Electric Vehicle Drivetrain Final Design Review

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Statement of Disclaimer

Table of Contents

List of Tables

List of Figures

Executive Summary

This document serves to outline the design and development of Sharpell Technologies' ST5 electric vehicle drivetrain. Background research was conducted on various drivetrain components and testing methods, leading to the development of our drivetrain's design requirements. Compiled in this report are the specifications of the components that will be integrated into our drivetrain design. All component calculations and specifications can be located in the appendices. An in-depth manufacturing plan features the modifications necessary to create several key components such as the shafts and housing. The assembly plan details a step-by-step process in which the components are arranged to ensure a safe and functional drivetrain unit. Our drivetrain design will be validated through a series of static and dynamic tests, confirming that all the design requirements have been met, or exceeded.

1.0 Introduction

Our team's task was to design a drivetrain for Sharpell Technologies. Our sponsor Zachary Sharpell, CEO of Sharpell Technologies, has formed a startup company focused on powerful, yet affordable electric vehicles. Sharpell first became inspired in 10th grade during an Auto Shop course when learning about various power sources. From this original inspiration, Zach began exploring ways to apply the concepts he had learned and eventually moved towards developing an electric car, founding Sharpell Technologies in June 2016. Zach elected for our senior project to focus on a performance oriented, mid-size Sports Utility Vehicle (SUV), that is also off-road capable. Very few companies have released affordable all-electric SUVs, with even fewer that are high performing. None of these existing options are off-road capable, creating an opportunity for Sharpell Technologies to fill the void in the market. This game changing SUV has the codename, ST5. The large SUV vehicle size will allow for a sizable battery that will sustain a 300+ mile range and have enough energy to produce the desired performance output. The ST5 will package all of this into a $40,000 sticker price. Our problem statement that defines the overall scope of our senior design project is presented beneath.

1.1 Problem Statement

Sharpell Technologies needs an affordable and reliable drivetrain to transmit power from an electric motor to rear wheels of an all-electric vehicle, resulting in faster accelerations and off-roading capabilities.

2.0 Background

To better understand the scope of a working drivetrain, our team focused our research on the following areas: drivetrain layout, types of gears and ratios in drivetrains, types of differentials, thrust loads on helical gear trains and bearings, and the SUV customer market.

2.1 Drivetrain Layout

The typical layout of the drivetrain in a rear wheel drive, front-combustion engine vehicle, consists of a multi-speed transmission that transmits power through a driveshaft to a differential. The power is then distributed to each of the rear wheels as shown in Figure 1.

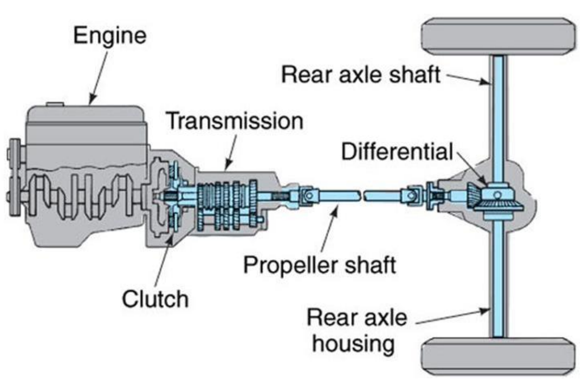

Figure 1. Typical drivetrain/powertrain layout in a rear wheel drive car [1].

Generally, normal transmissions incorporate helical gears for the forward driving, and spur gears for reverse. Spur gears are noisier than helical gears, and since little time is spent in the reverse gear position, the noise adds a tangible signal to the driver that the car will change direction. Utilizing the spur gears also helps to reduce the cost of manufacturing and wear is not as crucial due to lesser amount of time spent in reverse. The differentials use 90 degree hypoid gears. Hypoid gears are bevel-helical or spiral-bevel gears, and are often used when the axes of the power transmission are offset from one-another as seen in Figure 2. Hypoid gears allow the power to be transmitted along the perpendicular axis, which is the case in rear wheel drive, front-combustion engine vehicles. These different types of gears are displayed in Figure 4.

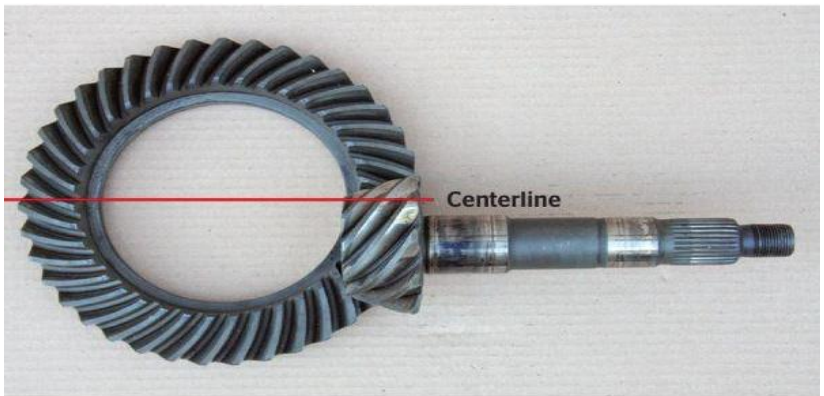

Figure 2. Hypoid gears transmit power offset from the centerline axis [2].

In modern electric vehicles, the drivetrain is less complicated and more compact than drivetrains seen in combustion engine vehicles. The electric motor is much smaller relative to a gasoline engine, allowing more possibilities for mounting locations and configurations. The standard layout has the electric motor transversely mounted, with the drivetrain consisting of both a gear reduction and a differential. The differential in most electric vehicles no longer uses any type of bevel gear, and rather, uses solely helical gears. The reasoning behind only using helical gears is that the axes of the shafts in the gear reduction, electric motor, and axle are all parallel with each other. One example that uses this previous configuration is the Tesla Model S as depicted in Figure 3. A more detailed look inside a transversely mounted drivetrain that is used in a Tesla Roadster, is shown in Figure 9.

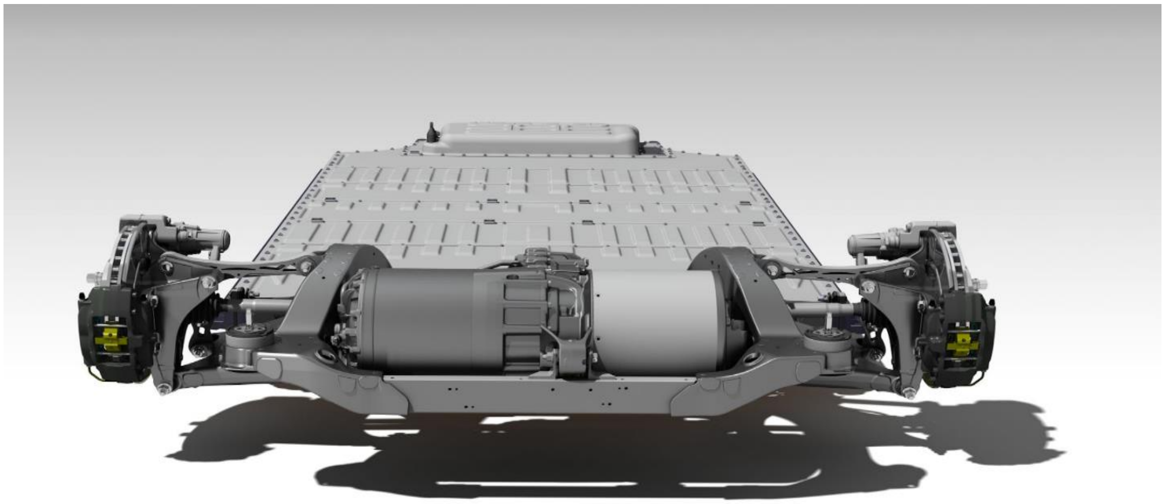

Figure 3. The Tesla Model S rear wheel drive vehicle with a drivetrain that is transversely mounted [3].

2.2 Gear Types

Another aspect found in the drivetrain of the electric vehicles we researched was the bearing type, which are designed based on the forces experienced by the gears. In spur gears, the teeth are cut parallel with the axis of rotation, and the components of the equivalent force only act perpendicular to the axis of rotation. Because helical gear teeth are cut in a helix pattern around the center of the gear, the equivalent forces imposed at the interface of two gears are not strictly perpendicular to the axis of rotation. Figure 4 shows how the spur, helical, and bevel gears are cut. One component of the equivalent force acts along the axis of rotation, subsequently called a thrust load or force, while the other two components act perpendicular to the axis of rotation as depicted in Figure 5.

Figure 4. Different types of gears [4].

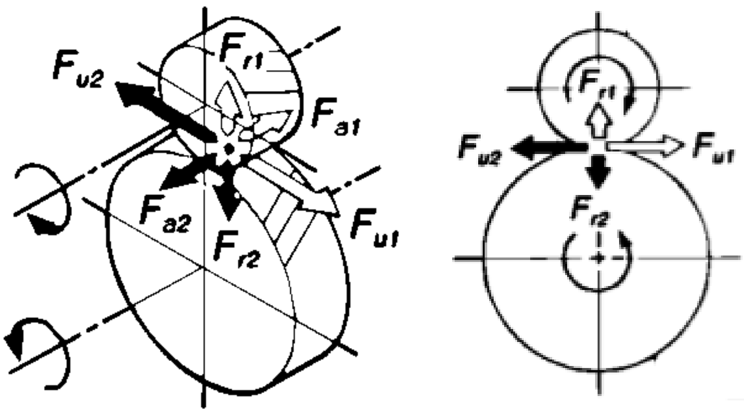

Figure 5. On left: The interface between two helical gears and the components of force imposed between them. On right: The interface between two spur gears and the components of force imposed between them. As explained above the spur gears have no thrust force acting on them [5].

2.3 Differentials

When a vehicle turns, the outside wheels have to travel a longer distance than the inside wheels. As a result, the outer wheels need to spin faster than the inside wheels in order to properly navigate the turn. A differential accommodates for this difference in driven wheel speed by decreasing the rotation of the inner wheel and delivering more rotational speed to the outer wheel. A rear differential can also manage the torque sent to each of the driven rear wheels, so that one wheel does not receive all the power.

All torque correcting differentials are iterations of the open differential as depicted in Figure 6. A pinion gear driven by a power source (a motor in the case of electric vehicles) transforms its rotary motion through the gear reduction to the ring gear. The ring gear is bolted to the differential carrier case, thus rotating the differential. The spider gears are located inside the differential and rotate with the side gears to accomplish the sending different speeds to each of the wheels. The side gears are connected through splines to the axles which supply torque to each of the rear wheels.

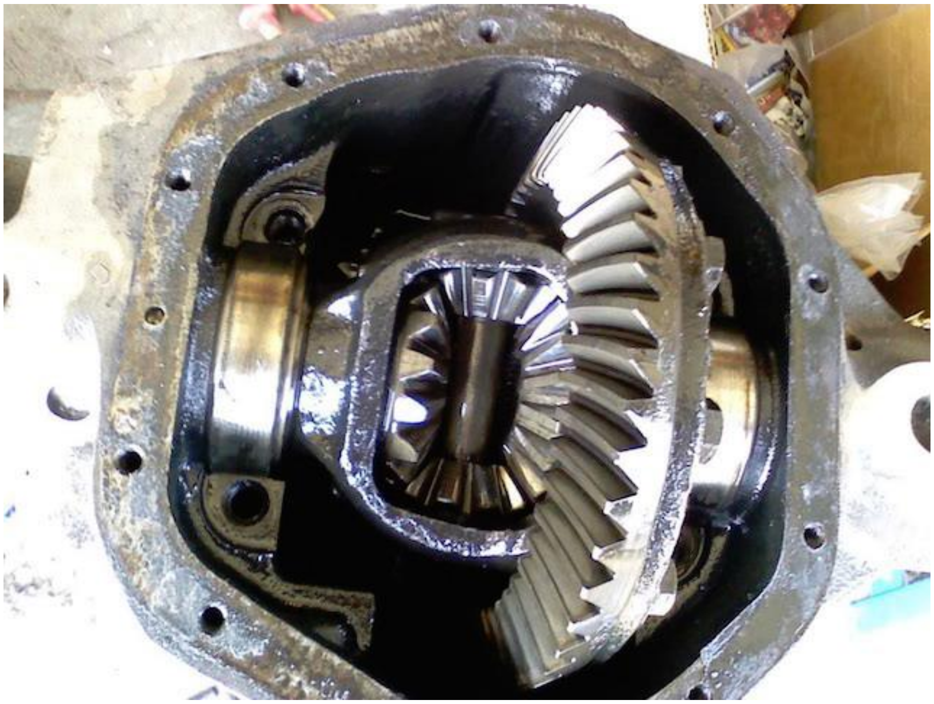

Figure 6. The layout of the gearing mechanisms used in open differentials [6].

The design of an open differential is more than adequate for normal road conditions; however, it has a poor quality when performing in off-roading scenarios. In such a situation where one wheel loses traction, the open differential will send all the power to the wheel with the least resistance. The possibility of losing all the power sent to one of the rear wheels is not ideal for any off-roading vehicle. Open differentials, however, have predictable handling and are the most common differential found on on-road vehicles. One solution to this issue is the locking differential which is used for severe off-roading applications. The locking differential can lock both of the side gears attached to the rear axle, enabling a solid axle with equal torque sent to each rear wheel. A locking differential operates as an open differential when not locked and therefore handles normal road conditions equally as well as a non-locking differential. However, when the differential is in the locked position, the vehicle should be driven only at low speeds and should not undergo tight turns [7].

A compromise between the open differential and the locking differential is the limited slip differential (LSD). A LSD allows for a difference in rotational wheel speed, but limits this difference so that the slower wheel always receives a given amount of power. This means the car can still vary wheel speeds to undergo turns, while never sending all the power to the single, free-spinning wheel. Because of this, LSDs perform better than open differentials for on and off-road driving. Higher speeds can be achieved with a LSD than a locked differential, and the stresses on the axle shaft and differential gears are also reduced. The main disadvantage with a LSD is that when cornering a turn, the car will oversteer due to torque at both of the wheels instead of just the slipping wheel. Also, wear and tear is higher in LSDs because there is always an element of friction acting within the differential.

There are many variations of limited slip differentials in the market today. Traditionally, LSDs were strictly mechanical systems, but now they can incorporate electrical systems that monitor the torque applied to each wheel and intervene when one of the driven tires begins to lose traction. One simple way of creating a LSD includes active braking systems that apply the brakes on the free-spinning wheel, allowing for some torque to be redirected to the other wheel. An electronic limited slip differential (eLSD) uses the vehicle's engine control unit (ECU) to determine the optimum amount of torque for the given driving conditions. Many eLSD systems have programmed settings for different driving surfaces such as snow, gravel, sand, and, of course, paved roads. We are consciously excluding the active braking method and eLSD. These systems integrate many sensors into the drivetrain and require constant communication between the wheels, differential, and the ECU, which is beyond the scope of what is needed for this drivetrain design. Our team narrowed our research accordingly by focusing on mechanical drivetrains and explored options of creating a limited slip differential necessary for the on and off-road driving conditions the ST5 will experience. Ways of achieving limited slip in a differential through purely mechanical systems include: Automatic Torque-Biasing (ATB), Clutch Type LSD, and Bair-Ling Technologies Bi-directional LSD (BT-B LSD) [8].

ATB differentials transfer more torque to the wheel with greater traction, and perform well when heavy braking is applied to the system. They are also durable and do not have issues with understeer. However, these upgrades in function are more expensive while still having issues similar to the ones experienced in an open differential. Another negative is that failure can arise when traction is abruptly regained at a wheel due to sudden loading, making ATB poor in regard to impulses.

Clutch Type LSDs add a spring pack and a set of clutch plates to an open differential configuration, as shown in Figure 7. The clutches activate when the wheels require differing rotational speed. The Clutch Type LSD desires each rear wheel to spin at the same rate, and as differing wheel rotations occur (as is the case in every turn), the frictional resistance is surpassed. Once one wheel surpasses the frictional resistance of the clutch plate, the plates will lock the wheels and the additional torque is sent to that wheel trying to restore the system equal wheel rotation. The spring stiffness and frictional coefficient of the clutch determine the amount of torque required to send more power to a single wheel. In addition, the slower spinning wheel will also receive torque when the faster spinning wheel overcomes the clutch friction [8].

Figure 7. This Clutch Type LSD shows similarities with the open differential, but has an added spring between the side gears, and clutch discs between the side gears and housing [9].

We had the opportunity to explore and review new designs by Bair-Ling Technologies detailing how to modify an open differential to a LSD by slipping on additional tolerance rings and installing housing units to help transfer torque. Bair-Ling Technologies has two new mechanical, torque-correcting devices that they were generous enough to let us examine. The two devices are the Constant Value Torque-Limiter (CVTL) and the mechanical, reactive, speed-sensitive clutch Blair-Ling Technologies Bi-directional (BT-B). Both are creative solutions to alter an open differential to allow for torque differences on each wheel powered out of the differential side gear connections. Once Bair-Ling Technologies began developing the CVTL, they saw that by modifying the shape of the grooves to cause the tolerance ring to wedge between the shaft and housing it acted as a purely mechanical, speed sensitive clutch. This new design, called the Bair-Ling Technologies Bi-directional (BT-B), limits the torque applied to each wheel based on internal friction. If one wheel spins too quickly, the BT-B will lock the axle to the differential housing, sending power to the wheel with traction. When both wheels regain traction, the BT-B unlocks and reverts back to functioning as an open differential. This technique minimizes the wear and tear on the components of the car since the general response will be identical to an open differential and allow the wheels to rotate unhindered [8]. To summarize the different characteristics of each LSD type we compared the LSD types against criteria specified in Table 1 below. This helped us gauge which LSD will integrate best into our drivetrain.

Table 1: Display of the various advantages and disadvantages of the common differentials discussed [8].

	Open	Clutch Type LSD	ATB	BT-B LSD
Complexity	Low	Moderate-High	Very High	low
Cost 150k mi.	$65	$385	$250	$74.75
Life	very long	short	very long	very long
Rebuild Cost	low	high	high	moderate
Understeer	no	yes	no	no
Tire Wear	no	yes	no	no
Traction	poor	good	good-poor	good
Impulse Survival	very good	very good	poor	very good

2.4 Effects of Loads on Bearings

Several different types of bearings were considered to be utilized which included hydrodynamic bearings, tapered roller, and ball bearings. Hydrodynamic bearings utilize an oil film which acts as a fluid damper and greatly reduces the wear on the metal interface.

To combat the thrust loads experienced by helical gears, tapered roller bearings are used because they are designed to specifically counteract those type of loads. Tapered roller bearings can handle radial loads and much greater thrust loads at a greater weight relative to a typical ball bearing because of their tapered races as shown in Figure 8 [10]. On the other hand, ball bearings still support both thrust and radial loads but are usually used in applications where smaller weight loads are applied. Contrary to what one might believe based on the previous information, ball bearings are used in the BorgWarner designed Tesla Roadster drivetrain, shown in Figure 9, rather than a tapered roller bearing. If ball bearings are sized appropriately (large enough), they can handle a somewhat substantial loads despite the fact that ball bearings are meant for smaller forces. This is important because tapered roller bearings are generally more expensive due to their complex nature and using them would increase overall production costs.

Figure 8. A cutaway view of a tapered roller bearing showing the tapered races [10].

Figure 9. The 31-03 eGearDrive gearbox made by BorgWarner for the Tesla Roadster, Ford Transit Connect Electric, and Coda Electric [11].

2.5 Housing and Sealant Options

The housing encompasses the components surrounding the gears, differential, bearings, and also contains an oil bath to lubricate the inside for thermal transfers. The housing of most common drivetrains is made of aluminum or steel and has a vertical transverse parting line that runs parallel to the rear axle (please see Figure 10 below).

Figure 10. Vertical, transverse parting line as shown by the plate bolted on the back of the differential [12].

To reduce weight and increase performance at high temperatures, aluminum is the likely candidate for our design. Both A413 and A360 aluminum alloys work well for keeping constant pressure and resisting corrosion. A typical surface finish of 100 to 200 micro inches should be used so that all the parts pressed into the housing will mate properly and completely contain the fluid inside. Typical methods to cast an aluminum housing are die casting and sand molds. If sand casting is pursued, the form must be larger than the housing dimensions to allow for shrinkage. Negative forms will be inserted inside to create the hollow cavities needed, and the mold can be broken after the positive is produced. Sand casting is the cheaper method since simple equipment is needed, even though it requires CNC machining to meet the tolerances.

Compared to sand casting, die casting is the more precise and accurate method. Die casting is used in industry settings when a large volume of parts needs to be created. High pressure die casting in-particular lends itself well to applications where tight tolerances are needed. This will also reduce the machining needed after the part has been cast. In this process, liquid metal is directly injected into the die via a shot sleeve and piston. However, the machine limits the size of the part that can be cast and is more expensive than other methods. Depending on the resources available, other methods of casting may be utilized in future steps to design the housing [13] [14].

In addition to the housing, seals are needed to prevent leakage and to prevent dirt and grime from entering the bearings. The most common type of seal used in drivetrains are rotary-single lipped seals as depicted in Figure 11. below.

Figure 11. Rotary-single lipped seal [15]

The seals are fitted directly into the housing bore and next to the bearing. The spring in the lip seal pre-loads the seal, and allows it to withstand higher heats by preventing thermal deformation in the material.

The type of lubricant determined for the oil bath affects the oil seal material. For typical engine oil (SAE 30 Wt. or SAE 10 Wt.) or gear oil (SAE 80 Wt.), nitrile rubber or polyacrylate rubber resist the oils and greases well. Nitrile and polyacrylate can withstand a temperature range from -40° to 250°F and -30° to 300°F respectively [16] [17]. Given that the housing will most likely be cast in aluminum, softer sealant rings such as the nitrile or polyacrylate are also more compatible versus making the seal out of a rigid material that cannot conform to the housing material. The diameter of the housing bore will also affect the tolerance needed for the seal to keep a tight fit. See below for Table 2 and Table 3 in regards to the housing bore diameters and the tolerances needed for the future housing design.

Table 2. Displays various tolerances for the outer diameter of the lip seal based on different housing bores [16].

Bore Diameter - Inch	**Nominal Press-Fit**		**O.D. Tolerance**	
	Metal O.D.	**Rubber O.D.**	**Metal O.D.**	**Rubber O.D.**
Up to 1.000	0.004	0.006	+/- 0.002	+/- 0.003
1.001 to 2.000	0.004	0.007	+/- 0.002	+/- 0.003
2.001 to 3.000	0.004	0.008	+/- 0.002	+/- 0.003
3.001 to 4.000	0.005	0.01	+/- 0.002	+/- 0.004
4.001 to 6.000	0.005	0.01	-1.5	+/- 0.004
6.001 to 8.000	0.006	0.01	-1.5	+/- 0.004
8.001 to 10.000	0.008	0.01	-2	+/- 0.004
10.001 to 20.000	0.008	0.01	-3	+/- 0.004
20.001 to 40.000	0.008	0.01	-4	+/- 0.004
40.001 to 60.000	0.008	0.01	-5	+/- 0.004

Table 3. Seal Width Tolerances [16]

	Width Range	**Tolerance**
Inch	All	+/- 0.015
Metric	Up to 10	+/- 0.20
	Over 10	+/- 0.30

2.6 Customer Market

The market for the ST5 must be fairly broad in order to see an acceptable number of sales. Our sponsor, Zach, filled us in on his initial market research he conducted when he first formulated this idea. Some key points from his initial study are that the electric SUV market is increasing as more manufacturers put these vehicles on the road. At the time of Zach's research, an electric SUV that had a 300 mile range for under $40,000 did not exist. Studies also show that "15% is the industry standard for 'number of 4x4s that actually go off-road" [18]. Given this very low percentage, the ST5 must still be capable of off-road use in order to attract the most customers possible. While the design of the drivetrain itself doesn't directly affect who will buy the ST5, it will still inadvertently play into the overall cost and capability of the vehicle.

2.7 Motor-Drivetrain Interface

Another crucial part of the design is the motor-drivetrain interface. Commonly, a machined steel interface isolates the motor shaft and drivetrain input shaft to maintain proper alignment and ensure only pure torsional load from the motor is transferred. Keeping the weight of the system off of the driveshaft assembly ensures longer bearing life and minimizes mechanical losses of the system.

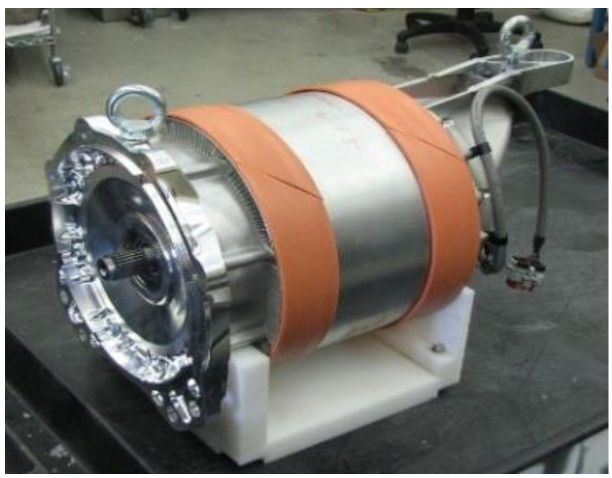

Figure 12: Motor-Drivetrain interface in a Tesla Model S [3]

The motor shaft interface consists of two components, the 21 spline attached to the rotor extruding from the motor shaft collar, and the helical pinion gear. This connection point is responsible for transferring all of the torque produced from the motor shaft to the differential shaft. The spline was determined to be sufficient based on torque rating and minimum permissible diameter. In Cal Poly Motor Car Association's Electric Car Re-design's final report, the distortion energy theory was used to find the shear strength of the spline assuming a yield strength of the steel. The factor of safety for both bending and contact stresses were found and determined to be adequate for the specifications of their design. In addition, future whole system designs can incorporate other motors to be connected to the drivetrain.

3.0 Requirements

Our goal is to design a drivetrain to deliver 700 lb-ft of torque at 6,000 rpm and 400 horsepower to the rear wheels of a 4,500 lbf vehicle. At the end of the year we will have a completed physical deliverable to present to our sponsor. The specifications listed above were received directly from our sponsor and were verified by benchmarking against existing quantifiable parameters from electric cars on the market such as the Roadster, Tesla's Model S and X, and the Chevy Bolt. Other capabilities that are desired for the drivetrain are off-road driving in rugged terrain, and accelerating 0-60 mph in 5 seconds.

The drivetrain layout must be compatible with emergency brakes and motor connection points. The housing will have areas to mount to the chassis and will be cast out of the best material found after further research and analysis. Additionally, the ability for future mass production will be considered when developing the manufacturing processes. A boundary sketch, Figure 12, was produced to show how the drivetrain interacts with the motor and rear axle assembly.

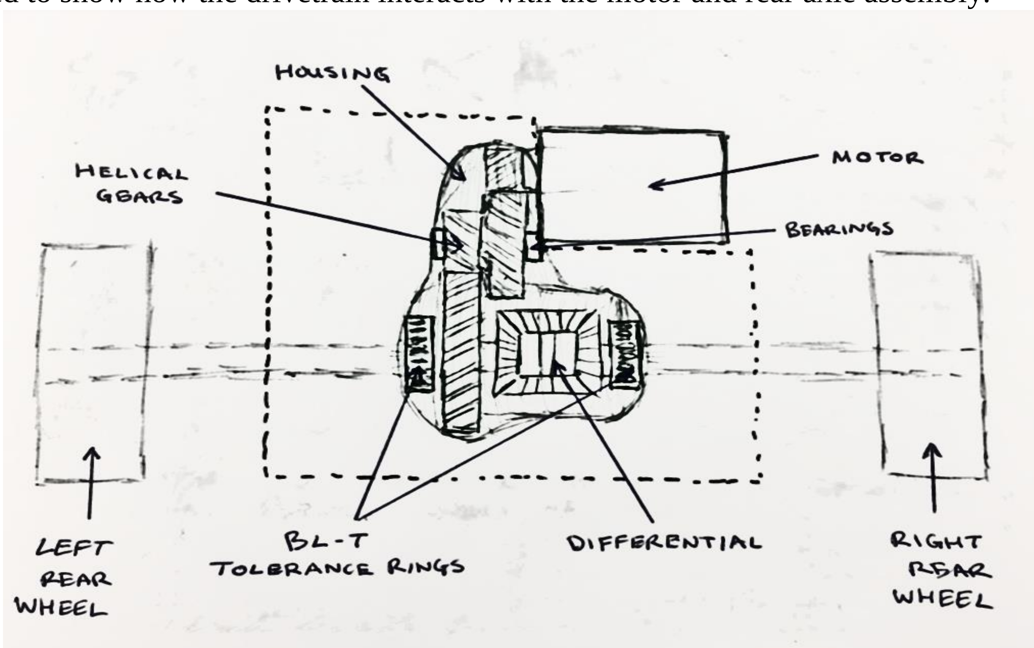

Figure 13. Boundary sketch showing the scope of our design which includes bearings, shafts, gears, differential, tolerance rings, and the housing.

3.1 Quality Function Deployment Discussion

The QFD model breaks the development of engineering specifications into 9 steps listed below:

1. Who: Customers affected by the design
2. What: Customer requirements as customer sees them
3. Who vs What: Customer priorities, weighted and added to 100
4. Now: Benchmarking against competitors who satisfy similar needs
5. Now vs What: How well does the competitor meet the need?
6. How: How will the design meet the requirements?
7. What vs How: Target for engineering specifications

8. How Much: How well does the competition meet the specifications?
9. The Roof: Check for redundancy in specifications

Our customer requirements came directly from our sponsor who performed a substantial amount of market research prior to developing the engineering objectives. We then modified these to suit a broader audience. Besides meeting our sponsor's requirements, we selected several other categories of common users to add to our customer base and listed them in the "WHO" column of the QFD. After adding the constraints from Zach Sharpell, we weighted them according to the functional value we felt each client type would choose. The results of the QFD output percentages that ranked the importance of the different specifications and tests based on our clientele. From this product assessment of our abilities versus the current market, the ST5 fills in gaps in other producers and can better satisfy the customers' needs. We took the specifications listed in the QFD matrix and assessed how we would perform tests to ensure our design was capable of meeting the requirements. The table also displays the level of risk in running each of the test and feasible measures to test each. For example, to ensure the design life of the drivetrain, we have a minimum mileage for the vehicle of 100,000 miles. Since this specification will not be able to be tested during the period of this project (giving the physical test a risk of high) our team chose to verify this area via analysis. Several specifications are not essential to the development of the drivetrain. These nonessential specifications include accelerating from 0-60 mph in 5 seconds maximum time, being capable of a specific grade or terrain, and discovering the minimum torque to wheel that the differential provides. However, although they are non-essential they could still be specifications that additionally benefit the customer and thus could increase the market demand.

Table 4: Original Specification table outlining test targets and tolerances to verify that each requirement is satisfied. Please see Appendix A.1 for the QFD model.

Spec #	Specific Description	Target [Units]	Tolerance	Risk	Compliance
1	Output rpm required for 155 mph top speed		min	M	A,S, T
2	Input Power	400 [HP]	min	M	A,T
3	Input Torque	700 [lb-ft]	min	H	A,T
4	Rpm Range	0-11,000 [rpm]	min	M	A,T
5	Design Life	100,000 [mi]	min	H	A
6	Output torque to achieve 0-60 time of 5 [s]		min	L	A, T
7	Cost	$10,000	max	L	S,I
8	Housing Leakage	0	+ 1 ml in 3 days	L	T,I
9	Torque to Rear Wheels	250 [lb-ft]	min	L	A, T

Key
A = Analysis
S=Similarity
T = Test
I = Inspection
Requires Detailed Analysis

4.0 Design Development

4.1 Concept Generation

To start the design development of our electric drivetrain, we first began with the ideation method of brain-writing. Due to the scope of our project, the specifications put forth from the beginning were very specific, but after forced creativity we found numerous ideas that related to the function of transmitting power to the rear wheels. We consciously avoided the typical method utilizing gear reductions, and instead ideated around chains, belts, and multiple driving motors at each wheel to expand our options. During the second ideation session, we used Legos to build the different layouts for the drivetrain that we had in mind. The three major concepts developed are discussed in detail next.

Concept 1:

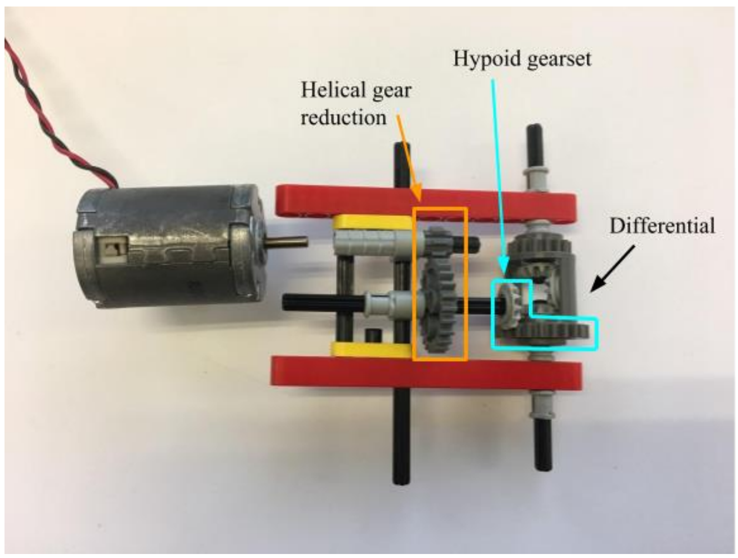

Figure 14. Concept 1: Longitudinally mounted motor with helical gear reduction and stock hypoid differential gear set.

Concept 1 mimics the layout of a drivetrain in a front engine, rear wheel drive vehicle. The electric motor's shaft runs in the longitudinal direction of the vehicle. This requires a change in direction of the power transfer to the rear wheels, which is done by the differential. The differential remains in its stock form, minus the stock housing. A gear reduction coming off of the electric motor utilizes helical gears, where one helical gear shares the intermediate shaft with the pinion gear of the differential. The helical gear reduction along with the differential are packaged in one housing.

Concept 2:

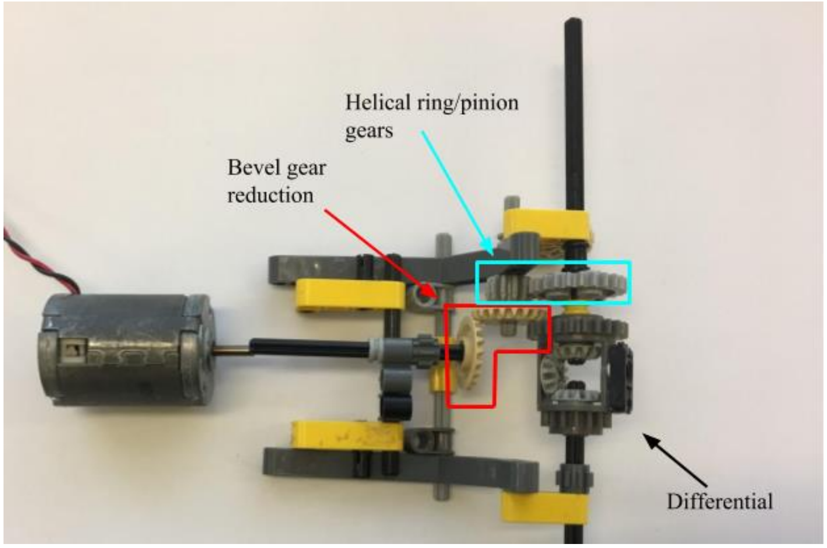

Figure 15. Concept 2: Longitudinally mounted motor with spiral bevel gear reduction and helical ring gear differential.

Concept 2 is similar to concept 1, but the types of gears used for the reduction and differential are exchanged. The gear coming off of the electric motor is a spiral bevel gear. The complementary spiral bevel gear is mounted on the same shaft as the helical pinion gear of the differential. The original hypoid ring gear is replaced by a helical ring gear. The bevel gear reduction and the helical gear differential are packaged in one housing.

Concept 3:

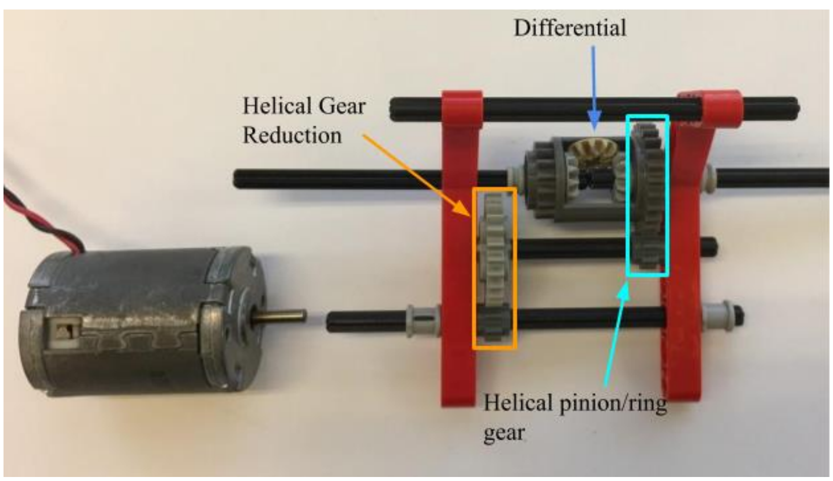

Figure 16. Concept 3: Transversely mounted motor with helical gear reduction and helical ring gear differential.

Concept 3 differs the most from the typical layout of a rear wheel drive vehicle. The motor itself is mounted transversely to the centerline of the vehicle. The first gear from the electric motor is a helical gear, and the complementary helical gear shares a shaft with the helical pinion gear of the differential. Again, the stock hypoid ring gear is replaced with a helical ring gear. The reduction and differential are also packaged in one housing.

Having the physical models helped us determine what bearing placement would look like and where the housing would be split for ease of assembly, casting, machining, etc. Along with that, the Lego models gave a better general idea of how each layout looked and its level of complexity. In the end, these ideation sessions solidified the fact that we wanted to continue to pursue a drivetrain design involving one of the gear concepts.

4.2 Idea Selection

The weighted decision matrix that can be seen in Appendix A.2 illustrates our justification for a total of nine layouts. Each layout concept from above had three variants in the form of how the cast housing was split. The split options were vertical and longitudinal, vertical and transverse, and a split along the horizontal plane. We also compared system functions (such as bearings, gears, and shaft and gear interfaces to support each of the layouts) in Pugh matrices listed in Appendix A.3 – A.6. We then pulled the best components evaluated in the Pugh matrices and combined these ideas into a cohesive system which was assessed via a decision matrix of the various overall drivetrain package.

One criteria that all helically geared layouts excelled in meeting was the ability to handle input power and torque. In the layouts involving at least one set of bevel or hypoid gears, one gear in a set of bevel or hypoid gears will have to be mounted outboard of its respective bearings. Because of this, the shaft may experience more deflection and stress than mounting the gear within the bearings, resulting in a bigger shaft diameter to handle these issues. This still needs in depth analysis to completely prove, but from our previous knowledge of gear systems, this can be an issue depending on the designed sizes for the shafts. As seen in Table 5 below, our top concept is the transversely mounted motor with the helical gear reduction and the helical ring gear on the differential.

Table 5. Top three concepts from weighted decision matrix, also relating to the Lego concepts.

Concept 1	Longitudinal Motor Layout Helical & Hypoid Gears, Vertical & Longitudinal Housing Parting Lines, BT-B LSD	7.36
Concept 2	Longitudinal Motor Layout Bevel & Helical Gears, Vertical & Transverse Housing Parting Lines, BT-B LSD	6.62
Concept 3	Transverse Motor Layout Helical Gears, Vertical & Longitudinal Housing Parting Lines, BT-B LSD	7.97

Concept 3 best meets our specifications and the goals of this project. Tentatively, we are choosing the vertical and longitudinal split for the housing based on the results of our decision matrix. As we get farther along and develop CAD models to view a more detailed layout, our housing decision will be solidified.

4.3 Design Process

Our team will follow the Gantt Chart we developed (Please see Appendix B.1 – B.8 for specific tasks and dates) to stay on track to meet crucial deadlines and continue to make steady progress in our design - the Gantt Chart is described in more detail in Section 5.0. Thus far, we have done a quick calculation to find the necessary gear reduction to meet 155 mph at ground speed. This calculation determined a 6:1 reduction. However, this does not account for the vehicle's 4,500 lb weight. Moving forward with our chosen design, we will begin on more detailed analysis on the gear reduction. Our next calculations will integrate vehicle weight, data from the ST5 engine power and torque curves, and the vehicle’s tire size. The overall gear reduction will be based on achieving the ST5’s performance parameters - 155 mph top speed and 0-60 mph in under 5 seconds. The overall gear reduction will be broken up among 4 gears, in order to keep the gear sizes smaller and create a more compact drivetrain.

Gears will then be sized using an iterative process and equations based on the information in Shigley's Mechanical Engineering Design input into an Excel tool that we developed to meet the following requirements. The gears will be designed to have a life of at least 100,000 miles, and withstand a minimum of 700 lb-ft of torque and a minimum of 400 hp. The shafts and bearings will then be designed with the same minimum 100,000-mile design life and for handling the axial loads developed from the helical gears. Excel tools will also be utilized to run several iterations of these calculations. Based on the loading characteristics, we will determine the appropriate gear and shaft coupling method.

Due to the overall complexity of a differential, we have decided that we will outsource one. Designing a differential would utilize a significant amount of our budget and would require a significant amount of time and technical expertise. This is simply not feasible given the nine-month design timeframe. However, we will be swapping out the stock hypoid ring gear that normally mounts to the differential. It will be replaced with a helical ring gear that will be designed using our Excel tool.

We have begun to develop CAD models, as seen in Figure 16, with equation based geometries allowing a swift iteration process for sizing and testing forces. We will then develop a CAD model that packages our gears, shafts, and bearings into a working drivetrain.

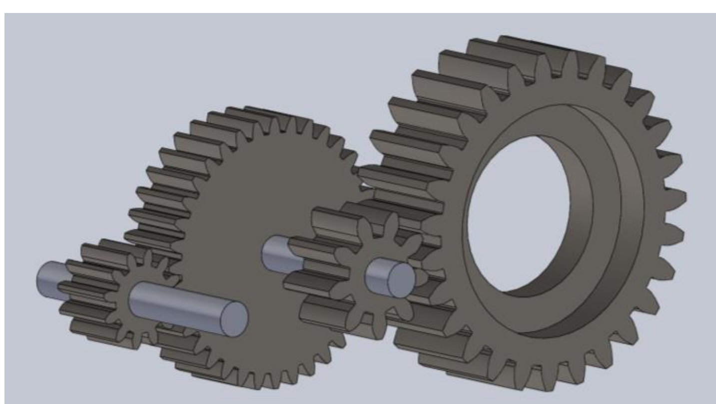

Figure 17. Initial CAD model of layout. Spur gears used instead of helical gears to simplify the geometry in the CAD program.

This CAD model will help us visualize our drivetrain and determine each individual part's location with respect to one another. While this iteration is not close to a final design, it will help to troubleshoot any early issues such as bulky layouts or interference issues, and we can then adapt our model to correct any potential problems. Once all issues have been rectified, we will look into prototyping our gears and shafts via 3D printing. This will be yet another attempt to find any additional design issues. A finalized CAD model will be developed of the gearing system, and our drivetrain housing will be modeled around the gear layout. The housing will then be rapid prototyped, and analyzed. The housing will be scrutinized and changes will be made to the CAD file, if need be.
Areas which require further research will be how the seals and bearings will mount to the housing. Additional research will be needed for the bolts which will hold the two halves of the housing together.

4.4 Construction Plan for Drivetrain

Once the design calculations and CAD models are finalized, we will look into purchasing existing gears, shafts, and bearings that closely match the ones we initially designed. The shaft diameter may need to turned down on a lathe to the match our designed diameter. The gear teeth and shafts will be made of steel and undergo post processing including hardening.

The intermediary helical gears will be mounted onto shafts through splines or keyhole connections. Both mounting options are used in industry, and calculations of torque limits and deflections will be used as criteria for selecting either shaft connection design. The differential used will be an open differential coupled with Bair-Ling Technologies' tolerance rings to convert it to an LSD. We will acquire our differential at the earliest time possible, so that Bair-Ling Technologies has adequate time to incorporate their technology to our drivetrain design.

We will reach out to Martin Koch and rely on his expertise to aid in the design of our housing and ensure that the design is feasible to cast. We will cast the housing in aluminum to cut the weight of the drivetrain down. Post manufacturing will also need to be considered for the design as the housing will need to be mounted in a CNC mill. Once the casting and post processing is complete, we will package the gears, shafts, bearings, seals, and differential into the housing to create our working drivetrain design. The gasket will be designed specifically for our housing design, and the manufacturing will be outsourced.

