XEBRA MUSCLE

ME 450 WINTER 2006

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<u>TEAM 28</u>
PATRICK FRANKLIN
CORIN GIANNOLA
TOMMASO GOMEZ
DAVID PLATT
JAMES WOLBERS

INSTRUCTOR: DR. MUAMMER KOÇ

SPONSOR: DAVID SWAIN, ENVIRONMENTAL PROTECTION AGENCY

Abstract

ME 450 Project Statements:

(Team #28) Xebra Muscle
Team members: James Wolbers
Patrick Franklin
Tommaso Gomez
Cory Giannola
David Platt

Sponsor: Mr. David Swain, Environmental Protection Agency

Due to increasing levels of CO_2 within the earth's atmosphere, there is a growing demand for alternative forms of energy in the automotive industry. Among these alternatives are electric powered vehicles, which offer highly efficient highway cruising, but very poor acceleration. With the help of Mr. David Swain from the Environmental Protection Agency, our team is designing a hydraulic launch assist system for the Xebra Electric Vehicle. This system will give the vehicle an additional boost of 50 Hp to be used during acceleration, thereby increasing performance of the vehicle without compromising energy economy. The hydraulic system has been implemented on a '92 Ford Ranger and will be installed on the Xebra vehicle during the Fall '06 semester.

Special Needs: (List anything below that you will need for the expo in addition to one standard sized table. This includes extra space, a wide door to move your project into the Duderstadt Center, electrical cords, outlet(s), water, etc.)

We need a place to put the Ford Ranger on display during the expo, either outside or inside. If it is outside, we would like it in view of our booth. We would also like a power outlet for a laptop.

Executive Summary

Our objective for this semester is to add a hydraulic launch assist system to a light, rear-wheel-drive vehicle, a Ford Ranger, since the Xebra car is not available to us. This project will ultimately lay the foundations for the integration of hydraulic launch assist in the Xebra car, since the two vehicles have similar characteristics. The system will entail a small, efficient fill up module that would run off the original powertrain to fill up the hydraulic accumulators. The goal for this term will be to incorporate a design that activates only when full throttle is applied, and apply an additional 50 horsepower to this vehicle to improve its acceleration.

Since we are no longer using the Xebra car for our project, it is less important to preserve the interior of the Ford Ranger. The main focus will be trying to incorporate this system as compactly as possible, in order to facilitate future integration in the Xebra car. The largest components of the design are two 8-liter accumulators that measure roughly 5in. in diameter and 42 in. long. The two pump motors are roughly the size of a fist and will be incorporated into the undercarriage of the vehicle. The two accumulators are the heaviest items and weigh a combined weight of 150lbs. We estimate the total weight of the system will not exceed 400lbs and there will be a focus to make the system as light as possible. In addition we will be focusing on the safety of the passengers. Our team will focus solely on adding the hydraulic launch assist system. However, we will design our system to accommodate future design modifications such as a two-stage regenerative braking system, computer control with continuously variable braking, and more complex variable gearing system. The estimated prototype cost of roughly $5000 will be covered by a grant given to the University of Michigan by the EPA.

After considering several variations on our initial design concept, we decided to utilize a single manifold (Appendix 1: Figures 16 through 18) that combines the flows of the accumulators in order to equalize the system pressure and then re-separate them. Afterwards two high efficiency manual ball valves will direct the flow through the system. These manual valves are much more efficient than electronically actuated solenoid valves.

We used a chain and sprocket system in order to couple the hydraulic motors with the truck's driveshaft. Both motors were mounted on a 1/2" steel plate (Appendix 1: Figures 24 and 25) 180^{o} from each other. This design provides incredible strength in order to resist torques applied by the motors. In addition, the motor orientation reduces the stresses incurred on the sprockets and reduces the chances of breaking the teeth. This design also takes into account how the suspension flexes. The motor plate is directly mounted to the rear differential, allowing it to move with the driveshaft.

We were unable to test our system due to time and financial constraints. In order to completely route the hydraulic lines, we would have to spend $2000 to purchase the correct fittings and custom length hoses. Since our system will be integrated into the Xebra Electric vehicle next semester, many of the fittings and custom hoses would not be utilized. In addition we would only be able to collect performance data for one day. With these factors in mind we decided not to perform acceleration testing.

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Introduction

ZAP describes the Xebra Electric Vehicle as a fun, compact, and affordable electric vehicle. The Xebra is a three-wheeled, four-passenger vehicle, powered by a DC electric motor, and designed for city driving. It has a top speed of 40 mph and range of up to 40 miles.

ZAP stands for Zero Air Pollution and was founded on 23 September 1994. ZAP was one of the first companies in the world to market electric transportation on the Internet through its website Zapworld.com in 1995. Since then, ZAP has expanded its product range to include electric motor bikes in 1999, electric drive scooters in 2001, electric automobiles in 2003, and the fuel-efficient Smart Car in 2004. Recognized as an industry leader by Time Magazine, The Wall Street Journal, Industry Week, and Advertising Age Magazine, ZAP owns several patents and trademarks and continues to develop electric, fuel-efficient vehicles.

The EPA is also directly involved in the implementation of this hydraulic launch assist system. In previous semesters ME 450 students have worked with the EPA and ZAP, installing a similar system onto a bicycle. These students will also contribute to our project this semester.

While electric motors are highly efficient and ideal for highway cruising, they fail to provide the swift acceleration required for city driving. As a solution, ZAP would like to add a hydraulic launch assist to the Xebra, to produce a swift, efficient acceleration. The hydraulic motor will be activated only under full throttle conditions. During acceleration from a stop, the hydraulic motor will produce most of the torque and then disengage when cruising speeds are reached. With the addition of the hydraulic launch assist to the current electric drive train, the goal is to improve the 0 - 30 mph acceleration without compromising the maximum driving range of the Xebra.

Because of logistic and financial complications in material transfer agreements between the University of Michigan and ZAP, we will not receive a Xebra car in time for this semester project. As an alternative, the system will be installed in a Ford Ranger, a rear-wheel drive vehicle with similar characteristics. We have provided a comparison of these two vehicles in Table 1, below. Clearly, the greatest difference between the two vehicles is their weight. This project will ultimately lay the foundations for the integration of hydraulic launch assist system in the Xebra car for a future ME450 project team, most likely in the fall semester of 2006.

Table 1: Comparison of Ford Ranger and Zap Zebra

	Ford Ranger	ZAP Xebra Car
Power Source	Gasoline Engine	DC Motor
Platform	4-wheel	3-wheel
Number of Passengers	3	4
Weight	3100lbs	1400lbs
Weight Distribution	Front biased	50/50 Split
Transmission	Automatic	N/A, direct drive

Customer Requirements

Developing Customer Requirements and Target Specifications
After discussing our design task with the EPA and ZAP, we established the following design requirements and determined some preliminary target values for some requirements (see Appendix 1: Figure 1):

50hp Power Increase: The fundamental role of the hydraulic launch assist system is to deliver a swift acceleration, while the electric system will provide cruising power. If the system must ultimately accelerate a fully loaded Xebra car with four passengers to 40mph safely, hydraulic launch assist must add 50hp to the vehicle. A less powerful system would result in very sluggish performance for a street legal vehicle. We determined that such a powerful system would require high operating pressures and efficient power transfer.

Safety: Because it is incorporated in a passenger vehicle, the hydraulic launch assist system must be safe. Thus, we must over design a system that protects passengers from potential fluid leaks or high temperatures under all conditions, including vehicle accidents. This will require resilient materials for the construction of accumulators and fluid lines.

Accommodate Design Additions/Modifications: Our task is to design the hydraulic launch assist system and install it on a Ford Ranger, as a precursor to the Xebra car. Ultimately, the final product will feature a clutch, regenerative braking, and numerous other modifications or additions. Thus, we must leave room in our design for these future changes and create a compact design.

Compact Packaging: In order to allow easy installation and future integration into the current ZAP car design, the hydraulic launch assist system should be as compact as possible. We established a preliminary target volume of $5ft^3$ and determined that we should minimize the number of interconnecting pipes and junctions in our design, for better utilization of space.

Minimize Impact on Vehicle Dynamics: The hydraulic launch assist system will not heavily impact the handling characteristics of the Ford Ranger, since its suspension is designed to accommodate rear loads. However, we anticipate that the Xebra car will be far more sensitive to the added weight of the system. Thus, we must make the total system as light as possible, so that the Xebra car will not require significant redesign of its chassis and suspension in the future.

Efficient Power Transfer: In order to provide a swift acceleration without wasting power, the hydraulic launch assist system must be coupled with a transmission that allows for smooth, efficient power transfer.

Durability: The hydraulic launch assist system will be used frequently in environments that require “stop & go” driving (e.g. city traffic), so the hydraulic launch assist system must be reliable and not fatigue under heavy operational loads and pressures. We determined that we must minimize the system weight, operational temperatures, and pressures. Additionally, we must choose light, strong materials for accumulators and fluid lines that can tolerate the high pressures and temperatures.

Relative Importance of Design Requirements

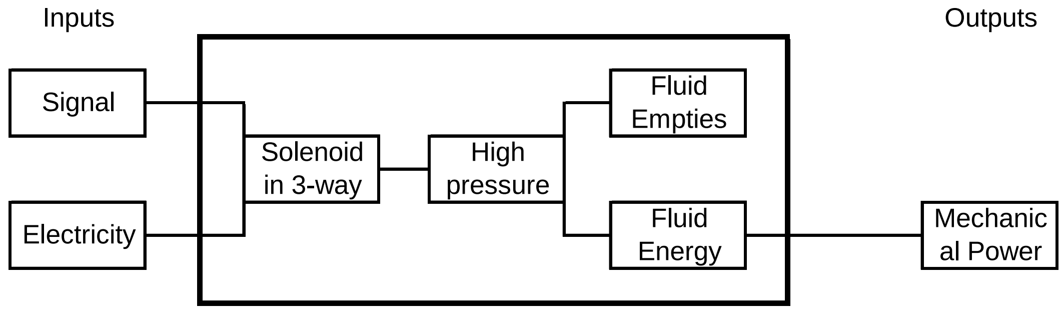
The main function of the hydraulic launch assist system in the future will be to improve the acceleration of the Xebra car so that electrical power is only used for cruising. Thus, the delivering 50hp power increase to the Ford Ranger is the most important customer requirement. Safety is of equal importance because this design will eventually be mass produced for passenger transportation all over the world. Our next priority is to make this system as compact as possible. We realize that the Ford Ranger and the Xebra car have totally different architecture and dimensions. However, if we restrict the size and weight of our hydraulic launch assists system, we can facilitate its future integration in the Xebra car. Since we are in the early stages of development, it is also important to leave room for modifications and additions. The impact on vehicle dynamics is less important since the Xebra is an economy vehicle, not a sports car. Finally, the least important requirement is efficient transmission and durability. ZAP can fulfill these requirements in later versions of the vehicle after we have optimized the use of hydraulic launch assist technology in a rear-wheel-drive-vehicle. Table 2 outlines these requirements.

Table 2: Summary of Customer Requirements and Relative Importance

Customer Requirement	Relative Importance (10=most important)
50hp Power Increase	10
Safety	10
Compact Packaging	7
Accommodates Design Modifications/Additions	6
Minimized Impact on Vehicle Dynamics	5
Efficient Power Transfer	5
Durability	4

Concept Generation and Functional Decomposition

For our project, concept generation is a much smaller aspect of the design process. The “Xebra Muscle” project builds off the hydraulic bike project of the previous term. As a result, the general layout of the hydraulic system has already been designed for us. It consists of two eight-liter accumulators powering two hydraulic motors. These motors deliver a combined displacement of approximately $50^{cc}/_{rev}$. Figure 2 in Appendix 1 provides a schematic of this layout with a description of the hydraulic symbols used. Below is the Functional Decomposition of this layout. Figures 3 and 4 in Appendix 1 are two different designs we created for our hydraulic system. We are very limited in designing our system due to the general layout provided to us. As a result, component selection and integration become the biggest challenges when choosing a design.



The first concept design (Figure 2 in Appendix 1) utilizes one high flow 3-way valve. The two high pressure accumulators discharge the hydraulic fluid, which combines and equalizes the system pressure at the first Y-Block (Appendix 1: Figures 13 through 15) adaptor. It then passes through the 3-way valve at a max speed of 168 $^{liters}/_{min}$ (see Appendix 3 for calculations). The hydraulic fluid is then split and flows through each motor, combining again after the motors. The fluid then empties into the low pressure accumulator until the high pressure accumulators are fully discharged.

The second design utilizes two high flow 3-way valves. The two high pressure accumulators discharge the hydraulic fluid, which combines and equalizes the system pressure at the first Y-Block adaptor. It then is redistributed into two streams through another Y-Block adaptor. The fluid in each stream passes through one 3-way valve at a max speed of 99 $^{liters}/_{min}$, while the other valve sees 69 $^{liters}/_{min}$ (see Appendix 3 for calculations). The hydraulic fluid then flows through each motor, combining the two flows again after the motors. The fluid then empties into the low pressure accumulator until the high pressure accumulators are fully discharged.

Concept Selection Process

Since the general design is already created for us, the most important part of our selection process is component selection. There are many different hydraulic motors and accumulators, so it is very important to compare the relative advantages and disadvantages of each. It is also very important to apply this method to the 3-way valves we select. Outlined below are the different components of the hydraulic system and how we selected each one.

High Pressure Accumulators
There are two main types of accumulators: (a) the piston type and (b) the bladder type. In the piston type, there is a metal piston inside the accumulator which separates the nitrogen precharge gas from the hydraulic fluid. When no fluid is present, the piston is forced to one side of the accumulator by the expanded nitrogen. The piston slides toward the nitrogen side compressing the gas when hydraulic fluid is stored. A bladder accumulator uses a rubber bladder instead of a piston to keep the nitrogen gas separated from the hydraulic fluid. The fluid is absorbed in a foam material when stored. We chose a piston accumulator because it is cheaper, more readily available and more compact. By compact, it has a smaller external volume while maintaining an equivalent internal volume.

We also have some freedom in the precharge pressure of the accumulators, which can be set at either 2100 psi or 2400 psi. If we make the assumption that the work done on the piston inside the cylinder (essentially power out of the accumulator = power output from motors minus all losses), then an ideal situation is a polytropic process with n=1. Then we can use:

$$W = P_1 V_1 \ln\left(\frac{P_2}{P_1}\right) \qquad \text{Equation 1}$$

By applying this equation, with no losses a precharge of 2100 psi produces 27 hp per accumulator, while a precharge of 2400 psi will produce 33.28 hp per accumulator. The 2400 psi precharge produces more power, and after system losses are included should produce more than our desired 50 hp. These calculations are shown in Appendix 2.

The accumulators we are using are 8-liter piston accumulators with a precharge of 2400 psi. These are Parker Hannifin products and have been ordered through Exotic Automation and Supply.

Hydraulic Motors/Pumps

There is little difference between what is classified as a hydraulic motor and what constitutes a hydraulic pump. Essentially, a motor has an integrated bearing used to resist heavy torques. If the shaft of a pump or motor has a force applied on it in the radial direction, which could result from a large torque, it could severely damage the unit. A pump does not have this bearing which makes it susceptible to damage. Since our design will incur heavy torques, we chose to use a motor instead of a pump.

The two major types of hydraulic motors are (a) gear and (b) piston. In a gear motor, the hydraulic fluid is forced through a set of gears, causing them to turn. This in turn drives the shaft and powers the system. A piston motor rotates similar to an internal combustion engine to displace fluid. Since we are using relatively low flow rates (33 $^{cc}/_{rev}$ and 23 $^{cc}/_{rev}$), gear motors are much more efficient. Gear motors also give us the option of pulsing the fluid flow to achieve a slower flow rate, which translate to a lower acceleration of the vehicle. Piston motors cannot be pulsed effectively since at any moment the piston could be on a compression or expansion stroke.

There are many other factors we considered when choosing the motors. They needed to be bi-directional, since they will be directly linked to the drive shaft and we do not want to restrict the vehicle's motion to forward only. Also we need the motors to be able to spin at high rpm's without pressure. This will be necessary once our system is integrated into the Xebra vehicle next semester. We also wanted a motor without a drain. This would simplify our system and give us the option to run a pressurized low side. It needs to withstand intermittent pressures of 3900 psi and about 3100 psi continuous pressure.

Table 3 below is a comparison of the two motors we were considering. We selected two different sizes of the PGM 517 series by Parker Hannifin products for because they were readily available and did not have drains (unlike the Marzocchi models). The absence of a drain will eventually allow for the precharging of the low-side accumulator in the Xebra car. These are gear motors capable of operating under 3988 psi intermittent pressure and 3625 psi continuous pressure. They are rated to run up to 3100 rpm without pressure, but experts at Zwei (Parker Manufacturer) believe they should be able to run higher without problems.

Table 3: Comparison of Hydraulic motor brands and specifications

	Parker PGM 517	Marzocchi 3A M
Max displacement	52 cc/rev	86 cc/rev
Max Pressure	3988	3900
Continuous Pressure	3625	N/A
Bi-Directional	Yes	Yes
Drain	No	Yes
Price	$555 - $570	$460 - $480
Distributor	Exotic Automation and Supply	Federal Fluid Power
Availability	10 days	10 - 12 weeks

3-Way Valves

3-way valves are very important to the operation of each design concept. They allow the high pressure fluid to flow through the hydraulic motors during launch, in addition to the circulation of fluid during standard operation of the vehicle. Heavy losses in efficiency can be incurred without properly choosing these components. In our first design, the one high flow 3-way valve must be capable of displacing up to 168 $^{liters}/_{min}$ efficiently. It will be very expensive and bulky (if even possible) to obtain a valve capable of handling this flow rate. For example, the EPA has some high flow valves rated at 70 gpm (265 $^{liters}/_{min}$). But, if we used these same valves at 168 $^{liters}/_{min}$, we would have losses in efficiency of 50%. Loss in pressure drop increases exponentially with increased flow rate. We decided to use a design with two 3-way ball valves. This will greatly reduces the flow rate each valve has to handle and increase the overall efficiency. Also manual ball valves are much more efficient than electronically actuated solenoid valves.

First Selected Concept Description

Due to logistic complications and delays in finding a suitable alternative to ZAP's Xebra car for this project, we produced and defined our first concept much later than expected (March 9, 2006). As a result, this initial concept remains up to date and unchanged (see Final Design Description below).

Engineering Design Parameter Analysis

The overall system design for the Xebra project was a carryover from previous semesters; however, component sizing and selection for this semester were crucial. Additionally, a combination manifold/3-way valve assembly needed to be designed to minimize the system size, cost and weight.

Hydraulic motor sizing was performed in close cooperation with the team's sponsor while also taking available sizes and specifications into account. The overall motor displacement goal, as defined by the sponsor, is approximately 50cc/rev for both motors combined, with the smaller motor being approximately 2/3 the size of a larger one. This criterion was a carryover from previous semester's work where the current design has been scaled up 15 times from a winter 2005 project. With these criteria in mind, we settled on a 33 cc/rev and a 23 cc/rev motor pair for a combined total displacement of 56 cc/rev.

Motor selection was not based solely on flow specifications. Max motor RPM, the motor body material, and the max pressure across the motor were also considered. Very few off the shelf hydraulic motors had acceptable peak angular speeds for vehicle use. The lower speed units would need some form of speed changer (a gear or chain and sprocket) in order to work on the vehicle. Devices that change angular velocity have the effect of reducing output torque while increasing angular velocity. Vehicle acceleration relies more on motor torque so the low speed units would have done little good in our application. The other major consideration was body material. The sponsor requested an aluminum motor body to minimized motor weight, thus keeping the overall system weight to less than 500lbs. Finally, the pressure drop across the motor (from the high pressure hydraulic accumulators to the low pressure reservoir) was also considered because this parameter dictates how much work the system can do and how much initial torque the motors can make. We finally settled on a pair of motors manufactured by Parker Hydraulics because these units have a max speed of 3100 RPM and a peak operating pressure of 4000 PSI.

Manifold (Appendix 1: Figures 8 through 11) design is a crucial part of the overall project. The manifold is the device where the pressure and flow from the two high pressure accumulators will combine and equalize to ensure equal pressure distribution to the two hydraulic motors. This component is key because it combines three sub assembles into one unit; however, it must do so in a safe fashion. The three sub assembles consist of a manifold body, and two 3-way valves (one for each of the hydraulic motors). This design was selected because the manifold efficiently combines the two inlet flows and then splits those into two outlet flows. Previous designs that were considered utilized a two hydraulic T's (one T to combine the inlet flows and equalize accumulator pressures, and a second T to separate the out flow to the individual 3-way valves) and two 3-way valves; however, this design was soon passed over due to the added tubing needed to implement the design along with the inefficiency associated with the T's.

