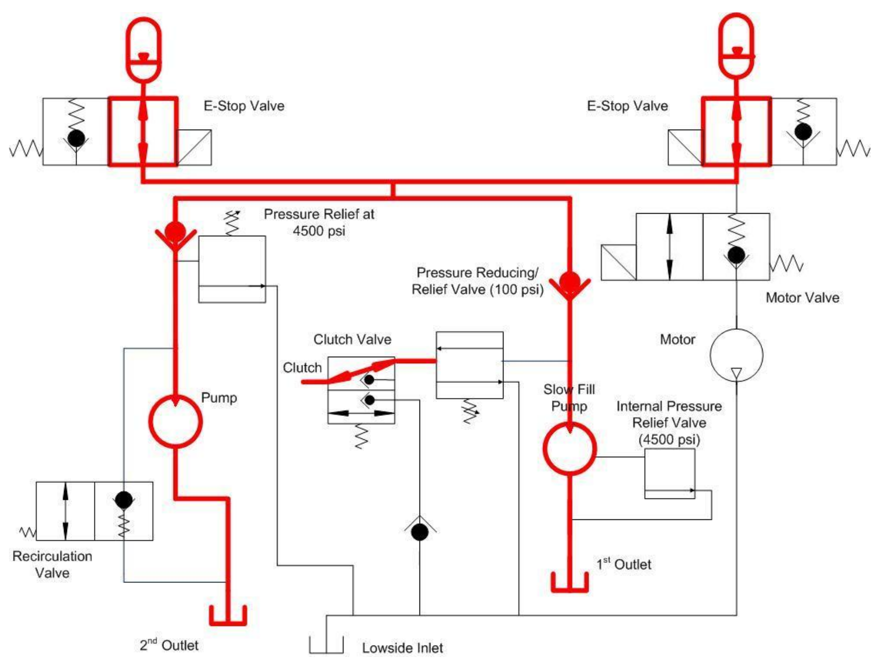


Figure 13: Steady-State Braking Event

6.3 Electrical

Our electrical design integrates into the existing circuitry and computer controls previously on the Xebra vehicle. During summer 2008, a computer program was written by Zachary Salzbank, a computer engineering student, which has the capability of monitoring and controlling aspects of the vehicle. A "base station" mounted to the vehicle consisting of a circuit board and PIC microprocessor connects to relays that turn on and off components.

As configured, it has the capability of monitoring four digital and six analog signals. Currently, two digital signals are being used by the hydraulic motor valve and the slow fill pump. Two analog signals are being used by a flow meter and a pressure sensor both at the outlet of the slow-fill pump. In addition, a wireless connection is available with a laptop. This allows for monitoring, controlling, and data logging away from the vehicle. The RPM sensor and a second pressure transducer can be wired into the base station so that these values can be monitored on a remote laptop.

A Hall Effect sensor was added to measure the frequency of sprocket teeth passing. This frequency was then calibrated to monitor the rotational rate of the hydraulic pump using the Newport P6430A Tachometer. The tachometer has a 6-digit readout and an analog current output that can be read by the base station. This sensor can be used to ensure that the pump rotational rate does not reach an unsafe level.

The vehicle is wired to allow full vehicle functionality even if there is a problem with the PIC microcontroller. This is achieved by hard wiring the accelerator and brake pedal touch sensors directly to relays 1 and 2. Relays 1 and 2 also allow for the appropriate valves and components to operate at the desired time. There is also a switch to set the vehicle in manual or computer control mode. See Figure 14 on the next page for the complete wiring diagram.

Relay 1: Hard wired to the accelerator pedal
- Accumulator E-stop valves (x2)
- Motor Valve

Relay 2: Hard wired to the brake pedal
- Accumulator E-stop valves (x2)
- Slow fill pump
- Clutch two position 3 way valve
- Time-delay relay pump recirculation

Relay 6: Emergency stop
- Buzzer

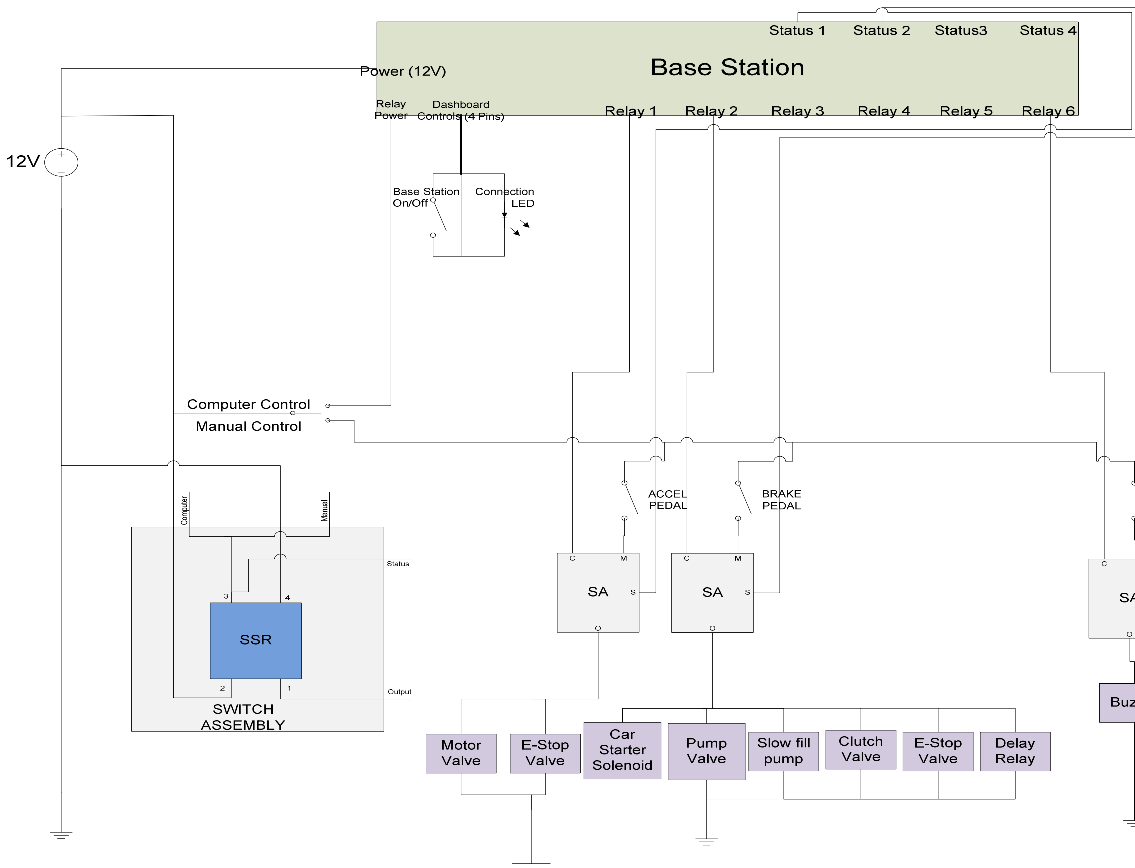


Figure 14: Vehicle Wiring Diagram

7. MANUFACTURING PROCESS

Mounting Plates. The manufacturing process involved different processes such as milling, welding, drilling, lathing, and cutting. We mounted our pump on a 7”x7.25”x0.5” piece of 6061 aluminum. This piece was milled and drilled to the specifications detailed in Figure 26 of Appendix B. We also used a second 6061 aluminum piece to mount the bearing that supports the other side of the pump’s shaft. This plate was originally milled and drilled to the specifications laid out in Figure 25 of Appendix B. However, we milled down the arc section an extra 0.15 inches to leave a thickness of 0.1 inches in this region. Also, we extended the arc region from a diameter of 5.375 inches to 5.5 inches. These changes will allow the chain to have more clearance so that it does not hit any part of the plate during operation. We have also milled out the inner arc region so that only a 3 inch diameter circle exists. Therefore, the entire inner part of the plate has been milled down to a 0.1 inch thickness except for the 3 inch diameter circle which still is at a thickness of 0.5”. The desired dimension of separation between our two plates was 0.875 inches. To stabilize the two plates, four 0.75” diameter aluminum spacers with lengths of 0.875” were created on the lathe and bolted between the plates. We manufactured both aluminum plates in the machine shop with the assistance of Marv Cressey.

Crossbar. We purchased a 3’ piece of 14 GA 2”x1.25” steel tubing for our crossbar. We welded a 15.5-inch piece of the steel next to the current steel tubing in the rear of the vehicle. To cut this piece down to 15.5 inches, we used a band saw in the machine shop and then welded the cross member into the vehicle with the assistance of Bob Coury. Since we planned to bolt the aluminum mounting plates to the cross member, we drilled the crossbar’s holes with the mounting plates as guides to assure accuracy. Note that this involved moving the motor assembly out of the way.

Low-side Outlet. Next in the manufacturing process was adding the new outlet to the low-side reservoir. This involved removing it and one of the high pressure accumulators from the vehicle and cleaning out the inside. A three inch round stock was welded 6 inches from the rear and 4 inches from the side with Bob Coury’s assistance. We drilled and tapped the stock to make a 0.75” female pipe thread outlet. Finally, we made four 0.75” diameter aluminum spacers with lengths of 0.5” to raise the low-side reservoir up for additional hose clearance.

Pump Sprocket. The sprocket that we ordered that sits on the pump shaft had a hub that was too large. Therefore, we faced the sprocket to a length of 0.5” using the lathe. The bore sized was also increased to 0.875” on the lathe and a 0.25” keyway was added. We also lathed a thin spacer out of spare round stock.

Drive Shaft Sprocket. We also increased the bore size to 1.875” on the drive shaft sprocket using the lathe. We used the mill to create the bolt pattern of 6 0.25” diameter holes distributed evenly on the sprocket at a radius of 1.3125 inches. Once the sprocket was on the bearings on the drive shaft, we also made a lip on each side in order to prevent the sprocket from sliding off the bearings.

Drive Cup. Likewise, we added the exact same bolt pattern to the drive cup with the mill. After that, we also used the mill to counter-sink the holes to keep the bolts flush with the drive cup.

RPM Sensor Bracket. We used the band saw, drill press, and a little bit of welding to cut, drill, and shape the L-shaped RPM sensor bracket.

Drive Shaft. To put all the necessary components on the drive shaft, we first removed it from the vehicle. We added a keyway groove for the clutch using the mill. Next, with Marv Cressey’s help, we smoothed the shaft and added a groove for the retaining clip on the lathe. We slide the drive cup sprocket on the shaft and then pressed the bearings between the shaft and sprocket. After that, we pressed the drive cup on the shaft and bolted it to the sprocket. We pressed on the hydraulic clutch, keyed it to the shaft, and mated it with the drive cup. Finally, we added the support bearing and re-installed the shaft.

Assembly. Initial assembly involved keying the sprocket onto the pump and press-fitting the bearings and plates together. Next the pump was bolted to one plate and the other plate was bolted to the first plate with the spacers defining the distance apart. Note that the pump bolt heads were lathed down to provide extra chain clearance. Once the pump assembly was complete, we installed it on the vehicle using two 0.375” diameter, 8” long bolts to support both the motor and pump.

Plumbing. A major part of the manufacturing process was plumbing in the hosing for the hydraulic system following the hydraulic layout of Figure 10 on page 17. Initially, we had to remove all of the old hosing and hydraulic components. In collaboration with the hose doctor technician, we arranged all of the hydraulic components and determined hosing sizes. The hose doctor fabricated the hoses and connected all the hydraulic components.

