ELECTRIC-HYDRAULIC HYBRID MOTOR COUPLING

FINAL REPORT
12/15/10

ME 450 – PROJECT 5
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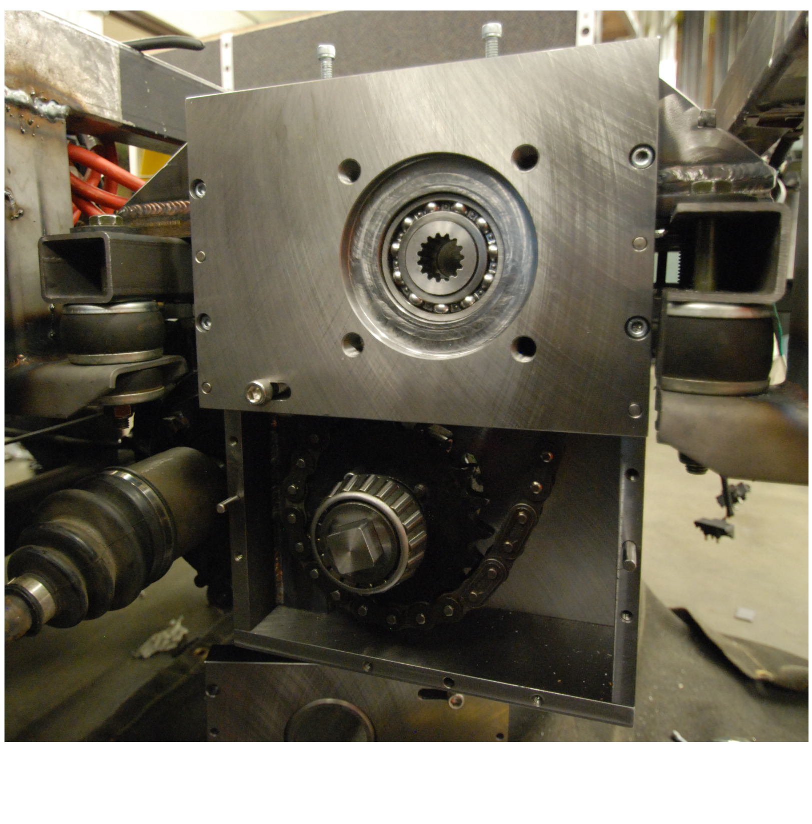
SECTION INSTRUCTOR – GORDON KRAUSS

DAVID FOK
ANDREW GAVENDA
ANUJ SHAH
MAT WECHARATANA



EXECUTIVE SUMMARY
Due to the continued effort to reduce vehicle emissions and increase the fuel efficiency of motor vehicles, the U.S. Environment Protection Agency (EPA) is working with University of Michigan (U of M) students and researchers to develop an electric-hydraulic hybrid vehicle by redesigning a ZAP! Xebra electric truck. Previously, other ME 450 teams have worked with the EPA on similar Xebra hybrid projects, but the lack of continuity as a result of multiple iterations created an inefficient design. Therefore, our sponsors (EPA) and customers (U of M) are creating a new hybrid using a new, "Blue Xebra" truck. Specifically, we have been tasked with creating the system that couples the new hydraulic pump/motor and the existing DC electric motor in this truck. This coupling system must also contain a gear reduction ratio between the hydraulic pump/motor and the existing DC motor gearing.

This power transmission coupling system must meet several requirements (detailed on page 7). Most importantly, the system must properly transmit torque and power safely when subjected to disturbances. The design must also allow for at least 3 gear ratios to be installed and easily changed. Additionally, the system must be reliable, low risk, and fit within the frame of the existing vehicle. It cannot negatively affect the performance or stability of the vehicle, and should cost less than $1000, pending sponsor approval of additional cost. Finally, all work on this project must be clearly documented so that future work and development can be conducted. The detailed correlation of these customer requirements to engineering specifications, as well as the relative importance of each specification, can be found in the Quality Function Deployment on page 11.

With the requirements of our design clearly defined, we generated concepts by decomposing our design into four functional areas: the reduction system, the coupling mechanism, the pump/motor mount, and the reduction case mount. We created multiple concepts for each functional area, and from these concepts we developed three "overall" concepts that encompassed all of these functional areas. Using both quantitative and qualitative analyses and an iterative process, we ultimately selected an initial "Alpha" design that utilized a rigid spline coupling, a roller chain and sprocket gear reduction, a horizontal hydraulic pump/motor mount, and a tab and gusset reduction case mounting method.

The Prototype/Final Design closely resembles the Alpha design. The hydraulic pump/motor is mounted directly to one side of the gear reduction case and a splined shaft connects the output of the pump/motor to the gear reduction system. The reduction system is composed of a chain, an upper sprocket that is attached to a hub on the pump/motor output shaft, and a lower sprocket that is interchangeable and accessible through a removable panel on the reduction case. The interchangeable sprocket is bolted to a hub, which is connected to the existing DC motor reduction through another shaft. The housing and associated reduction components are mounted using tab and gusset supports that are bolted to the reduction casing. These tab and gusset supports are bolted to a sub-frame, which in turn is mounted on dampeners to brackets that are welded to the vehicle's main body frame.

To fabricate this design on schedule, a detailed fabrication plan was developed and executed and can be found on page 46. Some machining was completed externally due to the capabilities of our resources. A bill of materials outlining the components procured and their costs is located on page 71.

While workspace limitations mainly drove the design, the Prototype design meets most of the customer requirements and engineering specifications outlined above. Most importantly, the design safely (without fracture) withstands 144 ft-lb of torque (limited by differential slip), with the intended bearing selection and chain slack. Also, the desired gear ratios and interchangeability time were achieved, and the reliability criterion of the design was satisfied through material selection and calculation verification. Although the final project cost exceeded $1000, it was approved by the sponsor. Finally, the vehicle stability was maintained, the components fit within the desired space, and all project information will be delivered to the sponsors and customers. A detailed discussion of the tests conducted and the results can be found on page 51.

In reflection of the completed design, while our finished product fulfilled the engineering requirements and successfully passed the validation process, there were issues that in hindsight could have been resolved. We would have avoided or improved upon our weaker design modifications, such as cutting into the sub-frame, introducing new components to join parts, and ineffective alignment. However, in the end, we machined and assembled a sub-frame, reduction system and case, that coupled the hydraulic pump/motor to the DC motor's output, while providing interchangeability in the gearing ratio, in a more space efficient and concise form than previous designs.

TABLE OF CONTENTS

1. INTRODUCTION

Due to the increasing push to reduce vehicle emissions and increase the fuel efficiencies of road-going vehicles, the United States Environment Protection Agency (EPA) is interested in increasing the efficiency of electric vehicles by integrating electric systems with hydraulic launch assist systems. Typically, there is a large efficiency drop, between 60% and 90%, during the acceleration of electric vehicles. Thus, by incorporating a hydraulic launch assist during the acceleration phase, the torque and power burden on the batteries is reduced and overall efficiency of the vehicle is improved [10]. The EPA is conducting small-scale testing on several vehicles including a ZAP! Xebra electric truck to validate the practicability of this concept. The focus of our project is to design a power transmission coupling system for the ZAP! Xebra electric truck, to enable it to operate using an electric-hydraulic hybrid drivetrain. This coupling connects the new hydraulic system with the existing electric system to allow the combined hybrid system to transmit power and torque to Xebra truck's single driveshaft.

Our sponsors, Andrew Moskalik and David Swain of the Environmental Protection Agency, have been working to develop electric-hydraulic hybrids for several years. Projects involving the University of Michigan began with the integration of a hydraulic system into a bicycle wheel. The EPA sponsors continued developing this technology with several ME 450 teams that made modifications to a ZAP! electric truck designated as the "White Xebra" (because of its paint scheme) by adding a hydraulic launch assist system to it. Over the course of several semesters, teams successfully designed a hydraulic launch assist system with a regenerative breaking mechanism for this White Xebra. The end result of these projects was a hybrid vehicle, which incorporated an electric-hydraulic power train. However, the numerous design changes to the White Xebra and lack of structural continuity to the changes caught the attention of our sponsors. They now want to redesign the system with increased performance and efficiency, using a more organized approach.

To accomplish this, Dr. Moskalik and Mr. Swain are working to develop a working hydraulic-electric test vehicle in conjunction with our primary customers, Andrej Ivanco and Xianke Lin of the University of Michigan. Dr. Ivanco and Mr. Lin are working on a new, "Blue Xebra" truck. The main goal of our project is to design a power transmission system that will enable the Xebra to operate using a hydraulic-electric hybrid drivetrain. As a part of this transmission system our most important task is to design a coupling mechanism that joins the DC electric motor and a hydraulic pump/motor to the wheels through a gear reduction case. We must design a stable mounting system to secure the motors and reduction gear case to account for the motors' relative motion under external disturbances. Torques and loads could lead to potential damage, inefficiency, and/or reduced life of the components. Another major requirement of our customers was to design a gear reduction case that allowed for the implementation of multiple gear ratios. To quantify our specifications we conducted patent research, referred to academic journals, previous team reports, and additional sources to better understand hydraulic and electric-hydraulic hybrid systems. Using the technical expertise and experience of our sponsors, we gained a better understanding of the functionality of our vehicle, the Blue Xebra.

The outcome of our project was the installation of the system that will couple the hydraulic and electric drives of the vehicle. Our customers from the University are simultaneously working on completing the hydraulic system of this hybrid. The end result of our joint efforts is to produce a fully functioning electric-hydraulic hybrid that will be tested to serve as a benchmark for our sponsors' and customers' future endeavors with electric-hydraulic hybrid vehicles. In a broader sense, the successful completion of this project will allow for testing of the validity and large-scale benefits of electric-hydraulic hybrid technology. It is important to note that upon completion of our project, we produced a coupling mechanism that is capable of coupling the electric and hydraulic systems, but the hydraulic system was not ready for implementation yet. Thus, this report presents detailed documentation to allow others to complete this coupling once the hydraulic system has been completed.

2. ENGINEERING SPECIFICATIONS

Our customer requirements were developed through several meetings with the customers and sponsors of our project, the University of Michigan researchers and our two EPA contacts. Using an iterative process with continual feedback from our sponsors, we translated the customer requirements into engineering specifications that quantified their requirements for a successful project. Ultimately, the specifications embody the overall purpose of our design in detail. Each specification is discussed in detail below. Additionally, the relationship between each customer requirement and engineering specification and the relative importance of each requirement was correlated using Quality Functional Deployment. Examination of the specific requirements that comprise our system design allowed us to investigate how well competitive products satisfy these requirements when compared to our system.

2.1 TRANSLATION OF CUSTOMER REQUIREMENTS

Our sponsors explicitly stated several of our significant customer requirements. They expressed their need for a fully functional system, complete with a changeable reduction ratio. We formulated other specifications utilizing engineering judgment to determine criteria for a successful, functional design. This included safety, reliability, spatial concerns, design risk, stability, and cost. Through continual discussion and verification with our sponsors, we were able to designate the relative importance, or weight, of each requirement. While the highly weighted requirements describe critical objectives of the overall design, the lower weighted requirements describe general features of the overall system design.

For some engineering specifications directly provided by our sponsors, we encountered some difficulty in accurately quantifying certain criteria using a single value. Distinctly, our highest weighted requirement that "the system must transmit power and torque when subjected to disturbances" could not be encompassed fully with a single value. Therefore, we decomposed the requirement into separate parts that would combine to satisfy this requirement. We judged that the customer requirement could be broken down into two basic engineering specification areas. The first accounted for fracture under torque and the second considered part engagement. Combined, these two fully encompassed the requirement that the system functions properly while under torque. Similarly, we correlated the customer requirement of "reliability" to multiple engineering specifications. We determined that reliability was fully encompassed by an "infinite" fatigue cycle life and a proper safety factor

For the requirements that were not explicitly stated by our sponsors, we referred to a variety of information sources to develop quantifiable values for targets such as center of gravity and space restrictions. We incorporated measurements of the frame geometry to restrict our workspace and researched industry standards for part clearance limits. Other specifications were based off of previous Xebra team targets and were translated and verified by our sponsors.

2.2 CUSTOMER REQUIREMENTS AND CORRESPONDING ENGINEERING SPECIFICATIONS

A table summarizing the customer requirements and the corresponding engineering specifications can be found in Table 2.2 on page 8.

2.2.1 System Transmits Power and Torque When Subjected to Disturbances

Invariably, the purpose of our design is to successfully transmit power and torque from the hydraulic motor to the rear axle while being subjected to any mechanical disturbances. Our sponsors also explicitly expressed this customer requirement. The coupling system must be able to do this under all external and internal loads and torques that the system may encounter that may create relative motion between the motors and the coupling system. Using engineering analysis, the greatest disturbance, internal or external, the system would face is the 206 ft-lb (see Appendix L on page 248) of combined torque output by the hydraulic pump/motor and electric motor. In order to translate this into a quantifiable engineering requirement, we divided this requirement into two areas. 1) Under this torque, the system components must not fracture, and 2) the components must remain engaged.

While the fracture specification is well defined for any system, the engagement specification is dependent on the final system design. Each design has specific components that need to remain engaged. For example, for a system with bearings, the bearing should not fail and thus must not operate at an rpm higher than ½ of the maximum rated rpm [27] [28]. For proper engagement in a roller chain and sprocket system, the chain slack distance must not be greater than 2% of the center-to-center distance between the sprockets [40]. For spur gears, a contact ratio of 1.2 must be maintained, and for a belt drive, the belt slip value must not exceed the system's threshold value [41]. Each of the quantitative values for these examples of part engagement was sourced from industry standards that apply to the particular parts. If any of these conditions are not met, the system could be considered not engaged.

Weight	Customer Requirement	Description	Engineering Specification	Value
1 (least) -10 (most)	*Desired features in end users' language*		*Quantitative values*	
10	System transmits power and torque when subjected to disturbances	When the electric motor and hydraulic pump/motor, and thus the vehicle, are operating, the system must perform its intended function and transmit torque and power. This must occur even when the system is subjected to relative motion between electric motor, driveshaft, hydraulic pump/motor, and gear coupling, or any internal or extrenal loads and torques are applied to the system. Thus, components must remain engaged and not fracture during performance. Engagement is defined independently for each design. Several examples are given, but a specific specification can not be defined until the design is chosen.	Under maximum torque experienced (206 ft-lb), the system components (gearbox, shafts, mount, etc.) do not fracture (0 fracture). (Note: 206 ft-lb is the maximum disturbance, internal or external that the system is expected to experience)	206 ft-lb, 0 fracture
			Parts remain engaged. EXAMPLE: The system does not operate over half of the bearing's maximum rated rpm	< 1/2 maximum rated rpm
			Parts remain engaged. EXAMPLE: The amount of slack in the roller chain is less than 2% of the center-to-center distance between the sprockets.	< 2% slack
			Parts remain engaged. EXAMPLE: The contact ratio between gears is ≥ 1.2	≥ 1.2 contact ratio
			Parts remain engaged. EXAMPLE: The belt slip value (BSV) is < system threshold ($BSV_{threshold}$)	$BSV < BSV_{threshold}$
10	System transmits power and torque safely	The system must operate safely. In addition to the above engineering specification that the system must not fracture when subjected to maximum torque, two additional safety concerns are required: 1) no parts should be expelled from the vehicle and 2) each openly exposed component must be fully enclosed	If parts were to break, they must not be expelled from the vehicle. For example, if a chain drive were used and it breaks, no parts should be flung from the vehicle. Therefore, all components of the design must remain within the restriction planes designated to create boundaries in the workspace (as specified in the requirement "Components fit within frame") at all times.	Parts always contained within restriction planes of existing vehicle frame
			The system must be designed so that no body parts and/or clothing could get caught in the machinery. Proper material selection, sizes, and clearances must be used pursuant to MIOSHA R408.10751 - R408.10754. From these rules, the maximum allowable opening (mesh opening) for metal is 0.5 inches.	≤ 0.5 inch opening
9	Configurable gear reduction	The system is capable of installing and testing multiple gear ratios (gearing the pump/motor rpm up to the driveshaft) in order to determine the most efficient gear ratio The hydraulic system must also be capable of disengagement from the driveshaft so that the vehicle can operate using only electric power.	Gear ratios between the pump/motor and the wheels must include 2.5, 3.0 and 3.6 to 1. The ratios are subject to change by request of the sponsor.	2.5:1 - 3.6:1
			There are 3 or more functional gear ratios over the range of 2.5 to 3.6:1, between the pump/motor and wheels, as well as a neutral position	≥ 3 ratios & neutral position
8	Easy to change gear ratios	Without a complete teardown of the system or disconnecting the system mount, the gear ratio can be changed between the above ratios.	Time to change components and reengage proper connections must be ≤ 3 hours.	≤ 3 hours

7	Reliable design	The components can withstand "infinte" fatigue cycling which is due to material and geometric (part shape) considerations.	Parts are designed to withstand cyclical loading for an "infinite" number of cycles. The maximum stresses faced by parts must be less than the endurance limit of the material	< Endurance limit of material (infinite fatigue life)
		To ensure parts will not fail under any expected loads, we have implemented a safety factor of 2.	Safety Factor of all parts must be ≥ 2	SF ≥ 2
6	Components fit within frame	When installed, the components added as part of the system must fit within the frame of the vehicle and must not contact or interfere with the vehicle's existing components.	The volume of the components added as part of our system must total ≤4.4 ft^3. Additionally, added components must not interfere with existing components. Specifically, they must not be located above the bottom of the truck bed or below the bottom of the current location of the DC motor. The engine rock clearance (distance between parts not intended to be in contact) must be ≥ .52".	≤ 4.4 ft^3
				No (0) unintended contact or interference with existing components
				≥ .52 inch
5	Design risk considered	An inherent amount of design risk is associated with any design and must be considered. This risk should be quantified and minimized. Risk takes into account performance and reliability factors as well.	A quantitative system is developed to quantify the risk associated with certain aspects of the design. Risks include but are not limited to time, assembly, complexity, tolerance, new application (novel), and machining risks. Only the designs we considered feasible to produce were scored against this system	Lowest (minimal) risk score
3	Vehicle stablity and dynamic performance maintained	The addition of the coupling system components and additional structures must not negatively impact the vehicle's stability and performance. The location of the truck's center of gravity must not move vertically higher than it currently is in order to maintain rollover stability.	Location of center of gravity (CG)	Between rear axle and the current CG. No higher (0 in.) than current vertical location and no lower than wheel contact with ground
2	Total cost	Project cost must be justified, detailed, documented, and provide the customer with the desired performance.	Total cost of components should be less than $1000. *(However, if increased performance can be achieved with increased cost, over $1000 may be spent. This is subject to sponsor approval.)	≤ $1000*

1	Transferable for future development	Enough information must be provided so that this project can be continued and optimized in the future by the EPA and UM teams, and possibly other ME 450 teams, continuing this research.	Proper documentation is provided and acknowledged. Documents that contain detailed description and explanations of our customer requirements, engineering specifications, concept generation, alpha and final concept selection, CAD files, analysis, testing reports and final report are provided.	Written acknowledgement that our team has satisfied this requirement (The team provided sufficient documentation)

Table 2.2: Customer Requirements and Engineering Specifications

2.2.2 System Transmits Power and Torque Safely

Another requirement directly stated by our sponsors was that not only must the design function properly, it must do so safely. In this usage, safety encompasses an area different than reliability. While reliability, as discussed below, focuses on material and parts' material and yield reliability, safety is focused more on ensuring that these parts do not fail and will not hurt operators of the system. We translated this into two parts. 1) No parts can be expelled from the vehicle in case of fracture, and 2) all openly exposed components must be fully enclosed.

In particular, the first specification requires that all parts of the system must remain within the bounds of the vehicle at all times. This means that all parts, functional or broken, must stay within the planes created by the frame and wheels of the vehicle so that ejected parts do not hurt anyone. The requirement that all components must be fully enclosed serves a similar purpose. All parts that are exposed to the environment (not enclosed by other parts) must be fully enclosed. Specifically, "enclosed" can be described by less than ½ an inch of spacing between parts of the same enclosure. As an example, for a box with six sides that contains other parts within it, no two sides can have a gap larger than ½ inch wide. This value stems from the Michigan Occupational Safety & Health Administration requirement for Guards for Power Transmission (Rules R408.10751-R408.10754) [31]. The ½ inch requirement represents the gap allowed for a metallic mesh for a power transmission guard.

2.2.3 Configurable Gear Reduction

The project customers specifically stated that the coupling system's gear reduction should allow for at least three separate gear ratios and a neutral position. Additionally, the range of ratios must fall between 2.5:1 and 3.6:1 (between the pump/motor output and the wheels), with ratios near 2.5:1, 3.3:1, 3.6:1 specifically included. The multiple gear ratios will allow the system to be tested for the optimal gear ratio. The specific gear ratios were specified by our customers, who conducted research to determine the range of the optimal gear ratio. The neutral position allows the hydraulic system to be fully disconnected from electric system without applying any loads to either motor during disengagement and reengagement. This is to allow for the electric system to be tested alone as a baseline test.

2.2.4 Easy to Change Gear Ratios

In addition to the previous configurability requirement, the gear ratio must be easy to change. As directly stated by our sponsors and customers, this translates to less than 3 hours required to change the gear ratio from one to another. However, these gear ratios need not be changed while the truck is in motion [12].

2.2.5 Reliable Design

Our customers wanted a reliable design. We subsequently decomposed this reliability requirement into two sub-sections, both of which are related to material reliability. 1) Parts must be able to withstand "infinite" fatigue cycling, and 2) all parts must be designed with a yield Safety Factor of at least 2.

Our sponsor directly specified the first specification of infinite fatigue cycling. Infinite fatigue life relates the material choice and part geometry to the stress experienced. The number of cycles a material can endure before breaking is a function of the stress it experiences, the material and the treatments applied to it, and other additional factors. Studies on the relation between applied stress and the number of fully reversed cycles to failure was conducted by August Wohler who produced S-N (stress vs. cycles) curves. These diagrams characterize the behavior of materials under stress cycling, which is used to evaluate the failure strength of materials [32]. As a standard, "infinite fatigue life" is defined as being able to apply 10 million cycles of force to a part without it permanently deforming or failing [33]. This is characterized with the concept of endurance limit, which describes the amplitude of cyclic stress that can be applied to a material without casing fatigue failure. If the applied stress is below the endurance limit of the material, the structure is said to have "infinite fatigue life" [34]. Typical values for the endurance limit of steels are ½ the ultimate tensile strength [35].

The second specification regarding Safety Factor was determined by exceeding industry standards [36] in consultation with our sponsors. From this analysis, we agreed on a Safety Factor in yield of at least 2.

The overall system reliability and redundancy had been considered as specification. However, we determined that this was unnecessary for two reasons. First, the reliability of each part could not feasibly be determined because it would be impractical to create multiple parts to conduct failure testing in order to produce reliability statistics such as a Weibull distribution. Aside from the lack of reliability statistics, our first reliability requirement of "infinite" fatigue life renders system reliability analysis useless because the reliability of each part is 100% in theory. Thus the overall system reliability due to material considerations would be 100%. It is obvious that this would not be equal to the system's overall reliability because there are other factors such as assembly and outside factors not relating to material considerations that cannot be quantitatively considered. These factors must be accounted for in a "risk" requirement as discussed below.

2.2.6 Components Fit within Frame

We used engineering judgment to determine that the system's components must fit within the vehicle's frame and should not interfere with any of the existing parts, particularly the DC motor and suspension system. Therefore, we limited our design to 4.4 ft^3 in total volume and established non-contact "planes" that our added parts should not cross. The total volume was calculated using measurements of the available workspace that our system must fit within at the rear of the truck, below the bed. The planes that were established were the horizontal bounds (left to right) of the truck's frame and no higher vertically than the frame or lower than the current DC electric motor.

Our customers also wanted us to keep in mind the relative motion of the hydraulic lines and additionally added components. Therefore, our design must account for appropriate clearances. To quantify this, we worked with our sponsors to establish a required clearance between parts non-internal to our design of 0.52 inches. More specifically, parts not internal to the design means parts that are not considered fully enclosed as specified in the "System Transmits Torque and Power Safely" requirement above.

To derive the value of 0.52 inches of clearance, we first used a baseline value of 1 inch of clearance for a full-sized vehicle. This value was deemed an industry standard by our EPA sponsors. We then selected a full-sized vehicle that is similar to the Blue Xebra truck to compare to. In this case, we selected a small pickup truck, the 2002 Ford Ranger (the vehicle that had been used in past reports as a comparison [42]). From this model, we developed a scaling ratio based on the vehicles' length and width dimensions. Using the overall area of each vehicle, we calculated a scaling ratio of (A_{Ranger} = 90.36 ft^2) / (A_{Xebra} = 46.6 ft^2) = 52%. By multiplying this scaling ratio to the initial baseline value of 1", we determined a clearance specification of 0.52 inches.

This method characterizes the critical factor of clearance, the limiting dimensions of the vehicle, using a linear correlation. Although load/weight and torque of a vehicle influence the amount of clearance needed between parts to prevent unintended contact, loads and torques will constantly change while the dimensions of a given vehicle will remain the same. In past iterations of this specification, we had used a direct scaling ratio between the combined hydraulic and electric motor torque of 206 ft-lb and the torque output of a Ford Cologne V6 engine (used in the 1992 Ford Ranger, regular cab, short wheelbase) of 240 ft-lb. 240/206 = 85.6%, which when multiplied with 1 inch, produced the 0.85 inch clearance requirement.

However, we realized that the torque is not the deciding factor when considering vibration and clearance issues because our system will be directly mounted to the motor, which will ideally produce no relative motion between parts. Therefore, it is more important to consider the dimensions of the vehicle, external to this coupled system of the motor and our motor coupling, rather than the torque generated by an engine or motor. Torque generated by engines and motors vary greatly, even within vehicles of the same size. Thus, it is not a very accurate scaling method. So although using dimensions alone may not consider all possible influences, for our purposes, this method is suitable because torque influences were deemed smaller than dimensional influences.

2.2.7 Design Risk Considered

Our sponsors requested that we quantitatively consider the relative risk between design concepts. Specifically, how likely is one design expected to perform properly with respect to another? This allows for a comparison between designs and gives another quantitative comparison and explanation as to why the particular design was chosen. An inherent amount of risk is associated with any design. Generally, this risk should be minimized. However, design performance and reliability should be factored in as well. In relation to this project, risks are present in areas including, but not limited to, assurance that the design will work, error in the design assembly, the complexity and number of parts in the design, the time and effort required to machine the parts, how tight tolerances must be for the design to function properly, the lead time and availability of parts, the novelty or uniqueness of the design (in general new, untested designs are riskier), and whether the design's materials fit the application.

Therefore, we created a system to quantify these risks. Each risk was given a weight, and using Pugh analysis, we rated three feasible designs. We only considered designs that we believed could properly satisfy all of the engineering specifications and were feasible to construct and implement. The results can be found below in Appendix D on page 149. As an engineering specification, we required that the design with the lowest "risk score" be chosen. This engineering specification only considers the risk of each design as defined by the eight identified factors. The specification is then used in conjunction with all other specifications during the overall design concept selection as described below.

2.2.8 Vehicle Stability and Dynamic Performance

Using information collected from past reports and our sponsors' recommendations, we determined that the vehicle's stability and dynamic performance should remain the same after our alterations were implemented. We translated this customer requirement to an engineering requirement in terms of the center of gravity (CG) of the vehicle [9] [10]. The vertical location of the center of gravity is closely related to stability (particularly rollover stability) of the vehicle. Similarly, the horizontal location of the CG dramatically affects the dynamic performance of the vehicle. Therefore, it is required that the vehicle at least retain its original stability with the horizontal component of the center of gravity no more forward from the rear axle than the current CG and no further back than the rear axle [4] [7]. Likewise, the vertical component of the CG must be above the ground and no higher than the current CG. The center of gravity was measured prior to the addition of our system to establish a baseline for comparison. Previous teams have attempted to measure similar qualities on different vehicles. However, based on analysis of their procedures, we did not feel that their baseline would be accurate for our project, mainly because our vehicle is entirely new.

2.2.9 Total Cost

Our sponsors have estimated a baseline restriction on the cost of our project, stating that our project's budget be no more than $1000. However, they have expressed flexibility in this value depending on the effectiveness of the system. A higher budget would be allowed if the performance matched the cost. This is subjective and dependent on approval from our sponsor.

2.2.10 Transferable for Future Development

As in the past, the results of this Xebra project must be transferable to our sponsors so that the EPA, UM, other departments, companies, or design teams can continue development on the vehicle and build off of our findings. Proper documentation must be provided, and a written sponsor acknowledgement will be obtained to verify our sponsor's satisfaction with the documents and files we have provided. Documentation must include material on customer requirements, engineering specifications, concept generation, alpha and final design selection, CAD files, analysis, testing reports, and a final report.

2.3 QUALITY FUNCTION DEPLOYMENT

A Quality Function Deployment (QFD) was developed for this project in order to correlate the different design specifications and gauge their interaction with one another (Table 2.3, pg. 11). By establishing their interrelation, we can correctly determine the key quantifiable measurements we should strive for and which would contribute the most to satisfying our customers. To determine these relations, we refined our design specifications through numerous iterations, incorporating feedback from our sponsors and customers.

In terms of the specific workings of the QFD, we identified customer needs that were deemed essential by our sponsors and associated a quantifiable value that correctly evaluated our success. Then, in analysis of the different requirements accumulated, we found that some variables, such as gear ratio related specifications, correlated highly with performance and many of the primary customer needs. However, lower priority needs such as total cost and aesthetics did not attribute highly to our main objectives. In keeping constant contact with both our University and EPA associates, the team was able to determine the priority of needs. Therefore, by incorporating research, information, and calculations done by past teams, searching for credible documents on work in this field, and speaking with our sponsors, and using general engineering judgment, we developed the specification correlations of the QFD.

Project:	Electric-Hydraulic Hybrid Power Coupling	
Date:	11/5/10	

Survey Legend	
A	Purely Electric Blue Xebra
B	White Xebra

#	Technical Requirement	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	Torque capable of being withstood (No fracture)															
2	Parts remain engaged (bearing rpm)	++														
3	Parts remain engaged (chain slack)	++														
4	Parts contained within restriction planes															
5	Enclosure opening gap															
6	Range of installable gear ratios	+			-											
7	Number of installable gear ratios						++									
8	Time to change gear ratio						+	+								
9	Infinite fatigue cycles	+	+													
10	Safety Factor	+	+													
11	Total volume of added components				+											
12	No unintended contact					++						++				
13	Clearance between parts			+	++				+							
14	Lowest Risk			+						+	+					
15	CG position											+				
16	Project cost						-									

Technical Requirements / **Customer Opinion Survey**

#	Customer Needs	Customer Weights	Torque capable of being withstood (No fracture)	Parts remain engaged (bearing rpm)	Parts remain engaged (chain slack)	Parts contained within restriction planes	Enclosure opening gap	Range of installable gear ratios	Number of installable gear ratios	Time to change gear ratio	Infinite fatigue cycles	Safety Factor	Total volume of added components	No unintended contact	Clearance between parts	Lowest Risk	CG position	Project cost	1 Poor	2	3 Acceptable	4	5 Excellent
1	Transmits power when subjected to disturbances	10	9	9	9						9			1	1	3					B	A	
2	Transmits power safely	10	9	9	9	9	9				9	9		1		9	1			B		A	
3	Configurable gear reduction	9	3	3	3			9	9				1		3	1				B		A	
4	Easy to change gear ratios	8			3	3		9	9	9			1	1	9						B		A
5	Reliable design	7	9	9	9		1				9	9	1			9					AB		
6	Components fit within frame	6				9	3	9	9				9	9	9		3				AB		
7	Design risk considered	5	9	3	3	1	3		3	1	3	3		3	3	9	1				B	A	
8	Vehicle stability and performance maintained	3	3	1	3	3			3		1	1	9				9				B	A	
9	Total cost	2	1					1	3		3	3				3		9				B	
10	Transferable	1	3	1				3	1	1		1				1	3	3					
	Raw score		329	289	318	209	130	212	238	78	267	178	105	97	178	244	63	21					
	Scaled		1	0.878	0.967	0.635	0.395	0.644	0.723	0.237	0.812	0.541	0.319	0.295	0.541	0.742	0.191	0.064					
	Relative Weight		11%	10%	11%	7%	4%	7%	8%	3%	9%	6%	4%	3%	6%	8%	2%	1%					
	Rank		1	3	2	8	11	7	6	14	4	9	12	13	9	5	15	16					
	Technical Requirement Units		ft-lb	rpm	% of center-to-center distance between sprockets	volume of parts outside restriction planes	inches	gear ratio	number of ratios	hours	Mpa	Safety Factor	ft^3	amount of contact	inches	rank	position	$					
	Technical Requreiment Targets		≥ 206	1/2 max rpm	≤ 2	0	≤ .5	2.5:1 - 3.6:1	≥ 3	≤ 3	< Endurance limit	≥ 2	≤ 4.4	0	≥ .52	Lowest	rear axle < CG < current CG	≤ 1000					

Table 2.3: Quality Function Deployment

2.4 RELATIVE IMPORTANCE OF SPECIFICATIONS

Creation of the QFD allowed us to properly organize our requirements and specifications and clearly see how their relative importance would factor into our design considerations. We found that the transmission performance and gear ratio details have the highest positive correlation. The reliability requirements had some positive correlation with the power transmission as well.

Using the QFD, we determined the rankings of engineering requirements that were the most important for achieving a successful design that meets our designated customer requirements. After incorporating all requirements, we found that the torque transmission requirement was identified as the most significant, which followed our intuition. This was because this specification is the primary motivation for this project and is dependent upon all of our components functioning properly. The next ranks showed the importance of the reduction system, specifically the two specifications that outline the engagement of the system. Again, this is expected since these specifications form a large part of the project goal of a functioning system. An interesting observation found from the QFD was the importance of the infinite fatigue cycles and low risk requirements. Although these would not seem as important intuitively, we realized that both are closely related to the main requirement of a functional system. Infinite fatigue cycles are required for a reliable design and an unreliable design is not likely to perform properly. Likewise, a design that is higher risk is also less likely to perform properly.

2.5 COMPETITIVE PRODUCTS

Using the engineering requirements specified above, we examined different comparable products to contrast with our design. Because of the recent development of this hydraulic-electric hybrid system, the only comparable vehicles were the purely electric stock ZAP! Blue Xebra and the modified hybrid ZAP! White Xebra modified by previous teams. The rating scale (as seen in the QFD) allowed us to determine the quality level and general expectations of our design. While the Blue Xebra (purely electric) scored well in the customer requirements, the purpose of the hydraulic implementation is to improve fuel efficiency, which is not covered in the scope of our project. The White Xebra implemented this fuel efficiency improvement, but sacrificed some customer requirements in the process. While overall the vehicle scored well, it lacked the configurability that is required in our project (gear ratio changes). Also, the requirement that all components must fit within the frame was not satisfied because the vehicle's bed was lifted to accommodate hydraulic components.

3 CONCEPT GENERATION

3.1 FUNCTIONAL DECOMPOSITION

We generated concepts by first decomposing our project into functional units. The purpose of our project is to transmit torque, power, and rotation from the hydraulic pump/motor to the Xebra truck's secondary gearbox while implementing an interchangeable gear ratio. To accomplish this, the project can be broken down into three main functional areas. The areas are 1) the interchangeable gearbox, 2) the couplings, and 3) the mounting system.

The gearbox functional area includes the gear reduction system, gearbox enclosure, and related supporting features including bearings, hubs, and collars. The coupling functional area encompasses the physical shaft connections between the gearbox and the DC and hydraulic motors, as well as the connection methods between each. Finally, the mounting system is comprised of the structure that connects the gearbox and couplings to the rest of the Xebra truck frame. These three areas fully satisfy all aspects of the customer requirements if the engineering requirements are successfully met.

Each team member then independently generated concepts for two of the four functional areas using a variety of sources including online parts catalogs and vendors [16], research of past designs and similar applications [21], logical engineering judgment, and previous knowledge. By limiting concept generation to only two of the four areas, team members could perform more in-depth research and analysis for those areas, rather than being concerned about every aspect of the project. Additionally, because each team member generated concepts for two areas, there was overlap and two team members produced concepts for each functional area. After each team member brainstormed independently, the team as a whole compiled all of the ideas and identified duplicates. Combined team discussion also allowed us compare different concepts from different team members so that some options might be combined together for a better concept.

During the iterative process, the team frequently reconvened to discuss concepts across their designated groupings to compile a wide variety of opinions and viewpoints. Although each member was defined a certain area of expertise to research options, every member often gave their perspective on every functional area. The initial concepts generated were all discussed, combined, and even adapted in different ways to form entirely new ideas. The team evaluated these ideas, particularly the members responsible for the functional area, who had more expertise and background knowledge of the problem. As a result, we generated several unique designs that are described below.

3.2 REDUCTION GEARBOX

3.2.1 Roller Chain and Sprocket with Idler/Tensioner

A roller chain and sprocket is a conventional method of transmitting power and torque through a gear reduction that is used in many applications including bicycles and motorcycles (Figure 3.1, below). An input shaft has one sprocket mounted to it and the output shaft has the other sprocket mounted to it. The roller chain connects the two sprockets and is driven by the teeth on each sprocket. The most important factors for the application of this system are the sprocket sizes and ratios, chain length and working load, and the tensioning system.

Chains can be custom fit to a sprocket system by removing links in the chain, but there can still be some residual slack in the system. Slack in a system can cause premature wear in a chain, greatly decreasing its life. If enough slack is present in a system, it can cause the chain to slip on the sprockets potentially damaging the system. Therefore implementing a tensioning system as shown in Figures 3.2-3.4 would remove a great deal of slack in the system preventing any of these issues.

Because the idler/tensioner is such an important aspect to the gear and sprocket system, we examined several types. A spring-tensioned idler uses a spring to keep tension in the string (Figure 3.2). A notched idler uses a pre-cut pattern to change the position of the idler, similar to how a lawn chair changes backrest inclination angle (Figure 3.3). A free-rotating idler sprocket spins freely as the system rotates. A motorcycle-style chain tensioner uses an adjustable arm and bumper to keep tension on a chain as seen in Figure 3.4 [20]. Each of these systems must provide adequate tension on both sides of the chain. Because each sprocket (smaller and larger sprocket) is the driven sprocket at any given time, both sides of the chain could be the slack side at any given time.

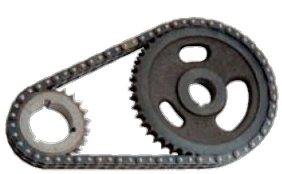


Figure 3.1: Roller Chain and Sprocket

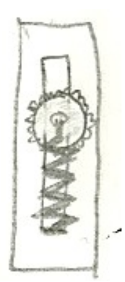


Figure 3.2: Spring-Tensioned Idler

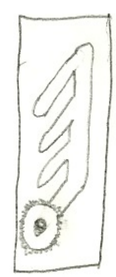


Figure 3.3: Notched Idler

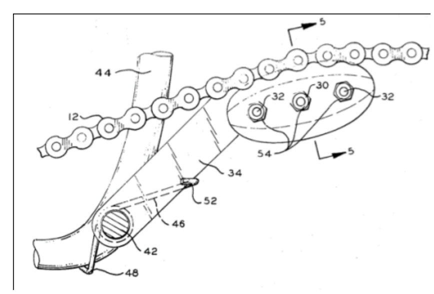


Figure 3.4: Motorcycle Chain Tensioner

For our design, the sprocket sizes and chain size would need to be selected for our application. Also, the interaction between the idler/tensioning system and the chain and sprocket system would have to be precisely coordinated.

3.2.2 Planetary Gear System

A planetary gear system is a unique method of transmitting power and torque through a gear reduction (Figure 3.5). It is used in several common applications such as an automatic transmission system and hybrid vehicle transmissions. In this system, a central "sun" gear engages two or more smaller "planet" gears that are engaged inside of a larger outer ring. Planetary gears turn on a movable center and the sun gears turn on a fixed center. This allows for a drastic gear reduction ratio and allows for greater stability due to the even distribution of mass and increased rotational stiffness [22].

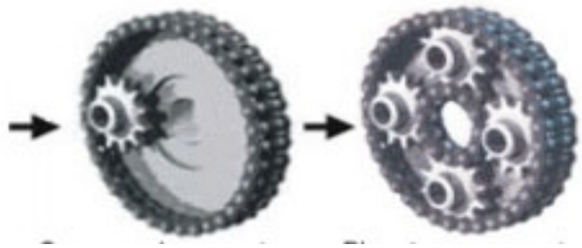


Figure 3.5: Planetary Gear System

For our design, the location of the input and output would have to be precisely selected due to the space limitations for our system. Also, the size of the sun and planet gears would need to be accurately controlled for the interchangeability of the gear ratios.

3.3 COUPLING

3.3.1 Rigid Sleeve/Muff Coupling

A sleeve coupling consists of a tube with an inner bore that is the same size as the shaft size (Figure 3.6). Installing a key between the inner bore and shaft transmits torque. Threaded holes with set screws are used to lock the coupling and shafts in place. In this case, a keyway would not be necessary because the spline profile serves the same purpose [19]. This connection could be used between the DC motor and the gear reduction and between the hydraulic pump/motor and the gear reduction.

3.3.2 Rigid Spline Coupling

A spline coupling consists of a male shaft being inserted into a female shaft with the same spline cross section. The torque is transmitted by each of the teeth around the spline. The shafts do not lock together so there is potential for disengagement if the shafts shift too much axially.

3.3.3 Rigid Flange Coupling

A rigid flange coupling is similar to the rigid sleeve coupling. The only difference is that the instead of threaded holes with set screws locking the coupling together, the two separate ends of the coupling have flanges where the bolts are secured along the same axis as the shafts that are coupled together (Figure 3.7).

3.3.4 Flexible Spider/Jaw Coupling

Spider/Jaw couplings contain an elastomeric insert between the two ends of the coupling that allows for vibration dampening (Figure 3.8). The insert, commonly referred to as a spider due to its shape, allows for a "zero-backlash" fit [15].

3.3.5 Flexible Magnetic Coupling

A flexible magnetic coupling is a unique coupling system that transmits torque without direct contact. A magnetic coupling consists of two nested rotors, each with its own set of rare earth magnets (Figure 3.9). The outer rotor mounts to the drive, and the inner rotor is connected to the output. As the drive turns, the magnets in the rotors transmit torque to the output [13]. The non-contact allows misalignment between each side of the coupling.

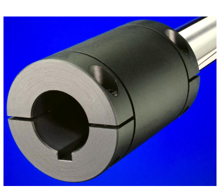


Figure 3.6: Rigid Sleeve/Muff Coupling

Figure 3.7: Rigid Flange Coupling

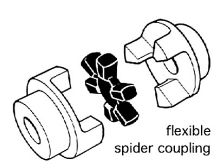


Figure 3.8: Flexible Spider/Jaw Coupling

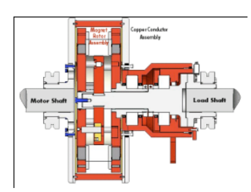


Figure 3.9: Flexible Magnetic Coupling

3.4 PUMP/MOTOR MOUNTING

3.4.1 Horizontally Mounted

The more obvious mounting method is to orient the pump/motor horizontally, with the DC motor output shaft and pump/motor output shaft parallel to each other (Figure 3.10). Thus, the gear reduction can be implemented in a single plane. For this method, the pump/motor is directly attached to the reduction casing, which is in turn directly attached to the existing intermediate gearbox that is attached to the DC motor output. This system rigidly connects both motors together so that relative motion is minimized. Spacing and the orientation of all the components may be an issue with this method. In particular, the width (oriented vertically with this method) of the pump/motor may be a clearance issue.

3.4.2 Vertically Mounted

The pump/motor may also be mounted vertically. In this orientation, the pump/motor output shaft and DC motor output shaft are along perpendicular axes. With this method, the pump/motor is mounted rigidly to a frame (separate from the truck's main frame) at the output shaft flange and below the pump/motor (Figure 3.11). Similar to the horizontal mount, the pump/motor output shaft is directly connected to the reduction system (sprocket, gear, etc.). However, with this method the reduction must undergo a 90° transmission using bevel gears or some other operation. When mounted vertically, the entire weight of the pump/motor can more easily supported by a support below the pump/motor. Also, the height of the pump/motor is minimized, thus optimizing the space available.

3.4.3 Undercarriage Cradle Support

The pump motor could also be mounted in a fashion similar to the DC motors attachment. Essentially the pump would have some sort of clamp band around the complete pump and then this band would be mounted to a beam from below. This method does not use the mounting locations on the pump motor but is safely supported by the band. This method also places the output shaft in the horizontal position, parallel to the DC motor's current output. Figure 3.12 below shows a rough idea of how this cradling system would work.

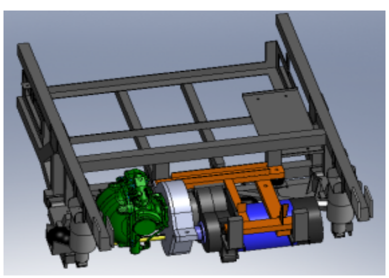


Figure 3.10: Pump/Motor Horizontally Mounted to Frame

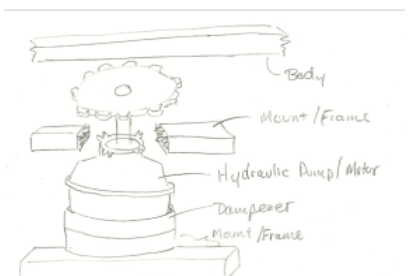


Figure 3.11: Pump/Motor Vertically Mounted to Frame

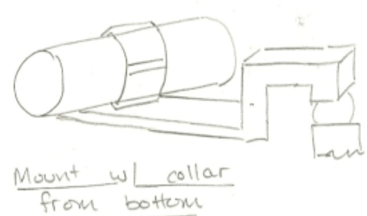


Figure 3.12: Pump/Motor Undercarriage Cradle Support

3.5 GEARBOX SUPPORT

3.5.1 Tabs

The gearbox could be mounted to a sub-frame, which in turn is mounted to the truck's main body frame, using tabs connected to the gear reduction casing (Figure 3.13). The tabs would be bolted to the sub-frame from above or below.

3.5.2 Side Bolt Mount

The gearbox could also be mounted from the side of the sub frame. Instead of having tabs that sit on top of the frame, a bolt would go through the side of the sub-frame and into the gearbox. Figure 3.14 below displays how the gearbox would be mounted.

3.5.3 Gusset/Plate
Gussets can be used to connect beams or plates to load bearing columns (Figure 3.15). Gussets are normally used as supporting members, often in support of truss members. Fastening methods are an important consideration for installing gusset plates [26].

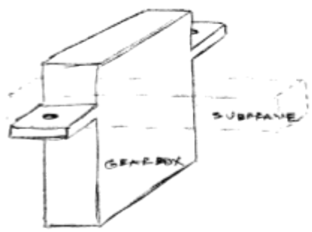


Figure 3.13: Tabbed Gearbox

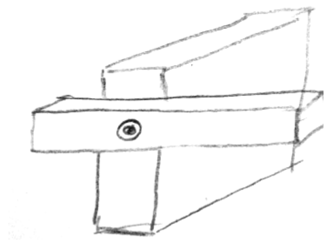


Figure 3.14: Side Bolt Mount

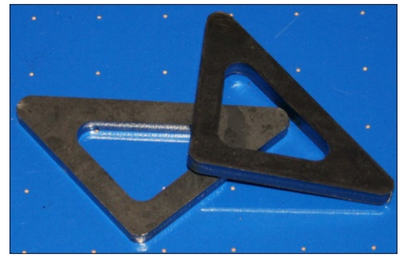


Figure 3.15: Gusset Plates

3.5.4 Tab and Gusset
The tab method combined with gussets demonstrates a separate concept. This concept incorporates the combined characteristics of both the tab and the gusset methods described above. In this method, the tabs and gussets would interact with the case, with the gussets supporting the tabs from shearing off of the case, while the tabs would contain the surface area to hold the loading on the frame.

4 CONCEPT SELECTION

4.1 CONCEPT SELECTION PROCESS
After identifying the four primary functional areas (reduction system, coupling, pump/motor mounting system, and gearbox support system) of our design and developing multiple solutions for each, we evaluated the concept options to determine the most effective choices. Based on our design specifications, which reflect the customer requirements, we quantitatively and qualitatively identified the advantages and disadvantages of our concepts.

We used Pugh chart analysis, with the following methodology, for each functional area to quantitatively review how well each concept fulfilled the customer requirements. The design selection criteria were directly translated from the customer requirements. Each requirement was weighted by importance (from 10 = most important, to 1 = least important), and each concept was rated for how well it fulfilled each requirement (with 5 = excellent, 3 = average, and 1 = poor). However, we neglected the risk and transferability specifications in the Pugh charts for the following reasons. Risk was considered for each overall design by accounting for assurance, assembly, complexity, machining, tolerance, procurement, newness, and material risks of an overall design. We only considered the risk associated with designs that contained all functional areas of a design. Likewise, we did not consider the transferability of each concept because we reasoned that transferability would be the same for each concept. Although it is true that some concepts might be more difficult to describe or document, we believed that these differences would be negligible for selecting concepts. We then obtained a weighted score for how well each concept fulfilled each requirement. These weighted scores were summed for a total score. Finally, we then ranked each concept by their total scores, from highest to lowest.

After ranking each concept for each functional area, we used a morphological chart to qualitatively select high-ranking concepts from each area. The morphological allows us to visualize how each area and concept fit into the overall function of the entire system. Simply because a concept is ranked highest in the Pugh analysis does not mean that it would work the best in the overall system. Thus, we selected multiple concepts from each functional area that we believed would work well together, using the morphological chart (Figure 4.7).

Detailed analysis of the concept selection for each functional area is discussed below.

4.2 REDUCTION SYSTEM
During the concept generation phase, we identified five valid ideas for the reduction system, each of which has advantages and disadvantages. These concepts are qualitatively discussed below. We also used the Pugh chart

below (Figure 4.1) to quantitatively rank the concepts. After reviewing each concept we decided that the roller chain and sprocket with a tensioning system is the strongest.

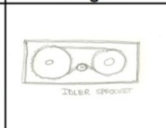
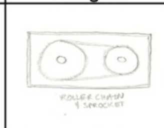
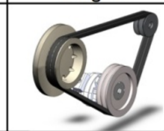
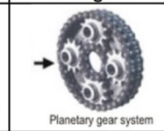
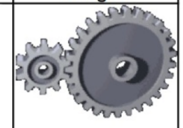
		Design 1		Design 2		Design 3		Design 4		Design 5	
	Sketches										
		Roller Chain & Sprocket w/ Idler		Roller Chain & Sprocket w/o Idler		Belt Drive		Planetary Gears		Spur Gears	
Selection Criteria	Weight	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score
System Transmits Power	10	4	40	4	40	3	30	4	40	4	40
Configurability of Reduction	9	4	36	3	27	3	27	2	18	2	18
Easy to Change	8	4	32	4	32	4	32	2	16	3	24
Components Fit in Frame	7	3	21	3	21	3	21	4	28	3	21
Reliability	5	3	15	4	20	3	15	4	20	4	20
Vehicle Stability / CG	3	3	9	3	9	3	9	3	9	3	9
Total Cost	2	4	8	4	8	4	8	2	4	3	6
Transferability	1	3	3	3	3	3	3	3	3	3	3
Total Score	N/A	164		160		145		138		141	
Rank		1		2		3		5		4	

Figure 4.1: Reduction System Pugh Chart

4.2.1 Roller Chain and Sprocket with or without Idler

The roller chain and sprocket concept with an idler gear or tensioner was the strongest concept after completing our Pugh chart analysis. This is a result of several factors of the design. Roller chains are very durable and can withstand large loads. In addition, they are relatively inexpensive and very configurable because sprockets with different numbers of teeth can be easily exchanged to alter the gear ratio. These attributes are the same for systems with and without an idler sprocket or tensioning system. After researching roller chain systems, it became clear that properly tensioned chains are crucial to the successful operation of the system. Therefore, the roller chain and sprocket with an idler scored slightly higher than the system without the idler.

Of the idler/tensioning options discussed previously, each has advantages and disadvantages. While the idler gear method provides adequate tensioning, the idler uses considerably more space and also is more difficult to configure compared with the motorcycle-style bumper method. It is also easier to install one bumper on each side of the chain to prevent chain slack from occurring no matter which sprocket is driven. Therefore, the motorcycle-style bumper was chosen as the chain tensioning method.

4.2.2 Belt Drive System

A belt drive system operates similarly to the roller chain and sprocket system. The belts themselves are more flexible than roller chains and can achieve the same gear ratios. However, there are several drawbacks to this concept. One of the primary disadvantages of a belt system is that the belt can slip (as described in the Engineering Specifications) [40]. Considering that our system will be under a substantial amount of torque from the hydraulic pump/motor, there is a chance that slipping could occur. Slipping greatly reduces the efficiency of a system, and because a primary goal of all hybrids is to optimize efficiency, using this type of system could be counterproductive. Because slippage is an issue, we also investigated belts with teeth, which would ideally remove any chance of slipping. While this might reduce the slippage, wear on belt teeth is generally worse than that of roller chains [43]. Since we are designing a robust system to test hydraulics on, using a belt system does not seem effective.

4.2.3 Planetary Gear System

A planetary gear system has several advantages versus the other concepts. One advantage is that the input and output shafts are in line with each other. This allows the entire system to remain in line, eliminating the need to offset shafts or span gaps. Planetary gear systems are typically quite rigid since they are based on gear contact interactions. Unlike the roller chain and belt concepts where the input and output shafts are connected by a flexible member, a planetary gear setup has a frame containing the gears and keeps the two shafts rigidly mounted.

Typically, having rigid, inline shafts would be an advantage, but this is not the case in our application. Space is extremely limited in the Xebra truck and as a result, mounting the hydraulic pump inline with the existing DC motor would put the pump too close to the ground with no protection. Planetary gear systems are also not easily configurable for different gear ratios. Since they are a rigid system with precise gear meshing, it is not easy to swap out gears to create a different ratio. Some research has been conducted regarding reconfigurable planetary systems, but cost is an issue as units cost upwards of $2000. This is beyond our budget and the system has few advantages. All of these drawbacks led us to not select this concept.

4.2.4 Gear-to-gear (Spur Gears) System

A gear-to-gear system allows for efficient torque transfer from the input to the output shaft. It also eliminates any issues of slack or slippage since there is only a tooth-to-tooth interaction. Finally, it can accommodate multiple gear ratios by swapping out different gear pairs. However, there are several disadvantages that became apparent when conducting the quantitative Pugh analysis. Meshing gears correctly is a difficult task itself due to the precise alignment required. Additionally, designing a system that allows for different gear sets while maintaining the correct center-to-center distance and gear mesh is difficult to achieve for our system due to the availability of the proper gear sizes. We were unable to find the appropriate gears to fit the requirements requested by our customers and sponsors. These disadvantages, as shown in the "Configurability of Reduction" and "Easy to Change" criteria of the Pugh chart (Figure 4.1), demonstrated that a gear system was inferior to a roller chain and sprocket system.

4.3 COUPLING

To connect the hydraulic pump to the gearbox and the gearbox to the existing transmission, a coupling is needed. We decided to investigate both rigid and flexible couplings. The quantitative analysis is shown below in the Pugh chart in Figure 4.2. After reviewing each concept, we decided that the splined coupling is the most advantageous concept. It has a very low profile, adequately transmits the torque, and can use the existing spline that the pump output shaft has. Initially, it was thought that a flexible coupling would be necessary to connect the pump to the gearbox. This was to prevent any loads or shocks subjected on the system from affecting the hydraulic pump. After further review however, we realized that a rigid shaft coupling with a bearing mount, if properly configured and aligned, would be sufficient to mitigate this issue. The bearings would offload the shaft before any load could affect the motor. Therefore, the strongest solution is the spline shaft. It will also easily transmit the torque output by the motor and will be easy to integrate into the system because the pump motor and DC motors both already have splined output shafts.

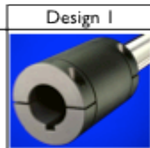
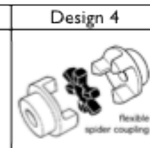
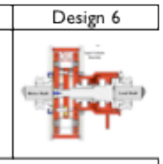
		Design 1		Design 2		Design 3		Design 4		Design 5		Design 6	
	Sketches												
		Rigid Shaft						Flexible Shaft					
		Sleeve / Muff		Splined		Flange		Spider/Jaw		Universal Joint		Magnetic	
Selection Criteria	Weight	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score	Rating	Score
System Transmits Power	10	3	30	4	40	4	40	3	30	3	30	3	30
Safely Transmit Power	10	3	30	4	40	4	40	3	30	3	30	3	30
Configurability of Reduction	9	3	27	4	36	3	27	4	36	3	27	3	27
Easy to Change	8	4	32	4	32	4	32	4	32	3	24	2	16
Reliable Design	7	4	28	4	28	4	28	3	21	3	21	3	21
Components Fit Within Fram	6	3	18	3	18	3	18	3	18	3	18	3	18
Vehicle Stability / CG	3	3	9	4	12	4	12	3	9	2	6	1	3
Total Cost	2	3	6	3	6	3	6	3	6	3	6	3	6
Total Score	N/A	180		212		203		182		162		151	
Rank		4		1		2		3		5		6	

Figure 4.2: Coupling Pugh Chart

4.3.1 Rigid Couplings (Sleeve/Muff, Splined, & Flange)

We investigated three different rigid coupling methods: sleeve/muff coupling, splined coupling, and a flange coupling. All of these couplings provide the same end result: a rigid connection from the input to the output. Each style has specific advantages. The sleeve/muff coupling connects two shafts using the compression between the two halves of the coupling, or by keying the shaft to the coupling. Because the coupling only has to be tightened, this coupling would be the easiest to use. However, this allows some slipping to occur. Keying the shaft requires additional machining time as well.

The splined shaft is another rigid connection. Splined shafts are difficult to machine but are extremely strong and designed to transfer torque. In addition, our supplied hydraulic pump/motor already has a splined output shaft that we could connect our system to. One drawback is that because the shafts are not locked together, they could potentially decouple if enough axial displacement occurred. However, when implemented in a rigid system where there is no shaft displacement, this is unlikely to occur.

The last rigid option is the flange coupling. Due to the size of the flange itself, a drawback of this coupling is that it is larger than the sleeve/muff and significantly larger than a spline coupling. In addition, the shafts to be coupled using the flange coupling must be keyed.

4.3.2 Flexible Couplings (Spider/Jaw, Universal Joint, Magnetic)

We also considered three types of flexible couplings: a spider/jaw coupling, universal joint, and magnetic coupling. The spider/jaw coupling is advantageous because it allows for many different displacements between the shafts while still remaining engaged. The universal joint only allows for angular displacements and all other forces must be transmitted through the coupling. The magnetic coupling has a rigid connection but is flexible in the sense that the coupling can disconnect if over-torqued. Considering that our shafts will all be horizontal and we are integrating our system with an existing horizontal shaft, angular displacement is not a main concern. Thus, the universal joint has no advantage over the other two flexible couplings. The spider/jaw pin coupling and universal joint are moderately sized while the magnetic coupling is quite large which is a significant disadvantage due to our space restrictions. Of all the flexible couplings, the spider/jaw pin coupling was the best at meeting our requirements according to our Pugh charts. This was due to its flexibility, moderate size, and adequate performance. However, we determined that the spider/jaw coupling was still not as strong of a concept as the splined or flange coupling due to its ability as a flexible coupling to safely transmit power when subjected to disturbances.

4.4 PUMP/MOTOR MOUNTING

The hydraulic pump/motor that our system must interface with only has one output shaft and one mounting location (at its output shaft flange). This limits our mounting options. We investigated mounting the pump horizontally, vertically, and using an undercarriage cradle support. Qualitative analysis is discussed below, and quantitative analysis is show in the Pugh chart in Figure 4.3 below. Our analysis concluded that a horizontal pump/motor mounting system would work best. The space constraints as well as the assembly ease associated with a horizontal shaft made it the best design solution.

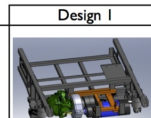
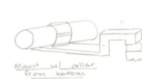
		Design 1		Design 2		Design 3	
	Sketches						
	Description	Horizontal Pump/Motor		Vertical Pump/Motor		Undercarriage Cradle Support	
Selection Criteria	Weight	Rating	Score	Rating	Score	Rating	Score
System Transmits Power	10	4	40	3	30	3	30
Configurability of Reduction	9	4	36	3	27	4	36
Easy to Change	8	3	24	3	24	3	24
Components Fit in Frame	7	4	28	3	21	4	28
Reliability	5	3	15	4	20	3	15
Vehicle Stability / CG	3	3	9	3	9	3	9
Total Cost	2	4	8	3	6	3	6
Transferability	1	3	3	3	3	3	3
Total Score	N/A	163		140		151	
Rank		1		3		2	

Figure 4.3: Pump/Motor Mounting Pugh Chart

4.4.1 Horizontal Mount
Mounting the pump/motor horizontally has several of advantages. It allows the output shaft to be parallel with the DC motor's output shaft, which makes aligning the two systems together much simpler. Horizontally mounting the pump/motor also makes it easier to connect hydraulic lines to the rest of the vehicle's hydraulic system since the flanges will face inwards towards the other hydraulic components. The last main advantage is that mounting the pump/motor horizontally allows for adjustment of the pump/motor's placement within the frame more easily. Due to the dimensions of the pump/motor, this orientation allows for more horizontal clearance between components.

4.4.2 Vertical Mount
While it is convenient to mount the pump/motor horizontally, we investigated a vertically mounted system to eliminate a problem present in the horizontal method. Because the pump/motor must be mounted using a flange near its output shaft, a large moment would be generated at this location if the pump/motor was mounted horizontally. The pump/motor weighs about 60 lbs., which puts a large moment on the mounting pins as shown in Figure 4.4, below. This moment could be reduced using a vertically mounted system. This would require the pump motor to hang from its mount, eliminating most moments and leaving the mount to support the 60 lbs. in the system.

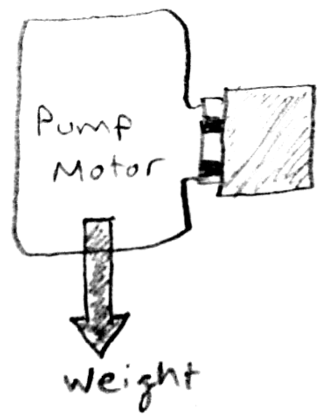


Figure 4.4: Weight of pump/motor produces large moment

While this concept works well for removing the moment, after some discussion it became clear that there would be many issues involving converting the rotation of the vertical shaft into a horizontal shaft. It would involve a bevel gear system or right-side drive. This conversion could work, but ultimately the solutions would be too large to fit within the small space we have to work with. Additionally, after discussing with our EPA and University of Michigan sponsors, it was determined that this would be of little concern. The hydraulic pump/motor was designed to mount solely using the mounting flanges. Therefore, the advantages of the vertical mount were negated.

4.4.3 Undercarriage Cradle Support
The last mounting option we investigated was a cradle support system. This concept is similar to the existing mount for the DC motor, which uses a clamp band to wrap around the casing, which suspends the motor from the top. This method produces a sturdy mount, which is ideal for a system with high torques like ours. However, due to the pump/motor's uneven outer surface and numerous inlet and outlet ports, mounts that make contact with the pump/motor's surface cannot be easily fabricated. There is no even surface to clasp tightly to. Thus, a cradle support as described is not ideal for mounting the hydraulic pump/motor.

4.5 GEARBOX MOUNTING
The final group of concepts discussed was options for mounting the gearbox to the truck's main body frame. Because the gearbox is the mounting point for the hydraulic pump/motor, it is very important that the gearbox be mounted securely. The three major gearbox mounting concepts are qualitatively discussed below and the concepts are quantitatively compared below in Figure 4.5. Additionally, the support tab and gusset concepts were combined as a separate concept because of their complementary attributes. As a result of this comparison, a tab design with gussets was chosen as the gearbox support system.

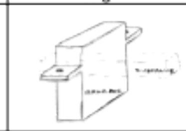
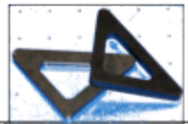
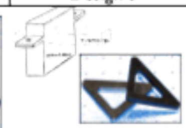
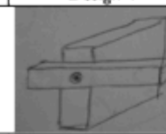
		Design 1		Design 2		Design 3		Design 4	
	Sketches								
	Description	Support Tabs		Gussets / Braces		Support Tabs / Gusset		Bolt from Sides	
Selection Criteria	Weight	Rating	Score	Rating	Score	Rating	Score	Rating	Score
System Transmits Power	10	3	30	3	30	3	30	3	30
Safely Transmit Power	10	3	30	3	30	4	40	3	30
Configurability of Reduction	9	4	36	3	27	4	36	4	36
Easy to Change	8	3	24	3	24	3	24	3	24
Reliable Design	7	3	21	4	28	3	21	3	21
Components Fit Within Frame	6	4	24	4	24	4	24	3	18
Vehicle Stability / CG	3	3	9	3	9	3	9	3	9
Total Cost	2	3	6	3	6	3	6	3	6
Total Score	N/A	180		178		190		174	
Rank		2		3		1		4	

Figure 4.5: Gearbox Mounting Pugh Chart

4.5.1 Support Tabs

The first gearbox mounting method uses tabs that extend from the side of the gearbox and rest on top of the sub-frame. One advantage of using tabs is that they distribute the loads over the whole face of the tab. Another advantage is that because the tabs are wide, they can counter the moments put into the system. However, attaching the tabs to the gearbox itself may be an issue.

4.5.2 Gussets / Braces

Gussets or braces can be applied in various places on the gearbox. Since there is a large moment created by the weight of the pump/motor, a gusset or brace would greatly reduce the moment applied to the other mounting location by distributing the weight across an additional surface.

4.5.3 Support Tabs & Gussets

As described above, support tabs are effective in distributing loads across the entire surface of the tab. Additionally, gussets and braces can further lessen the moments caused by the weight of the pump/motor by distributing the weight across an additional surface. Therefore, a combination of a tab design that is reinforced by gussets would be even more effective as shown in the Pugh chart analysis above (Figure 4.5).

4.5.4 Bolt from Sides

Another concept was to secure the gear casing from the side of the frame by bolting directly into the gear casing itself. An advantage of this method is that it is easy to align the gearbox and the sub-frame, which are attached by bolts. However, a large disadvantage is that no aspect of this concept can counter moments applied to the gearbox. The gearbox would simply twist around the bolts. Another disadvantage is that this concept relies on the shear strength of the bolts to provide all of the support.

4.6 OVERALL DESIGN SELECTION AND MORPHOLOGICAL CHART

As previously stated, the morphological chart serves as a link between the qualitative and quantitative analysis of the functionally decomposed areas of our project. Although the Pugh chart provides a direct ranking of the concepts based on our weighted customer requirements, our qualitative review of the advantages and disadvantages must be considered as well when selecting our overall alpha design. The morphological chart allows us to review how well each highly ranked concept could be incorporated into an overall system design and how well they would work with each other. The morphological chart is shown below in Figure 4.6.

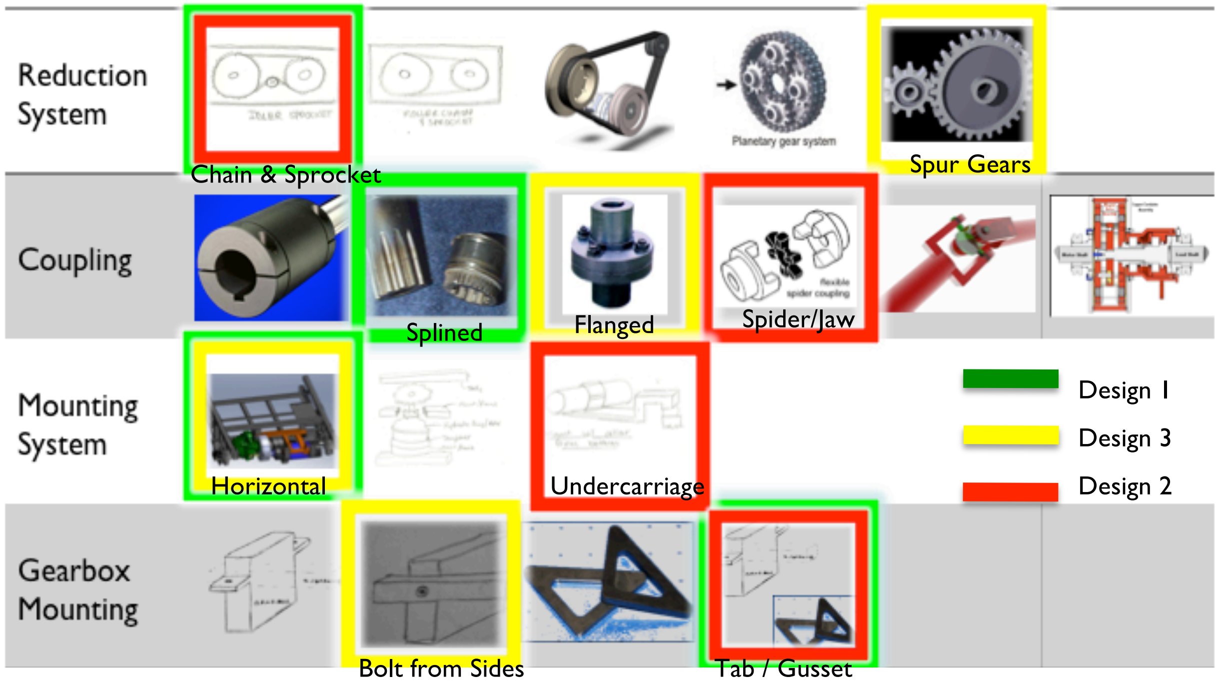


Figure 4.6: Morphological Chart

The morphological chart shows the selections of highly ranked concepts from each of the four functional areas. These concepts were merged into three overall system concepts based on a qualitative judgment of how well highly ranked concepts from each functional area would fit with each other. Using this method, we produced three overall design concepts, which we believed were feasible to produce (shown in Figure 4.6). As seen above, there is overlap involved with the design selections. For example, two designs incorporated the chain & sprocket reduction system, two used the horizontal mounting system, and two used tab and gusset gearbox mounting concept. This is because we determined that each of these designs were capable of creating a successful overall concept and could function well with other concepts from other functional areas.

The three overall designs generated are shown above in Figure 4.6, and below in Figure 4.7. The main advantage to Design 1 is that the entire system is rigidly connected to itself (with fixed spline shafts) and to the sub-frame (with tabs and gussets). Design 2 centers around the flexible spider/jaw coupling that allows for misalignment of internal shafts. The main distinguishing factor of Design 3 is its use of a spur gear reduction system.

These overall concepts were then quantitatively compared using Pugh chart analysis (Figure 4.7, below) with the previously mentioned selection criteria that were translated from the customer requirements. Note however, that the design risk criteria is now included because this Pugh chart includes overall design concepts that embody all of the previously mentioned risk factors.

As a result of this quantitative analysis, we determined that Design 1 would be the most likely to succeed. The scoring margin between Designs 1 and 3 was small. This is because the two designs were nearly the same except for the major difference between the chain and sprocket and spur gear reduction systems. Design 1 scored higher in the changeability, risk, and cost criteria, mainly because of the chain and sprocket system's advantages over a spur gear system.

	Functional Area:	Design 1		Design 2		Design 3	
	Reduction System	Chain and Sprocket		Chain and Sprocket		Spur Gears	
	Coupling	Fixed Spline		Flexible Spider		Flange	
	Pump/Motor Mount	Horizontal		Undercarriage		Horizontal	
	Gearbox Mount	Tab and Gusset		Tab and Gusset		Bolt from Sides	
Selection Criteria	Weight	Rating	Score	Rating	Score	Rating	Score
System Transmits Power	10	3	30	3	30	3	30
Safely Transmit Power	10	3	30	4	40	4	40
Configurability of Reduction	9	4	36	3	27	4	36
Easy to Change	8	4	32	3	24	3	24
Reliable Design	7	3	21	3	21	3	21
Components Fit Within Frame	6	4	24	3	18	4	24
Design Risk	5	4	20	2	10	3	15
Vehicle Stability / CG	3	3	9	3	9	3	9
Total Cost	2	4	8	3	6	3	6
Total Score	N/A	210		185		205	
Rank		1		3		2	

Figure 4.7: Overall Design Pugh Chart

5 DESIGN DESCRIPTION

After distinguishing our most effective design concepts, we created our final design, taking three main iterations of design reviews and concept progressions. To begin, we identified the main components of our design, and then found the appropriate components necessary to accomplish our specified goals. Through this concept progression, more details and part interactions were established, as a baseline assembly plan influenced the construction of our design. To get to our final design, we first created our Alpha Design, demonstrating the overall idea of our entire concept. Then we created the stack-up of our components (every part interaction) and generated our design as of Design Review 3. As a final step, we incorporated engineering changes based on assembly and realistic machining considerations not identified previously to obtain our final design.

5.1 CONCEPT DESCRIPTION

Our final design concept, conceptually broken down in Figure 5.1, accomplishes all of our desired tasks, while remaining as space efficient as possible. In order to effectively integrate a hydraulic drivetrain to create a hybrid system, we interfaced the splined output shaft of the pump/motor and DC gearbox through the use of two splined shafts, connected by the inner components of the reduction casing. The inner components mate the two shafts using a gear reduction system with interchangeable sizes in the lower gearing component to provide three unique gear ratios. The reduction case itself acts to house the gearing components, provide bearing support for the two shafts, and mount the drivetrain on the sub-frame of the vehicle's complete motor system. The case has access panels at the top and side-bottom for the component interchangeability, and also supports these components on the sub-frame.

As shown by the final picture (bottom-left) in Figure 5.1, the reduction case, power transmission system, hydraulic pump/motor, and DC system are all supported by a sub-frame, modified slightly to accommodate the geometry and weight of the additional components. The sub-frame sits on four dampeners, which isolate the components from the motion of the main vehicles structure.

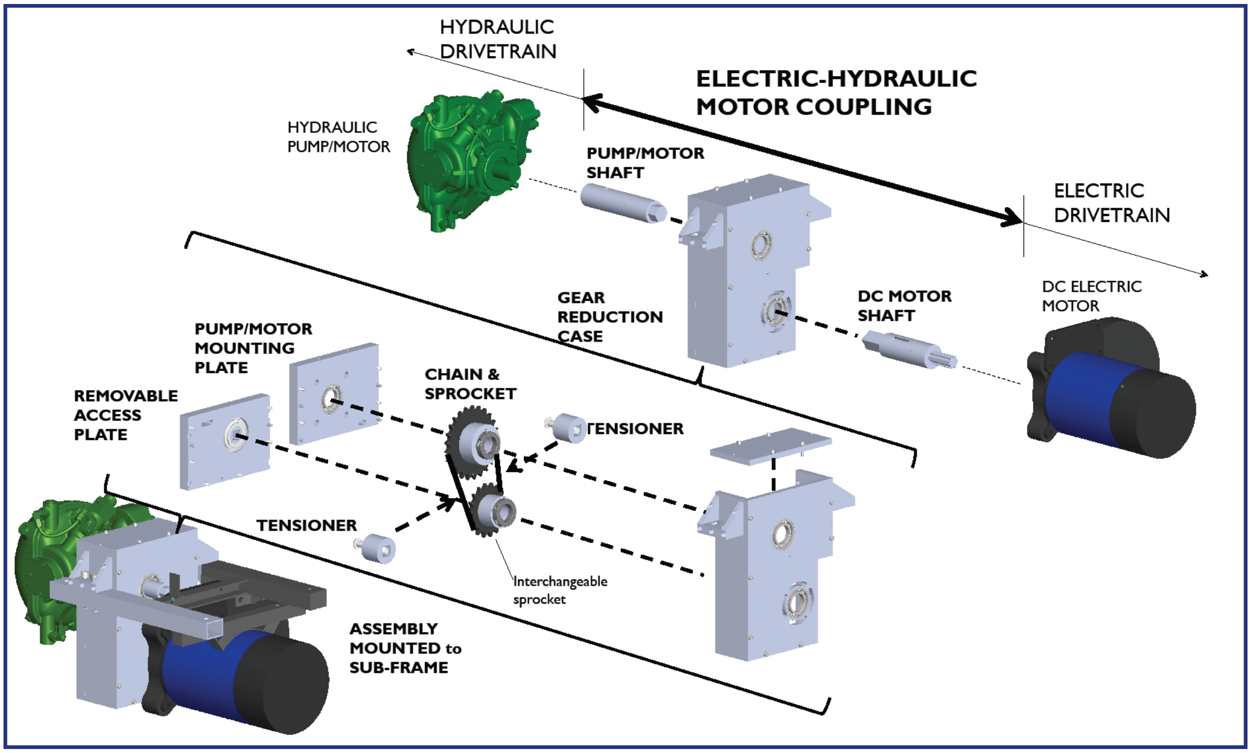


Figure 5.1: Concept breakdown of final design

Figure 5.1 below shows the stack-up of the inner components, which transmit torque between the hydraulic pump/motor and DC motor output. The output of the pump/motor fits with a splined shaft, mated with the gearing component that transmits the torque to the lower system. This lower system mirrors the stack-up of the upper system, in which there is another gearing component secured to the lower splined shaft. The shaft is bearing mounted to the reduction housing and interfaces with the DC motor's gearbox via a spline. This lower gearing component is interchangeable, to match the desired gear ratio range for testing. Figure 5.1 also shows the reduction case's bearing fits, which hold the shafts and offset any radial loading taken by the shafts during performance.

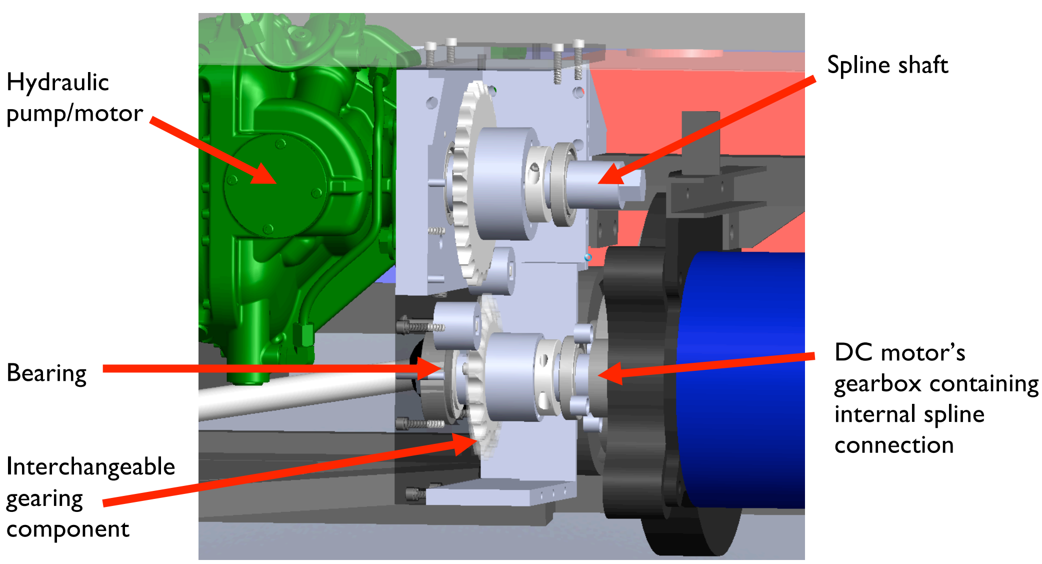


Figure 5.2: Power transmission system component layout

In Figure 5.3, the functionality of the reduction case is shown through its multiple features. The tabs and gussets on each side of the case act to hold the weight of the hydraulic pump/motor and case (including internal components) on the sub-frame. In addition, the transparent sides of the case both identify the accessibility locations in the case, so that gearing changes can be made and tensioning can be adjusted. Finally, the lower shaft's bearing mount is clearly shown, demonstrating the load relief and alignment feature provided by a double bearing mount design.

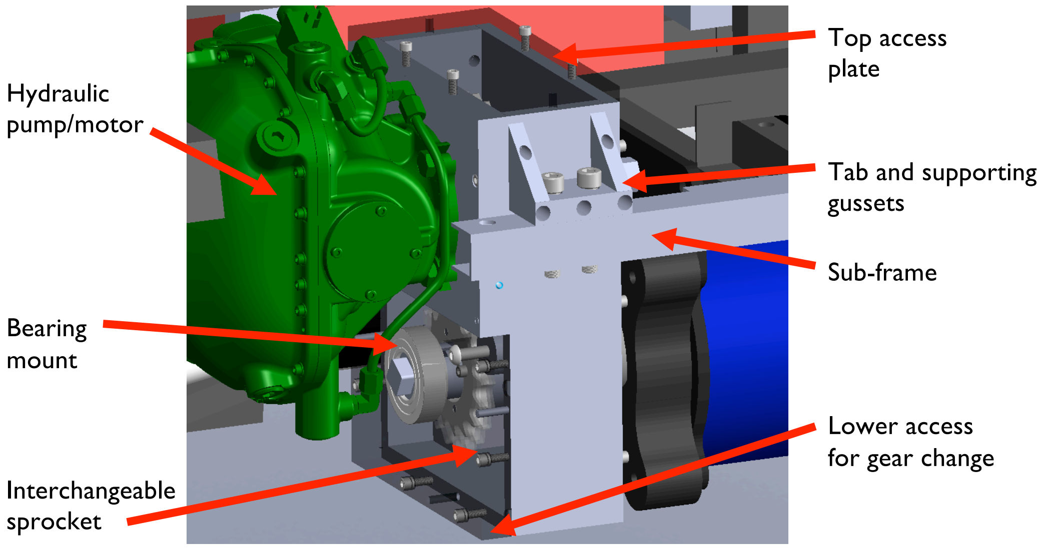


Figure 5.3: Various features of reduction case design

In terms of the sub-frame, which holds not only the additional hydraulic pump/motor and reduction case (through the slotted holes in the crossbars), but the previously installed DC motor and gearbox, our design modifies this structure slightly to integrate the new components. Figure 5.4 shows the framing, which is identical to the previous system, except for the extension of the cross bars shown in light grey to support the reduction case. To do this, two of the sub-frame mounts were removed, re-machined, and welded at different locations to shift with the extended frame dimensions. This framing rests on the preexisting dampeners that isolate the motor components to the motion of the vehicle.

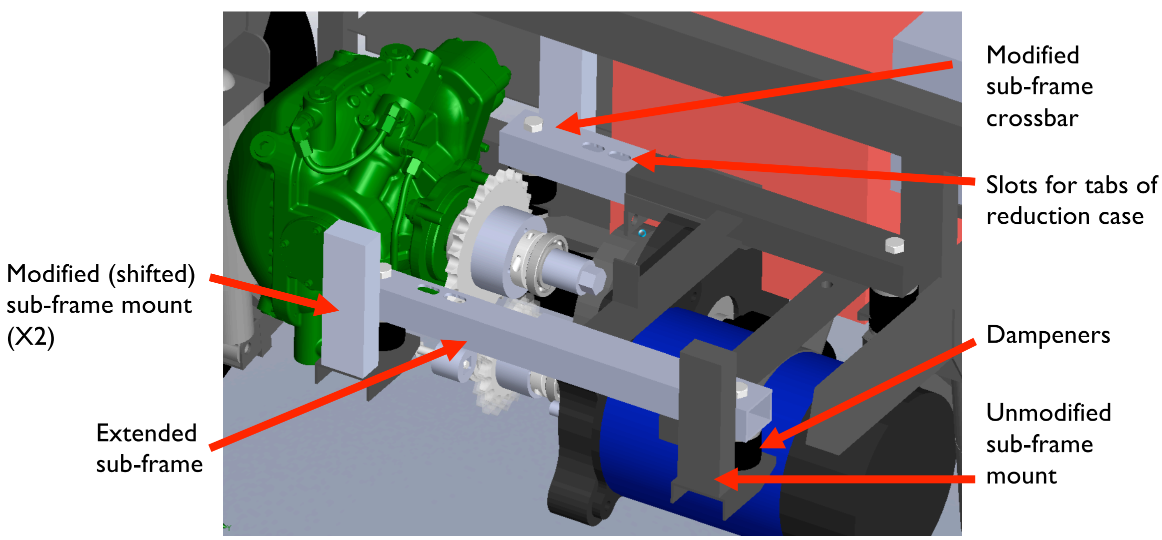


Figure 5.4: Modifications made to frame and frame mounts

5.2 ALPHA (PRELIMINARY) DESIGN

From Design Review 2, to begin development of our generalized concept, we reached our first iteration, Alpha Design. The Alpha Design that we selected was created as a first pass in the process, demonstrating fits and overall component relations. It integrates the best combination of concepts from each of the four functional groups. Although it is possible that the concepts other than the highest ranked may produce a better overall design, this was not true as seen in the Pugh chart for overall designs (Figure 4.7). To couple both the hydraulic pump/motor and the existing DC motor, our design uses a chain and sprockets as its gear ratio reduction system. The use of a chain and sprocket system offsets the center-to-center distances of the sprockets (as opposed to gears) so that the ratios can be easily interchanged. The outputs of the both motors are connected to the sprockets using splined shafts, enabling a rigid contact, which reduces relative motion between the output shafts while aiding with the interchangeability of the sprockets. Externally, a reduction case is designed to hold three different chain and sprocket ratios, and also to allow accessibility to the chain and sprocket from outside of the case. As previously stated, our design is ideally a completely rigid system, with the hydraulic pump/motor mounted horizontally onto the reduction case, allowing the output shaft to face toward the DC motor. The design also requires redesigning the current sub-frame on which the DC motor and the reduction case are mounted. The beam lengths are extended to accommodate the pump/motor and reduction case interfaces. The casing will be mounted on the sub-frame using a combination of tabs and gussets to structurally support the frame. Initially, a supporting brace was added to the design to help counteract specific system torques. Figure 5.5 shows a conceptual mock-up of our design.

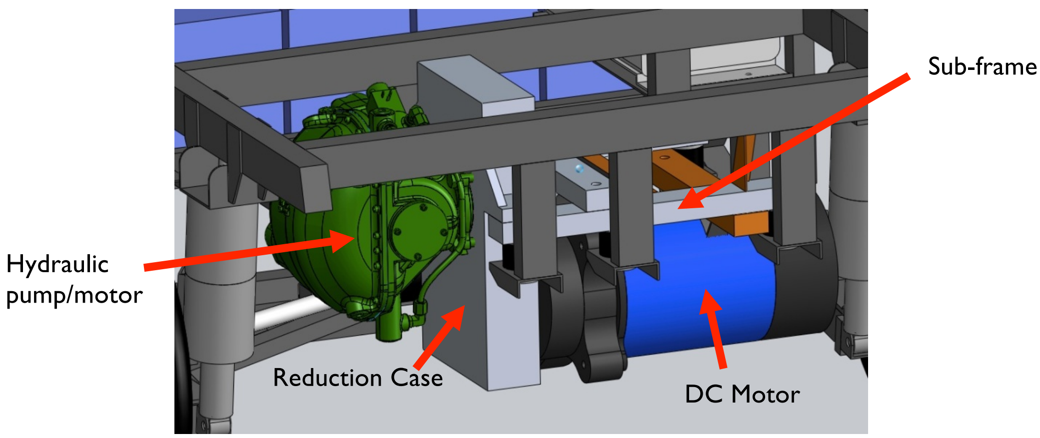


Figure 5.5: Proposed Alpha Design

As shown in Figure 5.5 above, the design mounts the pump/motor horizontally. The pump has been rotated about the output shaft to position the inlet and outlet hydraulic lines towards the front of the vehicle. This allows for a more direct connection to the hydraulic manifold and the rest of the hydraulic components, thus minimizing losses in these lines and increasing efficiency. Additionally, the horizontal mount (as opposed to other considered concepts such as the vertical mount) reduces the number of power transmission joints that the system requires. The pump/motor bolts directly onto the gear reduction casing using the flange containing four mount locations shown in Figure 5.6 below. The pump/motor is supported completely by one side of the reduction casing, which requires careful consideration of material selection and geometry design.

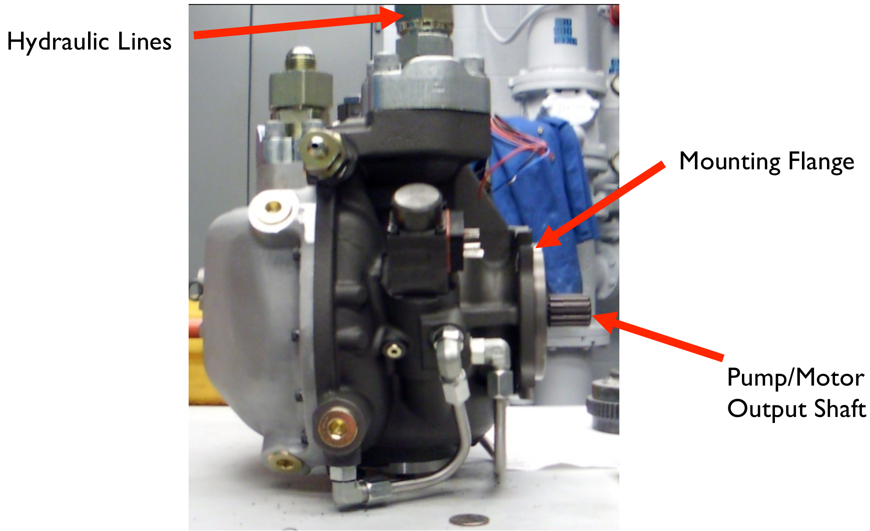


Figure 5.6: Proposed Pump/Motor Assembly

One of our customer requirements was that the system must provide three different gear ratios from pump/motor to the wheel within the range of 2.5:1 to 3.6:1. Our gear reduction must output torque and power to an existing transmission attached to the DC motor, which has a ratio of 4.5:1 from the DC motor to the wheel. So, for example, to achieve a 2.5:1 ratio from pump/motor to wheel, we would have to design our gear reduction to be 2.5:4.5 since it is routed through the DC motor transmission of 4.5:1 creating an effective ratio of 2.5:1. Therefore we have selected the lower sprockets necessary to achieve three ratios of 2.5:4.5, 3.3:4.5, and 3.6:4.5. These sprockets will provide overall pump/motor to wheel ratios of 2.5:1, 3.3:1, and 3.6:1, which were specifically requested by our sponsor. Although 3.3:1 is the theoretical "optimal" gear ratio (from pump/motor to wheel), we are applying the other two ratios to allow for empirical testing of the most efficient gear ratio. To accommodate this interchangeable gear ratio, our design incorporates a roller chain and sprocket power transmission system where one of the sprockets is interchangeable.