A second design concept utilized a "Y-block" to more efficiently combine the flows and equalize the pressures. This design would have required two Y's and two 3-way valves. This concept would improve the system efficiency due to reduced flow losses; however, this design would have been more costly to produce because the Y's would have to be custom manufactured (Y's are not an off the shelf item).

The final design combines the two functions of pressure equalization and flow control into a single device. The selected manifold efficiently equalizes the accumulator pressures while minimizing the tubing needed to assemble the system; moreover, the new design allows for the use of a cheaper and lighter valve assembly because the manifold will incorporate the support functions normally built into 3-way hydraulic valves which prevent valve binding during operation.

The next step in the design process required a more in-depth analysis of the forces on the manifold due to the 4000 psi hydraulic pressures. Finite Element Analysis (FEA) was performed a number of times on different manifold shapes and materials. Initially aluminum 6061-T6 was going to be used to reduce the weight; however, a rib structure would have been needed to support the manifold at the point of maximum stress. This point can be seen in Figure 10 as a bulge in the middle where the two flows combine; additionally, an aluminum manifold would be thicker than an iron or steel manifold of the same safety factor.

The final manifold design utilizes a simple flat face design and will be made from 1020 mild steel. This alloy and design were selected based on our FEA analysis of the various manifold shape and material selections. The flat design will be much easier to manufacture than a ribbed-manifold and will be thinner than aluminum or iron design. Figure 11 (Appendix 1) shows the stress distribution within the manifold caused by the high working pressures of the hydraulic system. The final design has a safety factor of 2.96 (as calculated by Cosmos analysis software).

Another point of consideration for manifold shape/material selection was the displacement induced in the manifold body by the high working pressures. Excessive displacement would have caused shear loading on any and all fasteners used to assemble the manifold/3-way valve assembly. These shear forces would have called for the use of larger fasteners than what would otherwise be required. Additionally, the shear forces induced in the manifold could have translated to forces on the 3-way valves, thus resulting in an unsafe system. Figure 12 (Appendix 1) shows the displacements of the manifold (exaggerated) with a maximum displacement of 1.059×10^{-05} inches. This maximum displacement occurred at the “bump” thus it will not affect the fasteners or 3-way valves. Displacements on the manifold faces where the JIC-16 couplers and the 3-way valves mate (see Appendix1: Figure 10) are on the order of 6×10^{-06}, which is insignificant.

A Y-block will be used to combine the motor discharge lines into a single flow to the low pressure accumulator. This feature will minimize the total amount of tubing needed to implement the system and will maximize flow efficiency. A T could have been used here but as noted before, T’s have a greater pressure drop than a gradually convergent flow device such as the Y-block, or the manifold. Stress and displacement analysis has been performed on the Y-block as with the manifold and the results can be seen in Figures 14 (Appendix 1) and Figure 15 (Appendix 1). The safety factor associated with the Y-block is 3.69 as determined by the Cosmos analysis software package.

Tubing size considerations were addressed using some basic assumptions, those being:

1) Fully developed flow
2) No boundary layer
3) No heat transfer

The above assumptions simplified tubing size selection in that a basic fluid flow equation could be used to determine the necessary tubing sizes for the design.

$$\text{Flow Velocity} = \frac{\text{Flow Volume}}{\text{Cross-sectional Area}}$$

The most limiting portions of the system have been analyzed (that being the section of hose from the manifold to the 33 cc/rev motor, and the combined flow from the Y-block to the low pressure accumulator) as noted below.

$$\text{Flow Velocity}_{\text{33cc motor}} = \frac{.05826 \dfrac{ft^3}{\sec}}{.0157\, ft^2} = 3.70\, ft/\sec$$

$$\text{Flow Velocity}_{\text{combined flow}} = \frac{.100 \dfrac{\text{ft}^3}{\sec}}{.0157} = 6.36\, ft/\sec$$

Both cases resulted in an acceptable flow velocity through the system.

Final Design Description

Our final design still consists of the same hydraulic launch assist system mounted on top of a Ford Ranger chassis. The hydraulic portion of the launch system consists of two high pressure accumulators (eight liters each), a manifold to tie the two accumulators together in order to equalize the pressures, two hydraulic motors, and two 3-way valves to control the motors. Appendix 1: Figures 5, 6, and 7 shows the component layout for the hydraulic system as it will sit in the Ranger chassis.

The Ranger will need to have two minor modifications to accommodate the hydraulic system. The first modification will be the removal of the exhaust system. The stock muffler on the Ranger would normally be located just below the chosen high pressure accumulator location. The second modification will be the addition of a bracket system to the rear differential to hold the hydraulic motors. We will use a steel plate and mount the motors 180^{o} from each other and the drive shaft. This design is shown in Figures 24 & 25. Figure 26 shows a stress analysis performed on the motor plate. This modification is necessary because the hydraulic motors must be mounted so that they do not move relative to the differential input shaft. A chain and sprocket system will be utilized to transfer power form the hydraulic motors to the driveshaft. In addition, there will be two idler sprockets to keep tension on the chain. Additional modifications will consist of drilling several holes in the frame in order to mount the accumulator brackets. Figures 6 and 7 more clearly show the compact layout of the hydraulic system.

Figures 8 and 9 are the combined manifold/3-way valve assembly. This sub-system will be used because the 3-way valves that will be used are high-flow units; however, they are very sensitive to physical loading and may stick. The hydraulic system needed some form of a Y-block or manifold to equalize accumulator pressures, so the combined valve/manifold assembly was a natural solution to the problem. The alternative separate, high flow, 3-way valves are large and relatively expensive. Thus, we reduced the cost and weight of the system substantially by combining the 3-way valves and manifold. We tabulated a current Bill of Materials in Appendix

4, listing each component and its respective information, quantity, cost, and delivery date. The current total of expenditures amounted to $4740.

In order to engage the hydraulic launch assist, the vehicle must be brought to an initial launch speed of approximately 6mph, so that the hydraulic powertrain is not entirely stationary. At the speed, the valve/manifold assembly opens the fluid lines of the high side accumulators, allowing the launch assist phase to begin. The huge pressure gradient across the pair of motors causes them to spin as fluid is discharged to the low side accumulators, and power is transferred to the wheels of the vehicle. Over the course of the launch assist sequence, the system will theoretically deliver 54hp and help the vehicle attain a reasonable cruising speed of 30mph. Because the Ranger is more than twice the weight of the Xebra car, we don't anticipate a particularly swift acceleration on this pick up truck.

The two motors will be mounted onto a machined steel plate (Appendix 1: Figures 24 through 26) and a 60 series chain and sprocket system will be used to transmit the power from the motors to the differential input. 60 series roller chains were selected because they can accommodate a working load of 1950 lbs. The working force the drive train places on the chains will be 800 lbs giving a safety factor of approximately 2.4. A single chain will snake around the drive sprockets and the driven sprocket simplifying system design.

Prototype Description

Due to the fact that our final design involves an entire vehicle, we cannot afford to have an alpha prototype on a separate vehicle. Thus, we only have a single, final design, which is described above.

Manufacturing Plan

This design project is a unique, intermediate step that will lay the foundations for the integration of a hydraulic launch assist system in the Xebra car. Thus, the initial manufacturing plan only pertains to our final design, with no consideration of mass production of the system. Nearly all of the components are pre-manufactured and ordered from distributors, including the motors, accumulators, and piping. We have included a description of modifications to the original Ford Ranger above (see Final Design Description). Additionally, we have included engineering drawings for the manufactured parts in Appendix 1: Figure 29-34, as well as a full Gantt chart for our manufacturing plan in Figure 35.

The motor plate (Appendix 1: Figures 24, 25 and 26) needed to be milled on a CNC machine due to its large size (12"x20"x1/2" 1020 mild steel). Machining a part this large would have been exceedingly impractical on a conventional mill using a turn table and the precision afforded by CNC manufacturing made it the best choice for the project. The CNC machine was initially set up with a 1/2" diameter, three flute, high speed steel end mill, with an Aluminum Titanium Nitride (AlTiN) coating. The spindle speed was initially set to 1250 RPM; however this speed was adjusted throughout the milling process finally settling at 1000 RPM. The program used to make the motor plate was set to remove 0.050" of material with each pass and the feed rate of the material was set to 8 IMP for the rough cuts and dropped to 3 IMPO for the final surfacing cut. The motor plate needed to have extensive amounts of material removed to accommodate the motor flanges, the outboard bearings and the hole for the drive line components of the truck. Two end mills were used to remove all the required material in order to maintain an acceptable surface finish for the machined surfaces. Finally, four holes were bored to accommodate the motor flange mounting holes. A canned pecking cycle was used to drill the 3/8" holes in two steps. The first step used a number 3 center drill set to peck in 0.05" inch increments at 2 IMP to a depth of 0.125". The next step used the same canned pecking cycle to finish the drilling process with a 3/8" diameter Jobber's twist drill with a black oxide coating. The 3/8" drill was set to remove material at 3 IMP per peck and a peck depth of 0.100".

The arms, seen in Appendix 1: Figure 27, (side supports used to hold the motor plate to the Ford Ranger differential) were cut to length on a band saw with an appropriate metal cutting blade. The supports needed to have semi-circles of material remove to fit around the Ranger's axle shafts. These semi-circles were cut by hand using an abrasive cutoff wheel mounted in an appropriate air powered cutoff tool. The Compressor was set to 90 PSI and the cutoff tool speed was set to approximately 20,000 RPM. Production versions of these supports could be produced using a 3" hole cutter mounted in a mill/drill; however, time constraints and tool availability necessitated the use of hand tolls to cut the appropriate shape. Slots were cut in the arms wide enough to accommodate the 1/2" thickness of the motor plate with 3/8" space on either side to accommodate flanges using the same cutoff tool described above. 3/8" thick flanges were welded into the slots and holes were drilled through the flanges to accommodate 1/2" grade 8 bolts. The motor plate was suspended into the correct position and the arms were put into place. The flange holes were marked and the holes were drilled trough the motor plate. Additionally, holes were bored through the arms accommodate 3" U-bolts used to fasten the arms to the axle shafts.

The driven sprocket, Appendix 1: Figure 25, that was mounted to the drive shaft also needed to be CNC milled due to the complex shape of the interior cutout section. This shape was needed to clear the existing U-joint of the truck while giving us an appropriate means of bolting the driven sprocket to the differential input shaft. A "blank" 60 series, 25 tooth sprocket was purchased and loaded into a CNC mill when it arrived. The sprocket was made of mild steel so the same procedure outlined for the motor plate above was used. One new end mill was used to machine this part.

The manifold was machined on a conventional mill using a 7/8" 2 flute HSS end mill. A block of 1020 steel (4.5"x4.5"x6.5") was purchased and cut to 4.5"x4.5"x2.625" on a horizontal cutoff machine. That stock was loaded into a mill with the spindle speed set to 1000 RPM and the auto feed set to 5 IPM. 0.030" of material was removed per pass to get the block to a final size of 4.2"x4.2"x2.5". This process took approximately four hours to complete resulting in a steel block with the desired dimensions. The next step in the process was to bore the 1" diameter hole through the centered of the block faces. A single tool long enough to complete this task in one pass was not available so a series of drilling operations was performed in 1/4" steps from 1/4" to 1". This process was performed on four faces due to the short (2.5") 1" drill bit used. The four holes drilled into the block formed a cavity in the center of the manifold where pressures could equalize with a minimum of flow losses. The final step in manifold production required a little assistance from the EPA. The special seat cutter needed to make the 37 degree SAE hydraulic seats was not readily available to the team and outside sources wanted upwards of $1800.00 to cut the seats and thread the block. The EPA had the needed cutter on hand and offered to help. The block was loaded into a mill in the EPA machine shop and each of the holes was enlarged to the correct size for threading with an SAE 12 TPI 1" tap and the seats were cut. The newly enlarged holes were then tapped the accept SAE 16 hydraulic fittings. This process was performed on all for faces of the manifold.

The Y-bloc was machined from the remaining 1020 mild steel left over after the manifold was cut from the stock. The piece of steel was loaded into a turntable mounted on a conventional mill bed plate. The top surface was milled flat and the mill datum was set to the middle of the block. The turntable allowed the machinist to rotate the stock in 60 degree increments to form the hexagonal shape of the Y-block. After the material was shaped it was flipped over and the excess stock used to clamp the piece into the turntable was milled off. The block was drilled in a similar fashion as the manifold except the holes were not bored all the way through. The combined flow hole was bored to 1" and then a 1-1/8" ball mill was used to form a hemispherical volute within the Y-block. 1-1/8" was used to allow for the use of SAE 20 hydraulic fittings on the combined flow outlet to minimize pressure losses within the Y and hosing from the Y to the low pressure accumulator. The final step in the process would have been to take the piece to the EPA and have threads cut; however, a machine malfunction (the mill bedplate was walking along the X-axis due to a bad jamb nut) caused the final Y-block to be out of spec and unusable.

The hydraulic fittings were originally purchased from Federal Fluid Power, but their fittings took up too much space to make conversions between pipe and tube sizes. As a result we decided that custom fittings would have to be ordered from Tompkins Industries at a cost of $1500. The JIC 16 size hydraulic flex line hosing capable of withstanding 4000 psi has a bend radius of 12” making it extremely difficult to fit in the frame without making large sweeping turns that greatly reduce the efficiency of the system.

The majority of this manufacturing plan will not be applicable to the Xebra car, however the future project teams may decide to make a similar motor mounting system, a hydraulic manifold, and Y-blocks. The integration, however, will be an entirely different process and will likely be much more difficult than in the Ranger. We recommend using custom bend piping and using a lighter more compact aluminum motor mounting plate.

Description of Validation Approach

Many of our engineering specifications can be easily measured without any complicated tests. A large component of our design is compact packaging, which ultimately corresponds to a small volume and small increase in weight. Most of the major components of our system have already been weighed and measured. In order to obtain the total system volume, we would simply have to measure the size of each component that we do not already have specifications on and add up the total volume. Similarly, the weight of each component we do not already have data on can be measured and then added up to obtain the total system weight.

Fluid Pressure is another aspect that we will be able to measure easily. We will be incorporating Noshok pressure gauges throughout the system in order to monitor pressure. Additionally, these parts have a specific feature that records the max pressure achieved during operation. Before we run a test of our system, we will reset the max pressure needle on the pressure gauge. Then after a successful run, we will measure and record the max pressures reached throughout the system. This will allow us to monitor our system pressure in order to avoid damage or failure.

Power output will be the most difficult engineering specification we need to measure. The best way is to do dynamometer testing. Due to costs and availability, this will most likely not be possible. There are several ways we can estimate the power produced by the system, but many involve specific assumptions. Power could be calculated by taking the theoretical work divided by the actual time it takes for the accumulators to empty. This method does not take into account losses in the system and would prove to be inaccurate. If we assume the system provides a constant acceleration, we could calculate acceleration with the time it takes to empty the accumulator and distance the vehicle travels in this time using equation 1. We would then have to measure the total mass of the vehicle. This can be done at a weigh station or a junk yard. With mass, distance, acceleration and time we can calculate the power produced with equation 2.

(1) $x = V_o t + \frac{1}{2} a t^2$

(2) $P = \frac{W}{t} = \frac{F * x}{t} = \frac{m * a * x}{t}$

The second most accurate option, and our chosen route, would be to purchase an accelerometer. This would eliminate the assumption of constant acceleration making our calculations much more accurate. This device would allow us to download acceleration data during the entire run to a computer. Using the measured mass of the vehicle as described above, we can use this data to calculate the power produced by our design.

The rest of the engineering specifications do not have as strict of guidelines. Fluid Temperature will increase slightly when the accumulators are filled. Ultimately this would be hard to avoid and will cause energy losses in our working fluid. We can determine a rough estimate of the magnitude of this loss based on a comparison between the theoretical work calculated and the actual work measured. It will not be very accurate due to many other loses incurred throughout the system. We could make rough estimations in order to account for these losses but this will provide us little benefit.

The number of tubes and junctions simply needs to be minimized. Our prototype will be selected based on the most efficient design, which requires a minimal amount of piping. Material strength is similar in the fact that there is no specific value we need to meet. All of the components we purchase are designed to withstand the pressures and flow rates of our system. Therefore determination of strength is not needed. On parts we fabricate, we will know the initial material strength and then design the part in order to withstand the pressures associated with our system. It would be hard to test the strength of a part we fabricate without breaking it, which would be impossible in our given time frame.

Test Results

In the limited time we had, we had to consider the feasibility and benefits of actually testing our system. After we integrated all the major components of our design into the Ford Ranger, it would cost an additional $2000 to purchase all the correct fittings and hydraulic hosing to make our project operational. Once our project was completely assembled, we would be left with only one day to collect performance data. In addition, many of the fittings and custom hosing will not be used next semester. We determined that it was not worth spending $2000 in order to test our product for one day. A bill of materials for the required fittings is in Appendix 5. The prices listed do not include tax or shipping and handling.

Problem Analysis

Fundamentals Requirements
In order to produce a functional, reliable solution to this design challenge, our team continues to build upon our foundations of fluid dynamics applied to hydraulic systems. While many components in the system are rated to perform at calculated pressures and flow rates, our manifold will require a rigorous application of finite element analysis to ensure that it can withstand operation pressures. We must strike a balance between reducing the weight of the design without compromising its strength and safety.

Additionally, we have also had to develop an understanding of the mechanics of transmission systems, to ensure a smooth power transfer from the hydraulic motors during acceleration. The project team will have to become familiar with bearing and gear design and selection.

Critical Issues
While our project is taking shape rapidly, we still have some design concerns and potential challenges ahead. Hydraulic component selection, such as the accumulators, hydraulic motor, and hydraulic piping, must be selected based on customer requirements, distributor availability, and component cost. Meanwhile, we will machine the essential structure of the manifold in the machine shop, and then send it to Protomatic no later than March 23 for tapping. Since ZAP has not provided us with any dimensions or specifications for the Xebra car, we can only speculate how well our design will apply to it. Thus, we have focused on minimizing the size and weight of the design without compromising safety.

Once we have installed the basic hydraulic system, we will have to couple the hydraulic motors to the Ford Ranger’s original power train. This design aspect will be entirely specific to the Ford Ranger, but we hope to gain some insight on how to couple the electric powertrain of the Zebra in the future.

Additionally, we still have some logistic concerns at the moment. While the EPA can provide us with a facility to work on our project during normal business hours, we have yet to find a place to store the vehicle overnight.

Design Critique: Flaws and Potential Solutions

Our design was created with the intent on implementing the system on a Xebra Electric Vehicle. Due to complications in material transfer negotiations between the University of Michigan and ZAP, we were unable to procure this vehicle. With this in mind, we installed our system on a ’92 Ford Ranger in order to validate its operation. During our project, we discovered several potential problems:

Size Constraints
The twin 8.0L high pressure accumulators are 44” long, which is roughly the width of the Xebra Car. In order for them to be integrated into the chassis of the vehicle, the accumulators would have to be oriented lengthwise, horizontally, as in our prototype. We can only speculate whether this is feasible with the Xebra car, but we suspect that the accumulators would have to be resized, perhaps with a larger diameter and shorter length.

As we learned with our prototype, there is insufficient space to accommodate the network of flexible tubing that links hydraulic components. This heavy duty, semi-rigid tubing has a bend radius of 12 inches, which creates problems when linking nearby components like the accumulators and the manifold. Thus, the Xebra car will require custom bent metal tubing, which would certainly raise production costs.

Finally, the slow fill pump unit includes a large motor and battery system that would be difficult to house in the Xebra car, given the space constraints. Ideally, we could eliminate the bulky battery cells by running the motor off the main batteries of the Xebra, or using regenerative breaking to power the pump.

Weight Constraints
Many of the critical components like the motors, slow fill pump, steel plate, and accumulators are very heavy, so the total weight estimate of the hydraulic launch assist system would be in excess of 500lbs. If this system were to be integrated in the Xebra car, it would seriously compromise the cruising range of the vehicle, operating on electric motors. Additionally, the added weight would require a suspension redesign for the rear portion of the car. Since the system must withstand such high pressures, it would be difficult to reduce this weight significantly without seriously compromising the safety and durability of the system.

Power Coupling
In our prototype on the Ford Ranger, the driveshaft was easily accessible, so we could easily couple the hydraulic motors to the original drivetrain via a system of chains and sprockets. However, the Xebra does not have a drive shaft, since its motors are housed in close proximity with the rear axel. As a result, the Xebra vehicle will require a new transmission to couple the hydraulic and electrical powertrains. With the appropriate gearing, this transmission would most likely be more efficient than the chain and sprocket system.

