Final Report

Optimizing a Hydraulic Regenerative Braking System for a 20” Bicycle Wheel

The University of Michigan
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Executive Summary

With a growing concern of climate change and decreasing availability of fossil fuels, the U.S. Environmental Protection Agency (EPA) has been researching hydraulic hybrid transportation systems. For seven years, the EPA and ME450 students at The University of Michigan (U-M) have collaborated on projects developing Hydraulic Regenerative Braking Systems (HRBS) for bicycles. These systems conserve energy that is normally lost during friction braking. The bike's kinetic energy is used to drive hydraulic fluid into an accumulator via a pump, braking the vehicle. This stored energy is later released to accelerate the bike forward.

This semester we have refined previous HRBS designs by optimizing the mechanical systems and improving safety. A key goal for our team was to build a functioning prototype 20" wheel that weighs less and has fewer moving parts than previous generations. Our team has made minimal changes to the extant hydraulic system, as the parts have been well-researched and recommended by our sponsor, David Swain of the EPA. Working with Mr. Swain, we created a list of customer requirements for this project. Table 1 below lists many of our key engineering specifications that were created to meet these requirements, as well as the final characteristics of the prototype. Our four categories for engineering specifications are safety, cost, weight, and functionality. Due to the conflicting nature of these specifications, it has been difficult to improve many of the bike's systems without adversely affecting others. Compromises have been necessary in order to create a feasible design.

Table 1: Summary of key engineering specifications

Characteristic	Target	Prototype
Front wheel assembly weight	≤ 30 lbs	24.75 lbs
Bicycle load rating (rider weight)	≤ 160 lbs	> 200 lbs
System pressure as limited by relief valve	≤ 4200 psi	≤ 4200 psi
Bicycle deceleration target	3.4 m/s^2 – 3.6 m/s^2	not available
Bicycle acceleration target	2.0 m/s^2 – 2.5 m/s^2	not available
Number of moving/rotating parts inside hub	< 11	7
Prototype cost	≤ $1400	$1338

Many of the main hydraulic components have long acquisition lead times. To meet our goal of having a functional prototype by the end of the term, we expedited concept generation and selection so as to leave enough time to order and receive these parts. We created a detailed plan for the semester based on expected task requirements as well as these lead times.

In reducing the weight of the prototype compared to previous designs, we have significantly reduced the number of gears, replaced the bulky fiberglass hub support system with a lightweight aluminum spoke system, and removed excess material from the internal support plate ("superbracket"). These modification choices were made from a broad number of concepts, based on a thorough analysis of the forces and torques required of each of the components. The main engineering obstacles to implementing these design improvements have been dealing with the nonstandard interface between metric and non-metric components, and determining the routing of the hydraulic circuit.

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1 Abstract

The U.S. Environmental Protection Agency (EPA) is researching hydraulic hybrid transportation systems in an effort to address the growing concerns about global climate change and insatiable fossil fuel demands. Hydraulic hybrid vehicles use regenerative braking to store energy in pressurized fluids. This energy is then released to assist in vehicle acceleration. For the past seven years, ME450 students at The University of Michigan (U-M) have been developing designs for hydraulic hybrid bicycle systems. This semester we refined the design of a hydraulic hybrid system enclosed in a 20" bicycle wheel, with a focus on decreasing weight, improving safety, and reducing the number of moving parts.

2 Introduction

This section outlines the origins of the hydraulic hybrid bicycle system concept at the EPA as well as the driving force for its development. A brief outline of the project's scope for the Winter 2009 semester of ME450 is also presented below.

2.1 Background and Motivation

Founded in 1970, the United States Environmental Protection Agency is a federal body tasked with correcting environmental damage and establishing guidelines to help protect the natural environment of the United States [1]. Research into clean energy, particularly for use in transportation, is the focus of several of the EPA's efforts [2]. In cooperation with Eaton Corporation, United Parcel Service, Ford, International, and the U.S. Army, the EPA has developed several hydraulic hybrid vehicles for the purposes of improving fuel economy and reducing environmental impact [3].

The primary concept of hydraulic hybrid technology is to capture and utilize the energy that would otherwise be lost during braking and use it to accelerate the vehicle. As the vehicle brakes, a hydraulic pump connected to the drivetrain pumps hydraulic oil into the high-pressure accumulators. During vehicle acceleration, the energy stored in the accumulators is released back into the drivetrain, as the fluid flows through a hydraulic motor. This significantly lowers the amount of fuel needed to accelerate back to normal operating speeds [3]. The result of this regenerative braking is a marked improvement in fuel economy – a feature that is not just better for the environment, but also reduces fuel costs for the owner. A diagram showing this hydraulic regenerative braking system (HRBS) is shown in Figure 1 on page 6.

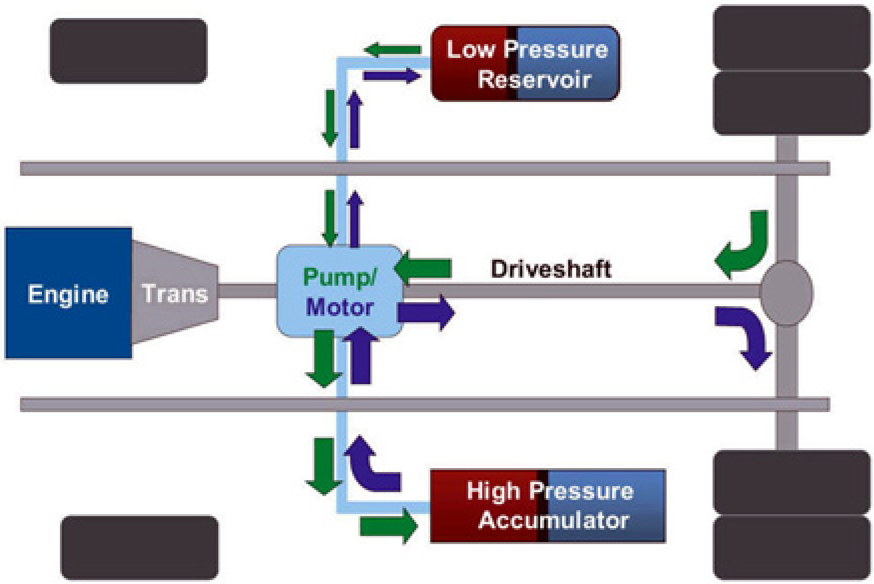


Figure 1: The hydraulic fluid's path in an HRBS [4]

The use of bicycles for commuting reduces fossil fuel use, greenhouse gas emissions, roadway congestion, and vehicle miles traveled while increasing the user's physical health [5]. The EPA has demonstrated 20-40 percent fuel economy improvements by installing HRBS on vehicles with internal combustion engines [3]. The possibility of clean, efficient transportation with hydraulic assistance bears exploration. The EPA has been working with U-M students on hydraulic bicycle implementation since 2002, but the project has produced only one functional product.

2.2 Project Description

The goal of this project is to develop a hydraulic regenerative braking system for a children's 20" bicycle. Due to the difficult nature of scaling down a hydraulic system, and the comparative ease of scaling upwards, the intent of using a 20" bicycle is to analyze the weight, force, and torque issues inherent to the HRBS on a small scale.

The EPA has been working on HRBS bicycles with ME450 students for the past seven years. Previous ME450 teams have worked on fitting these systems in 26" and 20" bicycle wheels. The primary focus of our work on the HRBS is refining the existing designs by improving safety, reducing weight, ensuring functionality, and lowering cost. We are designing an HRBS for a 20" wheel. Notably, one of the main goals is to reduce the device weight to 30 lbs without sacrificing mechanical robustness or safe pressure containment. We plan to retain the majority of the hydraulic components from past designs, as this technology has been well-researched and documented by David Swain and previous teams. By focusing on reducing moving parts, decreasing weight, and improving safety, we are further developing the understanding and implementation of HRBS technology through the fabrication of a functional prototype.

3 Information Search

To gain a better understanding of hydraulic hybrid systems, our team surveyed a broad collection of information including research papers, previous ME450 reports, and EPA resources. This section of the report discusses the information we found regarding hydraulic hybrid vehicle technology.

Hydraulic systems are used in a variety of applications such as machinery, braking systems, and energy storage. They are often used because of their ability to transfer large forces and convert kinetic energy into potential energy efficiently. To safely utilize this technology, many precautions must be taken to prevent high-pressure systems from rupturing.

The EPA, U-M, and companies such as Eaton and Ford have been developing hydraulic hybrid systems for transportation applications including cars, trucks, and bicycles. Hydraulic hybrid bicycle technology has been pioneered through a partnership between the EPA and U-M. For seven years, ME450 students at U-M have been researching, designing, and building hydraulic hybrid bicycle systems using HRBS. These systems require improvements in safety, functionality, and performance.

3.1 Automotive Research and Applications

The EPA has been developing hydraulic hybrid systems for three main automotive sectors: conventional vehicles, urban delivery trucks, and large SUVs and pickup trucks [3]. Each of these three sectors utilizes parallel and series hydraulic hybrid systems.

Parallel hydraulic hybrid systems utilize HRBS to pump incompressible hydraulic fluid into high-pressure accumulators. Much of the energy that is lost through conventional braking systems is recovered through this technique. This energy is then released by directing the pressurized fluid through a hydraulic motor. This hydraulic motor powers the car during acceleration. These systems are ideal for vehicles that operate under frequent stop-and-go driving. Eaton uses parallel hybrid technology in its Hydraulic Launch Assist™ (HLA®) system designed for refuse trucks and buses [6]. A computer model of the HLA® system can be seen in Figure 2 on page 8.

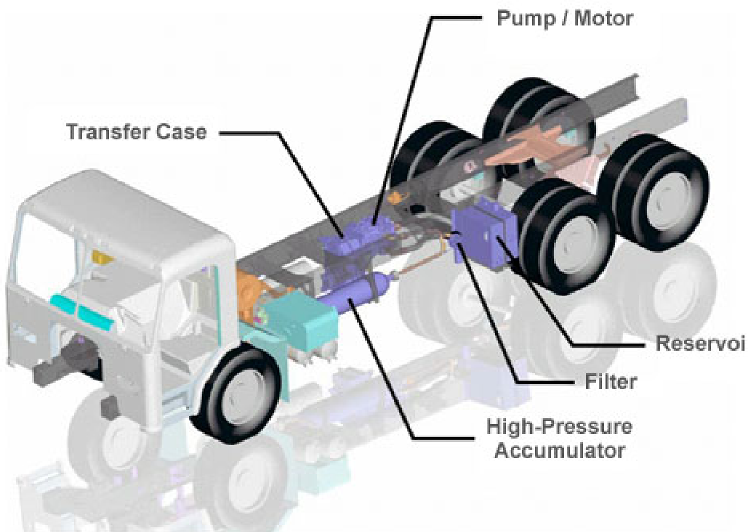


Figure 2: Computer model of Eaton Corporation's HLA® system for a refuse truck [6]

Series hydraulic hybrid systems replace the conventional drivetrain with a hydraulic drivetrain [3]. The EPA applied this technology to a Ford Expedition. In this setup a pump powered by an internal combustion engine is used to force fluid through a hydraulic motor. The motor uses the energy stored in the fluid to power the vehicle. Along with a hydraulic drivetrain, series vehicles use HRBS to recover energy lost during braking. The regenerative system is directly plumbed with the hydraulic drivetrain and helps the vehicle accelerate. The EPA estimates that full hydraulic drive vehicles could result in 30-40 percent improvements in combined city/highway fuel economy and lower emissions [3]. The EPA-modified Ford Expedition obtained a combined city/highway fuel economy rating of 32 miles per gallon (mpg) compared to its standard 14 mpg rating [7, 8].

The EPA has teamed up with UPS, Eaton, International, and the U.S. Army to develop the next generation of urban delivery vehicles using hydraulic hybrid technology. UPS is currently the main customer of these vehicles and has provided the EPA with delivery trucks to retrofit. Using hydraulic hybrid technology, it has been shown that the fuel economy of these trucks can be increased from 10 mpg to 18 mpg. Delivery trucks are strong candidates for hybrid systems due to their frequent stops.

3.2 Bicycle Research and Applications

Different forms of hybrid bicycles have been used for years, including mopeds and electric bicycles. Most of these systems are powered by gasoline or batteries. Some electric bicycles use regenerative braking, but this feature is not common. ME450 students at U-M have been developing hydraulic hybrid bicycles using regenerative braking as the energy source. This system is unique because it conserves kinetic energy by converting it to mechanical potential energy rather than refilling or recharging an energy source.

In December 2006 a team of engineers from U-M and the EPA filed for a patent on hydraulic regenerative braking for a vehicle [9]. This patent is based on the functional hydraulic hybrid bicycle designed and built through ME450 in Fall 2005 and a research project in 2006. This bike has been important for benchmarking purposes. The HRBS was enclosed in a modified 26" bicycle wheel. Testing this bike provided information on optimal acceleration and deceleration speeds, accumulator pressures, and hydraulic component size. In Winter 2007, an ME450 team created a redesign of the internal components of this bike. The final product was not a functional bike, but rather a model that demonstrated the process of hydraulic regenerative braking.

Beginning in the Winter 2008 term, ME450 projects regarding HRBS began focusing on implementing a regenerative braking system into a 20” bike wheel. This is a standard wheel size for kids’ bikes. By shifting research towards smaller wheels, the goals were to lighten and refine the system. The 20” wheel containing the hydraulic hybrid system currently weighs about 70 lbs [10, 11]. As of the date of this report’s publishing, the system is not yet operational.

In 2005 Parker-Hannifin Corporation began a design competition called the Parker Chainless Challenge [12]. For this contest, students design bicycles powered using hydraulic pumps and motors rather than chains. These bikes are series hydraulic systems. So far no team has entered a bicycle with regenerative braking. Students at U-M are working with the EPA to develop a hydraulic drive in parallel to an HRBS. U-M’s research for this design challenge began in 2007 as an ME450 project.

3.3 Previous Bicycle Design Information

ME450 project teams have created numerous HRBS bicycle designs. The most notable work has been accomplished during the past four years. The first functional HRBS bicycle was finished in 2006 for a 26” wheel. This system utilized a single gear reduction between the motor/pump and a large gear rigidly connected to the wheel hub. This system did not have any method to disengage the gears from rotating during operation, meaning that while a rider was pedaling the bike, he was also pumping fluid through the hydraulic loop. This increased the pedaling resistance and decreased the bike’s efficiency. The hubs on this bike were made of carbon fiber and were connected to a custom machined aluminum rim. Both the covers and the rim were quite heavy. The custom rim decreased serviceability and increased cost. The thicknesses of the material used to make the superbracket (4mm) and front bike fork (0.125”) resulted in heavy structural components.

In 2007, two teams refined the HRBS for manufacturability. A physical system was built; however, this system was not attached to a bike. The system that was built is housed within a display case and connected to a hand-powered hydraulic pump. Overall, the components used were comparable to those used in 2006.

Teams in 2008 further developed the design by incorporating an HRBS into 20” bicycle wheel. Using a 20” wheel was chosen to constrain the design and motivate innovation. This design contained a few notable improvements, but also opened the door for our team to make more changes. The first main improvement was creating a system that no longer used bevel gears. Earlier designs had used bevel gears to redirect rotation. The bevel gears were difficult to align which decreased overall efficiency. In this design, the motor and pump were rotated 90° and only spur gears were used to transmit the rotational energy. The motors and pumps used in designs prior to 2008 were too large, so in 2008 smaller motors and pumps were selected for the system. Additionally clutches were installed to disengage the pump and motor from rotating when neither the braking nor the launch system was engaged.

These improvements were beneficial, but there is still much room for improvement. The 2008 bike is not yet functional. Work is currently being done by two former ME450 students to complete the bike, improve the design, and test its performance. The hubs used on the 2008 bicycle were made of ½" thick fiberglass. Combined, the two hubs weighed about 40 lbs. The gears used on this system were ½" thick solid steel weighing a total of 12 lbs. While these gears were robust, they were overdesigned for the number of cycles seen in this system. The front fork and superbracket were also unnecessarily heavy components. The superbracket was made of 4 mm thick 1018 steel and the front fork was made of tubes with 0.125" wall thickness. Excluding the fork, these components resulted in an HRBS weighing approximately 70 lbs.

The 2008 system also posed problems for users effectively operating and maintaining it. The switches to engage the braking and launching systems were mounted on the bike frame directly in front of the seat tube. This setup would have required the rider to let go of the handlebars to engage and disengage the system. The bike was also difficult to service, as most of the components were welded together. There was an additional safety concern posed by the lack of a pressure relief device on the system. This could lead to unsafe operation, as the hydraulic components were only rated to 4000 psi. While much work has been done over the last seven years, more work is needed to effectively implement a hydraulic regenerative braking system on a bicycle.

3.4 Future Research

In order to further develop a hydraulic hybrid system, more research will need to be completed. The main topics of research this term focused on gear design, hydraulic component sizing, hub strength, and superbracket stiffness. In order to decrease the overall system weight, we investigated gear layouts and strength. Hydraulic component sizing was important in reducing the number and size of fittings. We analyzed the hub strength using FEA to ensure our design is strong enough to safely support a rider. More research into superbracket stiffness will be necessary to prevent gear disengagement and component vibration.

4 Project Requirements & Engineering Specifications

To outline the specifications for this project, we began by defining our customer requirements. We then translated these requirements into engineering specifications. This section of the report details these requirements and the resulting specifications.

4.1 Customer Requirements

The customer requirements for this term, as outlined by our sponsor David Swain, are continuations of the past two semesters with an added emphasis on three major underlying themes–safety, performance, and cost– to guide the formation of our engineering specifications. Table 1 on page 11 shows a listing of our customer's requirements, as grouped by the three major themes and their relative importance in each.

Table 1: Customer requirements categorized and listed by importance

Relative Importance	Safety	Performance	Cost
High	User Safe	Lightweight	Inexpensive Manufacturing Processes
	Natural Braking Rate	Reliable	Fastenable to a Stock Bicycle
	Easy to Use	Efficient	Material Costs
	Easy to Service	Sufficient Launch Speed	
low			

4.2 Engineering Specifications

When translating the customer requirements into engineering specifications, cost and safety translated directly. However, performance split into weight and functionality, as we find both categories of high enough importance to be separate. The resultant engineering specifications are described in the following list. The interactions between these specifications and their correlation to the customer requirements can be seen on our Quality Function Deployment (QFD) in Appendix A on page 62.

Note: Items in italics are integral to a complete design solution, but are not slated for implementation during the W09 timeframe.

1. **Safety**
   a. Hydraulic System
      i. Design shall incorporate pressure relief valve and line to dump high-pressure fluid to the low-pressure side in the event of overpressure (excess of 4200 psi).
      ii. High-pressure components and lines shall be sufficiently isolated from the user so as to prevent health and safety hazards in the event of a leak or rupture at 4200 psi.
      iii. High-pressure components and fittings shall be properly labeled for safety purposes.
      iv. Design shall provide a method by which the user may release the system pressure without accelerating the device.
   b. Power Transmission System
      i. Gears shall be sized to appropriately handle the torques/forces imposed upon them, without deforming or breaking themselves or the components to which they are mounted.
      ii. Moving/rotating components, especially those with teeth, shall be sufficiently isolated from the user so as to prevent mechanical safety hazards.
      iii. Moving/rotating components that present a mechanical hazard shall be properly labeled for safety purposes.

c. Electrical System
    i. Voltage and current sources shall be kept as far from the user as possible, such that the only interaction with the electrical system under normal circumstances is through use of well-insulated switches/toggles; in the event of an electrical system failure, the maximum exposure to electricity would result in less than 4 mA entering user.
    ii. No user interface devices such as switches/toggles shall be mounted in such a manner as to cause unsafe operation of the device.
    iii. Electrical components that impose dangerous voltage/current levels during normal operation, or could impose dangerous voltage/current levels in the event of an electrical malfunction, shall be properly labeled for safety purposes.

d. General Safety
    i. Mechanical connection between front tire and front fork, regardless of modifications to original bicycle design, shall be robust enough to support a rider weighing 160 lbs based on the weight of a 95th percentile 14-year-old boy [13].
    ii. Device deceleration should be limited to 3.6 m/s^2 to prevent user from losing control during braking.
    iii. Device should retain stock rim brakes on the rear wheel to allow for braking in the event of a complete front-wheel system failure.
    iv. Brake controls should be integrated in such a way that conventional friction braking may be imposed by actuating the RBS control with more force (e.g. squeezing harder in the event of the RBS failing to decelerate the bike properly).

2. Weight

a. Target
    i. Total front wheel assembly, including tire and rim, should weigh not more than 30 lbs.

3. Functionality

a. Pedaling
    i. Impediment to pedaling when the HRBS is disengaged should be minimized through use of clutches or other disengagement components.

b. Braking
    i. Device should respond within 500ms to call for braking.
    ii. Braking should be swift, but not violent; target range is 3.4 – 3.6 m/s^2 based on testing completed on the functional 2006 bike.
    iii. Upon full stop, device should refrain from accelerating backwards (the system must utilize a check valve to prevent the release of high-pressure flow from the accumulator).

c. Launch
    i. Device should respond within 500ms to call for launch.
    ii. Launch should be swift, but not rapid enough for the front tire to lose grip; target range is 2.0 – 2.5 m/s^2 based on testing completed on the functional 2006 bike.

        iii. Call for launch when device is already in motion should not damage or bind clutch, which could cause violent braking.
    d. Servicing
        i. Design should allow the typical user to perform standard maintenance operations, including replacing a worn/damaged inner tube.
        ii. Device should be able to be disassembled (i.e. no major components permanently connected).
        iii. Tire filling should not require disassembly.

4. **Cost**
    a. Target
        i. Total expended cost of the prototype (not including labor or components provided by sponsor) shall be not more than $1400.

5. **Interactions Between Specifications**
    a. Conflicting Specifications
        i. The weight target of 30 lbs (2.a.i) directly opposes several safety targets (1.a.i-ii, 1.b.i-ii, 1.d.i). Ensuring device safety by using larger, more-robust components will increase the total weight of the device.
        ii. The cost target of $1200 (4.a.i) directly or indirectly opposes several other targets (1.a.i, *1.a.iv*, 1.b.i, 1.d.i, *1.d.iv*, 2.a.i, 3.a.i). Components necessary to maximize safety, robustness, desirable operation, and weight reduction tend to be more costly.
    b. Consistent specifications
        i. Several safety targets (1.a.ii, 1.b.ii, 1.c.i) may be met simultaneously through proper device shielding.

5 Concept Generation

To effectively generate a broad collection of concepts, we began by decomposing the main subsystems of the HRBS. After breaking down the subsystems, we listed the main components of each. Each team member then created a list of concepts for each of the components. We then met as a team to build on one another's ideas and we created a master concept list.

5.1 Functional Decomposition

Based on the unique history and relative complexity of our project, we followed a slightly different concept generation process than most teams. We began by decomposing the bicycle HRBS into five functional subsystems. These subsystems are hydraulics, powertrain, hub, superbracket, and user interface. Each of these subsystems contained at a minimum two major components. Figure 3 is a functional decomposition tree showing which components fall under which subsystem.

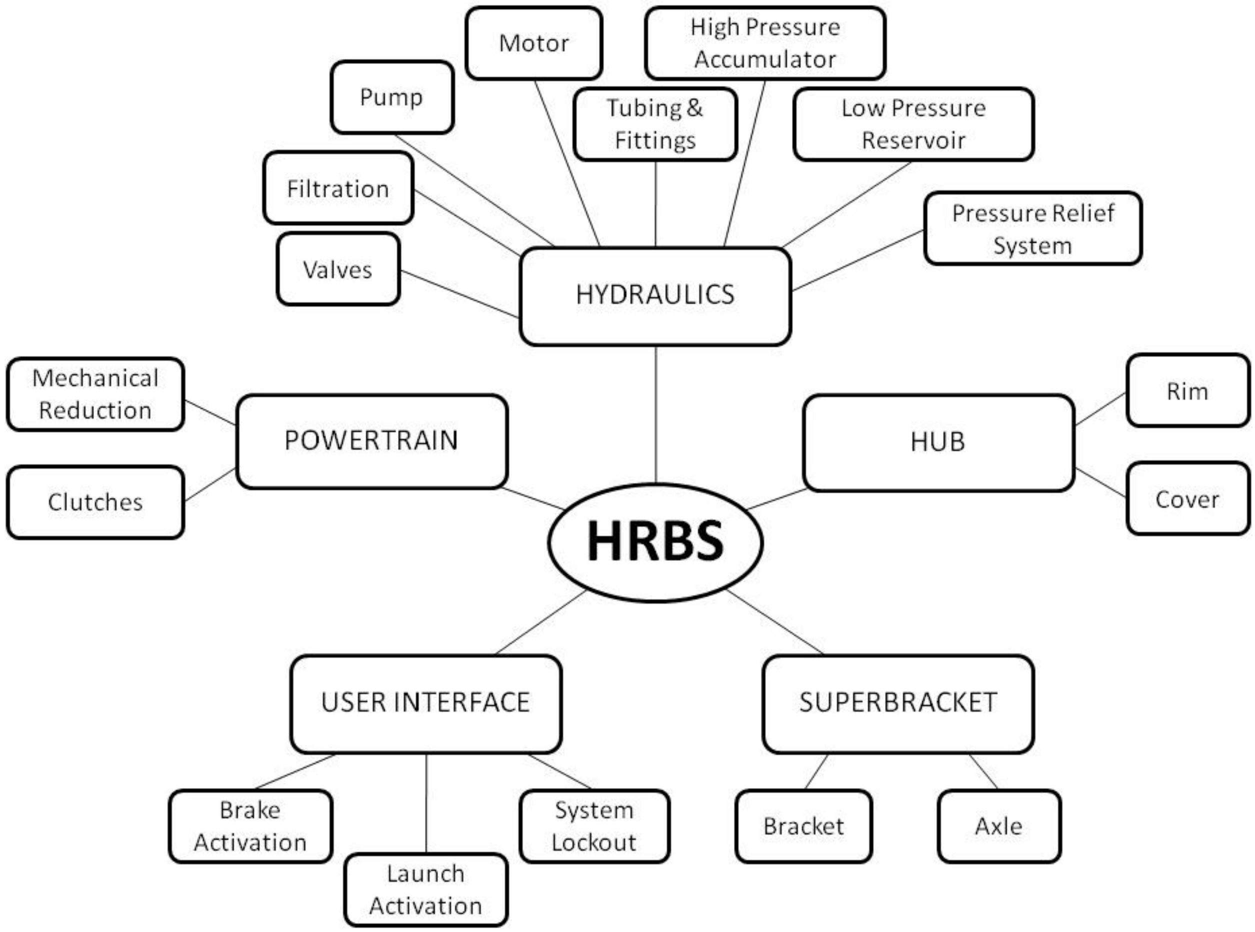


Figure 3: Functional decomposition tree outlining main components of each subsystem

After completing the functional decomposition, we generated concepts for each of the subsystem components. By individually creating concepts and analyzing them as a team, we were able to attack each design problem from multiple angles.

5.2 Hydraulics

The subsystem most refined by previous teams is hydraulics. This is also the subsystem with the longest lead-time items. As a result, many of our hydraulic components—including the pump, motor, high pressure accumulator, tubing & fittings, and low pressure reservoir—will remain the same as those specified by previous teams.

In addition to the systems used on previous generations, it is important to include a pressure relief system to prevent over-pressurizing the system. This can be achieved by including a variable pressure relief valve or a burst disc.

The valves category is made up of a check valve preventing high pressure flow from entering the pump and a directional valve to start and stop the launch process. There are various types of check valves that respond better to different pressures. The directional valve could either be a two-way or a three-way electronic valve. There are different types of each of these valves that vary in their sealing method. Poppet valves seal quite well, leaking only a few drops per minute; spool valves can leak multiple milliliters per minute.

5.3 Powertrain & Packaging
Powertrain decomposes into only two component categories, but it is very complicated due to the packaging constraints of a 20” bicycle wheel. In the past, the mechanical reduction was created using steel spur gears. We generated many concepts including plastic gears, phenolic gears, sprockets & chain, cogged belts, cables & pulleys, and friction rollers like those used to launch roller coasters.

The second powertrain category is clutch mechanisms. A system is needed to disengage the pump and motor from the rotating hub when braking and launching are not engaged. Concepts to complete this task included electromechanical clutches (benchmark), mechanical clutches, roller clutches, and a custom clutch utilizing a linear actuator.

5.4 Hub
The hub’s main roles on the bike are to support the rim, to interface with the mechanical reduction, and to enclose the system’s moving components. This hub rotates around the bike’s axle, which is stationary. Previous teams have created hubs made of carbon fiber and fiberglass. We included these in our concept list as well as aluminum sheet metal, vacuum formed plastic, and spokes with a thin cover. We developed another concept by combining the spoke and vacuum form designs. In this design a rigid skeletal structure would be used to support the bicycle and a thin plastic cover would enclose the system.

5.5 Superbracket
The superbracket subsystem is made up of the superbracket and the bike’s axle. These components are rigidly connected together. The hub rotates on the axle and electric wiring exits the hub through the center of the axle. Designing the superbracket is a material selection and thickness optimization problem. The bracket needs to support the hydraulic and mechanical components and prevent the pump and motor’s output/input shafts from being loaded radially. To meet these criteria we created a list of potential materials, including steel, aluminum, fiberglass, tooling board, wood, carbon fiber, and plastic. Along with material selection we have discussed methods of increasing the bracket’s stiffness by using dimple dies, adding gussets, and adding angle iron reinforcements.

5.6 User Interface and Controls
Previous designs incorporated a switchbox for controlling the brake and launch functions. This box was mounted on the frame of the bike directly in front of the seat. While functional, this forces the rider to let go of the handlebars with at least one hand to activate either system. In the event of a system braking failure, the rider would have to quickly adjust his hand position to activate the hand brake on the handlebar. One concept that could potentially solve this problem is to integrate the switch and the preexisting hand brake. This could be done by splicing a toggle switch into the cable. A light squeeze on the hand brake could activate the HRBS, while a hard squeeze would be enough to engage the friction brakes. Another option, provided that the bike is equipped with front and rear brakes, is to leave the rear hand brake unmodified and splice a toggle switch into the front hand brake cable. The launch activation could potentially be switched via a toggle switch mounted on the handlebars, or a pushbutton mounted on the

handlebars. If two switches are wired in parallel, there is the advantage that both switches must be activated for the launch to be triggered – this could be beneficial from a safety standpoint.

6 Concept Selection

The concepts described above are suitable for general applications. For example, a belt & pulley power transmission is able to transfer torque and rotational speed. However, in our small and lightweight bicycle, such a mechanism is unreasonable due to the added difficulty in developing and sustaining tension. Because of this, reasonable feasibility was the primary criterion for all of our concept selections.

Next, we compared each reasonable concept to a benchmark. In our case, the benchmark was the corresponding sub-assembly of the previous bicycle (Fall 2008). All concepts that were inferior to the benchmark were discarded because our design needs to improve on the previous iteration. The criteria used for comparing concepts included design characteristics, manufacturing, weight, safety, and cost.

6.1 Hydraulics
For the hydraulic subsystem, the components remain relatively unchanged except for the 2-way valve, filter, and relief valve. The new 2-way valve has the same functionality and size as the benchmark, but leaks at a much lower rate (5 drops per minute vs. 5 ml per minute). The new filter is rated for particles as small as 2 microns, and is able to withstand pressures of up to 6000 psi (the previous design was only rated to 1500 psi). The addition of a relief valve will allow fluid pressurized above the recommended accumulator limit of 4000 psi to be dumped safely into the low pressure accumulator. No such mechanism was present on the benchmark subsystem.

6.2 Powertrain
For the powertrain subsystem, we use drilled-out steel spur gears. These are much lighter than the current solid steel gears and offer similar strength properties – the lightening holes are strategically placed so as to remove excess material without sacrificing structural integrity.

To disengage the pump from the rotating hub, we plan to use the same 24V electromechanical clutch used by previous teams. This clutch was selected because it is relatively small in size and can transmit a larger amount of torque than comparably sized clutches.

We will use a Timken one-way clutch bearing to disengage the motor. When the motor is not engaged, it is disconnected from the gears. This clutch bearing requires no electrical power and weighs only $1/100^{th}$ of a second clutch. Using the clutch bearing adds a precision machining process to create the proper press fit.

6.3 Hub
The hub subsystem will not be made out of the benchmark material (thick fiberglass), but will be comprised of outer metal spokes covered by a vacuum formed ABS plastic cover. This substantially reduces the weight of the bicycle wheel and allow for quicker, more precise fabrication. A mold for the hub cover will be created out of Renboard using a CNC router. The vacuum forming will be done in the Art and Architecture woodshop.

6.4 Superbracket

The superbracket will be made from a thin stainless steel plate with strategically placed reinforcements and lightening pockets. The circular envelope of the rim will be similar, but more pockets and unnecessary material will be cut out of the plate for significant weight savings. We will use thin angle iron to increase the rigidity while maintaining the weight reduction. The stiffness of the superbracket should not be compromised as the previous iteration was overdesigned.

6.5 User Interface & Controls

The user interface will be reworked from the current setup of a box mounted between the rider's legs. A handlebar mounted system will utilize two pushbutton switches that must be engaged simultaneously to activate the launch process. Instead of the front hand brake connecting to calipers on the front wheel, it will connect to a momentary switch that will activate the regenerative braking process. This lends itself to improved stability for the rider, as s/he does not have to remove a hand from the handlebars to operate either of the systems. Wires will run through the hollow shaft of the handlebars so that charged components are kept away from the user. Also, for additional safety, a lock and key setup will prevent unauthorized users from engaging any of the operations of the bicycle involving stored energy.

7 System Model

One of our goals this semester was to more rigorously define the theoretical model of the HRBS through the use of computer simulation (Simulink), something that has not been attempted in previous generations. Though we have used it primarily to evaluate performance characteristics and design selection of the HRBS, we realize that this model will provide a valuable source of information to existing research as well as to future works.

The theoretical model is constructed around the pump and motor transmission, from which the other subsystems developed overtime. The architecture of the theoretical model is shown in Figure 4 with the complete model shown in Figure 5 on page 18. All variables used in the model are listed in Table 2 on page 19.