The pump/motor's splined output shaft interfaces with the chain and sprocket in the top half of the gear reduction casing, as shown in Figure 5.7. The gearbox itself has a counterbored hole to interact with the pump/motor's mounting flange, and will contain threaded holes for bolts from the pump/motor's mounting points. The output shaft in the gearbox is designed to have a female splined shaft, which will connect with the pump/motor's male output shaft. It is securely mounted onto bearings on either side to completely offload the output shaft from any disturbances in the system. By using a roller chain and sprocket reduction mechanism, the output shafts face mainly radial loads. These radial loads are offloaded by the bearings, while the axial loads will ideally be eliminated because the system is designed as a completely rigid system. All remaining axial loads, of indeterminate value, can be supported internally by the hydraulic pump/motor, which can withstand over 1000 psi of stress/pressure.

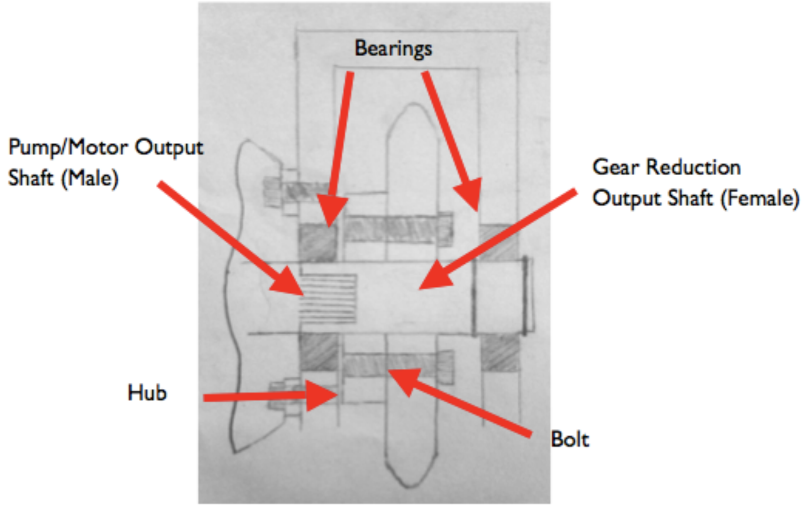


Figure 5.7: Proposed Shaft Assembly

The sprocket is directly mounted onto this upper shaft through a hub that is keyed in two places to the shaft. We selected this configuration of a fixed upper shaft system so that radial loads on the pump/motor would be minimized through the two-bearing system. Alignment issues are also minimized with a fixed upper shaft and sprocket. The selected upper sprocket has 25 teeth and is roughly 0.5" thick. To make sure the sprocket is solidly attached and receives all the torque generated by the hydraulic pump/motor, we used a hub to attach the sprocket to the pump/motor output shaft. As a result of the upper sprocket being permanently attached, any interchangeability of the gear ratios of the system must be achieved by modifying the lower portion of the reduction casing.

While the hydraulic pump/motor's output sprocket is permanently fixed, the gear ratio interchangeability must be present in the lower shaft assembly (Figure 5.4). The lower sprocket must be easily removable for quick replacement to achieve the different gear ratios. To accomplish this, a hub is rigidly attached to the shaft. Different sprockets can be bolted to this hub from the removable side of the reduction casing (the access door shown below in Figure 5.8). Each sprocket is fabricated to have identical hole patterns to be able to all bolt onto a single hub. To ensure the shaft is adequately supported, it has also been designed to use a double bearing system, similar to the upper shaft.

An issue that might arise is the difficulty involved with removing a bearing from one of the shafts to access the sprocket, which has an interference fit between the bearing and the shaft. Furthermore, accessibility to switch out sprockets might be difficult with the double bearing mounting system. Both issues were resolved by allowing the lower half of the gearbox, on the pump/motor side, to be removable (Figure 5.9 below). The lower shaft is split into two shafts that are connected with a splined coupling. One "intermediate" shaft is connected directly to the removable panel and mounted to a bearing in the panel. This shaft is attached to the DC motor shaft that is directly attached to the DC motor output. This method allows for removal of the part of the gearbox containing the splined shaft and bearing, while the hub remains mounted inside the gearbox. The hub bolts directly onto the housing and will be solely used for changes to the gear ratio (sprocket switch).

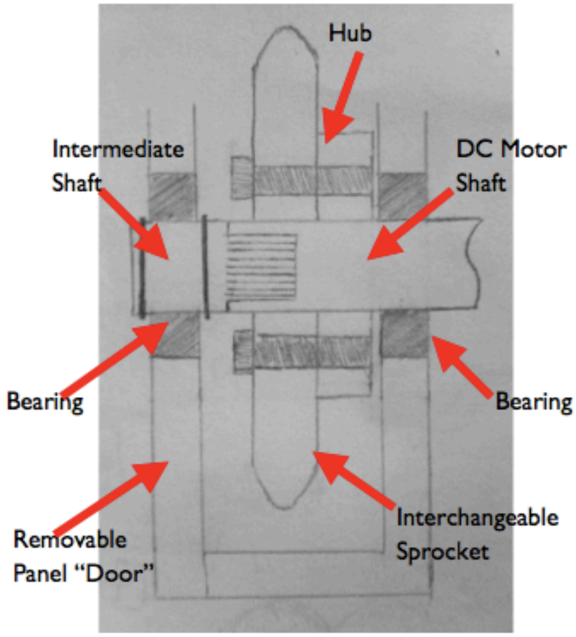


Figure 5.8: Proposed Lower Shaft Assembly

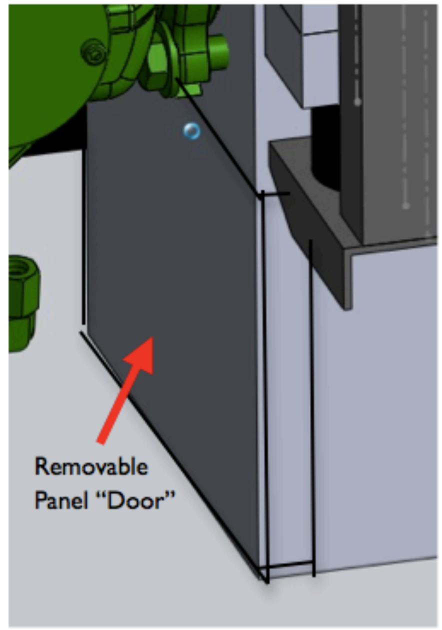


Figure 5.9: Location of Removable Panel "Door" for Lower Sprocket and Hub

We also decided to allow the top of the gear reduction casing to be removable to allow for additional access to the chain and sprocket. In the event that maintenance is required on the chain, reseating the roller chain on the upper sprocket would be difficult to accomplish without access to the top of the reduction casing. With an opening at the top of the case, the chain can simply be pulled over the permanently fixed sprocket to re-engage the chain and sprocket system.

To mount the entire gear reduction case to the sub-frame, we have decided to utilize the tab and gusset concept. This robust design distributes the load from the weight of the components across a large surface area from the tab to the sub-frame rails and dampeners. This design, shown below in Figure 5.10, contains tabs extending off the sides of the gearbox by 2-3" and rests on top of the existing sub-frame rails. These rails in their current design are not the correct lengths and would interfere with some components, so we will be manufacturing a new sub-frame to allow the tabs to be fully supported at our desired location. The sub-frame is the same as the existing sub-frame but will be extended to support our additional components. As shown in Figure 5.10, we selected a tab and gusset system because of the benefits of both types of mounting support. The gussets ensure the tabs are structurally secure and help support the tabs in bending as a result of the weight of the components. While the tabs could withstand the moments due to their strength and geometry, we initially did not want to rely solely on these two methods. Therefore, we designed a brace to counteract some of these moments. Figure 5.11 below shows the location of the brace, on the side of the gearbox opposite the pump/motor. The view in the figure has the sub-frame support rails removed for a clearer view of the brace. The upper crossbar of the brace rests on top of the sub-frame and is bolted into place.

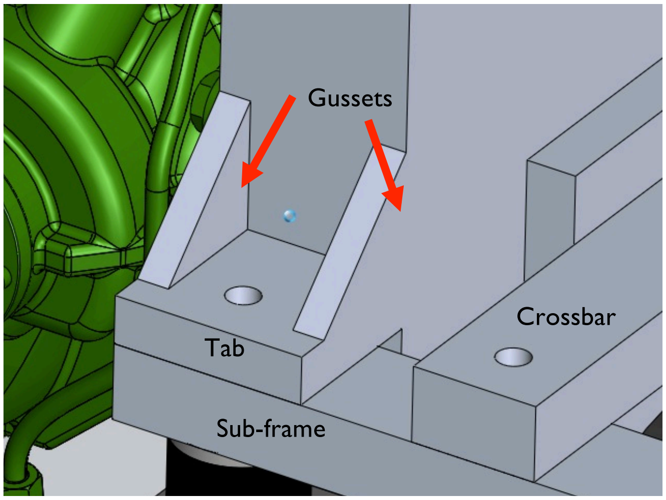


Figure 5.10: Tab Mounting System with Gussets

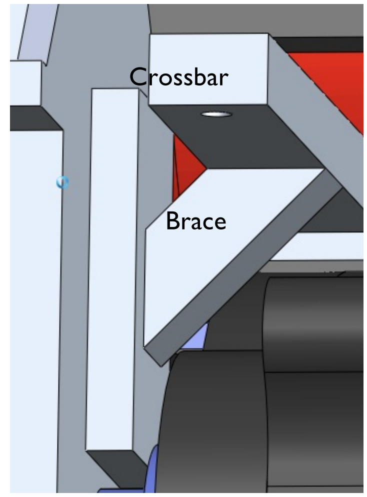


Figure 5.11: Gearbox Brace Mounting Location

With the pump/motor mounting, the gear reduction, the gearbox support, and coupling methods selected, the final important consideration was the system's rigidity. Each of the aforementioned functional areas were bolted or welded together, and thus rigidly connected to each other. The hydraulic pump/motor is mounted to the gear reduction casing, which is internally connected to the DC motor through a splined shaft. In addition to the splined connection that mates the lower shaft of the gearbox to the existing DC motor's female spline port, we added a spacer to connect the DC motor intermediate gear reduction to our gearbox, as shown in Figure 5.12 below. This spacer will protect the shaft, ensure the gearbox does not interfere with the existing transmission, and rigidly connect both reduction casings to prevent relative motion, which could place unwanted loads on our system.

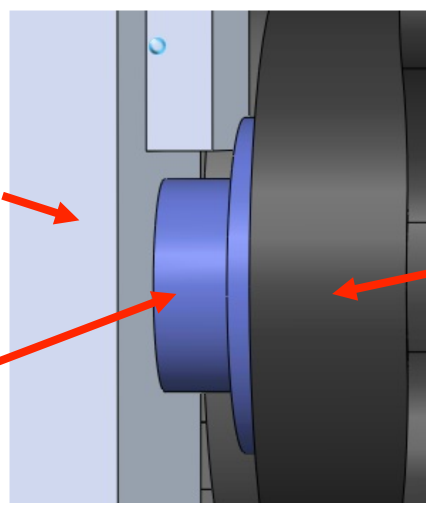


Figure 5.12: Spacer between the gearbox and the DC motor

5.3 ENGINEERING PARAMETER ANALYSIS

Several design parameters were decided upon during the generation phase of our Alpha design without any proper analysis. A detailed engineering analysis was carried out on a number of these parameters to validate our decisions. As a result of this analysis, some design criteria were subject to change, which are explained in more detail below. The purpose of this analysis is also to help determine the dimensions and shapes of certain fabricated components to make sure they meet our customer requirements. It is important to note that all our analysis is carried out under static loading conditions. The dynamic load analysis should be conducted once the entire hydraulic system of the vehicle is designed, and the analysis will summarize an overall result for the end user who will be using both the hydraulic and electrical systems together.

5.3.1 Force analysis on the reduction case mounting

A force analysis was carried out on the mounting system of the reduction case to make sure it does not fail under normal loading condition. We are using tabs along with gussets to support this reduction case. The gussets will enhance the capability of the tabs to withstand the load and stresses due to the reduction case. Thus to design for an extreme case we planned on conducting our force analysis on just the tabs without the gussets. The tab design must be designed to withstand the overall weight of the components it is supporting, the combined load torque applied by the pump/motor and the DC motor and the bending moments due to the pump/motor mounting. It is important to remember that the overall weight the tabs will support include the weight of the entire reduction case, the pump/motor, the bearings in the case, the shafts and the chain and sprockets. This was estimated to be around 150 lbs. To make sure the tabs do not yield due to the stresses applied, we used Von Mises Stress criterion to design for the appropriate thickness of these tabs that can withstand these stresses, which can be seen discussed in more detail in Appendix F.1. The tab dimensions were to be designed for a safety factor of at least 2. Using the stress analysis, we determined that a tab thickness of 0.75 inches would be appropriate to support the reduction system giving us a safety factor of 5.24.

The weight of the reduction case was found using the density of A36 steel and the volume of the entire case. Similarly, the weights of the shafts were found using the density of 1018 steel and its volume. The bearings were estimated to have a combined weight of 4 lbs. Lastly, the chain and the largest sprockets combinations were estimated to have a combined weight of 15 lbs.

The tab is designed such that it is bolted onto the side of the steel plate. Figure 5.13 shows the free body diagram of the reduction case with the resultant loads, moments and torques acting on it in three dimensions. Once we calculated the overall reaction forces and moments acting on the tabs, we began our analysis on the tabs separately. Figure 5.14 shows a free body diagram of the shorter of the two tabs. Using the free body diagram (FBD), we determined the different stresses (normal, bending and shear) acting on the tabs in all three dimensions. We were then able to determine the principal normal stresses acting on the tabs using equation 1. Applying the Von Mises yield criterion, equation 2, using the principal stresses we were able to determine the effective stress, (σ_H), acting on the tabs. Throughout the analysis the thickness of the tabs was inputted as a variable. Thus we designed an excel file with the required quantities and equations where we were able to input different thickness values for the tabs until we met our safety factor requirement of at least 2. The safety factor of the design was determined using equation 3, where (σ_Y) is the yield strength of A36 steel.

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^\wedge 2} \quad \text{Eq 1} \qquad \sigma_H = \sqrt{\frac{1}{2}((\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2} \quad \text{Eq 2}$$

$$\text{Safety Factor} = \frac{\sigma_Y}{\sigma_H} \quad \text{Eq 3}$$

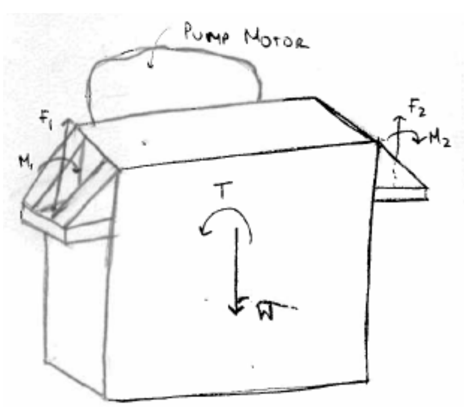


Figure 5.13: Free Body Diagram of entire case

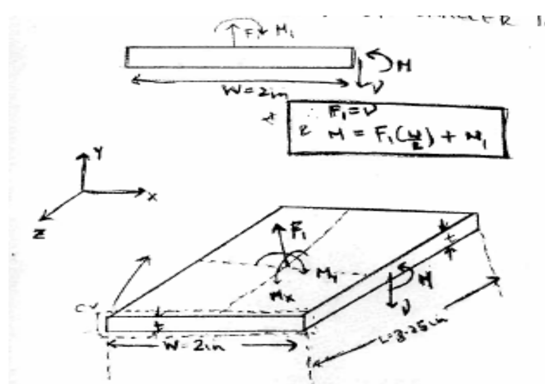


Figure 5.14: Free Body Diagram of the tab

To test for extreme conditions, it was difficult for us to quantify the kind of stress our system would experience. On further discussion with our customer we came to the conclusion that we would test for a bump in the street, which could be modeled for by increasing the weight of the suspended components by a factor of 3. Thus we carried out the same calculations as above with the new weights and we found that the safety factor of the tab is 4.26. Thus confirming that our design can withstand this sudden disturbance in the system.

This mounting system analysis was carried out for a worst-case scenario where the gussets were excluded. The gussets will help counteract the stresses and increase the overall stability of the mounting mechanism. The fabrications of these tabs should be a simple manufacturing process completed on the mill, and joined to the case by welding. With the correct technique and proper planning the fabricated tab should be very similar to the final design generated. Thus making our analysis accurate and acceptable.

5.3.2 Determining the thickness of reduction case plate for the hydraulic pump/motor

The hydraulic pump/motor will be mounted directly onto one side of reduction case using bolts. This wall of the reduction case will be thicker than the remaining walls to account for the flange in hydraulic pump/motor and to also support the bending moment stress due to weight of the pump/motor. It is also important for us to have a plate with a large thickness to have maximum interaction with the bolts used to mount this pump/motor.
The hydraulic pump/motor has an overall weight of 60lbs. When mounted onto the plate it will apply a bending moment, which we plan on resisting by designing the plate using A36 steel and a thickness of 1 inch. We carried out a stress analysis to determine the plate thickness that would be adequate to support this bending moment, which can be seen in Appendix F.2. A safety factor of 2 was applied to the design, which helped determine the appropriate plate thickness to support this bending stress to be 0.12 inches. Thus designing a plate with a thickness of 1inch will be more than sufficient to support the pump/motor.

We conducted an initial informal material selection process using the CES software. We then held discussion with our sponsors regarding the materials that we had short listed to get a more professional opinion through their experiences in industry. We then conducted some research in terms of the prices and availability of these materials, and finally we conducted our analysis using the available options. Thus we picked A36 as it met our overall parameter requirements and was easily available at a reasonable price that met our budget.

5.3.3 Chain and Sprocket Analysis

We are using a roller chain and sprocket system to transfer torque and power through our system. To pick the right combination of chains and sprockets there are a number of factors such as chain length, chain strength and many other factors that play a key role in the selection process and are discussed in more detail below. It is important to note that our team was provided with the required gear ratios. The limitation of workspace area made it even harder to find appropriate sprockets for these ratios and still meet the following criteria. It is important to choose the right sprocket to match a particular chain. Usually they are a function of the type of chain used and by making sure they are of the same pitch.

Center to center distance***:*** Sprockets use a chain to transmit torque, thus it is clear that the center to center distance should be at least greater than the sum of the radii of the outer diameters of each sprocket such that the two sprockets do not mesh. When the sprocket ratios are greater than 3:1 it is necessary that the center-to-center distance should be greater than the sum of the diameters of the two sprockets. In our configurable gear options, our largest ratio is 2.25:1 thus the above rule does not apply to our case. The final design has a fixed center-to-center design that is defined by the shaft locations from the pump/motor and the DC motor. Since our DC motor has a predetermined location, we were faced with the constraints of how high our pump/motor shaft would be. Thus finally the two centers for the sprockets are in one plane but not perfectly in one line, with a distance that is greater than the sum of the radii of the two largest sprockets.

Length of chain: The length of a roller chain is defined in multiples of twice the pitch of the chain, this is important, as the ends must be connected with an outside and inside link. A fairly long chain is preferred in comparison to the shortest one allowed by the sprocket diameters, because the rate of elongation in the chain due to natural wear is inversely proportional to the length. The longer length adds to the elasticity due to the longer strand and tends to absorb irregularities of motion and decreases the effect of shocks. The length of the chain in our case will be limited by the fact that we are designing for three different sprocket ratios. We are planning on purchasing a roller chain with interchangeable links to increase or decrease its length accordingly. Equation 4shows the calculation of chain length where, L = chain length in pitches; C = center distance in pitches; N = number of teeth in large sprocket; n = number of teeth in small sprocket

$$L = 2C + \frac{N+n}{2} + \frac{0.1013(N-n)^2}{4C} \quad \text{Eq 4}$$

Ratio	C (pitches)	N (teeth)	N (teeth)	L (pitches)
2.5:4.5	8.46	25	14	36.80
3.3:4.5	8.46	25	18	38.60
3.6:4.5	8.46	25	20	39.50

Table 5.1: Shows the chain length required for each ratio

Note: Due to engineering changes during the manufacturing phase, there was a new small sprocket of ratio 2.9:4.5 and the calculation for its chain length can be found in the new engineering analysis section.

Alignment: The accurate alignment of shafts and the sprocket tooth is important to provide a uniform distribution of load across the chain width and it also contributes substantially to optimum drive life. In our design we have bearings that will be used for precise alignment of the two shafts.

Lubrication: It is shown that just like journal bearings a separate wedge of fluid lubricants are formed in operating chain joints. Thus to minimize metal-to-metal contact a fluid lubricant must be applied to the joints. Lubrication also provides effective cooling and helps dampen impacts at higher speeds if applied in adequate quantities. It is also important to protect chain drives from dirt and preventing the lubricant from getting contaminated. Our reduction system is installed in a confined casing, which will prevent dirt from affecting the lubricant and the chains. We are planning on lubricating with appropriate non-detergent petroleum based oil.

Chain strength: Chain strength is the main deciding factor of whether the system will fail or not. A chain always fails before a sprocket if both its tensile and working loads are exceeded. The most common rating of roller chains strength is its tensile strength. Tensile strength represents how much load a chain can withstand under a one-time load before breaking. The other important factor is the chains fatigue strength, but this depends on different manufacturing techniques. To determine the strength of a chain, it is important that the chain load should not exceed $1/6^{th}$ of the tensile strength of the chain. Roller chains that experience forces larger than those do tend to fail prematurely via link plate fatigue failure.

We will be using an ANSI 60 ¾" pitch chain in our system. The tensile strength rating of this roller chain is 7,030 lbs. We determine the tensile load acting on the chain, 400lb, using the maximum torque applied due to the DC motor and the pump, and the radius of the fixed sprocket using Equation5below. We notice that this load is less than $1/6^{th}$ the tensile strength load which confirms that our chain will not fail due to loading conditions.

5.3.4 Force analysis on bolts used to mount the hydraulic pump/motor to the mounting plate

As mentioned earlier the hydraulic pump/motor is mounted to the reduction case using 4 bolts. These bolt locations are fixed based on the design on the pump/motor. The hydraulic pump/motor applies a maximum load torque of 206 lb-ft under any condition. The four bolts used to mount the pump/motor will face a shear force due to this torque. We will work backwards in this analysis by determining the shear force acting on these bolts and determining if they are sufficient based on their rated load capacity.

Using the locations of these bolt holes and the maximum torque in equation 5 we can determine the force acting at the bolt locations. Then we divide this total force by four to determine the effective force at each bolt location. We then match this force to the rated force of the bolts we will be using to determine if they will shear. Our calculations determined that the shear force acting at each bolt location is 135 lbs. The bolts that we will be using are designed for 1,375 lb. of shear force, based on the area of intersection. Thus these bolts should be adequate for their purpose without shearing.

$$T = r \; x \; F \quad \text{Eq 5}$$

Where r, is the radius of the bolt locations, F is the shear force acting at these locations and T is the load torque applied by the pump/motor.

5.3.5 Force analysis on dowel pins used to transmit the torque between the upper sprocket and hub the lower sprocket and hub

The upper sprockets used in our design will be mounted on the shaft by bolting onto a hub. We will be using two dowel pins to transmit the torque from the hub to this sprocket. The sprocket on the shaft from the pump/motor will face a larger load torque in comparison to the lower sprocket. Therefore we have conducted a force analysis on the dowel pins used to transmit the torque between this sprocket and hub. We will perform a similar calculation using equation 5 above, and determine the shear force acting on each dowel pin. Our system applies a maximum load torque of 206 lb-ft. In equation 5, r, signifies the location of the dowel pins on the hubs. From our calculations we determined that the shear force acting on each dowel pin is 500 lbs. The dowel pins that we will be using are designed for 12,800 lbs of shear force, based on the area of intersection. Thus these dowel pins should be more than adequate for their purpose.

The lower sprocket and hub will be transmitting a max torque of 165 lb-ft. Since we require this lower sprocket to be interchangeable, we are using dowel pins to transmit torque from the hub to the sprocket but with a clearance fit. Using equation 5 we determine that a shear force of 420 lbs acts on these pins. The dowel pins are the same as those used above and can withstand 12,800 lbs of shear force. Thus we conclude that these dowel pins should be more than adequate for our purpose.

5.3.6 Force analysis of the sub-frame

We designed a sub-frame that will support the DC motor, the reduction case with all its internal components and the hydraulic pump/motor. This sub-frame will be supported by attaching it to the main frame of the vehicle through a dampening system. It is important for us to determine whether the fabricated frame is sufficiently strong enough to hold the components rigidly under normal and extreme loading conditions. Under normal conditions the frame will face a bending moment due to the weights of the various components attached to it. We have conducted a stress analysis on the sub-frame structure to solve for its safety factor, which can be seen in more detail in Appendix F.3. Under extreme conditions, it was difficult for our team to quantify what kind of stress the frame would experience. On further discussion with our sponsor we came to the conclusion that we would model for a bump in the road, in which the weights of the suspended components can be scaled up by a factor of 3. We conducted the same analysis as under normal condition but with larger bending stresses.

Figure 5.15 below shows a free body diagram of one of the two sub frames. In the figure W1 represents the weight of the reduction case with all its internal components and W2 signifies the weight of the DC motor. Each weight is divided by a factor of two, as there are two sub-frames, which is a reasonable assumption for analysis as the two frames are almost equidistant from the center point from where these weights act. R1 and R2 denote the reaction force acting due to the sub-frame mounts. We made the assumption that the bending stress due to these weights will be the largest stresses acting on these sub-frames. We carried out a section-by-section bending stress analysis on beam to solve for the value and location off the max bending stress acting on it. We then used the safety factor equation mentioned above, along with the yield strength of A36 steel, to determine whether the frame would meet the customer requirement of safety factor of at least 2. From the analysis we found that under normal stress conditions our sub-frame has a safety factor of 52, and under extreme conditions it will have a safety factor of 17. Thus the sub-frame will easily meet our safety requirement while supporting the suspended components.

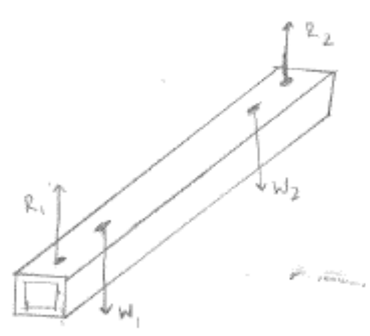


Figure 5.15: Shows a free body diagram of the fabricated sub-frame

5.3.7 Analysis on the keys used to transmit torque through the system

A key is a demountable machined part which, when installed in a key seat, provides a positive way of transmitting torque between a shaft and a hub or vice versa. We require a torque to be transmitted from our pump/motor shaft to a hub attached to it, and then from another hub (for lower sprocket) to the DC motor shaft. We are planning on double slotting each shaft to install two keys to increase engagement. These keys will be subjected to a maximum load torque of 206 ft-lbs. Thus we want to make sure these keys can withstand this load without shearing.

Using the load torque acting due to the pump/motor we can determine the shear force acting on the keys using equation 5, where T, is the load torque and r, is the radius of the shaft. The force will be divided amongst two keys, thus we have a resultant shear force of 3,275 lb-force. Using this shear force in equations 6 and 7 we can determine the shear stress and bearing stress acting on the key, where t, is the thickness of the square key, and l, is the length of the key. We then performed a 3 dimensional stress analysis on this key. Using equation 1 mentioned above we determined the principal stresses acting on the key. We then used the Von Mises stress analysis, equation 2 mentioned above to find the effective stress acting on the key. We wanted to make sure that the key does not shear under the applied load and met our safety factor requirement of 2. The safety factor can be found using the equation 3, where σ_Y, represents the yield strength of high carbon steel the key is made with. Thus the keys in the pump/motor shaft have a safety factor of 4.6, and confirm that they will not shear under this loading condition. These calculations can be found in Appendix F.4.

$$F = \frac{T}{d/2} \text{ Eq 6} \qquad \tau = \frac{F}{t*l} \text{ Eq 7} \qquad \sigma = \frac{F*2}{t*l} \text{ Eq 8}$$

To determine the shear stress acting on the keys in the lower shaft we needed to first determine the effective torque acting on the lower shaft, which is transmitted through the sprocket system. The maximum torque that will act on the lower shaft will occur when we use the gear ratio 3.6:4.5. This torque was found to be 224 N-m. Then we used equation 5 above and radius of the DC motor shaft to determine the shear force acting on the keys. The force will be divided amongst two keys, thus we have a resultant shear force of 2,540 lb-force acting on a key. We then perform the same analysis as above to find the keys in the lower shaft have a safety factor of 6.8.

5.3.8 Analysis on spline fitting

Splines are one of multiple ways of transmitting torque through a system. In our design the hydraulic pump/motor and the DC motor have splined shafts at their outputs. We will be using appropriate male/female splined shafts in our design that will transmit the required torque from the hydraulic pump/motor to the DC motor through our reduction system. We want to make sure that our splined shafts are strong enough to transmit the maximum torque that will be transmitted through the system. In case of splines too, the maximum allowed torque depends on the pitch diameter of the splines, the number of splines, their compressive strength and the length of these splines. To determine the theoretical torque capacity of a splined shaft, we use equation 9 below, where S_C, is the compressive strength of these splines, D, is the pitch diameter of the splines, and l, is the length of the splines. The torque capacity of the splines in the pump/motor shaft is 1,930 lb-ft. The maximum torque that needs to be transmitted through the system is 206 lb-ft. Thus confirming that our splines can safely transmit the required torque through the system.

$$T = 0.4\, x\, S_c\, x\, l\, x\, D^2 \quad \text{Eq 9 [45]}$$

5.3.9 Bearing Life Analysis

The engineering section of the Timken Bearing Selection Guide was used to help validate and determine if our bearings are adequate. In our design, the sprockets are mounted to the 1.5" diameter shafts and supported on both ends by the selected bearings. Because this is a chain and sprocket system, the only loads on the shaft are the result of an equivalent radial load. Figure 5.16 below shows the equivalent radial load, F_B acting on the sprocket.

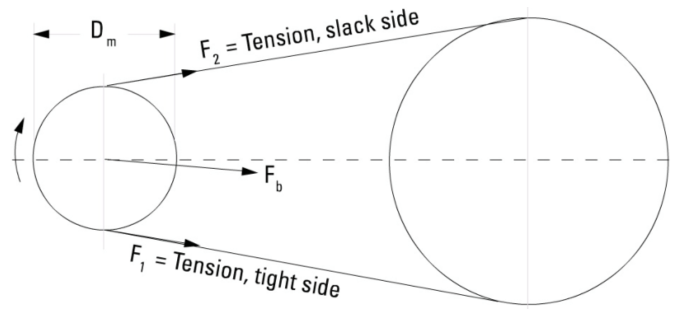


Figure 5.16: Shows equivalent radial load on bearings

This equivalent radial load takes into account the power of the motor (H), the torque (T), the rotation speed (n), and the mean bearing diameter (D_m). Using all of the worst-case values of our system (206 ft-lbs. torque and 2000rpm) we determined the maximum applied load to be 826.58 lbf. Since this is a sprocket and roller chain system, some misalignment can occur between the two sprockets where they no longer operate in the same plane. This off-angle loading could produce an axial force along the shaft. Because we will be carefully aligning the sprockets during installation, we determined that these sprockets could only be misaligned by a maximum of ½".

Considering that the center-to-center distance between the sprockets is around 6", a ½" misalignment would be quite noticeable. This ½" misalignment was resolved into an axial load of 68.83lbf. According to the Timken guide, if the ratio of the axial load to the radial load is ≤ 0.11, the equivalent dynamical load is equal to solely the radial load. This ratio was calculated to be .083 for our design and therefore the equivalent dynamic radial load used for calculation is 826.58lbf.

The L_{10} rating life is the lifetime that 90% of a set of identical bearings will complete before failing. This system is used in many industries including the automotive sector and therefore we were confident in applying it to our design. After plugging in all of our parameters into the L_{10} equation we determined that the three identical bearings in our system would last 94.2 million revolutions (Eq 11) and the one tapered roller bearing would last 326 million revolutions. Using a safety factor of 2, a 3.3:1 gear ratio, and the diameter of the tires we calculated the bearings to be good for a minimum of 14,100 miles. Considering that this is a proof of concept vehicle for the EPA and will be used only for testing for a short period of time these bearings will be more than adequate. The remaining equations can be found in Appendix F.5.

$$L_{10} = \left(\frac{C}{P_r}\right)^e (1*10^6) revolutions, where\ e = 3\ for\ roller\ bearings \quad \text{Eq 11}$$

$$L_{10} = \left(\frac{3{,}761 lbs}{826.58 lbs}\right)^3 (1*10^6) = 94.2*10^6 revolutions$$

5.4 FINAL DESIGN DESCRIPTION

In further conceptualizing the Alpha Design, taking into account manufacturability and performance, we extensively considered specific component interactions in terms of the manufacturing process necessary and the assembly/integration of our design into our available workspace. During this path, we discovered various issues, warranting modifications, which led to our design as of Design Review 3. As we completed machining of our design, engineering changes were needed, which ultimately brought us to our final design.

5.4.1 Identification of Workspace

Throughout this process, in order to distinguish the internal components of the existing Xebra truck from our design, Figure 5.17 demonstrates the available workspace that our team was provided to implement our system. Although an existing DC motor (and associated mount) currently engages the rear axle, we reestablished this mounting mechanism in addition to incorporating the hydraulic drivetrain in this rear space of the vehicle.

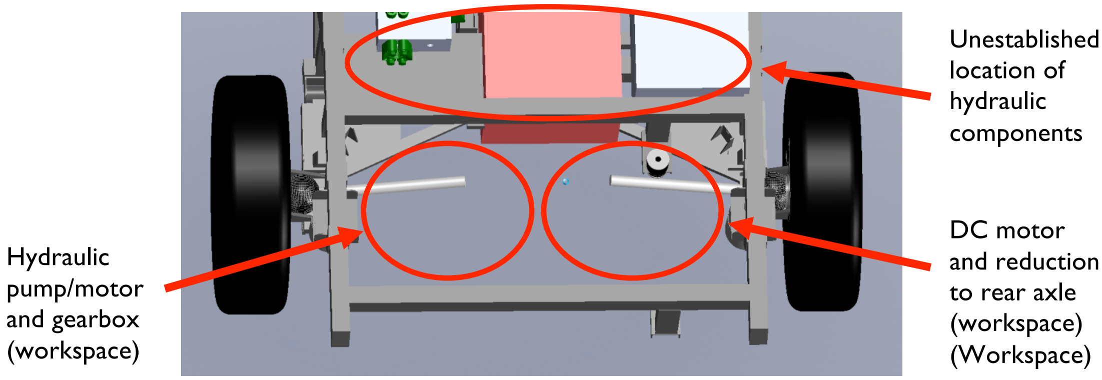


Figure 5.17: Available workspace for our final design

5.4.2 Sub-frame Modifications

In analysis of the sub-frame that holds the DC motor, we retained a majority of its structure and modified its geometry to accommodate the hydraulic system in the available volume. This severely decreased the redesign and machining time associated with completely overhauling an already effective mount. Therefore, Figure 5.18 shows the system of beams used to mount the old DC motor, its gearbox, and the new pump/motor and reduction case.

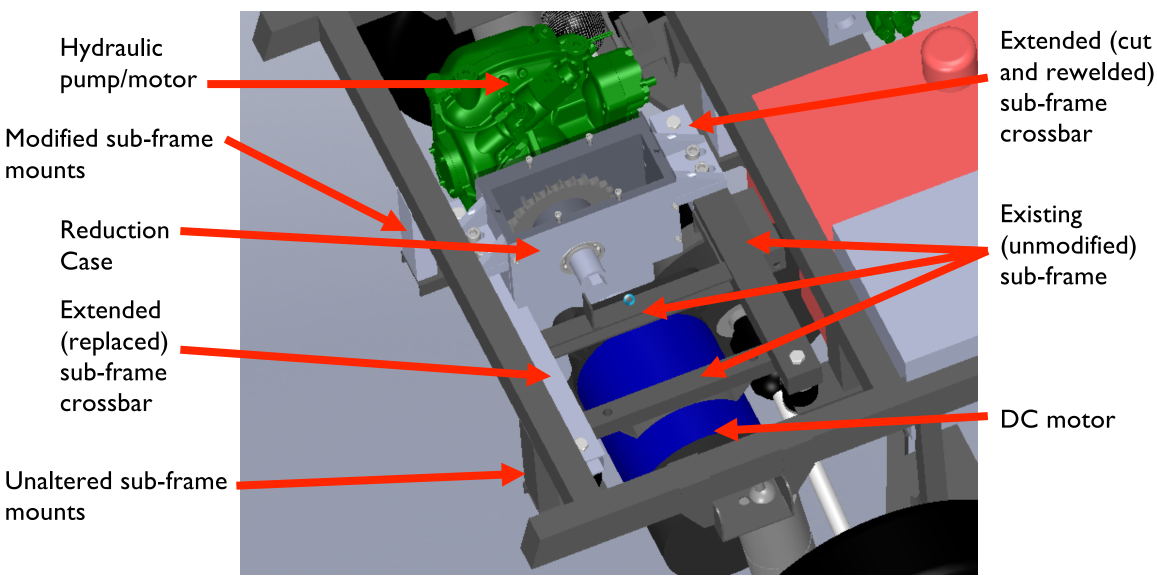


Figure 5.18: Shows the fabricated sub-frame location to support the components

On the crossbars of the vehicle's body, four L-shaped sub-frame mounts were originally welded to the beams to hold the complete motor layout. As shown by Figure 5.19, the dark grey mounts on the right remained at their previous positions, while our design altered the location of the left two light grey mounts so that the hydraulic pump/motor and reduction case could be properly installed. This change was made because the lengths of the support beams, which run across the Xebra, were extended to a final length of approximately 19". Other than this minor adjustment, engineering analysis of the structural stability of the support beams and its ability to hold increased load from the hydraulic components showed that the beams (made of A36 Steel) safely hold the loads applied by the components. So, we mimicked the general sub-frame system of the original Xebra for our design.

For our design's extension of the sub-frame and mounts, we machined parts of the sub-frame beams (cutting to length and drilling holes) and mount beams, as opposed to our original plan to reconstruct the entire sub-frame. After speaking with Bob Coury (University of Michigan Engineering Technician) at the start of machining, we gauged how much material from the sub-frame and sub-frame mounts could be retained from the original frame. To reduce any unnecessary machining of already existing components, we hoped to avoid machining and re-welding the entire frame by simply adjusting the current structure. Because the sub-frame mounts could not be cut without eliminating a portion of its length, as of the machining process, we removed the mount from the vehicle's main body, and kept the lower segment to be reused (Appendix B). Using the same stock as the prepared A36 steel for the sub-frame modifications, we reestablished the correct mount lengths and re-welded the mount (beam and lower segment) at the desired locations. For the sub-frame, we were able to retain a majority of the framing, so as to only deal with the extended lengths of the crossbars (shown in light grey in Figure 5.18). To complete this, we removed the entire back crossbar and one end of the front crossbar, machined the longer replacements, and re-welded the bar stock to the original locations. The engineering change on these crossbars also included slotting the holes for the tabs to introduce adjustability in the event our previous measurements were not accurate (Appendix B). Finally, a concluding engineering change on the back crossbar required us to machine .5" into approximately 8"of the bar during due to interference of the reduction case during assembly.

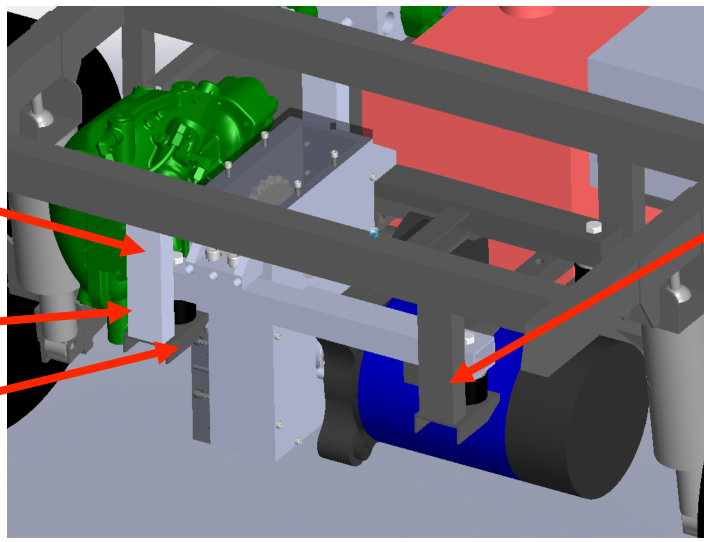


Figure 5.19: Shows the position for the mounts of the sub-frame

Another component that remained consistent in the design was the dampening material/method used to dampen the motion felt on the motors. With the new distribution of weight given the addition of the hydraulic components on one side of the frame, we analyzed the available options (cylindrical stock rated to the specified weight), which included a similar system as the current dampeners and one that does not use a through bolt to secure the components. However, after speaking with Bob Coury to determine the appropriate material, we decided to retain the previous materials, as he stated they are typically overrated and capable of holding significantly more weight than the original system plus our added components.

5.4.3 Power Transmission System

In terms of the power transmission system of the vehicle shown in Figure 5.20, we followed through with the Alpha Design concept of using a chain and sprocket to allow for easy adjustability of the gearing ratios. From the pump/motor, the male spline of the pump/motor's output shaft interacts with the female spline of the upper shaft that we machined. This shaft was turned down by our team, and sent out to a third party (Riverside Spline & Gear) to be broached. The shaft is keyed 1.5" in length in two locations, which allows the keys to transmit torque to a hub, where the sprocket is bolted. Torque is then transmitted through two dowel pins to the sprocket, where the chain carries the motion into the DC output system. The use of dowel pins is primarily to allow the dowel pins to withstand the shear during activity, rather than the threading of the bolts that secure the sprocket. As shown in Figure 5.20, both shafts were bearing mounted at two locations (walls of the reduction case not shown) to provide radial relief to any loading seen on the shafts.

The same keyed shaft and hub mechanism is applied to the lower shaft (attached to the DC motor), however significant changes to the design were made since Design Review 3. Originally as of our presentation, we developed a system using two splined shafts attached to the DC motor's female spline output in series. This was to allow for accessibility of the lower sprocket, where one of the shafts and the door would disconnect for clearance to configure the gear ratio. After presenting though, our sponsors preferred an alternative route, as they were concerned with the multiple spline interactions and the structural stability of outputting rotation through many joints.

Thus, our final design only uses one long shaft, and instead of the rigid connection through a ball bearing pressed into the wall of the access door, the shaft rests in a tapered bearing, which gives the freedom of a disconnecting joint between the inner and outer race of the bearing. Therefore, the access door can be removed, taking with it only the outer race and leaving space to replace the lower sprocket. As of the machining process however, we became aware of another issue, as the inner race of the bearing ran interference with sprocket. To resolve this, we machined an additional shaft cap with an internal square to be pressed into the inner race of the tapered bearing. By using John Mears' (University of Michigan Instrument Fabrication Specialist) CNC, we machined the cap and an external square on the shaft to provide a slip-fit for the tapered bearing's inner race/shaft cap assembly to be removable from the shaft, so the sprocket can still be interchanged.

Also, the hub on the lower shaft (DC connection) utilizes a split-fit between the dowel pins and the sprocket to transmit the torque to/from the sprocket, which allows for proper transmission but still provides configurability though its removability. This design was a significant change from our original concept, which contained multiple step downs to house the key, hold the sprocket, and also align the hub on the bearing. With the elimination of the multiple shafts in series, this concept was simplified to contain a larger portion for key transmission and sprocket bolts, and a smaller portion for alignment (Appendix B). Overall, a majority of these design details were only considered after Design Review 2, when we evaluated, in detail, the assembly and manufacturing feasibility of the design.

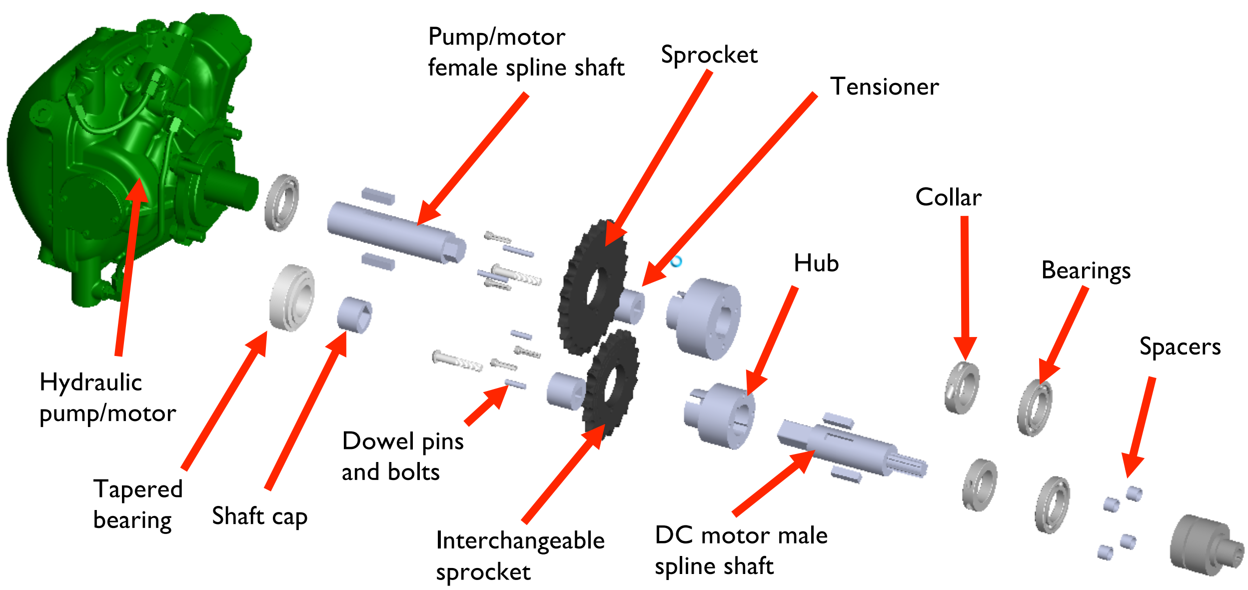


Figure 5.20: Exploded view of power transmission layout

In terms of the various parts shown in Figure 5.20, although a majority of them were purchased (keys, ball bearings, ¼"-20 bolts, ¼" dowel pins, collars), certain crucial components were machined. In particular, the shafts, hubs, and tensioners were machined by our team using 1018 Steel round stock (shafts and hubs) and UHMWPE (ultra high molecular weight polyethylene plastic tensioners). The sprockets were purchased, but bored out to fit the hub diameter specified and drilled for bolt and dowel pin placement.

For some of the particular features shown in Figure 5.20, additional machining and components were done for performance/testing purposes. On the pump/motor shaft, an external hex was machined by John Mears so that a torque wrench could be used for torque testing of our system. Also, this shaft was extended an extra 2" from the original specifications for future installation of an optical encoder (Xianke Lin). For the tensioners shown, we cut to size and bored through for the securing bolt. John Mears then used his CNC to mill an internal square to the round stock of UHMWPE so that a square nut could be used to lock the position of the tensioners and avoid rotation during performance. The collars shown are solely used to reinforce the position of the hubs and avoid any drift of the hubs to rub against the walls of the reduction case.

5.4.4 Reduction Case Design

In regard to housing both the transmission system as well as the hydraulic pump/motor, we developed the concept from Design Review 2 and modeled the casing with structural, assembly, and manufacturing concerns in mind. We first gathered different methods of machining the case and assembling, including milling it out of a large stock, or assembling the plates of the case using dowel pins and bolts. In terms of feasibility, environmental considerations, and time, we chose to machine plates that would be assembled using dowel pins for alignment. This option gave us flexibility in regard to our manufacturing plan, and was also more accessible in terms of purchasing the stock. Our end result was the case shown in Figure 5.21-22, which has a top and side access door to reach the interior components, tabs to engage the sub-frame for mounting, and gussets for additional support at the tab-to-wall joint. Also included as of our Design Review 3 presentation was an extended plate to be welded onto a wall for increased thickness at the pump/motor mount location. As of Design Review 2, another feature was created, as a cutout from the case was needed so that it could clear the existing gearbox of the DC motor.

When we described our model in Design Review 3, the internal concerns with the transmission system (multiple shafts and alignment) caused modifications to the case. In resolving the lower shaft concern regarding multiple splined shafts in series, the addition of the tapered bearing created necessity for a thicker plate on the pump/motor side of the case. So, instead of welding yet another plate to the case, the final design uses 1" thick A36 Steel on the pump/motor side to support both the pump/motor as well as house the tapered bearing. Because of the added weight of the increased thickness, we changed the thickness of the tabs to meet our safety factor concerns for yielding, so the final tab thickness changed to .75". The remaining components of the gearbox are made of .5" A36 Steel stock. These changes can be seen in Figure 5.22, which is an accurate model of the revised design (14" by 9.75" by 4.75"). The final design still utilizes two access plates, a tab and gusset system, and dowel pins/bolts to secure and align the different plates. The plates on the pump/motor side have slotted holes to adjust the position of the tensioners depending on the gear ratio desired (Appendix B).

A significant concern we discussed throughout was the alignment of the bearing holes and how to accurately machine these so as not to create initial loading from the plates of the reduction case. We had numerous conversations with both Bob Coury and our EPA resources concerning this issue. As a resolution, we sent out this portion of the machining to a machine shop (Lidell) capable of the accuracy we need. The hole was rough drilled after case assembly, so that one pass would drill through both plates. The holes were then indicated and bored to the correct size as a finish. This eliminated the inaccuracies associated with removing the work piece and reestablishing any reference locations from a datum surface. Although completed with relative ease, we found some design modifications made by Lidell that were not in the specified drawings we supplied them. They had concerns regarding welding the tabs and gussets to the case, so they instead bored bolt holes at 5 locations on each side and bolted the tabs/gussets to the case using 3/8"-13 bolts (Appendix B). Furthermore, they missed drilling some of the bolt holes, which luckily had no structural impact on our final design (Appendix B).

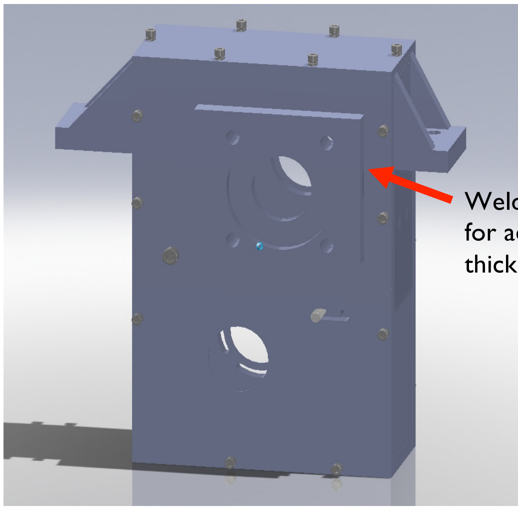


Figure 5.21: DR 3 Reduction Case Design

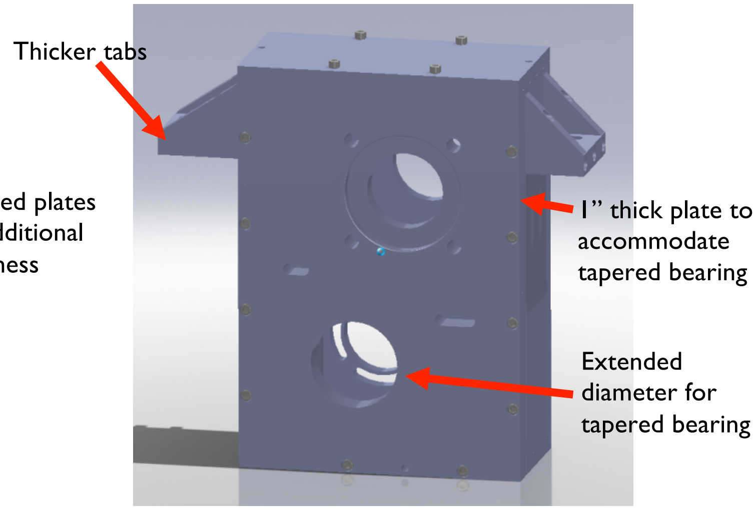


Figure 5.22: Prototype reduction case design

5.4.5 Overall Layout and Component Interaction

To rigidly secure the reduction case relative to the other components at the rear of the vehicle, we conceptualized different means of accomplishing this. As of Design Review 2, a truss extended across the sub-frame crossbars and rested on the DC motor side of the case. In the event of bending/deflection of the case in relation to the sub-frame, this would counteract the motion. However, the difficulty in machining far outweighed its benefits, as it did not reach low enough on the case to counteract the motion. Therefore, we decided to create a four-part spacer that would join the DC motor's gearbox with the reduction case of the pump/motor. These spacers are bolted around the DC motor's output shaft, and create a rigid connection between the two cases (Appendix B). This accomplishes the same objective as the truss support, but is space efficient and more effective (Figure 5.23).

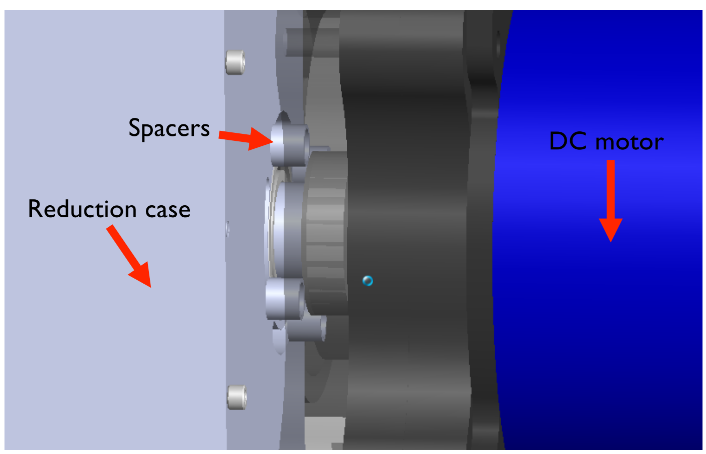


Figure 5.23: Spacer interaction with DC motor's

Figure 5.24 shows the different internal components, and how they fit into the support structure around it. Specifically shown, the two cylindrical pieces bolted to the wall of the case. These UHMWPE plastic tensioners, are bolted and act on both sides of the chain loop to create tension when either sprocket acts as the driver. The slotted bolt placement allows for adjustable tension for the different sprocket sizes.

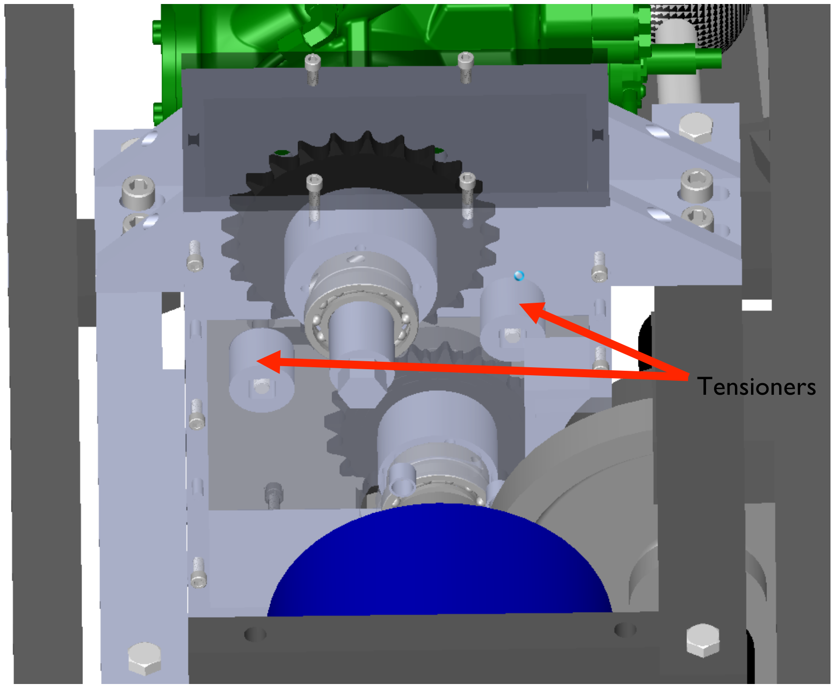


Figure 5.24: Layout of the internal components in the reduction case

Figure 5.25 demonstrates our completed assembly as it is fully installed into the Xebra truck. Both the DC motor and hydraulic pump/motor actuates the rear axle, although in reality the hydraulic components will not be installed until after our project. Appendix G shows the engineering drawings related to all parts to be machined.

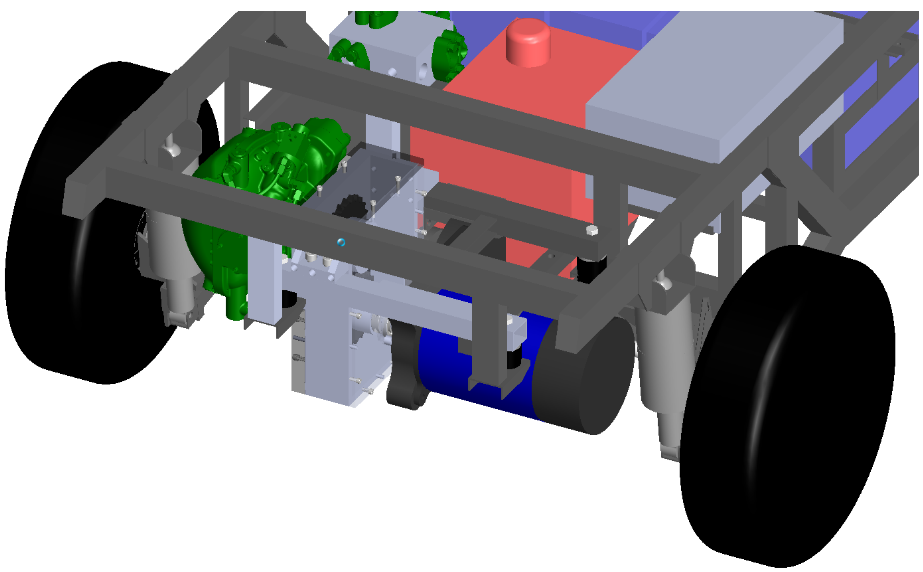


Figure 5.25: Represents the overall layout of our prototype design with the available workspace

5.4.6 Additional Engineering Analysis

5.4.6.1 Force analysis on the bolts used to mount the tabs and gussets to the reduction case
The tabs and gussets as per design review 3 were going to be welded to the reduction case. But due to concerns of deflection in the material due to welding, we decided to use bolts to mount the tabs and gussets onto the reduction case. The reduction case along with its internal components (shafts, sprockets, hubs, collars) and the hydraulic pump/motor has a combined weight of about 150 lbs and are supported by these tabs and gussets. Figure

5.26 shows a free body diagram of one of the tabs and shows the direction of the shear force acting on it. We then use equation 12 to determine the overall shear stress acting at each bolt location. "A" represents the area of the bolt locations and F is the shear force that acts at these locations. Please refer to Appendix F.6 for a more detailed calculation. Using the above calculations we determined the shear stress acting on each bolt is 326 psi. Typically a bolt is rated to withstand shear stresses of about 60% of their tensile strengths. The bolts that were used have shear strengths of 108,000 psi. Thus we can conclude that these bolts will be adequate to support our design under various conditions.

$$\tau = \frac{F}{A} \quad \text{Eq 12}$$

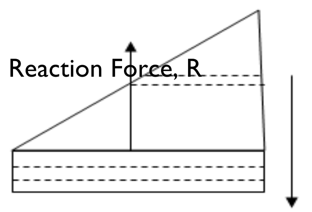


Figure 5.26: Shows the shear force acting on the tab

5.4.6.2 Force analysis on Hex end of the pump/motor shaft

To validate our final design, we needed to test our system under the influence of a load torque. We determined that the maximum torque that the system can experience is 206 ft-lb (2472 in-lb). To apply this torque we decided to use a torque wrench test, which is explained in more detail in the validation section of this report. We designed a hexagonal shape cross section of 0.65" length at one end of the shaft to apply this torque using a wrench and a force gauge. The hexagon has a 0.5625" side, and we wanted to determine if it could withstand this max load torque without failing under torsion. We used equation 13 below to determine the maximum torsional moment (T_{MAX}) that this hexagonal shaft can resist, where "b" represents the side of the hexagon, and τ_{max}, represents the maximum shear stress that acts on this cross section. In order to determine this shear stress due to torsion, we modeled the shaft with a circular cross section that would fit within the side of the hexagon (0.5" radii). This will act as a worst case as the overall material over which the shear stress acts is less compared to actual and the assumption seems reasonable for the sake of analysis. We used equation 14 to determine the shear stress due to torsion, where T, represents the maximum torque that acts on the system, r, is the radii of the circular cross section (0.5 in), and J, represents the polar moment of inertia of the circle. Thus we determined the hexagonal cross section shaft can withstand 6892 in-lb of torque which is much larger than the torque we wish to apply to validate our system and thus it will not fail under the applied torque.

$T_{max} = \left(\frac{1}{1.09}\right) \tau_{max} b^3$ Eq 13 [45]

$\tau_{max} = \frac{Tr}{J}$ Eq 14

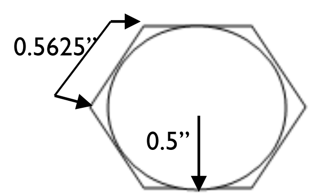


Figure 5.27: Shows the cross section of the pump/motor shaft

5.4.6.3 Deflection in DC and the pump/motor Shafts under the influence of torque

Torque is transmitted between the DC motor shaft and the pump/motor shaft using a chain and sprocket system. In order to transmit torque most effectively under the influence of disturbances, it was important that the chain slack should be less than 2% of the center-to-center distance between the two shafts. The measurement method of this chain slack was a concern, and what we needed to determine was whether we needed to measure the slack under the influence of a torque being applied to the system. We felt this was important as we wanted to determine if the deflection between the shafts due to the applied torque would actually affect our chain slack. To calculate the deflection on these shafts we modeled the shafts as simple supported beams that are loaded at the chain location. Figure 5.28 below shows a free body diagram of the pump/motor shaft, where you can see the force due to the chains and the reaction forces due to the reduction case acting at its ends. The DC motor shaft has similar loads acting on it but in opposite directions. For a more detailed calculation of the deflection in the shafts please refer to Appendix F7. Using bending analysis as shown in the appendix we were able to determine the bending at the point of the chain location in the pump/motor shaft and the DC motor shaft to be around 8.82×10^{-5} inches. This deflection is extremely small and as a result of this analysis we concluded that we could measure the chain slack in the system without the presence of an external torque.

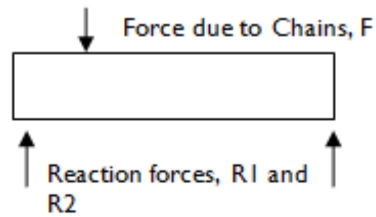


Figure 5.28: Shows the free body diagram of the pump/motor shaft

5.4.6.4 Chain Length for the new smallest sprocket
The smallest sprocket in our configurable gear ratio options was subjected to a change in size, as we had overlooked the interference between the chain and the hub. This engineering change is explained in more detail in the engineering changes section in Appendix B. As we will be using a new sprocket, our new smallest gear ratio is now 2.9:4.5 and we use equation 15 below to determine the new chain length required for this ratio.

$$L = 2C + \frac{N+n}{2} + \frac{0.1013(N-n)^2}{4C} \quad \text{Eq 15}$$

Ratio	C (pitches)	N (teeth)	N (teeth)	L (pitches)
2.9:4.5	8.46	25	16	37.66

Table 5.2: Shows the chain length for the new smallest sprocket

5.4.7 Prototype Design Considerations
In the case of our project, there is little to no distinction between final design and prototype. Due to the fact that our project is meant to facilitate testing of the hydraulic technology as applied to a specific vehicle, large-scale production is not of concern nor is applicable to this situation. Therefore, in considering our prototype design, we refer to our final design description since they are essentially the same. As a "one off" test fixture to be used by the EPA, our design is specified for these circumstances. Our original plan/the ultimate goal of the semester's completion was to integrate our mount, casing, and gearing into the current system. Although hydraulic components were not available to be fully integrated for our components to perform test, this portion is possibly the closest our project will come to full-scale assembly and integration into vehicles. In addition, the fabrication plans for both our prototype and final design are identical because of this fact. Any design changes we see as beneficial that were not already applied to the final design/our prototype will be discussed in the Recommendations section from a hindsight perspective.

5.4.8 Final Bill of Materials
The bill of materials is a complete list of materials that was required for the completion of our project. A significant number of the materials were sourced from McMaster Carr. We were able to find all of the required sprockets, bearings, collars, dowel pins, bolts and nuts in their catalog at acceptable cost. The sprockets, bearings and collars were the essential items to be purchased from McMaster though. The rest of the hardware for assembly could be purchased from various other retailers such as Stadium Hardware. Even though McMaster typically ships extremely fast, we purchased some extra non-essential items at Stadium Hardware for ease of access at comparable prices.

Our bill of materials also includes costs for raw materials from ALRO Metals, a local metal supplier. For small metal purchases like ours, ALRO does not provide exact cost estimates before purchase. They provide a small range of where the cost, but no exact value. The main reason this occurs is because of what ALRO calls "drop" or scrap material. When ALRO produces larger orders, they keep all the excess scrap metal after cutting the stock cut. They then sell this material at discounted prices. The issue with drop material though is that it is not always available, which is why exact costs are not always available. However, after purchasing the material stock, we calculated that buying from ALRO was cheaper than all other retailers we had previously investigated.

The complete Bill of Materials is attached in Appendix A.

6 FABRICATION PLAN

The five main plates of the reduction case that were crucial in terms of alignment were outsourced to a machine shop called Lidell Machining. Our team was responsible for the rest of the necessary machining, which was performed using the following fabrication plan. We followed the manufacturing schedule outlined in the Gantt chart in Appendix H on page 185.

6.1 MACHINING

6.1.1 Sub-Frame – A36 Hot Rolled Steel

The sub-frame was cut to 18.750 ± 0.100 inches using a band saw and the ends were end-milled to exact dimensions. The four ½ inch through-holes were drilled using a 17/32-inch drill running at 720 rpm for a clearance fit at the four locations on each rail.

6.1.2 Pump/Motor Shaft – 1018 Cold Rolled Steel

A two-inch diameter stock piece of steel was placed into a lathe. The lathe was set at 220 rpm and we removed 0.04 inches of material per pass until the diameter reached 1.6 inches. Then, we took progressively smaller cuts until we reached 1.500 inches, within tolerance. The shaft was then cut to 4.500 ± 0.010 inches in length using the cutoff tool.

The shaft was then sent to Riverside Spline and Gear to have a SAE B-B female spline broached into the end of the shaft. After the outsourced machining was completed on the shaft, we cut 3/8-inch keyways on opposite sides (180° apart) of the shaft using a 3/8-inch end mill running at 900 rpm. The keyway was a 1.500 ± 0.010-inch long slot in the shaft. In addition, a 1-1/8" male hex head was machined 0.65 in. deep onto the opposite end of the shaft using a CNC mill.

6.1.3 DC Motor Shaft – 1018 Cold Rolled Steel

The DC motor shaft was also fabricated using the two-inch diameter piece of steel stock using a lathe. The lathe rotated near 220 rpm and we removed 0.04 inches of material per pass until the outer diameter reached 1.6 inches. We then proceeded to take progressively smaller cuts until we reached 1.500 inches, within tolerance. The shaft was then cut to 7.250 ± 0.010 inches in length using the cutoff tool.

The shaft was then sent to Riverside Spline and Gear to have a 6-tooth male spline cut into the end of the shaft. After the outsourced machining was completed on the shaft, we cut two 3/8-inch keyways (180° apart) on the shaft using a 3/8-inch end mill running at about 900 rpm. The keyway was a 1.875 ± 0.010-inch long slot in the shaft. In addition, on the non-splined end of the shaft, a 7/8 in male square head was milled 1.625 inch deep.

6.1.4 Pump/Motor Hub – 1018 Cold Rolled Steel

The pump/motor hub was manufactured from a 4-inch diameter round steel stock using a lathe. The lathe was set to rotate at about 110 rpm and we took off 0.04 inches of material per pass until the diameter reached 3.6 inches. Then, we took progressively smaller cuts as our finishing passes until we reached 3.500 ± 0.010 inches, which was within our tolerance limit. Additionally, one end of the hub (an inch long) was turned down to 1.750 ± 0.001 inches using the same procedure.

The hub was then bored out to 1.5 inches (inner diameter). This was achieved by first drilling a ¾ inch hole into the shaft with a ¾ inch drill bit. From here, 0.04-inch passes were made with a boring tool to take off material until the inner diameter was 1.4 inches. Finally, progressively smaller passes were made until the inner diameter was 1.500 inches within tolerance. The shaft was then cut to 2.500 ± 0.001 inches using the cutoff tool.

The hub was then placed into a mill and secured. The center of the bored hole was found using a dial indicator. Four ¼ inch diameter holes were drilled and two were threaded for attaching the sprockets. The two holes to be threaded were first drilled using a 7/32-inch drill bit at 1400 rpm at the two locations in the drawing. Then the two holes hand tapped using a ¼-20 tap. The other two holes, to be pressed with dowel pins, are first drilled to 15/64 inch at 1400 rpm. Then the holes were reamed with a 0.2495" reaming bit at 900 rpm.

The final operation completed on the pump/motor hub was broaching two (180° apart) 3/8-inch keyways into the inner diameter of the hub using an arbor press.

6.1.5 DC Motor Hub – 1018 Cold Rolled Steel

A 4-inch diameter piece of steel stock was placed into a lathe. The lathe rotated near 110 rpm and we removed 0.04 inches of material per pass until the diameter reached 2.8 inches. Then, we took progressively smaller cuts until we reached 2.750 ± 0.001 inches, within tolerance. Additionally, 1 inch long of the end of the hub was turned down to 1.750 ± 0.001 inches using the same procedure.

The hub was then bored out to 1.5 inches (inner diameter). This was achieved by first drilling a ¾ inch hole into the shaft with a ¾ inch drill bit. From here, 0.1-inch passes were made with a boring tool to take off material until the inner diameter was 1.4 inches. Finally, progressively smaller passes were made until the inner diameter was 1.500 inches within tolerance. The shaft was then cut to 2.500 ± 0.001 inches using the cutoff tool.

The hub was then placed into a mill and secured. The center of the bored hole was found using a dial indicator. Four ¼ inch diameter holes were drilled and two were threaded for attaching the sprockets. The two holes to be threaded were first drilled using a 7/32-inch drill bit at 1400 rpm at the two locations in the drawing. Then the two holes hand tapped using a ¼-20 tap. The other two holes, to be pressed with dowel pins, are first drilled to 15/64 inch at 1400 rpm. Then the holes were reamed with a 0.2495" reaming bit at 900 rpm.

The final operation completed on the pump/motor hub was broaching two 3/8-inch keyways into the inner diameter of the hub using an arbor press.

6.1.6 Spacers – 5/8inch OD ½inch ID 6061 Aluminum Pipe

The spacers were rough-cut to length using a band saw. They were then placed in a mill and faced milled down to 0.500 ± 0.001 inches using a ¾ inch end mill running at 500 rpm.

6.1.7 Tab and Gusset – A36 Hot Rolled Steel

The tab and gussets were first rough cut to a slight oversize using a band saw from ¾ inch and ½ inch thick stock, respectively. The pieces were then milled to exact dimensions as specified on the engineering drawing using a 1-inch end mill running at 600 rpm, removing 0.1 inches of material per pass.

We then drilled two ½ inch holes through each of the two tabs to allow the ½-13 bolts to attach the gear reduction case to the sub-frame. This was achieved using a 17/32-inch drill bit running at 720 rpm. After these holes were drilled, a ½ inch end mill bit was used to create slots out of these holes, as shown in the drawing.

After these slots were complete, the gussets were TIG welded to the top of the tabs. TIG welding was chosen because it provides a small uniform weld that has a small heat affected zone (HAZ) compared to other methods. Since this was a critical junction in our system, we felt that TIG welding provided us with the strongest joint. Since we did not have experience with TIG welding, we asked Bob Coury assist us with the process.

This process was repeated for both of the tab and gusset assemblies. Each was then attached to the Gear Reduction Case Side Plate as discussed below.

6.1.8 Gear Reduction Case – Cut-Out – A36 Hot Rolled Steel
The cutout was an assembly composed of three ¼ inch plates with overall dimensions of 1.750 ± 0.001 by 1.500 ± 0.001 by 6.750 ± 0.001 inches. These three plates were rough-cut to size using a band saw. Then they were end milled to exact dimensions using a 1-inch end-mill running at 600 rpm, removing 0.1 inches of material per pass. The three plates were then welded together along the three edges shown in the drawing. We used TIG welding for this process for the same reasons as discussed above in the tab and gusset section. The assembly of this cut-out to the rest of the gear reduction case is discussed below.

6.1.9 Gear Reduction Case – Top Plate – A36 Hot Rolled Steel
The top plate was rough-cut to size from ½ inch stock using a band saw. An extra ½ inch of material was left around all edges for machining purposes. The piece was then end-milled to 9.750 ± 0.010 by 4.750 ± 0.010 inches using a 1-inch diameter end mill spinning at 600 rpm. 0.1 inches of the material was removed on each pass. Progressively smaller passes were taken until the desired dimension was reached. Through holes for the ¼ inch bolts were drilled at the locations indicated in the engineering drawing using an "F" drill running at 1400 rpm to produce a clearance fit.

6.1.10 Sprockets – Upper Sprocket – 25 Tooth – Low Carbon Steel
Four holes were drilled into the 25-tooth sprocket (to be attached to the pump/motor shaft). Two clearance-fit holes, for bolt connections, and two interference-fit holes, for the dowel pin connections, were drilled. This was accomplished by mounting the sprocket onto a mill. For the clearance fit holes, an "F" drill was used to drill the holes at 1400 rpm. For the interference-fit holes, a 15/64 hole was first drilled at 1400 rpm. Then the hole was reamed using a 0.2495" reaming bit at 900 rpm.

The sprocket also had a 2.000 ± 0.010-inch hole drilled into the center of it. First, a 1.5-inch diameter drill bit was used at 200 rpm to make a smaller diameter hole. Then, a 2-inch diameter bit used at 200 rpm to finish drilling the hole.

6.1.11 Sprockets – Lower Sprockets – 16-, 18-, 20-Tooth – Low Carbon Steel
The following operations were performed for each of the three sprockets listed above.

Four holes were drilled into each sprocket using appropriate drill sizes. Two clearance-fit holes, for bolt connections, and two close-fit holes, for the dowel pin connections, were drilled. This was accomplished by mounting the sprocket onto a mill. For the clearance fit holes, an "F" drill was used to drill the holes at 1400 rpm. For the close-fit holes, an "E" drill was used to drill a hole at 1400 rpm.

Similar to the largest sprocket, each small sprocket also had a 2.000 ± 0.010-inch hole drilled into the center of it. We first used a 1.5-inch diameter drill bit running at 200 rpm to make an under-size hole. Then, a 2-inch diameter bit was used at 200 rpm to finish drilling the hole.

6.1.12 Tensioner – Ultra High Molecular Weight Polyethylene
The tensioners were cut to length on the band saw out of 1" purchased stock. The faces were end-milled flat using a 1-inch diameter end-mill running at 600 rpm. Then, a ¼ inch through hole was drilled through the center to allow a bolt to attach it rigidly to the reduction housing. This hole was drilled with an F drill running at 1400 rpm. Finally, a square counterbore was cut into the tensioner to allow for a nut to rest within the tensioner. This counterbore was made using a CNC program.

6.1.13 DC Motor Shaft Cap – 1018 Steel
The square cap was to be fitted into the tapered bearing. Thus it was to be fabricated with a square internal and a circular outside profile. A two-inch diameter stock piece of steel was placed into a lathe. The lathe rotated near 220 rpm and we removed 0.1 inches of material per pass until the diameter reached 1.6 inches. Then, we took progressively smaller cuts until we reached 1.500 inches, within tolerance. The piece was then cut to 1.000 ± 0.010 inches in length using the cutoff tool. Finally, a 7/8 square hole, as per the attached engineering drawings, was cut into the piece using a CNC mill.

6.2 FINAL MACHINING AND ASSEMBLY

6.2.1 Sub-Frame Installation and Assembly

The existing sub-frame mounts on the frame of the Blue Xebra were not in locations necessary for our design. Therefore, we removed two of the mounts, using a reciprocating saw, and repositioned them to fit our new sub-frame. Bob Coury reattached of these mounts to the frame using TIG welding. After reattaching these mounts, the old sub-frame needed to be modified. The rear bar of the existing sub-frame was cut off using a reciprocating saw, then ground down to allow the new longer rail to be TIG welded in place. Similarly, the front bar of the existing sub-frame was removed and a new longer rail was welded into position. In Figure 6.1 below, the unpainted parts are the components that we have modified. The lower bar (lower in the figure, back in this description) was completely replaced, as was the upper bar (upper in the figure, front in this description). After these modifications, the gear reduction case was ready to be mounted to the sub-frame and DC motor.

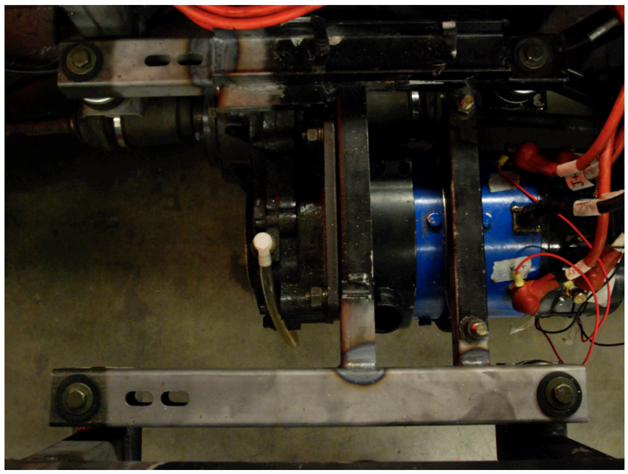


Figure 6.1: Modified Sub-frame Assembly

6.2.2 Pump/Motor Shaft Assembly

The pump/motor shaft assembly included attaching all the required components, the pump/motor hub and sprocket in particular, to the pump/motor shaft. The first task was to press the keys into the keyways on the pump/motor shaft using an arbor press. After this was completed, we slid the pump/motor hub onto the shaft and pressed the hub down over the keys, securing it to the shaft. After the hub was secured, the 25-tooth sprocket was attached to the hub using two ¼-20 bolts and two ¼ inch dowel pins. The dowel pins transfer the torque while the bolts keep the sprocket aligned against the hub. The bolts were fit through the sprocket and screwed into the DC motor hub, while the dowel pins were pressed through the sprocket and into the hub using an arbor press. A picture of the upper shaft assembly is shown below in Figure 6.2.

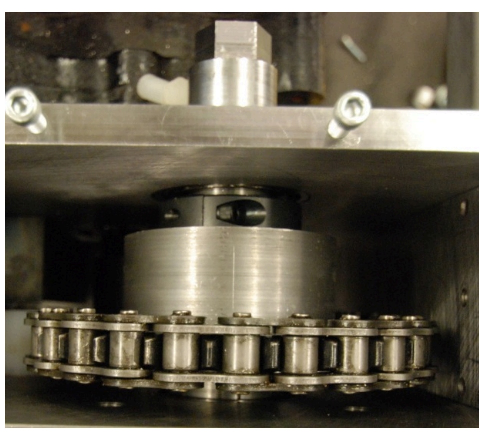


Figure 6.2: Upper Shaft Assembly (as viewed from above)

6.2.3 DC Motor Shaft Assembly
The DC motor shaft assembly included attaching all the required components, the DC motor hub and sprocket(s) in particular, to the DC motor shaft. The first task was to press the keys into the keyways on the DC motor shaft using an arbor press. After this was completed, we slid the DC motor hub onto the shaft and pressed the hub down over the keys, securing it to the shaft. After the hub was secured, one of the three potential sprockets was attached to the hub using two ¼-20 bolts and two ¼ inch dowel pins. The dowel pins transfer the torque while the bolts keep the sprocket aligned against the hub. The bolts were fit through the sprocket and screwed into the DC motor hub, while the dowel pins were pressed through the sprocket and into the hub using an arbor press. A picture of the lower shaft assembly is shown in Figure 6.3 below.

6.2.4 Bearing and DC Motor Shaft Cap Assembly
The machined DC motor shaft cap was pressed into the tapered bearing inner race using the arbor press. Care was taken to ensure that the taper faced down so as not to press the roller pins against the outer race. An addition piece of tube stock (metal or plastic) was used to press the cap flush with the bottom the bearing. This was needed because the cap is shorter than the height of the roller bearing.

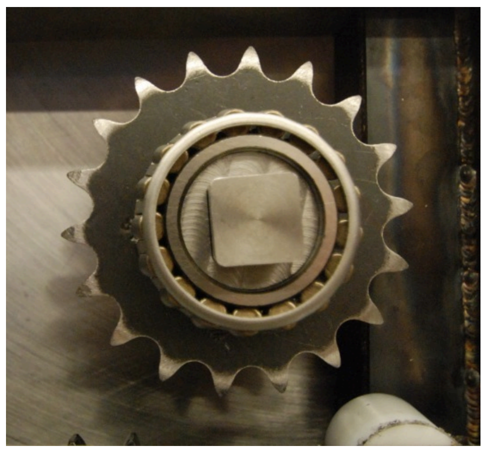


Figure 6.3: Tapered Roller Bearing and Square Cap Assembly on DC Motor Shaft Assembly

6.2.5 Final Assembly
The final assembly includes assembling all of the above components within the gear reduction case. The front plate of the reduction case was removed to allow access to the bearing bore holes. The two bearings for the pump/motor shaft and the one bearing for the DC motor shaft near the DC motor all needed to be press fit into position. Additionally, the outer race of the tapered roller bearing used with the DC motor shaft was also pressed into the housing. These bearings were pressed in using an arbor press additional round stock to ensure that we pressed only on the outer race of the bearings so as not to damage them. After the bearings were press fit, the pump/motor shaft was pressed into the pump/motor plate while the DC motor shaft was pressed into the DC motor plate. Then, the two tensioners were installed using 3/8-16 bolts that were fastened from outside the case. The pump/motor plate and DC motor plate were then aligned and pressed to the front plate of the gear reduction case, forming a fully enclosed gear reduction case.

The chain was installed on the sprockets from the open top. The chain was adjusted to the correct length (dependent on the sprocket size) by adding or removing chain links. The tension of the chain was also adjusted by moving the placement of the tensioners on the inner face of the pump/motor plate. Finally, all of the ¼-20 bolts were installed on the gear reduction case and the assembly was complete.

Finally, the tab and gusset assemblies were bolted to the sides of the outside of the gear reduction case.

Figure 5.20 on page 42 shows an overall exploded view of the final assembly Figure 6.4 below shows the fully assembled reduction housing before final installation.

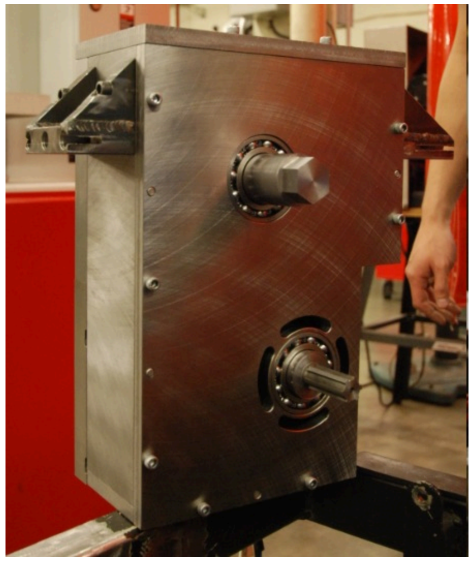


Figure 6.4: Complete Final Assembly of Gear Reduction Case

6.2.6 Integration onto Vehicle

The gear reduction case was then installed onto the sub-frame. This included installing the spacers between the DC motor shaft and the DC intermediate gear reduction (attached to the DC motor). We accomplished this by first aligning the DC motor shaft to the DC gear reduction female spline input. After the gearbox was correctly aligned, the male DC motor shaft was inserted into the spline. We rested the tabs on the front and back sub-frame bars. Then, the four ½-13 bolts were inserted through the tabs and tightened. Finally, the lower access door was removed and bolts were inserted into the DC gear reduction with the fabricated spacers in between the DC gear reduction and the gear reduction case. Figure 6.5 below shows a picture of the reduction housing fully integrated into the Blue Xebra truck. If we had had access to the hydraulic pump/motor, we would have attached it to the case using the pump/motor's mounting flange and four bolts.

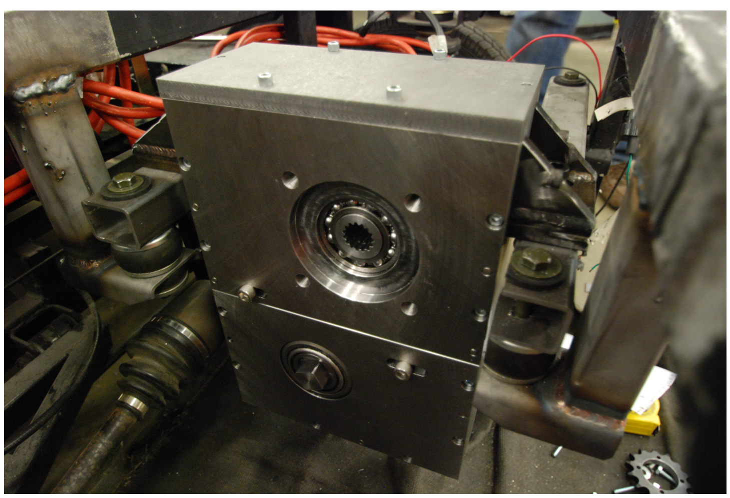


Figure 6.5: Gear Reduction Case Fully Integrated onto Vehicle

7 VALIDATION RESULTS

The following tests were used to validate that our design satisfied the customer requirements and engineering specifications above listed in Section 2.2. Each of the customer requirements and engineering specifications directly correlates to a validation result listed below. A table summarizing these results is shown below in Table 7.

Customer Requirement	Value	Validation	Testing	Tested Value
Desired features in end users' language		*How to verify the solution satisfies the customer requirements and engineering specifications*		
System transmits power and torque when subjected to disturbances	206 ft-lb, 0 fracture	Ensure system can withstand max torque without fracturing by applying the torque. Fracture is determined using visual inspection.	1)Torque wrench & wheel chocks, 2)Long shaft and add appropriate load to create torque	Reached 144 ft-lb successfully for 3 min but differential started to slip even with wheel chocks and e-brake on.
	< 1/2 maximum rated rpm	Bearing design selection for rated rpm	Design selection	9000 rated rpm for all bearings
	< 2% slack	Physically measurement and calculation of chain slack	Measurement and calculation	1.98%
	≥ 1.2 contact ratio	Calculation of ratio based on number of teeth and diameter of gears	Measurement and calculation	X
	$BSV < BSV_{threshold}$	Calculation of BSV based on type of belt drive and size & type of pulley	Measurement and calculation	X
System transmits power and torque safely	Parts always contained within restriction planes of existing vehicle frame	Measurement of restriction planes of the vehicle to and visual inspection for parts exceeding the boundary	Visual inspection	No parts higher than bottom of truck bed and no parts are lower than the bottom of the DC motor
	≤ 0.5 inch opening	Measurement of openings of parts that are considered enclosed	Measurement	Using measurement, there are no openings greater than .5"
Configurable gear reduction	2.5:1 - 3.6:1	Multiple gear ratios within range are installable	Rotate input shaft manually and visually monitor the output of each gearing ratio option to ensure proper function	Rotation test confirms accurate gear ratios between pump/motor shaft and DC motor shaft
	≥ 3 ratios & neutral position	Multiple gear ratios can be installed, one of which must be a neutral position		There are 3 gear ratios (2.9:1, 3.3:1, 3.6:1)

Easy to change gear ratios	≤ 3 hours	Able to change gear ratio to a different ratio in less than 3 hours	Perform physical gear ratio change and time accordingly	Timed test was conducted. All changes were performed in under 20 minutes (1/3 hour)
Reliable design	< Endurance limit of material (infinite fatigue life)	Material properties selection, hand calculations and/or FEA	Hand calculations	A36 steel endurance limit = 21.61 ksi, 1018 steel endurance limit = 23.06 ksi
	SF ≥ 2	Verify that applied load and torque calculations can be withstood by parts using hand calculations and/or FEA	Hand calculations	Calculations for plates (in bending), shaft (in torsion), bearings (dynamic radial load), chain (max tensile strength (also covers sprockets)
Components fit within frame	≤ 4.4 ft^3	Calculate total volume of added components using CAD or physical measurements	CAD measurements of parts's contained volumes	1.04 ft^3 for all parts
	No (0) unintended contact or interference with existing components	Measurement and visual inspection that parts are not unintentionally contacting each other	Visual inspection and measurement	No unintended contact is present. In particular, the wiring fits as needed and the electrical components are not interfered with. The existing components that were not modified are not interfered with.
	≥ .52 inch	Measurement of clearance between non-internal parts	Visual inspection	All non-internal components that are not intended to contact each other do not.
Design risk considered	Lowest (minimal) risk score	Create a risk rating system and quantify risk of each design and/or component	Calculation and explanation of risk and design choice (report)	Lowest risk design was chosen (shown in Appendix D)
Vehicle stablity and dynamic performance maintained	Between rear axle and the current CG. No higher (0 in.) than current vertical location and no lower than wheel contact with ground	Measurements and calculation of new CG to meet values specified.	Measuring the weight of the vehicle without the existing hydraulic components on it and then take into account added components and locations	Vertical: above ground (22.47" baseline, 22.06" current), Horizontal: forward of back of truck (45.97" baseline, 42.71" current)
Total cost	≤ $1000*	Approved spending above $400 by EPA	Final spending report	$2722 total cost (approved by sponsor)

Transferable for future development	Written acknowledgement that our team has satisfied this requirement (The team provided sufficient documentation)	Each of the components detailed in the specification is transmitted to the sponsors and customers and all acknowledge, in writing, that satisfactory documentation had been received.	Documents that contain detailed description and explanations of our customer requirements, engineering specifications, concept generations, final concept selection, CAD files, analysis, Testing reports and final report.	Completed on transfer of this report

Table 7: Validation Results Summary

7.1 SYSTEM TRANSMITS POWER UNDER DISTURBANCE

Three separate tests were used to validate that our design met this customer requirement.

