ME 450 Winter 2008: Project 13
Design of Hydraulic-Electric Hybrid Vehicle

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TABLE OF CONTENTS

ABSTRACT

The University of Michigan has purchased a 3-wheeled electric vehicle called the Xebra. The Environmental Protection Agency is sponsoring the conversion of the Xebra into a hydraulic-electric hybrid. Acceleration demands of city traffic drops the efficiency of an electric vehicle from 90% to 60%. The addition of hydraulics to the electric vehicle will allow for efficient energy use during acceleration, dramatically improving the range of the Xebra. This project is currently spanning several semesters. The focus of this term was to re-work the hydraulic system's layout, to couple the hydraulic system to the drive shaft, and to integrate the electric, hydraulic, and mechanical components. The ultimate goal for this term was to have a working hydraulic-electric hybrid vehicle for display at the end term Design Expo.

EXECUTIVE SUMMARY

Project 13 is sponsored by the EPA. The goal of the project is to create the world's first hydraulic-electric hybrid vehicle. This term's goal was to deliver a working hydraulic-electric hybrid vehicle by the end of the semester. This project is a continuation of previous semesters' work. Therefore, some key components of the system have already been purchased and installed.

Design Problem

Electric vehicles experience extreme inefficiency under heavy loading due to both motors and batteries becoming less efficient with high current use. The goal of the Xebra hydraulic- electric hybrid is to accelerate the vehicle at low speeds using hydraulic power, thus reducing the load on the electric driveline during the times when the largest amount of power is needed. Due to the scope of the project, and the desire to have a working concept at the end of the semester, a regenerative braking system will not be included in the current design iteration. This feature, which should dramatically increase the efficiency of the hybrid system, will be designed and implemented in future semesters.

Specifications

The primary goal of our sponsors at the EPA was to have a running hybrid at the Design Expo. In addition, a target speed of 27 mph was set for the hydraulic drive system and an acceleration goal was set. Our primary task is therefore to integrate the hydraulic components into the Xebra vehicle with a gear reduction that closely meets the performance criteria.

Concept Generation

Initially, we brainstormed ideas on how to integrate the mechanical reduction into the hydraulic system. After narrowing down the choices to ones that could be implemented onto the Xebra vehicle, we used Pugh charts to determine which main reduction system we would utilize. Additionally, we were able to select individual components of the system like clutches and pumps utilizing Morphological charts. Designs considered included gear, sprocket, and belt reduction. Different mounting locations, including a hydraulic bike style front wheel, were considered.

Final Design

Our design utilizes a sprocket reduction that is integrated into the input shaft of the existing Xebra differential. Thus, it utilizes the existing reduction, reducing the parts count and weight of our system. One sprocket will be pressed onto a one-way bearing, which will in turn be adapted to fit the output shaft of the hydraulic motor. The other sprocket will be keyed onto the driveshaft that is splined on one end to mate with the differential. Additional members will be added to the chassis to support the driveshaft and hydraulic motor.

Conclusion

After a semester of work, we successfully designed and built a working hydraulic-electric hybrid. Due to the desire for an emergency shutoff at the high pressure reservoirs and a limit on power at the Design Expo, only one of the reservoirs was used, limiting the top speed and average acceleration of the vehicle. However using these parameters the Simulink model we developed closely matched the performance of the Xebra, indicating that at full capacity the top speed should meet the 27 mph goal of our sponsor.

1 PROBLEM DESCRIPTION

This section will introduce the project by providing background information, our sponsor's motivation, and the goals for this term.

1.1 Background

The sponsor for our project is the U.S. Environmental Protection Agency (EPA). The EPA was established in 1970 to protect human health and the environment. Part of the EPA's Office of Transportation and Air Quality (OTAQ) is the National Vehicle and Fuel Emissions Laboratory (NVFEL). The OTAQ works to advance clean fuels technology while the NVFEL carries out laws to control air pollution from motor vehicles, engines, and their fuels. Our team is working directly with David Swain, an engineer at the NVFEL, which is located in Ann Arbor, MI [1]. The EPA has sponsored several design projects at the University of Michigan, dating back to the fall of 2004. The EPA has sponsored previous projects for the Xebra, a hydraulic-electric hybrid vehicle. Most of their work with the University, however, has been involved in the development of a regenerative braking system for bicycles. The EPA has future plans for the implementation of this regenerative braking scheme on the Xebra.

The University of Michigan purchased a Xebra (Figure 1) – a three-wheeled truck that was made in China by ZAP (Zero Air Pollution). It is legally classified as a motorcycle and meets federal requirements for use on public streets. The Xebra was built to serve as a utility or maintenance vehicle, for example, in a closed community.

Since the concept of this project is based on previous ME 450 projects, a great deal of background information and data analysis were readily available for us. The hydraulic system implanted on the Xebra during last semester is a large-scale version of the hydraulic system used in a bicycle regenerative braking system, another ongoing ME 450 project. An exception to this is the use of an additional slow-fill pump powered by electricity to recharge the high pressure accumulator consistently. More detailed information and help were also available from David Swain of the EPA, and two U of M students, Alex Lagina and Jason Moore who worked on the regenerative braking bicycle project for several semesters.

Figure 1: ZAP's Xebra Electric Truck

1.2 Motivation

The current design of the Xebra utilizes an electric motor to power the truck. The range of the Xebra is only 25 miles per charge of the electric motor. This is due to the fact that the electric motor is only about 60% efficient under heavy loads, such as accelerations seen from city traffic. However, the electric motor is about 90% efficient when powering a vehicle to maintain a constant speed. Our team, in collaboration with David Swain of the EPA, plans to utilize a hydraulic motor to power the Xebra under these heavy loads. While hydraulic motors are approximately 85% efficient under all loads, hydraulics are not as effective when powering a vehicle at a high velocity. Hydraulic power at high velocities requires a greater amount of hydraulic fluid, which in turn requires a larger system, ultimately increasing the weight of the system. Therefore, the hydraulic motor will accelerate the vehicle up to 27 mph while the electric motor will accelerate the vehicle above 27 mph and maintain any cruising speeds. We see the concept of a hydraulic-electric hybrid vehicle being consistent with the mission of the EPA.

1.3 Winter 2008 Goals

The overall goal for our team this semester will be to accelerate the Xebra from 0 to 27 mph using the hydraulic motor. To achieve this goal, we first plan to implement a new efficient layout of the hydraulic and electric components. This layout will account for the future implementation of a regenerative braking system, which is planned to be a project for next semester. The installation of the hydraulic system will require us to couple a motor, pump, and slow-fill pump to the drive shaft.

1.4 Potential Impact of a Successful Design

A successful design of the world's first hydraulic-electric hybrid vehicle could have a great impact on several things. Mass producing this vehicle could revolutionize the automobile industry, causing major automakers to possibly develop their own hydraulic-electric hybrids due to market demands. If the sale of these vehicles were great enough, the oil industry could take a large hit due to a great decrease in gasoline sales. Also, if many consumers were to purchase a vehicle of this type, the environment would see a great improvement due to the use of these vehicles since they do not produce any type of air pollution. These hybrids are also much quieter than an average car running on gasoline, which would decrease the noise level on streets and highways.

2 PROJECT REQUIREMENTS AND SPECIFICATIONS

This section explains the customer's project requirements and technical specifications, both used to form a Quality Function Diagram (QFD).

2.1 Customer Requirements

After meeting with David Swain, our sponsor, we were able to determine what the customer's requirements were for Xebra. Table 1 on p. 3 shows a complete list of the requirements, along with a description of what each requirement entails.

Table 1: Customer Requirements

Customer Requirement	**Description**
Transferable to future semesters	Thorough documentation; layout designed with future goals in mind
Comfortable feel during acceleration	Smooth acceleration; natural feel to driver
Sufficient acceleration to top speed	Hydraulic motor able to accelerate vehicle to 27 MPH
Efficiency in plumbing	Minimal frictional loses by avoiding unnecessary bends
Lightweight	Components need to be as light as possible
Reliable components	Components should not fail
Aesthetics	Vehicle needs to be pleasing to the eye; professional look
Safety	Vehicle is safe to drive under normal conditions
Easy to use	Use and feel should be familiar to a common car
Easy to service	All components must be accessible
Maintains vehicle function	Passenger and load amounts remain unchanged

The customer requirements that the EPA put the most emphasis on were: aesthetics, sufficient acceleration to top speed, and that the vehicle maintained its function.

2.2 Engineering Specifications

After receiving the customer requirements from our sponsor, we were able to determine the engineering specifications needed to complete the project. Table 2 on p. 4 shows the specifications we came up with, along with the target values and units for which they were associated with. The target values have been slightly modified throughout the semester as we developed a better understanding of the system and the requirements needed to meet our goals.

Table 2: Engineering Specifications

Engineering Specifications	Target Value	Units
Flow rate of slow-fill pump	0.04 – 0.05	L/s
Gear ratio from the motor and pump to the drive shaft	7.9:1	n/a
High-side accumulator pressure	15 - 30	MPa
Side loading on pump	0 – 5	N
Side loading on motor	0 – 5	N
Strength of materials for gears	400 – 1000	MPa
Pressure loss through geometry of fittings	0 – 20%	%
Working temperature of the fluid	275 – 325	K
Size of hydraulic fittings	8 – 24	mm
Space for all the components (layout)	0.2 – 0.4	m^3
Compatible components	Yes	Yes/No

The engineering specifications were created through interpreting the customer requirements and regular discussion with our sponsor. Also, a number of specifications were made that were required for our system to have a reasonable chance of working. For example, two constraints are the amount of space for all components and the necessary strength of materials. These basic requirements have caused us to focus on designs that take the least room and are made using parts or materials that can handle the desired loads.

2.3 Quality Function Diagram (QFD)

After determining the customer requirements and engineering specifications, we were able to create a QFD for our project (Appendix D). First, the weight percentage for each of the customer requirements was found by comparing each requirement to every other requirement. Next, the correlation matrix was created, showing how much each specification influences the customer requirements. We were then able to determine which customer requirements and engineering specifications ranked the highest for our design. For the customer requirements, the three items that ranked most important were safety, maintaining vehicle function, and transferability of the project to future semesters. The three engineering specifications ranked most important were space for all of the components, ability to achieve high-side accumulator pressure, and strength of the material for the gears.

3 CONCEPT GENERATION

In this section, the concept generation process is discussed, and several different concept designs are introduced.

3.1 FAST Diagram

Since it is very important for our concept design to meet all the customer requirements, we to develop a FAST (Function Analysis System Technique) diagram for our project. The FAST diagram takes the main function, or "functive," of the product and breaks it down into specific sub-functions that are needed to describe the main function. These sub-functions are then used as guidelines for generating design concepts. The FAST diagram of out project is shown in Figure 2.

We determined that the main "functive" of this project was to improve performance of the electric vehicle. This was then broke down into the subcategories of assist launch, captures energy, assures safety, assure convenient, assure reliability, please senses, and enhance the product.

Then next two columns specify the methods used to achieve the goals listed in subcategories. For example, vehicle launch is assisted using the hydraulic motor driven by fluid released from high pressure accumulator. Energy is captured as a pressurized fluid in high pressure accumulator by using regenerative brake or slow-fill pump. To assure safety, we can use a pressure release valve as a type of emergency switch. The rest of the diagram can be interpreted same way.

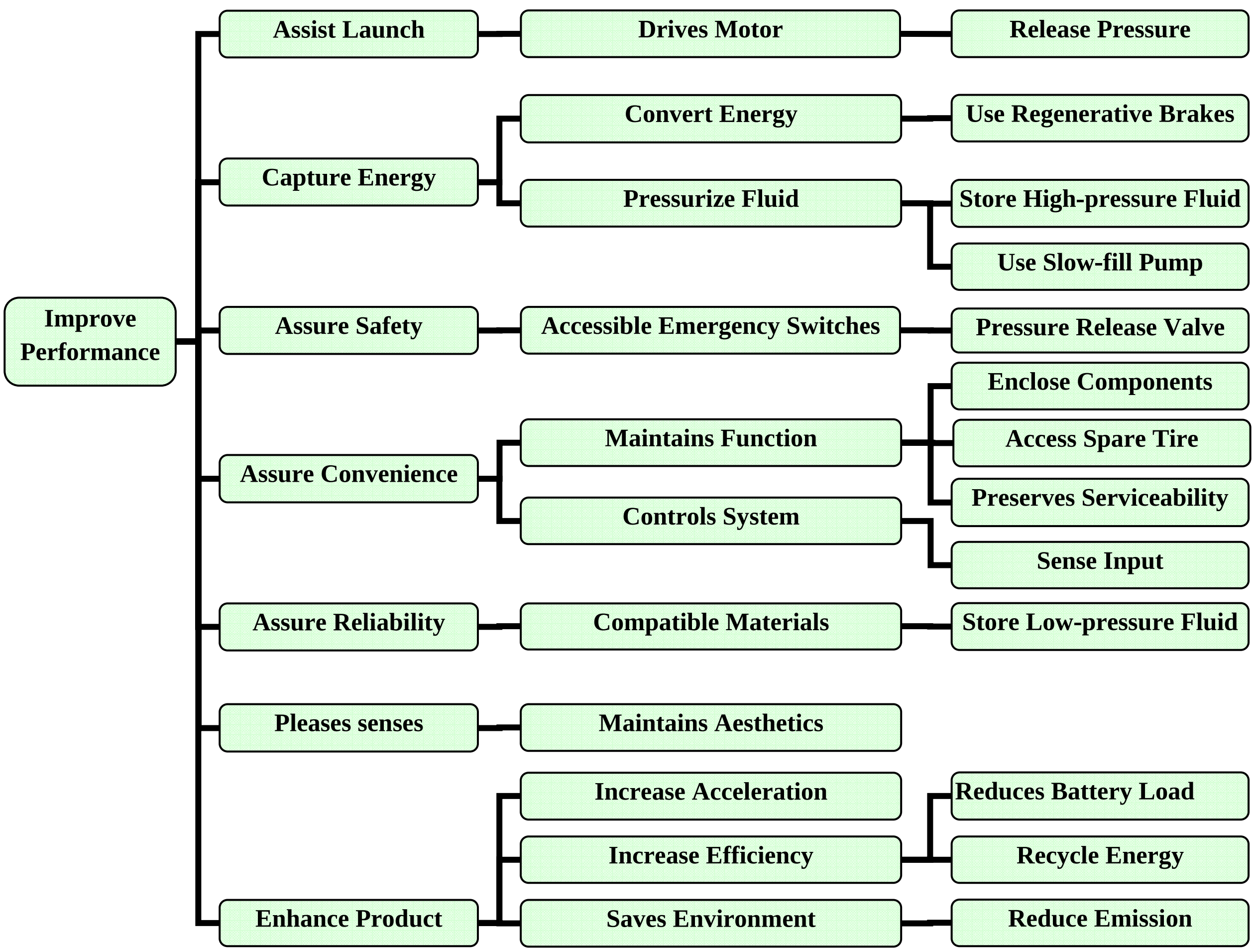
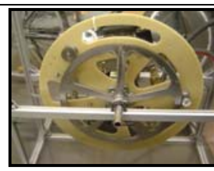


Figure 2: FAST Diagram

3.2 Morphological Chart

A Morphological chart was created to generate high-level concepts within the main design parameters of the project. Since previous group already mounted the hydraulic system on the electric vehicle, our main design parameters are focused on the concept reduction box design. Table 3 shows the complete morphological chart of the project.

Table 3: Morphological Chart

Design Parameters	Concept 1	Concept 2	Concept 3	Concept 4
Reduction Type	 Spur gears	 Bike Hub Wheel	 Chain and Sprockets	 Belt and pulleys
Clutch selection	 Overrunning clutch	 One way bearing	 Ratchet and Pawl	
Reduction Box Placement	 Extended Shaft	 Welded to Transmission	 Motor drives Rear Axle	

Based on the FAST diagram, three main design parameters were generated; reduction type, clutch selection, and reduction box placement. Some of the design parameters from the FAST diagram are omitted because those are already determined and installed on the vehicle by the group from last semester. In addition, regenerative braking concepts will be reserved for future semesters.

The concept designs for each design parameter were generated from brainstorming, taking apart the vehicle's main components such as transmission and suspensions (reverse engineering), and reviewing customer requirements in the QFD.

For the reduction type, we have four concept designs: spur gear, bike hub wheel, chain and sprockets, and belt and pulleys. Each reduction type works in a similar way, with the exception of the bike hub wheel – one side mounted to hydraulic motor, and another side mounted to spline shaft. However, bike hub wheel has to be installed as a front wheel of the vehicle. The overall design would stay the same for the bike hub wheel, but the pump and motor displacement and accumulator size will be different.