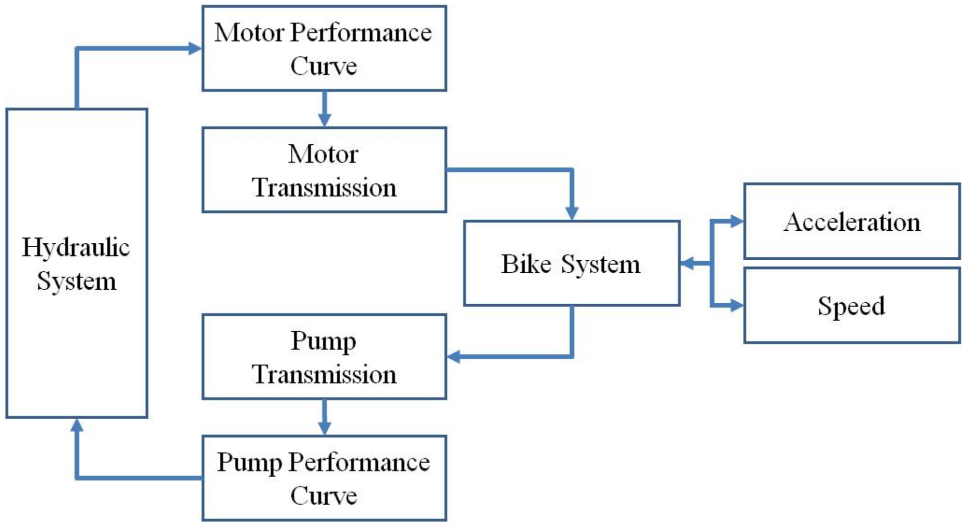


Figure 4: Architecture of theoretical model

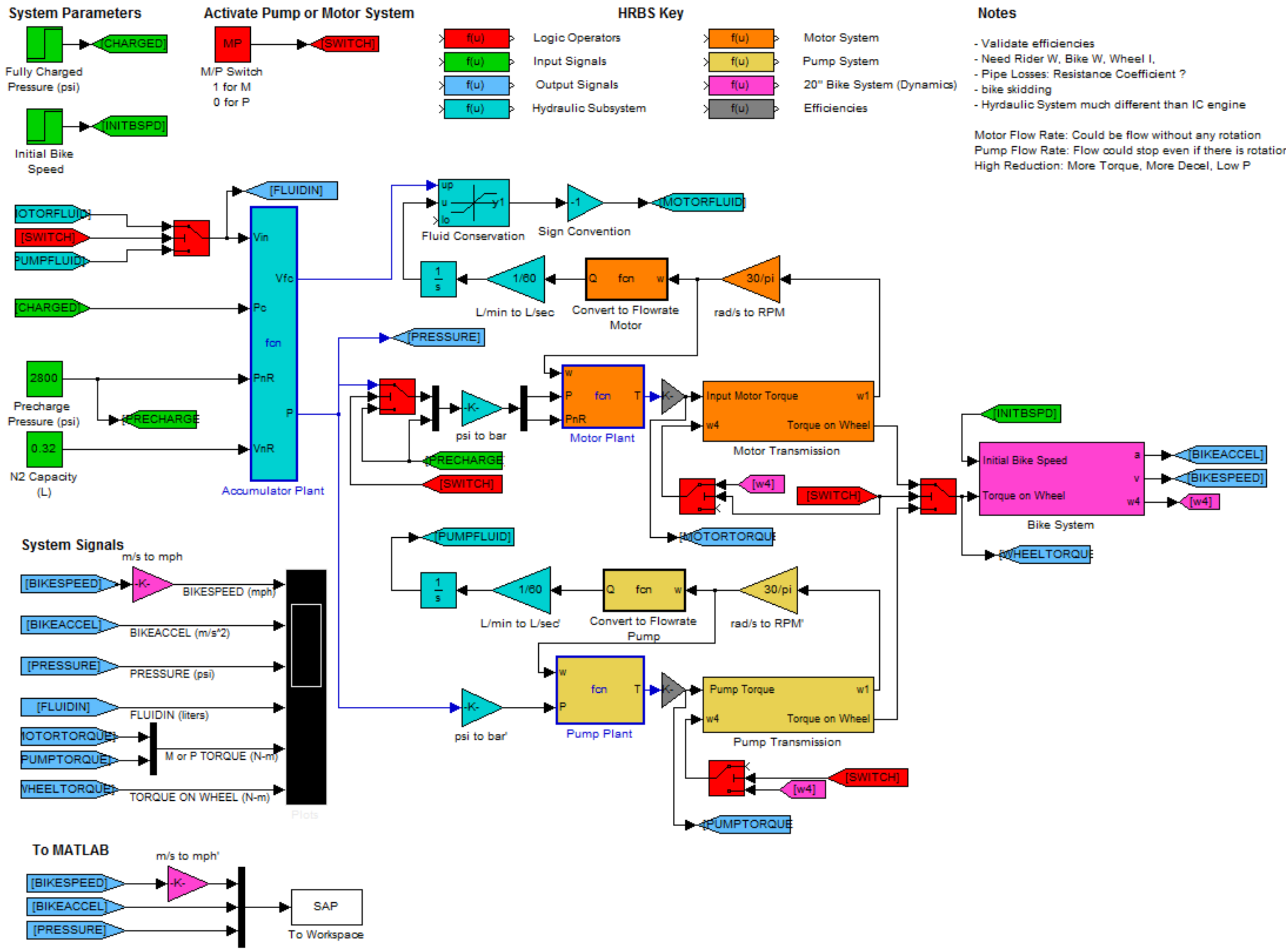


Figure 5: Complete HRBS model

Table 2: List of all variables used in theoretical model

Variable	Subsystem	Description
ω_1	Pump/Motor Transmission & Performance Curves	Angular speed of pump or motor shaft and gear
ω_2	Pump/Motor Transmission	Angular speed of 2nd gear on first reduction
ω_3	Pump/Motor Transmission	Angular speed of clutched gear
ω_4	Pump/Motor Transmission	Angular speed of main gear and front wheel
ω_5	Pump/Motor Transmission	Angular speed of meshed 5th gear from other transmission
R_1	Pump/Motor Transmission	Pitch radius of pump or motor gear
R_2	Pump/Motor Transmission	Pitch radius of 2nd gear on first reduction
R_3	Pump/Motor Transmission	Pitch radius of clutched gear
R_4	Pump/Motor Transmission	Pitch radius of main gear
R_5	Pump/Motor Transmission	Pitch radius of meshed 5th gear from other transmission
I_1	Pump/Motor Transmission	Rotational inertia of pump or motor gear
I_2	Pump/Motor Transmission	Rotational inertia of 2nd gear on first reduction
I_3	Pump/Motor Transmission	Rotational inertia of clutched gear
I_4	Pump/Motor Transmission	Rotational inertia of main gear
I_5	Pump/Motor Transmission	Rotational inertia of 5th gear from other transmission
T_{in}	Pump/Motor Transmission & Performance Curves	Torque applied by motor and pump shaft to transmission
T_{out}	Pump/Motor Transmission	Torque applied to wheels
T_s	Pump/Motor Transmission	Torque applied to clutched shaft
F_{12}	Pump/Motor Transmission	Tangential force between motor or pump gear to 2nd gear on first reduction
F_{34}	Pump/Motor Transmission	Tangential force between main gear and 1st gear on 2nd reduction
P	Performance Curves	Pressure on high side of pump or motor
I_w	Bike System	Inertia of front (and rear) wheel
M	Bike System	Total mass of bike (w/ rider)
R_w	Bike System	Radius of the front (and rear) wheel
a	Bike System	Acceleration of the bike
P_{nR}	Hydraulic System	Precharge pressure of hydraulic accumulator
V_{nR}	Hydraulic System	Volume of nitrogen gas for empty accumulator
P_g	Hydraulic System	Instantaneous nitrogen gas pressure
P_c	Hydraulic System	Charge Pressure (for motor system)
V_g	Hydraulic System	Instantaneous nitrogen gas volume
V_{fo}	Hydraulic System	Initial fluid volume
V_f	Hydraulic System	Instantaneous fluid volume
Q	Hydraulic System	Flow rate of fluid into the accumulator

7.1 Pump and Motor Gear Reductions

Figure 6 shows the geometry of the pump or motor transmission model (since both transmissions are the same). The transmission is a double reduction system (1 to 2 and 3 to 4) that amplifies the torque transmitted to the front wheel in two stages.

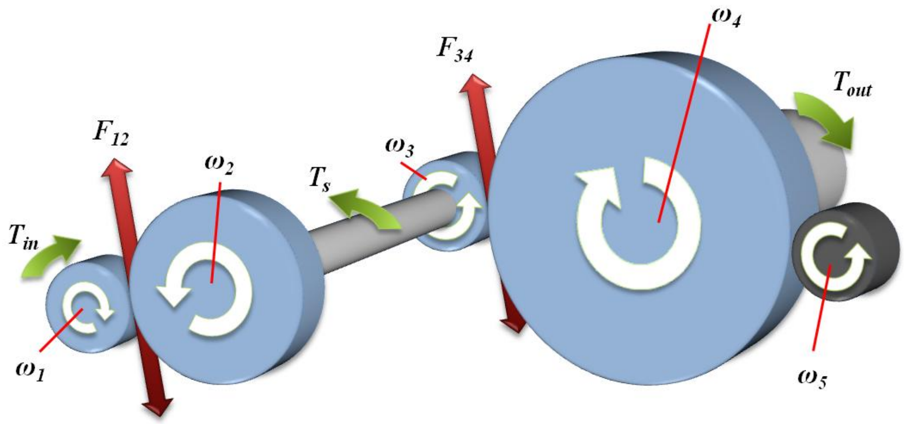


Figure 6: Geometry and loadings of transmission system

Using free body diagrams, the corresponding dynamics of each gear is determined in Equations 1-4.

$$T_{in} - F_{12}R_1 = I_1\dot{\omega}_1 \tag{1}$$

$$F_{12}R_2 - T_s = I_2\dot{\omega}_2 \tag{2}$$

$$T_s - F_{34}R_3 = (I_2 + I_s)\dot{\omega}_3 \tag{3}$$

$$F_{34}R_4 - T_{out} = \left(I_4 + I_5\left(\frac{R_4}{R_5}\right)^2\right)\dot{\omega}_4 \tag{4}$$

Although we could have largely simplified the above equations by algebraically combining them together, we chose not to in order to create separate subsystems for each gear. By doing this, we have the ability to incorporate additional information (frictional losses of tooth grinding for example) more efficiently if the need arises. Notice the inclusion of the 5th gear in the above equations and Figure 6. This is the satellite gear of the transmission not in operation. Though the two transmissions are independent, this satellite gear is meshed and will always rotate when either transmission is active. Alongside the dynamic analysis, a kinematic analysis reveals the mechanical reduction of the system, as shown in Equation 5.

$$\omega_1 = \frac{R_2}{R_1}\omega_2 = \frac{R_2}{R_1}\omega_3 = \frac{R_2}{R_1}\frac{R_4}{R_3}\omega_4 \tag{5}$$

Utilizing Equations 1-5, we formed the transmission model for the motor and pump as shown in Figure 7.

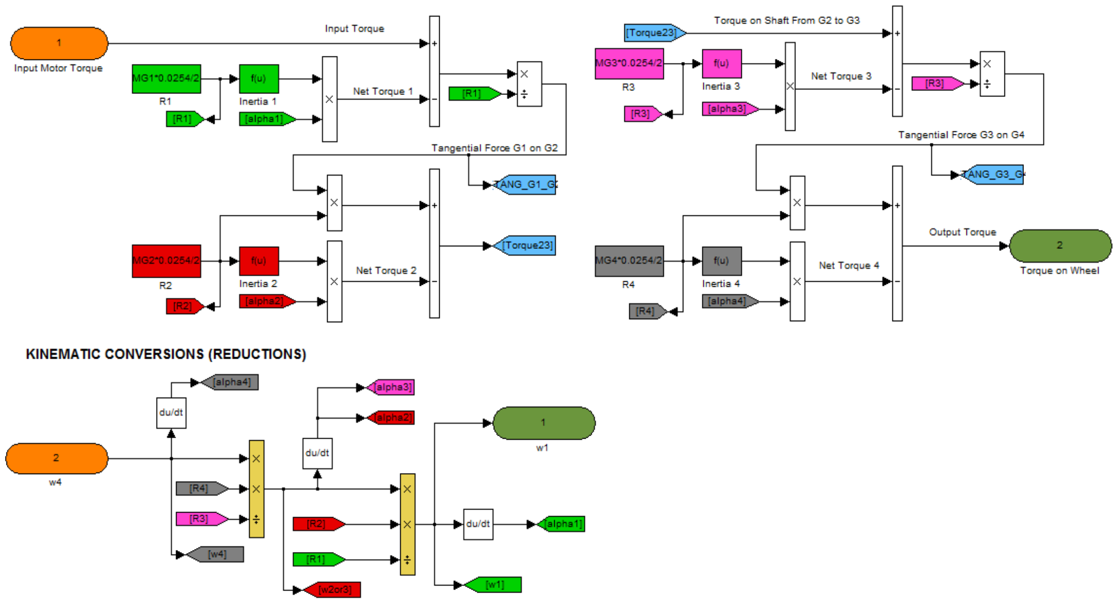


Figure 7: Motor (or Pump) transmission model

7.2 Motor and Pump Performance Curves

The motor and pump generates unique torques based on pressure and rotational speed. Fortunately, the manufacturer of the pump and motor, Marzocchi, provides steady-state performance curves [14] that trace their behavior for various operating points (see Figure 8). Thus by mapping these performance curves, we could obtain the "plants" of the motor and pump.

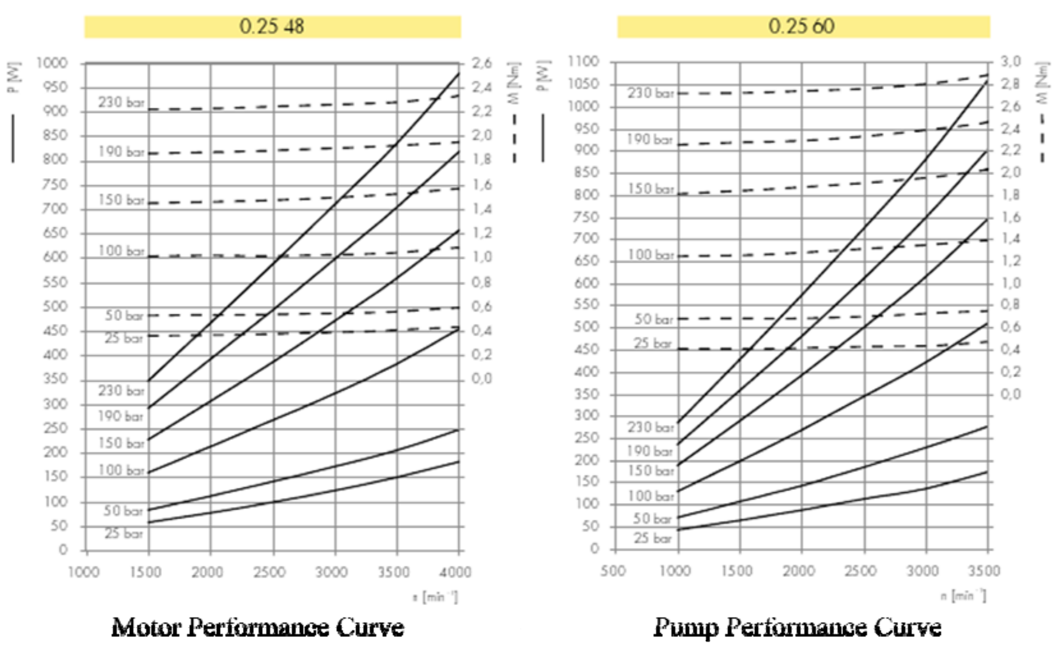


Figure 8: Motor and pump performance curves [14]

While a look-up table could have represented these curves, the operating pressure of our system is above the values shown on the graphs in Figure 8. This meant most of the data would have to be extrapolated and thus, a curve fit was considered. Noticing that the torque on the motor and pump shafts is a strong function of pressure and a weak function of speed, the curve fit was chosen by assuming that torque varied strongly with respect to pressure. The variation due to speed was assumed to be a percentage change from the average torque at the constant baseline pressure curve. The baseline pressure curve is the constant pressure line (on the performance curves) from which the curve fits are extrapolated. The selection of this baseline is chosen such that the extrapolated curve fits match as closely as possible to the actual performance curve data. The speed variation was modeled as a quadratic function due to its consistency with the shape of the performance curves. The generic curve fit is shown in Equation 6.

$$T_{in}(\omega, P) = \frac{\left(a\omega_1^{\,2} + b\omega_1 + c\right)}{d + eP_{BASE}}\left(d + eP\right) \tag{6}$$

Here, ω is the angular speed of the motor/pump shaft (in rpm), P is the pressure on the high side of the pump/motor (in bars), P_{BASE} is the baseline pressure, and the variables $\{a, b, c, d, e\}$ are fit parameters. The fit parameters for both the motor and pump are shown in Table 3. The baseline pressures were chosen such that the extrapolated curves matched as closely as possible to the actual performance curves.

Table 3: Curve fit parameters for motor and pump

	Motor	Pump
P_{BASE}	230 Bar	190 Bar
a*	$1.400(10)^{-8}$	$2.571(10)^{-8}$
b*	$-3.000(10)^{-5}$	$-5.286(10)^{-5}$
c	2.216	2.260
d	0.2000	0.1219
e	0.01748	0.01130

** These parameters become extremely significant for pump and motor speeds exceeding 4000 rpm.*

7.3 Bike System

The analysis of the bike system was conducted using the free body diagram (FBD) in Figure 9, from which the governing equations of the bike were determined (Equation 7).

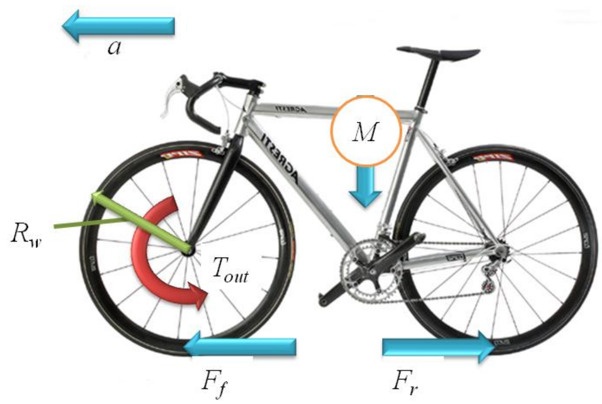


Figure 9: FBD of bike

$$T_{out} = \left(2I_w + MR_w^{\ 2}\right)\frac{a}{R_w} \tag{7}$$

It is important to note that the FBD analysis assumes rolling without slip, that is, the bike does not skid during operation. Though this might be added later to validate the system model, it has not been included since rolling without slip generates the greatest loadings, making them useful for designing against failure. Also, since the front wheel is rigidly attached to the fourth gear of the transmission, its angular speed is ω_4. The model of the bike system is shown in Figure 10.

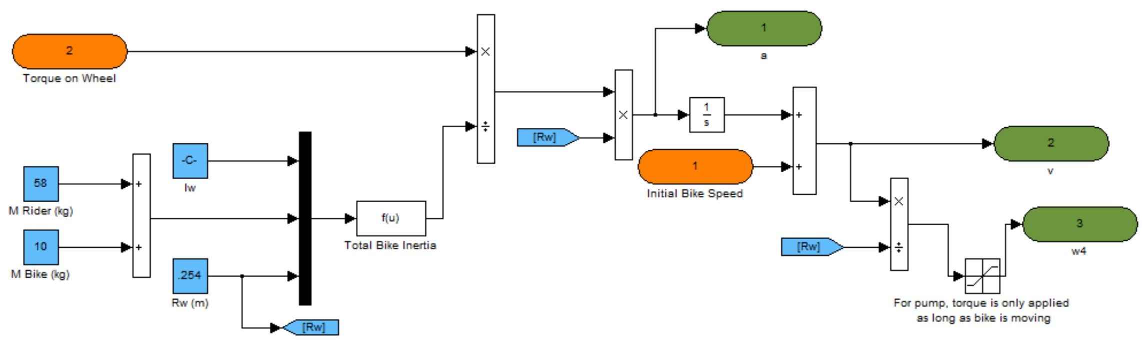


Figure 10: Bike system model

7.4 Hydraulic System

The hydraulic accumulator was modeled under the assumption that the nitrogen gas inside the accumulator behaved isothermally. This is a reasonable assumption, as there shouldn't be significant temperature variations in the accumulator during operation. Utilizing conservation laws, we determined the fluid and gas behavior of the fluid during operation (Equations 8-11). Definitions of the variables used in these equations are found in Table 2 on page 19.

$$P_g V_g = C = P_{nR} V_{nR} \tag{8}$$

$$V_g = V_{nR} - V_f \tag{9}$$

$$V_f - V_{fo} = \int_0^t Q\,dt \tag{10}$$

$$V_{fo} = \left(1 - \frac{P_{nR}}{P_c}\right)V_{nR} \qquad (11)$$

The flow rate through the system is dependent on the displacement of the pump and motor (assuming no energy losses in the pipes). Conveniently for us, Marzocchi also included flow rate curves [14] with the performance curves. By mapping these curves (Figure 11), we found a method of relating angular speed of the pump and motor to the flow rate. We chose a linear curve fit for the flows. Here Q is the flow rate (L/min) and ω is the angular speed of the pump/motor shaft (rpm). The corresponding fit parameter for both the motor and pump is shown in Table 4.

$$Q = a\omega \qquad (12)$$

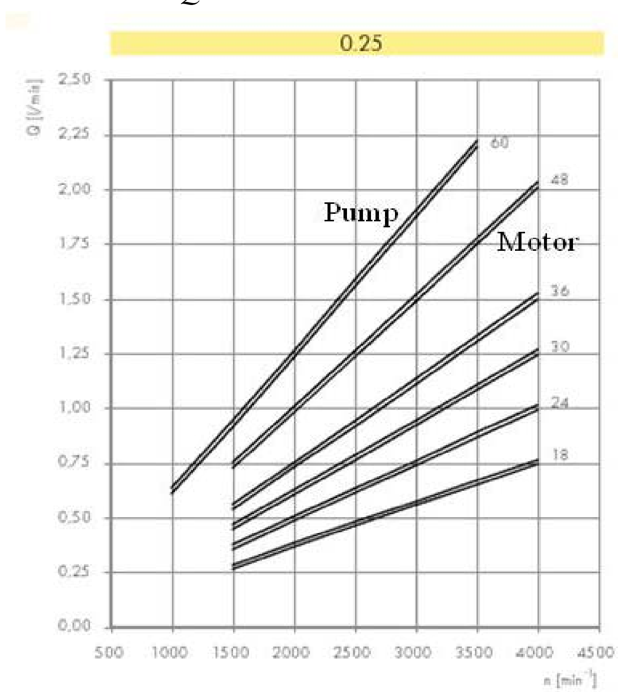


Figure 11: Pump and motor flow curves

Table 4: Fit parameters for pump and motor flow rates

	Motor	Pump
a	0.00051	0.000635

7.5 Discussion of System Losses

System losses are important to discuss since they directly impact the validity of the system model. Table 5 documents losses that have not been taken into account in each subsystem.

Table 5: Losses not modeled in the theoretical model

Subsystem	Losses not modeled
Pump/Motor Transmission	- Frictional losses and stored deformation energy of meshed gear teeth - Frictional losses in both clutches - Frictional losses in all bearings
Performance Curves	- Information uncertainty associated with back calculating transient behavior from steady-state
Bike System	- Frictional losses in all bearings - Air drag - Vibration of components
Hydraulic System	- Entrance/Exit effects at small openings (valves, pump, motor, accumulator, fittings) - Air pockets and or instantaneous cavitations due to motor/pump activity (model assumes fluid is continuous and is present at all times in the lines) - Heat and viscous losses in lines and hydraulic accumulator.

7.6 Transmission Tuning Analysis

With an automobile, one must tune the transmission to obtain the highest efficiency from the engine. The HRBS is no different. Utilizing the model, we tested different gear reductions of the transmission to tune the system. The observed behavior of the system is documented in Table 6 for higher and lower reductions.

Table 6: System behavior for variations in transmission reduction

Reduction Changes	Pump System Behavior	Motor System Behavior
Higher Reductions	- Lower final charge pressure - Larger pump shaft speed - Larger deceleration - Larger loadings on gears	- Slightly higher final speed - Larger motor shaft speed - Larger acceleration - Larger loadings on gears
Lower Reductions	- Larger final charge pressure - Lower pump shaft speed - Lower deceleration - Lower loadings on gears	- Slightly lower final speed - Lower motor shaft speed - Lower acceleration - Lower loadings on gears

Based on the information presented in Table 6, we can argue that to maximize performance we need to have as low a gear reduction as possible on the pump system (to acquire the largest accumulator pressure) and to have the highest gear reduction as possible on the motor system to take advantage of the higher final speed. However, there are limitations. For one, the high accumulator pressure is limited to the design constraint of 4000 psi. This limited how low of a reduction we could go on the pump side. Another constraint is that the maximum operating speed of the pump and motor is 7000 rpm. This places an upper limit on possible gear reductions for our HRBS. Further complicating the matter is the fact that our maximum acceleration and decelerations levels are limited to those levels comfortable to a rider. These were previously determined by our sponsor David Swain and are listed in the engineering specifications.

Testing stock gear sizes, we determined that a gear reduction of 17.5:1 for the both the pump and motor transmission satisfied all design constraints (see Table 7). We chose to use the same reduction for both the pump and motor to reduce the number of machining operations required for different gear geometries. Although we sacrifice some performance on the motor side (a slightly lower final speed), it does not outweigh the benefits of simpler machining schedules as well as reduced loadings on the transmission components.

Table 7: Final gear sizes (pitch diameters) for transmission systems

Pump Gear Reduction			To Wheel	Motor Gear Reduction		
G1	*G2*	*G3*	*G4*	*G3*	*G2*	*G1*
1”	3.5”	1”	5”	1”	3.5”	1”
Final Pump Reduction		17.5:1		*Final Motor Reduction*		17.5:1

8 Design of Selected Concept

Our system model created load, speed, and pressure information for our system. From this information and our component restrictions we created our final design. This section of the report outlines our final design and an analysis of it including such parameters as shape, material, and dimensions.

8.1 Parameter Analysis

Throughout our design process we have analyzed concepts for performance. This section of the report documents these processes as they pertain to each subsystem of the HRBS. The selection of purchased and custom parts is described.

8.1.1 Hydraulics

For safety purposes, the high-pressure side of the hydraulic system has been designed to withstand pressures of 6000 psi, even though the maximum expected system pressure is 4000 psi. Due to material considerations, this effectively excludes the use of brass fittings and components; while brass is commonly used in low-pressure systems, it is not strong enough to safely manage 6000 psi. Likewise, most aluminum fittings are not rated for this high of pressures. This leaves steel and stainless steel–the latter being preferred for its corrosion resistance.

The gear pump and gear motor are manufactured by Marzocchi. Since they are 12-week lead-time items, there was only one parameter to decide: use the pump and motor available through (and preferred by) our sponsor or order a new pump and motor with the knowledge that neither would arrive before the end of the semester. Since the W08 team had a significant problem with the Marzocchi lead time, we used the components we had on-hand.

Impurities in the hydraulic fluid will cause accelerated wear on the pump and motor, and will prevent the poppet valves from seating properly. According to Norman Filter Company, LLC, the most damaging contaminants in hydraulic systems are in the range of 3-20 microns [15]. As such, we have chosen a 2-micron stainless steel filter manufactured by Swagelok. Since the filter is to be placed on the high-pressure side of the system, it is rated for pressures up to 6000 psi. If the filter was placed on the low-pressure side of the system, it would present a restriction to flow resulting in possible cavitation or vacuum generation inside the pump, both of which are damaging. Placing the filter directly after the pump maximizes the number of components receiving freshly-filtered fluid.

To prevent backflow into the pump, the system utilizes a check valve after the filter. The check valve also keeps the fluid from back-flushing contaminants out of the filter and into the system. To reduce the resistance to opening, the crack pressure was chosen to be 1 psi. Manufactured by Swagelok, the check valve is made of stainless steel and is rated to 6000 psi.

The hydraulic accumulator is another component made available by our sponsor; the only parameter to decide was the nitrogen pre-charge pressure, which was selected to be 2200 psi based on data collected in previous semesters. This pre-charge is lower than previous generations so that system functionality may be tested under reduced loading conditions.

The system control valve had several options. Previous designs have used three-way spool valves, which leak at a rate of multiple mL/min. These valves are also available in poppet valve form, a style that leaks at a rate of drops/min. The two-way electronically-actuated poppet valve was chosen because it is smaller and leaks less than previous selections. The steel valve housing option was chosen over the aluminum option as the steel housing is rated to 6000 psi, whereas the aluminum housing is rated to 3300 psi. The main drawback of this selection is the larger weight of steel vs. aluminum.

The requirements for the low-pressure reservoir are markedly less severe than the requirements for high-side components. The reservoir must be large enough to handle 150 mL of hydraulic fluid while not monopolizing too much space in the superbracket envelope. The material must also be nonreactive with hydraulic fluid. We chose a polyethylene terephthalate (PET) honey bottle, mainly due to size and cost considerations.

8.1.2 Powertrain

Once the gear reductions were optimized using our Simulink model, we analyzed potential gear sizes and materials for our HRBS. We did this using the gear sizes listed in Table 7 on page 26. Selecting gears for this system was a complicated process, as gears are generally rated for around 10 million cycles. Selection was also made more difficult because gears generally fail through fatigue caused by sliding, rolling, bending, and compressing. Ultimately we utilized three different calculations to verify the performance of our gears. Our gear selection focused on two main characteristics: material and face width. Two secondary characteristics—pressure angle and diametral pitch—were directly related to the availability of the gears. Based on lead time and cost concerns, we ruled out custom gears as an option for this project.

Gears are made from a variety of materials so we focused our selection on steel, aluminum, phenolic, and nylon based on part availability, cost, and performance. The face width, or thickness, of the gear was important during selection as the hub is limited to four inches in width. Using the previous team's design of ½" steel gears as a benchmark we limited the overall thickness of each gear to ½". This value was also used in creating our initial CAD model, so we knew this was an upper bound.

Table 8: Gear design requirements used for material selection

Functions	- Lightweight, strong gears
Constraints	- Must not yield
	- Maximum face width of 0.5"
	- No Custom gears
Objectives	- Minimize weight
	- Minimize cost
Free Variables	- Face width (gear thickness)
	- Diametral pitch (number of teeth per inch diameter)
	- Pressure angle (20° preferable over 14.5°)

The gear material and pitch diameter drove the options for the pressure angle and the diametral pitch. When options were available, a 20° pressure angle was chosen over 14.5° for increased durability. Also, when choosing the diametral pitch the smaller value was selected to increase the size of each gear tooth.

Following the basic limitations of gear availability we began calculating gear strength using the Lewis Equation with the Barth Revision [16]. This was recommended to our team by numerous companies including Emerson and Boston Gear. The Lewis equation determines the maximum allowable stress in a gear based on the tangential tooth load, face width, Lewis form factor, velocity factor, and allowable material stress. The allowable material stress is generally calculated as one-third of the material's yield stress. This one-third factor allows for a wide range of variability and is designed to increase the life of the gear [17].

While using the Lewis equation we concurrently utilized equations describing the compressive and tensile stress applied to a gear tooth under given operation conditions. Once again, these calculations included a fractional factor on the allowable material stress.

These equations swiftly eliminated the possibility of using phenolic and aluminum gears, but based on the fractional stress factors, we were not content with their accuracy based on our low cycle application. This led us to a third equation, which came from the American Gear Manufacturers Association (AGMA). This equation, called the AGMA strength equation, utilizes calculates the stress in a gear based on five operating parameters, a geometry factor, the face width, the allowable stress, and the diametral pitch [18]. We chose to base our gear selection on this equation because its five user-defined operating parameters provide a more accurate stress calculation.

The AGMA gear strength formula showed us that, while undesirable, the loading on our system requires the use of steel gears. Our calculations found that the first reduction required gears with a 3/8" face width and our second reduction required ½". The diameter and face width requirements limited our gear options to a pressure angle of 14.5°. A summary of the selected gears is located in Table 9. Screenshots of the Excel worksheets used to make these calculations are located in Appendix E on page 67.

Table 9: Gear selection summary

	Pump Gear Reduction			To Wheel	Motor Gear Reduction		
	G1	*G2*	*G3*	*G4*	*G3*	*G2*	*G1*
	1"	3.5"	1"	5"	1"	3.5"	1"
Material	Steel	Steel	Steel	Steel	Steel	Steel	Steel
Face width (in)	0.375"	0.375"	0.5"	0.5"	0.5"	0.375"	0.375"
Diametral Pitch	20	20	16	16	16	20	20
Pressure Angle	14.5°	14.5°	14.5°	14.5°	14.5°	14.5°	14.5°
Weight (lbs)	0.059	.707	0.077	2.006	0.077	.707	0.059

The powertrain subsystem also includes two clutching mechanisms. The hydraulic system needs to disengage from the mechanical system so that normal riding does not pressurize the fluid or drive the motor shaft. The clutch device on the pump side needs to take ~80 in-lb of torque, and the clutch device on the motor side must be able to transmit a load of 56 in-lb (both figures are derived from the Simulink model). This can be accomplished through the use of an electromechanical clutch and a one-way needle bearing. The electromechanical clutch allows its two input shafts to rotate independent of one another when there is no voltage across it. When placed under a predetermined voltage differential, the shafts are effectively joined together in the desired direction. The pump gear train is therefore selectively isolated or joined to the hydraulic system while the bicycle is in motion. The one-way bearing serves the purpose of allowing the motor gear train to drive the bicycle but not the other way around (i.e. the bicycle's forward motion cannot drive the motor gear train).

8.1.3 Hub

The hub is a critical component of the HRBS since it supports the Superbracket, connects the front wheel to the bike, and transmits the power between the powertrain and bike. Its endurance to the rider weight as well as the torque applied by the pump or motor is crucial to rider safety and system functionality. Although the 2008 hub concept was able to meet this requirement, its implementation resulted in a total hub weight of 40 lbs which decreased system performance and rider comfort. The main focus of this term's hub design is to reduce weight to improve performance by utilizing a skeletal structure and a non-structural cover. By making a skeletal hub design, we eliminated the structural aspect of the hub cover, thus drastically reducing the weight of the hub system compared to the 2008 design. This also leads to improved serviceability as a person does not have to completely disassemble the structural components of the hub system to access the hydraulic components.

The challenge of designing the hub emerges from the fact that the typical location to mount a spoke structure is occupied by a hydraulic system. To further complicate matters, the HRBS front wheel is required to endure torque loads, in which case it will receive larger loadings than a typical bike wheel. The large size of the hydraulic system eliminated the possibility of running straight members from the bearing carrier to the rim like a typical tension spoke system of a bike. Thus, the spokes had to sustain both compression and tension.

Two different spoke layouts were considered: a radial layout where the spokes are arranged along the radius of the wheel and a tangent layout where the spokes are arranged tangent to the mounting holes of the bearing carrier (similar to a bicycle).

Radial Spoke Layout: Due to its geometry (Figure 12 on page 30), it has good triangulation to support vertical loads. However, the resultant forces from the torque applied to the bearing carrier will be directed normal to the axis of the spokes, in which case loadings can only be transmitted by the spokes shifting slightly about the fixed point on the rim. This will induce a wobble in the spoke structure, inherently accelerating fatigue and decreasing life.

Tangent Spoke Layout: Similar to a standard bicycle spoke design, this layout will direct torque loads along the axis of the spokes, eliminating the wobble inherent in the radial spoke layout. Furthermore, with the spokes being offset from the center of the wheel, the forces transmitted to the rim have a moment arm about the center. This allows for more efficient torque transmission to the wheel. The drawback to this design is that it is not as effective in supporting vertical loads since one spoke may receive a large percentage of the total loadings for certain wheel orientations.

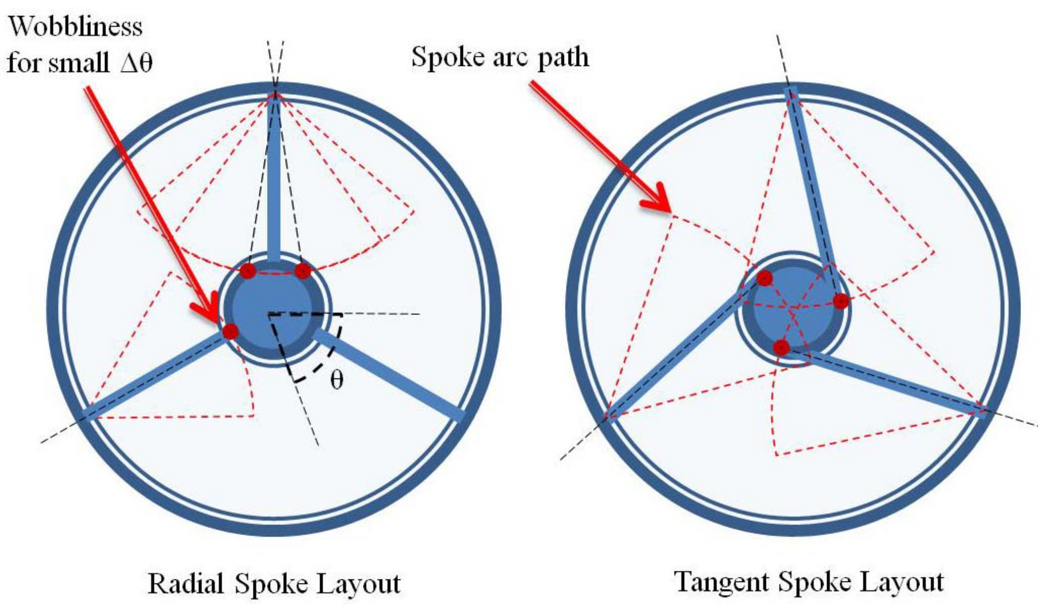


Figure 12: Spoke geometry layouts

A question of optimal geometry for loading arose for the tangent layout. Since the spokes are essentially long slender members, their ability to withstand tension is greater than their ability to withstand compression. Torque loads from deceleration are much higher than the torque loads from acceleration. Taking advantage of this fact, we decided that we could minimize possibility of failure by designing the orientation of the spokes such that they will be in compression for acceleration and in tension for deceleration.

A combination of the two spoke layouts was initially considered. This allowed the strengths of one layout to offset the weaknesses of the other. However, we realized that by using the identical tangent layout on both sides, not only were we able to cut down on manufacturing complexity, but we were able to form triangulations of opposite spokes. This made the hub system more effective at supporting vertical loads.

Since the rim is rigid and not prone to deformation from radial loads, we did not have to design the spokes to retain the arc shape of the rim. This eliminated the need for a large number of spokes. The smallest number of spokes we could chose on each side to properly constrain the geometry was three, which is what we chose.

Due to the urgency of constructing a functional and stable structural hub to contain the HRBS, we were limited to materials that were readily available locally and easy to process. Accordingly, our material selection process bypassed a traditional analysis based purely on design constraints, and instead favored a real world analysis of specific stock materials from local vendors. Metals were chosen over all other materials since our available fabrication resources (Wilson Center and ME Shop) were specifically equipped to deal with these materials. The types of metals readily available to us were aluminum, copper alloys, steel alloys, and stainless steel.