The first test validated that our design could withstand 206 ft-lbs of torque without fracture. This torque value was determined to be the maximum possible output torque of the pump/motor and DC motor combined (and hence the largest amount of torque our system would need to withstand at any moment). To perform this validation test, we had planned on applying a torque to the machined hex head on the upper pump/motor shaft using a torque wrench supplied by the EPA. However, after installing the entire reduction housing and receiving the torque wrench from the EPA, we were unable to use the provided torque wrench due to space limitations and part interference.

To remedy this problem, we designed a new method for applying torque. We used a 1-1/8" wrench and slip fit a 44 in. long pipe over the end of the wrench. We then attached a spring load gauge to the other end of the pipe and torqued using this moment arm (Figure 7.1). The wheels of the truck were chocked and the emergency brake was also fully applied to ensure no unwanted movement in the vehicle. Additionally, a piece of metal was placed over the top of the vehicle frame (as pictured) in the event of possible fracture. All tests were conducted under the supervision of Bob Coury.

Figure 7.1: Applying torque to reduction housing

The spring gauge displayed increments of 5 lbs, which equates to 18.33 ft-lbs using the equation T=F*R (torque = force * distance). We first applied 5 lbs, then 10 lbs, 15 lbs, 20 lbs, 25 lbs, 30 lbs, 35 lbs and then 40 lbs. At each increment, we held the load for 3 minutes. This time interval was due to the fact that typically in the aerospace industry, proof load tests are held for 3 minutes. Thus, we felt that this would be sufficient for our application. When we reached a load of 40 lbs (equivalent to 144 ft-lbs of torque), the system began to slip. Initially we believed that it was the wheels that were slipping, but upon further investigation, we determined that the differential in the existing transmission was slipping.

Although we did not achieve the 206 ft-lb requirement, we believe that our design is more than sufficient for its application in this vehicle for several reasons. There are two inherent "fail-safes" already in place in the Xebra truck system: the first is the tires and the second is the differential. In the unlikely event that the pump/motor and DC motor did output 206 ft-lbs of combined torque, the differential would begin to slip. If this did not occur and the differential remained rigid, the tires would begin to slip on the road surface. It is extremely unlikely that both the tires and differential would lock up. While we cannot quantify the likeliness of this occurrence, we are confident that our gear reduction case and contained components would be able to withstand 206 ft-lbs due to our design choices and material selection. A static test could be conducted, using only the gearbox that we fabricated, to validate that it can withstand the maximum torque.

The second validation of this customer requirement was completed by design. The specification states that the operating rpm of the system should be less than half the maximum rpm rating of the bearings. Our sponsor informed us that in an absolute worst case, the system could operate at 2000 rpm. Therefore, we selected bearings rated for 9000 rpm, which easily satisfies the engineering specification.

The third and final validation test for this customer requirement involved measuring chain slack, which needed to be less than 2% of the center-to-center distance between the sprockets. In our system, the center-to-center distance measured 6.3 in., resulting in an allowable slack of 0.126". Figure 7.2 below shows the four locations where we measured chain slack.

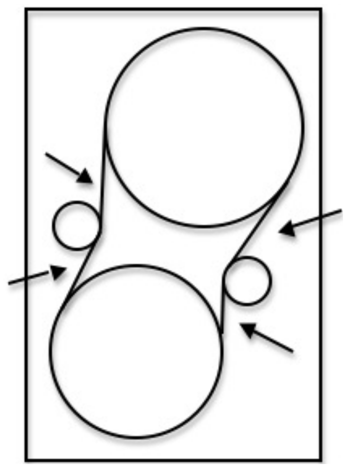


Figure 7.2: Chain slack measured at areas with arrows

The chain slack was measured using a scale. The scale was held steady while the chain was moved from side-to-side measuring the maximum deflection. We measured the chain slack to be 0.125" at three of the four indicated locations. Due to access restrictions, we were unable to measure the slack at the lower right location. This measurement was not possible because it was inaccessible through the top access panel of the gear reduction case. Additionally, with the lower access door removed, the tensioner would also be removed, providing an inaccurate measurement that would have more slack present that in normal operation. Therefore, we reasoned that the slack in this lower right area would be lower than the others because the tensioner is closest to the sprocket at this location. This results in the shortest length of chain between the tooth and tensioner, which correlates to less chain slack. The measurements made were overestimates, giving us confidence in this assumption. Using this method with each of the three gear ratios, we measured a maximum slack of 0.125" at each of the three locations (total of 9 measurements).

7.2 SYSTEM TRANSMITS POWER SAFELY

Two tests were conducted to validate that our design met this customer requirement.

The first validation test was used to verify that all of the components of the system always remained below the top of the vehicle frame and above the lowest part of the DC motor. Remaining inside these restriction planes ensures that no interferences with the road or the vehicle itself exist. More importantly, this ensures that in case of any failure, no parts would be ejected from outside of these planes at any time. Because we had no failure of fracture of any components during our testing or assembly of the design, this specification was satisfied after visual inspection. Our reduction case was designed to meet this specification and after visually inspecting the case as it was installed on the vehicle, we confirmed this.

The second test for safety was based on a Michigan OSHA specification stating that all openings must be less than 0.5 in. This specification guarantees that no body part, clothing, or other hazardous articles can fit into the gearbox and contact a rotating part. Again, our reduction case was designed to meet this specification and after measuring all gaps in the case manually, we confirmed that there are no openings larger than 0.5 in. The only area of concern was the joint between the walls and the cutout, in which the largest gap size was measured to be 0.125".

7.3 CONFIGURABLE GEAR REDUCTION
Two tests were conducted to validate that our design met this customer requirement.

The first test validated that our design satisfied the engineering specification that three gear ratios and a neutral ratio were present in the system. For our system, this translated to allowing for three different sprockets to be mounted within the system and for the entire system to be disconnected so that it would be freely rotating. Initially, we had designed the system to accommodate ratios of 2.5:1, 3.3:1 and 3.6:1. However, after finishing fabrication and assembly of the design, we determined that the 2.5:1 ratio would face chain and hub interference issues. Therefore, after discussion with our customers and sponsors, we used a 16-tooth sprocket (generating a 2.9:1 ratio) instead of the previous 14-tooth sprocket. While we did not produce the exact ratios we had originally intended to, we were able to provide three different gear ratios within the desired range that satisfied the needs of our customers.

The second validation test consisted of rotating the pump/motor shaft to verify that our sprockets would produce the correct amount of rotation at the wheels. This would have been moderately difficult to accomplish because even with the vehicle's rear wheels (the driven wheels) lifted off the ground and allowed to freely rotate, the existing differential would not always split the rotation evenly between the two wheels. Therefore, we decided to use a different testing method. We first made the assumption that the existing intermediate gearbox contained an exact 4.5:1 ratio between the DC motor output and the vehicle's wheels. We then verified the rotation of our designed gear reduction case by manually rotating the DC motor shaft and measuring the number of rotations output by the pump/motor shaft. By rotating the DC motor shaft a given increment of whole rotations, the observed rotation on the pump/motor shaft allowed us to calculate the actual gear ratio. The measurement method is shown below in Figure 7.3. The reverse was also completed by rotating the pump/motor shaft and measuring the DC motor shaft. This method was applied for each gear ratio and each ratio was validated to be accurate.

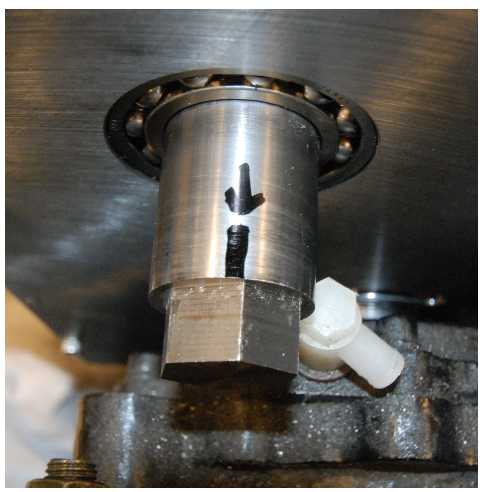


Figure 7.3: Marker used for alignment purposes during rotation test

7.4 EASY TO CHANGE GEAR RATIOS
To validate this requirement, we manually changed the gear ratio from one to another, while measuring the amount of time it took to complete each test. The change process is outlined below, with a more detailed description located in Appendix J.

- Loosen the tensioners and move them outwards
- Remove the six socket cap screws on the lower access door
- Pry off the lower access door using a flat-head screwdriver on the four machined notches
- Remove the four top access plate socket cap screws and the top access plate
- Remove the tapered roller bearing
- Remove a quick-connect link on chain and disconnect the chain

- Remove the two socket cap screws on sprocket
- Remove the sprocket and replace it with a different sprocket
- Replace the two socket cap screws on sprocket
- Add or remove chain links depending on the size of the new sprocket
- Reattach the chain
- Replace the tapered bearing
- Hold the slack chain with a wire from above
- Replace the lower access door and lightly hammer it into place
- Replace the six socket cap screws
- Replace the top access plate and four top socket cap screws

After performing this process multiple times, we recorded a maximum time of less than 20 minutes, which was well below the required time of 3 hours. The first sprocket change took 19 minutes and 36 seconds. Thus, it is assumed that even someone without extensive experience using our system would be able to change the sprockets in less than 3 hours. Figure 7.4 below shows one stage of the sprocket change.

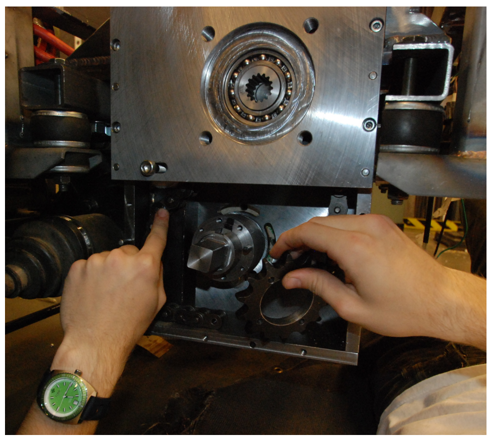


Figure 7.4: Gear change being performed

This picture demonstrates that the gear ratio changes we performed without the pump/motor installed. Since the pump/motor was not available to us during the test, we used a 3D rapid prototype print of the pump/motor to simulate the clearances and allowable space we would be allowed if the pump/motor was installed. With the pump/motor mockup in place of the real pump/motor, we were still able perform the sprocket swap in a maximum change time of 19 minutes and 53 seconds.

7.5 RELIABLE DESIGN

To ensure our design was reliable, we satisfied two design criteria. The first involved material selection. All of the materials we selected had endurance limits larger than the maximum stress the system is expected to face. The two materials we selected were A36 steel and 1018 steel, which have endurance limits of 21.61 ksi and 23.06 ksi, respectively. Accordingly, the maximum expected stress was 15.23 ksi, found to be located at the keys. The second criterion is that all designed components must have a safety factor of at least 2 in yield. This means that any applied stresses must be less than half of the material's yield strength, while also considering the part's geometry. This safety factor was applied to all the components that were designed including the reduction case plates, the shafts, the sprockets, and the chain. This analysis is detailed previously in Section 5.3, Section 5.4.6, and Appendix F.

7.6 COMPONENTS FIT WITHIN FRAME

Three different requirements needed to be satisfied for this customer specification.

The first component requires that the total volume contained by the designed components must measure less than 4.4 ft^3. We validated that our design satisfied this requirement by using CAD software to calculate a total contained volume of 1.04 ft^3. It is important to note that we did not simply measure the volume of the components themselves but instead measured the enclosed volume. The difference is most relevant in discussing the gear reduction case. The volume of the components would simply be the volume of each of the plates that compose the case. However, the enclosed volume is the total volume of the case itself. These factors are important in determining the total "footprint" of our design in the space of the vehicle.

The second validation involves part interference. Due to the numerous amount of mechanical and electrical components located in the rear of the Xebra truck, it was important that no part of our design interfered with any previously existing components. Therefore, we designed our reduction case to easily fit onto the existing vehicle's frame with only minor mechanical changes to the sub-frame. Our design also allowed all of the existing electrical connections to be easily rerouted and connected to the DC motor again with no other modifications. After completing the installation of the reduction housing onto the vehicle we observed the clearances of the components added and were unable to identify any interference issues.

The third component of this requirement mandated a clearance of at least 0.52" between non-enclosed components. This was another specification that we met by design. The location and orientation of the parts in our system were created so that at least 0.52" were located between parts that were not directly attached to each other. We have been able to confirm this measurement for all of our components except for the clearance between the pump/motor and the sub-frame. As previously stated, we did not have access to the pump/motor and could only measure our CAD model and rapid prototype mock-up. Although we achieved the clearance using these measurements, we informed our sponsors so they are aware of our methods and recommend that they conduct the measurement using the actual pump/motor once it becomes available. An example of a clearance we were able to measure was the distance between the reduction case and the existing gearbox of the DC motor, which was found to be .531".

7.7 DESIGN RISK CONSIDERED

As requested by our sponsor, we created a risk rating system (Appendix D) to quantify the relative risks of each of our design concepts. This design risk rating system accounted for assembly, complexity, machining, tolerances, newness and our overall confidence in the design, and was used for selecting concepts for a final design. Each of these areas received a weighted score against which each design concept was compared. As previously discussed in the Design Selection section, after rating all of the concepts, we selected the design with the lowest risk score for our final design.

7.8 VEHICLE STABILITY AND DYNAMIC PERFORMANCE MAINTAINED

The engineering specification for this customer requirement states that the horizontal center of gravity (CG) must remain between the existing CG and the rear axle. It also states that the vertical CG must remain at or below the current CG. We received weight measurements at each of the three tires from the EPA and used these to determine the existing CG. The total vehicle weight was 1660 lbs, which was distributed across the three wheels. Using the back edge of the car as a base and measuring the wheelbase of the car, we determined that the baseline CG was 45.9" forward of the back of the truck. After adding the weight of our added components (including gear reduction case and estimated pump/motor weight) and using the same method, we determined that the new horizontal CG was 42.7" forward of the back of the truck. This shows that the horizontal center of gravity remained between existing CG and the rear axle (located 15" forward of the back of the truck).

The vertical center of gravity was measured using the recorded weights at each of the wheels and allocating the distribution of weight amongst the existing components of the vehicle. We estimated the proportion of weight each component accounted for (such as the DC motor, sub-frame, etc.) based on the amount of weight at each wheel. We then determined the approximate vertical location of each of these components. Using the ground as the reference point, we calculated the baseline vertical center of gravity to be 21.7" above the ground. After adding the weights our added components, we calculated the new vertical CG to be 21.3" above the ground. As desired, the CG moved no higher than the existing CG. A thorough explanation of the process taken to calculate the CG can be found in Appendix M.

Other alternative methods of calculating vertical CG such as lifting to tip the vehicle or to individually weigh each component were not used because extremely accurate measure of the CG was not necessary according to our sponsors and customers. This is because many hydraulic components will be added to the vehicle in the future, including two large accumulators, which will dramatically affect the CG more than our system did.

7.9 TOTAL COST

At the onset of the project, our sponsors allocated an estimated budget of $1000 to us, which we had previously used as an engineering specification. However, it was noted that this value could be higher, pending sponsor approval of need. As the project progressed, the accuracy necessary to achieve a successful design led to cost increases. This was mainly due to having several of our gear reduction case plates machined by a third party, Lidell Machining, whose machining cost were $1855 dollars alone. Ultimately, we spent $2,722.25 with about $300 of that value sourced from the University of Michigan. We reserved much of the $400 U of M budget for smaller purchases because the comparatively longer lead time required to obtain funding from the EPA. The complete Bill of Materials can be found in Appendix A on page 73.

7.10 TRANSFERABILITY

The engineering specification for this customer requirement states, "all documents containing detailed descriptions and explanations of our customer requirements, engineering specifications, concept generation, alpha and final concept selection, CAD files, analysis, testing reports and final report are to be provided." After completing this final report and turning over all required documentation and the Xebra vehicle to our EPA sponsors and U of M customers, we will have our sponsors acknowledge receipt of all materials, thus satisfying this requirement.

8 DESIGN CRITIQUE

In retrospect, although our design was promptly completed and fulfilled our requirements, there were some process and design aspects that would have been approached differently had we knew what we do now. We believe our design excelled in many areas, and could definitely be considered successful given the circumstances. Yet there were potential improvements that were identified for future efforts.

In terms of the successes of our design, given the highly constrained system including extremely limited space and prior applicable experiences, we were able to complete modification and machining of a functional sub-frame system, power transmission, and transmission housing. Our design and machining resulted in a sub-frame that is robust and holds the weight of our components. Through the outcome of our validation tests, we have proved that our design is capable of transmitting as much torque as can be applied manually to a static system. The gear ratio can be quickly changed and functions properly for all ratios designated, giving proof that the access plates, tensioning system, and shaft-hub-sprocket system all act as predicted. Therefore, as an end result, our design can be seen as successful in that the EPA has fulfilled their goal of obtaining a working power transmission system to interface the hydraulic system and current DC motor.

Evaluating our design, there are some specific issues, which would have been improved if given the opportunity. As a final design change to resolve a special issue caused by measurement inaccuracies, the back crossbar of the sub-frame and dampener was cut into to shift the reduction case so that it could be properly aligned with the DC motor's spline. For this problem, there are two critiques that arise. First, we would have determined a more accurate measuring process to dimension our workspace from the beginning. Having not done so, all of our measurement variation accumulated in our stack-up and caused approximately 3/8" of accumulated error. In speaking with Bob Coury, instead of relying on a tape measure, he suggested possibly using a machinist scale. This resolution could potentially eliminate the need for a cutout from the sub-frame, if the case was adjusted accordingly. The second resolution that could improve upon the current design is the addition of a support for the open bar that currently supports the back sub-frame. If a piece was welded into the gapped section for additional stability, this would structurally support the sub-frame from collapsing over time.

Another weakness of the design came due to a missed assembly/maintenance clearance. Although we had successfully adjusted to eliminate the need for multiple splined shafts in series, clearance of the interchangeable sprocket to move passed the inner race of the tapered bearing was neglected. Because of this, the sprocket had no way of passing over the tapered bearing, even though the door could be removed separately from the shaft. With this issue, a new solution had to be made, in which a square cap and fitting external square on the shaft were machined to remove the inner race of the tapered bearing. Rather than solving both issues at once, this iterative process added more components to the design that were not necessary. If a tapered bearing were found that allowed for the sprocket to slip passed the inner race, or a ball bearing was used with a square cap, then there wouldn't be two removable joint locations, and thus would reduce the friction of the system. Additionally, an option considered but not used was a needle bearing rather than a tapered bearing, which would have allowed a complete removal of the bearing from the shaft. In hindsight, the needle bearing would have been the primary choice given the opportunity to redesign the components.

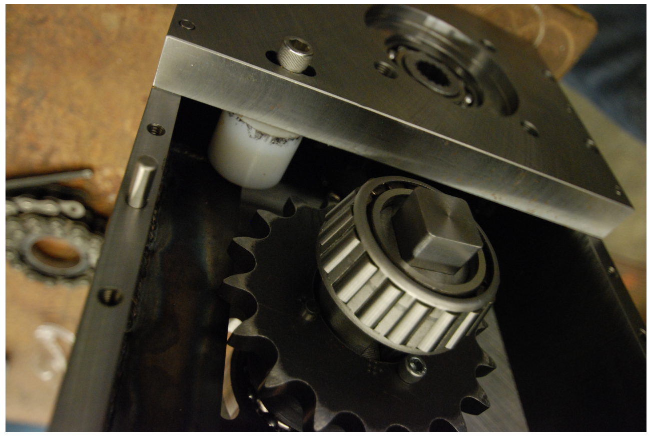


Figure 8.1: Resolution to clearance issue to remove lower sprocket

An observation that is more of a system critique rather than a design critique is the misalignment of the DC motor. We observed that the DC motor, when mounted using the existing sub-frame, had such large tolerance to move after being secured. Because of the opportunity for deviation, the DC motor could not be positioned correctly so that it would be parallel to the crossbars of the sub-frame. With such large misalignment, this places significant loading on the splined connection between the DC gearbox and the reduction case of the pump/motor. Furthermore, the misalignment created a variable distance between the reduction cases, so as not to allow our spacers to fit between and rigidly define/support the connection. While out of our control, if the DC motor and associated gearbox could be more accurately positioned to provide a consistent reference to interact with, then our spacer design could be implemented properly, and act as designed. Replacing the current DC motor mounting mechanism or reinforcing the connection could benefit the system alignment and interaction. At this point however, with severe tilt in the DC motor, the four spacers have been machine, but not all fit within the .5" gap.

When the vehicle was driven to transport it from the machine shop, a chattering noise was observed due to the chain. The noise was later attributed to the tensioning of the chain, a problem that had not clearly been fixed to that point. Because the vehicle was only validated using relatively static conditions, performance qualities such as noise and tensioning were never really addressed properly. At this point, an accurate and reliable method for determining the appropriate chain tension has not been defined. Therefore, the overall extent of tensioning the system can be improved if given the opportunity, since this general field of dynamic analysis has not been extensively reviewed. By completing this analysis, an optimal chain tension can be found at each gear ratio, thereby allowing the most efficient system performance.

In a more general sense, there are various processes in our design development, which, if done differently, could have allowed us to bypass some of the machining/assembly problems that arose. If we had taken a more detailed approach to outlining step by step the assembly of the case during the design process, we may have designed our fit-up to be more easily assembled. Overall though, while these issues have been described, detailed explanations and possible recommendations for these design problems are described in the following section.

9 RECOMMENDATIONS

There are several recommendations for future projects and recommended design changes that we recognized through our design critique section. It is important to note that these recommendations do not change the overall final design of our project.

Currently, the DC motor is supported on the sub-frame using two cross bars. One of the crossbars has a U-bolt that supports the DC motor as a cradle around it. We recommend that this DC motor mount be fixed. Currently, this mounting mechanism is tightened too far such that the DC motor is mounted at a slight angle, which affects the overall alignment of our design. Currently the connection between the DC motor reduction case and our designed reduction case is through a splined shaft. Due to this slight angle, there is added stress at the point of this connection and may cause the splines on the shaft to wear faster than normal. One way we suggest you can fix this is by using a slightly larger U-bolt that can hold the DC motor at a horizontal, even when tightened fully. The current U-bolt when tightened all the way tugs on the motor, making it mount at a slight angle. But when loosened until it is horizontal, it is too loose. Thus fixing the overall DC mounting system to be parallel to the sub-frame eliminate concerns of failure at the connection between the two reduction systems.

To validate our design it was important to test our system under the maximum load torque that was calculated. We conducted our torque test explained in the validation portion of this report. While we were conducting this test we were limited at a torque of 144 lb-ft. This happened due to the slipping in the differential causing the tires to turn, and limiting us from applying additional torque. To complete this task we suggest you can conduct one of the two possible solutions. These two recommendations could not be implemented by us with the facilities and funds at our disposal. As a possible solution you could switch out the current differential to one that has the capability to lock thus easily allowing you to torque the system to desired torque. You can also detach the reduction case from within the vehicle, and perform a torque test on it while it is outside the vehicle.

We recommend looking into more methods allowing interchangeability of the lower sprocket. The lower sprocket is mounted on a hub that is attached to the DC motor shaft. This shaft is supported at both ends using bearings for alignment purposes. You can achieve this interchangeability by using a tapered bearing technique where the tapered bearing detaches and the sprocket slides over the inner race of the tapered bearing. Another method you can use is by using a square cap with a slip fit on an external square on the shaft. Our recommendation is to use a needle bearing on the side of the access door that enables you to directly slide the shaft in and out of it.

The sub-frame was cut in order to add more adjustability to align the DC motor shaft with the electrical systems gearbox. We recommend welding a square plate in the gap of the sub-frame to increase the overall strength of the sub-frame that was cut.

We also recommend checking the tensioning on the chains, making sure it is set at the right amount depending on the sprocket that is installed. We suggest you do so by adjusting the tensioner while the vehicle is tested dynamically. The chain also needs to be lubricated from time to time, to prevent rusting and improve performance.

Alignment of various fabricated components is always a very important concern. We used a tape measure to make initial measurements of the existing workspace and ended up having an error stack up in our overall assembly. After consulting with a machining expert we recommend using a machinist ruler when taking dimensions directly off of the vehicle.

10 SUMMARY AND CONCLUSIONS

The Environment Protection Agency is developing a hydraulic-electric hybrid car by redesigning a Zap, Xebra electric car. Previously other ME 450 teams have worked on this project but the lack of continuity and multiple iterations of these projects led to an inefficient design. Thus our sponsors from the EPA and primary customers from the university have asked us to start over by working on a new Xebra (Blue Xebra) model, which is currently only electric. The goal of our project is to design a power transmission system that will enable the Xebra to operate using a hydraulic-electric hybrid drive train. The main end result of this project is to have a fully functioning electric-hydraulic hybrid, which will serve as a benchmark for our sponsors' and customers' future endeavors with electric-hydraulic hybrid vehicles. It is important for us to point out that at the end of our project we will have a coupling mechanism that combines the electric and hydraulic systems together, but the complete hydraulic system will not be ready to be implemented yet as our customers are still working on completing that system.

This power transmission coupling system must meet several customer requirements. Most importantly, the system must properly transmit torque and power safely when subjected to disturbances. The design must also allow for at least 3 gear ratios to be installed easily changed. Additionally, the system must be reliable, low risk, and fit within the frame of the existing vehicle. The detailed list and correlation of all the customer requirements to engineering specification, as well as the relative importance of each specification, can be found in the Quality Function Deployment Chart. The translation of our customer requirements to engineering specifications was approved by both our sponsors and customers.

With the requirements of our design clearly defined, we began generating concepts by functionally decomposing our design into four functional areas: the reduction system, the coupling mechanism, the pump/motor mount, and the reduction case mount. We created multiple concepts for each functional area. The reduction systems included spur gear, chain and sprocket, and planetary gear systems. Coupling mechanisms included fixed and rigid options such as flange, spline, and spider/jaw couplings. Pump/motor mounting methods varied from horizontal and vertical orientations to undercarriage supports. Reduction case mounting concepts included tabs, gussets, and bolting methods.

We were then able to create 3 concepts that included all of the above-mentioned functional areas in different ways. Using a Pugh chart analysis and a qualitative morphological chart we were able to determine how each one of these concepts reacted with one another in totality, creating a concept that meets our engineering requirements as closely as possible. Using this process and with constant feedback from our sponsors and customers we selected an initial Alpha Design.

Our Prototype/final design very closely resembles our Alpha design. The hydraulic pump/motor is directly mounted onto one side of the reduction case using bolts. The reduction case consists of a chain and sprockets system that helps transmit torque and power between the hydraulic and electric systems. We will be using multiple gear ratios to test for the optimal reduction system. The sprocket on the pump/motor shaft is permanently attached to it using a hub and the lower sprocket is interchangeable through an access door on one side of the reduction case. This reduction system along with the pump/motor is supported by a tab and gusset mount of appropriate thickness. The tab and gusset mounts sit on a fabricated sub-frame that is connected to the vehicles main frame through a dampening system. Each one of our new components that we will be adding to current vehicle is validated using engineering parameter analysis.

Throughout the project the workspace limitations drove our design. Our final design is safe and reliable based on the materials we have selected and designed geometries. We meet our customer requirements of fitting within our workspace, having configurable gear ratios, interchanging these ratios in a time frame, having a system with low risk and designed within our budget. On the whole a detailed explanation of how our design meets our engineering specifications and customer requirements can be seen in the "Testing and validation" and "Design Parameter Analysis" sections of this paper.

For future projects we have made several recommendations that our sponsor and customers should consider. We have recommended fixing the current DC motor mounting mechanism, recommended to prevent the differential from slipping to test the system under maximum torque, to check for better methods for the interchangeability of the lower sprocket, recommended welding a square plate to the part of the sub-frame that was cut to increase its strength and making sure the chain is properly tensioned and lubricated.

As an outcome of our project we were able to design a gear reduction system that was housed in a gear reduction case with accessibility. This case was mounted on a sub-frame that we fabricated and joined to main frame of the vehicle through a dampening system. We successfully designed a coupling mechanism that will join the hydraulic system to the currently existing electrical system of the vehicle. We met the optimal gear ratio range that our customer required through the multiple gear ratio options. Finally as an improvement to past projects were able to create a design that was more space efficient, organized and concise.

11 ACKNOWLEDGEMENTS

Our team would like to acknowledge the many individuals who generously offered their time, effort, and expertise to us throughout this semester. Without their help, we undoubtedly would not have been able to achieve success in this project to the level desired.

Special thanks to:

Dr. Andrew Moskalik and Mr. David Swain for being a constant source of technical information and guidance throughout the semester. It was invaluable to have sponsors that were will to discuss the many issues we had with the design throughout the entire design process.

Dr. Andrej Ivanco and Mr. Xianke Lin for continually working with us to clarify the goals and technical details related to the entire Xebra truck system. Their understanding and flexibility, especially during our many design alterations, was extremely helpful.

Mr. Bob Coury and Mr. John Mears for all their help during the machining and manufacturing process. Without exaggeration, our project could not have been completed without them. Not only were Bob and John's machining advice crucial, but special help in welding, CNC milling, and simple machining time allowance allowed us to finish our project on time and with a high degree of professionalism.

Professor Gordon Krauss and Mr. Dan Johnson for helping us keep our project on track throughout the semester. They asked the tough questions to ensure that our design process was thorough and well investigated and helped clarify any questions related to class material.

Mr. Matt Navarre and Dr. Zoran Filipi for helping us move the Xebra truck between its multiple storage locations and for allowing us a storage and workspace for the vehicle. This contributed immensely to our project's success as it allowed us to constantly work in the actual vehicle instead of a close substitute.

12 TEAM BIOGRAPHIES

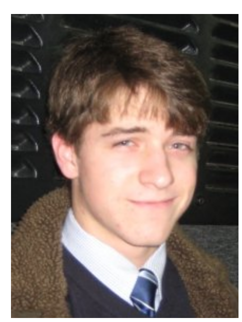
David Fok

I was born in Hong Kong and raised near Toledo, OH. I came to the University of Michigan for its engineering program and football team (the second of which has not panned out as successfully as I would have hoped). I chose engineering because I was always interested in building things (playing with Legos and building model cars and planes). In particular, I chose mechanical engineering because of the variety of opportunities available through the degree. I was originally interested in aerospace, which remains an open possibility as I have learned. I have always enjoyed using my hands in work and still do to this day which is why I am currently interested in field work.

Outside of the classroom, I am a part of the Human-Scale Interactive Rubik's Cube Team, which is building a Rubik's cube similar to the "Alamo" cube on Central Campus next to the Michigan Union. During the summers between semesters, I have worked as an intern in various industries. In 2008, I was held a business related position in marketing and strategic planning at Rexam (makers of aluminum cans and plastic packaging). A year later in 2009, I worked as a design engineering intern at GE Aviation on military jet engine augmenters and exhaust nozzles. This past summer, I held a project engineering role at Marathon Oil where I was able to experience fieldwork in an oil refinery.

In my spare time, I enjoy playing and watching almost all sports, particularly golf and football.

Contact: dfok@umich.edu
419-276-3838



Andrew Gavenda

I was born in Princeton, New Jersey and raised in Cranbury, New Jersey all of my life. I applied to Michigan 2 days before the deadline only because my mom's friend had mentioned it was a really good engineering school. After applying and visiting, I knew I would go here and was so glad that I had applied. I've wanted to be a Mechanical Engineer since grade school, and once I came to Michigan I knew it was the right major for me.

I enjoy working with people and with my hands more than classroom work so I'm involved in a lot of extracurricular activities. I'm on the Design Competition committee for Tech Day (the student run engineering open house for high school students), I'm the historian/photographer for ASME, I'm the Vice President of Pi Tau Sigma (the Mechanical Engineering Honor Society), and I also mentor and Eng100 class on Underwater ROV design. Also, for the past two summers I have worked for Lockheed Martin Space Systems Company on satellite designs and assembly and integration. I plan on living out in California after graduation mostly because I lived out there this past summer and absolutely loved it.

Contact: agavenda@umich.edu
609-216-2358

Anuj Shah
I was born in Mumbai, India and came to the University of Michigan to pursue my undergraduate degree in Mechanical Engineering. All my life I was surrounded with engineering since my father and my brother are Mechanical Engineers and run a family owned company. I was originally going to come to United States only if I got into my top choice of schools, meaning to say I was simultaneously applying to engineering schools in India. Luckily everything worked out and I am here taking classes in my final two semesters of my engineering degree.

Outside of school, I was always into sports growing up; I was in competitive swimming and played handball, and football (soccer as you would call it here) all throughout my school career. I take part in the annual Indian American Student Association Cultural Show here at the university, which is the largest student-run show in North America. I have also gained some industry experience through my past work experiences. Since the summer, I worked with the Department of Energy, Industrial Assessment center with whom I conducted energy audits in companies throughout Michigan and Ohio. Through these visits we helped make these companies more energy efficient by reducing their electrical energy usage and natural gas consumption. It is because of this opportunity that I am leaning towards consulting in the future. In the immediate future I would like to pursue a masters degree in the field of Industrial and Operations Engineering.

Contact: adshah11@gmail.com
734-223-1900

Mat Wecharatana
I am currently an undergraduate senior at the University of Michigan, studying Mechanical Engineering. I was born in Livingston, New Jersey, and have lived in Parsippany, New Jersey all of my life, until I left for Ann Arbor. I am interested in physical health, specifically sports, as I have played basketball since I was in 4th grade, and try to maintain a consistent workout and healthy diet. I originally wanted to become an Aerospace Engineer, as I had an interest in the aircraft industry. However to broaden my job opportunities, I selected Mechanical, and have been opened to the different industries I never knew of before. Since then, I have shifted toward an interest in Healthcare technology. Aside from school, I have had an internship working in a Fabrication Lab of a semiconductor manufacturing company. During that summer, I worked on the production line integrating some film deposition machinery into production. The last two summers, I have worked with GE Healthcare, with one summer working as a Sourcing Intern, and the last working in part with the X-Ray team. Last summer I performed reliability testing on a new portable X-Ray device, and machined some test fixture components for life testing. I will likely be returning full-time once I graduate.

Contact: mwechara@umich.edu
973-868-2946

13 INFORMATION SOURCES

Because of the unfamiliarity of our group with hydraulic hybrid technology, a significant amount of research was required. Therefore, we began investigating journal articles, previous reports, and websites to gain information about hydraulic systems. Despite our focus on an electric-hydraulic hybrid system, we also investigated the advantages of internal combustion (gas)-hydraulic hybrids. The concept is similar, as the hydraulic system is used for launch assistance while the other electric input is typically used for steady state cruising. An online article describes UPS's implementation of hydraulic hybrids in its delivery trucks [11]. Through our investigations, hydraulic hybrids were identified to be extremely advantageous for stop-and-go situations because of the high power capability of the hydraulic system, which is needed during acceleration. This system integrates a regenerative braking system, which increases the efficiency by redirecting energy dissipation from braking back into the hydraulic system. Furthermore, we noted that hydraulic components used in the large-scale UPS truck were directly downsized for integration into the ZAP! Xebra truck's smaller test bed. Immediately, the similarities between the two systems became obvious.

Additionally, by reviewing various previous ME 450 reports, we gained some understanding of the different types of systems possible (parallel, series, power split) [6]. Establishing a general background about hydraulic technology before we began looking into specific project application, our design process took into consideration previous solutions and their successes and failures.

Although we developed an increased understanding of the components and functions of a hydraulic system, our specific task remained somewhat difficult to comprehend, given the large opportunity and freedom for design improvements on the baseline model. After meeting with Dr. Andrew Moskalik and Mr. David Swain of the EPA (roughly 10 years of experience with hydraulic technology), they clarified and refocused our design objectives immediately [5]. They were able to explain the previous projects with more clarity and outlined the specific responsibilities of our project. In addition to our meeting with the EPA sponsors, we also met with our University of Michigan customers, Dr. Andrej Ivanco and Mr. Xianke Lin, both of whom are ultimately responsible for development and integration of a new hydraulic system into the Blue Xebra truck. A majority of our background knowledge, including up-to-date innovations and progression in technology, were pulled from these personal resources.

In order to establish a baseline for comparison of our vehicle, we considered the purely electric stock (Blue) ZAP! Xebra truck which was mainly sourced from the truck's company website. We also utilized performance information from the modified White Xebra, which was previously the ME450 project truck, to obtain a comparison with a similar hydraulic-electric hybrid. The differences between just these two vehicles established the current progress achieved and gave us targets for design improvements. Notably, we did not benchmark our design against other electric vehicles, such as the ZENN car or GEM car, as previous teams had. This was because we wanted to compare our design to a system with a hydraulic and electric component, not just electric. The modified White Xebra serves as this comparison, and the stock electric Blue Xebra serves as a baseline for the changes made to our specific vehicle. Our objective is to dramatically increase the level of success on the Blue Xebra, as compared to these other vehicle options. The primary objective from this analysis was to use baseline numbers and specifications from the previous reports.

Engineering information useful to our design was procured from multiple sources. Operating specifications for the hydraulic pump/motor including displacement, weight, torque values, and engineering drawings were provided by the EPA courtesy of the pump/motor manufacturer, FEV. Similarly, the specifications for the electric DC motor were provided from the motor manufacturer, via Mr. Xianke Lin. This information was used to specify the operating conditions that our design must fulfill.

Additional engineering information was found for the purpose of concept generation. Specifically, various types of couplings, reduction systems, and bearing considerations [27, 28] were researched to determine the applicability of each to our design. Research included additional patents [19, 20, 21] for tensioning systems and shaft couplings, as well as observations of the previously modified White Xebra truck.

Particular information regarding engineering specifications was obtained from various sources. Specifically, information on the reliability of bearings and chain and sprocket systems was sourced from several vendors and hand calculations. Material selection information was gathered from textbooks [37] [38] [39] and Ashby's material considerations and Cambridge Engineering Selector [29]. Using the textbook material, hand calculations were conducted to verify the validity of our material selection and geometries for various parts including bolts, plates, and tabs. Furthermore, chain length and tension [30] and bearing life calculations [44] were conducted for design choice validation.

Marketing information was also gathered as part of design research in order to determine the availability of components. This was an important consideration because we needed to determine the feasibility of obtaining certain parts, including cost and lead-time. Thus, we directly contacted or consulted online catalogs of several vendors including ALRO Metals Plus, McMaster-Carr [16], Martin Sprocket [30], Motion Industries [23], Stock Drive Products/Sterling Instrument, and SmallParts to determine the availability of our components such as sprockets and couplings. Information on hubs for the shafts was sourced from Hub City Inc., and splined shaft information was found from Diamond Broach Company, Inland Broach & Tool Company, Riverside Spline and Gear Inc., and GROB. Additionally, bearing information and selection was found from The Timken Company. These vendors and manufacturers were found using both personal research and references provided by our EPA contact, David Swain. Additional information related to the external manufacturing of some components was obtained from Lidell Specialty Products Inc. and Protomatic, Inc.

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APPENDIX A: BILL OF MATERIALS

Bill of Materials

Item	Description	Manufacturer/Distributor	Part #	Unit	QTY	Price/QTY	Total Price
Upper Sprocket	25 Tooth ANSI 60 Sprocket	McMaster	2299K77	1	1	$29.13	$29.13
Lower Sprocket 1	14 Tooth ANSI 60 Sprocket	McMaster	2299K65	1	1	$21.70	$21.70
Lower Sprocket 2	18 Tooth ANSI 60 Sprocket	McMaster	2299K69	1	2	$24.94	$49.88
Lower Sprocket 3	20 Tooth ANSI 60 Sprocket	McMaster	2299K72	1	1	$25.44	$25.44
Chain	ANSI 60 Chain 2 Ft	McMaster	6261K472	1	1	$11.06	$11.06
Chain Connecting Link	ANSI 60 Chain	McMaster	6261K195	1	8	$1.21	$9.68
Chain Adding Link	ANSI 60 Chain	McMaster	6261K245	1	8	$1.25	$10.00
Bearing	1.5ID Bearing	McMaster	60355K512	1	3	$33.24	$99.72
Tapered Bearing Inner	1.5ID Tapered	McMaster	5709K270	1	1	$25.87	$25.87
Tapered Bearing Outer	3"OD outer Race	McMaster	5709K64	1	1	$12.38	$12.38
Gearbox Sides/DC Plate	1/2" A36 plate 1'x4'	ALRO		1	1	$50.00	$50.00
Gearbox Pump Plate	1" A36 plate 1'x1.5'	ALRO		1	1	$50.00	$50.00
Gearbox Cutout Piece	1/4" A36 Plate 1'x1'	ALRO		1	1	$17.15	$17.15
Hub Stock	4" 1018 Round 6"L	ALRO		1	2	$19.80	$39.60
Shaft Stock	2" 1018 Round 2'L	ALRO		1	1	$30.80	$30.80
Subframe Main Rail	2"x1.5"x.1875"T 4'L	ALRO		1	1	$13.20	$13.20
Collars	9/16" 1.5"ID 2.375"OD	McMaster	6436K23	1	2	$6.17	$12.34
Gearbox Dowel Pins	1/4"D 1"L	McMaster	98381A542	50	1	$9.85	$9.85
Gearbox Dowel Pins - Long	1/4"D 1.5"L	McMaster	98380A546	10	2	$11.70	$23.40
Gearbox Bolts - Long	1/4"D 1.25"L	McMaster	94912A470	5	3	$5.53	$16.59
PM Mount Bolts	1/2"-13 D, 1 1/4" L	McMaster	92865A714	25	1	$9.19	$9.19
PM Mount Bolt Washers	1/2" ID, 1 1/8" OD	McMaster	98370A033	5	1	$9.58	$9.58
UHMW Tensioner Plastic	2"D 1'L UHMW	ALRO		1	1	$11.24	$11.24
Hub Shoulder Bolts	10-24, 1/4 D, 3/4 L	McMaster	91259A535	1	4	$0.98	$3.92
Hub Bolts	1/4"-20 D, 1" L	McMaster	90128A247	25	2	$6.13	$12.26
Tab Bolts	1/2" - 13 D, 2 3/4" L	McMaster	91251A723	5	1	$5.52	$5.52
Tensioner Square Nuts	3/8" Nut	McMaster	92891A300	10	1	$9.63	$9.63
Tensioner Bolts	3/8"-16 D, 2" L	McMaster	92196A632	10	1	$7.17	$7.17
Key Stock	3/8" x 3/8" x 12"	McMaster	98535A170	1	1	$7.42	$7.42
DC Bolts	M8x1.25 Bolts	Stadium Hardware		1	15	$0.57	$8.53
Reduction Housing Machining		Lidell Maching Services			1	$1,855.00	$1,855.00
Splining Shafts		Vertical Machine Services			1	$225.00	$225.00
						TOTAL:	$2,722.25

APPENDIX B: ENGINEERING CHANGES

Multiple engineering changes were made between the final design described in Design Review 3 and the final, prototyped design. The changes were made for ease of manufacture (time savings and tool availability), assembly fit-up, testing, necessary design modifications for the proper function of the design, or other reasons. All engineering changes are detailed in this appendix. For each, a summary is provided that details the change and rationale, a drawing that outlines the change, the parts impacted, an analysis of the effect of the change (if necessary), and the change authorization.

ACCESS DOOR

Change and Rationale:

For assembly and functional use (accessibility), slots were machined into the access door. These were added to allow for the door to be more easily removed using the milled slots to pull or pry on. This aids the accessibility for gear ratio changes.

Sketch:

See attached.

Part Impacted:

This change affects only the access door plate of the gear reduction case. It allows for easier removal of this plate from the rest of reduction case.

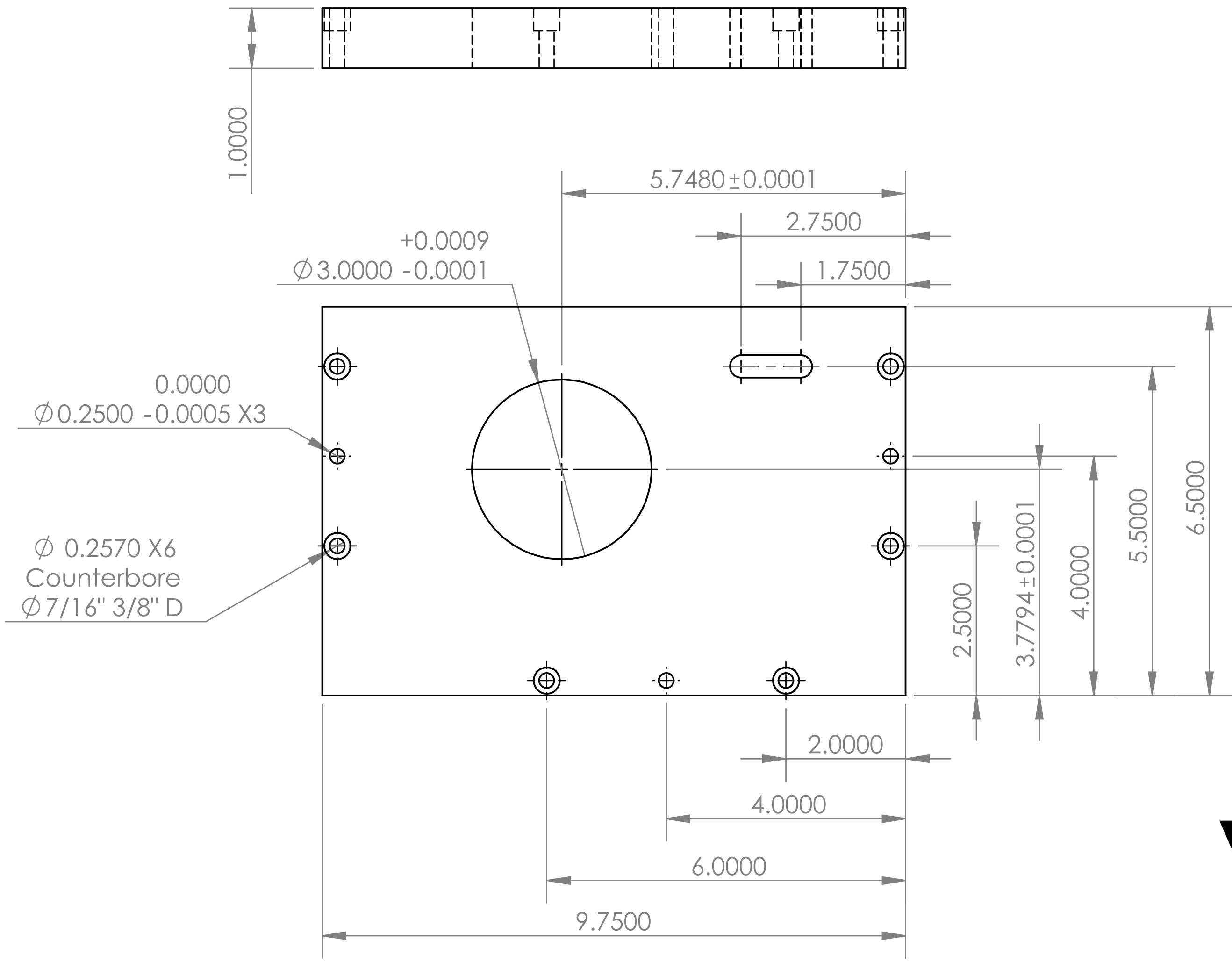
Analysis:

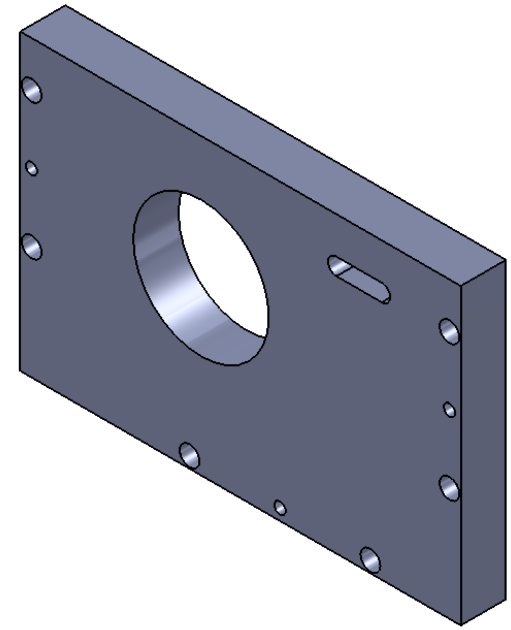
No structural or static analysis was necessary because the location and/or size of the slots do not affect the structural integrity of the plate or gear reduction case.

Change Authorization:

TEAM – Anuj Shah: 12/3/10

SPONSOR – Andrew Moskalik:





All 1/4" Holes are
1/4" from nearest
edge

WAS

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
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			CHECKED			Reduction Case: Access Door
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:3 · WEIGHT: · SHEET 1 OF 1

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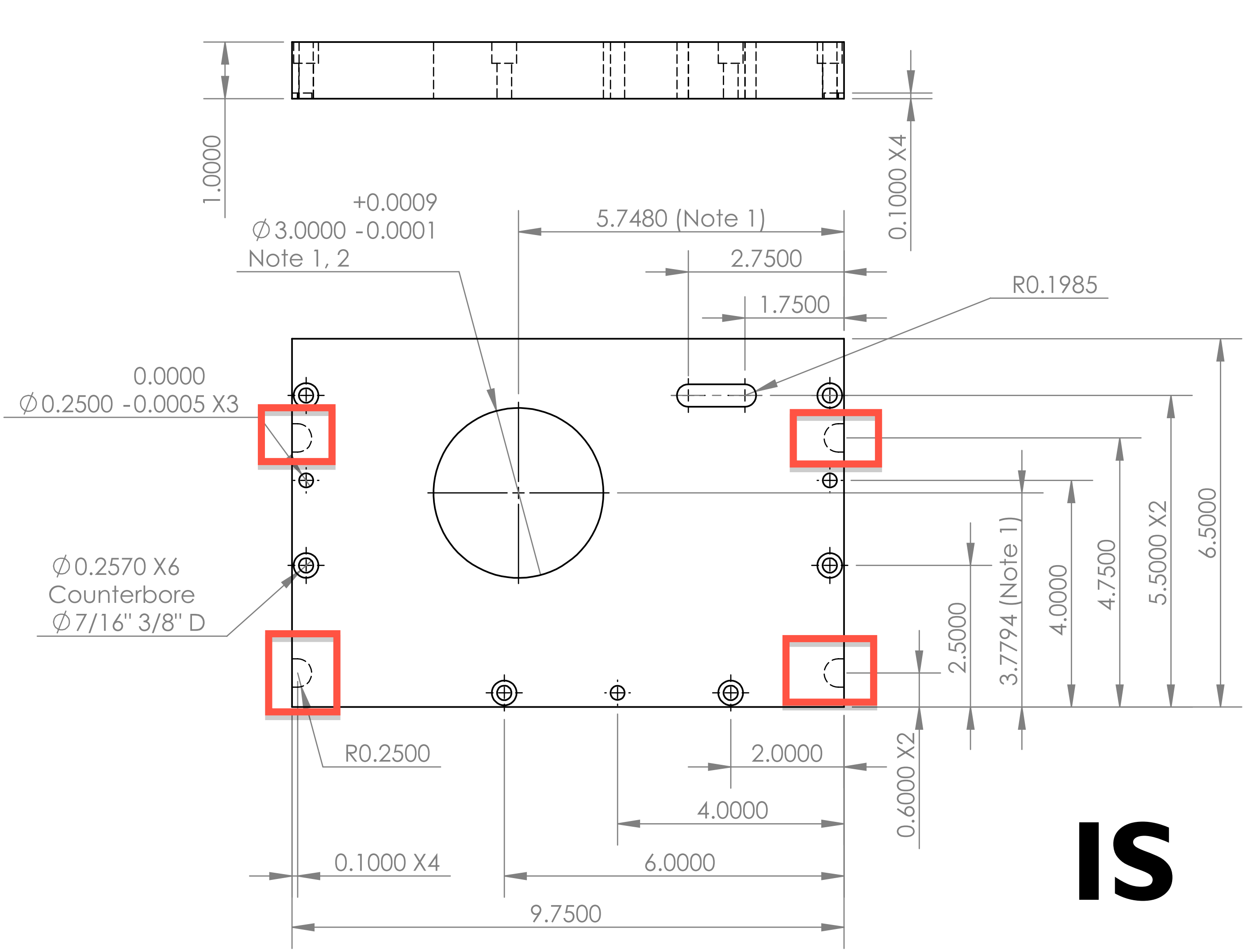

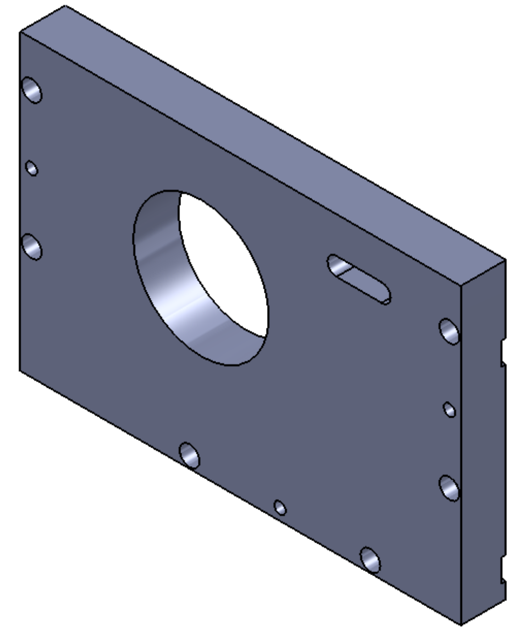

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All 1/4" Holes are
1/4" from nearest
edge

Note 1: Concentric ±**0.001in**
to bore on DC Plate
(opposite side of case)

Note 2: Bore hole for tapered
bearing

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL A36 STEEL

FINISH

DO NOT SCALE DRAWING

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DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

Reduction Case:
Access Door

SIZE	DWG. NO.	REV
A		1

SCALE: 1:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

DC MOTOR HUB

Change and Rationale:

The DC motor hub changed from a 3-tiered step down design to a 2-tiered step down design. The top-most tier (closest to the bearing on the pump/motor side of the gear reduction case) was eliminated and the remaining two tiers were increased in length. As a result, the overall length of the hub was also increased.

These changes were made for ease of machining and as a result of a design change carryover. The 3-tiered design would have been more difficult to manufacture than the 2-tiered design. Therefore, it was reasoned that since the third tier (previously implemented for the alignment of the hub to the bearing on the pump/motor side of the gear reduction case) was not actually necessary. Instead, the outer diameter of top-most tier (of the 3-tiered design) was used as the outer diameter of the new 2-tiered design. This allowed for the same alignment with the bearing. The overall length did not affect his alignment since the additional length was simply extended along the DC motor shaft.

Previously, the DC motor shaft had been made of two pieces. However, after the design change was made (as of the DR3 final report) to create the shaft out of one piece, the DC motor hub had not been changed accordingly. These changes reflect the corresponding shaft change.

Sketch:

See attached.

Part Impacted:

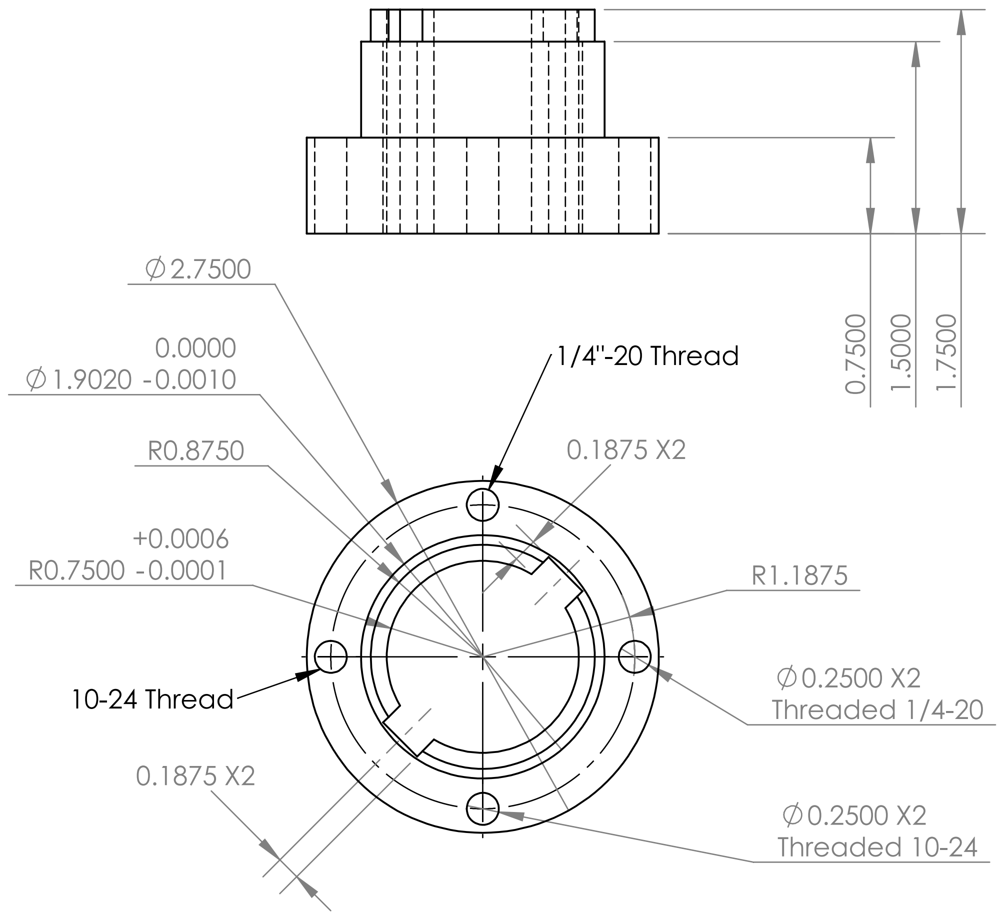
These changes affect the DC motor shaft because the hub mounts directly to the shaft through two keys. Therefore the keys needed to be changed as discussed in the DC motor shaft keys document. Although the length of the hub increased, the fit-up with the shaft remains the same. There is simply more engagement surface area which may affect the difficult of assembly (press-fit).

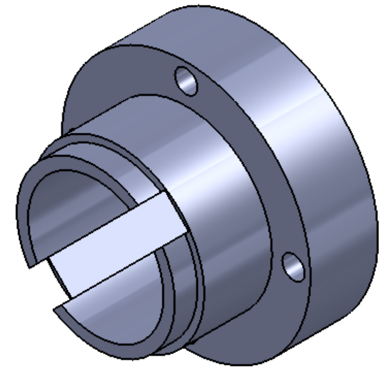
Analysis:

No new analysis was necessary because more material was added, and thus the torsional strength increases as well.

Change Authorization:

TEAM – Mat Wecharatana: 11/15/10
SPONSOR – Andrew Moskalik:





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			CHECKED			
			ENG APPR.			DC Hub
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 1018 STEEL	COMMENTS:			SIZE A	DWG. NO.	REV 1
	NEXT ASSY	USED ON	FINISH				
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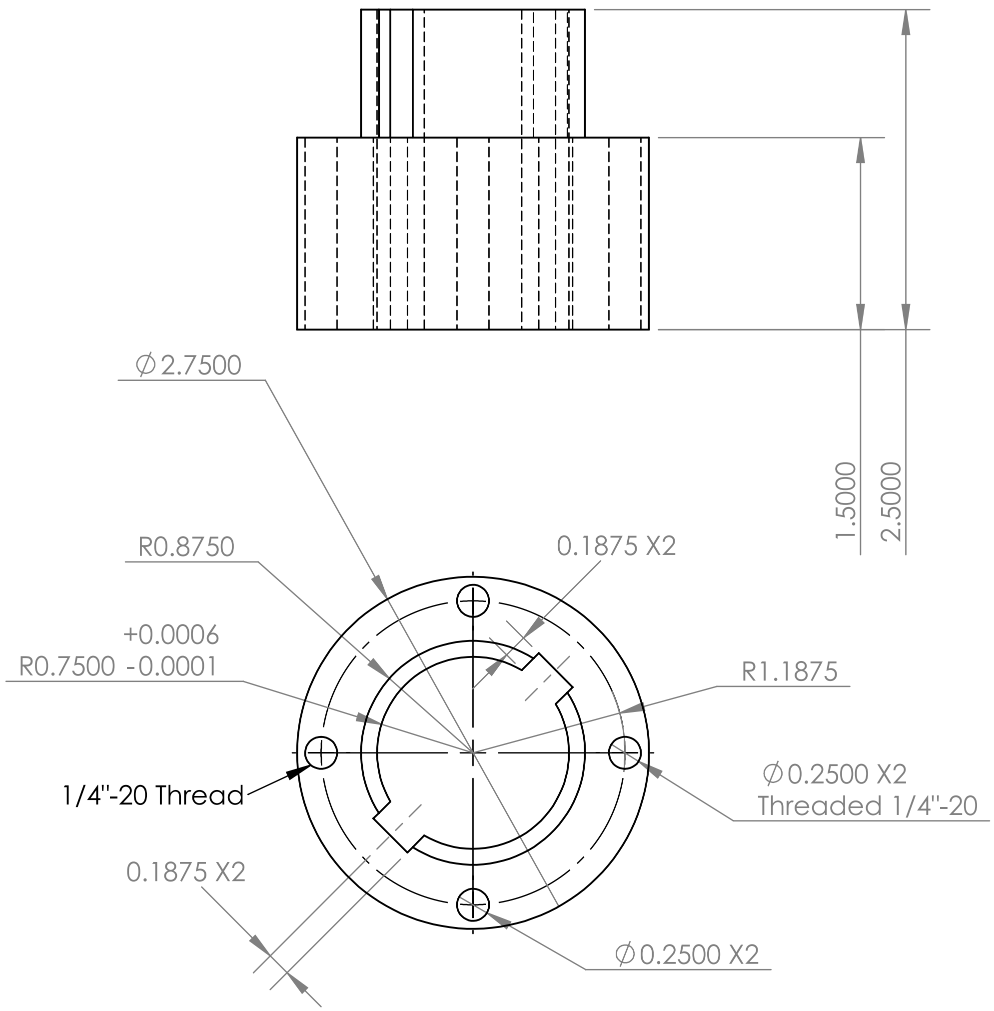

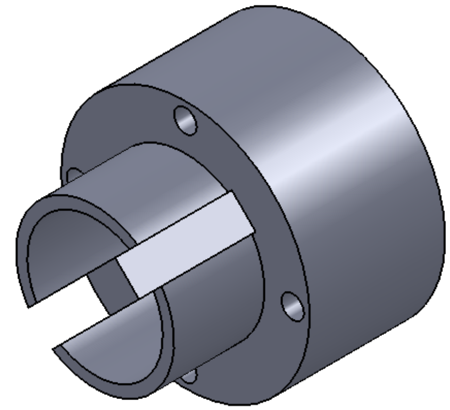

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third tier is eliminated and length

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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

TITLE:

DC Hub

SIZE	DWG. NO.	REV
A		1

SCALE: 2:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

DC MOTOR SHAFT KEYS

Change and Rationale:

The length of the keys has been decreased from 1.75" to 1.375". This change is tied directly to the changes to the DC motor hub and the replacement of the two-piece DC shaft with the one-piece DC shaft. This decrease in length is due to the elimination of a gap between the two pieces of the shaft (because the design changed to a one-piece design). Accordingly, the increase in length of the larger diameter tier of the DC hub allowed for a shorter key to engage only this tier, rather than the two larger diameter tiers of the 3-tiered design (please see DC Motor Hub analysis for more information).

Sketch:

See attached.

Part Impacted:

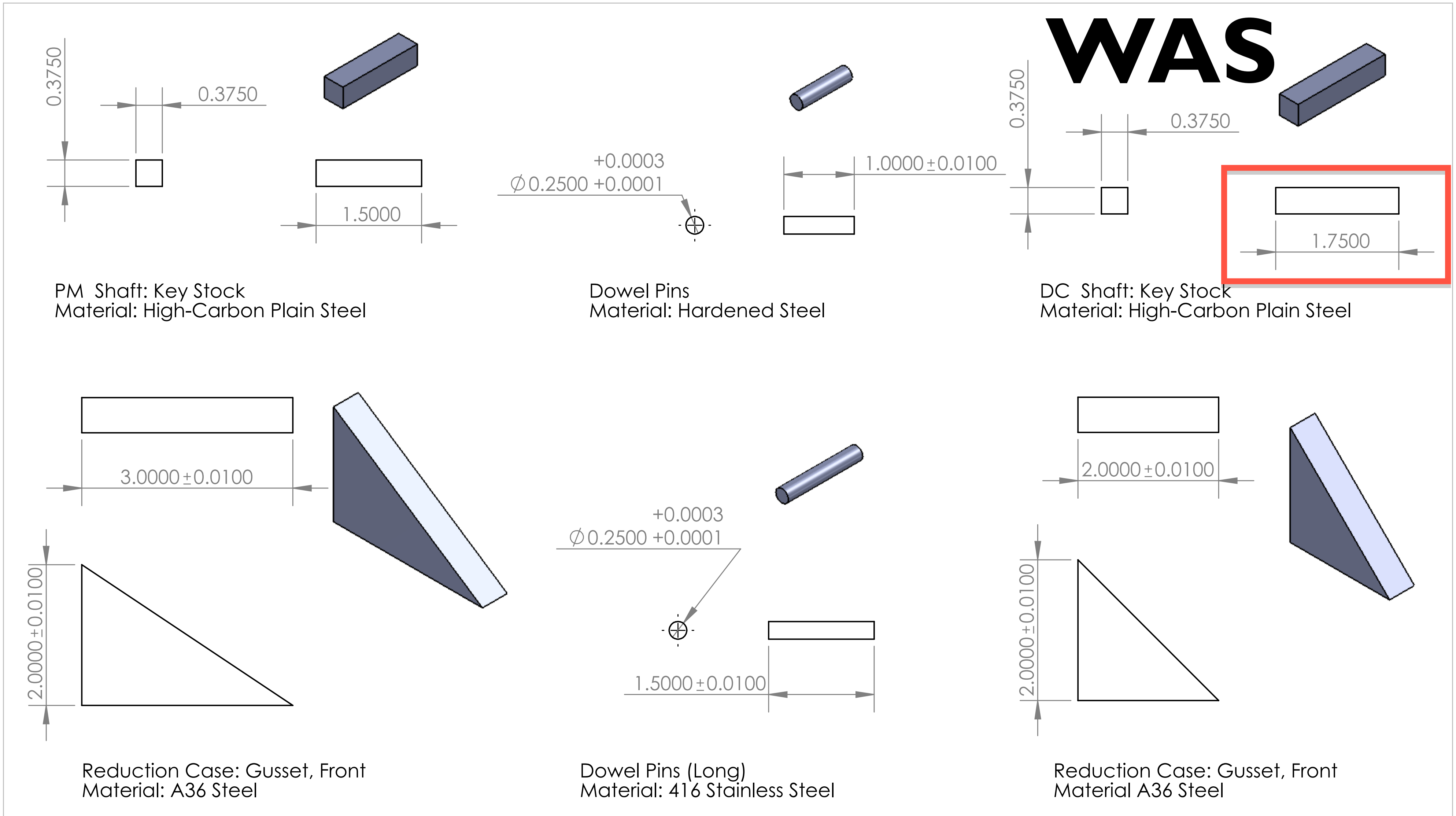
These changes affect the DC motor hub and the shaft to which they keys are directly attached. Although the length is reduced, no changes need to be made to the shaft because the keyways were already long enough to handle the previously longer keyways. As mentioned before, the change in the key length was directly related to the DC motor hub changes.

Analysis:

Analysis on the new key length was conducted. Please see attached.

Change Authorization:

TEAM – Mat Wecharatana: 11/15/10
SPONSOR – Andrew Moskalik:



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
SPECIFIED PER PART

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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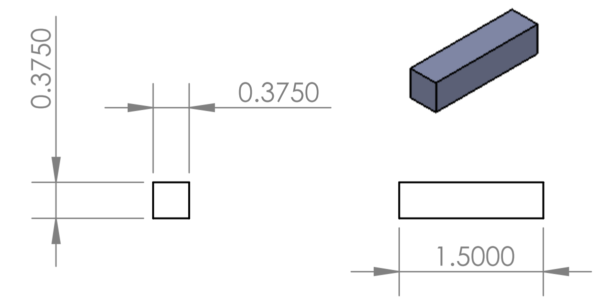
TITLE:

Misc Parts: Gussets, Keys, Dowel Pins

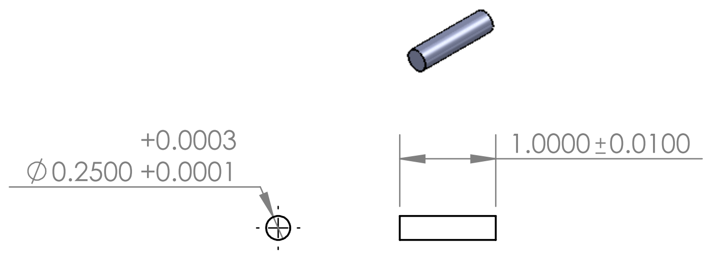
SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

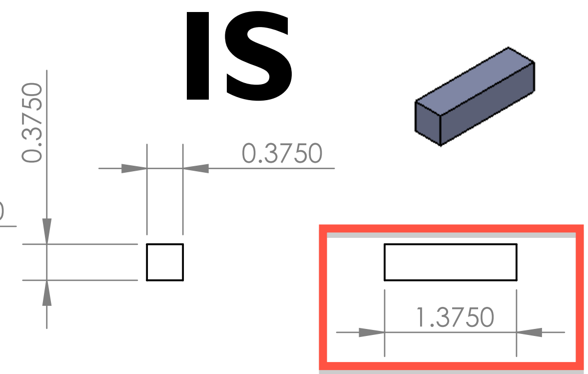
5 4 3 2 1



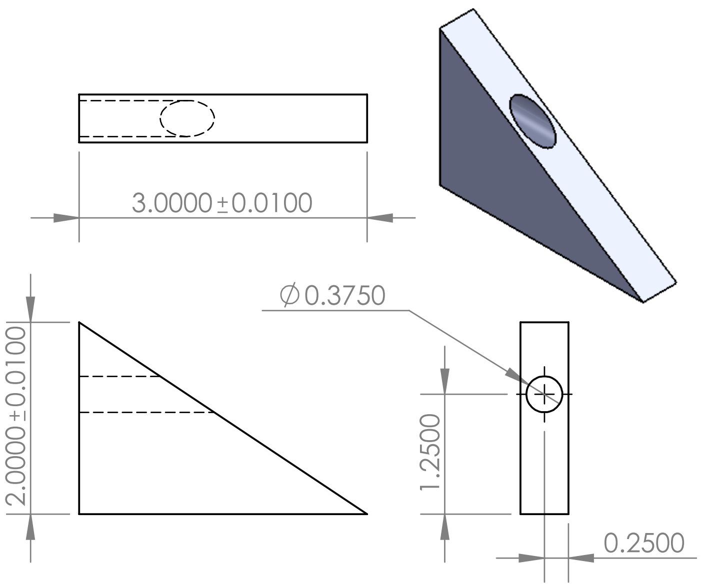
PM Shaft: Key Stock
Material: High-Carbon Plain Steel



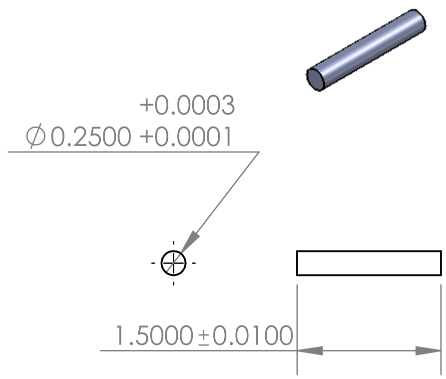
Dowel Pins
Material: Hardened Steel



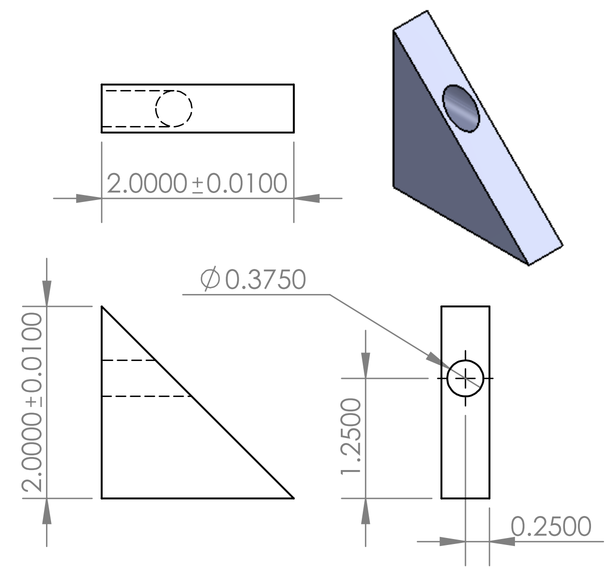
DC Shaft: Key Stock
Material: High-Carbon Plain Steel



Reduction Case: Gusset, Front
Material: A36 Steel



Dowel Pins (Long)
Material: 416 Stainless Steel



Reduction Case: Gusset, Front
Material A36 Steel

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
	CHECKED			Misc Parts: Gussets, Keys, Dowel Pins
	ENG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	MFG APPR.			
	Q.A.			
MATERIAL SPECIFIED PER PART	COMMENTS:			SIZE A · DWG. NO. · REV 1
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

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Key Analysis

Pump/Motor Shaft			Key Dimensions		
T	279.5	N-m	L	0.0381	m
d	0.01905	m	t	0.009652	m
F	14671.92	N			
τ	39897395	Pa			
σx	79794791	Pa			
σy	0				
σz	0				
σ1	96320833				
σ2	-1.7E+07				
σ3	0				
σH	1.06E+08				
	105.5586				
Yield Strength high Carbon steel	495				
Safety Factor for Keys	4.689339				

DC Motor Shaft			Key Dimensions		
T	223.6	N-m	L	0.034925	m
d	0.01905	m	t	0.009652	m
F	11737.53	N			
τ	34819545	Pa			
σx	69639090	Pa			
σy	0				
σz	0				
σ1	84061818				
σ2	-1.4E+07				
σ3	0				
σH	92123857				
	92.12386				
	495				
Safety Factor for Keys	5.373201				

DC MOTOR SPROCKETS

Change and Rationale:

For ease of manufacturing (availability of tooling), the inner bore diameter of each of the DC motor sprockets (2.5:4.5, 3.3:4.5, 3.6:4.5) was increased to 2" from 1.9020".

Sketch:

See attached.

Part Impacted:

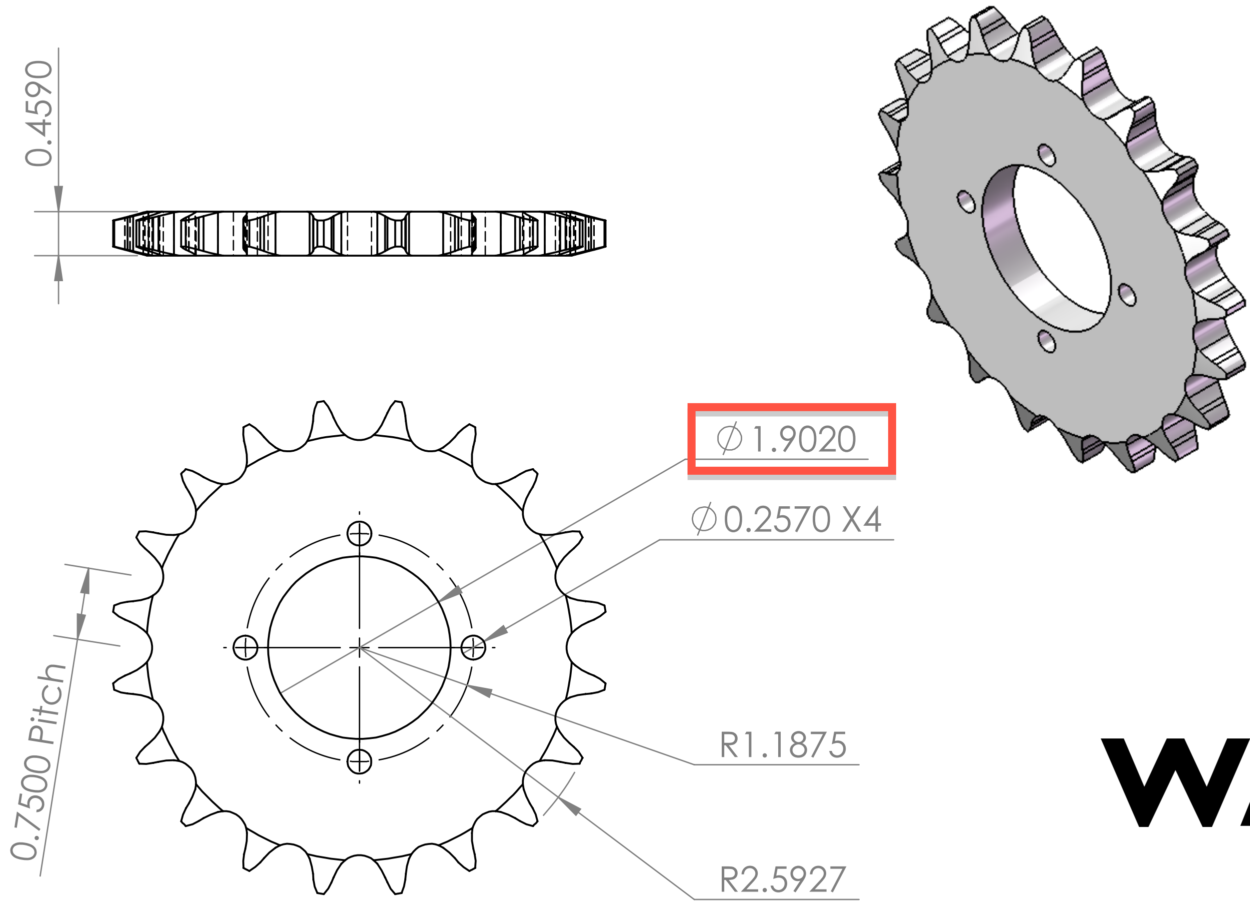
This change impacts no other parts. Although each sprocket is attached directly to the DC motor hub, the inner bore diameter does not contact the hub itself. The inner bore hole of sprocket is oversized and kept in place using dowel pins and bolts so that no contact occurs with the hub.

Analysis:

No additional analysis is necessary because no load is expected to be placed to this inner bore diameter. Although the amount of material between the inner bore and the drilled holes is decreased, this is not a concern because any loading will occur in the radially outward and angular directions. The holes and bolt/dowel connects will remain the same in these locations, even if material has been removed closer to the radially inward direction.

Change Authorization:

TEAM – David Fok: 11/22/10
SPONSOR – Andrew Moskalik:



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			CHECKED			3.6:4.5 DC Sprocket
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

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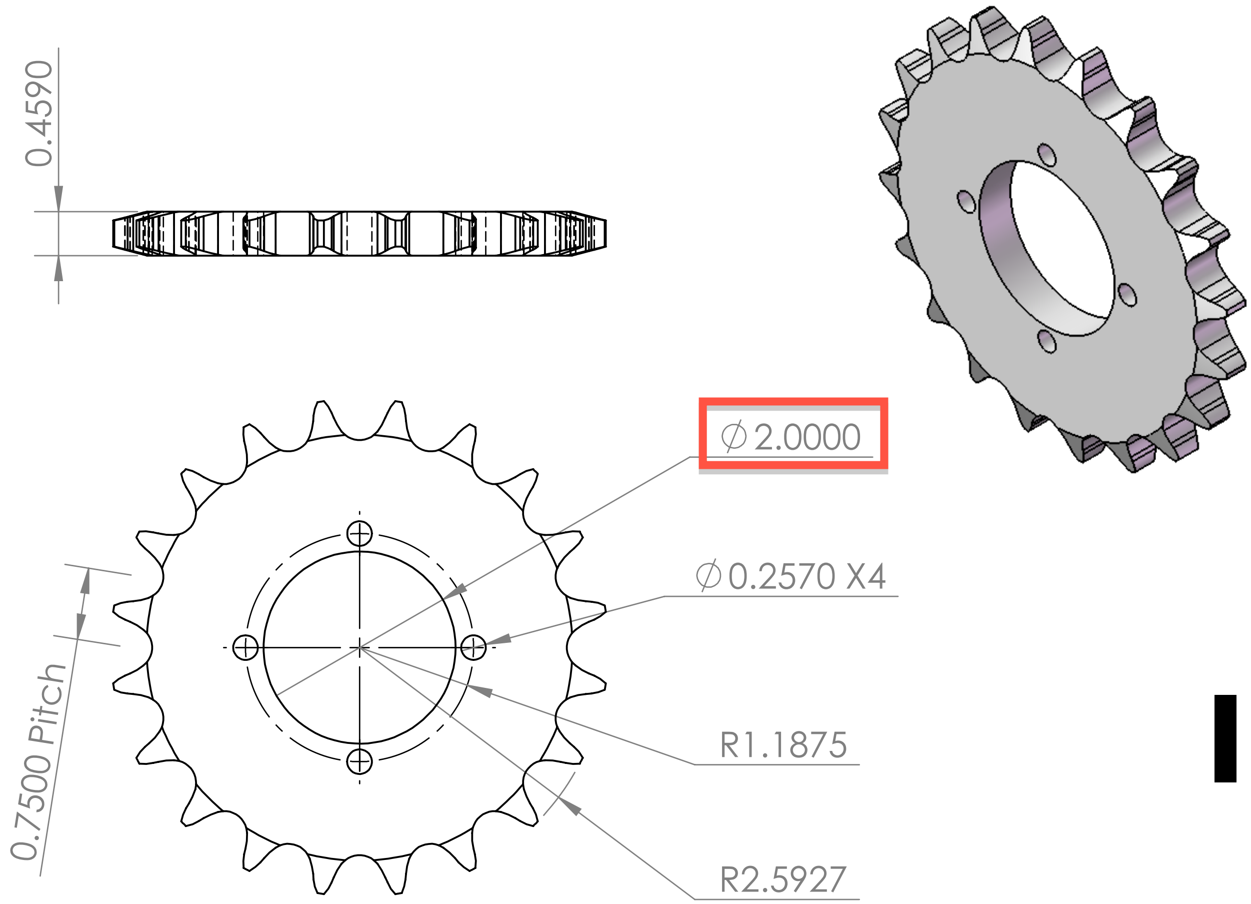

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		INTERPRET GEOMETRIC TOLERANCING PER:
		MATERIAL STEEL
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ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

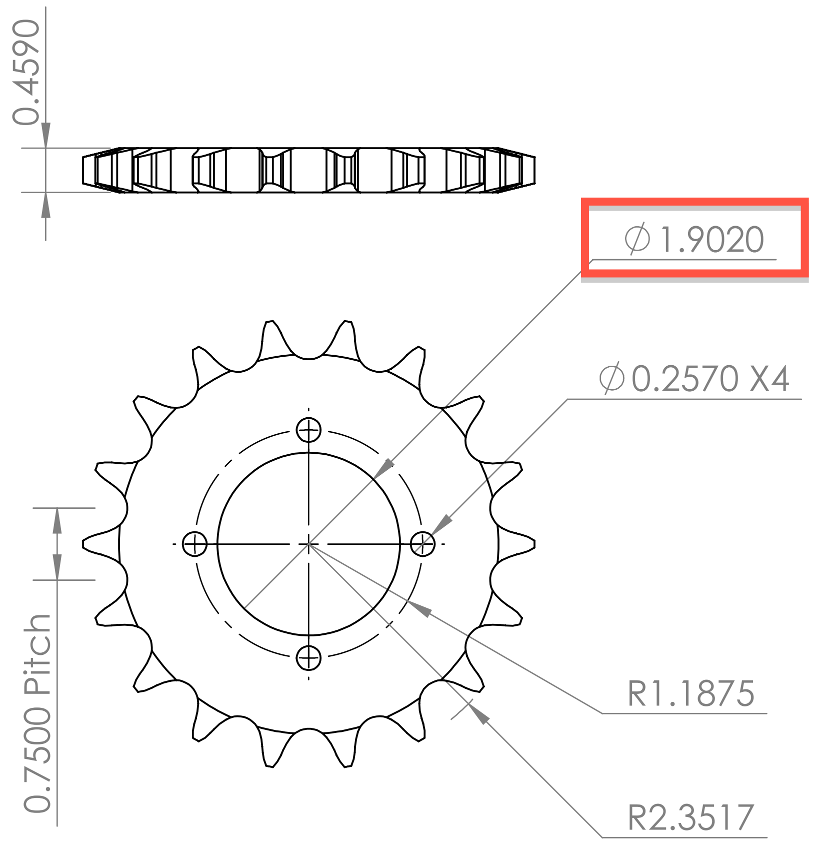
TITLE:

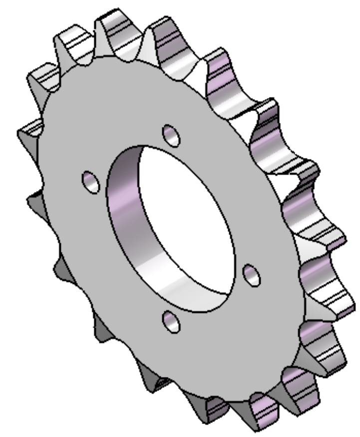
3.6:4.5 DC Sprocket

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1





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			CHECKED			3.3:4.5 DC Sprocket
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

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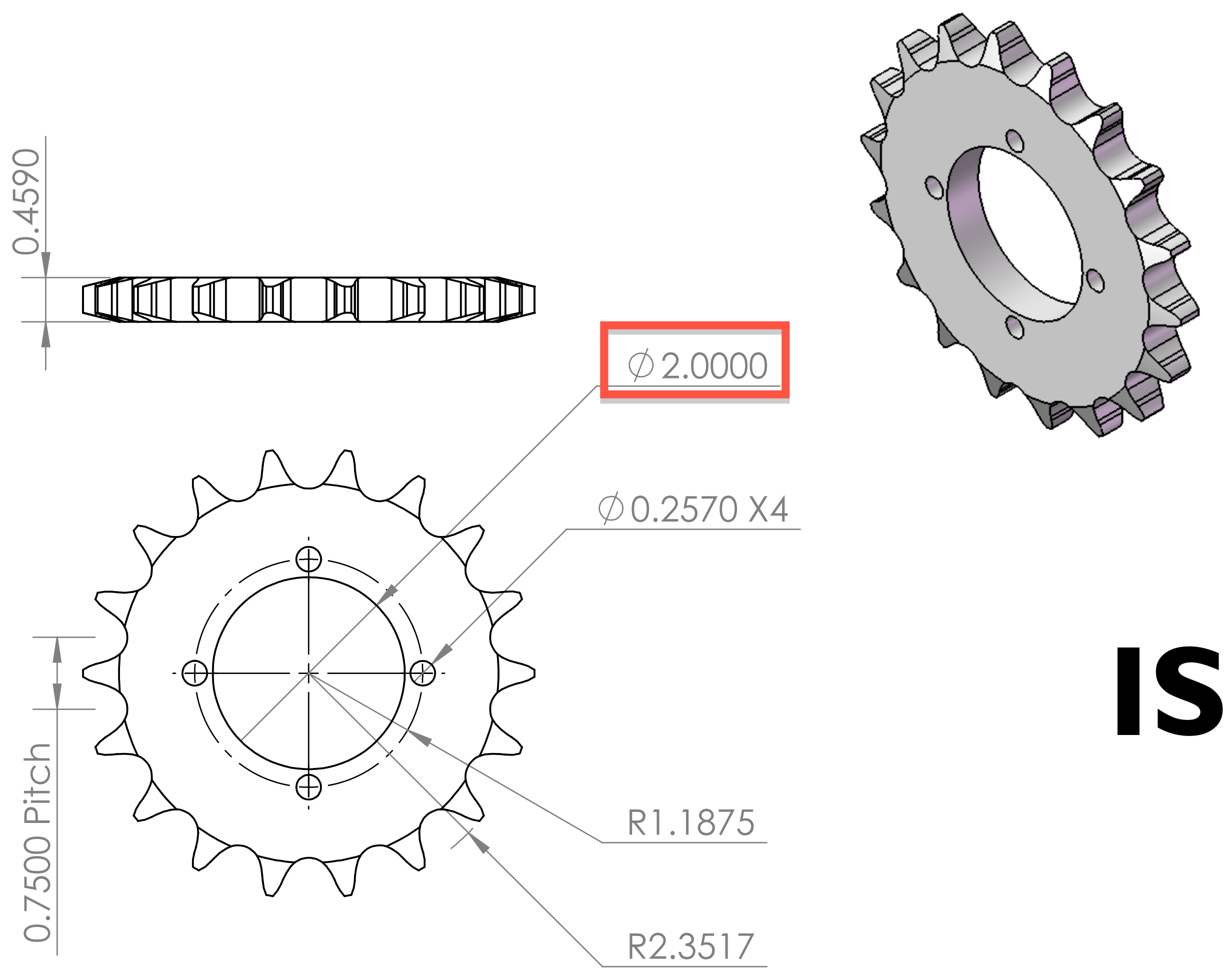

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	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STEEL

FINISH

NEXT ASSY	USED ON
APPLICATION	

DO NOT SCALE DRAWING

U of M Xebra Team Fall 2010

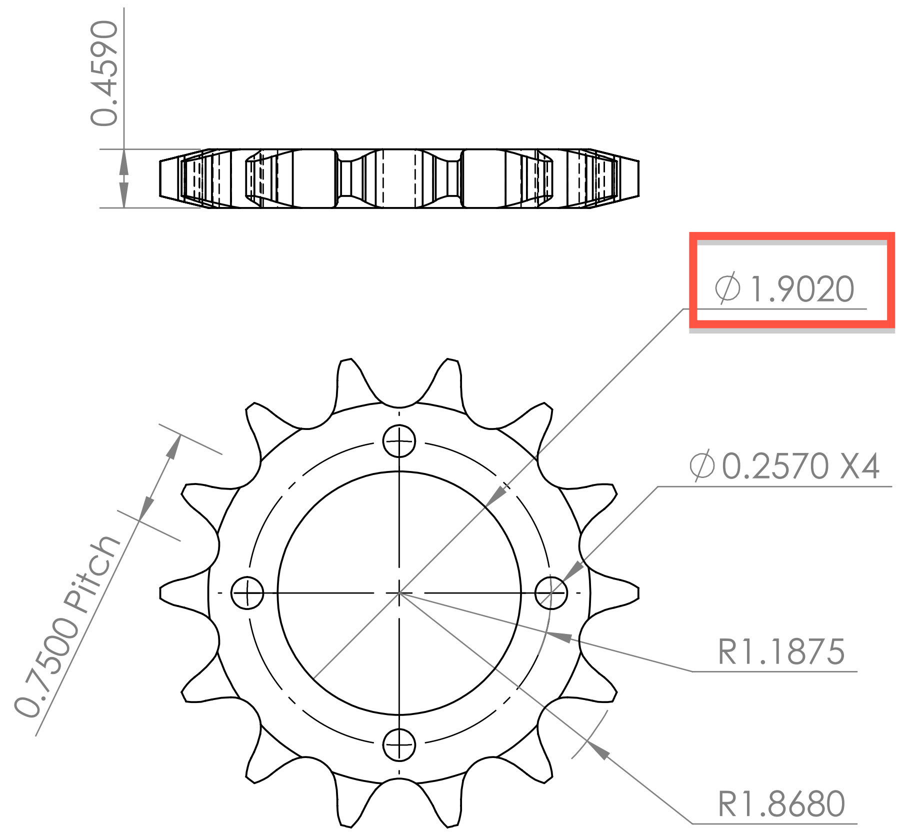
TITLE:

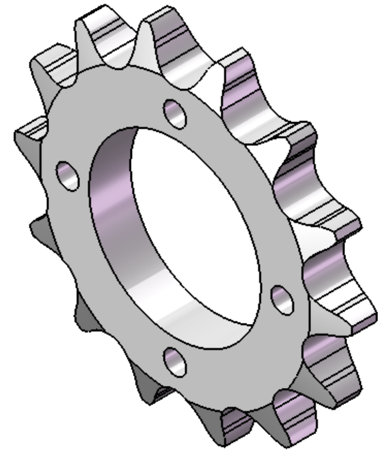
3.3:4.5 DC Sprocket

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1





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			CHECKED			2.5:4.5 DC SPROCKET
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A — DWG. NO. — REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE 2:3 — WEIGHT: — SHEET 1 OF 1

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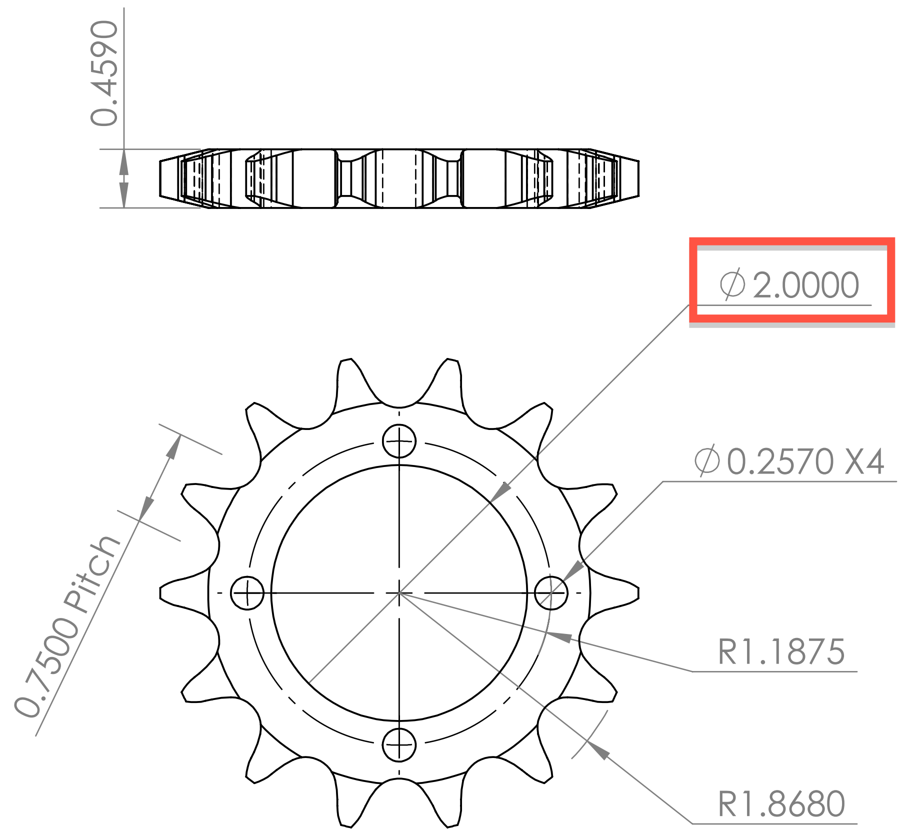
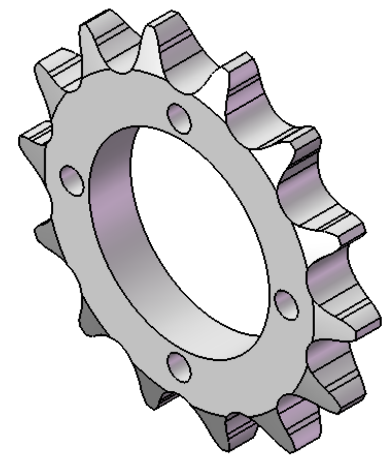




IS

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			2.5:4.5 DC Sprocket
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE 2:3 / WEIGHT: / SHEET 1 OF 1

5 4 3 2 1

GEAR REDUCTION CASE TOP LID

Change and Rationale:

Bolt holes originally intended to be drilled into the top plate were neglected during outsourced manufacturing by Lidell. This change was unintended, but did not affect the fit-up or structural integrity of the gear reduction case.