Losses in Power and Efficiency

In order to meet our budget constraints, we were unable to fully optimize our design to maximize the efficiency of the hydraulic launch assist. Our current prototype employs T-junction fittings where opposing flows meet with considerable pressure losses. Ideally, we would replace these T-junctions with Y-junctions ($900 each), if we could afford them. In addition to this, the length of tubing between the accumulators and the motors must be minimized, so that there is minimal pressure loss across the tubing. Again, custom bent tubing would be an ideal solution to this problem.

When linking hydraulic components with tubing, it is important to minimize the pressure drop from the accumulators to the motors, otherwise there will be a significant power loss. This can be accomplished by shortening the lengths of tubing (e.g. with custom bent tubing), and using more efficient flow junctions (e.g Y-blocks instead of T-junctions).

Design Inconsistencies in Components
When we designed the steal plate to hold the motors, we discovered a slight difference in the motor shaft diameters near the flange. In order to accommodate the difference on one of the motors, we had to attach a separate beating plate on the other side of the sprocket to house the steal ball bearing. This design consideration could have been avoided if the manufacturer standardized the specifications of the motor shafts for both motors.

Component Availability
Our design mainly consists of custom components (e.g. custom bent tubing, accumulators, etc.) that require long lead times for fabrication and shipping. If this design must be integrated into the Xebra car in the relatively short time frame of a semester, these components must be ordered as soon as possible. In our case, many of these potential problems were difficult to anticipate without possessing a Xebra Electric car. In order for this project to succeed, the Xebra car must be maid available to the project team immediately at the start of the semester.

Design Critique: Advantages of Current Design

In addition, our design provides many benefits:

Robust Design
Our design challenge consists of converting mechanical power from fluid flows at high temperatures and pressures that could be hazardous to passengers in the event of a component failure. To avoid such hazards, we took great precautions in our design process to ensure that each component could withstand the high pressure. In particular, we conducted an FEA analysis of the manifold/valve assembly so that we could produce a safety factor of 2.44:1, shown in figures 17 & 18. We also conducted FEA analysis of the steel motor plate, Y-block, and clutch mechanism.

Additionally, we anchored the motors and chain and sprocket system on to a 1/2” AISI 1018 mild steel plate that could withstand high torques and radial loading. We also installed, steel ball bearings around the motor shafts to prevent damage of the hydraulic motors due to radial loading. Overall, we have engineered a robust design that would prolong the lifespan of major components like the motors, valves, and accumulators. Future project teams should be able to reuse these components. All other critical components were custom ordered with specifications that met our safety requirements.

Safety Release Mechanism
Finally, our design is equipped with a safety release valve that enables the high pressure flow to bypass the motors and flow straight to the low pressure accumulator. This key feature would allow the driver to abort the hydraulic launch sequence in the event of unforeseen driving conditions, like and obstacle in the road.

Recommendations

Logistic Considerations
Overall, the scope of this project remains quite daunting and we feel it should be broken down into smaller projects that could either be spread across several semesters or assigned to multiple teams to tackle over a single semester. While the task of fitting a hydraulic system into a Ford Ranger was already difficult, the additional task of designing a transmission and interconnecting hydraulic lines made the project unmanageable for a single semester. A better approach would be to divide the project into to assignments: (a) designing a transmission to couple the hydraulic and electric powertrains and (b) selecting components for the hydraulic launch assist system. These two teams would work together to accomplish their tasks in tandem, so that each group could focus on a smaller piece of the overall project.

In order for future teams to accomplish these objectives, it is imperative that they receive the ZAP Xebra Electric Vehicle within the first week of the semester, along with an appropriate work facility. Our main stumbling blocks for the project were procuring a suitable vehicle for the integration of the hydraulic launch system and locating an appropriate facility to work in.

In addition, this project requires a more direct, efficient process for the allocation of funds and approval of purchases. In our case, accessing the funds from the EPA grant was a multi-step process where (1) a component had to be selected and located, (2) the component number had to be sent to a secretary who then ordered the part, and (3) our group had to pick up the part from the secretary's office upon delivery. This process was not particularly efficient because the team had to rely on a third party and there were several instances when we could not contact this secretary to order parts. This process was particularly restrictive to our progress because we could not continue to work on weekends if parts were needed. For future projects, we suggest that each team be provided with a credit card with a predetermined spending limit. Perhaps each team could appoint a treasurer to do the ordering and have financially responsibility for any purchases made with the card.

Many key structural components like steal plate, bearing plate, and manifold were hand machined out of steal, in separate processes that required up to 11 hours of work, as shown in Table 4. Given the limited availability of the CNC-mill, we were unable to machine all of our components in a timely fashion.

Table 4: Estimated Machining Times

Component	Manual/CNC	Estimated Machining Time (hrs)
Y-block	Manual	8
Manifold	Manual	11
Steel Motor Plate	CNC	8
Driven Sprocket	CNC	1

In order to make steady progress, future project teams will need regular access to CNC and manual machining equipment.

Technical Considerations

In additional to logistic recommendations, we have several technical recommendations as well. The twin 8.0L high pressure accumulators are 44" long, which is roughly the width of the Xebra Car. In order for them to be integrated into the chassis of the vehicle, the accumulators would have to be oriented lengthwise, horizontally, as in our prototype. We can only speculate whether this is feasible with the Xebra car, but we recommend that the accumulators be resized with a larger diameter and shorter length.

We would advise future teams to integrate a clutching mechanism in their design and select the motor and transmission accordingly, so that there will be space to accommodate a clutch. We designed a one-way clutch mechanism shown in Appendix 1: Figures 19 and 20, and performed FEA analysis on it shown in Figures 21-23. Unfortunately, we did not have the time or resources required to manufacture it. The University of Michigan 3D lab could help with the manufacturing of this piece by providing access to the stereo lithography machines (see Appendix 1: Figure 28). The resulting prototypes could then be cast by a foundry. Perhaps future teams could request access to a local foundry to fabricate these parts as cheaply as possible.

These teams should select hydraulic components as early as possible, in order to estimate the high costs. In particular, hydraulic fittings can easily consume a large part of a budget, so future teams should plan accordingly. We did not finish the project because the required fittings would have cost $1520 without including the additional cost of custom length flexible hydraulic tubing.

Also, there is insufficient space to accommodate the network of flexible tubing that links hydraulic components. This heavy duty, semi-rigid tubing has a bend radius of 12 inches, which creates problems when linking nearby components like the accumulators and the manifold. Because the system would require more than 12 feet of flexible tubing to link components, the pressure drop from the accumulators to the motors would be very large, resulting in significant power loss. Thus, the Xebra car will require shorter, custom bent metal tubing, which would certainly raise production costs.

The steel plate that secured the motors had a safety factor of 6.5 as shown in Appendix 1: Figure 26. We chose steel because it was easy to weld and relatively cheap, however future project teams should consider lighter materials such as aluminum. The use of aluminum would also decrease the time needed to machine the motor plate and make maneuvering the plate into position much easier.

Finally, we have concluded that the marginal efficiency gains of the manifold do not justify the 11 hour of machining that this component required. As an alternative, crosses can be purchased from Tompkins industries for a comparable price of $40. However, we would recommend assembling two T's together to form an H instead of using a cross flowing device. The H would allow fluid to flow straight through with only minimal cross flow and minimal flow collision.

Conclusions

After we encountered some logistic hurdles in the procurement of the ZAP Xebra Electric Vehicle, the scope of our project changed so that the ultimate goal became to lay the foundations for the integration of hydraulic launch assist system in the Xebra car, by installing it in a similar, lightweight, rear-wheel drive vehicle. We selected a 1992 Ford Ranger and devised a system that could be integrated into the chassis, underneath the truck bed.

The goal for this term was be to incorporate a design that activates only when full throttle is applied, and apply and additional 50 horsepower to the vehicle to improve its acceleration. Our design consisted of an efficient fill up module that would run off battery power to fill up the twin high pressure accumulators. Once the accumulators reached the desired pressure of 4000psi, two valves would release the hydraulic fluid into separate lines that would combine at the manifold assembly and then pass through two separate hydraulic motors. These motors would in turn be linked to the driveshaft via a system of chains and sprockets that would allow for efficient power transfer. After the fluid passed through the motors, it would discharge into the low pressure accumulator.

Our proposed design evolved into the most efficient solution for the given resources. We ensured that our design was safe for passengers, with careful FEA design of custom machined components, and the addition of an emergency release valve. Also, we secured the shafts of the hydraulic motors with steal bearings, for added support and durability.

In our attempt to implement this design, we encountered several logistic dilemmas since our system required many expensive, custom built components with long fabrication and shipping delays. Additionally, other key structural components required very long machining times on an automated CNC mill, which we had limited access to. In spite of these setbacks, our group persevered and assembled a large portion of the design. Eventually, we realized the scope of the design was well beyond our short time frame and financial resources. A low cost implementation of this design would consist of copious lengths of flexible tubing that would result in significant pressure loss in the system, and hence a significant power loss. The cost of the fittings alone would be in excess of $1500 and thus, outside of our budget. In addition, the weight volume of the total system rapidly exceeds initial expectations, and led us to question whether such a design could be implemented in the Xebra car in the future.

This is not to say the ultimate goal of integration in a Xebra car is impossible, or that our design attempts were futile. With some access to the Xebra car, an expanded budget to produce custom shaped accumulators and customized fittings and tubing, the design could be implemented in a relatively light weight, compact package that would deliver the promised 50hp. The depth and technicality of this design challenge exceed the resources of a single senior design capstone, and would warrant a more serious effort like that of the Challenge X project. The concept of a hybrid hydraulic-electric car certainly warrants such an effort, and we hope to see the project succeed in the future.

Information Sources

Since hydraulic launch assist is a new technology, especially in electric vehicles, there are few useful information sources and no similar products that could be benchmarked. We found some general information about hydraulic launch assist systems at the EPA's[1] and Eaton's[2] websites and we also obtained some hydraulic system information from dealers such as Federal Fluid Power[3]. We also investigated other hydraulic systems such as hydraulic presses[4], hydraulic steering systems[5], and hydraulic brake systems[6] to gain more general knowledge about hydraulic systems. However, these low-speed applications of hydraulics were not very relevant to our project.

A patent search yielded no results, however the team is currently working under a provisional patent submitted by the EPA. Most of our information regarding hydraulic launch assist systems and how they can be applied in an electric vehicle came from our project sponsor Mr. David Swain from the EPA. Hydraulic fluid calculations and equations were derived from a standard fluid mechanics textbook, Fundamentals of Fluid Mechanics 5th Ed. by Munson, Young, and Okishi.

We will not be able to benchmark our system against a stock setup on the Xebra car because ZAP has not provided us with any performance specifications.

References

ZAP Power Systems
www.zapworld.com

Adam Maurer
ME 490 Business Consultant
admaurer@umich.edu

David Swain
U.S. Environmental Protection Agency
swaindm@umich.edu

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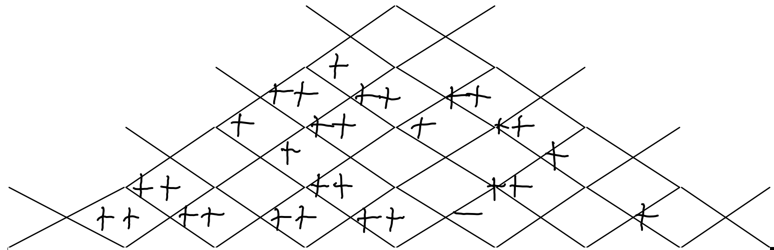
[1] www.epa.gov
[2] www.eaton.com
[3] www.federalfp.com
[4] www.magnumpress.com
[5] www.seastarsteering.com
[6] www.mico.com

Munson, Bruce R., Young, Donald F., and Okiishi, Theodore H. Fundamentals of Fluid Mechanics 5th Ed.
Hoboken, New Jersey: John Wiley & Sons, Inc., 2006

Appendix 1: Figures

QFD chart for project



		Volume (-)	Weight (-)	Power Output (+)	Fluid Temperature (-)	Fluid Pressure (-)	Number of Tubes/Junctions (-)	Material Strengths (+)	Transmission (+)	HLA Prototype
Compact Packaging	7	9	3				9	1	3	
50 Hp Increase	10			9	3	9		1	9	
Reduce Impact on Vehicle	5	3	9	9					3	
Efficient Operation	5			9				3	9	
Durability/Design Fatigue	4		9	9	9	9	3	9	9	
Upgradeability	6	9					9		3	
Safety	10			1	9	9	3	3		
Measurement Unit		ft^3	lb	Hp	°F	psi	#	MPa		
Target Value		5	400	50		4000				
Importance Rating		3	2	1	4	4	8	4	7	

Figure 1: QFD Showing Design Requirements and Engineering Specifications

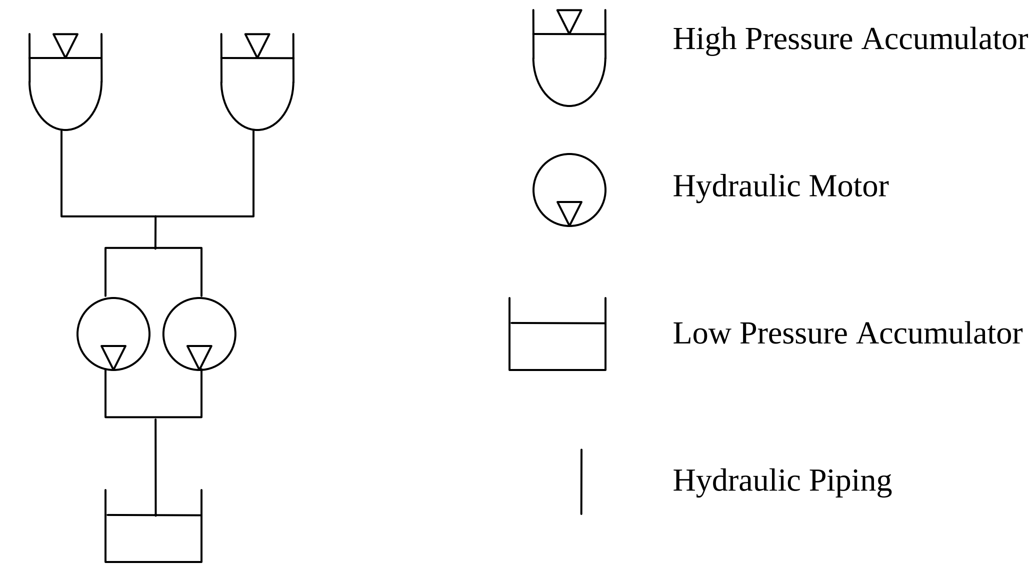


Figure 2: General layout of hydraulic system and the symbols used in all schematic