A difficult limitation to overcome was that stock material came in pre-specified cross sections. Our width limitations required us to use a 0.25” by 0.5” rectangular cross section. This proved to be a difficult constraint since it entailed a small cross section for the spokes to withstand buckling during compression, requiring a material with a large Yield Strength and Young’s Modulus. In addition, to keep the hub as light as possible, the material needed a low density. A summary of these design requirements is listed in Table 10.

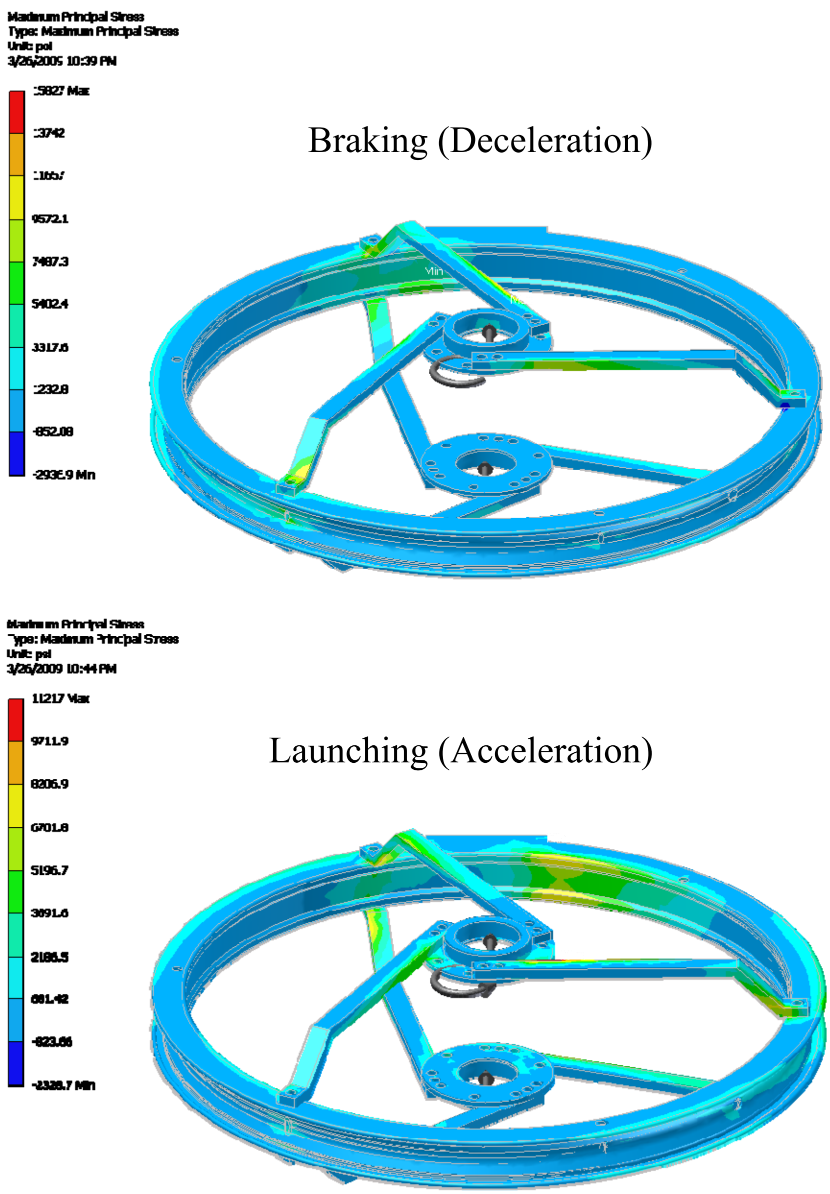
Table 10: Spoke requirements used for material selection

Functions	- Strong, lightweight spokes
Constraints	- Must not yield
	- Must not buckle
	- Must be metal (pre cut stocks available readily)
Objectives	- Minimize weight
	- Minimize lead time
Free Variables	- Hollow vs. solid

We ultimately decided to use 6061-T651 aluminum with a 0.25”x0.5” rectangular cross section. 6061-T651 aluminum was chosen due to its low density for a lightweight design, high Young’s

Modulus to resist buckling, high Yield Strength, and good machinability. Though steels, stainless steels, and coppers all passed the loading criteria, the aluminum was by far the lightest.

We conducted failure analysis by listing possible failure mechanisms in all loading conditions, determining the dominant failure mechanisms, and then designing against them. For compressive loadings, buckling was a concern. We analyzed yield by using an FEA package within Autodesk Inventor and buckling with the AISC Standard Buckling Equations [19], which are extensions of the Euler Elastic buckling and Johnson's Inelastic buckling equations to account for structural defects and eccentricities in loading. The results of the FEA and buckling analysis are shown in Figure 13 and Table 11.



Note: The loading scales on the two images are different.

Figure 13: FEA analysis for failure by yield for the hub

Table 11: Failure analysis parameters and results for 6061-T651 Al.

Negative values denote compression.

Loading Parameters	Magnitude
Vertical Load	400 lb
Torque (Deceleration)	600 lb-in
Torque (Acceleration)	300 lb-in

Localized Yield Analysis	Deceleration	Acceleration
Failure Stress (Yield) [20]	34900 psi (241 MPa)	34900 psi (241 MPa)
Actual Stress	13740 psi (94.7 MPa)	9700 psi (66.9 MPa)

Buckling Analysis	Deceleration	Acceleration
Failure Stress	-7300 psi (50.3 MPa)	-7300 psi (50.3 MPa)
Actual Stress	-852 psi (5.87 MPa)	-824 psi (5.68 MPa)

8.1.4 Superbracket

The superbracket provides support for all of the system's internal components. Contained within the hub, it must be smaller in diameter than the rim – including the inner tube's Schrader valve stem. A CAD image of the bracket is shown in Figure 14 on page 34. Since the overall width of the system is to be minimized, the superbracket must also be thin while retaining its strength. Based on these requirements, a maximum envelope for the superbracket was defined: it shall be no larger than 13" in diameter, and no thicker than 1/8". This bracket will be made of steel. An alloy will be decided based on stiffness, cost, and availability at Alro. A non-corrosive steel alloy is ideal.

To further reduce weight, as much unused material as possible will be removed from the superbracket. To determine where these unused sections will be, the system components were modeled in Autodesk Inventor and assembled within the diameter of the superbracket. The advantage of this method is that the components can be quickly and easily repositioned within the wheel. The primary constraints of this layout are the relative positions of the gears, and the position of the low-pressure reservoir relative to the pump. Gears must be positioned so as to mesh properly and connect the pump and motor to the drive gear. The low-pressure reservoir must be oriented so that fluid is driven into the pump upon braking.

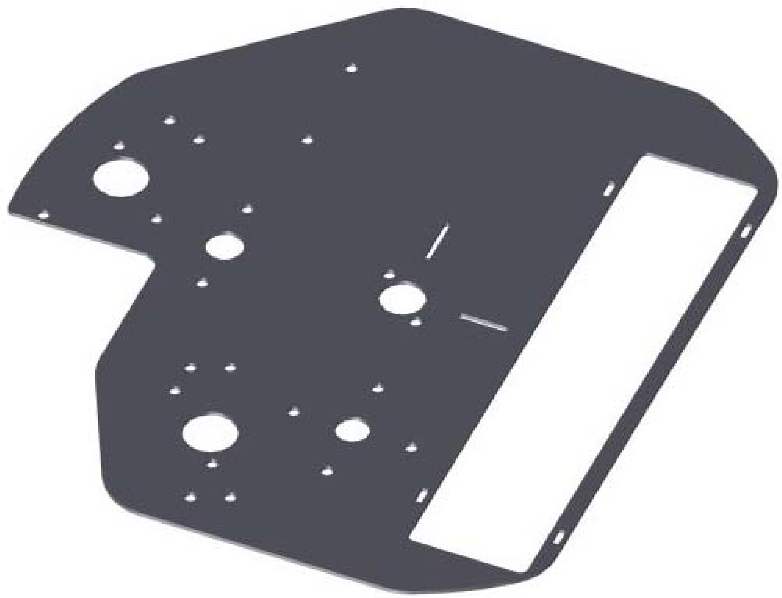


Figure 14: Isometric CAD view of superbracket (overall diameter of 13")

8.1.5 User Interface & Controls

The user interface consists of two points of interaction: the existing front brake lever and two pushbuttons mounted on the handlebar. The brake lever will initiate the regenerative braking process by actuating a toggle switch. This completes the braking circuit and allows the electromagnetic clutch to engage, which connects the powertrain to the hydraulics. The pushbuttons will activate the hydraulic pressure release system via the two-way valve to drive the powertrain, propelling the bicycle.

User safety is very important in developing parameters for this subsystem. We want to minimize the electric current flowing near the user at all times. One solution is to design a low-current signal circuit to activate the appropriate high-current power circuit. This way the user is not exposed to potentially dangerous levels of electrical energy. Also, switching the high power circuit should not use any mechanical movement because of potential corrosion or other wear on the contacts from repeated engagement and disengagement. A great way to accomplish this is through the use of transistors, with a control circuit activating and deactivating its corresponding power circuit.

The electromagnetic clutch and valve are on separate power circuits and have fixed electrical requirements. Both the clutch and valve require 24VDC, but they draw different amounts of current (250mA and 880mA, respectively). This constrains the electric system and forces all other electrical components to be designed and specified around these parameters.

There are several power options for our user interface, but the use of batteries is the simplest method that we have found. They are portable and have acceptable energy density for our application. A problem with batteries is that they are rated based on open circuit tests and the actual voltage that they can maintain under electric load is nontrivially less and decreases with time. To combat this, a voltage regulator will be used. A negative consequence of this is that the input to the regulator must be somewhat higher than the expected 24VDC output in order to guarantee a consistent output from the regulator. We will restrict the type and number of batteries to those combinations that yield a final voltage of 27VDC or more.

8.2 Final Design

Our final design is broken into the five subsystems of our HRBS. The four subsystems located within the wheel are shown in an exploded view in Figure 15. This section of the report describes the design selection and the component functionality.

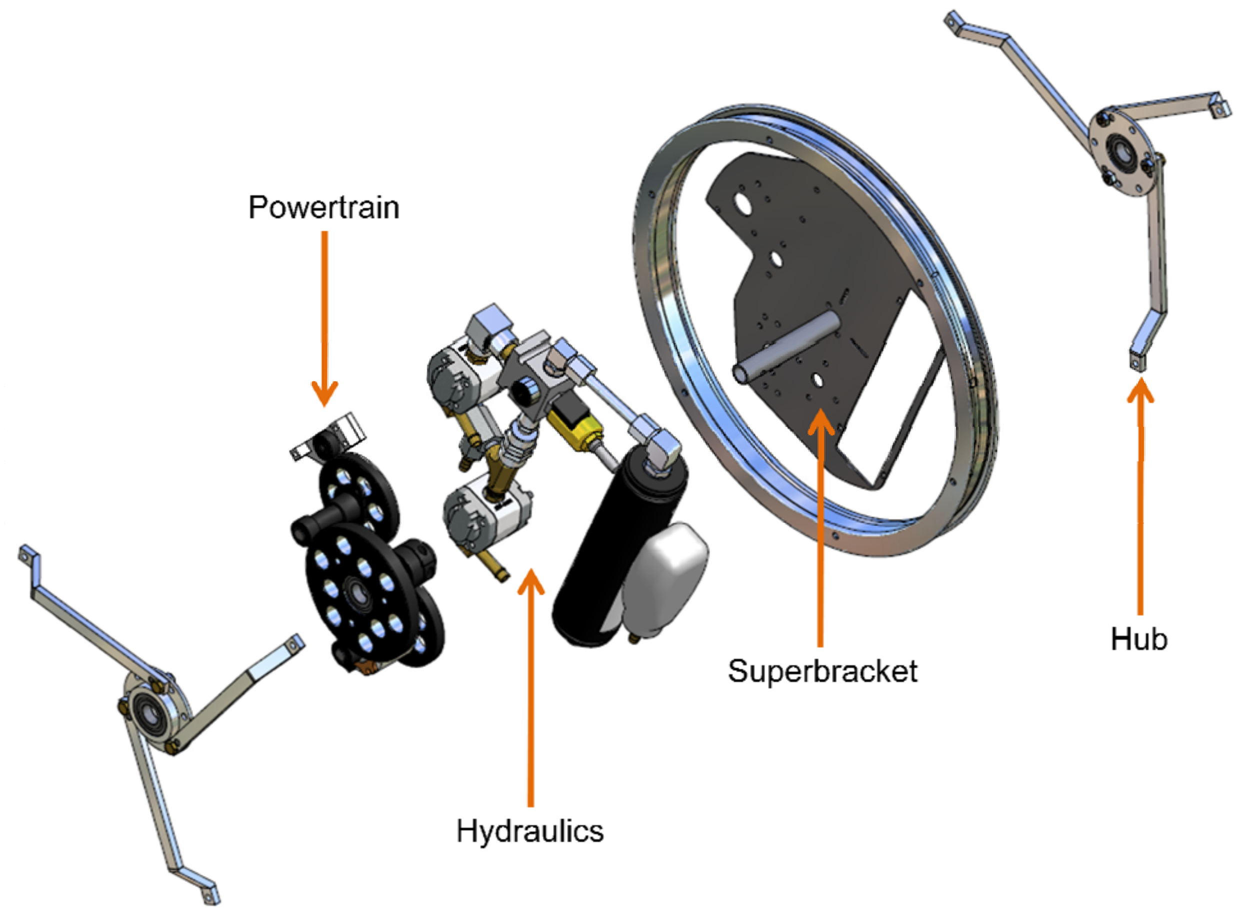


Figure 15: Exploded view of CAD assembly showing four subsystems

8.2.1 Hydraulics

The final hydraulic circuit is shown in Figure 16. An annotated drawing of this assembly is shown in detail in Figure 31 on page 51. The high-pressure accumulator and low-pressure reservoir are placed next to each other to save space. Upon braking, fluid will be pushed into the pump via gravity and bike deceleration. The filter is placed directly after the pump to maximize the system protection from particulates. The pressure relief valve is teed off of the line connecting the pump to the filter, so that the lines will not rupture in the event of the filter becoming clogged. The check valve is placed after the filter, followed by the two-way valve inlet. This inlet is connected to an always-open outlet on the other side of the two-way valve – which then goes to the high-pressure accumulator. When the two-way valve is triggered, the third port opens and fluid flows into the motor, and then back to the line from the low-pressure reservoir. The downside of working within such a tight envelope is that the circuit incorporates several right-angle bends, across which will cause pressure losses in the fluid. Ideally, the system would be arranged in a straight line, so as to present minimal impediment to flow.

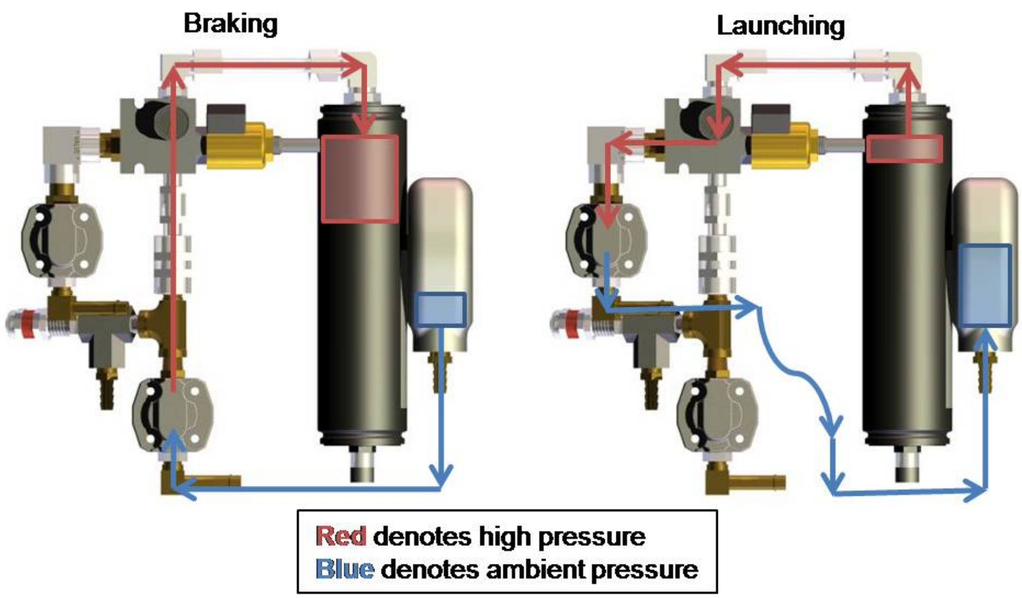


Figure 16: CAD model of hydraulic circuit in 20" wheel

8.2.2 Powertrain

Once the gear sizes were selected (see Table 7 on page 26), the main design concerns focused on attaching the gears to shafts. Based on gear availability and attachment goals, different torque transmission methods were chosen. A summary of the attachment methods is in Table 12.

Table 12: Summary of gear to shaft attachments in radial and axial directions

	Pump Gear Reduction			To Wheel	Motor Gear Reduction		
	G1	*G2*	*G3*	*G4*	*G3*	*G2*	*G1*
	1"	3.5"	1"	5"	1"	3.5"	1"
Radial Attachment	Key & Coupler	Key	Key	Pinned	Key	One-way Bearing	Key & Coupler
Axial Attachment	Retaining Clip	Retaining Clip	Retaining Clip	Set screw coupler	Retaining Clip	Retaining Clip	Retaining Clip

The most complicated gear attachment locations are on the pump/motor shafts. These shafts are metric (6mm diameter) with a key and M6 threads. Attached to each of these shafts is a gear with a 1" pitch diameter. Unless custom made, these gears cannot be purchased with a 6mm bore. We chose to purchase gears with a ½" finished bore with a keyway. To connect the gear to the shaft we will be using a set screw collar (6mm ID, ½" OD) as a coupler. On the outside of this collar we will machine a 1/8" keyway which matches that of the 1" gear. A support bearing on the end of this shaft will axially retain this gear. The bearing will be in a housing connected to the superbracket.

The 3.5” diameter gear on the pump reduction will be attached to a shaft connected to the electromechanical clutch. The shaft will use a 0.001” interference fit to transmit torque to the clutch. The gear will be connected to this shaft radially using a key and axially using a retaining clip.

On the opposite side of the electromechanical clutch, torque will be transferred to a shaft through a 0.0002” press fit and a cross pin. Attached to this shaft will be a 1” diameter gear. Torque will be transferred through a 1/8” key. Axial motion will be limited by a retaining clip on the end of the shaft.

The 3.5” diameter gear on the motor reduction will have a one-way clutch bearing pressed into it. This bearing will be the main means of torque transmission. A retaining clip will be used to axially restrain the gear from falling of the shaft and a Delrin spacer will be machined to offset the gear from the superbracket. On the other end of the shaft will be a 1” diameter gear connected with a key and a set screw.

The main gear on our system has a diameter of 5”. A bearing will be pressed into the center of this gear. This bearing will rotate around the bike’s axle (which is stationary). On the inboard side of the gear, a set screw collar will be used to position the gear axially, with a thrust bearing between the gear and the collar. On the outboard side, the gear will be retained by the hub bearing carrier, to which it will be rigidly connected via three dowel pins pressed into the gear. These pins will slide into matching holes on the hub bearing carrier. Torque will be transmitted through these pins.

The number of electromagnetic clutches on the market capable of supporting our system’s large torque and small size requirements is very limited. Several clutches were able to handle the torque, but they were large and heavy. Others were small enough, but their maximum torque rating was too small for expected operation. Ultimately we chose a clutch manufactured by Reell Precision Manufacturing which is the same clutch used by 2008 teams. It is rated to take 75 in-lbs – close to the maximum operating load of 80 in-lb. Selecting the one-way bearing was easier and was a simple constraint search with the correct shaft diameter and torque requirements. The final selection was a Timken one-way bearing sold by McMaster.

8.2.3 Hub

The final hub skeleton design is shown in Figure 17. This hub consists of 3 spokes on each side, with the spoke layouts on one side forming triangulations with the spokes on the other. This geometry combination increases the effectiveness of supporting vertical loads. The spokes are made of solid 0.25”x0.5” 6061-T651 aluminum. This material has a relatively high stiffness that retains structural integrity and is still light enough to reduce weight. To reduce weight, the bearing carriers are also made of aluminum and have extra “lightening” holes drilled into them, which is also utilized on the drive side to attach to the main gear. The hub cover will be made from thermoformed ABS plastic that will be mounted over the hub skeleton. The CAD image of the hub cover is shown in Figure 17. A photograph of the final hub assembly is shown in Figure 18.

Figure 17: CAD of final hub design

Figure 18: Side view of the final hub assembly without covers

8.2.4 Superbracket

The layout of the components on the superbracket was dictated by the requirements of the powertrain as well as the design of the hydraulic circuit. The largest components – the high-pressure accumulator and the low-pressure reservoir – were placed in the model first. These were followed by the pump and motor, which are constrained to be in contact with the gearing of the powertrain. After these components were organized, the remaining smaller components were positioned so as to satisfy the hydraulic circuit requirements and the mechanical space considerations of the powertrain.

8.2.5 User Interface & Controls

For engaging and disengaging the different modes of our system, Metal-Oxide-Semiconductor Field-Effect Transistors (MOSFET) are used because of their ability to switch high-current circuits on and off based on a control static voltage differential (i.e. no current flow). This is good primarily for improved safety of the user in all operation modes (see Table 13 on page 39). Normal braking and acceleration do not require any user inputs and do not employ any of the improved HRBS functionality. In fact, the HRBS is disengaged during these operating modes and the bicycle can be operated like a standard bicycle. In the regenerative braking mode, the front-brake lever is pulled and a toggle switch is activated. This triggers the electromagnetic clutch to engage the pump, which uses the bicycle's kinetic energy to pressurize the hydraulic fluid. To activate the hydraulic acceleration, the rider pushes two pushbuttons which energize the two-way valve circuit. This valve directs pressure through the motor, which drives the bicycle forward. When there are no user control inputs, the system reverts to a state where the HRBS is not engaged.

Table 13: HRBS Operation Modes

Mode	Setting	Result
Normal Braking	Pull rear-brake lever	Rear friction brake engaged, HRBS disengaged
Normal Acceleration/Cruise	-	HRBS disengaged
Regenerative Braking	Pull front-brake lever	Hydraulic pump engaged
Hydraulic Acceleration	Depress two pushbuttons	Fluid forced through motor

Based on the expected current through the electromagnetic clutch and valve solenoid, we have specified all of our electrical components to withstand at least 1A. This includes the voltage regulator and transistors. We have put fuses in place that are rated to 1 A as a precautionary measure. The entire circuit will fit inside the wheel hub, save the user interface controls as described above. A schematic of the circuit is shown in Figure 19.

The best combination and type of batteries for our application is four 9V batteries. They have a relatively small size and can fit into the hub. In addition, they have adequate capacity so that the batteries do not have to be replaced excessively often.

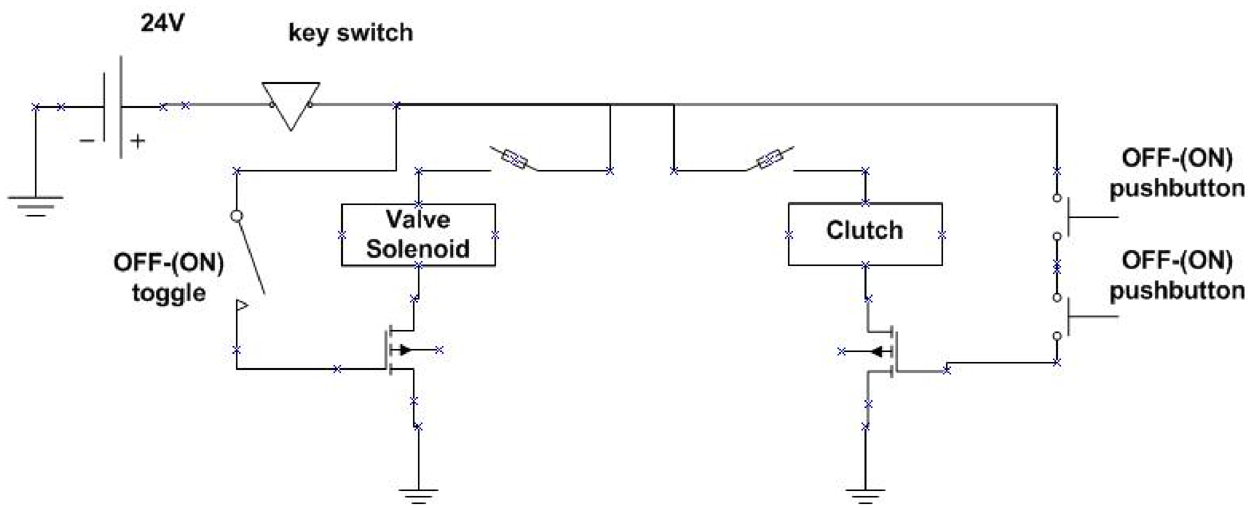


Figure 19: Electric schematic of user interface control system

8.3 Prototype Design

The prototype that our team created is a full-scale, functional HRBS for a 20" bicycle that is developed according to our final design. Consequently, our prototype proves the design from concept selection to assembly and fabrication. The working device and accompanying analysis are the main deliverables for our sponsor and can be used to further holistic and practical hydraulic regenerative systems. We met or exceeded all of the established engineering specifications, and therefore fully satisfy each customer requirement, except for final functionality. These metrics have yet to be verified through our bicycle's performance in validation tests (outlined in section 8.5).

The one main design difference between the final design and our prototype is the electrical system. We removed the voltage regulator, transistors, and fuses from the electrical system of our alpha prototype to make sure that all components worked as required. This made the circuit much easier to test and allowed for easier assembly/disassembly. The actual circuit in the prototype is shown in Figure 20.

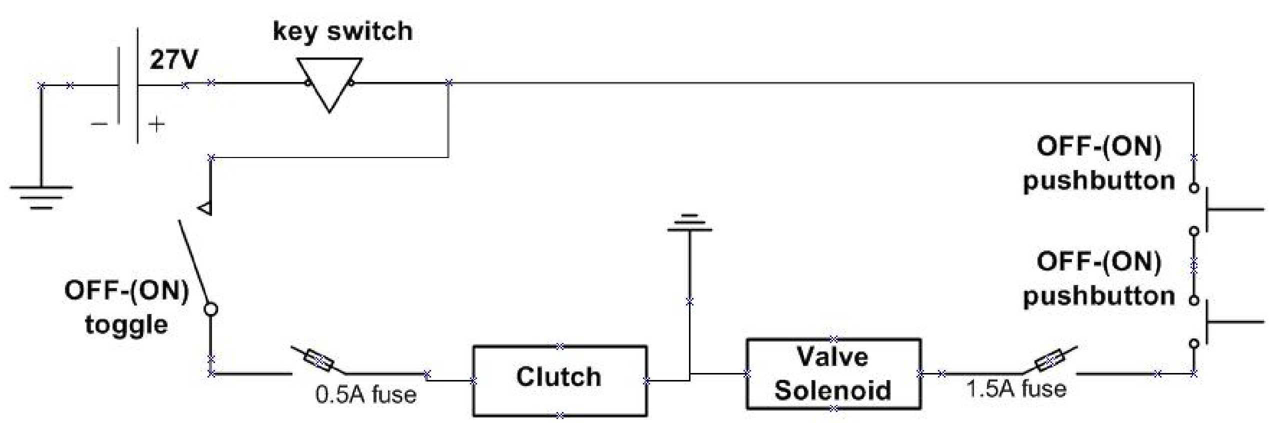


Figure 20: Wiring diagram of electrical system used on prototype

8.4 Fabrication Plan

Many components were fabricated for our HRBS. A summary of these components and the machining processes involved are covered in this section of the report and Table 14. Our assembly plan is listed in section 8.5.

Table 14: Summary of HRBS machining processes

Component	**Material**	**Required Tools & Machines**	**Speed**	**Location**
Hub mold	Renboard; MDF	CNC Router; sandpaper; Hand drill	N/A	Wilson Center (WC)
Hub cover	ABS plastic	Vacuum Former	N/A	Art School woodshop
Hub bearing carriers	6061 Aluminum	Lathe; Arbor press	1150 rpm	WC
Hub spokes	6061 Aluminum	Welder; Band Saw	30 fpm	WC & ME
Superbracket	304 Stainless steel	Water jet cutter	Determined by CAM program	ERC
Axle	Steel tube	Mill; Lathe	600 rpm	WC
Fork	Steel tube	Mill; Tubing Notcher; TIG Welder	600 rpm	WC
Rim	Aluminum	Mill; Rotary table	600 rpm	ME
Gear lightening	Steel	Mill; Rotary chuck; Lathe	600 rpm (mill); 950 rpm (lathe)	WC
Gear keyways	Steel	Broaches; Arbor press	N/A	ME
Gear shafts	Steel	Lathe; Mill	950 rpm	WC
Button carriers	6061 Aluminum	Lathe; Tap	1150 rpm	WC
Electrical system	Wires; solder	Soldering iron	N/A	WC & ME
Clutch stabilizer housing	6061 Aluminum	Lathe; Mill	1150 rpm	WC
Pump/Motor stabilizer collars	6061 Aluminum	Lathe	1150 rpm	WC

8.4.1 Hydraulics

Most of the hydraulic subsystem components are purchased parts. This helps guarantee reliability of components under high pressures. However, three of our hydraulic components needed modifications. These modifications were completed by Federal Fluid Power (FFP), as they have experience modifying hydraulic components.

Both the pump and the motor were modified to remove metric fittings. FFP rebored the M10 ports and threaded them for SAE 6 fittings. This removed four fittings from our assembly.

FFP also modified our two-way valve housing by boring two new holes. One is a ¼" national pipe thread (NPT) to allow the housing to serve as a tee-fitting in addition to being a valve housing. The other hole is a 1/8" NPT diagnostic port where our pressure gauge is attached.

The only other hydraulic component requiring machining is our low pressure reservoir. We machined a Lexan cap using the CNC Router at the Wilson Center and threaded a hole for a tube fitting.

8.4.2 Powertrain

All of our gears required machining processes. The required processes are summarized in the following bullets and in Table 15.

- **Bore (lathe 950 rpm):** Some of the gears were bored on a lathe. This allowed for the gears to fit on the shafts or have bearings pressed into them.
- **Keyway (broach):** Keyways (1/8") were broached using an arbor press and broach set in the ME shop.
- **Removal of hub projection (lathe 950rpm):** Each of our gears came from the manufacturer with a hub projection extending off the center of the gear. All of the gears had this projection removed on a lathe to reduce thickness and weight.
- **Lightening holes (mill 600rpm):** Some of the gears had unnecessary material that was milled away to reduce weight. This is seen in Figure 21 on page 43.
- **Bearing Press-fit (arbor press):** Two of our gears have bearings pressed into them. This was done using an arbor press.

Table 15: Machining processes (and tool) required for each gear

	Pump Gear Reduction			To Wheel	Motor Gear Reduction		
	G1	*G2*	*G3*	*G4*	*G3*	*G2*	*G1*
	1"	3.5"	1"	5"	1"	3.5"	1"
Bore (lathe)		x		x		x	
Keyway (broach)	x	x	x		x		x
Removal of hub projection (lathe)	x	x	x	x	x	x	x
Lightening Holes (mill)		x		x		x	
Bearing Press-fit (arbor press)				x		x	

In addition to machining the gears, all of our shafts required machining processes to cut them to length and diameter. We added keyways to many of the shafts as well as retaining clip grooves to all of them.

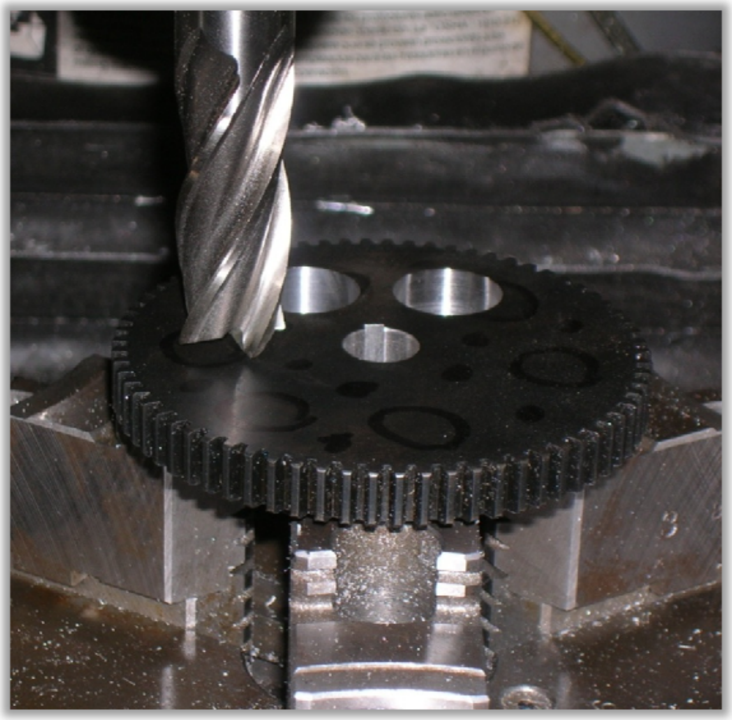


Figure 21: Milling lightening holes in a 3.5” gear

8.4.3 Hub

The hub is composed of four main components: the hub covers, the bearing carriers, the spokes, and the rim. Each of these components required its own fabrication process.

The ABS plastic hub cover was vacuum formed over a mold made of Ren Board. This mold was rough machined using the Wilson Center’s CNC Router and finished with sandpaper. Pictures of this process are in Figure 22. A base for the mold was made of ¾” thick medium-density fiberboard (MDF). The line separating the base from the mold was visible in the plastic following the vacuum forming. This created a cut line for post machining the hub cover. The vacuum forming process is shown in Figure 23.

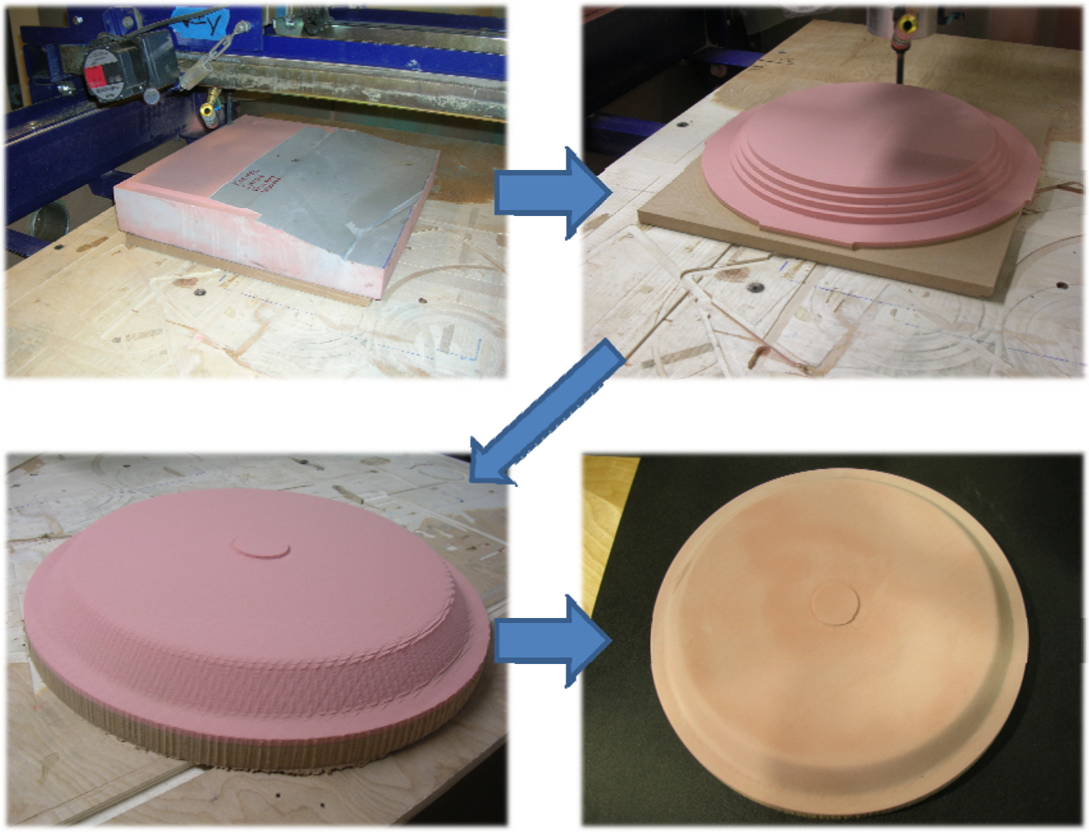


Figure 22: Machining of hub mold on CNC Router

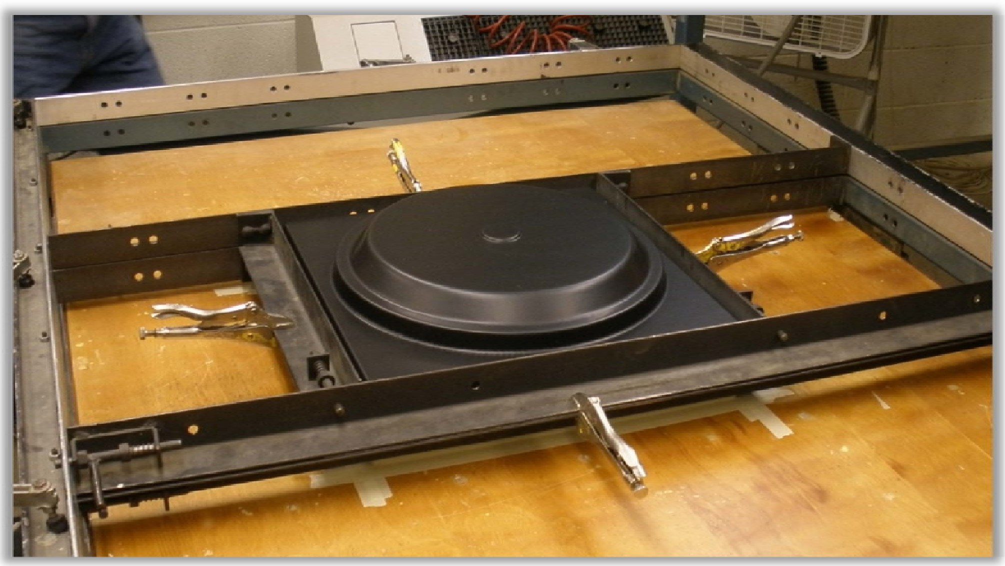


Figure 23: Vacuum forming hub covers in the A&A Woodshop

The bearing carriers (qty 2) were machined out of 6061 aluminum. Both lathe and mill processes were required to give the part its shape. The lathe processes included turning the outside profile and boring an inside profile with a 0.0005” press fit for a bearing. Once the lathe operations were complete, the holes in the flange were drilled on a mill. The bearing carrier is shown in Figure 24.