There are three possible choices for one-way clutch that meets the customer requirements. Essentially the overrunning clutch (Concept 1) and one-way bearing (Concept 2) are very similar, with the bearing being cheaper and simpler. Ratchet and pawl (Concept 3) also locks one way and runs freely the other way. For the reduction box placement, we found three possible locations around the transmission. Concept 1 uses an extended shaft mating with the shaft for the gear in the transmission box. For Concept 2, reduction box is welded to existing transmission box. Concept 3 uses rear axle to be coupled to the hydraulic motor.

4 CONCEPT SELECTION

After we generated the Morphological chart, we evaluated and selected a final concept design. The selected concept designs are shaded in light green and shown in Table 3. The reduction type was selected using Pugh chart and discussed in the next section. For the clutch selection, we chose the one-way bearing (Concept 2) because of its compact size, manufacturability, and low price. This bearing meets all of our technical requirements such as max torque (193 Nm) and max speed (4000 RPM). The reduction box is determined to be placed on an extended shaft mating one of the transmission gears (Concept 1). Concept 2 for the reduction box placement requires complicated machining, which is not feasible with limited time and machineries to access. Concept 3 for the reduction box has potential problem associated with a differential. Because of the existing differential in the transmission box, torque cannot be applied to one of the rear axle directly. Therefore, concept 3 had to be avoided.

To create more objective selection process, a Pugh chart is developed and shown in Table 4. A Pugh chart helped evaluate several concept designs by setting up a list of customer requirements and rating each design in terms of the individual criteria. Each customer requirements is given a weight (%) based on the importance relative to each other. Then each concept design is given a rating of 0 to 5 based on its affect on the customer requirements, with 5 having the most affect. The weight for each customer requirement is multiplied to the rating for each design, and these weighted values are added to produce the weighted total for each design, with the highest weight being the best. The Pugh chart determined that sprocket and belt reduction are the two best concept designs. After we researched on the sprocket and belt, we determined that chain and sprocket reduction would be the best selection for our project because we have a significant amount of torque produced from the hydraulic motor and belt reduction type could slip under the high torque.

Table 4: Pugh chart

	Weight (%)	**Hydraulic Bike Wheel**		**Gear Reduction**		**Sprocket Reduction**		**Belt Reduction**	
		Rating	Weighted	Rating	Weighted	Rating	Weighted	Rating	Weighted
Acceleration	9	4	36	5	45	5	45	5	45
Acceleration Jerk	3	4	12	4	12	4	12	4	12
Lightweight	7	2	14	2	14	4	28	3	21
Reliability of components	5	3	15	5	25	3	15	3	15
Aesthetics	2	5	10	3	6	3	6	3	6
Stability	5	2	10	5	25	5	25	5	25
Safety	10	2	20	5	50	5	50	5	50
Easy to use	3	5	15	5	15	5	15	5	15
Easy to service	7	3	21	3	21	5	35	4	28
Maintains Vehicle function	8	5	40	5	40	5	40	5	40
Final drive ratio	9	5	45	5	45	5	45	5	45
Strength of material for reduction	4	5	20	5	20	4	16	4	16
Component Layout	8	2	16	3	24	4	32	3	24
Prototype manufacturability	20	3	60	3	60	5	100	5	100
	TOTAL:	5	334	3	402	1	464	2	442

5 ENGINEERING DESIGN PARAMETER ANALYSIS

In this section, the key engineering parameters of the final design are evaluated. These calculations will ensure that all the components meet our specifications. We have created a model in Simulink that is able to predict the response of the vehicle under different hydraulic drive parameters. In particular, we are able to vary drive ratio, maximum pressure, pre-charge pressure, accumulator volume and motor displacement, to evaluate the differences in the vehicle response. The model can be seen in Appendix G. The model allows us to analyze the fluid required, maximum motor RPM, vehicle acceleration, vehicle speed, motor torque, and accumulator pressure. Additionally, tire and road load models are included to predict a more accurate response. The model is used to optimize the hydraulic drive system with the components that are available to us.

The material and manufacturing process selection were evaluated with CES software and provided in Appendix C.1. We selected drive shaft and motor mount to be analyzed with this software. Then the Design For Assembly (DFA) chart are created, and the design efficiency is evaluated. The old design had an efficiency of 32%. Then we redesigned it and evaluated in the DFA chart, and the efficiency of the design increased to 48%. We basically modified the sprocket from sprocket and flange mounted with bolts and nuts to A type sprocket with hub. To see the impact of the material (aluminum and steel) on the environment, we found materials from SimaPro and evaluated them. The result showed that aluminum is likely to have more impact on the environment. However, after we calculated the Ecoindicator point values, we found that steel is likely to have a bigger impact when the life cycle of the whole product is considered. We determined that the production volume of our project to be 1000. Then the manufacturing process of the drive shaft and motor mount was determined to be machining such as milling and lathing.

5.1 Hydraulic Motor

For the motor we determined to use 23cc Parker PGM 517 hydraulic motor (Appendix K). As the displacement of the hydraulic motor increases, the maximum speed of the motor decreases, and this is shown in Appendix I. To meet the sponsor requirement with medium range of gear reduction and acceleration, the displacement of the motor is determined to be 23cc , which has 3300rpm of maximum speed and 3988psi of maximum pressure. Also, its compact size compared to the larger displacement motors gave a good reason to select this motor. The biggest reason to use Parker hydraulic motor was that our sponsor has been used this motor and could get a discount on this. The flow rate using this motor is calculated and presented in the section 5.5. In addition, the gear reduction required to reach 27mph is evaluated in the next section.

5.2 Gear Reduction

For the vehicle to reach 27mph with hydraulic motor selected, approximately 7.272 : 1 reduction is required. With 20 in diameter tire (0.508 m), the vehicle travels 1.59593 m per revolution. Therefore, when the vehicle reached 27mph (12.07 m/s), the drive shaft should be rotating 453.783rpm.

$$1.59593\ m * 453.783\ rpm = 12.07\ m/s = 27\ mph$$

Given the maximum speed of hydraulic motor of 3300rpm, which is likely happen in our system with 4000psi inlet and the atmospheric pressure in the outlet, the reduction is then calculated to be 7.272. We chose the reduction to be 7:1 in our final design due to the limited availability of the number of teeth on sprockets. Therefore, this reduction ensures the vehicle to reach the speed of 27 mph. The material selection analysis of the sprockets can be found in the section 5.10.

5.3 Slow-fill Pump

Since the slow-fill pump is not the main component to recharge the high pressure accumulator, it is only required to pressurize up to 3500 psi. After few weeks of search, we finally found one company, Monarch, that still produce 72V DC hydraulic pump. Since Xebra uses 72V DC, it is required to use 72V DC hydraulic pump to avoid using an electric converter, which adds more weight and complexity to the vehicle. The maximum pressure and flow rate of the pump was determined by the sponsor, and the electric energy usage of this specific pump was then evaluated and shown in the section 5.7.

5.4 Total amount of hydraulic fluid required with air drag effect

The hydraulic system accelerates the Xebra from 0 to 27mph, or v_{ss} = 12.1 m/s. Since the drag force is a function of the velocity in our case, the maximum drag force is generated at the maximum speed. Using the Bosch handbook for calculations [7], the energy due to drag on the vehicle was interpolated to be 1.60 kW/m^2 using a vehicle frontal area of 2.07 m^2. Therefore, the maximum power to overcome due to the drag was found to be P_d = 1.65 kW. Then the average acceleration of the vehicle was determined to be, $a_{avr.}$ = 1.92 m/s^2, with an acceleration time of t = 6.19 seconds by using Eq. 1 and 2.

$$a_{avr.} = \frac{6E_r}{M\pi D} - \frac{2P_d}{Mv_{ss}} \quad \text{Eq. 1}$$

$$t = \frac{v_{ss}}{a_{avr.}} \quad \text{Eq. 2}$$

As a result, using Eq.3, the total energy loss due to drag, E_d, was calculated to be 10.40 kJ. Then the energy required from hydraulic system is determined to be 94.14 kJ by using Eq.4. Thus it requires a volume of 4.57 L of hydraulic fluid for accelerating from 0 to 27mph. This volume is 23% larger than that of our preliminary calculations which did not include the air drag.

$$E_d = P_d t \quad \text{Eq. 3}$$

$$E_h = \frac{SE_d}{\pi D / 6} \quad \text{Eq. 4}$$

5. 5 Maximum Flow Rate of the Hydraulic fluid

The Flow rate of the hydraulic fluid during acceleration reaches its maximum when the vehicle traveling at 27 mph (velocity, or $v_{vehicle}$). At this speed, the hydraulic motor rotates at ω_m =3176 rpm calculated using Eq.5 given the tire radius of 10 inches (r_{tire}) and total gear reduction of 7:1. Using 23 cc hydraulic motor running at 3176 rpm, the volumetric flow rate was determined to be 19.3 gallons per minute.

$$\omega_m = \frac{7v_{vehicle}}{r_{tire}} \quad \text{Eq. 5}$$

5. 6 Time to charge high pressure accumulator

From the performance graph of Monarch 72V 17-190 DC motor (Appendix L), the average flow rate of the motor to charge from 2500 psi to 3500 psi is determined to be 0.0379 liter/s. The performance graph of our slow-fill pump is shown in Appendix L. The amount of hydraulic fluid needed to accelerate from 0 to 27mph was determined to be 4.57 liters in previous section. Therefore, time to charge high pressure accumulator from 2500 psi to 35 psi is determined to be 120.58 s = 2.01 min using Eq.6.

$$t_{charge} = \frac{4.57\ liter}{0.0379\ \frac{liter}{s}} = 120.58s = 2.01\ min \quad \text{Eq.6}$$

5. 7 Electrical energy required to recharge the high pressure accumulator with slow-fill pump

From the performance graph of Monarch 72v DC motor, the power consumption graph is produced and shown in Figure 3. Then we integrated this to get the total energy to charge the high pressure accumulator. The total electric energy consumed during recharging was determined to be 159.82 kJ.

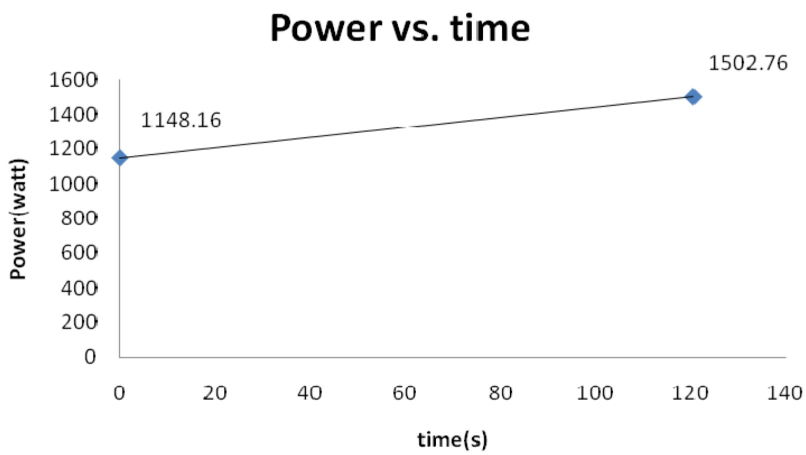


Figure 3: Slow-fill pump power consumption graph

5. 8 Electrical energy to accelerate 0 to 27 mph with stock electric motor

From the HOT 505 test performed in the previous semester, the average power consumed to accelerate 0 to 27mph was determined to be 129.62 kJ. This was done by integrating HOT 505 test data during the vehicle accelerating from 0 to 27mph. We repeated integration at several different locations of HOT 505 test data and averaged them to determine average power consumption of the stock electric motor. The test data is not included in this report, but it is available upon request.

5. 9 Comparison between hydraulic system with slow-fill pump and stock electric motor

In conclusion, the hydraulic system with only slow fill pump installed consumes slightly more energy (23.3%), but overall performance improved so much. Therefore, when we account all the improvements we achieved, we concluded that using hydraulic system with slow fill pump is a lot more efficient than using the stock electric motor. The results are shown in Table 5. The motive for this increase in energy consumption was to achieve the acceleration requirement of the EPA. While the vehicle does consume 23.3% more energy using the hydraulics compared to the electric motor, the implementation of regenerative breaking in a future term will greatly reduce the amount of energy consumption to maintain the same acceleration.

Table 5: Various performance factors are compared

	0-27mph	Energy	Power	Average Acceleration
Electric Motor	12.7s	129.62 kJ	13.67 HP	1.08 m/s^2
Hydraulic system	6.19s	159.82 kJ	34.62 HP	1.92 m/s^2
%change	51.3% faster	23.3% increase	153.3% increase	77.8% increase

Appendix M displays efficiency curves for our electric battery and the potential improvements utilizing hydraulics. The plots will show where inefficiencies occur using electric batteries with a electric motor, and how hydraulics can assist the electric components, allowing for an improvement in overall efficiency.

5.10 Material selection of drive shaft, sprocket and chain, supporting frame, and bearings
In order to determine the material for the extended shaft, we performed Finite Element Analysis (FEA) using Unigraphics NX 5.0. This analysis showed that we have the maximum stress at the edge of spline, and this value was determined to be 57MPa. Therefore, we chose 1117 low carbon steel rod which has 400MPa yield strength giving us safety factor of 7. The FEA of the splined shaft is shown in Figure 4.

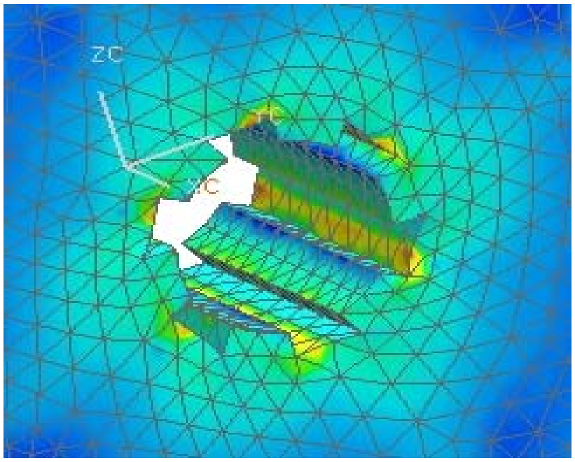


Figure 4: The Finite Element Analysis (FEA) of the splined shaft

The sprocket and chain are chosen based on the load calculation. The maximum torque created from the hydraulic motor is 71 ft lb. Thus, the maximum load on the chain can be calculated by dividing the maximum torque by the radius of one of the sprocket, and it was determined to be 543lb. We chose the single strained ANSI 40 roller chain, which has breaking point load of 3934 lb and working load of 437 lb. The number of teeth on the sprockets is determined based on the gear reduction we needs, and the material is chosen to be the same as the chain since the sprocket does not fail before chain does. The bore size (5/8") of the sprocket was chosen to be smaller than the diameter of the motor shaft (7/8") so that it could be bored out to the size of the motor shaft.

The supporting frame for the shaft and the motor are chosen to be the same as the Xebra frame made of. This frame is made of mild steel, which has at least 240Mpa yield strength. However, the stress on the frame due to the gear components is almost negligible compared to the load on the frame during the normal operation of the vehicle. Thus, it does not require further analysis.

The bearings and one-way bearing were chosen based on the working speed and the load. First, the ball bearings to support both motor and drive shaft will run at maximum rotational speed of 3300 rpm. The spec of the bearings we chose has maximum rpm of 9000 rpm and maximum load of 2271 lb, which gives us about safety factor of 2.7. Once again, the load on the bearing will be so much smaller than 2271 lb, so it does not require further analysis.

We also have spacer on the motor shaft since the bore of one-way bearing was larger than the diameter of the motor shaft. We also did the FEA using Unigraphics NX5.0, and the result is shown in Figure 5 on p. 12. The stress on the key way of the spacer was determined to be 38MPa. The yield strength of 1117 low carbon steel is 400MPa, and it gives us a safety factor of 10.5. A complete bill of materials can be found in Appendix A.

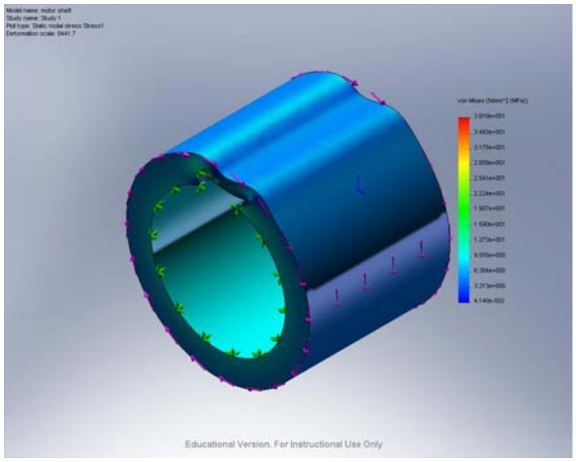


Figure 5: The Finite Element Analysis of the motor shaft spacer

6 FINAL DESIGN DESCRIPTION

Hydraulic drivetrains can be used to quickly and efficiently store and release great amounts of energy, making them particularly attractive for vehicle applications that entail a significant amount of stop-and-go driving. A hydraulic hybrid vehicle can capture and use a large percentage of the energy normally lost in vehicle braking. Hydraulic hybrid technology can increase fuel economy as well as vehicle acceleration performance. The hydraulic launch system will store energy via the hydraulic pump during braking of the vehicle. This energy is later released through the hydraulic motor, accelerating the vehicle. With the addition of a hydraulic launch system for vehicle acceleration, the electric batteries will not be burdened during accelerations and the overall efficiency and range of the Xebra will be improved. The main goal of our design is to allow the hydraulics to accelerate the vehicle. The addition of regenerative braking in future semesters will ultimately improve the efficiency.