Electronics. Using our wiring diagram, we wired up the electronics and valves with the assistance of Professor Epureanu. We also had to drill and bolt on an additional wood board for the mounting of new terminal blocks. Wiring the components involved measuring and cutting appropriate wire lengths, routing the wires to the valves, crimping on the appropriate connectors, and connecting them to the correct terminal blocks. Finally, we wired the other side of the terminal blocks to their respective relays.

Note that other engineering drawings can be found in Appendix C and a Bill of Materials in Appendix E.

8. COMPLETE PROTOTYPE

This section contains pictures that show the various components of the completed prototype.

Figure 15 shows the new outlet installed on the bottom of the low-side reservoir.

Figure 15: New Low-Side Reservoir Outlet

Figure 16 shows the RPM sensor and bracket installed above the motor sprocket.

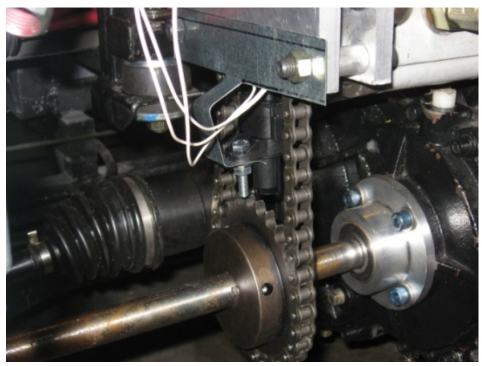


Figure 16: Hall Effect RPM Sensor

Figure 17 shows the wiring of the electronics. Note that the white box contains the base station circuit board, the black boxes are solid state relays, and the component with the dial is the time-delay relay.

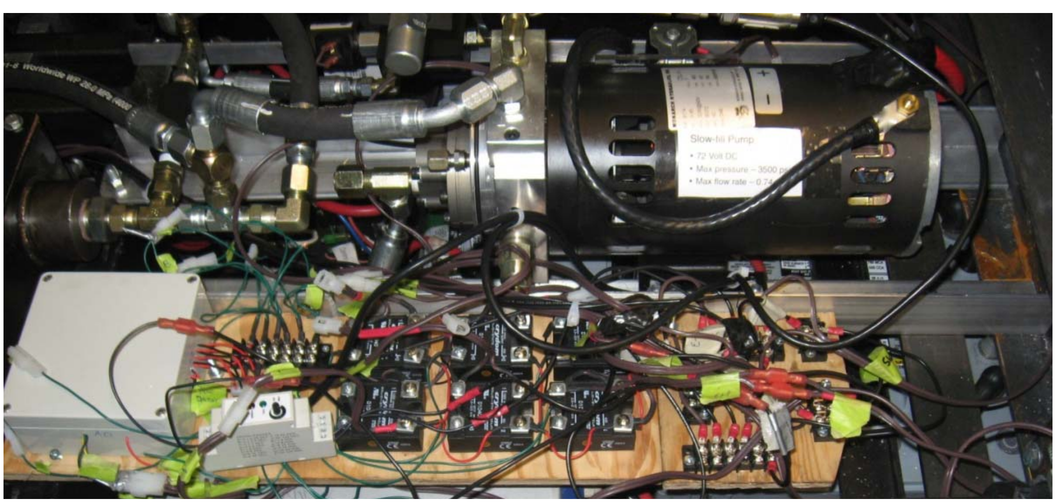


Figure 17: Final Wiring of the Electronics

Figure 18 shows the mechanical layout of the completed prototype.

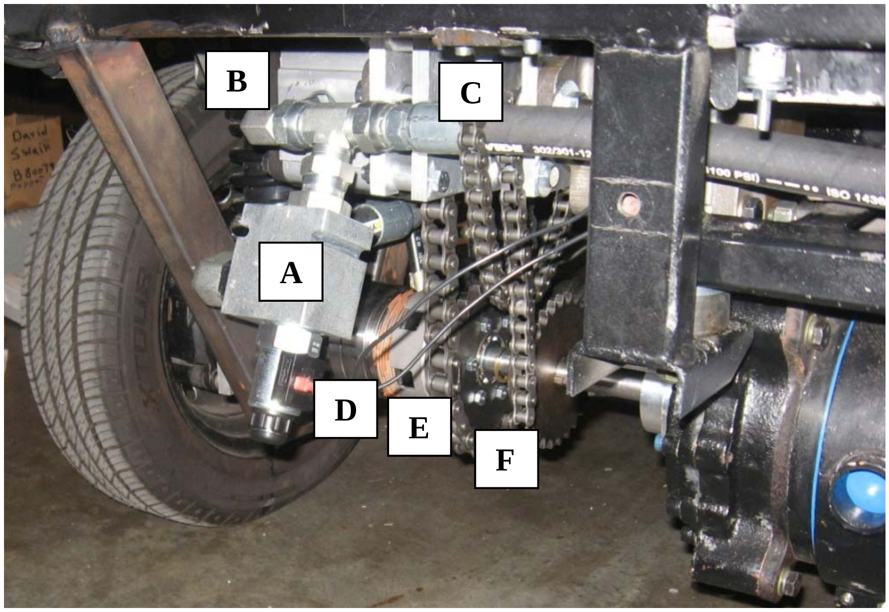


Figure 18: Completed Prototype Mechanical Layout

A) Recirculation Valve
B) Pump
C) Mounting Plates
D) Hydraulic Clutch
E) Drive Cup
F) Drive Sprocket

Figure 19 shows the slow-fill pump and clutch hydraulic layout.

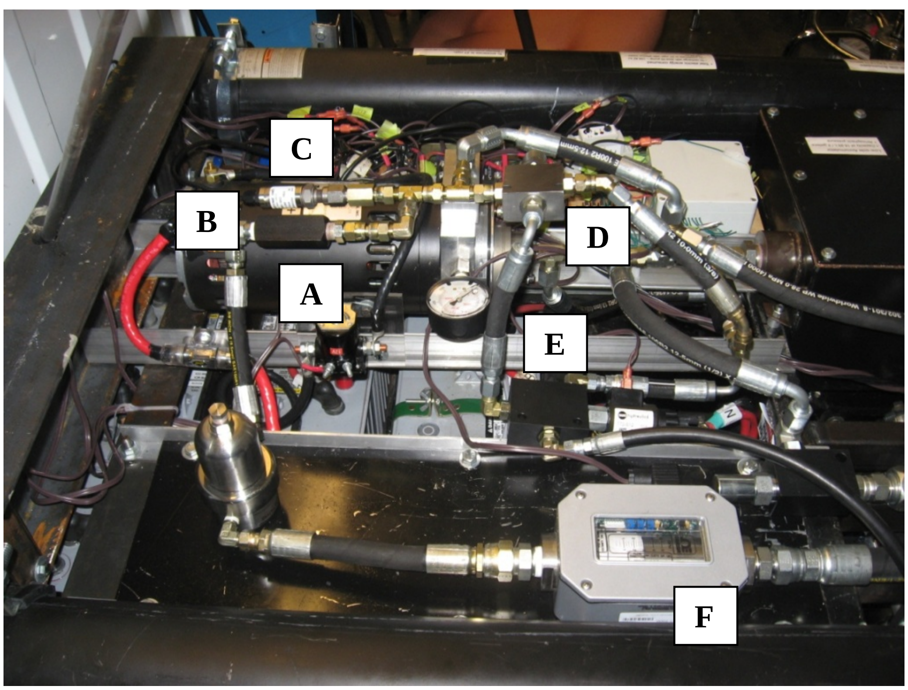


Figure 19: Slow-Fill Pump and Clutch Hydraulic Layout

A) Slow-Fill Pump
B) Check Valve
C) Pressure Transducer
D) Pressure Reducing Valve
E) 2 way, 3 pos Clutch Valve
F) Flow Meter

Figure 20 shows the pump hydraulic layout.

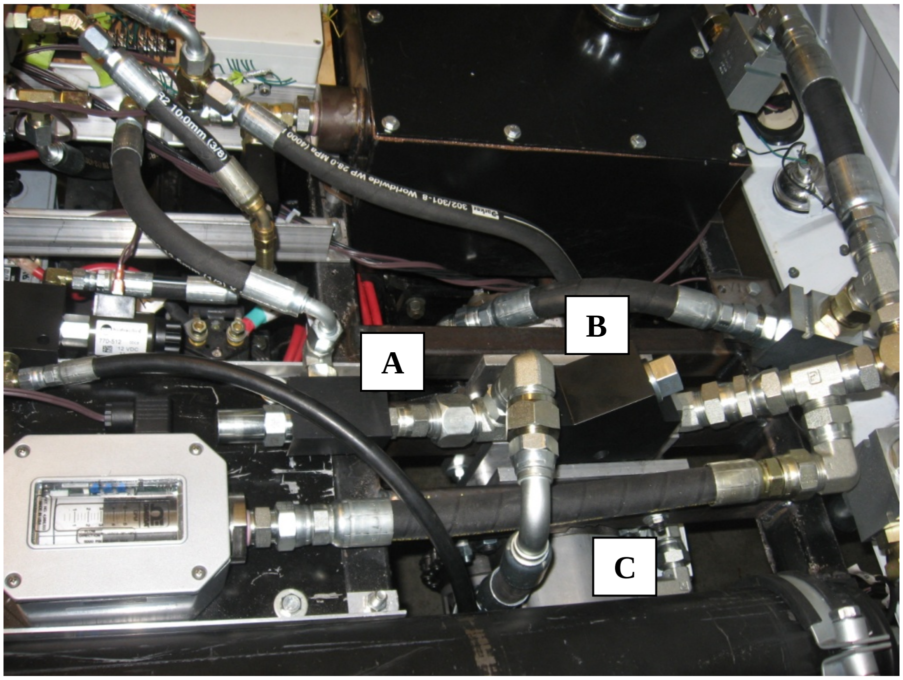


Figure 20: Pump Hydraulic Layout

A) Pressure Relief Valve B) Check Valve C) Pump

Figure 21 shows the high pressure line hydraulic layout.

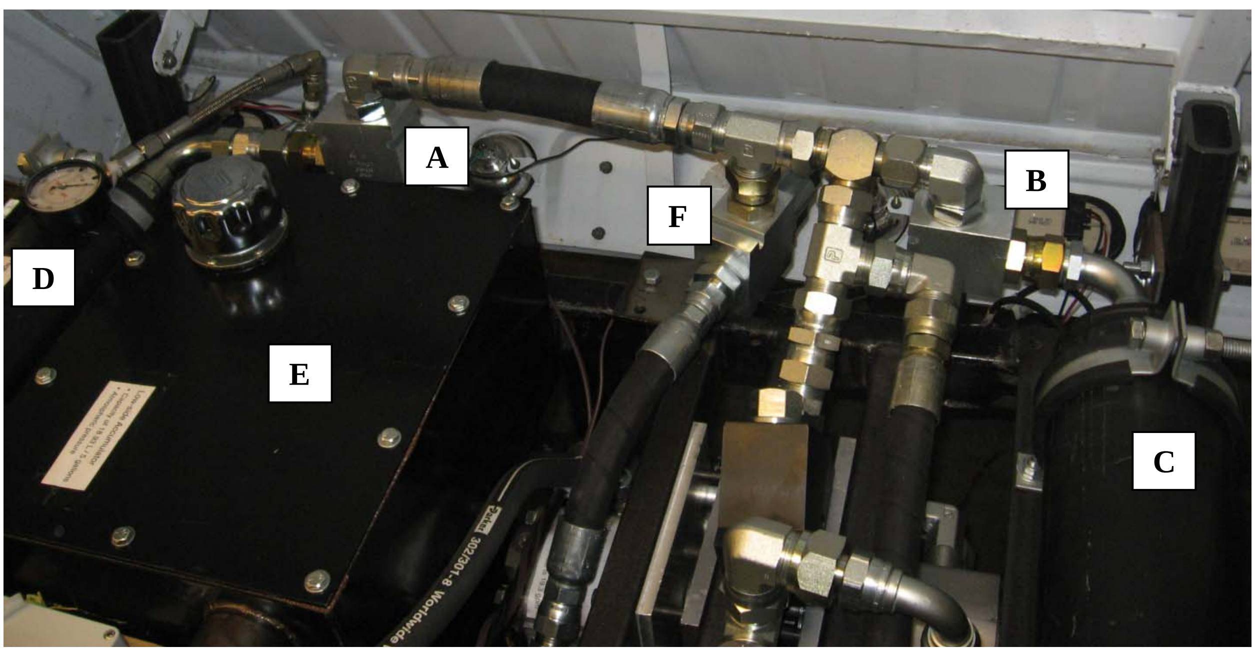


Figure 21: High Pressure Line Hydraulic Layout

A) E-Stop Valve
B) E-Stop Valve
C) High-Side Accumulator
D) High-Side Accumulator
E) Low-Side Reservoir
F) Motor Valve

9. TESTING

To test the performance of our prototype, two tests were scheduled.