Sketch:

See attached.

Part Impacted:

The lack of drill holes in the top lid resulted in no longer needing the corresponding drill holes in the two side plates.

Analysis:

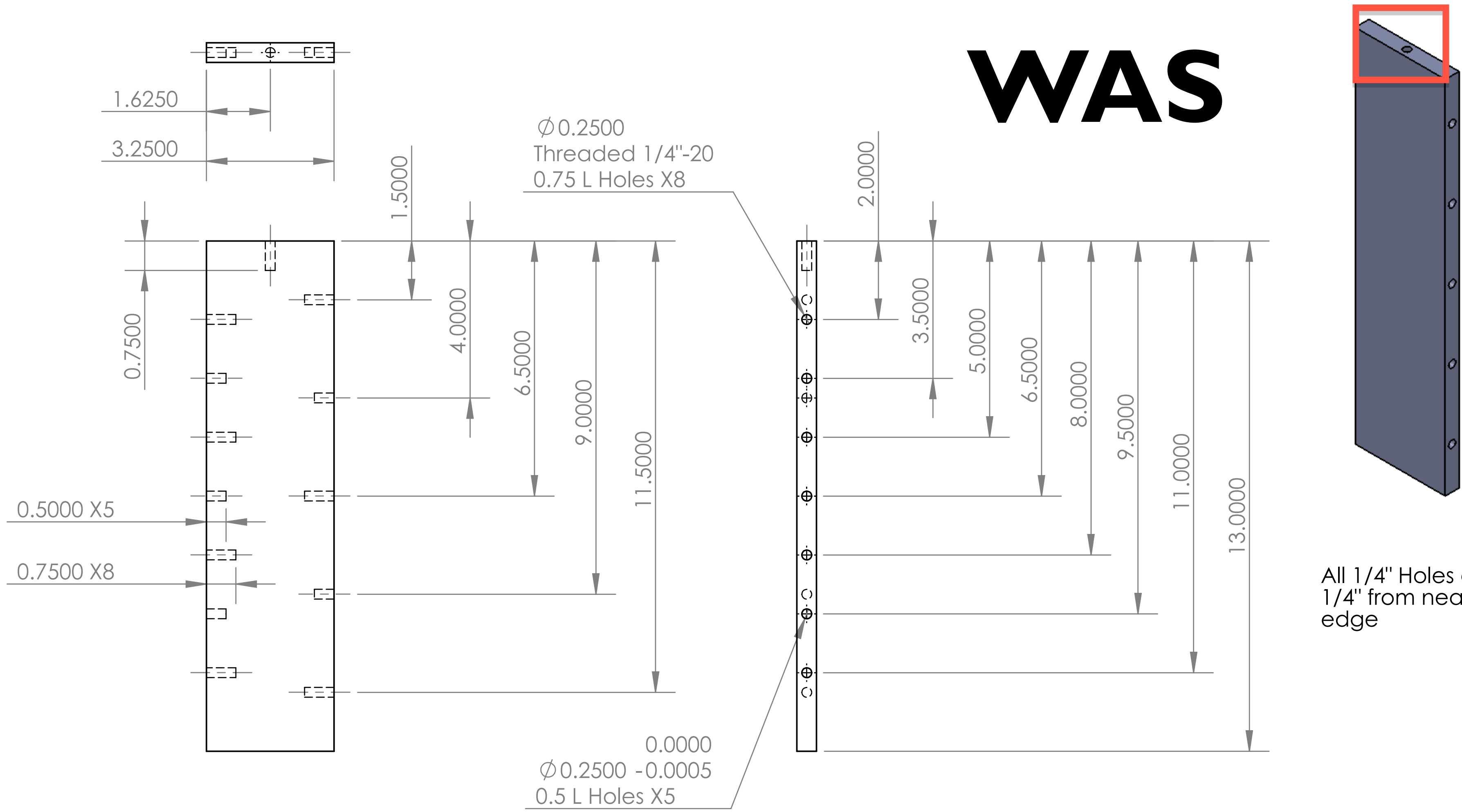
No additional analysis was necessary because the top lid of the gear reduction case still functions the same and is non-load bearing.

Change Authorization:

TEAM – Steve Lidell: 11/30/10 (NOTE: changes were not pre-approved by the team before outsourced machining)

SPONSOR – Andrew Moskalik:

WAS

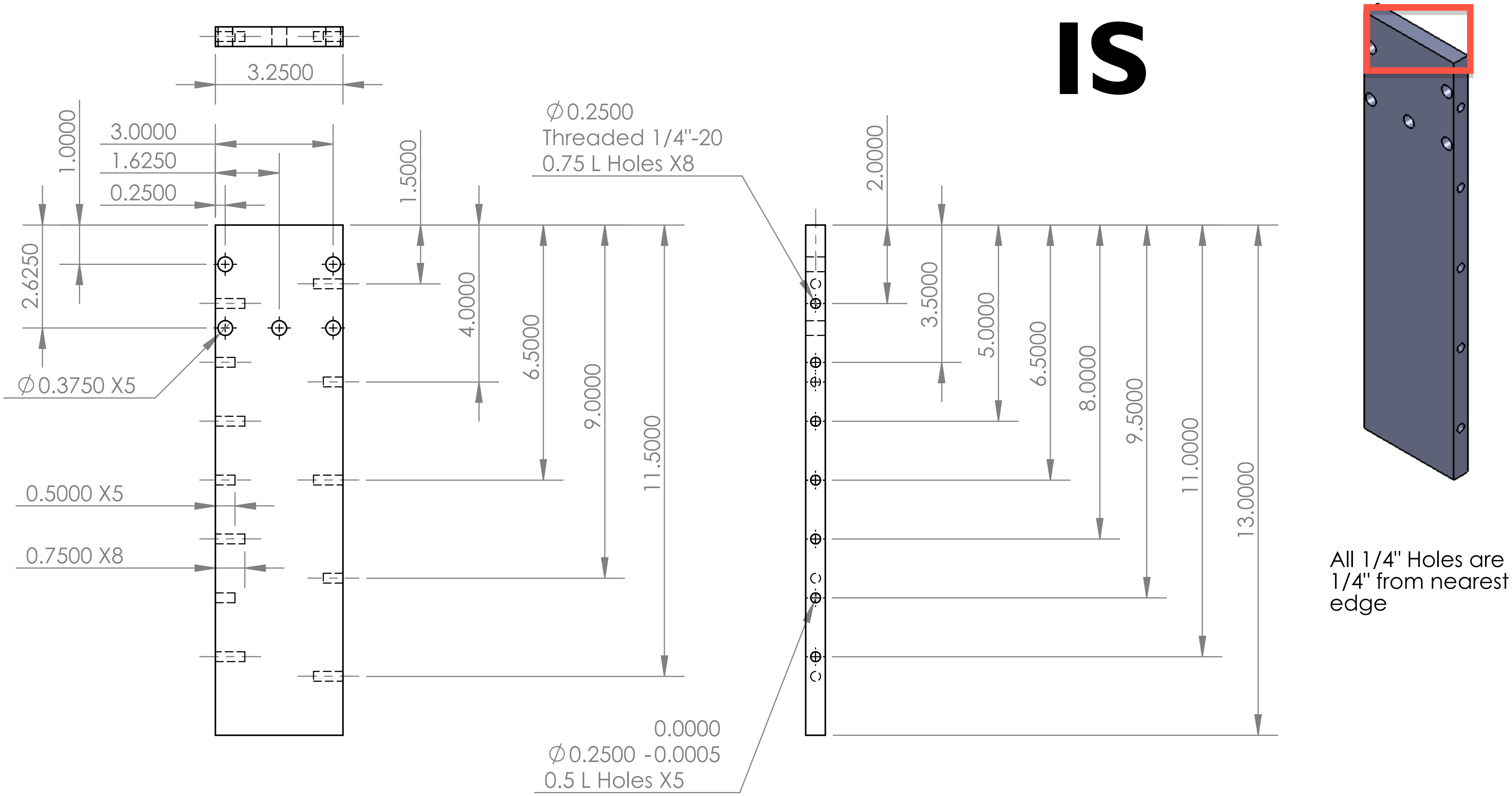


All 1/4" Holes are 1/4" from nearest edge

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:
			CHECKED			Reduction Case: Side Back
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4 / WEIGHT: / SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
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	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL A36 STEEL	COMMENTS: Threads not shown.		
FINISH			
DO NOT SCALE DRAWING			

NEXT ASSY	USED ON
APPLICATION	

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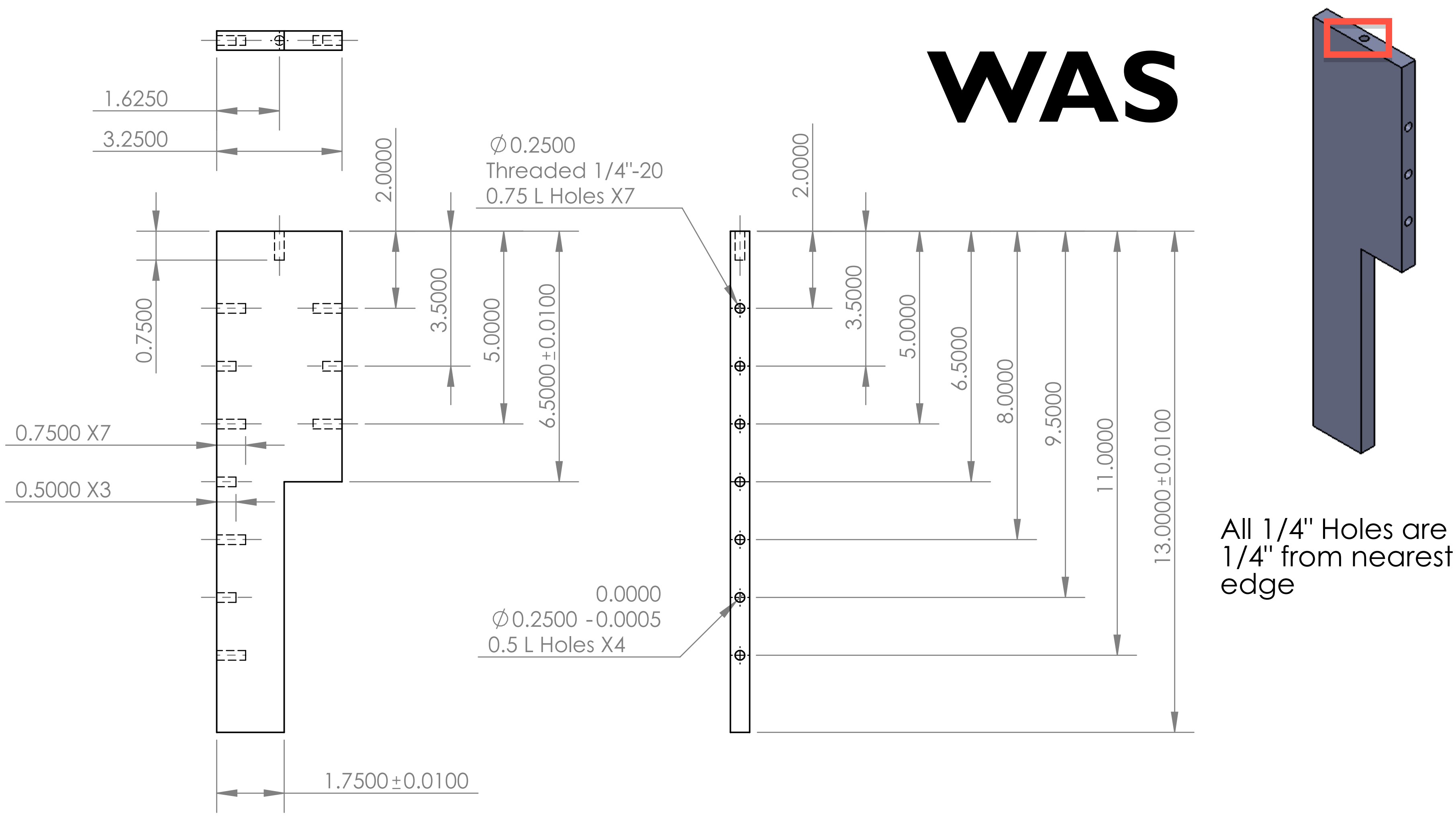
U of M Xebra Team Fall 2010

TITLE:

Reduction Case: Side Back

SIZE	DWG. NO.	REV
A		1
SCALE: 1:4	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



				NAME	DATE	
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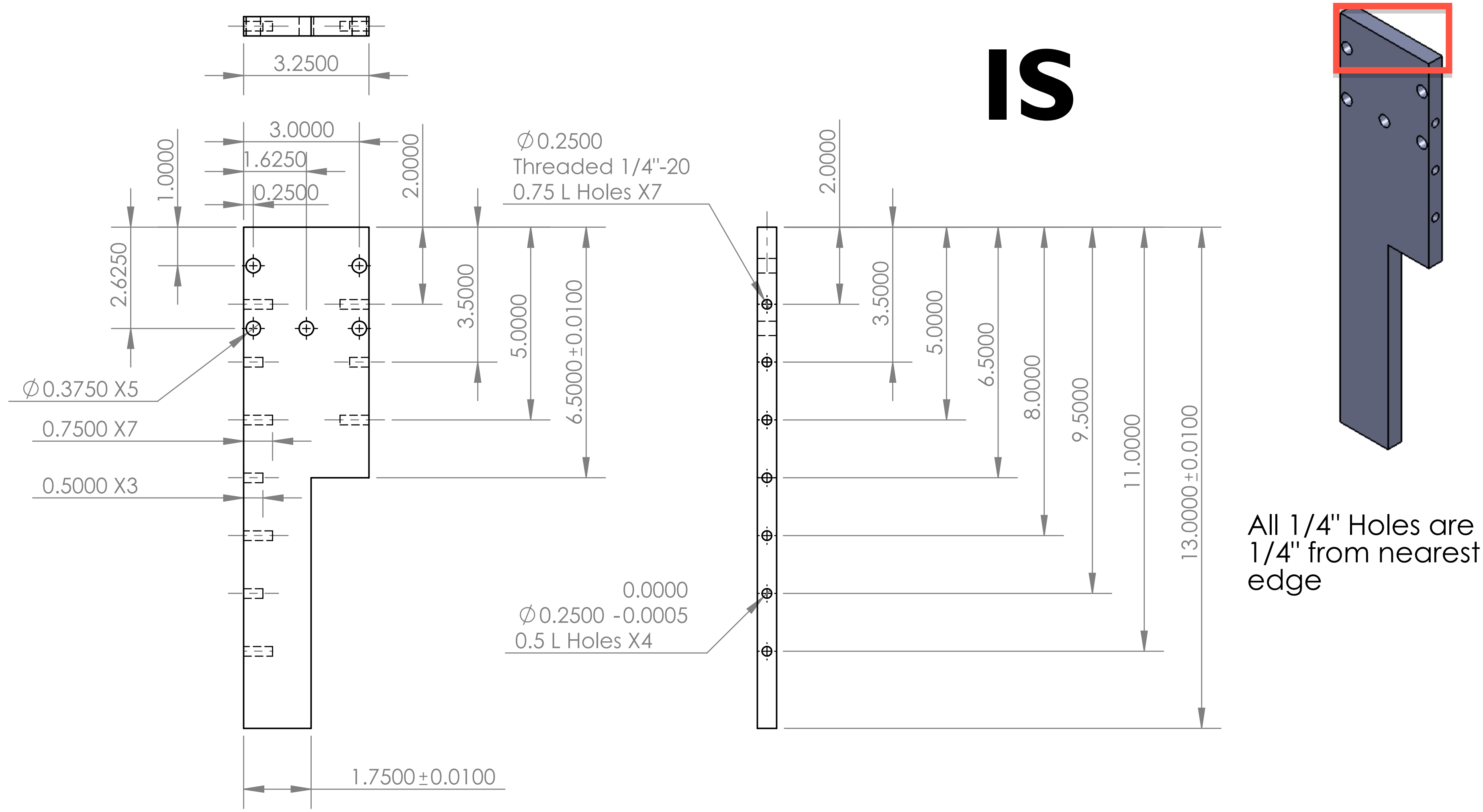
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:
			CHECKED			
			ENG APPR.			Reduction Case: Side Front
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



		NAME	DATE	U of M Xebra Team Fall 2010
	DRAWN	MW	12/6	TITLE:
	CHECKED			Reduction Case: Side Front
	ENG APPR.			
	MFG APPR.			
	Q.A.			
	COMMENTS: Threads not shown.			SIZE A — DWG. NO. — REV 1
				SCALE: 1:4 — WEIGHT: — SHEET 1 OF 1

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL A36 STEEL

FINISH

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE DRAWING

5 4 3 2 1

PUMP/MOTOR HUB

Change and Rationale:

For ease of machining, a through hole was used for the two .25” holes.

Sketch:

See attached.

Part Impacted:

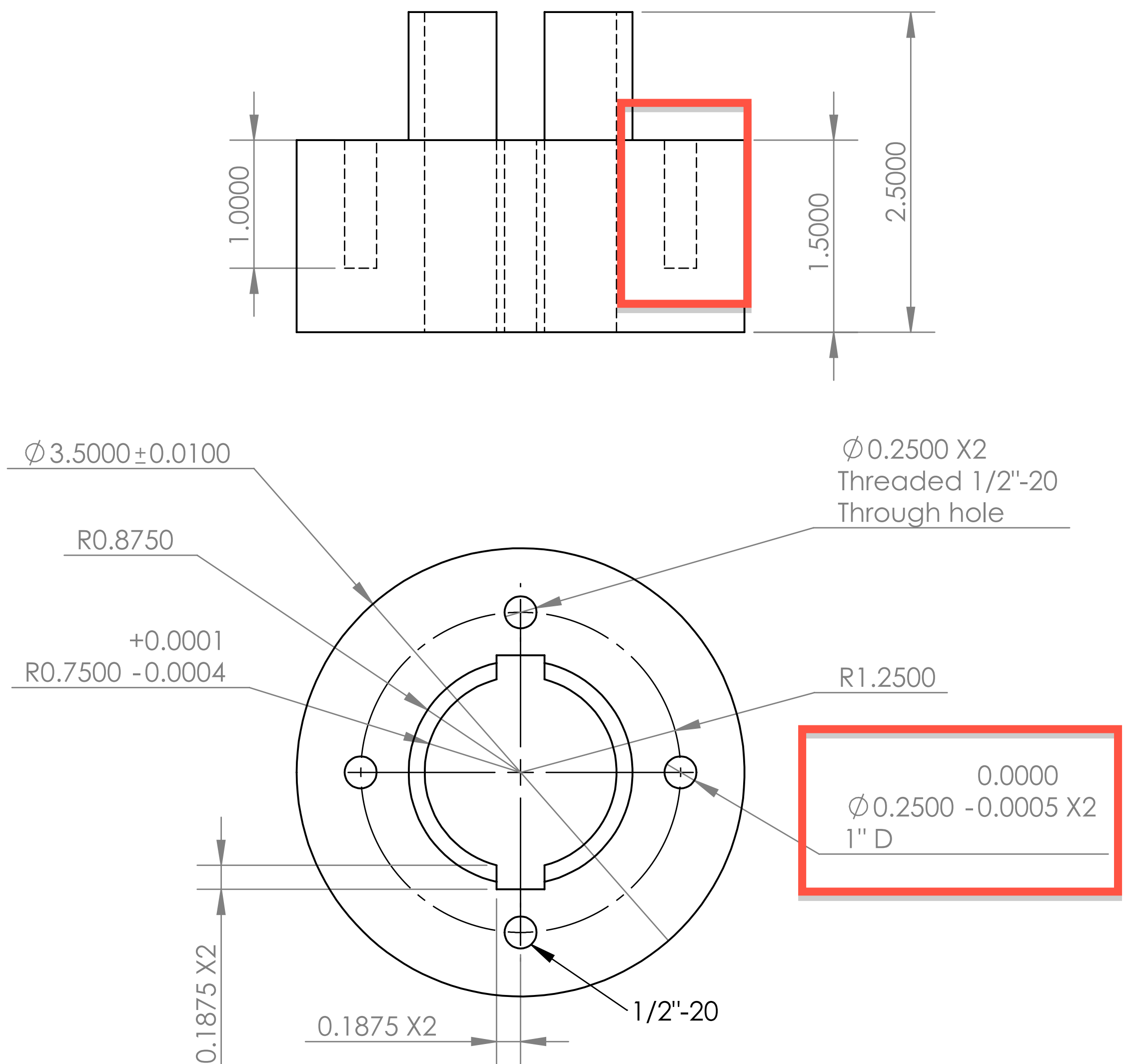
This change impacts the sprocket that is attached to this pump/motor hub using dowel pins at the holes. However, this does not affect the function of either part.

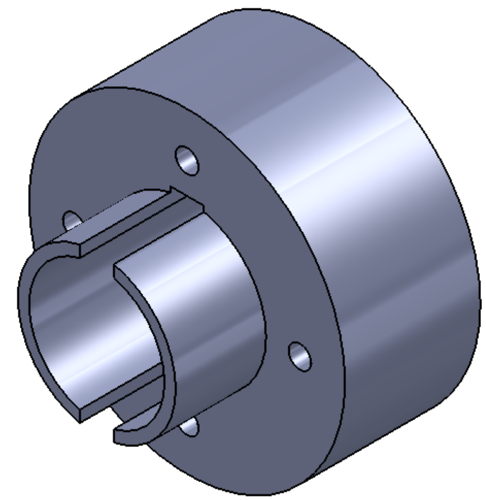
Analysis:

As stated above, this change has no affect on the function of any affected parts except for under thrust loading which these holes and dowel pins are not expected to face.

Change Authorization:

TEAM – Mat Wecharatana: 11/24/10
SPONSOR – Andrew Moskalik:





WAS

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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

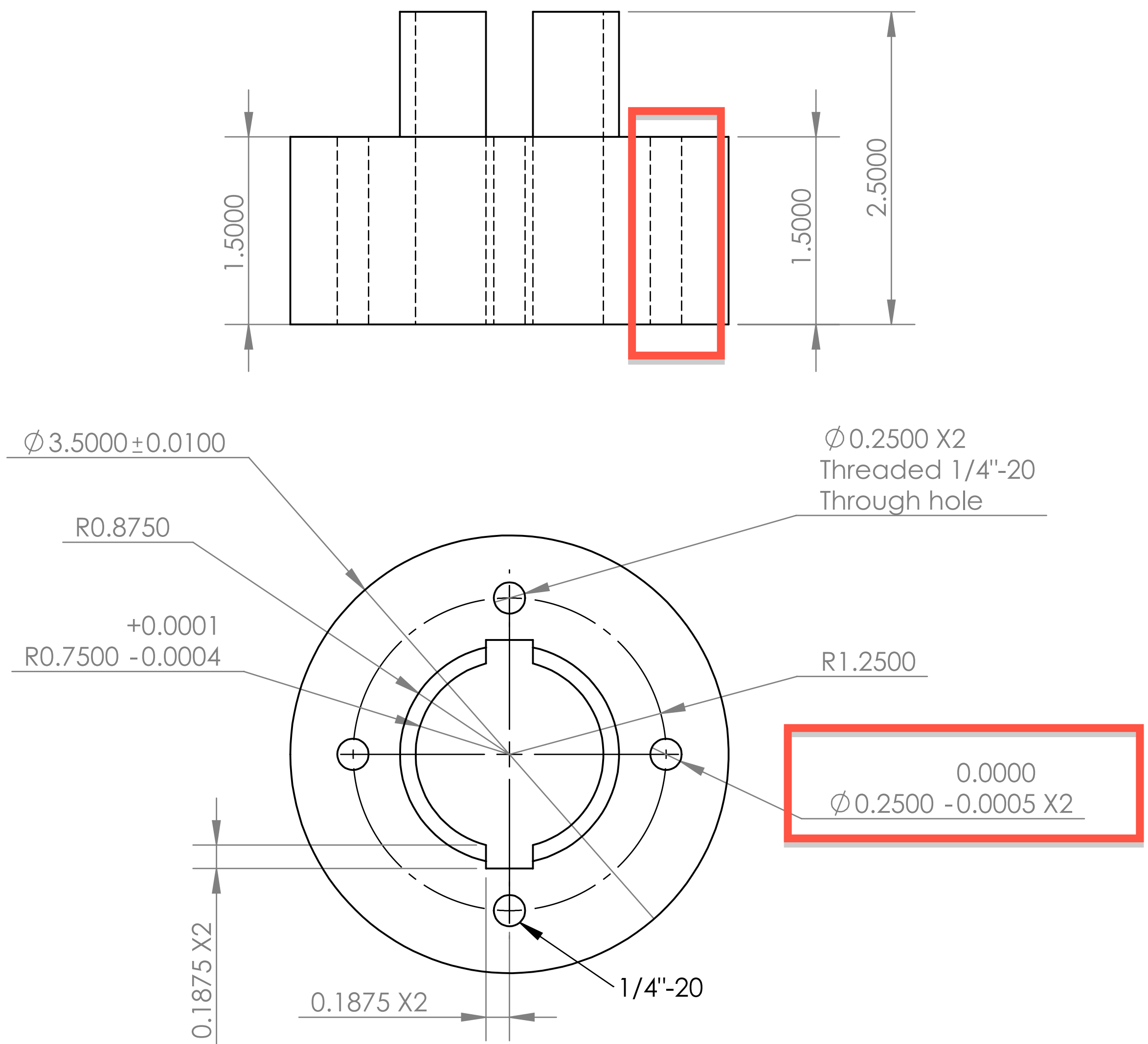
U of M Xebra Team Fall 2010

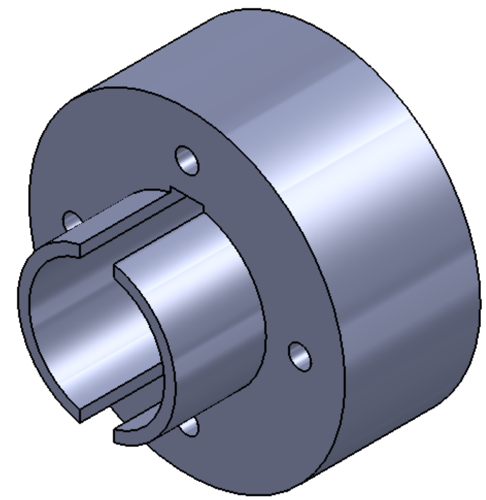
TITLE:

PM Hub

SIZE	DWG. NO.	REV
A		1
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1





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		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

PROPRIETARY AND CONFIDENTIAL

U of M Xebra Team Fall 2010

TITLE:

PM Hub

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

PUMP/MOTOR SHAFT

Change and Rationale:

For testing purposes, a hex head was machined in addition to the previously designed pump/motor shaft. A male hex head was needed to attach a torque wrench to the system for validation testing. This location was needed to apply torque to the system. Additionally, the shaft was increased in length as well. This was done to allow for an optical encoder or other method of rotation rate recording (at the request of the sponsor) to be added to the end of the shaft.

Sketch:

See attached.

Part Impacted:

This change only impacts the pump/motor shaft itself and does not affect the functionality of the part. However, it was verified with the sponsors that the machining of the shaft (creating a non circular surface) was acceptable for the other uses for the shaft (including adding an optical encoder or Hall effect sensor to the shaft to read rotation rate).

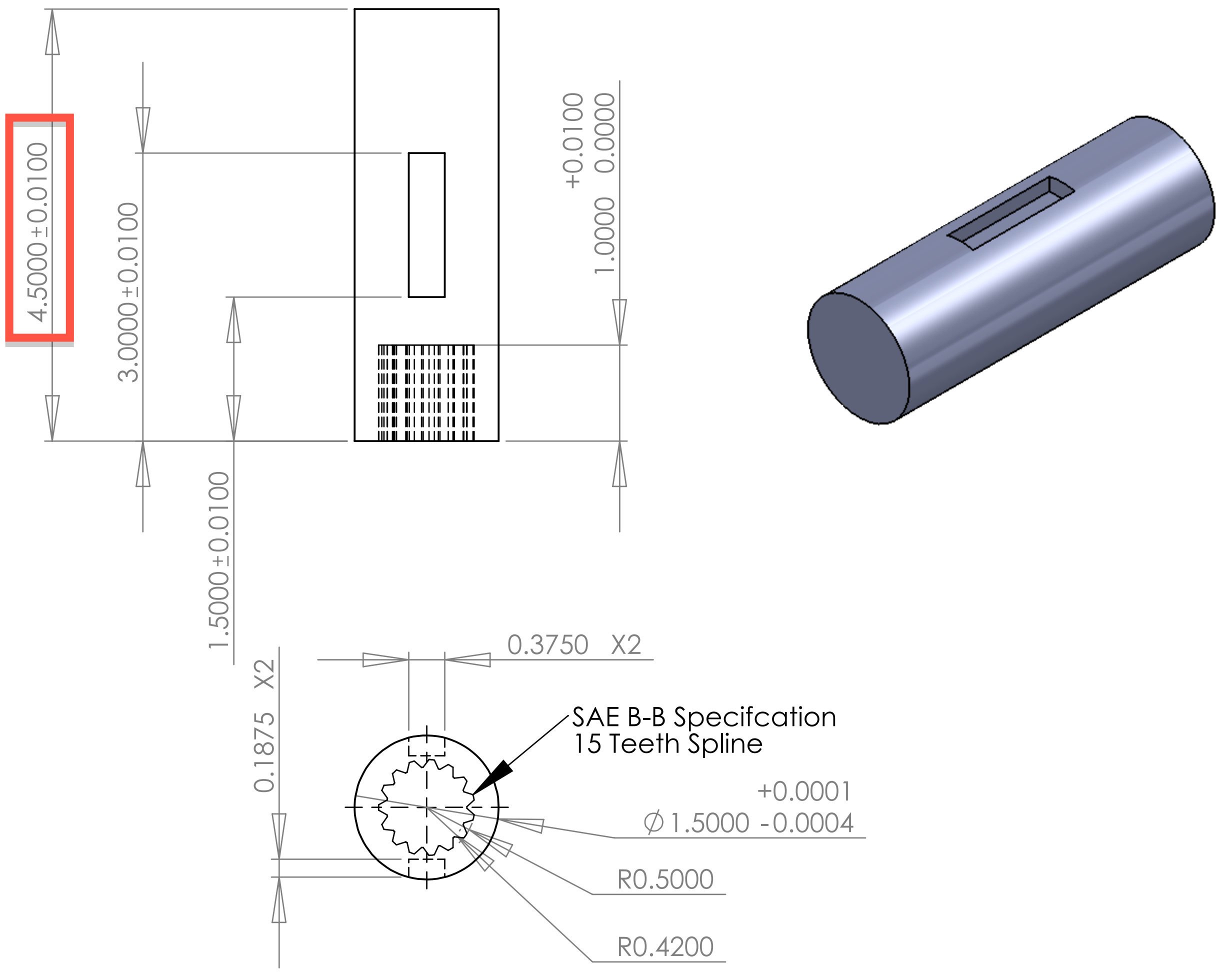
Analysis:

Analysis was conducted on the hex head to determine if the shaft under this new configuration could withstand the applied torque. Please see attached.

Change Authorization:

TEAM – Andrew Gavenda: 12/1/10
SPONSOR – Andrew Moskalik:

WAS



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		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

Xebra Team

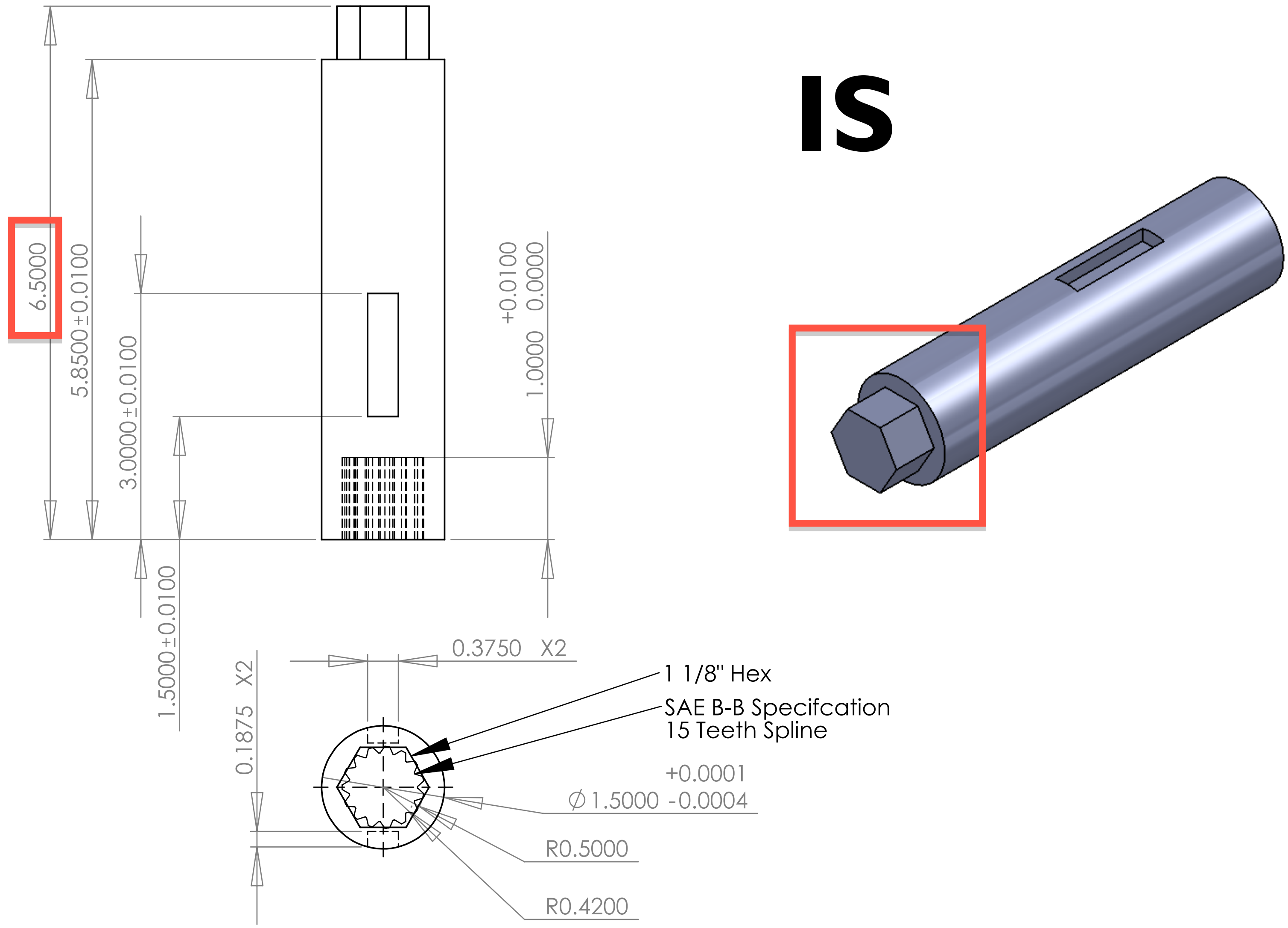
TITLE:

Output Shaft for Pump Motor
(Female)

SIZE A | DWG. NO. | REV

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			Output Shaft for Pump Motor (Female)
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 1018 STEEL	COMMENTS:			SIZE A / DWG. NO. / REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

5 4 3 2 1

PUMP/MOTOR SPROCKET

Change and Rationale:

For ease of manufacturing (availability of tooling), the inner bore diameter of the pump/motor sprocket was increased to 2" from 1.75".

Sketch:

See attached.

Part Impacted:

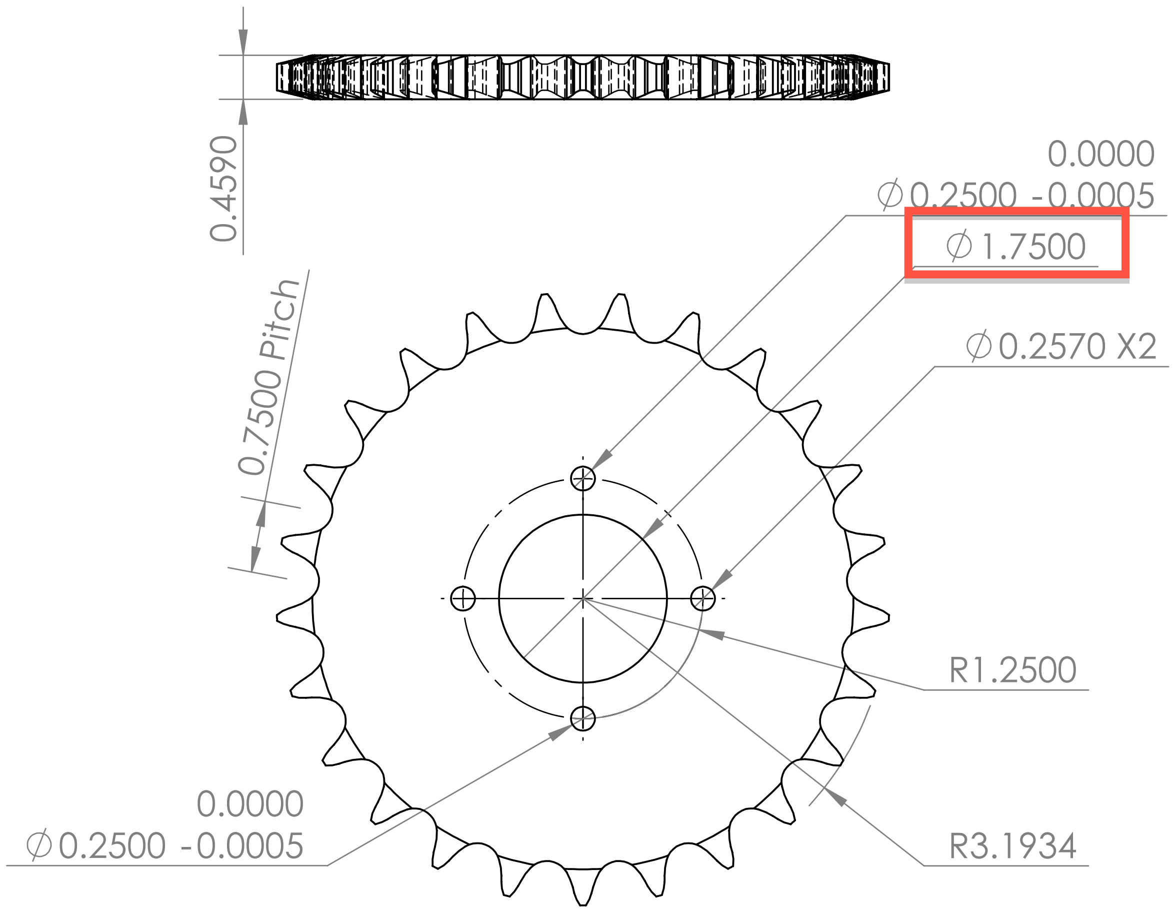
This change impacts no other parts. Although the pump/motor sprocket is attached directly to the pump/motor hub, the inner bore diameter does not contact the hub itself. The inner bore hole of sprocket is oversized and kept in place using dowel pins and bolts so that no contact occurs with the hub.

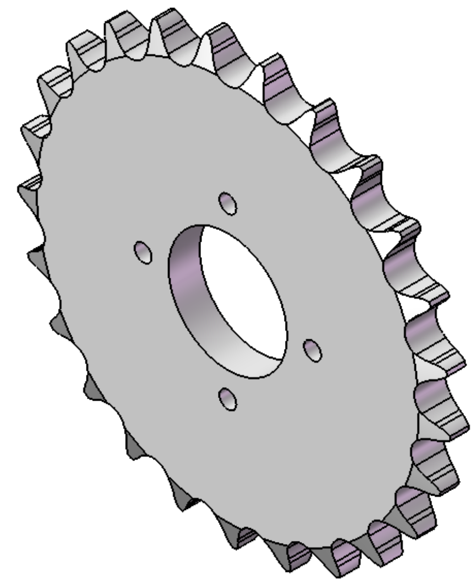
Analysis:

No additional analysis is necessary because no load is expected to be placed to this inner bore diameter. Although the amount of material between the inner bore and the drilled holes is decreased, this is not a concern because any loading will occur in the radially outward and angular directions. The holes and bolt/dowel connects will remain the same in these locations, even if material has been removed closer to the radially inward direction.

Change Authorization:

TEAM – David Fok: 11/22/10
SPONSOR – Andrew Moskalik:





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		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

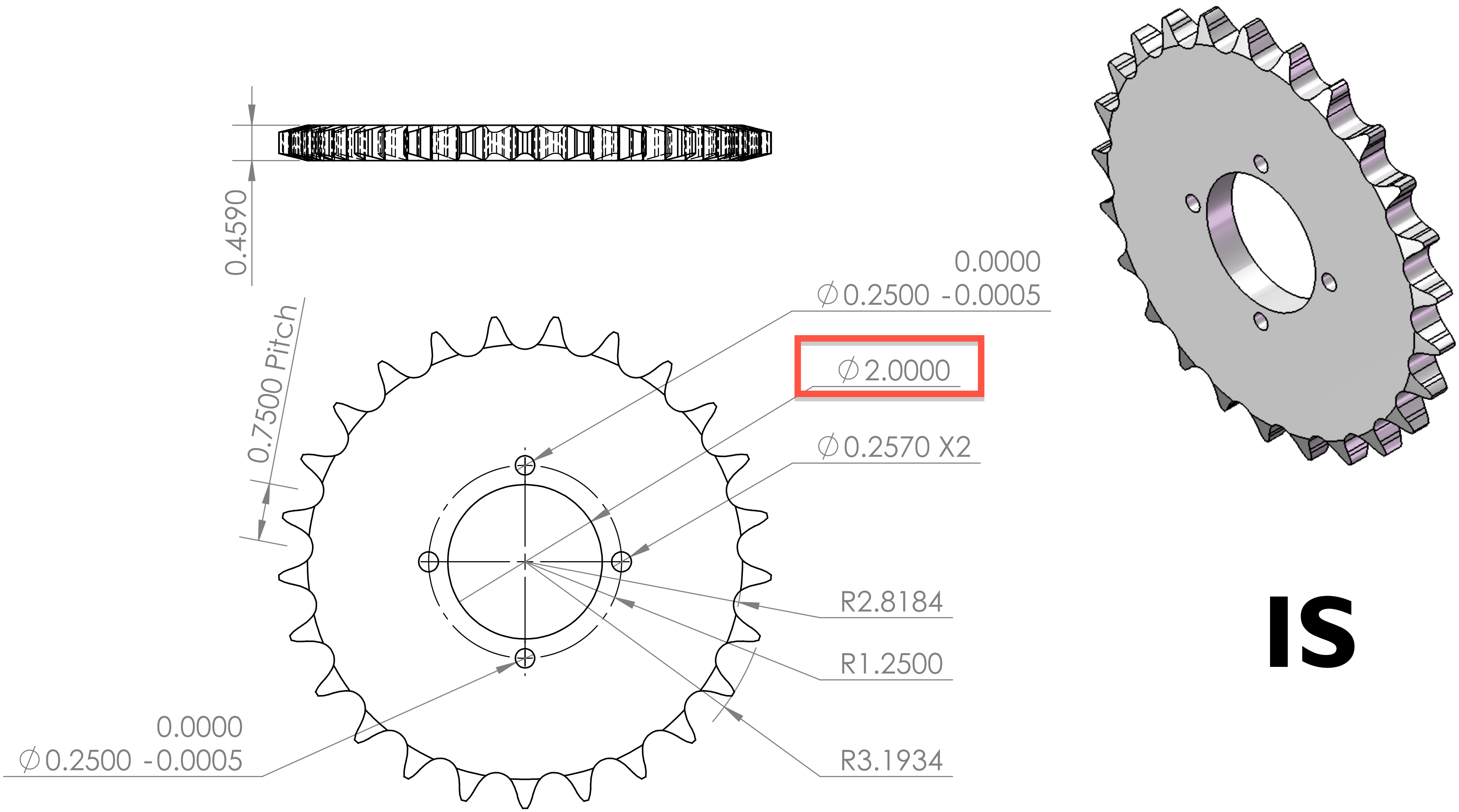
TITLE:

PM Sprocket

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			PM Sprocket
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A — DWG. NO. — REV 1
	NEXT ASSY	USED ON	FINISH			
	APPLICATION		DO NOT SCALE DRAWING			SCALE: 1:2 — WEIGHT: — SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1

SPACER

Change and Rationale:

For ease of manufacture, the space retention between the gear reduction case and the intermediate gear reduction case was maintained by creating four (4) small spacers, rather than one larger spacer.

Sketch:

See attached.

Part Impacted:

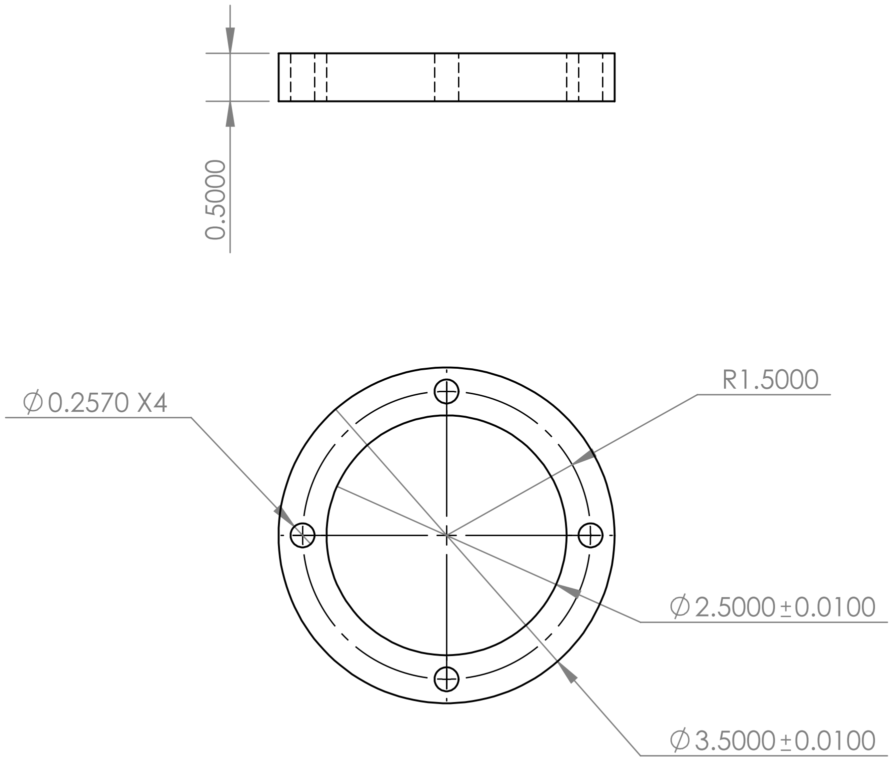
The spacers contact the intermediate gear reduction case (attached to the DC motor) and the gear reduction case (newly fabricated case)

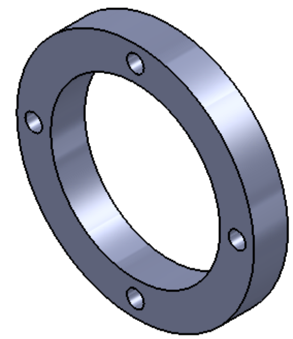
Analysis:

No additional analysis was necessary because the spacers are non-load bearing components

Change Authorization:

TEAM – David Fok: 12/1/10
SPONSOR – Andrew Moskalik:





WAS

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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

TITLE:

Spacer Mount

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1

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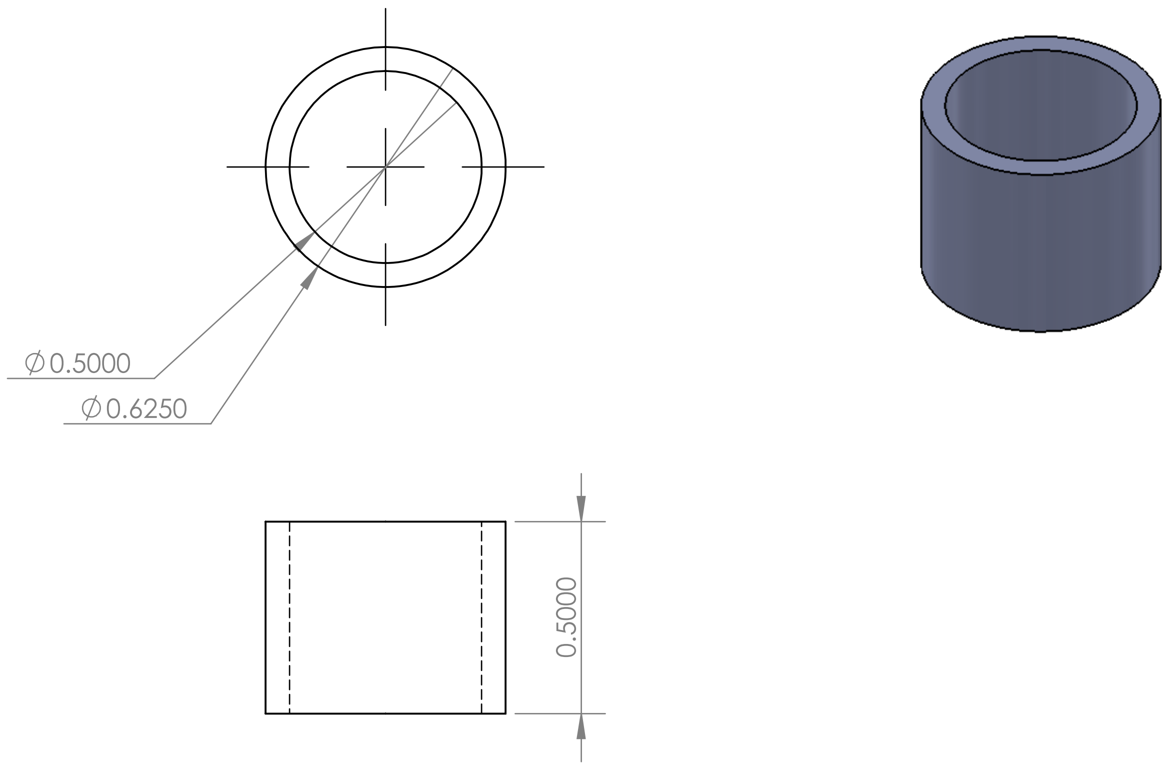

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		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			
			ENG APPR.			Spacer Mount
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 1018 STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

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5 4 3 2 1

DC MOTOR SHAFT AND DC MOTOR SHAFT CAP

Change and Rationale:

Due to a fit-up issue, a male square head was machined into the DC motor shaft opposite of the spline. To match this head, a new female square cap piece was created that will be mounted to the inner race of the tapered bearing. This change was necessary because the outer diameter of the inner race of this bearing is larger than the inner bore of each of the three DC motor sprockets. With this fit-up issue, the sprockets would not be interchangeable. This is because they would not be removable since the sprocket could not fit over the inner part of the bearing to slide off the shaft. Therefore, the new cap piece was created so that the inner race of the bearing can slide out along with the removable access panel of the gear reduction case.

Sketch:

See attached.

Part Impacted:

This change impacts both the tapered bearing fit and the DC motor shaft. The DC motor shaft has had additional material removed from it to create the male square head. Additionally, the tapered bearing is no longer mounted directly to the DC motor shaft, but instead to the new shaft cap piece. However, the fit is still the same in that the outer diameter of the cap is the same as the outer diameter of the DC motor shaft.

Analysis:

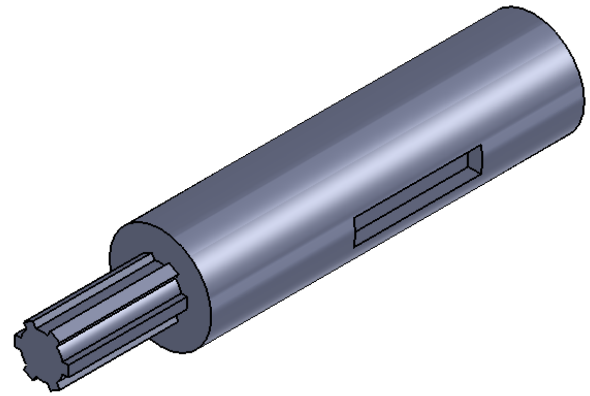
The slip fit between the cap and the DC motor shaft functions in the same way as a solid shaft would in that both will transmit angular displacement (rotation). The new cap piece will not need to transmit any torque because it is fit into the bearing that rotates freely. In static loading, the stress is increased due to the decreased cross-sectional area of the shaft because of the removed material. See attached for the calculation.

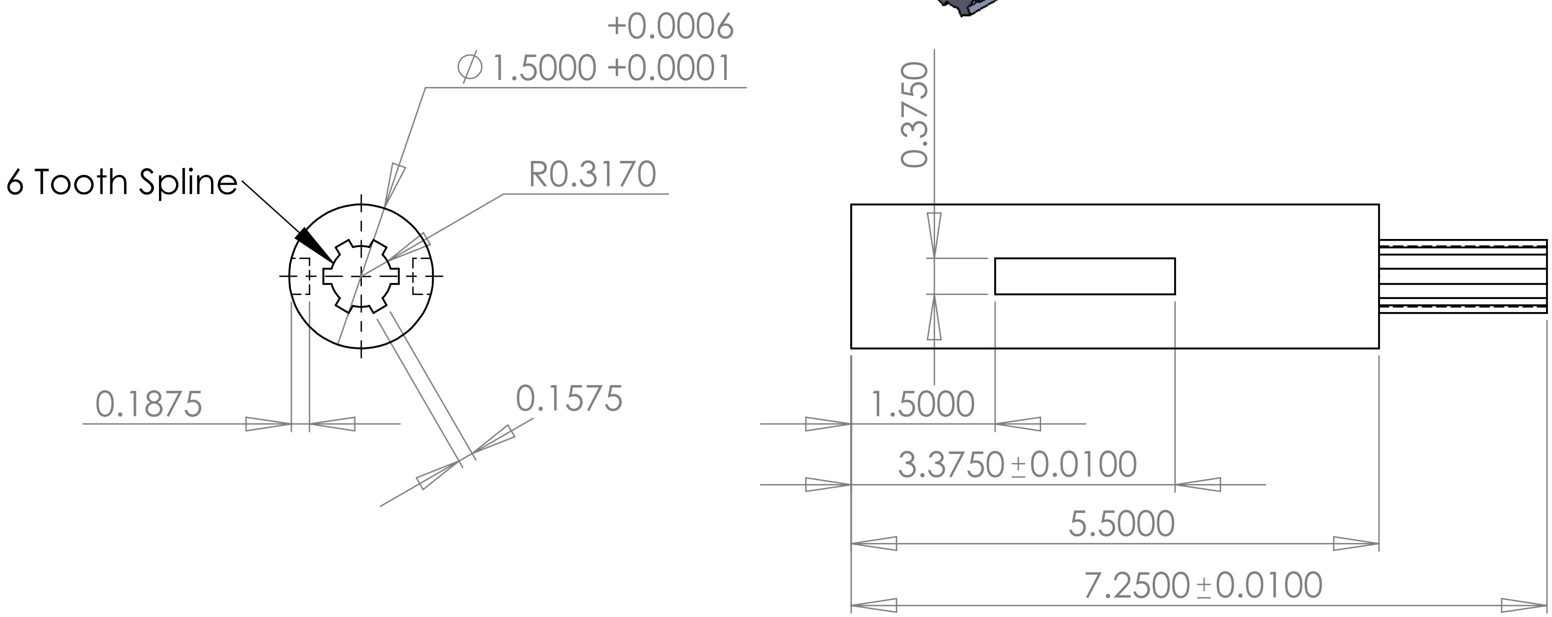
Change Authorization:

TEAM – Mat Wecharatana: 12/3/10

SPONSOR – Andrew Moskalik:

WAS

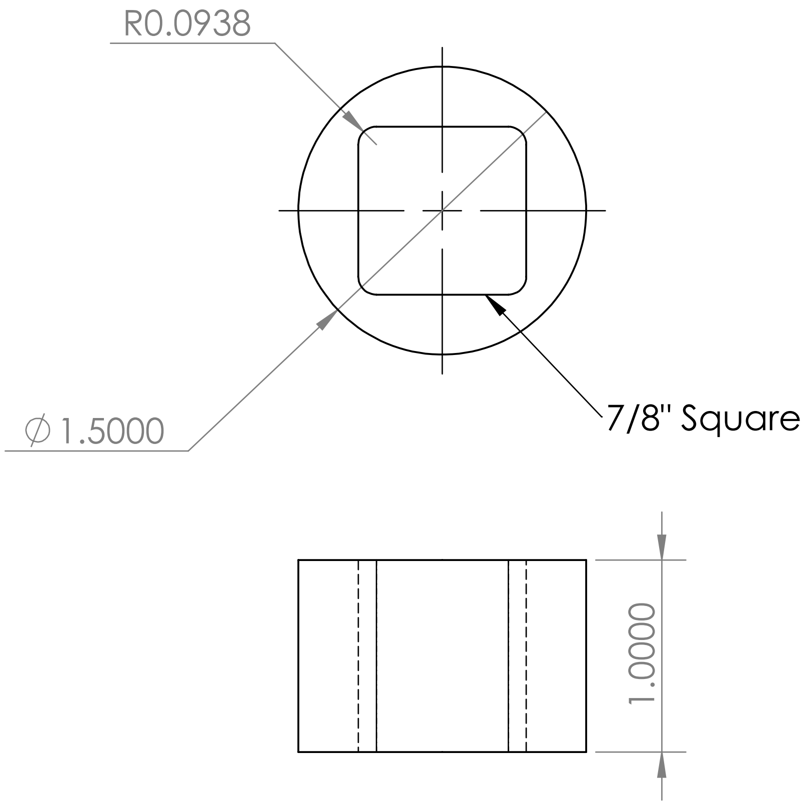


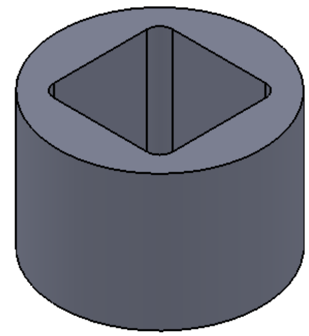


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:
			CHECKED			Output Shaft for DC Motor (Male)
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 1018 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1





NEW

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			DC Shaft Cap
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL	COMMENTS:			SIZE A | DWG. NO. | REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1 | WEIGHT: | SHEET 1 OF 1

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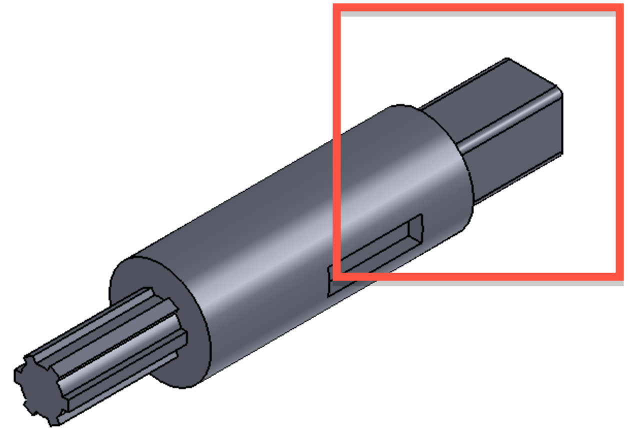
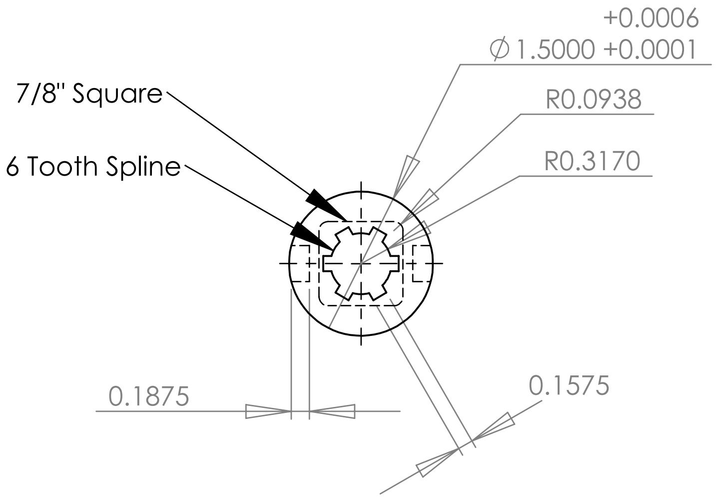
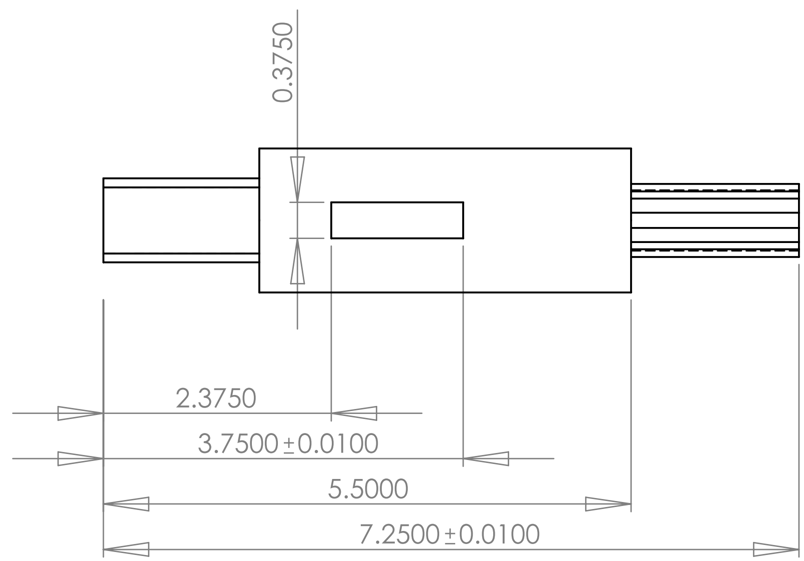




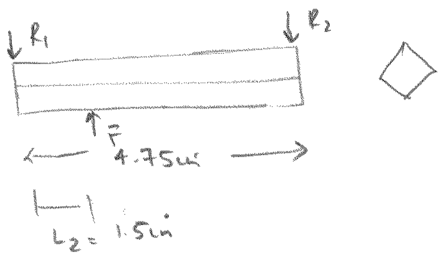


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			Output Shaft for DC Motor (Male)
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 1018 STEEL	COMMENTS:			SIZE A / DWG. NO. / REV
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

5 4 3 2 1

Bending / Deflection in DC motor shaft

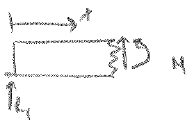
* Since the change of our design we now have a square cross section on one end of the DC motor shaft.
* The main concern with this change is the bending of this shaft under the applied forces.
* As a worst case scenario, we will determine the deflection in the DC shaft by modeling it with a square cross section throughout.



$$\Sigma M_{R_1} = 0 = -(1.5)(826.58) + 4.75 R_2$$

$$\therefore R_2 = 261.025$$

$$R_1 = 565.555$$



$$0 \le x \le L_2$$

$$M_{(x)} = R_1 x = 565.555(x)$$

$$\therefore M(L_2) = 848.333 \text{ lb in}$$

$$I = \frac{1}{12} bh^3 \qquad b = h \qquad b = 1\tfrac{7}{8}''$$

$$= \frac{1}{12} b^4 = \frac{1}{12} \times (1.875)^4$$

$$= 1.03 \text{ in}^4$$

$$EI \frac{d^2v}{dx^2} = M(x) \qquad E = 29000 \text{ KSI}$$

$$(29000)(1.03) \frac{d^2v}{dx^2} = 565.555x$$

$$\therefore \int \frac{d^2v}{dx^2} = \int 0.01893\, x$$

$$\frac{dv}{dx} = \frac{0.01893\, x^2}{2} + C_1$$

$$v(x) = \frac{0.01893\, x^3}{6} + \cancel{C_1}x + \cancel{C_2}$$

$$v(1.5) = \frac{0.01893 \times (1.5)^3}{6} = \underline{\underline{0.01 \text{ in}}}$$

This is worst case, and it will be smaller than the current value.

SUB-FRAME

Change and Rationale:

For ease of manufacture, parts of the existing sub-frame were retained. These changes can be broken down into four (4) components.

1) Sub-frame back bar
    a. Slots for the gear reduction case bolts were created
    b. These slots had just been holes
    c. This created adjustability and the ability to compensate for possible misalignment when the gear reduction case is mounted to the sub-frame
    d. A notch was cut into the bar to allow for proper alignment of the gear reduction case on the sub frame
        i. Due to a fit-up issue of the DC motor shaft not aligning properly with the DC motor, this notch was needed.
2) Sub-frame front bar
    a. Part of the original sub-frame is retained
        i. Thus, a material change has occurred
        ii. Previously, the entire front bar was planned to be manufactured using new stock (A36 steel)
        iii. Now, part of the front bar is retained from the previous sub-frame front bar (1018 steel)
        iv. The remaining part of the front bar (newly fabricated from A36 stock) is now slotted for the same reason as the sub-frame back bar is slotted
3) Sub-frame DC motor U-bar
    a. Entire part is retained from original U-bar
    b. Hence material change has occurred from planned (new A36 steel stock) to now (previously used 1018 steel)
4) Sub-frame DC motor bar
    a. Entire part is retained from original U-bar
    b. Hence material change has occurred from planned (new A36 steel stock) to now (previously used 1018 steel)

Sketch:

See attached.

Part Impacted:

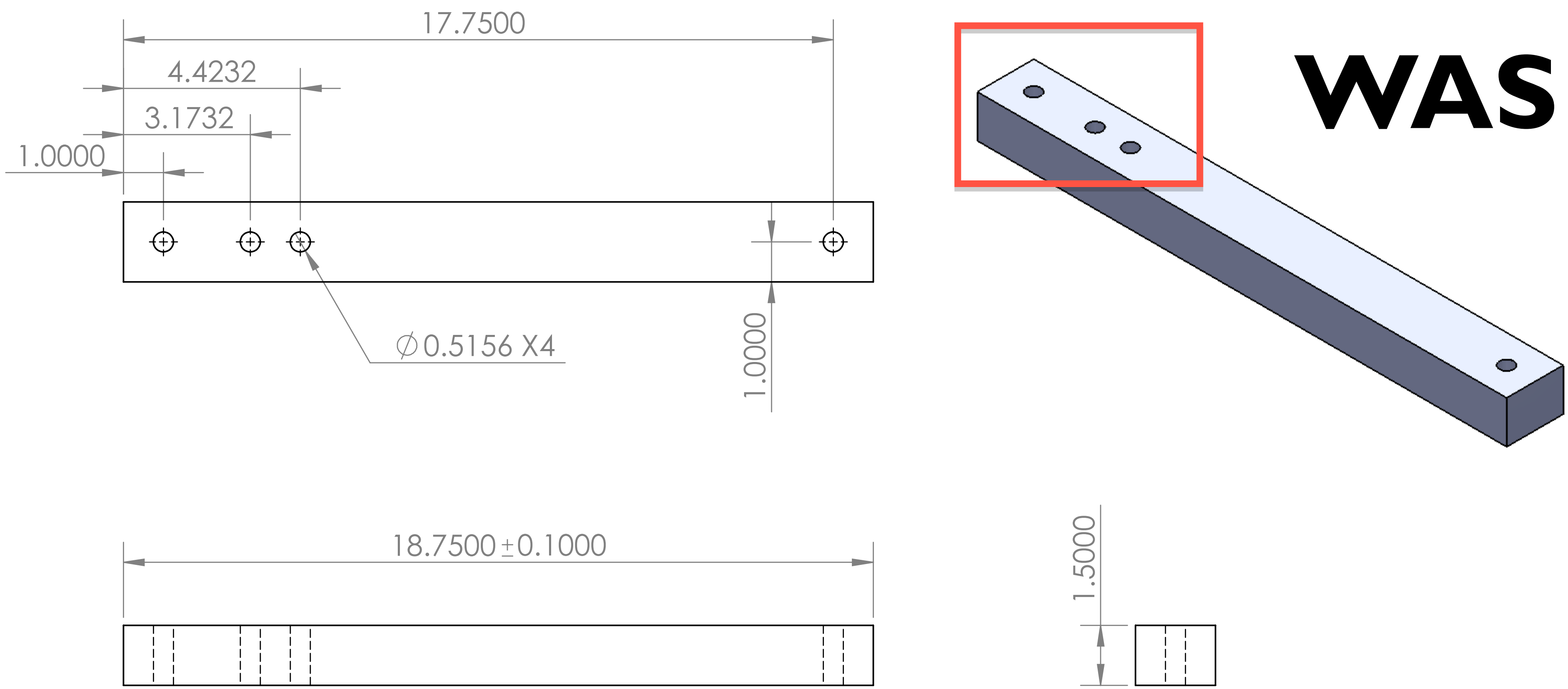
The parts impacted are mentioned above. These parts combined form the sub-frame that attaches directly to the sub-frame mounts and holds the DC motor and the intermediate gear reduction case.

Analysis:

Sub-frame back bar static analysis for geometry changes was conducted. Please see attached. No additional testing for sub-frame front bar was not conducted because only the material was changed. The material was changed from A36 to 1018 steel that has a higher yield strength. Similarly, additional material analysis was not needed for the sub-frame back bar.

Change Authorization:

TEAM – Andrew Gavenda: 11/23/10
SPONSOR – Andrew Moskalik:



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

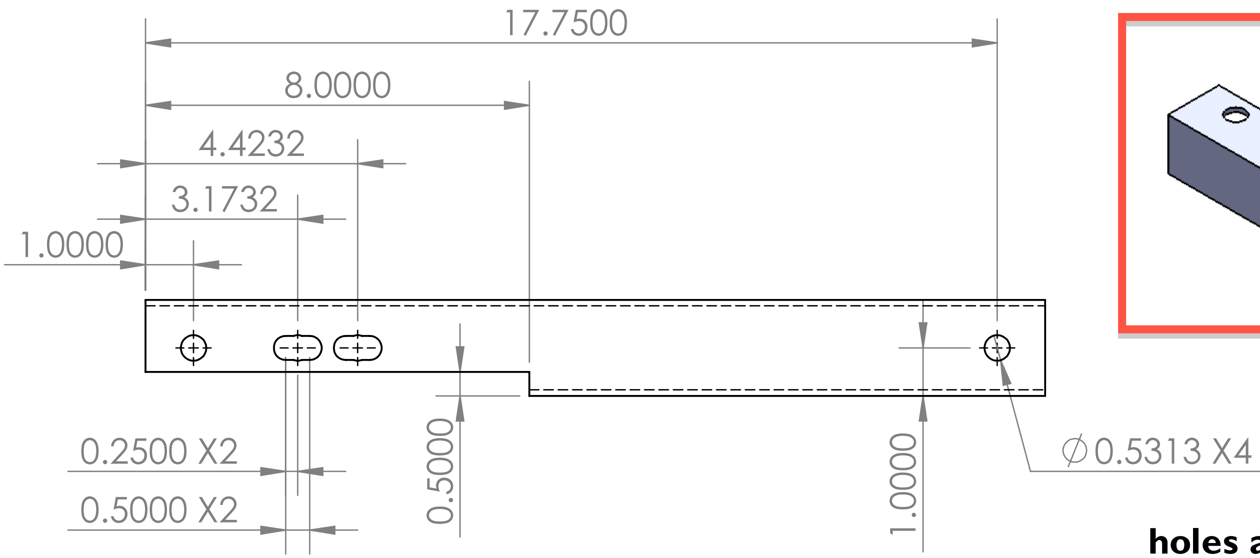
TITLE:

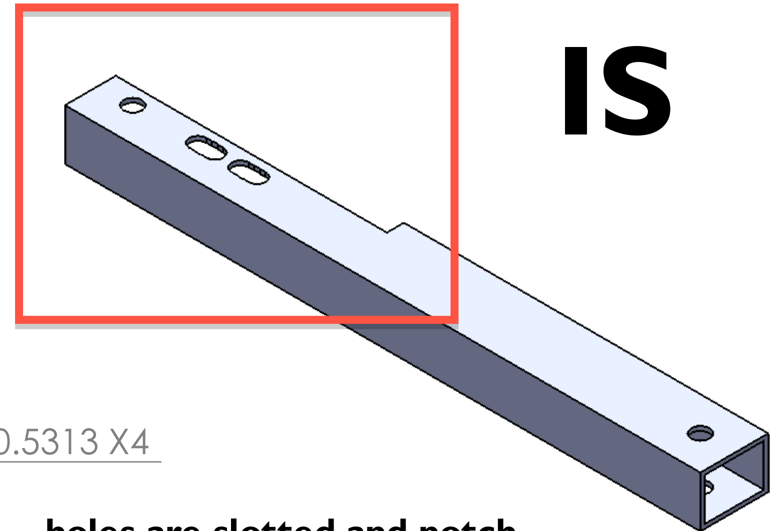
Subframe Back

SIZE	DWG. NO.	REV
A		1

SCALE: 1:4 WEIGHT: SHEET 1 OF 1

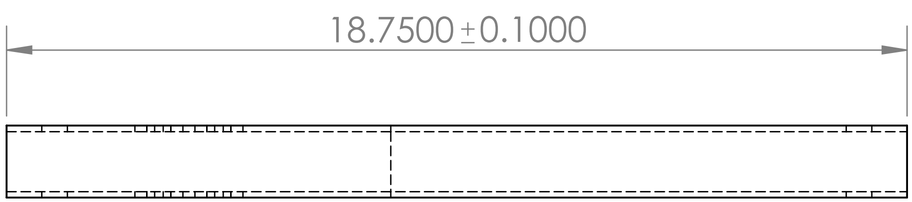
5 4 3 2 1

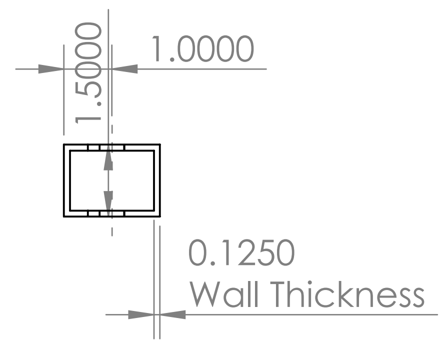




IS

holes are slotted and notch is cut into the bar





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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

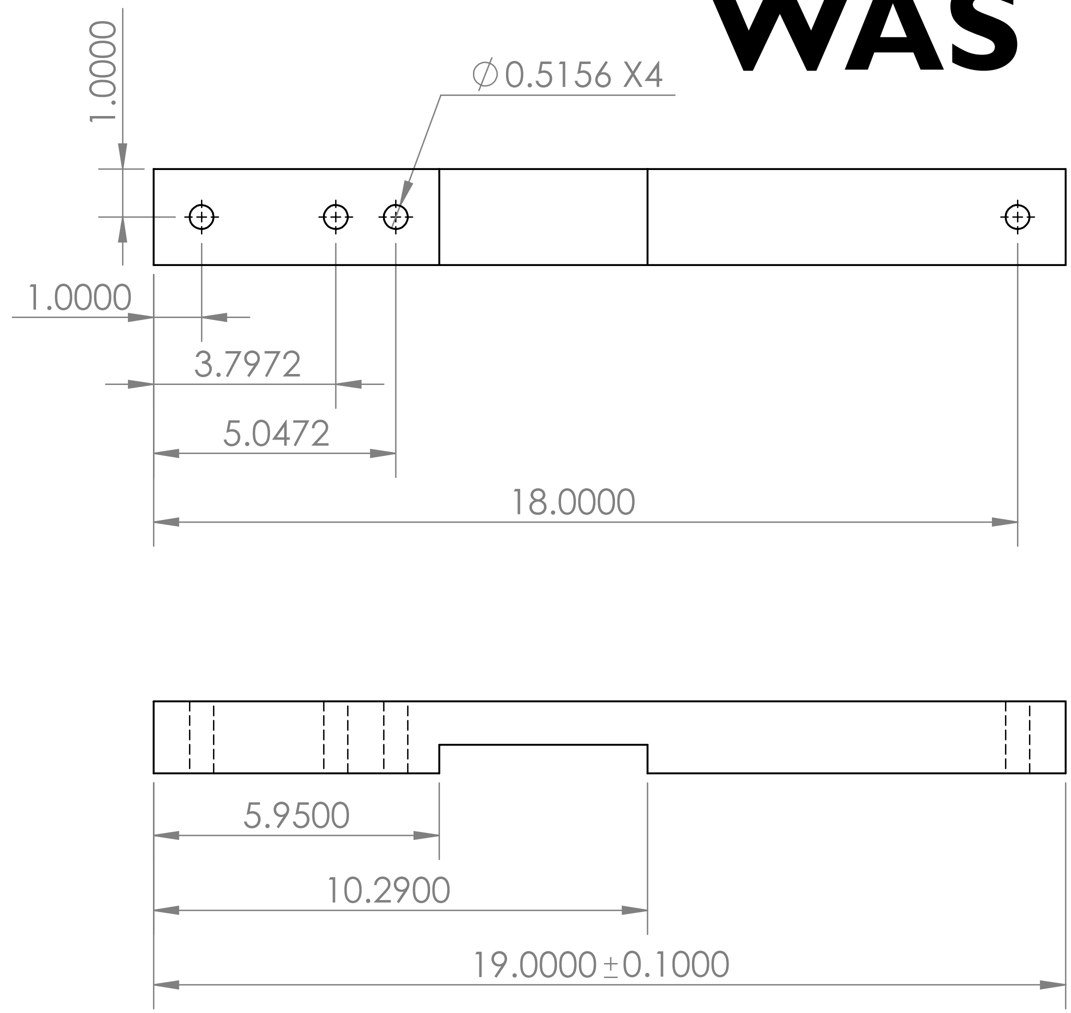
Subframe Back

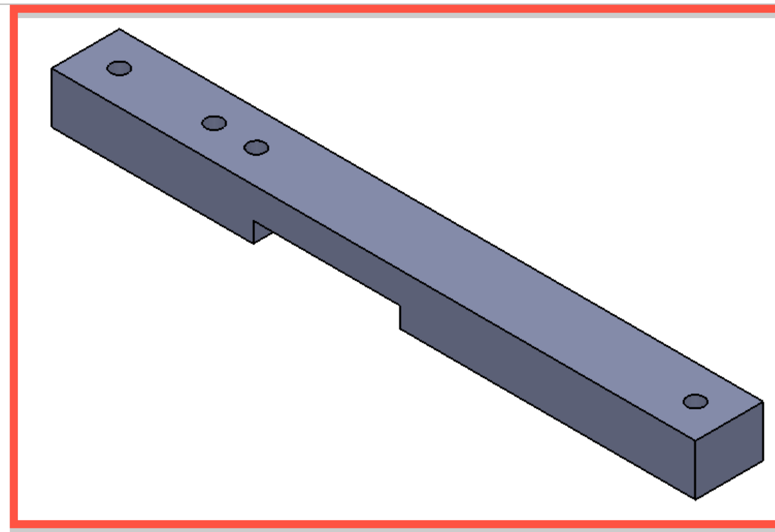
SIZE	DWG. NO.	REV
A		1

SCALE: 1:4 WEIGHT: SHEET 1 OF 1

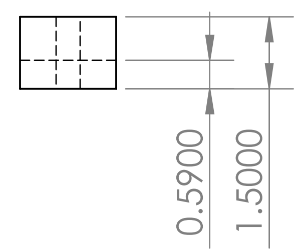
5 4 3 2 1

WAS





previously, the entire bar was to be newly made of A36 steel



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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

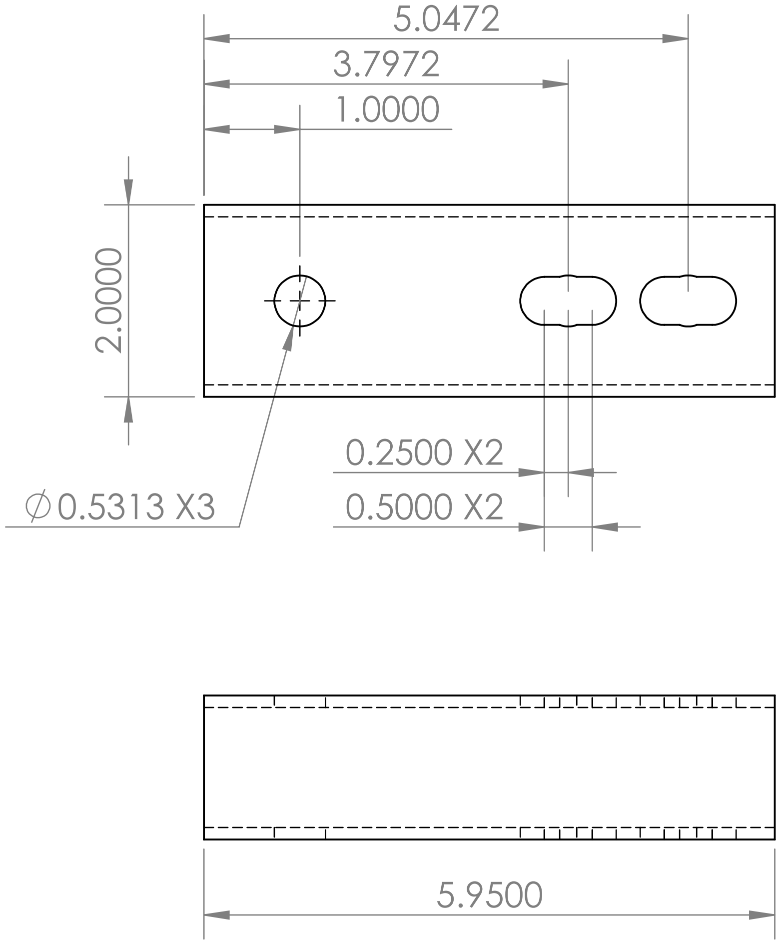
Sub-Frame: Front

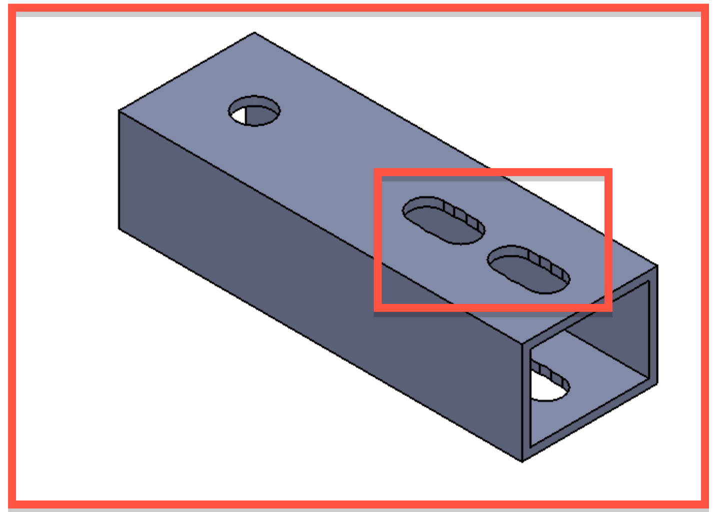
SIZE	DWG. NO.	REV
A		1

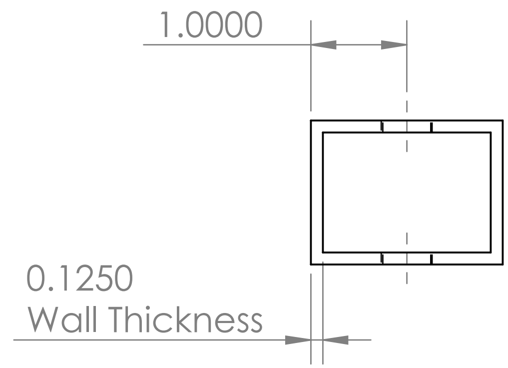
SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

IS







now, slots have been made and only this (shorter) portion is made of new material (A36 steel)

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TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

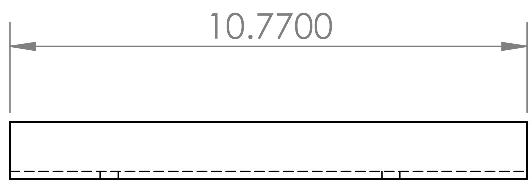
Sub-Frame: Front

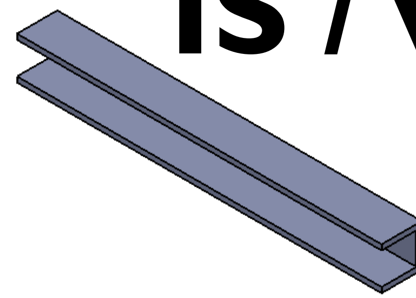
SIZE A | DWG. NO. | REV 1

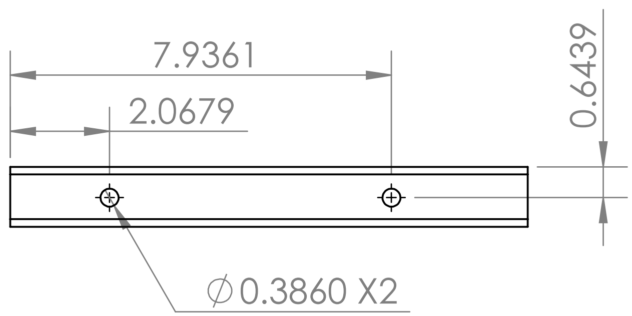
SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

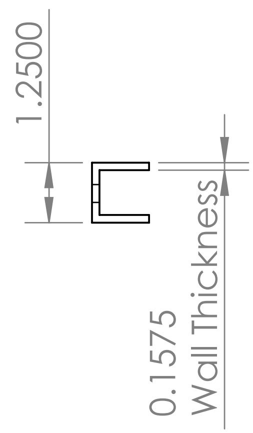
5 4 3 2 1

IS / WAS









was to be made of new A36 stock, now retained previously used 1018 part

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

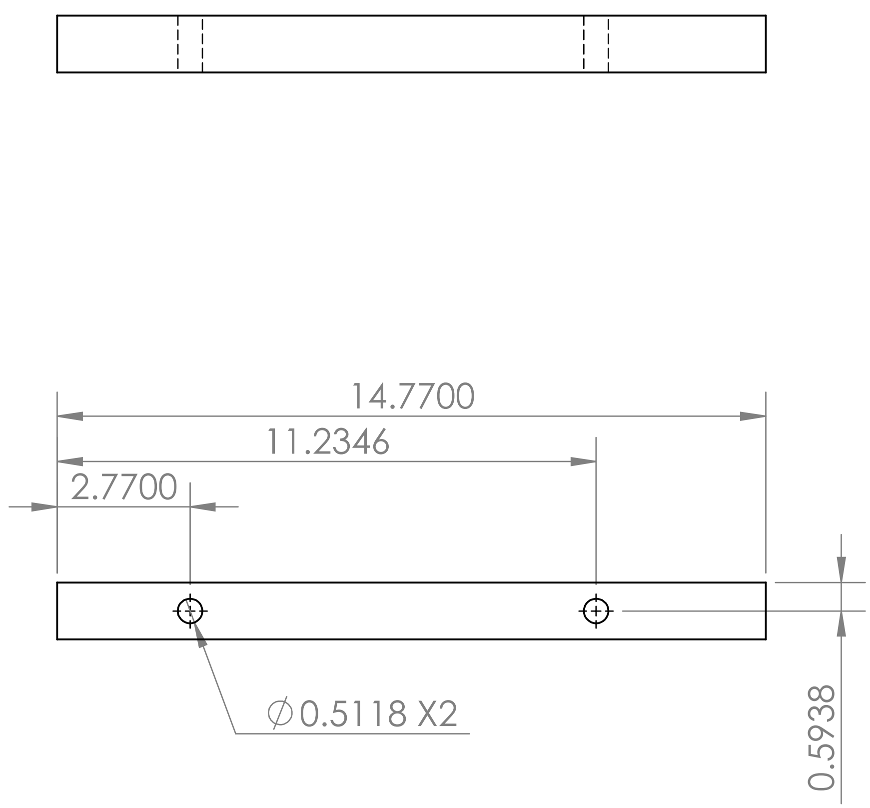
Subframe for DC Motor (U-Shaped)

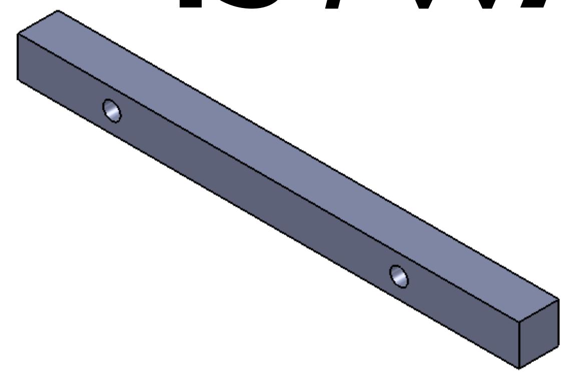
SIZE A | DWG. NO. | REV 1

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

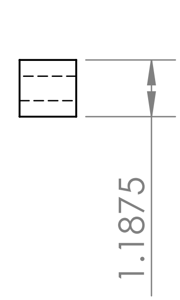
5 4 3 2 1

IS / WAS





was to be made of new A36 stock, now retained previously used 1018 part



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.01 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:
			CHECKED			Subframe for DC Motor
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

SUB-FRAME MOUNT

Change and Rationale:

For ease of manufacture, the existing sub-frame mount base was retained, rather than creating a new base out of new stock material. Thus, the material of the base is 1018 mild steel rather than the previously planned A36 steel.