During accelerations, after flowing through the hydraulic motor, the hydraulic fluid loses its pressure and is stored in a low-side accumulator (Figure 6). When the hydraulics are not being utilized for vehicle acceleration, the slow-fill pump will re-charge the high-side accumulators. Future terms implementing regenerative braking will allow the slow-fill pump in conjunction with the regenerative brake pump to re-pressurize the two high-side accumulators until it is needed for acceleration. When the vehicle is using the electric motor (while cruising), valves are configured such that the fluid is freely circulating in the system without flowing through the pump or motor. For the Design Expo, an emergency-stop valve was directly connected at the end of one of the high pressure accumulators and the second high pressure accumulator was capped off for safety purposes. Engaging the emergency-stop valve will disallow the high pressure accumulator to release fluid. This can be utilized if a leak occurs while the accumulator is filled.

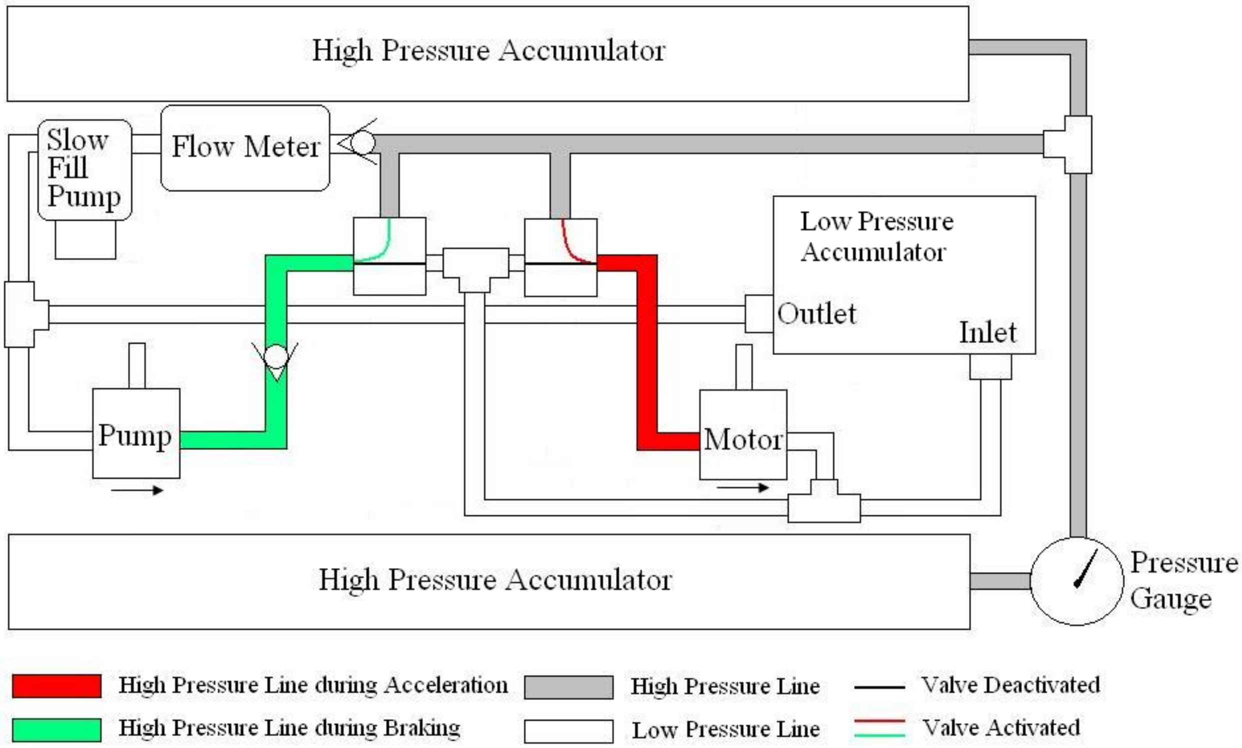


Figure 6: Schematic of Hydraulic Plumbing

The final hydraulic layout can be seen in Figure 7 on p. 14. "A" marks the slow-fill pump, "B marks the low-side accumulator, "C" marks the high-side accumulators, "D" marks the three way valves, "E" marks the pressure gauges, "F" marks a filter for the hydraulic fluid, "G" marks a flow meter, and "H" marks the emergency-stop valve.

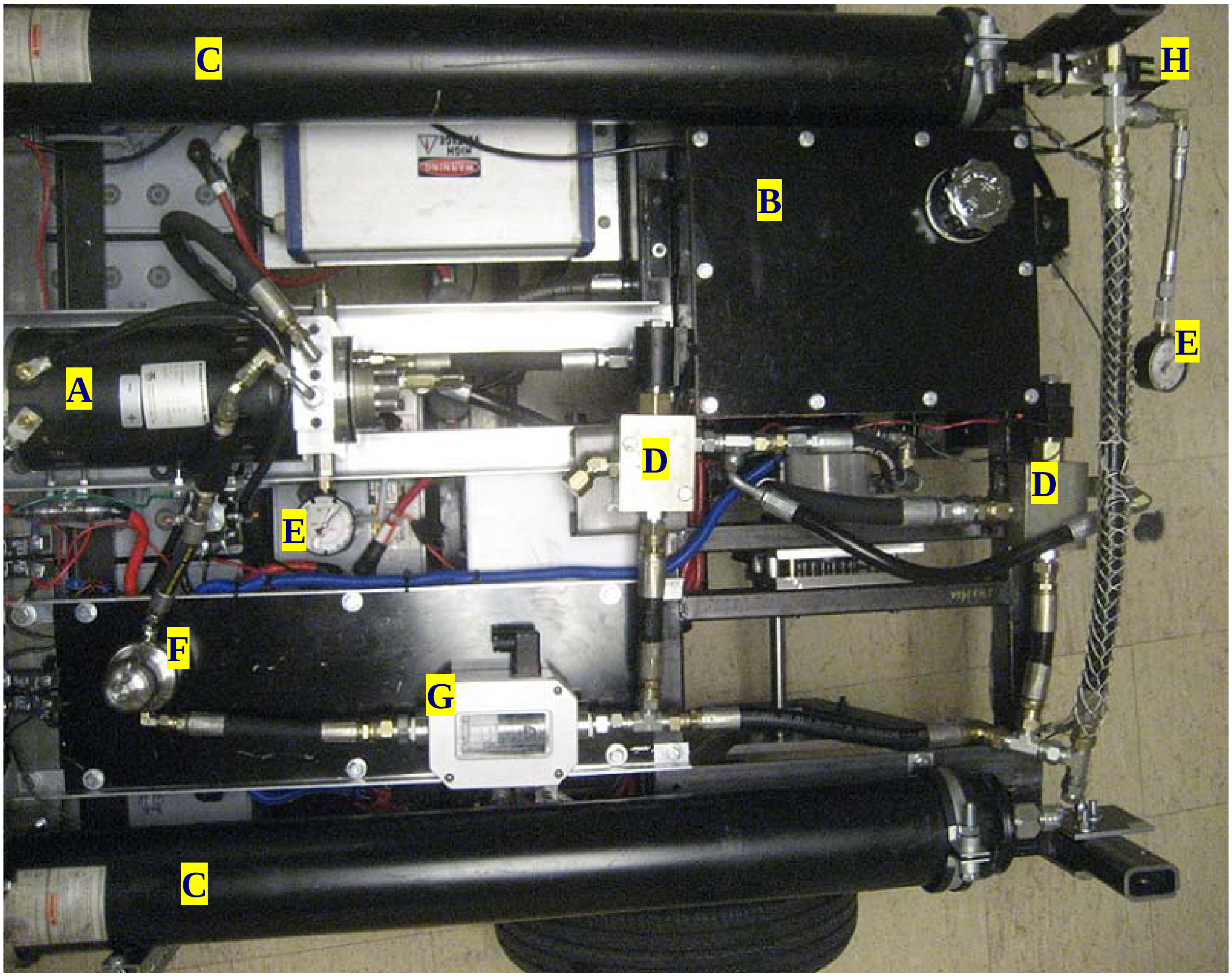


Figure 7: Final Hydraulic Layout of the Xebra

The final design assembly can be seen in Figures 8 and 9 on p. 15-16. The final concept is very similar to that of the initially selected concept, using chain and sprockets reduction type. The only deviation from the initial concept is a modified layout of several components. A one-way bearing was pressed into the small sprocket, and the large sprocket was mounted on the spline shaft. The spline shaft was mated to one of the gears in the differential.

The major modifications to the initial layout at the beginning of the semester and the initial selected concept are the relocation of the hydraulic motor, slow-fill pump, and low pressure reservoir to give enough space for the hydraulic motor and reduction box. It will also provide easier access to the motor and reduction box, which preserves the serviceability. The hydraulic motor will be mounted onto the extended frame, and the shaft of the motor will be supported by two bearings to avoid side loading on the motor shaft. Hydraulic hoses and fittings will be used to connect all of the components of the hydraulic system. The layout of all the components leaves space around the reduction box allowing future semesters to add regenerative braking to the vehicle.

The large sprocket is mounted on the spline shaft by bolting onto a flange with an internal spline, and the shaft is also be supported by two bearings to prevent failure of the bearings due to bending loads. The bearings to be used for our design will be all radial bearings, which cannot withstand bending. The chain will stand up to 1000 lb of load, which is selected with the safety factor of 2.

As mentioned in DR2, the slow-fill pump will be replaced since the old pump could only pressurize up to 1500 psi [4]. As previously mentioned, the new model, a Monarch 08174 DC motor, can pressurize up to 3500 psi with average flow rate of 0.037 L/s [5].

The hydraulic motor, high and low pressure accumulators were pre-selected by the previous group. The flow meter, pressure sensor and gauge, and oil filter were provided from the EPA and were installed during assembly.

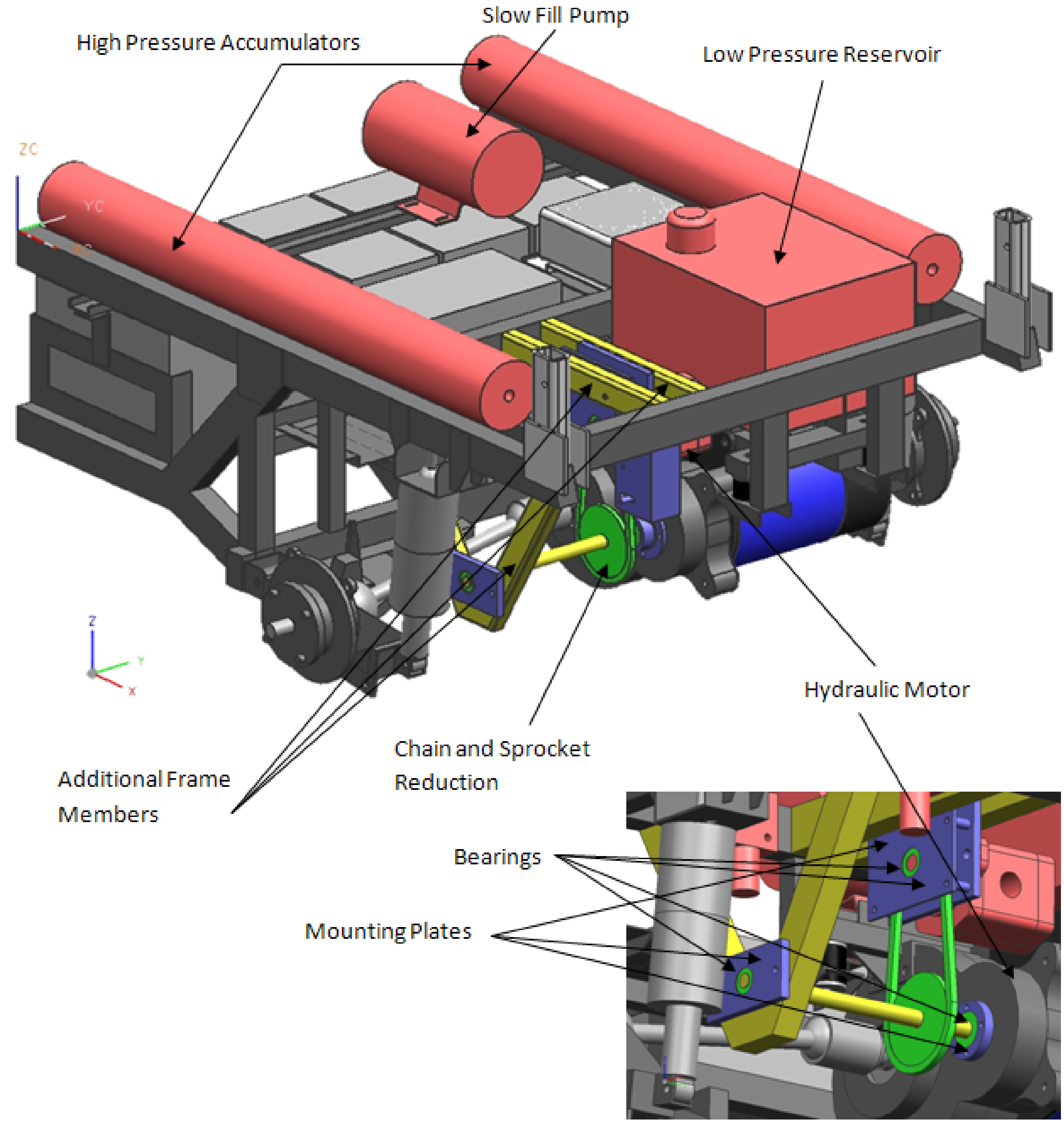


Figure 8: Final CAD Model of the Xebra's Component Layout

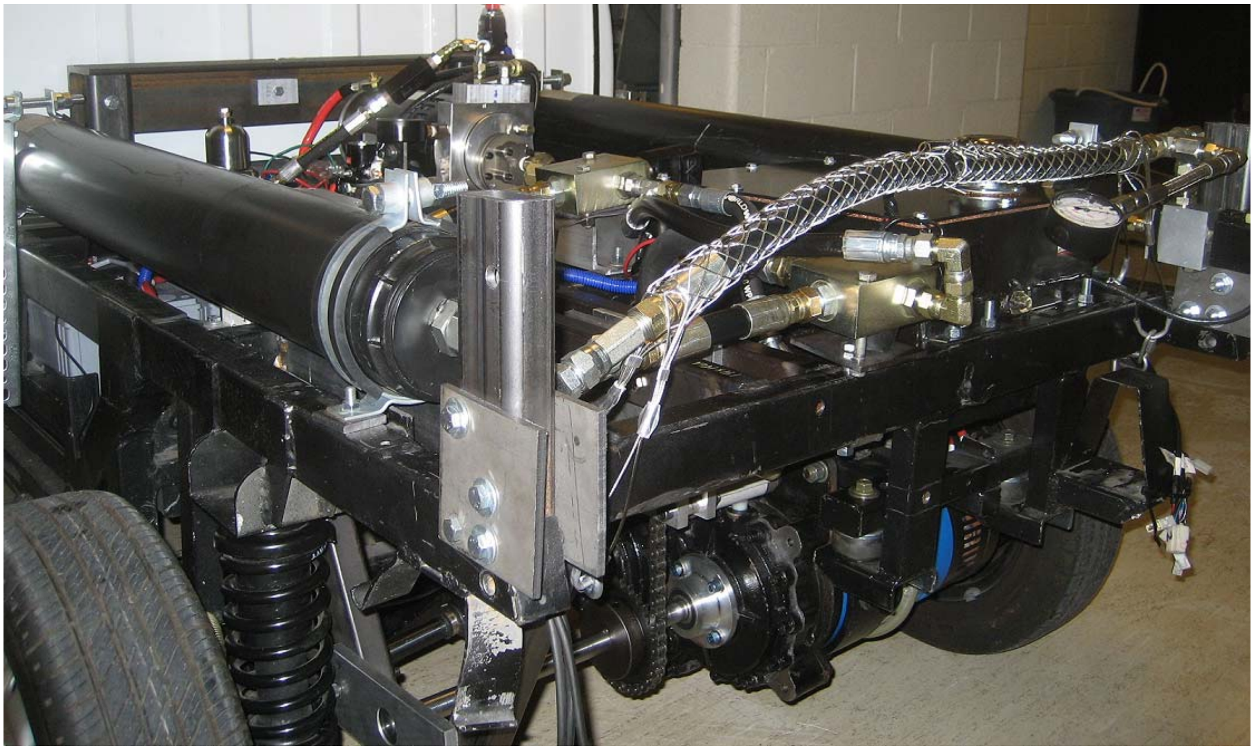


Figure 9: Final Assembled Design of the Xebra

7 FABRICATION PLAN

The entire design has been completed so that all components can be machined using the available lathes, mills, and saws. No CNC machining is required. Below is a breakdown of the particular machining concerns for each major component of the final design. A complete parts list can be found in Appendix A.