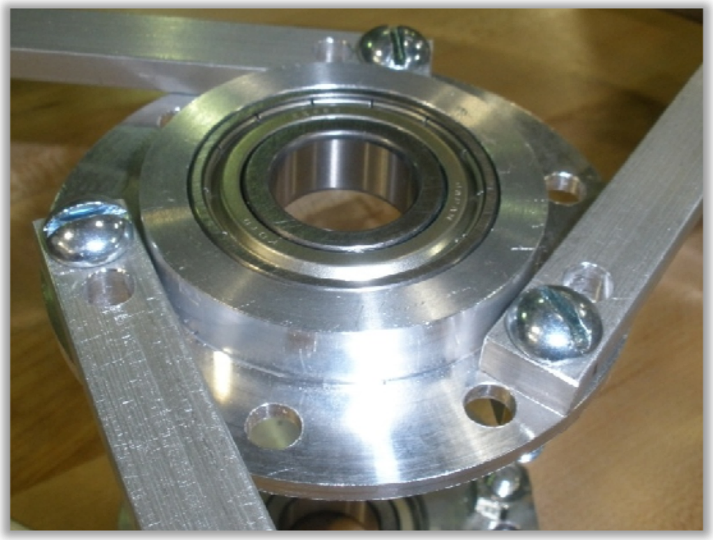


Figure 24: Hub bearing carrier with bearing and spokes

Spokes were machined out of ½” x ¼” rectangular 6061 aluminum stock. The spokes are composed of three pieces. Each side of the assembly requires three spokes. To simplify the manufacturing processes we chose to use the same spoke design for all six spokes. Maintaining machining tolerances on the spokes is important in guaranteeing wheel alignment. If spokes are of different lengths or if they mount poorly, the wheel will not spin true. Because of this, the spoke sections were milled to length and a mill was used to drill attachment holes. The sections were welded together. A jig, seen in Figure 25, was used to hold the assembly in place during welding.

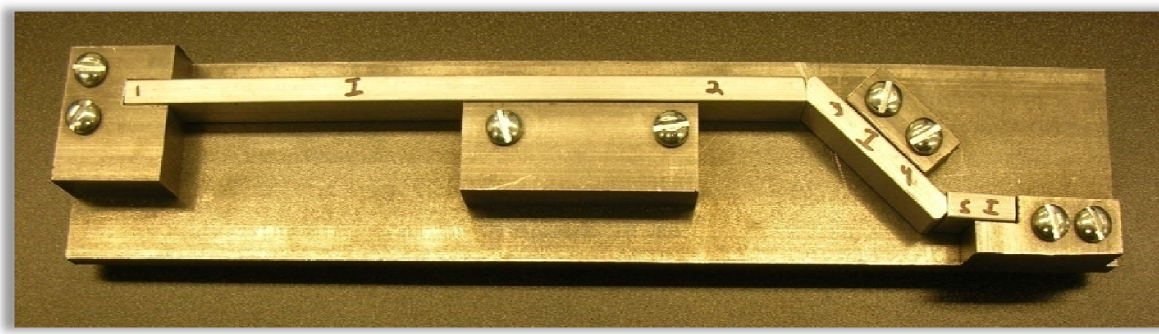


Figure 25: Hub spoke jig with three spoke segments mounted prior to welding

The fourth component of the hub assembly is the rim. Holes were drilled through the rim for mounting the spokes and covers. For this process a mill with a rotary table was used. The manufacturing process is shown in Figure 26.

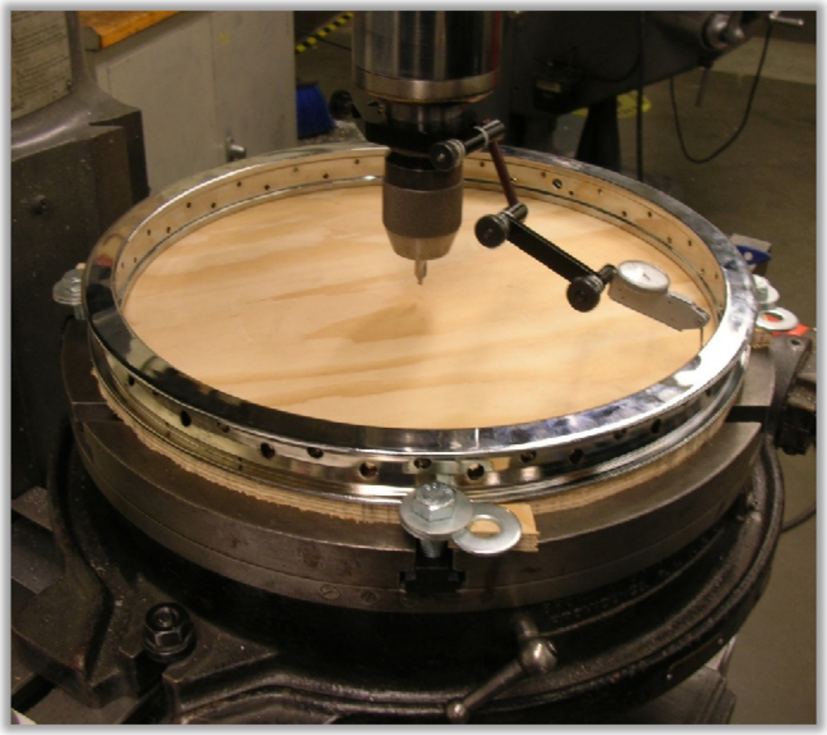


Figure 26: Aligning the rim on the mill in the ME Shop using a dial indicator

8.4.4 Superbracket

To accurately cut all of the holes in our superbracket, we used the water jet cutter in the ERC. Utilizing the final CAD model of our system we created a 2D model of the superbracket that was the computer input in the ERC. Using a water jet cutter saved time, improved accuracy, and allowed us to machine more complex shapes to lighten the bracket.

Along with the superbracket, a fork was made using steel tube. Tubes were welded onto the ends of the horizontal members of our current fork. This is shown in Figure 27 on page 47. Vertical tubes were fish-mouthed (shown in Figure 28 on page 47) and tee welded to the bottom of the horizontal tubes. Prior to welding, slots and through holes were milled into the bottom of the tubes to allow for axle placement and attachment. The fork was then painted and clear coated in the Wilson Center's paint booth. This is shown in Figure 29 on page 48, next to the CAD model of the fork.

Figure 27: Horizontal fork members prior to welding

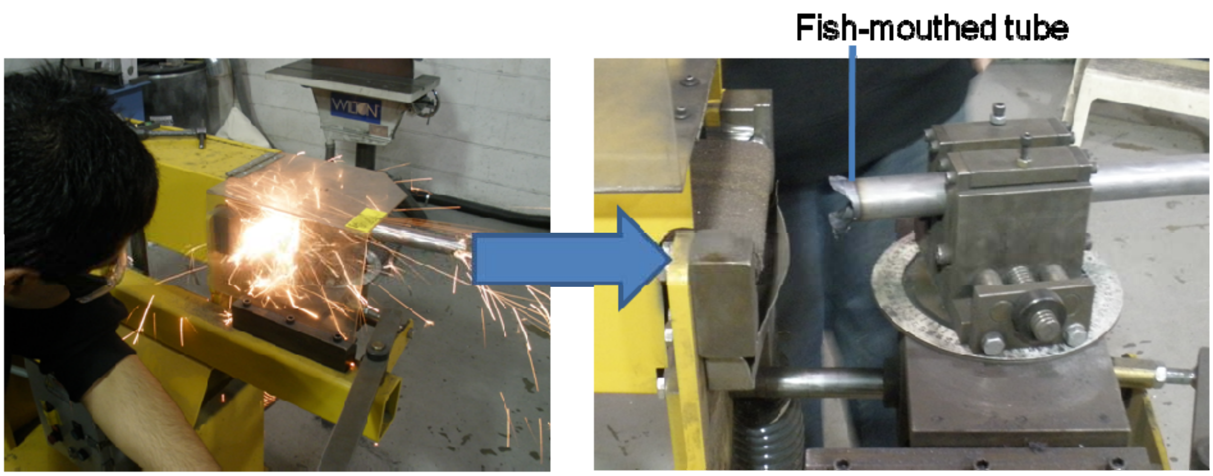


Figure 28: Fish-mouthing fork tubes in the Wilson Center

Figure 29: Front fork – From CAD model to reality

The axle was machined from ¾" × 0.083" steel tube. Holes were machined into either end for a through bolt that connects it to the fork. The tube was cut to length using a horizontal band saw. A third hole was drilled near the center of the tube to allow for wires to pass through the axle and out of the cover.

8.4.5 User Interface & Controls

The launch push-buttons located in the bar ends of the bicycle handlebar are mounted in carriers. These push-button carriers are made of 6061 aluminum. They were turned and drilled on a lathe at 1150rpm. The carriers were pressed into the ends of the bar ends (0.0005" interference fit). The master key switch was mounted in a similar manner, but on the horizontal end of the handlebar.

The wiring for the user interface was contained inside the handlebar and run into the hub through the axle. The wires were placed into position in the handlebar and the ends were soldered to their respective components. This way, there is some slack and the wires will be less susceptible to damage or wear. Corrugated plastic tubing surrounds the wires that are outside of the handlebar (e.g., wire running to the axle) for tightness and safety. The toggle switch was mounted in a bracket attached to the handlebars. Electrical components were assembled in the Wilson Center and the ME Shop.

8.5 Prototype Assembly
Once all components are fabricated or purchased, they are assembled into three main subassemblies: the hubs, the hydraulics, and the powertrain.

The hub assembly is shown below in Figure 30 on page 50. Two of these assemblies are made – one for each side of the rim. The rim and axle are used for alignment assistance during assembly. Proper alignment of these assemblies is critical – if the bearing carriers are not properly centered on the rim, the wheel will not spin true. One hub assembly is affixed to the rim while the other is left off, so that the powertrain and hydraulic assemblies may be placed inside the rim once complete.

The hydraulic assembly is shown in Figure 31 on page 51. Each SAE fitting is tightened firmly so that the o-ring is properly compressed, and each pipe fitting is coated with Jomar thread sealant prior to assembly. Components are bolted to the superbracket to ensure proper alignment of the motor and pump. As the motor and pump interface directly with the powertrain, their alignment is critical. The high-pressure accumulator is placed into its trough on the superbracket, and then secured with zip ties. Low-pressure vinyl tubing and hose barb fittings, not shown in the diagram, are assembled using hose clamps. To ensure correct hose routing, all low-pressure components are connected after the powertrain and hydraulic subassemblies are combined. It is especially important to route these lines after the clutch has been assembled on the superbracket, as they must be routed around it.

The powertrain assembly, by far the most complicated, is shown below in Figure 32 on page 52. The hydraulic assembly (not shown) is first mounted to the superbracket, followed by all of the powertrain components. Gear shafts are secured using c-clips and keys (not shown). It is at this point that the low-pressure and electrical components are assembled. This entire assembly is then inserted into the prepared hub-rim assembly, and the remaining hub is attached. Hub covers are placed on the outside of the hubs, and the axle is mounted into the bike fork. Care must be taken to ensure that bolt heads and nuts do not interfere with the fork uprights; any interference can be corrected by adjusting the shaft collars that constrain the superbracket to the axle.

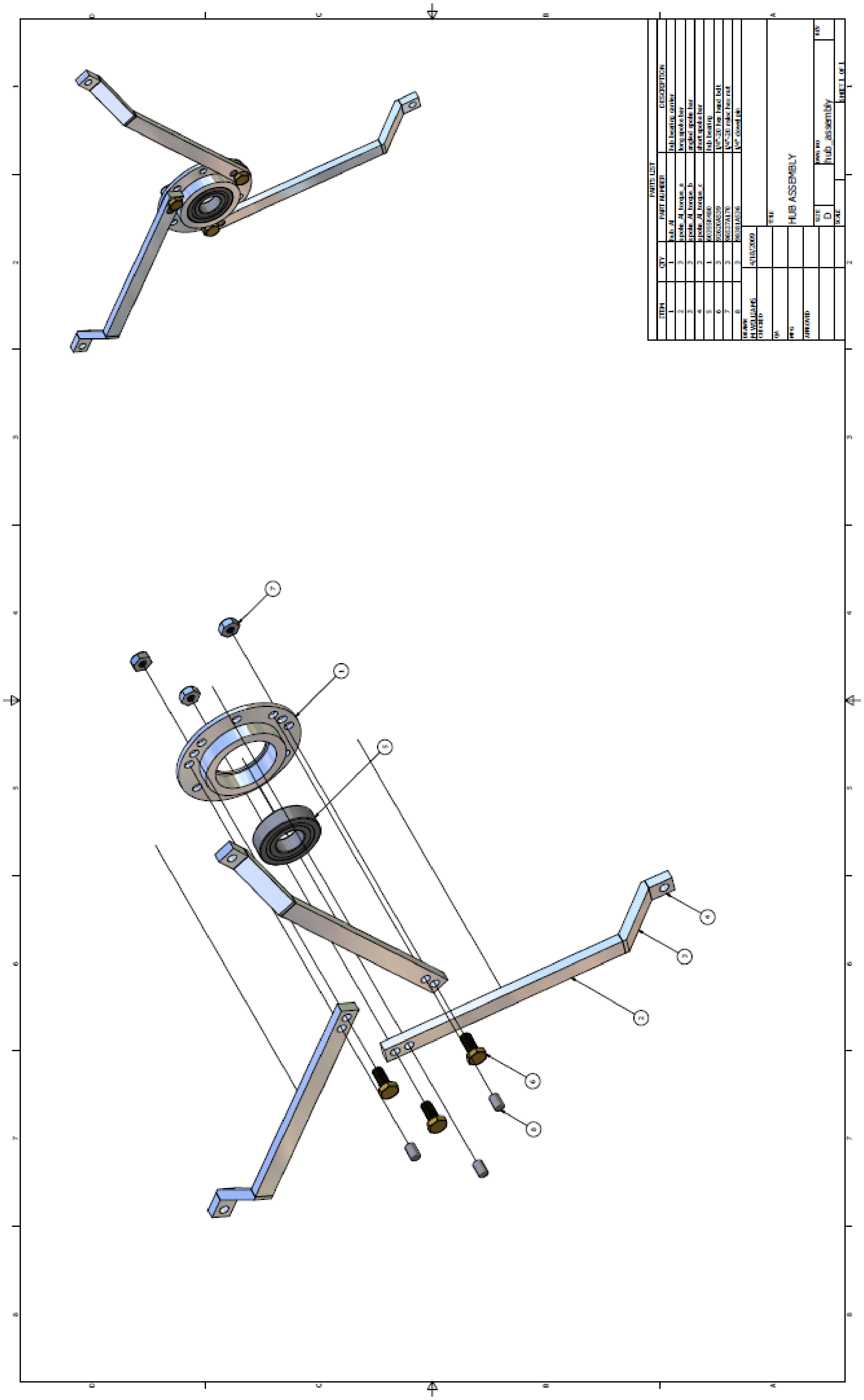


Figure 30: Hub assembly drawing

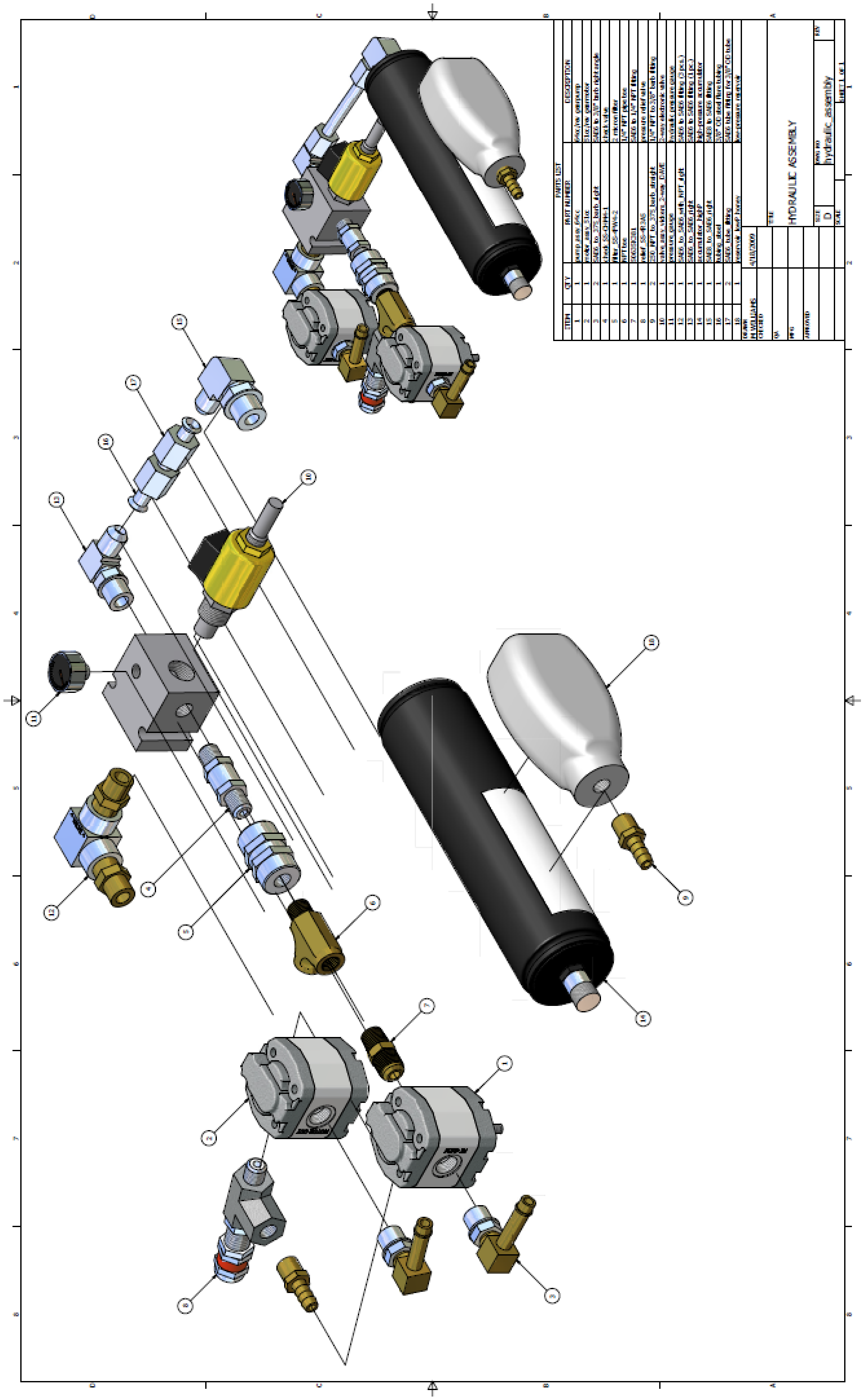


Figure 31: Hydraulic assembly drawing

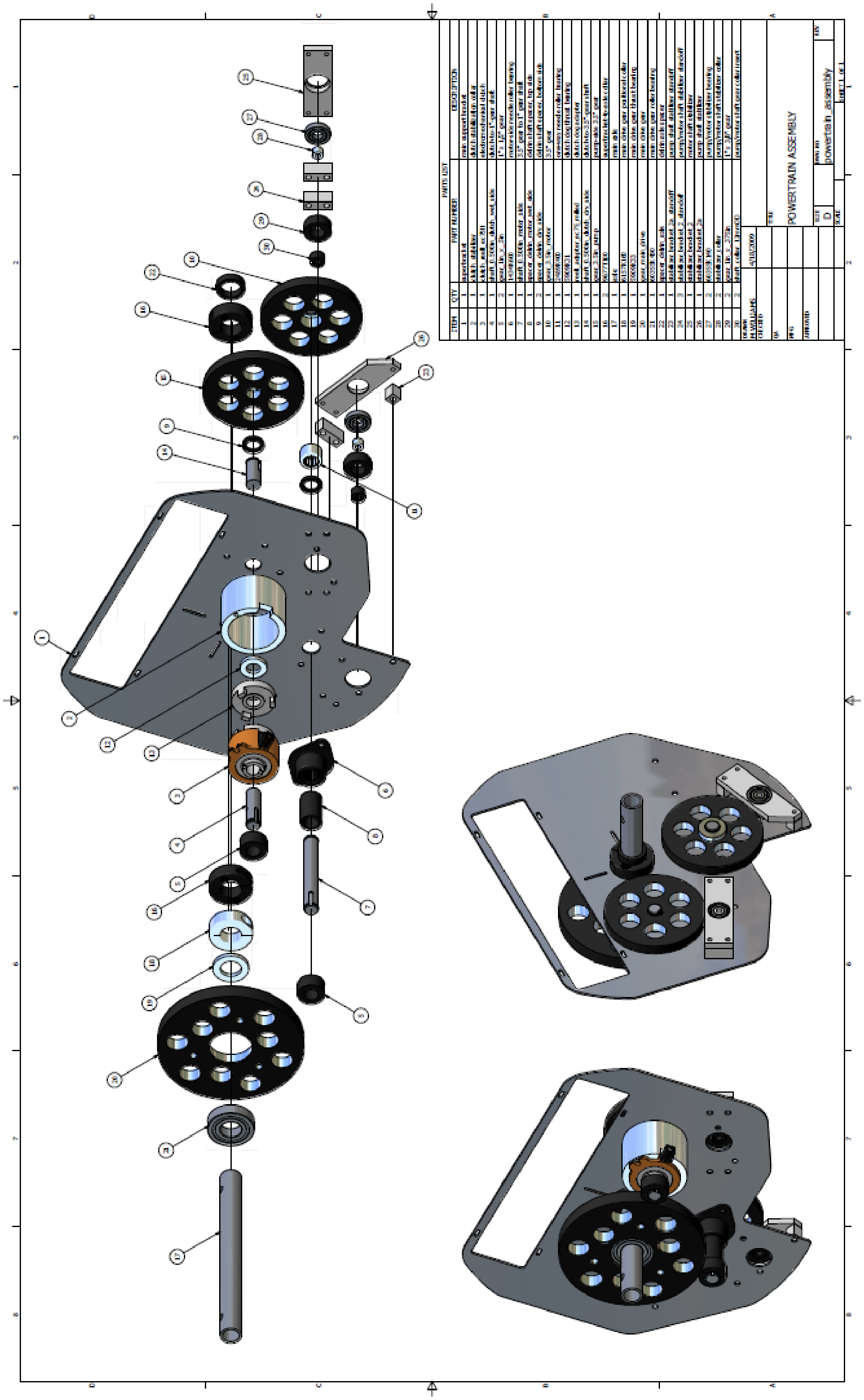


Figure 32: Powertrain assembly drawing

8.6 Validation Plan

One important aspect of this project is the validation of the prototype and the theoretical model. Testing the prototype will verify whether we satisfy our engineering specifications as well as validate the prediction capabilities of the model. Table 16 documents the basic testing layout we will perform to validate the system. The engineering analysis we will be testing is the storage pressure, acceleration, and final speed, since all other engineering specifications will be met through component selection, fabrication and assembly (i.e. hydraulic safety, weight, electrical safety, front fork).

To validate the system, we will be using one of our team members to ride the bike. Although we will not be using the 58 kg rider for which this bike is optimized, we have the Simulink model to predict the deceleration/acceleration and speeds of the bike for a heavier rider. One important result of the testing is to see how much of an impact losses in the system (of which only inertial loads and pump/motor efficiencies were modeled) have on the performance of the HRBS.

Table 16: Prototype testing scheme

Stage 1	*Performance Test of Isolated HRBS Wheel*
Objective:	The objective of this test is to measure the wheel speed of the HRBS for designated charge pressures to see how system behaves without rider and bike inertia.
Special Notes/Cautions:	Lack of bike and rider inertia may cause front wheel to spin faster than max operating conditions. High speed (close to 7000 rpm) on motor shaft may be a concern. Hand cranking to charge the system at higher pressures may not be possible in which case testing at higher pressures may be avoided or a different procedure involving a more powerful input (electric motor) may be required. If the latter is chosen, additional precautions and setup apparatus modifications may be required to safely constrain the bicycle.
Simulink Validation:	Compare measured final bike speeds at a given charge pressure to those predicted by the Simulink model (bike and rider inertia removed). Modify Simulink model accordingly (variable gain to represent all losses).
Parameters to Vary:	P_c(Charge Pressure)
Parameters to Measure:	V_f(Final Speed)
Measuring Devices:	Pressure Gauge (attached to 2 way valve) and Speedometer (Retrofit)
Engin. Spec. Validation:	$P_c < 4000$ psi *Important for testing pressure relief settings*
Test Procedure:	Raise and support front wheel on stand apparatus such that the only component allowed to move on the bike is the front wheel. Hand crank front wheel with HRBS brakes engaged up to a charge pressure of 200 psi. Activate system and measure peak speed of front wheel from speedometer. Perform this same test for charge pressures of 400 psi, 600 psi, 800 psi, up to 4000 psi.

Stage 2	*Performance Test of Fully Operating HRBS System*
Objective:	The objective of this test is to test the performance of the fully operating HRBS system.
Special Notes/Cautions:	In this stage, the bike is operating under real conditions. In no time during operation shall a person, other than the rider, be within 15 ft while the bike is moving. The covers must remain on the HRBS at all times unless modifications must made, in which case the system must be released of any stored pressure. We will be testing with a 180 lb rider in which case the peak operating velocities of the prototype must be reduced so that components are not overloaded with the additional inertia. Max operating speed for our testing will be 15 mph.
Simulink Validation:	Compare measured final bike speeds at given initial speeds to those of the Simulink model. Since rider weight is different, determine theoretical speeds and pressures for 180 lb rider and compare to measured results.
Parameters to Vary:	V_i (Initial Bike Speed)
Parameters to Measure:	P_c (Charge Pressure) and V_f (Final Speed)
Measuring Devices:	Pressure Gauge (attached to 2 way valve) and Speedometer (Retrofit)
Engin. Spec. Validation:	$P_c < 4000$ psi $a_{avg,d} = 2.6$ m/s^2 (1.8 m/s^2 for 180 lb rider) $a_{avg,a} = 1.4$ m/s^2 (1 m/s^2 for 180 lb rider) Values from Theoretical Model
Test Procedure:	Find an isolated location such that there are very few pedestrians. Make sure that no one comes near 15 ft of the bicycle when it is in motion. Pedal bike up to 5 mph and hit the brakes. (Measure charge pressure). Activate launch and measure final velocity of bike. Repeat procedure for initial speeds of 7, 9, 11, 13, and 15 mph.

Additional Safety Measures	
1	Operation of HRBS shall be permitted only if system is fully enclosed by hub cover (unless a pressure gauge and part rotation are being analyzed).
2	Testing area shall be clear of any pedestrians.
3	All modification of the hydraulic components will strictly follow measures outlined by Parker Hannifin corporation safety procedures. [21]
4	Rider must wear protective gear such as helmet and protective gloves in the case of falling from the bike.

9 Project Plan

We developed a plan of action to guide us through each stage of the design process, qualitatively separated into phases. The detailed components of these phases, as well as the timeline for their completion, are represented graphically in a Gantt chart in Appendix B. As of April 21, 2009 Phase IV is complete.

9.1 Phase I: Project Background & Specifications

Information was gathered from many sources, including:

- David Swain (EPA) – HRBS fundamentals, optimization, and customer requirements
- Prof. Steven Skerlos of U-M – project management and documentation
- ME490 HRBS team – mechanical assembly complications and sourcing issues
- Scholarly articles on regenerative braking, vehicle hydraulics, and energy storage

Based on these sources, we generated a set of customer requirements, and from these, developed a set of engineering specifications. The information gathered was used to ensure that these specifications are realistic and properly quantified. The correlation between the specs and the customer needs is documented in a QFD. This design phase was completed on January 26, 2009.

9.2 Phase II: Concept Generation & Selection

Design concepts were generated, based on information and specifications from Phase I. These concepts were refined through further discussions with our sponsor and ME490 students, as well as hands-on experience with the ME490 bike. Concept selection and preliminary 3D models/engineering drawings are complete. This required knowledge of the fundamental principles of dynamics (especially powertrain design), material selection, hydraulic design and static stress analysis. During this phase we began sourcing parts with long lead times to ensure that all parts will be on hand for our prototype build. We have acquired a bicycle and have collected quotes for hydraulic components. This design phase was completed on February 16, 2009.

9.3 Phase III: Final Design

The 3D model has been further refined, and a safety study has been performed on the device. This study does not just include user safety, but also team and stakeholder safety during the fabrication, assembly, and testing processes. Specifications based on components that are on order or on hand have been finalized. Parker Hannifin provided our team with Hydraulic SafetyWorks training on March 13, 2009. This design phase was completed on March 17, 2009.

9.4 Phase IV: Alpha Prototype

Upon design finalization, the prototype was fabricated and assembled. It was then validated against the technical specifications. Because the bicycle was not fully functional (the introduction of hydraulic fluid into the system generated superbracket deflection higher than expected), we were only able to test a subset of the original specifications. These include weight, cost, and a few safety considerations. The alpha prototype was assembled on April 14, 2009 and validation continues as of April 21, 2009. The cost of parts for this prototype is approximately $1730 (of which our team spent about $1340). We have divided the parts into four main categories: hydraulics, powertrain, support housing (superbracket and hub), and user interface. A more detailed description of the groups and individual parts can be found in Appendices C & D.

10 Recommendations

Throughout the course of the semester our team compiled a list of recommendations for future projects related to ours. This section of the report outlines these recommendations.

10.1 Motor/Pump Selection

While the hydraulic motor and pump used for this project were dramatically improved from those used prior to 2008, we feel that we have reached the smallest possible wheel width of an HRBS enclosed in a 20” bicycle. We recommend that our sponsor and section instructor further research hydraulic pumps prior to assigning a similar project. Since the lead times for these pumps are often very long (12 + weeks), it is recommended that these parts be purchased prior to the start of the semester. If at all possible, we recommend using a single pump to reduce the number of hydraulic components and weight. We believe that the greatest amount of weight reduction at this stage of the HRBS can come from removing more components of the hydraulic subsystem. Also, finding motor and pumps that have shaft sizes that can easily be attached to stock gears would greatly simplify assembly. The M6 shaft size on our pump and motors proved difficult to work with since we had to custom make adapters to connect the gears to the motor and pump shafts.

10.2 Clutch Selection

The electromechanical clutch that we chose is not an ideal component. First of all, the bore dimensions for the input shafts are significantly larger than specified. This forced us to create a keyway in the clutch in order for it to transmit torque properly. Another potential solution could be the implementation of a pin connection. Second, the three-dog drive hub fits loosely together with the rest of the clutch. A more desirable option would involve tighter specifications or a simple one-piece design. Both of these issues caused several problems in terms of gear alignment and proper meshing.

10.3 Hydraulic Manifold

Our discussions with Federal Fluid Power at the midway point of the semester resulted in an interesting concept regarding hydraulic routing. FFP suggested designing one large hydraulic manifold out of a steel or aluminum block. With proper drilling and tapping, the fluid can be routed through the manifold using cartridge-style valves and filters. The end result would be a marked reduction in fittings. Possible downsides include weight (using one large metal manifold will likely be heavy), routing (likely resulting in multiple 90-degree bends), and size (space needs to be allotted for the axle, accumulator, and pumps). FFP expressed some interest in providing technical assistance with designing such a manifold.

10.4 Superbracket stiffness analysis

Our team failed to analyze the stiffness of the superbracket prior to purchasing material. Ultimately the material was not stiff enough which resulted in an initial failure to make a functioning system. When the braking system was engaged, the loads on the system were large enough that they flexed the superbracket. This caused the gears to misalign and jam.

We recommend not overlooking such components based on engineering judgment. While we found our material to be stiff on the showroom floor, it was based on limited formal analysis that dramatically delayed our prototype.

10.5 Work with experts

Our team strongly recommends that students working on similar projects meet with hydraulic experts prior to designing a system. We waited too long to discuss our design with Federal Fluid Power (FFP) in Plymouth and in turn spent valuable time researching components without clear direction. The engineers at FFP have been very helpful and are supportive of this project.

Additionally, when selecting mechanical components, it is useful to talk with distributors such as Applied Industrial in Romulus. Applied works directly with manufacturers and is able to provide parts not listed on their website. These include custom gears. When looking at online descriptions, be careful not to accept distributor specifications, but rather call and ask. For example, we ordered gears that were listed to have keyways. The gears that we received did not have keyways. This was an error on Applied Industrial's website.

Ultimately what determines a working prototype is the quality of the mechanical components such as the hub structure, gears, bearing support, and superbracket. We recommend early on in the semester to get in touch with a mechanisms expert and a materials expert and maintain these contacts throughout the semester. We found that our ME450 graders did not have the in-depth hydraulic and gear train experience to provide in-depth critiques early on in our mechanical designs, especially with a system that is so compact.

10.6 Aggressive sourcing and manufacturing schedule

The only reason we were able to complete as much as we did was by aggressively sourcing parts at the beginning of the semester. This project contains many specialized components with long lead times. By ordering parts early in the semester we considerably extended our manufacturing time. This was necessary to have an assembled prototype for the design expo. Hydraulic components, in particular, typically have long lead times, so aggressive sourcing was critical to keeping the project on schedule.

10.7 Accurate CAD model

Due to the extremely small working envelope inside a 20" wheel, the tolerances between components must be quite small. Accurate CAD modeling is therefore critical to the successful design and completion of this project. We very highly recommend that the system be modeled in its entirety before any fabrication is attempted. Last-minute modifications due to unplanned shaft connections and unforeseen bolt head sizes resulted in loss of working time; making sure that every single component (down to the smallest bolt and retaining ring) is accounted for and modeled will save time in the long run. Not only is accurate CAD modeling vital to proper tolerances and clearances, it is also an effective way to generate images for presentations and reports.

11 Conclusion

This semester we designed and built a hydraulic regenerative braking system enclosed in a 20" bicycle wheel. We used hydraulic hybrid technology that was proven by the EPA and previous ME450 teams. Using the vast resources available to our team, we redesigned the mechanical and electrical systems on the bike. The hydraulic component specifications did not change from previous iterations of the bicycle. We reduced weight, improved safety, and increased functionality with our design and were motivated by those driving factors during manufacturing and assembly. We were able to meet the deadlines of our project by sourcing parts aggressively and scheduling proactively throughout the semester. In such a short design cycle, adherence to a methodical and thoughtful approach was necessary to avoid confusion and misguided efforts. It also allowed for each team member to have an intimate knowledge of the system and its components, resulting directly in a significant leap forward in the evolution of this project.

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13 Biographies

13.1 Bryan D'Souza

Bryan grew up in Troy, Michigan in a musical environment. He started playing the piano when he was five years old and began viola training at age nine. He is still very active musically, and considers it a positive outlet for creative expression. Since 2007, he has interned at General Electric Transportation in the finance department and General Electric Energy in accessories. He was particularly excited by the assignment at GE Energy, which involved the development of a hybrid-fuel gas turbine system. This has drawn him to focus on energy systems and their processes as a whole, and he hopes to pursue this interest into graduate school at the University of Michigan.