9.1 Dynamometer Testing

The testing plan consisted of running the vehicle on the Urban Dynamometer Driving Schedule (UDDS), commonly called the "LA4" or "the city test" [13]. A representation of this cycle can be seen in Appendix F. Since this same test was run on the unaltered electric Xebra vehicle in the Fall 2007 ME450 term, we have a baseline with which to compare our results and quantify the improvements made. Estimations from previous ME450 terms show the possibility of a 40% improvement in efficiency during this city cycle with the implementation of regenerative braking. Due to the lack of availability of a current probe, this test had to be postponed to a later semester.

9.2 Spin Down Test

In our final design meeting with our sponsors at the EPA, they expressed great interest in the damping the hydraulic clutch adds to the system and the associated loss in overall efficiency. Since Logan Clutch Corporation was unable or unwilling to provide us with damping data, we designed a set of experiments in order to quantify the damping. This section discusses our spin down experiments, the associated analysis, and makes recommendations for future teams in order to completely answer this question.

Our idea involved jacking up the car and running two spin down tests: one without and one with the clutch. By recording drive shaft rotational rate versus time data, one should be able to calculate the time constants and damping coefficients for each situation. We used a time-delay relay to switch between forward and neutral in order to have a consistent energy input into the system. Figure 22 presents basic pictures of the systems.

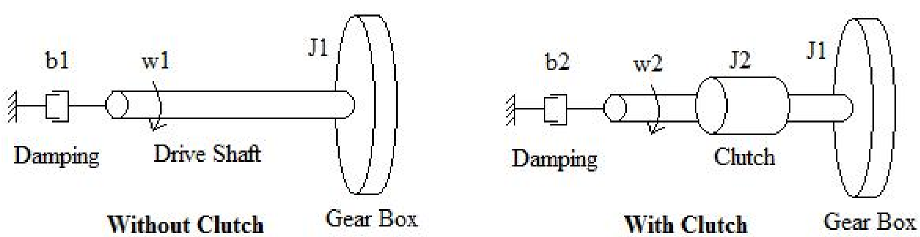


Figure 22: Basic Pictures of Spin Down Test Systems

Analysis of these systems yields two differential equations and two energy equations presented as Equations 19 and 20 respectively. Since the energy inputs were equal for both cases, the energy equations can be set as equals and solved to find the change in inertia.

$$J\dot{\omega} + b\omega = 0 \qquad \text{(Eq. 19)} \qquad\qquad E = J\omega^2 \qquad \text{(Eq. 20)}$$

The damping coefficient is found by finding the solution of the differential equation and the time constant of the solution. For a ramp down situation, the differential equation solution is Equation 21 with Ω_0 as the initial rotational rate and τ as the time constant, which is equal to J/b. Rearrangement of this equation leads to Equation 22.

$$\omega(t) = \Omega_0 e^{-t/\tau} \qquad \text{(Eq. 21)} \qquad\qquad -t/\tau = \ln\left[\frac{\omega(t)}{\Omega_0}\right] \qquad \text{(Eq. 22)}$$

A plot of the right hand side of Equation 22 versus time gives a line with a slope equal to the negative inverse of the time constant, which is a function of the damping coefficient. An illustration of this can be seen in Figure 23. The graph shows that the spin down data is more linear and less exponential than expected from the analysis. These facts lead to an inaccurate time constant of 2.12 ± 0.25 s for the system without a clutch.

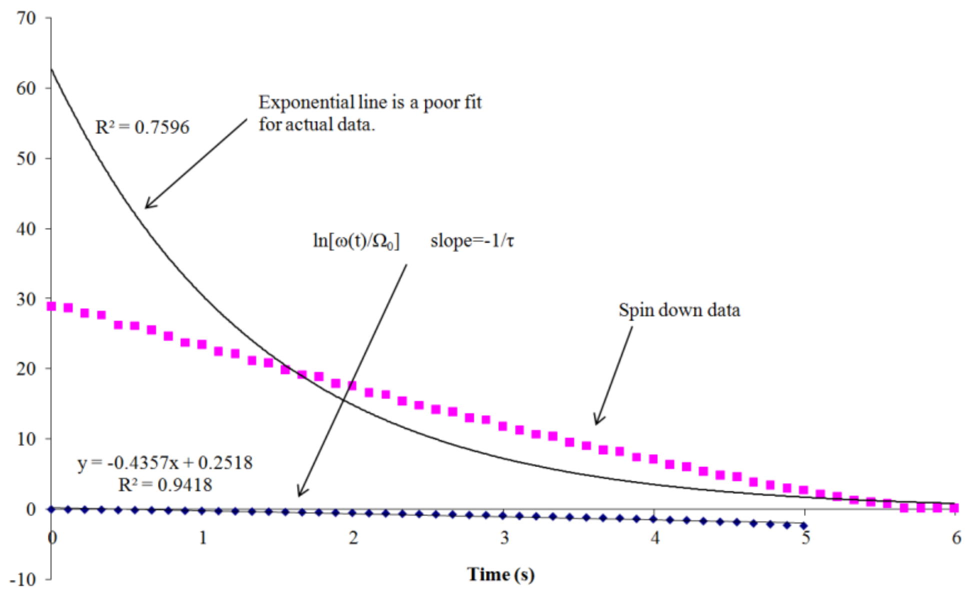


Figure 23: Spin Down Data Analysis Graph

We recommend that future teams use a ramp up test in order to determine the clutch damping. The ramp up data we obtained, like in Figure 24, looked much more like an exponential ramp up function than the spin down or ramp down data. This fact should hopefully yield more accurate time constants and damping coefficients. Similar analysis can be performed to develop the new equations, but the time-delay relay should be set for a longer time to allow the system to reach a steady state rotational speed. Finally, if system damping is so important to the EPA, they should consider de-coupling the electric motor during braking because we suspect that it causes the large damping found in the spin down experiments. The damping of the ramp up and spin down systems could even be compared to confirm and quantify this suspicion.

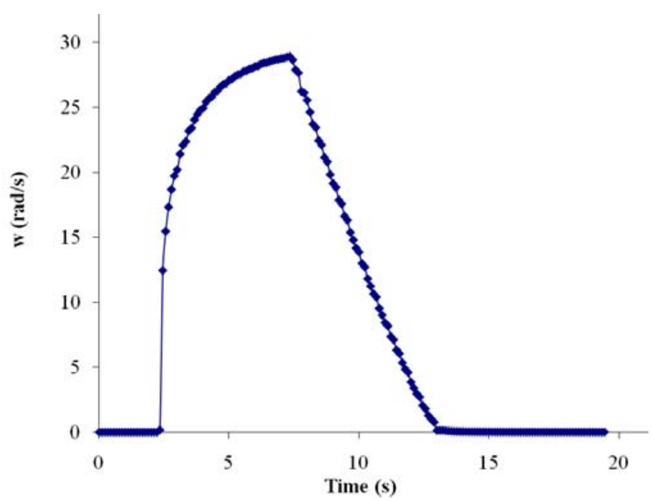


Figure 24: Ramp Up and Spin Down Graph

10. DISCUSSION

While writing this report and discussing the vehicle with people at the Design Expo, we have had a chance to reflect on our final design and the decisions that we made. This reflection has helped us understand the strengths and weaknesses of our design. Furthermore, we have also identified potential changes and improvements. Unfortunately, since we could not test our design, we can only speculate on the need for changes or improvements.

10.1 Design Strengths

Considering our initial knowledge of hydraulics, a strength of our design is the hydraulic system. We were able to spec out, find, purchase, and install seven hydraulic valves in our one semester on the project. These valves allow for the hydraulic system to be completely controlled by the pedals and also ensure system safety. Moreover, these valves, previous components, and the hoses were all fit into the back of the Xebra vehicle with a clean and professional look. The increased hose and valve sizes also reduce pressure losses and increase efficiency.

A second design strength is the mechanical packaging and layout. The vehicle had limited space for our pump, clutch, and sprockets, but with meticulous CAD modeling/planning and precise manufacturing we were able to fit all the components into the rear of the vehicle. When any packaging space issues arose, we quickly adapted our design and manufacturing to solve the problems. Our design uses the rear space of the vehicle in a very efficient manner.

10.2 Design Weakness

Based on a recommendation from Martin Sprocket and Gear, we designed the chain with 4% slack. During installation, we found this decision to be a weakness since it leads to the chain being too loose. It would be good for another team to remedy this problem in order to increase the transmission efficiency between the two sprockets.

In our haste to get ready for the Design Expo, we installed the electronics in a messy and jumbled manner. The valves open at the correct times, but the wire mess is still a weakness since it is confusing and takes away from the overall vehicle appearance. Color coded electrical wires and greater care during installation can easily fix the problem and make the vehicle design easier to understand.

10.3 Potential Changes

We chose to install an e-stop valve on both accumulators. However, technically, the system only needs one valve teed into the high pressure line to prevent high pressure fluid from flowing through the system during an emergency. Using one valve would reduce cost, but the system would lose the option of running on only one accumulator. Two valves are also much safer if the high pressure line is blown.

We chose to actuate the hydraulic clutch with the slow-fill pump as opposed to using a separate small electric feeder pump. It was convenient and easier to use the slow-fill pump since it was already on the vehicle. We expect the energy use to be about the same, but an electric feeder pump that requires less energy, costs less, and reduces the hydraulic system complexity may exist. Finally, a separate pump might also provide a higher flow rate that can actuate the clutch quicker, but since we have not done testing, we do not know if this is even necessary.

We were only able to find spool type pressure reducing/relieving valves. Spool type valves typically have greater leakage rates than poppet type valves. Our valve has a leakage rate of 82 mL/min at 85% of its crack pressure, implying that the maximum power loss due to valve leakage is 37.7 W. Considering the mass and speeds of the vehicle, this power loss is negligible so the valve leakage is really a nonissue. However, if a poppet type pressure reducing/relieving valve could be located, leakage would be virtually eliminated and the EPA would be happier.

A variable displacement motor and pump would provide smoother accelerations and decelerations. However, the Xebra vehicle uses a fixed displacement motor and pump to greatly reduce the complexity of the controls. Furthermore, if this vehicle were ultimately marketed to third world countries, consumers would most likely prefer a less smooth feel of the vehicle in return for the cheaper cost.