4.5 Dynamometer Testing Background

To test for our drivetrains compliance with specification 1-4, 6, 8, and 9 we will use a motoring dynamometer. The ST5 vehicle's motor is not yet developed, and so we will utilize a dynamometer as a substitute.

Two categories of dynamometers currently exist to measure force, torque, power, speed, and efficiency of systems. Dynamometers are frequently used in the automotive industry to test the power output of engines, as well as run simulated driving cycles to test the integrity of the engine build. The typical dynamometer is an absorption dynamometer and an example of one is shown in Figure 17. This type of dynamometer is used to develop power and torque curves over a range of angular shaft speeds for a given electric motor or combustion engine. The absorption dynamometer receives an input from a rotary shaft coupled to a motor or any other power producing system and then measures power related parameters with its various sensors.

Figure 18. Electric motor on the left coupled to an absorption dynamometer on the right [19].

The other type of dynamometer is the motoring or driving dynamometer. A motoring dynamometer requires a similar setup, but operates in the opposite manner. The dynamometer instead drives the tested component and the input sent to the component can be varied. As mentioned previously, this dynamometer can model various driving conditions. Many manufacturers develop their own testing criteria and can run their engines continuously for months in a manner that simulates the lifetime of an engine. Quickly ramping up the engine speed and running it at high rpms can wear the engine components, and create feedback on the structural integrity of individual components. A motoring dynamometer can find how much torque and power is required to overcome the coefficient of static friction of the engine to start it and the kinematic coefficient of friction as the motor continues to run. Using this information, the energy losses from the motor and thus the mechanical efficiency of the engine can be determined [20].

Universal or active dynamometers have the capability to be both an absorption or driving dynamometer. Due to the fact that the ST5 drivetrain simply transmits power, and does not create it, we will need to utilize a dynamometer with motoring capabilities in order to test it. Figure 18 shows the layout of how a drivetrain with axle shafts would be tested. We will need to locate a similarly designed drivetrain dynamometer that has two test stands capable of applying loads to the output shafts. The output shafts would be coupled to the test stands and the applied loads on the axle shafts will allow for torque to be transmitted through the drivetrain. Without the added loading, the dynamometer will simply spin the axle shafts without developing the large torque values that we wish to test for.

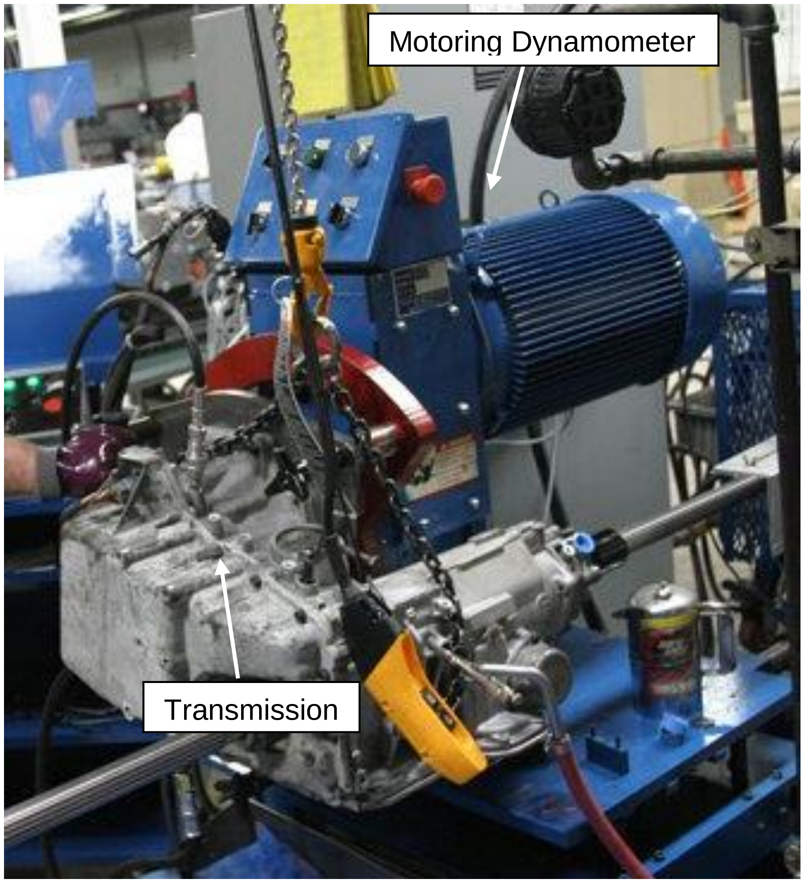

Figure 19. Motoring dynamometer testing a transmission [21].

4.6 Test Plan

To validate our design, we will test our drivetrain on a motoring dynamometer. If our design withstands the specified input values of torque given to us by Zach Sharpell, then our design will be ready to be implemented in the future ST5 (Please refer to the Specification Table in Section 6.2 – Table 8). We will locate and schedule time to use a motoring dynamometer. The dynamometer must be capable of producing the power output of the ST5's theoretical electric motor. While running the drivetrain, we will listen for any sounds that indicate grinding or bad alignment between gears. If the drivetrain operates as designed and in a safe manner, the drivetrain will be tested over its entire 11,000 rpm range (specification 4). The drivetrain will also be tested at the ST5 engine's 700 lb-ft of maximum torque (specification 3) and 400 hp (specification 2) at 6000 rpm. Our exact test plan on the dynamometer will be influenced by the expertise of the dynamometer technician and the dynamometers overall capabilities.

The 155 mph top speed (specification 1) and 0-60 mph acceleration (specification 6) will first be calculated analytically. From there, we will develop the appropriate gear reduction. On the dynamometer we can install the output shafts and measure the output rpm, to validate our 155 mph top speed design calculations. We will measure the rpms of the axle shafts using a laser tachometer. To validate our design calculations of the torque necessary to reach 0-60 mph in under 5 seconds, we will measure the output torque on the axle shafts using strain gauges or torque sensors.

Testing for the torque delivery at each wheel (specification 9) will also be done with strain gauges or torque sensors. This specification requires that the differential delivers 250 lb-ft of torque, at a minimum, to each wheel. The BT-B LSD system must meet this specification. To validate the differential LSD system, the rotation of one of the two axle shafts will be limited, allowing for the differential to distribute power unevenly between the axles. Strain gauges will reside on each output axle shaft and the data from the strain gauges will allow for the torque output to be calculated [22].

The quality of our housing seals (specification 8), will be tested for by filling the housing one quarter of the way full with oil. The housing will be left stationary for a several days, and a container will reside underneath it to catch any leaked fluid. The gasket will be validated if the amount of fluid that leaked measures less than 1 mL. The gasket will not be the only seal in the housing. Since there are three shafts in total exiting the confines of the housing, the final drivetrain assembly will have three lip seals. These lip seals will stop oil from leaking at these shaft exit locations. The seals must be effective, as the housing and oil will not be pressurized and will also be tested and evaluated during our dynamometer test.

The 100,000 mile design life requirement (specification 5) cannot to be tested for since that would require a time frame well beyond our project deadline. However, analysis will be conducted for our design to meet this specification.

Keeping the research and development costs of the total car under $10,000 (specification 7), will be met by developing a cost breakdown of our system early on. Certain portions of our funding will be allocated toward purchasing stock components, raw materials for casting the housing, any services that are to be outsourced, and for testing our completed drivetrain design.

4.7 Design Safety Hazard Identification

Our design hazard checklist is attached in Appendix C.1 – C.2. The drivetrain will not be a component that the consumer will directly interact with. However, the drivetrain will need to be designed with safety in mind, as components will be moving at speeds of up to 11,000 rpm. Due to these high speeds and high loads moving through the gears, the drivetrain components will reach high temperatures. As such, the drivetrain will sit in a bath of oil. The oil will act to reduce the friction between gears and thus decrease the temperature. The drivetrain has the possibility of undergoing large accelerations if the user decides to utilize the ST5's full torque capacity. The drivetrain can also undergo rapid deceleration under heavy braking. The drivetrain must be able to withstand these sudden loads. The drivetrain may be used in an improper manner if the user is operating the engine near redline, or the maximum RPM of the motor. Loud noises may also be generated near the upper rpm range. Other potential safety hazards include the following:

- The housing must also keep the gearing system free of dirt and debris. Contaminants getting into the oil and gears will increase wear and decrease performance. The gasket and lip seals must be designed to keep a tight seal to keep contaminants from entering the housing. These seals must also prevent oil from leaking out of the housing. Oil draining from the housing is an environmental hazard. Also, it would be hazardous to operate the ST5 vehicle if all of the oil had drained out of the housing as the gears would rub and possibly deform due to the heat generated at the tooth interface.
- The bolts joining the two opposing halves of the housing must be torqued appropriately, to ensure that road vibrations do not loosen them. A liquid adhesive will also be applied to the threaded bolt and nut, to help combat loosening.
- Since the ST5 vehicle will be required to be serviced, the drivetrain will be assembled in a secure way that will not fall apart once the housing is unbolted. The technician should not need to replace any of the components, but may be required to drain and change the oil.

4.8 Cost Estimation

A cost analysis was performed to meet the specification put forth from Sharpell Technologies of designing the total vehicle cost less than $10,000. After determining the general layout of a transversely mounted motor with helical gear reduction and helical ring gear differential, the next step in the design process was to derive a general cost estimation. The main components studied included: BT-B LSD, differential carrier cases, spider gear sets, axle bearings and seals, helical ring gears, and intermediary helical gears. Excel was used to document this cost estimation, and pricing was determined from multiple online auto part websites. Multiple differential configurations were considered, including limited slip differential carrier cases that implemented Dura Grip positraction and Trac Loc positraction. The price of these other limited slip options was compared to a BT-B integrated open differential. One potential issue that was discovered is that the differential carrier case could have difficulties being compatible with the Bair-Ling Technology limited slip differential. The issue would be altering the side bevel gears to fit the BT-B tolerance ring and machining a slot for this installation.

A summarized table of the preferred drivetrain layout with major component prices is displayed below in Table 6. A Ford 8.8” open differential carrier case was chosen due to its low price and ease of BT-B LSD integration. A USA Standard Gear spider gear set was chosen due to its compatibility with the Ford 8.8” open differential. For this estimation, an 8.8” helical ring gear was modelled as a 9-⅜” helical gear that could potentially be bolted to the Ford differential. Similar to this simplification, the three intermediary helical gears were chosen to establish a general cost of the final concept layout described previously. The axle bearings and seal kit are also compatible with the Ford differential, and two were priced out giving one set for each side axle.

Table 6. Initial Concept Cost Estimation for Major System Components.

Final Concept Cost Estimation			
Drivetrain Component	Model	Weight [lbs]	Price [$]
Ford 8.8" Differential Carrier Case	YC F880502 - Yukon Gear & Axle	10.59	$140
Spider Gear Set	ZIKF8.8-S-31 - USA Standard Gear	5.2	$120
Axle Bearings & Seal Kit x2	AK 1561FD - Yukon Gear & Axle	2.6	$110
8.8" Helical Ring Gear	2195V LH 92 tooth 9-3/8"		$300
1" Helical Motor Pinion	Boston Gear H1010L Plain Helical Gear		$58
4" Helical Intermediary Gear	Boston Gear H1040R Plain Helical Gear		$185
3" Helical Intermediary Gear	Boston Gear H1030R Plain Helical Gear		$135
Bair-Ling Technologies	BT-B LSD		$75
		TOTAL	$1,123

The total estimated price equaled $1,123 for these major system components, but excludes other components necessary for the completed drivetrain. The other components that were not considered were the costs to cast the aluminum housing, to purchase stock internal bearings and shafts for the intermediary gears, and also the price of other small components including nuts, bolts, washers, and lubrication oil. All these aspects will be looked into further and accounted for in the next cost analysis.

The total price estimated was only 11% of the total budget and our team believes that this represents of the bulk of the maintain component costs. Future costs will not only involve materials and components not yet specified, but also labor and other miscellaneous items that will be budgeted throughout the remaining design process. In comparison to the drivetrain developed by BorgWarner (the 31-03 eGearDrive gearbox shown previously above in Figure 9) that costs a total of $2,995, our initial estimate seems reasonable.

5.0 Management Plan

To ensure the success of our project the following roles have been assigned. These positions are flexible and have the ability to change each quarter. Team roles shall include those listed below with their designated responsibilities and shall have the support of the other team members to accomplish the work.

CAD Lead – Jimmy King

a. Main designer of SolidWorks parts and assemblies.
b. Integrate other team member's CAD parts into assemblies.

Communications Officer – Jimmy King

a. Be main point of communication with Sponsor.
b. Facilitate meetings with Sponsor.

Editor – Charissa Seid

a. Compile and edit reports.
b. Describe analysis and method of design.

Manufacturing Lead – Kevin Moore

a. Prototype components using 3D printing.
b. Seek out assistance from IME department to cast the drivetrain housing.
c. Organize shop time at Mustang 60 and The Hangar.

Meeting Coordinator – Kevin Moore

a. Create an agenda to detail topics of discussion for upcoming meetings.
b. Scheduling time and location of out-of-lab meetings.

Secretary/Recorder – Charissa Seid

a. Maintain information repository for team on Google Drive.
b. Document all sponsor discussions.

Team Treasurer – Charissa Seid

a. Maintain team's travel budget.
b. Maintain team's materials budget and ordering of parts.

With these set roles, we should be able to work in an organized and efficient manner, allowing us to complete all tasks by their due dates. The major deadlines in which deliverables are due are listed below [13].

October 25, 2016	Project Proposal
November 17, 2016	Preliminary Design Review
February 7, 2016	Critical Design Review
March 16, 2016	Project Update Report
May 2, 2016	Project Hardware/Safety Demo
June 6, 2016	Project Expo
June 6, 2016	Hardware Handoff
June 6, 2016	Final Design Report

Our Gantt Chart (Appendix B.1 – B.8) will keep us on track for the duration of this project. There are some major milestones we need to achieve in order to be successful in completing the drivetrain. The milestones are as follows:

Design Development
This milestone could potentially be the most important one we need to hit. It involves all of the analysis of the gears and shafts, including creating the CAD model for the entire system. Without completing this phase, we cannot move forward to the rest of the project.

Assemble Prototype
During this phase, we will see if the geometries we specified are correct. We will 3D print parts for the prototype model before we spend money on the actual parts. This will enable us to catch any flaws in the assembly of the drivetrain. We will ensure that the gears are meshing properly and are not clashing with the housing, bolt holes on the housing lining up, and the shaft align properly. If issues arise, we will adjust the CAD model and redo calculations as needed.

Cast Housing
In order to test the drivetrain, we must have a completed housing. None of us have enough experience to tackle this milestone on our own, so we will utilize some resources we have on campus. A professor in the IME department, Martin Koch, has knowledge of casting aluminum parts and designing parts properly to use this manufacturing method. We plan on reaching out to Mr. Koch immediately and have given ourselves 3 weeks, according to the Gantt Chart, to cast and machine the housing, knowing that we will likely have only one chance to manufacture this part.

Test Prototype
This will be the final way of ensuring our designed drivetrain meets the required specifications. A dyno test will cover the majority of the tests needed to validate our drivetrain. We will most likely have to travel out of town in order to find a company that performs the dynamometer tests for our high speed application. Once testing is complete, we can move on to evaluating our design based on its performance.

Delivering the Design
We will deliver a final report documenting our entire process, which includes all information we will have acquired, and our final design to our sponsor, Zach Sharpell. In addition, we will deliver rough prototypes and the molds for the housing. We hope he can incorporate our drivetrain in the ST5 successfully.

6.0 Adjusting to New ST5 Design

6.1 Changes to Design Specifications

Originally, our team was given specifications for a powerful motor; one that could output 700 lb-ft of torque. Several of our design specifications were focused around performance, which would be achievable with this powerful motor - an under 6 second 0-60 time as well as a top speed of 155 mph. Other specifications were created to ensure that the gears could withstand the motor’s maximum output speed and torque. We had known that our sponsor did not have this actual motor, and that these specifications were largely theoretical of future motor technology. To achieve these performance specifications, our drivetrain design was determined to have an overall gear reduction within the range of 5.5:1 to 5.76:1. However, our sponsor notified our team of a major change in the overall drivetrain design of the ST5, that would directly affect our overall gear reduction ratio. The change to the ST5 prototype is that it will now be a dual motor design, using a less powerful motor at both the front and rear axles. Each motor would be capable of producing 310 lb-ft of torque. The motor that will be coupled with our drivetrain is the REMY HVH250-115 motor that is depicted in Figure 20. Our drivetrain design will still be coupled with a single motor, and then will be later duplicated for the motor to be located on the front axle. This major change was a result of a change in Sharpell Technologies' market strategy. Sharpell Technologies felt that the ST5 vehicle would no longer have lucrative access to the affordable electric vehicle market due to the increasing amount of affordable, electric vehicles being developed by other major automotive manufacturers. Instead, the ST5 is now to be a performance vehicle, with a price-tag in the range of $250,000. Their customer acquisition strategy includes going to various track days, and showing off the performance capabilities of the ST5. This will help garner attention to the brand as well as allow customers to get an up close look at the ST5.

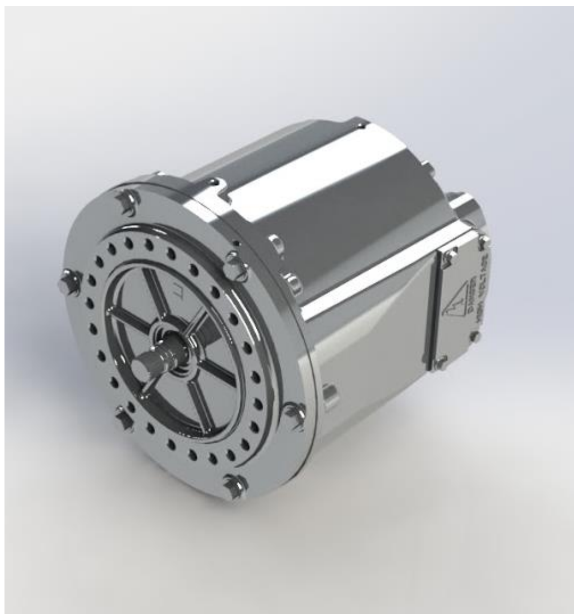

Figure 20. REMY HVH250-115 Motor

Although the final design of the ST5 vehicle will be performance oriented, the main focus of the ST5 prototype vehicle is to test the battery technology as well as the vehicle's control systems. Our drivetrain design is to be incorporated in the prototype and will now be designed accordingly to the output of the REMY HVH250-115 motor. Our sponsor directed us to design the drivetrain for the single motor, and not take into account that it will develop into a dual motor drivetrain. With this in mind, the gear ratio was increased to fall within the range of 8:1 to 8.5:1. This is because the new motor develops less torque, and the higher ratio will help to increase the torque at the wheels to achieve a quicker 0-60 mph acceleration time. Running calculations for this 0-60 mph acceleration time with an 8:1 gear reduction ratio yields a 0-60 mph acceleration time of around 7.2 seconds. The 8.5:1 gear reduction ratio yields a 0-60 mph acceleration time of around 6.8 seconds. (Reference ST5 Vehicle Performance Calculations) The driving factor for the acceleration is the REMY HVH250-115 motor's torque and horsepower output. The solid blue line shown in Figure 21. graphically depicts the peak output torque of the

REMY HVH250-115 while supplying 350V to the motor. This is the performance curve we are designing to, as instructed by Sharpell Technologies.

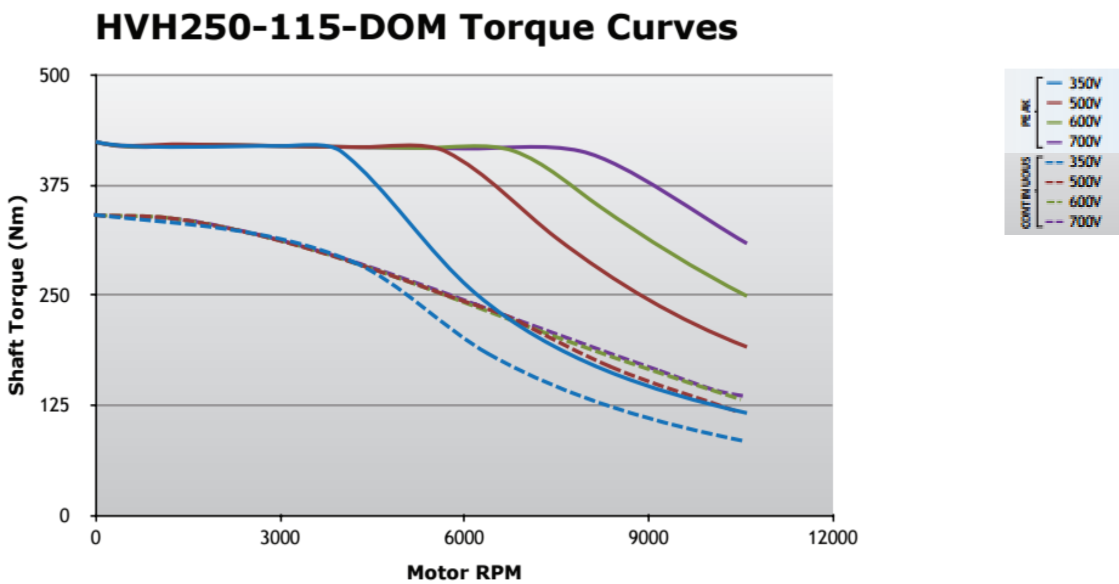

Figure 21. Torque output curves for the REMY HVH250-115 motor at various input voltages.

While these acceleration times do not meet our initial "under 6 seconds 0-60 mph acceleration specification," the ST5 will surely be able to do so when the vehicle is driven by both motors. The addition of the second motor at the front axle will allow for power to be transmitted to the front wheels, which were normally not driven. This additional power will accelerate the ST5 well under the 6 second mark. These 0-60 mph acceleration times were calculated utilizing the motor's torque output at various speeds, the overall gear reduction, as well as vehicle's effective mass which is determined through the ST5's physical parameters. Appendix D.1 – D.4 contains the analysis associated with these vehicle performance calculations.

The factors that contribute to the vehicle's effective mass is the overall vehicle's mass, tire size, and estimates of the rotational inertias of the drivetrain/axle and tires mounted on the wheels. The motor's rotational inertia is known to be 0.086 kg-m^2 as it is given from the supplier. A byproduct of the less powerful motor and increased gear reduction is a change to the vehicle's overall top speed. The initial 155 mph top speed is no longer achievable with the increased gear reduction being the limiting factor. The 8:1 gear reduction ratio yields a top speed of 112 mph and the 8.5:1 gear reduction ratio yields a top speed of 105 mph. A summary of the effect of the ratio on our performance parameters are tabulated below, in Table 7.

Table 7. Comparison of performance at the upper and lower limits of our desired reduction values.

Performance Specifications		
Reduction	0 - 60 mph acceleration time [seconds]	Top Speed Limit [mph]
8:1	7.2	112
8.5:1	6.8	105

The addition of a motor on the front axle, will not have the effect of increasing the top speed, as both identical gear reductions will limit the wheel speeds. The decreased top speed is not much of a concern to our sponsor. We were assured from our sponsor that the prototype will not be driven at speed above 100 mph.

Sharpell Technologies has purchased a BMW 330Ci (Figure 22) to serve as the earliest ST5 prototype, as the ST5 body and chassis construction has not been developed yet. The interior will be gutted and retrofitted with Sharpell motor controller and battery technology, the REMY HVH250-115 motor, and our completed drivetrain.

Figure 22. BMW 330Ci that will be the first application for the completed drivetrain.

6.2 Updated Specification Table

Table 8. Updated specifications adjusted for REMY HVH250-115 motor output. The reductions listed beneath where used as a good beginning range to shoot for while playing with various gear selections.

Spec #	Specific Description	Target [Units]	Tolerance	Risk	Compliance
1	Wheel speed for 100 mph top speed	< 9:1 Gear Reduction	min	M	A,S, T
2	Input Power	305 [HP]	min	M	A,T
3	Input Torque	400 [lb-ft]	min	H	A,T
4	Rpm Range	0-11,000 [rpm]	min	M	A,T
5	Reliability	100,000 [mi]	min	H	A
6	Achieve 0-60 acceleration time of 8 seconds or less	> 6.5:1 Gear Reduction	min	L	A
7	Cost	$10,000	max	L	S,I
8	Housing Leakage	0	+ 1 ml in 1 day	L	T,I

Key
A = Analysis
S=Similarity
T = Test
I = Inspection
Determined by Detailed Analysis

The design specification for the minimum torque applied to a wheel was removed. We are no longer utilizing the BT-B LSD system, which can be designed to always engage both wheels to a pre-determined minimum torque output. Instead we are incorporating a clutch type LSD in the form of a Ford 8.8” Traction-Lok differential. Since we have no involvement in the design of this pre-existing differential, it is a specification that is outside of the scope of our design. However, the incorporation of a LSD does meet the customer requirements that we are designing to meet. A more in-depth discussion about the differential can be found in section 7.1.5.

7.0 Description and Justification of Final Design

Our final drivetrain design features a two-stage reduction. The two stage reduction allows for the drivetrain to achieve an 8.04:1 gear reduction ratio in a relatively compact design. Figure 23 below shows a CAD model of our drivetrain which is coupled to the motor for size comparison.

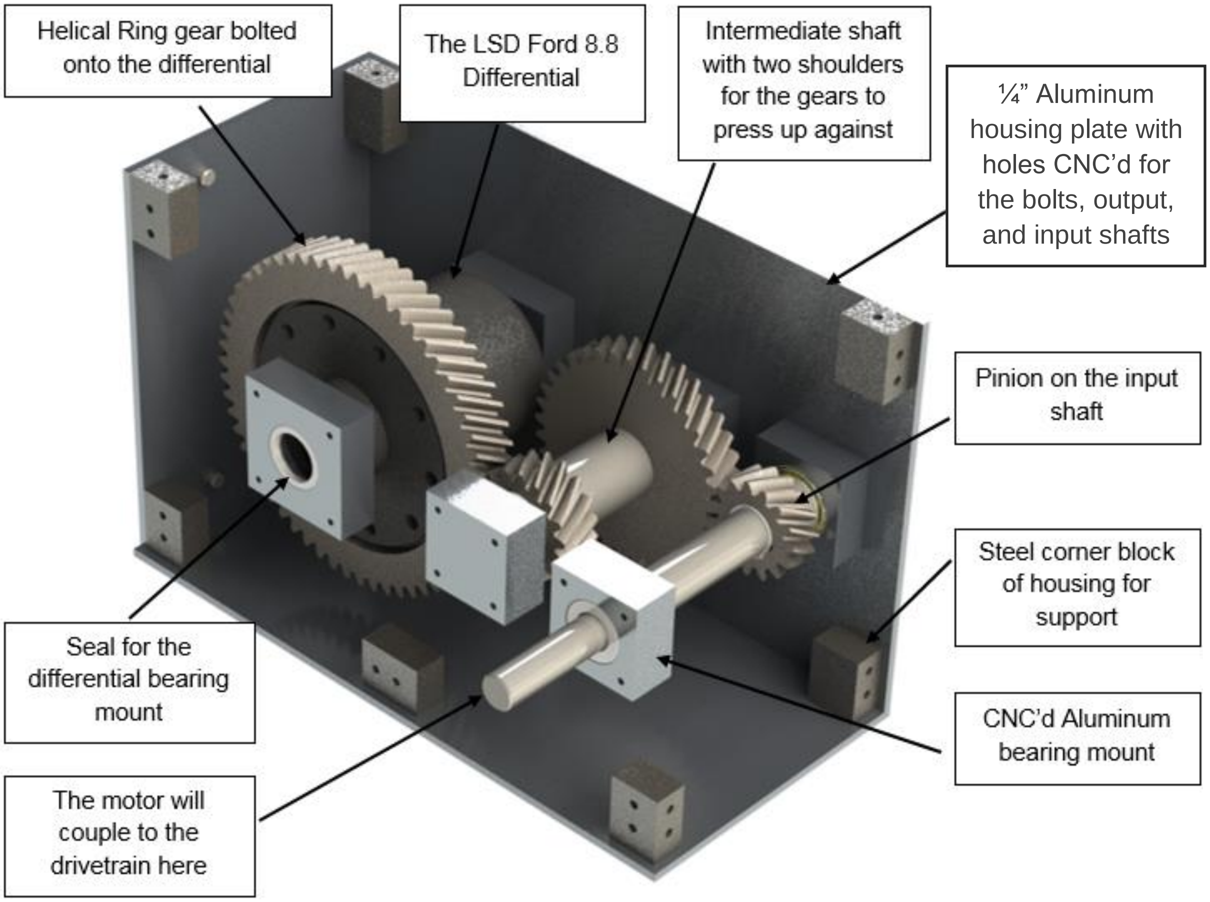

Figure 23. This figure displays our drivetrain layout including the four helical gears and the input shaft protruding from the housing.

The gears are able to handle the specified input torque of 400 lb-ft and increase this torque across the two gear sets. They are pressed up against the shoulders of the shafts on one side and held by a snap ring fit into a groove in the shaft on the other. Additionally, keys are set into slots to fix the gears orientation and keep them from freely spinning when a force is transmitted through the drivetrain. The shafts support the helical gears and keep them aligned and in the proper locations so that the teeth will mesh. The bearing blocks hold the bearings and were sized for a perfect fit between the two interfaces so the bearings would hold the shafts and mated gears in place. Tapered roller bearings help support the axial load and allow the shafts to spin freely. Seals were set in the counterbores next to all the output shaft locations to seal the oil in the drivetrain and keep debris out. The housing plates were used to enclose the drivetrain and provide the backbone in which all the smaller components could be fixed to and held in place. Corner blocks are used to thread into and mechanically join the housing plates together and fill and drain plugs for the oil are located on the back face of a housing plate. Thorough reasoning with quantification and analysis for each component of the drivetrain will be discussed in the next section.

Please see Appendix E.1 – E.20. for detailed CAD drawings for all the parts manufactured.

7.1 Component Selection

In the following sections, the design of each critical component will be discussed.

7.1.1 Gear Selection

After determining the requirements our drivetrain must meet, we began sizing the four gears needed for the overall reduction. In walking through the design process of the helical gear design, it is best to use a chronological order to discuss the iterations that occurred.

Appendix D.5 – D.11 provides a more detailed explanation including the reasoning and theoretical equations used to develop the iterative Excel tool which was employed for the gear selection and helped develop a systematic process. The AGMA strength and stress equations used to design the gear reduction are also referenced within this appendix. Also note that the gears were designed based off of a transverse diametral pitch versus a normal pitch diameter. These design factors, according to ANSI/AGMA standards 2001-D04 and 2101-D04, guard against bending fatigue failure and against pitting failure.

Several other constraints provided starting diameters for the pinion and helical ring gear. The helical ring gear size was dependent on the dimensions of the Ford 8.8 differential carrier. The mounting location has a diameter of 6.75 inches, which constrained the minimum size for our ring gear to be at least 8.5 inches in diameter to allow for the large bore at the mounting location. The ring gear bolting distances will allow the stress concentrations around the bored holes to dissipate, and not affected the forces transmitted through the gear.

Another factor in finding the appropriate gears for the drivetrain was maintaining an equal transverse diametral pitch (P_d) between the gears in both the first and second set to ensure that the teeth mesh between the two. The smallest transverse diametral pitch that is recommended is 6 teeth/in and was utilized in the motor pinion gear. For the second step of the gear reduction, the same transverse diametral pitch of 6 teeth/in was used and the total two step gear reduction was set at 8.04:1.

The minimum pinion gear diameter had to ensure that it could accommodate a bore equal to the size of the input shaft, which depended on the stresses the drivetrain had to handle. Based on the constraints on the helical ring gear and pinion gear we were able to begin the iteration process.

Originally, after developing the Excel tool we assumed the maximum torque of 700 lb-ft would occur throughout a majority of the drivetrains lifetime which gave us incredibly small safety factors for both wear and bending. This meant that the gears needed to be of enormous diameters and facewidths to handle the amount of stress. However, as our analysis progressed it was determined that our conservative estimates for lifetime and the duration that the car would operate at these conditions stacked. We proceeded to update the gear excel tool allowed for the more reasonable operating cases while maintaining valid design factors which allowed us to decrease the size of the gears and thus the cost.

In programming these spreadsheets numerous assumptions and estimations based off of AGMA graphs were made, which originally resulted in low factors of safety (around 0.6 and 0.7 for contact and bending respectively).

The quality factor we originally had set was a Qv of 7. However, this is only the upper end of commercial gearing. Since we are developing a drivetrain for a high end SUV we changed this value to 10 to reflect the precise gearing needed to transmit the power from the motor. We also updated J, the form factor parameter to the new diametral pitches chosen for each of the gear ratios. This slightly decreased the factors of safety but they still were greater than 1.

By increasing the Brinell hardness factor by changing the material from 4140 to 8620 steel again both the wear contact safety factor and the bending safety factor increased. To minimize the weight of the drivetrain, it was desirable to minimize the face widths of the gears. Being that face width is instrumental in numerous design coefficient factors, there was a minimum length needed to achieve acceptable safety factors necessary for a quality final design.

There were other various modifications made to our original assumptions to help design to an operating point that reflected the actual input parameters for the gear design. A major change made when improving the Excel tool was to decrease lifetime factor. In our original safety factor calculations, we were designing to the max torque, max revolutions per minute (rpm), for the entirety of the drivetrain's lifetime. As described previously discussed, this worst case scenario was not realistic, and thus the assumptions were re-evaluated.

Shortly thereafter, we were notified that the motors purchased would only output a maximum of 400 lb-ft. Since the motor will only output that much torque for a very small portion of its lifetime we decreased the input motor torque to 150 lb-ft for a more realistic design. We also ensured that the drivetrain can with stand this maximum stress generated for the shorter lifetime of 10,000 revolutions at 99% reliability.

For the prototype vehicle we changed the overload factor, K_o to uniform shock. The effect of this change was that the bending stress decreased proportional to the change from K_o equaling 1.25 to 1.0. Originally, the conservative medium shock overload factor demonstrated the vehicle was going to be used in more frequent off-road driving but the new vehicle will mainly be used on the road. The load distribution factor, K_m, which is influenced by the face width of the gear, was adjusted as well after the face widths decreased based on the new strengths gained from post processing the gears.

After appropriately manipulating the coefficients, the diameter and tooth count of each gear were determined. This allowed cross checking with the Excel bearing and shaft tools to ensure all the minimum requirements of bore sizes, baseline geometries, and stress factors were met. The final selections are shown in the table below.

Table 9. The final geometry selection for the helical gears in the drivetrain. All the gears are made from HR 8620

	Pinion	Primary Gear	Secondary Gear	Helical Gear
Tooth Cut	RH	LH	LH	RH
# Teeth	15	36	17	51
Face Width [in]	1	1	1.25	1.25
Transverse Diameter [in]	2.5	6	2.83	9.5

	Safety Factor Bending	Safety Factor Contact
Pinion & Secondary Gear	1.5	4.55
3rd Gear & Helical Ring Gear	1.1	2.9

Our final factors of safety for bending and wear are listed above between each other gear stages.

7.1.2 Right Hand versus Left Hand Gear Cut:

After viewing our system layout, it was determined that we wanted the gears to press up against the shoulders in the shafts to minimize the load the thrust bearing would carry. Since the direction of the thrust is dependent on which way the motor pinion spins we chose gears that would push up against the shoulders of the shafts when the car is moving forward since much more time will be spend in that spin than in reverse.

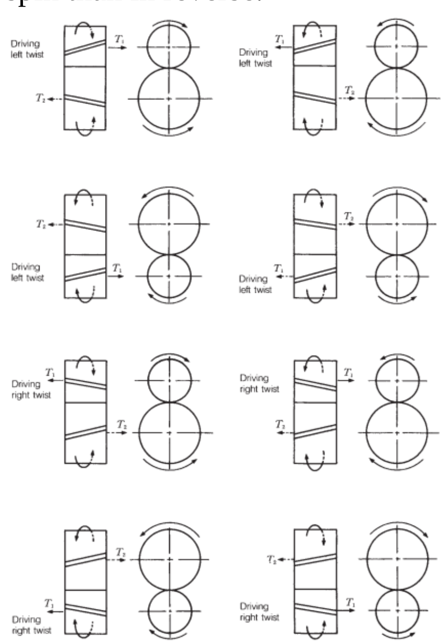

Fig. 4 Thrust direction

Figure 24. Determining RH vs LH cut of helical gears based on thrust load direction [25].

With these geometries, we then researched various AGMA gear manufacturers to try and find companies that had the stock sizes we wanted as well as the capabilities to CNC the custom helical ring gear. We first attempted to incorporate only stock gears to reduce the cost of our design, but after listing out the all constraints that simultaneously needed to be met using only stock gears was not a feasible option. We also briefly considered manufacturing the prototypes in house but concluded we do not have the equipment or skills required to machine the gears ourselves within the tolerances needed for an automotive application. Rush Gears was selected to manufacture the 4 helical gears in the drivetrain as they carry a variety of stock gears to choose from and can be modified with the proper bore sizes and keyways which helped decrease the cost instead of going fully custom. They also are able to make a completely custom gear which will be needed for the helical ring gear and the primary gear since the diameter and tooth count are not typical numbers. We will also have to modify the bore sizes according to the shaft diameters which will be another customization.

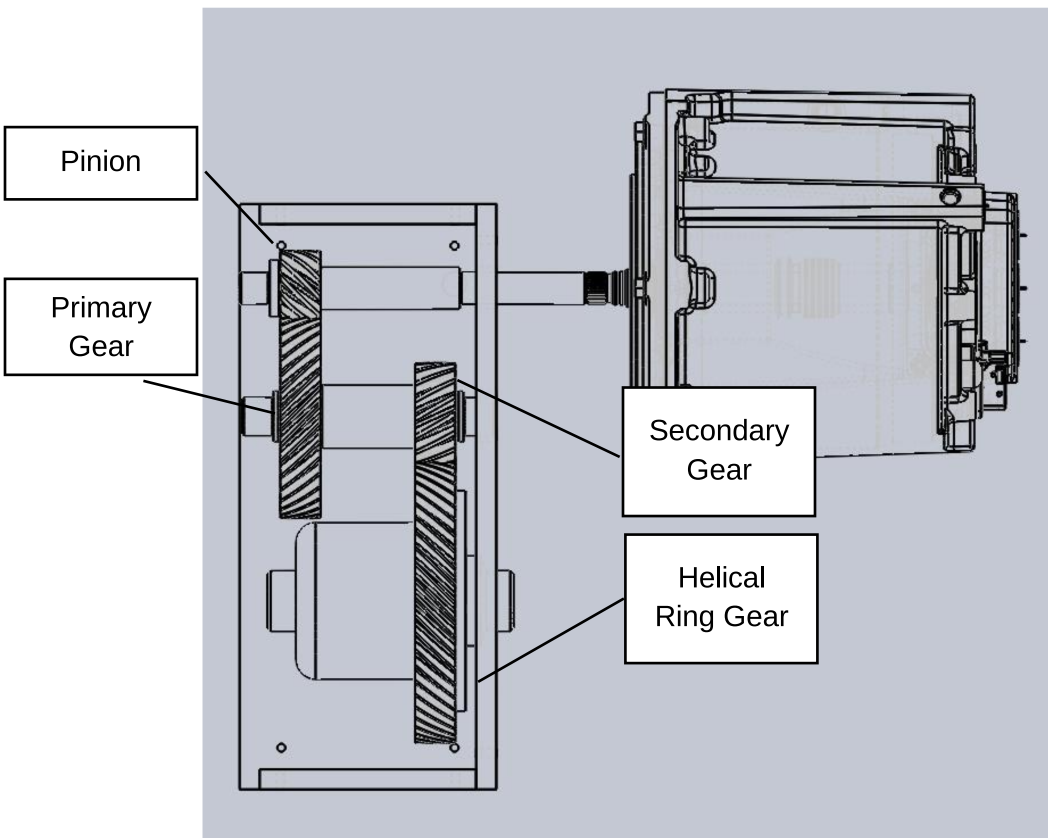

Figure 25. The layout of our final gear geometries in relation to the motor.

7.1.3 Gears Material Selection Justification:

The material we selected for our gears was 8620 steel. 8620 is a common alloy already used in many automotive, oil and gas drilling, and tool making applications. The reason behind this selection was due to the high strength, toughness, and fatigue resistance required for our gears.

7.1.4 Gear Processing:

In order to ensure our gears can handle the forces as discussed during the Excel verification of gear choices, proper material processing to increase the hardness will be specified for the manufacturing company.

Brinell Hardness	Rockwell Hardness			Tensile Strength (Approximate) Psi
Tungsten Carbide Ball 3000 KG	A Scale 60KG	B Scale 100KG	C Scale 150KG	
656	81.3	-	60.1	-
653	81.2	-	60.0	-
647	81.1	-	59.7	-
638	80.8	-	59.2	329,000
630	80.6	-	58.8	324,000
627	80.5	-	58.7	323,000
601	79.8	-	57.3	309,000

Figure 26. A Brinell hardness of about 627 for 8620 will be needed for all four of the helical gears [31].

The gears will be case hardened by the manufacturer to ensure the desired strength, ductility, and hardness for each gear. In this manner the material can handle the specific load in that portion of the drivetrain. Case hardening the gears requires re-heating the gear after it is cut to 780^{o}C - 820^{o}C, hold until temperature is uniform throughout the section, and then quench it in oil. The RC of the gears ordered are between 58-62 RC with the minimum Brinell hardness of 627.

Part# 329-2669

QTY	Net each	Total	>Standard< Prod. Time
1	$3350.00 ea	$3350.00	3 WEEKS
2	$1675.00 ea	$3350.00	3 WEEKS
4	$1507.50 ea	$6030.00	3 WEEKS
6	$1340.00 ea	$8040.00	3 WEEKS
10	$1172.50 ea	$11725.00	3 WEEKS

Description: MADE TO ORDER, 8620 CASE HARDENED (58-62 RC) STEEL, HELICAL GEAR, 6.000 DP, 20.0 PA, 57 HOBBED TEETH, 1.250" FACE, 5.125" BORE, A TYPE, 9.5000" PD , 9.789" OD:, 30 DEGREE, RIGHT HAND

Figure 27. Above is displayed one of the quotes for the helical ring gear from Rush Gears. The Rockwell Hardness corresponds to the desired Brinell hardness for 8620 and will be needed for all four of the helical gears as is noted in the description.

7.1.5 Shaft Design

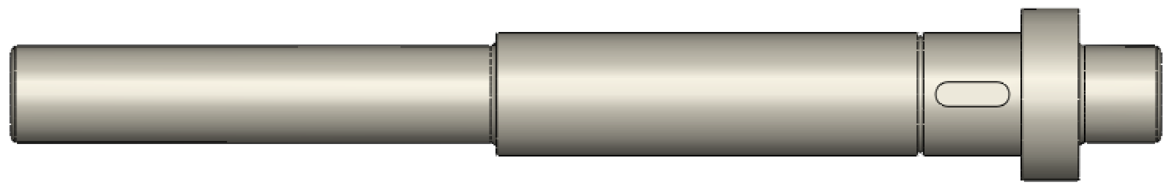

Figure 28. CAD model of input shaft.

To determine the shaft diameters of the input shaft in Figure 28 above and the intermediate shaft, many, smaller details had to be considered. After calculating the tangential and radial loads from the gears based 150 lb-ft of input torque, we developed shear and moment diagrams to determine the areas of highest stress for each shaft. Specific shaft features have a great effect on what the minimum diameters can be; therefore, several stress concentration factors also had to be taken into account when designing the two shafts. The first of these stress concentration factors was a shoulder fillet. On both shafts, there were shoulder fillets where the shaft is located by the bearing. However, due to how small the moment was toward the ends of the shaft, these locations were not a likely failure point. Shoulder fillets were also used to locate the gears. These points were more critical because the radial and tangential gear forces acted in close proximity to the shoulder causing greater moments. Another stress concentration factor we needed to look at was the keyway. While there is a reduction in material, the gear forces acted directly through the keyways causing a maximum moment on the motor shaft and local maxima at the two keyways on the intermediate shaft. Last, ring grooves used to retain the gears were also an important feature to look at due to a reduction in material and small fillets that caused high stress areas.

We used the distortion energy failure theory with the Goodman criteria to find the minimum diameter at the gear locations for each shaft. Goodman criteria was chosen because it was in between the more conservative Soderberg and the less conservative Gerber criteria estimates for the shaft diameter. For quick iterations, an Excel tool was created to calculate shaft diameters based on Goodman criteria and allowed us to spend minimal time calculating numbers and more time focused on the smaller details associated with this kind of analysis. Appendix D.12 – D.16 shows more detail on what needed to be calculated and then inputted into the Excel tool. Upon calculating for the motor shaft and intermediate shaft, the minimum allowable diameters with a safety factor of 1.5 were 1.28 inches and 1.56 inches respectively. On the motor shaft, the ring groove was the most critical point, giving the limit on the minimum diameter. The intermediate shaft had the critical point one of the keyways. The shaft itself is symmetrical, but the smaller gear that mates to the ring gear has higher forces imposed on it and allowed us to analyze only one side.