13.2 Andrew Kneifel

Andrew was born in Cincinnati, Ohio and was raised in an Ohio State family. After moving to Rochester Hills, Michigan in 1996, Andrew's passion for the outdoors and travelling grew dramatically. In 2003 he spent a month living with a host family in Bad Neustadt, Germany. Throughout his life he has been active in Scouting including two backpacking trips in New Mexico and sailing in the Bahamas. In 2005 he received the award of Eagle Scout. During a challenging university selection, Andrew decided to break allegiances and study Mechanical Engineering and German at U-M. Since beginning classes at U-M he has been an active member of the Baja SAE Racing Team. He is currently the Wilson Center's CNC Lathe trainer. Andrew co-oped twice for General Electric where he made clothes dryers dry faster and refrigerators cost less. Last summer he interned at Procter & Gamble's Green Bay plant in the Bounty paper towels department.

13.3 Victor Singh

Born in California but raised in rural Washington, Victor grew up on a 20 acre farm filled with odd ball phenomena such as warring birds, cows escaping and running onto the freeway, and the occasional rampaging headless chicken. This has no doubt led to his strange humor and thoughts. On the farm was where his passion for engineering ignited. He was fascinated by his family's bulldozer and tractor as they crawled across the fields, shaking the ground beneath as their shimmering hydraulic actuators raised the massive blades and arms attached to them. Unfortunately, this life didn't last as some issues with property development would prove to be an economic struggle later. So his family moved to city of Seattle; they brought the bulldozer and tractor of course. It was here where his engineering dream grew to full fruition. Taking classes at his local community college while still in high school, Victor entered the Human Powered Paper Vehicle Competition and learned how to build a bike out of paper and glue. At home, he and his father repaired semi-trucks, landscaped, and developed residential properties. After graduating from high school and staying an additional year at community college, Victor decided to leave home and pursue his engineering dream here at UM. During his first year, he joined the UM Baja SAE team and is currently an active member. He still entertains his strange thoughts, but now they are mixed with calculus.

13.4 Matthew Williams

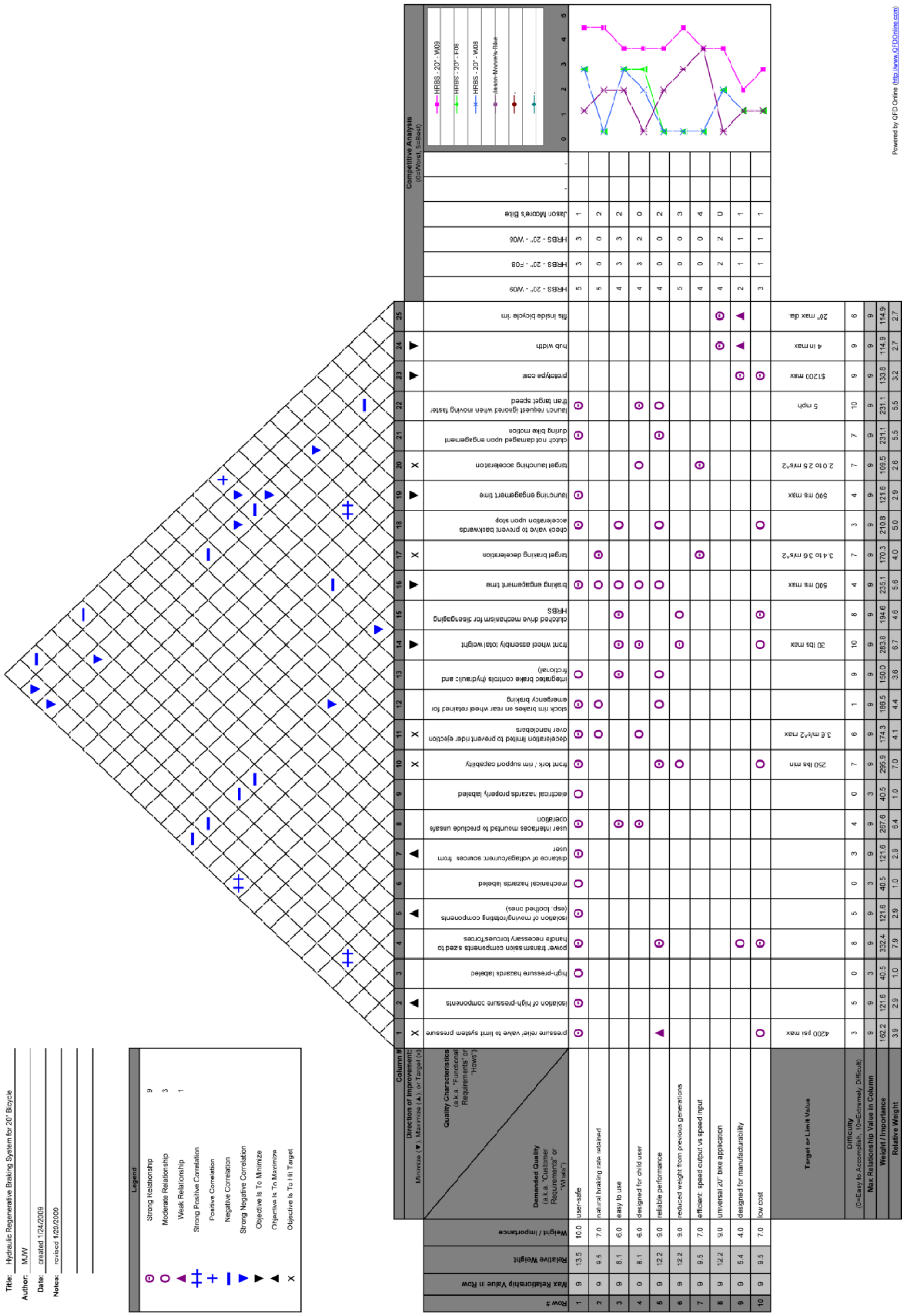
Born and raised in Troy, Michigan, Matt has always loved tinkering with broken appliances. Mechanical Engineering at Michigan was a natural choice, given his affinity for mechanical problem solving and driving in the snow. He worked at Boston Market for two years as a carver, and can quarter a cooked chicken in seconds. Matt currently works for Ann Arbor-based Solidica Inc. as a CAD consultant, designing enclosures for vehicle telematics sensors. His latest enclosure design was described as "having that satisfyingly robust feel of a poker chip." He enjoys thermodynamics, and has a working model of a Stirling-cycle engine that he found at a garage sale for $5 that he likes to take apart and put back together for no reason other than simply because he can. He also enjoys mechanical linkages and 3D modeling, and likes nothing better than a well-dimensioned drawing, except perhaps his fiancée.

14 Acknowledgements

We would like to thank those people and organizations that helped make this project a success:

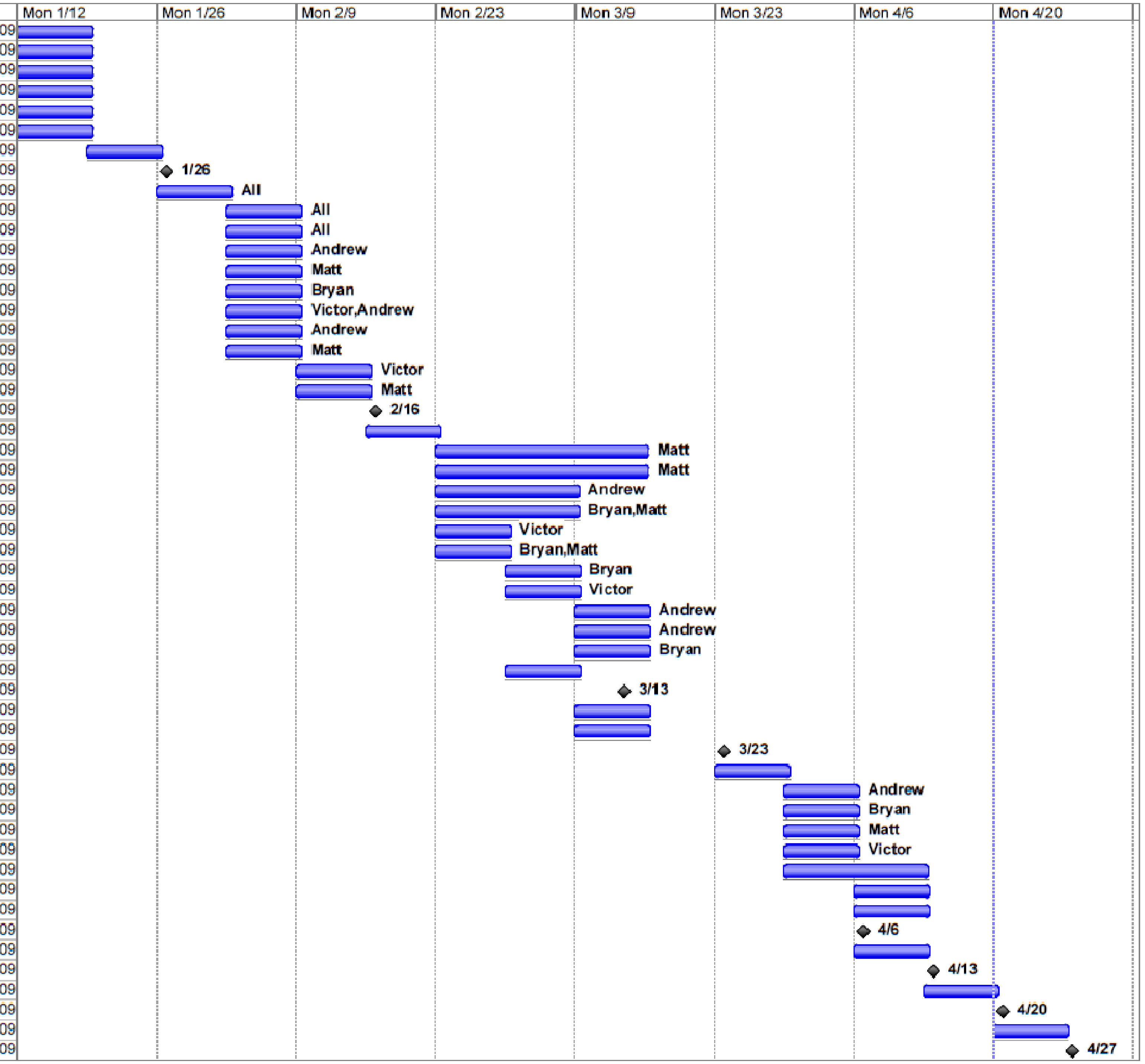
David Swain, EPA; Professor Steven Skerlos; Mark Krecic, A&A Woodshop; School of Art & Design; Taubman College of Architecture + Urban Planning; Steve Erskine, ERC; Federal Fluid Power; Professor Peter Washabaugh, Wilson Center; Bob Coury; Marv Cressey; Brian Min; Kenneth Tsang; Alex Ogdon; Great Lakes Cycling & Fitness

Appendix A – QFD



Appendix B – Gantt Chart

ID	Task Name	Duration	Start	Finish
1	Talk to Dave Swain, Skerlos, previous team for initial thoughts	1 wk	Mon 1/12/09	Mon 1/19/09
2	Information gathering (general and specific)	1 wk	Mon 1/12/09	Mon 1/19/09
3	Decide on project for this semester, receive customer requests	1 wk	Mon 1/12/09	Mon 1/19/09
4	Delineate roles for ME450 and ME490	1 wk	Mon 1/12/09	Mon 1/19/09
5	Design Engineering Specifications	1 wk	Mon 1/12/09	Mon 1/19/09
6	Discuss Budget Constraints, sourcing limitations	1 wk	Mon 1/12/09	Mon 1/19/09
7	Develop Gantt Chart, QFD	1 wk	Mon 1/19/09	Mon 1/26/09
8	Prepare Presentation and report for DR1 (1/29, 1/30)	1 day	Mon 1/26/09	Mon 1/26/09
9	Concept Generation, Engineering Fundamentals and Functional Decomposition	1 wk	Mon 1/26/09	Mon 2/2/09
10	Talk to Dave Swain, Skerlos, previous team for conceptual refinement	1 wk	Mon 2/2/09	Mon 2/9/09
11	Disassemble previous bicycle for further understanding	1 wk	Mon 2/2/09	Mon 2/9/09
12	Invite Dave Swain to DR2	1 wk	Mon 2/2/09	Mon 2/9/09
13	Confirm A/V setup for presentation	1 wk	Mon 2/2/09	Mon 2/9/09
14	Clutch research	1 wk	Mon 2/2/09	Mon 2/9/09
15	Gears/Transmission research	1 wk	Mon 2/2/09	Mon 2/9/09
16	Wheel Rim research	1 wk	Mon 2/2/09	Mon 2/9/09
17	Bicycle Acquisition, Filter/relief valve research	1 wk	Mon 2/2/09	Mon 2/9/09
18	Develop outline for presentation	1 wk	Mon 2/9/09	Mon 2/16/09
19	Initial Engineering Drawing (or Experiment?)	1 wk	Mon 2/9/09	Mon 2/16/09
20	Finalize Presentation, report for DR2 (2/19, 2/20)	1 day	Mon 2/16/09	Mon 2/16/09
21	Write up safety proposal with hazards and procedures	1 wk	Mon 2/16/09	Mon 2/23/09
22	Complete 3D CAD model	19 days	Mon 2/23/09	Mon 3/16/09
23	Component sourcing - 2-way valve, check valve, filter, hydraulic fittings, tubing	19 days	Mon 2/23/09	Mon 3/16/09
24	Gear Fatigue Analysis	13 days	Mon 2/23/09	Mon 3/9/09
25	FEA of hub designs	13 days	Mon 2/23/09	Mon 3/9/09
26	Mathematical model of powertrain	7 days	Mon 2/23/09	Mon 3/2/09
27	Finalize electrical system	7 days	Mon 2/23/09	Mon 3/2/09
28	Wiring harness assembly	7 days	Mon 3/2/09	Mon 3/9/09
29	Powertrain design	7 days	Mon 3/2/09	Mon 3/9/09
30	Create mold for vacuum forming hub	7 days	Mon 3/9/09	Mon 3/16/09
31	Component sourcing - gears, bike rim	7 days	Mon 3/9/09	Mon 3/16/09
32	Component sourcing - clutch, electrical components, hydraulic fluid	7 days	Mon 3/9/09	Mon 3/16/09
33	Safety Review	7 days	Mon 3/2/09	Mon 3/9/09
34	Hydraulic Safety Training with Parker-Hannifin	1 day	Fri 3/13/09	Fri 3/13/09
35	Discuss expected specifications based on final design	1 wk	Mon 3/9/09	Mon 3/16/09
36	Anticipate problems and create solutions	1 wk	Mon 3/9/09	Mon 3/16/09
37	Prepare for DR3 (3/26)	1 day	Mon 3/23/09	Mon 3/23/09
38	Talk to Dave Swain, previous team for assembly guidance	1 wk	Mon 3/23/09	Mon 3/30/09
39	Machine gears, shafts, rim; create vacuum-formed hub covers	1 wk	Mon 3/30/09	Mon 4/6/09
40	Construct fork, validate user interface, obtain superbracket, axle materials	1 wk	Mon 3/30/09	Mon 4/6/09
41	Hydraulic assembly, final CAD final, part drawings	1 wk	Mon 3/30/09	Mon 4/6/09
42	Machine spokes	1 wk	Mon 3/30/09	Mon 4/6/09
43	Final assembly	13 days	Mon 3/30/09	Mon 4/13/09
44	Initial test and validation of prototype and system model	1 wk	Mon 4/6/09	Mon 4/13/09
45	Final modifications and validation	1 wk	Mon 4/6/09	Mon 4/13/09
46	Prepare for DR4 (4/7)	1 day	Mon 4/6/09	Mon 4/6/09
47	Develop poster and presentation tasks, arrange for transport of bicycle	1 wk	Mon 4/6/09	Mon 4/13/09
48	Prepare for Expo (4/16)	1 day	Mon 4/13/09	Mon 4/13/09
49	Solidify information and documentation	1 wk	Mon 4/13/09	Mon 4/20/09
50	Prepare for Final Report (4/21)	1 day	Mon 4/20/09	Mon 4/20/09
51	Arrange for transport of bicycle, finalize reimbursements	1 wk	Mon 4/20/09	Mon 4/27/09
52	Prepare for Final Prototype Delivery (4/30)	1 day	Mon 4/27/09	Mon 4/27/09



Appendix C – Prototype Costs

Hydraulic		**User interface**	
*Motor	$80	Switches	$19
*Pump	$80	Electrical wiring/components	$22
*High-pressure accumulator	$150	Batteries	$11
2-way valve housing	$42	Shipping/tax	$25
*2-way valve cartridge/solenoid	$80		
Relief valve	$141		
High pressure filter	$86		
Check valve	$51		
Low pressure accumulator	$3		
Fittings, hydraulic lines	$74		
Hydraulic fluid	$8		
Re-boring operations	$122		
Shipping/tax	$9		
Subsystem Total	**$926**	**Subsystem Total**	**$77**

Powertrain		**Structural**	
Electromechanical clutch	$153	20" rim	$30
One-way bearing	$9	Bicycle frame	$20
Gears	$233	Plastic hub material	$28
Support accessories	$53	Spoke material	$7
Shipping/tax	$55	Fork & axle material	$20
		Superbracket material	$22
		Nuts, bolts, collars, etc	$55
		Bearings	$25
		Shipping/tax	$15
Subsystem Total	**$503**	**Subsystem Total**	**$222**

GRAND TOTAL	**$ 1728**

*Denotes items provided by David Swain, EPA

Appendix D – Bill of Materials

Italicized items were not used on the final bike

	Part (Qty)	Cost	Part Number	Vendor	Manufacturer
Hydraulic	Hose clamps	$ 13.86	408758	Carpenter Bros. Hardware	
	Tube Splices	$ 3.57		Carpenter Bros. Hardware	
	Tube Adapter	$ 0.79		Carpenter Bros. Hardware	
	Thread Sealant	$ 3.99		Carpenter Bros. Hardware	
	tax	$ 1.33			
	[Service] - rebore ports	$ 121.90		Federal Fluid Power	
	Hard tube + fittings	$ 34.23		Federal Fluid Power	
	soft tubing	$ 1.38		Carpenter Bros. Hardware	
	Tubing tee (2)	$ 4.08	048643074033	Home Depot	
	Tubing adapter (2)	$ 4.74	048643074491	Home Depot	
	Vinyl tubing (25')	*$ 7.34*	*048643120136*	*Home Depot*	
	tax	$ 0.97			
	Pressure Gauge	$ 9.80	9767T217	McMaster-Carr	
	Hydraulic fluid (Mobil 1 Synthetic)	$ 8.00		Autozone	
	Two-way housing	$ 41.68	SBV11-8V-CM-S6T-24BQP-00	RHM Fluid Power	Eaton Vickers
	Two-way cartridge/solenoid	$ 80.00	MCSCP024DQ000010	RHM Fluid Power	Eaton Vickers
	Filter	$ 85.60	SS-4FW4-2	Swagelok	
	Relief Valve	$ 137.30	SS-4R3A5	Swagelok	
	Spring kit	$ 3.90	177-R3A-K1-G	Swagelok	
	Check valve	$ 51.20	SS-CHM4-1	Swagelok	
	shipping	$ 7.07			
	Pump	$ 80.00	UO.25D60 TR RO		Marzocchi
	Motor	$ 80.00	UO.25S48 TR RO		Marzocchi
	Accumulator	$ 150.00	ACP05AA032E1KTC		Parker-Hannifin
	LP Reservoir	$ 3.00		Meijer	
User Interface	Keylock	$ 5.43	CKC8018-ND	Digi-key	
	Fuse 1.5A (4)	$ 2.68	507-1014-ND	Digi-key	
	Fuse 0.5A (4)	$ 2.68	507-1011-ND	Digi-key	
	Pushbutton switch (2)	$ 7.54	EG1932-ND	Digi-key	
	Toggle switch	$ 5.85	360-1902-ND	Digi-key	
	Voltage regulator 24V 1A	*$ 0.77*	*497-1458-5-ND*	*Digi-key*	
	tax	$ 1.50			
	shipping	$ 4.80			
	MOSFET (2)	$ 4.80	STP16NK60Z	Digi-key	
	9V battery snap (4)	$ 1.32	BS12I-HD-24AWG-ND	Digi-key	
	Voltage regulator 24V 2A	$ 0.68	497-1470-5-ND	Digi-key	
	tax	$ 0.41			
	shipping	$ 2.02			
	MOSFET	*$ 2.40*	*STP16NK60Z*	*Digi-key*	
	Female crimp connector (10)	$ 2.70	WM18235-ND	Digi-key	
	tax	$ 0.31			
	shipping	$ 1.85			
	9V Alkaline Battery (4)	$ 10.96	71455K56	McMaster-Carr	
	shipping	$ 4.75			
	24AWG wire (25')	$ 1.72	8073K617	McMaster-Carr	
	Female crimp connector (10)	$ 2.56	7060K58	McMaster-Carr	
	shipping	$ 9.00			
Powertrain	5" gear	$ 57.46	S1680	Applied Industrial Technologies	Martin Sprocket and Gear
	3.5" gears (2)	$ 81.86	S2070	Applied Industrial Technologies	Martin Sprocket and Gear
	1" 16DP gears (2)	$ 47.16	S1616BS1/2	Applied Industrial Technologies	Martin Sprocket and Gear
	1" 20DP gears (2)	$ 46.20	S2020BS1/2	Applied Industrial Technologies	Martin Sprocket and Gear
	tax	$ 13.96			
	shipping	$ 26.68			
	Thrust bearing 0.5" ID	$ 2.66	5909K31	McMaster-Carr	
	Thrust bearing washer 0.5" ID	$ 1.80	5909K44	McMaster-Carr	
	Thrust bearing 0.75" ID	$ 2.73	5909K33	McMaster-Carr	
	Thrust bearing washer 0.75" ID	$ 2.00	5909K46	McMaster-Carr	
	Flange-mounted needle bearing	$ 9.86	1434K6	McMaster-Carr	
	Ball bearing 0.375" ID (2)	$ 9.10	60355K14	McMaster-Carr	
	shipping	$ 4.75			
	Ball bearing 0.375" ID (2)	$ 10.20	60355K45	McMaster-Carr	
	Ball bearing 0.5" ID	$ 6.90	6384K61	McMaster-Carr	
	shipping	$ 4.50			
	key stock 0.125" zinc plated	$ 1.19	N180-158	Carpenter Bros. Hardware	
	tax	$ 0.07			
	Key Stock 0.1875"	*$ 2.74*	*98535A140*	*McMaster-Carr*	
	Motor/pump shaft collar 6mm ID (3)	$ 4.05	57485K66	McMaster-Carr	
	One-way needle bearing 0.5" ID	$ 9.46	2489K4	McMaster-Carr	
	Electrical Clutch 0.5" ID	$ 153.20	5156T4	McMaster-Carr	
	shipping	$ 4.75			
Structural	Nuts, bolts, nails, and screws	$ 12.09		Carpenter Bros. Hardware	
	tax	$ 0.73			
	Alex double-wall rim	$ 29.99		eBay	Alex
	shipping	$ 11.25			
	0.75" OD axle	$ 3.44	29110540	Alro Metals Plus	
	Spoke Material	$ 7.22	21436660	Alro Metals Plus	
	Stainless steel sheet 14 gauge	$ 13.75		Alro Metals Plus	
	1.25" OD fork	$ 16.85	29111450	Alro Metals Plus	
	Stainless steel sheet 16 gauge	*$ 8.25*		*Alro Metals Plus*	
	Solid Aluminum 3" bar	*$ 5.50*		*Alro Metals Plus*	
	Main gear-axle shaft collar	$ 3.36	6157K16	McMaster-Carr	
	Superbracket-axle shaft collar (2)	$ 14.86	9677T1	McMaster-Carr	
	Nuts, bolts, nails, and screws	$ 1.20		Carpenter Bros. Hardware	
	tax	$ 0.36			
	epoxy	$ 3.99	325847	Carpenter Bros. Hardware	
	Nuts, bolts, nails, and screws	$ 2.40		Carpenter Bros. Hardware	
	tax	$ 0.38			
	ABS plastic for hub covers	$ 28.00	1ZBT7	Grainger	
	tax	$ 1.68			
	Nuts, bolts, nails, and screws	$ 12.09		Carpenter Bros. Hardware	
	tax	$ 0.73			
	Bicycle	$ 20.00		Craigslist	
	Ball bearing 0.75" ID	*$ 8.49*	*60355K48*	*McMaster-Carr*	
	Ball bearing 0.75" ID (2)	$ 16.98	60355K48	McMaster-Carr	
	TOTAL	**$ 1,739.33**			

Table 17: Breakdown of component weights

Component	Weight (lb)
Rim	1.260
Spokes (6)	0.600
Tire	2.175
Superbracket	1.875
Electromechanical clutch	0.800
2-way valve assembly	1.904
High pressure accumulator	5.052
Relief valve	0.456
High pressure filter	0.249
Check valve	0.145
Low pressure reservoir	0.090
Fittings, hydraulic lines	1.553
Hydraulic pump	0.833
Hydraulic motor	0.798
Gears	3.727
Miscellaneous hardware	3.233
Total	24.75

Appendix E – Design Analysis Assignment

For our design analysis assignment, we chose to take a different approach on material selection. This was based on the nature of our project and the components involved in making it. The following section of this report discusses the material selection, environmental performance, and manufacturing processes for our spokes and gears.

Spoke Material Selection

Material selection for spokes is described in Section 8.1.3 Hub on page 29 of this report. The following figures and tables show the results of the calculations.

Figure 33: Comparison of Yield Strength and Young's Modulus for Aluminum, Steel Alloys, and Copper Alloys

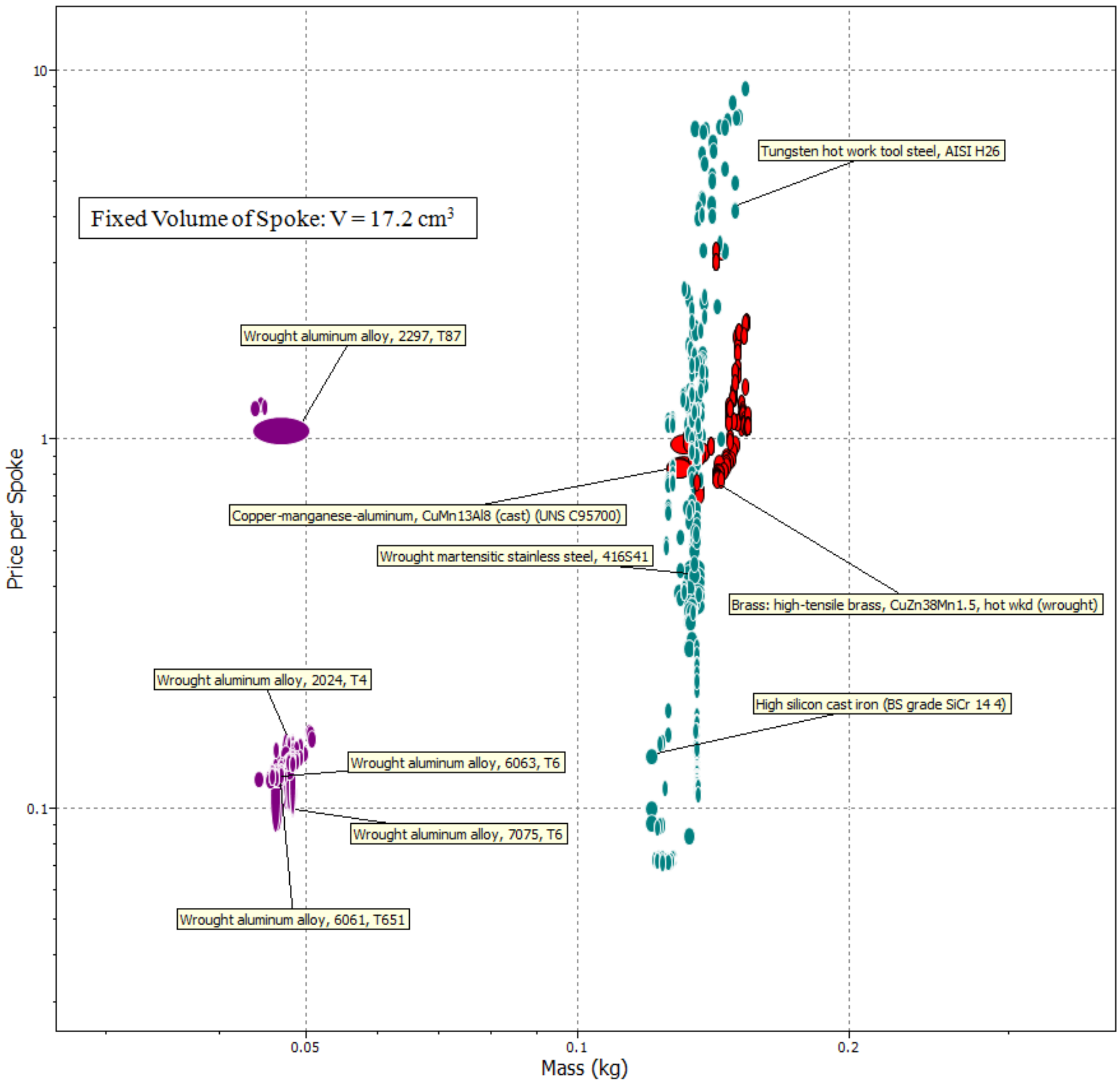


Figure 34: Comparison of masses and prices for Aluminum, Steel Alloys, and Copper Alloys

Equations used to make the following calculations are based on *Fundamentals of Machine Elements*

Table 18: Spoke Buckling Calculations

Loading based on acceleration from 4000 psi with 165 lb rider

Spoke Dimensions				
w	0.0127	m	0.5	in
t	0.00635	m	0.25	in
L	0.16256	m	6.4	in
A	8.065E-05	m^2		
I	2.71E-10	m^4		

Loading Conditions from FEA			
Torque	300	lbs-in	
W	400	lbs	> 2.4x Overload
Max σ	5183	psi	Determined from FEA
Eq. F	647.875	lbs	

Material Properties of Spoke (6061 AL)			
$S_{y,c}$	9.70E+07	Pa	*Compressive Yield*
E	6.8E+10	Pa	*Young's Modulus*
r_g	0.0018331	m	*Radius of Gyration*

Buckling Properties of Spoke		
k	0.5	*pinned-fixed*
C_c	117.63416	*Critical Slenderness Ratio*
L_e/r_g	44.340501	*Slenderness Ratio*
n_a	1.8013231	*Reduction Factor*

Possible Failure Modes		
Elastic Buckling		
Predicted Euler Buckling Failure Load	27528.65	N
	6188.687	lbs
Predicted AISC Buckling Failure Load	14362.77	N
	3228.88	lbs
Inelastic Buckling		
Predicted Johnson's Failure Load	7266.848	N
	1633.652	lbs
Predicted AISC Buckling Failure Load	4034.173	N
	906.9181	lbs

Failure loading			
$P_{critical}$	**906.918072**	**lbs**	**By Inelastic Buckling**
$\sigma_{critical}$	**7255.34458**	**psi**	

$$P_{cr} = \frac{\pi^2 EI}{(kL)^2} \quad \text{Euler Elastic Buckling} \qquad \frac{kL}{r_g} \geq C_c$$

$$P_{cr} = A\left(S_{y,c} - \frac{S_{y,c}^2}{4\pi^2 E}\left(\frac{kL}{r_g}\right)^2\right) \quad \text{Johnson's Inelastic Buckling} \qquad \frac{kL}{r_g} \leq C_c$$

$$C_c = \sqrt{\frac{2\pi^2 E}{S_{y,c}}}$$

$$r_g = \sqrt{\frac{I}{A}}$$

$$P_{cr} = A\frac{12\pi^2 E}{23(kL/r_g)^2} \quad \text{AISC Elastic Buckling} \qquad \frac{kL}{r_g} \geq C_c$$

$$P_{cr} = \frac{A\left(1 - \frac{(kL/r_g)^2}{(2C_c^2)}\right)S_{y,c}}{n_a} \quad \text{AISC Inelastic Buckling} \qquad \frac{kL}{r_g} \leq C_c$$

$$C_c = \sqrt{\frac{2\pi^2 E}{S_{y,c}}}$$

$$r_g = \sqrt{\frac{I}{A}}$$

$$n_a = \frac{5}{3} + \frac{3(kL/r_g)}{8C_c} - \frac{(kL/r_g)^3}{8C_c^3}$$

Gear Material Selection

Material selection for gears is described in section 8.1.2 Powertrain on page 27 of this report. The following tables show the results of the calculations. Please note that it is not known exactly which alloy of steel was used to make the gears as the manufacturer uses different alloys for different gear sizes. Most likely 1018 or 1020 steel was used.

Equations used to make the following calculations are based on *Shigley: Mechanical Engineering Design.*

Table 19: Gear selection symbols and their names [18]

Symbol	Name
BHN	Brinell Hardness
C_f	Surface condition factor
C_p	Elastic coefficient
d_p	Pitch diameter
F	Face width
I	Geometry factor of pitting resistance
J	Bending strength geometry factor
K_m	Load-distribution factor
K_o	Overload factor
K_R	Reliability factor
K_s	Size factor
K_T	Temperature factor
K_v	Dynamic factor
$N1$	# teeth on Gear 1
$n1$	rpm of gear 1
$N2$	# teeth on Gear 2
$n2$	rpm of gear 2
S_c	AGMA surface endurance strength
S_F	AGMA factor of safety
S_t	AGMA bending strength
W^t	Tangential tooth load
Y_N	Stress cycle factor
$\sigma_{b,all}$	Allowable bending stress on a gear tooth
$\sigma_{c,all}$	Allowable surface stress

Table 20: 1” to 3.5” Gear reduction calculations

Loading based on deceleration from 4000 psi with 165lb rider

Spur Gear Design Calculation - Based on Shigley & Mischke, ed. 6

Worksheet taken from http://courses.washington.edu/mengr356/daly/hand_outs.html

Common data:			Gear 1 (pinion)			Gear 2 (gear)		
Power	1.5	HP	N1	20		N2	70	
Gear ratio	3.5		Diameter	1	in	Diameter	3.5	in
Pitch	20		n1	5927	rpm	n2	1693.4	rpm
Circ. Pitch	0.1571	in	Pitch line vel.	1551.7	ft/min	Pitch line vel.	1551.7	ft/min
Press. Angle	14.5	degrees	W^t	31.90	lb.	W^t	31.90	lb.
Factor of safety	1		Torque	15.95	in-lb	Torque	55.83	in-lb
K_o	1.5		BHN (core)	315		BHN (core)	315	
Q_V	5		S_t (grade 1)	37149.5	psi	S_t (grade 1)	37149.5	12800
A	54.77							
B	0.91		BHN (surface)	500		BHN (surface)	500	
K_v	1.642		S_c (grade 1)	190100	psi	S_c (grade 1)	190100	psi
K_m	1.45							
k_b	1		J	0.27		J	0.395	
K_s	1.000		Y_N	1		Y_N	1	
Press. Angle	0.2531	radians	K_R	1		K_R	1	
			C_P	2300		C_P	2300	
			I	0.094		I	0.094	
					For grade 1 steel			
Enter all data shown in blue			$\sigma_{b,all}$	37149.5	psi	$\sigma_{b,all}$	37149.5	psi
Enter Pitch			F	0.227	in	F	0.155	in
Results in red			$\sigma_{c,all}$	190100	psi	$\sigma_{c,all}$	190100	psi
			F	0.177	in	F	0.177	in

Table 21: 1” to 5” Gear reduction calculations

Loading based on deceleration from 4000 psi with 165lb rider

Spur Gear Design Calculation - Based on Shigley & Mischke, ed. 6

Worksheet taken from http://courses.washington.edu/mengr356/daly/hand_outs.html

Common data:			Gear 1 (pinion)			Gear 2 (gear)		
Power	1.5	HP	N1	16		N2	80	
Gear ratio	5.0		Diameter	1	in	Diameter	5.0	in
Pitch	16		n1	1693.4	rpm	n2	338.7	rpm
Circ. Pitch	0.1963	in	Pitch line vel.	443.3	ft/min	Pitch line vel.	443.3	ft/min
Press. Angle	14.5	degrees	W^t	111.65	lb.	W^t	111.65	lb.
Factor of safety	1		Torque	55.83	in-lb	Torque	279.13	in-lb
K_o	1.25		BHN (core)	315		BHN (core)	315	
Q_V	5		S_t (grade 1)	37149.5	psi	S_t (grade 1)	37149.5	12800
A	54.77							
B	0.91		BHN (surface)	500		BHN (surface)	500	
K_v	1.347		S_c (grade 1)	190100	psi	S_c (grade 1)	190100	psi
K_m	1.45							
k_b	1		J	0.27		J	0.42	
K_s	1.000		Y_N	1		Y_N	1	
Press. Angle	0.2531	radians	K_R	1		K_R	1	
			C_P	2300		C_P	2300	
			I	0.101		I	0.101	
					For grade 1 steel			
Enter all data shown in blue			$\sigma_{b,all}$	37149.5	psi	$\sigma_{b,all}$	37149.5	psi
Enter Pitch			F	0.435	in	F	0.279	in
Results in red			$\sigma_{c,all}$	190100	psi	$\sigma_{c,all}$	190100	psi
			F	0.395	in	F	0.395	in

Spoke Environmental Performance

The first component we chose for DFES analysis is our aluminum spoke. The material choice on the spokes was between solid aluminum strip stock and rectangular steel tubing. With a total of six spokes, the aluminum option weighs approximately 0.28 kg while the steel option weighs around 0.52 kg. The environmental impact of these materials was explored using SimaPro 7, classifying the aluminum as Al99 I and the steel as Fe360 I. The EcoIndicator 99 I calculation method was used. The resultant estimated emissions from these material selections are shown in Figure 35 below. In all cases, the aluminum option indicates significantly higher emissions, even though the total mass of aluminum is around half that of steel.