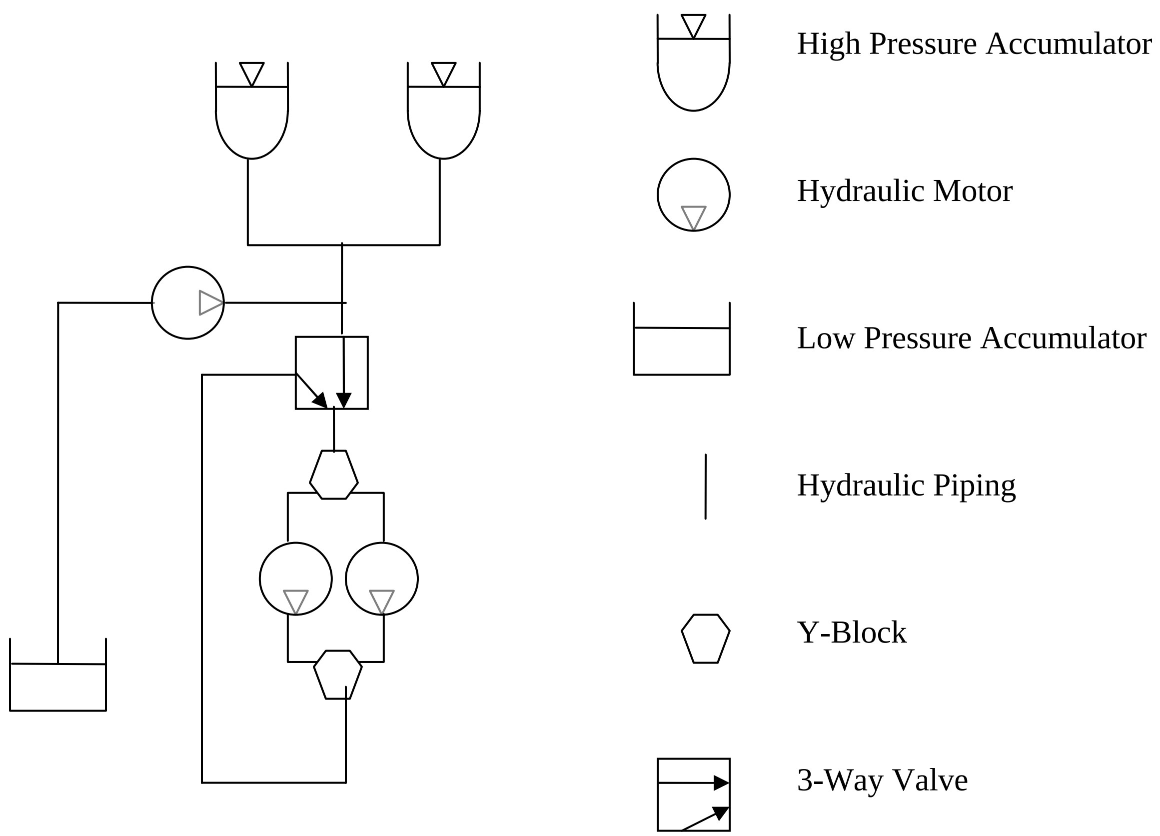


Figure 3: Hydraulic system using only one 3-way valve

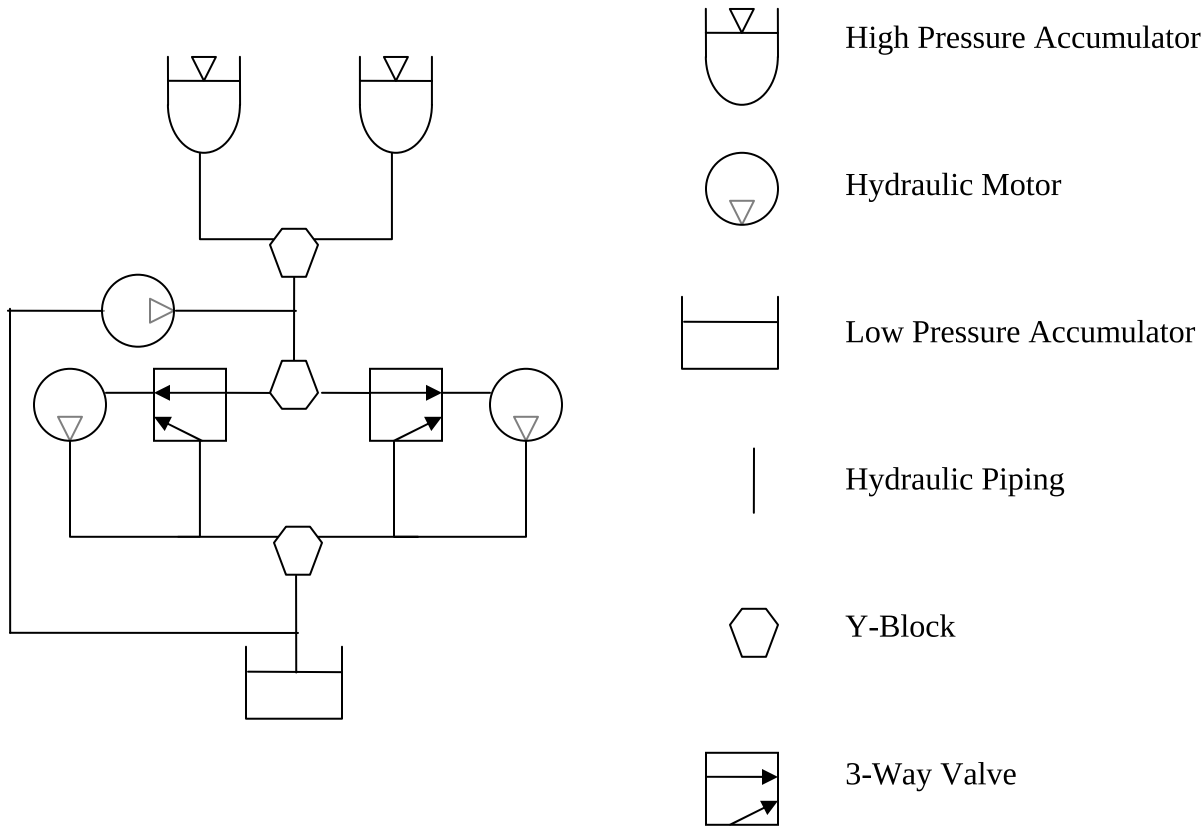


Figure 4: Hydraulic system using two 3-way valves

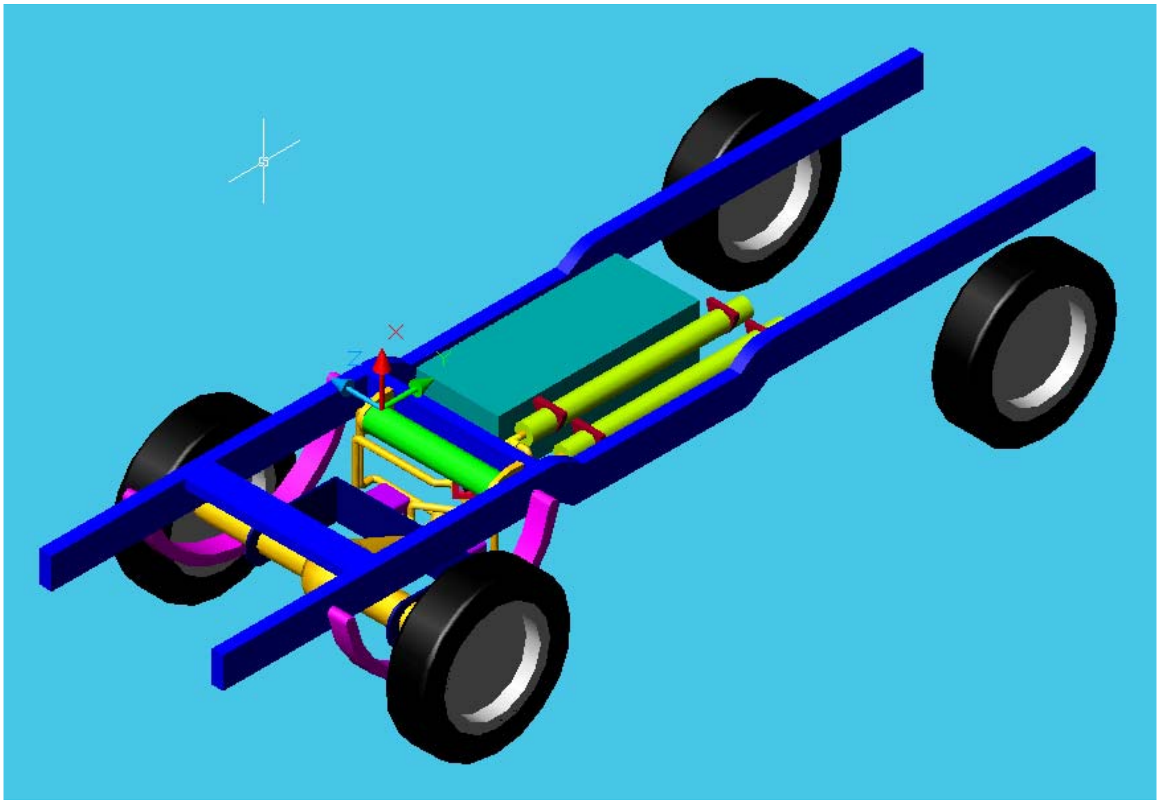


Figure 5: Hydraulic launch assist system shown on Ford Ranger chassis

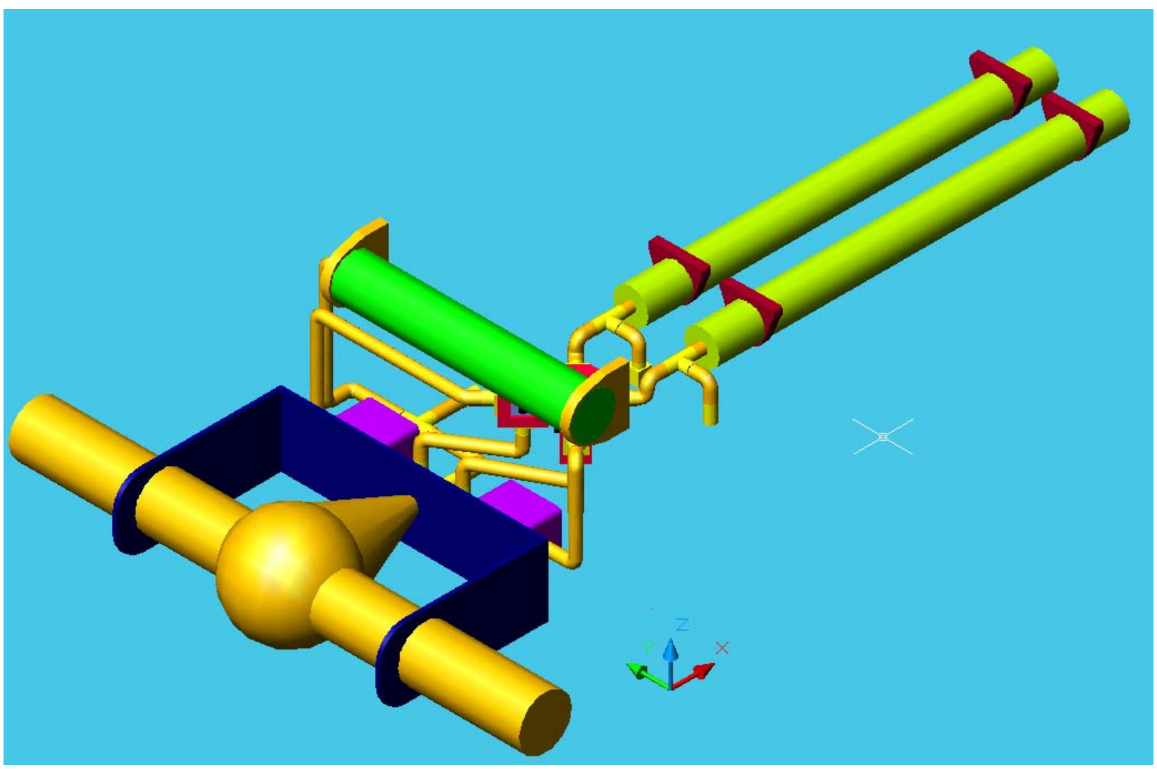


Figure 6: Compact layout of the hydraulic system, showing major components

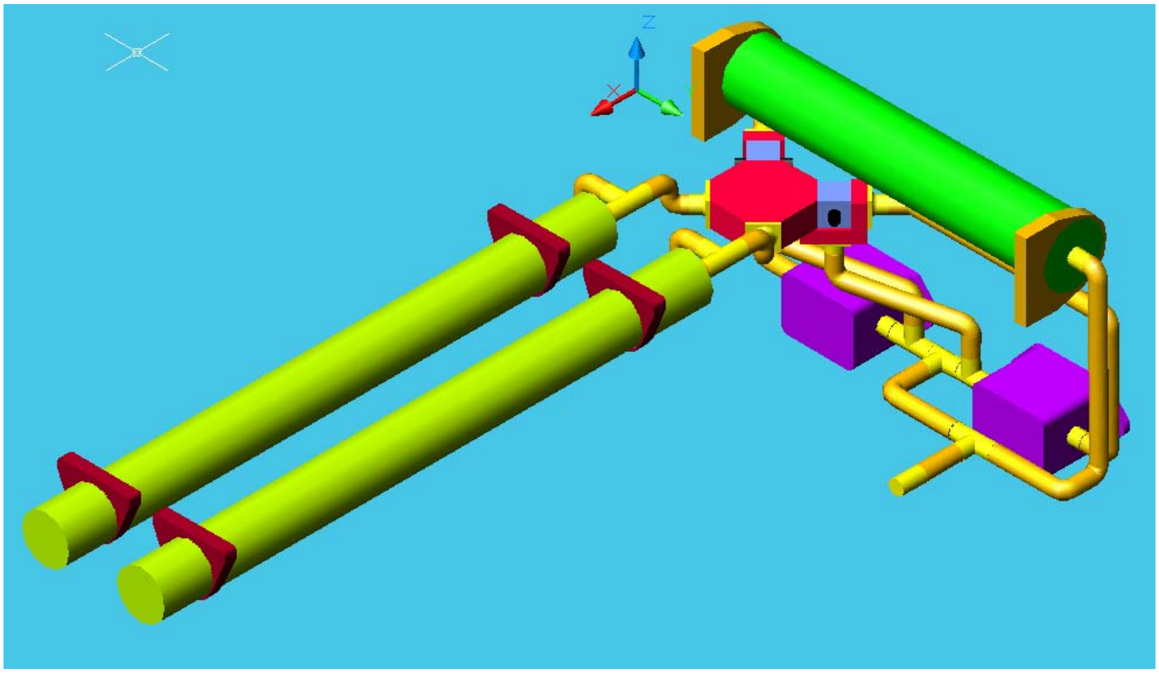


Figure 7: View of hydraulic hose layout

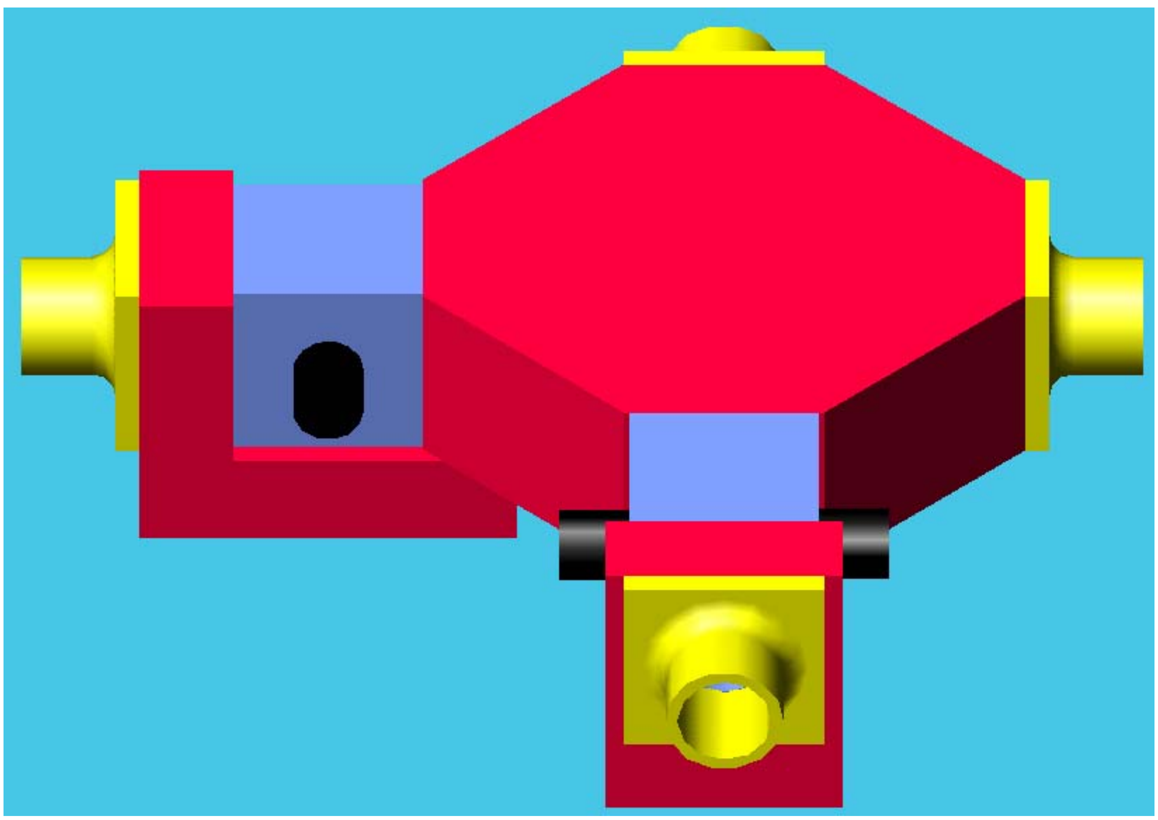


Figure 8: Combined manifold and 3-way valve assembly (note purple modules are fast-acting hydraulic control valves)