Sketch:

See attached.

Part Impacted:

The sub-frame mount is the only part affected because it is directly attached to the sub-frame mount base.

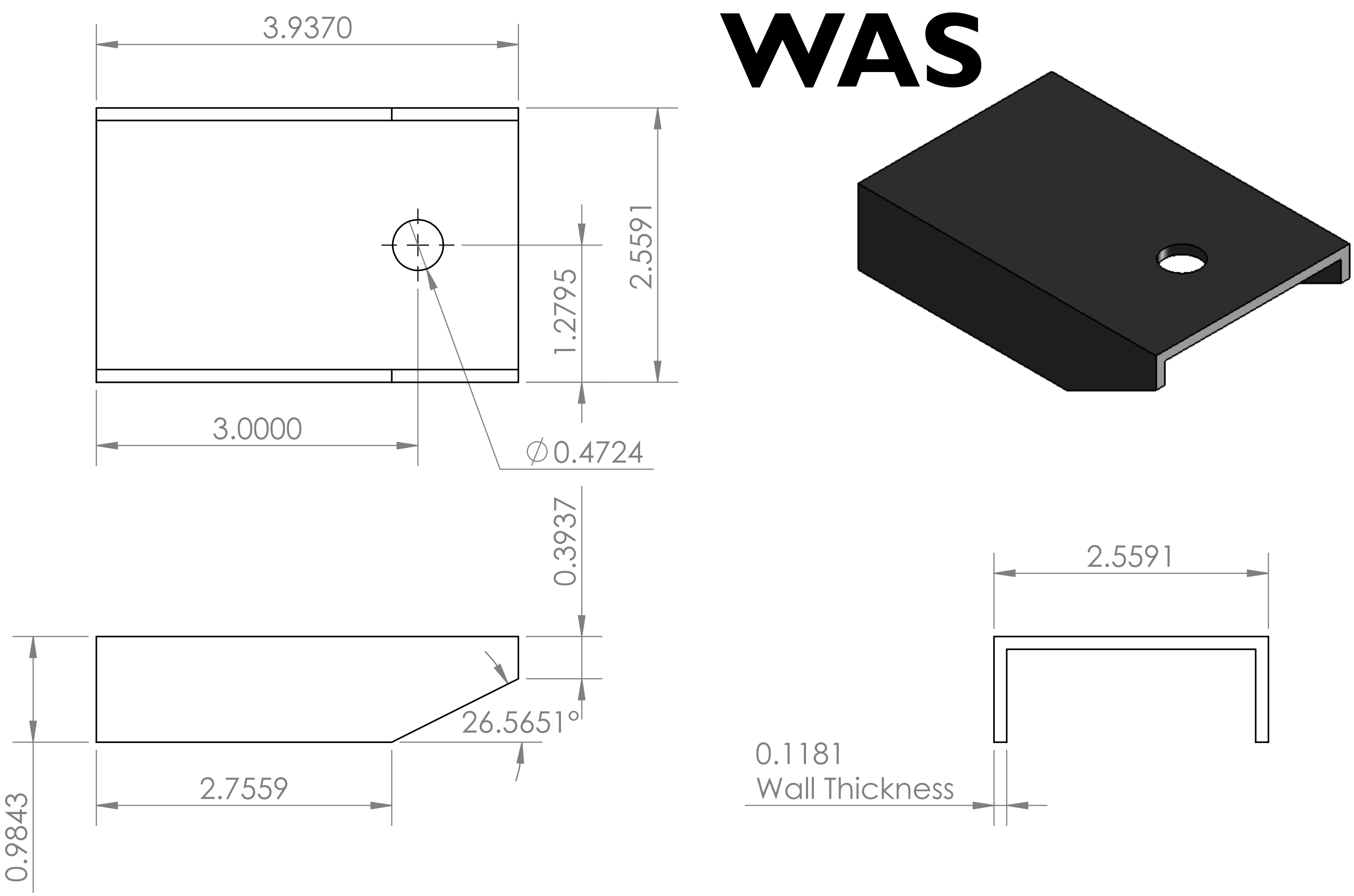
Analysis:

The material has been changed from A36 steel to 1018 steel. 1018 has a higher yield strength (53,700 psi > 36,300 psi (http://www.onlinemetals.com/steelguide.cfm)) meaning that this design changed resulted in a higher yield strength for the part than previously planned.

Change Authorization:

TEAM – Andrew Gavenda: 11/23/10

SPONSOR – Andrew Moskalik:



NEXT ASSY	USED ON	
APPLICATION		

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

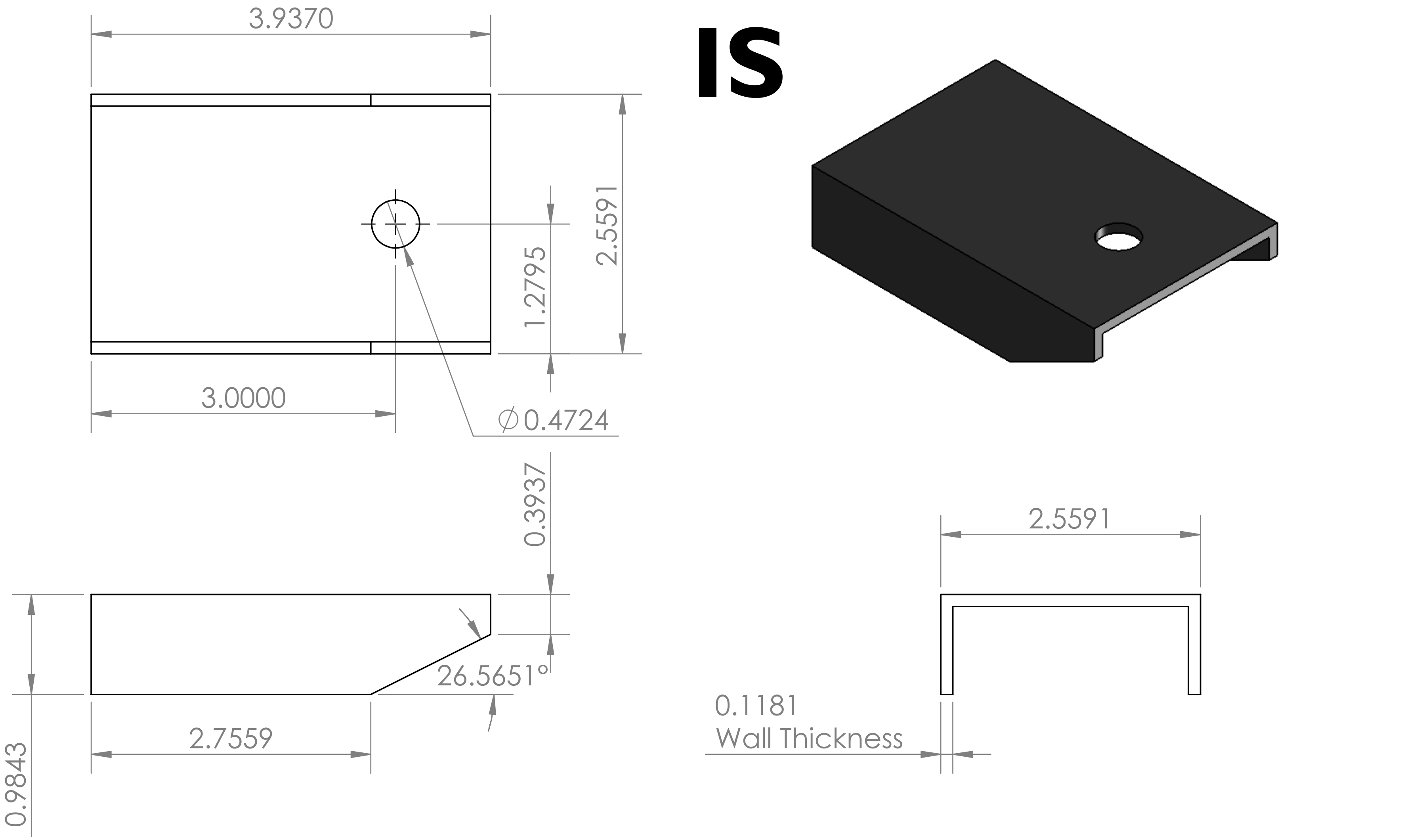
Subframe Mount Base

SIZE	DWG. NO.	REV
A		1

SCALE: 2:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

IS



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

Subframe Mount Base

SIZE	DWG. NO.	REV
A		1

SCALE: 2:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

TABS AND GUSSETS

Change and Rationale:

The following changes were made to the tabs and gussets

1) attachment method
    a. Instead of using welds to attach the tabs and gussets to the gear reduction case, 5 bolts were used instead
        i. This change was made because it was suspected that deformation could occur if welds were used
        ii. This is important because this could affect the alignment of the holes in the gear reduction case which affects the function of the case
    b. Closely related, holes were drilled for for bolts
        i. Material was removed from the tabs and gussets
    c. The holes were also counterbored to prevent the bolt heads from protruding from the surface of the tabs and gussets
    d. Consequently, holes were drilled and tapped into the side plates of the gear reduction case to accommodate the bolts
2) Additionally, the drill holes for attachment to the sub-frame were turned into slots
    a. This allows for accommodation of possible misalignment of the gear reduction case on the sub-frame

Sketch:

See attached.

Part Impacted:

These changes impact the mounting of the gear reduction case to the sub-frame

Analysis:

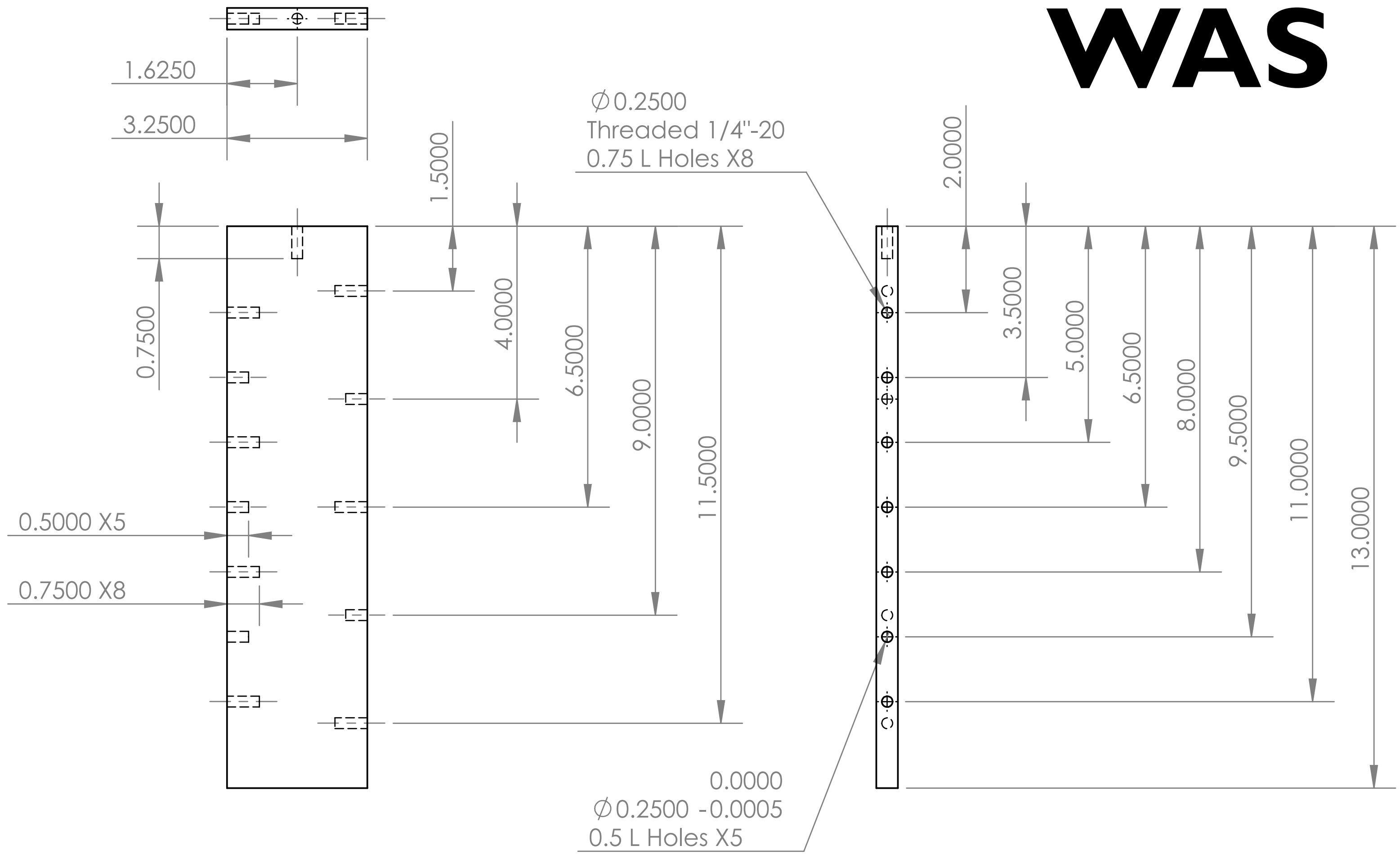
Static analysis of the tab and gusset mount to the side plate was conducted. Please see attached.

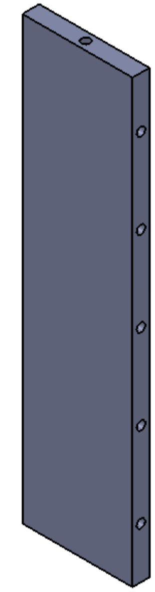
Change Authorization:

TEAM – Steve Lidell: 11/30/10 (NOTE: changes were not pre-approved by the team during outsourced machining)

SPONSOR – Andrew Moskalik:

WAS





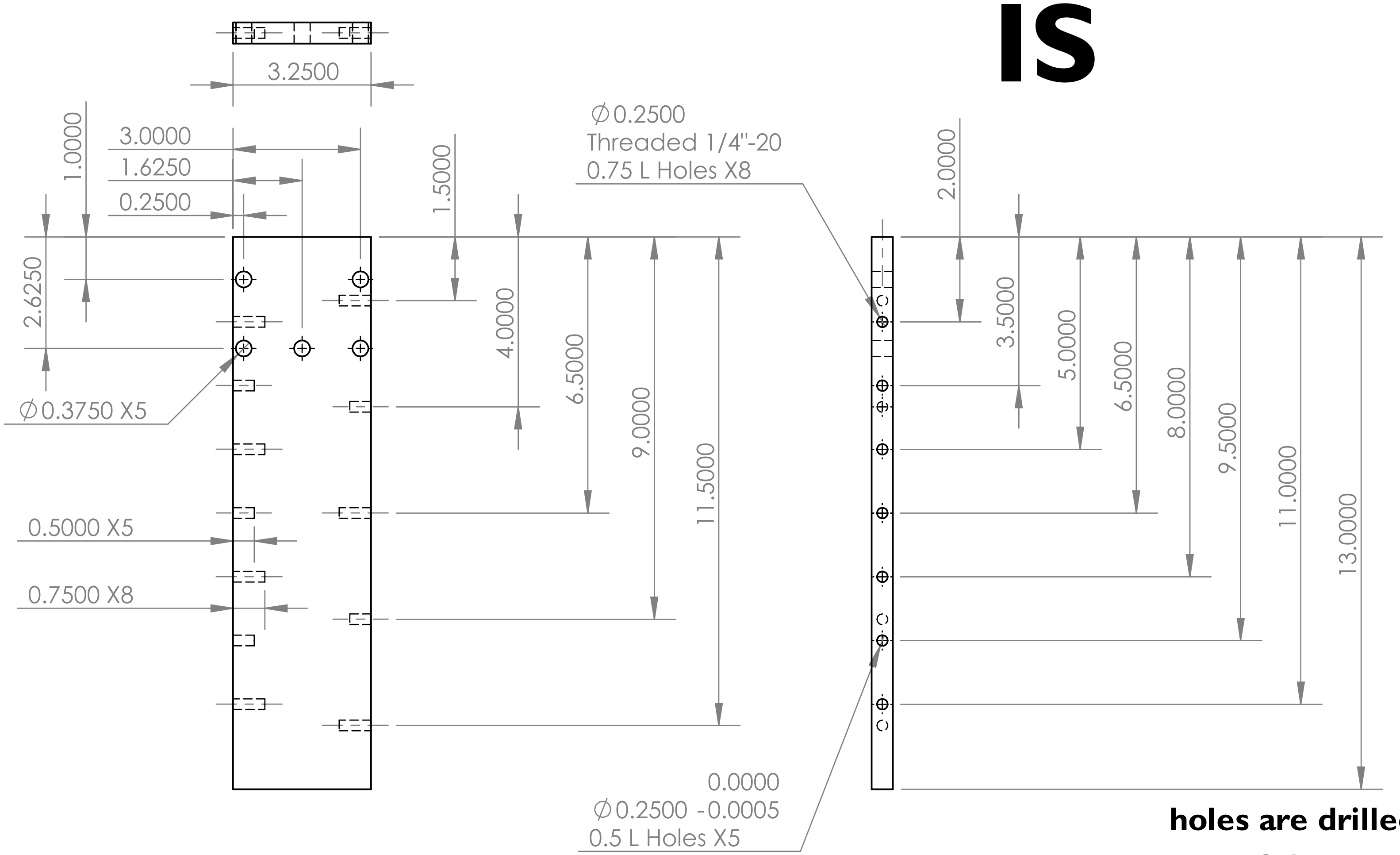
All 1/4" Holes are 1/4" from nearest edge

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010	
DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:	
	CHECKED			Reduction Case: Side Back	
	ENG APPR.				
	MFG APPR.				
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.				
MATERIAL A36 STEEL	COMMENTS:			SIZE A DWG. NO.	REV 1
FINISH					
DO NOT SCALE DRAWING				SCALE: 1:4 WEIGHT:	SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

5 4 3 2 1



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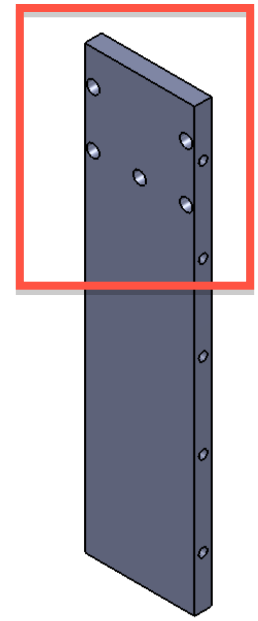

All 1/4" Holes are 1/4" from nearest edge

holes are drilled into the side plate of the gear reduction case

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS: Threads not shown.

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL A36 STEEL

FINISH

NEXT ASSY | USED ON

APPLICATION

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U of M Xebra Team Fall 2010

TITLE:

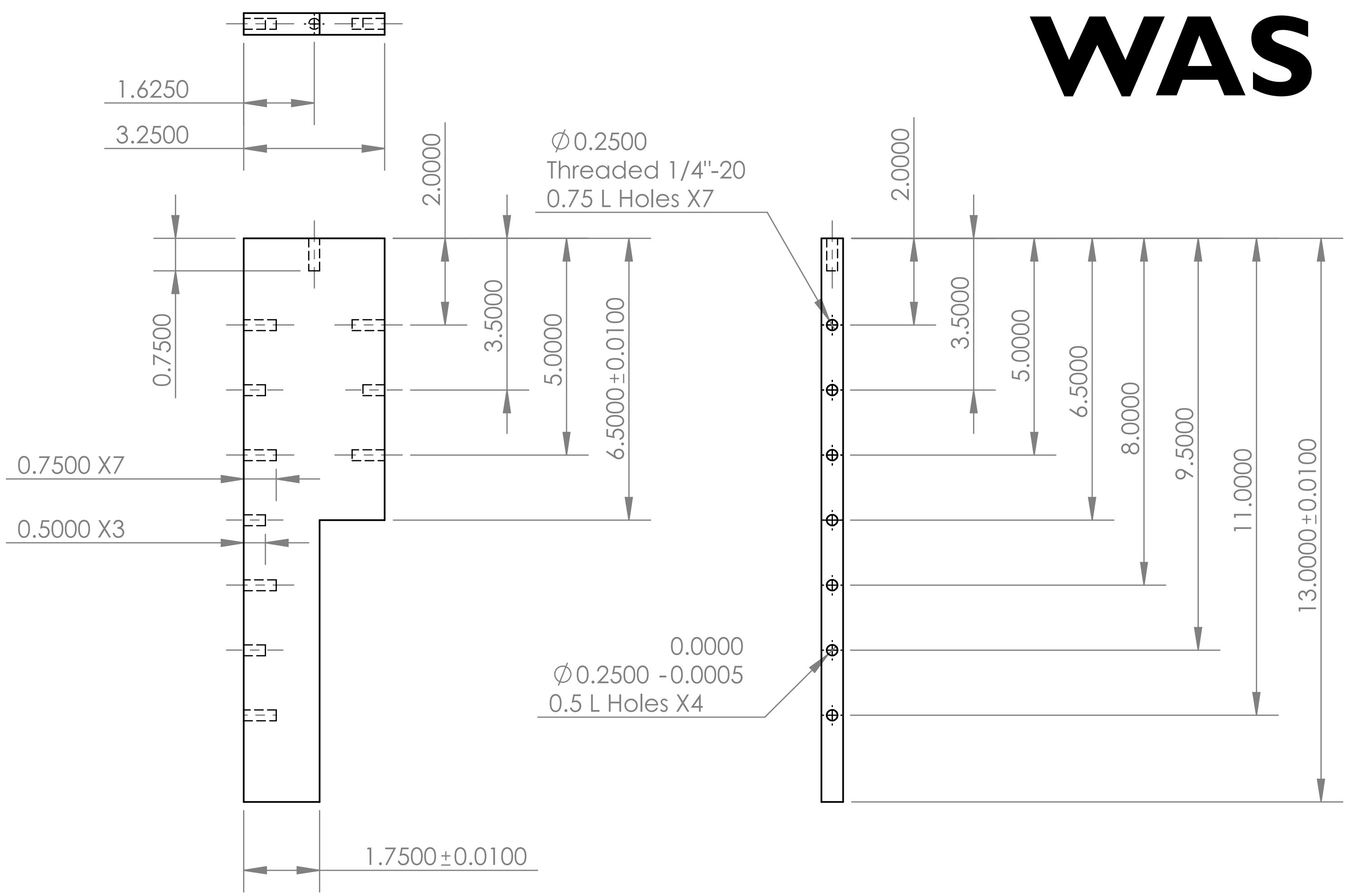
Reduction Case: Side Back

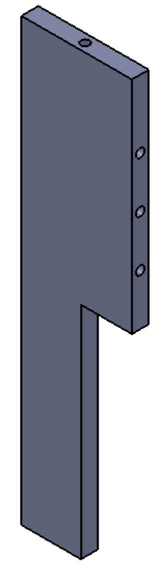
SIZE A | DWG. NO. | REV 1

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

WAS





All 1/4" Holes are 1/4" from nearest edge

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL A36 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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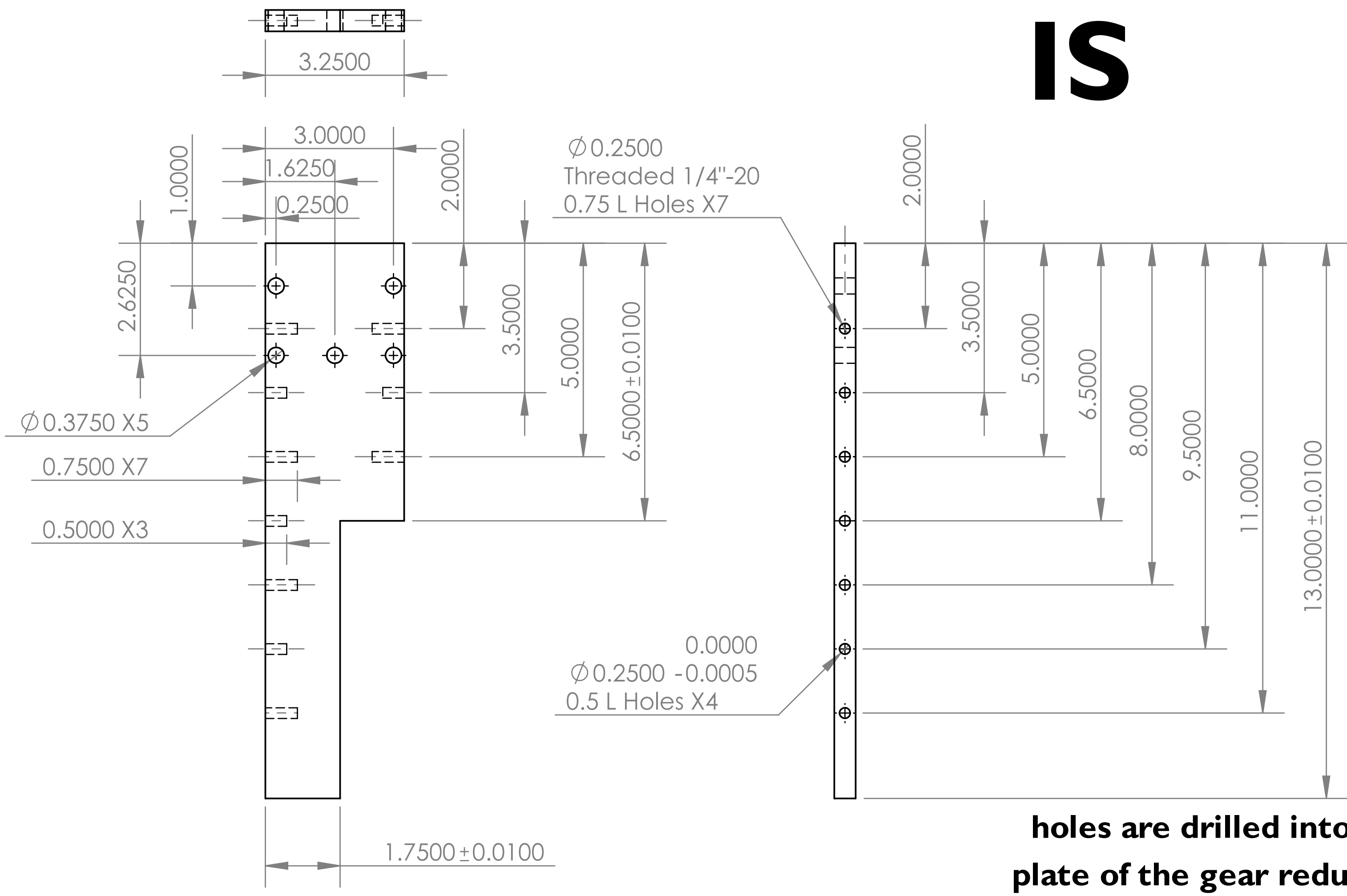
TITLE:

Reduction Case: Side Front

SIZE	DWG. NO.	REV
A		1
SCALE: 1:4	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



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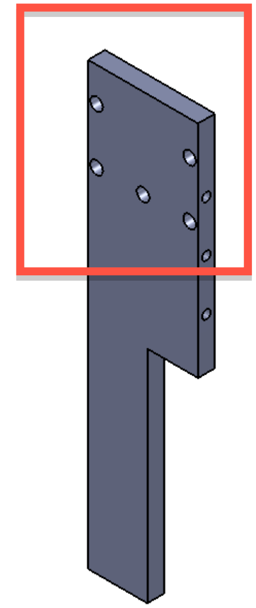
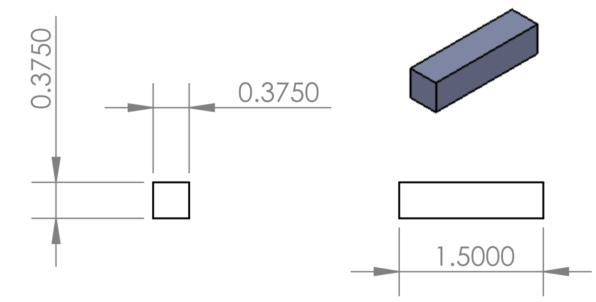


All 1/4" Holes are 1/4" from nearest edge

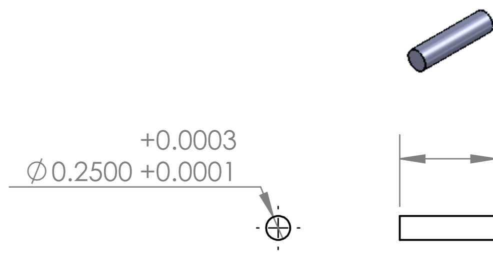
holes are drilled into the side plate of the gear reduction case

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			
			ENG APPR.			Reduction Case: Side Front
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS: Threads not shown.			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: · SHEET 1 OF 1

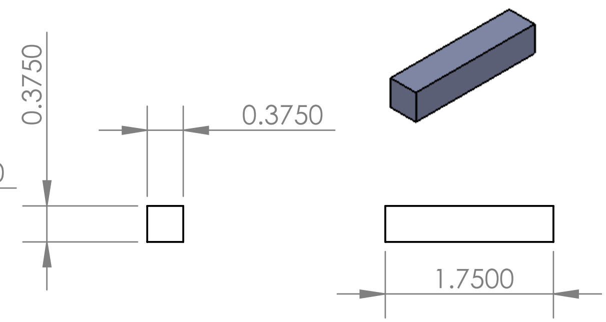
5 4 3 2 1



PM Shaft: Key Stock
Material: High-Carbon Plain Steel

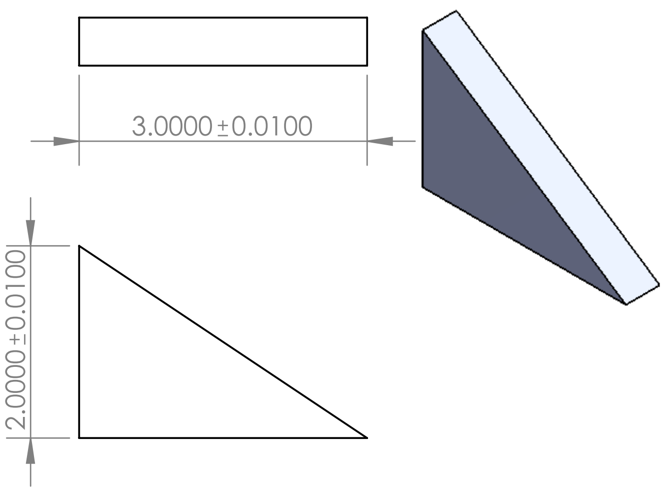


Dowel Pins
Material: Hardened Steel

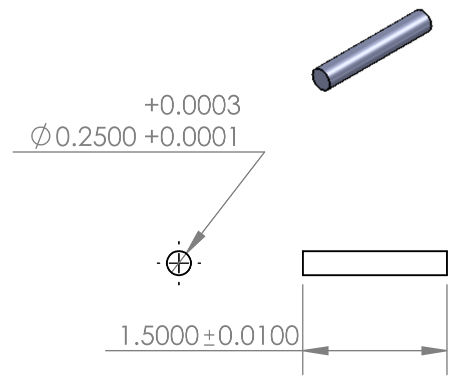


DC Shaft: Key Stock
Material: High-Carbon Plain Steel

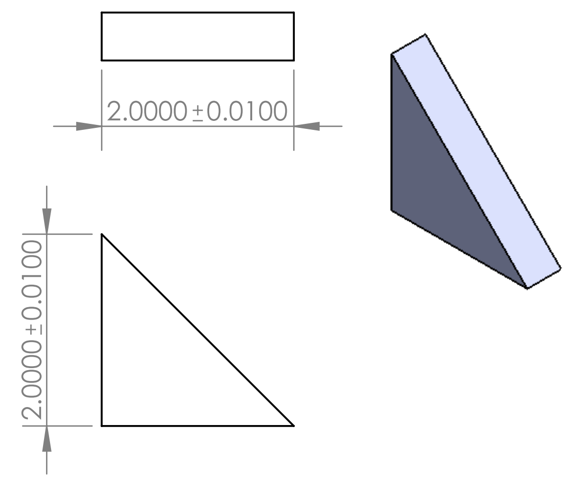
WAS



Reduction Case: Gusset, Front
Material: A36 Steel



Dowel Pins (Long)
Material: 416 Stainless Steel



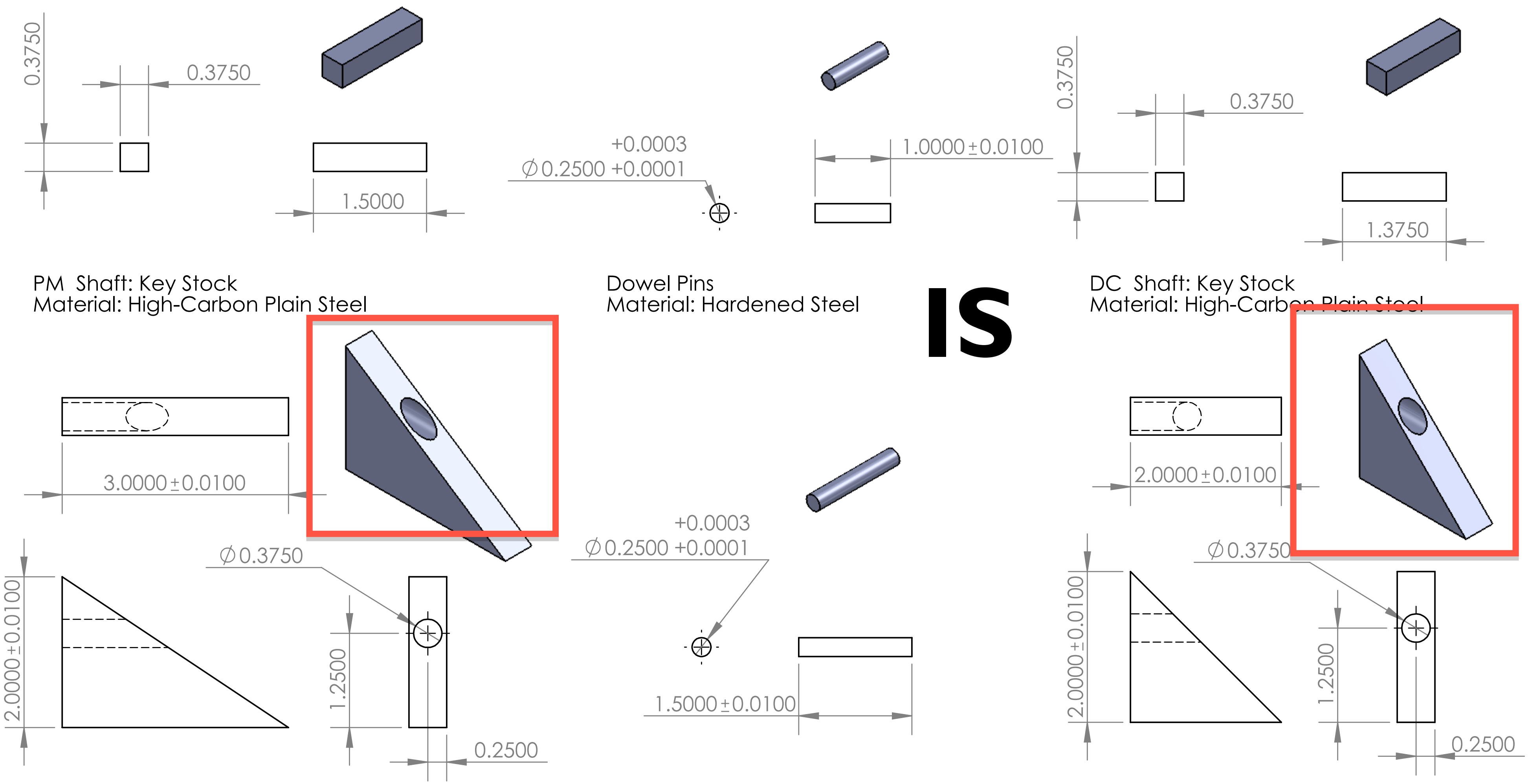
Reduction Case: Gusset, Front
Material A36 Steel

NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
SPECIFIED PER PART
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010
TITLE:
Misc Parts: Gussets, Keys, Dowel Pins
SIZE A | DWG. NO. | REV 1
SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1



PM Shaft: Key Stock
Material: High-Carbon Plain Steel

Dowel Pins
Material: Hardened Steel

IS

DC Shaft: Key Stock
Material: High-Carbon Plain Steel

Reduction Case: Gusset, Front
Material: A36 Steel

Dowel Pins (Long)
Material: 416 Stainless Steel

Reduction Case: Gusset, Front
Material A36 Steel

holes were drilled into the gussets for bolt holes

UNLESS OTHERWISE SPECIFIED:		
DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		
INTERPRET GEOMETRIC TOLERANCING PER:		
MATERIAL SPECIFIED PER PART		
FINISH		
DO NOT SCALE DRAWING		
NEXT ASSY	USED ON	
APPLICATION		

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

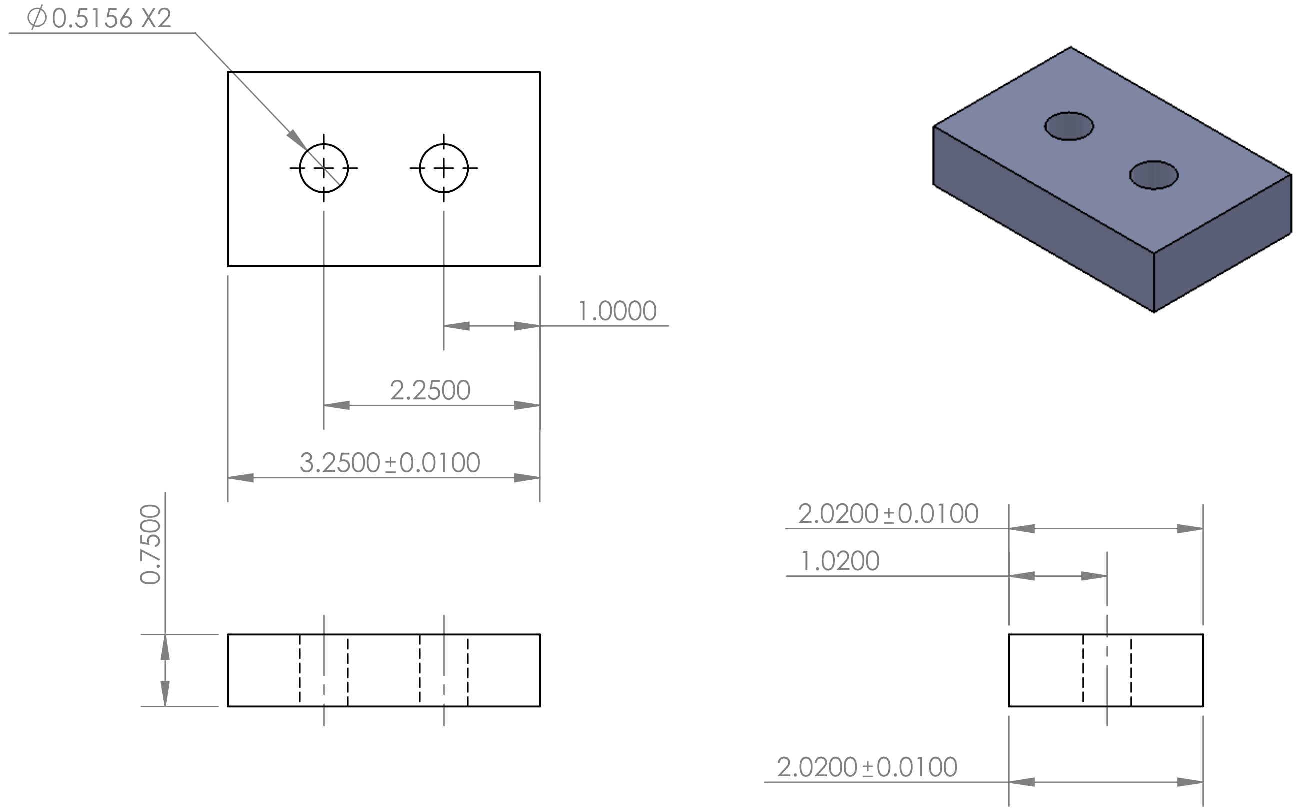
TITLE:

Misc Parts: Gussets, Keys, Dowel Pins

SIZE	DWG. NO.	REV
A		1
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1

WAS



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			CHECKED			
			ENG APPR.			Reduction Case: Tab, Back
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

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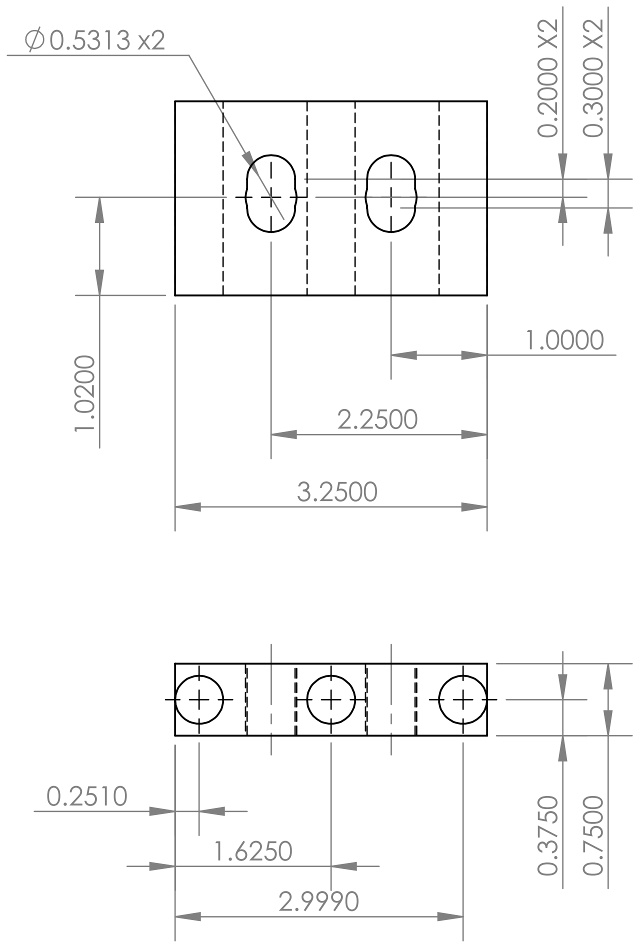
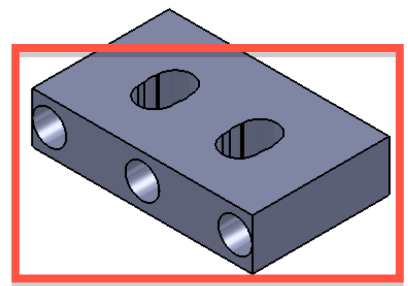
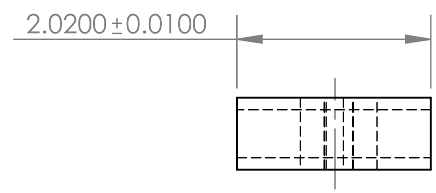
5 4 3 2 1

IS





holes were drilled for bolts and previous drill holes were slotted for alignment



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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL A36 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

TITLE:

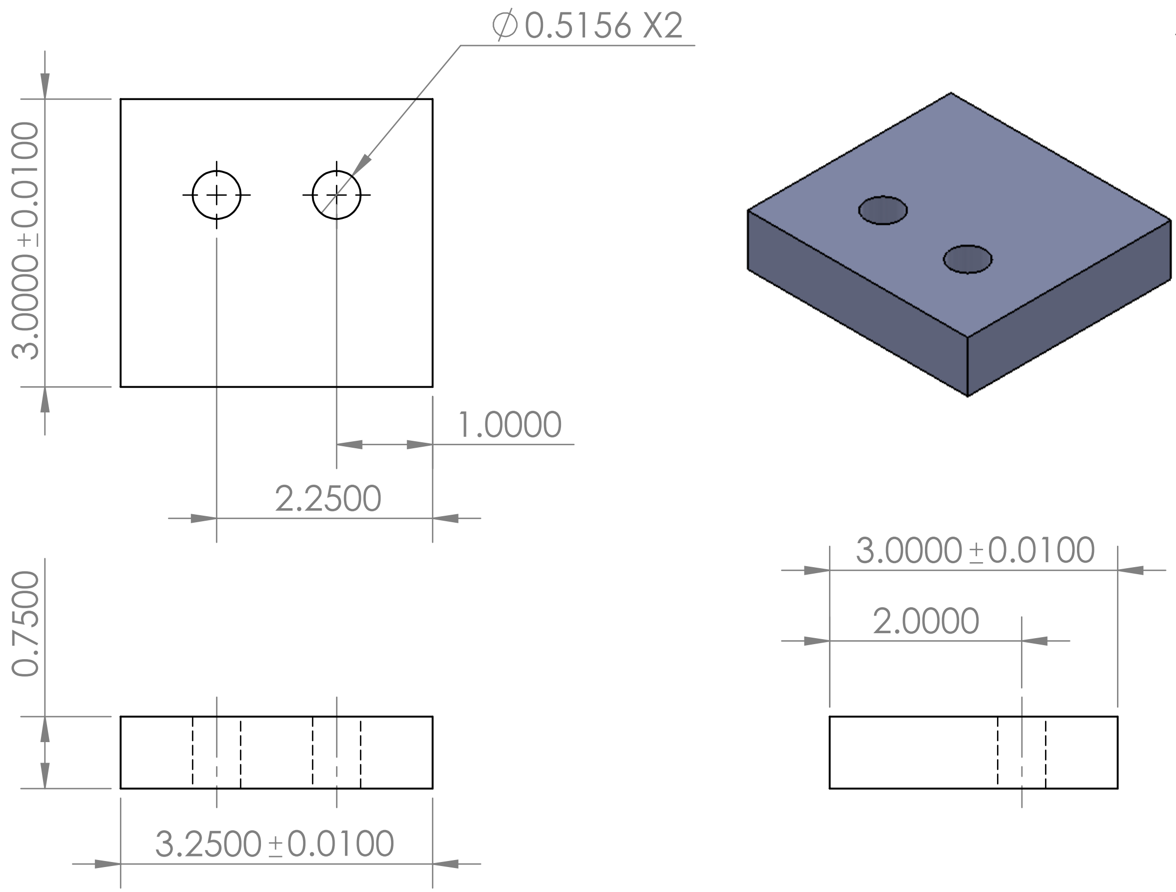
Reduction Case: Tab, Back

SIZE **A** DWG. NO. REV **1**

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

WAS



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
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			CHECKED			Reduction Case: Tab, Front
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 · WEIGHT: · SHEET 1 OF 1

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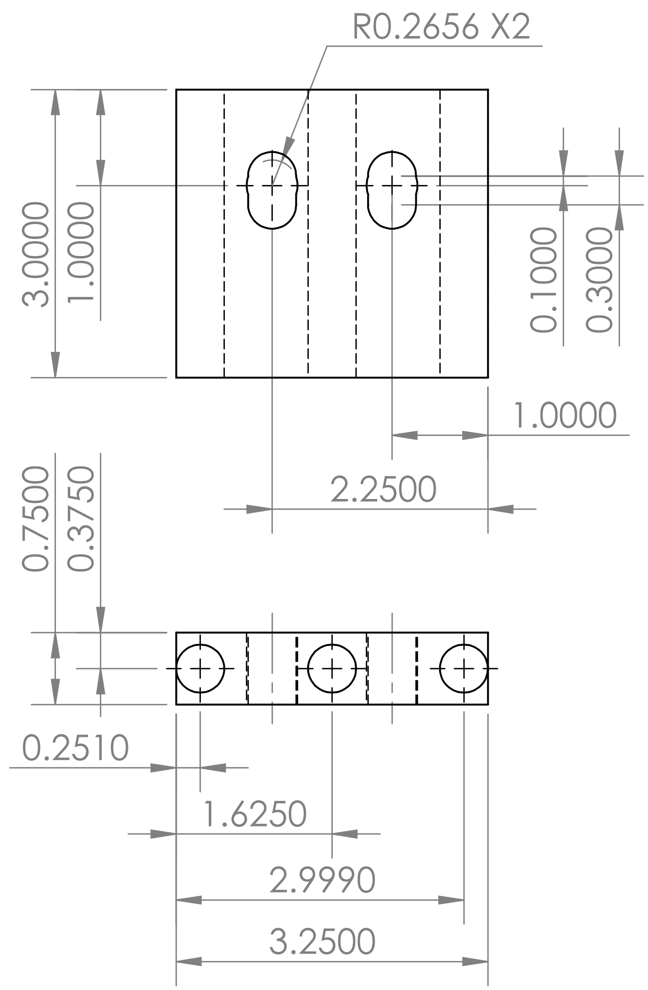
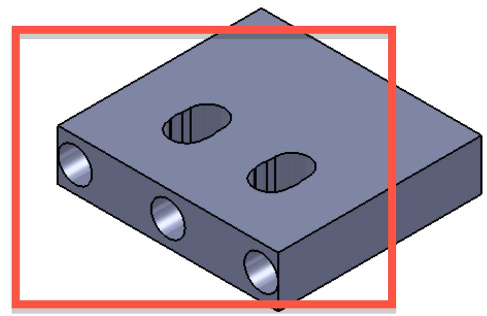
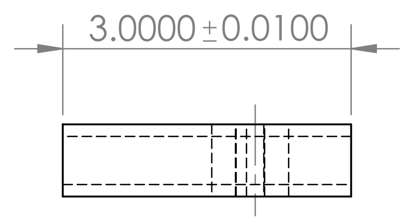
5 4 3 2 1

IS







holes were drilled for bolts and previous drill holes were slotted for alignment

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APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

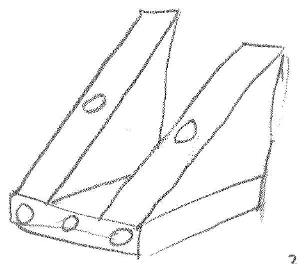
Reduction Case: Tab, Front

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

* TAB & GUSSET ANALYSIS ON THE NEW MOUNTS



1) The tabs and gussets are now mounted to the reduction case using 5 bolts instead of welding.

2) The tabs support the weight of the reduction case, the hydraulic pump/motor and the components within the frame.

Based on Analysis conducted previously on the strength of the tabs we determined the shear force acting on the tabs to be.

$V = 80\ lbs$

* This shear force is distributed over the area of 5 bolt locations of size 1/4" each.

∴ the shear stress distributed across these areas is,

$$\tau_{shear} = \frac{V}{5 \times Area} = \frac{80}{5 \times \pi \times (0.125)^2}$$

$$= 325.95\ lb/in^2$$

Bolts are typically rated for Tensile strength. But they can also withstand shears of values upto 60% of their tensile strength.

Tensile Strength = 180,000 psi

∴ 60% = 108,000 psi

∴ The bolts should be strong enough to withstand the applied shear force.

TENSIONERS

Change and Rationale:

For ease of machining, the square counterbore was changed to a rounded edge square. Also, to allow for proper fit-up of the nut and bolt attaching the tensioner to the gear reduction case, the counterbore depth was increased

Sketch:

See attached.

Part Impacted:

As previously stated, these changes impact the depth of the bolt that mounts the tensioner to the gear reduction case.

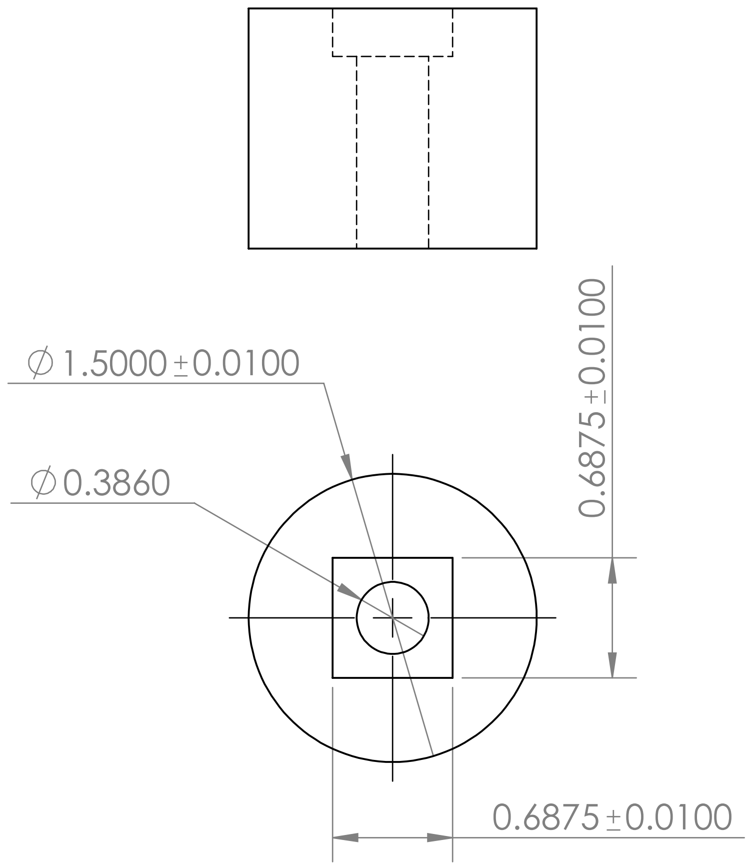
Analysis:

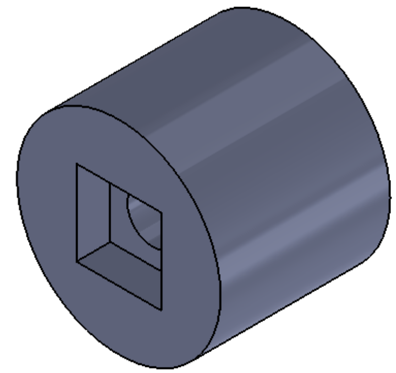
The changes have no affect on the function of the tensioners. They simply allow for the proper fit-up of the tensioners to the gear reduction case.

Change Authorization:

Anuj Shah: 11/29/10
SPONSOR – Andrew Moskalik:

WAS





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NEXT ASSY	USED ON
APPLICATION	

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DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
UHMW Tension Plastic

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

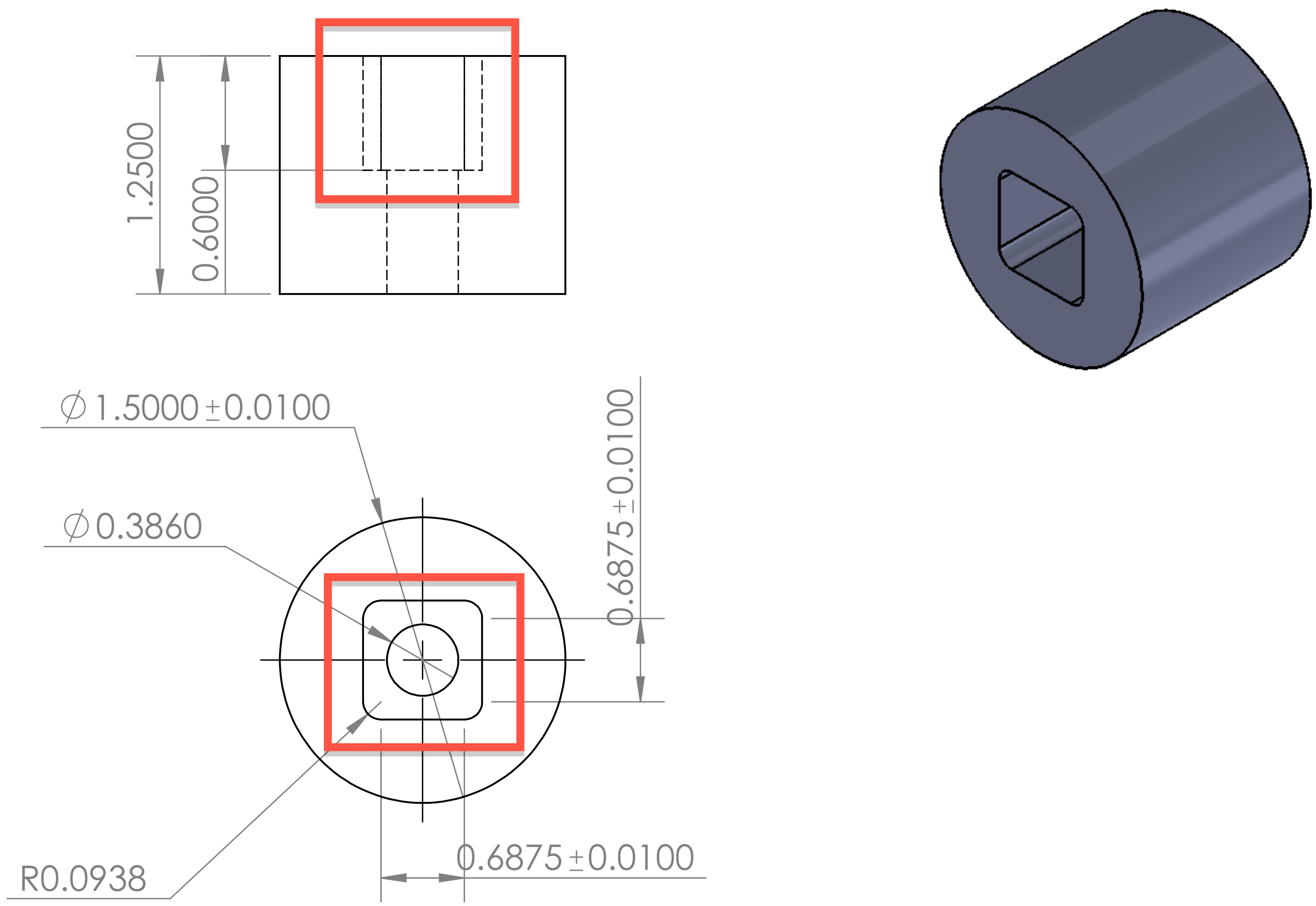
TITLE:

Reduction Case: Tensioner

SIZE	DWG. NO.	REV
A		1

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/6	TITLE:
			CHECKED			
			ENG APPR.			Reduction Case: Tensioner
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL UHMW Tension Plastic	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

DC MOTOR SMALL SPROCKET

Change and Rationale:

Due to fit-up issues, we have changed the size of the smallest sprocket from 14 teeth to 16 teeth. The chain links interfered with the location of the bolt heads and dowel pins on the previous 14-tooth sprocket, due to the close proximity of the bolt heads and dowel pins to the teeth of the sprocket. By increasing the size of the sprocket by using a sprocket with more teeth, the chain is further away from the bolts and pins. Although this alters the gear reduction ratio, the new ratio was approved by our customer (Mr. Xianke Lin).

Sketch:

See attached.

Part Impacted:

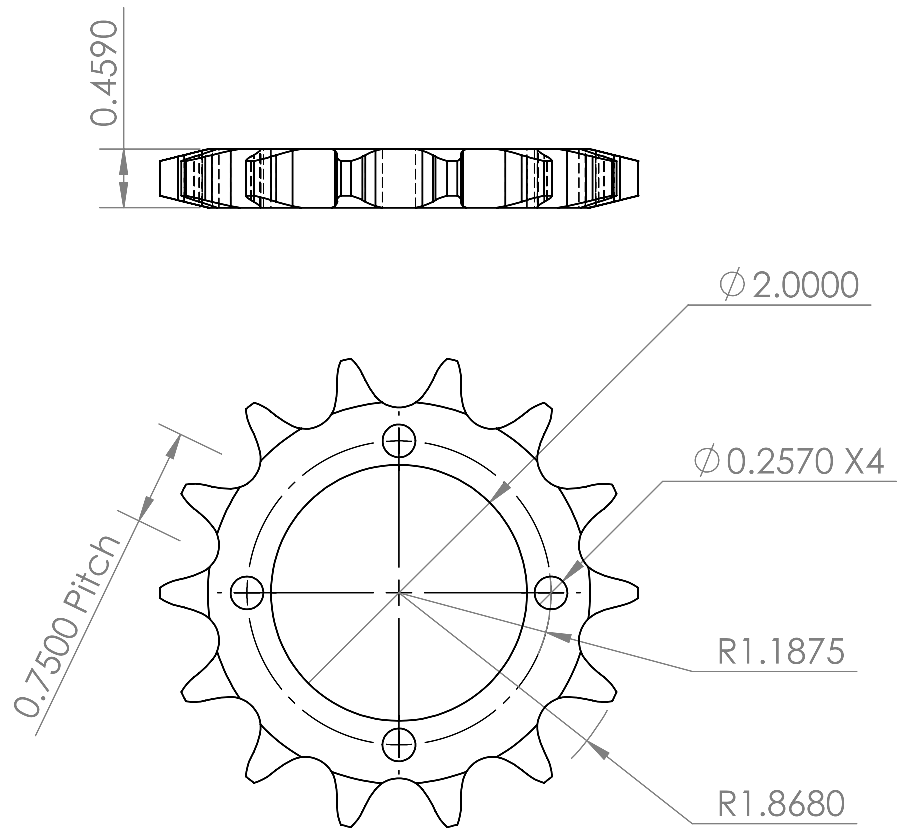
This change only impacts the sprocket itself (it is replaced by a larger sprocket). The DC motor hub is not affected because all of the dimensions remain the same.

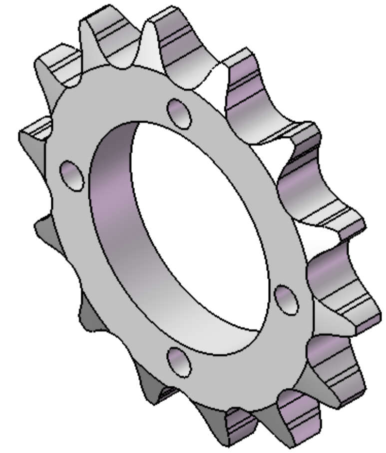
Analysis:

No additional analysis was necessary because this sprocket is larger than the previous sprocket.

Change Authorization:

TEAM – Andrew Gavenda: 12/7/10
SPONSOR – Andrew Moskalik:





WAS

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NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/6
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

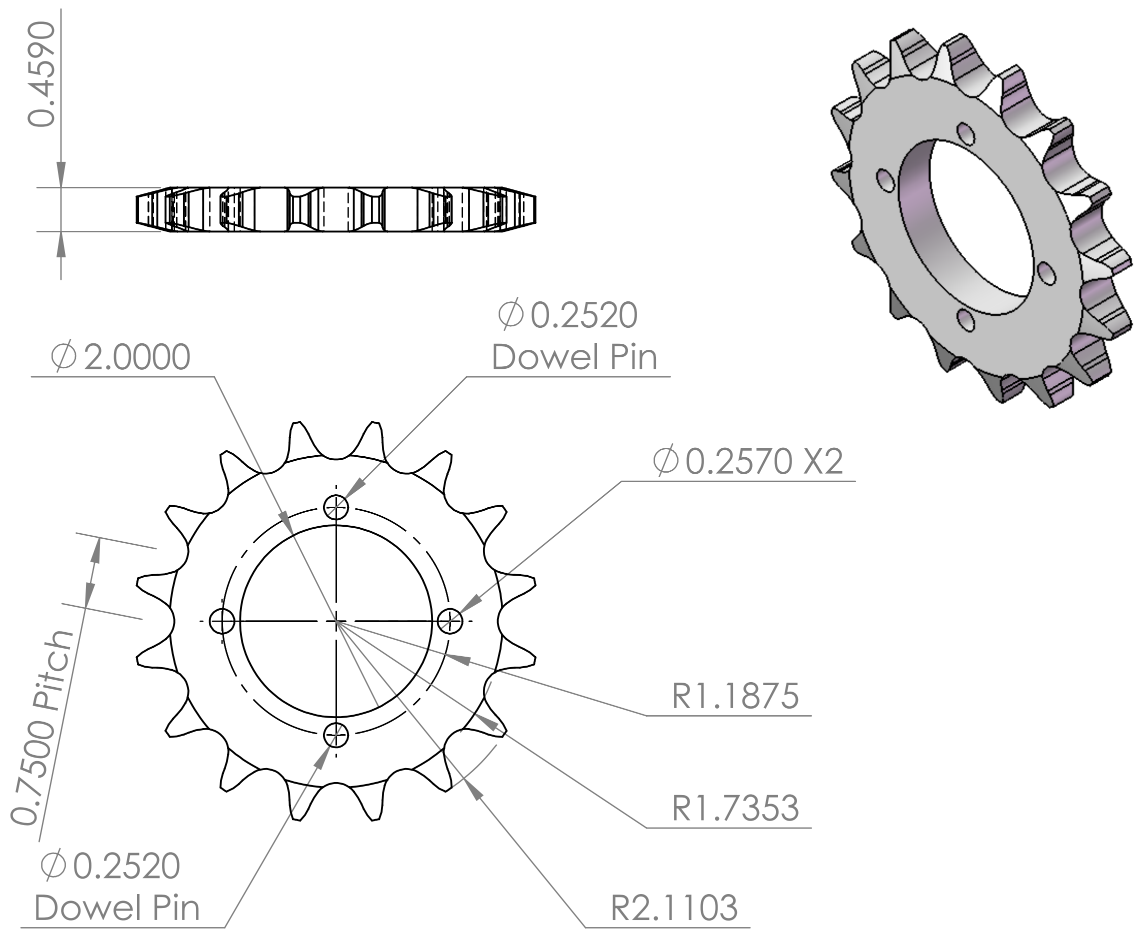
TITLE:

2.5:4.5 DC Sprocket

SIZE A | DWG. NO. | REV 1

SCALE 2:3 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



IS

	UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
	DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/12	TITLE:
		CHECKED			
		ENG APPR.			2.9:4.5 DC Sprocket
		MFG APPR.			
	INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
	MATERIAL Steel	COMMENTS:			SIZE A DWG. NO. REV 1
NEXT ASSY / USED ON	FINISH				
APPLICATION	DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

APPENDIX C: DESIGN ANALYSIS ASSIGNMENT

C.1 MATERIAL SELECTION – FUNCTIONAL PERFORMANCE

C.1.1 Sub-frame Back Bar

C.1.1.1 Function, Objective, and Constraints
Function:
The function of the back sub-frame bar is to mount the gear reduction case (and enclosed components), hydraulic pump/motor (which is directly attached to the gear reduction case) and the DC motor connection (which is directly attached to the gear reduction case) to the vehicle frame. The bar can be modeled as a beam. This beam is not limited by bending because its components (gear reduction case, hydraulic pump/motor, and DC electric motor connection) are mounted to the same sub-frame and are thus assumed to not move relative to each other. Therefore, it must support a particular load (the weight of all components it is carrying) without failure.

Objective:
We want this beam to withstand the specified load but not fail. Simultaneously, we want a beam that is as light as possible. Again, the deflection is not as important as ensuring the load can be carried without failure.

Constraints:
Due to the maximum allowable dimensions within the frame of the vehicle, the length (l) of the bar/beam is fixed. Also, the force (F) is fixed.

C.1.1.2 Material Indices
We are concerned with the material index that provides the material with minimum weight and a prescribed strength. Beginning with the mass of the part, $m = AL\rho$, we investigate the failure load of a beam, $F_f = C_2 \frac{I\sigma_f}{y_m l}$, where $C_2 = 4$ and $y_m = \frac{t}{2}$ for this geometry of a horizontal beam/bar. Additionally, $I = \frac{b^4}{12} = \frac{A^2}{12}$, assuming a square cross-section for the bar. Combining the force (F_f) mass (m), and moment of inertia (I), equations with the constants, we obtain the mass of a beam that supports a load F_f, $m = \left(\frac{6}{C_2}\frac{F_f}{l^2}\right) l^3 \left(\frac{\rho}{{\sigma_y}^{2/3}}\right)$. With $\left(\frac{\rho}{{\sigma_y}^{2/3}}\right)$ as the unconstrained parameter, the material index becomes $\left(\frac{{\sigma_f}^{2/3}}{\rho}\right)$ because we are trying to minimize this parameter.

C.1.1.3 Top Material Choices
Using the CES software, the top five material choices identified are BMI/HS carbon fiber, Cyanate ester/HM carbon fiber, Epoxy/aramid fiber, Epoxy/HS carbon fiber, and PEEK/IM carbon fiber (as seen below in Figure C.1.1.3).

C.1.1.4 Final Material Selection
The final material selected based on the CES recommendations would be Epoxy/aramid fiber. This is based on the material's lightweight characteristic and ability to resist impact. Both of those attributes would be useful in an application where normal usage (driving the vehicle on a road) might subject the component to occasional impacts.

However, intuitively we understand that it would be highly unlikely that this material would actually be utilized for our design. Cost and manufacturability are two detriments of this material. More importantly, we desire a material that produces higher safety factors in yielding, which is not completely characterized by this analysis. Additional considerations such as assembly also factor into the decision to not use Epoxy/aramid fiber. Unlike our selected material (carbon steel), epoxy cannot be welded to the frame of the vehicle (which is steel) and would make our design much more difficult to assemble. Thus if we relax our constraints (as seen below in Figure C.1.1.4), we notice that a high number of carbon steels are available of viable materials to select for our application.

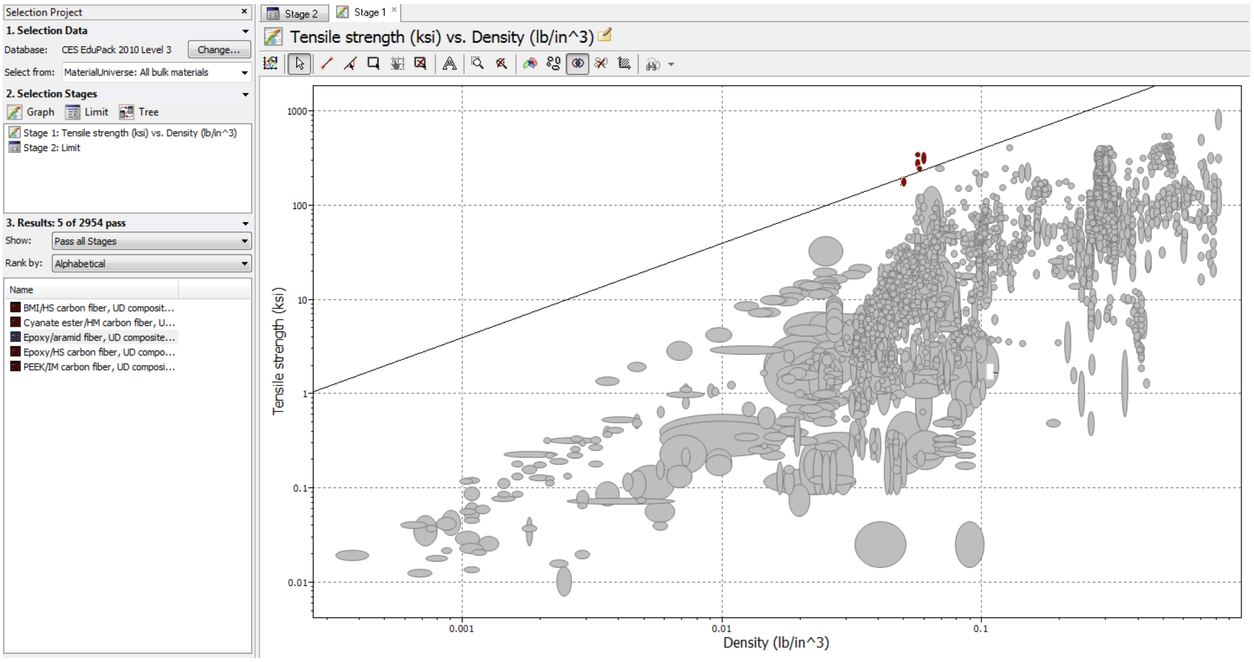


Figure C.1.1.3: CES Top Material Choices

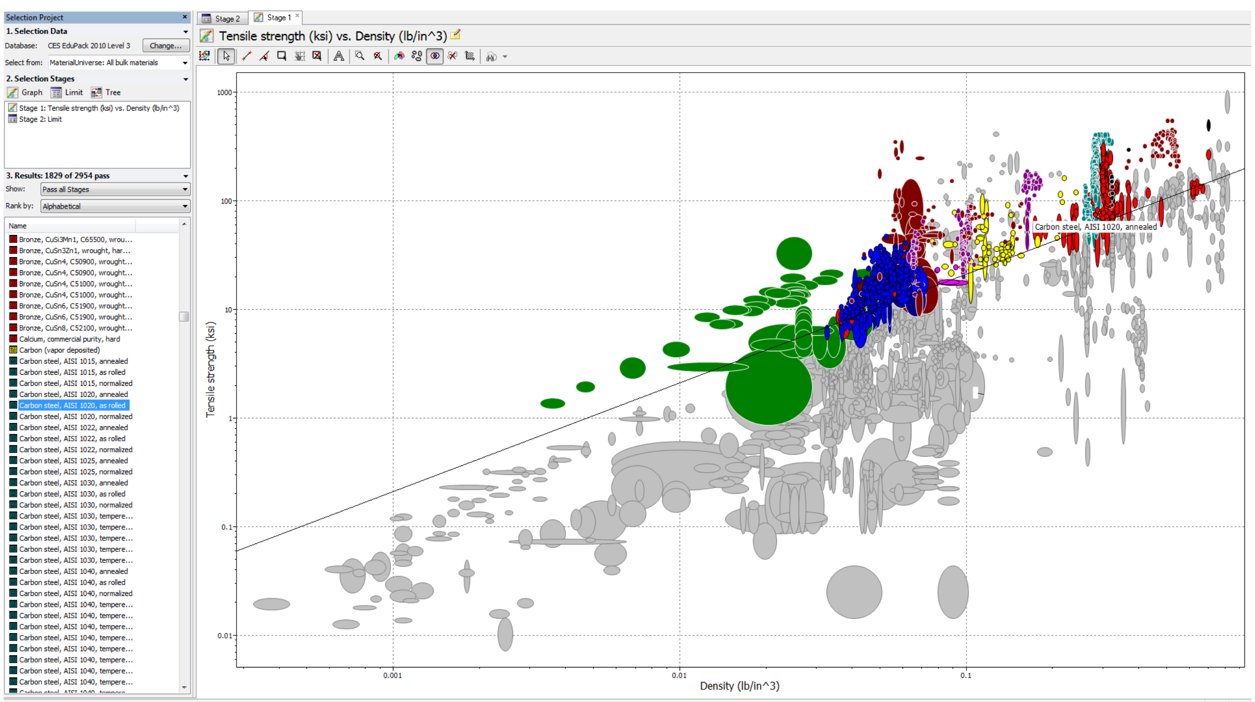


Figure C.1.1.4: CES Relaxed Material Constraints

C.1.2 Gear Reduction Case

C.1.2.1 Function, Objective, and Constraints

Function:

The major function of the gear reduction case is to support the load of all the components that are mounted to it. This includes the weight of case itself, but more importantly the weight of the hydraulic pump/motor. To support these loads means that the case and material must support bending moments, axial loads, shear, and torsion. Due to the weight and location of the pump/motor, the dominating behavior that the case must support is bending. In particular, we investigate the pump/motor plate that the pump/motor will be mounted to. This can be modeled as a wide, flat beam.

Objective:
The case must be as safe and lightweight (minimum mass) as possible (in that order).

Constraints:
The material is constrained by the dimensions of the maximum size of the gear reduction case. In this case, the constraint is the length of the plate/beam. The component must carry the bending without failure and thus its stiffness must be maximized.

C.1.2.2 Material Indices
We are concerned with the material index that provides a material with a minimum weight and a specific stiffness. Beginning with $S = \frac{F}{\delta} \leq \frac{CEI}{L^3}$, we incorporate the objective of minimizing the mass ($m = AL\rho$) of the part. Combining both equations, $m = \left(\frac{12S}{CL}\right)^{1/2} L^3 \left(\frac{\rho}{E^{1/2}}\right)$. Therefore, the material index is $M = \frac{E^{\frac{1}{2}}}{\rho}$ because we are trying to minimize the non-constrained parameter.

C.1.2.3 Top Material Choices
Using the CES software, the top five material choices identified are Cyanate ester/HM carbon fiber, Fir (abies procera), Palm, Spruce (picea rubens), and Willow (salix alba) (as seen below in Figure C.1.2.3).

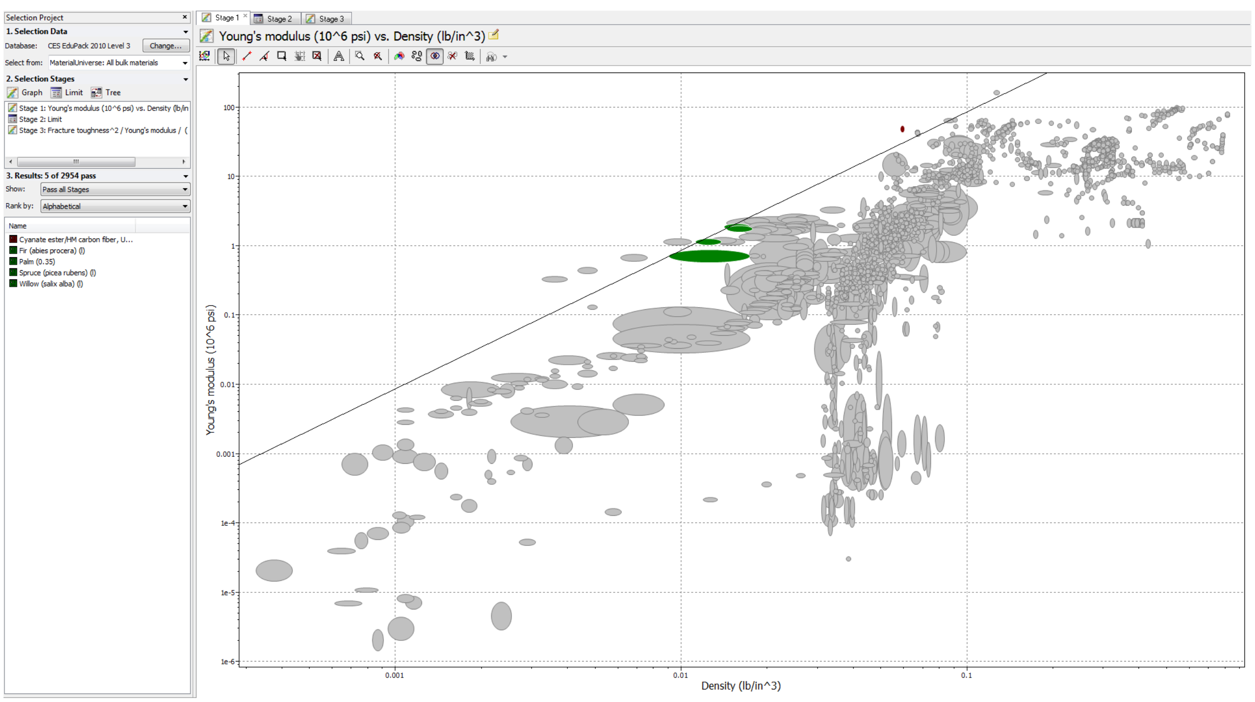


Figure C.1.2.3: CES Top Material Choices

C.1.2.4 Final Material Selection
The final material selection based on the CES recommendations would be Cyanate ester/HM carbon fiber. This is based on the material's lightweight characteristic and typical application in high performance spacecraft, aircraft, and missiles. Its ability to absorb moisture is also beneficial in an automotive setting.

Intuitively however, we understand that it would be highly unlikely that this material would actually be utilized in our design. Cost and manufacturability are two detriments of this material. Assembly considerations are also a factor in the practical application of Cyanate ester/HM carbon fiber in our design. Unlike our selected material (carbon steel), epoxy cannot be welded to the frame of the vehicle (which is steel) and would make our design much more difficult to assemble. Thus if we relax our constraints (as seen below in Figure C.1.2.4), we notice that a high number of carbon steels are available of viable materials to select for our application. Accordingly, we ultimately selected A36 steel and 1018 steel for our structural components including the sub-frame back bar and gear reduction case.

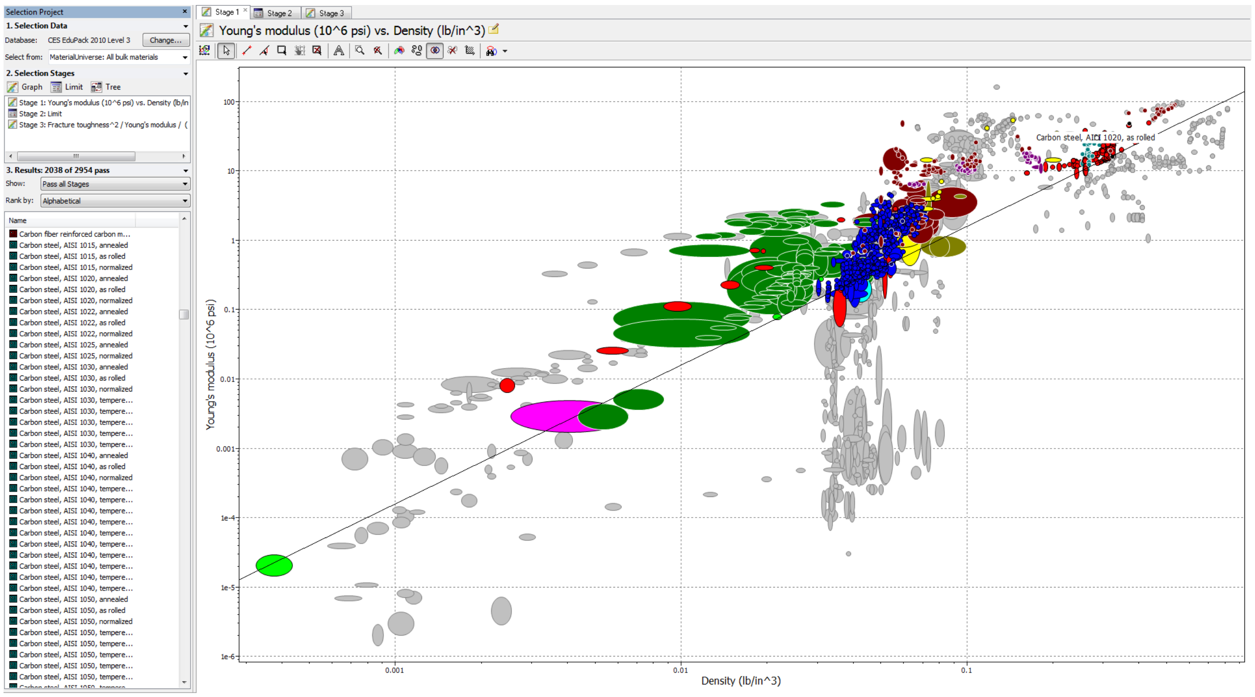


Figure C.1.2.4: CES Relaxed Material Constraints

C.2 MATERIAL SELECTION – ENVIRONMENTAL PERFORMANCE

The two materials that the CES software outputted most suitable for our design were Epoxy/aramid fiber and Cyanate ester/HM carbon fiber. But due to concerns related to manufacturing, availability, assembly and other reasons explained in more detail in the material selection for functional performance section above we have picked two other materials that best meet our requirements. We have conducted the analysis for environmental performance on A36 steel and 1018 steel.

Using the SimaPro software we were able to determine the overall environmental impact of the materials that we used in our final design. We had to find the steels within the software that closely represented the materials we used. For 1018 we used galvanized steel sheet and for A36 we picked the hot rolled sheet from the options available. We picked these two based on their properties and how they correlated to the materials that we used. Using the dimensions and the densities of these two materials we were able to determine that we used 40 kg of A36 steel and about 20 kg of 1018 steel in our entire design. These weight values also include the weights of the hubs, shafts and other parts that were manufactured out of steel.

Based on the results using the SimaPro software and seen in Figures C.2.1, C.2.2, C.2.3 and C.2.4 below we can conclude that Hot rolled Sheet steel's (A36 steel) on the environment in each of the EcoIndicator 99 damage classifications is worst compared to 1018 steel. Based on the EI99 point results the "minerals" damage meta-categories followed by the climate change category have the largest effects on the environment. Based on the EcoIndicator point value results seen in the single score graph below, you can see that A36 has the worst impact on the environment in comparison to 1018 steel over the entire life cycle of this product. This is a concept design, making it the only one of its kind. Even though in some categories the two materials impact on the environment is pretty similar, on the whole based on the results from the SimaPro program we determine that A36 is worst for the environment.

Results from SimaPro:

Figure below shows that A36 steel has more emissions in every compartment in comparison to 1018 steel.