7.1 Driveshaft

The driveshaft is the most complex part of the finalized design. The shaft is 17” in total length, and has two bearings pressed onto it for support. Due to the difficulty of supporting the center section of such a long shaft during lathing, all bearings were designed to be pressed onto the ends of the shafts, allowing tolerances of less than 0.001” for bearing seats. Additionally, the shaft is mated to an existing spline in the gearbox, and has 2 keyways to transmit torque from the hydraulic system. The driveshaft required 7 minutes on a manual lathe at 1700 RPM and a feed of 0.005” per revolution feed. Also, 5 minutes on a manual mill were required to cut the keyways.

7.2 Driveshaft Bearing Supports

The bearing supports consist of two pieces, a support and a plate. The supports were cut on a band saw according to CAD. Then, the supports were tacked together, and onto the existing frame, prior to fully welding them. This reduced the gaps between the components, allowing for easier manufacturability. The plate is made of 6061 aluminum and was made in one milling operation. The only critical dimension is that of the bearing bore. For assembly, the plate was located by truing the driveshaft to the existing gearbox to minimize oscillations. Then, the mounting holes were drilled to ensure they are placed correctly relative to the gearbox. The plate required 15 minutes to machine on a manual lathe at 2000 RPM at a feed of 22 inches per minute.

7.3 Driveshaft Gearbox Coupler

The driveshaft-gearbox coupler is made of 7075-T6 aluminum. It was lathed and parted to the correct length, with the critical dimension once again being the bearing bore. After lathing, a bolt circle was milled to match the one existing on the gearbox. The part required 8 minutes on a manual lathe at 2000 RPM and 15 inches per minute feed. Also, two minutes on a mill were required to drill the bolt circle.

7.4 Sprockets

The 18 tooth sprocket was turned to length and bored to the correct inner diameter to house the one-way locking bearing. This was done at 625 RPM and .004" per revolution feed, and took 10 minutes.

7.5 Hydraulic pump mounting plates

Similar to the driveshaft bearing plate, these two plates each house a bearing and are made of aluminum. Each required one milling operation. These were completed at the same feeds and speeds as the driveshaft bearing supports, and required 12 minutes each. Additionally, 4 aluminum spacers were fabricated on the lathe to ensure that the two plates are held parallel to one another, eliminating side load on the hydraulic pump. These each took 1 minute at 600 RPM, as the only needed to be drilled and parted to length.

7.6 Assembly

The design allows for all bearings to be installed on a press, prior to locating the sub-assemblies onto the vehicle. Once the hydraulic motor assembly and the driveshaft assembly were installed onto the vehicle, the connecting link of the chain was secured, and installation was complete.

7.7 Mass production considerations

For mass production, a few components would be constructed in different ways. First, all aluminum plates would be cast, and bearing bores would be final machined. Second, jigs would be made in order to place all supports that were welded to the existing frame. Third, the sprockets would be custom ordered to the correct inner diameter and length. Finally, a spline adapter would be utilized to match the existing spline in the gearbox, rather than having each driveshaft custom ground.

8 VALIDATION PLAN

With the main goal for this semester being to utilize a hydraulic motor to accelerate our vehicle, the validation plan will be simple. Upon completion of fabrication and reassembly of the components, we will test the vehicle to insure that all of the components are working properly. First, we will reattach the battery terminals in order to give power to the vehicle. Next, we will check that all of the components are receiving power when they are supposed to. For the valves that feed the pump and motor, we will make sure that one is turned on when the accelerator is engaged, and the other is turned on when the brake pedal is engaged. Also, we will make sure that the slow fill pump receives power when the brake pedal is engaged. Once we are assured that everything is wired correctly, we will check to make sure that the hydraulics are being used to accelerate the car. This will be done by activating the pressure sensor on the accelerator without actually engaging the pedal. If the tires spin, we are assured that the hydraulic system is indeed working. As a backup, we will take readings from the hydraulic pressure gauges and flow meter to verify that there is hydraulic fluid moving through the system.

Since this project spans over many semesters, we are unable to validate if the acceleration requirements have indeed been met. Once the regenerative braking is implemented in future semesters, the car will be sent out to be run on an EPA HOT505 test. The results from this test will then be compared to the baseline testing that was done in previous semesters. From these results, they will be able to determine whether or not the overall design specifications were met. They will be able to calculate acceleration, energy usage, power, etc. Please see section 10 for recommendations on validation testing.

9 DISCUSSION

The selection of the clutch, reduction type, and reduction location were chosen very carefully. For the scope and focus of our project, all selections proved to work well for our system.

After re-working the layout of the hydraulic components, we feel that we have more efficiently utilized the system's components and constraints. During the testing phase of the hydraulics, a few modifications were necessary. This forced us to use several fittings and valves that were not originally installed by the Hose Doctor, causing the hydraulic plumbing to be slightly less efficient. The Hose Doctor is a representative from Exotic Automation. He can be reached through Exotic Automation and Supply (248-477-2122). The next team should have the Hose Doctor re-install pipes that do not fit quiet as well due to the addition of the two check valves. The Hose Doctor should also provide more efficient fittings where our team had to use several connected fittings in series or fittings smaller than that desired due to limited time and resources. A few examples of this would be the fittings coming out of the slow-fill pump to the filter, which should be replaced with a longer hose and fewer fittings, and the emergency shutoff valve at the high pressure accumulator, which should be larger and duplicated on the other high side. Future teams should ensure that all hydraulic lines installed are rated at a sufficient pressure (note that not all lines experience high pressure). It is also recommended that all fittings are JIC with swivel head fittings.

Regarding safety, our team ensured that the hydraulic plumbing was properly installed to avoid any broken lines, fittings, etc. Our team also attached cable support grips to the hydraulic lines that experience high pressure to prevent these lines from being a potential danger if they were to break. Another safety addition to the Xebra was that of a emergency-stop check valve for the high pressure accumulators. Pressing the button for this valve prevents the high pressure accumulators from releasing pressure. This can be utilized if a line were to break, preventing any additional pressure in the accumulators from being released. Since this emergency-stop button is located in the back of the truck, there should be an additional button located inside the cab of the Xebra so that a driver could press it without having to exit the vehicle. This additional button can be simply connected in series with the other button. The Xebra already had another emergency-stop button located under the driver's seat, which shuts off the electrical components. Note that pressing this button would also turn off the hydraulic pumps.

Our design sufficiently accounted for the future implementation of regenerative braking. In order to accomplish this, we would recommend utilizing another sprocket reduction from the driveshaft to the regenerative motor. Unlike the drive we installed, though, a one way bearing will not be appropriate, and a more sophisticated clutch mechanism will be required.

The aforementioned validation plan will provide performance data on the Xebra that can be used to determine exactly how well the hydraulics work, which can provide guidance for further modifications and additions.

10 RECOMMENDATIONS

There are several possibilities for future modifications and additions. The re-installation of several hydraulic lines and fittings and the addition of another emergency-stop were discussed in the previous section.

Instead of using two high pressure accumulators, a single large accumulator could be used. This can possibly improve the efficiency of the hydraulic plumbing; however, there are several downsides to this approach. With the two current high pressure accumulators being located on each side of the rear, their weight is balanced. One large accumulator placed on either side of the rear could shift the center of gravity in an undesirable manner. To avoid this, it could be placed toward the center of the rear; however, this will cause complications with the current layout of the hydraulic components and it may be difficult to re-arrange the components accordingly.

The possibility of using a pressurized low-side accumulator could possibly improve the performance of the system. This would be a complicated task, however, since safety controls would need to be integrated so that the low-side accumulator does not get over-pressurized.

A variable displacement hydraulic motor could pose some advantages, but the implementation of this would not be needed until a later stage. It should be noted that very low and high speeds are not desirable since they can cause leakage and volumetric loses, respectively. Also at a later stage, the design for different levels of accelerations and braking would improve the usability of the Xebra.

The need for one of the hydraulic components (pump or motor) could be eliminated by utilizing a four-way valve. This would also improve the efficiency of the plumbing by removing a pump/motor and all of the associated plumbing complications associated with it. These hydraulic components can act as both a pump and a motor. There are a couple issues with this design, though. In typical city driving, a person may need to switch from pressing the accelerator to pressing the brake. If the component being used as both a pump and a motor were to lock in the sense that its function did not switch, this situation could be hazardous. The pump is actuated by the brake and the motor is actuated by the accelerator. The driver could potentially need to brake and the motor may continue to drive the vehicle. In our current design, the locking of the motor is accounted for. The combination of the slow-fill pump and the regenerative brake pump will overpower the motor in this case, ensuring that the vehicle will decelerate if the brake is applied. The other major issue with this design is the need for controls to actuate the four-way valve.

As discussed in the Validation Plan section, the Xebra's performance needs to be determined. The preliminary baseline performance test was performed at Lotus Engineering, therefore it is important to perform the final test here as well. You can contact Don Apple or Pat Barker for more information (734-995-2544). To quantify the improvements of the Xebra, perform as many HOT505 city driving tests that the vehicle can handle. Also, a current and voltage reader will need to be provided for the data analysis.

Before the baseline test can be performed, it is necessary to complete a coast-down test. The data from this will be used on the dynamometer at Lotus to provide more realistic data. The first test was performed at the Chrysler Proving Grounds with the help of Larry Webster from Car and Driver magazine. You will need to run the tests according the SAE Standard J2263, however, do not warm up the vehicle before the tests and take the data starting at the maximum speed of the Xebra vehicle. Note that the speedometer reads the top line in kilometers per hour.

11 CONCLUSIONS

This semester our group has successfully added a hydraulic drive to the electric Xebra vehicle. While this satisfies the short term goals of our sponsor, there remains much work to do. Among the tasks for future terms, a regenerative braking system should be added, the plumbing should be redone to minimize pressure drops and increase efficiency, and further measures should be taken to develop the controls.

In our implementation we have left room for a regenerative hydraulic pump to be mounted, using the same driveshaft that we installed for acceleration. In order to accomplish this, we would recommend utilizing another sprocket reduction from the driveshaft to the regenerative motor. Unlike the drive we installed, though, a one way bearing will not be appropriate, and a more sophisticated clutch mechanism will be required.

12 ACKNOWLEDGEMENTS

There are many people and organizations that we would like to thank. First, we would like to thank our two major sponsors: the University of Michigan Mechanical Engineering Department and the United States Environmental Protection Agency. From the Mechanical Engineering Department, we would like to thank Professor Steven Skerlos, Professor Albert Shih, Professor Guru Dinda, Professor Jyotirmoy Mazumder for their guidance and support. Also from the University of Michigan, we would like to thank Bob Coury and Marvin Cressey of the GG Brown Machine Shop, and Jessica Boria, Christine Vladu, and Mark Scott for their efforts toward planning the Design Expo. From the EPA, we would like to thank our sponsor David Swain, who was in constant contact with our team throughout the semester providing the vision for our project along with supervision, support, and necessary resources.

We would also like to give a thanks to the companies that our team and the last team had worked with: Ann Arbor Machine; ASAP Source; Batteries Plus; Butech; DC Hydraulics, Inc.; ETI; Exotic Automation; Federal Fluid Power; Ford Motor Company; Golden Customs; Great Lakes Cycling & Fitness; Hoffman Enclosures; Hydac; Light Electric Vehicle Technologies, Inc.; Lotus Engineering; Marzocchi USA; McAuliffe Co.; Parker Hydraulics; Rockwell Automation; Sem Cycle; Shupan Aluminum; Tsubaki; UM IP Technology Transfer Dept.; and ZAP.

13 INFORMATION SOURCES

This section will discuss the information collected during research of similar systems and previous ME 450 projects at the University of Michigan. There were no relevant or useful articles found when conducting a literature search.

13.1 Technical Benchmarks

Hydraulic-electric hybrid systems are still an unfamiliar concept on today's market, and there are currently no vehicles using this system. Therefore, instead of benchmarking this system against other products, we will benchmark against the technical specifications of this project. To improve the efficiency of the current electric system on Xebra, hydraulic motor will accelerate the vehicle from 0 to 27mph and then electric motor will power the vehicle during higher accelerations and cruising.

13.2 Patent Search

The regenerative braking technology applied to the ME 450 bicycle project has been filed as a patent (Patent #. 20070126284) by the University of Michigan in December 2006. We have also received information from the ME 450 Xebra team from Fall 2007 term. This information includes calculations, a list of materials purchased, performance data from the HOT505 test (EPA standard test that simulates city driving), and several other resources that we may utilize in the future.

In addition to the University of Michigan's regenerative braking patent, there also are some other patents related to our project:

- Hydraulically driven electrically powered vehicle with energy, 1991 (Patent #. 5064013):

  This system generates electric energy with a generator linked to a hydraulic rotor during braking of the vehicle. That energy is captured in a storage device and is used to drive the vehicle. This system shares the concept of capturing the kinetic energy of the vehicle and using it to drive the vehicle. However, this system stores the energy captured during the deceleration in the form of electric energy, but our system stores energy in a high pressure accumulator. Also, our system has a hydraulic motor to deliver the power to the vehicle not the electric motor. [2]

- Hydraulic assist propulsion apparatus for vehicle drive, 1996 (Patent #. 5542335):

  An electric motor driven hydraulic pump feeds the hydraulic fluid to a piston cylinder under pressure. Then a control system energizes the motion of a piston to a shaft providing power to drive the vehicle. This system is very similar to the system mounted on Xebra, but our system will also have the regenerative braking system to pressurize the hydraulic fluid. [3]

13.3 Information Gaps

The greatest lack of useful information for our system comes with the fact that there are currently no products on the market that utilize hydraulics and electric motors in the same fashion as we plan to. At the completion of Design Review 1, we were lacking information regarding supplied parts and components. Since then, we have acquired necessary information on many of the components supplied to us at the beginning of the term. The most notable information is technical and performance information for our hydraulic motor [6]. We also acquired information regarding the slow-fill pump supplied to us, leading us to discover that it was not adequate for our application.

14 REFERENCES

[1] http://www.epa.gov/

[2] http://www.google.com/patents?id=wuAgAAAAEBAJ&dq=hydraulic-electric%2bvehicle

[3] http://www.google.com/patents?id=LkkkAAAAEBAJ&dq=hydraulic+assist+propulsion+apparatus

[4] http://www.marzocchi.com/pompe/pompe.asp?LN=UK&sito=usa-pompe

[5] http://www.monarchhyd.com/index.htm

[6] http://www.parker.com/

[7] Bosch. *Automotive Handbook 4th Edition.* SAE, 1996

15 TEAMMATE BIOGRAPHIES

Doug Lambert

I am originally from Novi, MI, where I was on my high school's FIRST robotics team. After coming to UM, I joined the Solar Car team and have won a national championship with them and placed third and seventh in the World Solar Challenge in Australia. I'm hoping to graduate next December, and work on hybrid vehicles for whatever company will take me.

Rob Lawrance

I am a senior engineer at the University of Michigan. I will graduate in December 2008 with two BSE's: Mechanical Engineering and Industrial and Operations Engineering. I grew up in Livonia, MI attending both public and private schools before coming to U of M. I love trying and learning about new things. I have competitively played many sports, with my favorites being ice hockey and football. I was a member of two Division I state championship high school football teams. I spent the summer of 2006 studying abroad in Shanghai, China. I took two engineering courses and also served a one month internship for SKF. In the summer of 2007 I interned for Caterpillar in Peoria, IL. During my internship at Caterpillar, I received training as a Six Sigma Green Belt. I currently work part-time as a Peer Advisor for the Engineering Career Resource Center. I have little experience working with cars. I selected this project because, as I had mentioned before, I love trying new things.

Hyo Jong Lee (Jimmy)

I was born and raised in Seoul, Korea. I came to the US about 5 years ago and went to community college in California. I transferred to University of Michigan in Fall 2006. I am 4th year senior in mechanical engineering, and I am planning to start graduate study from this Fall at U of M. I served as a Marine in Korea for 26 months and I worked in my dad's iron casting company for a year. I also worked in a company making tuning parts for radio control cars in Korea. I am good at calculations/analysis and fabrication (not very good, though). I am fascinated in cars, so I like to work on learn about cars. I like fishing and playing with my R/C cars. I also like martial arts, and I used to be a Taekwondo and Hapkido instructor. And I am married.