Finally, the Xebra vehicle is optimized for the LA4 cycle so it uses a motor for accelerating and a separate pump for braking. A viable option involves using a single component with a valve that switches the inlet and outlet so that it can act as both a motor and pump. This option would save space and reduce plumbing losses, but the valve controls would be complex. In city driving, people switch between accelerating and braking often so if the switching valve or component did not switch or locked, it could be dangerous.

11. RECOMENDATIONS

11.1 Necessary Next Steps

Unfortunately, unforeseen and uncontrollable circumstances arose such that we were unable to get the regenerative braking system working. Moving forward with the project, several key items must be implemented to have a fully functioning prototype. Note that future teams can obtain a list of key contacts from David Swain to answer questions on these tasks. The next steps are as follows:

- **Chain Slack:** The slack in the chain should be reduced to increase transmission efficiency between the sprockets. This can be achieved by adding an idler pulley or by moving the pump mounting plates. Note that moving the pump could change some hosing lengths that are already fabricated and installed.
- **Clutch:** A retaining clip should be added to prevent it from translating on the shaft and the drive cup should be aligned with the clutch body. Moreover, a specific clearance distance should be maintained between the clutch and drive cup in accordance with Appendix G.
- **Anti-Rotation Bracket:** An anti-rotation bracket needs to be fabricated and installed to prevent drag from the clutch rotating about the shaft. This can be as simple as a piece of sheet metal around the clutch's actuation fitting that limits rotation as seen in the final mechanical design layout CAD model (Figure 9 on page 16). See Appendix H for more information.
- **RPM Sensor:** The RPM sensor and its bracket must be re-installed above the motor sprocket. There should be minimal distance between the sensor and sprocket teeth.
- **Tachometer:** The new tachometer needs to be installed and wired into the electronics and base station. Installing it so that the driver can read the tachometer's display would be ideal. Moreover, the tachometer will have to be calibrated and programmed according to its manual. If done correctly, the tachometer should be able to cut power to the hydraulic system if the pump's rotation rate rises above 4500 rpm, increasing safety.
- **E-stop Button/Buzzer:** To further increase the vehicle's safety, an e-stop button should be installed so that system power can be cut during emergencies. For safety, the e-stop button should probably be placed inside the cab for easy access. Finally, a buzzer could be added to signify when the e-stop button is tripped.

- **Clutch Drag:** Since drag and efficiency are of great concern to the EPA, the testing to quantify the added damping due to the clutch should be completed. See the testing section or consult a professor knowledgeable about dynamics in order to determine the best approach.
- **Computer Code:** It would be a good idea to modify the computer code to eliminate the power surge on vehicle startup with the computer. Furthermore, the newly installed RPM sensor and pressure transducer should be calibrated and incorporated into the software so that the user can read these values real time. Scott Hotz and Thomas Naylor of Southwest Research Institute would be reasonable people with whom to collaborate.
- **Recirculation:** Some calculations or experiments should be completed in order to determine how long to set the time-delay relay to re-circulate fluid around the pump while the clutch is engaging. This could involve a trial and error process. Ultimately, we expect the recirculation time to be a few milliseconds or less.
- **Electronics:** Admittedly, the electronics were done in haste and are somewhat jumbled. Therefore, cleaning up the electronics and color coding the wires would improve the transferability to future semesters. A single plug to connect all electronics would be ideal.
- **Oil:** All fittings must be tightened, the hydraulic fluid must be added, and all air must be evacuated from the system. To remove the air, the fittings around major components and valves must be loosened with the fluid flowing to "bleed" out the air with a little oil.
- **Coast-Down Test:** A new coast-down test should be performed to recalculate the new road load coefficients. These coefficients are used in the program that runs the LA4 dynamometer to achieve a better simulation of the Xebra driving on a road. The Fall 2007 team conducted the original coast-down test with the help of Larry Webster from Car and Driver Magazine.
- **LA4 Testing:** Once everything is installed, debugging is complete, and the vehicle is fully functioning, the improvement from the hydraulic system should be quantified by running the vehicle on the LA4 cycle. The team from Fall 2007 completed the original baseline testing so their procedure should be repeated. The contact for this is Pat Barker of Lotus Engineering.

11.2 Future Suggestions

In past semesters, the electronics and software were simple enough for the teams to easily handle. However, the vehicle has reached the point where the electronics and software are beyond what one can reasonably expect senior mechanical engineering students to successfully work with in less than a few months, especially considering the project's other demands and expectations. Therefore, we recommend that a multi-disciplinary team be formed that includes EECS and CSE students to collaborate with on the electronics and software. Besides this recommendation, we have several ideas that could be the focus of future ME 450 teams depending on the EPA's needs and wants. Note that the following ideas can be combined or modified to achieve a project with suitable scope for ME 450.

- One team could work on the controls of the vehicle. This team could not only develop controls for the hydraulic functions, but also integrate them with the electric drive motor. This could increase the system's overall efficiency and improve vehicle performance.
- Another option involves a team dissecting the current vehicle and decisions made by past teams in order to identify ways to optimize the vehicle and its efficiency.
- Likewise, a team can analyze the vehicle in terms of manufacturability. Considering the ultimate goal is to mass produce the vehicle, they can look for ways to simplify the system, reduce the number of parts/components, and cut costs.

- A text fixture could be designed and developed to test and quantify the efficiency of the pump and motor over the vehicle's full operating range. This information would be very valuable in the development of more accurate computer simulations and models.
- With the current vehicle design, the reverse feature will not work properly. When the accelerator pedal is actuated in reverse mode, the electric motor spins backwards while the hydraulic motor spins forwards. The system can be set up so the hydraulic system is shut off in reverse mode or the hydraulic system can be used to help drive the vehicle in reverse mode.
- Currently the slow-fill pump and several hydraulic components sit above the batteries, causing a large time investment in order to service or replace the batteries. One task involves designing a system or device to quickly move the slow-fill pump and other components for easy battery access. This could involve a hinged platform or sliding tray.
- In order to keep the truck bed raised, a metal doll rod is used as a prop. This is unsafe and the rod is very easy to lose. If another team could improve the truck bed propping design, accessing the hydraulic components would be easier and the vehicle would look nicer when on display.
- Finally, one last idea involves designing and installing a pair of permanent hydraulic jacks that can quickly and safely prop up the back wheels for spinning tests and displays.

12. CONCLUSIONS

The Environmental Protection Agency along with the University of Michigan is converting the Xebra Electric Vehicle to a Hydraulic-Electric Hybrid. The vehicle has been worked on in three previous ME450 semesters and our team worked on the regenerative braking system. It was our responsibility to integrate a hydraulic pump that pressurizes the high pressure accumulators during braking to recapture wasted braking energy. The goal is to minimize the battery usage during acceleration by storing the energy from braking and then use that energy to accelerate the vehicle. We completed much work while striving to reach our goal and complete our part of the project.

We consulted with David Swain of the EPA to develop the customer requirements and engineering specifications. Using these requirements and specifications, we generated concepts and selected the best ones as our Alpha design. We completed several calculations to select components and design the system. This process lead to a final mechanical, hydraulic, and electrical design. To create a prototype of this design, we used numerous manufacturing and assembling processes. Some tests are complete, but several more are required to completely validate the prototype. The Design Expo gave us a chance to analyze the vehicle's strengths and weakness and identify potential changes. Due to unforeseen and uncontrollable circumstances, we were not able to get the regenerative braking system working. Therefore, there are several steps for a future team to complete to achieve a fully functioning prototype. After that, the Xebra vehicle has the potential to offer many other challenging and rewarding ME 450 projects.

13. ACKNOWLEDGEMENTS

Team X-Gen would like to thank those who went above and beyond what was necessary in order to help the team. 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.

Dr. Andrew Moskalik, Environmental Protection Agency, thank you for being a source that we can learn from and someone we can turn to with any questions.

Scott Hotz and Thomas Naylor, Southwest Research Institute, thank you for all your help, guidance, and especially your patience with the electronic aspects of this project.

Thank you, Michael Woon for giving us practical ideas throughout the project.

We would like to thank Aidan Feldman for his help and support with the software and computer controls.

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Professor Bogdan Epureanu, 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 thank you to our sponsor, David Swain of the Environmental Protection Agency, for being a constant resource for any questions that arose and any problems that the team encountered. You have made all of us on Team X-Gen more knowledgeable of hydraulics and have been an outstanding sponsor. Thank you very much.

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APPENDIX A - QFD

Technical Specifications / Customer Requirements	**Weightage**	Max Flow-rate of pump	Accumulator Pressure	Pump Size	Valve Sizes	Braking Types	Corrosion Resistant	Losses due to heat/friction of brakes	Clutch Improvements	Location of pump	Location of valves	Flow rate of fluid into high accumulator	Location of hinge and platform	**Total (weighted CR)**	**Normalized to Total**	**Importance Rating**
Transferrable to Future Semesters	10	3	3	3	3	9	3	3	9	9	9	3	9	660	.15	15
Comfortable feeling during braking	2	3	9	9	9	9	1	3	3	1	1	3	1	106	.02	2
Sufficient braking until stop	8	3	9	3	3	9	1	3	1	1	1	9	1	352	.08	8
Easy to service	8	1	1	1	1	3	9	3	3	9	9	1	9	400	.09	9
Hinged platform for access to battery	9	1	1	3	3	1	3	1	3	9	9	1	9	396	.09	9
Maintains vehicle function	9	9	9	9	9	3	1	9	3	3	3	9	3	630	.14	14
Easy to use	7	1	1	1	1	9	1	1	3	1	1	3	1	168	.04	4
Variable Braking	2	9	9	3	3	9	1	9	3	1	1	9	1	116	.02	2
Aesthetics	5	1	1	3	3	1	9	1	3	9	9	1	9	250	.05	5
Safety	10	9	9	3	9	9	3	3	3	3	3	9	1	740	.16	16
Reliability	9	9	9	9	9	9	1	3	9	1	1	9	3	729	.16	16
Measurement Unit		L/s	kPa	cc	NA	sec	%	W	NA	NA	NA	L/s	NA			
Our target		1.5	3800	33	NA	12	100	0	NA	NA	NA	1.5				
Total (weighted technical specifications)		359	419	327	387	497	241	324	335	373	373	421	371			
Normalized to Total		0.08	0.09	0.07	0.09	0.11	0.05	0.07	0.08	0.08	0.08	0.1	0.08			
Importance Rating		8	9	7	9	11	5	7	8	8	8	1	8			

APPENDIX B – ENGINEERING CHANGES NOTICE (ECN)