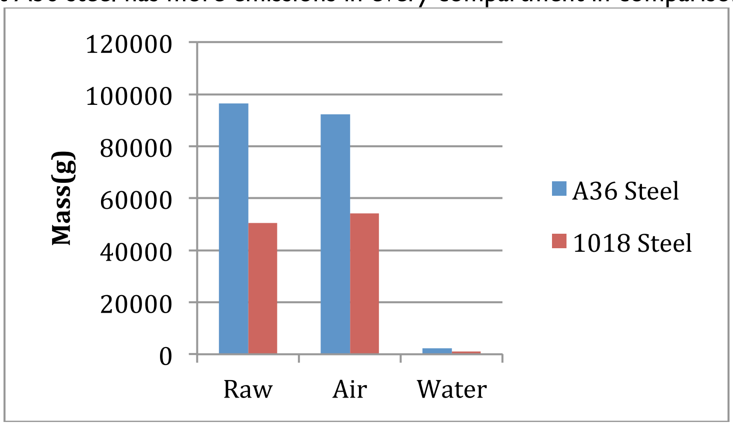


Figure C.2.1: Shows the total emissions in various compartments from both the materials used

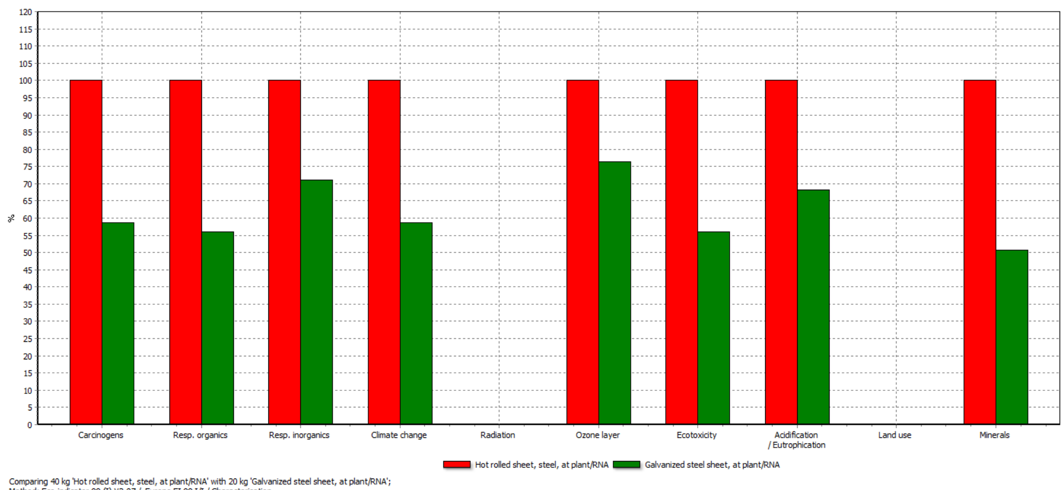


Figure C.2.2: Shows the impact of the materials in each of the EcoIndicator 99 categories

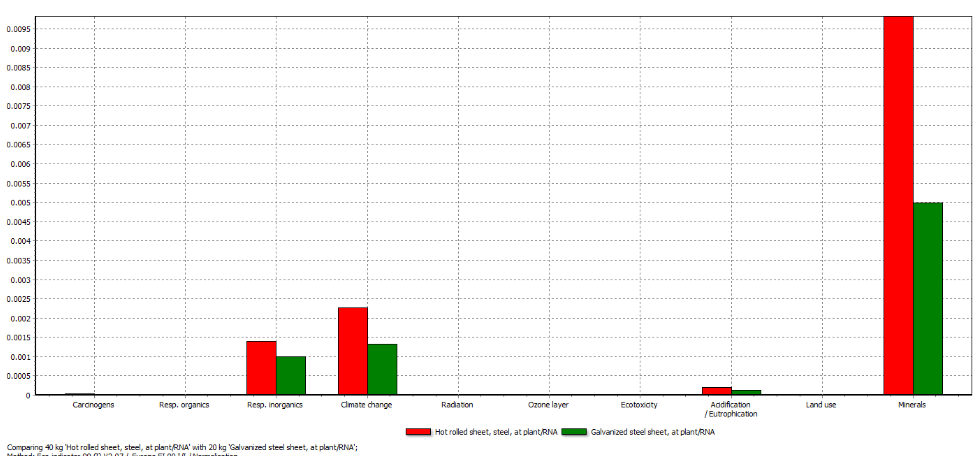


Figure C.2.3: Shows the point results for the damage meta-categories

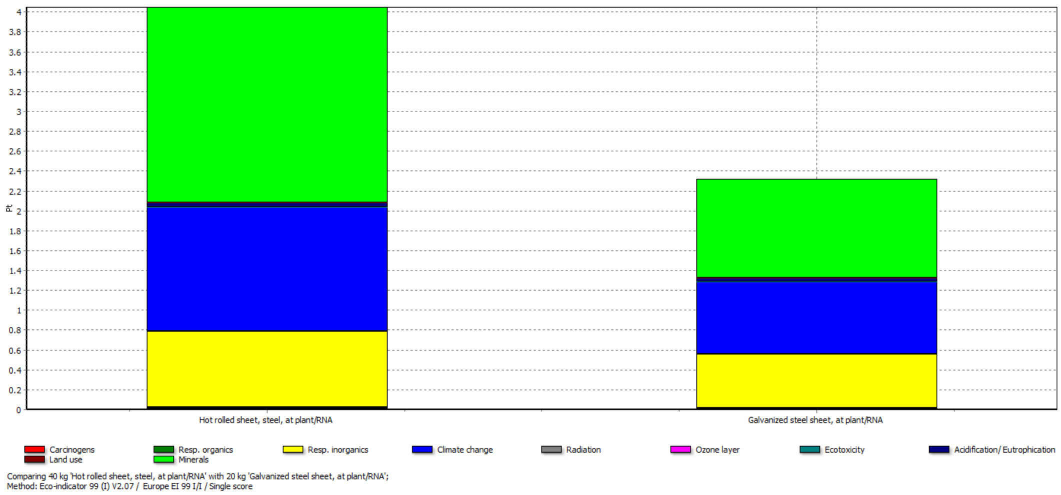


Figure C.2.4: Shows the results of the material performance in total E199 points

The results of this analysis make us aware of the impacts of both these materials on the environment. Based on the parts these materials are used to manufacture and their individual requirements we feel that the materials selected meet both needs adequately.

C.3 MANUFACTURING PROCESS SELECTION

C.3.1 Production Volume

The power coupling that we are designing for the EPA is solely for proof of concept purposes. The EPA wants to know if hydraulic-electric hybrids are worth investing more research, time, and money into. By modifying a small vehicle, such as the Blue Xebra, to run on hydraulic power as well as electric the EPA will be able to assess the effectiveness of this hybridization. The EPA has no intent of working with ZAP to mass-manufacture electric-hydraulic hybrids, it is only using the Xebra for a test bed. It is for these reasons that we have decided to set the production volume of our power coupling at one unit.

If the EPA had asked us to design the power coupling for a regular sized vehicle such as a sedan or a truck, we would have raised our production volume greatly. As it stands, the Zap Xebra is not a regular sized car that could be mass-produced and marketed to consumers. It is a small utility vehicle that is relatively inexpensive and provides the EPA with a solely electric vehicle to test on.

C.3.2 Process Selection

After deciding on the low production volume we began to look at potential shaping techniques to manufacture our gear reduction housing. CES EduPack 2010 was used to help determine adequate manufacturing processes. After selecting "shaping" as our selection library we began to apply certain limits.

The first limit that we applied was a limit on the shape. The reduction housing that we have designed is comprised of several plates that bolt together with dowel pins as well. Some of these plates are quite intricate and have holes and bores going through all three axes. It is for these reasons that we selected our shape to be a Solid 3-D. The other options for shape did not apply to our reduction housing as well.

The second limit that we applied to the search was a limit on batch size. Obviously when discussing manufacturing batch size is of great concern and this is why this was our second limit. As determined in the previous section, we set the batch size limit to a maximum of one. After applying the first two limits to the selection, we reduced our potential shaping processes from 60 down to 27. We decided that this was still too many processes to select from and applied another limit.

The third limit that we applied to the search was a limit on process tolerance. The alignment of the shafts in our system was of greatest concern. If the shafts were even the slightly misaligned a radial load could incidentally be applied to the pump/motor and damage it. It is for this reason that the tolerances of our reduction housing are extremely tight. For this limit, we set the max tolerance to be 1 thousandth or 0.001in. While not all of our machining will be held to such high tolerances, certain components need to be and therefore the machinery must have these capabilities.

After applying the limit on shape, batch size, and tolerance the list of acceptable manufacturing processes was presented as shown in Figure C.3.2.1 below.

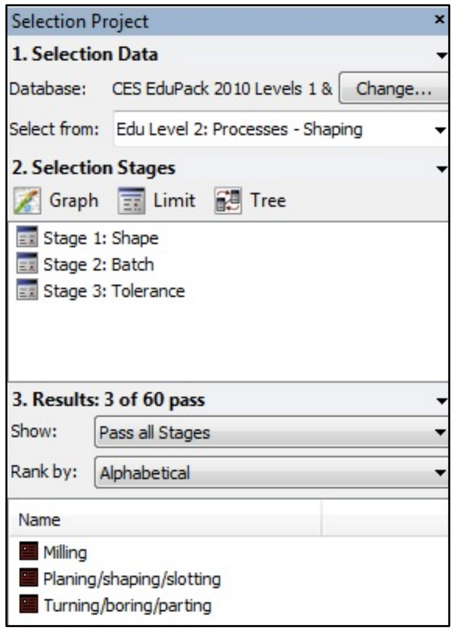


Figure C.3.2.1: After applying three limits, three separate machining processes remained

The three processes that resulted are Milling, Planning/Shaping/Slotting, and Turning/Boring/Parting. All of these processes will be necessary to fully machine our reduction housing. The plates will need to be planed or surface ground to a toleranced thickness, the shafts need to be turned, the plates need to be milled to size and finally bored to allow for bearings to be installed. The CES selection method allowed us to determine these processes would be well suited for our manufacturing operation.

APPENDIX D: DESIGN RISK ANALYSIS

			DESIGN 1		DESIGN 2		DESIGN 3	
RISK	DESCRIPTION	WEIGHT (0-1)	Score (1-5)	Weighted Score	Score (1-5)	Weighted Score	Score (1-5)	Weighted Score
Assurance	How much confidence do we have that the design will work?	5.0	2	10.0	3	15.0	3	15.0
Assembly	How much room for error is there in assembly?	3.4	3	10.2	3	10.2	3	10.2
Complexity	Number of parts and their interactions with each other.	2.8	2	5.6	2	5.6	3	8.4
Machining	Estimated time to machine parts.	2.6	4	10.4	3	7.8	2	5.2
Tolerance	How tight must tolerances need to be for design to work?	2.4	3	7.2	3	7.2	4	9.6
Procurement	Estimated lead time needed for all parts.	1.4	3	4.2	3	4.2	3	4.2
Newness (novel)	Has there been a design like this before?	1.4	1	1.4	2	2.8	2	2.8
Material	Material attributes (UTS, cost, weight, machinability) fit application.	1.0	1	1.0	1	1.0	1	1.0
		20	TOTAL	50.0	TOTAL	53.8	TOTAL	56.4
	(1=low risk - 5=high risk)							

FUNCTIONAL AREA	SELECTED CONCEPT	SELECTED CONCEPT	SELECTED CONCEPT
Reduction System	chain & sprocket	chain & sprocket	spur gears
Coupling	fixed spline	flexible spider	fixed flange
Pump/Motor Mounting	horizontal	undercarriage	horizontal
Gearbox Mounting	tab & gusset	tab & gusset	bolt from sides

max = 40	max = 100
min = 8	min = 20

APPENDIX E: CONCEPT GENERATION

Below are additional concepts that were not previously discussed.

E.1 REDUCTION GEARBOX

Roller Chain and Sprocket without Idler/Tensioner

This system is identical to the roller chain and sprocket described previously, except that no tensioner or idler would be implemented. In this system, the chain length is exact for each separate gear ratio or chain links are removed when gear ratios are changed. This allows eliminates the need for an additional idler/tensioner subsystem.

Spur Gear Mesh

Spur gears are a conventional method for transmitting torque and rotation through a gear reduction ratio (Figure E.1.1). The gears directly mesh with each other. Thus, the alignment of the input/output shafts must be very precise. Additionally, changing gear ratios requires both gears to be changed which could affect the center-to-center distance between the two motors' output shafts without proper calculation of gear sizes.

Belt Drive

This system is very similar to a roller chain and sprocket drive (Figure E.1.2). However, a belt drive does not require an idler or tensioning device. The belt does need to be exactly the correct length. Therefore, there must be a different belt for each gear ratio we use.

Figure E.1.1: Spur Gear Mesh

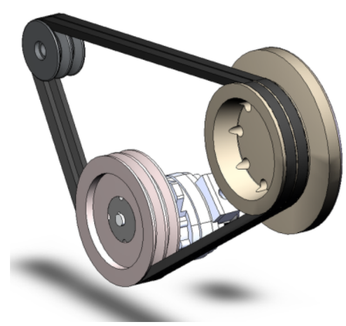


Figure E.1.2: Belt Drive

E.2 COUPLING [17]

Rigid Clamp/Split Coupling

A clamp coupling is nearly identical to the sleeve coupling except that a metallic ring clamp, occasionally lined with rubber or some other pliable material, tightens around the shafts to connect them (Figure E.2.1) [23].

Flexible Bushed Pin Coupling

A bushed pin coupling is identical to a rigid flange coupling except that instead of rigid bolt connections, the bolts are covered with rubber, leather, or some other flexible material [18]. Thus, there is some allowable radial misalignment (Figure E.2.2).

Flexible Universal Joint

The universal joint allows a two-part shaft to have angular misalignment due to a pair of hinges oriented 90° to each other and connected by a cross-shaft (Figure E.2.3) [18].

Figure E.2.1: Rigid Clamp Coupling

Figure E.2.2: Flexible Bushed Pin Coupling

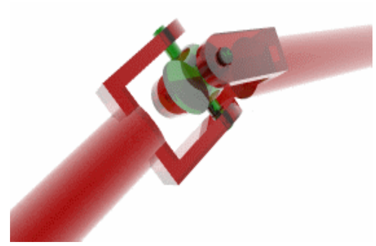


Figure E.2.3: Flexible Universal Joint

Flexible Oldham Coupling

The Oldham coupling allows for torque transfer between to non-collinear shafts [18]. The coupling has three discs, one coupled to the input, one coupled to the output, and the middle disc is joined to the others by tongue and groove (Figure E.2.4).

Flexible Constant Velocity Joint

Similar to the universal joint, the constant velocity joint allows for torque transmission through a variety of angles [18]. Whereas the U-joint uses two hinges, the CV-joint spheres in grooves (Figure E.2.5). True to its name, the CV-joint ensures constant velocity regardless of angular rotation angle.

Flexible Bellows Coupling

The bellows coupling is similar to a rigid sleeve coupling in that a solid connection is achieved using set screws [14]. However, the middle part of the bellows coupling is flexible, allowing slight misalignment in the radial, axial, and angular directions (Figure E.2.6). Thus, the bellows coupling allows for low backlash operation [24].

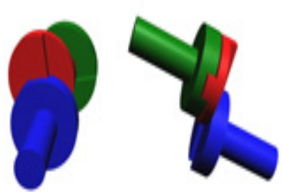


Figure E.2.4: Flexible Oldham Coupling

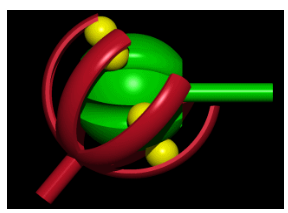


Figure E.2.5: Flexible Constant Velocity Joint

Figure E.2.6: Flexible Bellows Coupling

Flexible Thompson Coupling

The Thompson coupling is a type of constant velocity joint that can be loaded axially and maintains constant velocity over a variable range of input angles (Figure E.2.7). This is accomplished using a pair of special “cardan” joints [18].

Flexible Disc Coupling

A disc coupling transmits torque from a driving to a driven bolt tangentially on a common bolt circle [18]. The circle transmits the torque and misalignment is allowed due to the deformation of the material between the bolts (Figure E.2.8).

Flexible Diaphragm Coupling

Similar to the flexible disc coupling, diaphragm couplings transmit torque from the outside edge to the inside edge of a flexible plate, across a spacer, and then from the inside to the outside edge of another plate [18]. The deformation of the series of plates allows for misalignment (Figure E.2.9)

Figure E.2.7: Flexible Thompson Coupling

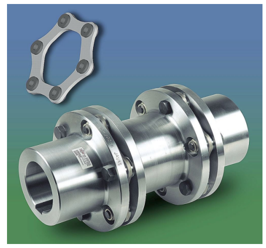


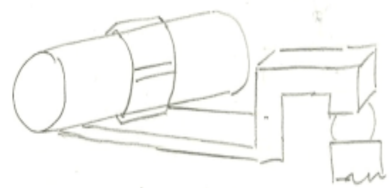
Figure E.2.8: Flexible Disc Coupling

Figure E.2.9: Flexible Diaphragm Coupling

E.3 PUMP/MOTOR MOUNTING

Undercarriage Cradle Support

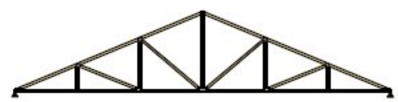
The undercarriage cradle support involves removing the current mount for the DC motor and replacing it with a support below the motor, as seen to the right. This allows for additional space above the DC motor for other components since the DC motor is the lowest point of the vehicle above the sway bars and suspension. This method affects the rigidity of the drivetrain components very little because the motor is still mounted to a sub-frame assembly through dampers.



E.4 GEARBOX MOUNTING

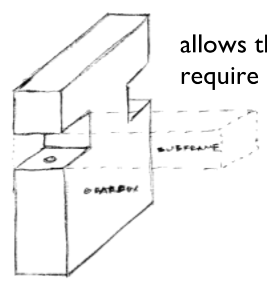
Truss

A truss structure uses triangular units made of straight members whose ends are connected at joints (as seen to the right). Forces are intended to act only at these joints. Thus forces in all members are strictly compressive or tensile with torques explicitly excluded [25].



Internal Tab

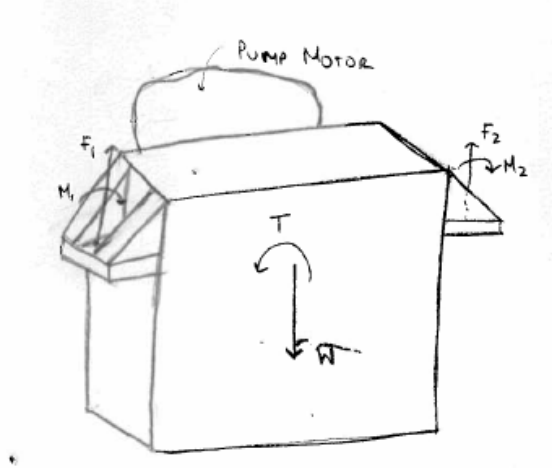
The opposite of the aforementioned tab concept, the internal tab concept allows the sub-frame to rest recessed inside of the gear reduction casing. This could require bolting the gearbox and sub-frame together internally or using some other connection method.



APPENDIX F: ENGINEERING PARAMETER ANALYSIS

F.1: TAB ANALYSIS

FREE BODY DIAGRAM OF REDUCTION CASE



$x_1 = 5.05$ in

$x_2 = 7.72$ in

$x_3 = x_1 + x_2 = 12.77$ in

$T = 206$ ft-lb

W = Weight of Case + Weight of pump + Weight of shafts + Weight of bearings + Weight of Chain & sprockets.

$\therefore W = 115$ lb

Force Balance on FBD of entire case,

$$F_1 + F_2 = W$$

$$M_1 = \frac{x_1}{x_1 + x_2}(T)$$

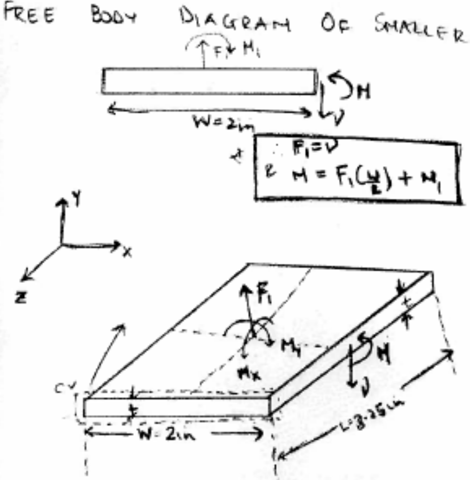
$$M_2 = \frac{x_2}{x_1 + x_2}(T)$$

Using Torque & Moment balance to solve for reaction forces F_1 & F_2 on the reduction cases,

$$F_1 = \frac{T + W(x_2)}{x_3} \quad \& \quad F_2 = \frac{W(x_1) - T}{x_3}$$

Once we find reaction forces & Moments we can use force analysis on the tabs separately

FREE BODY DIAGRAM OF SMALLER TAB.



M_x = Bending Moment due to suspended pump/motor

$\therefore \sigma_x = \frac{-M_x y}{I}$

$\sigma_y = \frac{F_1}{A_s}$

$\sigma_z = \frac{-M_1 y}{I} + \frac{M y}{I}$

$\tau_{shear} = \frac{V A^* y^*}{I t} = \frac{6V}{t^2}$

$A_s = W \times L$
$= 2 \times 3.25$
$= 6.5 in^2$

$A^* = t \times 3.25$

Principal Stresses :-

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_{x-y}}{2}\right)^2 + (\tau_{shear})^2}$$

$\therefore \sigma_1 = 34 \text{ MPa} \quad \sigma_2 = -9.2 \text{ MPa}$

$\sigma_3 = 13.36 \text{ MPa}$

Von Mises Stresses :- Normal Condition

$$\sigma_M = \sqrt{\frac{1}{2}\left((\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_1-\sigma_3)^2\right)}$$

$= 45.76 \text{ MPa}$

Yield strength of A36 Steel = 250 MPa

$\therefore$ Safety Factor $= \frac{\sigma_y}{\sigma_M}$

$= \frac{250}{45.76}$

$= \underline{\underline{5.4}}$

* Extreme Condition

$\sigma_M = 58.73$

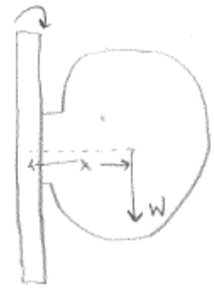
$\therefore$ Safety factor $= \frac{\sigma_y}{\sigma_M}$

$= \frac{250}{58.73}$

$= \underline{\underline{4.26}}$

F.2: FORCE ANALYSIS OF PUMP/MOTOR MOUNT PLATE

W = weight of hydraulic Pump/motor when fully filled.



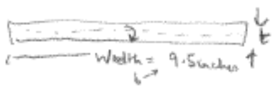
S.V.

Bending Moment Acting on plate = $W \times x$

$$= 60 \times \frac{4.22}{12}$$

$$= 21.1 \text{ lb-ft}$$

Thus to determine the bending stress acting on the plate we use,



TV

$$\sigma_b = -\frac{My}{I} \qquad I = \frac{bt^3}{12} \rightarrow (1)$$

$$y = \frac{t}{2}$$

We then implement a safety factor of two, to equation below to determine the corresponding stress to it.

$$\therefore SF = \frac{\sigma_y}{\sigma_b} \qquad 2 = \frac{250}{\sigma_B}$$

$$\sigma_B = 125 \text{ MPa}$$

$\therefore$ We can know plug this back in (1) to solve for the thickness of the plate required to support the weight of the pump/motor.

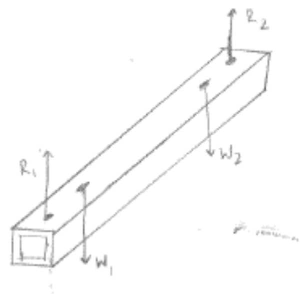
$$\therefore t = \sqrt{\frac{12(M)}{\sigma_{all} \times b}}$$

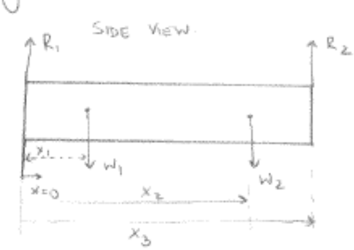
$$\therefore t = 0.11 \text{ in}$$

Thus the required thickness to support the bending stress is 0.11 in. We are using a plate that is 1 inch thick, which will meet our goals very easily.

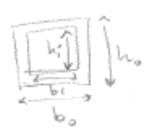
SUB Frame Analysis

- R_1 & R_2 are reaction forces at the bolt locations of the subframe.
- W_1 = Represents the weight of Reduction case + its internal components + weight of the Pump/motor = 115 lbs
- W_2 = Represents weight of DC motor = lbs
- W_1 & W_2 are divided by 2 for analysis sake as these weights are distributed between two bars in the sub-frame.





$x_1 = 3$ in
$x_2 = 14.52$ in
$x_3 = 17$ in



$b_i = 1.75$ in
$h_i = 1.25$ in
$b_o = 2$ in
$h_o = 1.5$ in

Force Balance on FBD gives us,

$R_1 + R_2 = W_1 + W_2$

Taking moments about $x=0$,

$\Sigma M_0 = 0 = (-W(x_1) - (W_2(x_2)) + R_2(x_3))$

$R_2 = \dfrac{W_1 x_1 + W_2 x_2}{x_3}$

Moments about $x = x_3$

$\Sigma M_{x_3} = 0 = -R_1(x_3) + W_2(x_3 - x_2) + W_1(x_3 - x_1)$

$\therefore R_1 = \dfrac{W_2(x_3 - x_2) + W_1(x_3 - x_1)}{x_3}$

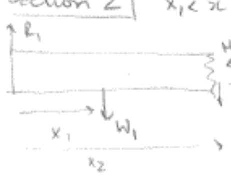
Conducting a sectional Bending Moment stress analysis on the bar.

Section 1

$M_1 = R_1(x_1)$

$\sigma_{x_1} = \dfrac{-My}{I}$, $y = h/2$, $I = \frac{1}{12}bh^3$

Section 2 $\quad x_1 < x < x_2$



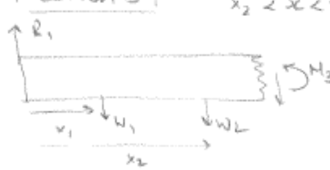
$M_2 = R_1(x) - W_1(x - x_1)$

Since the moment has a linear trend it is max at end points (x_1 & x_2)

$\therefore \sigma_{x_2} = \dfrac{(R_1(x_2) - W_1(x_2 - x_1))(y)}{I}$ $\quad y = h/2$

$I = \frac{1}{12}bh_o^3 - \frac{1}{12}bh_i^3$

Section 3 $\quad x_2 < x < x_3$



$M_3 = R_1(x) - (W_1(x - x_1) - W_2(x - x_2))$

Same as section 2, max moment occurs at end points of section

$\sigma_{x_3} = \dfrac{(R_1(x_3) - (W_1(x_3 - x_1) - W_2(x_3 - x_2))y}{I}$

$y = h/2$, $I = \frac{1}{12}bh_o^3 - \frac{1}{12}bh_i^3$

Thus max stress occurs at point x_2.

$\therefore$ Safety factor of sub-frame under normal condition is,

$$\text{Safety Factor} = \frac{\sigma_y}{\sigma_{x_2}}$$

Under extreme conditions, all stresses increase by a factor of 3. (Refer to Engineering Analysis section for detailed expression)

Stresses acting on sub frame

$\sigma_{x_1} = 4.48$ MPa

$\sigma_{x_2} = 4.72$ MPa

$\sigma_{x_3} = 0$ MPa

$$\therefore \text{safety factor} = \frac{\sigma_y}{\sigma_{x_2}} = \frac{150}{4.72} = \boxed{32}$$

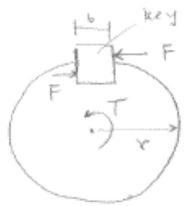
Normal Condition

$$\therefore \text{Safety Factor} = \frac{\sigma_y}{\sigma_2} = \frac{250}{14.15} = \boxed{17.7}$$

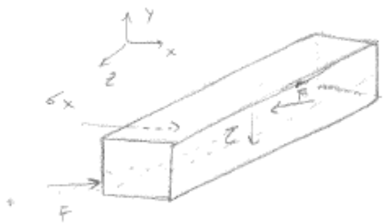
Extreme Conditions

F.4: PUMP/MOTOR SHAFT KEY

Key in Shaft for Pump/Motor



Shear Force acting on the Keys can be found using $F = \frac{T}{r}$, where T is the load torque and r is the radius of the shaft.



Using the FBD of the key we can then find the shear stress and bearing stress on the keys

Shear Stress $\tau = \frac{F}{tl}$

Bearing Stress $\sigma_x = \frac{F \times 2}{tl}$

$$\sigma_y = 0$$
$$\sigma_z = 0$$

∴ Then we can use Mohr's circle to find principal stresses,

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2}$$

$$\therefore \sigma_1 = 96.32$$
$$\sigma_2 = -1.07 \times 10^{7}$$
$$\sigma_3 = 0 \text{ MPa}$$

We then use Von Mises stress analysis to find equivalent stress on the key

$$\sigma_M = \sqrt{\tfrac{1}{2}\left((\sigma_1-\sigma_2)^2 + (\sigma_2-\sigma_3)^2 + (\sigma_1-\sigma_3)^2\right)}$$

$$= 106 \text{ MPa}$$

σ_Y, is the yield strength of the steel the key is made with.

$$\therefore \text{Safety factor} = \frac{\sigma_Y}{\sigma_M} = \frac{495}{106}$$

$$= \underline{\underline{4.68}}$$

Thus the key will not fail.

F.5 BEARING ANALYSIS CALCULATIONS

$$Torque = (206 ft - lbs)(12 in / ft) = 2472 in - lbs$$

$$Hp = \frac{2\pi nT}{395{,}877} = \frac{2\pi(2000 rpm)(2472 in - lbs)}{395{,}877} = 78.46 Hp$$

$$Mean\ Bearing\ Diameter = D_m = \frac{Pitch}{\sin\left(\frac{180}{N_s}\right)} = 5.98 in.$$

$$F_b = \frac{(1.26 * 10^7) Hf_B}{D_m n} = \frac{(1.26 * 10^5)(78.46 Hp)(1.00)}{(5.98 in.)(2000 rpm)} = 826.58 lbf$$

$$L_{10} = \left(\frac{C}{P_r}\right)^e (1 * 10^6) revolutions, where\ e = 3\ for\ roller\ bearings$$

$$L_{10} = \left(\frac{3{,}761 lbs}{826.58 lbs}\right)^3 (1 * 10^6) = 94.2 * 10^6 revolutions$$

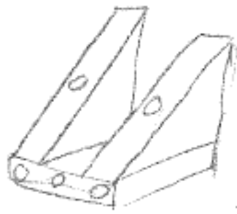
$$94.2 * 10^6 \left(\frac{1}{2}\right)\left(\frac{1}{3.3}\right)\left(\pi * \frac{20"\ wheel}{12 in / ft}\right)\left(\frac{1\ mile}{5280\ ft}\right) = 14{,}100\ miles$$

$$L_{10} = \left(\frac{4690}{826.58}\right)^{10/3} (1 * 10^6) = 326 * 10^6 revolutions$$

$$326 * 10^6 \left(\frac{1}{2}\right)\left(\frac{1}{3.3}\right)\left(\pi * \frac{20"\ Wheel}{12 in / ft}\right)\left(\frac{1 mile}{5280 ft}\right) = 49{,}000 miles$$

F.6 TAB AND GUSSET ANALYSIS ON NEW MOUNTS

F TAB & GUSSET ANALYSIS ON THE NEW MOUNTS



1) The tabs and gussets are now mounted to the reduction case using 5 bolts instead of welding.

2) The tabs support the weight of the reduction case, the hydraulic pump/motor and the components within the frame.

Based on Analysis conducted on the strength of the tabs previously we determined the shear force acting on the tabs to be.

$V = 80 \text{ lbs}$

* This shear force is distributed over the area of 5 bolt locations of size 1/4" each.

∴ the shear stress distributed across these areas is,

$$\tau_{shear} = \frac{V}{5 \times Area} = \frac{80}{5 \times \pi \times (0.125)^2}$$

$$= 325.95 \text{ lb/in}^2$$

Bolts are typically rated for Tensile Strength, But they can also withstand shears of values upto 60% of their tensile strength.

Tensile Strength = 180,000 psi

∴ 60% = 108,000 psi

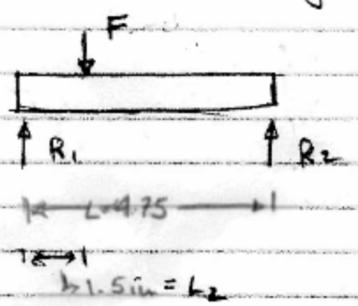
∴ The bolts should be strong enough to withstand the applied shear force.

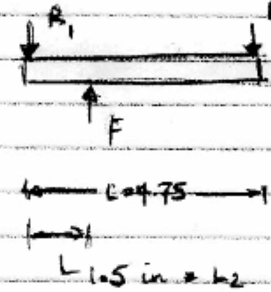
F.7 DEFLECTION IN PUMP/MOTOR AND DC MOTOR SHAFTS

Pump/Motor Shaft | DC MOTOR SHAFT

F = Force due to chains

R_1, R_2 - Reaction forces

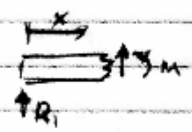




$$\Sigma M_{R_1} = 0 = -1.5 \text{ in } (826.581) + 4.75 \text{ in } (R_2)$$

$R_2 = 261.025$ | $R_2 = 261.025$

$R_1 = 565.555$ | $R_1 = 565.555$



$0 < x \leq L_2$

$M = R_1 x = 565.555(x) = M(x)$ | $M(x) = 565.555(x)$

$M(L_2) = 848.333$ lb in

$EI \frac{\partial^2 v}{\partial x^2} = M(x)$ | $E = 29000$ ksi

$(29000 \times 10^3)(.249) \frac{\partial^2 v}{\partial x^2} = 565.555x$ | $I = \frac{\pi}{4} r^4$

$\int \frac{\partial^2 v}{\partial x^2} = \int 7.832 \times 10^{-5} x$

$\frac{\partial v}{\partial x} = \frac{7.832 \times 10^{-5} x^2}{2} + C_1$ | $= \frac{\pi}{4}(.75 \text{ in})^4 = .249 \text{ in}^4$

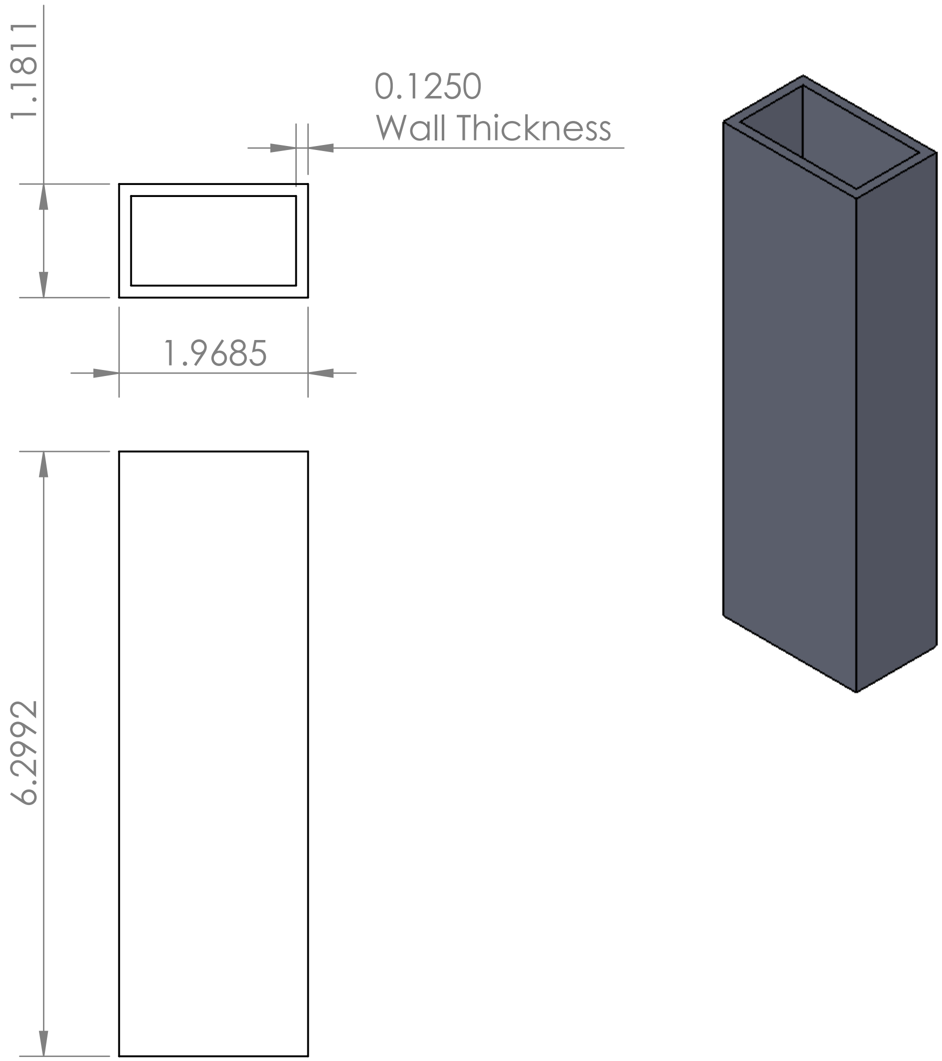
$V(x) = \frac{7.832 \times 10^{-5} x^3}{6} + C_1 x + C_2$ | BC's, $x=0$, $C_2 = 0$; $\frac{\partial v}{\partial x} = 0$, $C_1 = 0$

$V(x) = \frac{7.832 \times 10^{-5} x^3}{6} + \cancel{C_2 x}$

$V(1.5) = \frac{7.832 \times 10^{-5}(1.5)^3}{6} = .000044055$

$= 4.406 \times 10^{-5} \text{ in} \times 2 = \boxed{8.811 \times 10^{-5} \text{ in}}$

APPENDIX G: ENGINEERING DRAWINGS



		UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±
		INTERPRET GEOMETRIC TOLERANCING PER:
		MATERIAL A36 STEEL
NEXT ASSY	USED ON	FINISH
APPLICATION		DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

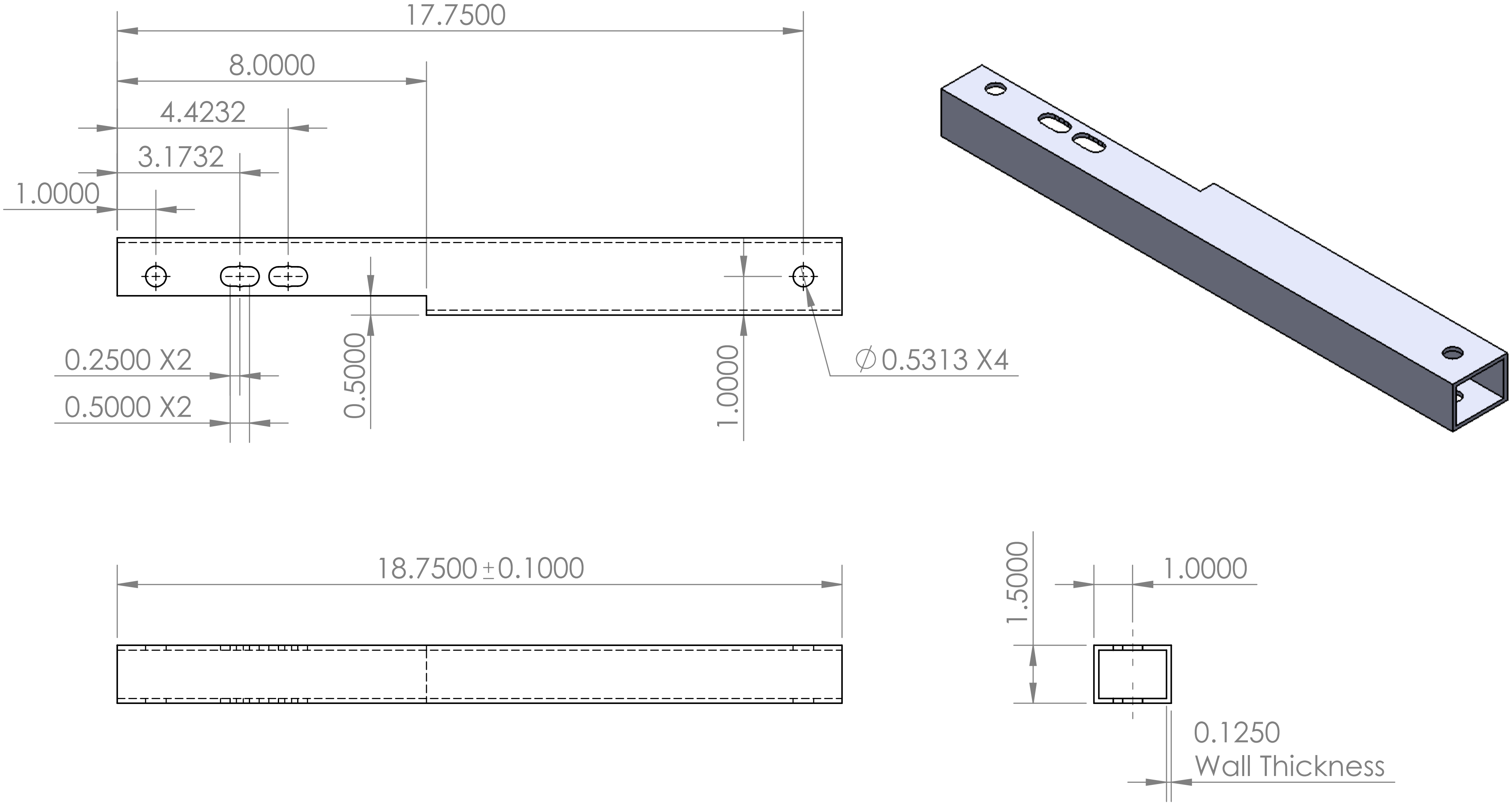
TITLE:

Subframe Mount Beam

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.01 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/12
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL A36 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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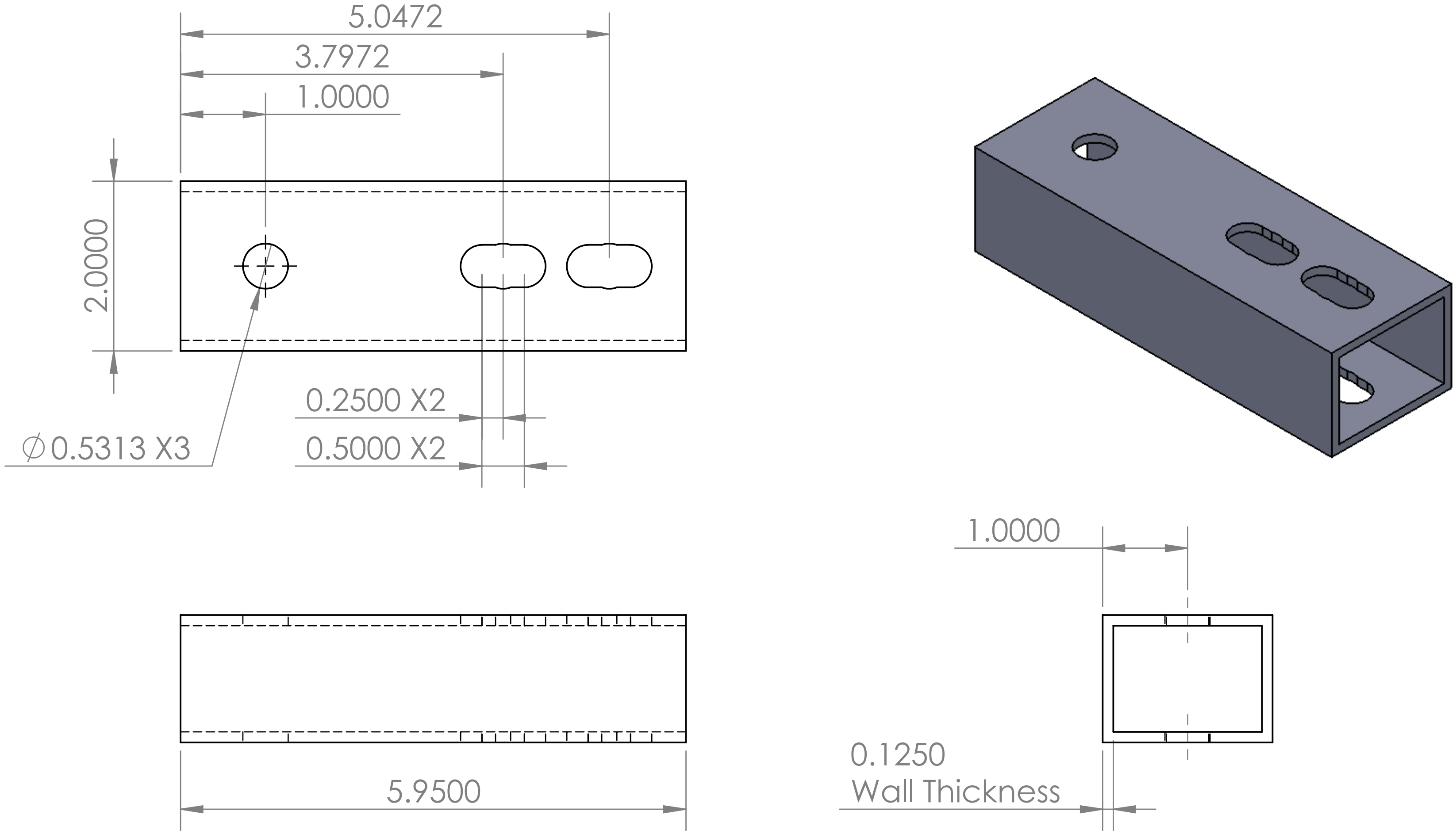
TITLE:

Subframe Back

SIZE	DWG. NO.	REV
A		1

SCALE: 1:4 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



			NAME	DATE
		DRAWN	MW	12/12
		CHECKED		
		ENG APPR.		
		MFG APPR.		
		Q.A.		
NEXT ASSY	USED ON	COMMENTS:		
APPLICATION				

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

U of M Xebra Team Fall 2010

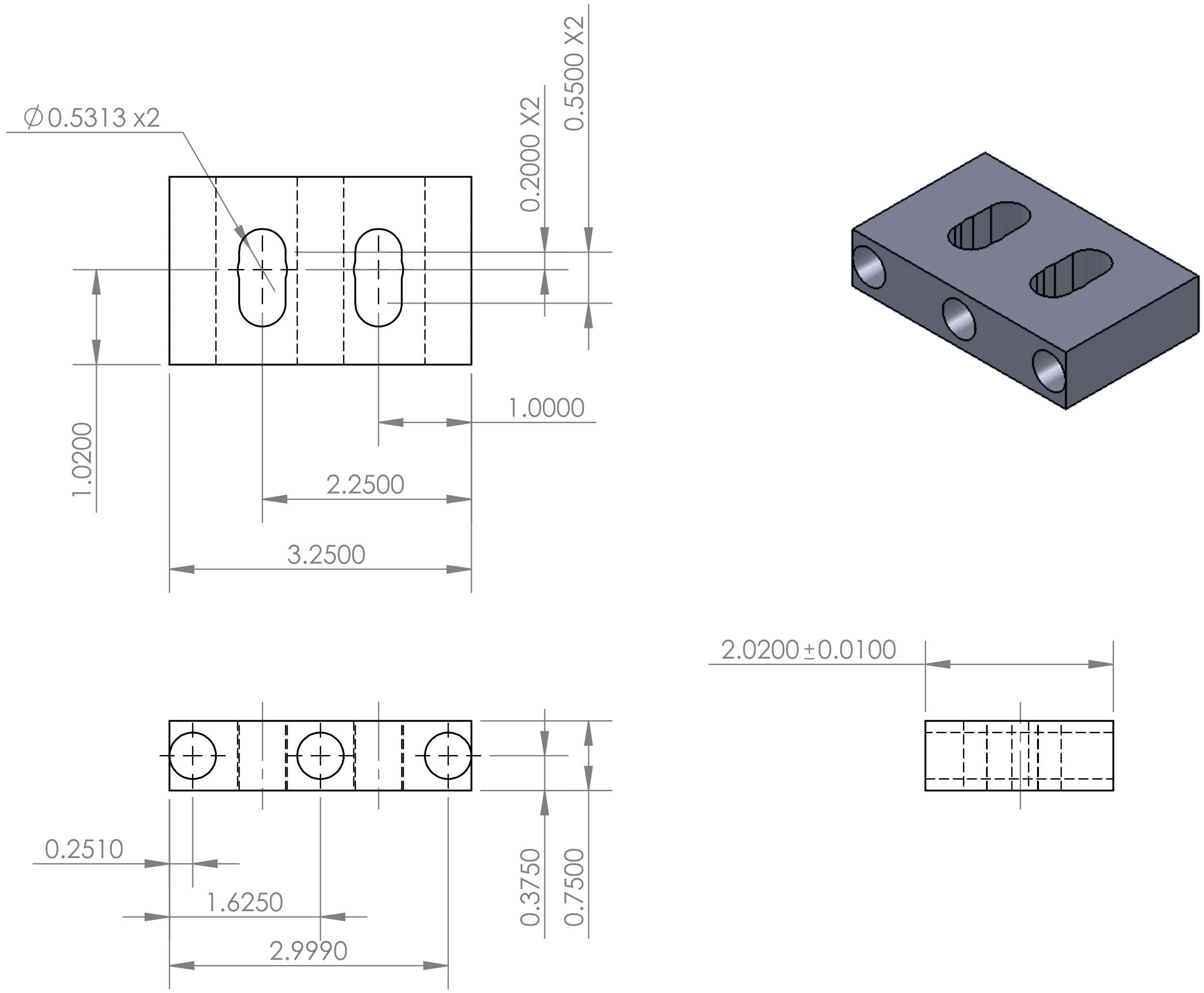
TITLE:

Sub-Frame: Front

SIZE A | DWG. NO. | REV 1

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



PROPRIETARY AND CONFIDENTIAL

NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

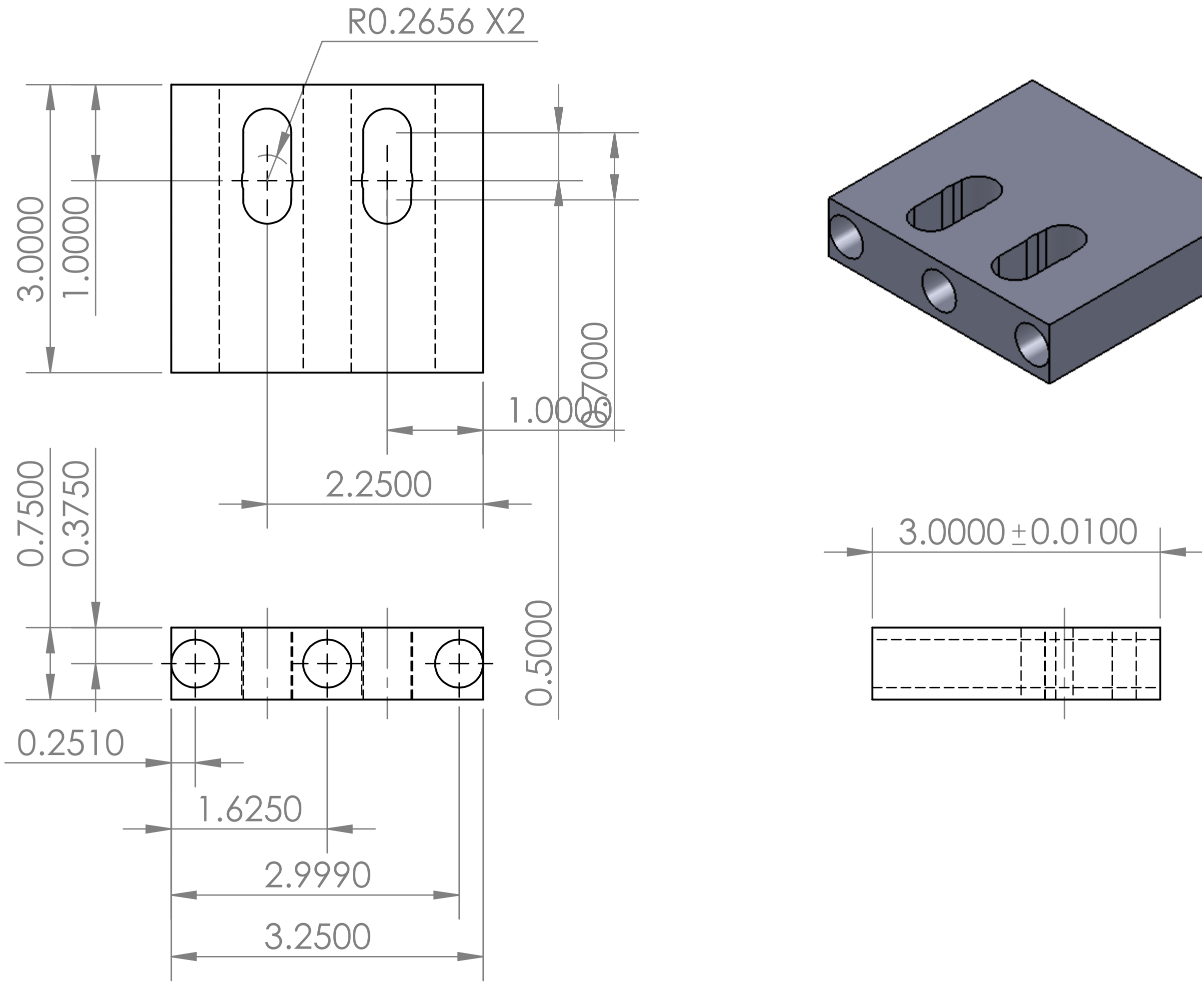
TITLE:

Reduction Case: Tab, Back

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



5 4 3 2 1

PROPRIETARY AND CONFIDENTIAL

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

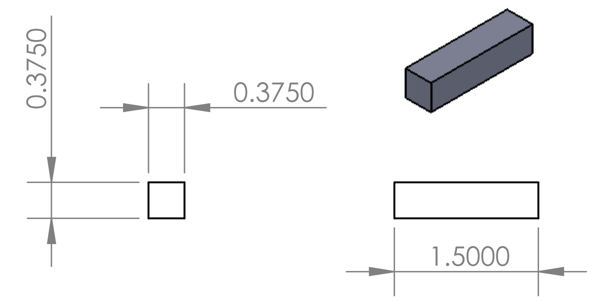
U of M Xebra Team Fall 2010

TITLE:

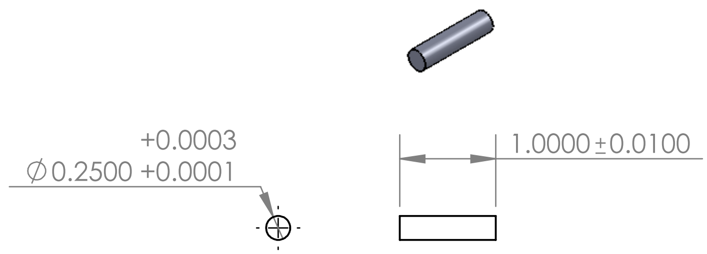
Reduction Case: Tab, Front

SIZE	DWG. NO.	REV
A		1

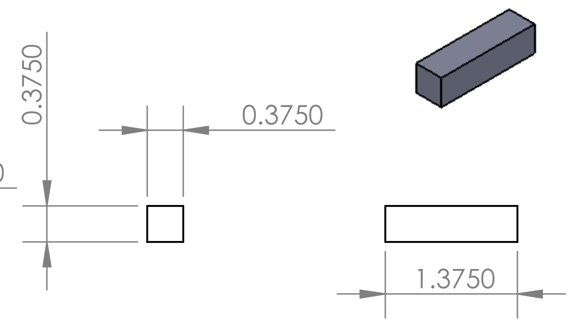
SCALE: 1:2 WEIGHT: SHEET 1 OF 1



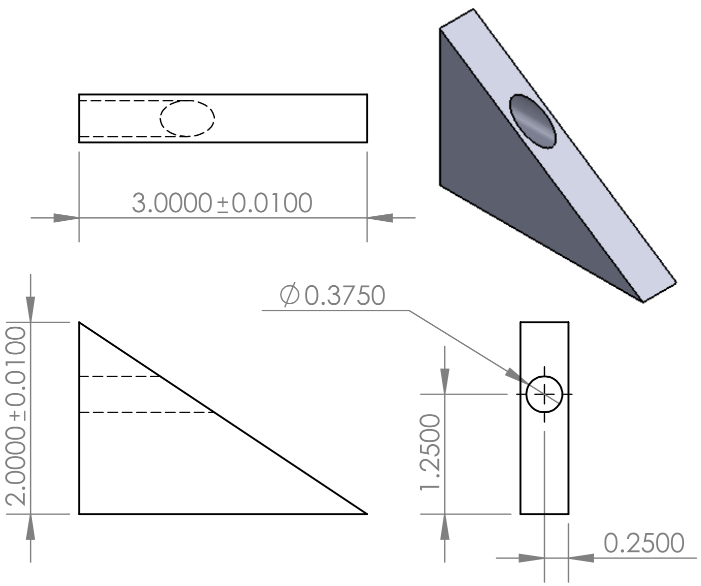
PM Shaft: Key Stock
Material: High-Carbon Plain Steel



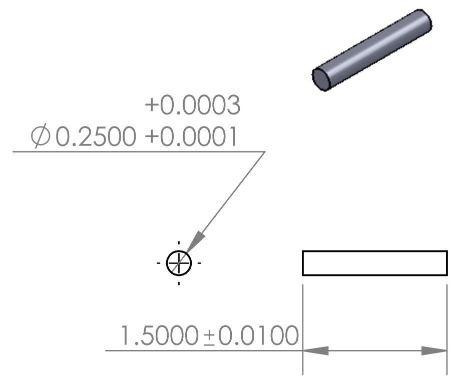
Dowel Pins
Material: Hardened Steel



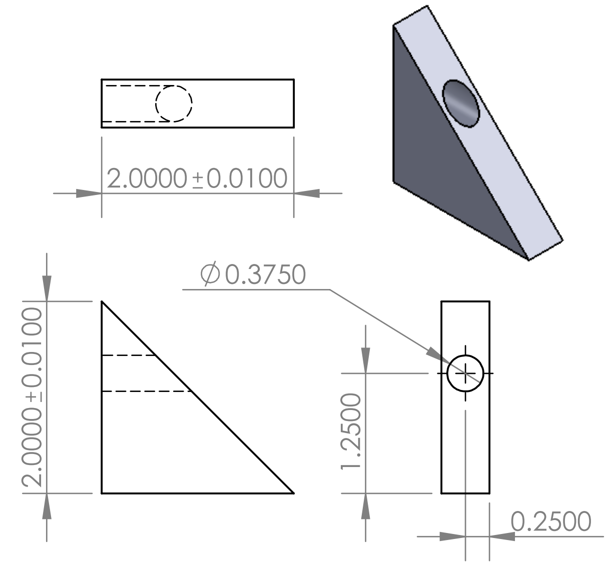
DC Shaft: Key Stock
Material: High-Carbon Plain Steel



Reduction Case: Gusset, Front
Material: A36 Steel



Dowel Pins (Long)
Material: 416 Stainless Steel



Reduction Case: Gusset, Front
Material A36 Steel

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
SPECIFIED PER PART

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

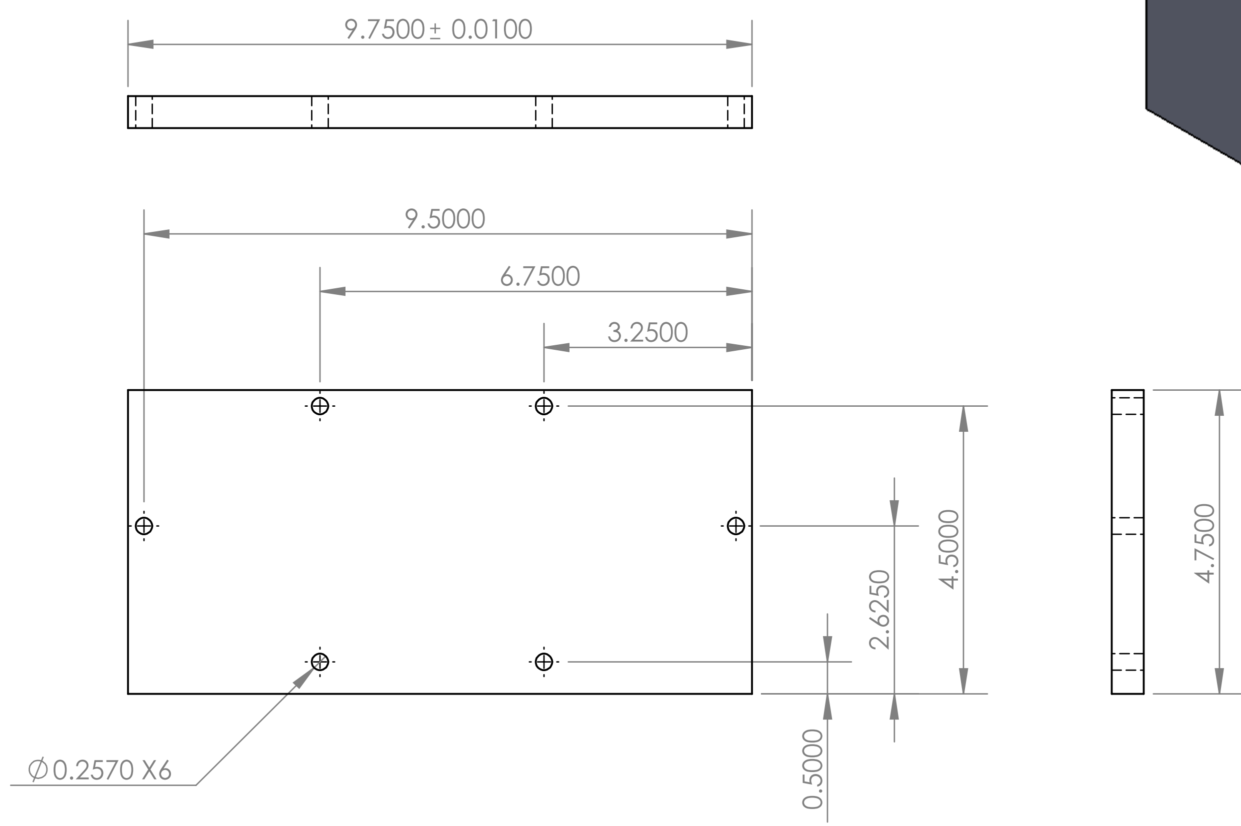
TITLE:

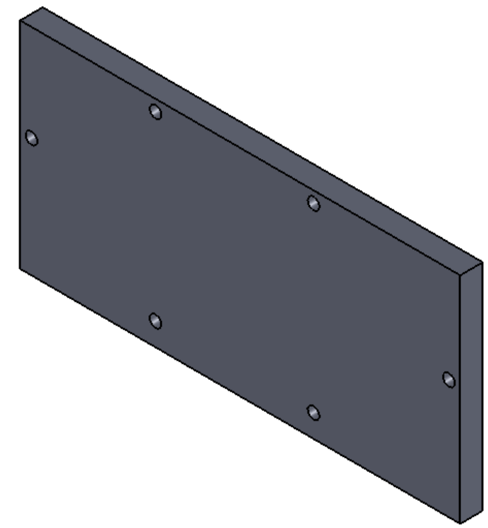
Misc Parts: Gussets, Keys, Dowel Pins

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

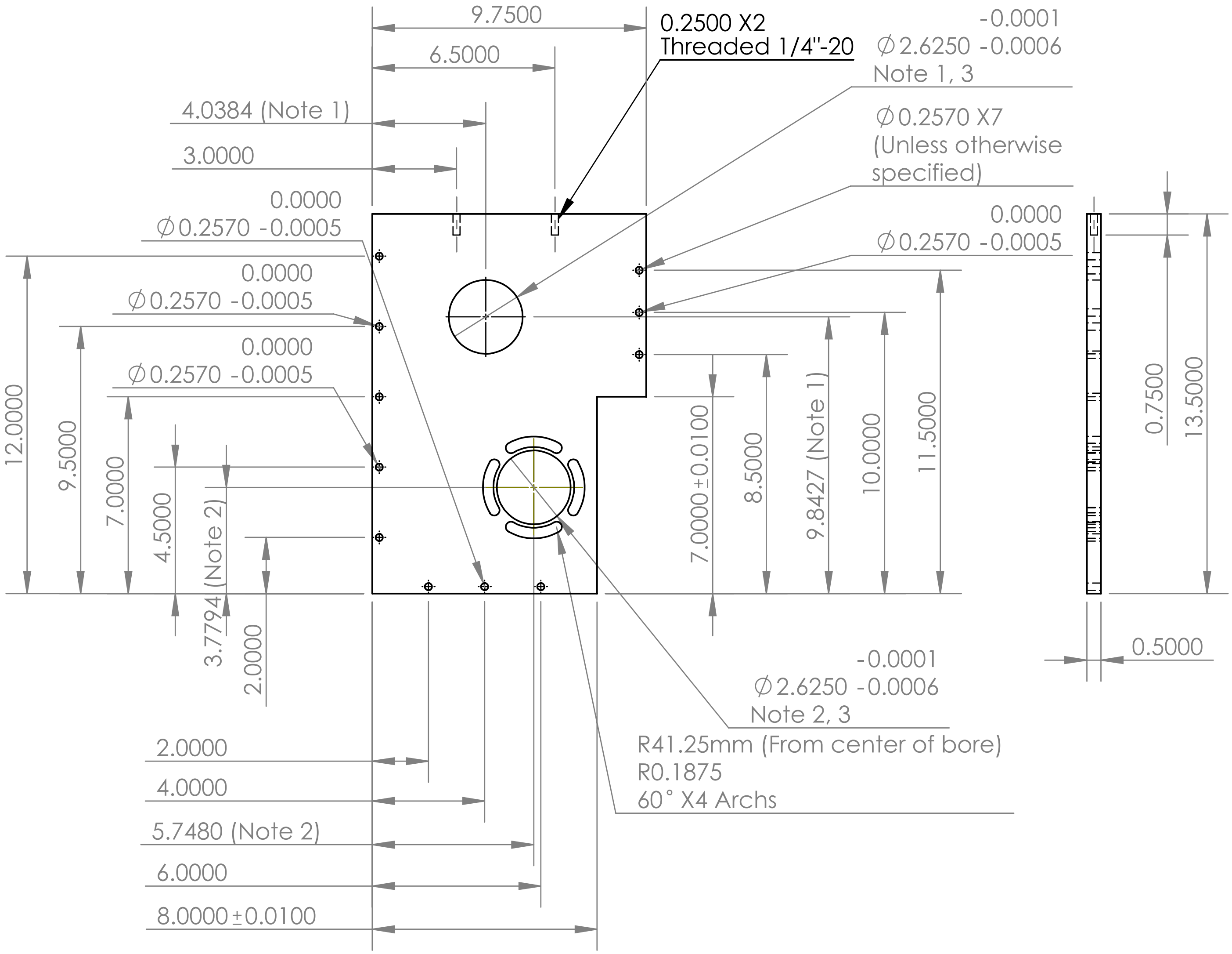


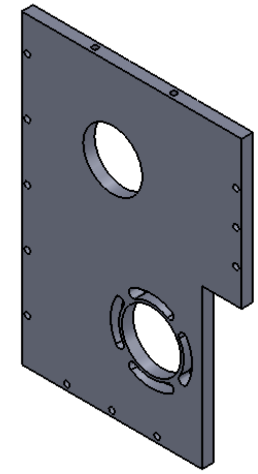


All 1/4" Holes are 1/4" from nearest edge unless otherwise specified

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/12	TITLE:
			CHECKED			Reduction Case: Top Access Door
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A | DWG. NO. | REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:3 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1





All 1/4" Holes are 1/4" from nearest edge

Note 1: Concentric ±**0.001in** to bores on PM Plate (opposite side of case)

Note 2: Concentric ±**0.001in** to bore on Access Door (opposite side of case)

Note 3: Bore holes for ball bearing

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS: Threads not shown.

U of M Xebra Team Fall 2010

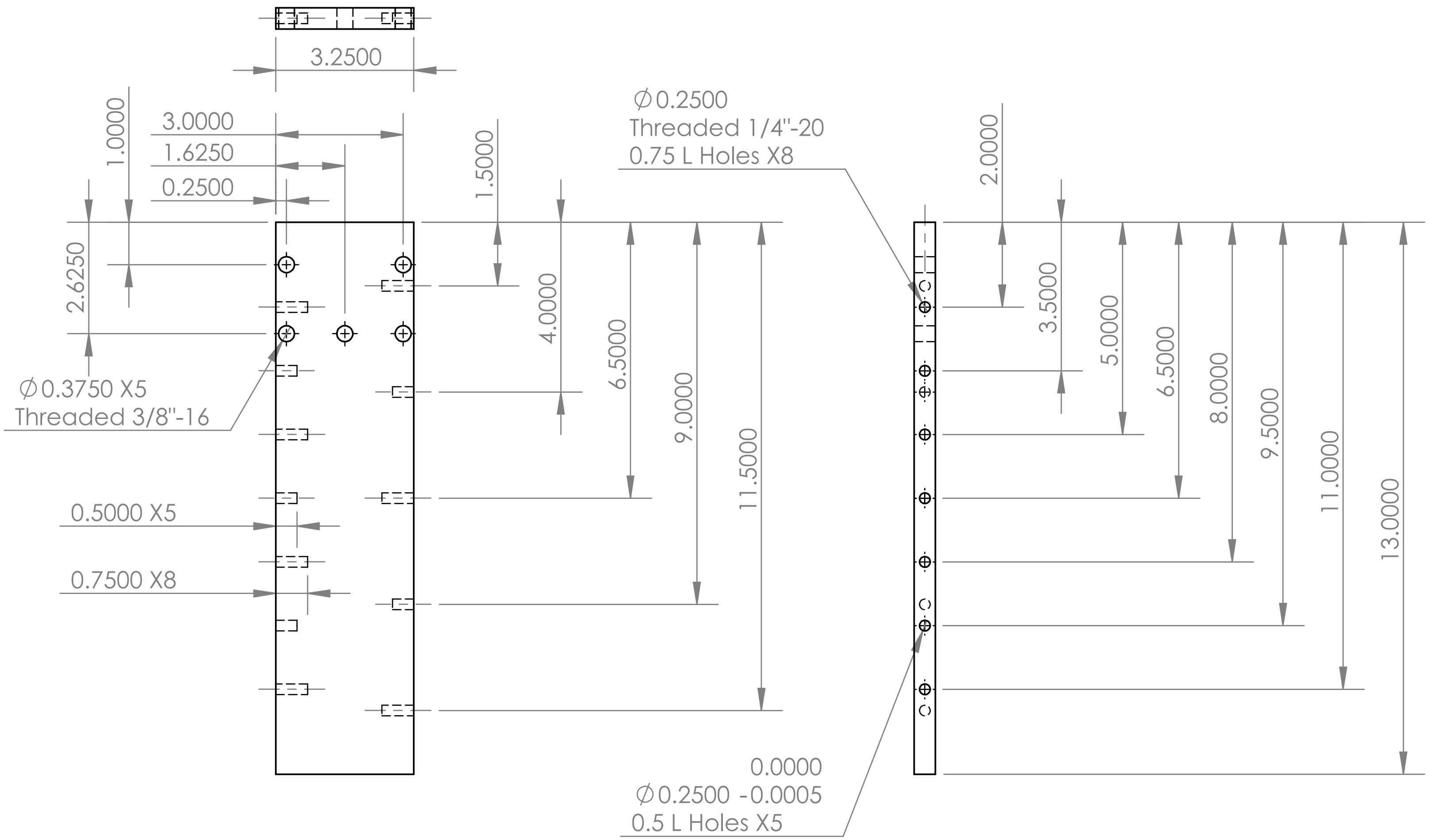
TITLE:

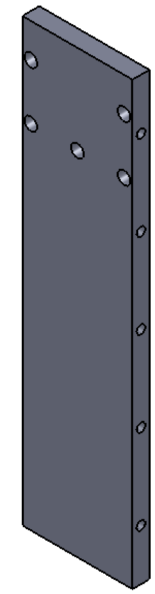
Reduction Case: DC

SIZE	DWG. NO.	REV
A		1

SCALE: 1:6 WEIGHT: SHEET 1 OF 1

5 4 3 2 1





All 1/4" Holes are 1/4" from nearest edge

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS: Threads not shown.

U of M Xebra Team Fall 2010

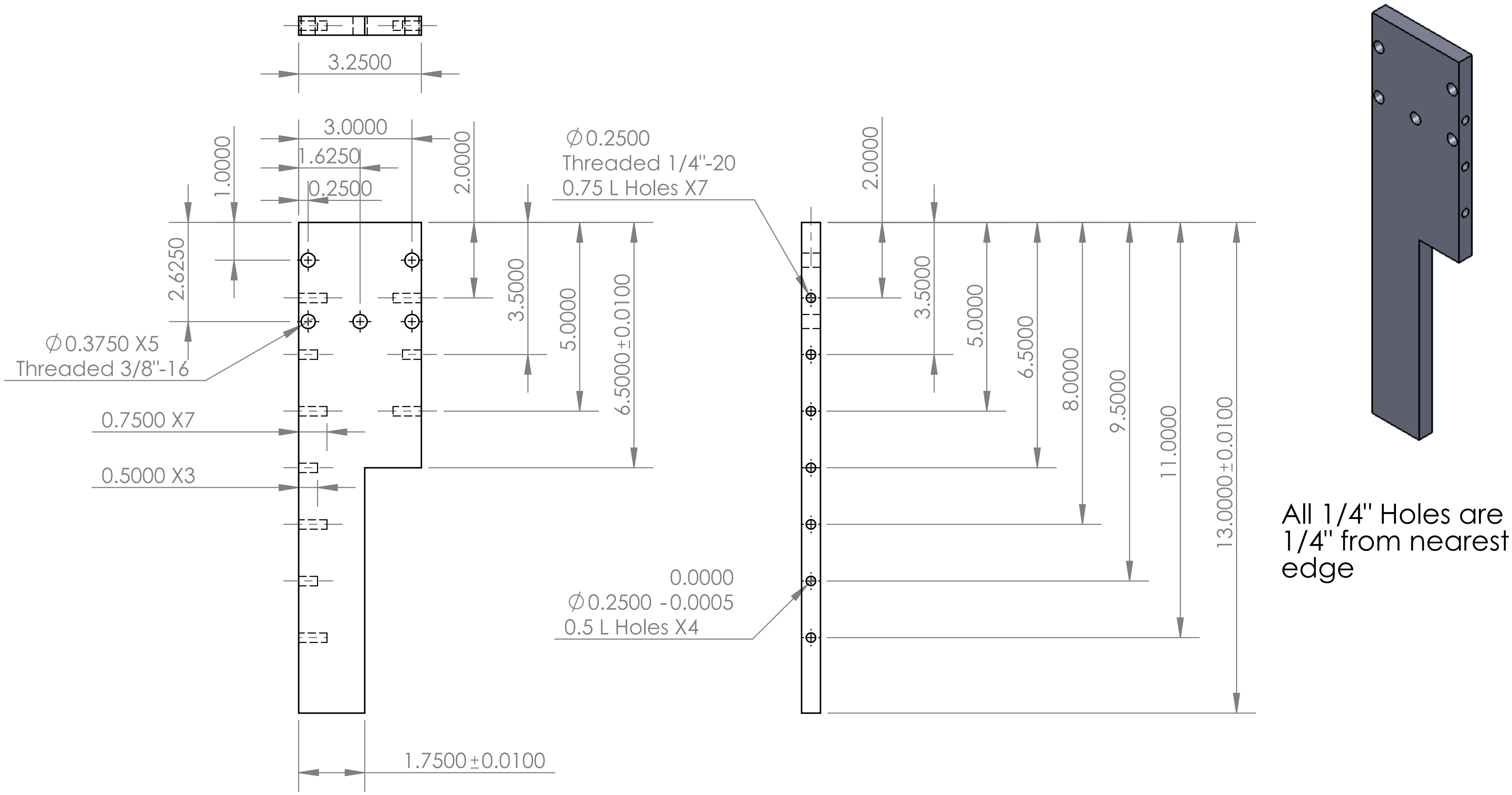
TITLE:

Reduction Case: Side Back

SIZE A | DWG. NO. | REV 1

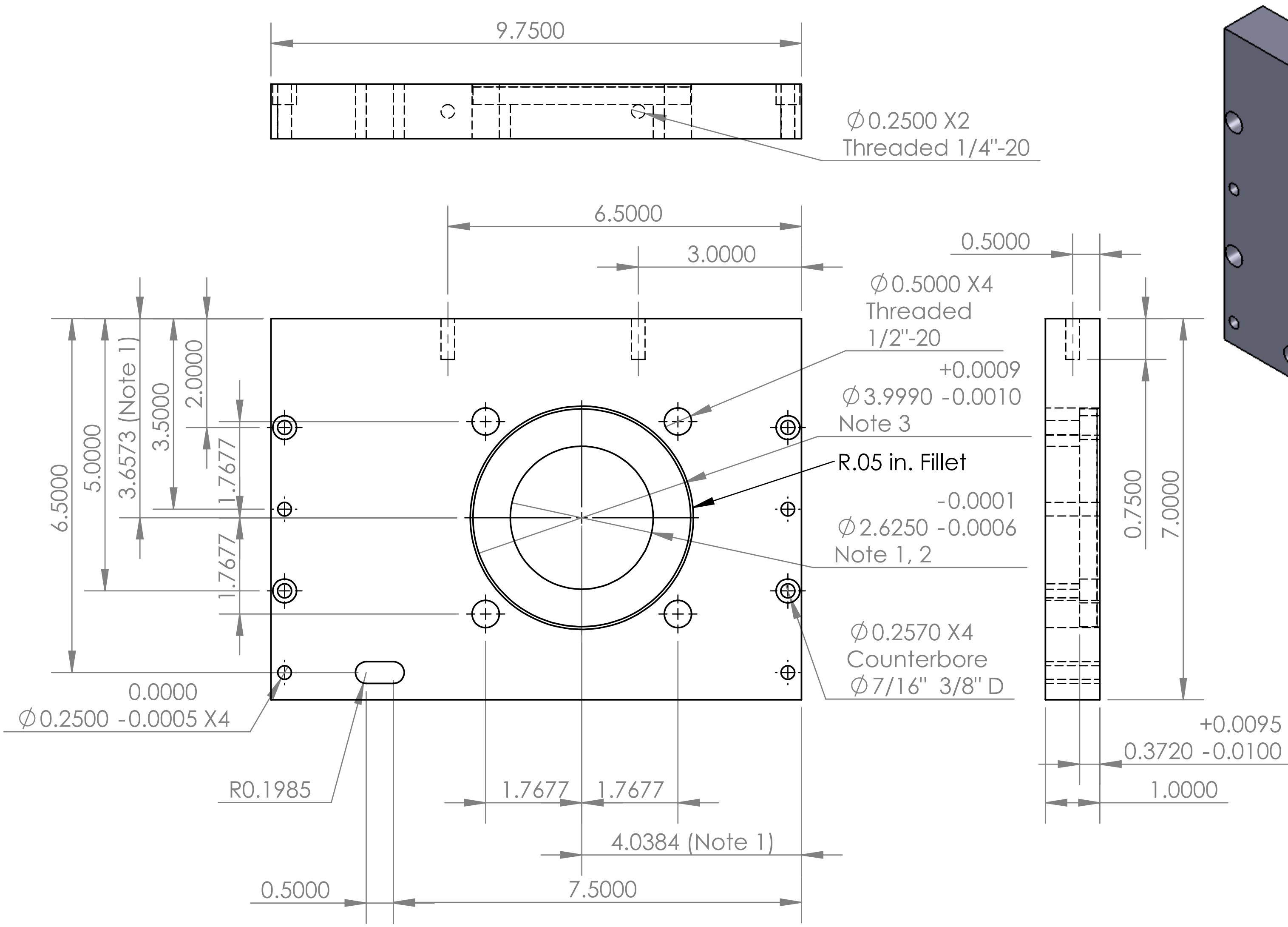
SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

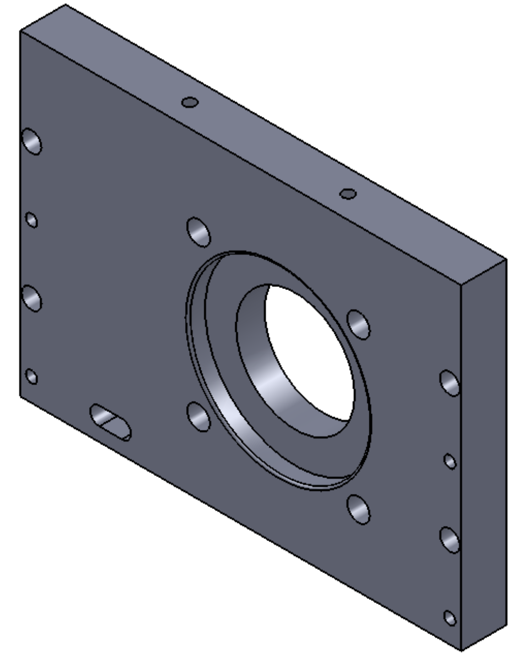
5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/12	TITLE:
			CHECKED			Reduction Case: Side Front
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS: Threads not shown.			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1





All 1/4" Holes are 1/4" from nearest edge

Note 1: Concentric ±**0.001in** to bore on DC Plate (opposite side of case)

Note 2: Bore hole for ball bearing

Note 3: Bore hole for pilot of pump/motor

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS: Threads not shown.