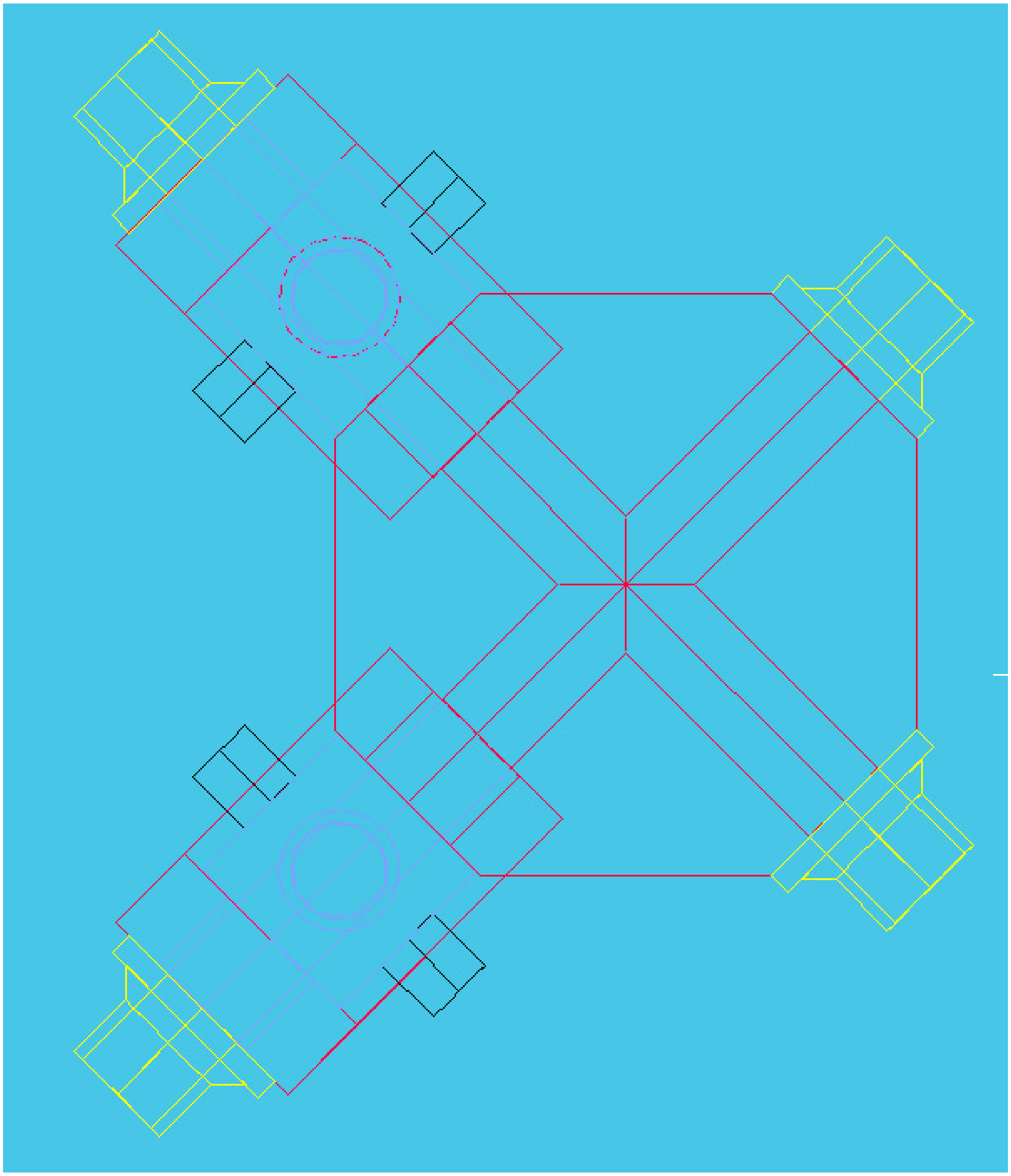


Figure 9: Top view of manifold and control valve assembly

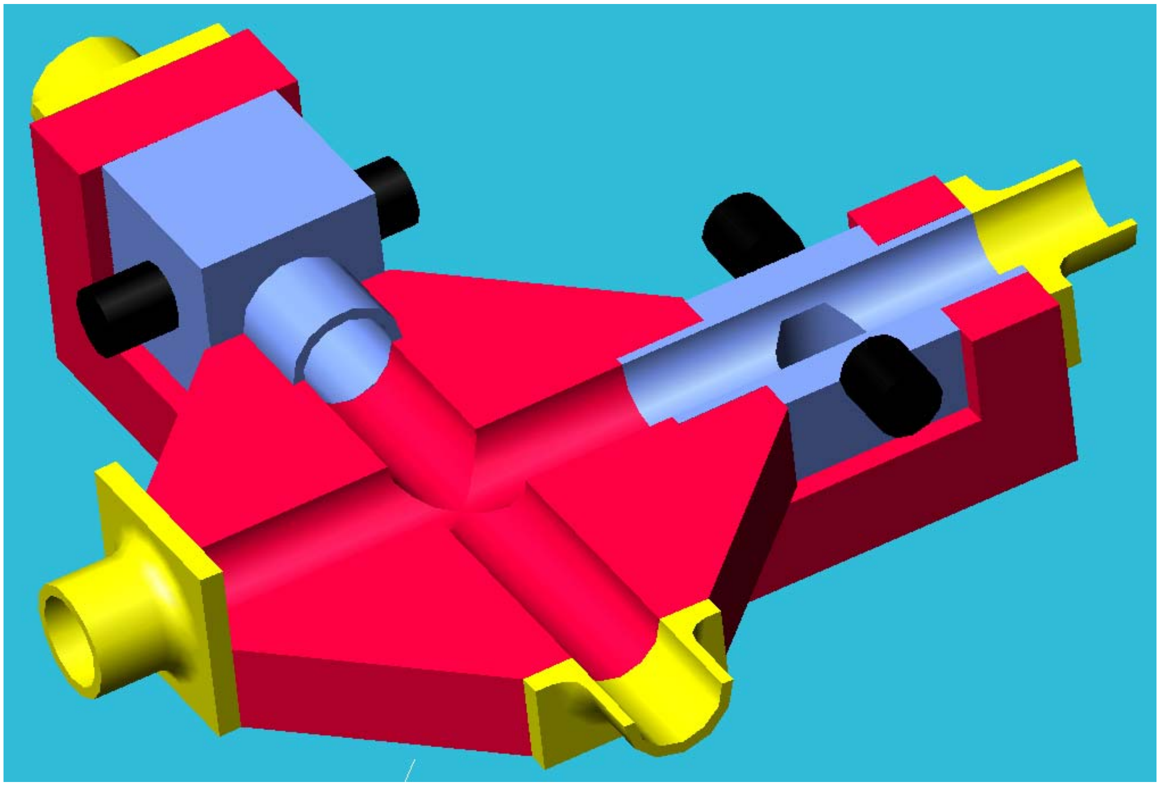


Figure 10: Cutaway View of the Manifold/Valve Assembly

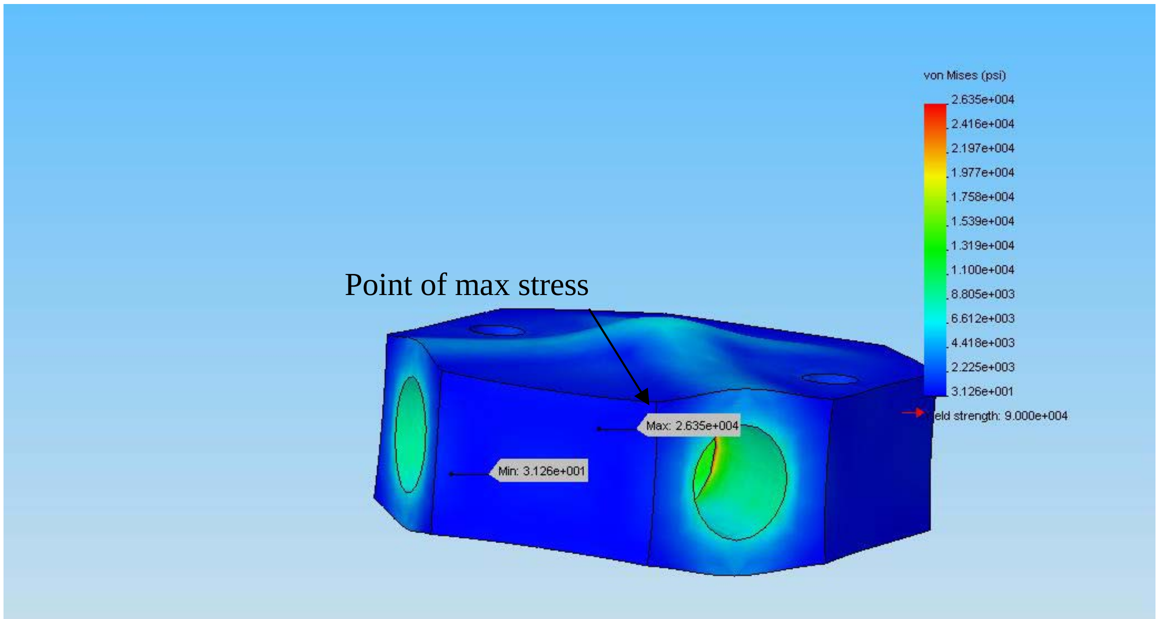


Figure 11: Stress Distribution on Manifold/Valve Assembly

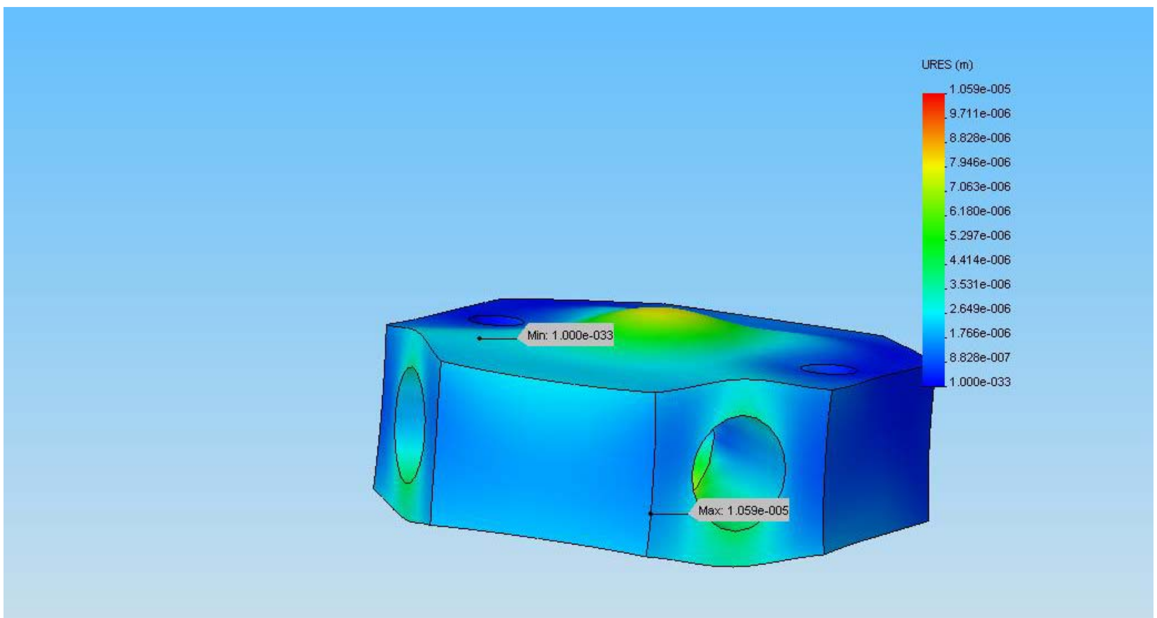


Figure 12: Displacement of Manifold/Valve Assembly Under Pressure

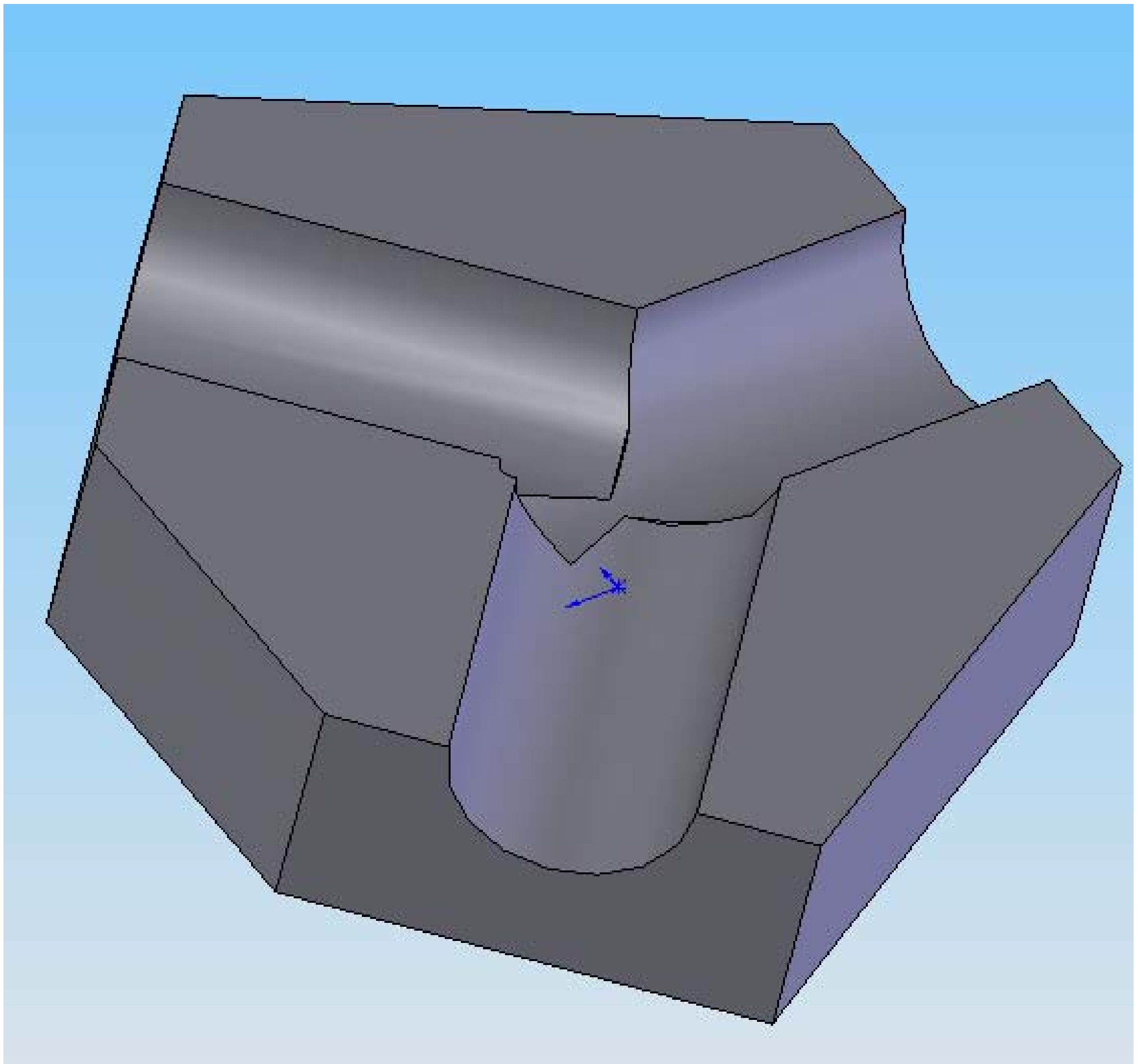


Figure 13: Cutaway View of Y-block

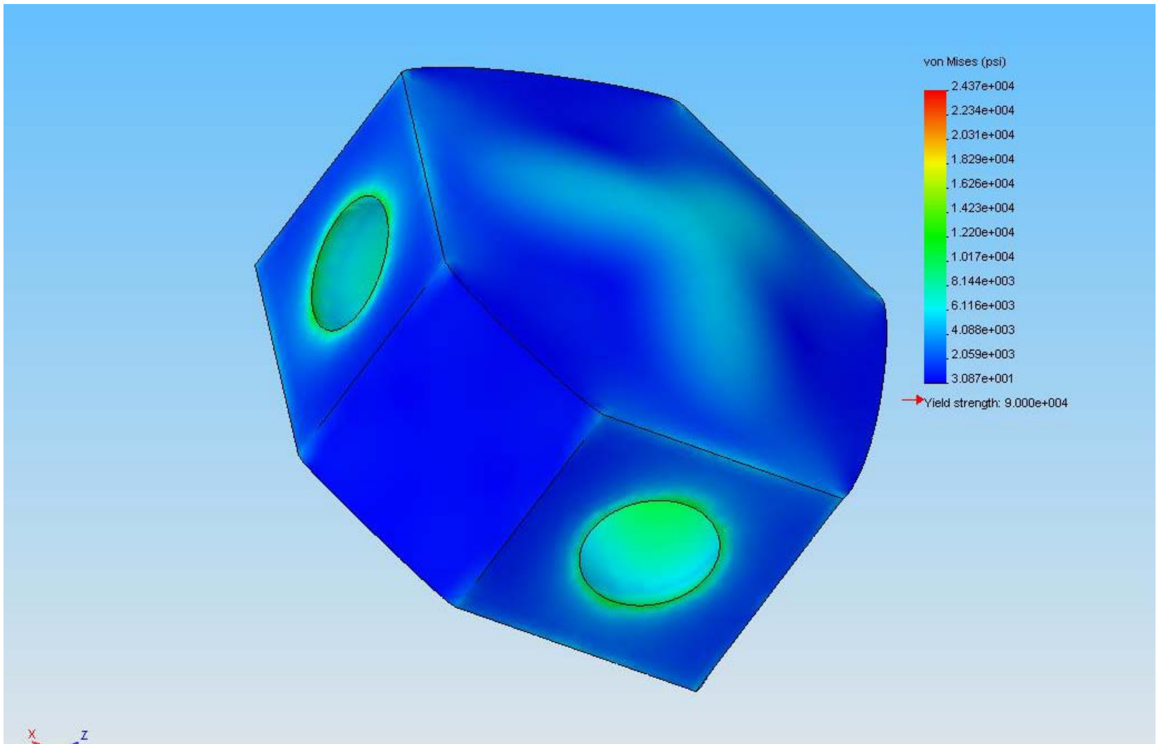


Figure 14: Stress Distribution on a Y-block

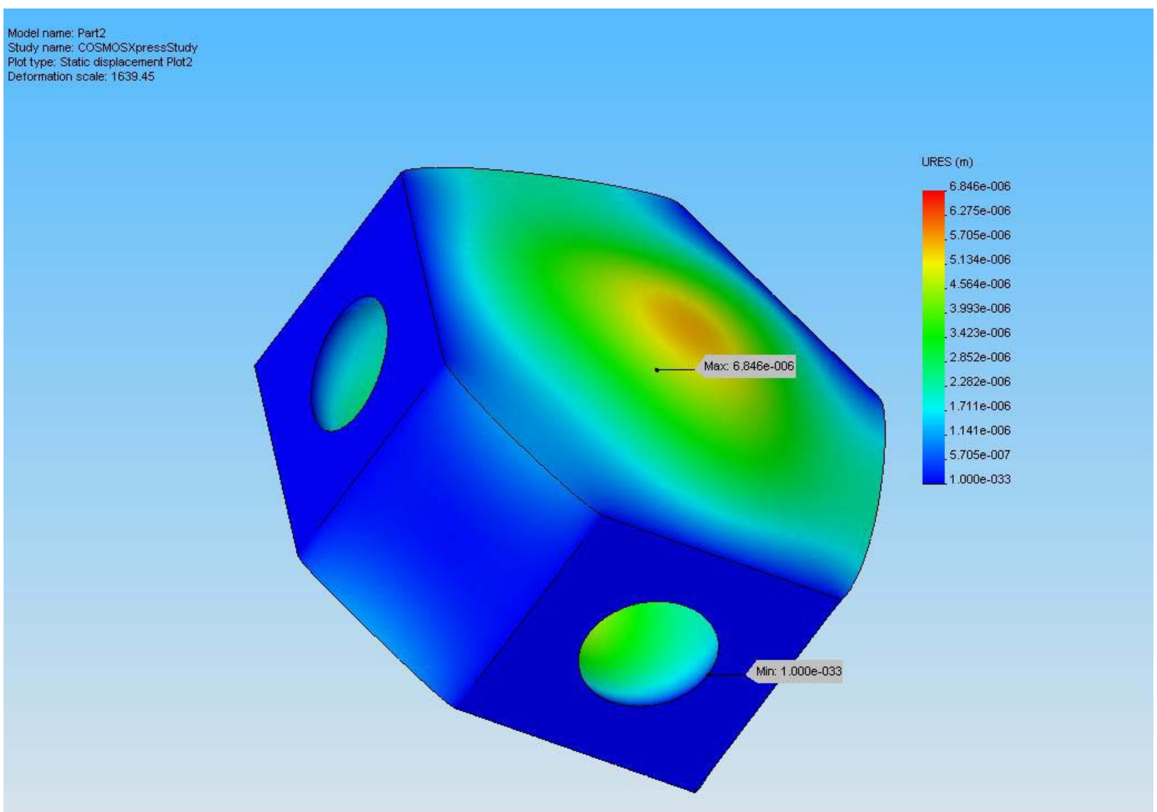


Figure 15: Displacement under Pressure of Y-block

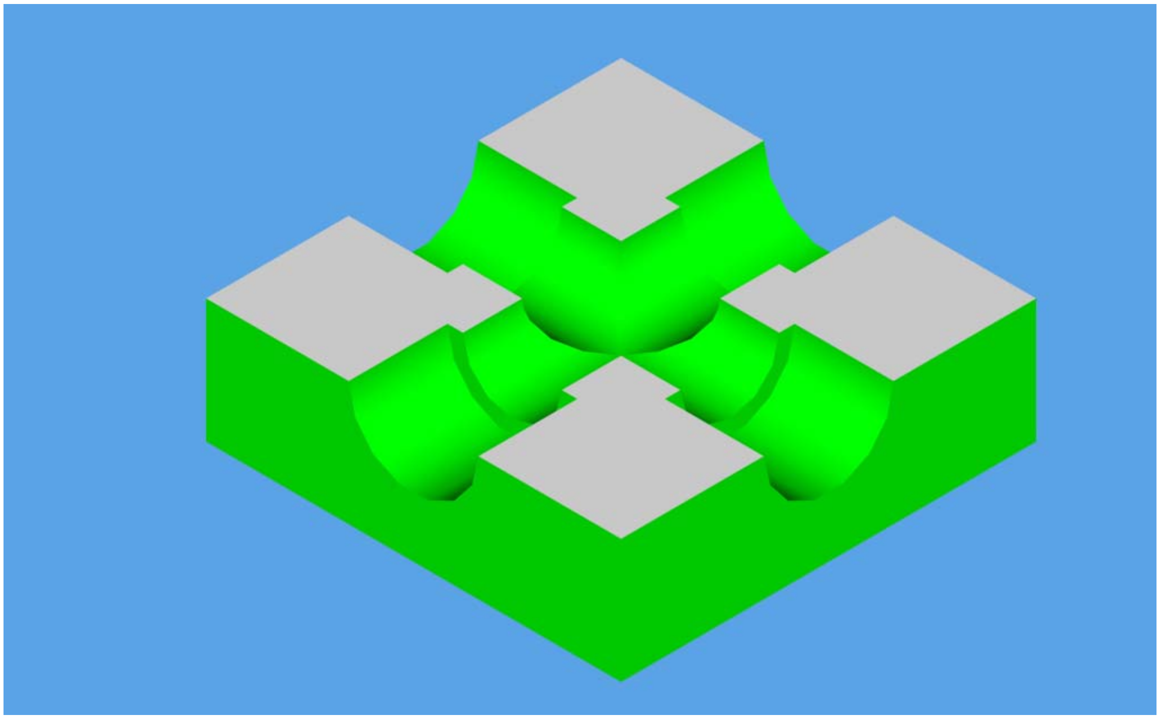


Figure 16: Cutout of our new manifold design

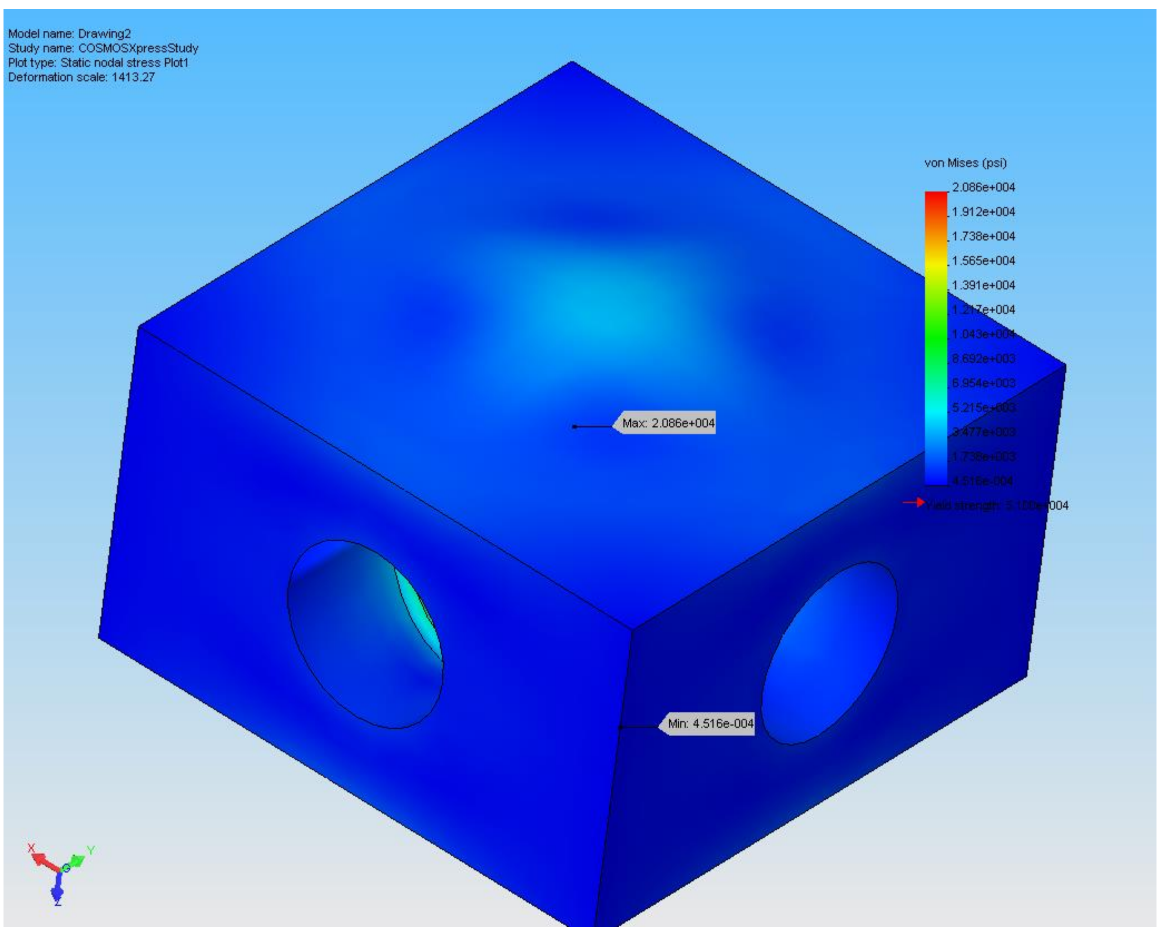


Figure 17: Stress Analysis on the new manifold design, produced safety factor of 2.44.