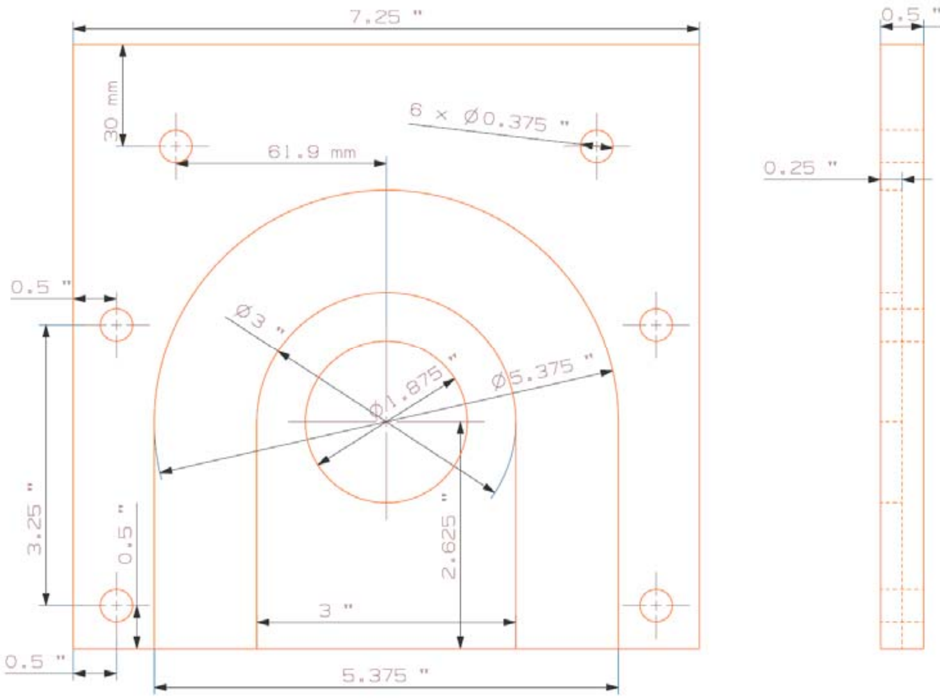


Figure 25: Original Bearing Mounting Plate

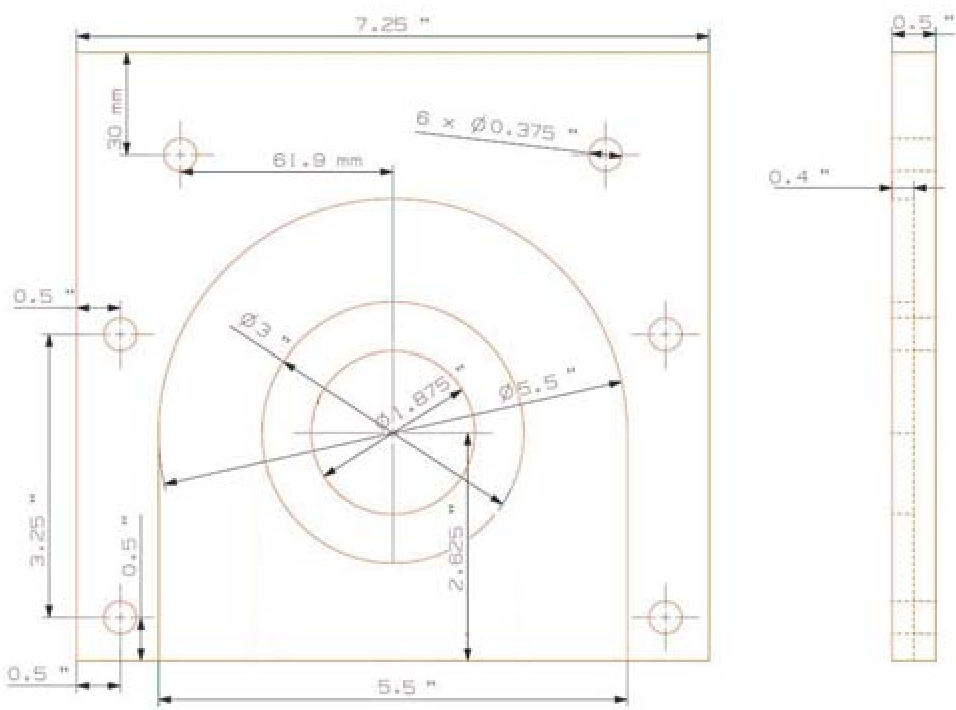


Figure 26: Modified Bearing Mounting Plate

Changes made to provide more clearance for the chain.

APPENDIX C – ENGINEERING DRAWINGS

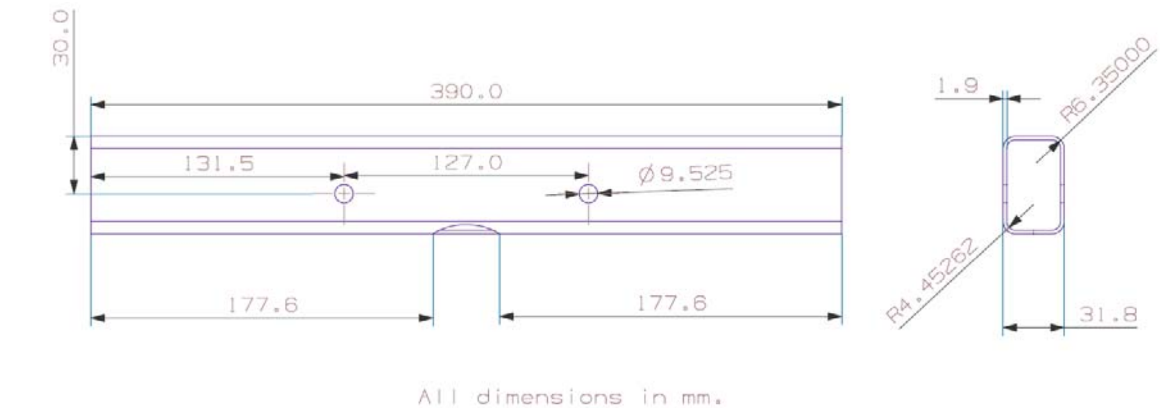


Figure 27: Cross Member

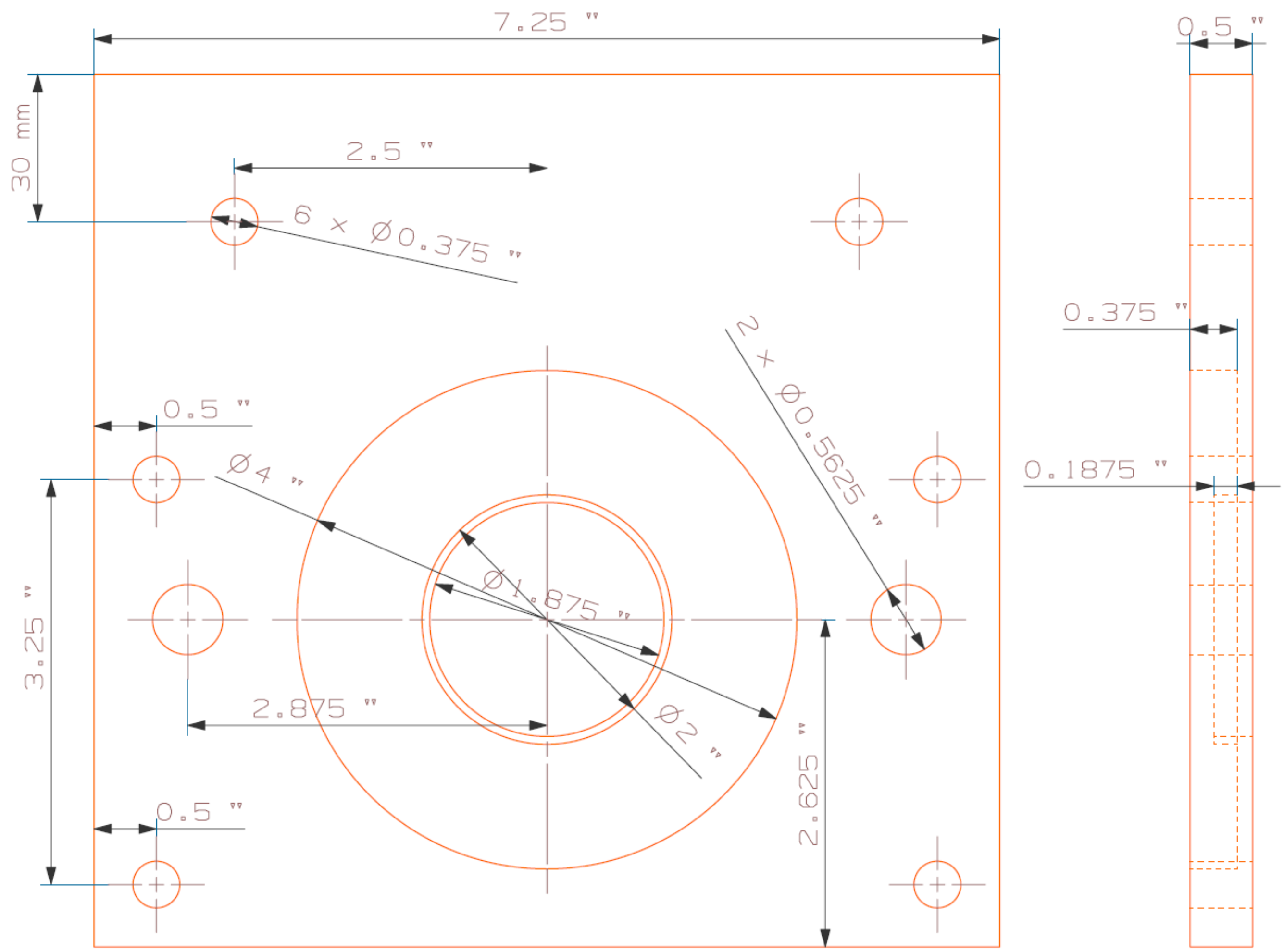


Figure 28: Hydraulic Pump Mounting Plate

APPENDIX D – FAILURE MODE AND EFFECT ANALYSIS

Identify Hazards | Assess and Reduce Risk

	Item Id	User	Task	Hazard Category	Hazard
1	1-1-1	All Users	All Tasks	mechanical	fatigue

	Item Id	User	Task	Hazard Category	Hazard
1	1-1-1	All Users	All Tasks	mechanical	fatigue
2	1-1-2	All Users	All Tasks	mechanical	machine instability
3	1-1-3	All Users	All Tasks	electrical / electronic	energized equipment / live parts
4	1-1-4	All Users	All Tasks	electrical / electronic	water / wet locations
5	1-1-5	All Users	All Tasks	slips / trips / falls	slip
6	1-1-6	All Users	All Tasks	slips / trips / falls	trip
7	1-1-7	All Users	All Tasks	ergonomics / human factors	repetition
8	1-1-8	All Users	All Tasks	ergonomics / human factors	duration
9	1-1-9	All Users	All Tasks	ergonomics / human factors	human errors / behaviors
10	1-1-10	All Users	All Tasks	fire and explosions	hot surfaces
11	1-1-11	All Users	All Tasks	fire and explosions	flammable liquid / vapor
12	1-1-12	All Users	All Tasks	material handling	motor vehicle movement

Cause/Failure Mode
Components may wear with time.