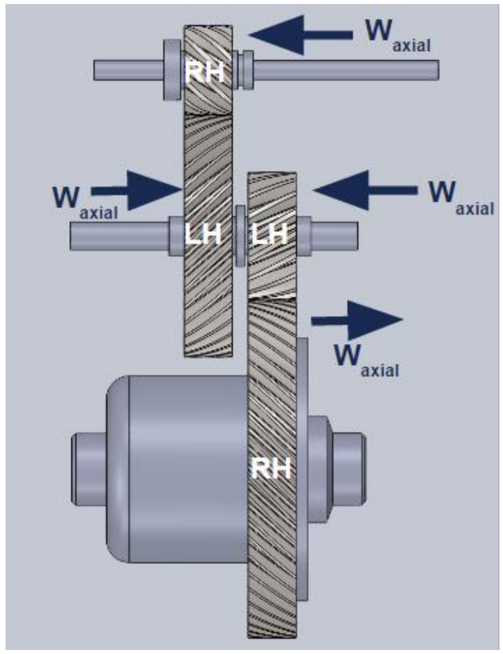

Figure 29. When operating in the forward direction, the axial loads from the gears will point towards the shoulders, providing enough support to constrain the shafts.

As noted, the helical gears impose axial forces on the shoulders of their respective shafts as seen in Figure 29. This is true when operating the vehicle in the forward direction, and when in reverse, the axial loads switch direction, causing a need for containment of the gears. To accomplish this, retaining rings will be used to constrain the gears on the shafts. In reverse operation, the drivetrain would be operating at a low speed and low torque scenario, applying relatively low axial loads. Upon simple shear stress calculations, we concluded that a retaining ring similar to Figure 30 would be sufficient in retaining the gears.

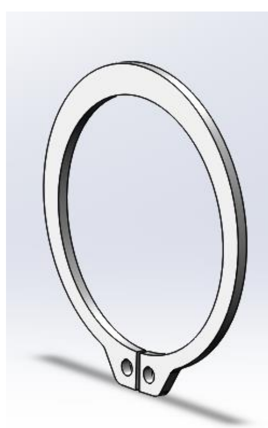

Figure 30. A retaining ring would be used to constrain the gears when the drivetrain is operating in the reverse direction because of the change in direction of the axial loads.

To fix the rotation of the gears with the shafts, we designed the shafts to use keys in keyways. Generally, keyway width is based on shaft diameter according to a table of ANSI standards. For our purposes, the widths of the keyways were based on 1-5/16" and 1-9/16" shafts. The stock key dimensions can be seen in Appendix F.1 – F.2. To calculate the key strength, a simple shear stress equation, $\tau = F/A$, was used. The force at the key was calculated by dividing the respective shaft torque by the radius from the center of the shaft to the base of the key. The area was calculated by multiplying the width of the key by the length of the key, which is the same as the face width of the respective gear in our case. After using the von Mises strength criteria to adjust the shear stress into a comparable tensile stress value for each keyway, the stress came in under the yield strength of 1018 steel, a common steel grade used for key stock.

We have decided to use 4140 HT steel for the shaft material. Round stock is readily available from online distributors, and this material boasts high tensile and yield strength. More detailed specifications can be seen in Appendix F.1 – F.15.

7.1.6 Bearing Selection

Bearings sizes were determined for both the input and intermediate shaft, as the differential already has its own bearings that are properly sized to fit the differential carrier. After the gear sizing and shaft lengths were determined, ball bearings and tapered roller bearings were examined. These bearings types were chosen for this application due to their prevalence in automotive drivetrains, and due to our team having previous experience designing ball and tapered roller bearings.

The forces at the bearing locations were calculated using the motor's torque input, gear sizes, and location of the gears on the shafts. The transmitted, radial, and axial loads were converted into an equivalent load which was used to calculated the C10 values for ball bearings and tapered roller bearings. The C10 value is a dynamic load rating, that is used to specify the upper limit of dynamic loading a given bearing can withstand, and is used to select bearings from a manufacturer's catalog. The first iteration of our gear design had caused large forces at the bearing locations, that were beyond the capability of available ball bearings. After lowering the helix angle to decrease the axial forces, the forces exerted on the bearings still were not manageable. The ball bearing's calculated C10 value still remained high after the helix angle changes, and would require shafts that are greater than 3 inches in diameter. So, it was decided to move forward with tapered roller bearings which can more adequately handle axial loads. Appendix D.17 – D.22 contains the analysis for determining the C10 ratings for both ball and tapered roller bearings.

The life of the bearings was determined based off of the vehicle's 100,000 mile warranty. Using the circumference of the tire, the output shaft's revolutions were found. The design life of the intermediate shaft, and input shaft were multiplied by the reduction of the gears. This resulted in the input shaft having the greatest design life, followed by the intermediate shaft.

Due to the variability of motor output and driving styles, we found that we would need to scale the life based on the amount of torque output from the motor. In other words, the ST5 vehicle will never be driven 100,000 miles utilizing the full 400 lb-ft that we are designing to. The maximum output of a motor is rarely used, so we determined that the bearing must only be able to handle this loading condition for 1% of the total 100,000 mile design life. Table 10 shows the various lifetime percentages that the bearings must handle at various motor outputs.

Table 10. C10 Dynamic Load Ratings Based on Different Life and Loading Conditions.

Motor Output Torque	Percentage of Lifetime	Lifetime	Input Shaft Design Life	Intermediate Shaft Design Life	Input Shaft Largest C10	Intermediate Shaft Largest C10
lb-ft	%	miles	revolutions	revolutions	kN	kN
400	1	1000	6.27E+06	2.09E+06	42.0	44.6
300	3	3000	1.88E+07	6.27E+06	43.4	46.6
200	10	10000	6.27E+07	2.09E+07	41.5	44.5
100	86	86000	5.39E+08	1.80E+08	39.6	42.5

The input shaft bearings must have a C10 value greater than 43.4 kN and a limiting speed greater than or equal to 11,000 rpm. The intermediate shaft needs a bearing with a C10 greater than 46.6 kN and a limiting speed greater or equal to 3667 rpm. SKF was chosen to be our bearing supplier, as their catalog has displays the limiting operating speed, unlike many other manufacturers. The tapered roller bearing chosen for the input shaft is SKF's 32305 J2 and the intermediate shaft will use SKF's 33206/Q tapered roller bearing. The SKF 32305 J2 tapered roller bearing (Figure 31) has a limiting speed of 12,000 rpm, and a dynamic load rating of 60.5 kN. This bearing will be installed at both location A and B, as shown in Figure 32. SKF's 33206/Q tapered roller bearing will be installed at location C and D, and has a limiting speed of 11,000 rpm and a dynamic load rating of 64.4 kN. More detailed specifications of these bearings can be seen in Appendix F.14 – F.15. Although both bearing locations on each shaft do not see the same force (axial load only seen at one location), both locations will use the same bearing which is rated to handle the highest C10 dynamic load rating. This will ensure that the drivetrain can handle axial loads at both bearing locations, as the axial loads will shift from A and C while driving forward, to B and D while driving in reverse.

Figure 31. SKF 32305 J2 Tapered Roller Bearing to be used on the Input Shaft [26].

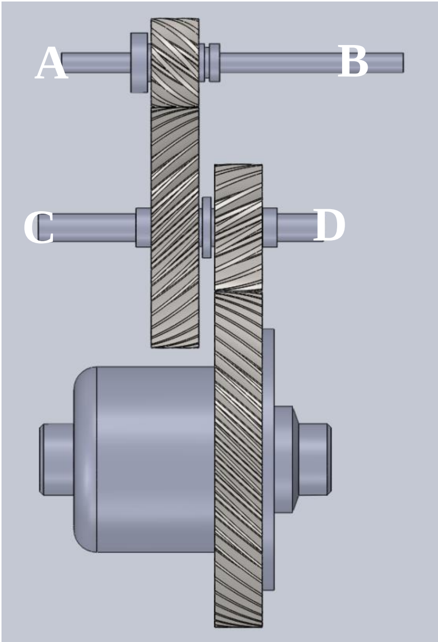

Figure 32. Locations where tapered roller bearings are being utilized.

7.1.7 Bearing Housing

Utilizing a flanged bearing housing will allow for the bearings to be seated in a very secure manner. The flanged housing will also be used to locate the shafts in their respective locations. Having a proper shaft alignment will help to ensure that the contact between the meshing gears is aligned properly. Some flanged bearing housing have seals integrated into their design, which would be beneficial at our input shaft location. A typical flanged bearing housing is depicted below in Figure 33. Bearing housings will allow for a decreased housing wall thickness, since we will no longer have to locate the bearings within the thickness of the housing wall.

Figure 33. Mounting bracket of a flanged bearing housing [27].

Flanged bearings housings typically have a shaft collar within the bearing retained in the housing. This further restricts the size of your shaft for a given bearing bore size. As such, we are planning on designing our own flanged bearing housing. We will start with a small billet block of aluminum and will use the CNC on campus to create our four flanged bearing housings. We will also design additional flanged bearing housings for the differential to mount to the housing, and these bearing housings will be reinforced due to the heavy weight of the differential.

7.1.8 Differential

We acquired a Ford 8.8" differential for no cost, courtesy of Blair-Ling Technologies – shown in Figure 34. The differential supplied to us was from a donor vehicle with roughly 110,000 miles on it. Due to the relatively high mileage on the differential, it was decided that we will swap out the bearings with new replacements. Additionally, the spider gears and pinion gears of the differential will need to be replaced with gears that designed to retain an Independent Rear Suspension (IRS) axle set-up. The spider gears will need to have 31 spline connections, if we plan on utilizing 31 spline axle shafts. The axle retainers will also need to be for an IRS set-up and not solid axle.

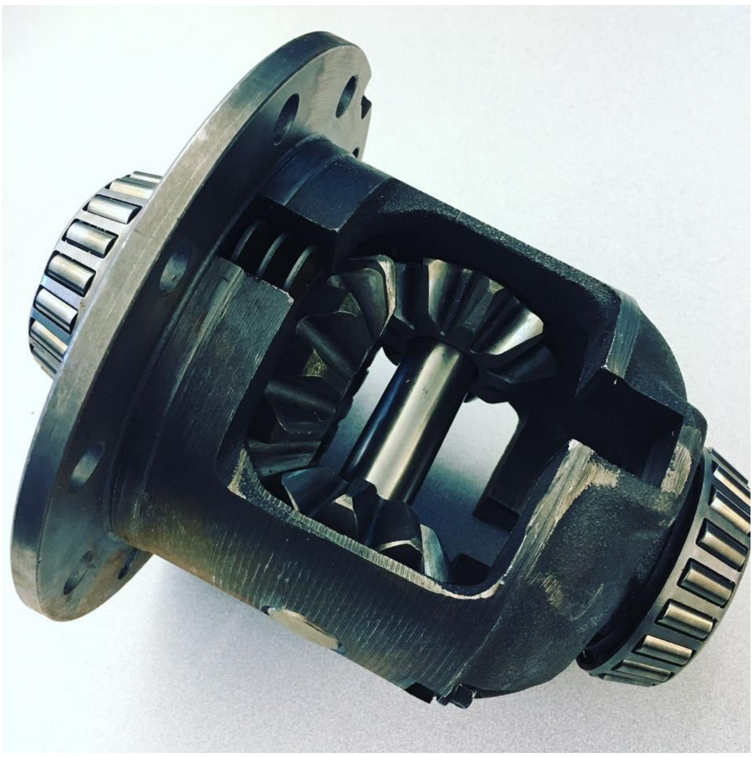

Figure 34. Ford 8.8" Traction-Lok Differential Gifted by Bair-Ling Technologies.

Our sponsor previously committed to incorporating Blair-Ling Technologies BT-B within our drivetrain, but Bair-Ling Technologies' BT-B system is still under development (refer back to section 2.3 for the detailed BT-B explanation). The differential given to our team was a clutch type LSD, and did not incorporate any of the initial BT-B prototype designs. However, the differential carrier housing is a design that they have worked with extensively and have designed their BT-B to fit within our specific differential carrier. Once Bair-Ling Technologies' BT-B system is finalized, it can easily be retrofitted into the differential. The BT-B system will replace the clutch plates and steel plates that make up the clutch type system. However, due to the BT-B still being in the development stage, we were given permission by our sponsor to proceed with the clutch type LSD in our final design, as it still meets the requirement of having a LSD.

7.1.9 Housing

Another piece of the project that has deviated from the original plans is the casting the drivetrain housing. After speaking with Martin Koch, an IME professor at Cal Poly, and pitching the idea of casting the aluminum housing, it determined that the amount of work to design the patterns, gateways, and other details would be equivalent to the work of another senior project as well as cause our costs to far exceed our budget. In reevaluating the time available to our team and noting that our primary focus is the power transmission from the motor it was determined to shift our design for a simpler rectangular geometry with the correct dimensions to house our drivetrain system. We brainstormed various methods of manufacturing this simpler design including CNCing a solid block of metal to the proper form or welding aluminum plates together. After formulating these new plans, we spoke to several more manufacturing professors at Cal Poly to enlist their expertise and will continue to work closely with two in particular; Trian Georgeou for design and CNC work and Kevin Williams jigging and bolting expertise. The new concept is mechanically joining the plates together and then fitting an acrylic sheet on top. There will be blocks made from 4140 steel at each of the corners as well as two additional blocks midway on the bottom of the housing to provide additional support when the drivetrain is filled with oil. In addition, holes for oil fill and drain plugs are located on the back wall of housing in the center of the facewidth of the helical gear so the oil will coat some of the gear while it is being poured in.

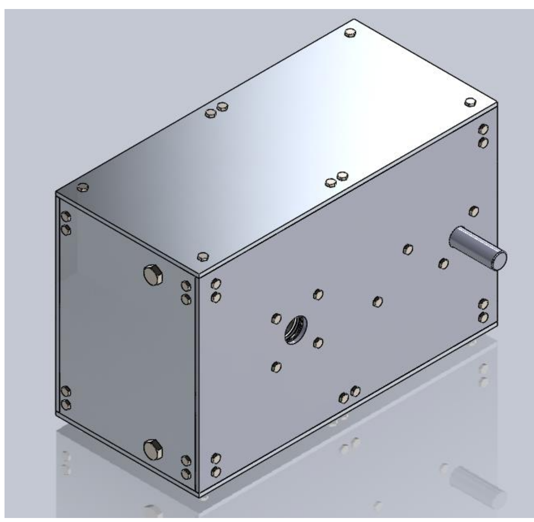

Figure 35. The much simpler geometry will a cheaper easier manufacturing method and still allow our team to test the entire drivetrain by containing an oil bath.

<u>List of Major Materials Required for Housing:</u>

- 6061 T6 Al
  - 19.25"x8.75"x12" (Overall Housing Dimensions)
    - AL Sheet Quantity and Specs: 2x1 (4), 1x1 (2)
  - Thickness 0.25"
- Acrylic Top 19.25"x 8.75" (Used for smaller testing and display only)
- 1X2 in, 6 ft. Aluminum rectangular stock
- 1/4" Grade 8 Bolts (minimum 6 bolts per corner)
- Engine Sealing Loctite
- Thread Fastener Loctite

The dimensions were chosen to make sure the entire assembly was encompassed as well as minimizing the amount of aluminum plate needed and keeping the sheets stock sizes available. Using the center to center distances of the gears we determined the minimum length needed to enclose the gears. To allow for plate thickness and clearance for the gear teeth the length of the side was specified at 18.25 inches. The clearance was also to ensure that if adjustments were needed to be made for alignment purposes there would be room for a tool to be inserted around the gears. The width is based off of the calculated shaft lengths and the clearance required for the bearing housing which will rest within the housing.

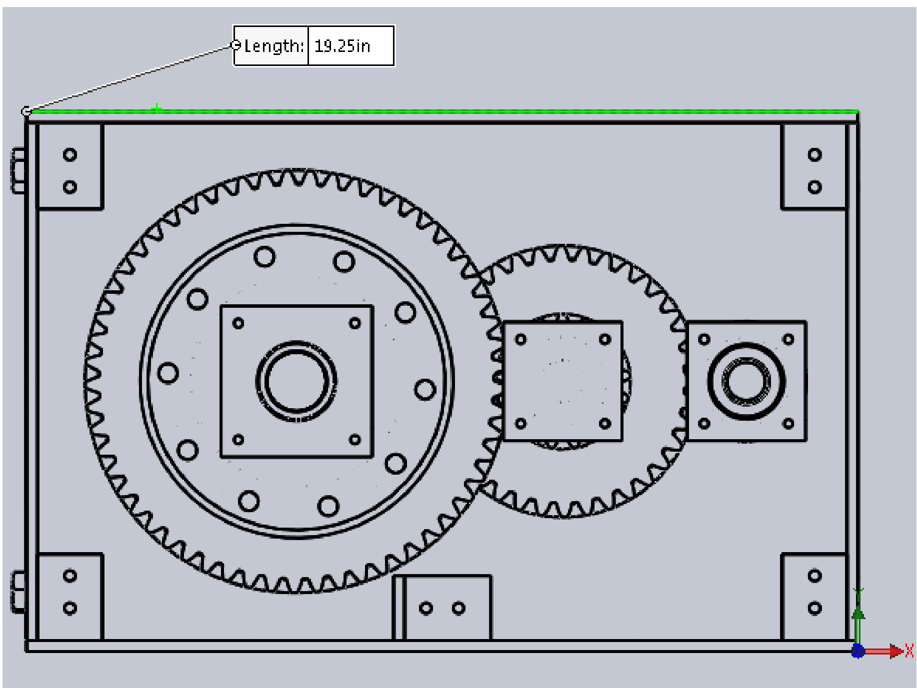

Figure 36. Length of 19.25 to allow for roughly an inch of clearance on either side of the gears.

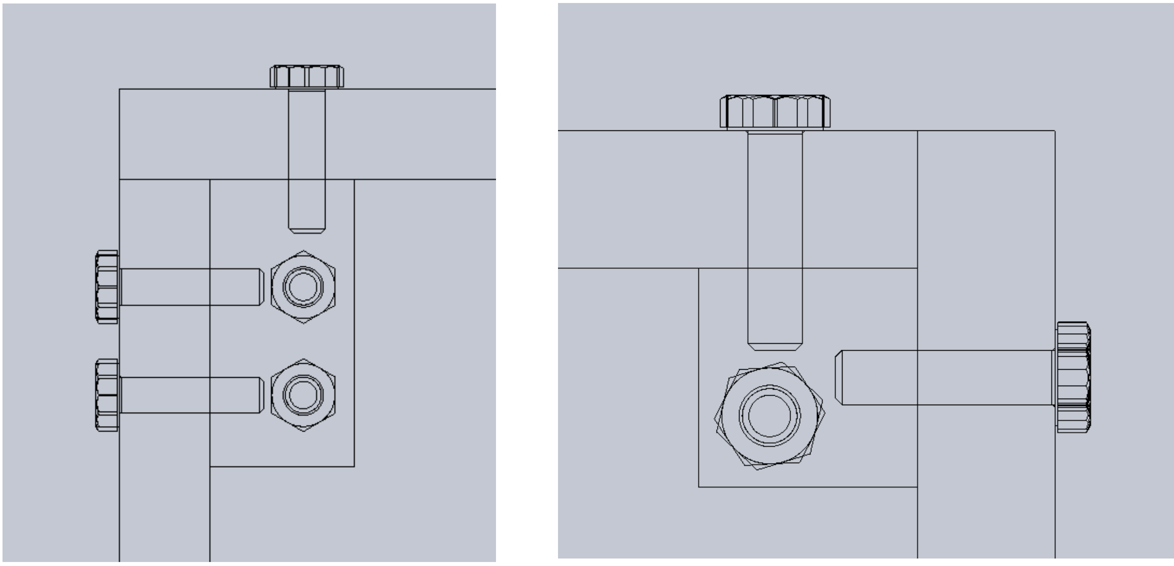

Figure 37. Bolt Clearances: Figure on right is a top view of the housing showing the clearance of the bolts once joined. Only one bolt is needed to fasten the top and bottom plates since they are not load bearing and are simply there to retain the oil.

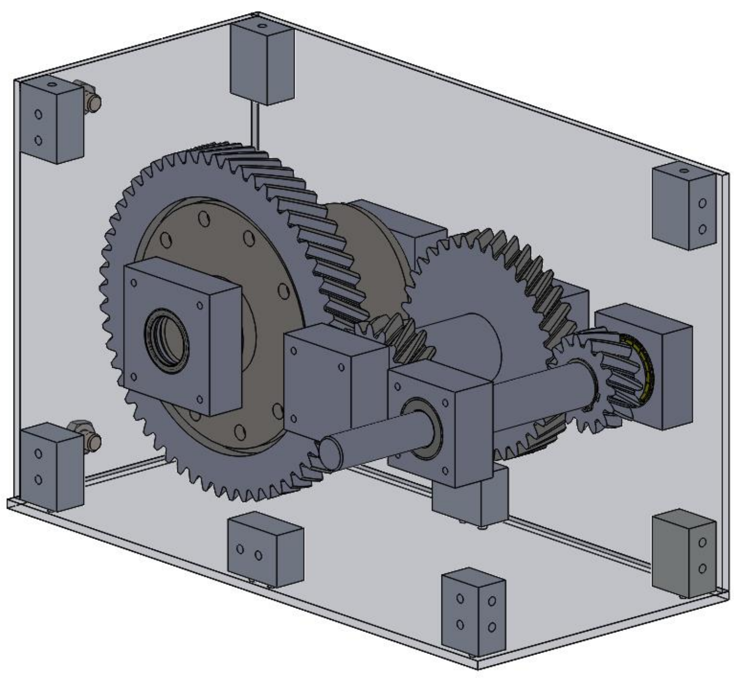

Figure 38. Shows the drivetrain encased in the rectangular housing without the front or top plate. This will be how the shafts and gears are inserted into the housing before bolting the remaining faces on.

As previously mentioned, the top of the housing may be replaced with an acrylic of the same dimensions as the aluminum plate so we have look inside the housing and see all of the components to ensure the gears are aligned and mesh properly while rotating during low level tests. This will be attached to the steel corner blocks with ¼"-20 hex head bolts. However, when the housing is implemented into the prototype vehicle or in running the tests with the dynamometer, this acrylic top will be replaced with a sixth sheet of aluminum to better support the top edges of the housing assembly. We numerically verified that the housing would minimally deflect due to the axial loads generated by the helical gears and calculated a very conservative estimate of 0.35 inches at the center of the long vertical side plate with the three locating holes (Please See Appendix D.23). To further minimize this number additionally support blocks were added in the center of the lower plates to help support. Also, the tapered roller bearings were designed to handle most of the load generated from the helical gears.

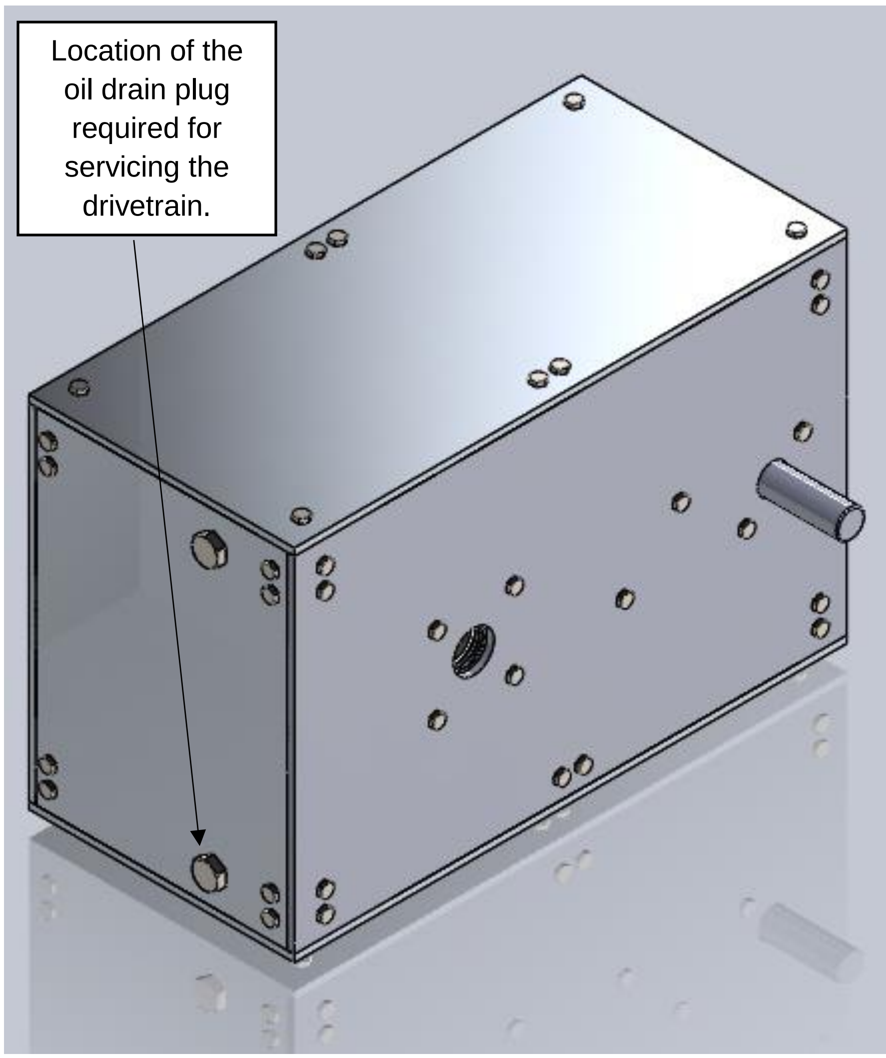

Figure 39. The housing plates with the addition of the CNC'd shaft locations and fill and drain plugs located on the back housing plate to allow for oil drainage and replacement.

The bearing will be housing by the flanged housing bearings discussed in section 7.1. The bolting locations are shown on the sides of the housing. A diagonal bolt pattern will be used to minimize the total number of bolts needed. In addition, if a bolt strips the threading in the aluminum block, the other corners of the block can be used to secure the bearing mount to the housing. The only additional manufacturing required would be to drill the diagonal holes through the housing plate as the bearing mounts have all four holes located in each other their corners.

7.1.10 Motor Coupler

In designing the motor shaft collar, assumptions were made that the company supplying the motor to the specifications of the ST5 would be able to have a reasonable diameter motor shaft for the connection point to the drivetrain. In other examples of motor shafts, the motor shaft collar was fixed to the motor shaft and then bolted to the transmission assembly connecting to the transaxle. The connection was made using a shaft key, a tapered lock collar and an outer mounting sleeve. In the example collar design, it required bolts to be threaded towards the face of the motor, which was problematic during assembly of the drivetrain. After our gear reduction was determined, helical stock gears were chosen to meet this value as discussed previously. The next task in our design was to combine a helical pinion gear with a motor shaft collar, while having the ability for a shaft key to be inserted and connected to the future motor shaft. This hole in the motor interface needs to allow a slight clearance fit around the motor shaft, the key and the shaft collar keyway. This allowed the helical pinion connection to the motor shaft assembly to not require fasteners, simplifying the drivetrain connections inside the housing. The method of shaft and gear connection could either be splined, bolted, or a threaded connection. The chosen helical gear will determine whether a bolting flange is machined into the motor interface or whether a spline connection is implemented.

7.2 Design for Safety

Initial safety considerations regarding our design were discussed in Section 4.7 along with the Design Safety Hazard Checklist. Many of the same safety considerations remain in our final design. The drivetrain is a component that will not see direct human interaction. When implemented, the drivetrain will be located on the underside of the ST5 prototype vehicle. The only time when human interaction is required, is during oil changes. However, the drivetrain will not be operating during servicing. Large loads and forces will be transmitted through our drivetrain, but all load bearing components will be enclosed in a structural housing. This housing will serve to prevent outside debris from coming in contact with our rotating components. The housing will also prevent oil leakage. Oil leakage is not only hazardous to the environment, but also to the gears and bearings. With a loss of oil, various drivetrain components can see increased wear that can decrease the life of the drivetrain. Leakage tests will be performed to ensure that our drivetrain housing is properly sealed. There are three rotating shafts that exit the confines of the drivetrain housing. While it is outside of our design scope, the prototype vehicle can incorporate “shields” around these shafts, to eliminate the possibility of unintentional contact with the rotating shafts.

7.3 Budget

Moving forward with the information stated in the cost analysis section, an Excel spreadsheet was created in order to further track how our expenditures fit within our sponsor’s budget. The full budget for the drivetrain build can be found in Appendix F.16. In the originally presented gear diameters, as seen in Table 11 below, the expensive nature of the over-designed custom gears was evident in the quote received from Rush Gears which is included in Appendix F.17 – F.20. The price total was around $11,500 and did not include shipping and handling. The second quote received from Rush Gears was also disheartening in that the effort to utilize stock gears did not net to a worthwhile cost saving. Since there was a minimal difference between the stock options and fully customized gears it was decided to order custom gears to ensure our requirements were met without having to constrain or hinder our designs to fit in with stock options. For a total price of 10,700 plus tax, we received two sets of gears (for a total of eight gears) since the rear set up would eventually be duplicated in the front for a dual motor system. Table 11 below shows the budget percentage of the four helical gears.

Table 11. Results from Budget

	Before Tax	**8% Tax**
Prototype Total Cost	$5,350.00	$5,778.00
Budget	$10,000.00	$10,000.00
Remaining difference	-$4,650	-4,222
Gear Price Percentage	53.50%	57.78%

The standard lead times in industry are three weeks, and expediting the shipping and production time would not be worth the added cost. As seen in the Figure 40 below, buying in bulk, or even 2 of each gear, drastically reduces the price per gear, and would be beneficial to the future dual motor drivetrain system, which as mentioned before, was the option that our team ended up pursuing.

Part# 1095-0699

QTY	Net each	Total	>Standard< Prod. Time
1	$2150.00 ea	$2150.00	3 WEEKS
2	$1075.00 ea	$2150.00	3 WEEKS
4	$967.50 ea	$3870.00	3 WEEKS
6	$860.00 ea	$5160.00	3 WEEKS
10	$752.50 ea	$7525.00	3 WEEKS

Description: MADE TO ORDER, 4140 HT STEEL, HELICAL GEAR, 6.000 TDP, (6.93 NDP), 20.0 PA, 18 TEETH, 1.000" FACE, 1.688" BORE, A TYPE, .375 X .1875" KEYWAY, 3.0000" PD , 3.289" OD:, 30 DEGREE, RIGHT HAND

Expedited Production Times: OEM prices (net each piece)

QTY	2 WEEKS	1-WEEK	72-hours
1	$3225.00 ea	$4300.00 ea	$6450.00 ea
2	$1612.50 ea	$2150.00 ea	$3225.00 ea
4	$1451.25 ea	$1935.00 ea	$2902.50 ea
6	$1290.00 ea	$1720.00 ea	$2580.00 ea
10	$1128.75 ea	$1505.00 ea	$2257.50 ea

Figure 40. First quote from Rush Gears for one of the intermediate gears.

8.0 Manufacturing Plan

The manufacturing of the housing will take place on Cal Poly’s campus utilizing the IME facilities, the hanger, and Mustang 60. For the CNC code needed to locate shafts and bearing locations, Trian Georgeou has graciously offered his expertise to help our team. Lathes and mills are also available to perform the other cuts of the stock metal for the drivetrain system and the manufacturing facilities are equipped with the other necessary tools required such as hydraulic presses to fit the bearings into bearing mounts, socket wrenches for housing assembly, drill presses for through holes.

Steps to Housing Assembly:

i. The aluminum plates for the side will first be stacked on top of each other and drilled through with a mill to locate the shaft positions and bolt locations.
ii. Secure all plates in rectangular form excluding acrylic top and front of housing
    a. Cut Al stock into 2x1 [ft] rectangles with the mill
    b. Cut Al bar into 1x1x2 inch blocks with the mill
    c. Thread Al blocks for ¼" Grade 8 bolts
    d. Insert 1x1 blocks of aluminum inserted in the all the corners
    e. Align ¼" Grade 8 bolts to the hole locations and drill in to secure
iii. Seal seam lines with Engine Sealing Loctite
iv. Seal bolts with thread Loctite

Steps to Machine Shafts:

i. Turn stock bar steel down to the appropriate diameters on a lathe.
ii. Face shafts to length
iii. Mill in key ways
iv. Spline input shaft

8.1 Assembly Plan

Beneath is the process in which we will go about assembling the final design.
Steps to Combine Gears and Shafts:

i. Insert the keys to the appropriate locations on the shafts.
ii. Slip the gears onto the shaft aligned with the proper the key ways for secure mating
iii. Install retaining rings to prevent gears from slipping out the end which is not pressed up against the shaft.

Steps to Combine the Housing to Gears and Shafts

v. Insert bearings into flanged bearing housing
vi. Bolt flanged bearing housing to aluminum plates
vii. With assembly resting on side plate insert shafts vertically into bearings and check to ensure shafts are parallel to the front and back of the housing as well as each other.
viii. Insert final plate with symmetric shaft and bolt locations to secure shafts and have gears aligned.
ix. Seal the remaining bolts with Loctite thread fastener.

8.2 Maintenance and Repair

Our system will require general, yet minimal maintenance once it is implemented in the consumer market. It will require the user to change oil regularly. For all other maintenance issues it is recommended that the consumer bring it into a shop since it is unlikely that they will have the equipment required to remove the other components of the car and reach the drivetrain system. In the event of gear, shaft, or bearing failure, the user would need to take the vehicle to a dealer where it would receive replacement components and/or a new drivetrain depending on the severity of the failure.

9.0 Design Verification Plan

9.1 Housing Leakage Test

Upon assembling the drivetrain, we will test the integrity of the bearing seals and sealing between the plates that make up the housing. The drivetrain housing will be filled with oil, until the oil level is mid-way up the second largest gear in the assembly. This oil level will test for gaps in the input shaft seal, as well as the axle shaft seals. These are vital locations to test for leakage, since these three areas have shafts extruding outward from our housing. We are aware that the seals may behave differently when these shafts are spinning, but it will be a baseline test that will also evaluate the effectiveness of the sealant/gasket material between the joined housing plates.

9.2 Gear Reduction Confirmation Test

We will perform simple static tests to verify the torque increase and speed reduction that is associated with our 8.04:1 gear reduction. The speed reduction test can be done by hand, where the input shaft will be rotated and verify that it will take roughly 8.04 rotations to produce a single rotation of the output shafts. The torque increase can be performed using digital torque wrenches as shown in Figure 41. A digital torque wrench will be placed on the input shaft and additional digital torque wrenches will be placed on the two output shafts. The torque wrenches on the output shafts will be fixed in position, holding the output shafts stationary. A load will then be applied with the torque wrench located on the input shaft. The applied torque will be read from the digital readout, and while keeping the input torque constant, the torque on the output shafts can be read from their respective digital torque wrenches. Assuming the differential does not distribute the load unevenly, the output shaft torques should each read 4.25 times that of the input torque. In this event, the differential would split the torque 50-50 between the two output shafts, and when combined, the torque output would be a multiple of 8.04 of the input shaft torque.

Figure 41. A common digital torque wrench [28].

An alternate to using three torque wrenches would be use strain gauges on the output shafts instead. This would only require the use of a single digital torque wrench, but would require a method of fixing the output shafts. The torque seen on the output shafts would be calculated from the based on the stresses and material properties of the shafts.

9.3 Dynamometer Testing

Although a transmission dynamometer has not yet been located, we still wish to test our drivetrain on a motoring dynamometer. However, we have set a deadline for ourselves to develop an alternate test plan if a dynamometer has not been located by February 23, 2017. Utilizing a dynamometer for validating our drivetrain design would be very advantageous, due to its instrumentation and variability in power delivery. The dynamometer's power output can be manipulated from either a control unit or computer software, and can allow us to test for the maximum motor specifications of the REMY HVH250-115 motor, if the dynamometer is powerful enough. The following tests would be performed with the help and guidance from a dynamometer technician:

1. Run the drivetrain at an input torque of 50 lb-ft. The drivetrain will be inspected, and if our drivetrain shows no signs of concern, we will increase the input torque in 50 lb-ft increments, until reaching the 400 lb-ft torque input that the drivetrain was designed to. Data will be recorded for each run at both the input and output shafts. As a result of this test we hope to extract the mechanical efficiency of our drivetrain
2. Run an RPM sweep from 0 - 11,000 rpm, and remain at 11,000 rpm for a 30 second duration. The RPM will then be ramped back down to 0 rpm, where the drivetrain will be inspected. During this test the torque being sent through the motor will not have to be of a significant value. This test is designed to test the bearing's limiting speed.

These tests also serve to test the integrity of the Ford 8.8" differential that was supplied by Bair-Ling Technologies. This, however, is not much of a concern, as Ford 8.8" differentials are integrated in the rear axle assembly of Ford Mustang GT's, which are capable of producing 400 lb-ft of torque [29].

The technician and company whose dynamometer we use will have the control to change the specifics of our test plan, if any of our tests are deemed to be unsafe.
Since a dynamometer has not been located, we are currently unsure of the cost to test our drivetrain and the hourly rate of the dynamometer technician. However, we would like to set aside $1,500 in our budget for this test

9.4 BMW 330Ci Implementation

Sharpell Technologies recent purchase of BMW 330Ci may serve as a back-up plan if a transmission dynamometer cannot be located. Our sponsor is working toward equipping the BMW prototype test vehicle with all the necessary drivetrain equipment by the end of Spring 2017. This would allow for our drivetrain to be coupled directly to the REMY HVH250-115 motor, and undergo dynamic loading conditions. The prototype vehicle can be run on a chassis dyno, allowing for the drivetrain to be run in a controlled environment. This testing would be run with a procedure similar to what would be done on the transmission dyno. This test would also allow us to collect output torque data from the chassis dyno, which could be compared to the motor's output torque to determine mechanical losses of the drivetrain.

9.5 Alternate Plan

Due to the nature of the dynamometer test and BMW prototype implementation possibly falling through, we will continue to brainstorm additional ideas for testing the functionality of the drivetrain. We wish to test the integrity of our design with loading situations comparable to that of daily driving conditions.

9.6 Drivetrain Specification Checklist

Table 13 lists the design specifications from our DVP&R which is also listed in Appendix G.1. We will be able to validate specifications 1, 2, 5, and 6 through our static testing, housing leakage test, and component calculations. Specifications 3, 4, and 7 will prove to be the most difficult specifications to validate. This is due to the possibility of not being able to conduct dynamometer testing. The other avenue that would verify these specifications would be in the BMW prototype vehicle. However, the 400 lb-ft torque verification would not be able to be tested for, as the REMY HVH250-115 motor is only capable of producing 310 lb-ft of torque. The 11,000 rpm test could be accomplished, if the vehicle's rear wheels were lifted off the ground through the use of a lift or jack stands. This however, would have the potential to be an unsafe test.

Table 12. Design specifications taken from DVP&R.

Item No	Specification	Test Description	Acceptance Criteria
1	Verify Speed Reduction	Apply an input speed while connected to a dynamometer, and verify that our reduction creates the appropriate output speed.	> minimum RPM to reach 100 MPH
2	Verify Torque Increase Through Gears	Attach drivetrain to motoring dynamometer, input 50 lb-ft, and read the load applied on the output shafts. The output torque will be calculated from the load, and compared against the 8.5:1 gearing reduction.	> 50 lb-ft
3	Maximum Input Torque	Input maximum torque of 400 lb-ft from dyno at low rpm (> 1,000 rpm). Inspect drivetrain to verify that no damage was done.	No Damage to Drivetrain
4	RPM range	Attach drivetrain to motoring dynamometer and verify that the drivetrain can withstand 11,000 rpm.	> 11000 RPM
5	Design Life	Analysis through calculations.	> 100000 miles
6	Housing Leakage	Fill housing with oil and collect any leaking oil.	< 1 mL in 1 day
7	Differential Load Verification	Verify that the Ford 8.8" differential and other rear axle components can handle the maximum 400 lb-ft of torque. This will be tested on a motoring dynamometer. Verify that no damage is done to differential.	No Damage to Differential

10.0 Manufacturing

We began manufacturing during the first week of spring quarter and started with the simplest pieces to manufacture to refine our CNC processes before attempting the more critical parts. Refer to Appendix B.9 to see the various manufacturing processes and their dates of completion.

10.1 Housing Blocks

Eight corner blocks were needed to support the sides of the housing and were made from a piece of stock 4140 steel.

The process consisted of first cutting the blocks to their rough desired dimension on a horizontal band saw. The next steps involved using a mill. We indicated the back jaw of the mill vise with a dial indicator to ensure the jaw was parallel and square with the machine's axes and proceeded to reference off the back jaw for the rest of the milling operations. The vise jaw simulated a datum to ensure the blocks were cut to the desired size properly. Next, we measured each of the blocks to determine how much is needed to face off from each block. After noting the amounts needed to be removed from each block, we then proceeded to face off one side of the block using an end milling operation.

To create the holes needed in the steel blocks we chose to CNC them for the precision required for the spacing of the holes and to make sure that they would align with the holes drilled in the housing plates. With the aid of Trian Georgeou, a professor in the IME department at Cal Poly, we were able to use Mastercam which created CNC operations based off our SolidWorks model. First, we probed the part to give the machine a datum to reference the operations off of as seen in Figure 42 below. Then, we peck drilled the hole locations. Peck drilling helps to minimize tool deflections and helps clear metal chips from the hole as it is drilled. We also used flood coolant to minimize the warping of the part (or tool) due to excess heat from the drilling operation. One slight issue that we ran into was drilling holes on the wrong faces of certain blocks. From the left and right sides of the drivetrain, the blocks are mirror images of each other. This means the blocks are not the same, and the holes are on different faces relative to its respective side of the housing. After realizing this we had to re-drill 4 of the blocks and sealed the unwanted holes with Quick Steel epoxy. The final step in finishing the corner blocks was de-burring the drilled holes and then hand tapping them for a 1/4-20 UNC bolts.

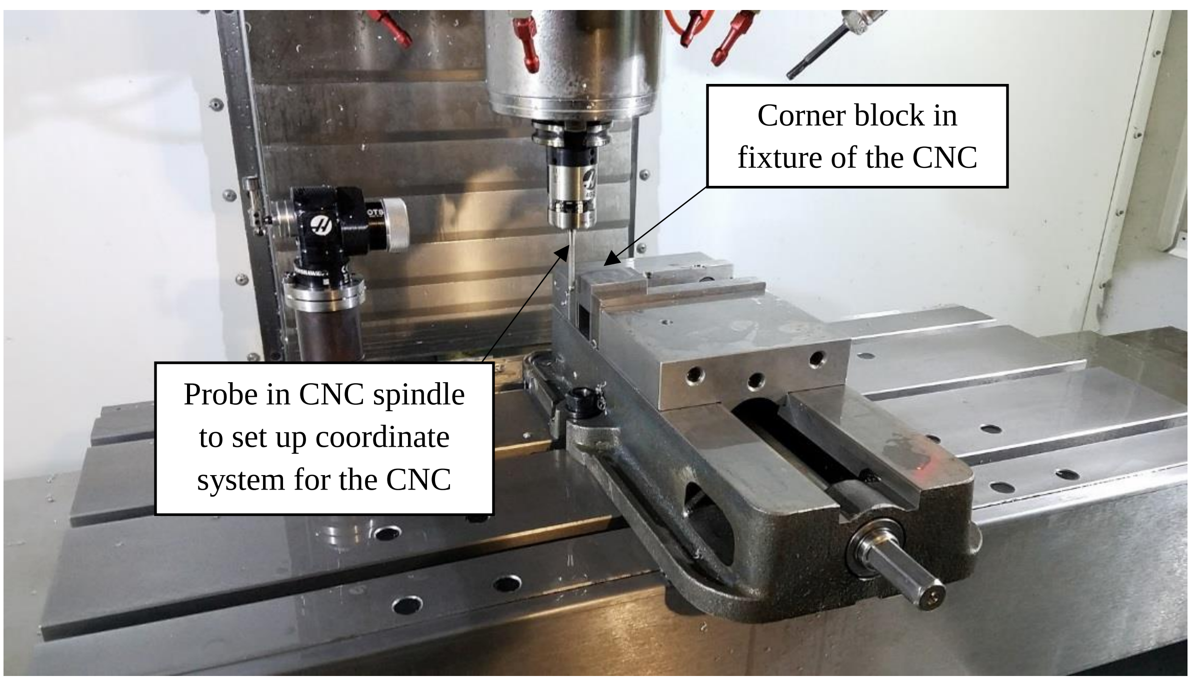

Figure 42. The probe located in the spindle is touching off a corner block in the CNC to give a reference location to perform the drilling operation.