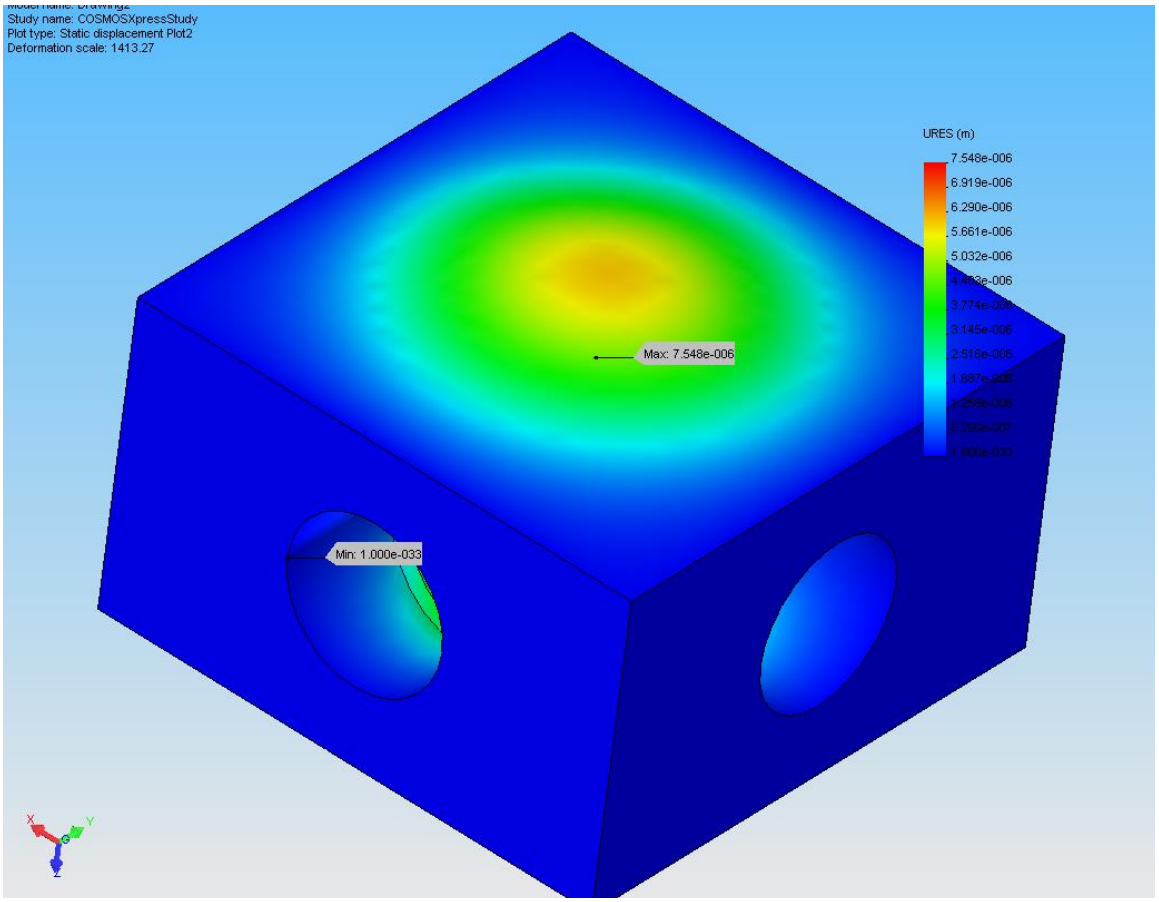


Figure 18: Displacement analysis on the new manifold design. Max displacements were negligible.

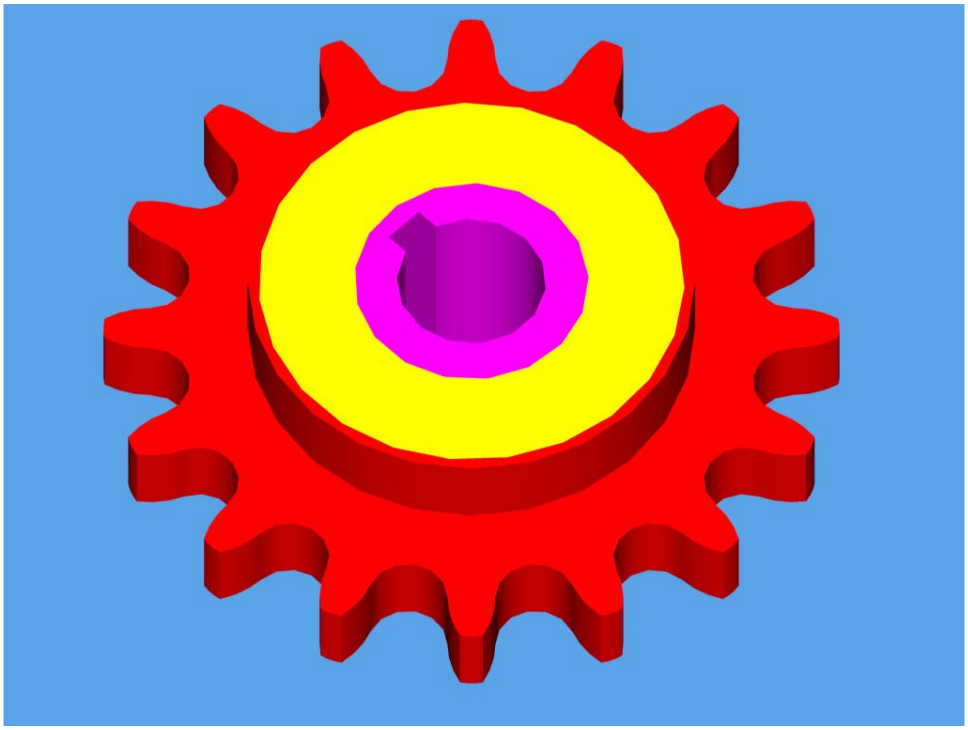


Figure 19: Assembled drawing of Mechanical Diode. The yellow section represents radial bearings to support loads.

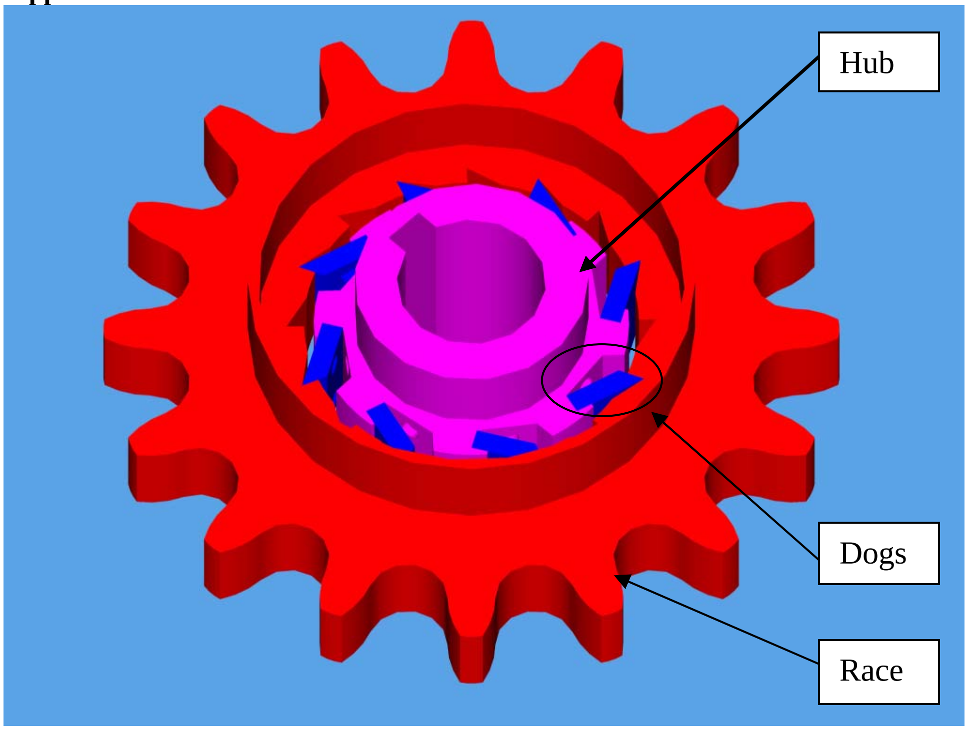


Figure 20: Inside view of Mechanical Diode. The purple center represents the Hub, the blue teeth represent the Dogs, and the red sprocket represents the Race.

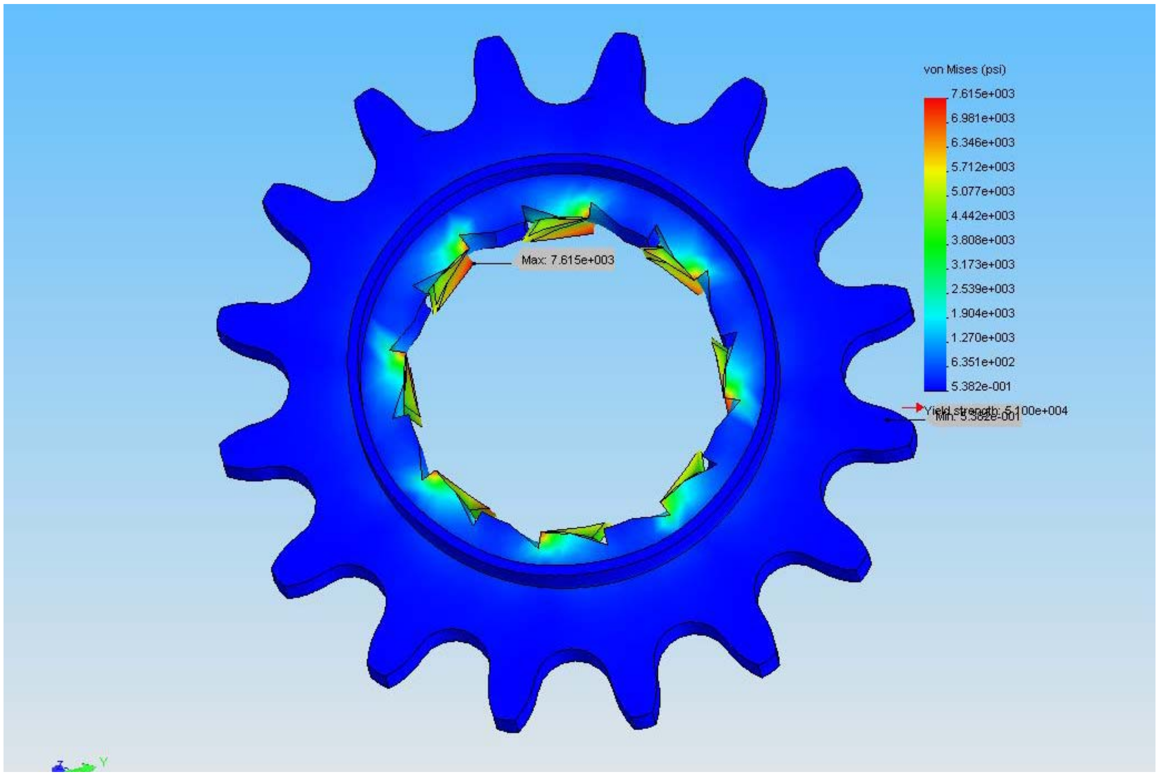


Figure 21: Stress analysis on the Race producing a safety factor of 6.6.

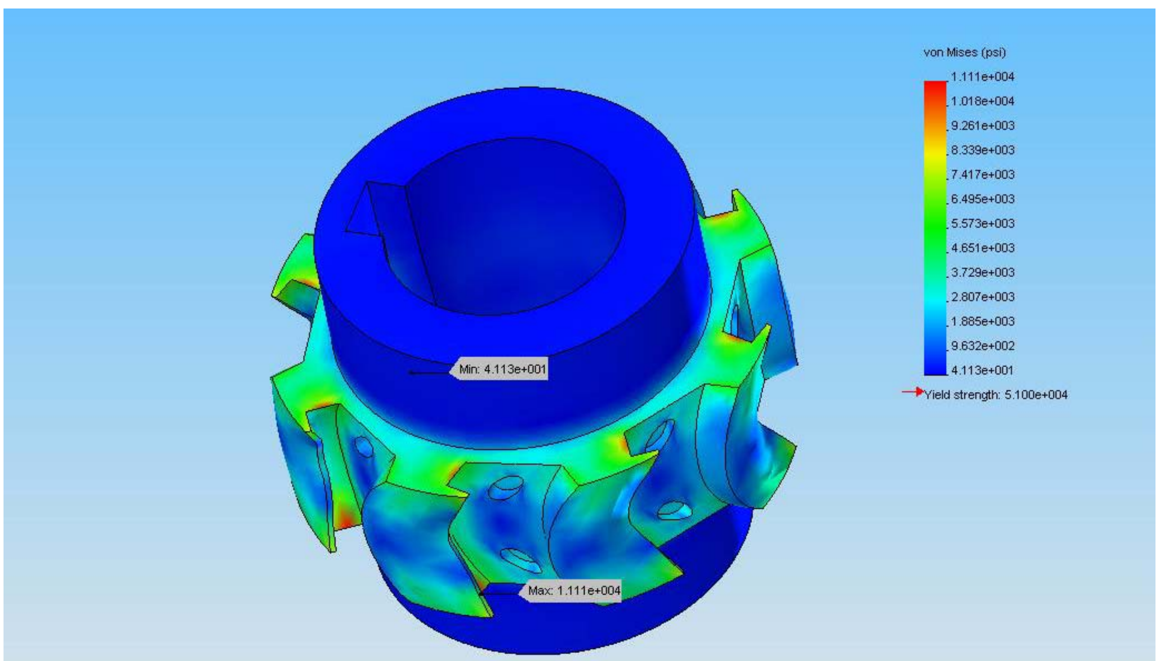


Figure 22: Stress analysis on the Hub producing a safety factor of 3.35.

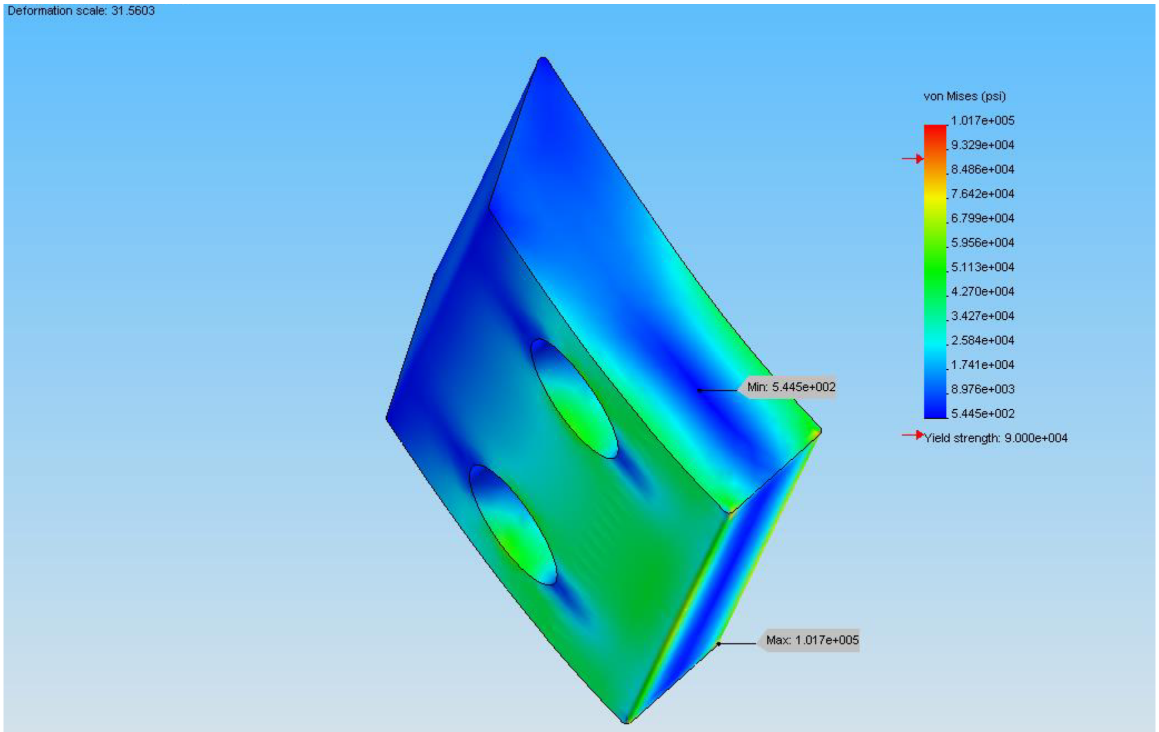


Figure 23: Stress analysis on a Dog producing a safety factor of 0.88. Fracture occurs at the corner, this part should be redesigned in order to produce a higher factor of safety.

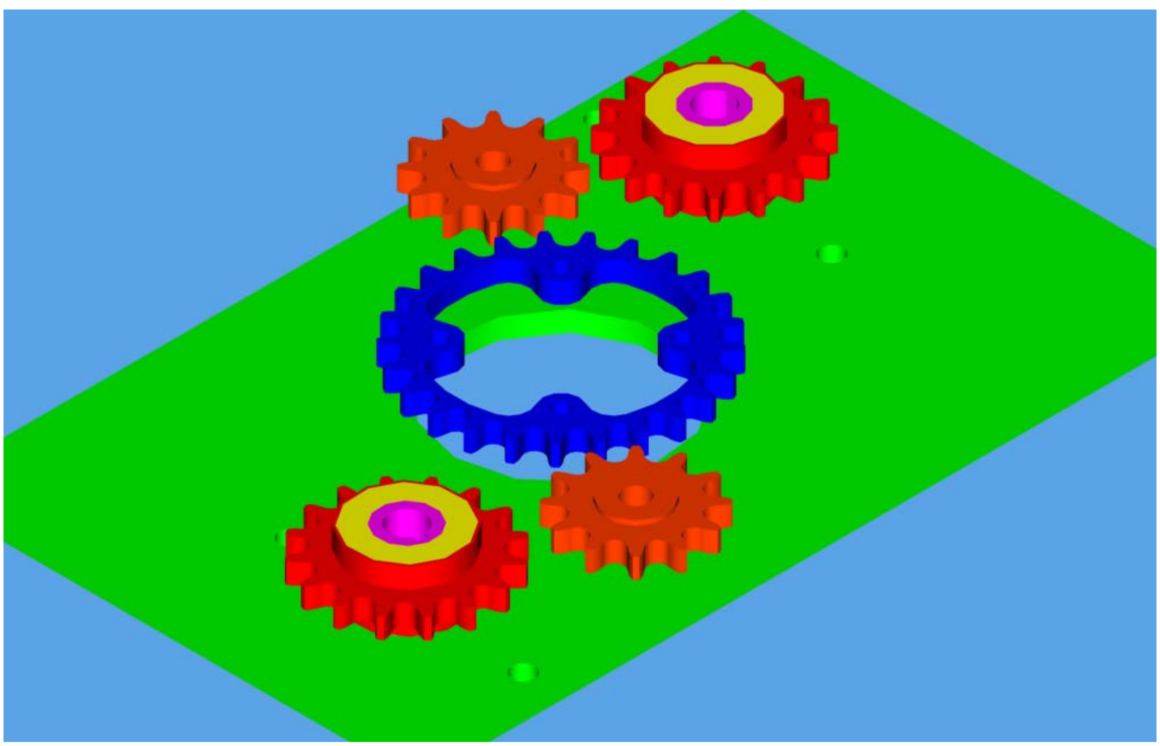


Figure 24: Isometric view of transmission system. The chain has been removed for clarity.

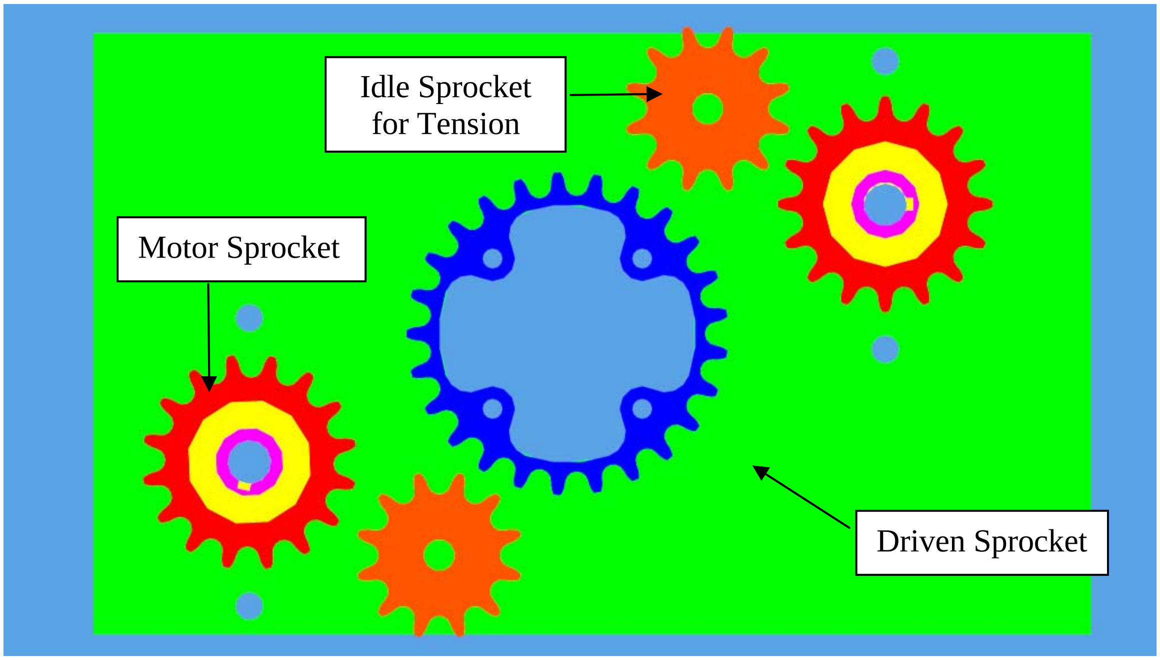


Figure 25: Top view of transmission system. The chain has been removed for clarity.