Cause/Failure Mode	Severity	Exposure	Probability
Components may wear with time.	Slight	Remote	Possible
Some components will vibrate during operation.	Serious	Frequent	Probable
Live electronic wiring throughout the system.	Serious	Remote	Unlikely
Greased up components, possible oil leakage between hoses.	Slight	Remote	Possible
Possible oil leakage on surfaces of components.	Slight	Occasional	Possible
Vehicle drives on even paths.	Minimal	Frequent	Negligible
Vehicle is constantly accelerating and braking.	Minimal	Frequent	Negligible
Vehicle as a limited duration.	Slight	Frequent	Negligible
Humans operate vehicle, so thier actions influence vehicle operation.	Minimal	Frequent	Negligible
Vehicle batteries are hot.	Minimal	Occasional	Negligible
Hydraulic oil is flammable.	Catastrophic	Remote	Negligible
Vehicle is constantly moving and stopping. Motor is constantly working.	Minimal	Frequent	Negligible

	Item Id	User	Task	Hazard Category	Hazard
1	1-1-1	All Users	All Tasks	mechanical	fatigue

	Risk Level	Reduce Risk	Severity	Exposure	Probability	Risk Level
1	Moderate	monitor components	Slight	Remote	Possible	Moderate
2	High	special tools or fixtures	Serious	Remote	Possible	Moderate
3	Moderate	monitor stability of wiring	Serious	None	Unlikely	Low
4	Moderate	contact strip	Minimal	None	Negligible	Low
5	Moderate	safety mats / contact strip	Minimal	Remote	Negligible	Low
6	Low	light acceleration/braking on uneven surfaces	Minimal	Frequent	Negligible	Low
7	Low	monitor components	Minimal	Frequent	Negligible	Low
8	Low	monitor components	Minimal	Frequent	Negligible	Low
9	Low	standard procedures, E-stop control	Minimal	Occasional	Negligible	Low
10	Low	monitor batteries	Minimal	Occasional	Negligible	Low
11	Moderate	safety mats / contact strip to absorb oil	Catastrophic	None	Negligible	Low
12	Low	standard procedures	Minimal	Frequent	Negligible	Low

APPENDIX E – BILL OF MATERIALS

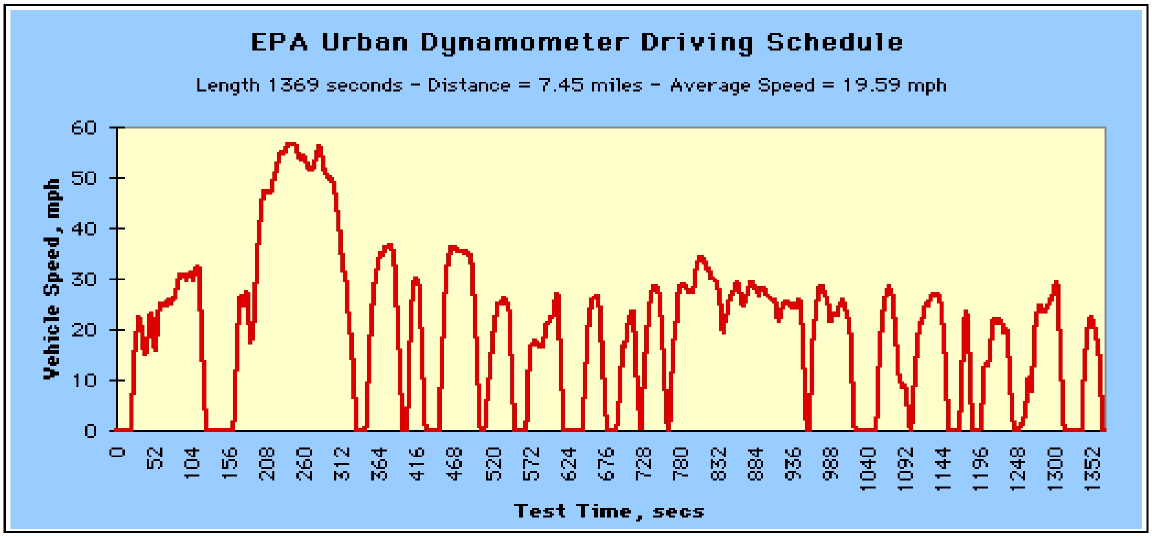
Items Purchased this Semester

#	Description	Supplier	Manufacturer	Part Number	Price
2	E-Stop Valves	Morrell Inc.	Hydac	WS16ZR-01-M-SS16-N-12-DS	$353.20
1	Recirculation Valve	Morrell Inc.	Hydac	WS16YR-01-M-SS16-N-12-DS	$176.60
1	Motor Valve	Morrell Inc.	Hydac	WS16ZR-01-M-SS16-N-12-DS	$170.01
1	Wireless Antenna	Newegg.com	Hawking Technology	HAI15SC	$38.94
1	Pressure Reducer	Psi Hydraulics	Command Controls Corp.	PRRS-08-N-T-06TS-15	$221.50
1	3-Way Valve	RHM Fluid Power	Sun Hydraulics	DWDA-MAN512-ECI/S	$209.18
1	Pump Relief Valve	RHM Fluid Power	Sun Hydraulics	RPGS-CWN-CAK/S	$148.06
1	Hydraulic Clutch	Logan Clutch Corp	Logan Clutch Corp	P35-0003 Industrial Clutch	$1,085.00
1	Drive Cup	Logan Clutch Corp	Logan Clutch Corp	016-0018	$163.55
1	Shaft Sprocket	McMaster-Carr	Martin	6793K196	$29.44
1	Pump Sprocket	McMaster-Carr	Martin	6793K194	$25.28
4	Open Ball Bearings	McMaster-Carr	Koyo	60355K18	$30.28
1	Time Delay Relay	Newark	Amperite	29K8891	$99.69
2	Mounting Plates	Alro Metals Plus	N/A	7" x 7.25" x 0.5" Al	$31.10
1	Bar for Spacers	Alro Metals Plus	N/A	0.75" Rd, 12" long Al	$6.22
1	Crossbar	Alro Metals Plus	N/A	2" x 1.25" x 36", 14 Ga	$14.90
1	Chain Breaker	McMaster-Carr	?	6051K15	$21.60
1	ANSI 60 Chain	McMaster-Carr	?	6261K473	$15.81
1	Sealed Ball Bearing	McMaster-Carr	Koyo	60355K39	$10.43
50'	14 Awg Elec. Wire	McMaster-Carr	?	7587K975	$14.80
1	Digital Tachometer	Newport	Newport	P6430A	$644.00
1	Pic Programmer	Digi-Key	Microchip Technology	PG164120-ND	$34.99
1	28-Pin Board Pic	Digi-Key	Microchip Technology	DM164120-3-ND	$24.99
2	Mounting Bolts	McMaster-Carr	?	91251A117	$8.46
6	Drive Cup Bolts	McMaster-Carr	?	91263A566	$7.06
2	Pump Bolts	McMaster-Carr	?	91251A698	$6.00
1	Lowside Tap Stock	Alro Metals Plus	N/A	2.5" Rd, 3" long	$9.96
8	Terminal Blocks	McMaster-Carr	?	7527K4*	$10.68
8	Terminal Covers	McMaster-Carr	?	Various	$15.36
25	Terminal Jumpers	McMaster-Carr	?	7527K59	$2.15
	Electrical Connectors	Ace Hardware	?	Various	$32.89
	Expo Tow Truck	Sakstrup's Towing	N/A	N/A	$209.00
	Hydraulic Fittings	Exotic Automation	Parker	Various	?
	Hose Installation Labor	Exotic Automation	N/A	N/A	?
				Total:	**$3,871.13**

Items Acquired from Sponsors or Previous Semesters

#	Description	Supplier	Manufacturer	Part Number
1	Pump	Exotic Automation	Parker	PGM517MA 0330 BM1H3ND6 D6B1B1B1
1	Pump Check Valve	RHM Fluid Power	Sun Hydraulics	CXHA-XAN-ICM/S
1	Clutch Check Valve	Exotic Automation	Parker	C600S
1	RPM Sensor	EPA/SWRI	BMW	
1	Pressure Transducer	Omega	Omega	PX309-5KG5V
	Bolts, Nuts, Washers	Machine Shop	N/A	Various sizes and lengths
	Hydraulic Fittings	EPA	Tompkins Ind.	Various sizes and fittings

APPENDIX F – EPA CITY CYCLE

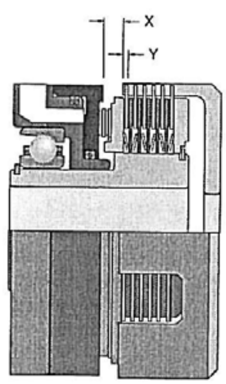


"The EPA Urban Dynamometer Driving Schedule (UDDS) is commonly called the 'LA4' or 'the city test' and represents city driving conditions. It is used for light duty vehicle testing [13].'

APPENDIX G – CLUTCH INSTALLATIN SPACING

Clutch Spacing and Installation

FIGURE 4.



Refer to Figure 4.

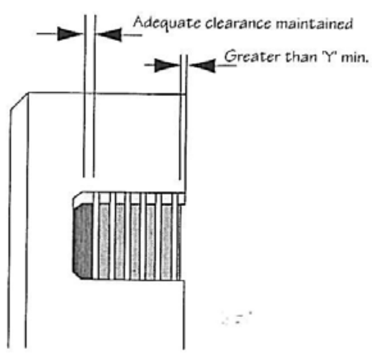
Clutch	S250	S300	S350	S400	S450
X Minimum	0.200	0.215	0.246	0.249	0.262
Y Minimum	0.030	0.030	0.058	0.073	0.087

These are general specifications. Your application may vary. In most applications there is allowance made for axial spacing adjustment of the drive cup, clutch, or both with either fitting spacers or locknuts.

It is important that "X" spacing and "Y" spacing be maintained according to the chart above. If "X" is less than specified, when the clutch wears, the cylinder will hit the drive cup and spin, causing lube and actuation lines to rip out. This could occur months or even years after installation. If "Y" is not maintained the last disc could come out of the drive cup and keep the clutch from disengaging. Please note: as the clutch wears "Y" will increase when engaged.

FIGURE 5.

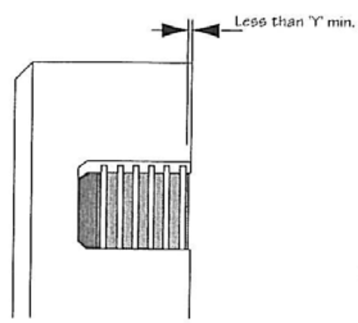
Good.



5 - A

Generally, a clutch and drive cup combination is designed so that the disc pack is centered in the drive cup slot. However, sometimes due to spacing or other considerations, central location of the disc pack is not possible. Acceptable spacing is when the 'Y' minimum is maintained and the and the clutch does not "bottom out" in the drive cup.

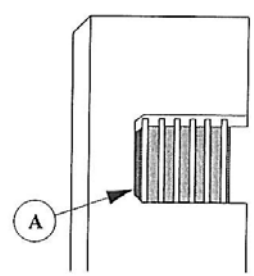
Not Acceptable.
Too close to the edge.



5 - B

An unacceptable installation is when the 'Y' minimum is not maintained. The last disc may come out of the drive cup and keep the clutch from disengaging.

Not Acceptable.
Bottomed out at the drive cup slot.



5 - C

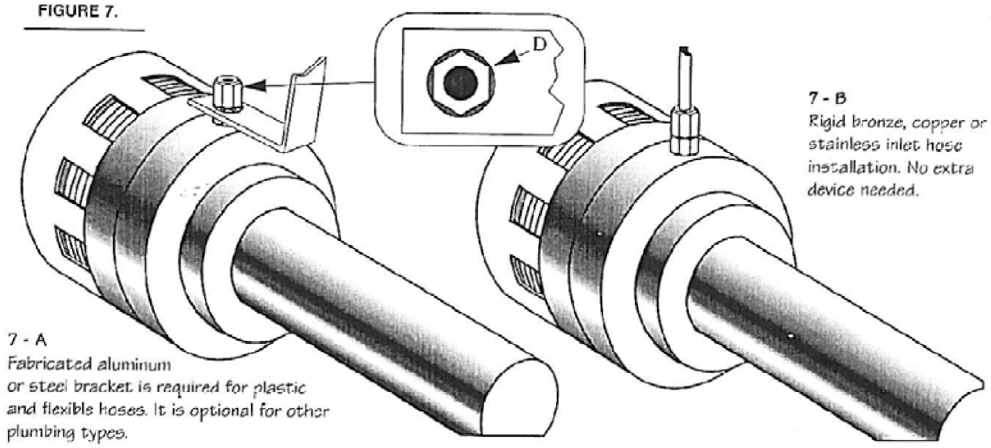
Another unacceptable installation occurs when the clutch is so deep in the drive cup that the deepest disc bears against the bottom or radius of the drive cup slot (space 'A' above).