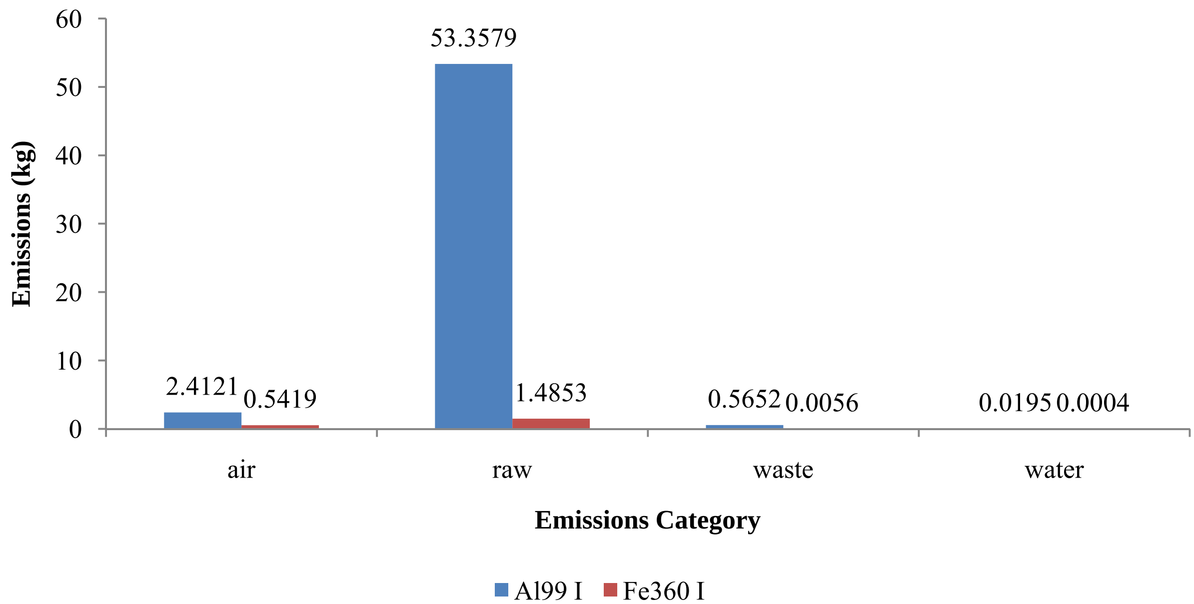


Figure 35: Estimated emissions for aluminum and steel spoke options

SimaPro also provides estimated data about the environmental usage and human effects of material processing. The relative effects of the aluminum option and steel option are shown in Figure 36 on page 73. As with emissions, the negative effects of the aluminum option are greater in all cases. In Figure 37 on page 73, these data are condensed and normalized into three categories: human health, ecosystem quality, and resources. Again, steel spokes are shown to be environmentally preferable. In Figure 38 on page 74, the data is condensed again, resulting in a single final "environmental score" for each option – the lower the score, the lower the negative environmental impact of the material.

The results of our DFES analysis clearly indicate that steel spokes have less negative impact on the environment than aluminum spokes do. However, our redesign of the HRBS was driven by material weight, cost, and availability: aluminum spokes made from strip stock are lightweight, inexpensive, and readily available. Conversely, steel spokes made from tube stock are twice as heavy, markedly more expensive, and difficult to obtain.

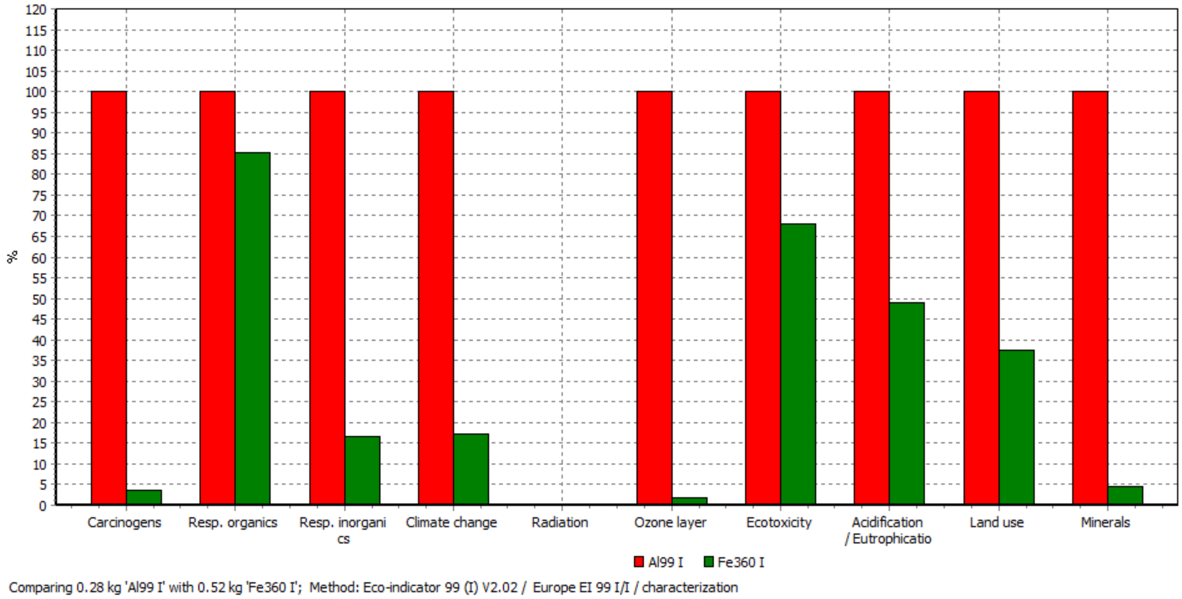


Figure 36: SimaPro 7 characterization results for aluminum and steel spoke options

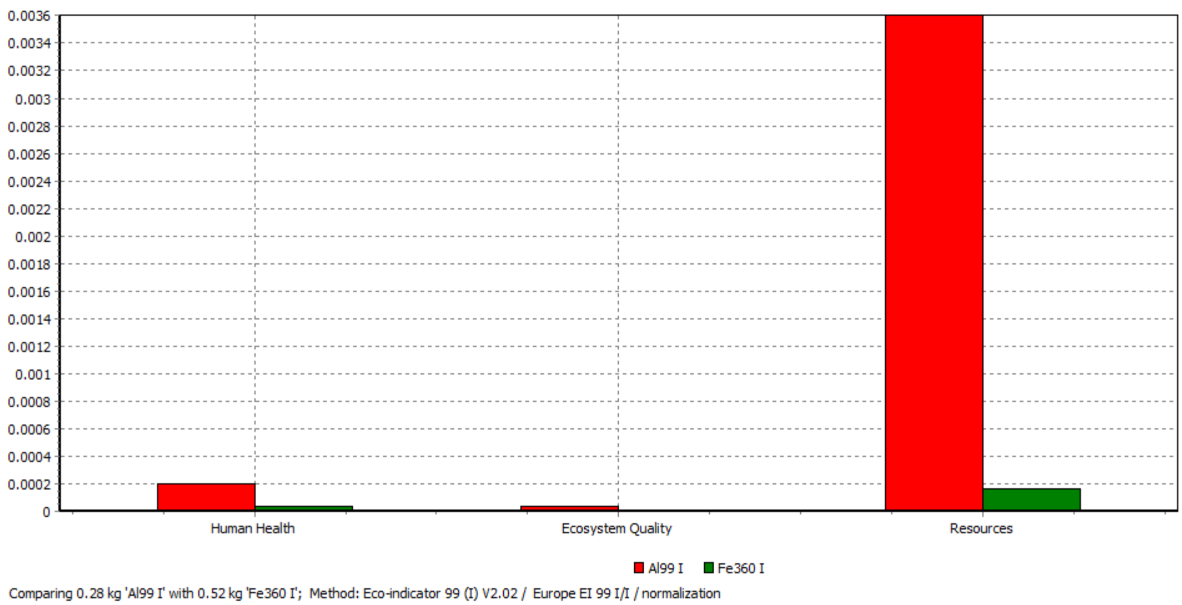


Figure 37: SimaPro 7 condensation and normalization of data in Figure 36

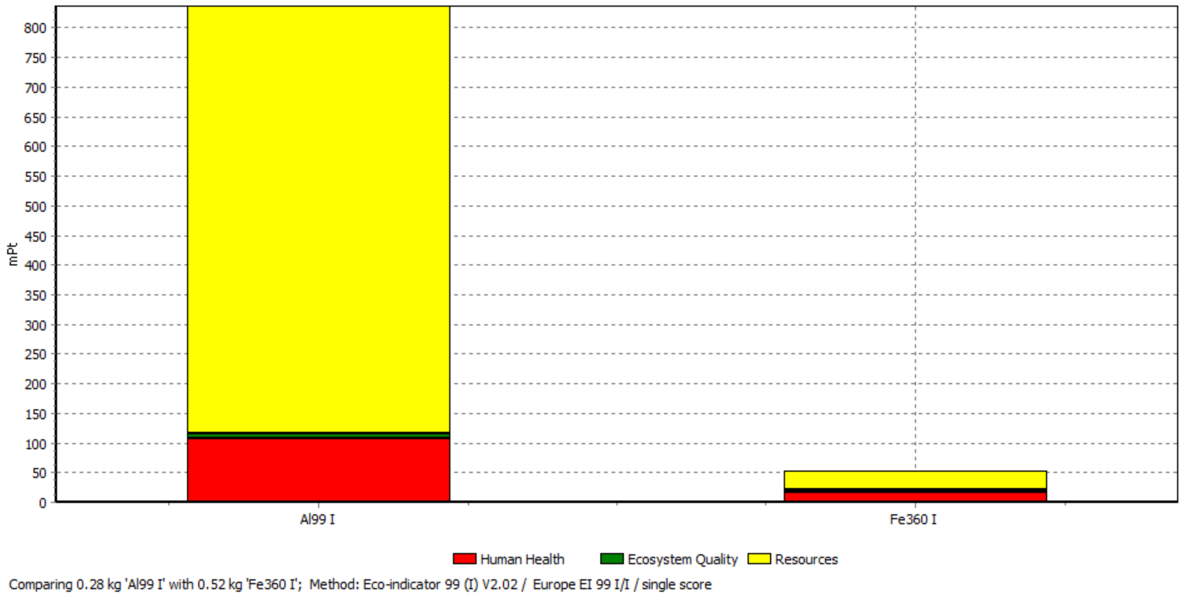


Figure 38: SimaPro 7 single-score condensation of environmental impact data for aluminum and steel spoke options

Gear Environmental Performance

The second component we chose for environmental analysis is our gear train. During the material selection process, we had a choice between phenolic resin gears and steel gears. The steel gears weigh a total of 1.7 kg, while the phenolic gears sum to 0.7 kg. As with the spoke material selection, these options were analyzed in SimaPro 7.

The emissions statistics are shown in Figure 39 on page 75. Except in the category of water emissions, the phenolic option releases markedly fewer emissions.

Environmental and human effects are detailed in Figure 40 on page 75. In all cases, the steel option has greater negative effects by at least a factor of two. These data are condensed into factor categories in Figure 41 on page 76, and further into a single environmental/health score in Figure 42 on 76. In all, the steel option is shown to be around 18 times as harmful as the phenolic option.

Due to the poor strength and wear of phenolic gears, as well as the sheer size that would be required due to these considerations, we opted to use steel gears in our design. Steel gears are also a fraction of the cost of phenolic gears.

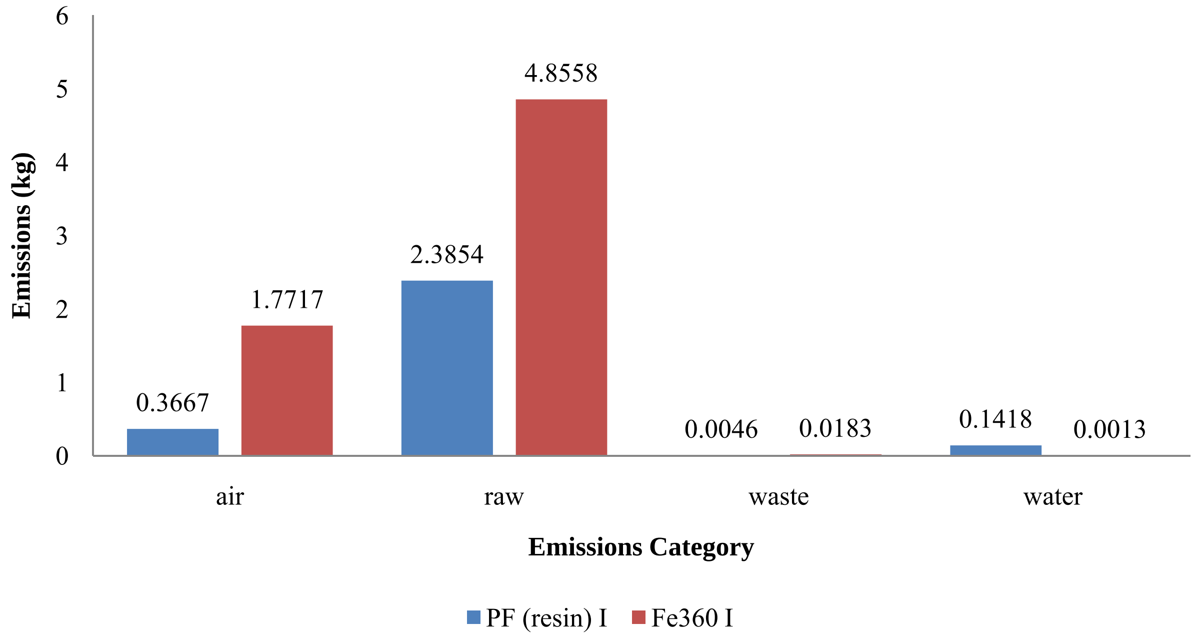


Figure 39: Estimated emissions for phenolic and steel gear options

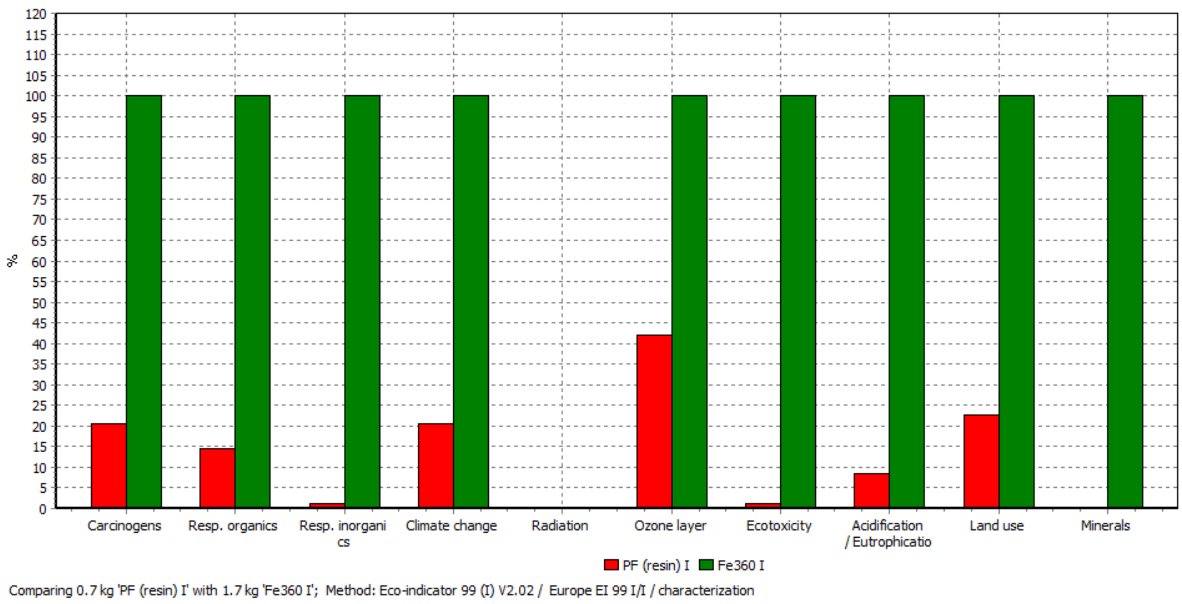


Figure 40: SimaPro 7 characterization of results for phenolic and steel gear options

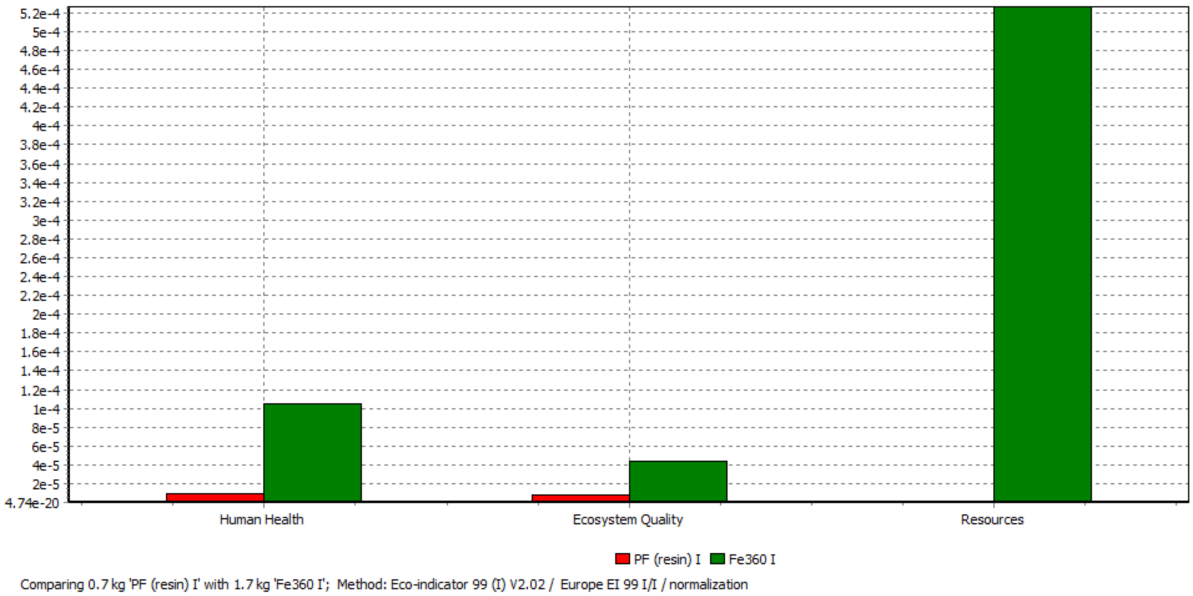


Figure 41: SimaPro 7 condensation and normalization of data in Figure 40

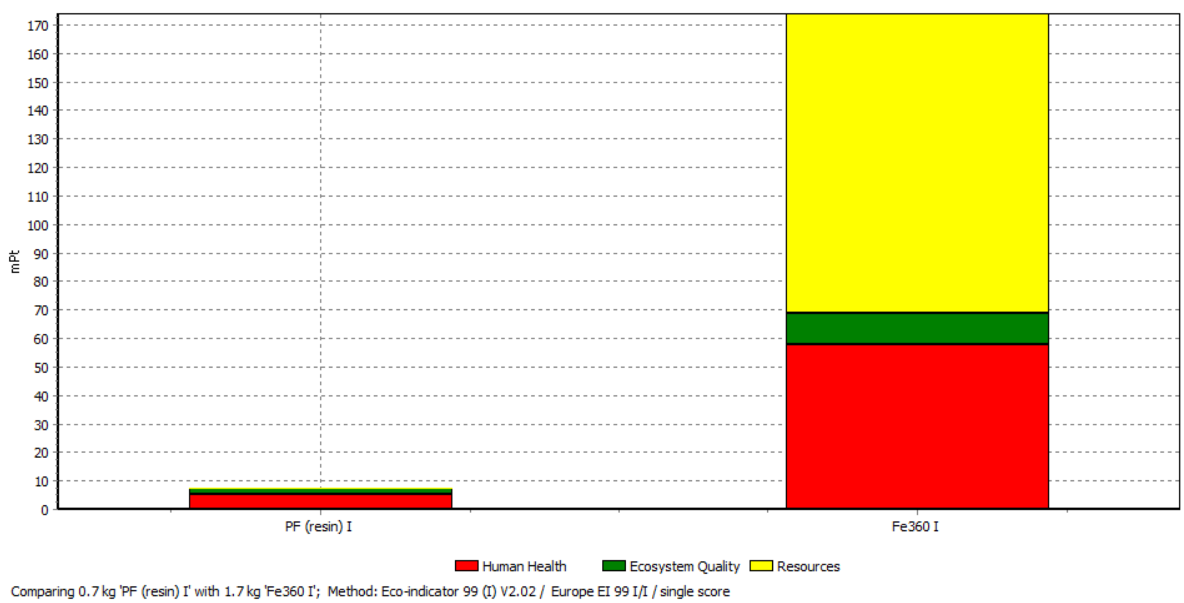


Figure 42: SimaPro 7 single-score condensation of environmental impact data for phenolic and steel gear options

Spoke Manufacturing Process

Based on how new this product would be to the market and the relatively high cost of making such a device, we feel that first year sales would be about 1000 units. This means 6000 spokes need to be manufactured. The aluminum would be initially extruded into 0.25"x0.5" rectangular stock and blanks would be cut to length. To simplify the process the spokes would be bent rather than welded. Because 6061-T651 aluminum is age hardened, 6061-T0 (annealed) would be used for the bending. Following the bending process, the material would be age hardened to increase its strength. CES agrees with this process and says 6061 aluminum can be extruded, cut, drilled, bent, and heat treated. These all fall within the economic batch size as they are simple processes that require little capital equipment.

Gear Manufacturing Process

Our HRBS utilizes four different gear styles. Based on the part quantity needed to produce 1000 gears, it is not economical for us to purchase a machine to make these gears in-house. It would be more appropriate to source the gears to a company with the capabilities to cut or hob gears. The gears used for this system are not customized to the point where the cost would be outrageous. Any post processing would be completed using a lathe or mill, but ideally the gears would be cut from blanks of the correct size and shape. CES agrees that steel can be cut into gears, turned, and milled.

Appendix F – DesignSafe Report

designsafe Report

Application:	HRBS Safety Analysis	Analyst Name(s):	Matthew Williams, Victor Singh, Andrew Kneifel, Bryan D'Souza
Description:		Company:	
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:			

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

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
operator normal operation	mechanical : unexpected start failure of 2-way valve or electromechanical clutch	Serious Frequent Unlikely	High				
operator normal operation	mechanical : break up during operation overloading of system components	Catastrophic Frequent Negligible	Moderate				
operator normal operation	mechanical : machine instability failure of rider to maintain control	Slight Frequent Unlikely	Moderate				
operator normal operation	mechanical : impact failure of rider to maintain control, resulting in impact	Serious Frequent Possible	High				
operator normal operation	electrical / electronic : lack of grounding (earthing or neutral) insulated rubber tires	Slight Occasional Negligible	Low				
operator normal operation	electrical / electronic : insulation failure wire insulation fraying due to vibration/contact with sharp edges	Slight Occasional Negligible	Low				
operator normal operation	electrical / electronic : improper wiring miswired system resulting in improper function	Slight Occasional Negligible	Low				

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
operator normal operation	electrical / electronic : unexpected start up / motion shorting of components resulting in activation of 2-way valve or clutch	Serious Frequent Unlikely	High				
operator normal operation	electrical / electronic : overvoltage /overcurrent shorting of components resulting in shorted battery	Serious Occasional Unlikely	Moderate				
operator normal operation	electrical / electronic : power supply interruption disconnection of electrical components resulting in deactivation of system	Serious Occasional Possible	High				
operator normal operation	ergonomics / human factors : posture poor rider posture due to improperly adjusted seat/handlebars	Slight Frequent Negligible	Low				
operator normal operation	ergonomics / human factors : human errors / behaviors improper use of system due to insufficient training	Serious Frequent Possible	High				
operator normal operation	fluid / pressure : hydraulics rupture failure of hydraulic fittings or tubing	Catastrophic Frequent Unlikely	High				
operator normal operation	fluid / pressure : explosion / implosion explosive failure of hydraulic accumulator	Catastrophic Frequent Negligible	Moderate				
operator normal operation	fluid / pressure : fluid leakage / ejection minor failure of hydraulic fittings or tubing at interfaces	Serious Frequent Unlikely	High				
operator lubrication	mechanical : pinch point moving/rotating gears	Slight Occasional Unlikely	Moderate				
operator lubrication	mechanical : unexpected start failure of 2-way valve or electromechanical clutch	Serious Occasional Unlikely	Moderate				

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
operator lubrication	fluid / pressure : hydraulics rupture failure of hydraulic fittings or tubing	Catastrophic Occasional Unlikely	High				
operator lubrication	fluid / pressure : explosion / implosion explosive faillure of hydraulic accumulator	Catastrophic Occasional Negligible	Moderate				
operator lubrication	fluid / pressure : fluid leakage / ejection minor failure of hydraulic fittings or tubing at interfaces	Serious Occasional Possible	High				
maintenance technician lubrication	mechanical : pinch point moving/rotating gears	Slight Remote Unlikely	Low				
maintenance technician lubrication	mechanical : unexpected start failure of 2-way valve or electromechanical clutch	Serious Remote Unlikely	Moderate				
maintenance technician lubrication	fluid / pressure : hydraulics rupture failure of hydraulic fittings or tubing	Catastrophic Remote Unlikely	Moderate				
maintenance technician lubrication	fluid / pressure : explosion / implosion explosive faillure of hydraulic accumulator	Catastrophic Remote Negligible	Moderate				
maintenance technician lubrication	fluid / pressure : fluid leakage / ejection minor failure of hydraulic fittings or tubing at interfaces	Serious Remote Unlikely	Moderate				
maintenance technician periodic maintenance	mechanical : pinch point moving/rotating gears	Slight Remote Unlikely	Low				
maintenance technician periodic maintenance	electrical / electronic : energized equipment / live parts system potential as high as 30V	Slight Remote Possible	Moderate				

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
maintenance technician periodic maintenance	electrical / electronic : lack of grounding (earthing or neutral) insulated rubber tires	Slight Remote Unlikely	Low				
maintenance technician periodic maintenance	electrical / electronic : insulation failure wire insulation fraying due to vibration/contact with sharp edges	Slight Remote Unlikely	Low				
maintenance technician periodic maintenance	electrical / electronic : improper wiring miswired system resulting in improper function	Slight Remote Unlikely	Low				
maintenance technician periodic maintenance	electrical / electronic : unexpected start up / motion shorting of components resulting in activation of 2-way valve or clutch	Serious Remote Unlikely	Moderate				
maintenance technician periodic maintenance	electrical / electronic : overvoltage /overcurrent shorting of components resulting in shorted battery	Serious Remote Unlikely	Moderate				
maintenance technician periodic maintenance	ergonomics / human factors : excessive force / exertion overtorqued / rusted fasteners	Slight Remote Unlikely	Low				
maintenance technician periodic maintenance	ergonomics / human factors : deviations from safe work practices improper handling of high-pressure components or improper lifting of heavy components	Slight Remote Possible	Moderate				
maintenance technician periodic maintenance	fluid / pressure : hydraulics rupture failure of hydraulic fittings or tubing	Catastrophic Remote Unlikely	Moderate				
maintenance technician periodic maintenance	fluid / pressure : explosion / implosion explosive failure of hydraulic accumulator	Catastrophic Remote Negligible	Moderate				

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
maintenance technician periodic maintenance	fluid / pressure : fluid leakage / ejection minor failure of hydraulic fittings or tubing at interfaces	Serious Remote Possible	Moderate				
maintenance technician parts replacement	mechanical : pinch point moving/rotating gears	Slight Remote Possible	Moderate				
maintenance technician parts replacement	mechanical : unexpected start failure of 2-way valve or electromechanical clutch	Serious Remote Unlikely	Moderate				
maintenance technician parts replacement	electrical / electronic : energized equipment / live parts system potential as high as 30V	Slight Remote Unlikely	Low				
maintenance technician parts replacement	electrical / electronic : lack of grounding (earthing or neutral) insulated rubber tires	Slight Remote Unlikely	Low				
maintenance technician parts replacement	electrical / electronic : insulation failure wire insulation fraying due to vibration/contact with sharp edges	Slight Remote Unlikely	Low				
maintenance technician parts replacement	electrical / electronic : improper wiring miswired system resulting in improper function	Slight Remote Unlikely	Low				
maintenance technician parts replacement	electrical / electronic : unexpected start up / motion shorting of components resulting in activation of 2-way valve or clutch	Serious Remote Unlikely	Moderate				
maintenance technician parts replacement	electrical / electronic : overvoltage /overcurrent shorting of components resulting in shorted battery	Slight Remote Possible	Moderate				
maintenance technician parts replacement	ergonomics / human factors : excessive force / exertion overtorqued / rusted fasteners	Slight Remote Possible	Moderate				

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
maintenance technician parts replacement	fluid / pressure : hydraulics rupture failure of hydraulic fittings or tubing	Catastrophic Remote Unlikely	Moderate				
maintenance technician parts replacement	fluid / pressure : explosion / implosion explosive failure of hydraulic accumulator	Catastrophic Remote Negligible	Moderate				
maintenance technician parts replacement	fluid / pressure : fluid leakage / ejection minor failure of hydraulic fittings or tubing at interfaces	Serious Remote Probable	High				

Appendix G – FMEA

Product Name: HRBS x System __ Subsystem Name: ______ Component	Development Team: Team 24	Page No. 1 of 1 FMEA Number ________ Date: 8-Apr-2009

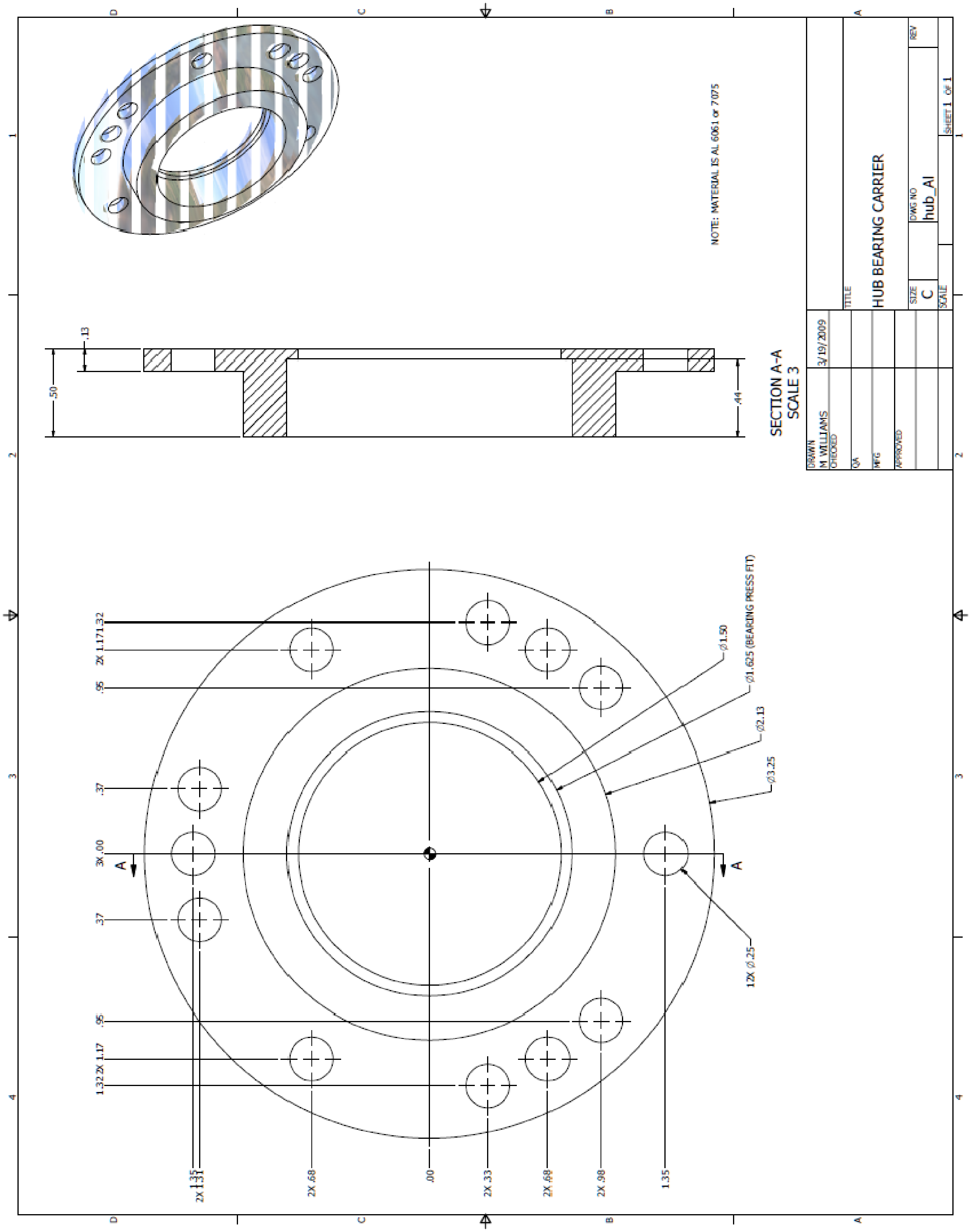
Hydraulic

Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Severity (S)	Potential Causes / Mechanisms of Failure	Occurrence (O)	Current Design Controls / Tests	Detection (D)	Recommended Actions	RPN
Low pressure accumulator - stores excess fluid	Leaks	System will eventually become inoperable	9	Fluid exits through vent; plastic bottle fails	1	Deflection plate; material interaction research	2		18
High pressure accumulator - stores pressurized fluid	Leaks	System will eventually become inoperable	8	Debris; excess pressure	1	Filter; pressure relief valve	2		16
	Bursts	Sudden release of energy	10	Poor construction	1	None	1		10
Fittings - connects hose to components	Leaks	System will eventually become inoperable	7	Poor assembly; excess pressure	2	Test assembly before use; pressure relief valve	2		28
	Bursts	Sudden release of energy	7	Excess pressure; defective	2	Pressure relief valve	1		14
Hose - channels fluid	Leaks	System will eventually become inoperable	7	Excess pressure; defective	2	Pressure relief valve	2		28
	Bursts	Sudden release of energy	7	Excess pressure; defective	2	Pressure relief valve	1		14
Pump - pressurizes fluid by rotation	Seizes	Wheel immediately stops; gear failure	7	Buildup of debris	2	Filter	2		28
Motor - generates rotation from pressure gradient	Seizes	Loss of HRBS	7	Buildup of debris	2	Filter	2		28
Filter - minimizes particulates in fluid	Resists all flow	Loss of HRBS	7	Buildup of debris	2	Periodic replacing of fluid	2		28
Check valve - prevents reverse flow through pump	Leaks	Less effective HRBS	5	Buildup of debris	2	Filter	3		30
	Wedges closed	Loss of HRBS; high pressures in system	7	Buildup of debris	1	Filter; relief valve	2		14
	Stuck open	Loss of HRBS; potential reverse operation	9	Buildup of debris	1	Filter	2		18
2-way valve - opens or closes pathway to motor	Fails closed	Pressure in HPA cannot be released	8	Buildup of debris	2	Filter; pressure relief valve	2		32
	Fails open	HPA cannot be pressurized	7	Buildup of debris	2	Filter	1		14
Relief valve - releases excess pressure	Fails to trigger	Excess pressure in system	10	Buildup of debris	2	Filter; pressure gauge	1		20
	Fails open	HPA cannot be pressurized	7	Buildup of debris	2	Filter	1		14