10.2 Bearing Mounts

A total of six bearing mounts were needed to house the bearings and support the shafts in the housing. Given the designed dimension from the SolidWorks model, only two of the bearing mounts had identical operations. For all the parts, we chose to reference off of the face in contact with the housing since the width was our most critical dimension and would ensure the proper spacing between the bearing mounts so the bearings would not bind. Each block required a counterbore for the bearing and three blocks required additional counterbores on the opposing sides for the seals of the input and output shafts. All blocks were also drilled and tapped for 1/4-20 UNC bolts. Each of these features can be seen in Figure 43 below.

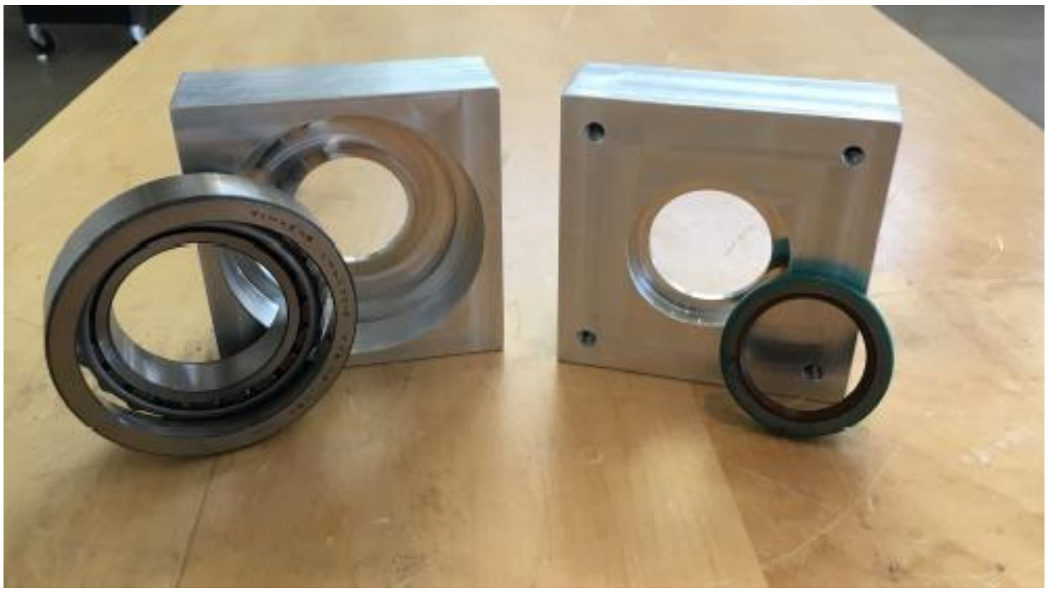

Figure 43. This bearing mount is one of the more complex designs due to having more features. A counterbore was required for the bearing as well as the seal for the input shaft.

One issue we ran into was that when we first ran the counterbore operation was the depth of cut that the tool took was too great, which allowed the tool to flex and did not provide a perfect circle. This was fixed with running a simple boring operation after where the depth of cut was much less and the tool has less tendency to flex. After each counterbore operation was completed, we deburred the hole and dropped the outer race of the bearing into the hole to ensure that the parts mated properly.

10.3 Input and Intermediate Shaft

First, we cut the stock on a horizontal band saw to the rough length of the intermediate (7.5”) and input shaft (10.5”). Again, due to the precision required we opted to use a Haas CNC lathe in the IME building to turn down a piece of cylindrical 4140 steel stock. With the assistance of Wyatt Hall, an assistant in the IME department, we used the CAD model as a reference to measure off the different desired diameters. Throughout the process and once each diameter was reached we checked that the dimension was within the specified tolerance and also slipped the inner bearing race or gear over the shaft as a second check that the parts fit properly. After the shafts were turned down to the correct diameters and faced off to the appropriate length, the next step was cutting the keyway in the shafts to mate the gear securely to the shaft.

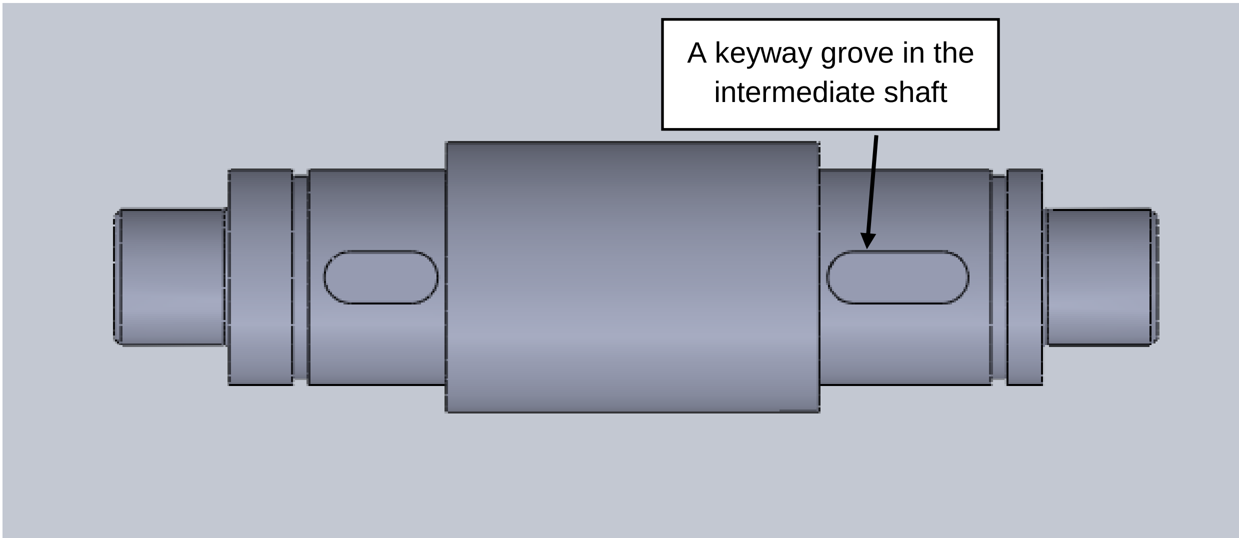

Figure 44. This SolidWorks model shows the keyway for each gear on the intermediate shaft. The face widths of the gears are slightly different which requires different keyway lengths.

To cut the keyways shown in Figure 44 above, we used the manual milling machines with digital readouts in the Hangar. The digital readout (DRO) provides coordinates of the milling tool relative to a specified origin. This helped to machine the keyways because the referenced origin lied along the centerline of the shaft, and would otherwise be more difficult and time consuming if we used the dials on the mill handles to machine the slot to the appropriate dimensions. After establishing the origin of the shaft for the keyway, we cut the keyways using end mills with a diameter equal to the keyway widths. The sizes are 0.375" and 0.250" in width for the intermediate and input shafts respectively. The length of each keyway was determined by the face width of each gear. The keyways have the profile of a slot because the end mills have a circular profile, and a very small end mill would have been necessary to create a squarer end for each profile. This would have been unnecessary for our application and taken more time away from other operations.

In order prevent axial motion of the gears on the shafts, retaining rings were used and required grooves to be cut into the shaft as seen in Figure 45. To do this we used the manual lathes in the Hangar as well. There are tools specific to cutting grooves of different sizes in shafts, but the Hangar did not carry the tools necessary. To combat this, we used a parting tool instead. The parting tool was 0.120" wide which was wider than the groove specified on the input shaft drawing. The extra width was accounted for by offsetting the groove partially toward the centerline of the gear such that the remaining visible groove with the gear installed was the correct width for our specified retaining rings.

Figure 45. This is the intermediate shaft depicting the two grooves for the retaining rings. The intermediate shaft has two gears, thus the need for two retaining rings and their respective grooves.

10.4 Keys

Keys are a necessary part of a keyway system and we had to manufacture these as well. Keyways are standard sizes depending on shaft diameter so key stock is also available in the same sizes. From our key stock, a portion was cut to rough length of the respective keyway that the key would be inserted into. From there, vise grips were used to hold the piece and a bench grinder was used to the shape the key to its final dimensions and shape as seen in Figure 46.

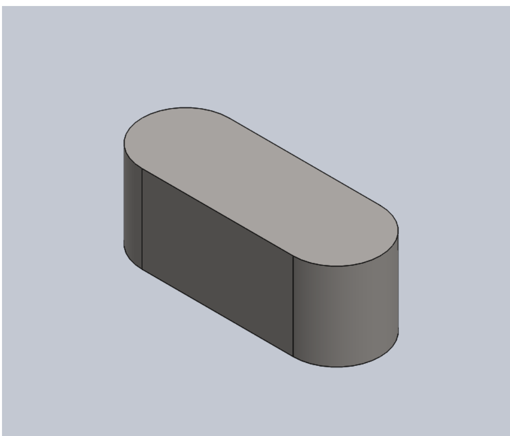

Figure 46. This is the key used for the input shaft. The key size depends on the shaft diameter so the intermediate shaft keys are slightly bigger.

10.5 Housing Plates

For the top, bottom, front, back, and acrylic top housing plates, a drill press was used to drill the holes in them. To locate where the holes were specified on the drawings, a ruler, square, center punch, and hammer were used to center punch marks where the holes would be. Center punching the holes prevents the drill bit from wanting to walk away from its desired location. Wooden blocks were placed underneath the plate and then vices were used to clamp the plate down to eliminate movement while the drilling operation was performed. A drill press was used to drill through 1/4" holes into the plate (a total of eight holes per plate for the top and bottom plates, four holes for the front plate, and six holes for the back plate).

Figure 47. The CNC'ed housing plates with all of the necessary holes to bolts the housing together and the bearing mounts. The corner blocks and bolts needed to fasten the plates together are shown in the right of the picture.

The side plates were machined on a CNC mill in Mustang '60. The holes that locate the bearing mounts reside in these plates and the accuracy of these is critical to ensuring proper shaft alignment. The through holes for the input and output shafts were also machined in the same CNC operation. The final machined housing plates are shown in Figure 47 above.

10.6 Ring Gear Bolt Pattern

Measurements were taken off of the Ford 8.8" differential to properly dimension the bolt pattern on the flange where the ring gear mounts. Once properly dimensioned, it was incorporated into CAD and a 1:1 scale drawing was made. The bolt pattern dimensioning was verified by lining up a printout of the 1:1 scale drawing to the actual bolt pattern on the differential. After the dimensions were verified, the coordinates of the 10 bolt hole locations were pulled from the SolidWorks model. A mock-up ring gear was then created from wood for additional verification of the bolt pattern dimensioning as well as a way to develop a method for fixturing the ring gear for the manufacturing. This mock-up ring gear was laser cut from a piece of plywood, and matched the inner and outer diameter dimensions of the actual 9.5" ring gear. The fixturing of the ring gear is depicted below in Figure 48, which utilizes four 3-2-1 blocks and toe clamps. 3-

2-1 blocks are steel blocks that have the dimensions of 3” X 2” X 1” and are often used in fixturing. Using a mill with a DRO, drilling operations were performed on the wooden mock-up ring gear at the locations determined by the bolt hole coordinates. As a final verification step, the wooden mock-up ring gear was bolted onto the differential. All of these verification steps were taken to ensure that no mistakes would be made when drilling into the $1,591.25 ring gear.

Figure 48. This is the fixturing used for drilling the holes in the ring gear. The 3-2-1 blocks raised the gear off the mill table to allow the drill to move all the way through without marking the table. The toe clamps held the gear in place.

Due to the case hardening of the 8620 steel ring gear, we were required to purchase a 7/16” carbide drill bit. After acquiring the 7/16” carbide drill bit, the actual ring gear was fixtured on the mill table. The ring gear was oriented so that the counterbore was facing upward. The reason for this is that the counterbored surface is not case hardened and thus easier to penetrate when starting the drilling operation. This is because when manufactured, the ring gear was case hardened prior to milling out the counterbore. As such, the counterbore surface was non-case hardened 8620 steel. It is much harder to start a drilling operation into a case hardened surface, so piloting the hole on the non-case hardened surface was determined to be the preferred course of action. Since the hole would already be started, it would be easier to guide the carbide drill bit through the rest of the case hardened material. This drill bit was still required to drill through the case-hardening on the back side of the ring gear because the equipment provided at the shops on Cal Poly’s campus were dull and not strong enough to use for the 8620 material. Once fixtured, a bore finder was used to center the bore of the ring gear with the chuck of the mill. After the ring gear bore was perfectly centered about the drill bit, the DRO was zeroed out. Using the determined bolt location coordinates, the 10 bolt holes were drilled. It was found that the best method for manufacturing was to start the hole by feeding the mill's quill by hand, and then engaging the auto-feed in the z-direction (vertical direction) to drill through the remaining material and case hardening. The auto-feed was necessary as the drill bit would occasionally lock-up when feeding the quill by hand. The final manufactured ring gear is shown below in Figure 49.

Figure 49. The holes drilled in the ring gear lined up perfectly with the holes on the flange of the differential. Not pictured here are the bolts used to mount the gear to differential itself.

<u>10.7 Differential</u>

One of the differential's outboard bearings was removed using a bearing puller as seen in Figure 50 below. The other bearing, however, was tightly secured onto the differential carrier, and required an angle grinder to cut off the bearing. A local auto-shop ground the stubborn bearing off, free of charge.

Figure 50. This bearing came off with the use of the bearing puller shown. The puller consists of the arms that have a lip to grab on the bottom of the bearing and a threaded rod to apply force to a stationary surface. We used steel blocks stacked inside the differential to provide a surface to apply force.

11.0 Assembly

Intermediate checks were performed throughout the manufacturing process to ensure that the proper tolerances were met for assembly. After all the housing parts were manufactured, we first bolted the housing together using a ratchet and socket set. Figure 51 shows the initial fit up of the housing. Fine tuning had to be done in order for the holes in the housing plates and corner blocks to line up while maintaining tight seams where no oil could leak out. Some holes in the housing plates had to be drilled one to two drill sizes bigger in order the align the plates with the blocks. Fortunately, the holes in the blocks located the housing plates so drilling the housing plate holes bigger had no adverse effects. A few of the blocks had to be faced off on the sides in contact with the bottom and side housing plates to provide better alignment. The blocks were all very close to the desired sizing, and we could fix them with a few more hours in the shop. We expected these issues beforehand as well, and we experienced no issues once we finally fit up the housing.

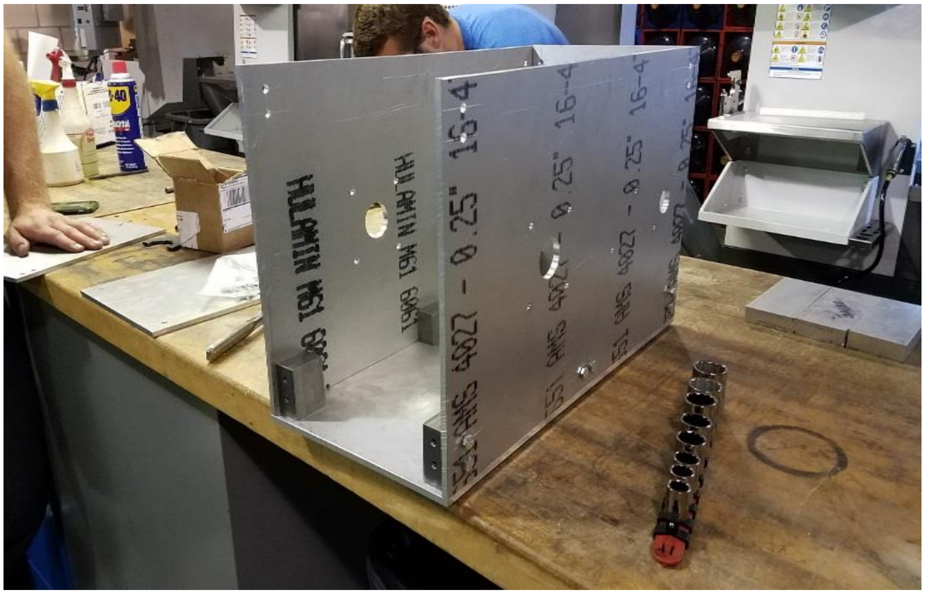

Figure 51. This is our initial fit up of the housing. Some of the holes in the housing plates had to be drilled a size or two bigger along with more facing done to the blocks to allow proper alignment of the housing.

To fit the seals and bearings into the bearing mounts we first tried using a butane torch to heat up the aluminum bearing mount and then press the outer race of the bearing into the counter bored hole. However, since it was difficult to uniformly heat the mount using this method, we ended up baking the bearing mounts in an oven at 250 °F and freezing the outer races. This was done to expand the diameter of the hole and shrink the outer race so we could mate the parts before both parts reformed at room temperature. This new method allowed us to easily slip the parts together. To fit the seals into the input and output bearing mounts, a hydraulic press was used.

To assemble the differential, we first had to remove the original bearings as described earlier. The old bearings were replaced with identical Timken bearings. They were installed by pressing the new bearings onto the differential using a hydraulic press. Figure 52 shows the bearings being pressed onto the differential carrier. The picture shows that we used scrap material to properly press the bearings onto the differential without damaging the component. A scrap piece of steel pipe was used as a spacer, and was of the proper diameter to sit inside of the cage containing the tapered rollers. Without the spacer, the force would be directly applied to the bearing cage as it is the uppermost surface of the bearing. Pressing directly onto the bearing would severely damage the bearing cage and could misalign the rollers. The flat piece of steel was used to as a surface for the press to apply force onto. Additionally, it served to disperse the force evenly across the spacer. With this set-up the bearings with installed in a very safe and easy manner. Bearing grease was also applied to the outer surface of the differential carrier for greater ease of installation. Excess grease was cleaned up and removed upon installing the bearings.

Figure 52. The hydraulic press with a small concentric tube to apply pressure to the inner race of the bearing without interfering with the differential. The plate on top is used to evenly apply pressure to the set up. We also greased the outer circular portion of the differential to allow the inner race of the bearing to slide on more easily.

The internal components then needed to be re-installed into the differential carrier. The clutch packs were installed into the differential carrier, and the spider gears were inserted, thus locking the clutch pack into position. Lastly, the S-spring was installed. To accomplish this, the S-spring was compressed in a vise, and then clamped down using needle-nosed vise grips. Once removed from the actual vise, the S-spring was positioned between the spider gears. Since it was not an easy fit, the S-spring was tapped into place using a mallet, while still being compressed with the needle-nosed vise grips. Once the S-spring was situated between the spider gears it released from the vise grips. However, the position of the S-spring still needed to be manipulated so that it was centered between the spider gears, as to not interfere with the cross pin. This was accomplished by lightly tapping on the S-spring with the mallet. Once centered the cross pin was installed and secured in place with the locking bolt. Please note that a face shield was worn during this process. The fully assembled differential is depicted in Figure 53.

Figure 53. Tools required to install the S-spring include a vise, needle-nosed vise grips, and a mallet.

The ring gear was then bolted onto the Ford 8.8” differential. As mentioned previously, through holes were drilled into the ring gear. As such, the ring gear was secured using both nuts and bolts. Permatex thread locker was applied to the nuts and bolts in order to combat loosening that may occur due to road vibration. Before fully tightening down the bolts on the ring gear, a feeler gauge was used to ensure equal clearance between the ring gear bore, and the outer diameter of the differential carrier. This centers the ring gear and eliminates the possibility of it being mounted non-centrally. If mounted non-centrally, the ring gear would cause the differential to be unbalanced, and would then cause the intermediate shaft to be unbalanced as well due to the mating gears. This would cause unsafe operation when coupled to the motor under load and speed.

Assembling the shafts were a relatively simple process. Gathering the input and intermediate shaft, retaining rings, keyways, and gears, we proceeded insert the key into the keyway milled into the shaft and then slipped the gear over the shaft. Next, we used snap ring pliers to snap the retaining rings over the shaft and into the groove machined next to the gear. The final input shaft assembly is shown in Figure 54.

Figure 54. Above is the assembled input shaft with the retaining ring holding the pinion in place pressed up against the shoulder on the left-hand side. The key prevents the gear from freely spinning on the shaft when experiencing a torque.

The final assembly of the housing required that we use RTV sealant to keep oil from leaking out. We first started with the bottom housing plate and housing blocks. We applied RTV at the mating surface of the plate and the housing reinforcement blocks before securing the blocks to the plate with their respective bolts. RTV was then applied on the top surface of the bottom housing plate along the left edge as seen in Figure 55 below.

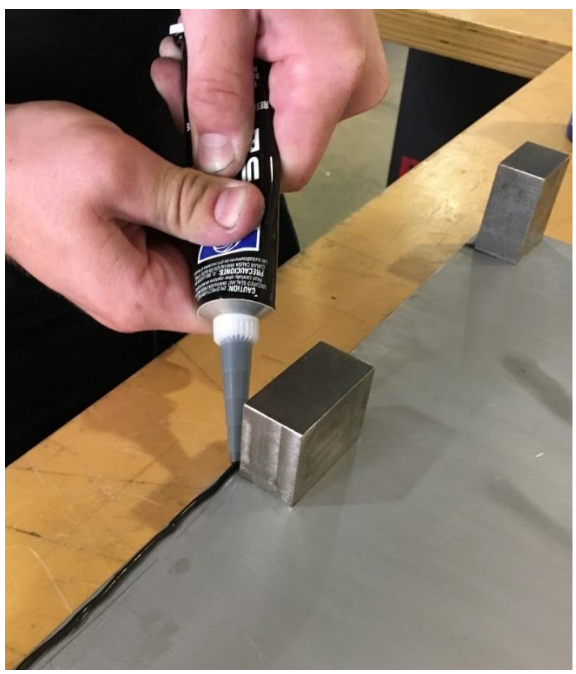

Figure 55. This how we applied RTV initially on the bottom plate. The left side plate was set directly on the line of RTV and bolted to the housing blocks.

The left side plate was placed on top of this line of sealant and then fastened with bolts. For the front and rear housing plates, there are two seams we needed to seal. These seams included the bottom plate and left side plate surfaces. Similarly, RTV was applied along these edges of the partially assembled housing, and the front and rear plates were installed and secured with bolts. Before we dropped in the shaft assemblies and differential, we applied RTV at the remaining mating surfaces of the housing blocks and the left, front, and rear housing plates. We also added extra RTV in the corners of each joint as seen in Figure 56 below.

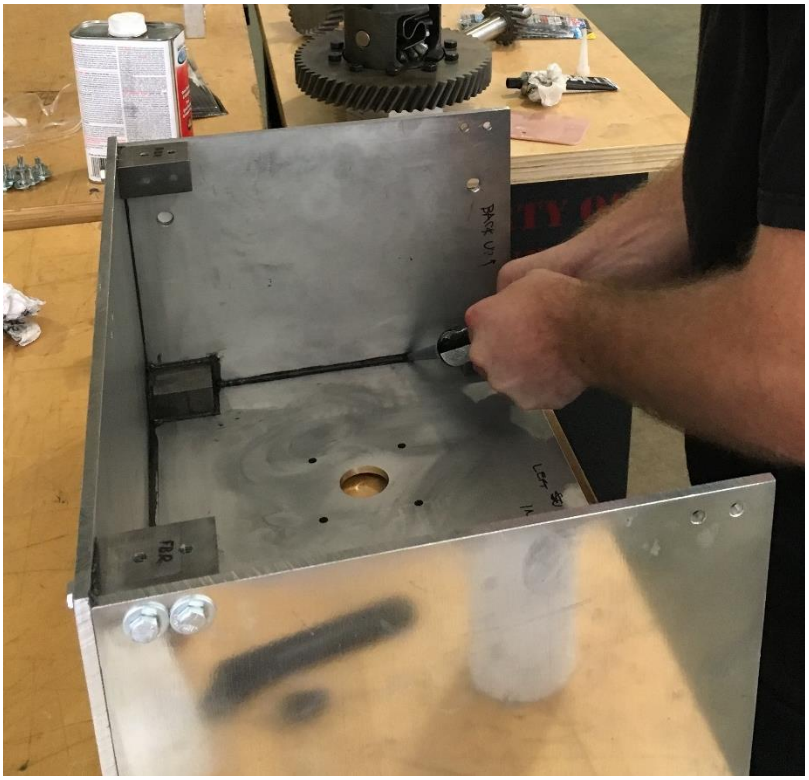

Figure 56. We put extra RTV in the corners of every joint and used a spreading tool to ensure the RTV made it all the way in the corner and clean any excess. The housing is sitting on the left side plate in this picture.

A spreading tool was used at every seam to clean up any excess sealant and ensure the sealant was pressed into the corner of each seam. Without shafts assemblies installed, it made for easy access to these areas that were otherwise difficult to access. Next, we bolted on the bearing mounts to the left side plate. We only applied RTV around the lower mounting holes knowing the upper holes were well above the oil level when the drivetrain is appropriately filled. With the right side housing plate still uninstalled, we bolted on the bearing mounts in the same fashion. Then, we set the bearings in their respective mounts and installed the differential and shaft assemblies. Before we installed the right side housing plate, we applied RTV to the right side housing blocks, and the three edges that the right plate would mate with. The right plate was set over the input shaft and down to mate with the front, rear, and bottom housing plates. Using the spreader proved to be more difficult with all of the components installed, but we managed a clean finish in each of the seam corners. The top plate was left uninstalled and without any RTV in the seams because of the fact that the oil level will not reach even half the height of the drivetrain.

12.0 Testing

All the testing discussed was performed on Cal Poly's campus in Bonderson.

12.1 Speed Reduction Test

The first test performed was a speed reduction verification. This test was just a simple check to make sure our gear ratio physically matched what we designed on paper. To perform the tests, we ran through the following test procedure:

Objective:

The purpose of this test is to verify the speed reduction achieved by the drivetrain. Since the drivetrain incorporates an 8.04:1 gear reduction, the input shaft should rotate 8.04 revolutions causing 1 revolution of the output shafts.

Equipment:

- Assembled Drivetrain
- Painter's tape
- Box cutter or other knife

Procedure:

1. Place assembled drivetrain on flat ground or a sturdy tabletop.
2. Using the painter's tape and knife, cut 4 small pieces of tape in the shape of an arrow. These will act as pointers to track the rotation of the input and output shafts.
3. Place one tape marker on the side housing plate just above each of the input and output shafts. Make sure the marker is pointing toward the centerline of the shafts.
4. Repeat step 3, but instead place the tape markers on the shafts themselves. The marker should point down the centerline of the shaft and line up with the tape on the side housing plate so that the tips of the arrows meet.
5. Turn the input shaft in either direction, continuing to rotate until the ring gear makes one complete revolution, according to the tape marker placed previously. Also take note of the number of rotations on the input shaft to complete one revolution of the ring gear.
6. Record this number in the table below. The designed gear ratio is 8.04:1. Compare the experimental value and the theoretical values. Seeing .04 revolutions of the input shaft may be hard to see, but the experimental ratio should be at least 8:1.
7. Repeat steps 5 and 6 if further verification is deemed necessary.

Results:

Our test results matched what we designed for and predicted. The output shafts rotated 1 revolution while the input shaft rotated 8.04 revolutions. Our experimental setup is shown below in Figure 57.

Figure 57. Our experimental setup was very simple and required very few additional supplies. The tape markers helped us keep track of the revolutions of each shaft.

12.2 Original Torque Increase Test

The second test performed was the torque increase test. We again followed the procedure beneath:

Objective:

The purpose of this test is to verify the torque increase achieved by the drivetrain. Since the drivetrain incorporates an 8.04:1 gear reduction, the torque at the output shafts should be 8.04 times that of the torque applied at the input shaft.

Equipment:

- Assembled Drivetrain with Acrylic Top Installed
- Axle Shafts (2) and C-Clip Retainers (2)
- Input Coupler
- Digital Torque Wrenches (3)

Procedure:

Set-Up

1. Place the drivetrain on the ground, and install the axle shafts into the differential spider gears. Then, set the retaining c-clips on the ends of the axle shafts, which will

prevent the axle shafts from sliding out of the spider gears. Next, place the coupler on the input shaft.

2. Connect two of the digital torque wrenches to the two output shafts*, and rotate the spider gears until the handles of both torque wrenches are touching the floor. Lastly, connect the third torque wrench to the input shaft coupler.

* Note: The input and axle shafts have nuts welded onto their ends, so that the torque wrenches can attach to the shafts.

<u>Statistical Uncertainty</u>

1. Have one individual located at each of the three torque wrenches. The two individuals located at the axle shafts will need to keep the end of the wrench's handle fixed to the ground. This will ensure that the axle shafts do not undergo any revolution and create loads being sent through the drivetrain's gearing.
2. With the output shafts fixed, the individual located at the input shaft will apply a torque using the torque wrench. Try to keep the applied torque constant for roughly five seconds. The torque applied should be 5 lb-ft.
3. As the desired 5 lb-ft of torque is being applied, all individuals will record the torque readout on their respective digital torque wrench.
4. Conduct a total of five trials and record torque measurements in the table provided

<u>Linear Correlation</u>

1. Take additional measurements for 7.5, 10, and 12.5 lb-ft torque inputs. For each torque input, conduct five trials and record data.

Data Processing:

The ratio of the sum of the output torques to the applied torque should be equal to the drivetrain's 8.04:1 gearing ratio. Summing the two output torque readings is necessary, as the differential creates a 50-50 torque split between the two axle shafts. If the ratio is found to be lower than the 8.04:1 gearing ratio, frictional losses are likely to be the root cause.

Run statistical analysis on the five trials that were conducted at the desired input torque to determine the mean torque increase ratio for the data sample. Additionally, find the error and standard deviation associated with the results.

Plot the experimental data collected and determine if a linear correlation is achieved, with a slope that falls within the error of the mean torque increase value determined from the statistical analysis.

Additional Information:

<u>Harbor Freight Torque Wrench Specifications</u>

- Torque range: 29.5-147.6 ft. lbs.
- +/- 2%

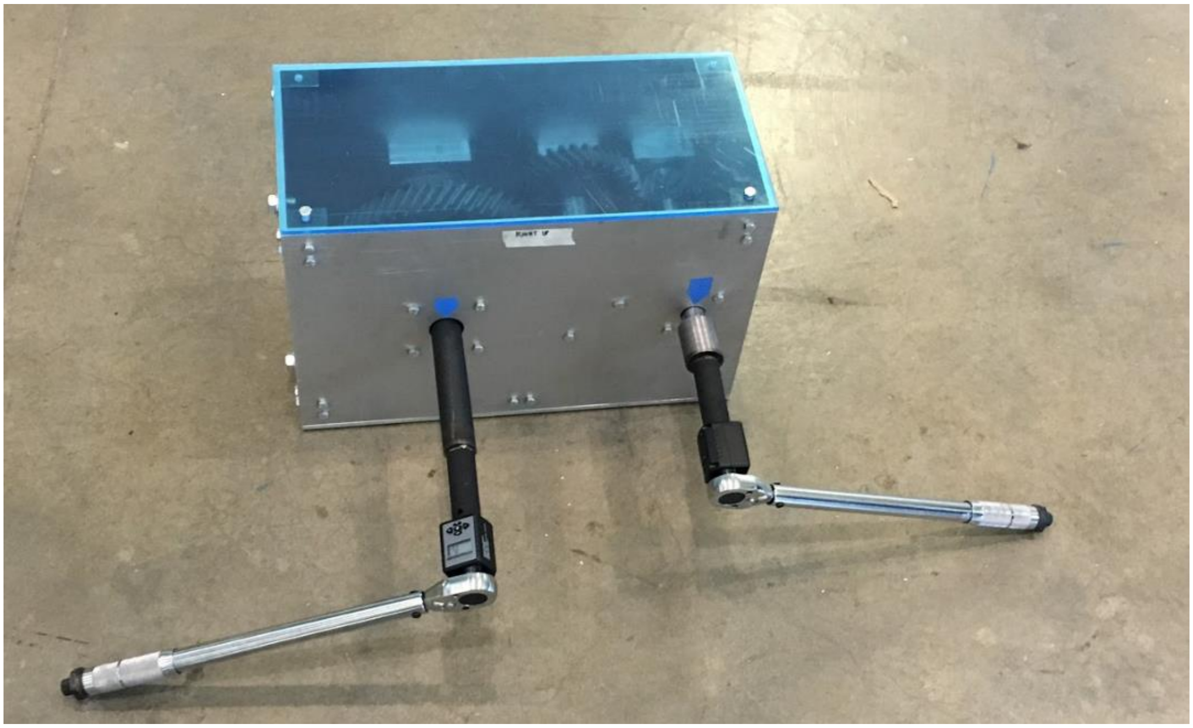

Figure 58. The input shaft with a torque wrench on the right-hand side is used to input the torque to the drivetrain. The corresponding output torque will read from the digital adapter coupled to the torque wrench on the left-hand side.

Results:

When following the above procedure, we found the torque wrenches on the output shafts were not receiving a 50-50 torque split. Occasionally, the torque values would split about 66% - 33% between the two output shafts. However, the torque would frequently be applied entirely through one shaft, while the other received zero torque. During some trials, the torque wrench on the left side received all the torque, and in other instances the torque wrench on the right side would receive all the torque. This was not due to the differential, as we visually confirmed that the spider gears in the differential were locked in place by the S-spring. We determined that this was due to the backlash in the torque wrenches, and whichever torque wrench removed it's backlash first, was the one which received all the torque. We do not believe that the varying torque splits from these trials reflect any issues with our drivetrain design. Instead, it showed that the equipment that we were using to conduct the test was not right for the job. We were unable to devise a way to simultaneously and equally remove the backlash from both torque wrenches, so we decided to conduct the test using only two of the three torque wrenches as seen in Figure 58. One remained at the input shaft to apply the torque, and only one output shaft was fitted with a digital torque wrench. This method for testing removed the variability in the torque split that we were receiving and allowed for consistent torque readings over the twenty trials. As seen in Table 13 we were able to neglect using the third torque wrench and found a range of output to input torque ratios of 6.84:1 – 7.59:1. The reason we were not able to achieve the designed 8.04:1 torque ratio experimentally is due to mechanical losses through the system.

Table 13. Truncated experimental data set from the twenty trials. The applied and output torques are averaged from five trials at each specified applied torque. The second output wrench was not used so we only took readings from one wrench at the output.

Trial #	Applied Torque	Output Torque at Wrench #1	Output Torque at Wrench #2	Sum of Output Torque Readings	Ratio of Sum of Output Torques to Applied Torque
	[lb-ft]	[lb-ft]	[lb-ft]	[lb-ft]	
1	5.1	35.3	N/A	35.3	6.92
2	7.5	53.6	N/A	53.6	7.14
3	10.0	68.4	N/A	71.6	6.84
4	12.5	94.9	N/A	94.9	7.59

The full experimental data set can be seen in Appendix G.2. With this data we performed statistical analysis to determine the uncertainty in our torque measurements at the input shaft and single output shaft. The sources of error stem from only two places. There is calibration error in the digital torque wrench from the manufacturer as well as uncertainty in the reading from the resolution of the digital torque wrench. For each set of trials at different input torque values (i.e. 5.1, 7.5, 10.0, 12.5 lb-ft), we averaged the output torque readings to enable us to perform the analysis. Figure 59, shows the average input torque values and the average output torque values for each set of trials. Error bars are included to shows the range at which the values could have been due to the uncertainty. Knowing the relationship between the input and output torque is linear, we used a linear curve fit line to find a value that best described what the gear ratio is with mechanical losses involved. The error also increases with increased torque input which can also be seen in the figure below. For example, the trials with input torque at 5.1 lb-ft, the output torque was 36.6 ± 0.11 lb-ft. For trials with 12.5 lb-ft of input torque, the output torque was 91.6 ± 1.83 lb-ft. The reason for increase in error is due to the calibration error. From the manufacturer, the uncertainty in the digital torque wrench was ± 2% of the reading. This means the error scales linearly with the torque reading whereas the error in the resolution remains constant regardless of the reading.

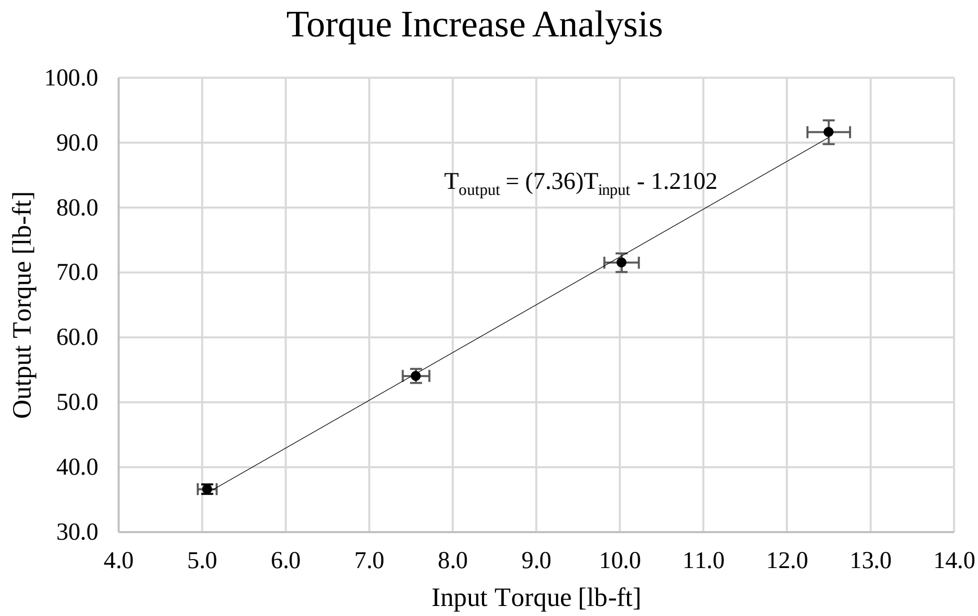

Figure 59. A linear fit of the torque test's experimental data determined a 7.36:1 ratio for our drivetrain, despite its actual 8.04:1 ratio.

12.3 Oil Leak Test

The final test performed was the leakage test as once the plates were sealed there would be no further disassembly of the drivetrain. Therefore, we wanted to ensure the function of our drivetrain with the previous two tests before taking this final step. To perform the final test, we followed the guidelines below:

Objective:

The purpose of this test is to ensure proper sealing between housing plates of the drivetrain. If the drivetrain leaks, it will not be useable due to the possible failure modes caused by leaking oil/low oil level.

Equipment:

- Assembled Drivetrain
- Specified gear oil from owner's manual
- Funnel
- Ratchet and socket to fit drain/fill plugs
- Input Coupler
- Plastic container big enough to fit the entire drivetrain housing

Procedure:

1. With the drivetrain completely assembled and resting on a flat surface inside a plastic container, verify the oil drain plug is tight. The plug is located on the back housing plate near the bottom.
2. Using the ratchet and socket, remove the oil fill plug. This plug is located on the back housing plate near the top.
3. Insert the funnel into the fill hole and fill the drivetrain with the specified 9 quarts of oil.
4. Replace the oil fill plug in its original location.
5. Let the drivetrain sit overnight.
6. The next day, inspect the drivetrain around the housing seams and input/output seal locations, visually inspecting for leaking areas and physically feeling for oil. The oil level does not exceed the input/output seal locations so there should be no sign of oil in these areas.
7. Take note and if desired, pictures of the leaking areas if any.
8. Proceed to recommended repair procedure if any repairs are necessary.

Performing the Oil Leak Test

After letting the drivetrain sit for the allotted time, we removed the drivetrain from the plastic container used to capture any oil leakage. Via visual inspection it appeared that no oil had leaked onto the plastic bottom. However, once lifting the drivetrain out of the container there was a slight residual oil on the back plate. Upon closer inspection it was discovered that two holes (one corner block threaded hole on the lower left side of the back plate and the drain plug) allowed slight oil seepage. To improve the sealing of these two holes in particular, teflon tape was used to wrap the threads of the bolts so that when they were reinserted it added extra compression and a much tighter sealing interface. For future dynamic testing purposes, we are certain that the drivetrain will contain oil and will not cause any issues.

Reach Goal Update

One reach goal our team originally had hoped to accomplish was a dynamic verification of our drivetrain system. This was to be accomplished by installing the drivetrain in the BMW test vehicle our sponsor has purchased, and by running the vehicle in a controlled test setting on a chassis dyno. The test vehicle was originally planned to be ready by the first week of May 2017, but since the timeline shifted and the prototype vehicle will no longer be ready in time for our system to be installed and tested. As such, we were unable to conduct a dynamic test where there would be both a speed input and load being transmitted through the drivetrain. We recommend that our sponsor still conduct a dynamic test on a chassis dyno upon the completion of retrofitting the test vehicle with its necessary powertrain systems.

13.0 Conclusions

13.1 Design Concept vs Final Prototype

Our drivetrain design concept came to full realization with our final prototype. No compromises or alterations to our design were made during the manufacturing stage and we were able to build our drivetrain concept exactly as planned because it was designed for ease of manufacturing. Additionally, when designing the drivetrain, we took into account the tools and resources available to us on campus, again allowing for the convenience and cost reduction of manufacturing with readily available equipment. Figure 60 portrays how the drivetrain build resembles the design concept.

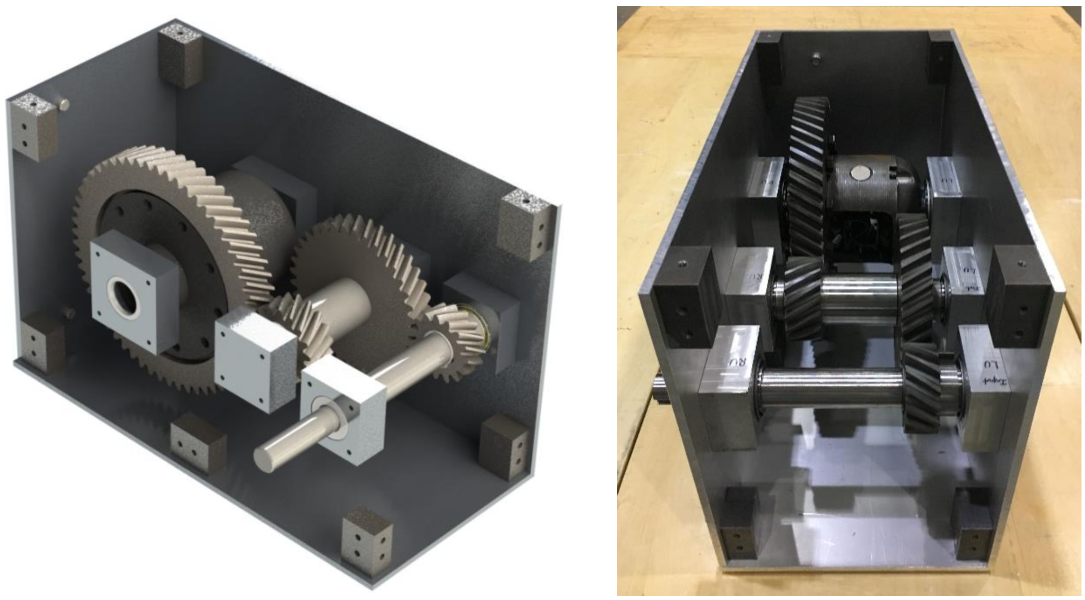

Figure 60. Side by side comparison of design concept (left) and the final prototype (right).

13.2 Prototyping Cost

Due to the change in the application of our drivetrain, the system now is built as a one-off prototype for testing purposes and will not be mass produced to incorporate into the ST-5. However, after our sponsor implements the drivetrain into his prototype vehicle our system may become the basis for a further improved drivetrain system. Since Sharpell Technologies plans on duplicating the drivetrain for installation on the front axle of the prototype vehicle to create an AWD powetrain, our bill of materials will be a useful tool to determine what components need to be purchased and the overall cost of the actual drivetrain itself. Without the inclusion of labor and manufacturing processes, the drivetrain would cost $6,514.60. Please refer to Appendix F.21 for the bill of material’s component cost breakdown and Appendix F.22 for a detailed list of the vendor list and contact information.

Additional costs were incurred, in the form of tools and testing equipment. All costs associated with our senior project totaled $6,764.62 as detailed in the overall budget – Appendix F.16. We were successfully able to stay within our sponsor's budget cap of $10,000.

13.3 Future Manufacturing Recommendations

For the manufacturing of any machined parts in the future, we recommend that carbide milling cutters are used rather than high speed steel (HSS) cutters. The parts that we used the HSS cutters on include the steel housing blocks and all six bearing mounts. Granted, the HSS cutters we used were not very sharp, but even so using carbide tooling may have relieved some of the issues we had when machining, specifically the bearing mounts. Carbide tools allow much faster cutting speeds that would take down our machine time, and they are harder than HSS which when sharp, will be more efficient at removing material. When the tool can properly remove material, the tool will flex less, making the features more accurate and not require additional operations to produce a finished part within the desired tolerances. The steel housing blocks would benefit from carbide tooling as well through less machine time and a much better surface finish. These both were lacking when using the HSS tools at Cal Poly's manufacturing locations. However, carbide tooling is expensive and as we were limited by budget, we made due with the tools available.