It is possible, due to differences in manufacturers and aftermarket designs of drive cups, that in rare instances the new clutch can not be installed while maintaining both "Y" minimum and adequate clearance from the bottom of the drive cup slot. In this instance, contact Logan Clutch Corporation.

APPENDIX H – ANTI-ROTATION BRACKET

"S" Clutch Anti-Rotation Bracket

FIGURE 7.



All Logan "S" Series Clutches have free spinning cylinders to allow for fixed lubrication and activation input lines. A cylinder anti-rotation device must be provided so that drag will not cause the clutch cylinder to spin. Figures 7-A and 7-B show suitable designs for anti-rotation devices. Make sure that the anti-rotation device does not place any load on the clutch cylinder (Point 'D').

 3

APPENDIX I - PRELIMINARY CALCULATIONS

Acceleration Energy

How much energy in Joules would it require to bring the vehicle up to 27mph (12.1m/s) from rest? For our preliminary calculations, vehicle weight is taken as 1800lbs with a 500lb payload (1043kg total). Using the kinetic energy equation, we can determine the energy required.

$$E = \frac{1}{2}mv^2 \quad \text{(Eq. 23)}$$

Substituting, Equation 23 yields E = 76.4KJ.

Hydraulic fluid volume

How much fluid will you theoretically need at 3000psi (20.7MPa) to get to 27mph (12.1 m/s) from rest?

The energy contained in the hydraulic fluid must be equal to the kinetic energy calculated above. From Thermodynamics, we have Equation 24.

$$E = P * dv \quad \text{(Eq. 24)}$$

Solving for Volume, we determine that the required amount is 3.69liters (0.00369 m^3).

Acceleration Calculation

What kind of acceleration would you get with a 23cc fixed displacement hydraulic motor assuming a 3000psi supply, a 12:1 motor-to-wheel gear reduction, and a 20 inch wheel diameter?

We first calculate the energy in each motor revolution using Equation 25 below and taking the ambient pressure as 101kPa.

$$E_r = V * dP \quad \text{(Eq. 25)}$$

Eq. 3 yields 473.42J for each revolution. We solve for motor acceleration using the energy equation and Newton's 2nd Law in Equations 26 and 27 below.

$$E = F * s \quad \text{(Eq. 26)}$$

$$F = m * a \quad \text{(Eq. 27)}$$

Solving for acceleration, we arrive at Eq. 28.

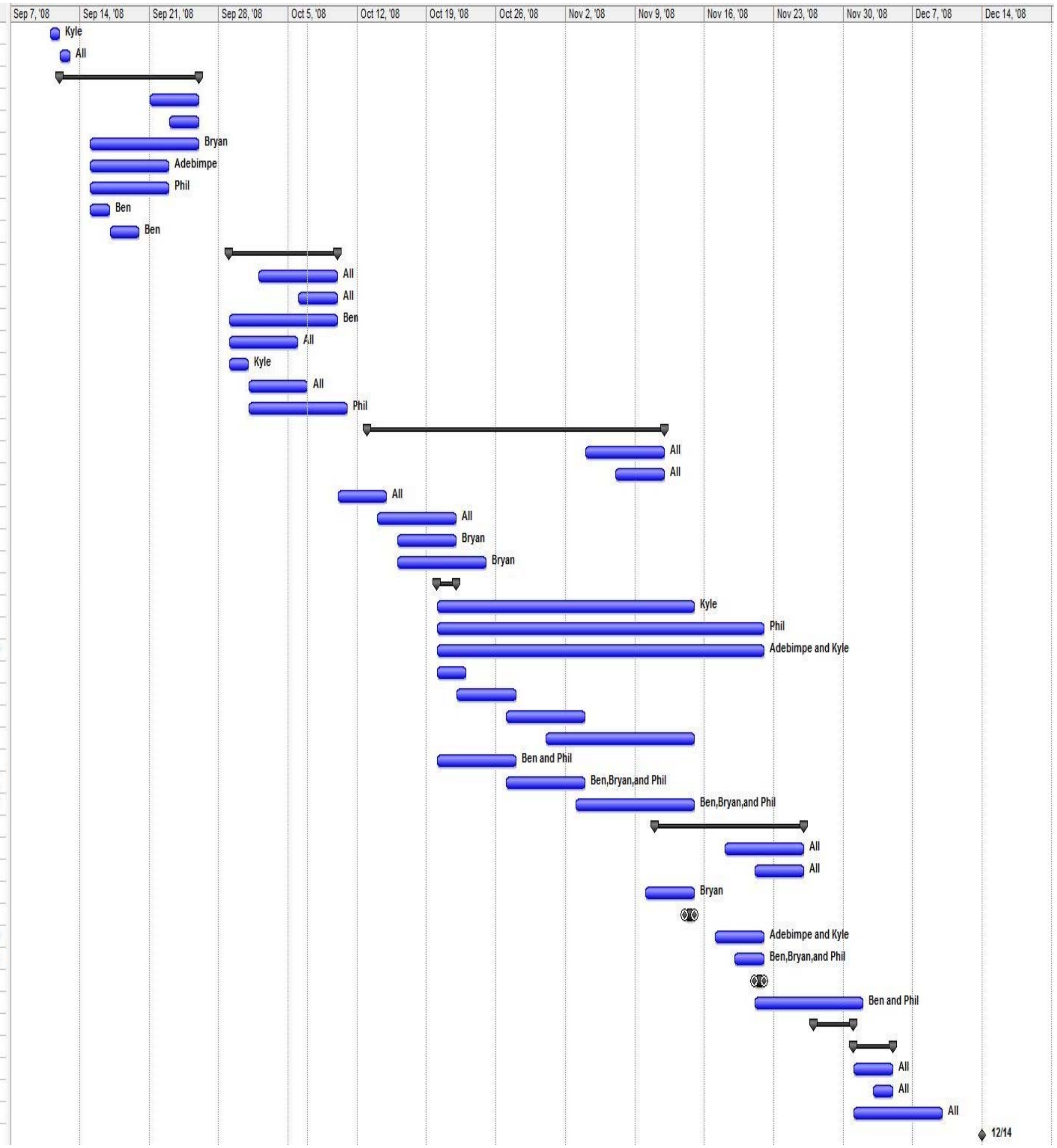
$$a = \frac{E}{m * s} \quad \text{(Eq. 28)}$$

$$s = \frac{\pi D}{12} \quad \text{(Eq. 29)}$$

Substituting for wheel diameter, and plugging into Eq. 29, we find the vehicle acceleration to be 3.41m/s^2.

APPENDIX J - GANTT CHART

	Task Name	Start	Finish	Resource Names
1	Initial Calculations	Thu 9/11/08	Thu 9/11/08	Kyle
2	Meet with Sponsor	Fri 9/12/08	Fri 9/12/08	All
3	**Design Review #1**	**Fri 9/12/08**	**Thu 9/25/08**	
4	Written Report	Sun 9/21/08	Thu 9/25/08	
5	Oral Presentation	Tue 9/23/08	Thu 9/25/08	
6	Brochure	Mon 9/15/08	Thu 9/25/08	Bryan
7	Preliminary Research	Mon 9/15/08	Mon 9/22/08	Adebimpe
8	Customer Spec/Eng. Req	Mon 9/15/08	Mon 9/22/08	Phil
9	Plumbing Losses	Mon 9/15/08	Tue 9/16/08	Ben
10	Deceleration Calculations	Wed 9/17/08	Fri 9/19/08	Ben
11	**Design Review #2**	**Mon 9/29/08**	**Thu 10/9/08**	
12	Written Report	Thu 10/2/08	Thu 10/9/08	All
13	Oral Presentation	Mon 10/6/08	Thu 10/9/08	All
14	E-Stop Valve Selection	Mon 9/29/08	Thu 10/9/08	Ben
15	Concept Generation	Mon 9/29/08	Sun 10/5/08	All
16	Shaft Torque Calculations	Mon 9/29/08	Tue 9/30/08	Kyle
17	Cavitation Calculations	Wed 10/1/08	Mon 10/6/08	All
18	Arrange Lotus Testing	Wed 10/1/08	Fri 10/10/08	Phil
19	**Design Review #3**	**Mon 10/13/08**	**Tue 11/11/08**	
20	Written Report	Tue 11/4/08	Tue 11/11/08	All
21	Oral Presentation	Fri 11/7/08	Tue 11/11/08	All
22	Finalize Design	Fri 10/10/08	Tue 10/14/08	All
23	Select Clutch, Gears, Bearings	Tue 10/14/08	Tue 10/21/08	All
24	Create Bill of Materials	Thu 10/16/08	Tue 10/21/08	Bryan
25	Order Parts	Thu 10/16/08	Fri 10/24/08	Bryan
26	**Fall Break**	**Mon 10/20/08**	**Tue 10/21/08**	
27	Arrange Hose Doctor	Mon 10/20/08	Fri 11/14/08	Kyle
28	Arrange Tow Truck	Mon 10/20/08	Fri 11/21/08	Phil
29	Electronics, Controls, Software	Mon 10/20/08	Fri 11/21/08	Adebimpe and Kyle
30	Stop Power On Surge	Mon 10/20/08	Wed 10/22/08	
31	Recalibrate Flow Sensors	Wed 10/22/08	Mon 10/27/08	
32	Add Speed Sensor	Mon 10/27/08	Mon 11/3/08	
33	Regen Brake Controls	Fri 10/31/08	Fri 11/14/08	
34	Finite Element Analysis	Mon 10/20/08	Mon 10/27/08	Ben and Phil
35	Machine Mounting Pieces	Mon 10/27/08	Mon 11/3/08	Ben,Bryan,and Phil
36	Install Components	Mon 11/3/08	Fri 11/14/08	Ben,Bryan,and Phil
37	**Design Review #4**	**Tue 11/11/08**	**Tue 11/25/08**	
38	Written Report	Tue 11/18/08	Tue 11/25/08	All
39	Oral Presentation	Fri 11/21/08	Tue 11/25/08	All
40	Finalize Plumbing Layout	Mon 11/10/08	Fri 11/14/08	Bryan
41	Hose Doctor Installation	Fri 11/14/08	Fri 11/14/08	
42	Finalize Electronic Controls	Mon 11/17/08	Fri 11/21/08	Adebimpe and Kyle
43	Final Testing/Debuging	Wed 11/19/08	Fri 11/21/08	Ben,Bryan,and Phil
44	Lotus Dyno Testing	Fri 11/21/08	Fri 11/21/08	
45	Lotus Data Analysis	Fri 11/21/08	Mon 12/1/08	Ben and Phil
46	**Thanksgiving Break**	**Thu 11/27/08**	**Sun 11/30/08**	
47	**Design Expo**	**Mon 12/1/08**	**Thu 12/4/08**	
48	Expo Poster	Mon 12/1/08	Thu 12/4/08	All
49	Prepare Xebra	Wed 12/3/08	Thu 12/4/08	All
50	**Final Report**	**Mon 12/1/08**	**Tue 12/9/08**	**All**
51	**Graduation**	**Sun 12/14/08**	**Sun 12/14/08**	