Kyle Martin

I am a 5th year Senior, and 5th year member of the SAE Baja team. This is my first year as team leader. I am the shop trainer for the mill and lathe at the Wilson Student Center, and have extensive experience machining and fabricating. After spending two years as an intern at Toyota Motor Engineering and Manufacturing North America, I have accepted a full time offer at John Deere in their construction and forestry division. I have experience designing drivetrain, including a custom CVT secondary that I designed last year, along with multiple gear reduction boxes for use on the Baja car.

Ian Murphy

I am a senior at the University of Michigan. I was born and raised in New City, New York, about 40 minutes northwest of Manhattan. I plan to finish out my undergraduate career here and graduate with a bachelor's degree in Mechanical Engineering. I hope to get a job in automotive design, as I have a deep interest in cars. I also plan to get a master's in business administration at NYU. I am a volunteer fireman for the New City Fire Department and have been since I was 16. When I'm home in NY I spend a lot of my time at the firehouse, both hanging out and going on calls. Things I like to do for fun include bowling, working out, and listening to music.

Appendix A – Bill of Materials

Item Description	Supplier	Part Number	Quantity	Price (each)	Total Price
1117 low carbon steel rod, 1" diameter, 3' length	McMaster Carr	8290T183	1	$18.83	$18.83
Steel needle roller clutch, 30mm shaft diameter, 37mm OD, 20mm width	McMaster Carr	6392K32	1	$29.24	$29.24
Steel machinable-bore sprocket, for #40 chain, 1/2" pitch, 18 teeth, 5/8" min bore	McMaster Carr	6793K149	1	$12.97	$12.97
Steel ball bearing ABEC-1, open bearing #R14 for 7/8" shaft diameter, 1-7/8" OD	McMaster Carr	60355K18	2	$7.57	$15.14
Steel ball bearing ABEC-1, double sealed #R14 for 7/8" shaft diameter, 1-7/8" OD	McMaster Carr	60355K39	1	$10.43	$10.43
Steel ball bearing ABEC-1, open bearing #R12 for 3/4" shaft diameter, 1-5/8" OD	McMaster Carr	60355K38	1	$8.35	$8.35
Standard ANSI roller chain, #40, single strand, 1/2' pitch, .312" diameter, 4' length	McMaster Carr	6261K444	1	$12.28	$12.28
Connecting link for #40 standard ANSI roller chain	McMaster Carr	6261K193	2	$0.68	$1.36
Roller chain idler sprocket, steel w/ ball bearing for #40 chain, 18 teeth, 5/8" bore	McMaster Carr	6663K42	1	$21.51	$21.51
12 gauge electrical wire	Advance auto Parts	85710L11212	5	$4.88	$24.40
60 Amp fuse	Advance auto Parts	BPAGU60GP	1	$3.97	$3.97
Plastic wire-protecting tube	Advance auto Parts	86651	2	$3.77	$7.54
81 peace terminal kit	Radio Shack	6403098	1	$9.99	$9.99
Electrical tape	Radio Shack	6402375	1	$3.99	$3.99
60 Amp fuse holder	Mr.Tunes Ann Arbor	SPD5101	1	$15.95	$15.95
72 volt DC hydraulic motor	DC Hydraulics	M326-190-8174-2000	1	$595.80	$595.80
Hydraulic pressure guage	DC Hydraulics	1790	1	$10.00	$10.00
SAE 75w-90 gear oil	AutoZone	373225 15901	2	$4.79	$9.58
CV boot	AutoZone	580339 614-001	1	$11.99	$11.99
Stalube CV grease	Murray's Discount Auto Stores	15112	1	$3.99	$3.99
Automatic transmission fluid (1 gal)	Meijer	n/a	6	$10.49	$62.94
1117 steel shaft	Vertical Machining	n/a	1	$100.00	$100.00
2 way hydraulic valve	Federal Fluid Power	DDL12 / 30102359	1	n/a	n/a
Emergency shut off switch	EPA	n/a	1	n/a	n/a
6061 aluminum, 3" x 6" x 0.5"	Alro Metals Plus	n/a	1	$10.89	$10.89
6061 aluminum, 6.625" x 6.875" x 0.5"	Alro Metals Plus	n/a	2	$13.73	$27.46
6061 aluminum, 1.75" x 0.5" round	Alro Metals Plus	n/a	4	$2.19	$8.76
6061 aluminum, 0.5" x 3" round	Alro Metals Plus	n/a	1	$10.46	$10.46
1117 steel, 1" x 1.25" round	Alro Metals Plus	n/a	1	$8.69	$8.69
1117 steel (rectangular tube, 0.125 thickness), 1.5" x 2.25" x 6"	Alro Metals Plus	n/a	4	$6.97	$27.88

*Various parts and tools used for fastening devices were provided by Bob Coury's shop

** All hydraulic hoses and fittings were provided by Exotic Automation/Federal Fluid Power/ EPA

*** All other parts were either supplied to us by our sponsors or purchased in previous semestrers

Appendix B – Engineering Change Notice

There were no mechanical design modifications made to our original design. The only modifications made were slight changes to the location of several of the hydraulic components for packaging purposes. From the system analysis of last semester and analysis done by our team, including extensive CAD modeling, calculations, and system simulation modeling in Simulink, our alpha design did not require any modifications.

Appendix C.1 – Material Selection

Shaft:

Function:	Transmit torque
Objective:	Durable, able to transmit needed torque
Constraints:	Torque, price, length, diameter
Material Indices:	$r = 0.011\ m$
	$J = \frac{\pi}{2}(r^4) = 2.40\ \times 10^8\ m^4$
	$\tau = \frac{Tr}{J} = 924\ MPa$
	$\sigma = \sqrt{3} \cdot \tau = 277\ MPa$
Top Materials:	All carbon steels:

- 1015
- 1020
- 1117
- 1022
- 1118

We chose to use 1117 Carbon Steel for our shaft. This decision was made due to the fact that all of the top materials listed above are very similar, and 1117 was available in the size and quantity needed for the shaft. Also, although price was one of our constraints, all of the materials cost about the same price per pound so this didn't factor into our final selection.

Motor Mount:

Function:	Support motor weight and hold bearing
Objective:	Easily machineable, minimal mass
Constraints:	Thickness, length, width
Top Materials:	Cast magnesium alloy AS21
	Wrought magnesium alloy Z6
	Wrought aluminum alloy 6061
	Wrought aluminum alloy 2024
	Cast aluminum alloy A356

We chose to use wrought aluminum alloy 6061 for our motor mount. This decision was made based on the availability of the material, as well as the ease of machineability. Also, for safety issues, we did not use magnesium as it is highly flammable and requires special fire extinguishers if it catches on fire.

Appendix C.2 – Design for Assembly

A. Original Design had 32 % Design of efficiency

1	2	3	4	5	6	7	8	9	Name of Assembly
Parts ID No.	Resistance to insertion	two degit manual handling code	Manual handling time per part	two digit manual insertion code	Manual insertion time per part	Operation time in seconds	Operation cost	Minimum number of parts needed	Motor mount (reduction box)
10	1	20	2	2	5	12	4.8	1	Motor mount plate
9	0	5	1	4	3	13	5.2	4	Spacers
8	1	20	2	2	5	12	4.8	1	Shaft supporting plate
7	2	7	1.2	12	8	36	14	6	Screws
6	1.5	15	1.8	0	10	10	4	2	Mount bars
5	0.5	11	1.4	10	6	8	3.2	1	Sprocket without hub
4	0	1	0.8	8	2	8	3.2	4	Washers
3	2	13	0.9	14	15	60	17	4	bolts and nuts
2	0.5	17	1.6	14	5	50	20	1	Flange for the sprocket
1	2.5	3	1.1	6	15	30	12	2	Bearings
									Design of Efficiency
						219	77	29	0.32

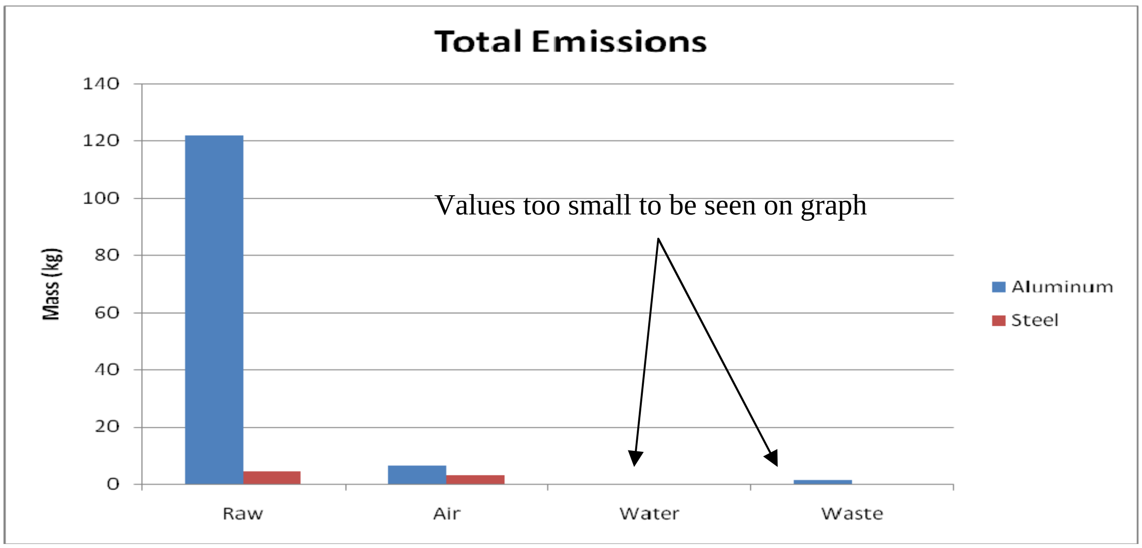
B. New Design has 49% Design of efficiency

1	2	3	4	5	6	7	8	9	Name of Assembly
Parts ID No.	Resistance to insertion	two degit manual handling code	Manual handling time per part	two digit manual insertion code	Manual insertion time per part	Operation time in seconds	Operation cost	Minimum number of parts needed	Motor mount (reduction box)
10	1	20	2	2	5	12	4.8	1	Motor mount plate
9	0	5	1	4	3	13	5.2	4	Spacers
8	1	20	2	2	5	12	4.8	1	Shaft supporting plate
7	2	7	1.2	12	8	36	14	6	Screws
6	1.5	15	1.8	0	10	10	4	2	Mount bars
5	0.5	11	1.6	10	6	8	3.2	1	A type Sprocket with hub
4	0	1	0.8	8	2	8	3.2	4	Washers
1	2.5	3	1.1	6	15	30	12	2	Bearings
									Design of Efficiency
						129	52	21	0.49

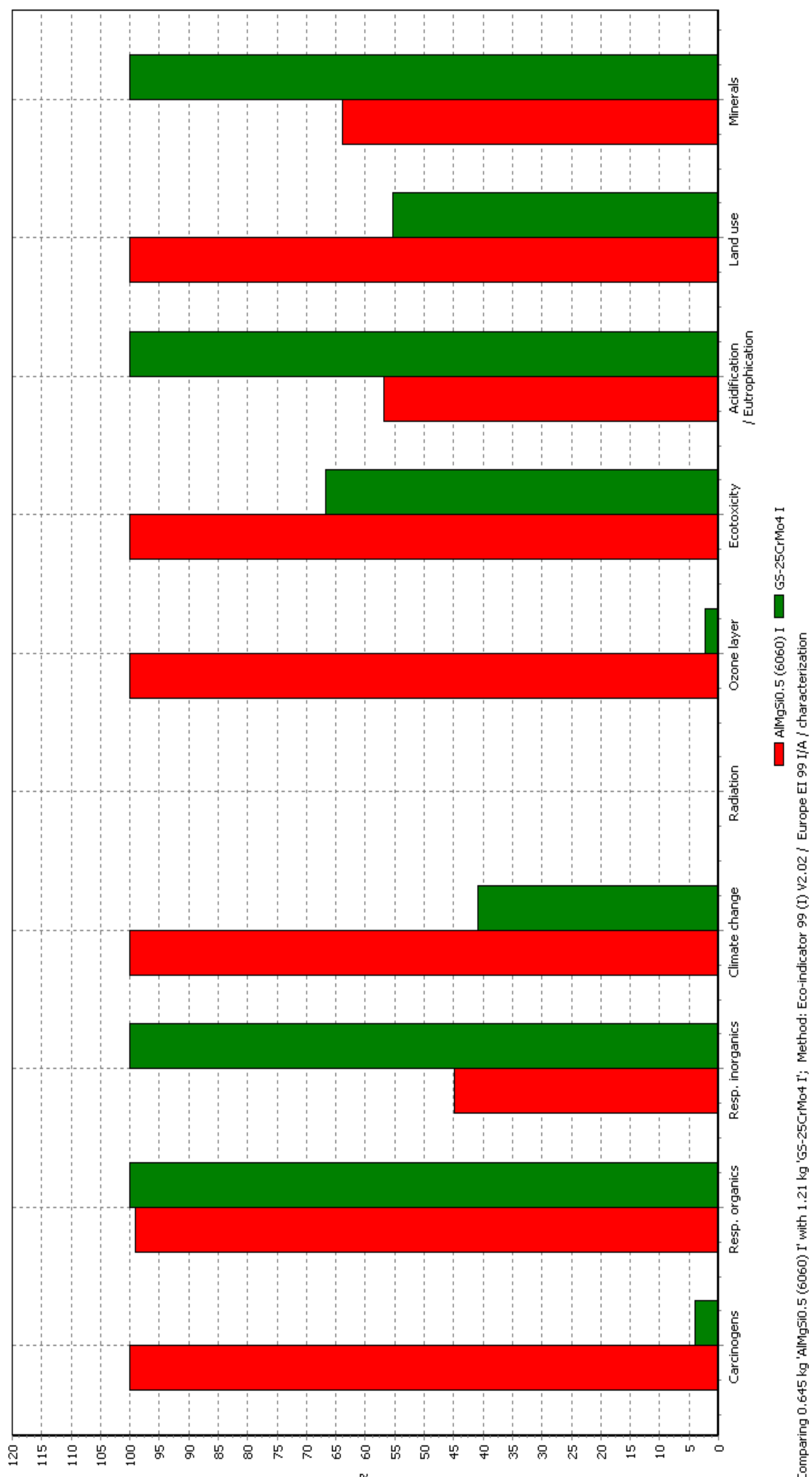
Appendix C.3 – Design for Environmental Sustainability

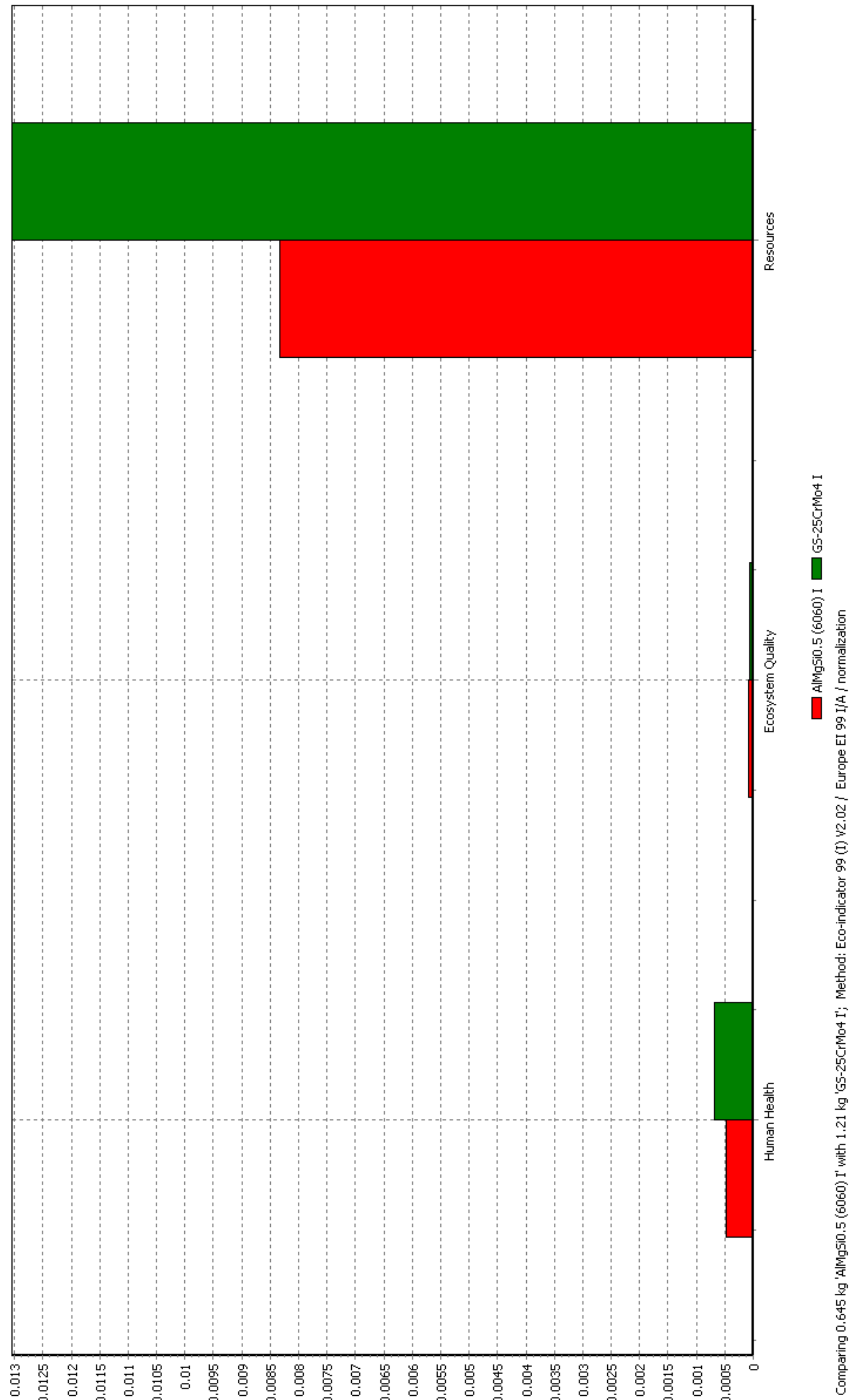
Total Emissions:

Material	Mass (kg)	Raw Emissions (kg)	Air Emissions (kg)	Water Emissions (kg)	Waste Emissions (kg)
6061 Aluminum	0.645	121.8	6.275	0.09	1.5
1117 Steel	1.209	4.6	3.4	0.001	0.015

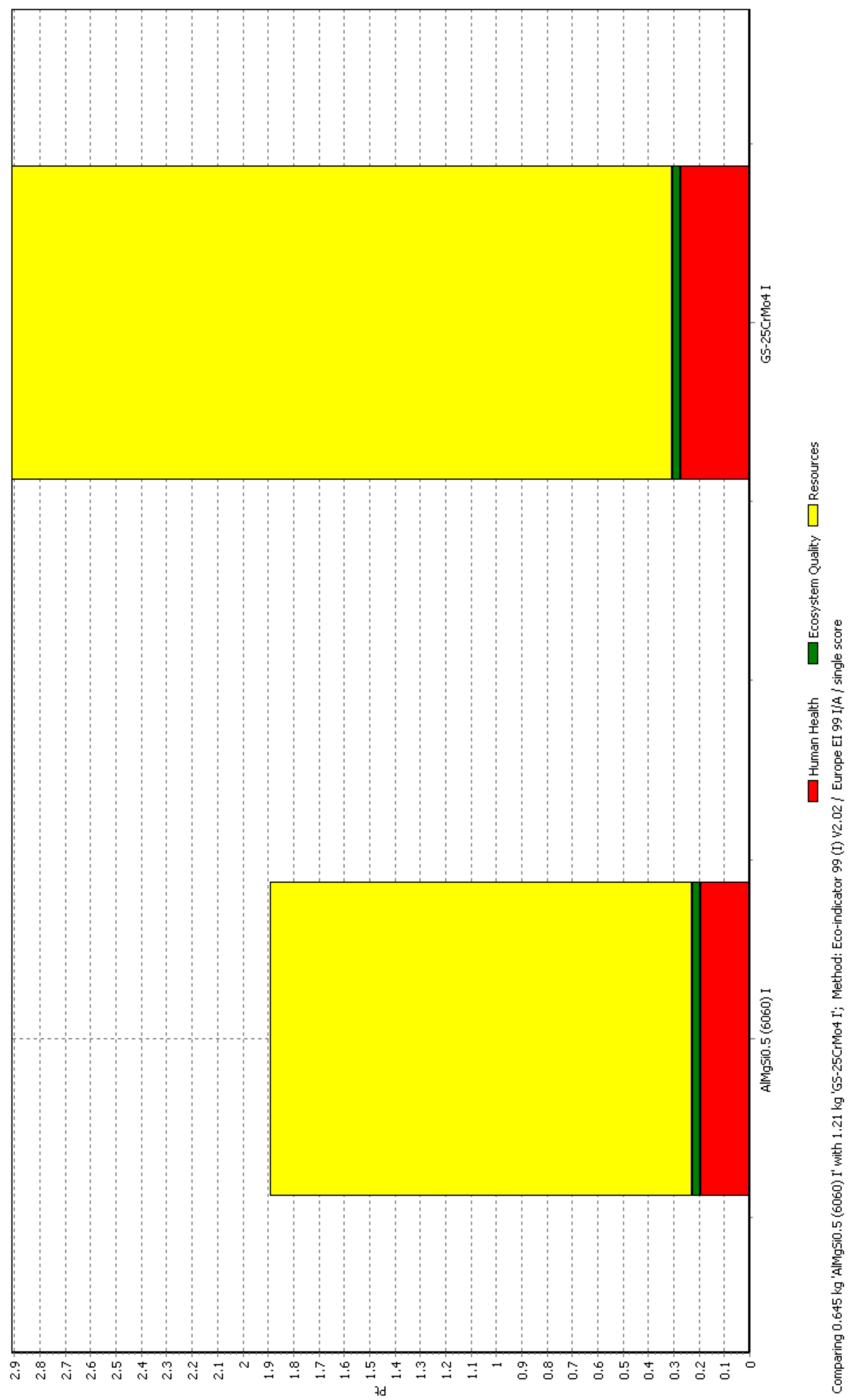


EcoIndicator 99 Damage Classifications:





0.013
0.0125
0.012
0.0115
0.011
0.0105
0.01
0.0095
0.009
0.0085
0.008
0.0075
0.007
0.0065
0.006
0.0055
0.005
0.0045
0.004
0.0035
0.003
0.0025
0.002
0.0015
0.001
0.0005
0
Human Health
Ecosystem Quality
Resources
AlMgSi0.5 (6060) I
G5-25CrMo4 I
Comparing 0.645 kg 'AlMgSi0.5 (6060) I' with 1.21 kg 'GS-25CrMo4 I'; Method: Eco-indicator 99 (I) V2.02 / Europe EI 99 I/A / normalization



Based on the graph shown above, the aluminum appears to have more impact on the environment than the steel. However, steel gets higher ecoindicator 99 point values, so this is likely to have a bigger impact when the life cycle of the whole product is considered.

Appendix C.4 – Design for Safety

Risk	Electrocution	Sharp edges	High pressure fluid	Projectile fitting	Bed falling	Throttle stuck	Chain
Severity	1	2	5	5	3	4	5
Liklihood	4	4	1	1	3	2	2
Score	**4**	**8**	**5**	**5**	**9**	**8**	**10**

The most major risks are sharp edges within the system causing cuts, the hydraulic motor switch becoming stuck, the bed of the truck falling on someone during maintenance, and someone getting caught in the chain. For the most part, these dangers apply to people doing maintenance on the vehicle, with bystanders and the driver being relatively safe.

The results of an analysis using DesignSafe software is shown on the next two pages. While none of the hazards found in this analysis are surprising, it is much more detailed than the Risk Analysis performed above.

The difference between Risk Analysis and FMEA is that Risk Analysis is more of a brainstorming based, subjective method of discovering all of the possible hazards inherent in a system. FMEA on the other hand is much more formalized and attempts to produce a quantitative output of what is most likely to fail, how it will happen, and how bad the results will be.

Zero risk does not exist in the real world. Acceptable risk is as low a risk as is reasonably possible. This plays into our project as displayed in the DesignSafe analysis. All of our risks are low or moderate, but given the tasks our system has to perform the hazards themselves are inevitable. All action possible has been taken to minimize the likelihood and severity of these risks, but they can never be completely eliminated.

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

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level
All Users All Tasks	mechanical : crushing When bending over the car, the user could be injured by the bed falling due to improper restraint of the bed.	Slight Occasional Possible	Moderate
All Users All Tasks	mechanical : cutting / severing When maintaining the vehicle, the user could be injured by sharp edges due to contact.	Minimal Frequent Possible	Moderate
All Users All Tasks	mechanical : drawing-in / trapping / entanglement When working near the driveshaft, the user could be injured by the chain and sprocket due to loose clothing and unexpected starting.	Catastrophic Remote Unlikely	Moderate
All Users All Tasks	mechanical : unexpected start When working around the vehicle, the user could be injured by the vehicle moving due to an unexpected start.	Serious Remote Negligible	Low
All Users All Tasks	mechanical : head bump on overhead objects When working on the vehicle, the user may be injured by frame members due to inattention.	Slight Occasional Possible	Moderate
All Users All Tasks	mechanical : impact When driving the vehicle, the driver may be injured by an impact due to crashing.	Serious Remote Unlikely	Moderate

Figure C.4.1 – DesignSafe Analysis for the Xebra

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level
All Users All Tasks	electrical / electronic : energized equipment / live parts When connecting the electronics the user could be injured by electrocution due to the 72V batteries.	Minimal Occasional Possible	Moderate
All Users All Tasks	slips / trips / falls : slip When walking near the vehicle, the user may be injured by slipping due to hydraulic oil spills.	Slight Occasional Unlikely	Moderate
All Users All Tasks	fluid / pressure : hydraulics rupture When working in the rear of the vehicle, the user may be injured by small hydraulic lines rupturing due to high pressure.	Serious Remote Negligible	Low
All Users All Tasks	fluid / pressure : explosion / implosion When near the vehicle, the user could be injured by the explosion of fittings due to improperly tightened fittings.	Catastrophic Remote Negligible	Moderate
All Users All Tasks	fluid / pressure : fluid leakage / ejection When near the vehicle, the user may be injured by high pressure fluid due to leaks in the lines.	Serious Remote Negligible	Low

Figure C.4.2 – Continued DesignSafe Analysis for the Xebra

Appendix C.5 – Manufacturing Process Selection

This project can be applied to any motorized vehicle. Therefore, our potential real world production volume can be hundreds of thousands. However, there can be many other professors and students who try similar projects. Therefore, we decided out production volume as 1000.

Both shaft and motor mount can be manufactured by machining due to its tolerances. Because of the spline mating and shaft alignment, the shaft has very small tolerances in its dimension. In addition, since the bearing is pressed into the motor mount, the dimensions are critical in the motor mount. Therefore, these parts have to be machined with mill and lathe.

Milling techniques.

Shape

Circular prismatic	✓
Non-circular prismatic	✓
Flat sheet	✓
Dished sheet	✓
Solid 3-D	✓

Physical attributes

Mass range	2.205e-3	-	2205	lb
Tolerance	7.874e-4	-	0.01969	in
Roughness	0.03937	-	0.9843	mil

Process characteristics

Machining processes	✓
Discrete	✓
Prototyping	✓

Economic attributes

Relative tooling cost	low		
Relative equipment cost	high		
Labor intensity	medium		
Economic batch size (units)	1	-	1e7

Turning, boring and parting operations performed on a lathe.

Shape

Circular prismatic	✓
Solid 3-D	✓
Hollow 3-D	✓

Physical attributes

Mass range	2.205e-3	-	1.213e5	lb
Tolerance	5.118e-4	-	0.01496	in
Roughness	0.01969	-	0.9843	mil

Process characteristics

Machining processes	✓
Cutting processes	✓
Discrete	✓
Prototyping	✓

Economic attributes

Relative tooling cost	medium		
Relative equipment cost	high		
Labor intensity	medium		
Economic batch size (units)	1	-	1e7

Appendix D – Quality Function Diagram (QFD)

Flow-rate of slow-fill pump											
Gear ratio from the motor and pump to drive shaft											
High-side accumulator pressure	9										
Side loading on pump											
Side loading on motor				9							
Strength of material for gears		9									
Pressure losses through geometry of fittings	9										
Working temperature of the fluid	9										
Size of hydraulic fittings			3				9				
Space for all the components (Layout)		3									
Compatible components											

Technical Requirements

Customer Needs	Customer Weights	Flow-rate of slow-fill pump	Gear ratio from the motor and pump to drive shaft	High-side accumulator pressure	Side loading on pump	Side loading on motor	Strength of material for gears	Pressure losses through geometry of fittings	Working temperature of the fluid	Size of hydraulic fittings	Space for all the components (Layout)	Compatible components	Relative Weight	Rank
Transferrable to future semesters	11	3	3	3	3	3	3	3		3	9	3	10.2%	3
Comfortablity during acceleration	2		9	9						3			0.9%	12
Sufficient acceleration to top speed	14	9	3	9					3	3	1		10.2%	4
Efficiency in plumbing	8	1		3				9	9	9	9		8.1%	6
Lightweight	2	3	9		3	3	9			1		1	1.2%	11
Reliablity of components	6	1	3	3	9	9	9	3		1	1	9	7.8%	5
Aesthetics	11		3					1		1	9	3	4.8%	8
Stability	3	3	9	3	9	9	9	1		1	3	3	4.1%	9
Safety	17	3	3	9	9	9	9	1	3	3	3	9	27.3%	1
Easy to use	5	1		3			3				3	3	1.6%	10
Easy to service	9	1		3	3	3	3	1			9	1	5.9%	7
Maintains Vehicle function	15	1		3	9	9	9		9		3	1	17.9%	2
	Relative Weight	7.0%	6.1%	12.2%	11.6%	11.6%	12.2%	4.2%	7.9%	5.8%	12.9%	8.5%		
	Rank	8	9	2	5	4	3	11	7	10	1	6		
	Units	L/s	n/a	Mpa	N	N	Mpa	%	K	mm	m³	Y/N		
	Values	.04-.05	7.9 to 1	15-30	0-5	0-5	400-1000	0-20	275-325	8 - 24	0.2-0.4	Y/N		

Appendix E – Preliminary Calculations

- The vehicle is 1800 lbs unloaded. With the estimated 500 lbs payload, the vehicle is 2300 lbs (1043 kg). The energy required to accelerate the vehicle from 0 to 27 mph (12.1m/s). Using the kinetic energy equation (7)

$$E = \frac{1}{2} \cdot M \cdot v^2 \qquad \text{Eq. 7}$$

substituting the mass and velocity yields E = 76.4 kJ.

- The volume of fluid at 3000 psi (20700 kPa) required for the acceleration was calculated using Bernoulli equation (8) for incompressible fluid

$$\frac{P_1}{\rho_1 g} + \frac{v_1^2}{2 \cdot g} + Z_1 = \frac{P_2}{\rho_2 g} + \frac{v_2^2}{2 \cdot g} + Z_2 \qquad \text{Eq. 8}$$

With $V_1=0$, $Z_1 = 0$, and $Z_2 = 0$, $\frac{P_1 - P_2}{\rho} = \frac{v_2^2}{2}$ was obtained.

Since $\rho = M/V$; where V is volume. The equation becomes: $V \cdot (P_1 - P_2) = \frac{m \cdot v_2^2}{2}$

Finally, Eq. 9 was derived.

$$V = \frac{E}{(P_1 - P_2)} \qquad \text{Eq. 9}$$

For the vehicle, we plan to use atmospheric pressure (101 kPa) at the low-end accumulator. With the energy found above, the volume of fluid needed is $0.00371 m^3$, or 3.71 liters.

- Vehicle acceleration, *a*, for a fixed displacement hydraulic motor at 23cc/rev ($2.2 \cdot 10^{-5}$ m^3)

From Eq. 4, we get $E = V \cdot (P_1 - P_2)$; thus the energy provided by the motor on a revolution was found to be 0.453 kJ.

From energy equation $E = F \cdot s$ and Newton's second law $F = Ma$, we obtained Eq. 10

$$a = \frac{E}{M \cdot s} \qquad \text{Eq. 10}$$

For one revolution of the motor, the wheels turn 1:10 revolution. As the result, the vehicle moves the distance $s = \frac{\pi D}{10}$ where *D* is the wheel's diameter. Finally, we found the vehicle acceleration, *a* = 2.72 m/s^2.

Appendix F – Brake Force Calculations

The total braking force on the rear axle can be determined by using the torque on the motor and the gear reduction through Eq. 11:

$$F = \frac{\tau}{r} \qquad \text{Eq. 11}$$

Where: τ = torque on the motor

r = radius of the larger gear

With τ = 850 in-lb and r = 3.15 in, the braking force was calculated to be 281 lbs.