Another recommendation we have is pressing the bearing caps into the bearing mounts with the hydraulic press from the start, rather than baking the mounts to expand the metal. We did not realize the press was available at the time, and using it would have sped up the process and made it less complex of a process.

Lastly, we also suggest buying a new differential from the beginning rather than trying to refurbish an older one. A good amount of time was consumed removing and reinstalling bearings, installing the spider gears and clutch pack, and fitting the S-spring. With a new differential, the only thing we would have had to do is press on new bearings which is fast and simple process.

Another large modification that would simplify the assembly is casting a housing. As mentioned early on, this was not feasible within the time frame of our project but being able to cast a housing would remove the need to mill and CNC all the corner and bearing blocks as well as minimize the housing deflection and also mitigate any leakage issue.

13.4 Safe Operation of Drivetrain

To ensure the safety of our sponsor and any of his associates, we created a user's manual that is in Appendix H.1-H.2. Included in the manual are multiple provisions to take note of when operating the drivetrain to minimize health risk to the user. A simple maintenance plan is also included to keep the drivetrain running properly which will result in longer drivetrain life, less risk for parts to fail, and ultimately keep the user safe. We developed the maintenance plan based on our design failure mode effect and analysis (DFMEA) in Appendix C.3 – C.6. The DFMEA is and organized list of possible failure modes of components in the drivetrain and how we would prevent them or combat them if failure happened to occur.

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15.0 Appendices

A. Design Decisions

B. Project Timelines

C. Project Safety

D. Excel Design Tools and Analysis

E. Part Drawings

F. Drivetrain Component Cost and Specifications

G. Testing Information

H. Guidelines for Safe Drivetrain Operation

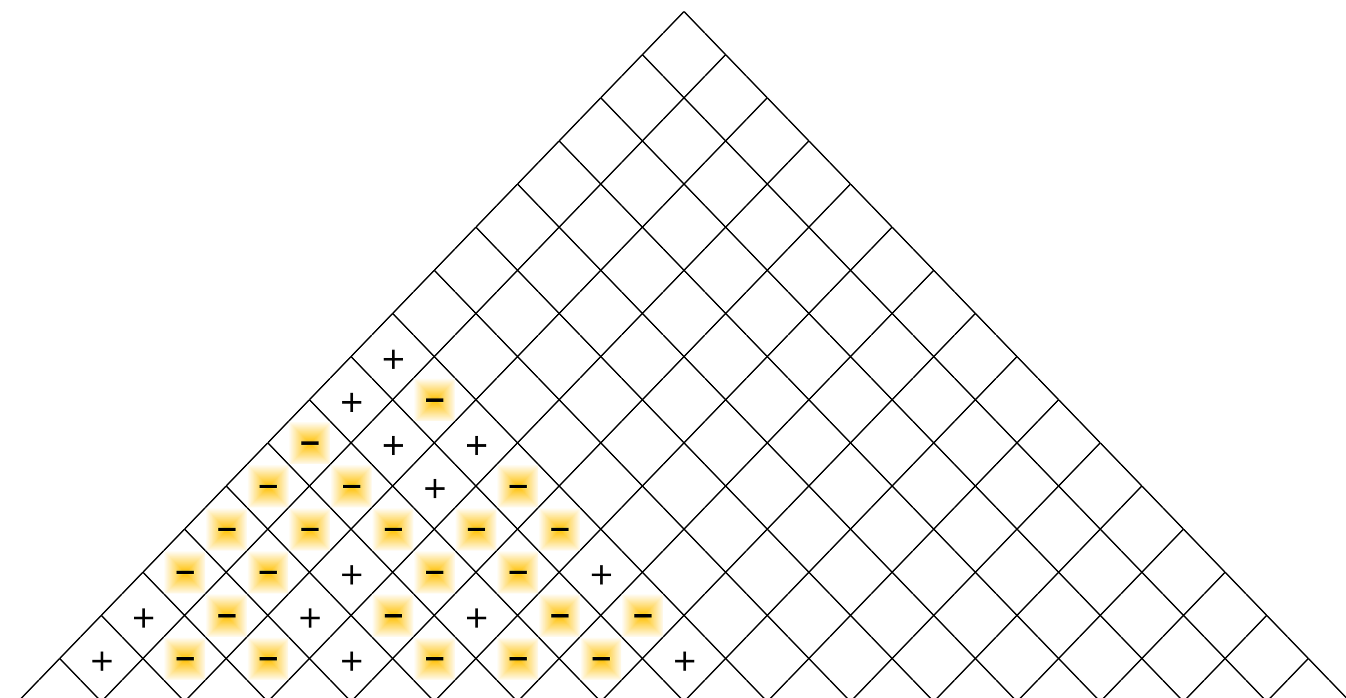
Table of Contents

QFD: House of Quality
Project:
Revision:
Date:

Correlations	
Positive	+
Negative	−
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼



Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Direction of Improvement	▲	◇	▲	▲	▼	▲										
HOW: Engineering Specifications	Max Wheel rotation under load	Dynamometer Test	Fatigue Lifetime Testing	Shaft Balance	Shaft Deflection	Lubrication Housing Seal	Source cost of material and labor	Drive car up various grades	Use test rig to measure torque on lo							

WHO: Customer — NOW: Current Product Assesment - Customer Requirements

Row #	Weight Chart	Relative Weight	Zach Sharpell (Sponsor)	Soccer Mom	Off-Roaders		Maximum Relationship	WHAT: Customer Requirements (explicit & implicit)	1	2	3	4	5	6	7	8	9	Our Current Product	BorgWarner 31-03 E-Drive	Tesla Model X	Tesla Model S	Chevy Bolt	Row #
1		10%	8	4	8		9	Top Speed = 155 mph	●	▽	▽	▽	▽	▽	▽	▽	▽	3	2	2	2	1	1
2		9%	10	2	8		9	Input power at 400 HP	○	●	▽	○	○	▽	▽	▽	▽	3	2	2	2	1	2
3		9%	10	2	8		9	Input torque at 700 ft-lbf (0-6,000 rpm at max torque)	●	●	○	○	●	▽	○	○	○	3	1	2	2	1	3
4		10%	8	4	8		9	rpm range of 0-11,000 rpm	●	○	○	○	●	▽	▽	▽	▽	3	4	4	5	2	4
5		8%	6	4	7		9	Capable of 0-60 mph in 5 seconds max time	●	▽	▽	▽	▽	▽	▽	▽	▽	3	3	4	2	1	5
6		14%	9	10	8		9	100,000 mile reliability target	▽	▽	●	▽	●	●	○	▽	▽	3	3	3	3	3	6
7		11%	7	10	3		9	Noise/Vibration/Harshness	▽	▽	○	○	○	●	▽	▽	▽	3	5	5	5	3	7
8		9%	6	3	10		9	Common grade (simple analysis)	▽	▽	▽	▽	▽	▽	▽	●	▽	3	2	4	3	1	8
9		8%	8	2	8		9	Min. Torque to wheel/Diff. Functionailty	○	▽	▽	▽	▽	▽	▽	○	●	3	2	5	5	1	9
10		12%	10	7	7		9	Max Cost = $10,000	▽	▽	○	○	▽	▽	●	▽	▽	3	5	5	5	4	10
11		0%																					11
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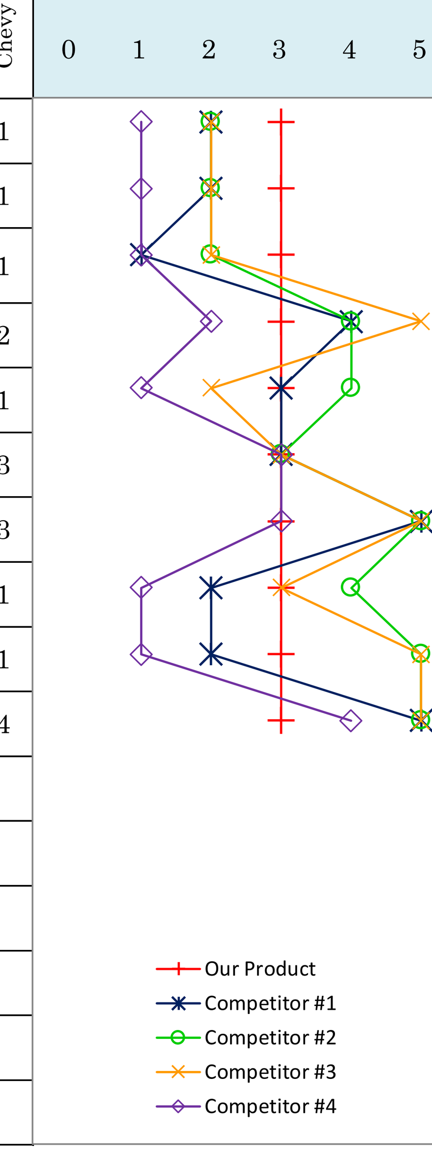
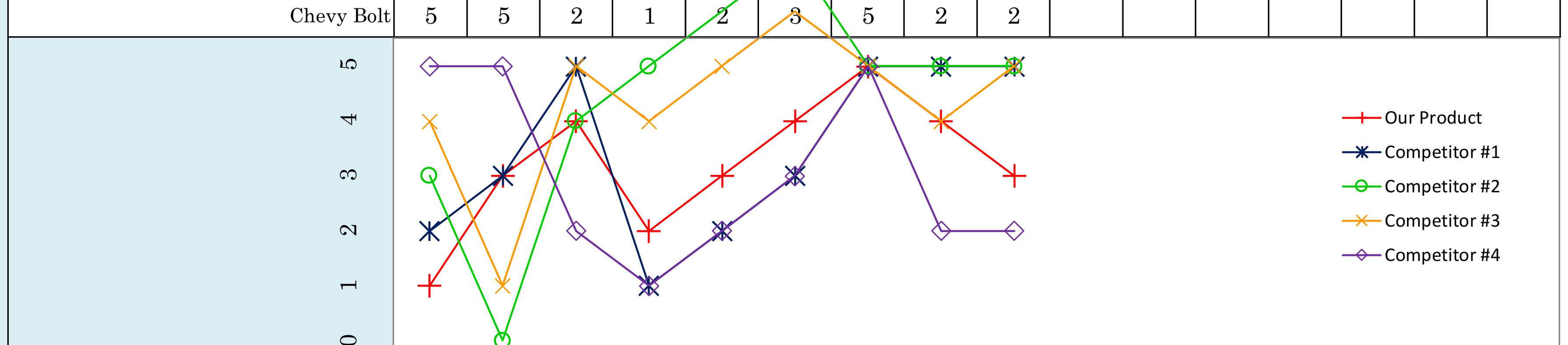


	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
HOW MUCH: Target	Requires detailed analysis	400 HP/ 700 LB-FT	>100000 miles	<1 oz-in of imbalance	<.01 INCHES	<.1 mL per 3 days	<$10000	>30 degree grade	Minimum of 250 LB FT per wheel							
Max Relationship	9	9	9	3	9	9	9	9	9							
Technical Importance Rating	426.48	263.32	296.78	201.53	402.29	302.25	242.63	206.15	183.59							
Relative Weight	17%	10%	12%	8%	16%	12%	10%	8%	7%							
Weight Chart																

Current Product Assesment - Engineering Specifications

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Our Product	1	3	4	2	3	4	5	4	3							
BorgWarner 31-03 E-Drive	2	3	5	1	2	3	5	5	5							
Tesla Model X	3	0	4	5	6	7	5	5	5							
Tesla Model S	4	1	5	4	5	6	5	4	5							
Chevy Bolt	5	5	2	1	2	3	5	2	2							



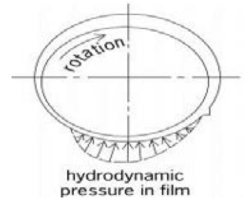
Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16

Template Revision: 0.9 Date: 4/23/2010
Christopher Battles

Team E- Drive Weighted Decision Matrices

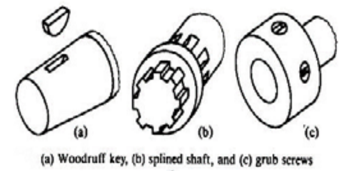
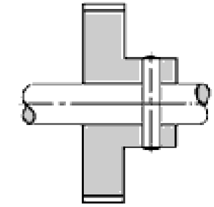
			Spec #	1	2	3	4	5	6	7	8	9	10	TOTAL
			Description	Output rpm required for 155 mph top speed	Input Power	Input Torque	Rpm Range	Reliability	Output torque to achieve 0-60 time of 5 [s]	Cost	Housing Leakage	Torsional Sealing Test	Torque to Rear Wheels	
	Concept Description	**Weight [%]**		8%	10%	18%	11%	19%	5%	3%	6%	4%	16%	100%
Concept 1	Longitudinal Motor Layout Helical & Hypoid Gears, Vertical & Longitudal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		6	8	8	7	7	6	8	6	9	8	
		Wgt x Rtg		0.48	0.8	1.44	0.77	1.33	0.3	0.24	0.36	0.36	1.28	7.36
Concept 2	Longitudinal Motor Layout Helical & Hypoid Gears, Vertical & Transverse Housing Parting Lines, BT-B LSD	**Rating [1-10]**		6	8	8	7	7	6	8	6	7	8	
		Wgt x Rtg		0.48	0.8	1.44	0.77	1.33	0.3	0.24	0.36	0.28	1.28	7.28
Concept 3	Longitudinal Motor Layout Helical & Hypoid Gears, Horizontal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		6	8	8	7	7	6	8	8	4	8	
		Wgt x Rtg		0.48	0.8	1.44	0.77	1.33	0.3	0.24	0.48	0.16	1.28	7.28
Concept 4	Longitudinal Motor Layout Bevel & Helical Gears, Vertical & Longitudinal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		9	7	7	4	5	9	6	6	9	8	
		Wgt x Rtg		0.72	0.7	1.26	0.44	0.95	0.45	0.18	0.36	0.36	1.28	6.7
Concept 5	Longitudinal Motor Layout Bevel & Helical Gears, Vertical & Transverse Housing Parting Lines, BT-B LSD	**Rating [1-10]**		9	7	7	4	5	9	6	6	7	8	
		Wgt x Rtg		0.72	0.7	1.26	0.44	0.95	0.45	0.18	0.36	0.28	1.28	6.62
Concept 6	Longitudinal Motor Layout Helical Gears, Horizontal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		9	7	7	4	5	9	6	8	4	8	
		Wgt x Rtg		0.72	0.7	1.26	0.44	0.95	0.45	0.18	0.48	0.16	1.28	6.62
Concept 7	Transverse Motor Layout Helical Gears, Vertical & Longitudinal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		8	8	8	9	8	8	6	6	9	8	
		Wgt x Rtg		0.64	0.8	1.44	0.99	1.52	0.4	0.18	0.36	0.36	1.28	7.97
Concept 8	Transverse Motor Layout Helical Gears, Vertical & Transverse Housing Parting Lines, BT-B LSD	**Rating [1-10]**		8	8	8	9	8	8	6	6	7	8	
		Wgt x Rtg		0.64	0.8	1.44	0.99	1.52	0.4	0.18	0.36	0.28	1.28	7.89
Concept 9	Transverse Motor Layout Helical Gears, Horizontal Housing Parting Lines, BT-B LSD	**Rating [1-10]**		8	8	8	9	8	8	6	8	4	8	
		Wgt x Rtg		0.64	0.8	1.44	0.99	1.52	0.4	0.18	0.48	0.16	1.28	7.89

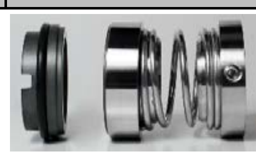
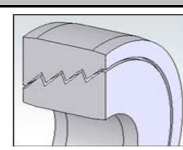
Pugh Matrices

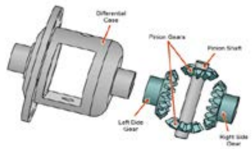
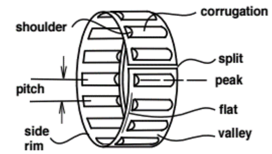
Bearing Types			
	Tapered Roller	**Ball**	**Hydrodynamics**
			
Thrust/Axial Load	+	0	+
Radial Load	+	0	+
Size	S	0	S
Weight Load Capacity	+	0	+
Simplicity	-	0	-
Cost	-	0	-
Σ +1	3	0	3
Σ -1	1	0	1
Σ S	2	0	2
Total	+2	0	+2

Key
Datum
Scale +1 or -1
Total

Gear Types			
	Spur	**Bevel**	**Helical**
			
Tooth Contact	0	+	+
Noise	0	+	+
Size of Thrust/Axial Load	0	-	-
Size of Radial Load	0	S	+
Cost	0	-	-
Σ +1	0	3	3
Σ -1	0	1	2
Σ S	0	1	0
Total	0	+2	+1

Gear and Shaft Interface					
	Press Fit	**Rectangular Key**	**Spline**	**Set Screw**	**Cotter Pin**
					
Gear Size (Hub)	0	S	S	-	-
Resistance to Shear Stress	0	-	S	S	S
Backlash	0	-	S	-	S
Availability of Shaft	0	S	-	S	S
Shaft Modifications	0	-	-	-	-
Additional Part Needed	0	-	S	-	-
Fit	0	-	+	+	-
Torque Transmission	0	-	S	S	-
Cost	0	-	-	-	-
Σ +1	0	0	1	1	0
Σ -1	0	7	3	5	6
Σ S	0	2	5	3	3
Total	0	-7	-2	-4	-6

Sealant Options				
	Unbalanced O-rings	**CF Series Labyrinth**	**Single Loaded, Single Lip Oil Seal**	**Bearing Grease Seal**
				
Wear and Tear	S	S	0	S
Leakage	S	S	0	S
Compatibilty	S	S	0	S
Cost	S	S	0	S
Σ +1	0	0	0	0
Σ -1	0	0	0	0
Σ S	4	4	0	4
Total				

Rear Differential				
	Open	Clutch Type LSD	ATB	BT-B LSD
				
Complexity	0	-	-	S
Cost 150k mi.	0	-	-	S
Life	0	-	S	S
Rebuild Cost	0	-	-	-
Understeer	0	-	S	S
Tire Wear	0	-	S	S
Traction	0	+	+	+
Impulse Survival	0	S	-	+
Σ +1	0	1	1	2
Σ -1	0	6	4	1
Σ S	0	1	3	5
Total	0	-5	-3	+1

Housing Parting Lines			
	Veritcal and Longitudinal	Vertical and Transverse	Horizontal
			
Ease to Cast Housing	S	0	S
Ease to Assemble	+	0	+
Less Area to Leak Oil	S	0	+
Post-Processing	S	0	S
Σ +1	1	0	2
Σ -1	0	0	0
Σ S	3	0	2
Total	+1	0	+2

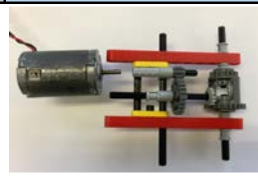
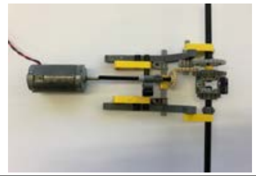
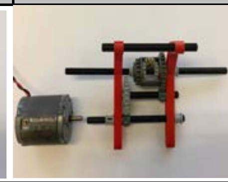
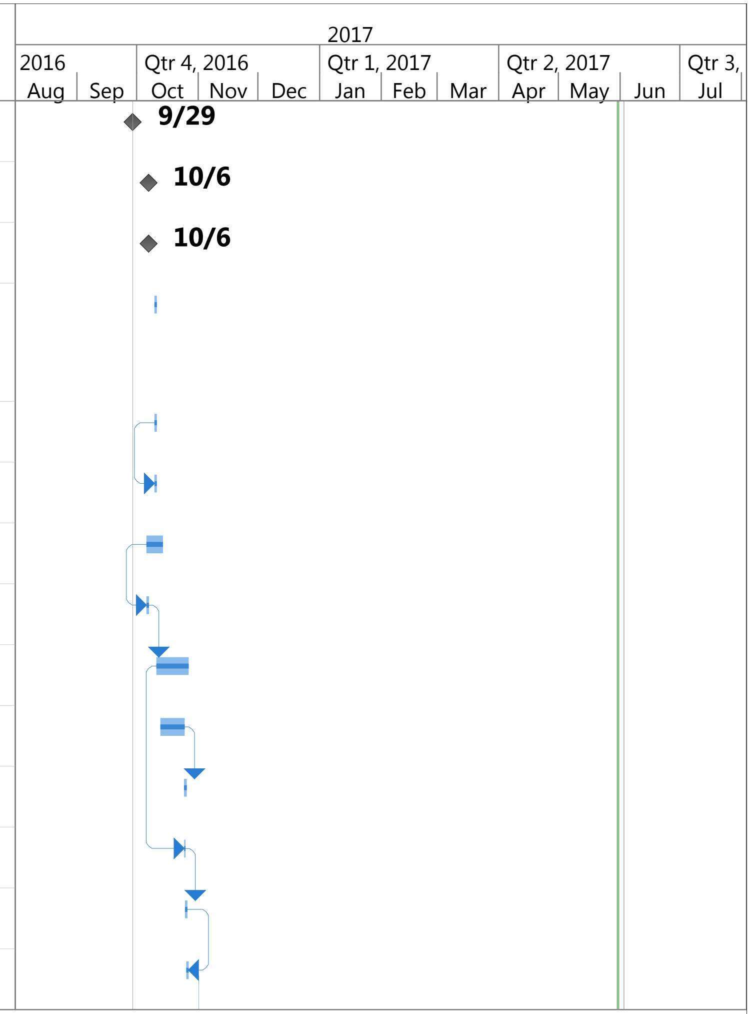
Drivetrain Layout			
	Longitudinal Motor Layout Helical & Hypoid Gears	**Longitudinal Motor Layout Bevel and Helical Gears**	**Transverse Motor Layout Helical Gears**
			
Compactness	0	S	+
Number of Gears	0	S	S
Number of Shafts	0	S	S
Number of Bearings	0	S	S
Efficiency	0	S	S
Ease of Gear Reduction	0	S	S
Availability of Stock Gears	0	-	-
Load transfer at high speed	0	S	S
Deflection of shafts	0	S	+
Shaft sizes	0	S	+
Range of common ratios	0	S	+
Wear	0	S	S
Cost	0	S	S
Σ +1	0	0	4
Σ -1	0	1	1
Σ S	0	12	8
Total	0	-1	+3

Table of Contents

Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
1	**Write and Send Intro Letter to Sponsor**	Thu 9/29	Thu 9/29
2	**Create Team Contract**	Thu 10/6	Thu 10/6
3	**Attend First Meeting with Sponsor**	Thu 10/6	Thu 10/6
4	Meet Outside of Class to Get to Know Group Members	Mon 10/10	Mon 10/10
5	Write Team Contract	Mon 10/10	Mon 10/10
6	Revise Team Contract	Mon 10/10	Mon 10/10
7	Conduct Preliminary research	Thu 10/6	Thu 10/13
8	Benchmark Similar Products	Thu 10/6	Thu 10/6
9	Write Project Proposal	Tue 10/11	Wed 10/26
10	Complete QFD/Specification Table	Thu 10/13	Mon 10/24
11	Complete Decision Matrices	Tue 10/25	Tue 10/25
12	Peer Review Project Proposal	Tue 10/25	Tue 10/25
13	Make Edits to Project Proposal	Tue 10/25	Wed 10/26
14	Submit Project Proposal to Sponsor	Wed 10/26	Wed 10/26

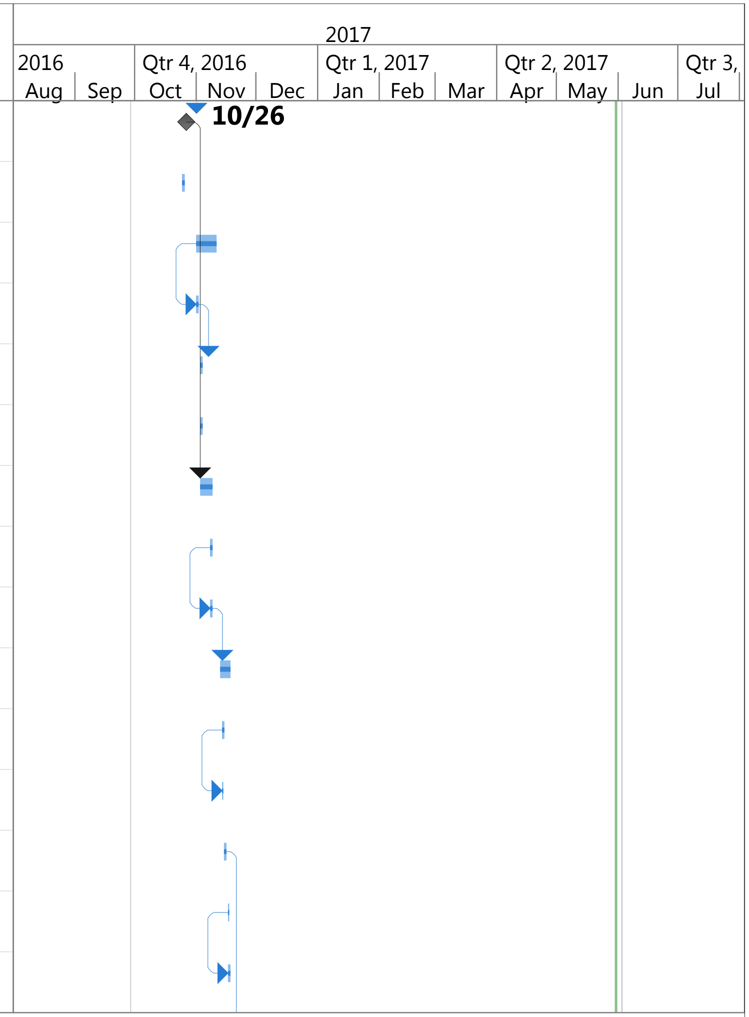


Project: E-Drive Gantt Chart-Ar
Date: Wed 5/31

Task	Inactive Summary	External Tasks
Split	Manual Task	External Milestone
Milestone	Duration-only	Deadline
Summary	Manual Summary Rollup	Progress
Project Summary	Manual Summary	Manual Progress
Inactive Task	Start-only	
Inactive Milestone	Finish-only	

Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
15	**Project Proposal**	Wed 10/26	Wed 10/26
16	Ideate For Power Transmission Function	Tue 10/25	Tue 10/25
17	Perform In-Depth Research on Concepts	Tue 11/1	Thu 11/10
18	Build Prototypes of Concepts	Tue 11/1	Tue 11/1
19	Evaluate Prototypes	Thu 11/3	Thu 11/3
20	Develop Gantt Chart	Thu 11/3	Thu 11/3
21	Make Project Proposal Edits Using Report Edit Log	Thu 11/3	Tue 11/8
22	Develop Pugh Matrices	Tue 11/8	Tue 11/8
23	Complete Final Decision Matrix	Tue 11/8	Tue 11/8
24	Draft PDR Report	Sun 11/13	Thu 11/17
25	Prepare Powerpoint	Mon 11/14	Mon 11/14
26	Practice PDR Presentation	Mon 11/14	Mon 11/14
27	Give PDR Presentation to Professor	Tue 11/15	Tue 11/15
28	Peer Review PDR Report	Thu 11/17	Thu 11/17
29	Edit PDR Report	Thu 11/17	Thu 11/17



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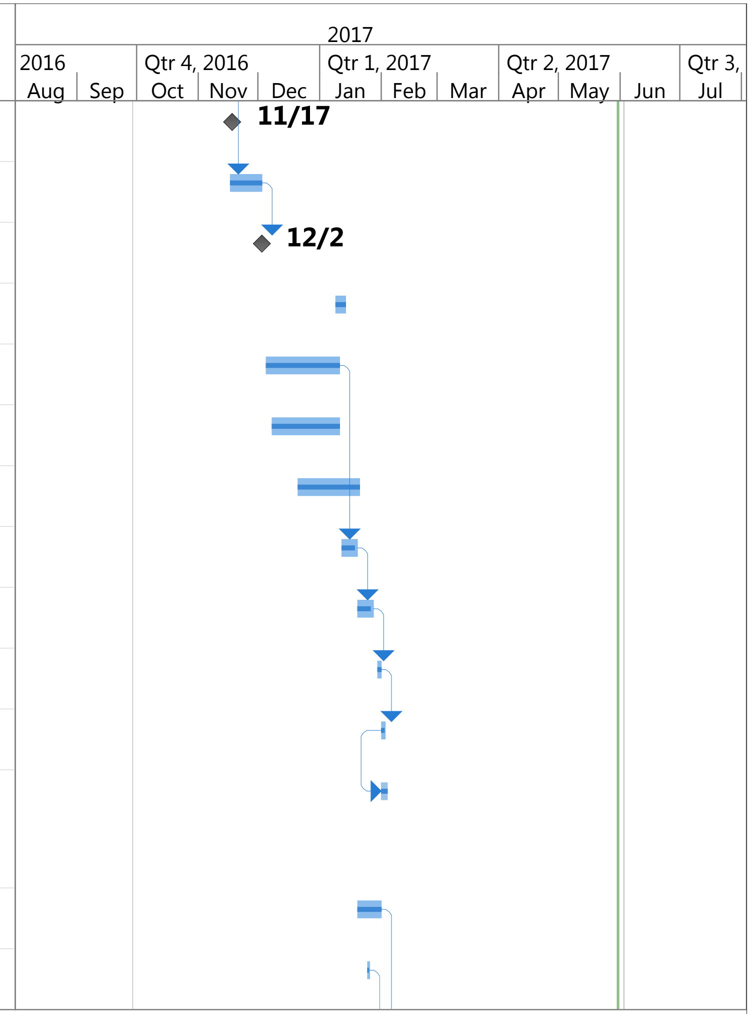
Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
30	**PDR Report**	Thu 11/17	Thu 11/17
31	Correct PDR Presentation Based on Feedback	Thu 11/17	Fri 12/2
32	**PDR Presentation w/ Sponsor**	Fri 12/2	Fri 12/2
33	Purchase Rear Differential	Mon 1/9	Fri 1/13
34	Validate Excel Tools	Mon 12/5	Tue 1/10
35	Complete Detailed Drivetrain Analysis	Thu 12/8	Tue 1/10
36	Research Testing Equipment	Wed 12/21	Fri 1/20
37	Use Excel Tools to Specify Parts	Thu 1/12	Thu 1/19
38	Find Availible Stock Part Sizes if Possible	Fri 1/20	Fri 1/27
39	Identify Other Critical Components	Mon 1/30	Tue 1/31
40	Adapt Solidworks Models Accordingly	Wed 2/1	Thu 2/2
41	Complete Solidworks Models and Detailed Drawings	Wed 2/1	Fri 2/3
42	Draft CDR Report	Fri 1/20	Tue 1/31
43	Complete BOM	Wed 1/25	Wed 1/25



Project: E-Drive Gantt Chart-Ar
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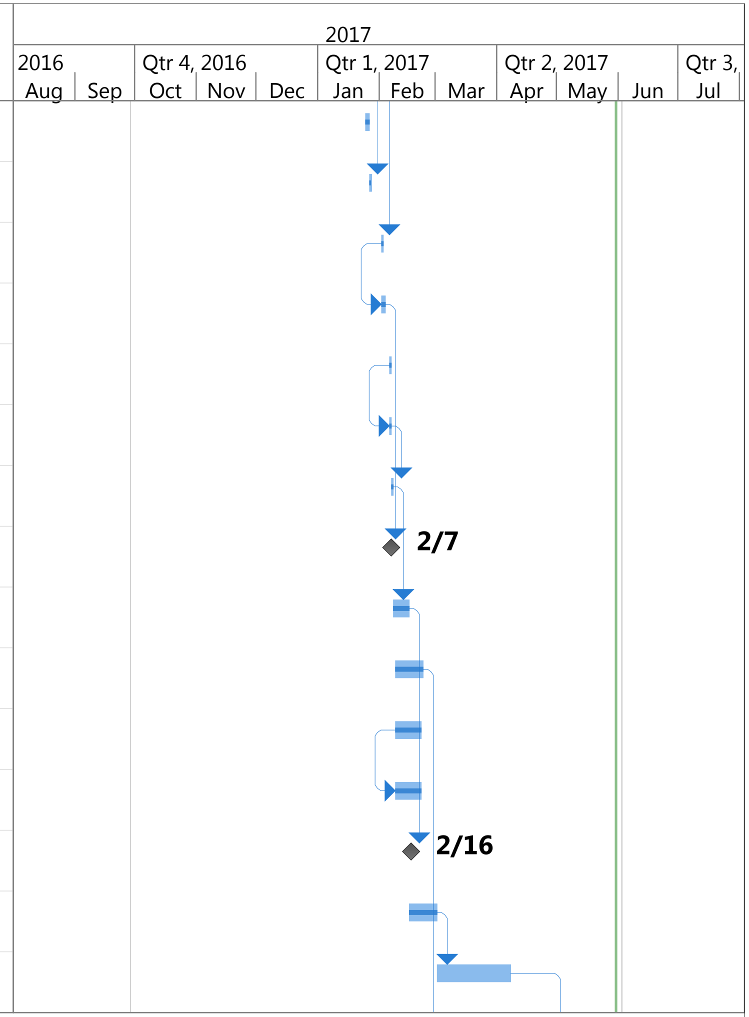
Task
Split
Milestone
Summary
Project Summary
Inactive Task
Inactive Milestone
Inactive Summary
Manual Task
Duration-only
Manual Summary Rollup
Manual Summary
Start-only
Finish-only
External Tasks
External Milestone
Deadline
Progress
Manual Progress

Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
44	Identify Safety Hazards	Wed 1/25	Thu 1/26
45	Complete Cost Analysis	Fri 1/27	Fri 1/27
46	Peer Review CDR	Thu 2/2	Thu 2/2
47	Make Edits to CDR Using Report Edit Log	Thu 2/2	Fri 2/3
48	Prepare Powerpoint for CDR Presentation	Mon 2/6	Mon 2/6
49	Practice CDR Presentation	Mon 2/6	Mon 2/6
50	Give CDR Presentation to Professor	Tue 2/7	Tue 2/7
51	**CDR**	Tue 2/7	Tue 2/7
52	Correct CDR Presentation Based on Feedback	Wed 2/8	Wed 2/15
53	Continue Transmission Dyno Search	Thu 2/9	Wed 2/22
54	Finalize Gear Sizing	Thu 2/9	Tue 2/21
55	Find New Gear Supplier	Thu 2/9	Tue 2/21
56	**CDR w/ Sponsor**	Thu 2/16	Thu 2/16
57	Order Gears	Thu 2/16	Wed 3/1
58	Gearing Manufacturing & Delivery	Thu 3/2	Fri 4/7



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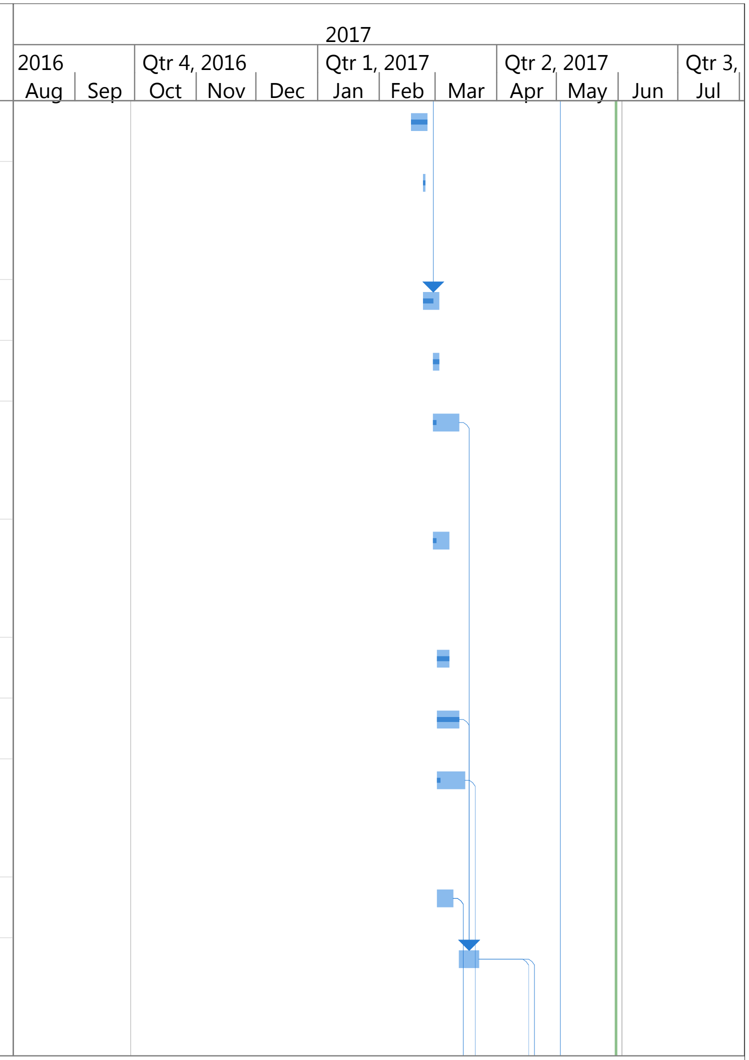
Inactive Milestone	Finish-only	

Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
59	Get Coupler Information From Zach	Fri 2/17	Fri 2/24
60	Call Timken - New Bearing and Bearing Housing Supplier	Thu 2/23	Thu 2/23
61	Develop Test Plan	Thu 2/23	Thu 3/2
62	Finalize Shafts	Tue 2/28	Thu 3/2
63	Finalize Housing Plate Sizes and Mounting Locations - Charissa	Tue 2/28	Sun 3/12
64	Start Editing CDR Report Based on Feedback - Drew	Tue 2/28	Tue 3/7
65	Determine Seal for Input Shaft - Kevin	Thu 3/2	Tue 3/7
66	Design Bearing Mounts - Kevin	Thu 3/2	Sun 3/12
67	Develop Specifics of Static Torque Increase Test - Kevin	Thu 3/2	Wed 3/15
68	Create First Draft of Operator's Manual - Drew	Thu 3/2	Thu 3/9
69	Order Raw Materials for Shafts, Housing, Bearing Mounts, and Keyways - Charissa	Mon 3/13	Wed 3/22



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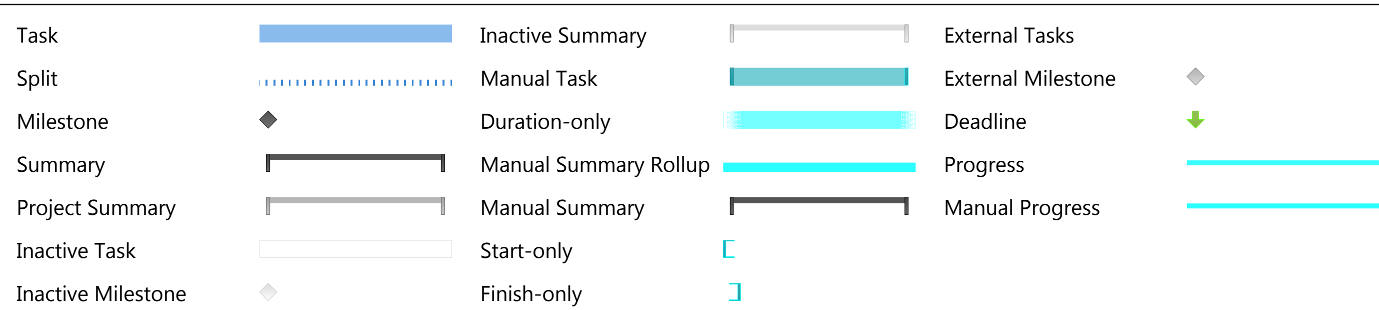
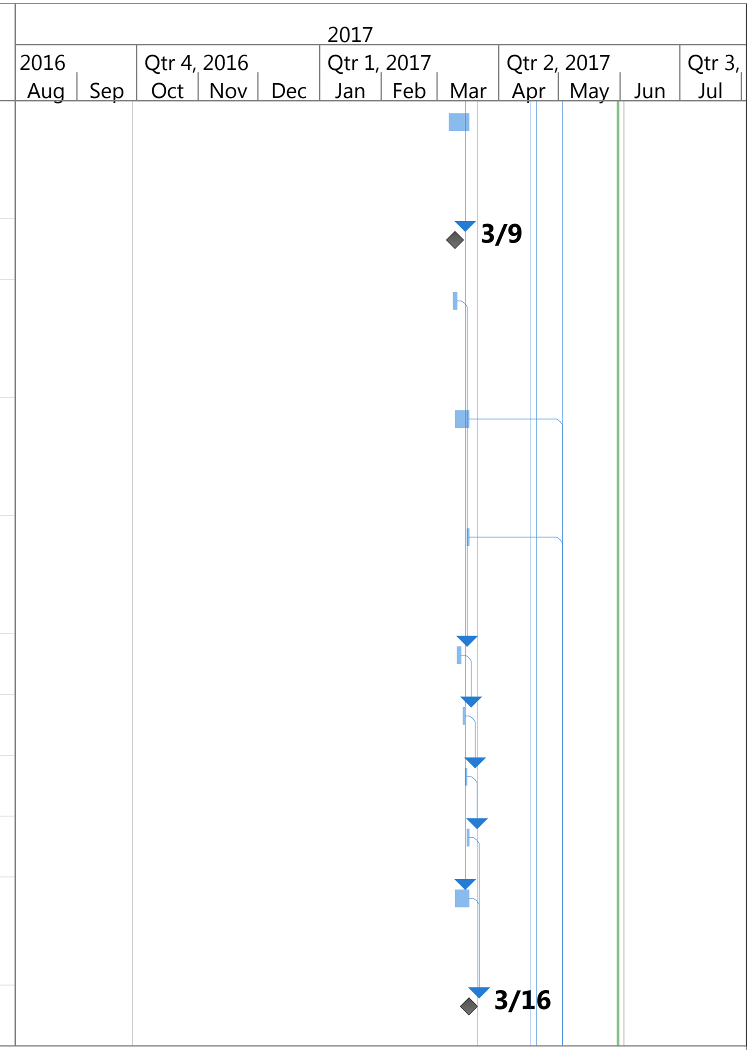


Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
70	Order Differential Bearings and IRS Spider Gears - Kevin	Tue 3/7	Thu 3/16
71	**Operator's Manual Draft Due**	**Thu 3/9**	**Thu 3/9**
72	Request Input Shaft Connection Information about Coupler	Thu 3/9	Fri 3/10
73	Verify Status of All Parts Needed for Assembly - Jimmy	Fri 3/10	Thu 3/16
74	Order Any Remaining Parts Needed for Assembly - Charissa	Thu 3/16	Thu 3/16
75	Write Project Update Report - Kevin and Jimmy	Sat 3/11	Sun 3/12
76	Peer Review Project Update Report - Charissa	Tue 3/14	Tue 3/14
77	Make Edits to Project Update Report - Jimmy	Wed 3/15	Wed 3/15
78	Send Project Update Report to Sponsor - Drew	Thu 3/16	Thu 3/16
79	Edit Operator's Manual and Complete Final Draft - Drew	Fri 3/10	Thu 3/16
80	**Project Update Report**	Thu 3/16	Thu 3/16