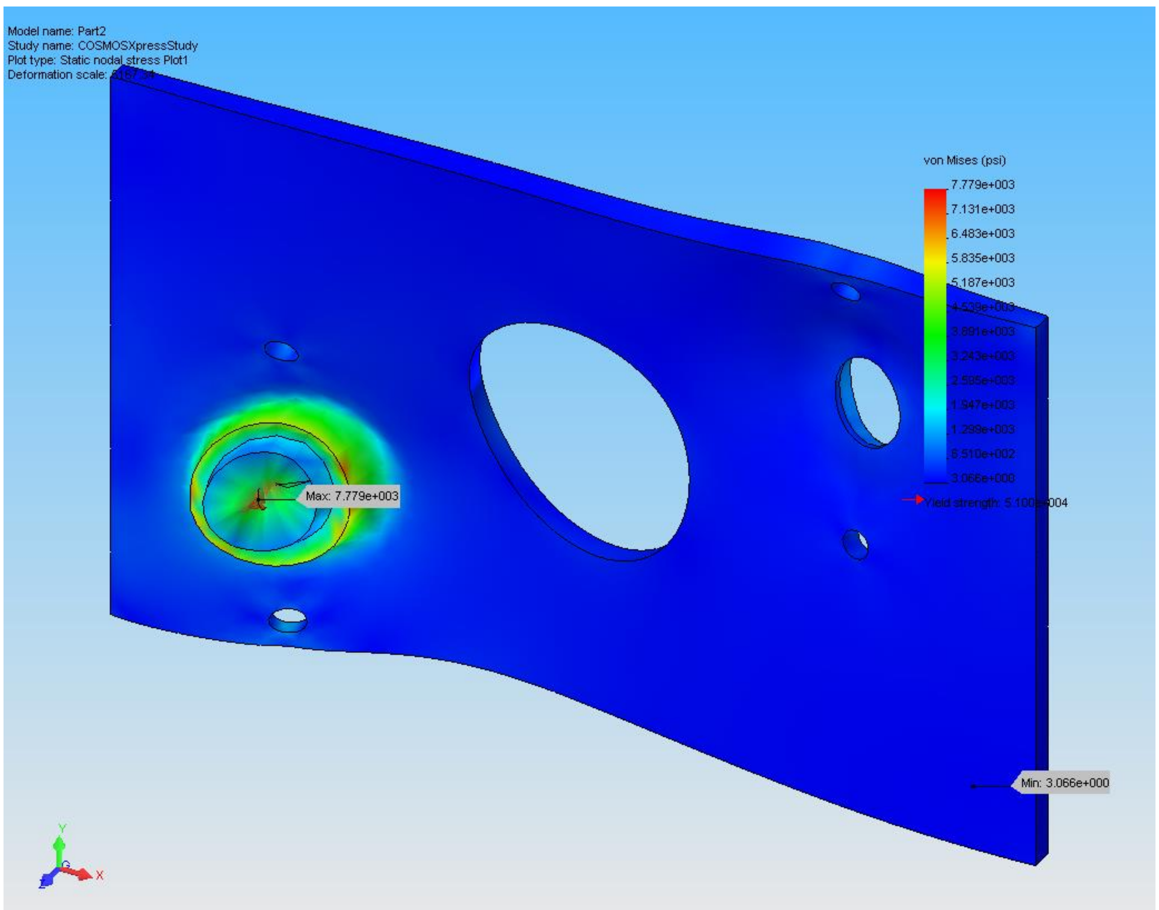


Figure 26: Stress analysis on the motor plate producing a safety factor of 6.5.

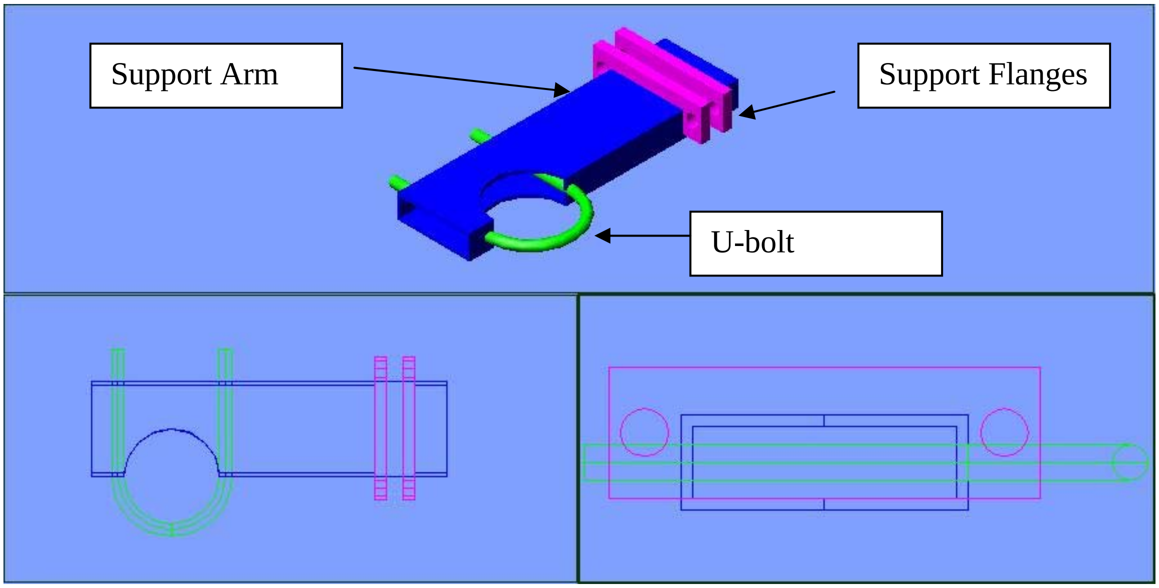


Figure 27: CAD drawing of motor plate support arm, showing the U-bold and mounting flanges.

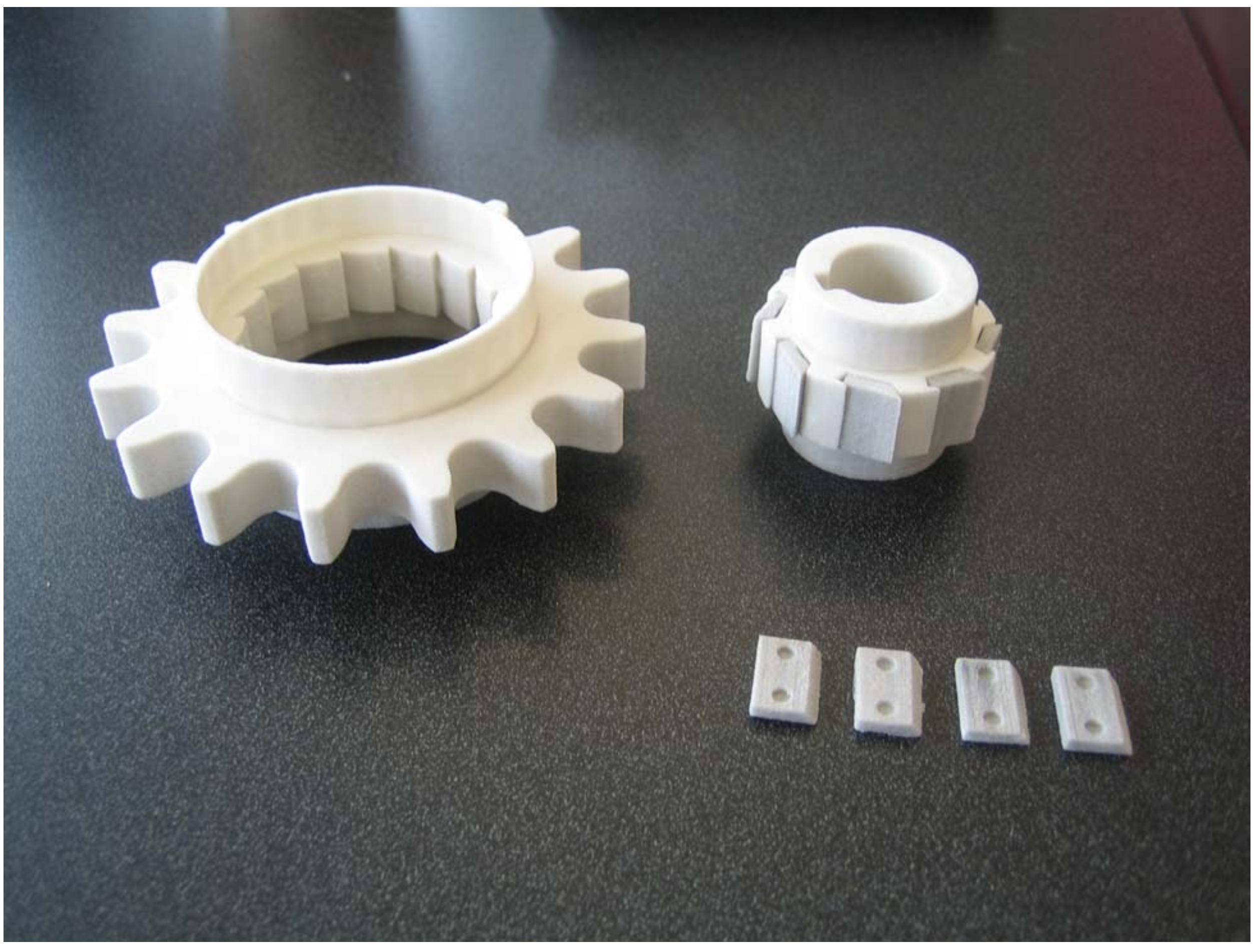


Figure 28: Rapid prototype of one-way clutch mechanism showing the race, the hub, and the dogs.

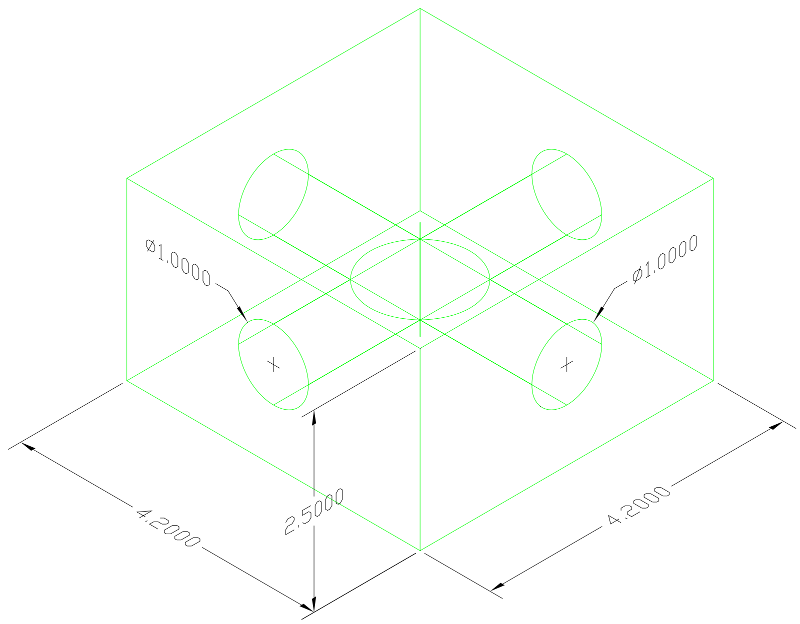


Figure 29: Engineering drawing for the redesigned manifold.

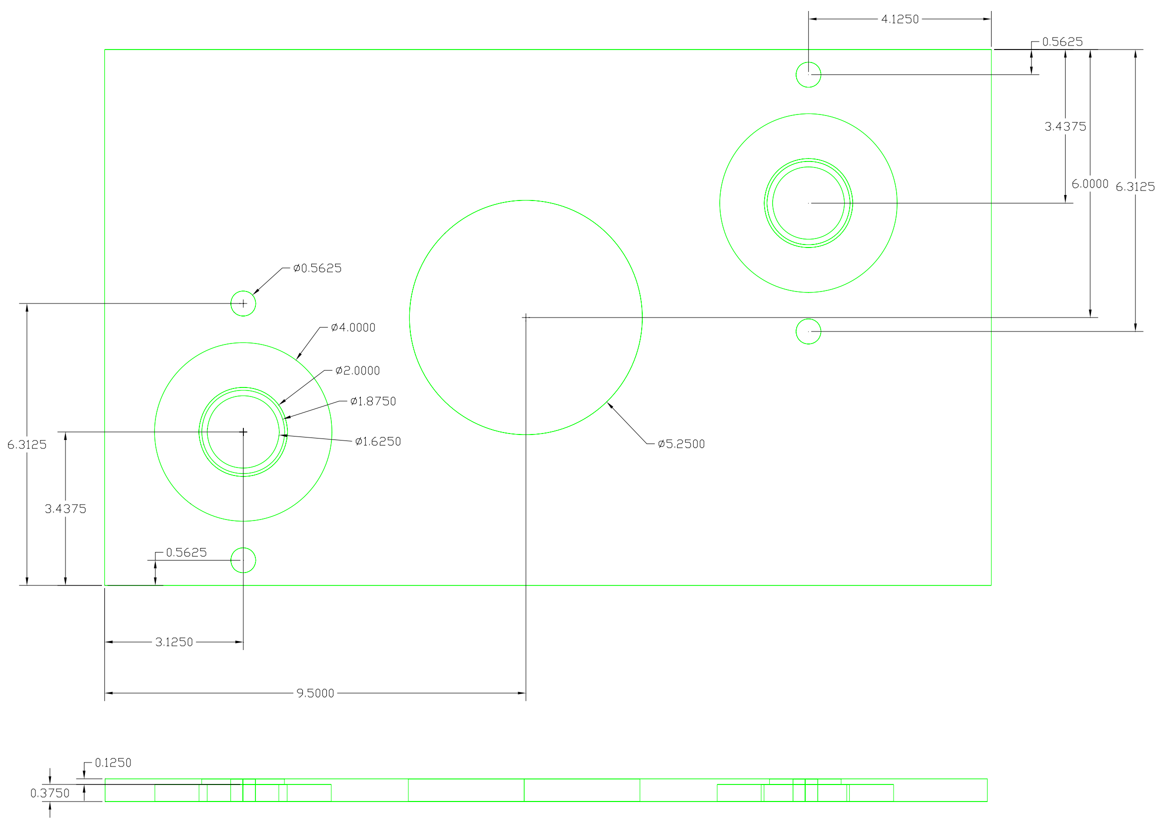


Figure 30: Engineering drawing for the motor plate.

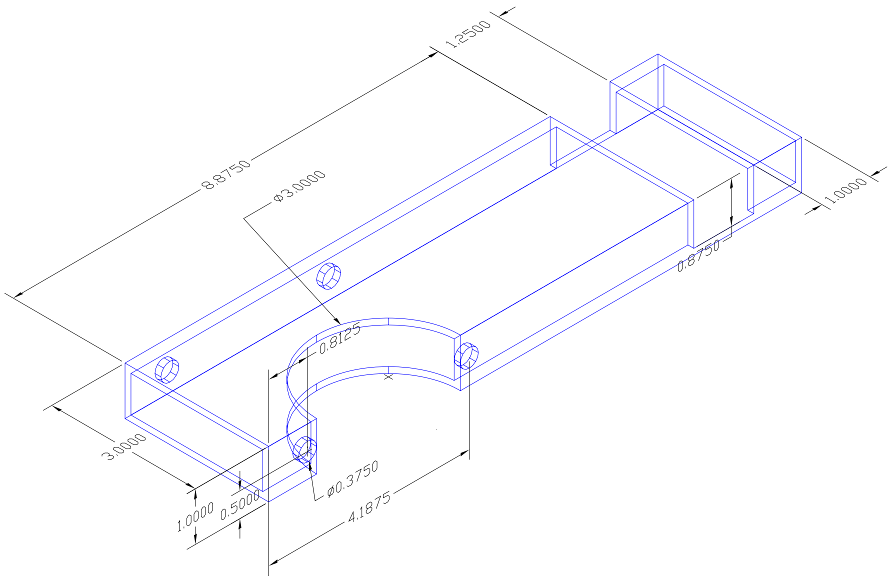


Figure 31: Engineering drawing for the support arm.

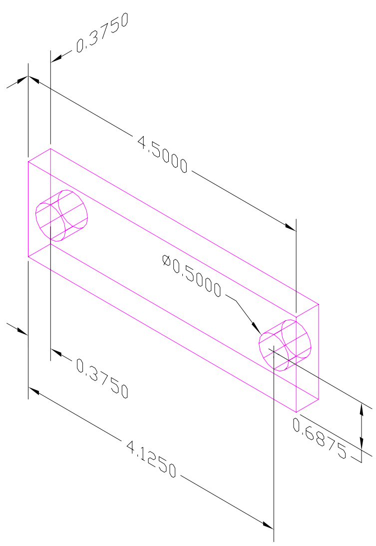


Figure 32: Engineering drawing for the support arm flange.

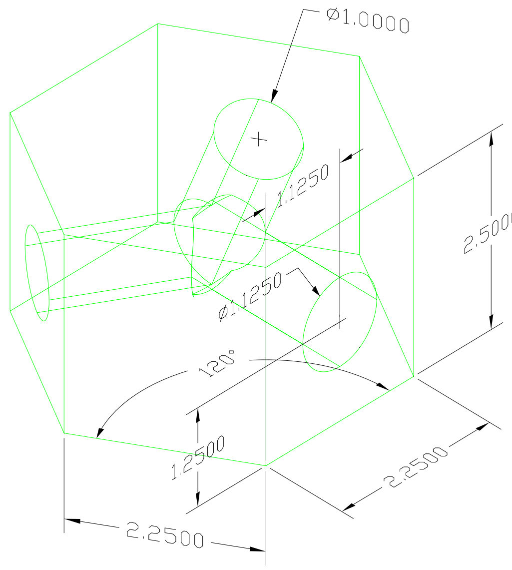


Figure 33: Engineering drawing for the Y-block assembly.

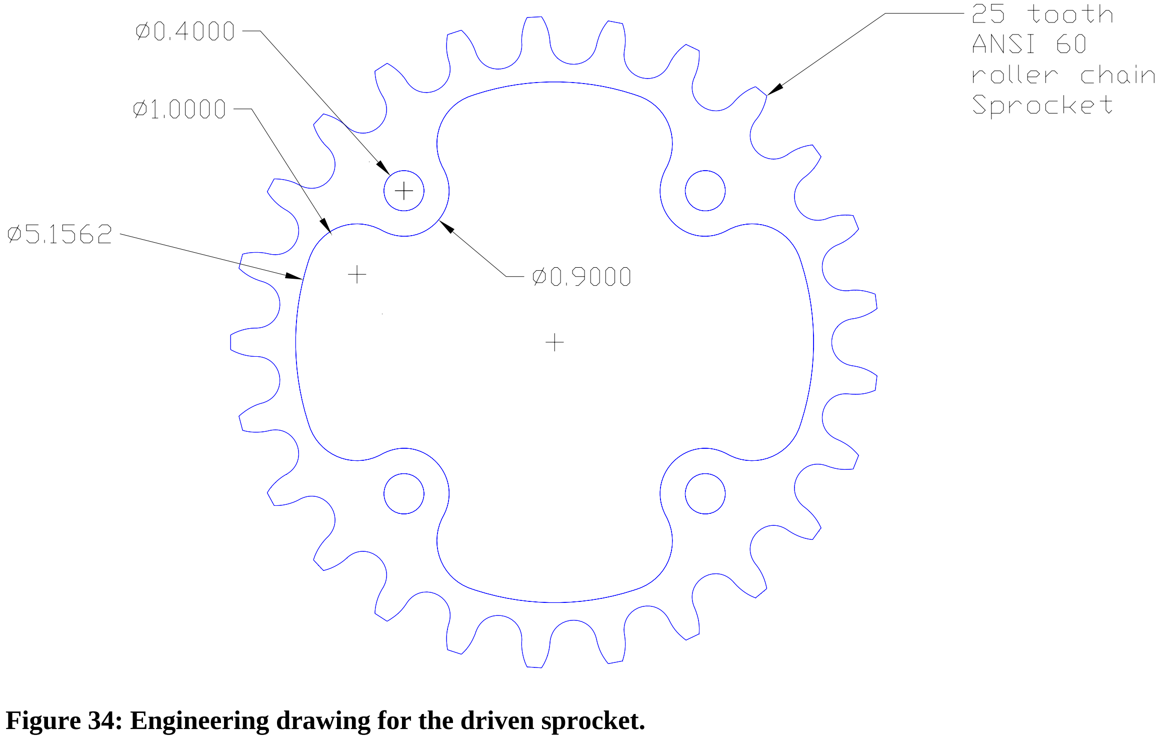


Figure 34: Engineering drawing for the driven sprocket.