The maximum brake force that can be applied to the rear wheels without lock-up is defined by Eq. 12:

$$F_{\max} = \frac{\mu \cdot F_{r,s}}{1 + \mu\left(\frac{h_{CG}}{L}\right)} \qquad \text{Eq. 12}$$

Where: μ = coefficient of friction between the tires and the road

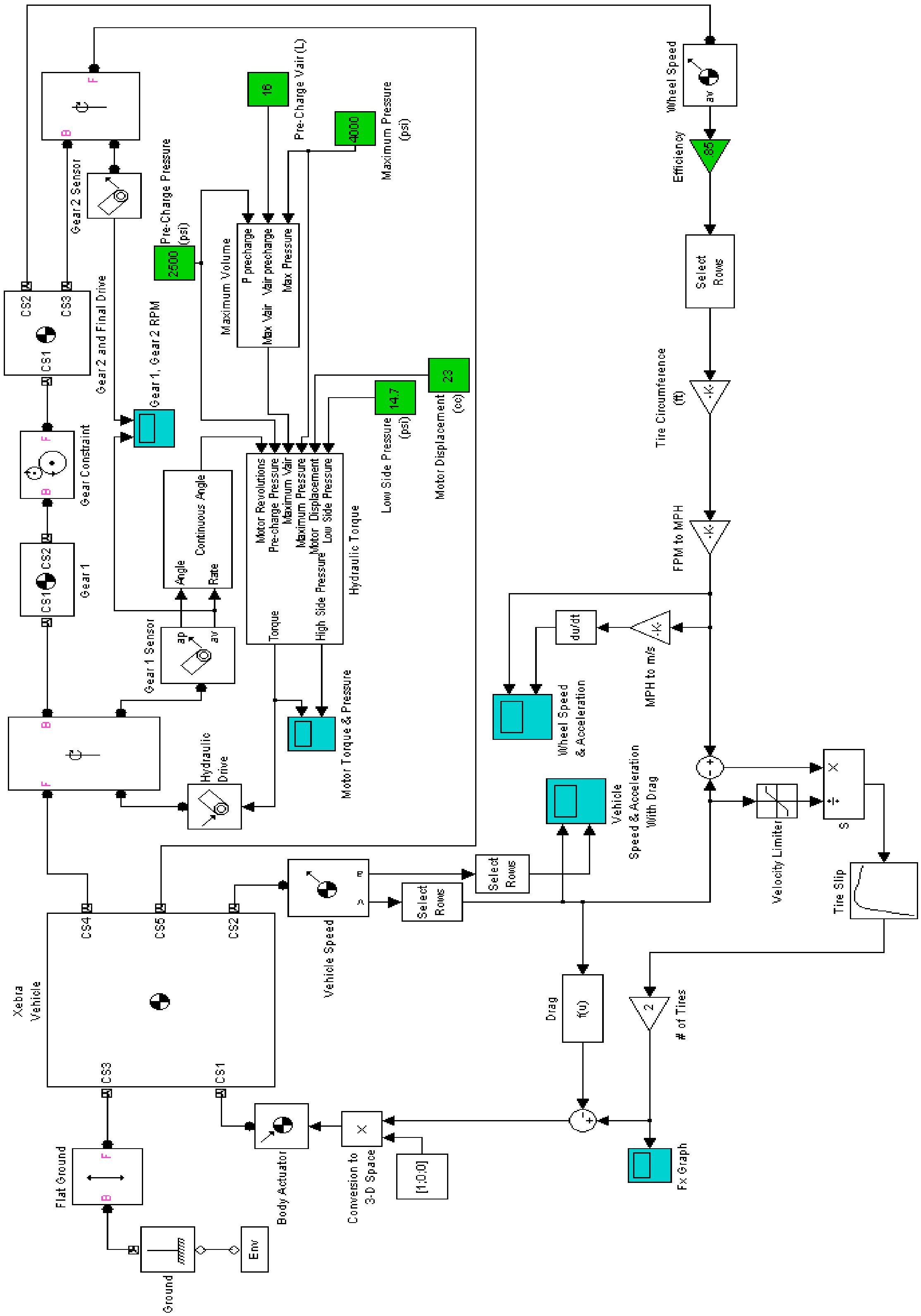
$F_{r,s}$ = static force on the rear axle

h_{CG} = vertical height of the center of gravity

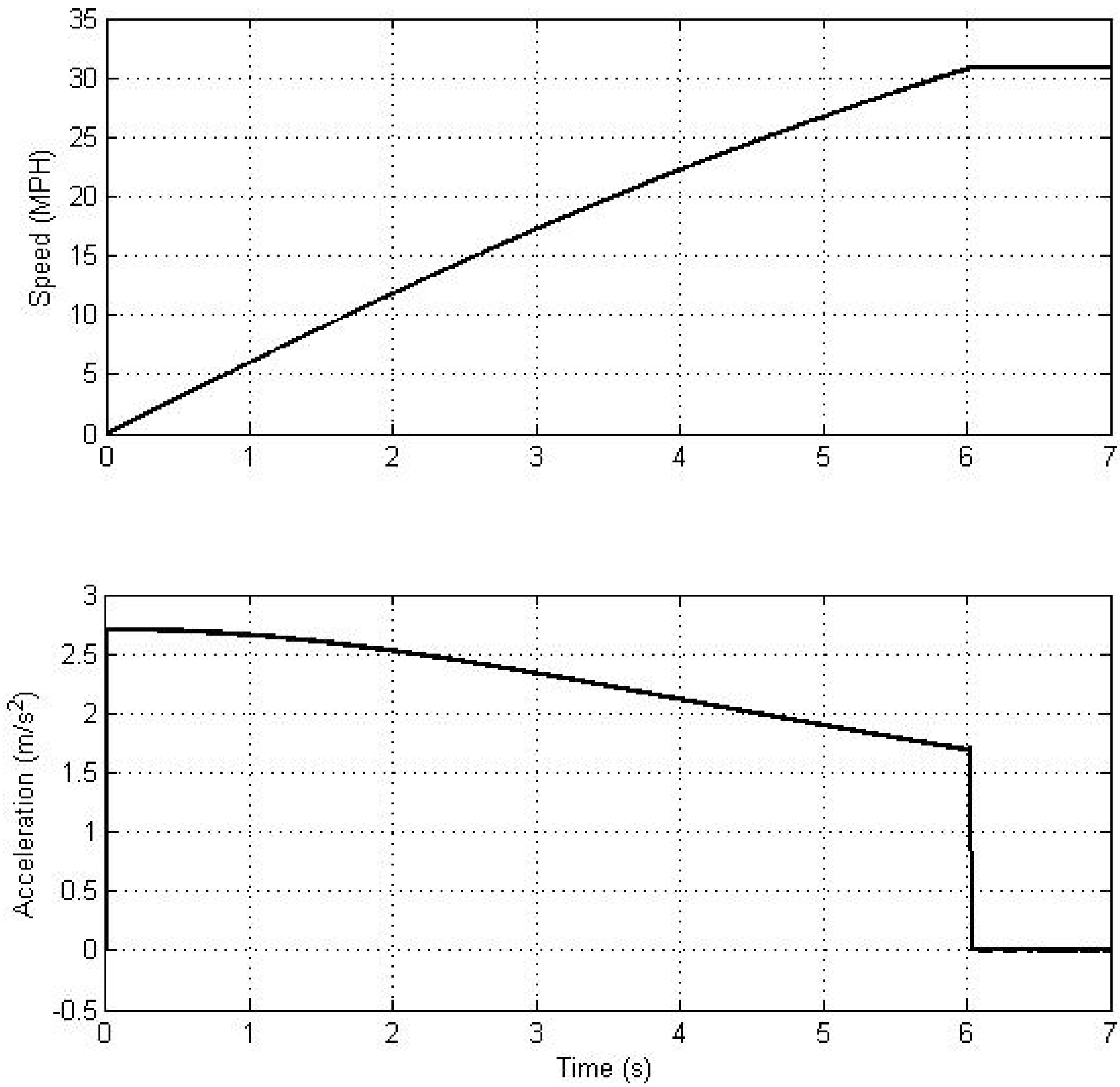
L = Wheelbase

With μ = 0.8, $F_{r,s}$ = 1622 lbs , h_{CG} = 23.62 in , L = 95 in, we determined the maximum breaking force that can be applied without lockup to be 1082 lb. Therefore, our design will not cause rear lock-up with a factor of safety of about 4.

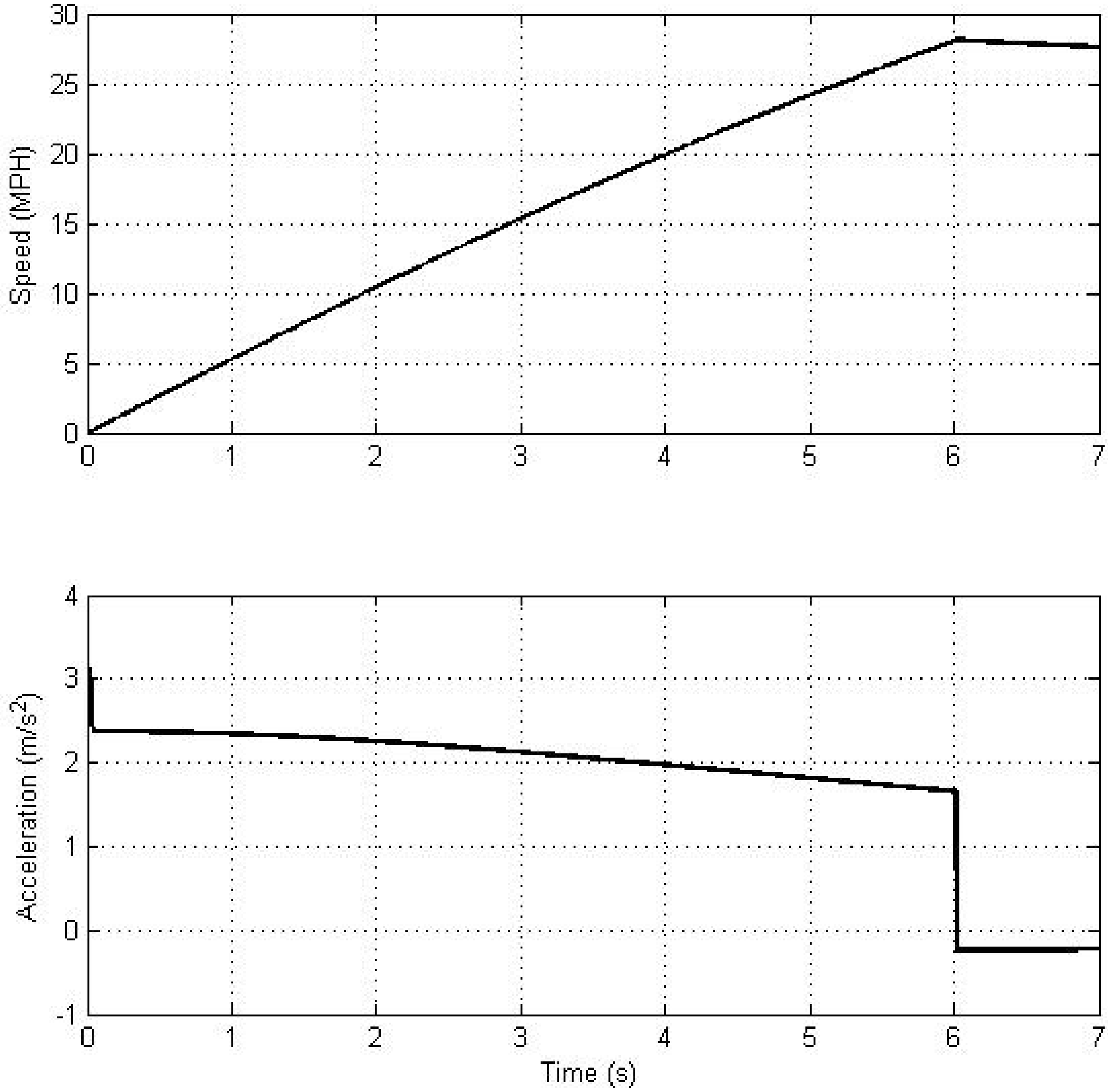
Appendix G – Simulink Model



Appendix H – Vehicle Response Without Road Loads or Tire Slip



Appendix I – Vehicle Response With Road Loads and Tire Slip



Appendix J – Sensor/Valve Circuit Diagrams

The following shows the circuit diagrams for the hydraulic actuation system in the Xebra.

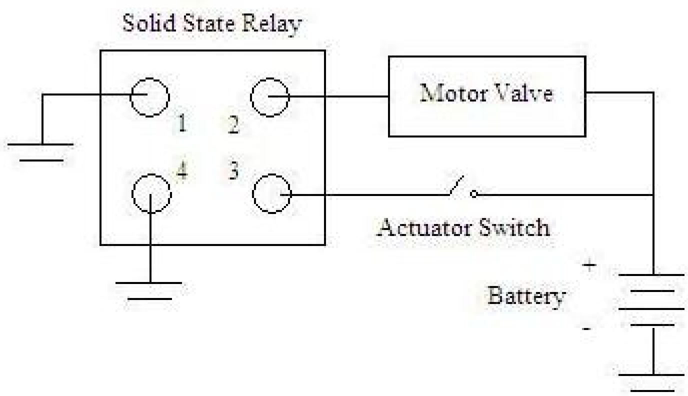


Figure J-1: Circuit Diagram for Motor Valve, Accelerator Pedal Sensor and Relay

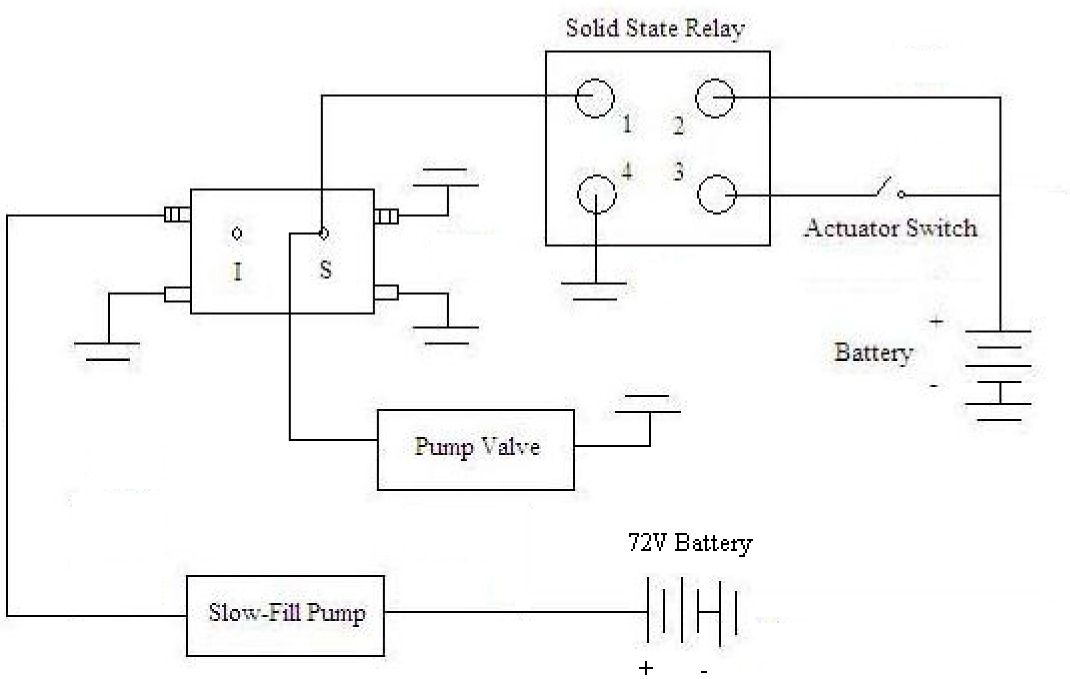


Figure J-2: Circuit Diagram for Pump Valve, Slow-fill Pump, Brake Pedal Sensor, Relay and Car Starter Solenoid

Appendix K – Hydraulic Motor Specification (see 0230 model)

Heavy-Duty Aluminum Pumps and Motors

PGP/PGM 500 Series

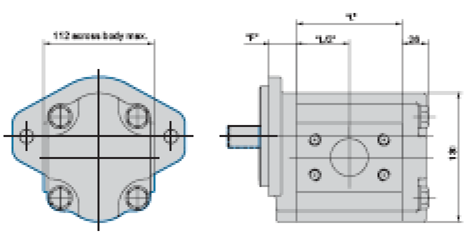
PGP/PGM 517 Specifications

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

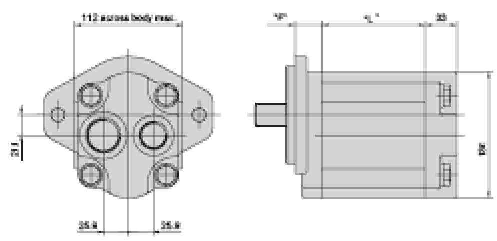
*Single pump with Shaft End Cover H3 and non ported Port End Cover.