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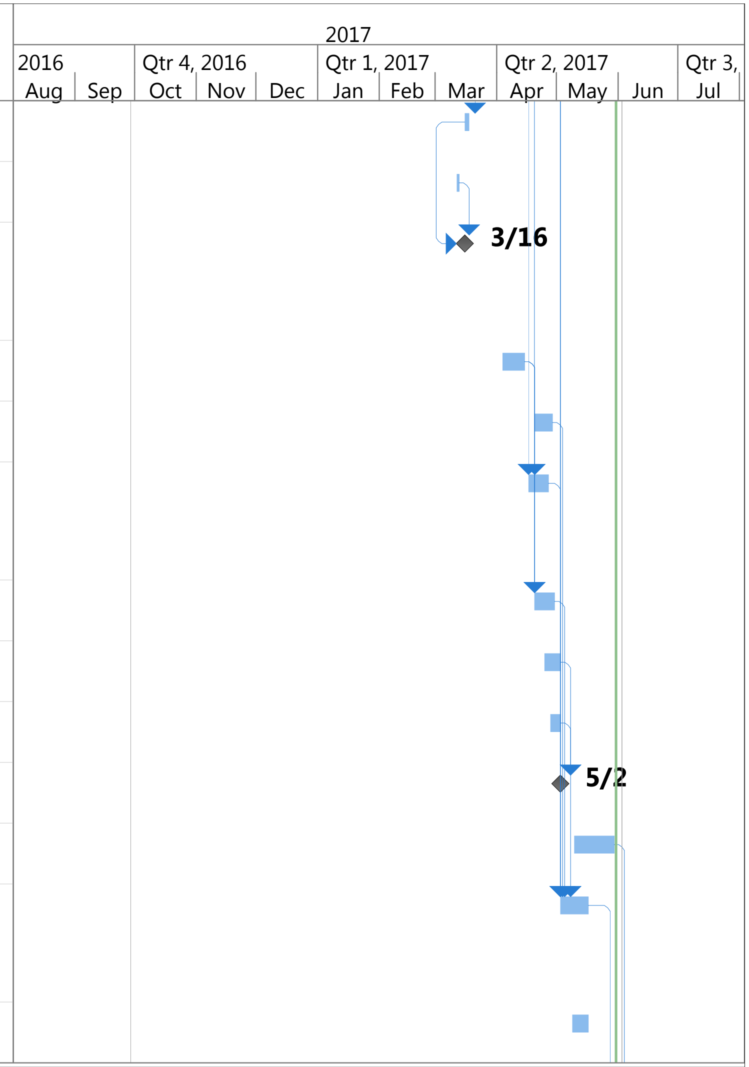
Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
81	Update Test Plan	Thu 3/16	Fri 3/17
82	Review Safety Hazards	Sun 3/12	Sun 3/12
83	**Manufacturing Status and Test Plan Presentation**	Thu 3/16	Thu 3/16
84	Write CNC Code for Part Manufacturing	Tue 4/4	Fri 4/14
85	Drill Mounting Holes in Ring Gear	Thu 4/20	Fri 4/28
86	CNC Housing Plates, Bolting Blocks, and Bearing Mounts	Mon 4/17	Wed 4/26
87	CNC Shafts for Keyway and Retaining Ring Groove	Thu 4/20	Sat 4/29
88	Cut Keyways and Drill Acrylic Top	Tue 4/25	Tue 5/2
89	Prepare Safety Demo	Fri 4/28	Tue 5/2
90	**Hardware and Safety Demo**	Tue 5/2	Tue 5/2
91	Write FDR Report	Wed 5/10	Mon 5/29
92	Assemble Prototype and Make Necessary Adjustments	Wed 5/3	Tue 5/16
93	Gather All Materials Necessary for Testing	Tue 5/9	Tue 5/16



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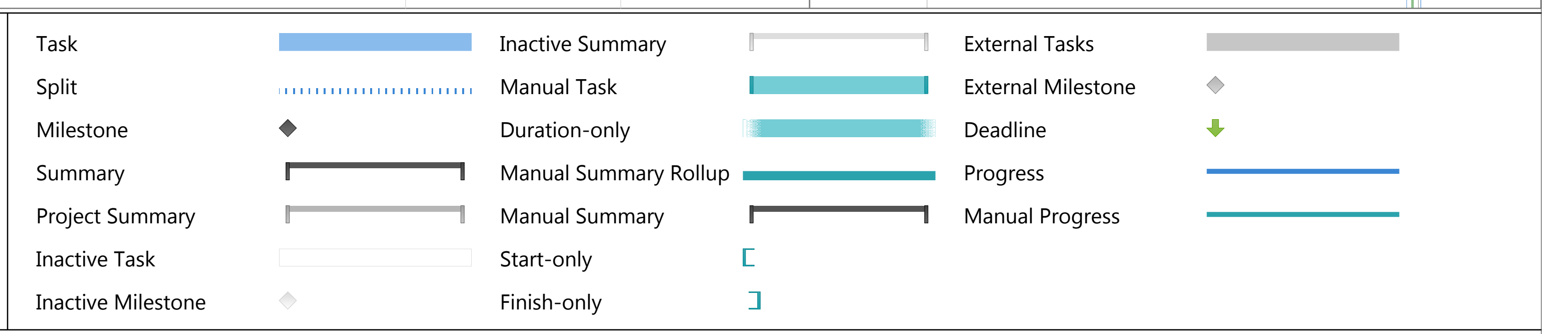
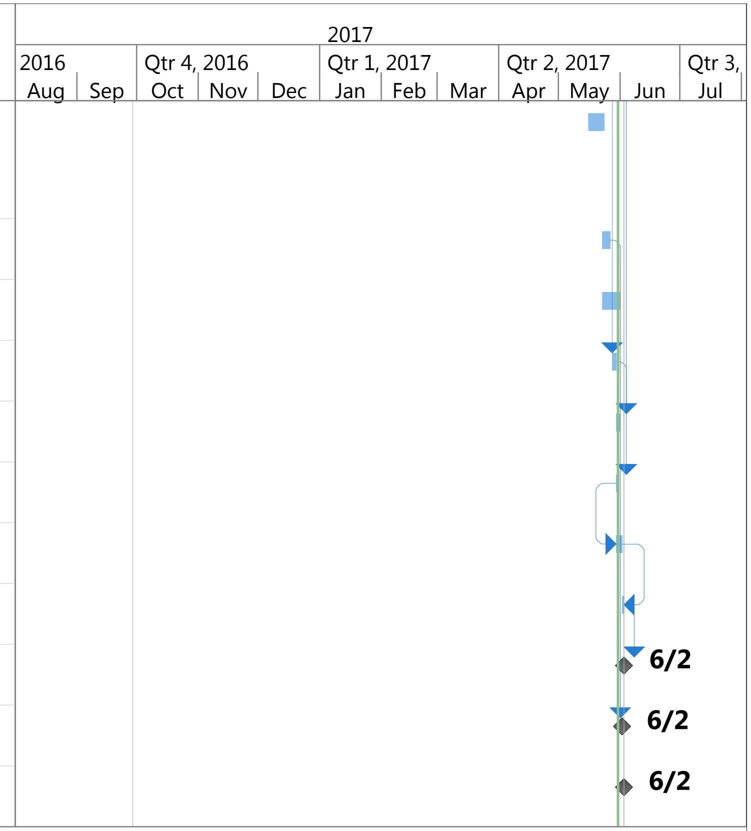


Team E-Drive Gantt Chart

ID	Task Name	Start	Finish
94	Conduct Static Speed Reduction and Torque Increase Tests	Tue 5/16	Tue 5/23
95	Create Presentation Board	Tue 5/23	Fri 5/26
96	Finalize Prototype by Adding RTV Sealant	Tue 5/23	Wed 5/31
97	Perform Leak Test on Housing	Sun 5/28	Mon 5/29
98	Fix Leaking Areas of Housing (If Necessary)	Tue 5/30	Wed 5/31
99	Peer Review FDR Report	Tue 5/30	Tue 5/30
100	Edit FDR Report	Tue 5/30	Thu 6/1
101	Send FDR Report to Sponsor	Fri 6/2	Fri 6/2
102	**FDR**	Fri 6/2	Fri 6/2
103	**Senior Project Expo**	Fri 6/2	Fri 6/2
104	**Deliver Design to Sponsor**	Fri 6/2	Fri 6/2



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Manual Summary Rollup
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External Milestone
Deadline
Progress
Manual Progress

Manufacturing Timeline

Part #	Part Name	Part Description	Operation Description	Operation Location	Operator	Planned Manufacturing Date	Date Completed
ST501-01	Input Shaft (1 3/4 X 12 in.)	Drivetrain input shaft	Cut to length	Hangar/Mustang 60	Charissa	4/22/17	4/22/17
			CNC gear and bearing shoulders	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/26/17	4/25/17
			CNC retaining ring slots	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/26/17	4/27/17
			Mill keyseat for pinion	Hangar/Mustang 60	Jimmy	4/26/17	4/29/17
			Mill keyseat for input shaft/motor coupler	Hangar/Mustang 60	Jimmy	5/10/17	5/13/17
ST501-04	Key Way Stock (12 in.)	Keyway to mate pinion and input shaft	Cut to proper facewidth of pinion	Hangar/Mustang 60	Jimmy	4/25/17	4/29/17
			Radius ends to match keyway	Hangar/Mustang 60	Jimmy	4/25/17	4/29/17
ST501-06/ ST502-07	Stock Aluminum Block	Input and intermediate shaft bearing mounts	Cut mounts to rough size on horizontal band saw	Hangar/Mustang 60	Kevin	4/17/17	4/11/17
			Face the mounts down to appropriate size on mill	Hangar/Mustang 60	Kevin	4/18/17	4/13/17
			CNC bearing slot	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/19/17	4/24/17
			CNC bolting holes	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/19/17	4/24/17
			Tap holes	IME Advanced Manfucturing	Charissa	4/19/17	4/24/17
ST502-01	Intermediate Shaft (2 X 12 in.)	Intermediate shaft	Cut to length	Hangar/Mustang 60	Charissa	4/22/17	4/22/17
			CNC gear and bearing shoulders	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/26/17	4/27/17
			CNC retaining ring slots	IME Advanced Manfucturing	IME Shop Tech: Wyatt	4/26/17	4/27/17
			Mill keyseats	Hangar/Mustang 60	Jimmy	4/26/17	4/29/17
ST502-05	Key Way Stock (12 in.)	Keyway to mate intermediate gears and shaft	Cut to proper facewidth of each gear	Hangar/Mustang 60	Jimmy	4/25/17	4/29/17
			Radius ends to match keyway	Hangar/Mustang 60	Jimmy	4/25/17	4/19/17
ST503-03	RH Cut, 9.5" Helical Ring Gear	Helical ring gear on differential	Laser cut a wooden gear prototype from plywood	Hangar/Mustang 60	Kevin	4/24/17	4/24/17
			Drill bolt pattern in wooden "ring gear" and test fit on differential	Hangar/Mustang 60	Kevin	4/25/17	4/26/17
			Drill bolt holes in actual ring gear to match Ford 8.8" differential carrier bolt pattern	Hangar/Mustang 60	Kevin	4/28/17	4/28/17
ST503-06	Stock Aluminum Block	Differential bearing mount	Cut mounts to rough size on horizontal band saw	Hangar/Mustang 60	Kevin	4/17/17	4/11/17
			Face the mounts down to appropriate size on mill	Hangar/Mustang 60	Jimmy	4/18/17	4/13/17
			CNC bearing slot	IME Advanced Manfucturing	IME Professor Trian Georgeou	4/19/17	4/21/17
			CNC bolting holes	IME Advanced Manfucturing	IME Professor Trian Georgeou	4/19/17	4/21/17
			Tap holes	IME Advanced Manfucturing	Charissa	4/19/17	4/21/17
ST504-01	Plates (0.625x12x24)	Left side plate for the housing	CNC output shaft hole	Hangar/Mustang 60	Shop Tech: Adam	4/24/17	4/25/17
			CNC bolt holes	Hangar/Mustang 60	Shop Tech: Adam	4/24/17	4/25/17
ST504-02	Plates (0.625x12x24)	Right side plate for the housing	CNC input shaft hole	Hangar/Mustang 60	Shop Tech: Adam	4/24/17	4/25/17
			CNC output shaft hole	Hangar/Mustang 60	Shop Tech: Adam	4/24/17	4/25/17
			CNC bolt holes	Hangar/Mustang 60	Shop Tech: Adam	4/24/17	4/25/17
ST504-03	Plates (0.625x12x24)	Top and bottom plates for the housing	Mark bolt hole locations	Hangar/Mustang 60	Jimmy	4/24/17	4/24/17
			Mill bolt holes	Hangar/Mustang 60	Jimmy	4/25/17	4/25/17
ST504-04	Plates (0.625x12x12)	Front end plate for the housing	Mark bolt hole locations	Hangar/Mustang 60	Jimmy	4/24/17	4/24/17
			Mill bolt holes	Hangar/Mustang 60	Jimmy	4/25/17	4/25/17
ST504-05	Plates (0.625x12x12)	Back end plate for the housing	Locate bolt holes and oil drain and fill plug holes	Hangar/Mustang 60	Jimmy	4/24/17	4/24/17
			Mill bolt holes and oil drain/fill plug holes	Hangar/Mustang 60	Jimmy	4/25/17	4/25/17
			Tap oil drain and fill plug holes	Hangar/Mustang 60	Charissa	4/26/17	4/25/17
ST504-06	Sheet	Clear acrylic top for housing display	Drill bolting holes	Hangar/Mustang 60	Charissa	4/24/17	5/9/17
ST504-07	Flat Stock, Al, 6061	Corner blocks and re-enforcement on housing	Cut blocks to rough size on horizontal band saw	Hangar/Mustang 60	Charissa	4/11/17	4/11/17
			Face the blocks down to appropriate size on mill	Hangar/Mustang 60	Jimmy	4/12/17	4/13/17
			CNC bolt holes	IME Advanced Manfucturing	IME Professor Trian Georgeou	4/26/17	4/18/17
			Tap holes	IME Advanced Manfucturing	Charissa	4/26/17	4/18/17
ST505-01	Coupler	To be used in the torque increase test	Turn down outside diamter on lathe	Hangar/Mustang 60	Kevin	5/9/17	5/9/17
			Drill hole sized to the outer diameter of the input shaft on a lathe	Hangar/Mustang 60	Kevin	5/9/17	5/12/17
			Cut internal keyway and make key	Hangar/Mustang 60	Kevin	5/15/17	5/20/17

Table of Contents

DESIGN HAZARD CHECKLIST

Team: E-Drive Advisor: Schuster

Y	N	
☒	☐	1. Will any part of the design create hazardous revolving, reciprocating, running, shearing, punching, pressing, squeezing, drawing, cutting, rolling, mixing or similar action, including pinch points and sheer points?
☒	☐	2. Can any part of the design undergo high accelerations/decelerations?
☒	☐	3. Will the system have any large moving masses or large forces?
☐	☒	4. Will the system produce a projectile?
☐	☒	5. Would it be possible for the system to fall under gravity creating injury?
☐	☒	6. Will a user be exposed to overhanging weights as part of the design?
☐	☒	7. Will the system have any sharp edges?
☐	☒	8. Will any part of the electrical systems not be grounded?
☐	☒	9. Will there be any large batteries or electrical voltage in the system above 40 V?
☐	☒	10. Will there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids?
☐	☒	11. Will there be any explosive or flammable liquids, gases, or dust fuel as part of the system?
☐	☒	12. Will the user of the design be required to exert any abnormal effort or physical posture during the use of the design?
☐	☒	13. Will there be any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?
☒	☐	14. Can the system generate high levels of noise?
☒	☐	15. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, cold, high temperatures, etc?
☒	☐	16. Is it possible for the system to be used in an unsafe manner?
☒	☐	17. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any "Y" responses, add (1) a complete description, (2) a list of corrective actions to be taken, and (3) date to be completed on the reverse side.

Description of Hazard	Planned Corrective Action	Planned Date	Actual Date
Rotating Shafts and Pinch Points Between Gears	Gears will be contained in a housing to eliminate possibility of entanglement or pinch points. Input and output shafts will exit housing, but will be located beneath the vehicle.	5/2/17	5/1/17
The shaft assemblies will undergo high accelerations and decelerations when coupled to motor.	Shafts and keyway connections will be designed to handle the loading conditions. In the event that the components break under loading, the housing will contain the fragments.	3/18/17	3/14/17
Large moving masses and large forces transmitted in drivetrain.	The gears, shafts, keyway connections, and bearings will be designed to handle loading conditions created from the motor's 400 lb-ft torque output.	3/18/17	3/14/17
The system may generate high levels of noise at the upper rpm range of the motor.	Helical gears will be used to reduce noise. The drivetrain will sit in an oil bath to lubricate the gears, which will also help in noise reduction.	3/18/17	3/14/17
The drivetrain may be subjected to extreme environmental conditions, including high temperatures.	The gear oil/lubricant will help to reduce heat created by friction of bearings and meshing gear teeth.	5/30/17	5/29/17
The system may be able to be used in an unsafe manner if motor is ran at redline for a prolonged period of time.	Note in user/repair manual to not operate the vehicle at top speed for a prolonged amount of time.	5/23/17	5/30/17
Debris and contaminants entering/leaving drivetrain.	The housing will be sealed and incorporate shaft seals to ensure no outside debris enters drivetrain. The seals and enclosed housing will also ensure no oil leakage.	5/30/17	5/28/17
Deterioration of oil after much use.	Note in user/repair manual to replace oil after a certain mileage. An oil drain and oil fill plug will be incorporated in the drivetrain design for ease of servicing.	5/23/17	5/30/17
Loosening bolts due to excessive vibrations.	Loctite will be applied to all nuts and bolts used in the assembly of the drivetrain.	5/30/17	5/28/17

Design Failure Mode and Effects Analysis

Product: ______________________________ Prepared by: ______________________________

Team: ______________________________ Date: ________________ (orig)

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Action Results: Actions Taken	Action Results: Severity	Action Results: Occurence	Action Results: Criticality
Transfer Power	Component Failure	Not Driveable	9	Bearing failure	2	18	Regulary perform maintanance and replace bearings	Kevin	Operator's Manual			
				Gear tooth shearing	3	27	If car and handing feels unstable check and replace gears	Kevin	Operator's Manual			
				Shaft cracks	2	18	Designed with FS of 1.5 Replace shaft	Kevin	Operator's Manual			
				Differential load failure	3	27	Spider gear replacement if load is too great	Kevin	Operator's Manual			
		Requires Repair	7	Bearing failure	3	21	Designed bearings to handle a load that exceeded the actual load it will experience. Replace bearings	Jimmy	Operator's Manual			
				Gear tooth shearing	4	28	Replace gear	Jimmy	Operator's Manual			
				Differential Failure	3	21	Spider gear replacement if load is too great	Jimmy	Operator's Manual			
				Shaft cracks	4	28	Replace shaft	Jimmy	Operator's Manual			
	Gear/Shaft Interface	Not Driveable	9	Material warping	2	18	Replace component	Charissa	Operator's Manual			
				Keyways/Splines shear	3	27	Re-cut keys and spline in shaft with thicker or stronger material if needed	Charissa	Operator's Manual			
		Will require repair	6	Material warping	5	30	Replace component	Charissa	Operator's Manual			
				Keyways/Splines shear	6	36	Re-cut keys and spline in shaft	Charissa	Operator's Manual			
	Shaft Deflection	Not Driveable	9	Unbalenced shaft assembly	2	18	Remanufacture shafts to be symmetric. Choose a stronger material or increase FS so shafts will no longer deflect	Jimmy	Operator's Manual			
		Will require repair	5	Unbalenced shaft assembly	3	15	Remanufacture shafts to be symmetric	Jimmy	Operator's Manual			

Design Failure Mode and Effects Analysis

Product: ____________________________ Prepared by: ____________________________

Team: ____________________________ Date: ________________ (orig)

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Action Results: Actions Taken	Action Results: Severity	Action Results: Occurence	Action Results: Criticality
	Housing deformation	Not Driveable	9	Impact loading	2	18	Remanufacture housing plates and add reenforcement to the walls or use stronger material. Outsource to cast a housing that is more suited for a typical drivetrain	Charissa	Operator's Manual			
		Will require repair	8	Impact loading	3	24	Remanufacture housing plates and add reenforcement to the walls	Charissa	Operator's Manual			
	Tooth Alignment/Contact	Not Driveable	9	Tolerancing	1	9	Designed with AGMA guidelines to ensure the meshing of gears. Realign shafts to reduce or eliminate seizing or gears/ Re-manufacture housing plates	Kevin	Operator's Manual			
				Manufacturing	1	9	Re-Manufacture components to ensure gears mesh/ Realign parts and check for proper spacing between shafts	Kevin	Operator's Manual			
		Will require repair	5	Tolerancing	2	10	Realign shafts to reduce or eliminate seizing or gears	Jimmy (5/8/17)	Enlarged the bore in a bearing block so the holes on either bearing block were concentric			
				Manufacturing	2	10	Re-Manufacture components	Charissa	Operator's Manual			
		Oil leakage on floor	2	Debris	4	8	Drain oil and clean entire drivetrain	Charissa	Operator's Manual			
				Seal/Gasket Wear	3	6	Replace brittle seals or gaskets	Charissa	Operator's Manual			

Design Failure Mode and Effects Analysis

Product: ______________________________ Prepared by: ______________________________

Team: ______________________________ Date: ________________ (orig)

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Action Results: Actions Taken	Action Results: Severity	Action Results: Occurence	Action Results: Criticality
Isolation of System	Leakage			Effects of weather	7	14	Replace brittle seals or gaskets	Charissa	Operator's Manual			
		Requires maintanence	3	Debris	6	18	Drain oil and clean debris from drivetrain	Charissa	Operator's Manual			
				Seal/Gasket Wear	5	15	Replace brittle seals or gaskets	Charissa	Operator's Manual			
				Effects of weather	8	24	Replace brittle seals or gaskets	Charissa	Operator's Manual			
		Reduced wear protection	8	Complete loss of oil	1	8	Remanufacter housing plates and check for cracks or gap in loctite sealant	Charissa	Operator's Manual			
		Oil leakage on floor	2	Debris	5	10	Drain oil and clean debris from drivetrain	Charissa	Operator's Manual			
				Wear	5	10	Replace brittle seals or gaskets	Charissa	Operator's Manual			
				Bolt Connection failure	4	8	Re-Manufacture components, re-thread holes and insert new bolts. Be careful of torquing bolt down too hard	Charissa	Operator's Manual			
				Effects of weather	7	14	Replace brittle seals or gaskets	Charissa	Operator's Manual			
		Requires maintanence	5	Debris	6	30	Perform regular maintanance. Drain oil and clean entire drivetrain	Charissa	Operator's Manual			
				Wear	5	25	Replace brittle seals or gaskets	Charissa	Operator's Manual			
				Effects of weather	8	40	Replace brittle seals or gaskets	Charissa	Operator's Manual			
				Overload on bolt	2	8	May replace corner blocks if shearing occurs and design with a higher bolt grade	Kevin	Operator's Manual			

Product: ____________________________

Design Failure Mode and Effects Analysis

Prepared by: ____________________________

Team: ____________________________

Date: _______________ (orig)

Item / Function	Potential Failure Mode	Potential Effect(s) of Failure	Severity	Potential Cause(s) / Mechanism(s) of Failure	Occurence	Criticality	Recommended Action(s)	Responsibility & Target Completion Date	Action Results: Actions Taken	Action Results: Severity	Action Results: Occurence	Action Results: Criticality
	Bolt connection failure	Noise	4	Manufacturing	2	8	Re-manufacture components as needed with stronger material	Kevin	Operator's Manual			
				Tolerancing	3	12	Re-Manufacture components as needed with better tooling for more accurate parts	Kevin	Operator's Manual			
		Alignment issues	5	Overload on bolt	1	5	Reduce load on individual bolt by increasing the number of bolts used or choosing a higher grade/thicker bolt	Jimmy	Operator's Manual			
				Manufacturing	2	10	Re-Manufacture components as needed	Jimmy	Operator's Manual			
				Tolerancing	2	10	Re-Manufacture components as needed to be within specified tolerance	Jimmy	Operator's Manual			

Table of Contents

ST5 Vehicle Performance Calculations

Road Load

$$Drag = (C_D A)\frac{1}{2}\rho v^2$$

$$Lift = (C_L A)\frac{1}{2}\rho v^2$$

$$Normal\ Load = W - Lift$$

$$R_{RR} = \mu_{RR} N$$

$$Road\ Load = Drag + R_{RR}$$

Maximum Traction

$$F_{T,\ max,\ RWD} = \frac{\frac{\mu_T}{L}[(I_1 - h\mu_{RR})(W - L_a) - M_a]}{\left(1 - \mu_T \frac{h}{L}\right)}$$

Maximum Tractive Effort

$$F_{T,\ max} = \frac{\eta_T \eta_A P_{E,\ max}}{V}$$

Tractive Effort

$$F_T = \frac{\eta_T \eta_A T_E * Drivetrain\ Ratio}{r_{tire}}$$

Effective Mass

$$m_{eff} = m\left[1 + \frac{(I_W + I_D(\xi_A)^2 + I_E(\xi_A \xi_T)^2)}{mr^2}\right]$$

Time To Speed

$$F_T = m\frac{dV}{dt}$$

$$\frac{dt}{dV} = \frac{m_{eff}}{F_T}$$

$$\Delta t_i = \frac{\left[\left(\frac{dt}{dV}\right)_{i-1} + \left(\frac{dt}{dV}\right)_i\right] * [(V)_i - (V)_{i-1}]}{2}$$

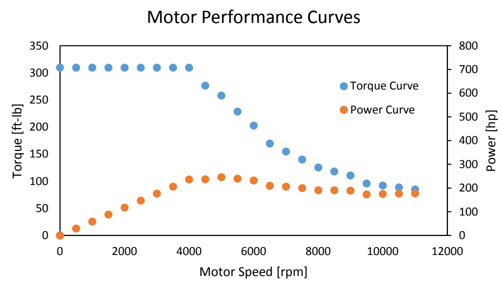
$$Time\ To\ Speed = \sum_{i=V_i}^{V_f} \Delta t_i$$

Vehicle Parameters

Description	Variable Name	Value	Units	Value	Units
Vehicle Weight	W	4500	lbf	139.7516	slug
Front Distribution	F	0.45			
Rear Distribution	R	0.55			
Tire Diameter	D_t	27.3	in	2.275	ft
Tire Radius	r_t	13.65	in	1.1375	ft
Height to CG	h_cg	18	in	1.5	ft
Wheelbase	L	110	in	9.166667	ft
Front Distance to CG	l_1	60.5	in	5.041667	ft
Rear Distance to CG	l_2	49.5	in	4.125	ft
	θ_s	0	deg		
Drive Assembly Eff.	η_A	0.95			
Transmission Eff.	η_T	0.95			
	I_motor	0.086	kg-m^2	0.063379	ft-lb-sec^2
	I_axle,finaldriv	0.03	ft-lb-sec^2		
	I_tire/wheel	1.398	ft-lb-sec^2		
	μ_RR	0.01			
Coefficient Traction Road	μ_road	0.7			
Drag Coefficient	C_D	0.3			
Lift Coefficient	C_L	0.2	upward		
	M_a				
	A	26	ft^2		
Air Density	ρ	0.002377	slug/ft^3		
	RWD				
	ft/s for 1 mph	1.466667			
	ft-lbf/s for 1 h	550			
	lbf for 1 kg	2.20462			
	ft^2 for 1 m^2	10.7639			
	gravity	32.2	ft/s^2		

Adapted From Model S P85D Power Curves

Engine Speed		Torque	Torque	Power	Power Available
rpm	rad/s	lb-ft	Nm	hp	hp
0	0	0	0	0	0.0
1	0	310	420	0	0.1
500	52	310	420	29	26.6
1000	105	310	420	59	53.2
1500	157	310	420	88	79.8
2000	209	310	420	118	106.5
2500	262	310	420	147	133.1
3000	314	310	420	177	159.7
3500	367	310	420	206	186.3
4000	419	310	420	236	212.9
4500	471	277	375	237	213.9
5000	524	258	350	246	221.8
5500	576	229	310	239	216.1
6000	628	203	275	232	209.1
6500	681	170	230	210	189.5
7000	733	155	210	206	186.3
7500	785	140	190	200	180.6
8000	838	125	170	191	172.4
8500	890	118	160	191	172.4
9000	942	111	150	190	171.1
9500	995	96	130	173	156.5
10000	1047	92	125	176	158.4
10500	1100	89	120	177	159.7
11000	1152	85	115	178	160.3



ST5 Traction Calculations

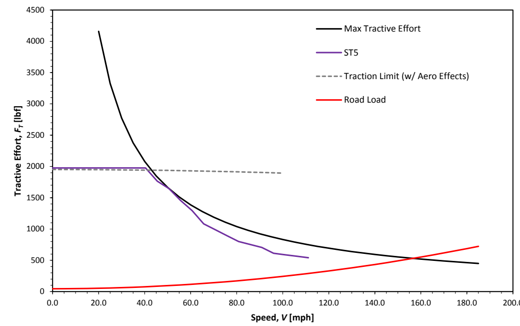
Speed mph	Drag lbf	Lift lbf	Normal Load lbf	R_RR lbf	Road Load lbf	Road Load hp	Max Traction lbf	Max Tractive Effort lbf
0.0	0.0	0.0	4500.0	45.0	45.0	0.0	1950.8	#DIV/0!
5.0	0.5	0.3	4499.7	45.0	45.5	0.6	1950.7	16634.90948
10.0	2.0	1.3	4498.7	45.0	47.0	1.3	1950.2	8317.454739
15.0	4.5	3.0	4497.0	45.0	49.5	2.0	1949.5	5544.969826
20.0	8.0	5.3	4494.7	44.9	52.9	2.8	1948.5	4158.727369
25.0	12.5	8.3	4491.7	44.9	57.4	3.8	1947.2	3326.981896
30.0	17.9	12.0	4488.0	44.9	62.8	5.0	1945.6	2772.484913
35.0	24.4	16.3	4483.7	44.8	69.3	6.5	1943.7	2376.41564
40.0	31.9	21.3	4478.7	44.8	76.7	8.2	1941.6	2079.363685
45.0	40.4	26.9	4473.1	44.7	85.1	10.2	1939.1	1848.323275
50.0	49.9	33.2	4466.8	44.7	94.5	12.6	1936.4	1663.490948
55.0	60.3	40.2	4459.8	44.6	104.9	15.4	1933.4	1512.264498
60.0	71.8	47.9	4452.1	44.5	116.3	18.6	1930.1	1386.242456
65.0	84.3	56.2	4443.8	44.4	128.7	22.3	1926.5	1279.608421
70.0	97.7	65.1	4434.9	44.3	142.1	26.5	1922.6	1188.20782
75.0	112.2	74.8	4425.2	44.3	156.4	31.3	1918.4	1108.993965
80.0	127.6	85.1	4414.9	44.1	171.8	36.6	1913.9	1039.681842
85.0	144.1	96.1	4403.9	44.0	188.1	42.6	1909.2	978.5240869
90.0	161.5	107.7	4392.3	43.9	205.4	49.3	1904.1	924.1616377
95.0	180.0	120.0	4380.0	43.8	223.8	56.7	1898.8	875.5215515
100.0	199.4	132.9	4367.1	43.7	243.1	64.8	1893.2	831.7454739
105.0	219.9	146.6	4353.4	43.5	263.4	73.7	1887.3	792.1385466
110.0	241.3	160.9	4339.1	43.4	284.7	83.5	1881.1	756.132249
115.0	263.7	175.8	4324.2	43.2	307.0	94.1	1874.6	723.2569338
120.0	287.2	191.4	4308.6	43.1	330.2	105.7	1867.8	693.1212282
125.0	311.6	207.7	4292.3	42.9	354.5	118.2	1860.8	665.3963791
130.0	337.0	224.7	4275.3	42.8	379.8	131.7	1853.4	639.8042107
135.0	363.4	242.3	4257.7	42.6	406.0	146.2	1845.8	616.1077584
140.0	390.9	260.6	4239.4	42.4	433.2	161.7	1837.8	594.1039099
145.0	419.3	279.5	4220.5	42.2	461.5	178.4	1829.6	573.6175682
150.0	448.7	299.1	4200.9	42.0	490.7	196.3	1821.1	554.4969826
155.0	479.1	319.4	4180.6	41.8	520.9	215.3	1812.3	536.6099832
160.0	510.5	340.3	4159.7	41.6	552.1	235.6	1803.3	519.8409212
165.0	542.9	361.9	4138.1	41.4	584.3	257.1	1793.9	504.088166
170.0	576.3	384.2	4115.8	41.2	617.5	279.9	1784.2	489.2620435
175.0	610.7	407.1	4092.9	40.9	651.6	304.1	1774.3	475.2831279
180.0	646.1	430.7	4069.3	40.7	686.8	329.7	1764.1	462.0808188
185.0	682.5	455.0	4045.0	40.5	722.9	356.7	1753.6	449.5921481

GEAR RATIO CALCS

Drivetrain Ratio
8.04

F_T lbf	Vehicle Speed mph
0	0
1976	0
1976	5
1976	10
1976	15
1976	20
1976	25
1976	30
1976	35
1976	40
1764	45
1647	51
1459	56
1294	61
1082	66
988	71
894	76
800	81
753	86
706	91
612	96
588	101
565	106
541	111

ST5 Top Speed



TIME TO SPEED

Effective Mass

m_eff lbf
146.63

dt/dv

1st
#DIV/0!
0.074
0.074
0.074
0.074
0.074
0.074
0.074
0.074
0.074
0.083
0.089
0.101
0.113
0.135
0.148
0.164
0.183
0.195
0.208
0.240
0.249
0.260
0.271

delta T

Vehicle Speed mph	Δt s
0.0	#DIV/0!
0.0	#DIV/0!
5.1	0.549
10.1	0.550
15.2	0.550
20.2	0.550
25.3	0.550
30.3	0.550
35.4	0.550
40.4	0.550
45.5	0.583
50.5	0.638
55.6	0.702
60.6	0.792
65.7	0.922
70.7	1.052
75.8	1.157
80.8	1.287
85.9	1.400
90.9	1.491
96.0	1.657
101.0	1.811
106.1	1.885
111.1	1.966
Sum 0-60	7.111 seconds

Helical Gear Analysis and Equations

HELICAL GEAR ANALYSIS

* Geometries

$P \equiv$ diametral pitch, teeth/in
$N \equiv$ Number of teeth
$d \equiv$ pitch diameter [in]
$p \equiv$ circular pitch [in]

Addendum $a = \frac{1}{P}$

Dedendum $b = \frac{1.25}{P}$

clearance $c = b - a$

* Contact Ratio

$$m_c = \frac{\left\{\left[(r_p + a)^2 - r_{bp}^2\right]^{1/2} + \left[(r_G + a)^2 - r_{bG}^2\right]^{1/2} - (r_p + r_G)\sin\phi\right\}}{p\cos\phi}$$

must be > 1.2

Interference check Eq (13-11) pg. 678 $m_G = \frac{N_G}{N_P} = m$ & > 1

* Smallest number of teeth on the pinion without interference

$$N_p = \frac{2k}{(1+2m)\sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)\sin^2\phi}\right)$$

Tooth proportions (ALL TYPES) pg 684

13-10 Parallel Helical Gears, reference Fig 13-22

ψ = helix angle

ac = transverse circular pitch p_t (usually called the circular pitch)

ae = normal circular pitch p_n

Eq. (13-16) $p_n = p_t \cos\psi$

ad = axial pitch p_x

Eq. (13-17) $p_x = \frac{p_t}{\tan\psi}$

$p_n P_n = \pi \Rightarrow$ normal diametral pitch

Eq. (13-18) $P_n = \frac{P_t}{\cos\psi}$

ϕ_n = pressure normal angle

ϕ_t = pressure angle in direction of rotation

Eq. (13-19) $\cos\psi = \frac{\tan\phi_n}{\tan\phi_t}$

virtual number of teeth: (can be shown by analytical geometry)

Eq (13-20) $$N' = \frac{N}{\cos^3 \psi}$$

Pg 687 → face-contact ratio → results in quieter gears

GEAR TRAINS

$$\text{train value} \equiv e = \frac{\text{product of driving tooth numbers}}{\text{product of driven tooth numbers}}$$ Eq. (13-30)

* pitch diameters can also be used

Pg 691 * A two-stage compound gear train can obtain e up to 100:1

* To minimize package size, keep portions as evenly divided between the stages as possible

POWER TRANSMITTED THROUGH A ROTATING GEAR

Eq (13-33) $$H = T\omega = \left(\frac{W_t d}{2}\right)\omega$$

V = pitch-line velocity, ft/min
d = gear diameter, in
n = gear speed, rev/min

Eq. (13-34) $$V = \frac{\pi d n}{12}$$

W_t = transmitted load, lbf
H = power, hp
V = pitch-line velocity, ft/min

Eq. (13-35) $$W_t = 33\,000 \frac{H}{V}$$

FORCE ANALYSIS - HELICAL GEARING

Eq. (13-39)
$$W_r = W \sin \phi_n$$
$$W_t = W \cos \phi_n \cos \psi$$
$$W_a = W \cos \phi_n \sin \psi$$

W = total force
W_r = radial component
W_t = tangential component, also called transmitted load
W_a = axial component, also called the thrust load

* Reference Figure 13-37
tooth forces acting on a RH (right hand) helical gear

Table 14-1: Symbols, Their Names, and Locations pg 727

* Examination of run-in teeth will show that the heaviest loads occur near the middle of the tooth. pg 730
Therefore the maximum stress probably occurs while a single pair of teeth is carrying the full load, at a point where another pair of teeth is just on the verge of coming into contact

DYNAMIC EFFECTS

* 14^{th} century, & current AGMA standards but b/c of date, it is possible the tests were conducted on teeth having a cycloidal profile instead of an involute profile pg 731

fatigue stress-concentration factor K_f

Eq (14-9) $$K_f = H + \left(\frac{t}{r}\right)^L \left(\frac{t}{l}\right)^M$$

$$H = 0.34 - 0.4583662\phi$$
$$L = 0.316 - 0.4583662\phi$$
$$M = 0.290 + 0.4583662\phi$$
$$r = \frac{(b - r_f)^2}{(d/2) + b - r_f}$$

*ref. Fig 14-1

AGMA elastic coefficient C_p

$$C_p = \left[\frac{1}{\pi\left(\frac{1-\nu_P^2}{E_P} + \frac{1-\nu_G^2}{E_G}\right)}\right]^{1/2}$$ Eq. (14-13)

surface compressive stress (Hertzian stress)

Eq (14-14) $$\sigma_c = -C_p\left[\frac{K_v W^t}{F\cos\phi}\left(\frac{1}{r_1} + \frac{1}{r_2}\right)\right]^{1/2}$$

pg 737 for helical assumptions

AGMA Stress Equations:

bending stress & pitting resistance (contact stress)

⇒ stress numbers in AGMA terminology

$W^t \equiv$ transmitted tangential load
$K_o \equiv$ overload factor
$K_v \equiv$ dynamic factor
$K_s \equiv$ size factor
$P_d \equiv$ transverse diametral pitch
$F \equiv$ Face width
$K_m \equiv$ load distribution factor

Items the designer should consider pg 738

K_f = root fillet stress-concentration

$Y \equiv$ Geometry Factor

AGMA STRENGTH EQUATIONS

strength ⇒ allowable stress numbers

gear strength ⇒ allowable stress numbers pg 739

gear bending strength (S_t) ⇒ Figs 14-2, 14-3, 14-4
Tables 14-3, 14-4

Bending stress:

Allowable

$S_t \equiv$ allowable bending stress $\frac{lb_f}{in^2}$

$Y_N \equiv$ stress-cycle factor for bending stress

$K_T \equiv$ temperature factors

$S_F \equiv$ AGMA factor of safety, stress ratio

Contact Stress:

Allowable

$S_c \equiv$ allowable contact stress $\frac{lb_f}{in^2}$

$Z_N \equiv$ stress-cycle factor

$C_H \equiv$ hardness ratio factors for pitting resistance

$K_T \equiv$ temperature factors

$K_R \equiv$ reliability factors

$S_H \equiv$ AGMA factor of safety, stress ratio

allowable contact stress (S_c) ⇒ Figs 14-5 & Tables 14-5, 14-6, 14-7

AGMA allowable stress numbers are for:
- Unidirectional loading
- 10 million stress cycles
- 99 percent reliability

, 742

or two-way (reversed) loading

FAILURE LOCUS

Goodman ⇒ $k_e = 1.33$

Gerber ⇒ $k_e = 1.66$

GEOMETRY FACTORS I & J

face-contact ratio $\equiv m_F \equiv \frac{F}{p_x}$

$F \equiv$ Face width

$p_x \equiv$ Axial pitch

LCR helical gears can have $m_F \leq 1$

conventional helical gears: $m_F > 1$

Bending-Strength Geometry Factor : AGMA perspective

$J = \dfrac{Y}{K_f m_N}$

Eq (14-20)

$Y \equiv$ obtained from calculations within AGMA 908-B89, often based on highest point of single-tooth contact

$K_f \equiv$ stress-correction factor, formula deduced from a photoelastic investigation of stress concentration in gear teeth over 50 years

load-sharing ratio : $m_N = \dfrac{P_N}{0.95 Z}$ for helical gears w/ $m_F > 2.0$

Eq (14-21) * conservative approximation pg 745

$P_N \equiv$ normal base pitch

$Z \equiv$ length of the line of action in the transverse plane (dist. L_{ab} in Fig. 13-15, p. 676)

Fig. 14-6 for 20° pressure angle & full-depth teeth

Fig. 14-7 " 20° normal pressure angle & $m_F = 2$ or greater

Surface-Strength Geometry Factor : AGMA perspective

$I \equiv$ also called pitting-resistance geometry factor by AGMA

* $\phi \Rightarrow \phi_t \rightarrow$ transverse

$$\text{speed ratio} \equiv m_G = \frac{N_G}{N_P} = \frac{d_G}{d_P} \qquad \text{Eq (14-22)}$$

Adding load-sharing ratio m_N

* valid for both spur and helical gears

$$I = \frac{\cos\phi_t \sin\phi_t}{2 m_N} \frac{m_G}{m_G + 1} \qquad \text{Eq. (14-23) EXTERNAL GEARS}$$

* solving 14-21 for m_N $\Rightarrow$ $P_N = p_n \cos\phi_n$ Eq (14-24)

(14-21) $p_n \equiv$ normal circular pitch

$$* \quad Z = \left[(r_P + a)^2 - r_{bP}^2\right]^{1/2} + \left[(r_G + a)^2 - r_{bG}^2\right]^{1/2} - (r_P + r_G)\sin\phi_t$$

r_P & $r_G \equiv$ pitch radii r_{bP} & $r_{bG} \equiv$ base-circle radii

com Eq (13-6) $r_b = r\cos\phi_t$ Eq (14-26)

THE ELASTIC COEFFICIENT

$C_p \equiv$ Eq (14-13) or Table 14-8 pg. 749

DYNAMIC FACTOR K_v

* Transmission error : the departure from uniform angular velocity of the gear pair

$Q_v \equiv$ Quality numbers   3 → 7 : Commercial
8 → 12 : Precision

$$K_v = \left(\frac{A+\sqrt{V}}{A}\right)^B \quad [ft/min] \qquad A = 50 + 56(1-B), \quad B = 0.25(12 - Q_v)^{2/3}$$

Eq (14-27)   Eq (14-28)

graph Fig. 14-9 pg 750

$$(V_t)_{max} = [A + (Q_v - 3)]^2 \quad ft/min$$

OVERLOAD FACTOR K_o

Figs. 14-17 & 14-18

SURFACE CONDITION FACTOR C_f

* only for the pitting resistance equation (14-16)

* standard surface conditions not yet established ⇒ set $C_f = 1$ ☆

SIZE FACTOR K_s

pg. 751

$$K_s = 1.192\left(\frac{F\sqrt{Y}}{P}\right)^{0.0535}$$

* Lewis's geometry incorporated into the Marin size factor

LOAD-DISTRIBUTION FACTOR K_m

for $F/d_p \leq 2$

face load distribution factor, C_{mf}, ⇒ $K_m = C_{mf} = 1 + C_{mc}(C_{pf} C_{pm} + C_{ma} C_e)$

$C_{mc} =$ 1 uncrowned, 0.8 crowned   Eq. (14-30)

$C_{pf} = \dfrac{F}{10 d_p} - 0.0375 + 0.0125F \quad 1 < F \leq 17 in$ Eq. (14-32)

$C_{pm} = 1$ for straddle-mounted pinion w/ $S_1/S < 0.175$

$C_{ma} = A + BF + CF^2$ Eq (14-34) & use Table 14-9

$C_e =$ 0.8, 1   Eq. (14-35)

HARDNESS-RATIO FACTOR C_H : only for the gear

$C_H = 1$ for the pinion

$$C_H = 1.0 + A'(m_G - 1.0) \qquad \text{Eq. (14-36)}$$

$$A' = 8.98(10^{-3})\left(\frac{H_{BP}}{H_{BG}}\right) - 8.29(10^{-3}) \qquad 1.2 \le \frac{H_{BP}}{H_{BG}} \le 1.7$$

* see Fig. 14-12 for a graph of Eq. (14-36)

$$\frac{H_{BP}}{H_{BG}} < 1.2 \qquad A' = 0$$

$$\frac{H_{BP}}{H_{BG}} > 1.7 \qquad A' = 0.00698$$

work hardening

$$C_H = 1 + B'(450 - H_{BG}) \qquad \text{Eq. (14-37)} \qquad \text{Fig. 14-12, 14-13}$$

surface-hardened pinions ⇒ run w/ → through-hardened gears

⇒ deals w/ surface finish f_P

w/ $B' = 0.000\,75 \exp[-0.0112 f_P]$

w/ f_P RMS roughness R_a in μ-in.