APPENDIX K – LOGAN CLUTCH HYDRAULIC CLUTCH

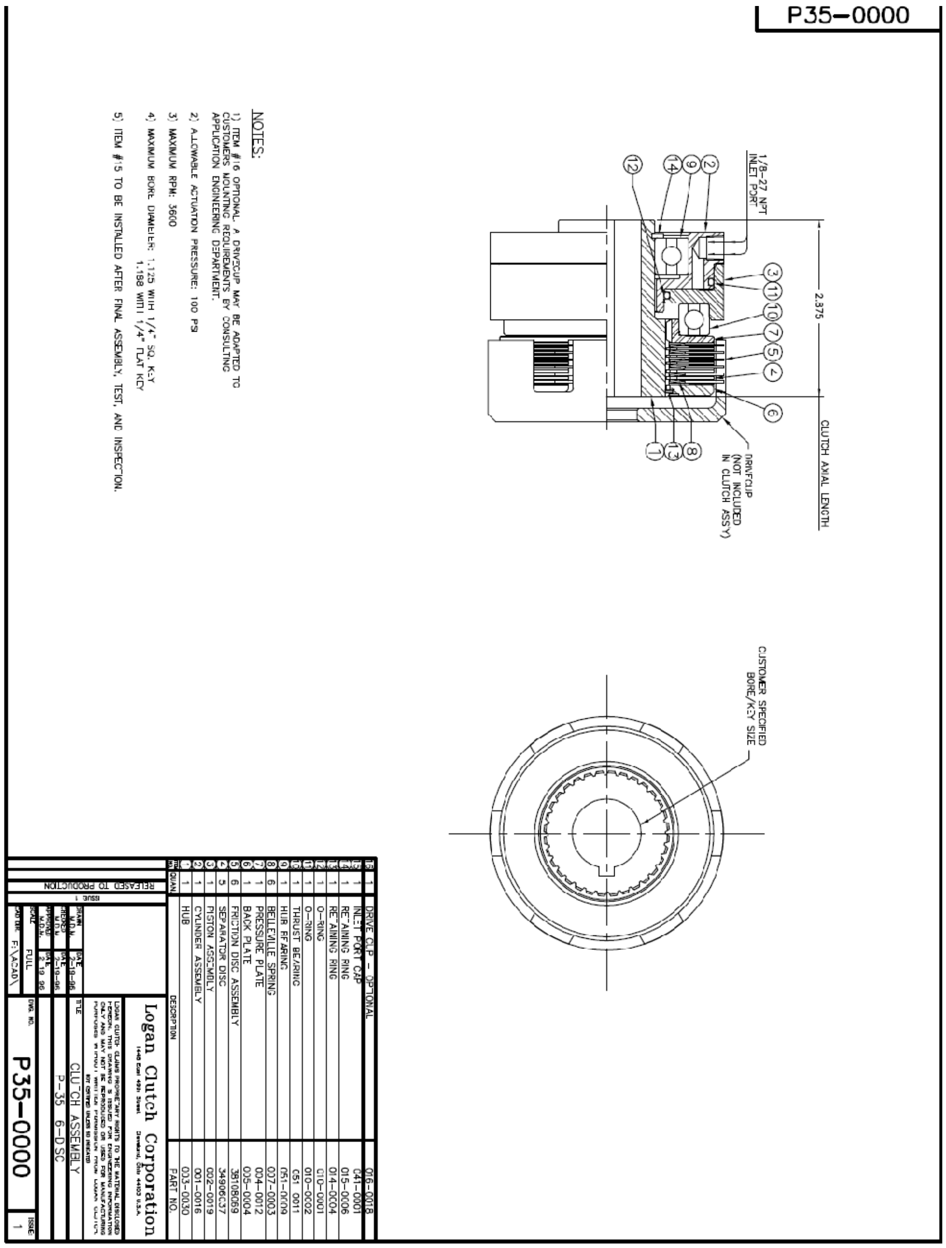


Figure 29: Hydraulic Clutch from Logan Clutch

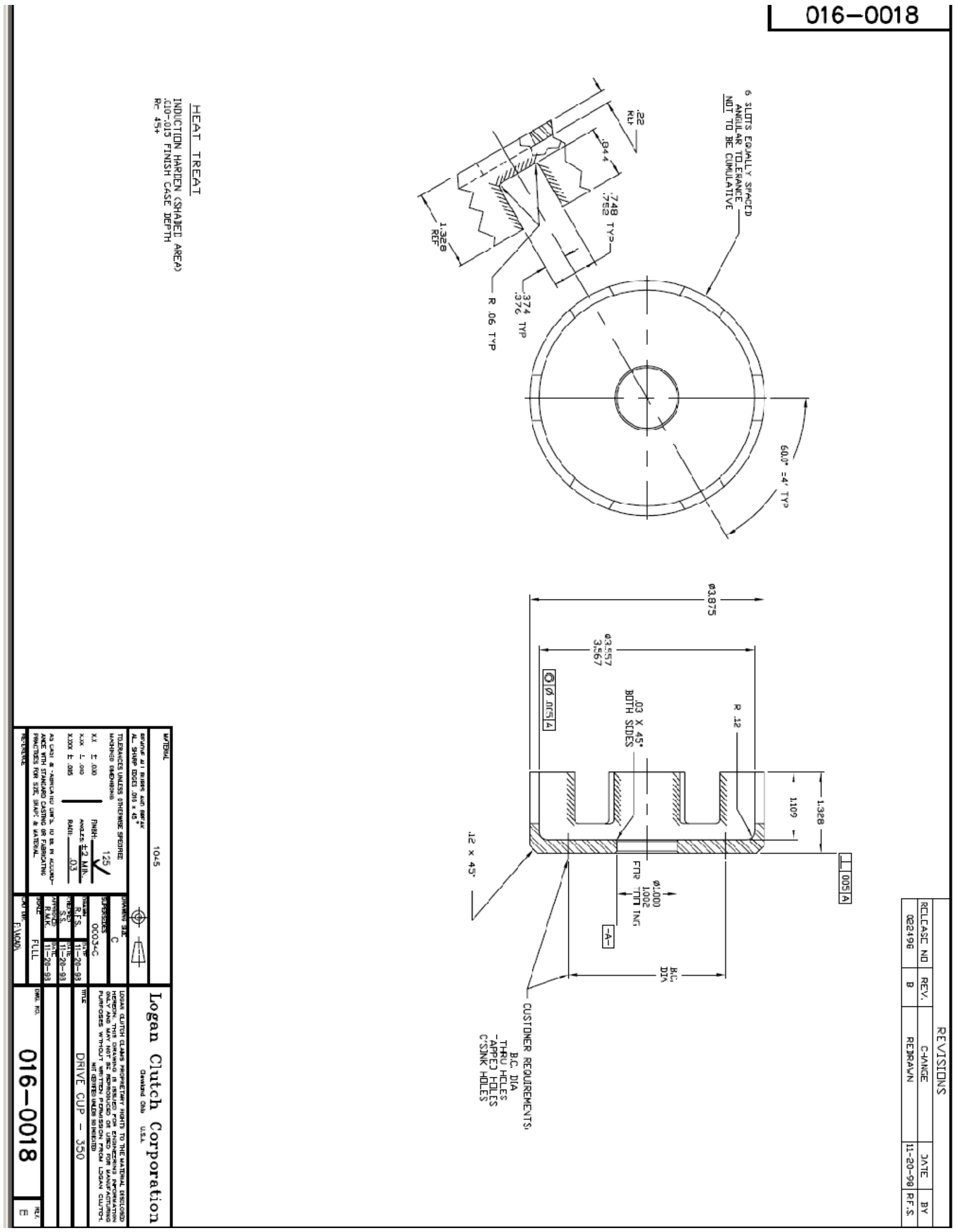


Figure 30: Clutch Drive Cup from Logan Clutch

APPENDIX L - TEAM MEMBER BIOGRAPHIES

My name is **Phillip Geisler** and I am a senior mechanical engineer at the University of Michigan. I am originally from St. Joseph, MI where I enjoy the beaches of Lake Michigan in the summer. Academically speaking, I enjoy finding out how things work, working with other people, and I also like to be a leader in a team situation. When I am not spending time with my school work, I enjoy spending time with friends, working out (including but not limited to playing racquetball, basketball, running, and lifting weights), watching movies, and meeting new people. When I am back home in St. Joseph, I frequently visit the beaches of Lake Michigan where I enjoy water sports like tubing, jet skiing, and wake boarding. However, this past summer I was not able to hit the beach because I was in New York City working in the realty and construction business. When I graduate, I would like to move back to New York and work in this industry.

Phone: 269-214-1060
Email: pgeisler@umich.edu

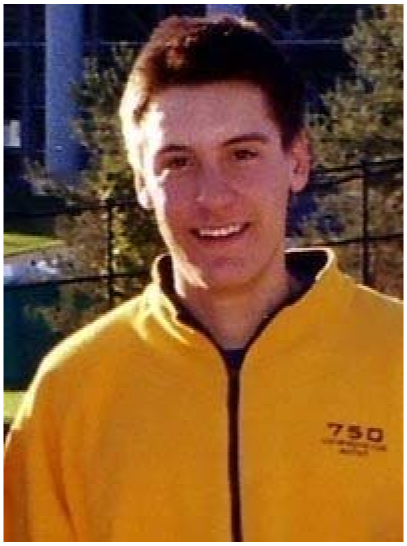
Kyle Anderson is in his final semester of study in Mechanical Engineering at the University of Michigan. Upon graduation, Kyle will work in Drivetrain Design at Toyota Technical Center in Ann Arbor, MI. Kyle enjoys the design process and developing new ideas in the pursuit of solving Engineering problems. Prior to studying Engineering, Kyle completed a Bachelor's and Master's degree in Music Performance. When not studying, Kyle enjoys spending time with his wonderful wife and cycling.

Email: kylema@umich.edu

My name is **Adebimpe Lawal**; I am a senior majoring in Mechanical Engineering at the University of Michigan and will be graduating in December 2008. My favorite subjects are thermodynamics, heat transfer and finite element analysis and I would ideally like to concentrate my career in a combination of those areas after I graduate. More specifically, I want to work in the aerospace industry. I also have a strong interest in computer programming and would like to become proficient in at least one language sometime in the future (C++ most likely). In my free time I like to read (although I'm phasing out of that hobby) and I also like to watch movies (though only with a serious movie-buddy and only the baddest most explosive action movies, but I digress). A recent addition to my *things I'd like to do sometime in the future* list is to learn glass-blowing. My dearest dream is to one day write a book, ***the*** book.

Cell: (248) 943-2511
Email: adebimpe@umich.edu

Ben Hagan



I am a senior mechanical engineering student at the University of Michigan and will graduate in December 2008. I plan on going to graduate school for a year to obtain a master's degree with a thesis on hydraulic oil de-aeration with the support of the EPA. I became an engineer because I like math and science and want to apply those subjects' principles to solve real problems.

I am good at calculations and have experience with AutoCAD, UniGraphics, Pro/E, and Hypermesh. I have limited machine shop experience, but I am fairly adequate with electronics for a mechanical engineer. Finally, I also work well with Microsoft Office.

I am from Kansas City, MO. In my free time, I enjoy watching sports and attending UM football and basketball games. I also like to play fantasy football and lift weights.

My name is **Bryan Hartman**. I'm 21 years old and I am a fourth year Mechanical Engineering student at the University of Michigan. I was born and raised in Fort Lauderdale, Florida. A fun fact about me is that I had never seen snow until the winter of my freshman year at Michigan. My hobbies include playing and watching all types of sports, watching movies, and learning about new technology. I am involved in many student-based groups, most notably as an active participant in Relay for Life, benefitting the American Cancer Society, and Dance Marathon, benefitting Motts Children's Hospital. After my years in college, I hope to settle up north, leaving the beaches of South Florida.