PGP/PGM 517 Dimensions

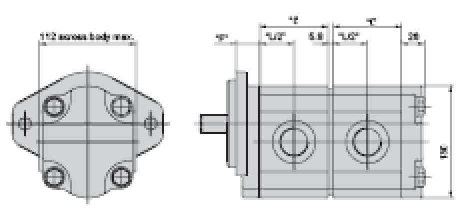
Single Unit PGP/PGM 517



Single Unit PGP/PGM 517 with rear ports



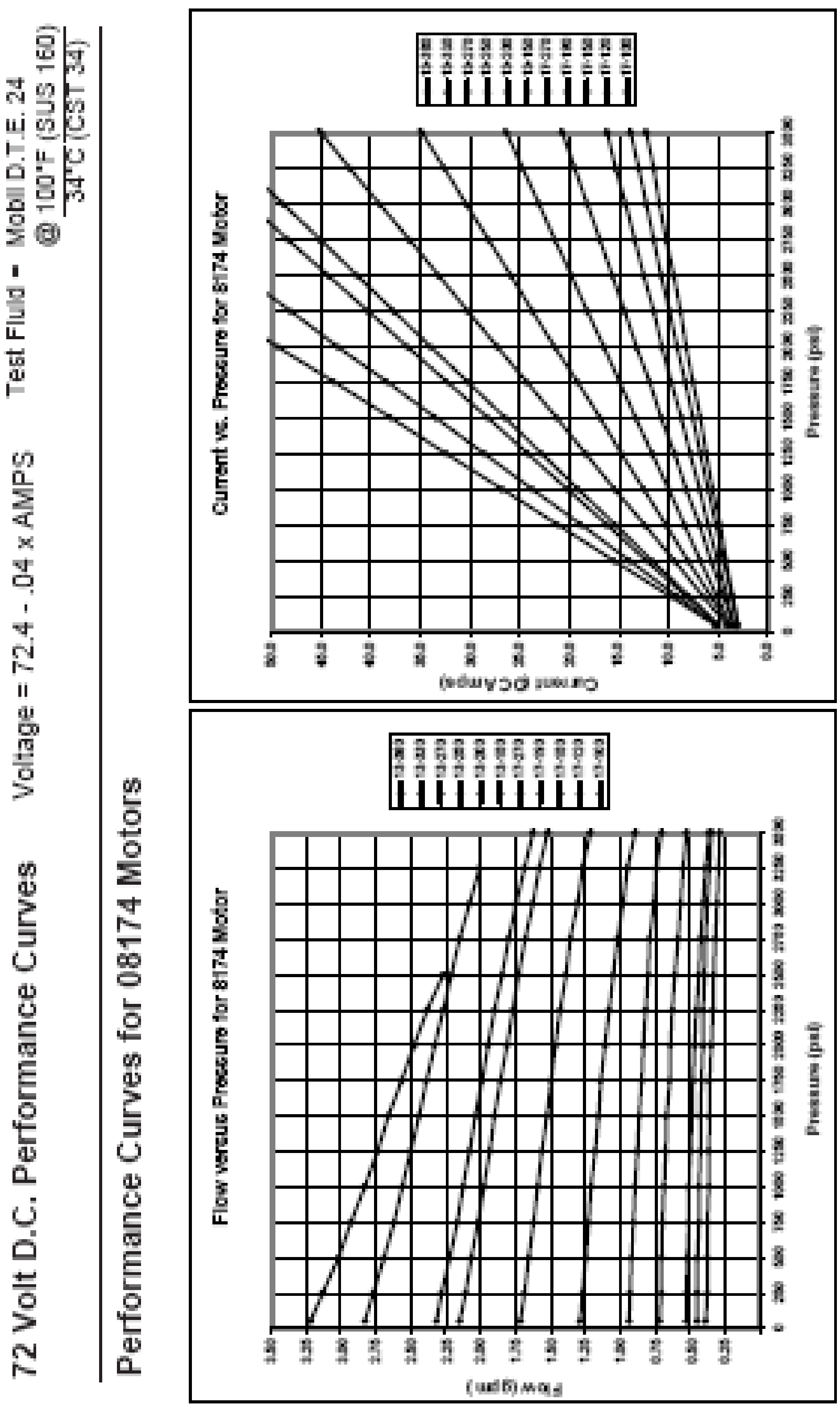
Tandem Unit PGP/PGM 517



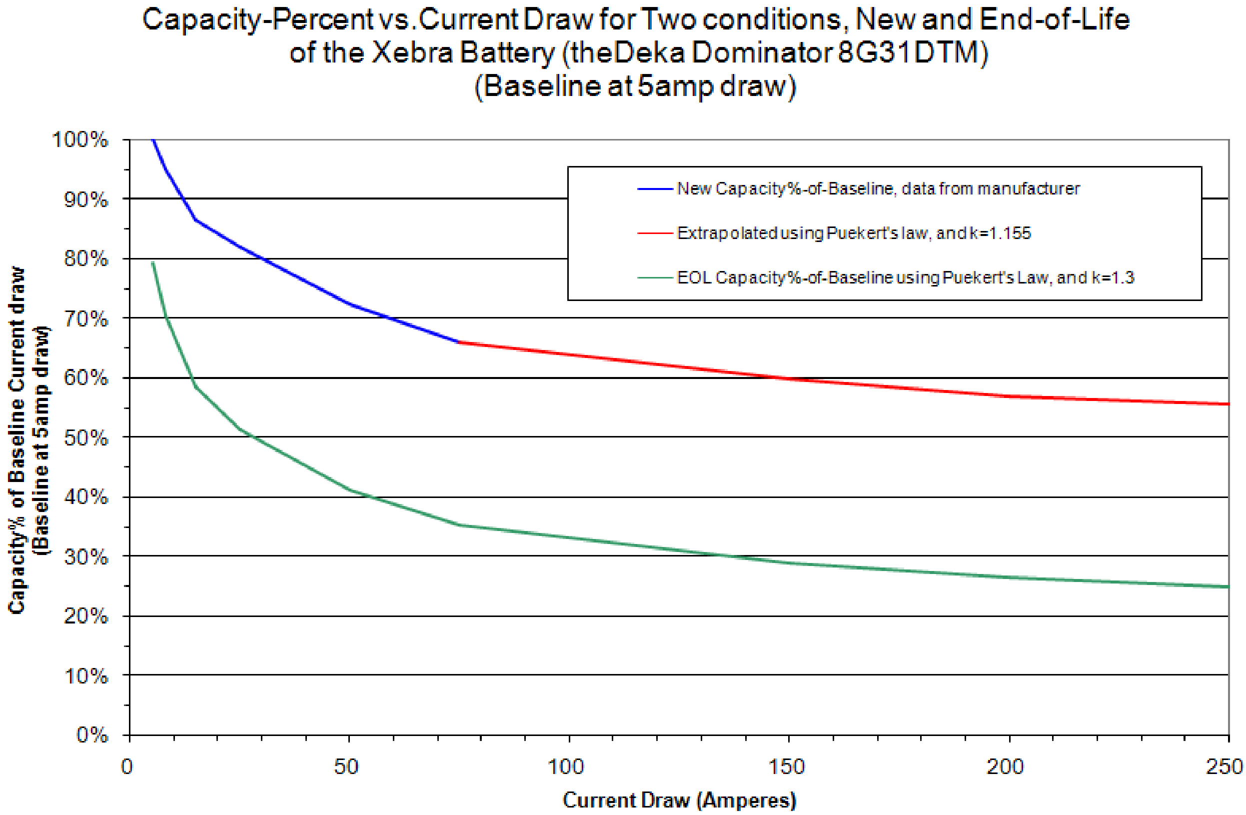
NOTE:
Dimension "F" see shaft end covers on page 21
Dimension "L" see table above

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

Appendix L – Performance curves for Monarch 72 Volt DC hydraulic pump (model 17-190)



Appendix M – Electric Battery Efficiencies and Potential Improvements From Hydraulics



*232amps measured in Ann Arbor on HOT505 max acceleration, 82.9amps at steady-state 35mph

Figure M.1 – Battery efficiency extrapolated using Pukert's Law

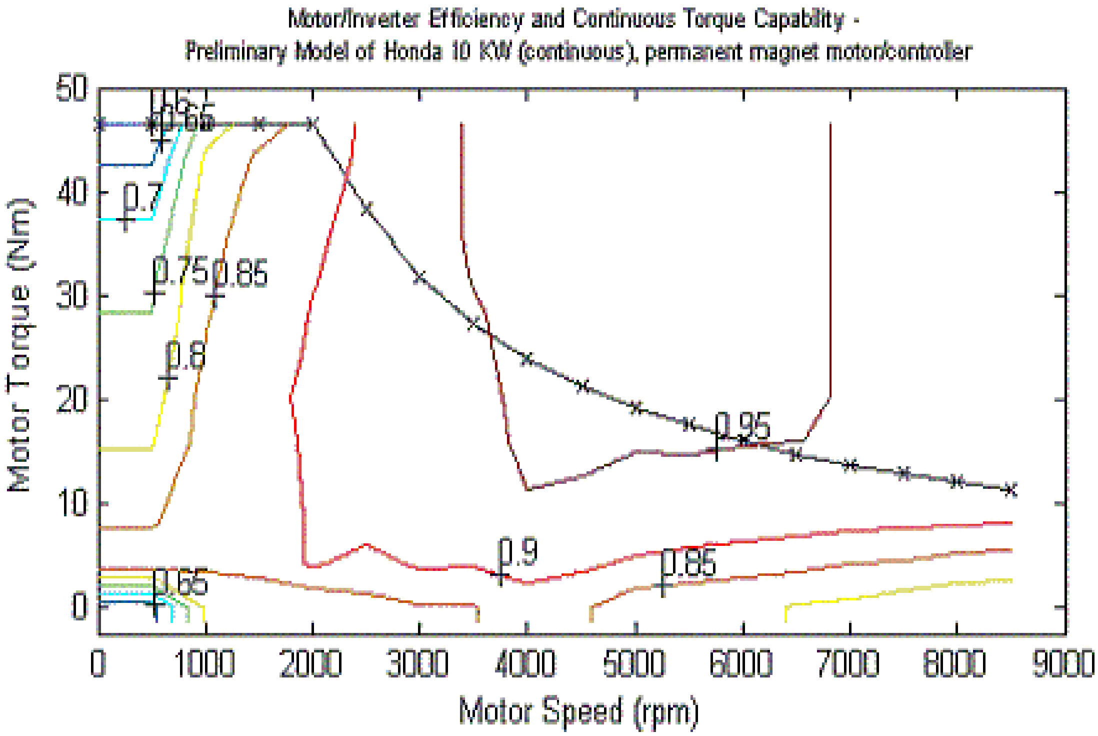


Figure M.2 – Comparable Electric Permanent Magnet Efficiency Curves

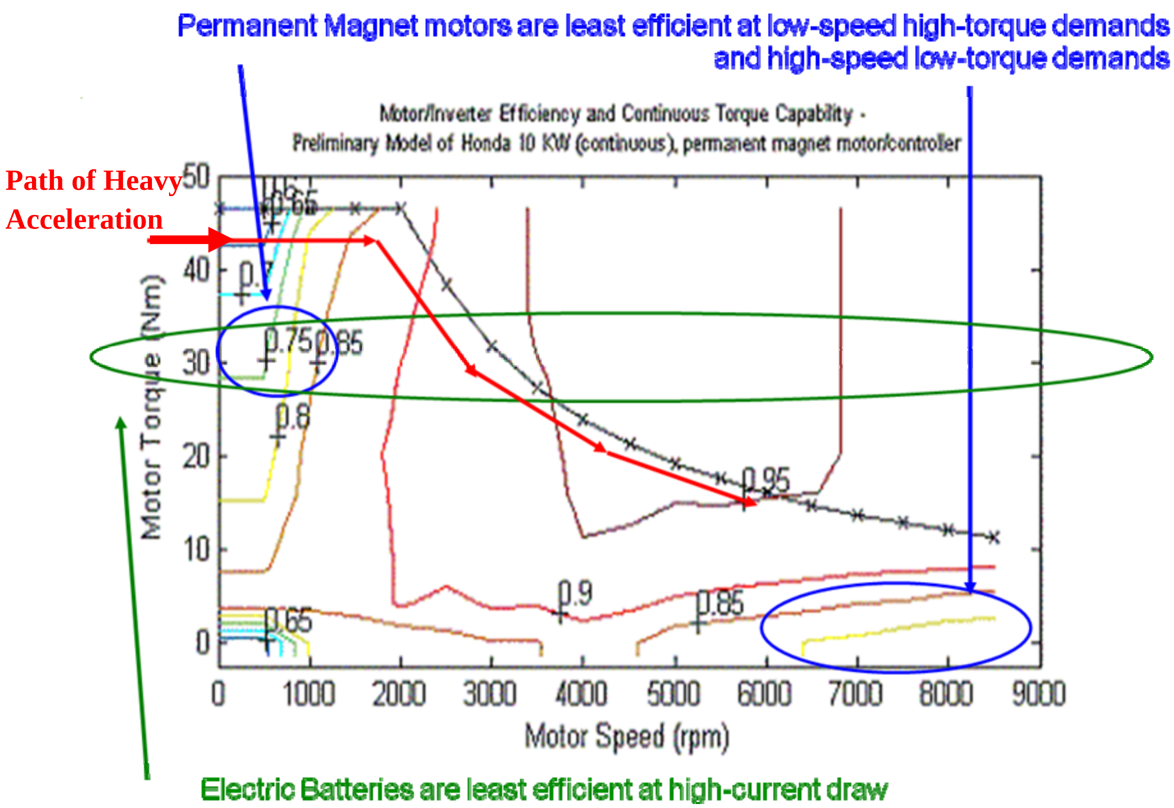


Figure M.3 – Inefficiencies of Electric Batteries and Motors under Heavy Acceleration

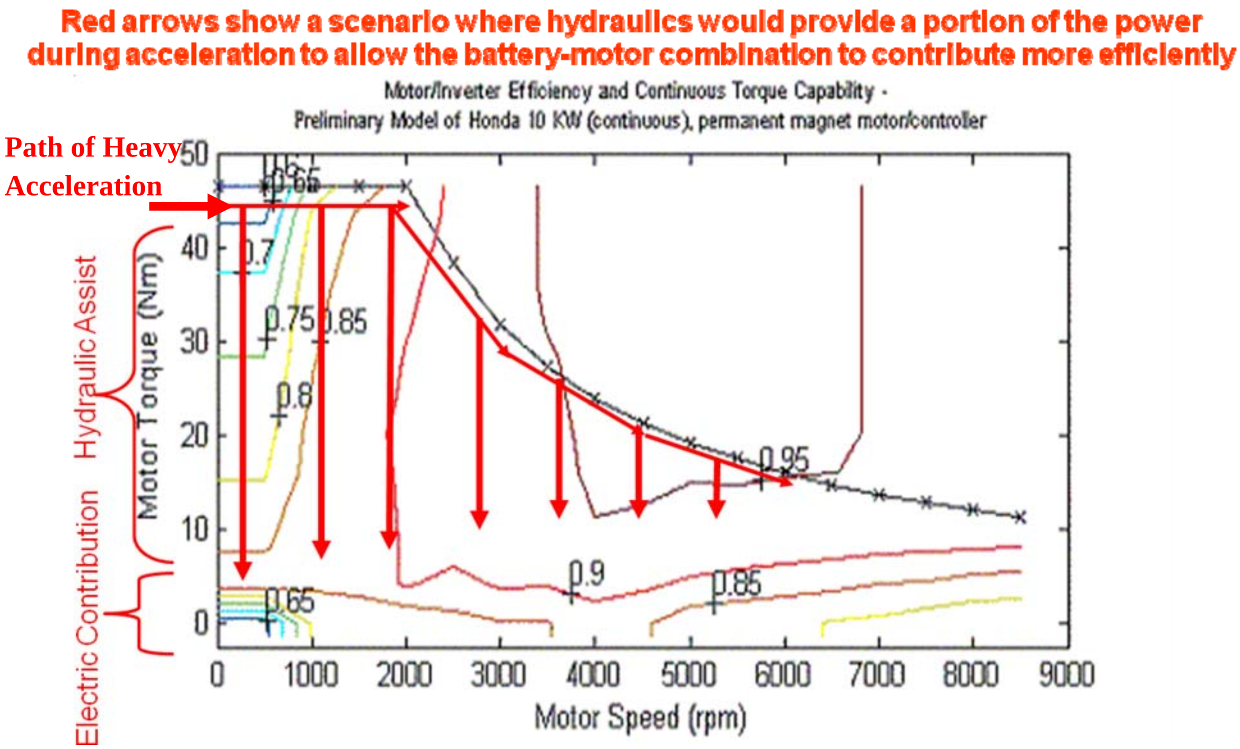


Figure M.4 – Hydraulic Assist Allows the Electric Batteries and Motor to Operate More Efficiently