U of M Xebra Team Fall 2010

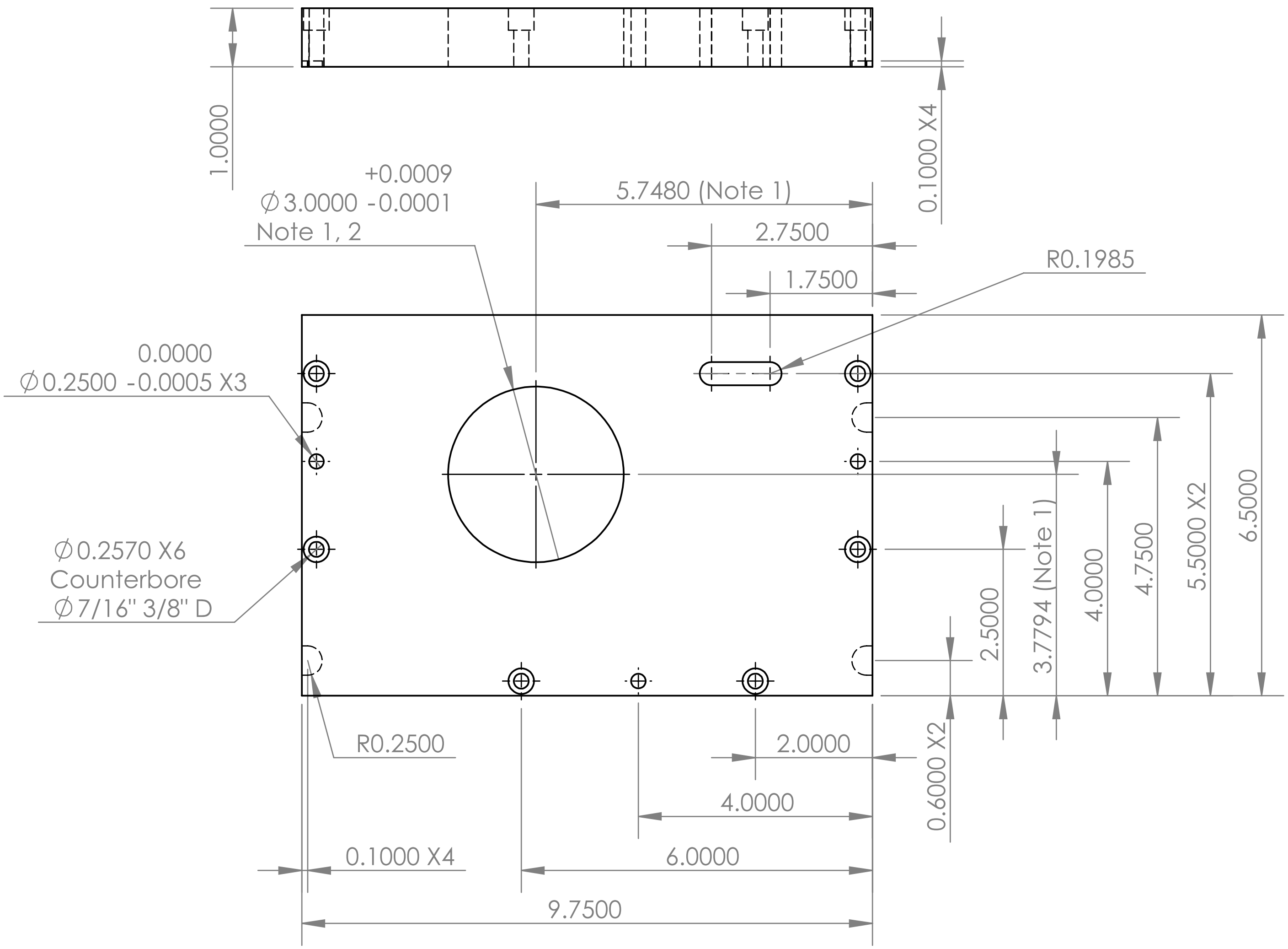
TITLE:

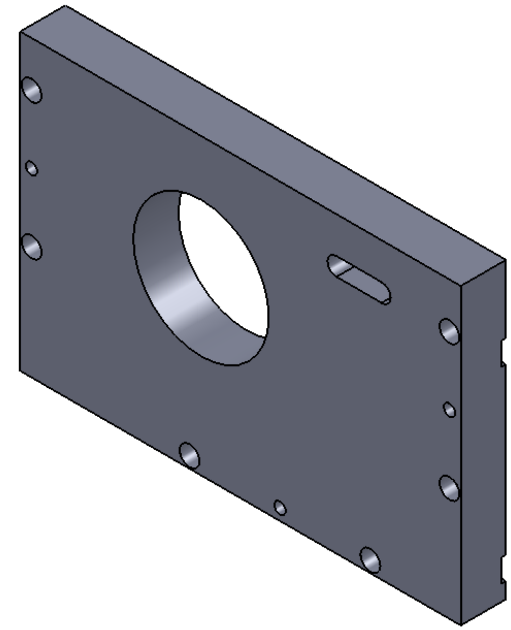
Reduction Case: PM

SIZE	DWG. NO.	REV
A		1

SCALE:1:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1





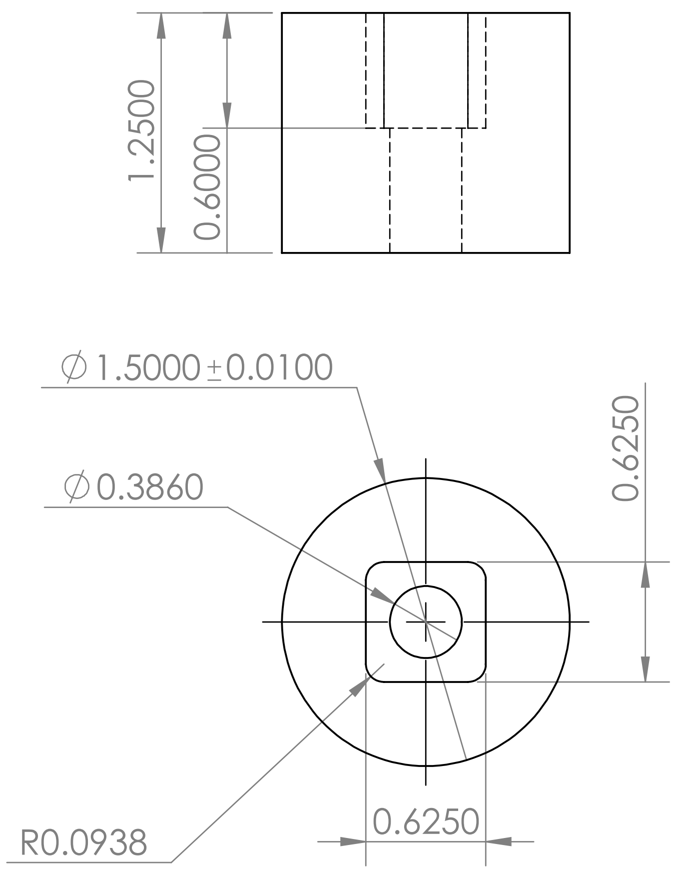
All 1/4" Holes are 1/4" from nearest edge

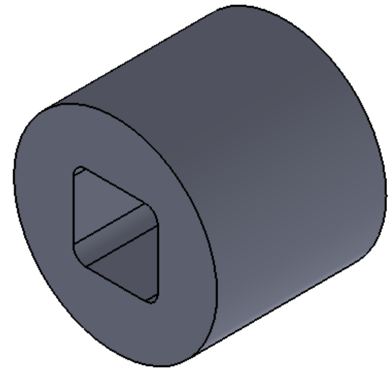
Note 1: Concentric ±**0.001in** to bore on DC Plate (opposite side of case)

Note 2: Bore hole for tapered bearing

		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	12/12	TITLE:
			CHECKED			Reduction Case: Access Door
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL A36 STEEL	COMMENTS:			SIZE A · DWG. NO. · REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:3 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1





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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
UHMW Tension Plastic

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

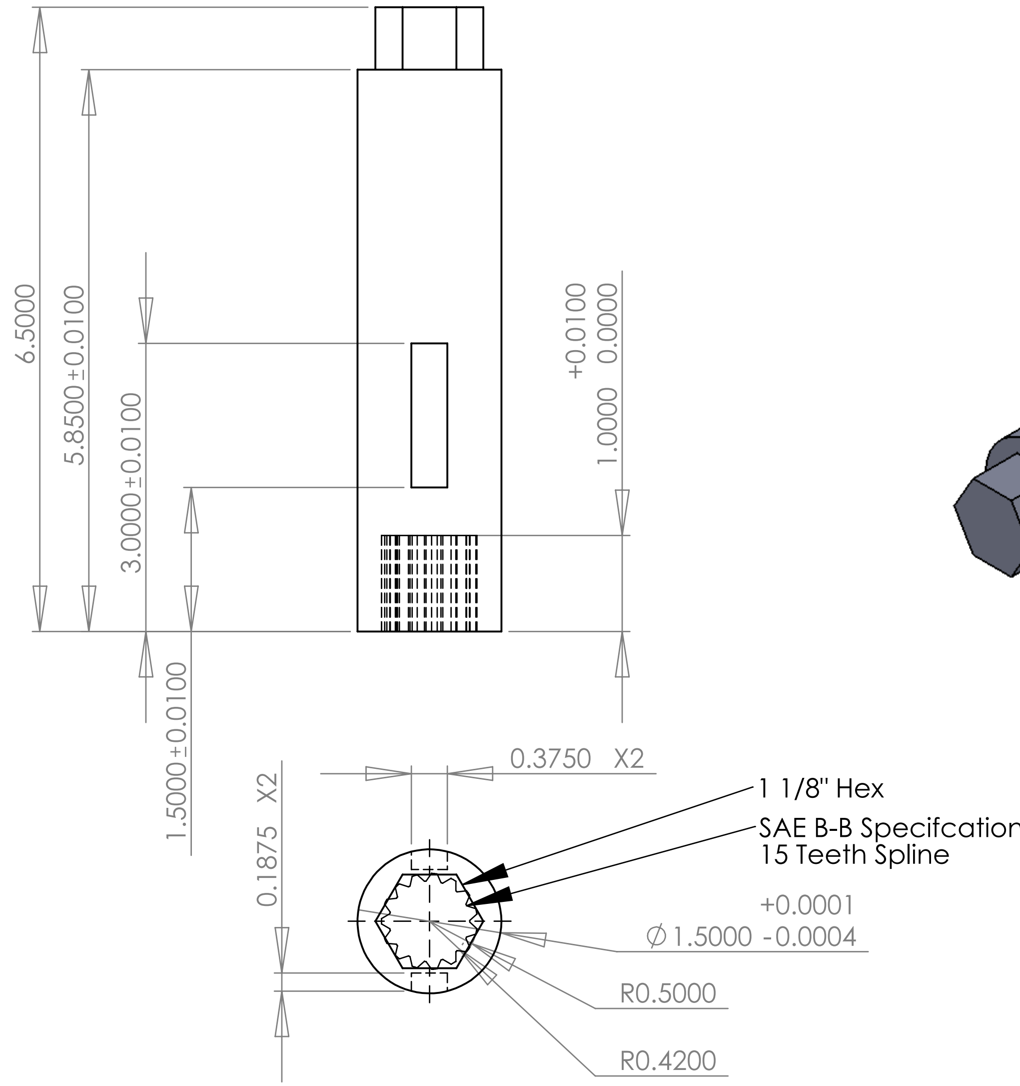
TITLE:

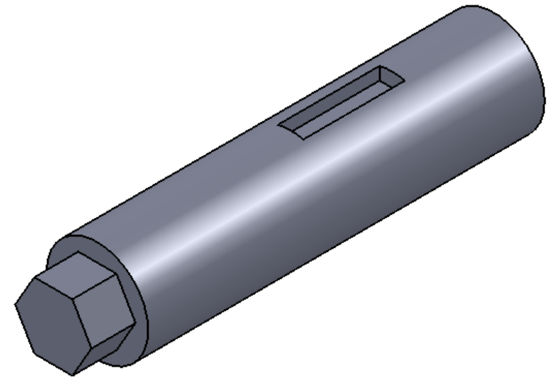
Reduction Case: Tensioner

SIZE	DWG. NO.	REV
A		1

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1





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NEXT ASSY	USED ON
APPLICATION	

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DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

TITLE:

Output Shaft for Pump Motor (Female)

SIZE	DWG. NO.	REV
A		

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

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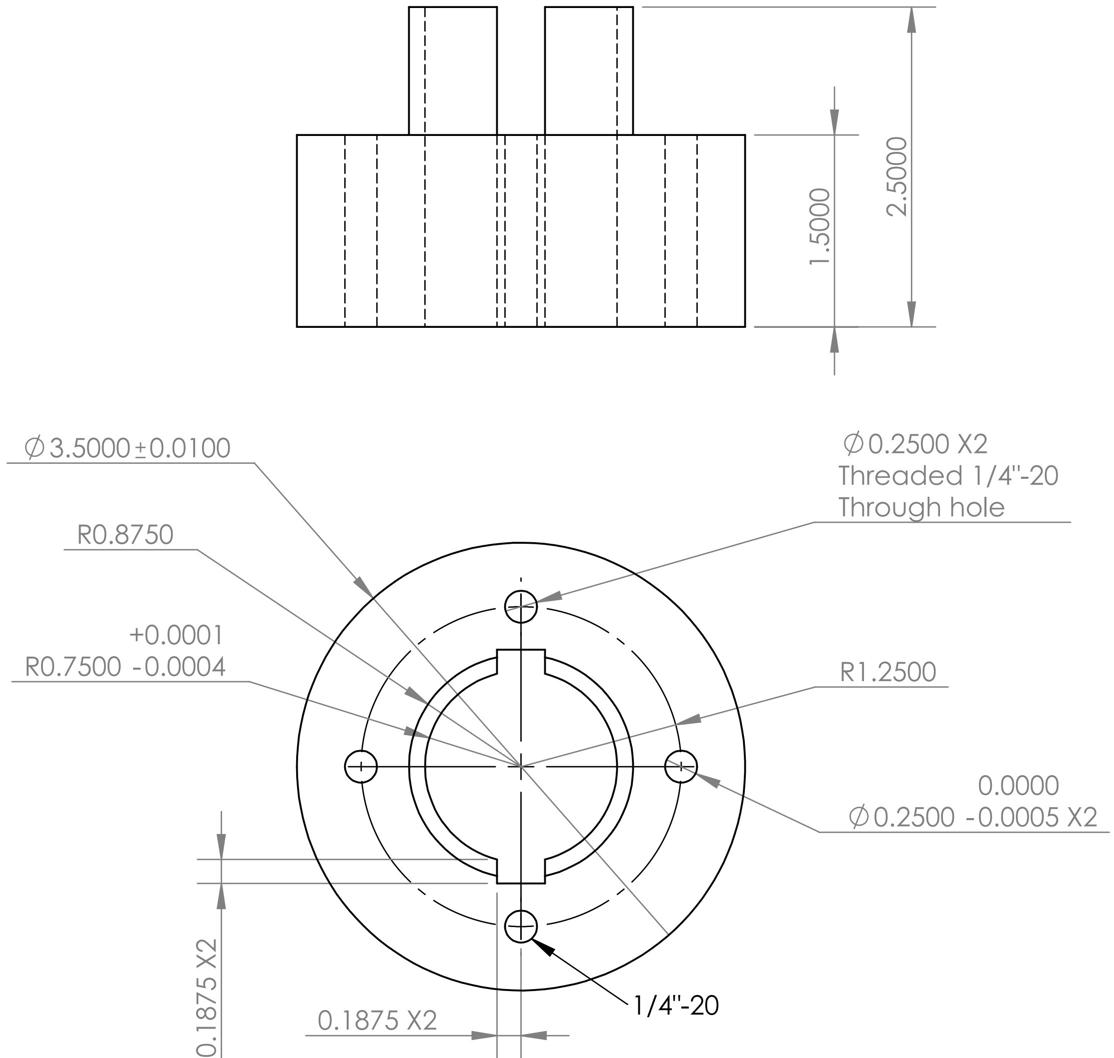

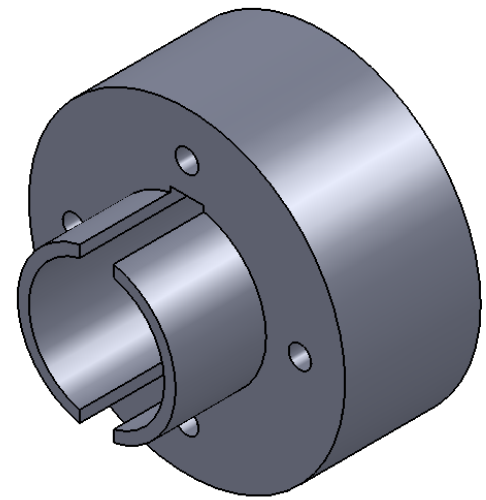

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		INTERPRET GEOMETRIC TOLERANCING PER:
		MATERIAL 1018 STEEL
NEXT ASSY	USED ON	FINISH
APPLICATION		DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

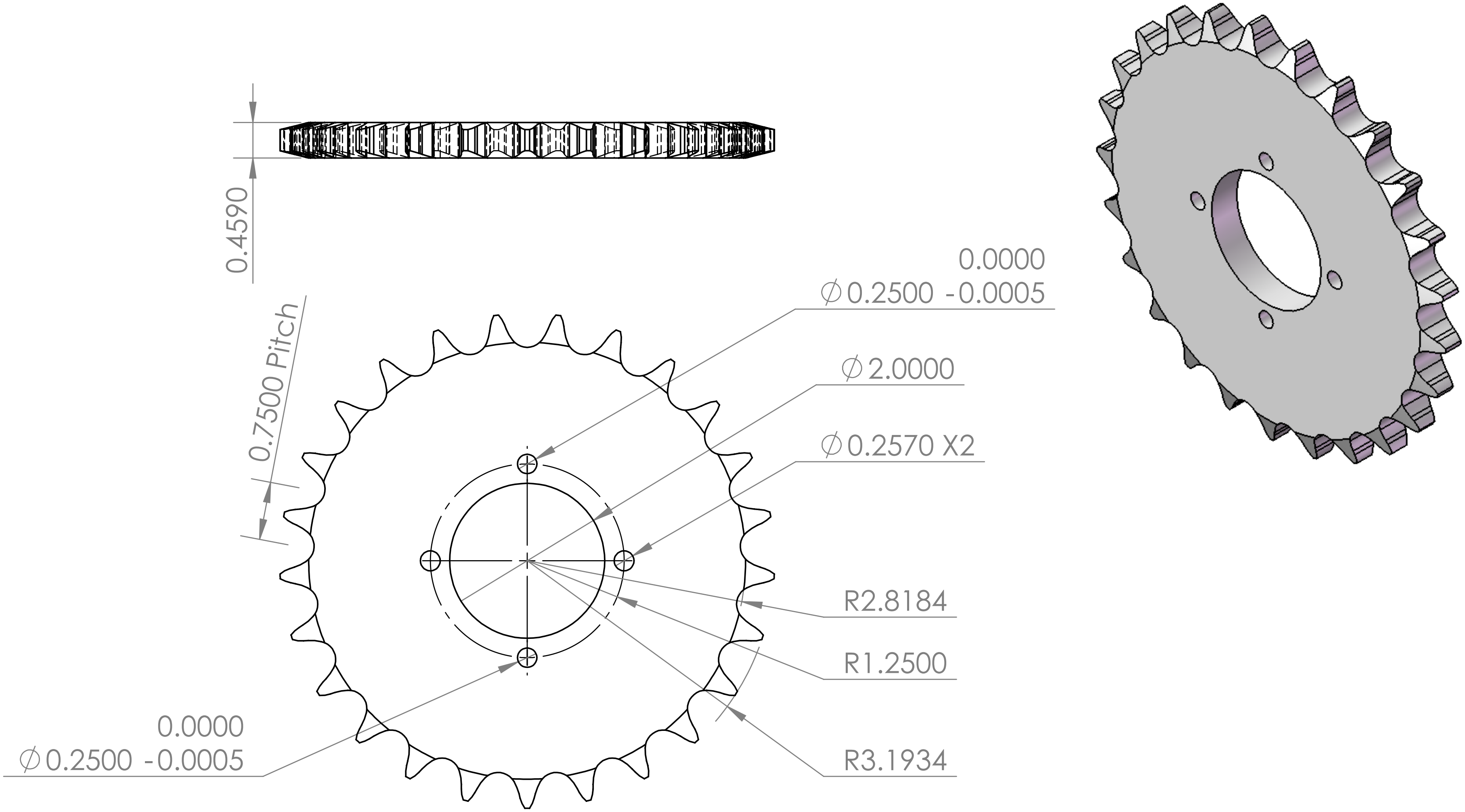
TITLE:

PM Hub

SIZE	DWG. NO.	REV
A		1

SCALE: 2:3 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

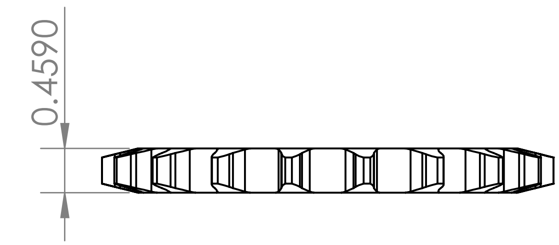
TITLE:

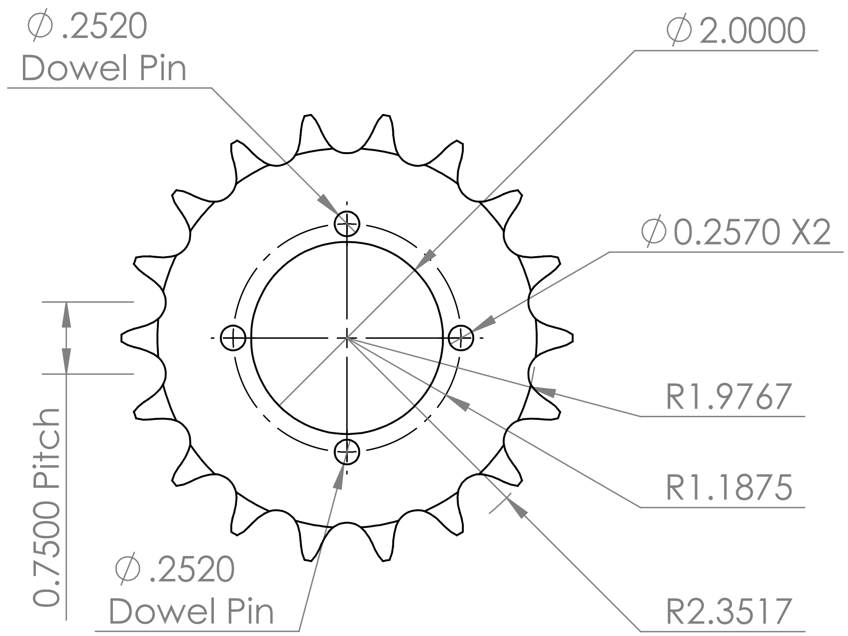
PM Sprocket

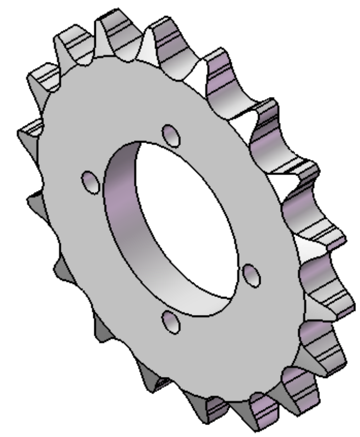
SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1







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			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

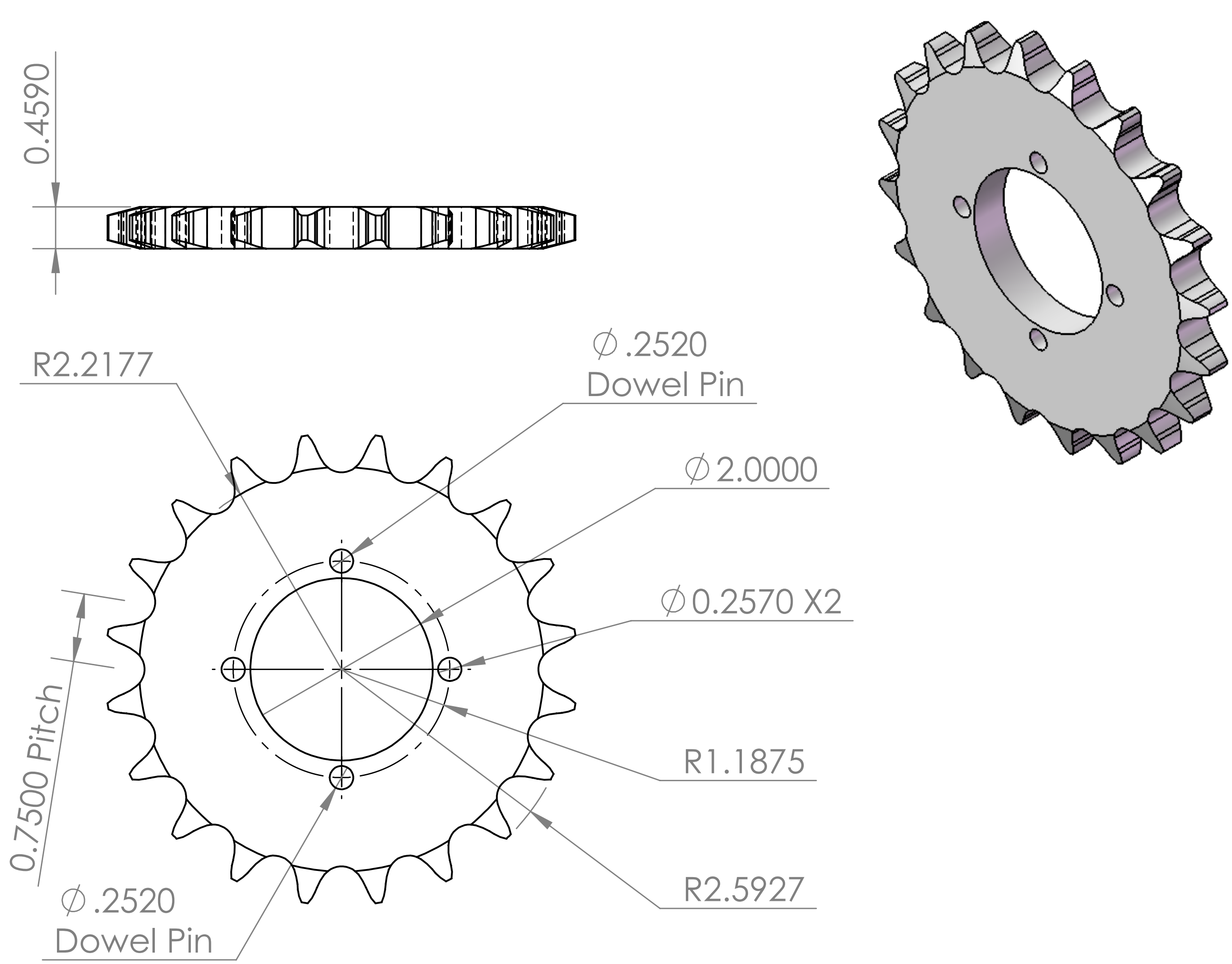
TITLE:

3.3:4.5 DC Sprocket

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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			CHECKED			3.6:4.5 DC Sprocket
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A / DWG. NO. / REV 1
	NEXT ASSY / USED ON	FINISH				
	APPLICATION	DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

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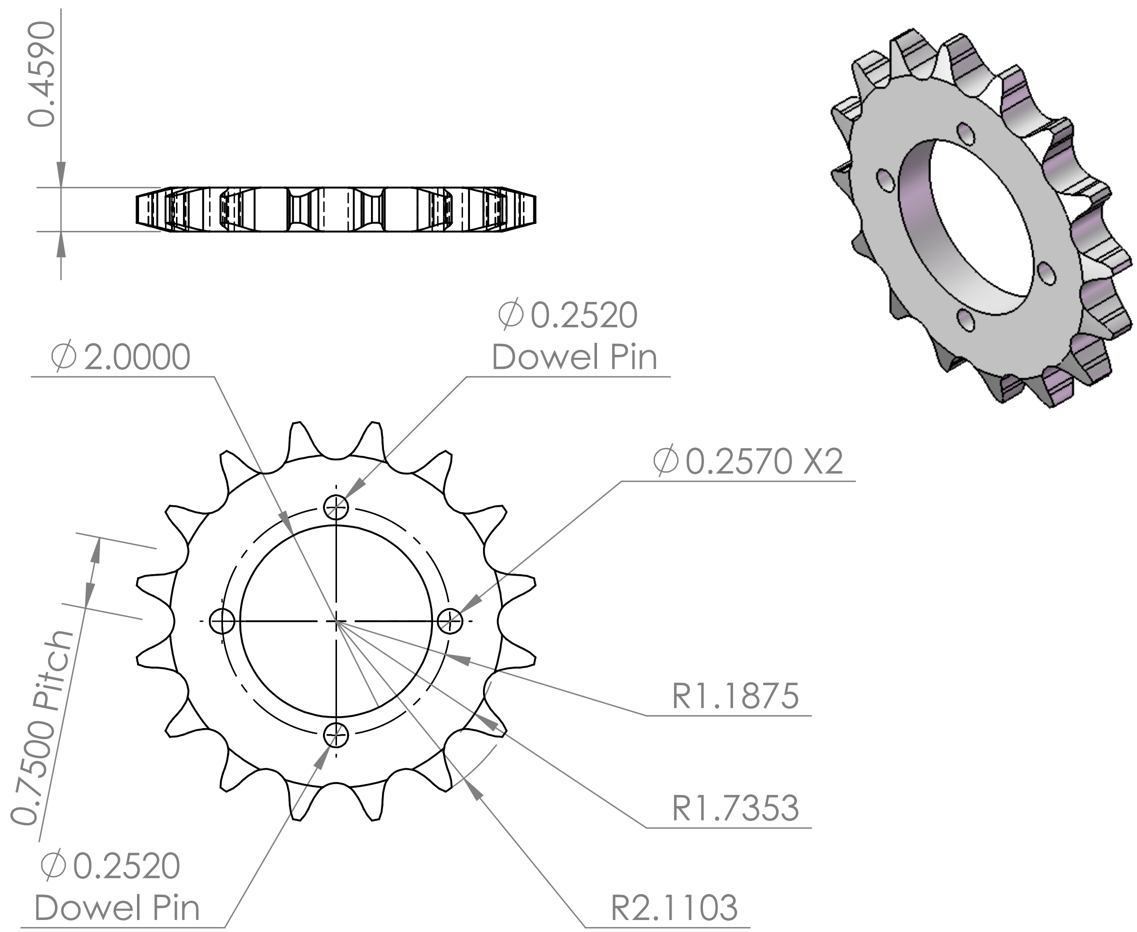
5 4 3 2 1



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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

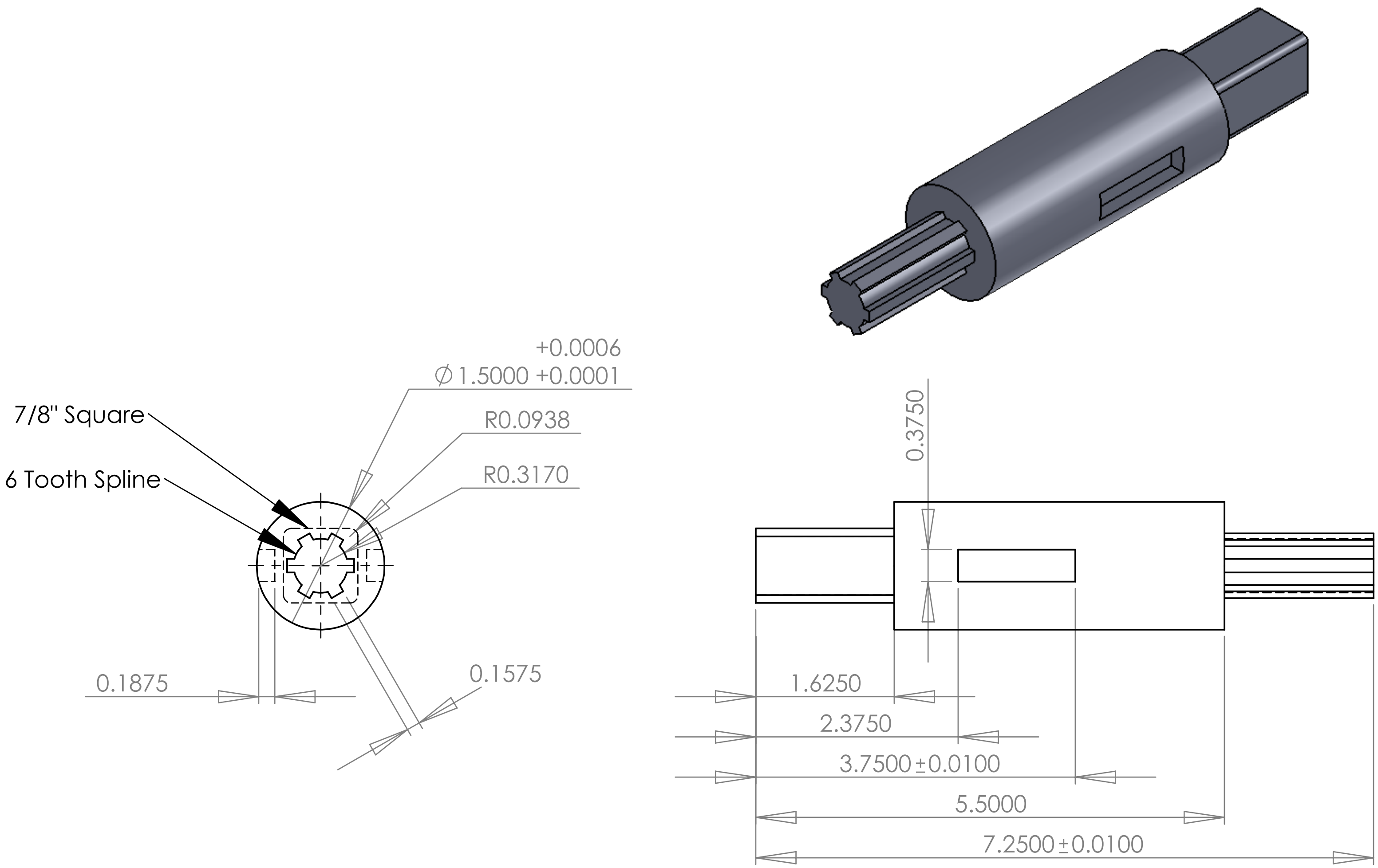
TITLE:

2.9:4.5 DC Sprocket

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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U of M Xebra Team Fall 2010

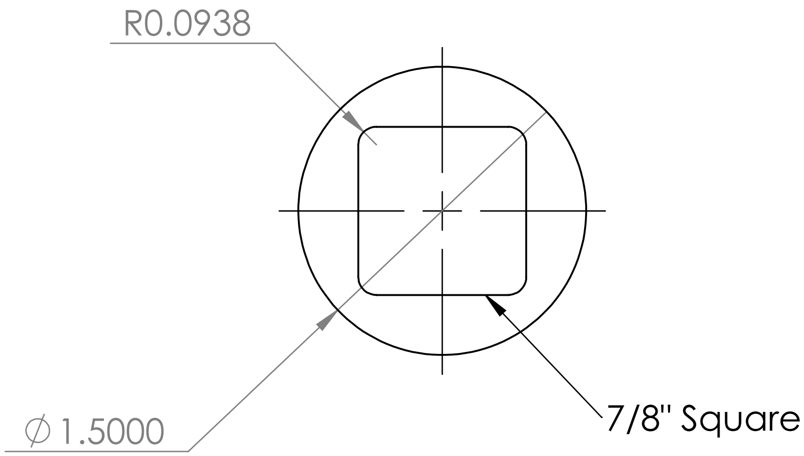
TITLE:

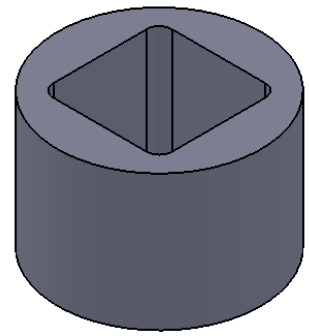
Output Shaft for DC Motor (Male)

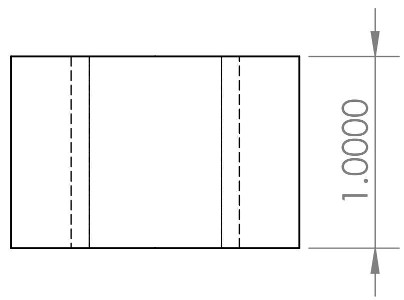
SIZE	DWG. NO.	REV
A		

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1







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NEXT ASSY	USED ON
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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	12/12
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

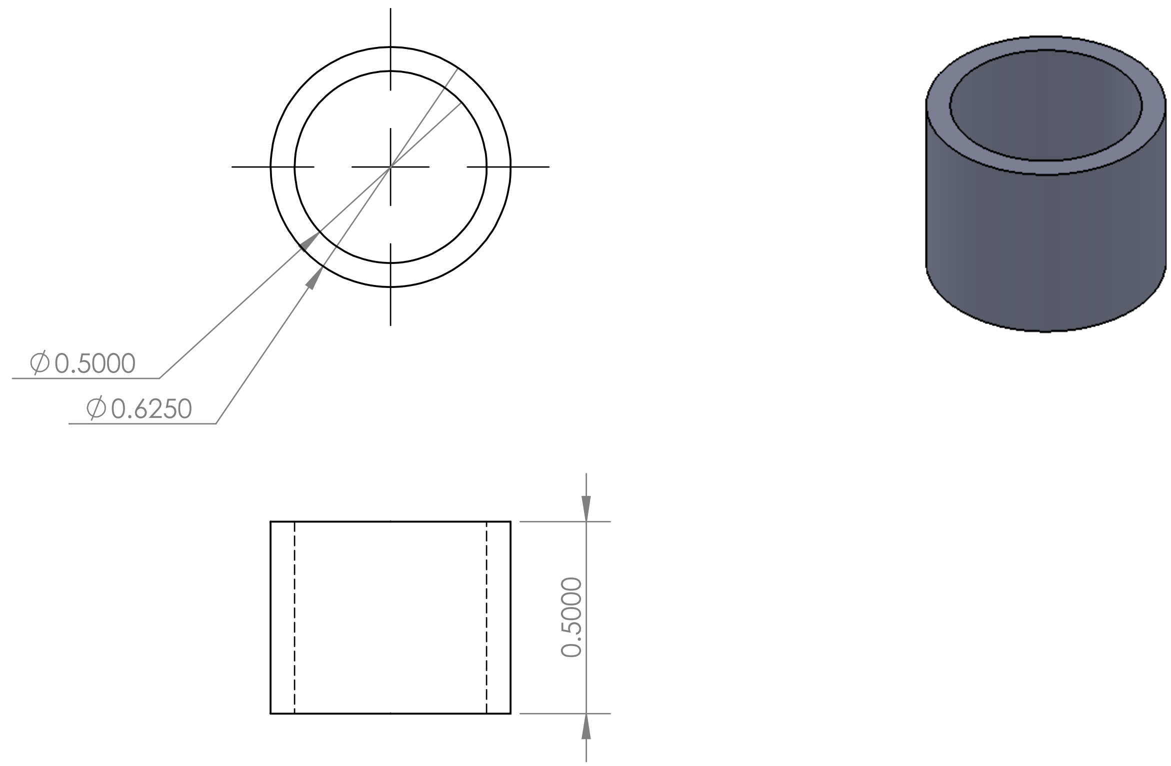
TITLE:

DC Shaft Cap

SIZE	DWG. NO.	REV
A		1

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL Aluminum	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

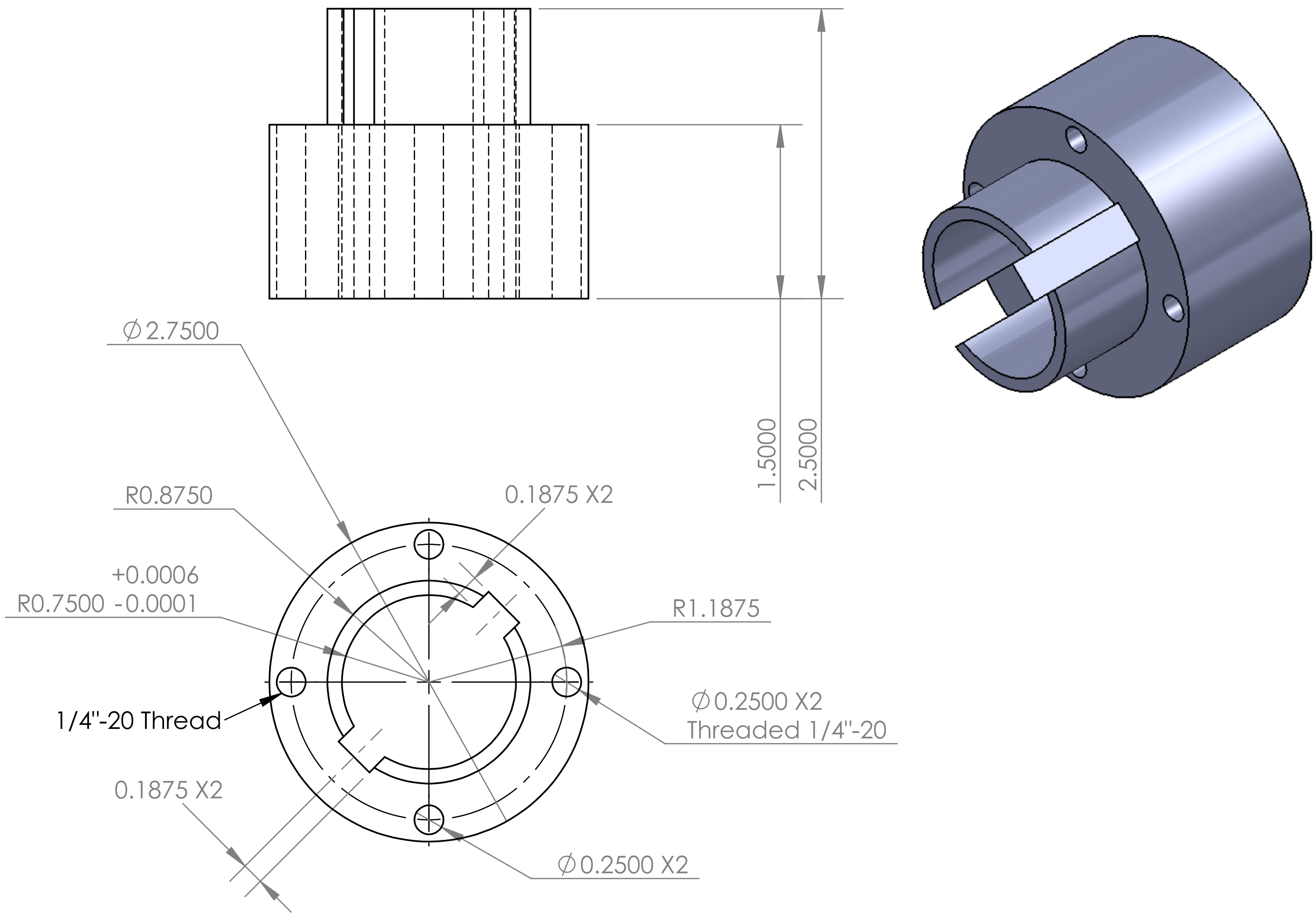
TITLE:

Spacer Mount

SIZE A | DWG. NO. | REV 1

SCALE: 2:1 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



	NAME	DATE
DRAWN	MW	12/12
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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 STEEL

FINISH

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE DRAWING

U of M Xebra Team Fall 2010

TITLE:

DC Hub

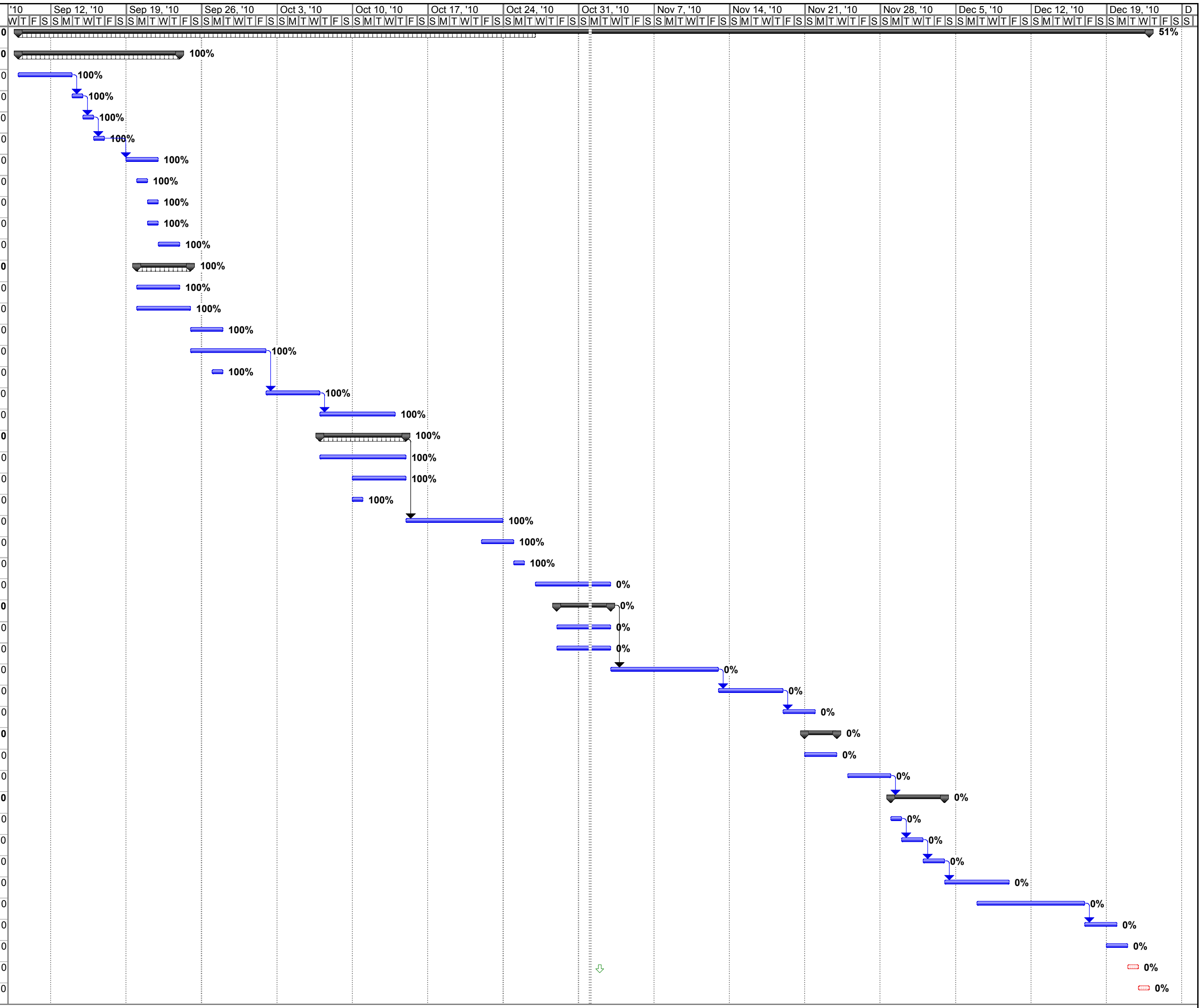
SIZE A | DWG. NO. | REV 1

SCALE: 2:3 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

APPENDIX H: GANTT CHART

ID	Task Name	Duration	Start	Finish
1	**ME450 XEBRA**	**105 days**	**Thu 9/9/10**	**Wed 12/22/10**
2	**Preliminary Tasks**	**15 days**	**Thu 9/9/10**	**Thu 9/23/10**
3	Gathering Literature	5 days	Thu 9/9/10	Mon 9/13/10
4	1st Sponsor Meeting-EPA	1 day	Tue 9/14/10	Tue 9/14/10
5	1st Sponsor Meeting-UofM	1 day	Wed 9/15/10	Wed 9/15/10
6	Summarize Cutomer Requirements	1 day	Thu 9/16/10	Thu 9/16/10
7	Discussing Design Specifications	3 days	Sun 9/19/10	Tue 9/21/10
8	2nd Sponsor Meeting-UofM	1 day	Mon 9/20/10	Mon 9/20/10
9	2nd Sponsor Meeting-EPA	1 day	Tue 9/21/10	Tue 9/21/10
10	Creating Gantt	1 day	Tue 9/21/10	Tue 9/21/10
11	Creating QFD	2 days	Wed 9/22/10	Thu 9/23/10
12	**Design Review 1**	**5 days**	**Mon 9/20/10**	**Fri 9/24/10**
13	Oral Presentation	4 days	Mon 9/20/10	Thu 9/23/10
14	Writing Paper	5 days	Mon 9/20/10	Fri 9/24/10
15	Go through existing CAD models	3 days	Sat 9/25/10	Mon 9/27/10
16	Concept Generation	7 days	Sat 9/25/10	Fri 10/1/10
17	Get Tools to Work on White Xebra	1 day	Mon 9/27/10	Mon 9/27/10
18	Design Comparisons	5 days	Sat 10/2/10	Wed 10/6/10
19	CAD Work	7 days	Thu 10/7/10	Wed 10/13/10
20	**Design Review 2**	**8 days**	**Thu 10/7/10**	**Thu 10/14/10**
21	Report 2	8 days	Thu 10/7/10	Thu 10/14/10
22	Oral presentation	5 days	Sun 10/10/10	Thu 10/14/10
23	Building Mock Up	1 day	Sun 10/10/10	Sun 10/10/10
24	Analysis	9 days	Fri 10/15/10	Sat 10/23/10
25	Getting Quotes / Setting the budget	3 days	Fri 10/22/10	Sun 10/24/10
26	Discussion of Budget with EPA/UofM	1 day	Mon 10/25/10	Mon 10/25/10
27	Hardware purchase	7 days	Wed 10/27/10	Tue 11/2/10
28	**Design Review 3**	**5 days**	**Fri 10/29/10**	**Tue 11/2/10**
29	Report 3	5 days	Fri 10/29/10	Tue 11/2/10
30	Oral presentation	5 days	Fri 10/29/10	Tue 11/2/10
31	Hardware manufacturing	10 days	Wed 11/3/10	Fri 11/12/10
32	System assembly and testing	6 days	Sat 11/13/10	Thu 11/18/10
33	System trouble shoot	3 days	Fri 11/19/10	Sun 11/21/10
34	**Design Review 4**	**3 days**	**Sun 11/21/10**	**Tue 11/23/10**
35	Oral presentation	3 days	Sun 11/21/10	Tue 11/23/10
36	Thanksgiving	4 days	Thu 11/25/10	Sun 11/28/10
37	**Validation Testing**	**5 days**	**Mon 11/29/10**	**Fri 12/3/10**
38	Testing for fit / Safety of Design / Slack	1 day	Mon 11/29/10	Mon 11/29/10
39	Configurable Gear Ratio / Change Gears	2 days	Tue 11/30/10	Wed 12/1/10
40	Torque Test on Gear / System	2 days	Thu 12/2/10	Fri 12/3/10
41	Design Expo	6 days	Sat 12/4/10	Thu 12/9/10
42	Final report	10 days	Tue 12/7/10	Thu 12/16/10
43	Clean work space	3 days	Fri 12/17/10	Sun 12/19/10
44	Prepare Reimbursements	2 days	Sun 12/19/10	Mon 12/20/10
45	Transport Both Xebra's to EPA	1 day	Tue 12/21/10	Tue 12/21/10
46	Prototype submission	1 day	Wed 12/22/10	Wed 12/22/10



Project: F10-Gantt
Date: Mon 11/1/10

Critical | Task | Baseline | Milestone | Project Summary | Deadline
Critical Split | Split | Baseline Split | Summary Progress | External Tasks
Critical Progress | Task Progress | Baseline Milestone | Summary | External Milestone

APPENDIX I: MACHINING SPEED CALCULATIONS

The following are the calculations used to determine spindle speeds during the manufacturing process

Shaft Lathe Speeds - $RPM = \frac{4 * Speed}{Diameter} = \frac{4 * 110\,fpm}{2"} = 220 RPM$

Hub Lathe Speeds - $RPM = \frac{4 * Speed}{Diameter} = \frac{4 * 110\,fpm}{4"} = 110 RPM$

Hub Drill Speeds - $N = \frac{4 * Speed}{Diameter} = \frac{4 * 90\,fpm}{0.25"} = 1400 RPM$

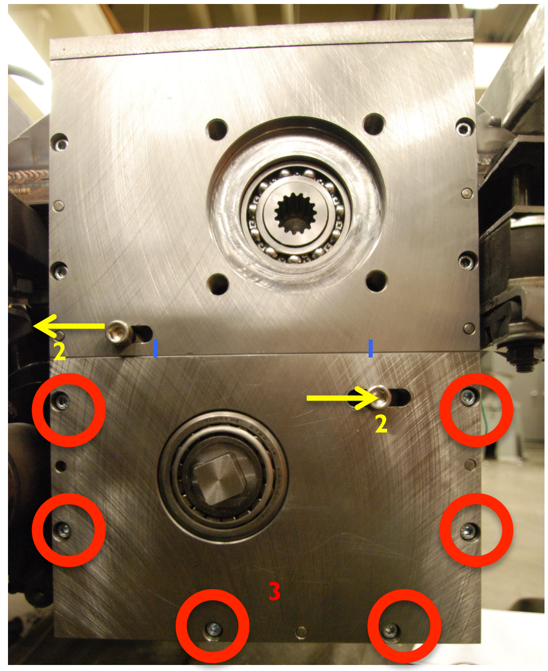
APPENDIX J: GEAR RATIO CHANGE INSTRUCTIONS

This appendix details the procedure to change the gear ratio within the reduction case between 2.9:1, 3.3:1, and 3.6:1 (from hydraulic pump/motor to the vehicle's wheels).

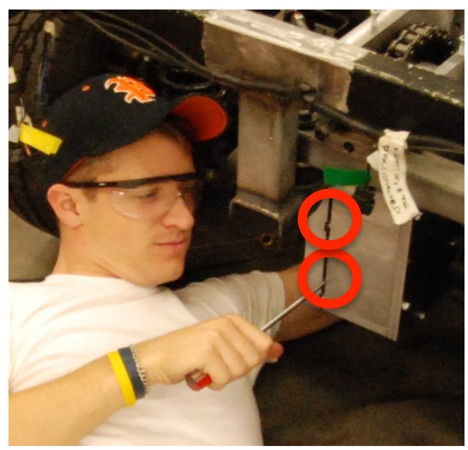
1) Loosen the tensioners by turning hex bolts (3/8"–16 D) counter-clockwise

2) Move bolts outwards towards edges of gear reduction case to relax the chain tension

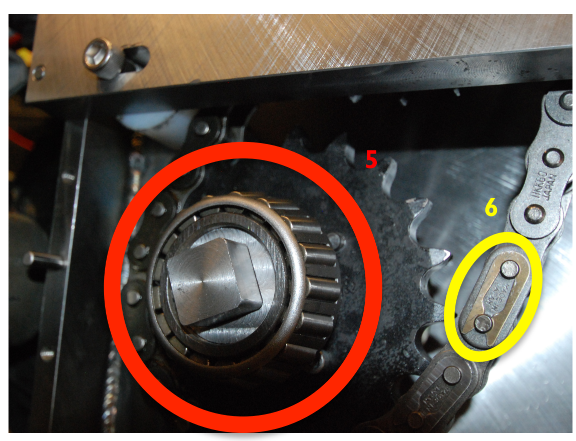
3) Loosen and remove the six hex bolts (¼"–20 D) on the lower access door



4) Remove the lower access door by prying the case open using a flat-head screwdriver, chisel, or other flat tool on the four machined notches
    a. There are two machined notches on each side of the case, one lower and one upper
    b. Be sure to evenly pry on each upper and lower notch on each side so that the plate comes out as flat as possible (not angled)

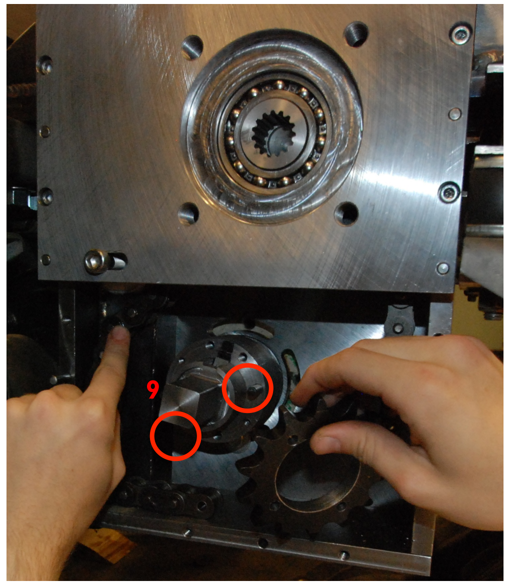


5) Once the plate is removed, remove the tapered roller bearing and the attached square cap (the cap should be pressed into the inner race of the bearing)
6) Then use pliers to remove a quick-connect link on the roller chain.
    a. There is more than one quick-connect link
    b. A quick-connect link has a bronze clip connecting two pins
    c. Remove this pin first, then the spanner piece
    d. Finally, slide the two pin assembly out (from behind)
    e. The chain should then be disconnected



f. Note: the quick-connect link may be in a difficult to access position based on the chain orientation
    i. This is the reason the top plate is also removable (four ¼"–20 D hex bolts) at the top of the case

7) Remove the two hex bolts (¼"–20 D) on the sprocket



8) Remove the current sprocket

9) Replace the new desired sprocket by placing the sprocket on the dowel pins and bolting the new sprocket in place
    a. Two of the holes (180° apart) on the sprocket will be larger than the other two
    b. These two holes will rest on the dowel pins
    c. The other two are where the bolts are located

10) Depending on the sprocket size, additional chain links may need to be added or removed
    a. Chain length is altered (lengthened or shortened) by using additional chain links
        i. Chain links are attached or removed as in Step 6 with the quick-connect links
        ii. Half-links are available for more precise chain length specification
    b. The smaller the diameter of the sprocket, the shorter the length of chain and the larger the diameter of the sprocket, the longer the chain

11) Replace the tapered bearing and the attached square cap
12) Lift the excess chain slack manually
    a. This is accomplished using a wire or other thin material
    b. Lower this wire down from the open top of the gear reduction case
    c. Then lift the chain taught where the tensioner will be placed once the lower access door is re-attached
13) While the chain slack is held in place manually, replace the lower access door by replacing the door on the three dowel pins
14) Completely close the access door by hammering the door shut
    a. Use a brass hammer or rubber mallet
    b. Some dowel pins may need to be hammered flush with the top of the access door depending on how they were removed
15) Release the chain slack and adjust the chain tension
    a. The chain should now rest on the chain tensioners (white cylinders)
    b. Chain tension is adjusted by moving the 3/8"–16 D hex bolts and then tightening in the desired position
16) Replace the six hex bolts (¼"–20 D) on the lower access door

APPENDIX K: SAFETY REPORT

Me 450 Safety Report: Fall 2010

Team Number: # 5

Project Title: Electric-Hydraulic Hybrid Motor Coupling

Team Members:
David Fok
Andrew Gavenda
Anuj Shah
Mat Wecharatana

APPROVAL

Name: ______________________________

Signature: ______________________________

Date: ______________________________

Safety Report Instructions

Executive Summary
Answer the following questions: What activities or designs are covered by this report? What hazards have you identified and eliminated? What analysis have you performed and why do you conclude that the activities/designs are low risk? Be sure you consider all aspects of your project: experimental data collection, component design, system design, manufacturing, assembly, and testing.

1. Experimentation Plans Prior to Design Completion
   For your experimentation, list what data you will be collecting and why. Are any experiments that might have safety risks unnecessary? Why/Why not?

2. Purchased Component and Material Inventory
   Provide an inventory of all materials (solid materials such as aluminum/wood/etc.) and purchased components you will be using. Why are these materials and components necessary?

3. CAD Drawings and Designsafe Summary for Deigned Parts
   Provide CAD drawings for components you have designed and will manufacture.

   a. Conduct a risk assessment using Designsafe software (available on CAEN) for each designed component and for the full assembly of components constituting your design. Provide the Designsafe output as an appendix to this safety report and summarize the results in your own words for the main report body.

4. Manufacturing
   Provide a list of all fabrication or manufacturing activities you will perform. Where will these activities take place? Why are these processes necessary

   a. CAD drawings for parts to be manufactured are required (per #4 above).

   b. For machining or forming processes, list special setup requirements and the operational conditions that will be employed (e.g., speeds, feeds, etc.).

5. Assembly
   How and where will your components be assembled? On what basis do you conclude that the assembly will not fail before use, during use, or after use?

6. Design and Validation
   How and where will your final design be tested? Which design specifications are being validated through the testing? Do you plan to test aspects of your design as you manufacture your prototype, or are you going to be validating a finished prototype after most/all manufacturing has been completed?

   a. What would you consider to be your first major test of the design?

   b. Have you arranged with your Section Instructor to have a cognizant individual present at your first major test? Who will this be? When do you expect this first test to take place?

7. Additional Appendices

   a. For every chemical (powder, liquid, gas – distinguished from a “material” defined in step 2 above as a solid) you propose for use in testing or design, you must supply a complete MSDS as an appendix.

b. I f relevant safety documentation is provided with a purchased component, include it as an appendix.

8. Submission

   a. Submit this report to your Section Instructor for signature. Please check with your Section Instructor to learn if a hard copy or an electronic copy is preferred for signature. Regardless, please create an electronic copy for filing and email to Bob Coury and Dan Johnson.

   b. After the report is signed, email a copy to Bob Coury (hornet@umich.edu) and our course GSI Dan Johnson (danijohn@umich.edu)

      i. Both Bob and Dan are expected to raise additional safety concerns that will be shared with the students and the Section Instructor. They have the authority to stop any activity they deem unsafe, regardless of whether a safety report has been signed. If this happens, the safety report will be revised and re-signed by the Section Instructor, then emailed with revisions to Bob Coury and Dan Johnson

Contents

1. Executive Summary

Through this safety report, and based on the knowledge of the existing vehicle, we will talk about the new components that we will purchase and those that we will fabricate to meet our customer requirements and engineering specifications. All our new components will be validated through the Design-Safe process. We have developed a detailed manufacturing and assembly plan to ensure that our design is completed efficiently and safely. Finally a validation and testing plan is provided for a detailed evaluation of our design.

Our sponsors at the EPA conducted preliminary weight tests on the vehicle to determine the position of the center of gravity of the Xebra vehicle. We will be using this data in the future to validate the stability of the vehicle by finding the new center of gravity of the vehicle using the weights of the new components we will be adding. On the whole we will be performing no experimentation on the vehicle before the completion of the design.

Once we had a thorough understanding of our customer requirements, we focused on the new components that we will be designing to meet our engineering specifications. The main components that we will be adding include; the reduction case that will house the chain and sprocket transmission system, the sub-frame which will support the DC motor and the reduction case, and the hydraulic pump/motor. The hydraulic pump/motor will be bolted on to the reduction case using a mounting plate. The reduction case is mounted on the sub-frame using a tab and gusset support. A thorough stress analysis was performed on the fabricated parts to ensure their safety and confirming whether they meet our safety factor requirement. We also conducted a Design-safe analysis to eliminate the potential danger brought by our new components.

Our design is completely mechanical in nature. We have a hydraulic pump/motor and an electric DC motor that we will be connecting, but both these systems are external to our design. Since there will be no electric equipment, the hazards due to fire and electrocution. Since the complete hydraulic system will not be ready by the end f our project, we will not be dealing the safety concerns of high pressure fluids. Lastly, our vehicle is not going to be mass produced, therefore there are no material handling or waste management concerns.

We also provided a detailed list of the components and materials that we need to purchase to ensure that we meet our cost limit. Once we validate our design through the engineering analysis and the design-safe processes we can move forward to the manufacturing and assembly processes. A step by step process of the manufacturing plan and the assembly plan is provided. The order of assembly is extremely crucial for our design to effectively be completed and functional. We have also provided detailed CAD drawings of each fabricated component to assist in the manufacturing our components.

After speaking with our sponsors from the EPA and our customers from the university, we have confirmed that only the testing of our fabricated design and not the entire hydraulic system-electric system is to be validated. As the hydraulic-electric hybrid system will not be completely finished within our timeline, we will not be affecting the overall performance of the car and testing on the EPA dynamometer will not be necessary. We have come up with an alternate validation plan to test how our design meets each one of our engineering specifications and customer requirements.

2. Experimentation Plans Prior to Design Completion

Our sponsors at the EPA conducted preliminary weight tests on the vehicle to determine the position of the center of gravity of the Xebra vehicle. We will be using this data to validate the stability of the vehicle by finding the new center of gravity of the vehicle using the weights of the new components we will be adding. On the whole we will be performing no experimentation on the vehicle before the completion of the design.

Safety Concerns

The entire hydraulic system will not be completed within our project timeline. Since we will not be working with high pressure hydraulic fluids during our validation tests, the overall safety concern of our design is reduced. Our validation procedure does not involve most dangerous types of failure, and all of our fabricated components have built in safety factors that account for normal and extreme loading conditions. Appropriate eye protection and insulated tools are planned to be used. Each team member will make sure the electric system is not powered, and safety precautions are taken when the vehicle is raised on a platform.

3. Purchased Component and Material Inventory

The bill of materials shown below is a list of all the required materials for completing our project. A number of items are being purchased from McMaster Carr while others are being purchased from online metal suppliers. All of the items on this list are essential for successfully constructing our power transmission system.

The sprockets and chain to be purchased from McMaster are essential for transmitting the torque from the pump motor to the existing DC gearbox. The bearings we are purchasing from McMaster are also necessary to ensure the radial loads being applied in our system are off-loaded to prevent any forces on the Pump motor. The raw materials we are purchasing include ¼ inch, ½ inch, ¾ inch, and 1 inch thick A36 steel plate, 1-½ by 2 inch A36 steel tube, and 2 inch and 4 inch diameter 1018 steel rounds. The A36 plate is necessary to manufacture the reduction casing as well as all of its support mechanisms. The A36 steel tube is for the new sub-frame that we will be machining. Finally, the 1018 steel rounds are for the shafts and hubs that we will be producing to transmit the torque from shaft to sprocket and sprocket to shaft. The rest of the dowel pins, bolts, shoulder bolts, nuts, and washers are all for the assembly of reduction housing, the two shafts, two hubs, and connections to the sub-frame. The use of all of these components will be detailed in the Manufacturing Plan in the following section.

Bill of Materials

Item	Description	Manufacturer/Distributor	Part #	Unit	QTY	Price/QTY	Total Price
Upper Sprocket	25 Tooth ANSI 60 Sprocket	McMaster	2299K77	1	1	$29.13	$29.13
Lower Sprocket 1	14 Tooth ANSI 60 Sprocket	McMaster	2299K65	1	1	$21.70	$21.70
Lower Sprocket 2	18 Tooth ANSI 60 Sprocket	McMaster	2299K69	1	1	$24.94	$24.94
Lower Sprocket 3	20 Tooth ANSI 60 Sprocket	McMaster	2299K72	1	1	$25.44	$25.44
Chain	ANSI 60 Chain 2 Ft	McMaster	6261K472	1	1	$11.06	$11.06
Chain Connecting Link	ANSI 60 Chain	McMaster	6261K195	1	8	$1.21	$9.68
Chain Adding Link	ANSI 60 Chain	McMaster	6261K245	1	8	$1.25	$10.00
Bearing	1.5ID Bearing	McMaster	60355K512	1	3	$33.24	$99.72
Tapered Bearing Inner	1.5ID Tapered	McMaster	5709K270	1	1	$25.87	$25.87
Tapered Bearing Outer	3"OD outer Race	McMaster	5709K64	1	1	$12.38	$12.38
Gearbox Sides/DC Plate	1/2" A36 plate 1'x2'	Metals Depot	P112	1	2	$52.68	$105.36
Gearbox Pump Plate	1" A36 plate 1'x2'	Metals Depot	F2112	1	1	$106.08	$106.08
Gearbox Tabs	3/4" A36 Plate 8"x12"	Metals Depot	F2348	1	1	$25.50	$25.50
Gearbox Cutout Piece	1/4" A36 Plate 1'x1'	Metals Depot	P114	1	1	$12.86	$12.86
Hub Stock	4" 1018 Round 1'L	Metals Depot	R24	1	1	$117.50	$117.50
Shaft Stock	2" 1018 Round 2'L	Metals Depot	R22	1	1	$40.08	$40.08
Subframe Main Rail	2"x1.5"x.1875"T 4'L	Metals Depot	T121316	1	1	$27.72	$27.72
Subframe Cross Rail	1.25"x1.25"x.1875"T 4'L	Metals Depot	T1114316	1	1	$24.04	$24.04
Collars	9/16" 1.5"ID 2.375"OD	McMaster	6436K23	1	2	$6.17	$12.34
Gearbox Dowel Pins	1/4"D 1"L	McMaster	98381A542	50	1	$9.85	$9.85
Gearbox Dowel Pins - Long	1/4"D 1.5"L	McMaster	98380A546	10	2	$11.70	$23.40
Gearbox Bolts - Long	1/4"D 1.25"L	McMaster	94912A470	5	3	$5.53	$16.59
PM Mount Bolts	1/2"-13 D, 1 1/4" L	McMaster	92865A714	25	1	$9.19	$9.19
PM Mount Bolt Washers	1/2" ID, 1 1/8" OD	McMaster	98370A033	5	1	$9.58	$9.58
UHMW Tension Plastic	1.5"D 1'L UHMW	Onlinemetals.com	N/A	1	1	$3.50	$3.50
Hub Shoulder Bolts	10-24, 1/4 D, 3/4 L	McMaster	91259A535	1	4	$0.98	$3.92
Hub Bolts	1/4"-20 D, 1" L	McMaster	90128A247	25	2	$6.13	$12.26
Tab Bolts	1/2" - 13 D, 2 3/4" L	McMaster	91251A723	5	1	$5.52	$5.52
Tensioner Square Nuts	3/8" Nut	McMaster	92891A300	10	1	$9.63	$9.63
Tensioner Bolts	3/8"-16 D, 2" L	McMaster	92196A632	10	1	$7.17	$7.17
Key Stock	3/8" x 3/8" x 12"	McMaster	98535A170	1	1	$7.42	$7.42
						TOTAL:	$859.43

4. Designsafe Summary for Designed Parts

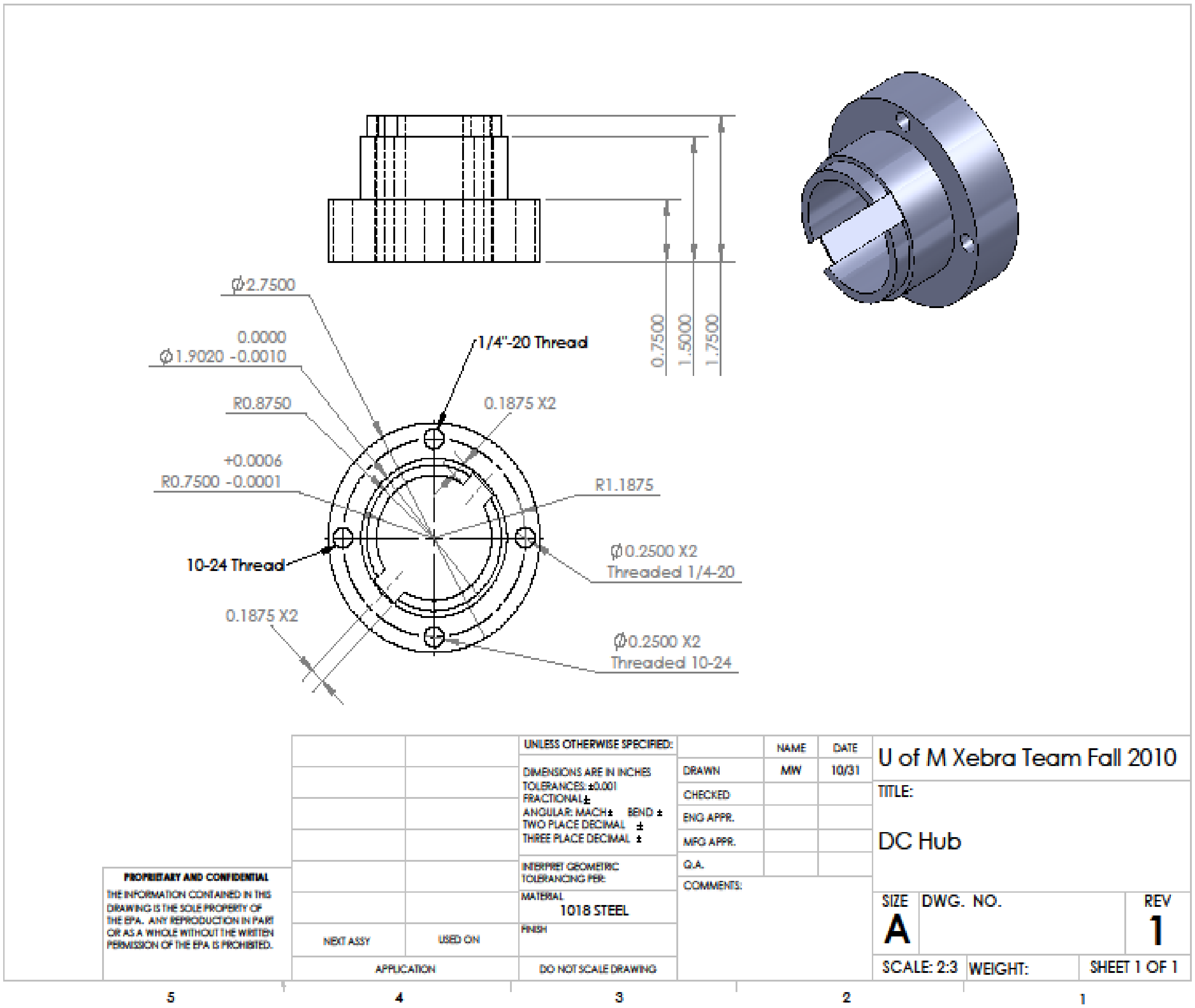
The DesignSafe analysis was broken down into main categories with different safety concerns: Static and Rotational components. Static components include the sub-frame, reduction case, tabs & gussets, and spacer. These parts are classified as static in the sense that they are not intended to move as part of their functionality. This is not to say that movement from vibration, external loads, or other sources of disturbances are not expected and designed for. However, the function of the parts themselves does not require movement. This leads to seven main hazards and failure modes which include 1) cutting/severing, 2) mechanical fatigue, 3) break up during operation, 4) impact, 5) vibration equipment damage, 6) vibration fatigue/material strength, and 7) corrosion. While the geometry of each part and the loads and torques each is subjected to varies, each of these hazards are common to each static component.

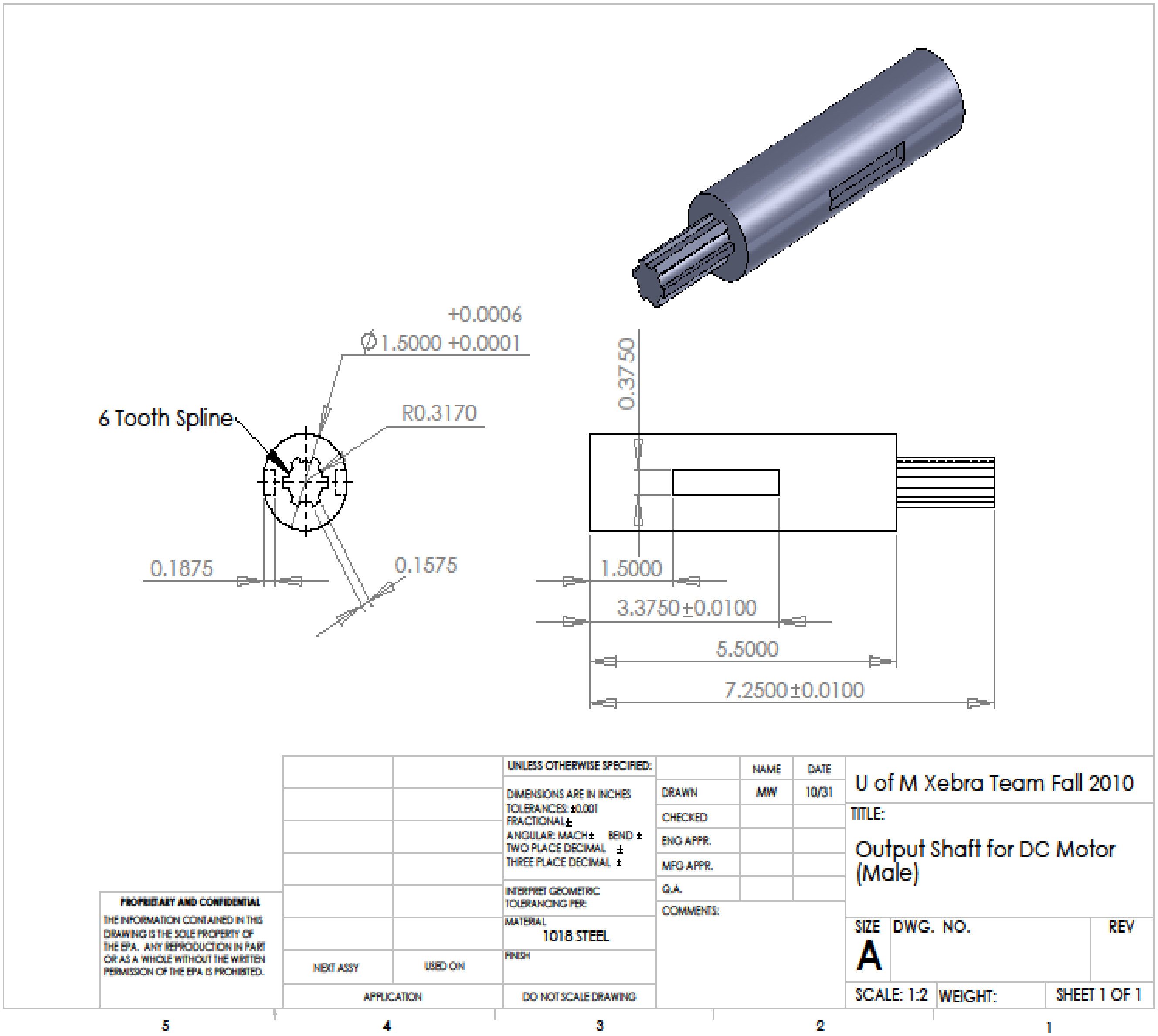
Slightly differently, rotational components are intended to rotate as part of their function. These parts include the hub, shafts, and sprockets. In addition to the hazards that the static components face (because those hazards still apply if the components are in motion), the rotational parts face 2 additional hazards in 1) vibration noise and 2) debris generation. Vibration noise could occur due to misalignment of rotating parts (eccentricity) which leads to the human factor of hearing damage. Likewise, the rotational aspect creates a larger possibility that debris could be ejected, whether the component is broken or intact.

Our design consists entirely of only mechanical components. Therefore, many risks are immediately diminished. The system is fully mechanical, meaning there are only solid part and no fluids (gases or liquids) that are contained in the system in any significant quantity. The only fluid that may be present in the system is lubricant for bearings or the chain and sprocket system. Both of these are assumed to be internal to their respective components and have not been considered. While the system is attached to the pump/motor that has major pressurized fluid concerns, it is assumed that it is fully enclosed and external to our system. Although an electric motor is attached to our system and components, it is fully enclosed and separated from our system through the entirely mechanical intermediate gear reduction casing. The lack of electrical equipment also eliminates fire and explosion hazards caused directly by our system itself. Additionally, there are no lasers or radiation hazards. Since the design will not be mass manufactured, there is no material handling concerns or waste concerns involving processes for mass manufacture (Lean). Additionally, there is no intended immediate human contact with the parts of the system while it is in operation so human factors such as ergonomics, biological / health, ingress / egress, industrial hygiene are mostly mitigated. Lack of chemicals, fluids, or electrical equipment directly in the fabricated parts eliminates those hazards.

Overall, the safety risks of the project are generally low due to the function of the system. Users are not expected to be near the components of the system while it is in operation, which is when the system is most likely to fail if at all. This is because when the vehicle is operating, the user is either someone conducting a test on the truck or someone driving the vehicle. In the first case, the operator usually stands several feet away from the components of the system, lessening any danger that might be associated with the design. Similarly, for the second case, the operator is enclosed by a cab and several other components that stand between the system and the operator. You can see a DesignSafe analysis of our components under Appendix A.

5. CAD Drawings





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Q.A.
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U of M Xebra Team Fall 2010
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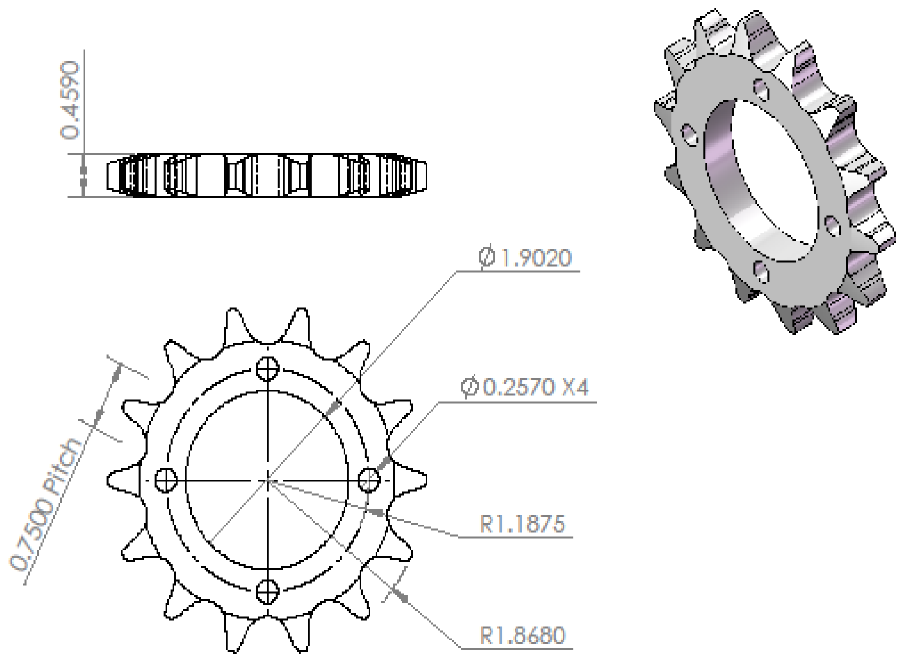

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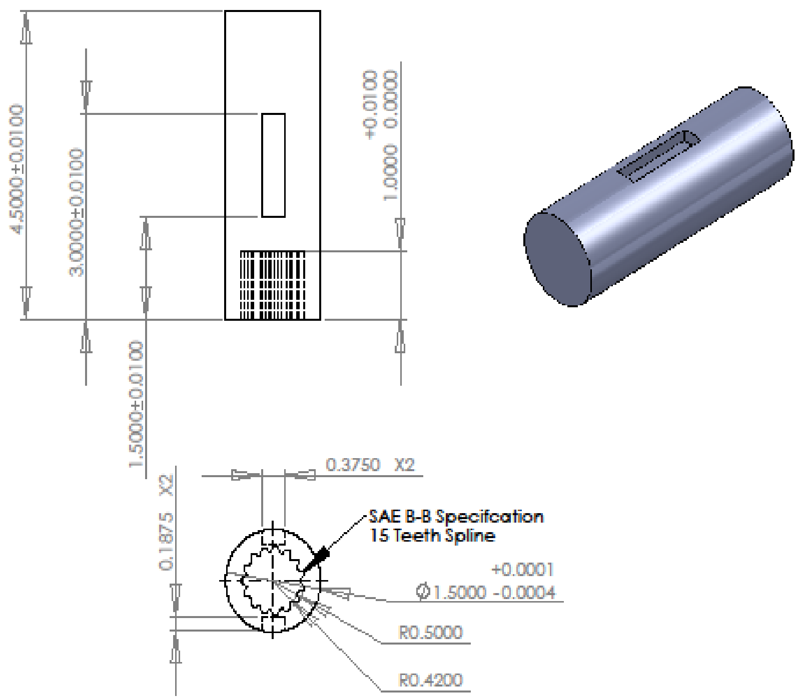

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Xebra Team

TITLE:

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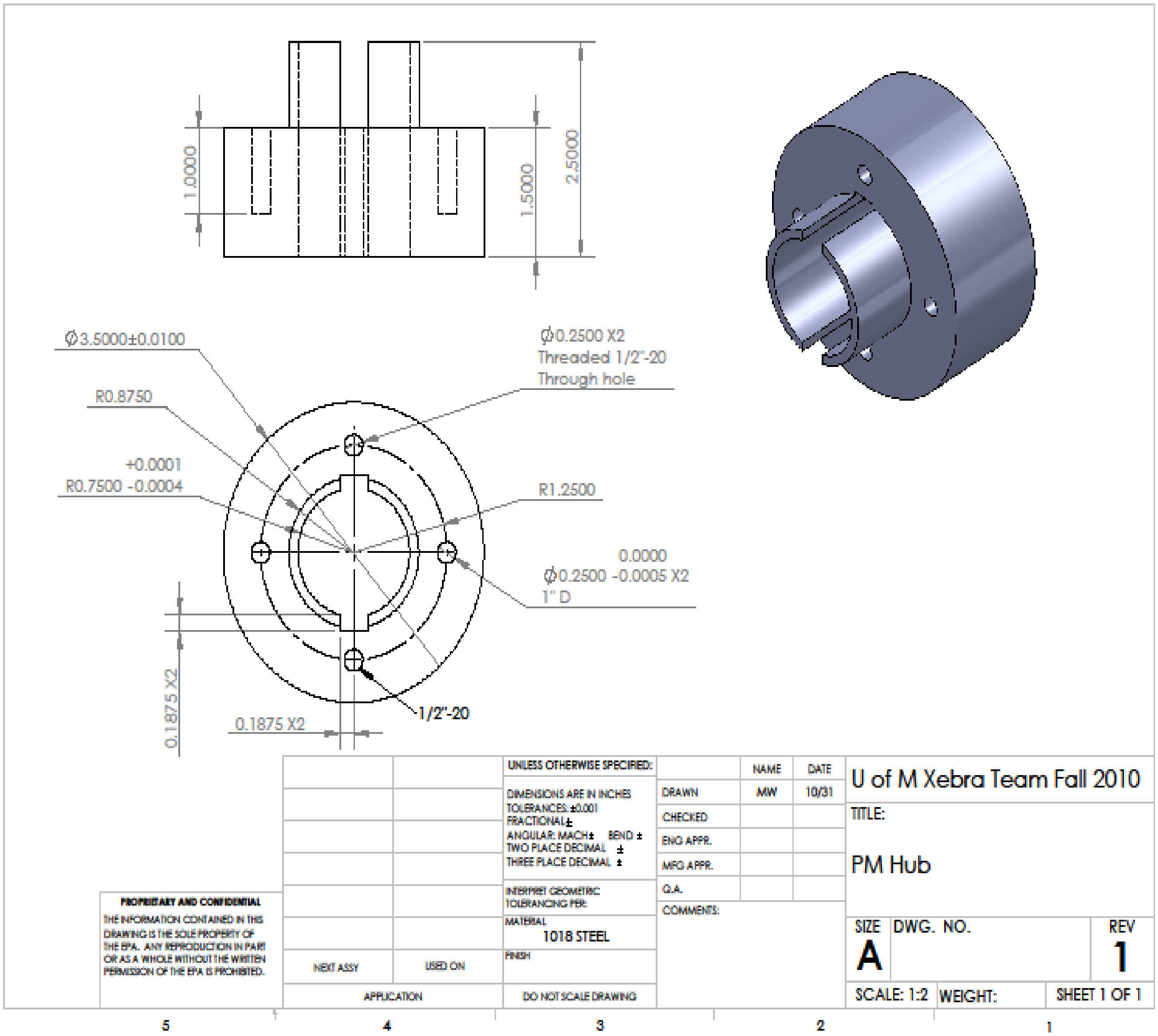
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U of M Xebra Team Fall 2010
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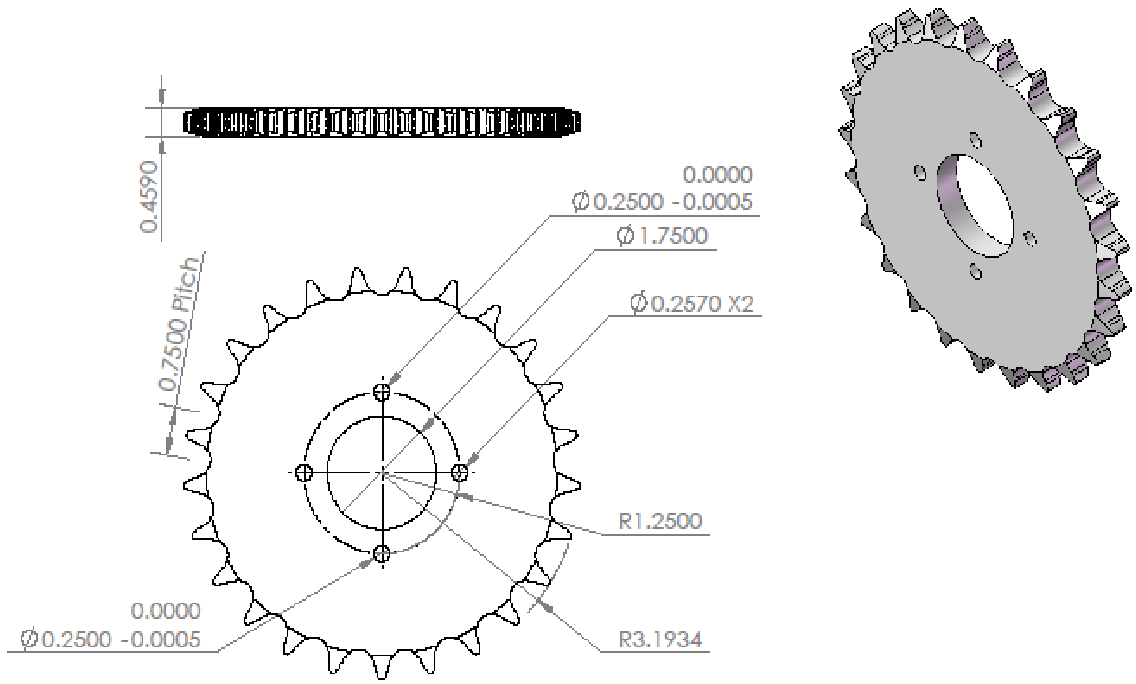

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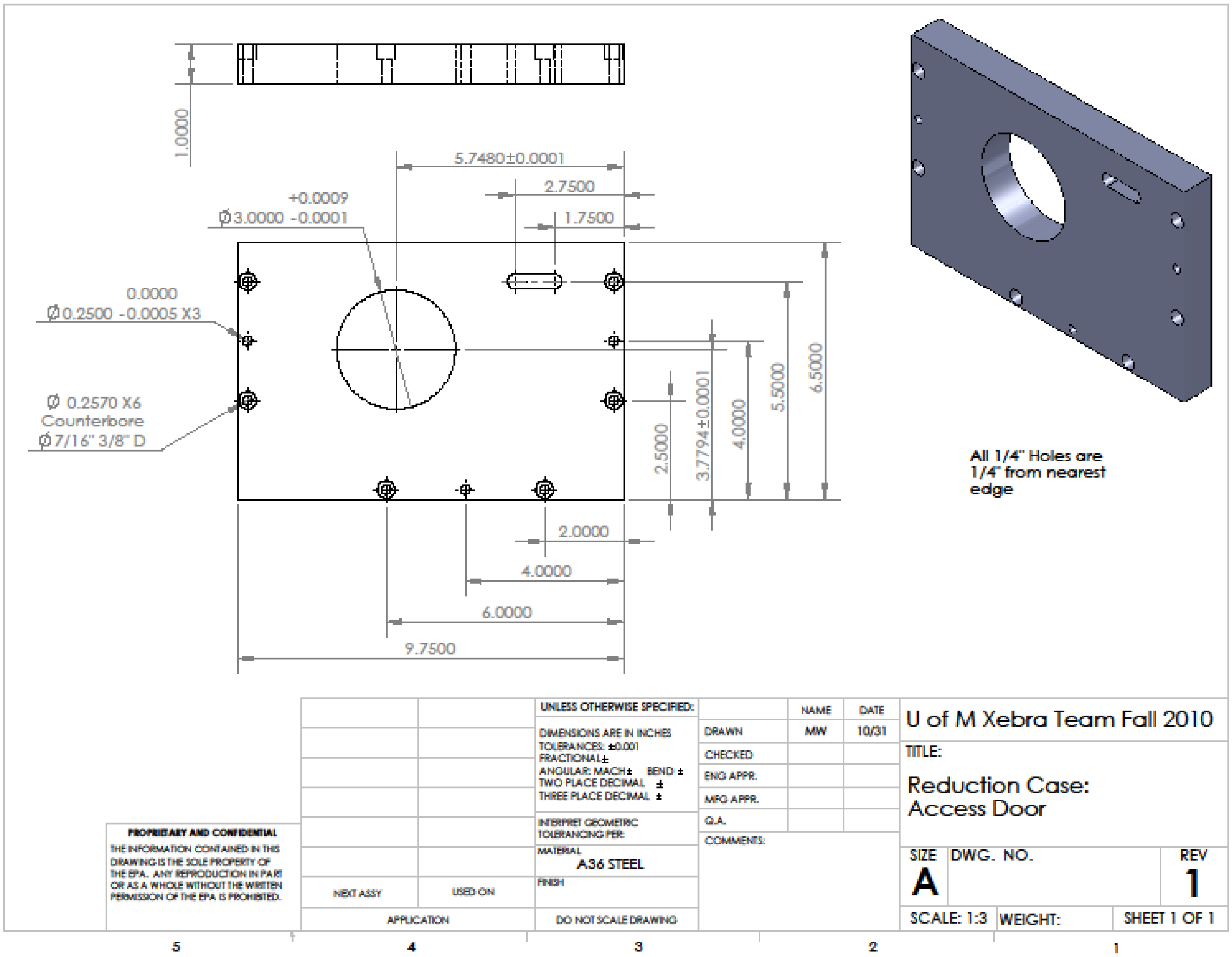

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All 1/4" Holes are
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U of M Xebra Team Fall 2010
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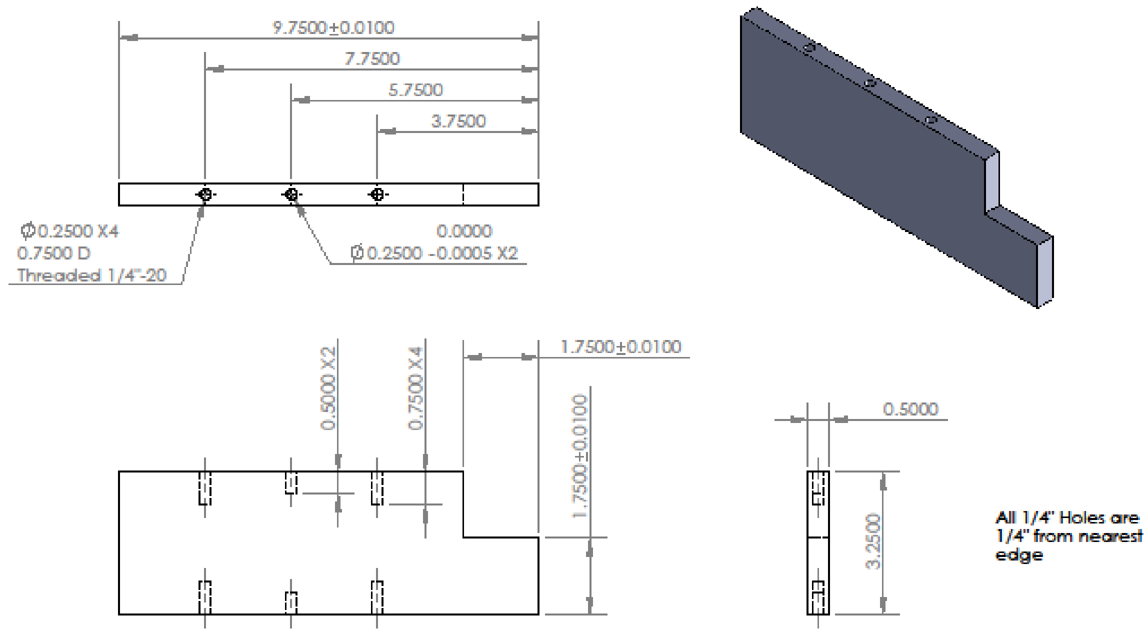

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COMMENTS:		

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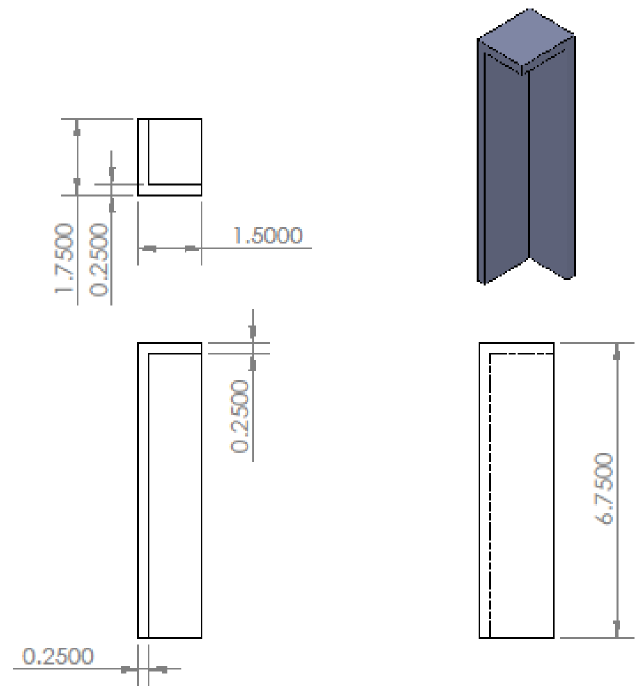