STRESS-CYCLE FACTORS Y_N & Z_N

* Figures based on 10^7 load cycles applied pg. 754

⇒ used to modify life other than 10^7 cycles

RELIABILITY FACTOR K_R

* linear interpolation is too crude * log transformation to each quanity produces a linear string

(14-38) $K_R = 0.50 - 0.109 \ln(1-R) \qquad 0.99 \le R \le 0.9999$

TEMPERATURE FACTOR K_T

oil or gear-blank temperatures up to 250°F (120°C), use $K_T = Y_\theta = 1.0$

If ↑ Temp the factor is greater than unity ∴ σ_{allow} ↓ ⇒ S.F. ↑

Shaft Analysis Sample Calculations

Governing Equation for Distortion Energy Theory with Goodman Criteria

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

Endurance Limit, S_e

$$S_e = k_a k_b k_c k_d k_e k_f S'_e$$

where
- k_a = surface condition modification factor
- k_b = size modification factor
- k_c = load modification factor
- k_d = temperature modification factor
- k_e = reliability factor[13]
- k_f = miscellaneous-effects modification factor
- S'_e = rotary-beam test specimen endurance limit
- S_e = endurance limit at the critical location of a machine part in the geometry and condition of use

Surface Factor, k_a

$$k_a = a S_{ut}^b$$

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

From C.J. Noll and C. Lipson, "Allowable Working Stresses," *Society for Experimental Stress Analysis*, vol. 3, no. 2, 1946 p. 29. Reproduced by O.J. Horger (ed.) *Metals Engineering Design ASME Handbook*, McGraw-Hill, New York. Copyright © 1953 by The McGraw-Hill Companies, Inc. Reprinted by permission.

INTERMEDIATE SHAFT

$\psi = 30°$

$\phi_t = 22.8$

$\phi_n = 20°$

FOR 6 IN GEAR

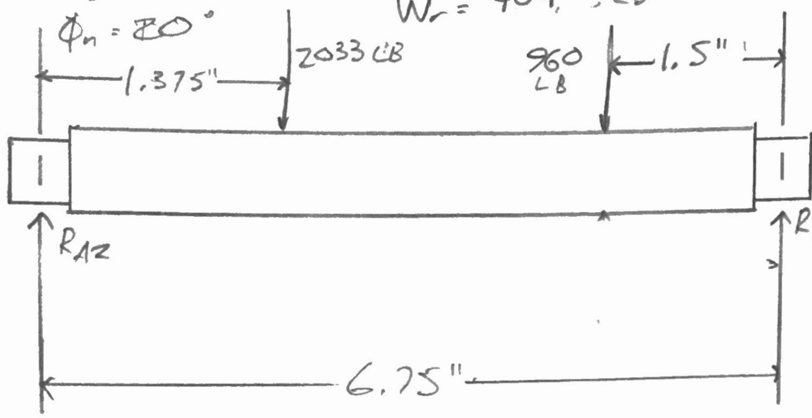
$W_t = 960$ LB

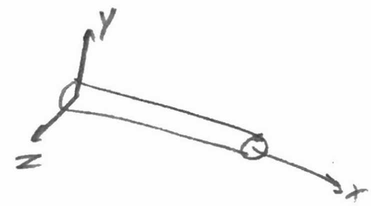
$W_r = 404$ LB

FOR 2.83 IN GEAR

$W_t = 2033$ LB

$W_r = 895$ LB



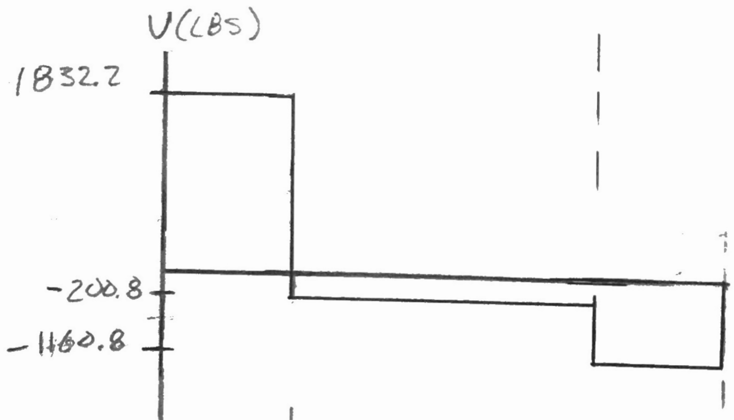


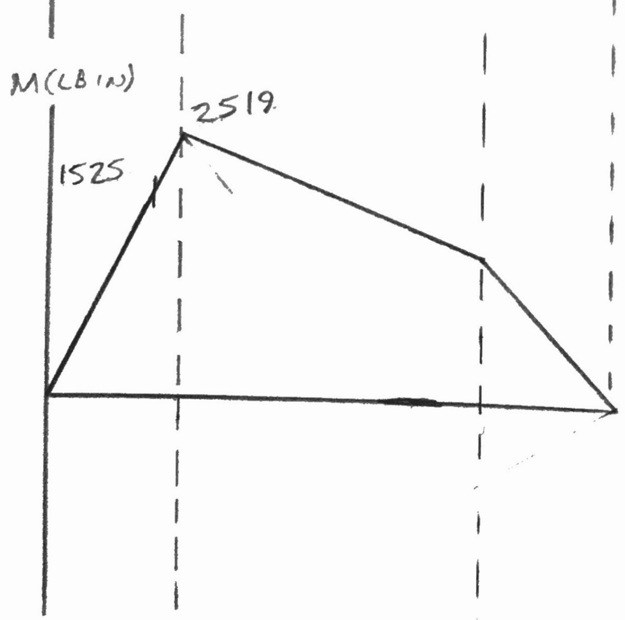
$\Sigma M_A = 0$

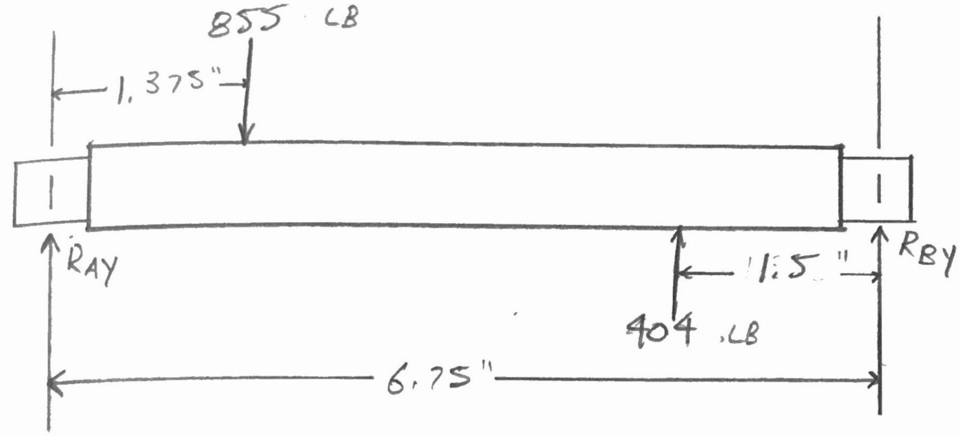
$-2033\,LB(1.375\,IN) - 960\,LB(5.25\,IN) + R_{Bz}(6.75\,IN) = 0$

$R_{Bz} = -1160.8$ LB

$R_{Az} = 1832.25$ LB







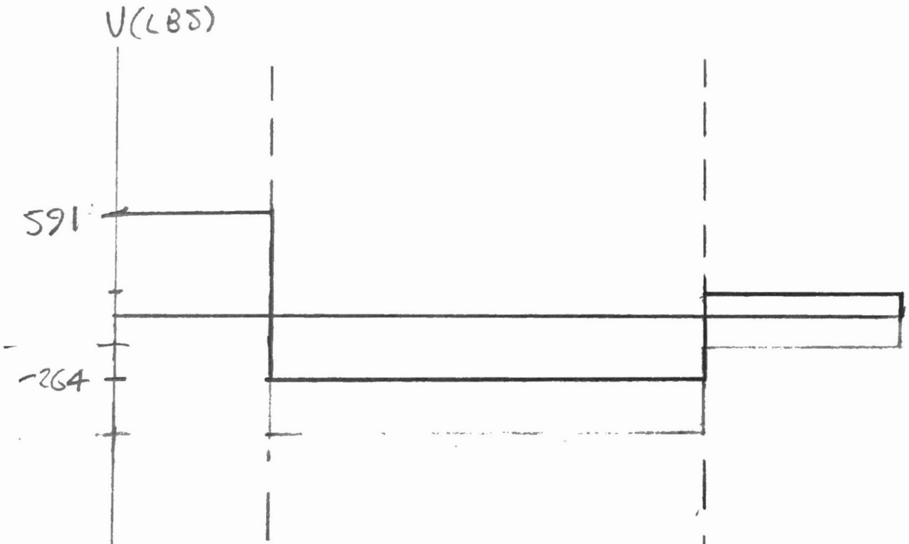
$R_{AY} - 855 + 404 + R_{BY} = 0$

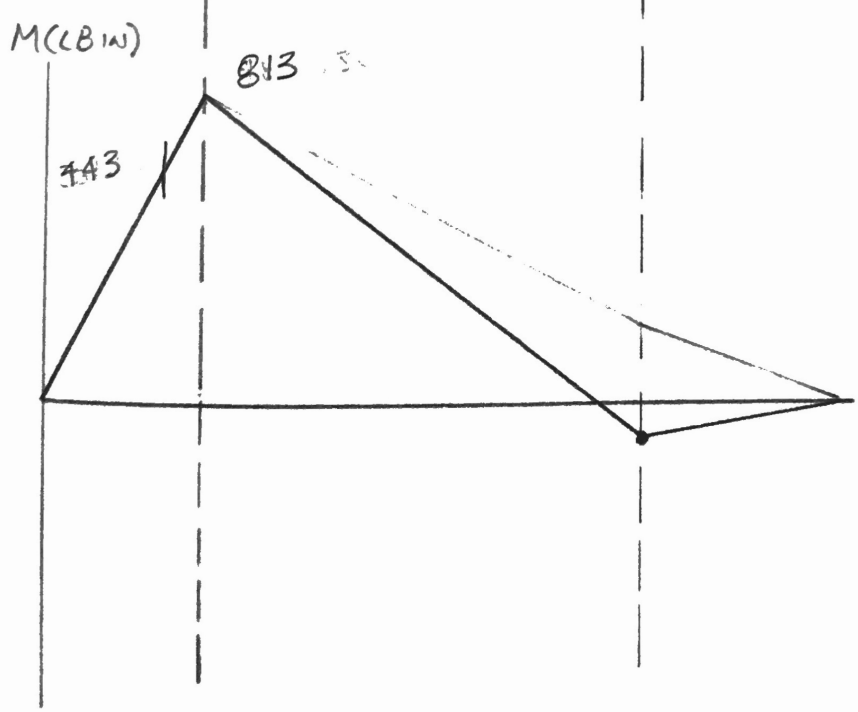
$\underline{\Sigma M_A = 0}$

$-855\ LB(1.375\ IN) + 404\ LB(5.25\ IN) + R_{BY}(6.75\ IN) = 0$

$R_{BY} = -140\ LB$

$R_{AY} = 591\ LB$





ALT. MOMENT, M_a

KEYWAY

$$M_a = \sqrt{2863.90^2 + 480.8^2}$$

$$= 3002.56 \text{ LB IN}$$

RING GROOVE

$$M_a = \sqrt{1616.67^2 + 261.98^2}$$

$$= 1637.76 \text{ LB IN}$$

DE Goodman	
n	1.5
K_f	2.0488
K_{fs}	2.84
M_a	3002.56
M_m	0
T_a	0
T_m	3600
S_e	2.97E+04
S_{ut}	2.00E+05
d	1.565905

DE Gerber	
n	1.5
K_f	2.0488
K_{fs}	2.84
M_a	3651
M_m	0
T_a	0
T_m	3240
S_e	2.97E+04
S_{ut}	2.00E+05
A	14960.3376
B	15937.63871
d	1.579487612

DE Elliptic	
n	1.5
K_f	2.0488
K_{fs}	2.84
M_a	3651
M_m	0
T_a	0
T_m	3240
S_e	2.97E+04
S_y	2.20E+05
d	1.57218

DE Soderberg	
n	1.5
K_f	2.0488
K_{fs}	2.84
M_a	3651
M_m	0
T_a	0
T_m	3240
S_e	2.97E+04
S_y	2.20E+05
d	1.638625

Key		Units
n	Safety Factor	
K_f	Bending Stress Concentration Factor	
K_{fs}	Torsion Stress Concentration Factor	
M_a	Alternating Moment	lbf-in
M_m	Midrange Moment	lbf-in
T_a	Alternating Torque	lbf-in
T_m	Midrange Torque	lbf-in
S_e	Endurance Limit	psi
S_y, S_{ut}	Yield Strength	psi
d	Diameter	in

Stress Concentrations		
K_t	2.14	Values found on page
K_{ts}	3	1028 of Shigley's
q	0.92	Values found on pages
q_{shear}	0.92	295/296 of Shigley's

Endurance Limit w/ Marin Factors		
S_e' (ksi)	70	Page 282
k_a	0.663118	Page 287
k_b	0.850084	Page 288
k_c	1	Page 290
k_d	1	Page 290
k_e	0.753	Page 293
S_e (ksi)	29.71293	

k_a		
a	2.7	For machined surfaces
b	-0.265	For machined surfaces
S_{ut} (ksi)	200	
k_a	0.663118	

k_b		
Diameter (in.)	1.367	Initial guess for $0.11 \le d \le 2$
k_b	0.850084	

Tapered Roller Bearing Calculations

Gear Reduction

$$Reduction\ 1 = \frac{N_{G_{intermediate}}}{N_{P_{motor}}}$$

$$Reduction\ 2 = \frac{N_{G_{ring\ gear}}}{N_{P_{intermediate}}}$$

$$Overall\ Reduction = Reduction\ 1 * Reduction\ 2$$

Gear Forces Calculation

Motor Pinion:

$$W_t = \frac{Input\ Torque}{\frac{r_P}{12}}$$

$$W_r = W_t * tan(\phi_t)$$

$$W_a = W_t * tan(\Psi)$$

Intermediate Gear:

$$Intermediate\ Shaft\ Torque = Input\ Torque * Reduction\ 1$$

$$W_t = \frac{Intermediate\ Shaft\ Torque}{\frac{r_G}{12}}$$

$$W_r = W_t * tan(\phi_t)$$

$$W_a = W_t * tan(\Psi)$$

Intermediate Pinion:

$$W_t = \frac{Intermediate\ Shaft\ Torque}{\frac{r_P}{12}}$$

$$W_r = W_t * tan(\phi_t)$$

$$W_a = W_t * tan(\Psi)$$

Helical Ring Gear:

$$Ring\ Gear\ Torque = Intermediate\ Shaft\ Torque * Reduction\ 2$$

$$W_t = \frac{Ring\ Gear\ Torque}{\frac{r_G}{12}}$$

$$W_r = W_t * tan(\phi_t)$$

$$W_a = W_t * tan(\Psi)$$

<u>Reaction Forces at Bearing Locations</u>

$$\sum M_{B_Y} = 0$$

$$\sum F_Z = 0$$

$$\sum M_Z = 0$$

$$\sum F_Y = 0$$

<u>Bearing Forces</u>

$$F_{A,\ radial} = \sqrt{{R_{A_Y}}^2 + {R_{A_Z}}^2}$$

$$F_{A,\ axial} = R_{A_X}$$

$$F_{A,\ induced} = \frac{0.47 * F_{A,\ radial}}{K}$$

$$F_{B,\ radial} = \sqrt{{R_{B_Y}}^2 + {R_{B_Z}}^2}$$

$$F_{B,\ axial} = R_{B_X}$$

$$F_{B,\ induced} = \frac{0.47 * F_{B,\ radial}}{K}$$

For $F_{A,i} < F_{B,i} + F_{A,axial}$:

$$F_{A,\ equivalent} = 0.4 * F_{A,\ radial} + K * \left(F_{B,\ induced} + F_{A,\ axial}\right)$$

$$F_{B,\ equivalent} = F_{B,\ radial}$$

<u>Dynamic Loading Rating</u>

$$L_D = \frac{miles * 5280 \frac{ft}{mile} * Overall\ Reduction}{\pi * Tire\ Diameter * \frac{1\ ft}{12\ inch}}$$

$$X_D = \frac{L_D}{L_{10}}$$

$$C_{10} = a_f * F_{A,\ equivalent} * \left(\frac{X_D}{\theta * (1 - Reliability)^{\frac{1}{b}}}\right)^{\frac{1}{a}}$$

Gears

	Helix Angle, Ψ	Normal Pressure Angle, ϕ_n	Transverse Pressure Angle, ϕ_t	Gear Cut	Diameteral Pitch	Teeth
	degrees	degrees	degrees		in	
Motor Pinion	45	14.5	20.1	LH	2	12
Intermediate 1	45	14.5	20.1	RH	6	36
Intermediate 2	45	14.5	20.1	RH	3	18
Ring Gear	45	14.5	20.1	LH	8.5	51

Ratios

Motor Pinion	Reduction 1	3	Overall Reduction	8.5
Intermediate 1				
Intermediate 2	Reduction 2	2.833333333		
Ring Gear				

Forces

	Torque	RPM	H	Velocity	Wt	Wr	Wa
	lb-ft		hp	ft/min	lbf	lbf	lbf
Motor Pinion	400	4,000	305	2,094	4800.103	1755.592	4800.103
Intermediate 1	1200	1,333	305	2,094	4800.103	1755.592	4800.103
Intermediate 2	1200	1,333	305	1,047	9600.207	3511.184	9600.207
Ring Gear	3400	471	305	1,047	9600.207	3511.184	9600.207

Design Parameters

Parameter/Variable	Value	Units
Warranty	3000	miles
Tire Diameter	27.35	in
L10	1.00E+06	
System Reliability	0.99	
Individual Bearing Reliability	0.994987437	
V (Inner Ring Rotation)	1	
Load Application Factor, a_f (Commercial Gearing/No Impact)	1.2	
x_0	0.02	
θ	4.459	
a	3	
b	1.483	
lbf to kN Conversion Factor	0.00444822	

Ratios

	Diametral Pitch in				
Motor Pinion	2	Reduction 1	3	Overall Reduction	8.5
Intermediate 1	6				
Intermediate 2	3	Reduction 2	2.833333333		
Ring Gear	12				

Key
Values Can Be Altered
Changes With Bearing

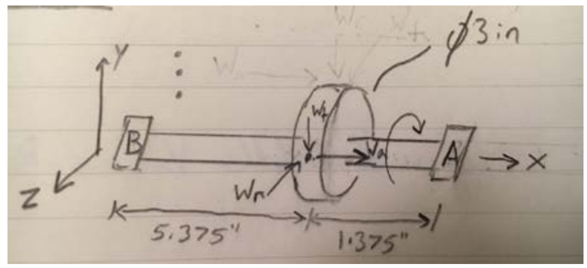
Forces

	Wt kN	Wr kN	Wa kN
Motor Pini	21.35192	7.80926	21.35192
8" Gear	21.35192	7.80926	21.35192
4" Gear	42.70383	15.61852	42.70383
Ring Gear	42.70383	15.61852	42.70383

Input Shaft

Distance to A	1.375	in
Distance to B	5.375	in

Bearing at Location A (See Figure)			**Bearing at Location B** (See Figure)		
R_{AX}	21.352	kN	R_{BX}	0	kN
R_{AY}	17.002	kN	R_{BY}	4.349	kN
R_{AZ}	9.382	kN	R_{BZ}	-1.572	kN
F_{radial}	19.419	kN	F_{radial}	4.625	kN
F_{axial}	21.352	kN	F_{axial}	0.000	kN
Table 11-1 (Shigley's Textbook)			**Table 11-1 (Shigley's Textbook)**		
X2	0.560		X2	0.560	
Y2	1		Y2	1	
Equivalent Force - Combined Radial and Thrust			**Equivalent Force**		
$F_{equivalent}$	32.227	kN	$F_{equivalent}$	N/A	
Bearing Calculation			**Bearing Calculation**		
L_D	18803909	revolutions	L_D	18803909	revolutions
X_D	18.80		X_D	18.80	
$C10_A$	195.8083589	kN	$C10_B$	28.1013607	kN
$C0_A$	38.12842122	kN	$C0_B$	0	kN

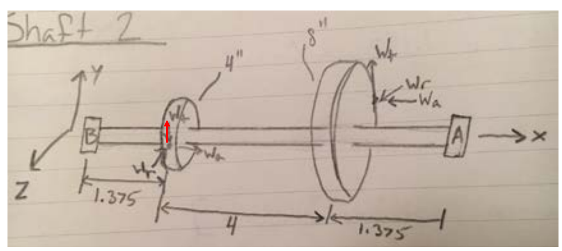


Head to Table 11-2 to Size Bearing (Max Out at 121 kN - Angular Contact / 108 kN - Deep Groove

Intermediate Shaft

4.5" Gear Distance to A	5.375	in
4.5" Gear Distance to B	1.375	in
9" Gear Distance to A	1.375	in
9" Gear Distance to B	5.375	in

Bearing at Location A (See Figure)			**Bearing at Location B** (See Figure)		
R_{AX}	-21.352	kN	R_{BX}	0	kN
R_{AY}	-25.701	kN	R_{BY}	-38.354	kN
R_{AZ}	15.943	kN	R_{BZ}	-8.133	kN
$F_{A,radial}$	30.244	kN	$F_{B,radial}$	39.207	kN
$F_{A,axial}$	21.352	kN	$F_{B,axial}$	0.000	kN
Table 11-1 (Shigley's Textbook)			**Table 11-1 (Shigley's Textbook)**		
X2	0.560		X2	0.560	
Y2	1		Y2	1	
Equivalent Force - Combined Radial and Thrust			**Equivalent Force**		
$F_{equivalent}$	38.289	kN	$F_{equivalent}$	N/A	
Bearing Calculation			**Bearing Calculation**		
L_D	6267970	revolutions	L_D	6267970	revolutions
X_D	6.27		X_D	6.27	
$C10_A$	161.3051113	kN	$C10_B$	165.174381	kN
$C0_A$	38.12842122	kN	$C0_B$	0	kN



Head to Table 11-2 to Size Bearing (Max Out at 121 kN - Angular Contact / 108 kN - Deep Groove

Design Parameters

Parameter/Variable	Value	Units
Warranty	1000	miles
Tire Diameter	27.35	in
L10	9.00E+07	
System Reliability	0.99	
Individual Bearing Reliability	0.994987437	
K	1.5	
Load Application Factor, a_f (Commercial Gearing/No Impact)	1.2	
x_0	0	
θ	4.48	
a	3.33	
b	1.5	
lbf to kN Conversion Factor	0.00444822	

Ratios

	Diametral Pitch in				
Motor Pinion	2	Reduction 1	3	Overall Reduction	8.5
Intermediate 1	6				
Intermediate 2	3	Reduction 2	2.833333333		
Ring Gear	8.5				

Key
- Values Can Be Altered
- Changes With Bearing

Forces

	Wt kN	Wr kN	Wa kN
Motor Pinion	21.35192	7.80926	21.35192
6" Gear	21.35192	7.80926	21.35192
3" Gear	42.70383	15.61852	42.70383
Ring Gear	42.70383	15.61852	42.70383

Input Shaft

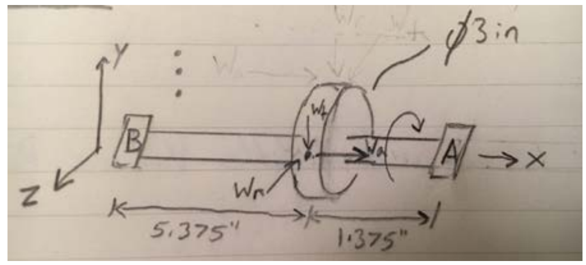
Distance to A	1.375	in
Distance to B	5.375	in

Bearing at Location A (See Figure)			**Bearing at Location B** (See Figure)		
R_{AX}	21.352	kN	R_{BX}	0	kN
R_{AY}	17.002	kN	R_{BY}	4.349	kN
R_{AZ}	9.382	kN	R_{BZ}	-1.572	kN
$F_{A,radial}$	19.419	kN	$F_{B,radial}$	4.625	kN
$F_{A,axial}$	21.352	kN	$F_{B,axial}$	0.000	kN
Induced Force			**Induced Force**		
$F_{A,i}$	6.085	kN	$F_{B,i}$	1.449	kN

Check that $F_{A,i} < F_{B,i} + F_{A,axial}$, then proceed

Equivalent Force - Combined Radial and Thrust			**Equivalent Force**		
$F_{A,equivalent}$	41.969	kN	$F_{B,equivalent}$	4.625	kN
Bearing Calculation			**Bearing Calculation**		
L_D	6267970	revolutions	L_D	6267970	revolutions
X_D	0.07		X_D	0.07	
$C10_A$	41.64686848	kN	$C10_B$	4.58946146	kN

Head to Figure 11-15 to Size Bearing (Max Out at 22.7 kN)



Intermediate Shaft

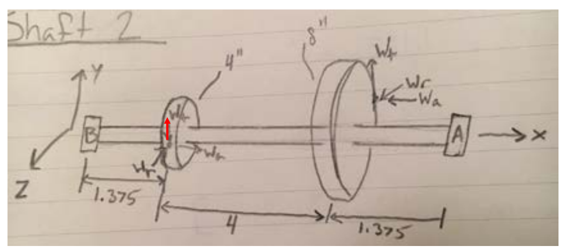
4.5" Gear Distance to A	5.375	in
4.5" Gear Distance to B	1.375	in
9" Gear Distance to A	1.375	in
9" Gear Distance to B	5.375	in

Bearing at Location A (See Figure)			**Bearing at Location B** (See Figure)		
R_{AX}	21.352	kN	R_{BX}	0	kN
R_{AY}	-25.701	kN	R_{BY}	-38.354	kN
R_{AZ}	15.943	kN	R_{BZ}	-8.133	kN
$F_{A,radial}$	30.244	kN	$F_{B,radial}$	39.207	kN
$F_{A,axial}$	21.352	kN	$F_{B,axial}$	0.000	kN
Induced Force			**Induced Force**		
$F_{A,i}$	9.477	kN	$F_{B,i}$	12.285	kN

Check that $F_{A,i} < F_{B,i} + F_{A,axial}$, then proceed

Equivalent Force - Combined Radial and Thrust			**Equivalent Force**		
$F_{A,equivalent}$	62.553	kN	$F_{B,equivalent}$	39.207	kN
Bearing Calculation			**Bearing Calculation**		
L_D	2089323	revolutions	L_D	2089323	revolutions
X_D	0.02		X_D	0.02	
$C10_A$	44.64402154	kN	$C10_B$	27.9821518	kN

Head to Figure 11-15 to Size Bearing (Max Out at 22.7 kN)



Housing Plate Deflection Analysis

Displacement

$$w(x,y)=\frac{4P_c}{\pi^4 D L_x L_y}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{\sin\left(\frac{m\pi a}{L_x}\right)\sin\left(\frac{n\pi b}{L_y}\right)\sin\left(\frac{m\pi x}{L_x}\right)\sin\left(\frac{n\pi y}{L_y}\right)}{\left(\left(\frac{m}{L_x}\right)^2+\left(\frac{n}{L_y}\right)^2\right)^2}$$

$w_{at\ loading} = w(a,\ b) = 0.354915467558\ \text{in} \approx 0.355\ \text{in}$

The above displacement is based on the first $4 \times 4 = 16$ terms of the series solution.

Stress

$$M_x(x,y)=\frac{4P_c}{\pi^2 L_x L_y}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\left(\left(\frac{m}{L_x}\right)^2+v\left(\frac{n}{L_y}\right)^2\right)\frac{\sin\left(\frac{m\pi a}{L_x}\right)\sin\left(\frac{n\pi b}{L_y}\right)\sin\left(\frac{m\pi x}{L_x}\right)\sin\left(\frac{n\pi y}{L_y}\right)}{\left(\left(\frac{m}{L_x}\right)^2+\left(\frac{n}{L_y}\right)^2\right)^2}$$

$$M_y(x,y)=\frac{4P_c}{\pi^2 L_x L_y}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\left(v\left(\frac{m}{L_x}\right)^2+\left(\frac{n}{L_y}\right)^2\right)\frac{\sin\left(\frac{m\pi a}{L_x}\right)\sin\left(\frac{n\pi b}{L_y}\right)\sin\left(\frac{m\pi x}{L_x}\right)\sin\left(\frac{n\pi y}{L_y}\right)}{\left(\left(\frac{m}{L_x}\right)^2+\left(\frac{n}{L_y}\right)^2\right)^2}$$

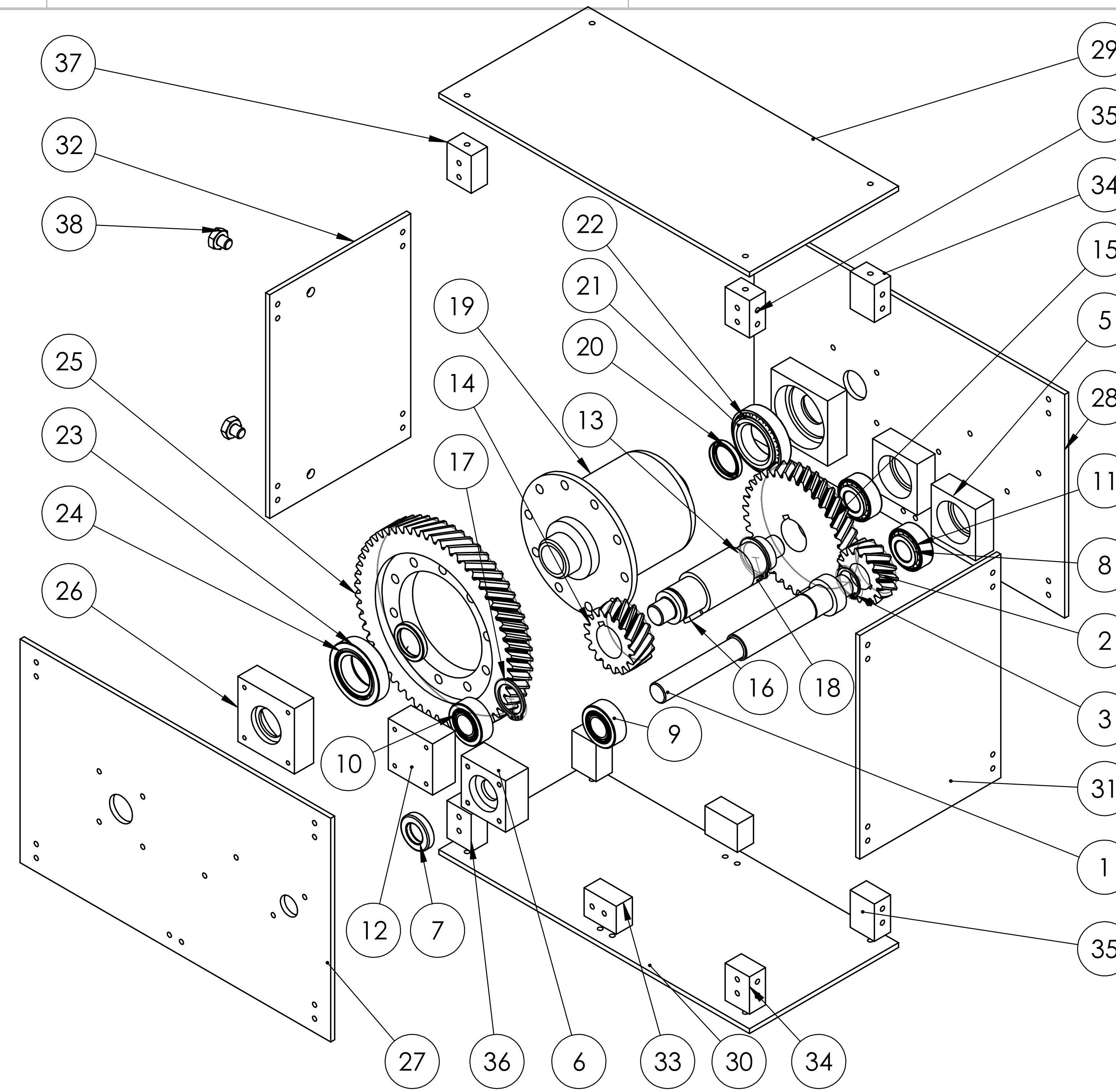
$\sigma_x\left(a,b,\pm\frac{h}{2}\right)=\frac{6}{h^2}Mx(a,b)= 50071.4845572\ \text{psi} \approx 5.01 \times 10^4\ \text{psi}$

$\sigma_y\left(a,b,\pm\frac{h}{2}\right)=\frac{6}{h^2}M_y(a,b)= 37336.3855347\ \text{psi} \approx 3.73 \times 10^4\ \text{psi}$

This calculation is for a single point load generated by the axial force from the helical gears and shows the worst case deflection if all the force concentrated at one point on the aluminum plates of the housing. The plate thickness is 1/4" and the load was placed on the longest side plate also as a worst case condition

Table of Contents

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	ST501-01	DRIVETRAIN INPUT SHAFT	1
2	ST501-02	2.5inchRushGears__1096-0868	1
3	ST501-03	Snap_Ring_Input_97633A340	1
4	ST501-04	Input Shaft Key	1
5	ST501-06 & ST502-07	Input Mounts	2
6	ST501-07	Input Bearing Mount With Seal	1
7	ST501-08	Seal 9803	1
8	ST501-06/07 & ST502-07/08	Tapered_Roller_33205_Q_PART1	4
9	ST501-06/07 & ST502-07/08	Tapered_Roller_33205_Q_PART2	4
10	ST501-06/07 & ST502-07/08	Roller_Bearing_33205_Q_PART4	4
11	ST501-06/07 & ST502-07/08	Tapered_Roller_33205_Q_PART3	4
12	ST502-04	Intermediate Seal Side Mount	1
13	ST502-01	DRIVETRAIN INT. SHAFT	1
14	ST502-02	2.83inchRushGears__1096-0857	1
15	ST502-03	6inch-RushGears__1096-0846	1
16	ST502-05	Intermediate_Shaft Key 1	1
17	ST502-04	Snap_Ring_Int_98585A136	2
18	ST502-05	Intermediate_Shaft Key 2	1
19	ST503-01	Differential	1
20	ST503-02	Seal 13510	2
21	ST503-02	Differential_Inner_Bearing_Race	2
22	ST503-02	Middle_Race_Bearing_Differential	2
23	ST503-02	Outer_Race_Bearing_Differential	2
24	ST503-02	Roller_Bearing_Differential	2
25	ST503-03	RingGear-RushGears__1096-0834	1
26	ST503-06/07	Output Mount	2
27	ST504-01	Housing.LongSide.Input	1
28	ST504-02	Housing.Side.Long	1
29	ST504-03	HousingSideTop	1
30	ST504-04	HousingSide.Bot	1
31	ST504-05	Housing.Front	1
32	ST504-06	Housing.Back	1
33	ST504-08	BottomBlocksSupport	2
34	ST504-08	ALCorner.2	2
35	ST504-08	ALCorner	2
36	ST504-08	ALCorner.3	2
37	ST504-08	ALCorner4	2
38	ST504-11/12	HHFBOLT 0.5000-13x0.5x0.5-N	2



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009

MATERIAL: 4140 HT STEEL

FINISH: NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	CHARISSA SEID	5/31/17
CHECKED	JIMMY KING	5/31/17
ENG APPR.		
MFG APPR.		
Q.A.		

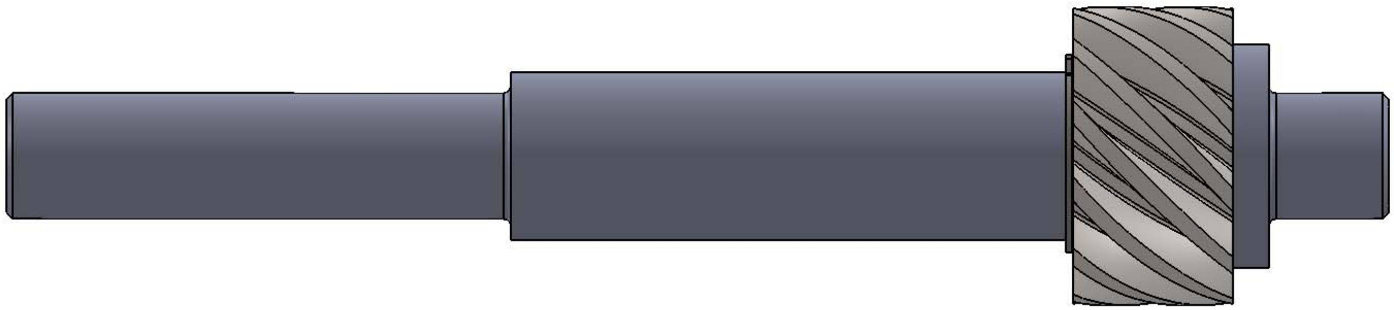
COMMENTS:

TITLE:

TOP LEVEL DRIVETRAIN ASSEMBLY

SIZE B | DWG. NO. 000 | REV A

SCALE: 1:16 | WEIGHT: | SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/32 ANGULAR ± 1° TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	DRAWN	JIMMY KING	2/9/17	TITLE:
	CHECKED	KEVIN MOORE	2/9/17	DRIVETRAIN INPUT SHAFT ASSEMBLY
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009	Q.A.			
MATERIAL -----	COMMENTS:			SIZE A
FINISH -----				DWG. NO. 100
DO NOT SCALE DRAWING				REV A
				SCALE: 2:3
				SHEET 1 OF 1

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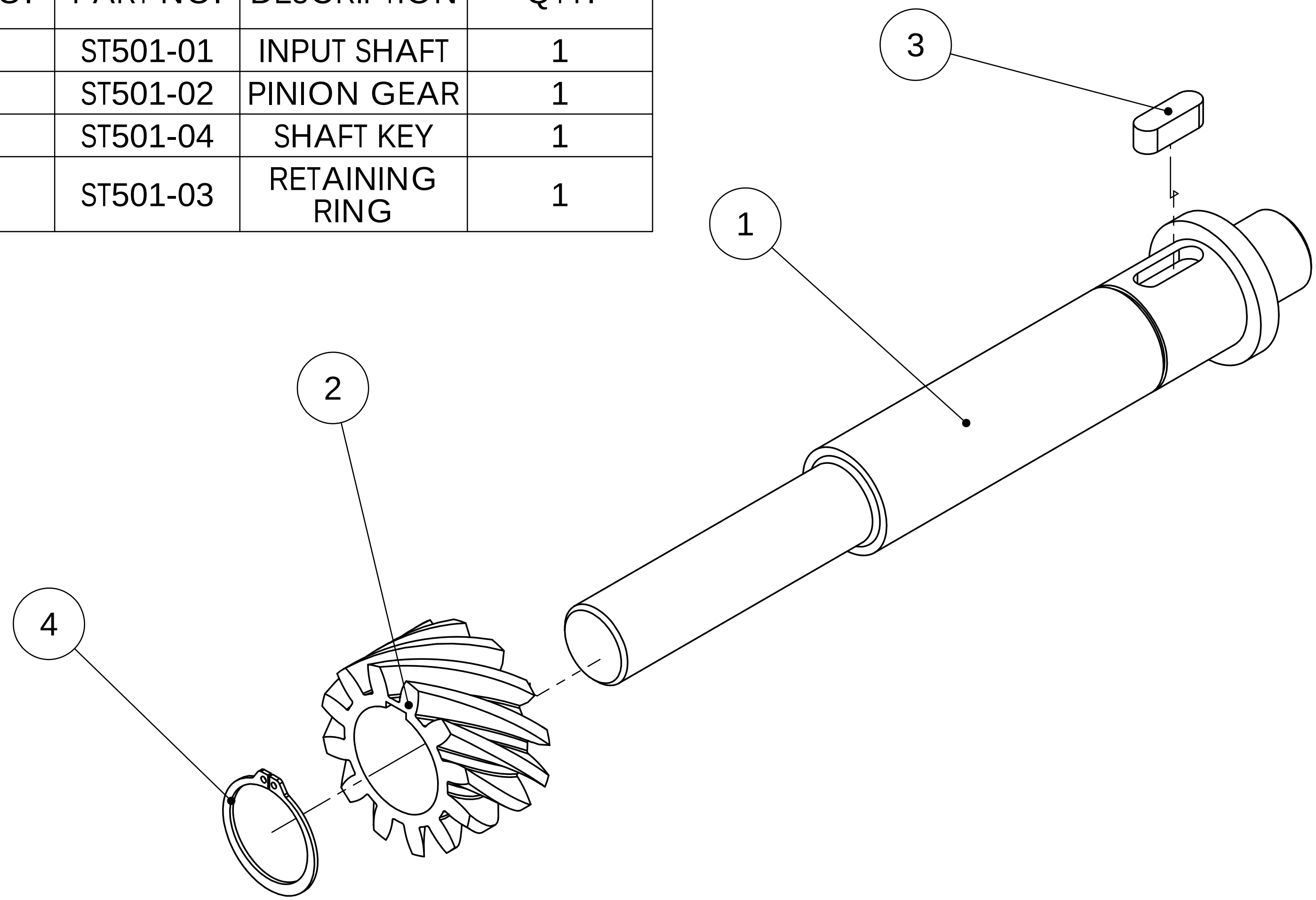
ITEM NO.	PART NO.	DESCRIPTION	QTY.
1	ST501-01	INPUT SHAFT	1
2	ST501-02	PINION GEAR	1
3	ST501-04	SHAFT KEY	1
4	ST501-03	RETAINING RING	1



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MATERIAL -----

FINISH -----

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JIMMY KING	2/9/17
CHECKED	KEVIN MOORE	2/9/17
ENG APPR.		
MFG APPR.		
Q.A.		