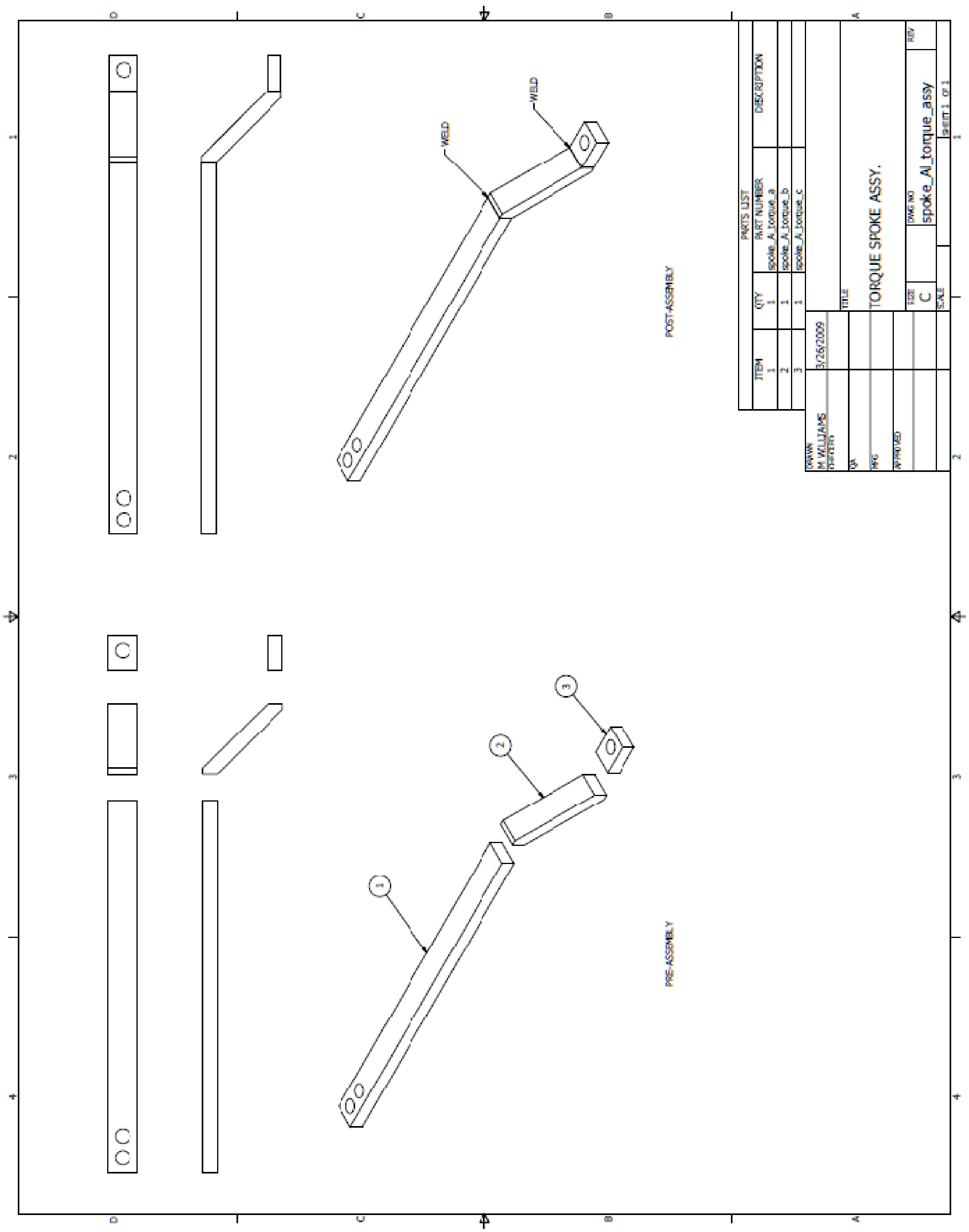
	Part # and Functions	Potential Failure Mode	Potential Effect(s) of Failure	Severity (S)	Potential Causes / Mechanisms of Failure	Occurrence (O)	Current Design Controls / Tests	Detection (D)	Recommended Actions	RPN
Electric	Wire - transmits electricity	Open circuits	Cannot engage HRBS features	10	Corrosion, frayed or broken wire	1	Preliminary electrical testing	2		20
	Toggle switch - switches power to electric clutch	Short circuits	Pump always engaged	10	Corrosion, frayed or broken wire	1	Preliminary electrical testing	2		20
		Open circuits	Cannot engage pump	10	Corroded terminals	1	Preliminary electrical testing	2		20
	Keylock - master on / off switch	Open circuits	Cannot engage HRBS	10	Corroded terminals	1	Preliminary electrical testing	2		20
	Pushbuttons - switches power to 2-way valve	Short circuits	Motor always engaged	9	Poor connection isolation	1	Preliminary electrical testing	2		18
		Open circuits	Cannot engage HRBS	10	Corroded terminals	1	Preliminary electrical testing	2		20
Structural	Spokes - support weight of bicycle and rider and transmit torques	Buckle or shear	Front wheel collapses	10	Excess torque	2	User weight limits; pressure relief valve	1		20
	Superbracket - supports all nonmoving components	Bend, buckle or shear	Internal components compromised	10	High vibrations, high loads	2	Preliminary structural tests	1		20
	Outer plastic hub - shields internal components from the elements	Cracks or shatters	Debris enters internal hub area	4	Impact from environment fragments	2	None	1		8
Transmission	Gears - transmit rotational torque and speed	Tooth shear	High velocity fragments released in internal hub area; loss of functionality	8	Excess torque or excess rotational speed	2	Lewis Equation	2		32
	Clutch - selectively engages pump gear	Fails to engage	Loss of braking functionality	7	Debris	1	Hub covers	2		14
		Fails to disengage	Constant braking	6	Debris	1	Hub covers	2		12
	One-way bearing - allows motor to rotate freely in one direction but not the other	Seizes	Constant applied acceleration torque	5	Buildup of debris	1	Hub covers	2		10
		Allows rotation in both directions	Loss of accelerating functionality	7	Excess torque	1	None	2		14