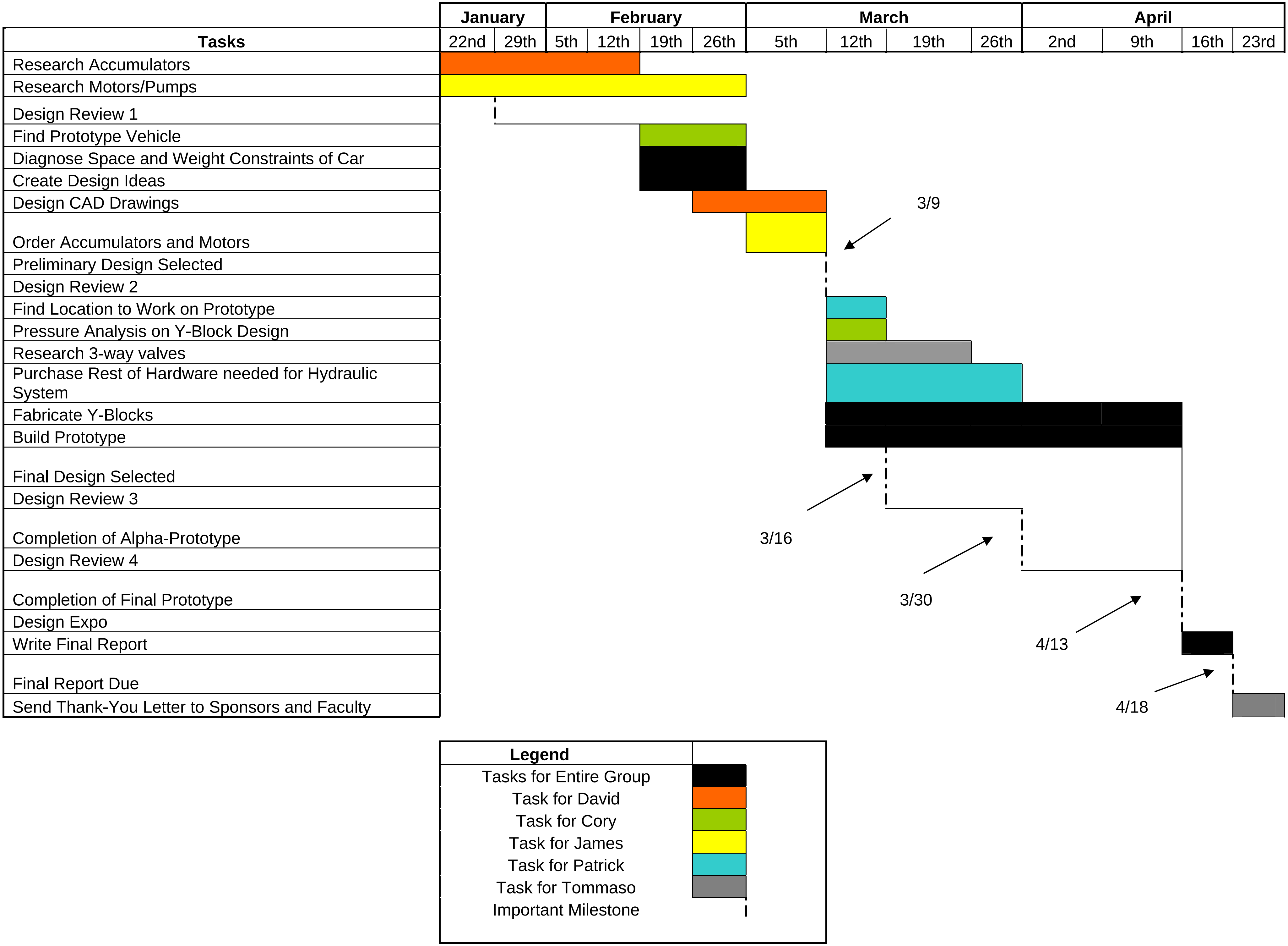


Figure 35: Gantt Chart

Appendix 2: Calculations of Accumulator Power

I used a 2100# pre-charge and a 3900# working pressure (motor/pump limited).

Rearranging the Ideal gas law--- $V_2 = \frac{P_1}{P_2} V_1 = \frac{2100\,lbs}{3900\,lbs} 8\,L = 4.307\,L$

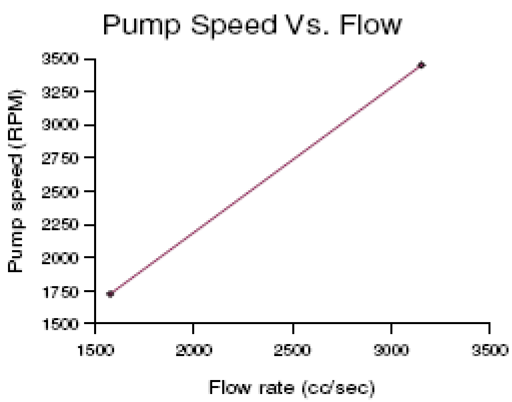
That means we have a working volume of 3.692L per accumulator. Two Accumulators yields a working volume of 7.386L. If we assume a 5% loss in the working volume due to temperature effects then we end up with an approximate working fluid volume of 7L.

The pump curves for the Marzocchi units have a linear flow speed relationship which can be derived from the data sheet.

Pump Models	*Displacement (cc)*	*Flow at (GPM)*	
		1725 rpm	3450 rpm
3A M 33	21.65	9.86	19.73
3A M 50	33.19	15.12	30.25
Combined	54.84	24.98	49.98

Converting the flow to SI units we get:

1576 cc/sec @ 1725 rpm
3153 cc/sec @ 3450 rpm



From the graph we see the flow rate needed to maintain 2000 RPM will be about 1850 cc/sec.

Using the equation for the line:

$$Flow\ Rate = \frac{Speed\,(rpm) - 1.094}{1.094}$$

We find the flow rate to be 1827 cc/rpm.

So, at 1850 cc/sec and a capacity of 7L we we find that we have 3.78 seconds of boost available.

To calculate the power output of the add-on system we would need to have some acceleration data available. We do not so a less accurate method of using average values is shown below:

$$Power = Disp\,(l/rev) \times rps \times Pressure\,(KPA) = 37432.75\,W = 50.198\,hP$$

A function for the pressure drop as a function of time can be easily derived once the speed-time relation ship is know; however, we would need some basic data to formulate a function relating motor speed as a function of time.

Torque is a little easier to determine. Again the calculations below will use averages; however, the pressure drop as a function of time will require the same data as needed for the power calculation. The average torque calculation is shown below:

$$Torque = \frac{Disp\,(l) \times Pressure\,(kPa)}{2\,\pi} = 180.5\,N - m = 133.2\,ft - lbf$$

Re-calcualted with 2400 lbs pre-charge:

$$V_2 = \frac{P_1}{P_2} V_1 = \frac{2400\,lbs}{3900\,lbs} 8\,L = 4.923\,L$$

That gives an effective working fluid volume of 6.154L which is a 16% reduction in working fluid volume. If we apply the same 5% volume losses due to temperature effects the working volume is further reduced to 5.846L and a resulting maximum working time of 3.16 seconds. This is again a 16% reduction.

The output power according to the supplied equations appears to go decrease with a decreased pre-charge; however, this flies in the face of reason because the same amount of work is done from 3900 pounds to 2400 pounds no matter what the pre-charge happens to be. In both cases the change in pressure per cc ends up being about the same and likewise the work done from 3900 to 2400 pounds should be about the same.

If we make the assumption that the work done on the piston inside the cylinder (essentially power out of the accumulator = power output from motors minus all losses) in an ideal situation is a polytropic process with n=1 then we can use:

$$W = P_1 V_1 \ln\left(\frac{P_2}{P_1}\right)$$

Using the above we see that we can only get 27 hp out of the accumulator at 2100 pound pre-charge and 33.28 hp with the 2400 pound pre-charge per accumulator. Again though these values were determined without directly integrating a series of equations relation pressure drop and acceleration and motor speed.

Using the 2400 pound pre-charge yields a higher power but that, in my opinion, comes from using simplified mathematical models. Additionally, the power output will be essentially the same for either pre-charge from 3900 to 2400 pounds.

Appendix 3: Calculations of flow rate through 3-way valves

1 valve
-motors displace maximum of 56 cc/rev
-run at a maximum of 3000 rpm

$$\frac{56cc}{rev} x \frac{3000rev}{\min} x \frac{1L}{1000cc} = 168 \frac{L}{\min}$$

2 valves
-motor displaces a maximum of 33 cc/rev
-run at a maximum of 3000 rpm

$$\frac{33cc}{rev} x \frac{3000rev}{\min} x \frac{1L}{1000cc} = 99 \frac{L}{\min}$$

Appendix 4: Bill of Materials for our Project

Item Description	Source	Model Number	Quantity	Price	Total	Contact
Smaller Motor	Exotic Automation	PGM517MA0230BM1H3ND6D6B1B1B1	1	$555.00	$555.00	Rick Roberts
Larger Motor	Exotic Automation	PGM517MA0330BM1H3ND6D6B1B1B1	1	$570.00	$570.00	Rick Roberts
Accumulator	Exotic Automation	ACP 10AA 800 E 1 K TE	2	$587.00	$1,174.00	Rick Roberts
Accumulator Brackets	Exotic Automation		4	$37.00	$148.00	Rick Roberts
Precharge for Accumulators	Televan	Precharge to 2400psi	2	$50.00	$100.00	Televan
3-Way Valves	Federal Fluid Power	HBV3-16-SAE	2	$188.00	$376.00	Mike/Grant
Manifold Steel	ASAP	1020 Steel	1	$80.00	$80.00	
Steel Ball Bearing	McMaster Carr	60355K18	6	$7.24	$43.44	mcmaster.com
Analog Pressure Gauges	McMaster Carr	25-300-5000i	4	$75.00	$300.00	mcmaster.com
Motor Plate Steel	ASAP	AISI 1018 Mild Steel	1	$320.00	$320.00	ASAP
Steel Ball Bearing	McMaster Carr	60355K22	6	$9.70	$58.20	mcmaster.com
18 Teeth Sprocket	McMaster Carr	2299K69	3	$19.59	$58.77	mcmaster.com
25 Teeth Sprocket	McMaster Carr	2299K77	3	$24.13	$72.39	mcmaster.com
Steel Roller Chain	McMaster Carr	6261K34	1	$86.40	$86.40	mcmaster.com
#60 Connecting link	McMaster Carr	6261K195	10	$1.04	$10.40	mcmaster.com
12 TeethIdler Sprocket	McMaster Carr	6663K206	3	$25.87	$77.61	mcmaster.com
16 Teeth Chain Sprocket	McMaster Carr	6280K473	2	$23.53	$47.06	mcmaster.com
16 Teeth Flat Sprocket	McMaster Carr	2299K67	3	$18.33	$54.99	mcmaster.com
3 gal Hydraulic Reservior	McMaster Carr	1364K33	1	$175.11	$175.11	mcmaster.com
Pressure Relief Valve	Federal Fluid Power	DERVDOO4000S	1	$39.75	$39.75	Mike/Grant
End Mill Bits	McMaster Carr	88815A45	3	$30.77	$92.31	mcmaster.com
End Mill Bit	McMaster Carr	2849A84	1	$38.22	$38.22	mcmaster.com
End Mill Bit	McMaster Carr	3046A24	1	$58.26	$58.26	mcmaster.com
1" Drill Bit	McMaster Carr	8947A235	1	$39.17	$39.17	mcmaster.com
9/16" Drill Bit	McMaster Carr	8947A221	1	$14.31	$14.31	mcmaster.com
2" Crescent Wrench	McMaster Carr	5386A7	1	$56.94	$56.94	mcmaster.com
3/8" ID Steel Pipe	ASAP	1/2" OD, 3/8" ID 1020 Steel	1	$9.36	$9.36	ASAP
MDF Particle Board	Home Depot	1/2" x 2' x 4'	2	$6.67	$13.34	Home Depot
3/8" Nuts, Bolts, & Washers	Home Depot		70	$0.50	$35.00	Home Depot
Drill Bits & Star Sockets	AutoZone		1	$15.00	$15.00	AutoZone
Saw Blades	Home Depot		3	$5.00	$15.00	Home Depot
Cutting Discs	Lowe's		1	$5.98	$5.98	Lowes
			Grand Total		**$4,740.01**	

Appendix 5: Bill of Materials for Fittings from Tompkins Industries

Item Description	Source	Model Number	Quantity	Price	Total	Contact
JIC Union Cross (all Male)	Tompkins Industries	2650-16-16-16-16	3	$40.42	$121.26	tompkinsind.com
O-Ring to JIC Swivel (M-F)	Tompkins Industries	6402-16-16	4	$29.47	$117.88	tompkinsind.com
JIC Reducer 16-4 (F-M)	Tompkins Industries	2406-16-04	4	$29.61	$118.44	tompkinsind.com
JIC Swivel to Pipe (F-F)	Tompkins Industries	6506-04-04	4	$9.25	$37.00	tompkinsind.com
JIC Swivel Elbow 45^{o} (M-F)	Tompkins Industries	6502-16-16	10	$22.76	$227.60	tompkinsind.com
JIC to O-Ring Elbow 90^{o} (M-M)	Tompkins Industries	6801-16-12	2	$20.55	$41.10	tompkinsind.com
JIC to O-Ring Elbow 90^{o} (M-M)	Tompkins Industries	6801-16-16	2	$23.04	$46.08	tompkinsind.com
JIC Swivel Run Tee (F-M-M)	Tompkins Industries	6602-16-16-16	4	$29.38	$117.52	tompkinsind.com
JIC to O-Ring Elbow 45^{o} (M-M)	Tompkins Industries	6802-16-12	2	$28.89	$57.78	tompkinsind.com
JIC to O-Ring Elbow 45^{o} (M-M)	Tompkins Industries	6802-16-16	4	$22.93	$91.72	tompkinsind.com
JIC Reducer 16-8 (F-M)	Tompkins Industries	2406-16-08	3	$16.58	$49.75	tompkinsind.com
O-Ring to JIC Swivel (M-F)	Tompkins Industries	6402-08-08	3	$8.72	$26.16	tompkinsind.com
JIC to O-Ring Elbow 90^{o} (M-M)	Tompkins Industries	6801-08-08	1	$6.79	$6.79	tompkinsind.com
JIC Union Tee (All Male)	Tompkins Industries	2603-24-24-24	1	$85.26	$85.26	tompkinsind.com
JIC Reducer (F-M)	Tompkins Industries	2406-24-16	3	$86.86	$260.58	tompkinsind.com
JIC Swivel Elbow 90^{o} (F-M)	Tompkins Industries	6500-16-16	2	$21.23	$42.46	tompkinsind.com
JIC to BSPP Adjustable Run Tee (All Male)	Tompkins Industries	3859-08-08-06	1	$41.85	$41.85	tompkinsind.com
JIC Swivel Run Tee (F-M-M)	Tompkins Industries	6602-16-16-16	1	$29.38	$29.38	tompkinsind.com
JIC to O-Ring (M-M)	Tompkins Industries	6400-08-08	1	$2.29	$2.29	tompkinsind.com
			Grand Total		**$1,520.90**	

Appendix 6: Description of Engineering Changes Since DR#3

There are two major differences between our Final Design and our prototype. One of these changes occurs in our choice of 3-way valves. We would like to use two, electronically actuated solenoid valves. Unfortunately our system requires very high flow rates, on the scale of 100 L/min. The most efficient solenoid valves produce a pressure drop of 250 psi across the valve at these flow rates. Our project is designed around efficiency, a loss of pressure that significant would reduce our working pressure by about 7%. We chose to use manually actuated 3-way ball valves instead. These valves only produce a pressure drop around 70 psi, which is only a loss of about 2%. In addition to being much more efficient, these valves are also much cheaper. The downside occurs with the method of actuation. We need to design a linkage between the two valve handles in order to actuate them at the same time.

The second major difference occurs in our manifold design. Since we were not directly attaching solenoid valves to the manifold, we could reduce the size and amount of material. We were able to simplify the octagonal shape to a square shape. This new design can be seen in Figure 16, while figures 17 & 18 show stress and displacement analysis. This block is designed to be 4" x 4" x 2.5" made from mild steel, and maintains a safety factor of 2.44. Other materials could be stronger and lighter, but harder to machine.

Appendix 7: Biographical Information

Patrick Franklin

I was born on Ft. Lewis Army Base near Seattle, Washington and lived in Spanaway, Washington until I was 14. I then moved to Ypsilanti, Michigan where I graduated from Ypsilanti High School in 2001. I attended Washtenaw Community College until I transferred to The University of Michigan College of Engineering in the winter 2004 semester. During the summer of 2004 I attended Leader's Training Course, a training camp for Army ROTC for cadets joining their junior year. During the 2004-05 school year I continued to pursue my Bachelors Degree in Mechanical Engineering and continued training for Army ROTC. During the summer of 2005 I attended Warrior Forge, the pinnacle of cadet training and assessment prior to commissioning. I plan on graduating and commissioning with the rank of Second Lieutenant in the U.S. Army in December 2006. I will serve active duty for four years and then reserves for four years. I do not know if I will pursue a military or civilian career. If I do opt for a civilian career I would like to work for General Dynamics, Boeing, or the automotive industry. I am interested in mechanical engineering because as a child I was the one who would dismantle radios and toys and put together new things. I also hold a part time job as a bus driver for UM.

James Wolbers

From: Northville, MI
Education: Mechanical Engineer, U of M
Hobbies: All sports, video games, movies, hanging out with friends, snowboarding.
Favorite Drink: Chocolate Milk, Guinness.

Favorite Car: Porsche 911 Turbo
Favorite Quote: “There are good ships and wood ships, and ships that sail the sea. But the best ships are friendships and may they always be.”

My name is James Wolbers and I’m from Northville, a suburb of Detroit about 30 minutes east of Ann Arbor. I always enjoyed math and science throughout high school which is why I became an engineer. I chose mechanical engineering specifically because I want to work in the auto industry. Mechanical engineering is more hands on than other departments and covers the topics I am more interested in including Thermodynamics, Heat Transfer, and product design.

Tommaso Gomez

My name is Tommaso Gomez and I am a 4th year senior from Orange, CT. I am interested in thermal/fluid sciences and hope to work in energy management someday. My favorite sports are soccer, mountain biking, and F1 racing. In addition to engineering, I am also interested in photojournalism and work as a photographer for The Michigan Daily.

Corin Giannola

I was born in Detroit, MI and lived in Farmington, MI for the first 20 years of my life. I graduated from Southfield Christian School in 2002. Currently, I live in Northville, MI and commute to school. I originally became interested in Mechanical Engineering because I love figuring out how things work and I also enjoy working on and playing with cars. In the future, I hope to open my own business running a custom car shop, building custom computers, or making custom drum kits.

David T. Platt

I was born in Bad Axe Michigan but my father was in the Air force at the time so I have lived all over the United States and two years in Izmir Turkey. I came back to Michigan in 1980 when my father retired from the Air Force and spent the next 14 years living in Monroe where I graduated from Jefferson High School. At age 20 I joined the US Navy as an enlisted sailor.

I spent six years in the Navy as an Electronic Technician/Nuclear Reactor Operator. I was stationed on a submarine (the U.S.S. Santa Fe) ported out of Pearl Harbor Hawaii. I qualified as an Engineering Watch Supervisor—the senior enlisted watch stander on a submarine. After my enlistment was up I went to work for an electronics manufacturing firm in Auburn Hills called Jabil Circuits. I was hired as a level 3 technician but advanced to Shift supervisor then to Test Engineer. As a Test Engineer I was responsible for writing software needed to test high-end internet backbone equipment as well as designing test hardware for said equipment.

I left Jabil in 2002 to pursue my education. I attended Oakland Community College then Transferred to Oakland University and then to the University of Michigan. I am currently attempting a dual major in Electrical and Mechanical Engineering because I eventually want to have a career in internal combustion engine design and control both from a mechanical perspective and from a computer controlled perspective.