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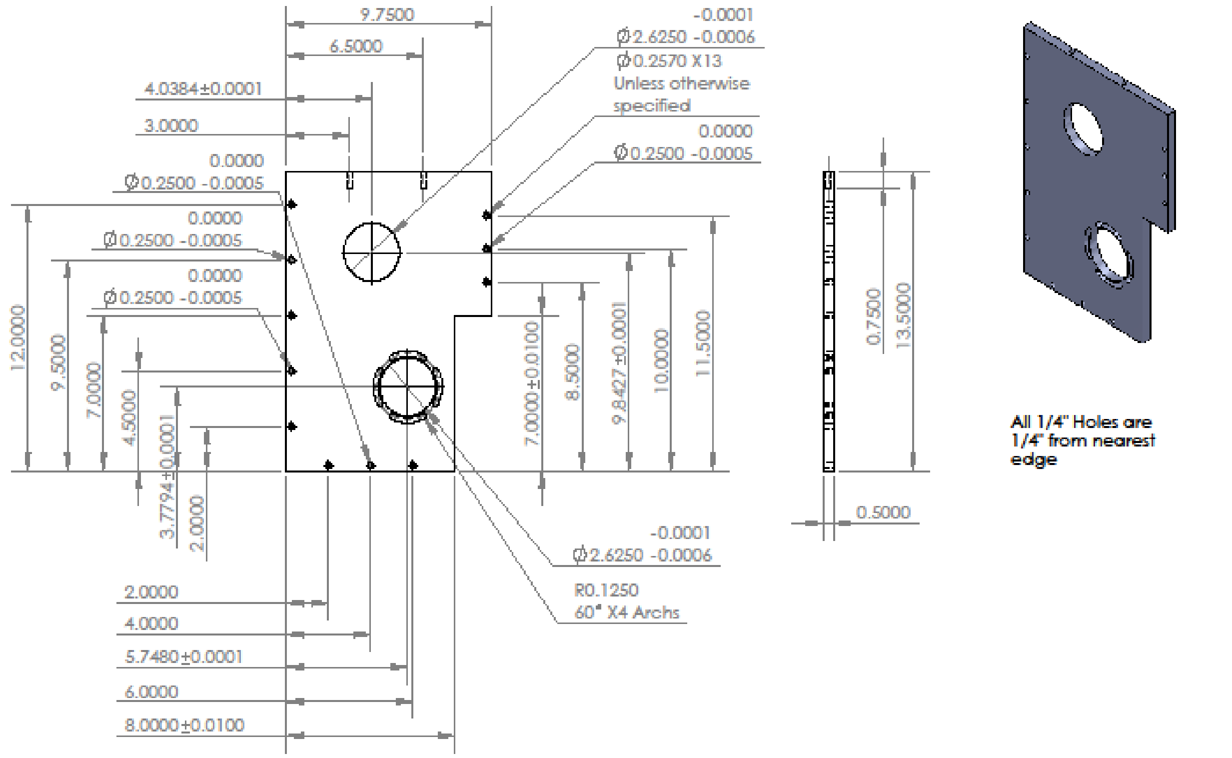

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INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

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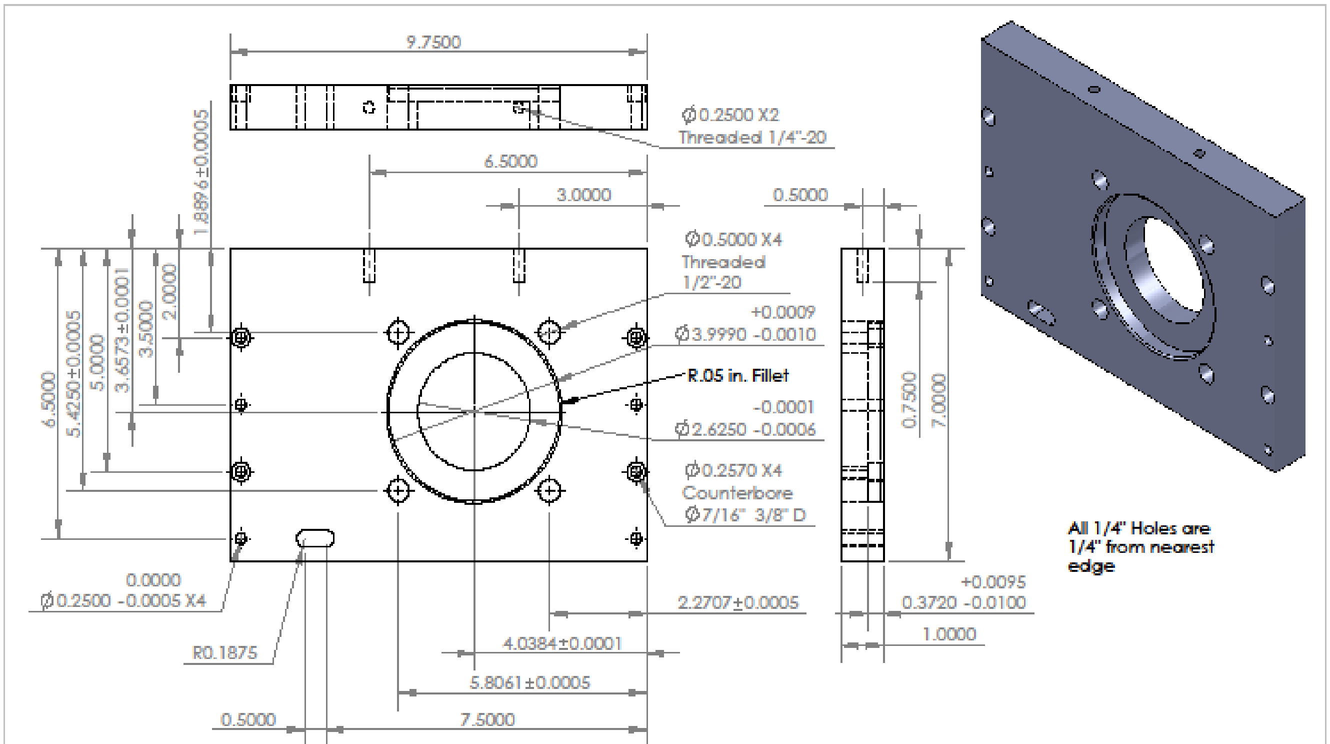

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INTERPRET GEOMETRIC TOLERANCING PER:

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DO NOT SCALE DRAWING

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COMMENTS:

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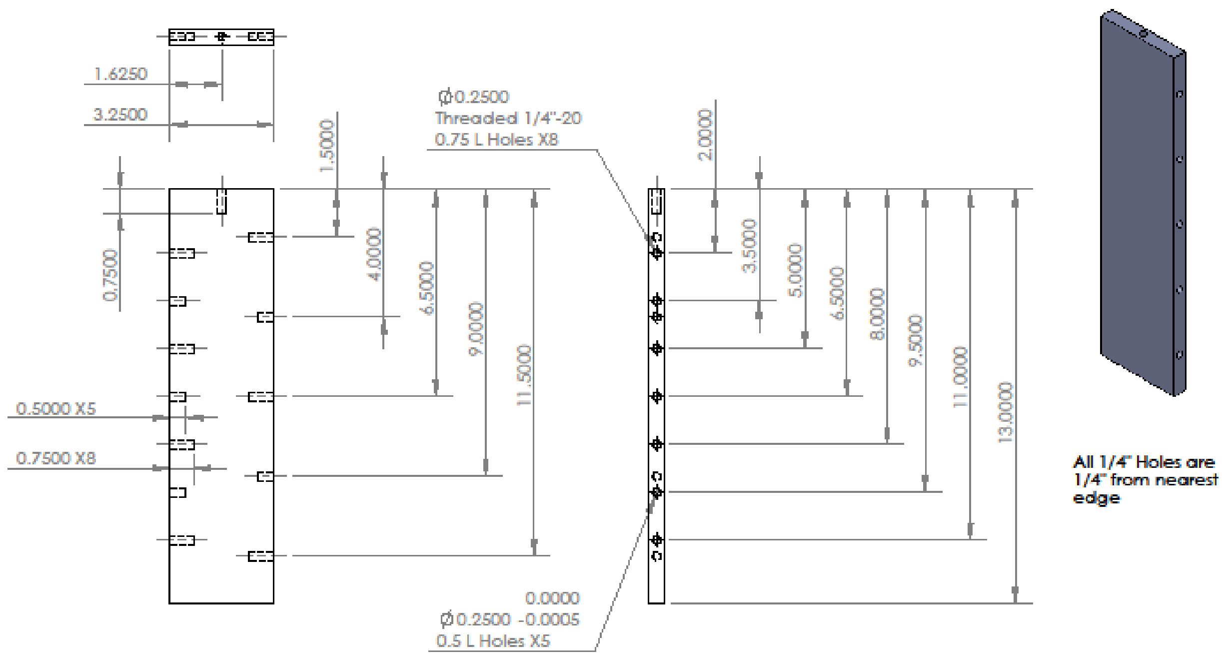

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THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
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FINISH

DO NOT SCALE DRAWING

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DRAWN	MW	10/31
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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

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TITLE:

Reduction Case: Side Back

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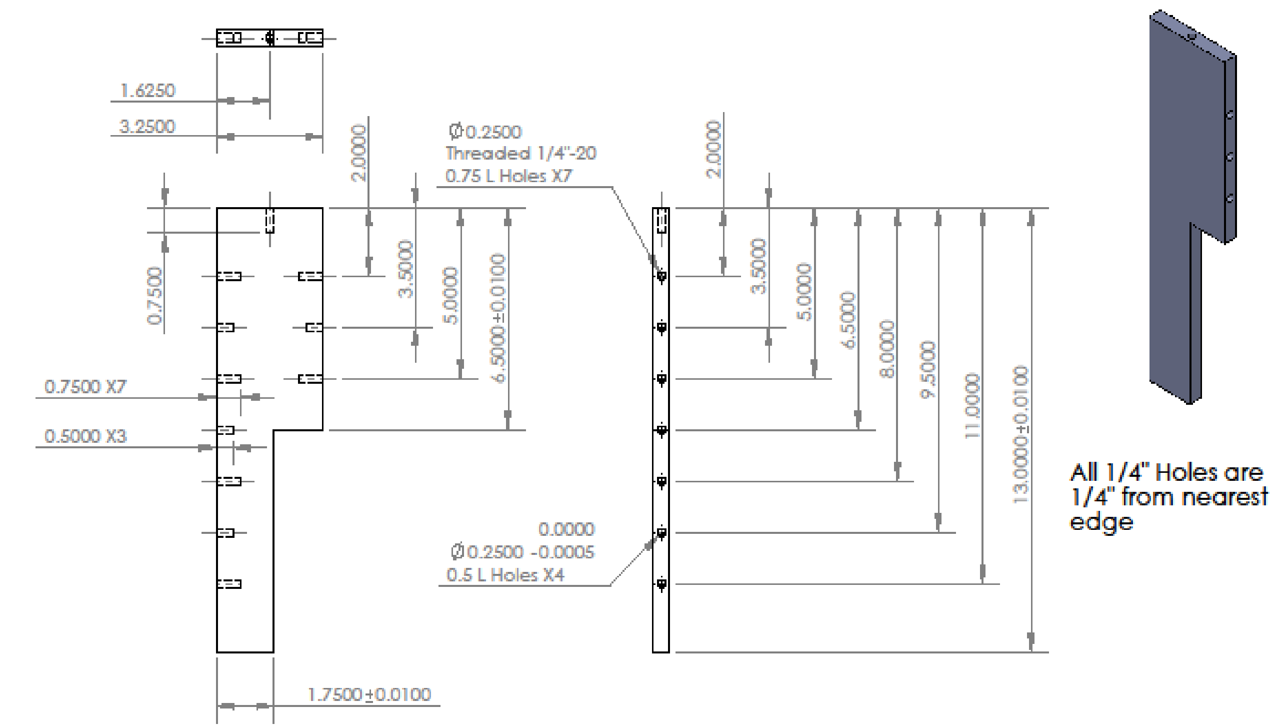

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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

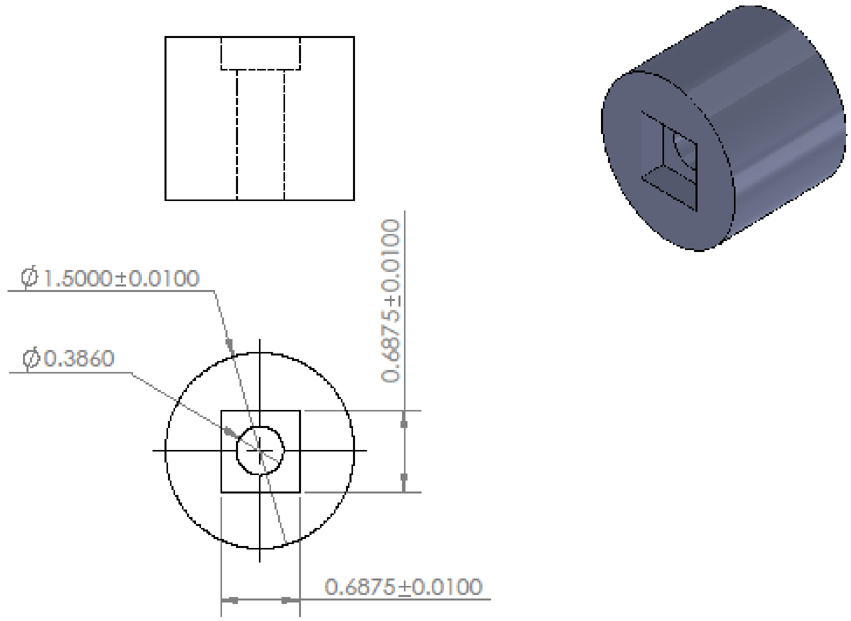
TITLE:

Reduction Case: Side Front

SIZE	DWG. NO.	REV
A		1

SCALE: 1:4 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



NEXT ASSY	USED ON	
APPLICATION		

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
UHMW Tension Plastic

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

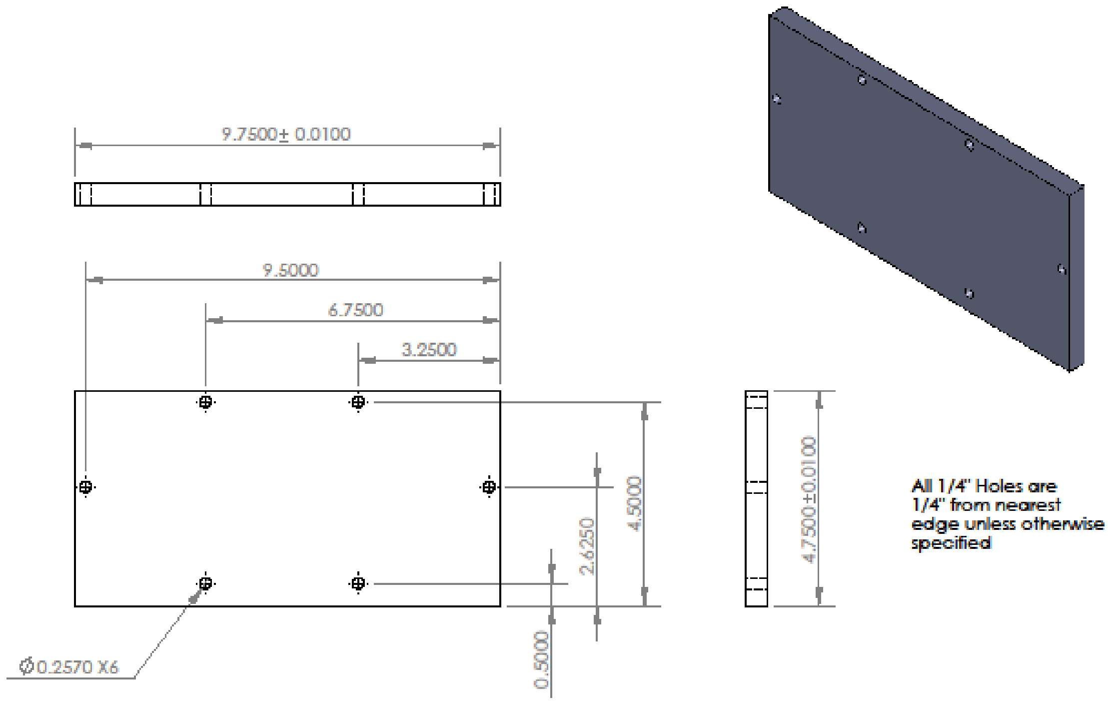
TITLE:

Reduction Case: Tensioner

SIZE	DWG. NO.	REV
A		1

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



PROPRIETARY AND CONFIDENTIAL

NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
A36 STEEL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

U of M Xebra Team Fall 2010

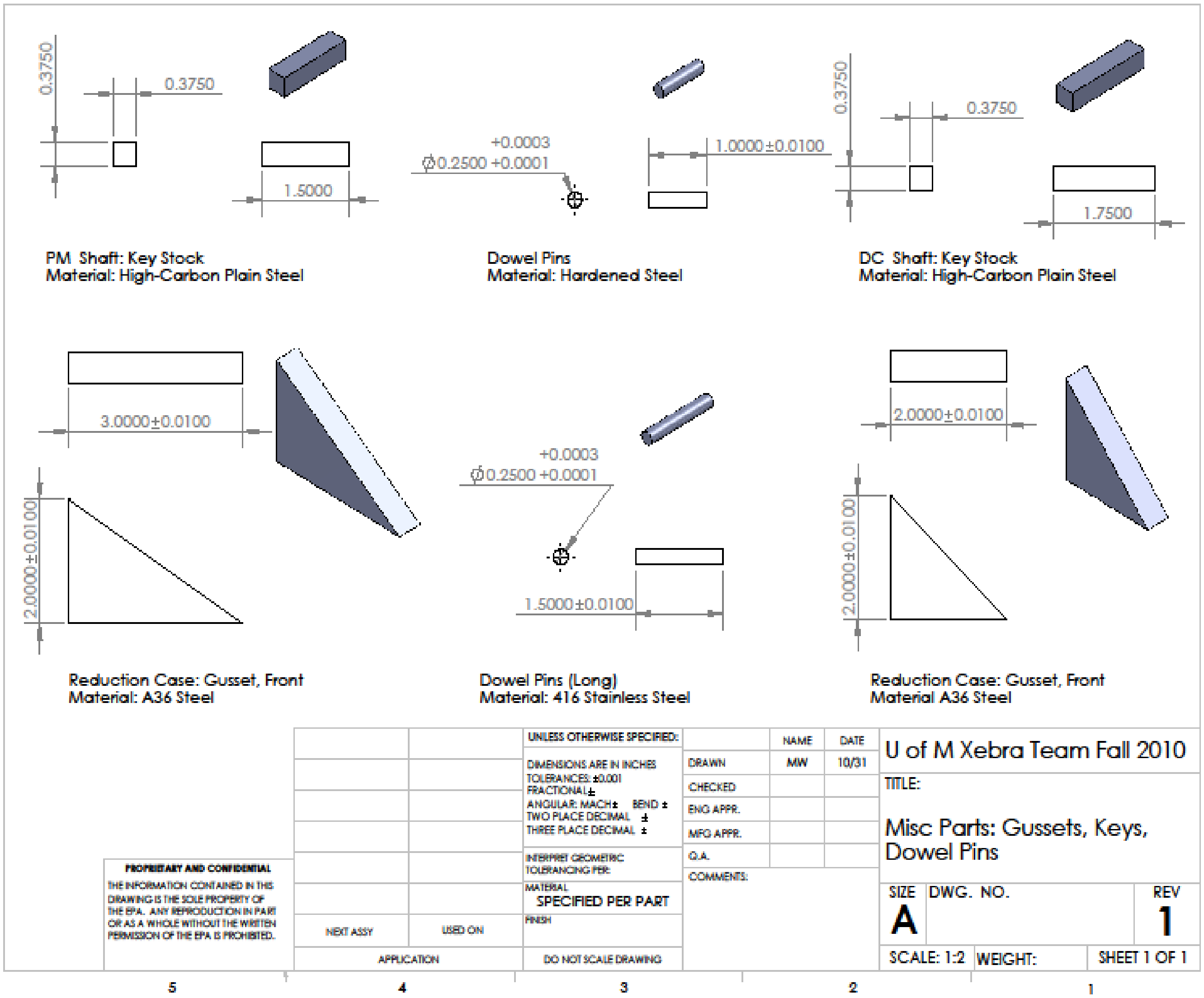
TITLE:

Reduction Case: Top Access Door

SIZE A | DWG. NO. | REV 1

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1


0.3750
0.3750
1.5000
PM Shaft: Key Stock
Material: High-Carbon Plain Steel
+0.0003
⌀0.2500 +0.0001
1.0000±0.0100
Dowel Pins
Material: Hardened Steel
0.3750
0.3750
1.7500
DC Shaft: Key Stock
Material: High-Carbon Plain Steel
3.0000±0.0100
2.0000±0.0100
Reduction Case: Gusset, Front
Material: A36 Steel
+0.0003
⌀0.2500 +0.0001
1.5000±0.0100
Dowel Pins (Long)
Material: 416 Stainless Steel
2.0000±0.0100
2.0000±0.0100
Reduction Case: Gusset, Front
Material A36 Steel
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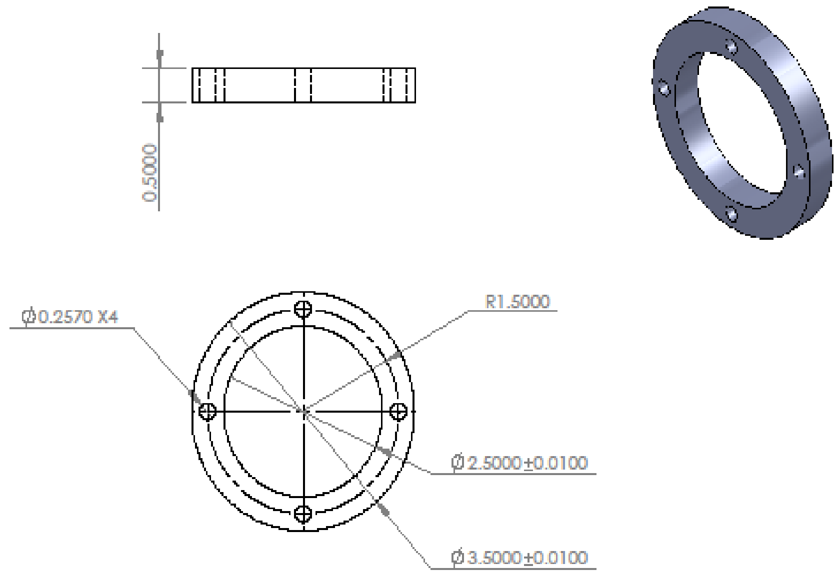
NEXT ASSY
USED ON
APPLICATION
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
SPECIFIED PER PART
FINISH
DO NOT SCALE DRAWING
NAME
DATE
DRAWN
MW
10/31
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
U of M Xebra Team Fall 2010
TITLE:
Misc Parts: Gussets, Keys, Dowel Pins
SIZE
A
DWG. NO.
REV
1
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
5
4
3
2
1



	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
1018 STEEL
FINISH

NEXT ASSY | USED ON
APPLICATION
DO NOT SCALE DRAWING

PROPRIETARY AND CONFIDENTIAL

U of M Xebra Team Fall 2010

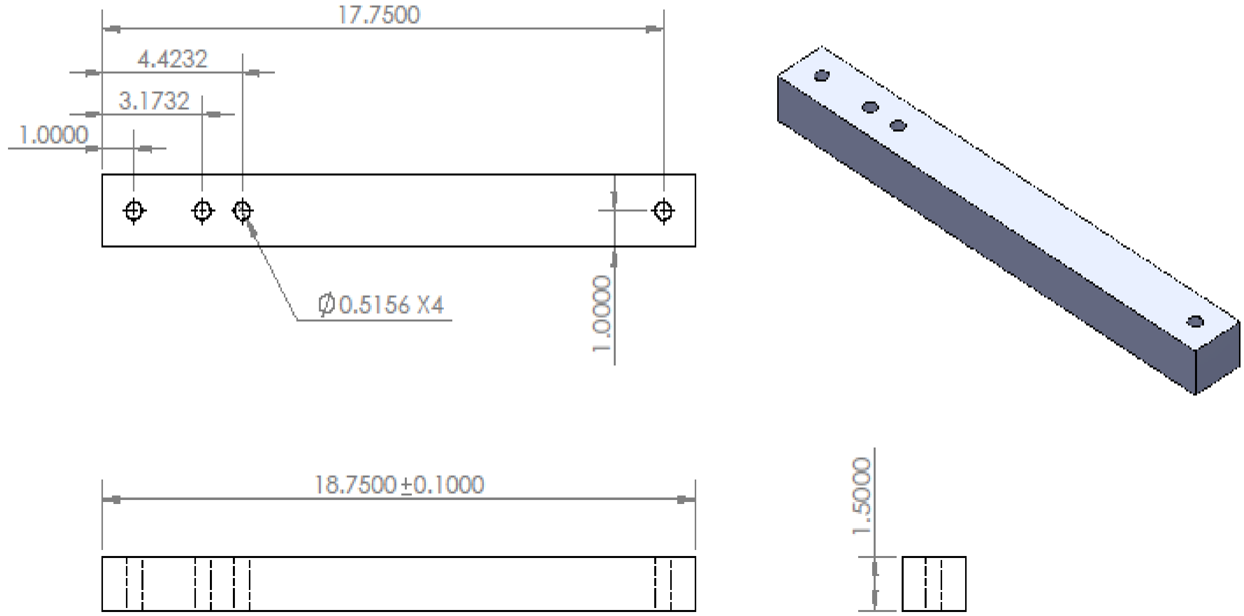
TITLE:

Spacer Mount

SIZE A | DWG. NO. | REV 1

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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NEXT ASSY | USED ON

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

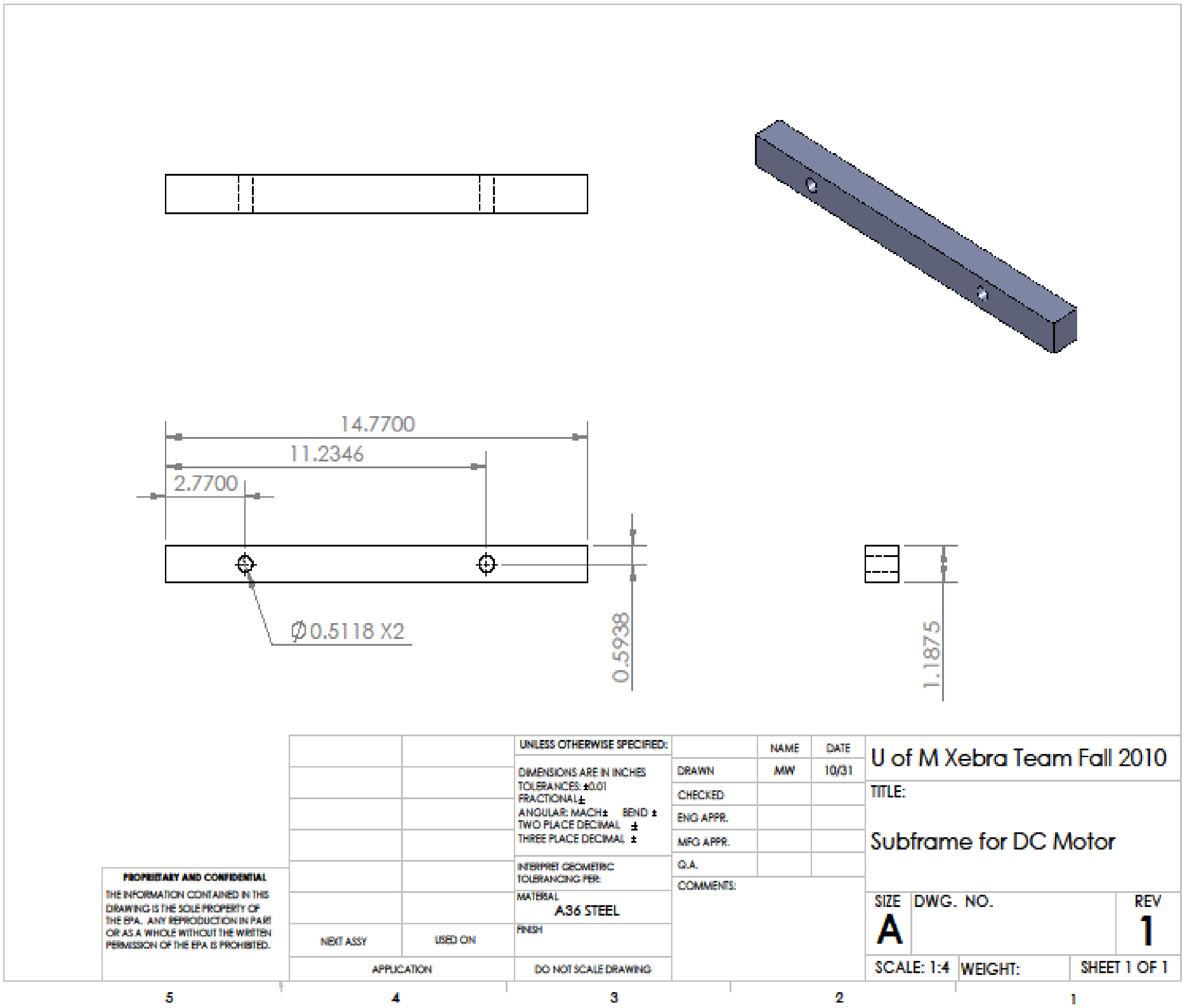
TITLE:

Subframe Back

SIZE A | DWG. NO. | REV 1

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1


14.7700
11.2346
2.7700
Ø0.5118 X2
0.5938
1.1875
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
A36 STEEL
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
MW
10/31
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
U of M Xebra Team Fall 2010
TITLE:
Subframe for DC Motor
SIZE
A
DWG. NO.
REV
1
SCALE: 1:4
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
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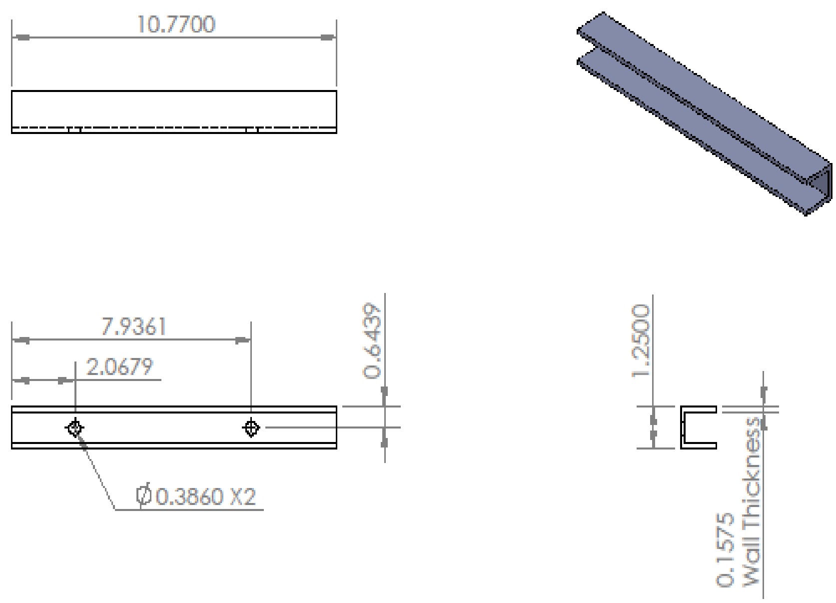
5
4
3
2
1



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: A36 STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

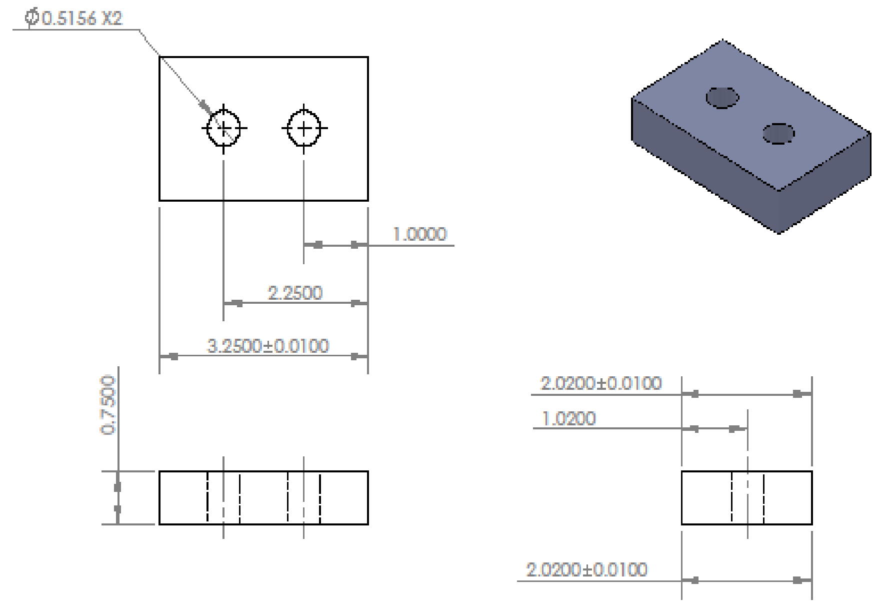
TITLE:

Subframe for DC Motor (U-Shaped)

SIZE	DWG. NO.	REV
A		1

SCALE: 1:4 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL A36 STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

U of M Xebra Team Fall 2010

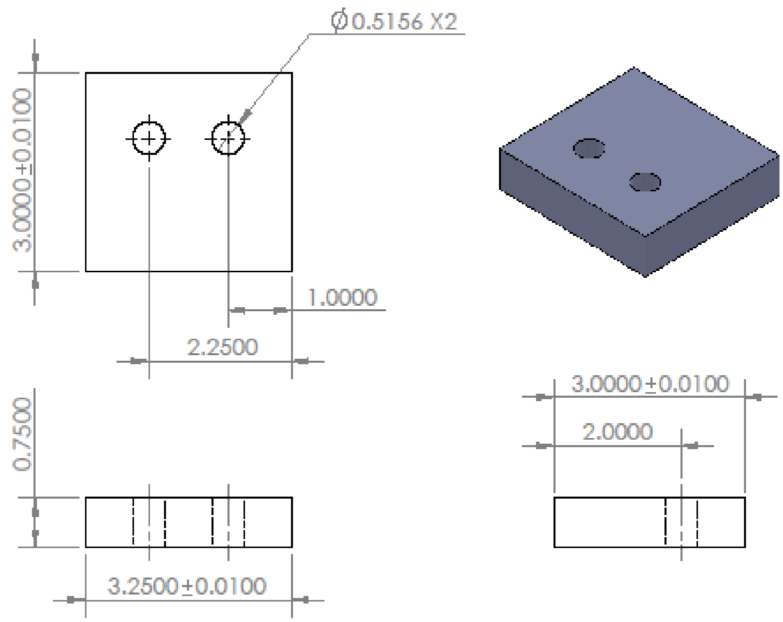
TITLE:

Reduction Case: Tab, Back

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL A36 STEEL	COMMENTS:		
	NEXT ASSY	USED ON	FINISH		
	APPLICATION		DO NOT SCALE DRAWING		

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U of M Xebra Team Fall 2010

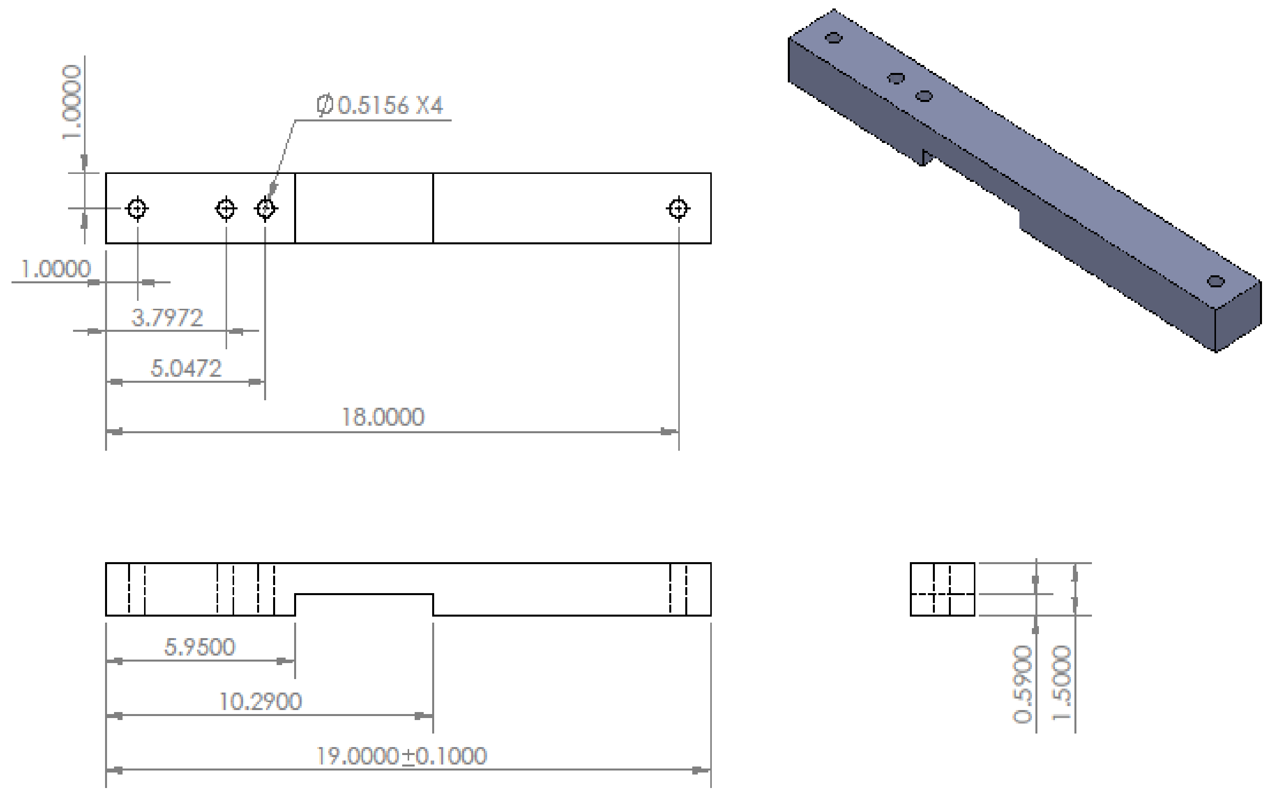
TITLE:

Reduction Case: Tab, Front

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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NEXT ASSY | USED ON
APPLICATION

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.01
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
A500 STEEL
FINISH
DO NOT SCALE DRAWING

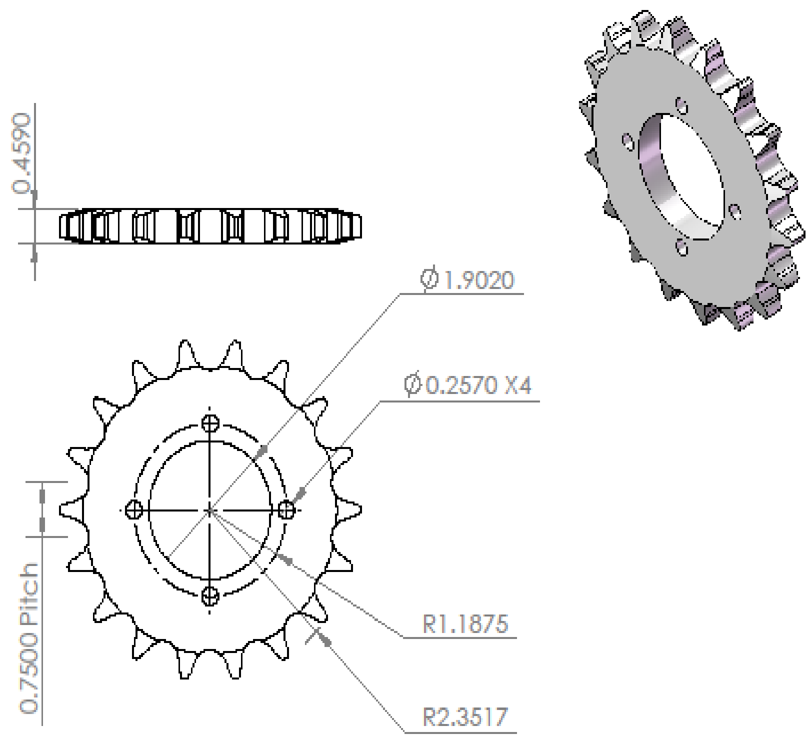
	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010
TITLE:
Sub-Frame: Front

SIZE A | DWG. NO. | REV 1
SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.001
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	MW	10/31
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

U of M Xebra Team Fall 2010

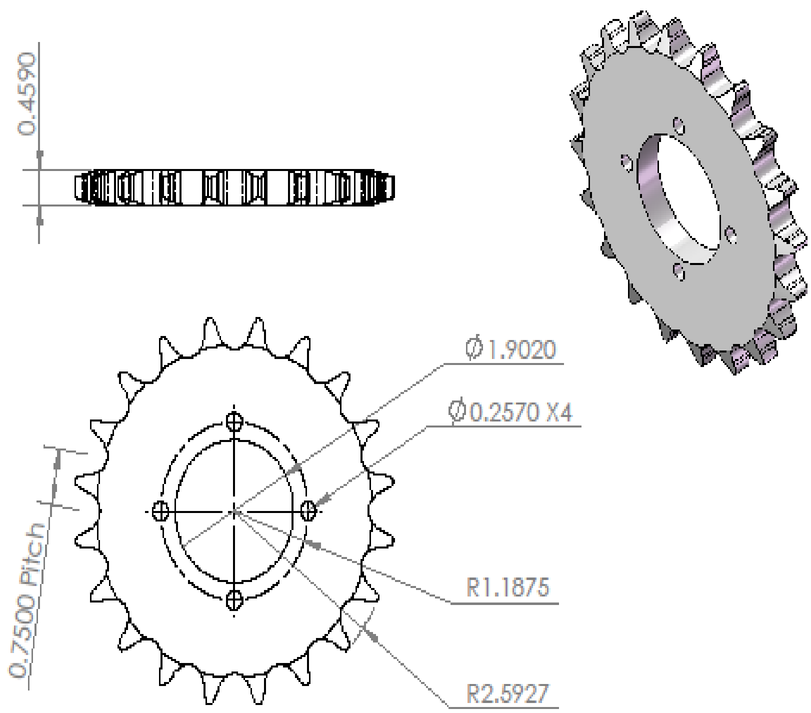
TITLE:

3.3:4.5 DC Sprocket

SIZE	DWG. NO.	REV
A		1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	U of M Xebra Team Fall 2010
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.001 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	MW	10/31	TITLE:
			CHECKED			
			ENG APPR.			3.6:4.5 DC Sprocket
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL STEEL	COMMENTS:			SIZE A DWG. NO. REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

6. Manufacturing

- *Sub-Frame – A36 Hot Rolled Steel*
  The sub-frame will be cut to 18.750 ± 0.100 inches using a band saw and the ends will be end-milled to exact dimensions. The four ½ inch through-holes will then be drilled using a 17/32 inch drill running at 720RPM for a clearance fit at the four locations on each rail.

- *Upper Shaft – 1018 Cold Rolled Steel*
  A two inch diameter stock piece of steel will be placed into a lathe. The lathe will be running around 220RPM and we will take off two 0.1 inch cuts to be at a diameter of 1.6 inches. Then progressively smaller cuts will be made until we reach 1.500 inches within tolerance. The shaft will then be cut to 4.500 ± 0.010 inches using the cutoff tool. The shaft will then be sent out to Riverside Spline and Gear to have a SAE B-B female spline broached into the end of the shaft. After the out sourced machining has been completed on the shaft, we will cut 3/8 inch keyways on opposite sides of the shaft using a 3/8 inch end mill running at 900 RPM to cut a 1.500 ± 0.010 inch long slot in the shaft.

- *Lower Shaft – 1018 Cold Rolled Steel*
  A two inch diameter stock piece of steel will be placed into a lathe. The lathe will be running around 220RPM and we will take off two 0.1 inch cuts to be at a diameter of 1.6 inches. Then progressively smaller cuts will be made until we reach 1.500 inches. The shaft will then be cut to 7.250 ± 0.010 inches using the cutoff tool. The shaft will then be sent out to have a 6-tooth spline cut into the end of the shaft. At this point we are still trying to determine the exact spline that exists in the female shaft of the DC motor. There is no documentation from previous reports and no documentation from the vehicle manufacturer either. After the out sourced machining has been completed on the shaft, we will cut 3/8 inch keyways on opposite sides of the shaft using a 3/8 inch end mill running at 900 RPM to cut a 1.875 ± 0.010 inch long slot in the shaft.

- *Upper Hub – 1018 Cold Rolled Steel*
  A 4 inch diameter stock piece of steel will be placed into a lathe. The lathe will be running around 110RPM and we will take off two 0.1 inch cuts to be at a diameter of 3.6 inches. Then progressively smaller cuts will be made until we reach 3.500 ± 0.010 inches. Additionally, 1 inch of the end of the hub will need to be turned down to 1.75 inches. This will be achieved by taking passes at 0.1 inch and progressively taking smaller cuts until the hub is exactly 1.750 ± 0.001 inches. The hub will then be bored out 1.5 inches. This will be achieved by first drilling a ¾ inch hole into the shaft with a ¾ inch drill bit. From here, 0.1 inch passes with a boring tool will take off material till the inner diameter is 1.4 inches. Finally progressively smaller passes will be made till the inner diameter is 1.500 inches within tolerance. The shaft will then be cut to 2.500 ± 0.001 inches using the cutoff tool. Now, the hub will be placed into a mill and secured. The center of the bored hole will be found using a dial indicator. Four ¼ inch diameter holes will need to be drilled and two will be threaded for attaching the sprockets. The two holes to be tapped will first be drilled using a 7/32 inch hole at 1400 RPM at the two locations in the drawing. Then the two holes will be hand tapped using a ¼-20 tap. The other two holes, to be pressed with dowel pins, are first drilled to 15/64 inch at 1400 RPM and finally reaming the hole with a 0.2495 reamer at around 900RPM.The machining will be concluded by cutting 3/8 inch keyways using the arbor press.

- *Lower Hub – 1018 Cold Rolled Steel*
  A 4 inch diameter stock piece of steel will be placed into a lathe. The lathe will be running around 110RPM and we will take off six 0.1 inch cuts to be at a diameter of 2.8". Then progressively smaller cuts will be made until we reach 2.750 ± 0.001 inches. 1 inch of the end of the hub will need to be turned down to 1.902 inches. This will be achieved by taking three passes at 0.1 inch and progressively taking smaller cuts until the hub is exactly 1.902 inches within tolerance. The a ¼ inch of the hub will be turned down to 1.750 ± 0.001 inches using the same method as before. The hub will then be bored out 1.5 inches. This will be achieved by first drilling a ¾ inch hole into the shaft with a ¾ inch drill bit. From here, 0.1 inch passes with a boring tool will take off material till the inner diameter is 1.4 inches. Finally progressively smaller passes will be made till the inner diameter is 1.500 inches within tolerance. The shaft will then be cut to 1.750 ± 0.001 inches using the cutoff tool. Now, the hub will be placed into a mill and secured. The center of the bored hole will be found using a dial indicator. Two ¼ inch diameter holes and two #10 holes need to be drilled and tapped for attaching the sprockets. The two ¼ inch holes are for the bolts and the 10-24 holes are for the shoulder bolts. This will be achieved by first drilling a 7/32 inch

hole at 1400 RPM at the four locations in the drawing. Then the two holes will be hand tapped using a ¼-20 tap. The two remaining holes will be machined by drilling a 0.1610 inch hole at around 1600 RPM and finally tapping the holes using a 10-24 hand tap. The machining will be concluded by cutting 3/8 inch keyways using the arbor press.

- *Spacer – 1018 Cold Rolled Steel*
  A 4 inch diameter stock piece of steel will be placed into a lathe. The lathe will be running around 110RPM and we will take off two 0.1 inch cuts to hit a diameter of 3.6 inches. Then progressively smaller cuts will be made until we reach 3.500 ± 0.010 inches. The spacer will then be bored out to 2.5 inches. This will be achieved by first drilling a ¾ inch hole into the shaft with a ¾ inch drill bit. From here, 0.1 inch passes with a boring tool will take off material till the inner diameter is 2.4 inches. Finally progressively smaller passes will be made till the inner diameter is 2.500 ± 0.010 inches. The shaft will then be cut to 0.500 ± 0.001 inch length using the cutoff tool. Now, the spacer will be placed into a mill and secured. The center of the bored hole will be found using a dial indicator. Four ¼ inch diameter through-holes will be drilled using a F drill running at 1400 RPM. This spacer is what connects the Reduction Housing to the DC gearbox.

- *Tab and Gusset – A36 Hot Rolled Steel*
  The tab and gussets will be roughly cut out from ¾ inch and ½ inch stock, respectively, with ½ inch extra around the edges using a band saw. They will be milled to exact dimensions given on the drawing using a 1 inch end mill running at 600 RPM taking off a max of 0.1 inches of material at a time. The two tabs will then have two ½ inch holes drilled on each tab to allow the ½-13 bolts to attach the reduction housing to the sub-frame. This will be achieved using a 17/32 inch drill bit running at 720 RPM. After these holes are drilled, the gussets will be welded to the top of the tabs. At this point we have decided that TIG welding will be our method of choice for welding the parts together. TIG can provide a small uniform weld that has a small heat affected zone when compared to other methods. Since this is a critical junction in our system we feel that TIG will provide us with the strongest joint. Since we do not have experience with TIG welding we plan on having Bob Coury assist us with this process. The manufacturing of this tab and gusset assembly will continue in the Reduction Housing – Side Plate section below.

- *Reduction Housing – Cut Out – A36 Hot Rolled Steel*
  The cut out is an assembly of three ¼ inch plates with an overall dimension of 1.750 ± 0.001 by 1.500 ± 0.001 by 6.750 ± 0.001 inches. These three plates will initially be roughly cut to size using a band saw. Then they will be end milled to exact dimensions using a 1 inch end-mill running at 600RPM taking of a max of 0.1 inches of material at a time. The three plates will then be welded together on the three edges as shown in the drawing. We decided to use TIG welding for this process for the same reasons as discussed in the tab and gusset section. The manufacturing of the cut out assembly will continue in the Reduction Housing – Side Plate Front section below.

- *Reduction Housing – Pump Face Plate – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the 1 inch stock using a band saw. The piece will then be end-milled to 9.750 ± 0.001 by 7.000 ± 0.001 inches using a 1 inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. Through holes for the ¼ inch bolts need to be drilled at various locations on the edges of the plate. This will be completed using a F drill running at 1400 RPM. These through-holes will be counter-bored to 7/16 inch diameter by 3/8 inch deep. This will be achieve by using a 7/16 inch bit running at 950RPM. In addition, on the top edge of the plate, holes need to be drilled and tapped to allow the top plate to be attached. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. Finally, they will be tapped by hand using a ¼-20 tap.

- *Reduction Housing – Pump Door Plate – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the 1 inch stock using a band saw. The piece will then be end-milled to 9.750 ± 0.001 by 6.500 ± 0.001 inches using a 1 inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. Through holes for the ¼ inch bolts need to be drilled at various locations on the edges of the plate. This will be completed using a F drill running at 1400 RPM. These through-holes will be counter-bored to 7/16 inch diameter by 3/8 inch deep. This will be achieve by using a 7/16 inch bit running at 950RPM. In addition, on the top edge of the plate,

holes need to be drilled and tapped to allow the top plate to be attached. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. Finally, they will be tapped by hand using a ¼-20 tap.

- *Reduction Housing – DC Face Plate – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the ½ inch stock using a band saw. The piece will then be end-milled to 9.750 ± 0.001 by 13.500 ± 0.001 using a 1inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. A 1.750 ± 0.010 by 7.000 ± 0.010 inch piece needs to be removed from the lower corner. This piece will be cut out by reorienting the plate in the mill and using the same end-mill. Through holes for the ¼ inch bolts need to be drilled at various locations on the edges of the plate. This will be completed using a F drill running at 1400 RPM. These through-holes will be counter-bored to 7/16 inch diameter by 3/8 inch deep. This will be achieve by using a 7/16 inch bit running at 950RPM. In addition, on the top edge of the plate, holes need to be drilled and tapped to allow the top plate to be attached. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. Finally, they will be tapped by hand using a ¼-20 tap.

- *Reduction Housing – Side Plate Rear – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the ½ inch stock using a band saw. The piece will then be end-milled square with a 1 inch end mill running at 600RPM taking off a max of 0.1 inches of material at a time. This milling will not be to specific dimensions though; extra material will be remain on purpose. At this point, the previously manufactured tab and gusset assembly will be welded to the plate in the location specified in the dimensioned drawing. The welding method using will be TIG and was chosen for the same reason as stated in the tab and gusset section. After completing this welding process, the side plate will be end-milled to 13.000 ± 0.001 by 3.250 ± 0.001 inches using a 1inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. The "final" machining of the side plates was completed after the welding to make sure that even if the plates were warped during the welding that they would be machined square. The next task is to drill and tap the holes on the two long edges and the top edge as show in the dimensioned drawing. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. Finally, they will be tapped by hand using a ¼-20 tap.

- *Reduction Housing – Side Plate Front – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the ½ inch stock using a band saw. The piece will then be end-milled square with a 1 inch end mill running at 600RPM taking off a max of 0.1 inches of material at a time. This milling will not be to specific dimensions though; extra material will be remain on purpose. At this point, the previously manufactured tab and gusset assembly will be welded to the plate in the location specified in the dimensioned drawing. The welding method using will be TIG and was chosen for the same reason as stated in the tab and gusset section. After completing this welding process, the side plate will be end-milled to 13.000 ± 0.001 by 3.250 ± 0.001 inches using a 1inch diameter end mill. The plate will also have a 1.625 ± 0.001 by 6.500 ± 0.001 inches section removed to allow for the reduction housing cut out to be attached. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. The next task is to attach the previously manufactured reduction housing cut out. This assembly will be attached by TIG welding one edge to the edge of the side plate. The reduction housing cut out assembly is not a load bearing plate and therefore the rigidity of the weld is not as big a concern as that of the tab and gusset assembly. The final task is to drill and tap the holes on the two long edges and the top edge as show in the dimensioned drawing. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. The holes will then be tapped by hand using a ¼-20 tap.

- *Reduction Housing – Top Plate – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the ½ inch stock using a band saw. The piece will then be end-milled to 9.750 ± 0.010 by 4.750 ± 0.010 inches using a 1inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. Through

holes for the ¼ inch bolts need to be drilled at various locations on the edges of the plate. This will be completed using a F drill running at 1400 RPM to produce the clearance fit.

- *Reduction Housing – Bottom Plate – A36 Hot Rolled Steel*
  This plate will be roughly cut to size with ½ inch extra around the edges from the ½ inch stock using a band saw. The piece will then be end-milled to 9.750 ± 0.010 by 3.250 ± 0.001 inches using a 1 inch diameter end mill. The mill will be running at around 600 RPM and will take off 0.1 inch of the material on each pass. When getting closer to the actual dimension, progressively smaller passes will be taken until the desired dimension is reached. A 1.750 ± 0.010 by 1.500 ± 0.010 inches piece needs to be milled out using the same 1 inch tool to allow for the reduction housing cut out to fit. The long edges of the plate need to be drilled and tapped to allow the Pump and DC Plates to be attached. After reorienting the plate in the mill and referencing to the edge a 7/32 inch bit running at 1400 RPM will be used to drill these holes. Finally, they will be tapped by hand using a ¼-20 tap.

- *Sprockets – Upper Sprocket – 25 Tooth*
  The 25-tooth sprocket has a small amount of machining to be completed. This sprocket is on the upper shaft and needs to have a 1.750 ± 0.010 inch hole bored into the center of it. This will be accomplished by mounting the sprocket onto a mill. If available, a 1-3/4 inch diameter bit will be used to drill the hole at around 200RPM. If this bit is not available, a simple tool path will be written on the CNC-mill to cut out this hole using a 1 inch diameter end mill running at 600 RPM. After completing the bore, four holes need to be drilled; two clearance-fit holes for the bolts and two interference-fit holes for the dowel pins to transmit torque. The two clearance holes will be drilled using a F drill running at 1400 RPM. The interference-fit holes will be achieved by first drilling a 15/64 inch hole at 1400 RPM and finally reaming the hole with a 0.2495 reamer at around 900RPM.

- *Sprockets – Lower Sprockets – 14-, 18-, 20-Tooth*
  The lower sprockets also have a small amount of machining to be completed. These sprockets are on the lower shaft and need to have a 1.902 ± 0.001 inch hole bored into the center of it. This will be accomplished by mounting the sprocket onto a mill. If available, a 1.75 inch diameter bit will be used to drill the hole at around 200RPM and bored to exact diameter with a boring bar. Another option is to write a simple tool path for the CNC-mill to cut out this hole using a 1 inch diameter end mill running at 600 RPM. After completing the bore, four holes need to be drilled; two clearance-fit holes for the bolts and two close-fit holes for the shoulder bolts to transmit torque. The two clearance holes will be drilled using a F drill running at 1400 RPM. The close-fit holes will be achieved by using an E drill at 1400 RPM for a 0.250 inch hole that will be very close to the slightly under-sized shoulder bolts.

- *Tensioner – Ultra High Molecular Weight Plastic*
  The tensioner is a very simply machined piece out of 1 inch purchased stock. The two tensioners will be cut to length on the band saw and end-milled flat using a 1 inch diameter end-mill running at 600RPM. The tensioner has a ¼ inch hole to allow a bolt to attach it rigidly to the reduction housing. This hole will be drilled with an F drill running at 1400RPM.

- *Reduction Housing Machining*
  To finish the machining on the Reduction Casing, the case must first be assembled using ¼-20 bolts and torqued to 60 in-lbs. The entire box will then be mounted on a mill and holes for the ¼ inch dowel pins will be drilled at various locations over the box. These holes will be drilled with a 15/64 inch hole at 1400 RPM and then reamed with a 0.2495 reamer at around 900RPM. The dowel pins will be inserted to ensure the box remains extremely rigid and can be realigned after the box is disassembled and reassembled. The 4 inch pilot hole for the pump motor, the 3 inch diameter hole for the lower bearing mount, and the 2-5/8 inch diameter hole for the other three bearing mounts all need to be drilled still. We have decided that because the alignment of these is so critical, we will be outsourcing this work to a machine shop. The capabilities of student shop simply do not allow us to achieve the tolerances in alignment that we are looking for. We are currently talking with the EPA's in-house machine shop and investigating potential machine shops to complete these tasks for us in a timely manner.

Safety Concerns During Manufacturing

Milling – Before starting a milling operation the user must ensure that all of the axis of motion have been locked down. A user must also be sure of the speed at which he is running the mill at to prevent damaging the part, the mill, or even the user. When inserting the collar with the mill bit, the user must ensure that the cutting tool has been tighten down to prevent any loosening from occurring during milling. Finally, when a user is milling a part he must be very aware of the rotating spindle. This is where all of the power of the mill is being applied to and therefore is the primary area of concern. If anything were to get caught in this area while spinning it could be catastrophic.

Lathe – While the safety hazards of a lathe are similar to a mill, they are increased because of the larger rotating piece. Because the part itself is rotating there is a much larger chance of something getting caught in the lathe while rotating. The user must be extremely aware of clothing and other items while machining. Finally, because the part is rotating at very high speeds, the user must be conservative about the amount material being removed. Calculations can be made to determine the cut depth, but if too much is removed, parts can be ejected from the lathe since it is traveling at such a high speed.

Welding – Though we will not be performing the welding processes on our own in the shop, there are plenty of safety concerns. Welding deals with extremely high voltages and temperatures. Caution must be taken when handling the equipment, ensuring that the user does not put anyone in danger. Because welding is physically melting metal the temperature of a piece can raise greatly. Users must be aware of the temperature of a part and cautiously move materials making sure to not burn oneself. Users must also be aware of burns from radiation of the arc. Because it is extremely bright sun burn can actually occur and therefore welding coats and gloves must be used.

Pressing – When pressing a bearing into place, the arbor press will be used. This machine can apply extremely high forces onto objects and therefore must be used with consideration. Fingers are the biggest concern and must always be out of harms when forcing a bearing into a part. Users must be very aware while applying force, looking for points of failure that could harm users in the area.

7. Assembly

- *Sub-Frame Installation and Assembly*
  The sub-frame mounts that exist on the frame of the Blue Xebra are not in the exact locations that we need them to be in. We will be removing two of the mounts and repositioning them to fit our new sub-frame. Since we have not had access to the Blue Xebra, we have not been able to make very accurate measurements for items such as these mounts. Therefore we do not know exactly where we will be remounting these supports yet. We have designed the sub-frame rails for a specific length and this will help determine where the mounts will be welded. These mounts will be welded back onto the frame using either MIG or TIG welding. We have not made a decision on which to use here because despite this being a critical mount, it is a relatively large piece to work on and the heat-affected zone is not of great concern. After reattaching these mounts, the two sub-frame rails will be mounted on top of new rubber dampers that we are still investigating. The DC motor and existing gearbox needs to be integrated with this new frame though. To accomplish this we will be cutting two supports off the old sub-frame to use on our new frame. We are trying to reuse many of these components because they are specifically designed to work with this DC motor system and this reduces our manufacturing load. This part of manufacturing actually needs to be completed early on and therefore we are trying to get the Blue Xebra as early as possible. After the DC motor and existing gearbox has been installed though, the Xebra is ready to receive the new powertrain.

- *Upper Shaft Assembly*
  The upper shaft assembly is essentially building up the shaft with the hub and sprocket. The first task is to press the keys into the keyways on the upper shaft using an arbor press. After this has been completed the upper hub is slid onto the shaft and pressed over the keys securing it to the shaft. After the hub has been secured, the 25-tooth sprocket is attached to the hub. The sprocket is attached by two ¼-20 bolts and two ¼ inch dowel pins. The dowel pins will transfer the torque while the bolts keep the sprocket pressed against the hub. The dowel pins are pressed through the sprocket and into the hub using an arbor press. Then the ¼-20 bolts are inserted and torqued to 60 in-lbs to ensure to correct clamping load exists in the system. This concludes the upper shaft assembly

- *Lower Shaft Assembly*
  The lower shaft assembly is essentially building up the shaft with the hub and sprocket. The first task is to press the keys into the keyways on the lower shaft using an arbor press. After this has been completed the lower hub is slid onto the shaft and pressed over the keys securing it to the shaft. After the hub has been secured, one of the three potential sprockets is attached to the hub. The sprocket is attached by two ¼-20 bolts and two ¼ inch shoulder bolts. The shoulder bolts will transfer the torque while the bolts keep the sprocket pressed against the hub. The shoulder bolts are fit through the sprocket and screwed into the lower hub. Then the ¼-20 bolts are inserted and torqued to 60 in-lbs to ensure to correct clamping load exists in the system. This concludes the lower shaft assembly

- *Final Assembly*
  After all machining has been complete only a small amount of assembly remains. The front face of the reduction housing will be removed to allow access to the bearing mounts. The two bearings for the upper shaft and the one bearing for the lower shaft near the DC motor all need to be press fitted into position. In addition, the outer race of the tapered roller bearing used with the lower shaft is also pressed into the housing. These bearings will be pressed in using an arbor press as well as some scrap rounds to ensure we press the outer race when installing. After the bearings have been press fit, the upper shaft is pressed into the Pump plate while the lower shaft is pressed into the DC plate. At this point the two tensioners can be easily installed with the 3/8-16 hardware. The pump plate and DC plate will then be aligned and pressed together finally connecting the reduction housing. The chain will be installed onto the sprockets from the open top. The chain will be adjusted to the correct length using the extra links we have purchased. The tension of the chain will also be adjusted by arranging the placement of the tensioners on the inner face of the Pump plate. At this point all of the ¼-20 bolts will be reinstalled into the reduction housing and the assembly will be complete.

Integration into Blue Xebra

The final integration into the Blue Xebra can be completed with or without the hydraulic pump motor. At this point we are unsure of the availability of the pump motor, but the reduction housing itself can be integrated into vehicle alone. The reduction housing will be installed onto the sub-frame and the spacer and lower shaft will be aligned with the DC gearbox. After the gearbox has been correctly aligned the four ½-13 bolts will be inserted through the tabs and torqued to 40 ft-lbs. This concludes the final integration of the reduction housing.

As discussed in the manufacturing plan, the majority of our machining and assembly will be completed in the student machine shop. The two shafts will have male and female splines shaped into them and will be sent out to a machine shop to have these splining operation performed. The other task that will not be performed in the student shop is the machining of the pump pilot and the two bearing holes.

8. Design Testing and Validation Plan

Once we have successfully manufactured and assembled our final design, we would like to conduct several tests to validate that our design meets each of our customer requirements and engineering specifications. All our validation will be performed on a completed final design, and not during different points in the manufacturing process.

After speaking with our sponsors from the EPA and our customers from the university, we have confirmed that only the testing of our fabricated design and not the entire hydraulic system-electric system is to be validated. As the hydraulic-electric hybrid system will not be completely finished within our timeline, we will not be affecting the overall performance of the car and testing on the EPA dynamometer will not be necessary. This also eliminates us testing with high pressure fluids, reducing the overall safety concern of our validation tests by a large amount.

Another concern that was just brought up in our sponsor meeting was the possibility of not receiving the actual hydraulic pump/motor to be mounted. The hydraulic pump/motor seemed to be over torquing in its preliminary test at the EPA. Thus as a back up our team is planning on designing a small frame that will fit the correct bolt locations designed for the pump/motor and add to it comparable weights (weight of pump/motor) to simulate the presence of the pump for validation purposes.

One of the main goals of our project was that the system transmits power and torque safely and when it is subjected to disturbances. Our team calculated that at maximum RPM the system would experience a maximum of 206 ft-lb of load torque. Thus we want to make sure that the system can withstand this torque without fracturing and can still transmit the power and torque. Our final design consists of a roller chain and sprocket system in the reduction case to transmit power between the two systems. We plan on testing this using an appropriate sized torque wrench and applying the max torque to our transfer case. Failure of our components is a possibility during this test. Based on our engineering judgment, our team feels if it were to then the chain in the reduction system is most likely to fail first. It is important to note that all of our components to which the torque will be applied will be enclosed within the reduction casing which will prevent any loose parts from flying out. The reduction case is designed to withstand these loads under normal and extreme conditions and thus should not fail. We plan on placing an aluminum sheet at the back of the vehicle and performing the test from behind this sheet just as a precaution. Fracture will be tested for by basic visual inspection. We also want to make sure that the reduction system we use will transfer torque effectively. Since we are using a chain and sprocket system the amount of slack in the roller chain must be less than 2% of the center-to-center distance between the sprockets for it to effectively transfer torque. We intend to physically measure this slack by using a micrometer or ruler and calculate the percent slack in the system.

Overall safety of our design is another major concern of our project. In addition to the above engineering specification that the system must not fracture when subjected to maximum torque, it is important that no parts are expelled from the vehicle and openly exposed components that could possibly cause damage must be fully enclosed. To avoid parts being expelled from the vehicle all components of the design must remain within the restriction planes designated to create boundaries in the workspace. To prevent any clothing or body parts getting caught in the components, there is an OSHA requirement for maximum allowable opening for metals (mesh opening) of 0.5 inches that will be administered. Validation of this engineering specification will include measuring restriction planes of the vehicle, and measuring the distance between the plates of the reduction case. We will require simple measurement instruments to measure distances and should take no more than an hour to complete.

Our next requirement was to include configurable gear ratios in our design. Our customer wanted us to have a total of four changeable ratios of which one is neutral for purely electric purposes. Once the reduction case is tested for torque and power transmission, we wish to test the multiple gear ratios using a simple rotational test. In this test we will manually rotate the input shaft and visually monitor the output using a marking. Our customer also wanted the above-mentioned gear ratios to be easily changeable within a time of less than three hours. Our plan is to test this by actually timing ourselves performing the gear swap out. The gear swap out test and the checking the configurable gear ratios rotation test will be performed simultaneously. But it is important to conduct the test for interchanging of gears separately, to make sure it can be done within three hours. The gear swap out test involves us working with simple working tools such as wrenches, screw drivers etc. The main safety concern will be components falling once loosened and injuring the operator. It is important to perform this swap out with full presence of mind and small safety issues can be avoided.

It was very important that our final design fits within the space available to us. We first tested this by preparing a mockup of our alpha design and check if it fits within the frame of the actual vehicle. This mockup served more as a form/shape of our design and helped us understand what kind of restraints we might have in the future. Acting as our baseline test and then we will test our actual fabricated components to fit within our frame. We have kept aside two days for preliminary tests for fitting each part, and a final testing period for the entire assembly of about 5 hours. It is important to measure the distances between parts that are not intentionally supposed to be in contact to meet the engine rock clearance of ≥ .52 inch. This will be conducted when the assembly is completed using measuring instruments (ruler, micrometer, etc) and we have estimated about two hours to complete this task.

The reliability of our design was the next important specification. We plan to design our parts to withstand cyclical loading for an "infinite" number of cycles (10^6 cycles). The maximum stresses faced by parts must be less than the endurance limit of the material. We will determine this endurance limit by using a S-N diagram of the respective materials used and determining the endurance stress limit for a duty cycle of 10^6 cycles. To ensure parts will not fail under any expected loads, we have implemented a safety factor of 2. This will be achieved by selecting the appropriate materials that can withstand the disturbances. We also conducted a thorough engineering analysis on each design parameter, to make sure the materials used and the dimensions of the fabricated components ensure a safety factor of at least 2. Some of this validation/analysis was conducted during the weeks preparing for DR3.

Vehicle stability and dynamics was the next important specification of our design. By adding new components to the current Xebra the center of gravity of the vehicle will shift. We plan on monitoring this change by first finding the current location of the center of gravity using scales under each wheel and calculating the location. The EPA has conducted this weight test and determined the current location of the CG. Once we have this baseline position of the CG, we will add the weights of the new components of our final design to the calculations of CG and find the new center of gravity location. Our goal is to design such that the new CG lies between the current location and the rear axle in the horizontal direction and the current location and the ground in the vertical direction. We have set aside an overall period if two days to find to location of the new center of gravity based on the additions we make to the current vehicle.

Lastly, the transferability of our project to our customers and sponsors is extremely important. We wish to test/quantify this by providing them with all the necessary documentation, CAD files, engineering analysis, etc, and would like to actually sign a written acknowledgement, confirming that our team has met these goals.

A practical challenge we will be facing during our validation is getting full access to the Blue Xebra vehicle. Currently, the vehicle is at the EPA where they are conducting tests on the batteries. We are constantly in contact with the EPA for the availability of the vehicle. As per the last update the vehicle will be available to us latest by November 15th. We have set aside a testing period of a week starting on November 29th which ends a few days before the design expo to keep some days for troubleshooting during our validation.

Most of our validation involves us measuring lengths, clearances and in one case rotation of a shaft. These tests involve using simple measuring instruments and do not involve any major safety concerns. Once we have our validation plan approved we would like to confirm with the faculty regarding the tests that will require supervision. For those tests we will arrange for an approved faculty member to be present during validation.

Designsafe Report

Application:	Hub	Analyst Name(s):	ME 450 Team 5
Description:	The hubs connect the shafts to the chain and sprocket gear reduction system. Failure of the hub leads to failure of the system. However, the hub is enclosed by the reductiong case which minimizes potential external injuries.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Part could sever due to shear loads or torque applied to the component	Moderate Likely	Medium	Part geometry and material selection must be designed so that expected loads are accounted for.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Rotation of component and continuous engagement with other components can cause material fatigue and failure	Moderate Likely	Medium	Infinite fatigue cycling must be planned for by selecting materials with higher endurance limits than stresses that could be faced by the components.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Rotational motion with forces applied to components can cause loading in unexpected directions and parts to break while operating	Serious Likely	High	Part geometry and material selection must be designed so that expected loads are accounted for.	Serious Likely	High	Complete [11/4/2010]
1-1-4	All Users Common Tasks	mechanical : impact Forces of unexpected magnitude may be generated during operation of the vehicle, creating impact forces to the components	Moderate Likely	Medium	Adequate safety factors under load must be built into the parts to sustain any load that is not expected but might occur.	Moderate Likely	Medium	Complete [11/4/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	slips / trips / falls : debris Pre-existing debris on the components may be ejected due to the rotational motion. Additionally, break-up of components creates more hazards for objects to possibly be ejected from the system	Serious Unlikely	Medium	Parts must be enclosed to contain debris generation. Parts that are not enclosed should not be rotational	Serious Unlikely	Medium	Complete [11/4/2010]
1-1-6	All Users Common Tasks	noise / vibration : noise / sound levels > 80 dBA Eccentricity and misalignment of rotating parts can cause excessive noise and chatter	Moderate Unlikely	Low	Rotational parts that might cause excessive noise should be contained so that most of the noise does not reach the external environment	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-7	All Users Common Tasks	noise / vibration : equipment damage Vibration of components due to the input torques from the DC motor and hydraulic pump/motor can cause equipment damage	Moderate Unlikely	Low	Vibration forces (likely from the road) must be included as part of the impact and cutting/severing hazards. Again, this is dependant on geometry and material	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-8	All Users Common Tasks	noise / vibration : fatigue / material strength Material fatigue in conjunction with mechanical fatigue effects must be considered. Endurance limit of materials must be great enough to withstand the combination of both.	Moderate Unlikely	Low	Material considerations for infinite fatigue apply. In conjunction with mechanical fatigue, vibration cycling must be included as well	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-9	All Users Common Tasks	environmental / industrial hygiene : corrosion Materials must withstand environmental conditions faced during operation including rain, road-borne chemicals, air, etc. without significantly corroding.	Minor Unlikely	Negligible	Material must be chosen to have minimal corrosion from the expected external conditions	Minor Unlikely	Negligible	Complete [11/4/2010]

designsafe Report

Application:	Reduction Case	Analyst Name(s):	ME 450 Team 5
Description:	The reduction case is composed of 6 pieces of steel that are bolted together. This is attached to the sub-frame through tabs and gussets that are welded to the case. The case encloses multiple components including shafts and sprockets.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Parts sever from shear loads induced by weight of components and internal and externally applied torques from the motor and other disturbances.	Moderate Likely	Medium	Design for expected loads and moments that the system will be subjected to by selecting material and part geometry.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Load cycling causes material failure	Moderate Likely	Medium	Material selection must include part geometry and be designed for "infinite" fatigue life under expected load cycles.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Cycling or impact causes failure of parts	Serious Likely	High	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Serious Likely	High	Complete [11/3/2010]
1-1-4	All Users Common Tasks	mechanical : impact Impact forces causes material failure	Moderate Likely	Medium	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Moderate Likely	Medium	Complete [11/3/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	noise / vibration : equipment damage Vibration caused the the rotational aspects of components causes damage to the equipment	Moderate Unlikely	Low	Vibration between noncontacting parts is reduced by establishing a clearance between parts.	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-6	All Users Common Tasks	noise / vibration : fatigue / material strength Vibration cycles causes material failure due to fatigue	Moderate Unlikely	Low	Material must be selected for expected vibration fatigue in addition to load fatigue cycles	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-7	All Users Common Tasks	environmental / industrial hygiene : corrosion Environmental exposure (air, water, etc.) causes corrosion of material	Minor Unlikely	Negligible	Non (or minimally) corrosive material must be chosen based on expected environmental conditions	Minor Unlikely	Negligible	Complete [11/3/2010]

designsafe Report

Application:	Shaft	Analyst Name(s):	ME 450 Team 5
Description:	The two shafts are at different locations but serve the same purpose and are subjected to the same hazards. Each is connected to a motor and through a spline connection. The rigid connection and rotational component present novel hazards.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Part could sever due to shear loads or torque applied to the component	Moderate Likely	Medium	Part geometry and material selection must be designed so that expected loads are accounted for.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Rotation of component and continuous engagement with other components can cause material fatigue and failure	Moderate Likely	Medium	Infinite fatigue cycling must be planned for by selecting materials with higher endurance limits than stresses that could be faced by the components.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Rotational motion with forces applied to components can cause loading in unexpected directions and parts to break while operating	Serious Likely	High	Part geometry and material selection must be designed so that expected loads are accounted for.	Serious Likely	High	Complete [11/4/2010]
1-1-4	All Users Common Tasks	mechanical : impact Forces of unexpected magnitude may be generated during operation of the vehicle, creating impact forces to the components	Moderate Likely	Medium	Adequate safety factors under load must be built into the parts to sustain any load that is not expected but might occur.	Moderate Likely	Medium	Complete [11/4/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	slips / trips / falls : debris Pre-existing debris on the components may be ejected due to the rotational motion. Additionally, break-up of components creates more hazards for objects to possibly be ejected from the system	Serious Unlikely	Medium	Parts must be enclosed to contain debris generation. Parts that are not enclosed should not be rotational	Serious Unlikely	Medium	Complete [11/4/2010]
1-1-6	All Users Common Tasks	noise / vibration : noise / sound levels > 80 dBA Eccentricity and misalignment of rotating parts can cause excessive noise and chatter	Moderate Unlikely	Low	Rotational parts that might cause excessive noise should be contained so that most of the noise does not reach the external environment	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-7	All Users Common Tasks	noise / vibration : equipment damage Vibration of components due to the input torques from the DC motor and hydraulic pump/motor can cause equipment damage	Moderate Unlikely	Low	Vibration forces (likely from the road) must be included as part of the impact and cutting/severing hazards. Again, this is dependant on geometry and material	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-8	All Users Common Tasks	noise / vibration : fatigue / material strength Material fatigue in conjunction with mechanical fatigue effects must be considered. Endurance limit of materials must be great enough to withstand the combination of both.	Moderate Unlikely	Low	Material considerations for infinite fatigue apply. In conjunction with mechanical fatigue, vibration cycling must be included as well	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-9	All Users Common Tasks	environmental / industrial hygiene : corrosion Materials must withstand environmental conditions faced during operation including rain, road-borne chemicals, air, etc. without significantly corroding.	Minor Unlikely	Negligible	Material must be chosen to have minimal corrosion from the expected external conditions	Minor Unlikely	Negligible	Complete [11/4/2010]

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Application:	Spacer	Analyst Name(s):	ME 450 Team 5
Description:	The spacer is unique in that it is a static component (a solid ring bolted to the intermediate reduction case and the reduciton gear casing) but could face rotation (from the shaft rotating within it) if failure were to occur.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Parts sever from shear loads induced by weight of components and internal and externally applied torques from the motor and other disturbances.	Moderate Likely	Medium	Design for expected loads and moments that the system will be subjected to by selecting material and part geometry.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Load cycling causes material failure	Moderate Likely	Medium	Material selection must include part geometry and be designed for "infinite" fatigue life under expected load cycles.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Cycling or impact causes failure of parts	Serious Likely	High	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Serious Likely	High	Complete [11/3/2010]
1-1-4	All Users Common Tasks	mechanical : impact Impact forces causes material failure	Moderate Likely	Medium	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Moderate Likely	Medium	Complete [11/3/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	noise / vibration : equipment damage Vibration caused the the rotational aspects of components causes damage to the equipment	Moderate Unlikely	Low	Vibration between noncontacting parts is reduced by establishing a clearance between parts.	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-6	All Users Common Tasks	noise / vibration : fatigue / material strength Vibration cycles causes material failure due to fatigue	Moderate Unlikely	Low	Material must be selected for expected vibration fatigue in addition to load fatigue cycles	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-7	All Users Common Tasks	environmental / industrial hygiene : corrosion Environmental exposure (air, water, etc.) causes corrosion of material	Minor Unlikely	Negligible	Non (or minimally) corrosive material must be chosen based on expected environmental conditions	Minor Unlikely	Negligible	Complete [11/3/2010]

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Application:	Sprocket	Analyst Name(s):	ME 450 Team 5
Description:	The sprockets must transmit torque while being attached to the hubs. The bolt connection to the hub presents safety hazards due to possible shearing. The chain also presents hazards due to the torque transmission and possible chain or tooth breakage.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Part could sever due to shear loads or torque applied to the component	Moderate Likely	Medium	Part geometry and material selection must be designed so that expected loads are accounted for.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Rotation of component and continuous engagement with other components can cause material fatigue and failure	Moderate Likely	Medium	Infinite fatigue cycling must be planned for by selecting materials with higher endurance limits than stresses that could be faced by the components.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Rotational motion with forces applied to components can cause loading in unexpected directions and parts to break while operating	Serious Likely	High	Part geometry and material selection must be designed so that expected loads are accounted for.	Serious Likely	High	Complete [11/4/2010]
1-1-4	All Users Common Tasks	mechanical : impact Forces of unexpected magnitude may be generated during operation of the vehicle, creating impact forces to the components	Moderate Likely	Medium	Adequate safety factors under load must be built into the parts to sustain any load that is not expected but might occur.	Moderate Likely	Medium	Complete [11/4/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	slips / trips / falls : debris Pre-existing debris on the components may be ejected due to the rotational motion. Additionally, break-up of components creates more hazards for objects to possibly be ejected from the system	Serious Unlikely	Medium	Parts must be enclosed to contain debris generation. Parts that are not enclosed should not be rotational	Serious Unlikely	Medium	Complete [11/4/2010]
1-1-6	All Users Common Tasks	noise / vibration : noise / sound levels > 80 dBA Eccentricity and misalignment of rotating parts can cause excessive noise and chatter	Moderate Unlikely	Low	Rotational parts that might cause excessive noise should be contained so that most of the noise does not reach the external environment	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-7	All Users Common Tasks	noise / vibration : equipment damage Vibration of components due to the input torques from the DC motor and hydraulic pump/motor can cause equipment damage	Moderate Unlikely	Low	Vibration forces (likely from the road) must be included as part of the impact and cutting/severing hazards. Again, this is dependant on geometry and material	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-8	All Users Common Tasks	noise / vibration : fatigue / material strength Material fatigue in conjunction with mechanical fatigue effects must be considered. Endurance limit of materials must be great enough to withstand the combination of both.	Moderate Unlikely	Low	Material considerations for infinite fatigue apply. In conjunction with mechanical fatigue, vibration cycling must be included as well	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-9	All Users Common Tasks	environmental / industrial hygiene : corrosion Materials must withstand environmental conditions faced during operation including rain, road-borne chemicals, air, etc. without significantly corroding.	Minor Unlikely	Negligible	Material must be chosen to have minimal corrosion from the expected external conditions	Minor Unlikely	Negligible	Complete [11/4/2010]

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Application:	Sub-frame	Analyst Name(s):	ME 450 Team 5
Description:	The sub-frame is composed of rectangular steel tube. It supports the gear reduction casing, its internal components, the DC motor, and the hydraulic pump/motor. Failure of this component leads to the failure of the system.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Initial Assessment Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Final Assessment Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Parts sever from shear loads induced by weight of components and internal and externally applied torques from the motor and other disturbances.	Moderate Likely	Medium	Design for expected loads and moments that the system will be subjected to by selecting material and part geometry.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Load cycling causes material failure	Moderate Likely	Medium	Material selection must include part geometry and be designed for "infinite" fatigue life under expected load cycles.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Cycling or impact causes failure of parts	Serious Likely	High	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Serious Likely	High	Complete [11/3/2010]
1-1-4	All Users Common Tasks	mechanical : impact Impact forces causes material failure	Moderate Likely	Medium	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Moderate Likely	Medium	Complete [11/3/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	noise / vibration : equipment damage Vibration caused the the rotational aspects of components causes damage to the equipment	Moderate Unlikely	Low	Vibration between noncontacting parts is reduced by establishing a clearance between parts.	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-6	All Users Common Tasks	noise / vibration : fatigue / material strength Vibration cycles causes material failure due to fatigue	Moderate Unlikely	Low	Material must be selected for expected vibration fatigue in addition to load fatigue cycles	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-7	All Users Common Tasks	environmental / industrial hygiene : corrosion Environmental exposure (air, water, etc.) causes corrosion of material	Minor Unlikely	Negligible	Non (or minimally) corrosive material must be chosen based on expected environmental conditions	Minor Unlikely	Negligible	Complete [11/3/2010]

designsafe Report

Application:	Tabs & Gussets	Analyst Name(s):	ME 450 Team 5
Description:	The tabs and gussets are welded to each other and then to the gear reduction casing. The materials are purchased then fabricated by us.	Company:	University of Michigan
Product Identifier:		Facility Location:	Ann Arbor, MI
Assessment Type:	Detailed		
Limits:			
Sources:			
Risk Scoring System:	ANSI B11 TR3 Two Factor		

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Parts sever from shear loads induced by weight of components and internal and externally applied torques from the motor and other disturbances.	Moderate Likely	Medium	Design for expected loads and moments that the system will be subjected to by selecting material and part geometry.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Load cycling causes material failure	Moderate Likely	Medium	Material selection must include part geometry and be designed for "infinite" fatigue life under expected load cycles.	Moderate Likely	Medium	Complete [11/3/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Cycling or impact causes failure of parts	Serious Likely	High	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Serious Likely	High	Complete [11/3/2010]
1-1-4	All Users Common Tasks	mechanical : impact Impact forces causes material failure	Moderate Likely	Medium	Material selection accounts for excessive loads including a safety factor in loading of at least 2.	Moderate Likely	Medium	Complete [11/3/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	noise / vibration : equipment damage Vibration caused the the rotational aspects of components causes damage to the equipment	Moderate Unlikely	Low	Vibration between noncontacting parts is reduced by establishing a clearance between parts.	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-6	All Users Common Tasks	noise / vibration : fatigue / material strength Vibration cycles causes material failure due to fatigue	Moderate Unlikely	Low	Material must be selected for expected vibration fatigue in addition to load fatigue cycles	Moderate Unlikely	Low	Complete [11/3/2010]
1-1-7	All Users Common Tasks	environmental / industrial hygiene : corrosion Environmental exposure (air, water, etc.) causes corrosion of material	Minor Unlikely	Negligible	Non (or minimally) corrosive material must be chosen based on expected environmental conditions	Minor Unlikely	Negligible	Complete [11/3/2010]

designsafe Report

Application: Power Transmission Coupling System

Analyst Name(s): ME 450 Team 5

Description: The overall coupling system consists of both static and rotational components. Each component faces varying loads and torques during different points of operation. Also, unexpected impact and shock loads could be subjected to any part of the system.

Company: University of Michigan

Product Identifier:

Facility Location: Ann Arbor, MI

Assessment Type: Detailed

Limits:

Sources:

Risk Scoring System: ANSI B11 TR3 Two Factor

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

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-1	All Users Common Tasks	mechanical : cutting / severing Part could sever due to shear loads or torque applied to the component	Moderate Likely	Medium	Part geometry and material selection must be designed so that expected loads are accounted for.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-2	All Users Common Tasks	mechanical : fatigue Rotation of component and continuous engagement with other components can cause material fatigue and failure	Moderate Likely	Medium	Infinite fatigue cycling must be planned for by selecting materials with higher endurance limits than stresses that could be faced by the components.	Moderate Likely	Medium	Complete [11/4/2010]
1-1-3	All Users Common Tasks	mechanical : break up during operation Rotational motion with forces applied to components can cause loading in unexpected directions and parts to break while operating	Serious Likely	High	Part geometry and material selection must be designed so that expected loads are accounted for.	Serious Likely	High	Complete [11/4/2010]
1-1-4	All Users Common Tasks	mechanical : impact Forces of unexpected magnitude may be generated during operation of the vehicle, creating impact forces to the components	Moderate Likely	Medium	Adequate safety factors under load must be built into the parts to sustain any load that is not expected but might occur.	Moderate Likely	Medium	Complete [11/4/2010]

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Probability	Risk Level	Status / Responsible /Reference
1-1-5	All Users Common Tasks	slips / trips / falls : debris Pre-existing debris on the components may be ejected due to the rotational motion. Additionally, break-up of components creates more hazards for objects to possibly be ejected from the system	Serious Unlikely	Medium	Parts must be enclosed to contain debris generation. Parts that are not enclosed should not be rotational	Serious Unlikely	Medium	Complete [11/4/2010]
1-1-6	All Users Common Tasks	noise / vibration : noise / sound levels > 80 dBA Eccentricity and misalignment of rotating parts can cause excessive noise and chatter	Moderate Unlikely	Low	Rotational parts that might cause excessive noise should be contained so that most of the noise does not reach the external environment	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-7	All Users Common Tasks	noise / vibration : equipment damage Vibration of components due to the input torques from the DC motor and hydraulic pump/motor can cause equipment damage	Moderate Unlikely	Low	Vibration forces (likely from the road) must be included as part of the impact and cutting/severing hazards. Again, this is dependant on geometry and material	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-8	All Users Common Tasks	noise / vibration : fatigue / material strength Material fatigue in conjunction with mechanical fatigue effects must be considered. Endurance limit of materials must be great enough to withstand the combination of both.	Moderate Unlikely	Low	Material considerations for infinite fatigue apply. In conjunction with mechanical fatigue, vibration cycling must be included as well	Moderate Unlikely	Low	Complete [11/4/2010]
1-1-9	All Users Common Tasks	environmental / industrial hygiene : corrosion Materials must withstand environmental conditions faced during operation including rain, road-borne chemicals, air, etc. without significantly corroding.	Minor Unlikely	Negligible	Material must be chosen to have minimal corrosion from the expected external conditions	Minor Unlikely	Negligible	Complete [11/4/2010]

APPENDIX L: MAXIMUM LOAD TORQUE ACROSS THE SYSTEM

) HYDRAULIC PUMP MOTOR - Torque

To determine the torque applied due to the hydraulic pump motor we can use the following calculations.

* Note: These are max conditions of the hydraulic pump/motor that will never occur under normal working conditions.

Max Pressure = 5000 psi

Displacement = 45 cc/Rev

Max RPM = 2000 RPM

∴ Hydraulic Power = Pressure × Displacement (gal/min)

Mechanical Power = Shaft speed × Torque

Equating the two, to solve for Torque,

∴ Shaft Speed × Torque = Pressure × Displacement

$$\text{Torque} = \frac{\text{Pressure} \times \text{Displacement}}{\text{Shaft speed}}$$

$$= 183 \text{ ft-lb of torque}$$

2) D.C. motor - torque

Max torque due to the DC motor is found using the spec sheet of D.C. motor provided.

T_{DC} = 23 ft-lb of torque

∴ Total or Maximum torque transmitted through our system is = 183 + 23

$\boxed{= 206 \text{ ft-lb}}$

APPENDIX M: CENTER OF GRAVITY CALCULATION

APPENDIX M: CG CALCULATION

Vetical CG Calculation			
Component	Weight,m (lbs)	Location (inches)	in-lbs
Battery	500	19	9500
Blue Body	700	28	19600
Backend	460	15	6900
Box	96	16.5	1584
Pump	60	19.5	1170

Horizontal CG Calculation			
Component	Weight (lbs)	Location (inches)	in-lbs
Back Right Wheel	549	15	8235
Back Left Wheel	518	15	7770
Front Wheel	590	102	60180
			76185
		Baseline Center of Gravity	45.97767049
Box	96	8	768
Pump	60	8	480
			1248
			77433
		New Center of Gravity	42.70987314

We used the basic center of gravity equation below to find the locations of the center

CG = (m1x1+m2x2+m3x3)/(m1+m2+m3)

Baseline Center of Gravity	21.68674699

New Center of Gravity	21.34030837

of gravities