Appendix H – CAD Drawings

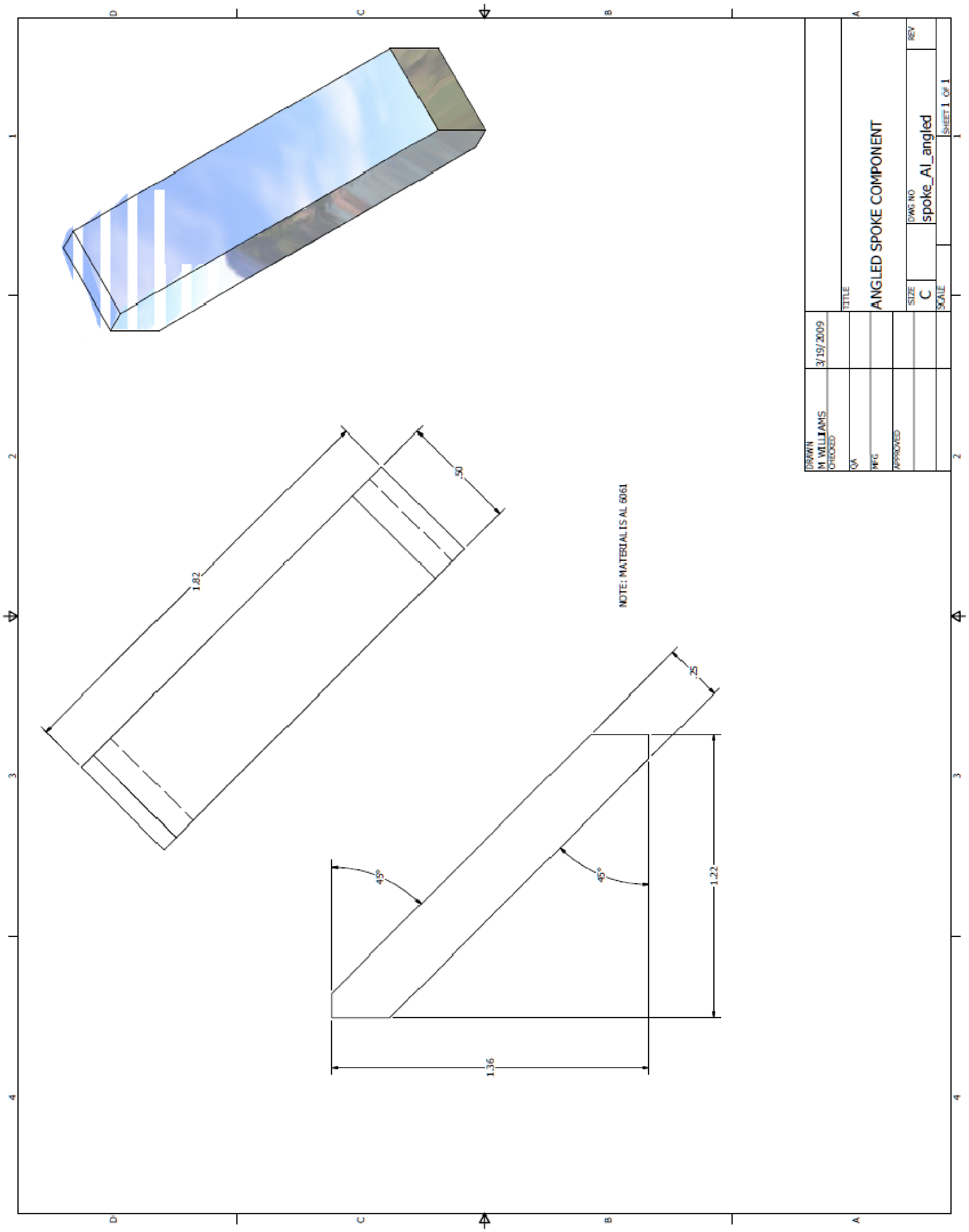
Hub bearing carrier



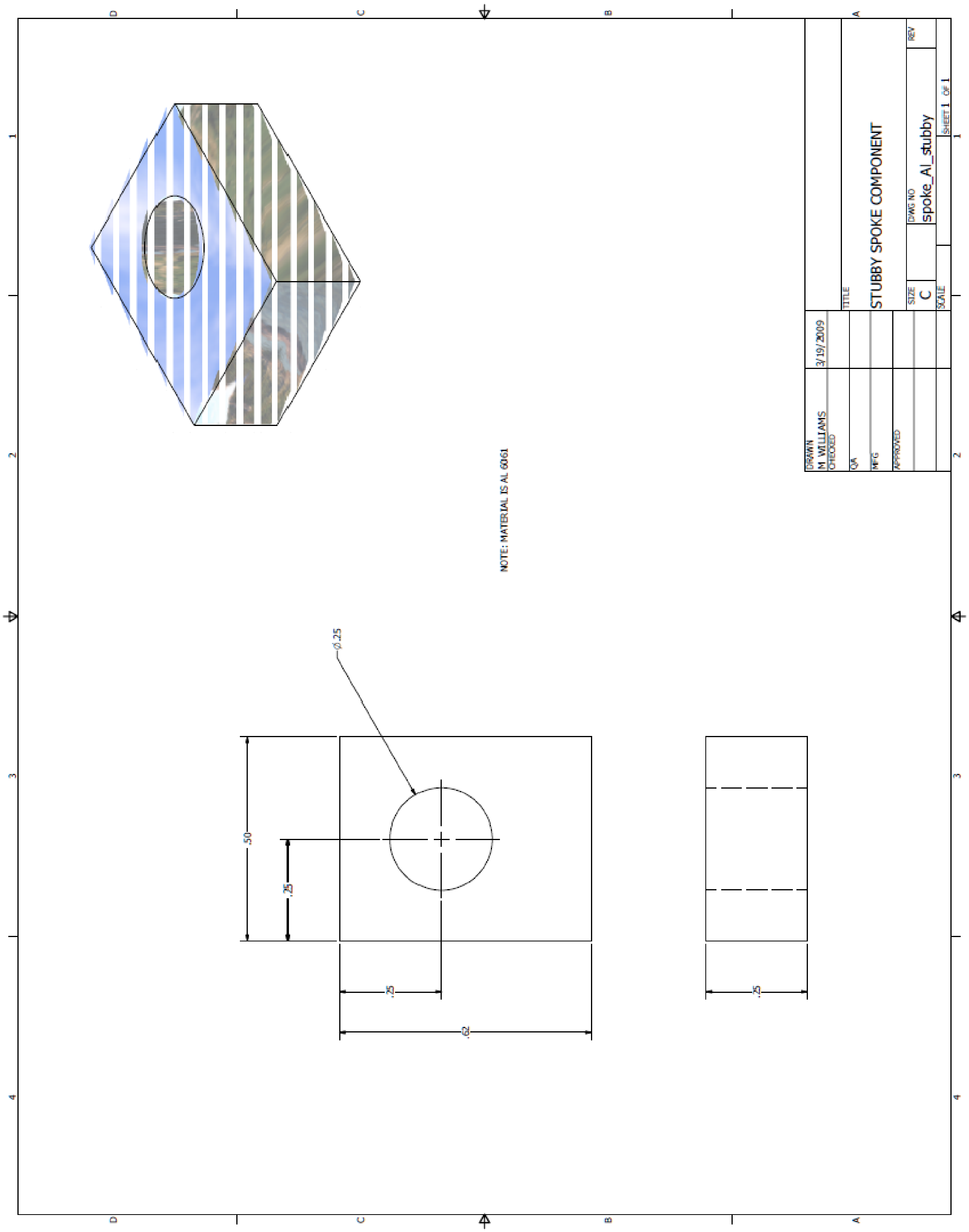
Hub spoke assembly



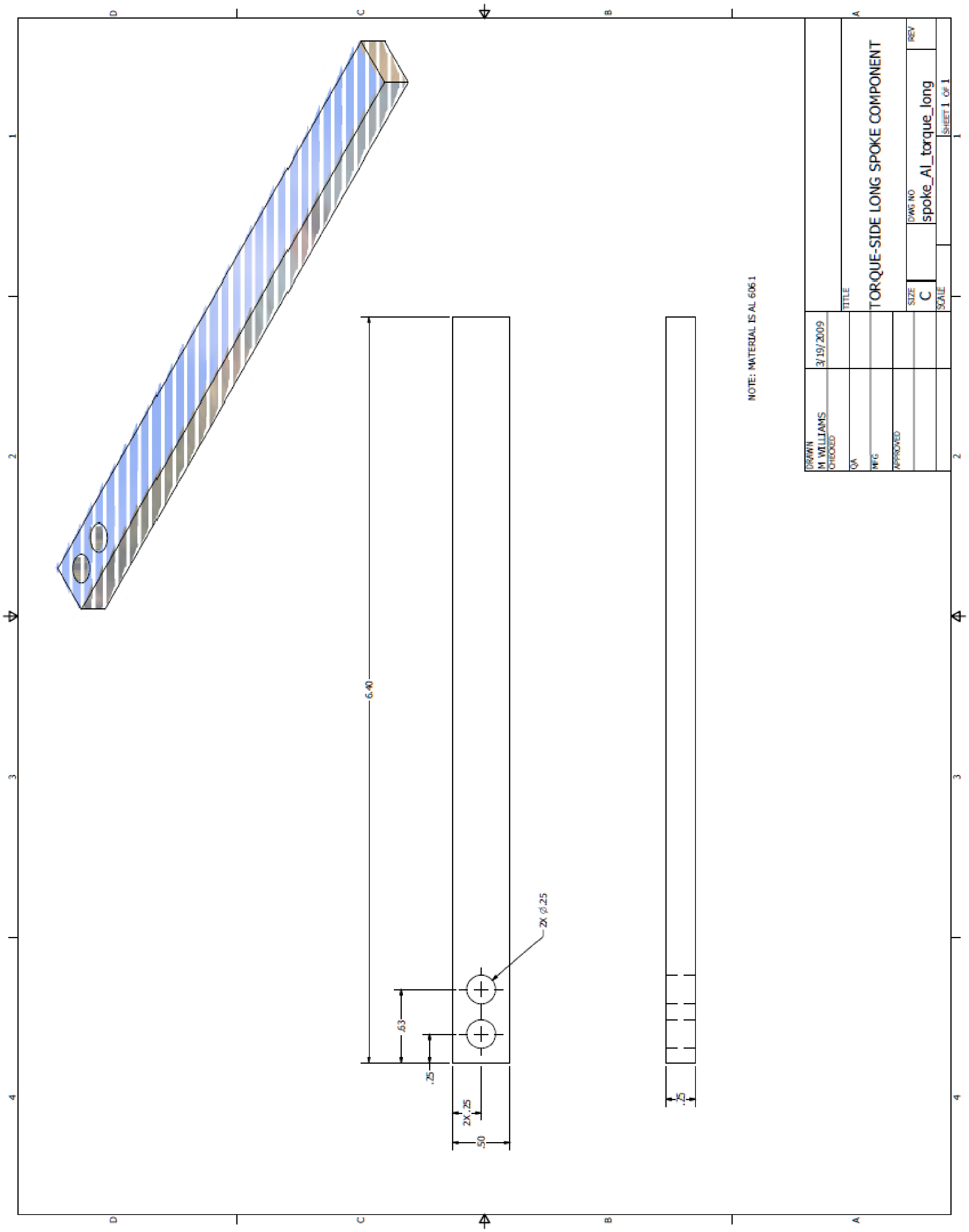
Angled spoke component



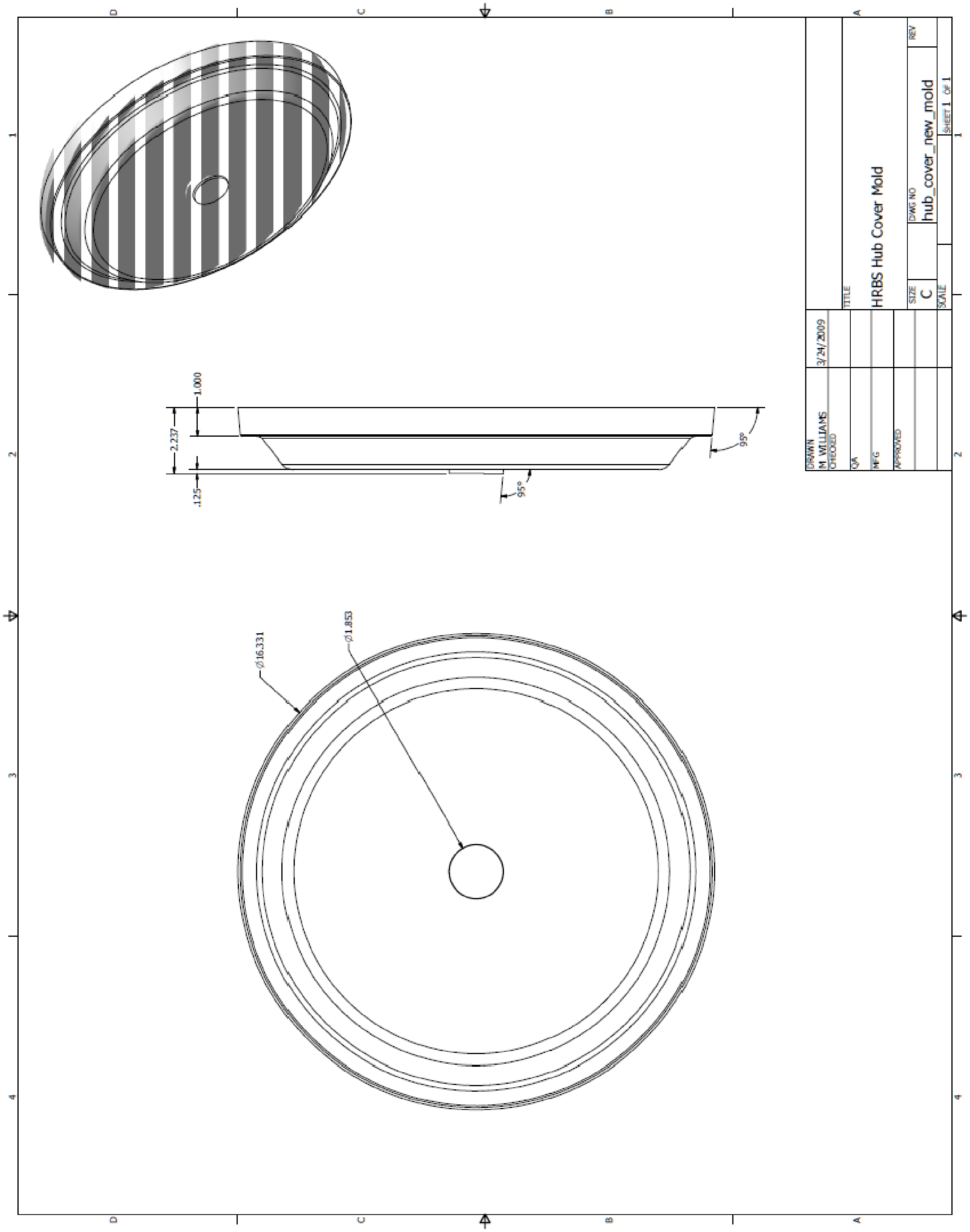
Stubby spoke component



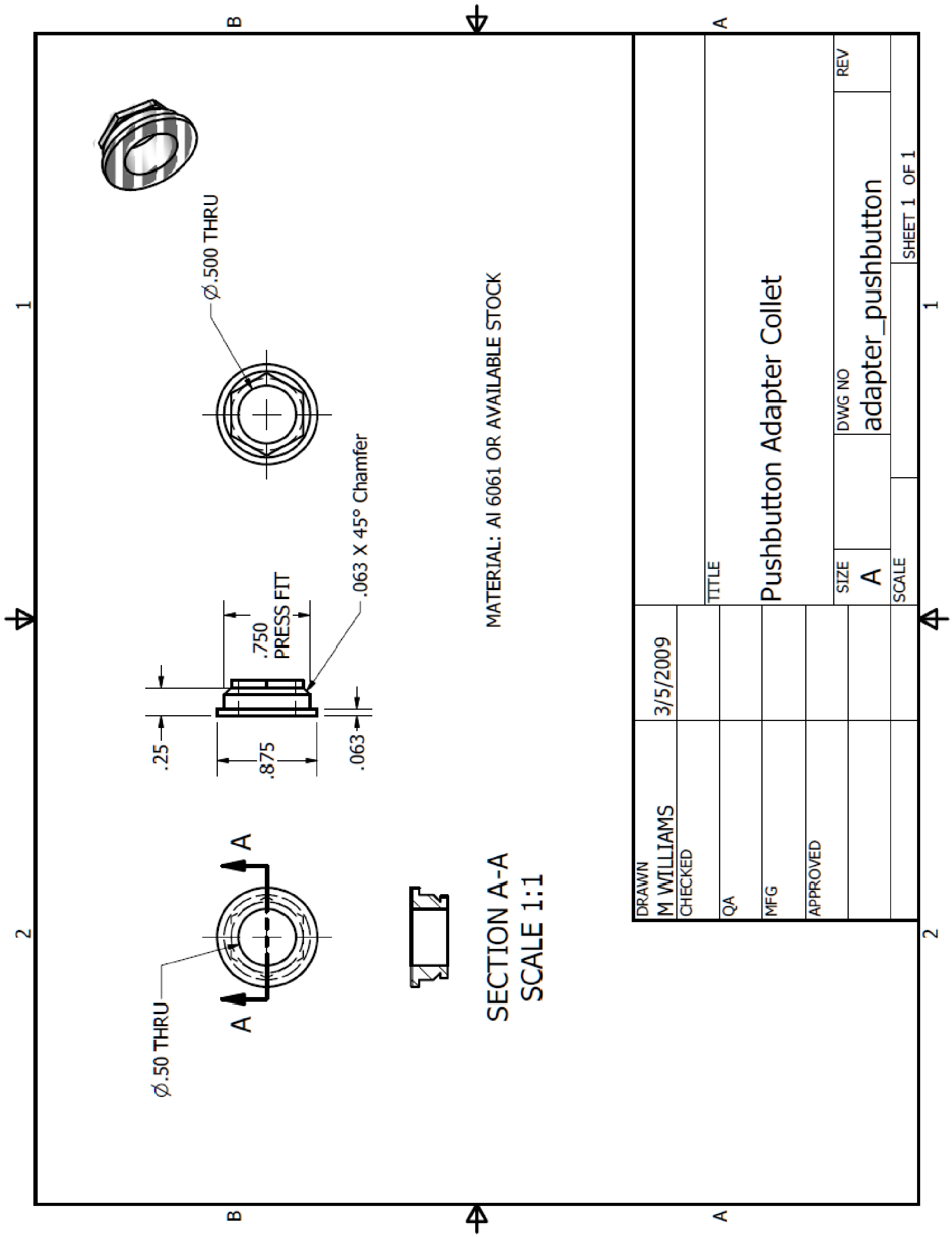
Long spoke component



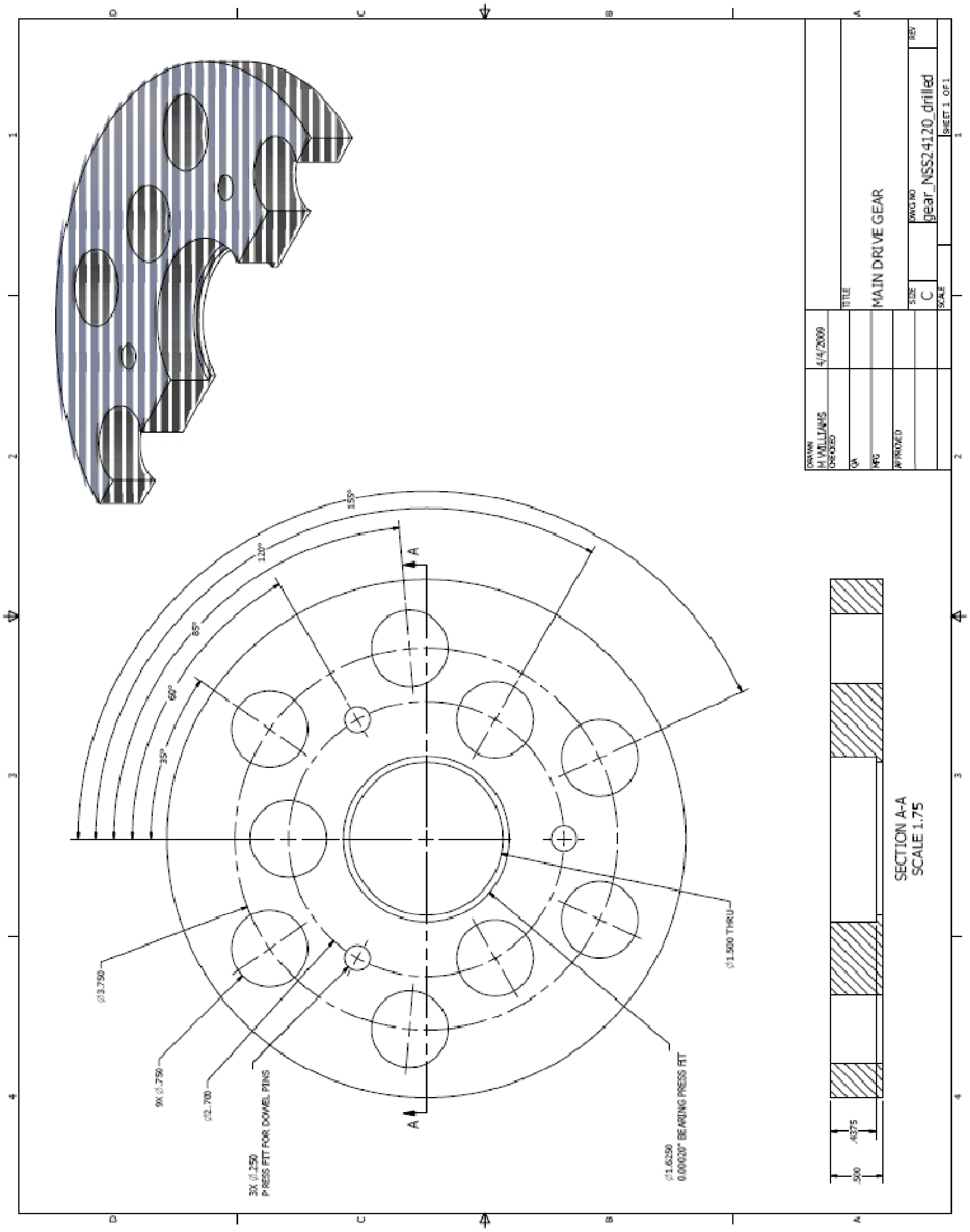
Hub mold – rough dimensioned drawing



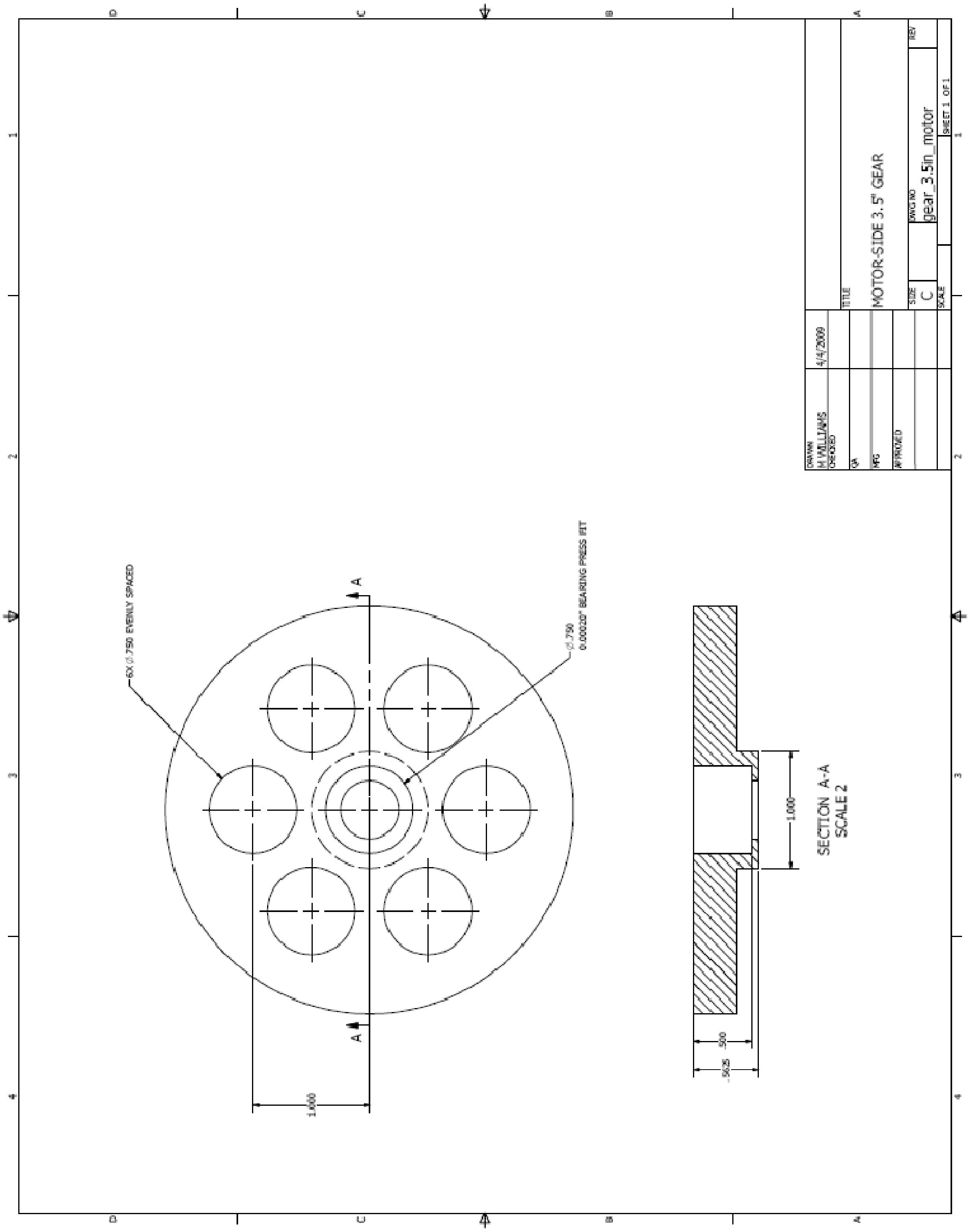
Pushbutton adapter collet



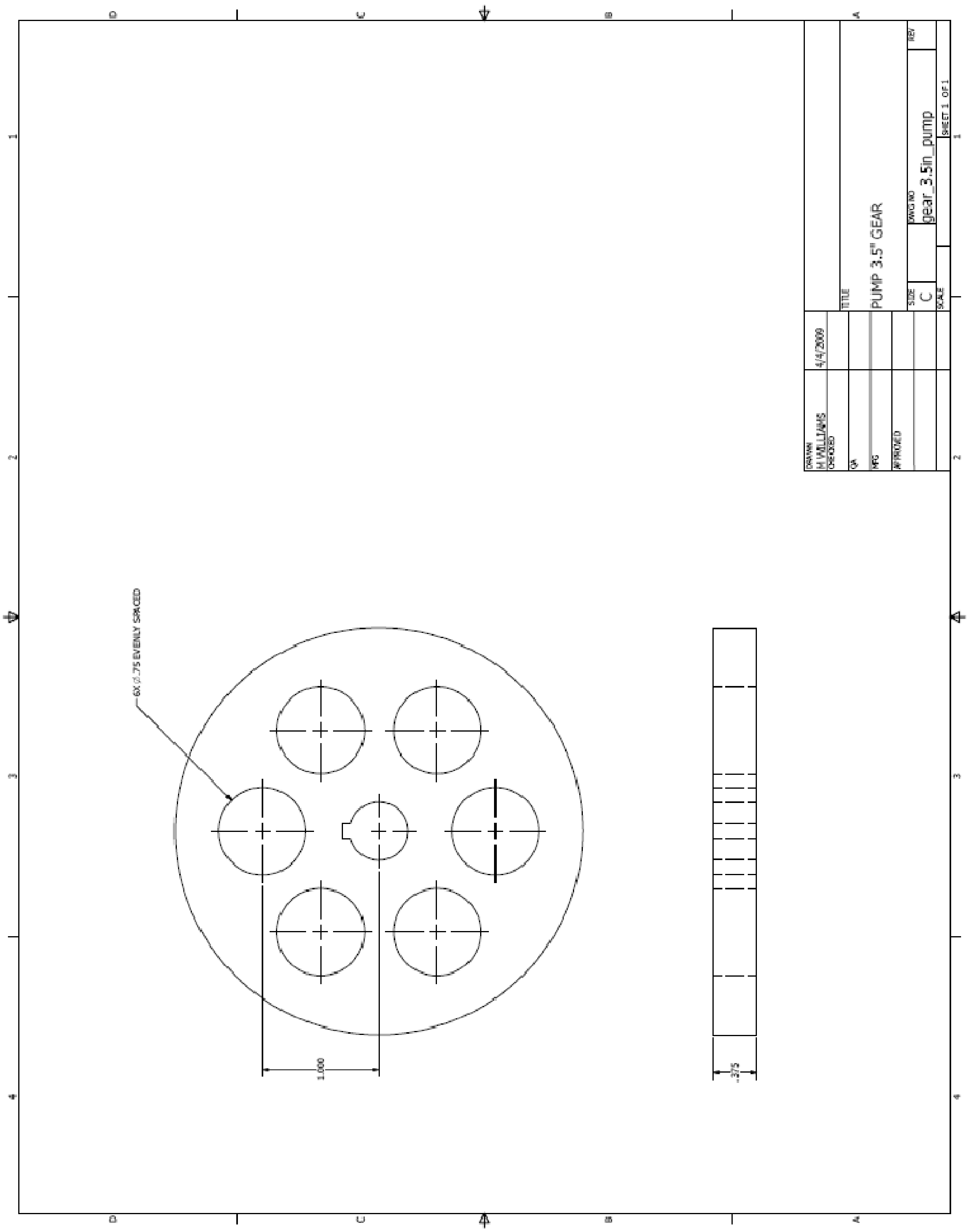
Main Gear – 5” – 16DP



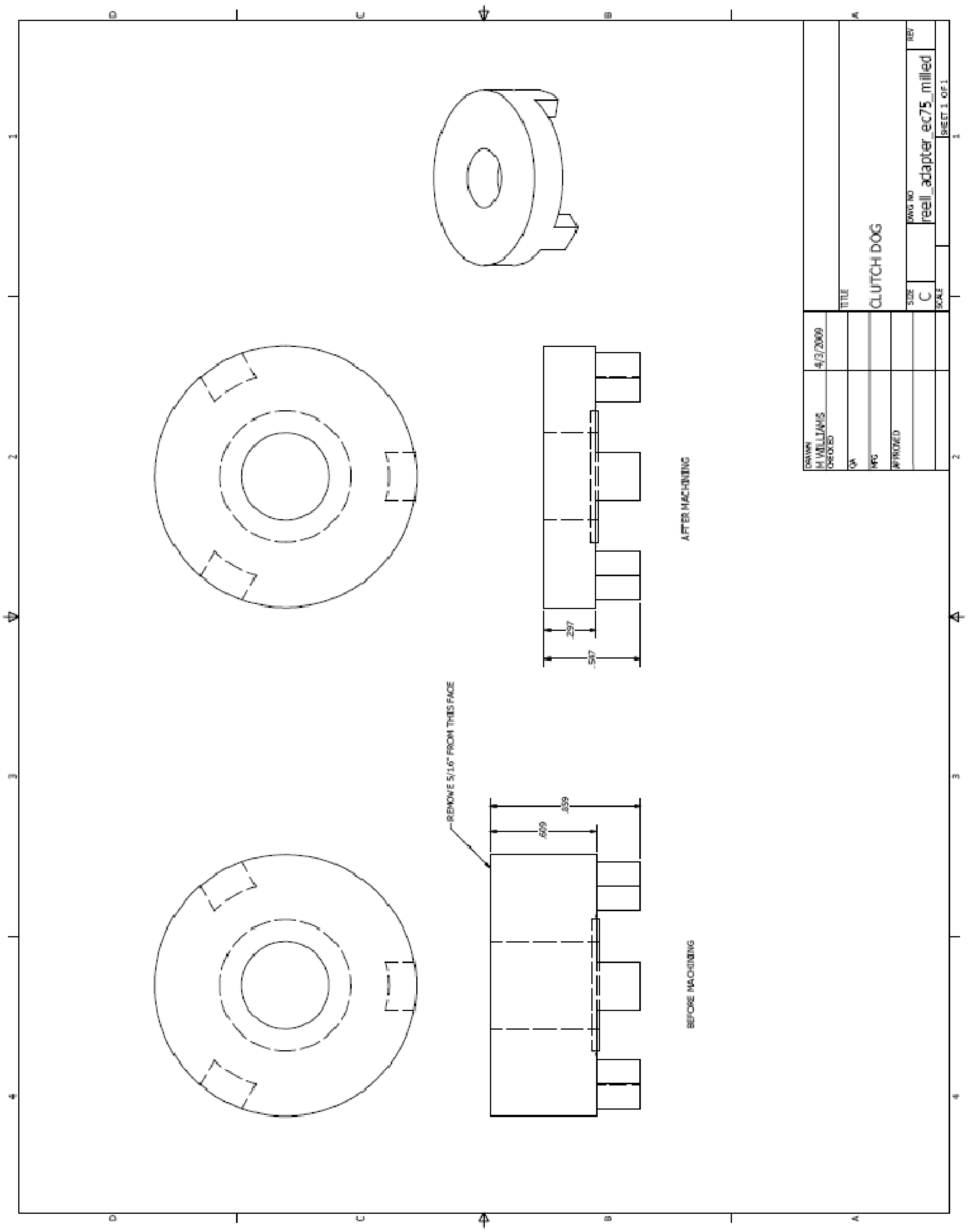
3.5” Gear – Motor side



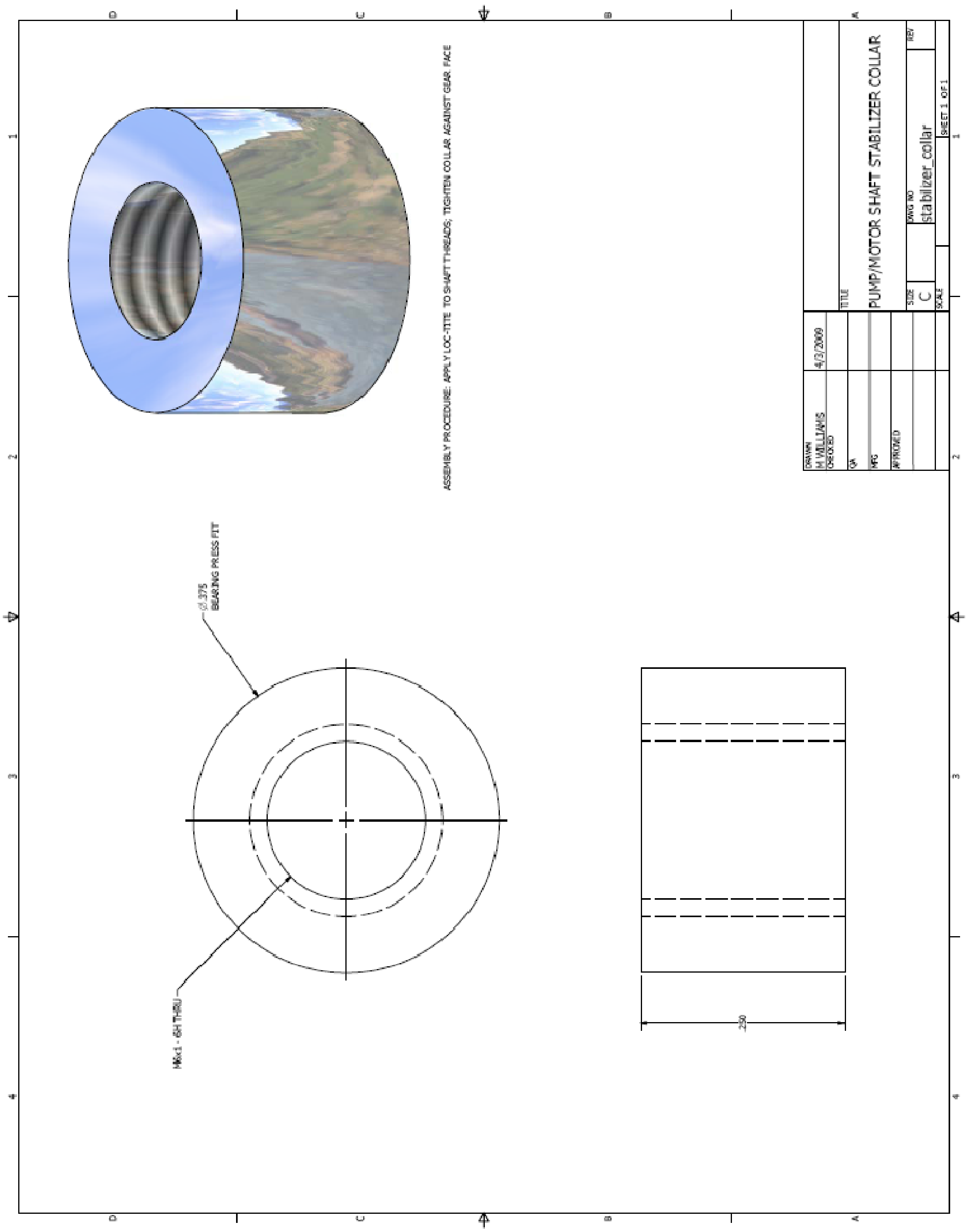
3.5” Gear – Pump side



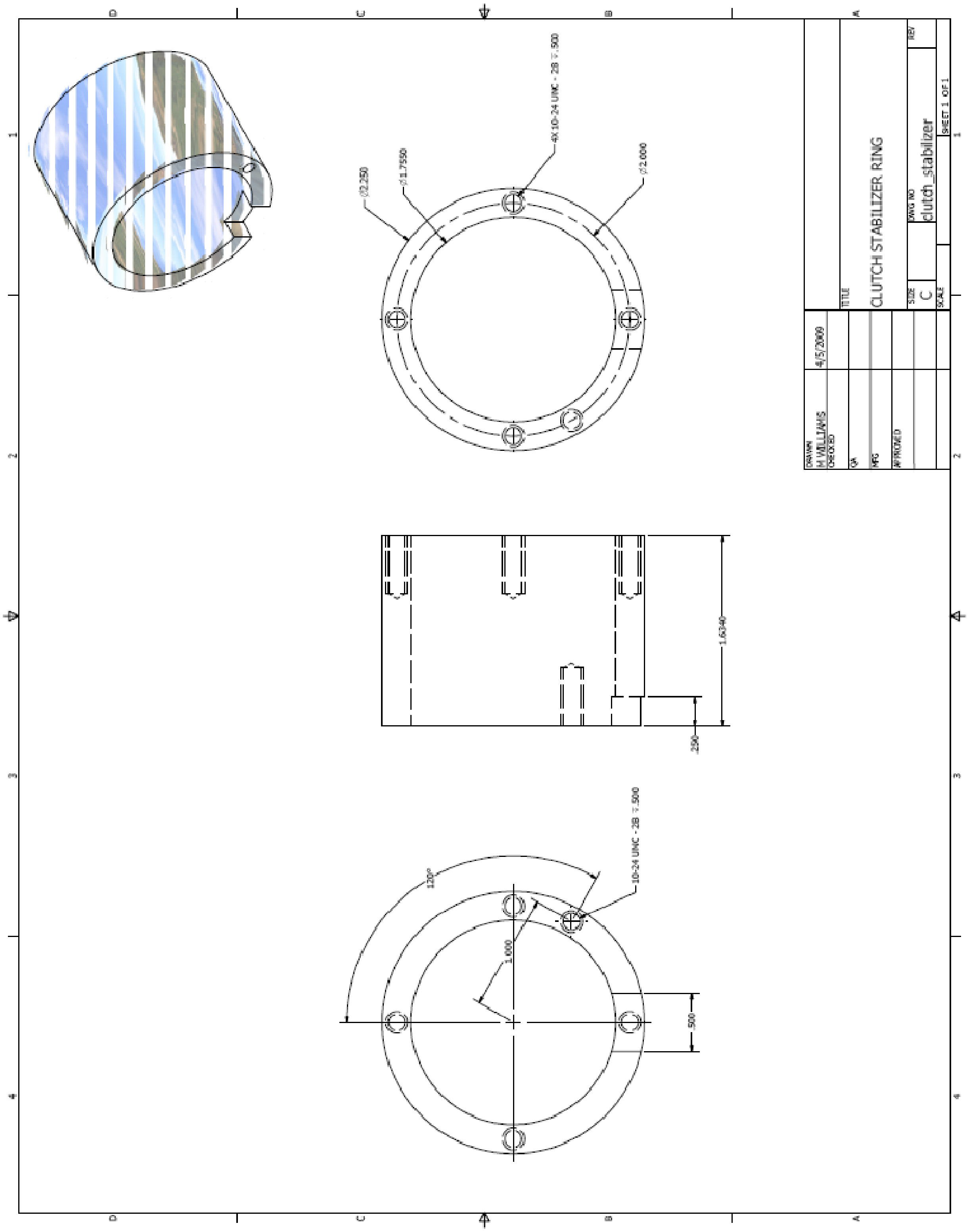
3-Dog Clutch Adapter – Modification of stock part



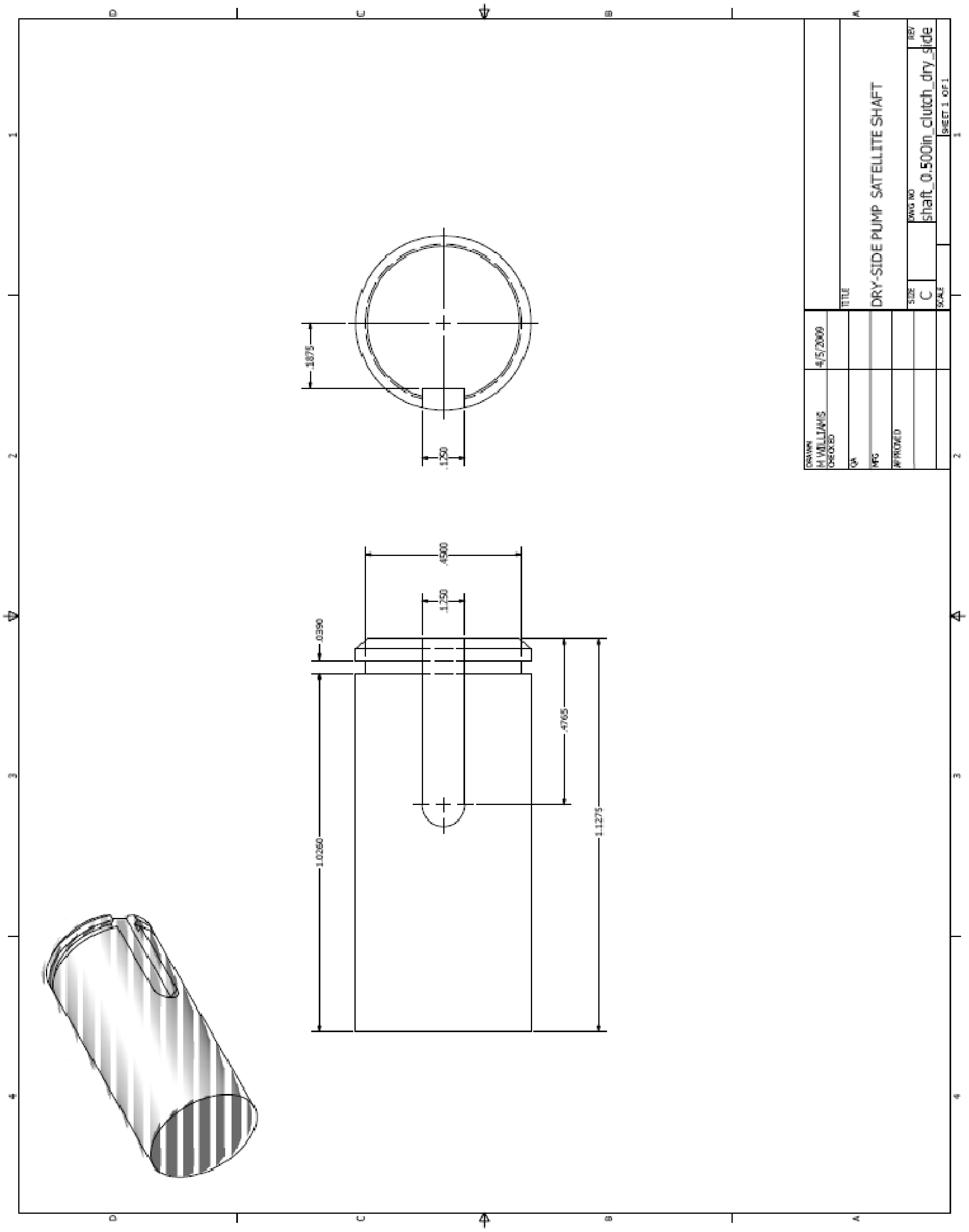
Motor/Pump stabilizer collar



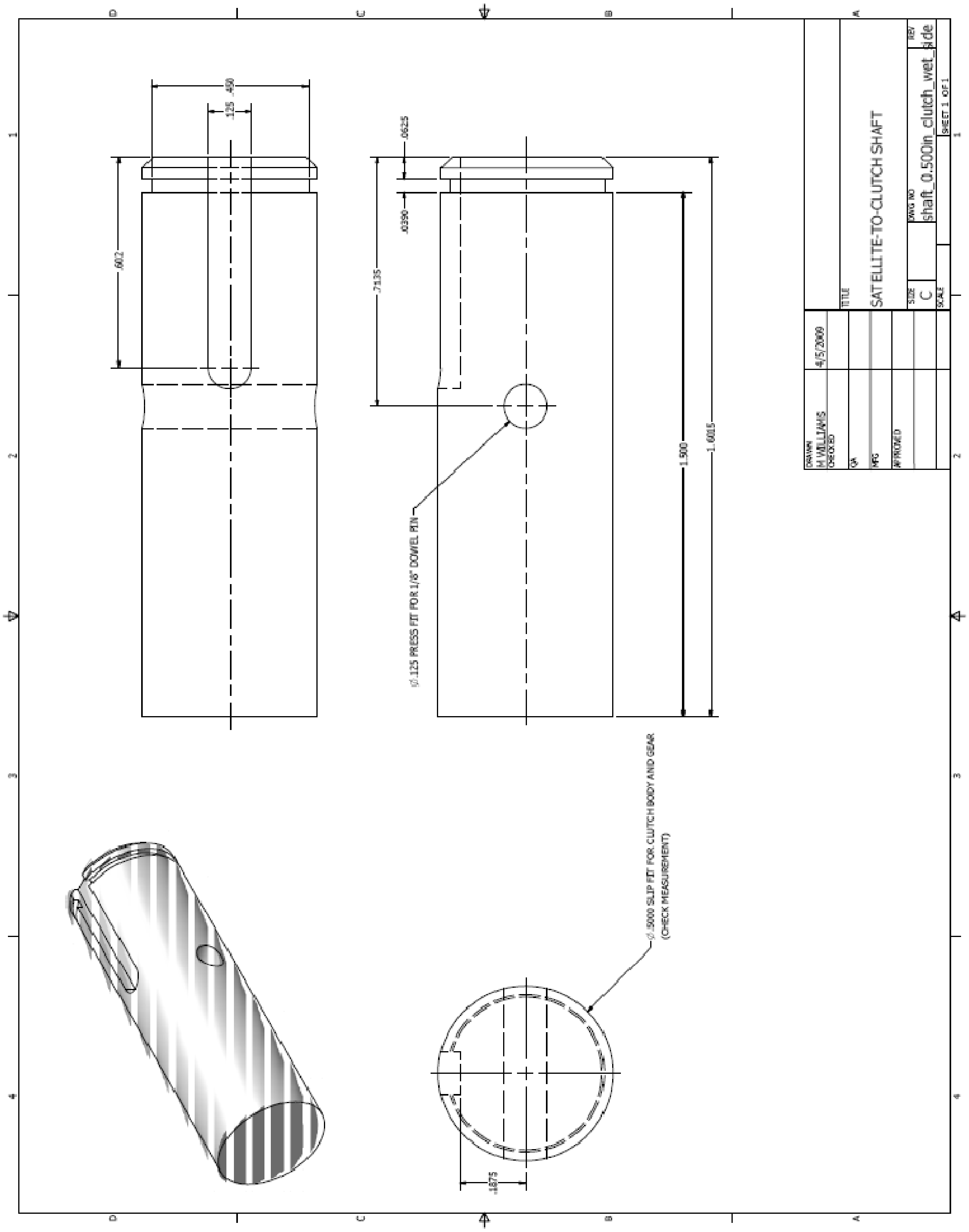
Clutch Stabilizer Ring



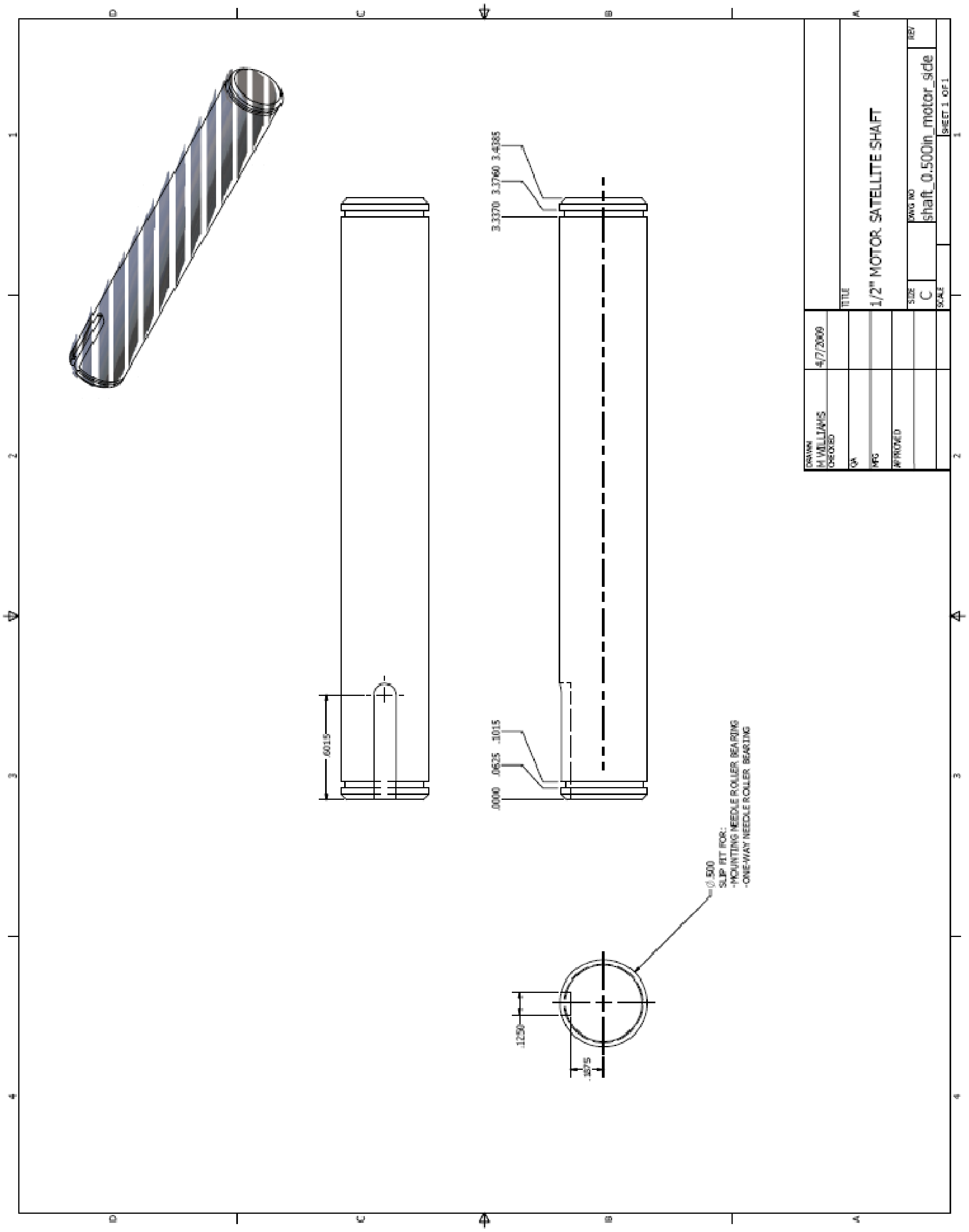
Pump dry-side satellite gear shaft



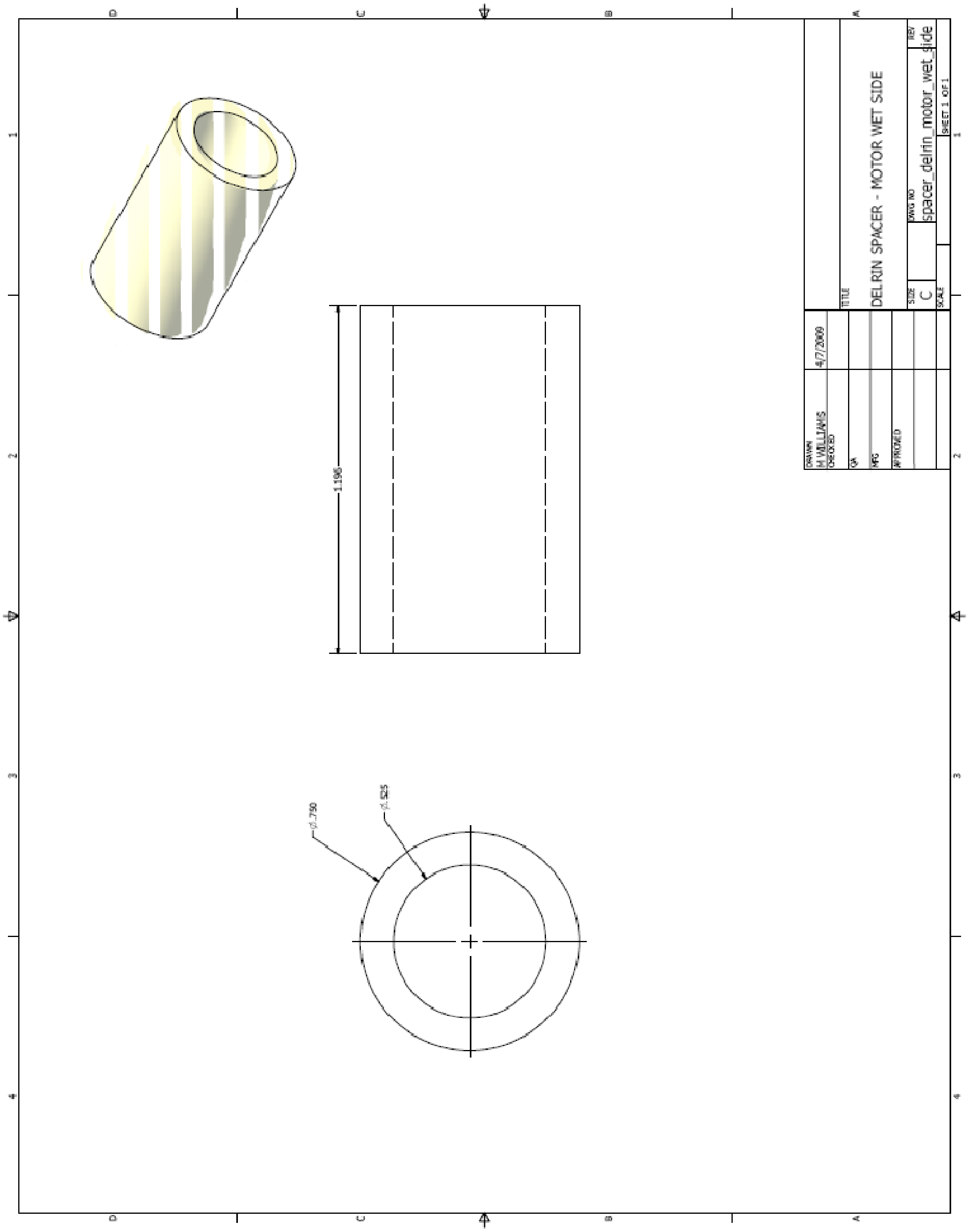
Pump wet-side satellite gear to clutch shaft



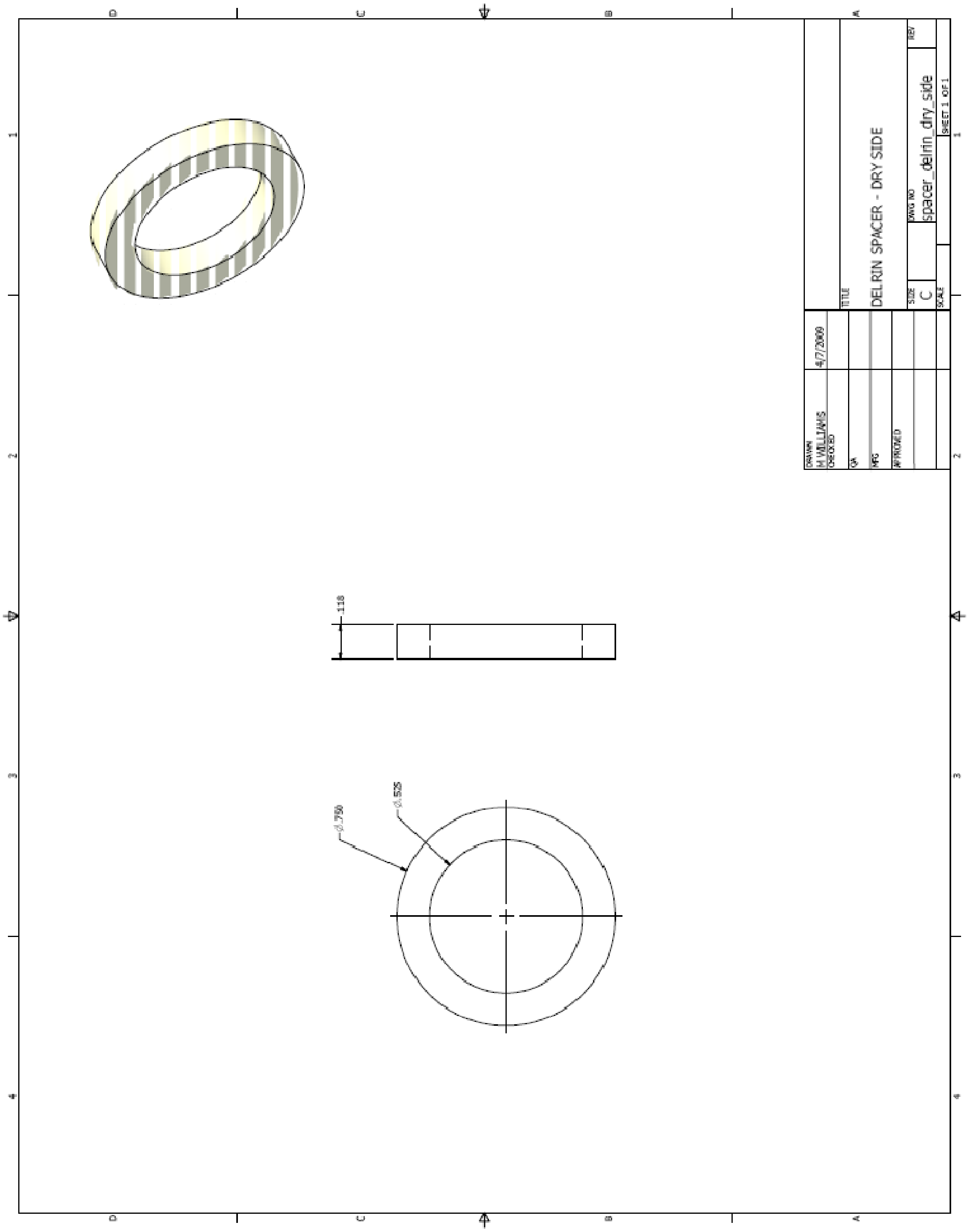
Motor-side satellite gear to 3.5” gear shaft



Delrin spacer – Motor wet side



Delrin spacer – Motor dry side



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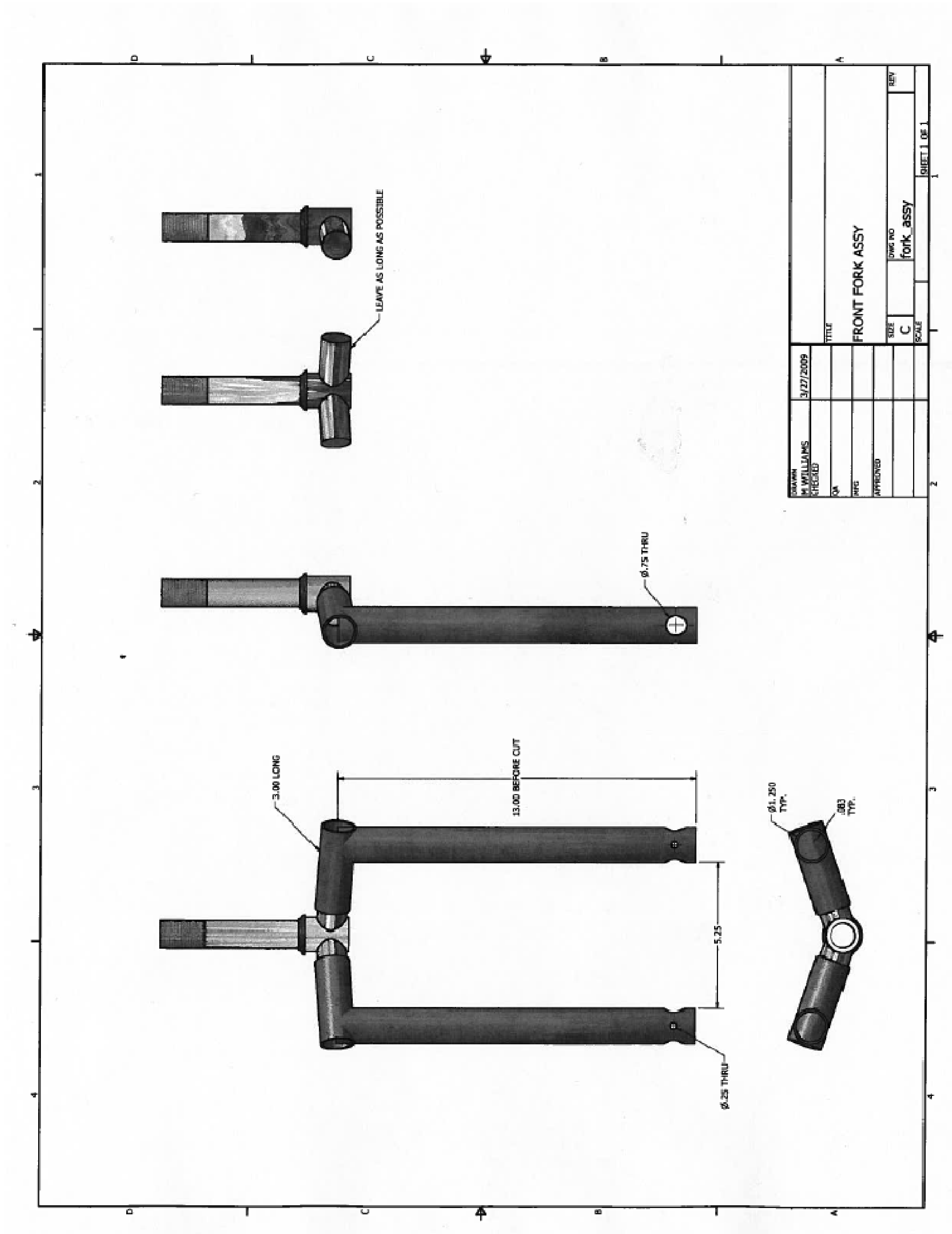
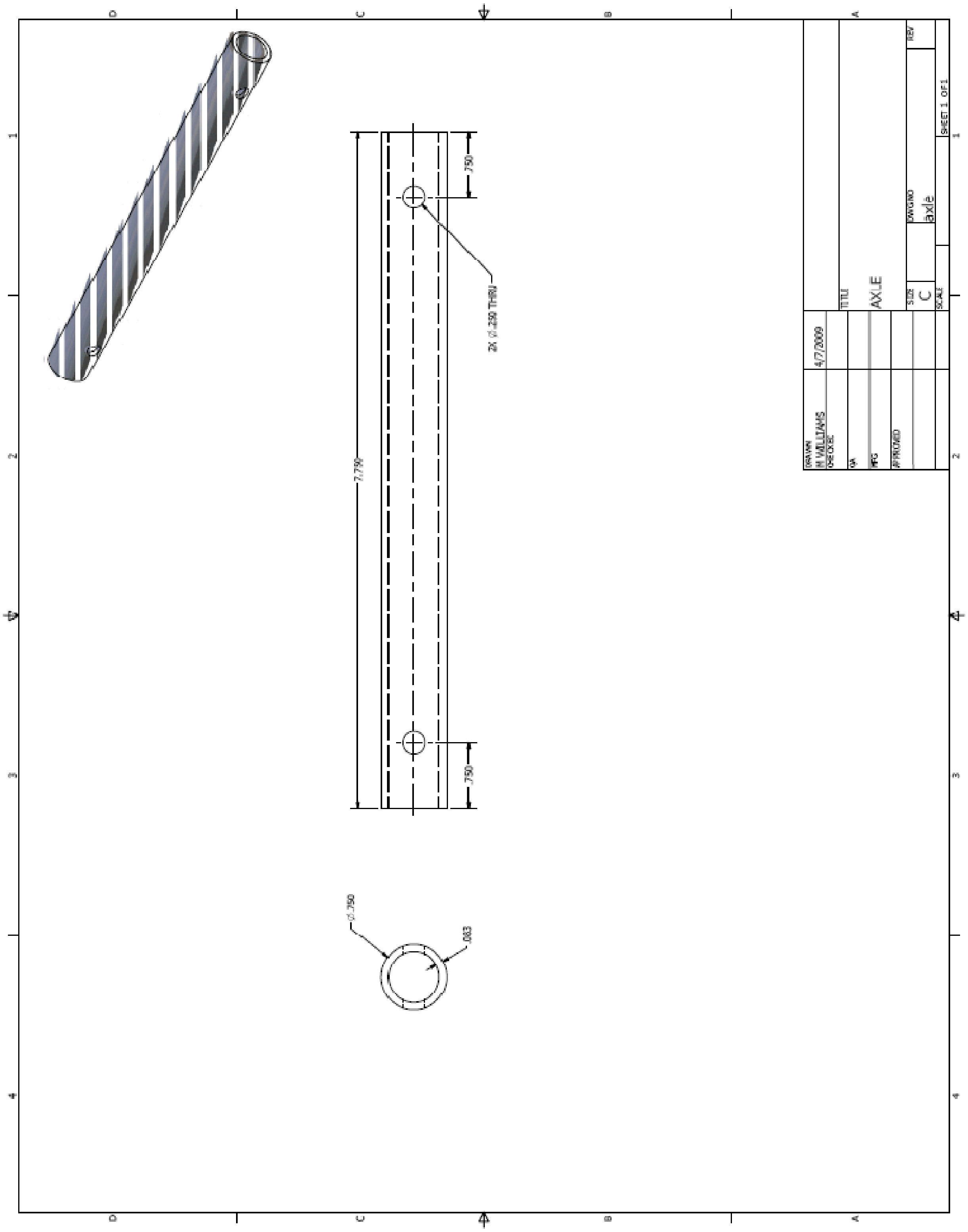
Fork Assembly



Axle


.750
2X Ø .250 THRU
7.790
.750
Ø .750
.083
DRAWN
4/7/2009
CHECKED
QA
MFG
APPROVED
TITLE
AXLE
SIZE
C
DWG NO
axle
REV
SCALE
SHEET 1 OF 1

Rim – 3 holes are #10 (spaced every 120 degrees) and the other 3 are 0.25”