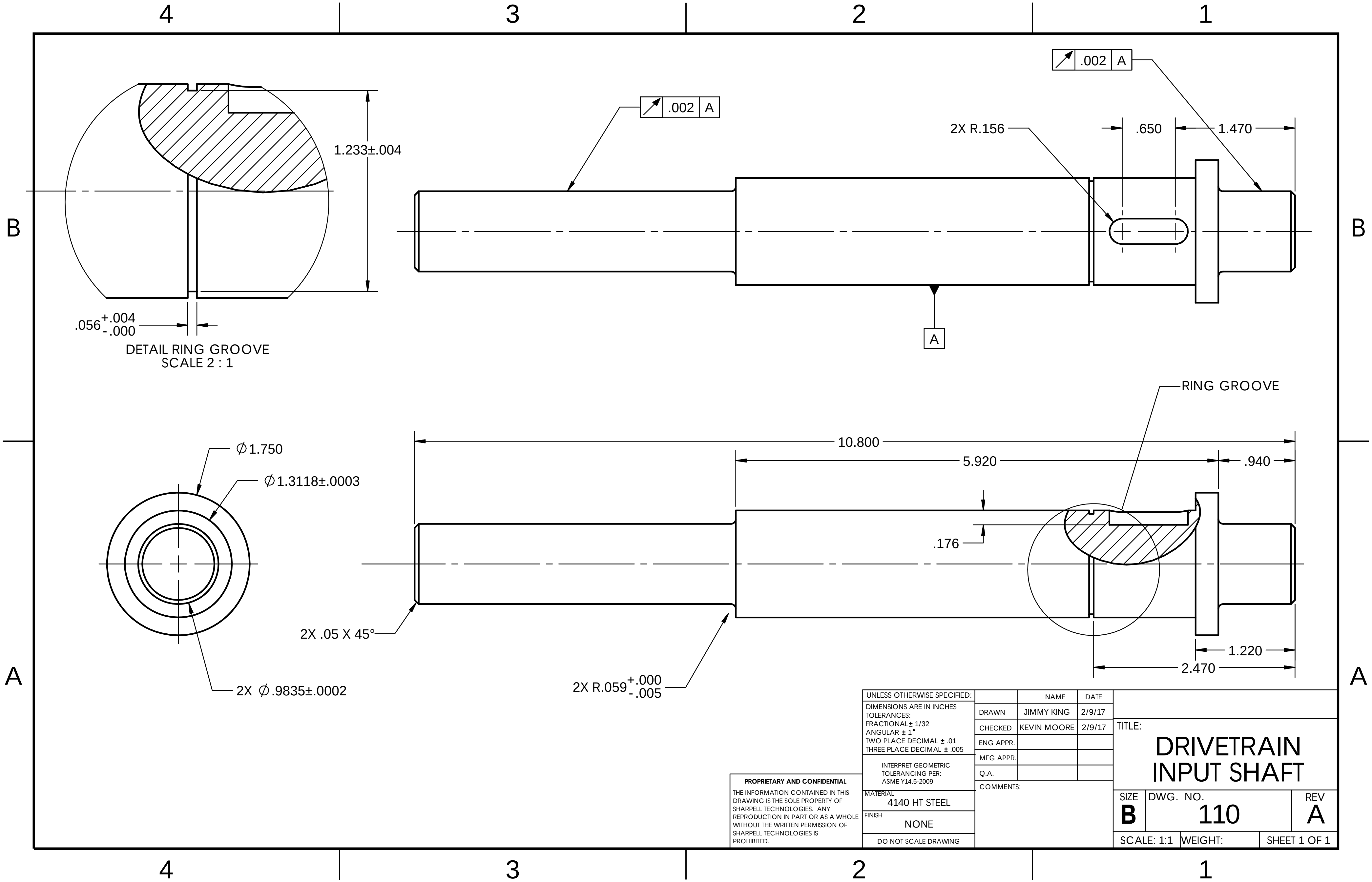
COMMENTS:

TITLE:

EXPLODED DRIVETRAIN INPUT SHAFT ASSEMBLY

SIZE	DWG. NO.	REV
A	101	A

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



.002 A
.002 A
2X R.156
.650
1.470
1.233±.004
.056 +.004 -.000
DETAIL RING GROOVE
SCALE 2 : 1
A
RING GROOVE
∅1.750
∅1.3118±.0003
10.800
5.920
.940
.176
2X .05 X 45°
2X R.059 +.000 -.005
2X ∅.9835±.0002
1.220
2.470
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009
MATERIAL 4140 HT STEEL
FINISH NONE
DO NOT SCALE DRAWING
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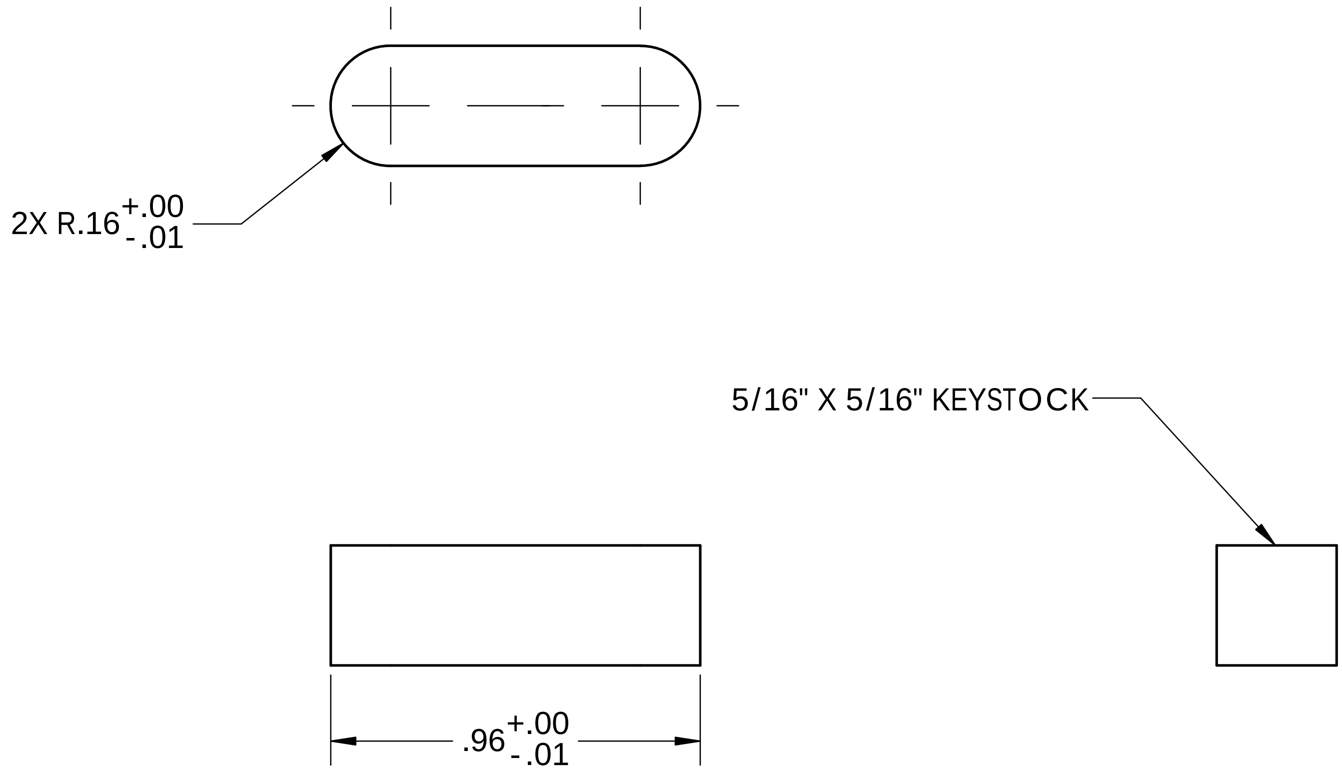
NAME DATE
DRAWN JIMMY KING 2/9/17
CHECKED KEVIN MOORE 2/9/17
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
DRIVETRAIN INPUT SHAFT
SIZE B
DWG. NO. 110
REV A
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1



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DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009

MATERIAL: LOW CARBON STEEL

FINISH: NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JIMMY KING	2/9/17
CHECKED	KEVIN MOORE	2/9/17
ENG APPR.		
MFG APPR.		
Q.A.		

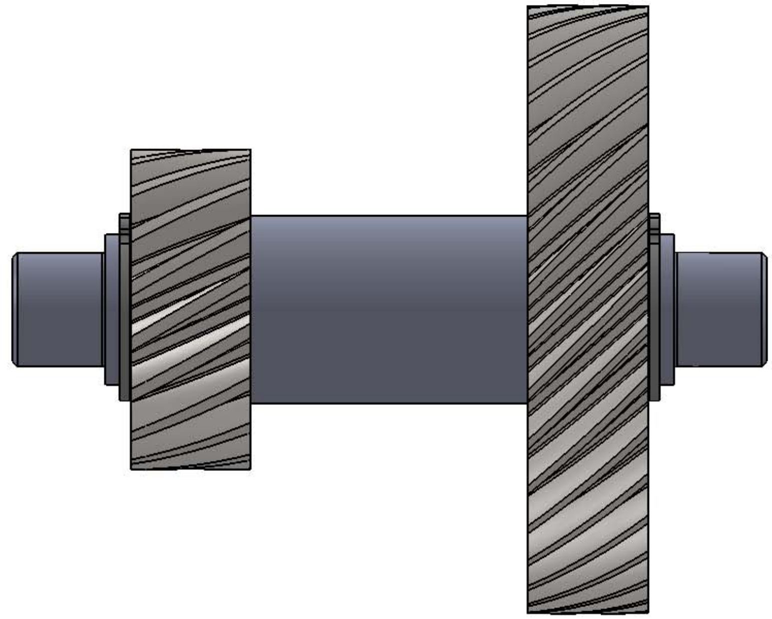
COMMENTS:

TITLE:

INPUT SHAFT KEY

SIZE	DWG. NO.	REV
A	111	A

SCALE: 4:1 | SHEET 1 OF 1



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INTERPRET GEOMETRIC TOLERANCING PER:		
MATERIAL -----		
FINISH -----		
DO NOT SCALE DRAWING		

	NAME	DATE
DRAWN	JIMMY KING	2/9/17
CHECKED	KEVIN MOORE	2/9/17
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:

DRIVETRAIN INTERMEDIATE SHAFT ASSEMBLY

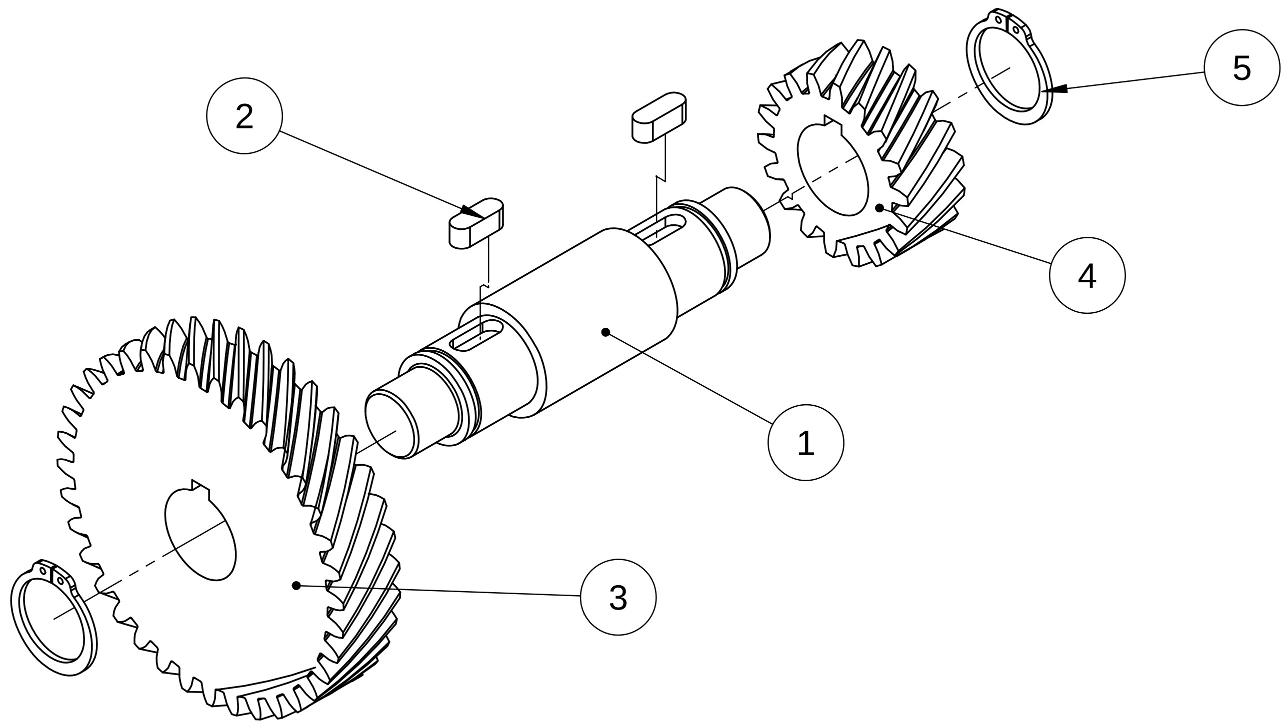
SIZE	DWG. NO.	REV
A	200	A

SCALE: 1:2	SHEET 1 OF 1

2 1

ITEM NO.	PART NO.	DESCRIPTION	QTY.
1	ST502-02	INTERMEDIATE SHAFT	1
2	ST502-06	SHAFT KEY	2
3	ST502-04	MATING GEAR	1
4	ST502-03	PINION GEAR	1
5	ST502-05	RETAINING RING	2

B B



A A

PROPRIETARY AND CONFIDENTIAL		NAME	DATE
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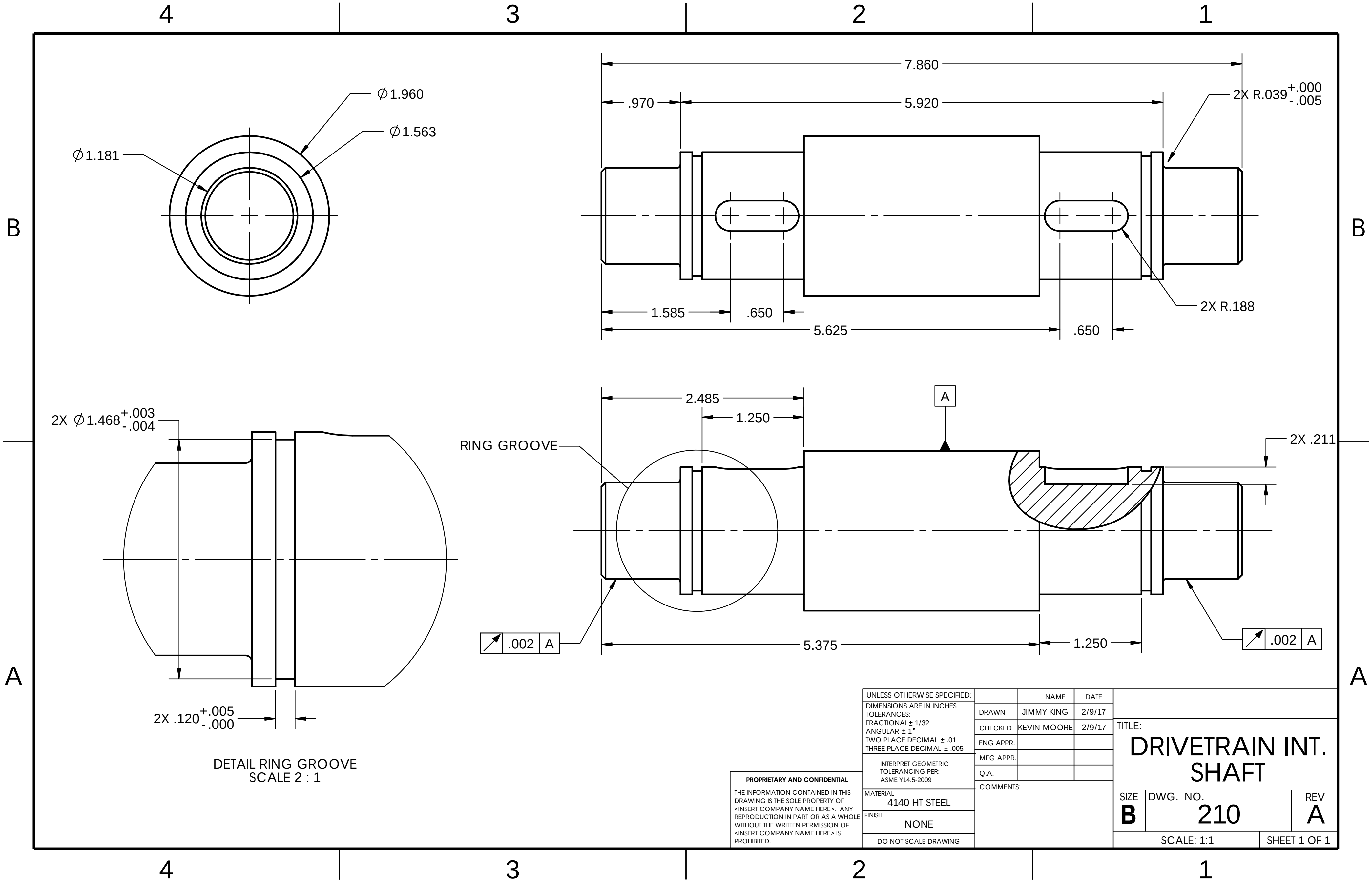
	CHECKED	KEVIN MOORE	2/9/17
	ENG APPR.		
	MFG APPR.		
	Q.A.		
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:		
MATERIAL -----			
FINISH -----			
DO NOT SCALE DRAWING			

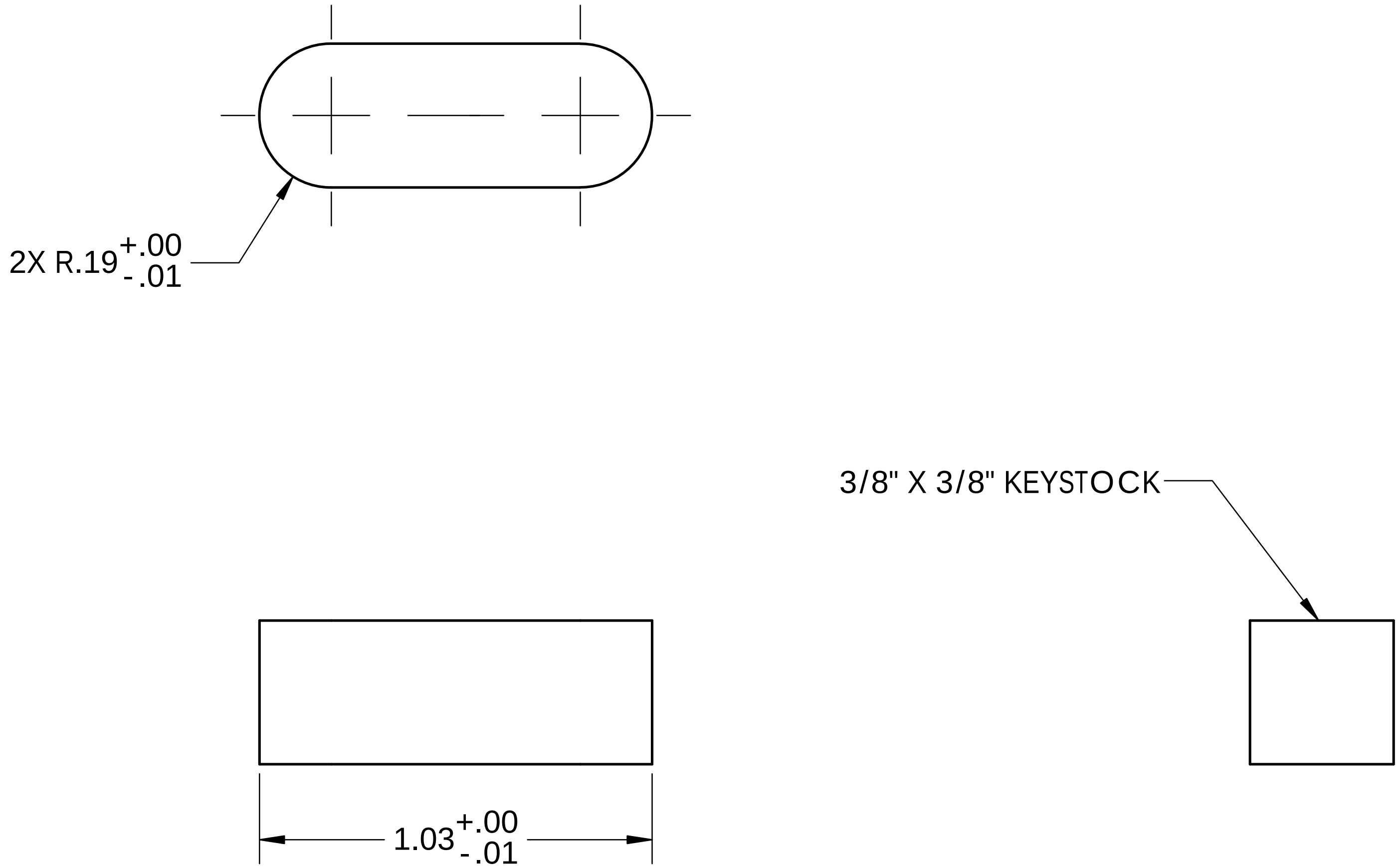
TITLE:
EXPLODED DRIVETRAIN INTERMEDIATE SHAFT ASSEMBLY

SIZE	DWG. NO.	REV
A	201	A
SCALE: 1:3		SHEET 1 OF 1

2 1



7.860
.970
5.920
2X R.039 +.000 -.005
∅1.960
∅1.563
∅1.181
2X R.188
1.585
.650
5.625
.650
2.485
1.250
A
RING GROOVE
2X .211
.002 A
5.375
1.250
.002 A
2X ∅1.468 +.003 -.004
2X .120 +.005 -.000
DETAIL RING GROOVE
SCALE 2 : 1
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009
MATERIAL
4140 HT STEEL
FINISH
NONE
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.
NAME DATE
DRAWN JIMMY KING 2/9/17
CHECKED KEVIN MOORE 2/9/17
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
DRIVETRAIN INT. SHAFT
SIZE B
DWG. NO. 210
REV A
SCALE: 1:1
SHEET 1 OF 1

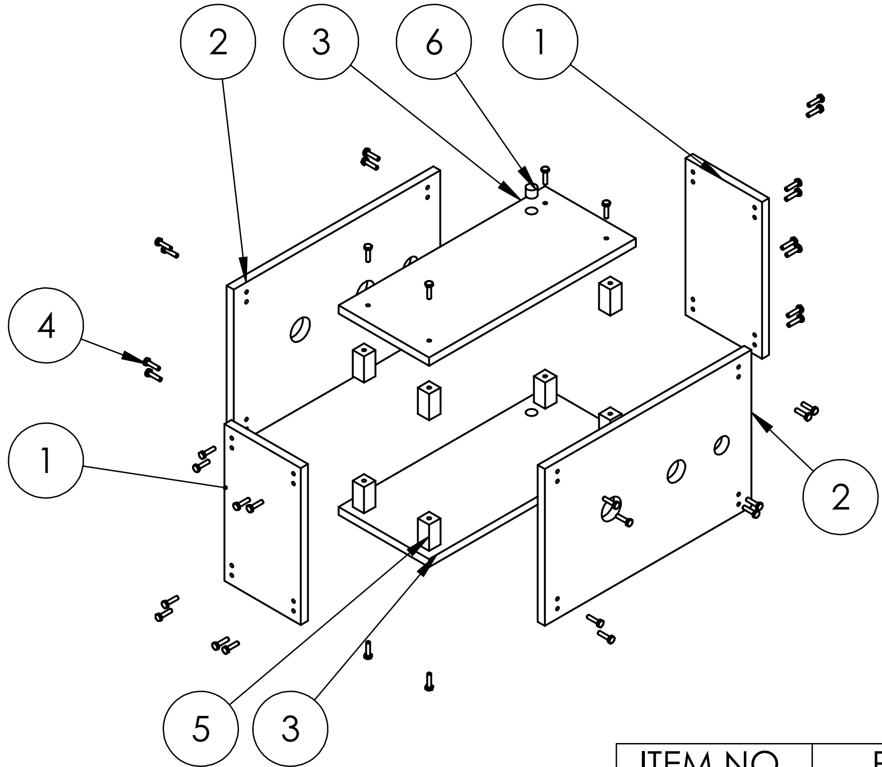


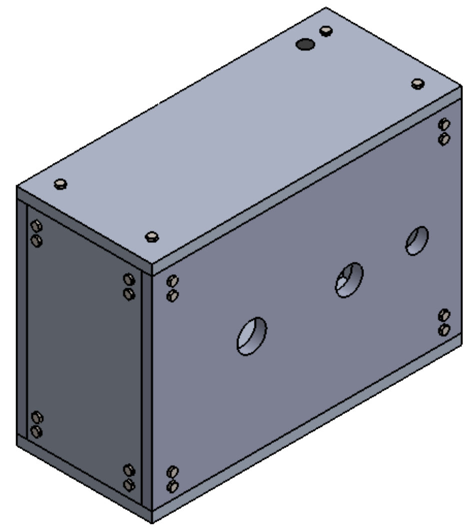
UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/32 ANGULAR ± 1° TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	DRAWN	JIMMY KING	2/9/17
	CHECKED	KEVIN MOORE	2/9/17
	ENG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009	MFG APPR.		
	Q.A.		
MATERIAL LOW CARBON STEEL	COMMENTS:		
FINISH NONE			
DO NOT SCALE DRAWING			

TITLE:

INT. SHAFT KEY

SIZE	DWG. NO.	REV
A	211	A
SCALE: 4:1	SHEET 1 OF 1	





ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	ST504-04	AL PLATE 0.625X12X12	2
2	ST504-02	AL PLATE 0.625X12X24	2
3	ST504-03	AL PLATE 0.625X12X24	2
4	ST504-11	HFBOLT 0.25-20x1x0.75-N	40
5	ST504-06	AL CORNER	8
6	ST504-13	DRAIN PLUG	2

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR: ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T6

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	SEID	2-9-17
CHECKED	MOORE	2-9-17
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
SEAL WITH LOCTITE ENGINE SEALANT AND THREAD FASTENER AFTER JOINING PLATES
WEIGHT GIVEN IN LBS

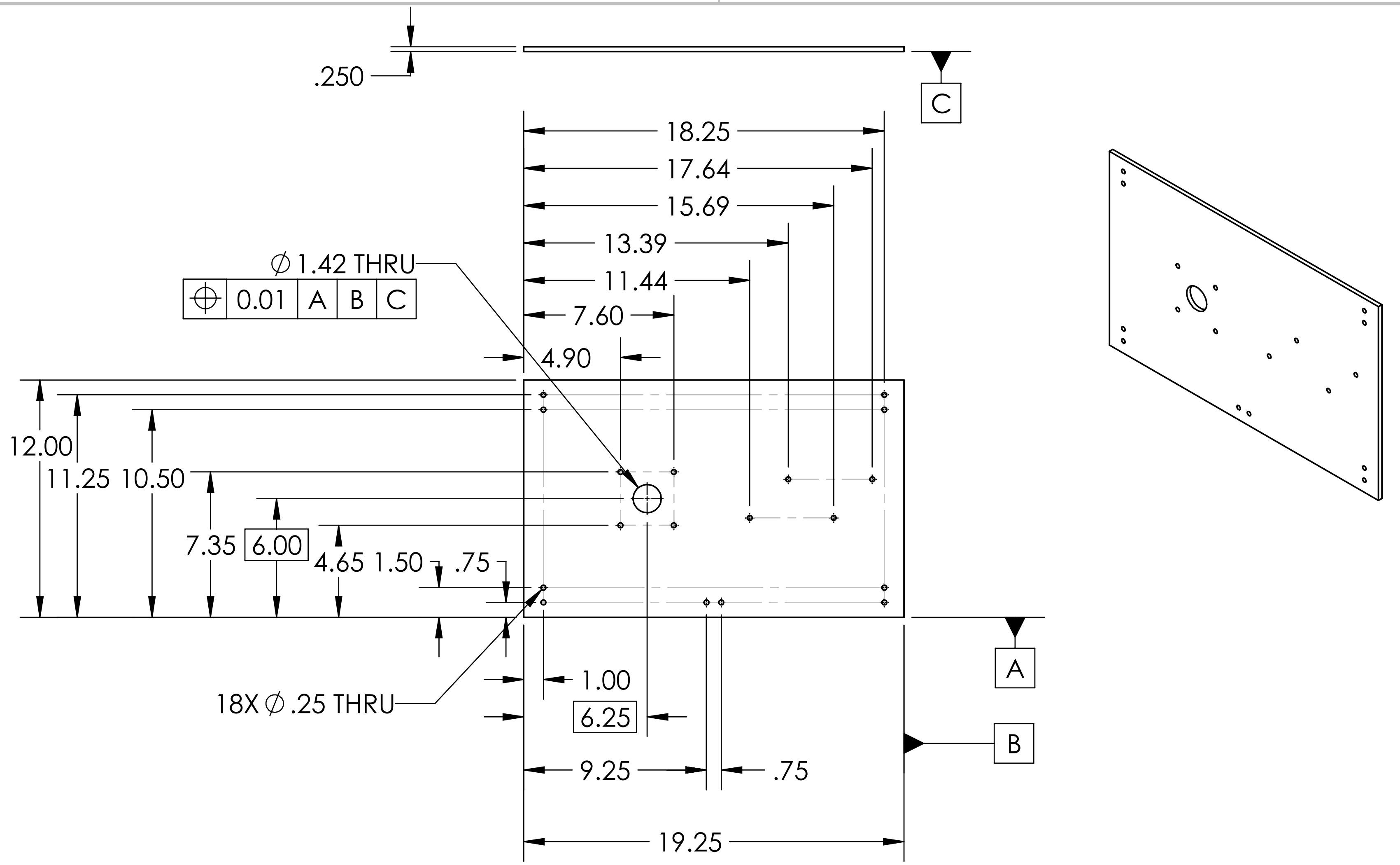
TITLE:

Housing ASSEMBLY

SIZE A | DWG. NO. 300 | REV A

SCALE: 1:16 | WEIGHT: 38.87 | SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ±1/32 ANGULAR: ±1° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	DRAWN	SEID	2-3-17	TITLE:
			CHECKED	MOORE	2-9-17	HOUSING LEFT VERTICAL SIDE
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL Aluminum 6061-T6	COMMENTS: DIMENSIONS RELATIONED TO ENSURE SHAFT AND GEAR ALIGNMENT FOR THE CENTER HOLE.			SIZE A · DWG. NO. 310 · REV A
NEXT ASSY	USED ON: HOUSING ASSEMBLY	FINISH NONE				
APPLICATION:		DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: 5.57LBF · SHEET 1 OF 1

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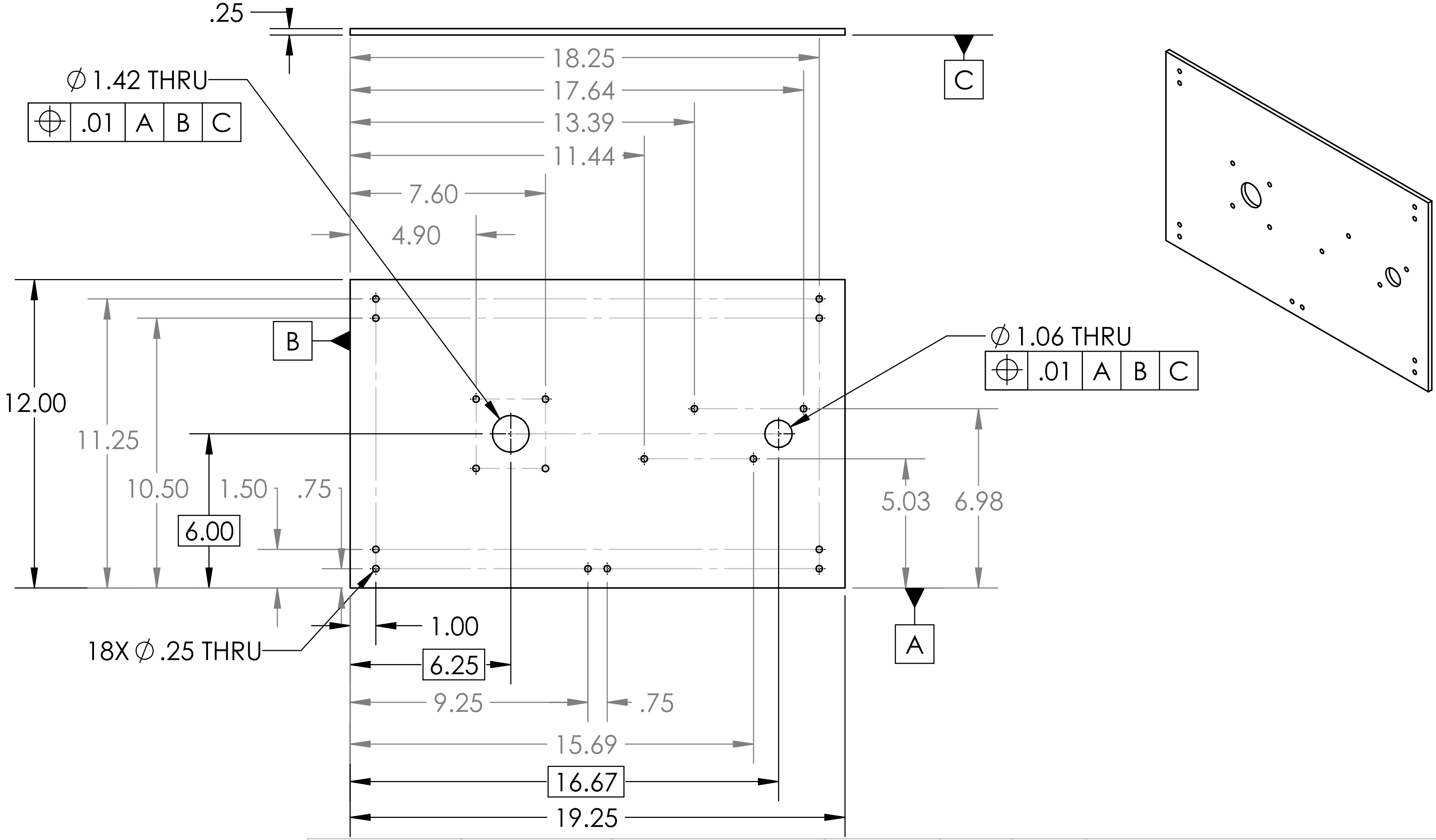

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NEXT ASSY | USED ON: HOUSING ASSEMBLY

APPLICATION:

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR: ± 1°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T6

FINISH
NONE

DO NOT SCALE DRAWING

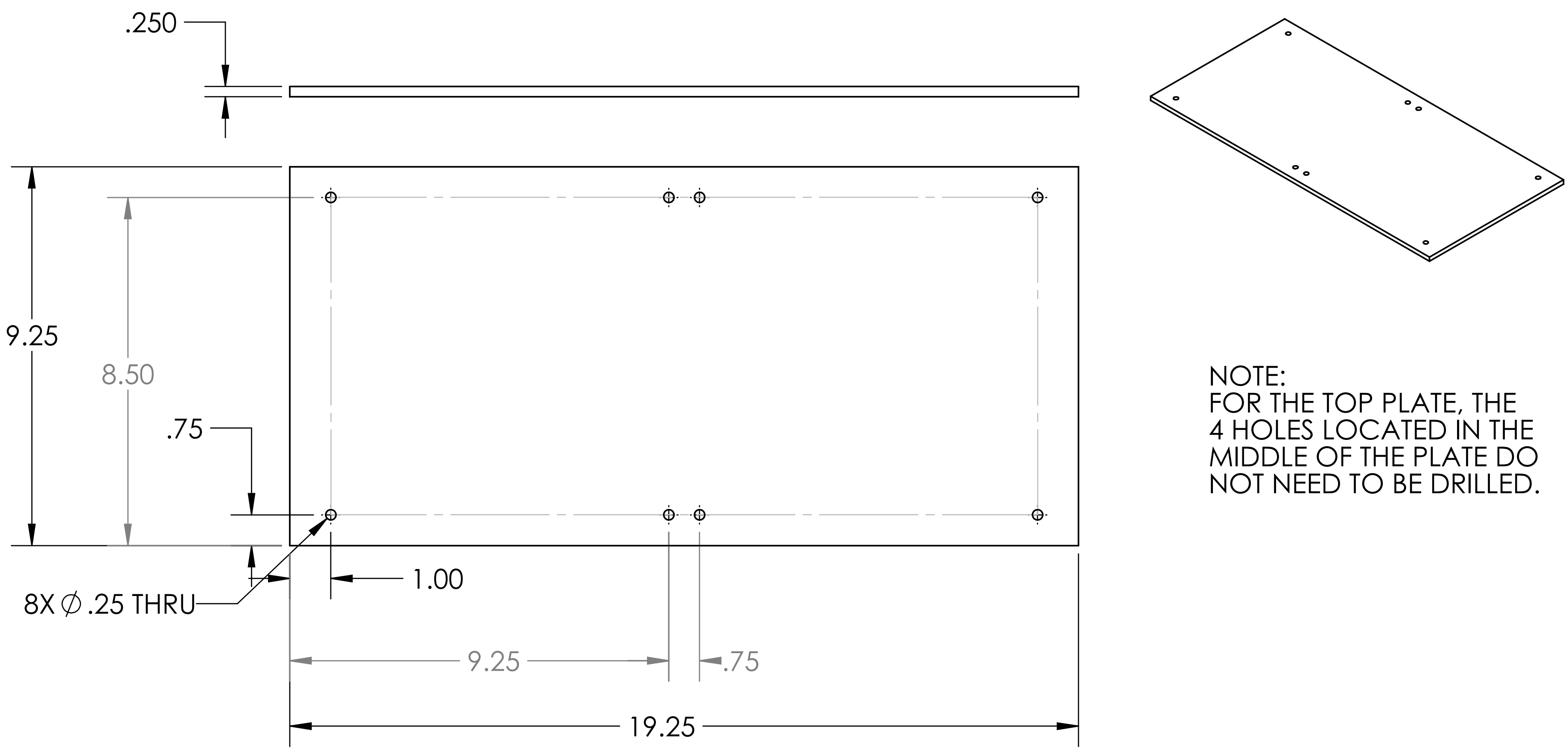
	NAME	DATE
DRAWN	SEID	2-9-17
CHECKED	MOORE	2-9-17
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
TOP PIECE WILL HAVE 2 OPTIONS WITH SAME SPECIFICATION. ONE ACRYLIC, ONE ALUMINUM
Weight IN LBF
LOCATE OIL SEAL OFF OF EDGE CLOSEST TO PINION

TITLE:
HOUSING RIGHT VERTICAL SIDE

SIZE A | DWG. NO. 311 | REV A

SCALE: 1:4 | WEIGHT: 4.34 | SHEET 1 OF 1



NOTE:
FOR THE TOP PLATE, THE 4 HOLES LOCATED IN THE MIDDLE OF THE PLATE DO NOT NEED TO BE DRILLED.

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NEXT ASSY	USED ON: HOUSING ASSEMBLY
APPLICATION:	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR: ± 1°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T6

FINISH
NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	SEID	2-9-17
CHECKED	MOORE	2-9-17
ENG APPR.		
MFG APPR.		
Q.A.		

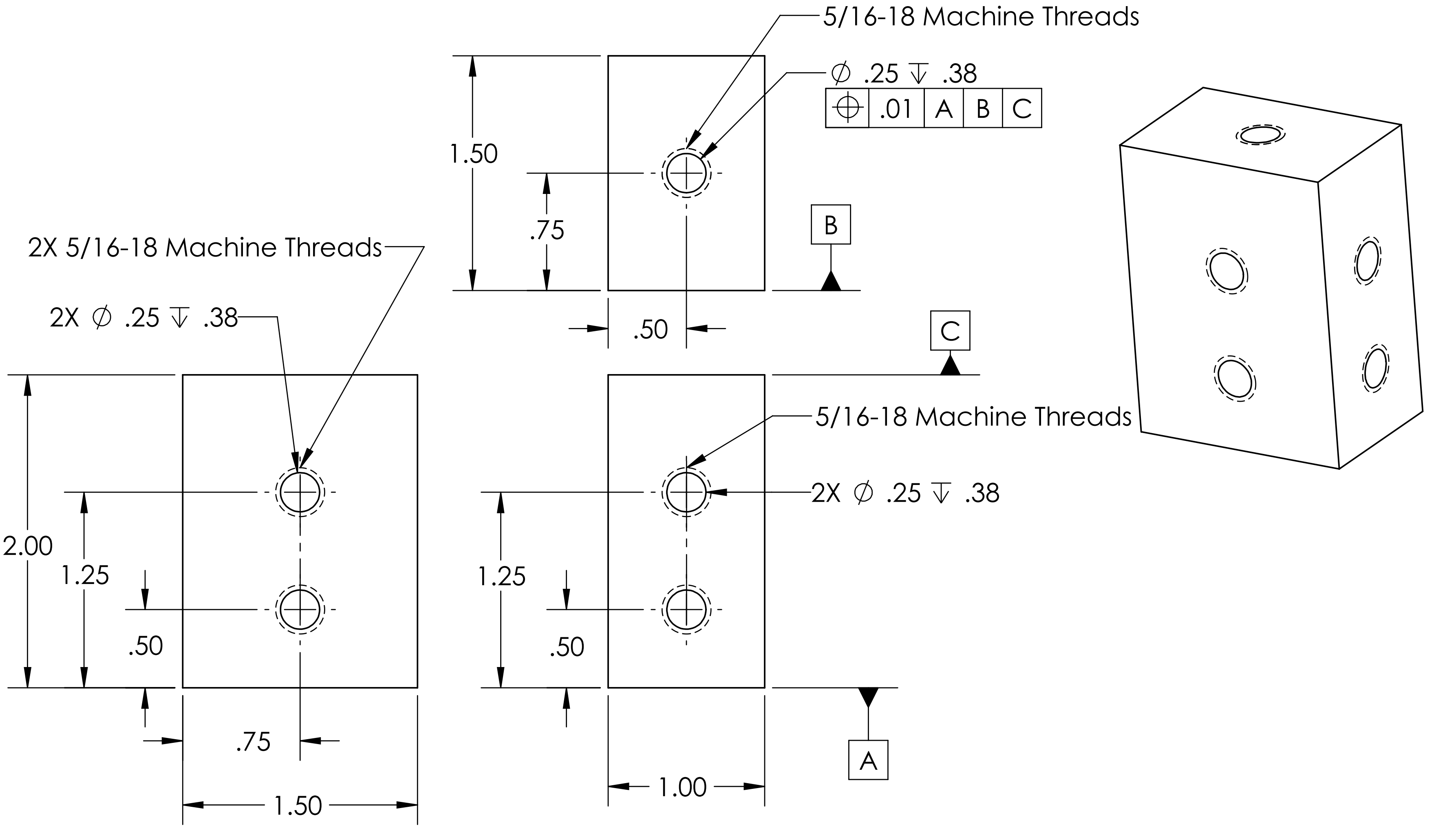
COMMENTS:
TOP PIECE WILL HAVE 2 OPTIONS WITH SAME SPECIFICATION. ONE ACRYLIC, ONE ALUMINUM
Weight IN LBF
LOCATE OIL SEAL OFF OF EDGE CLOSEST TO PINION

TITLE:

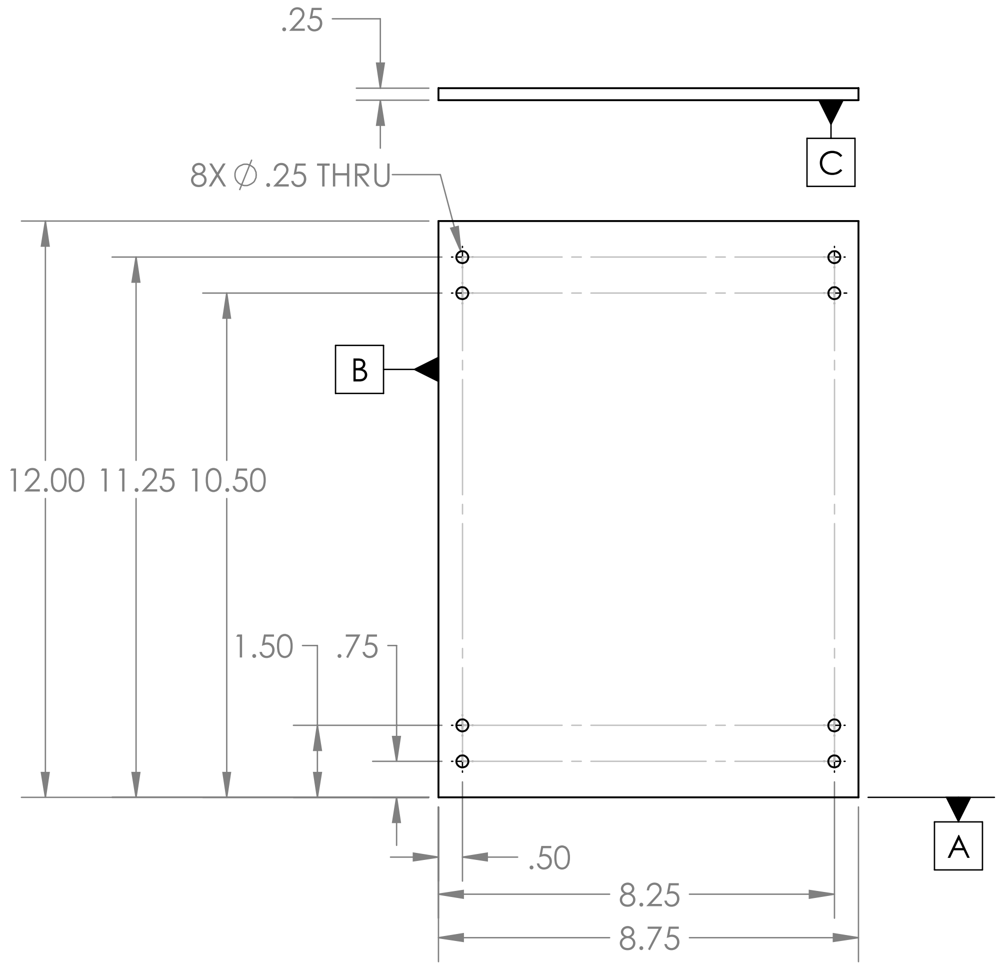
HOUSING TOP/BOTTOM

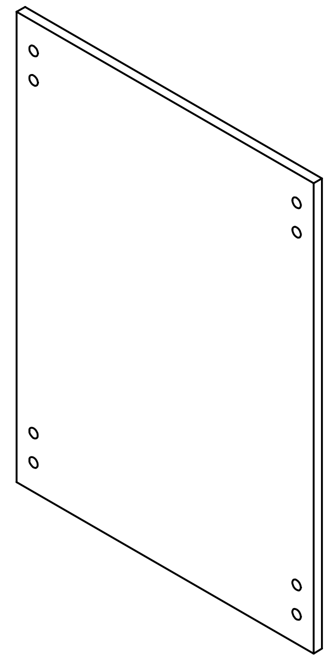
SIZE	DWG. NO.	REV
A	312	A

SCALE: 1:8 | WEIGHT: 4.33 | SHEET 1 OF 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/32 ANGULAR: ± 1° TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	DRAWN	SEID	2-9-17	
			CHECKED	MOORE	2-9-17	TITLE:
			ENG APPR.			CORNER BLOCK
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL 6160 AL	COMMENTS: WEIGHT IN LBF			SIZE A / DWG. NO. 313 / REV A
NEXT ASSY	USED ON	FINISH MILL PASS/MACHINE SMOOTH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:1 / WEIGHT: 0.19 / SHEET 1 OF 1





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NEXT ASSY	USED ON: HOUSING ASSEMBLY
APPLICATION:	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR: ± 1°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T6

FINISH
NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	SEID	2-9-17
CHECKED	MOORE	2-9-17
ENG APPR.		
MFG APPR.		
Q.A.		

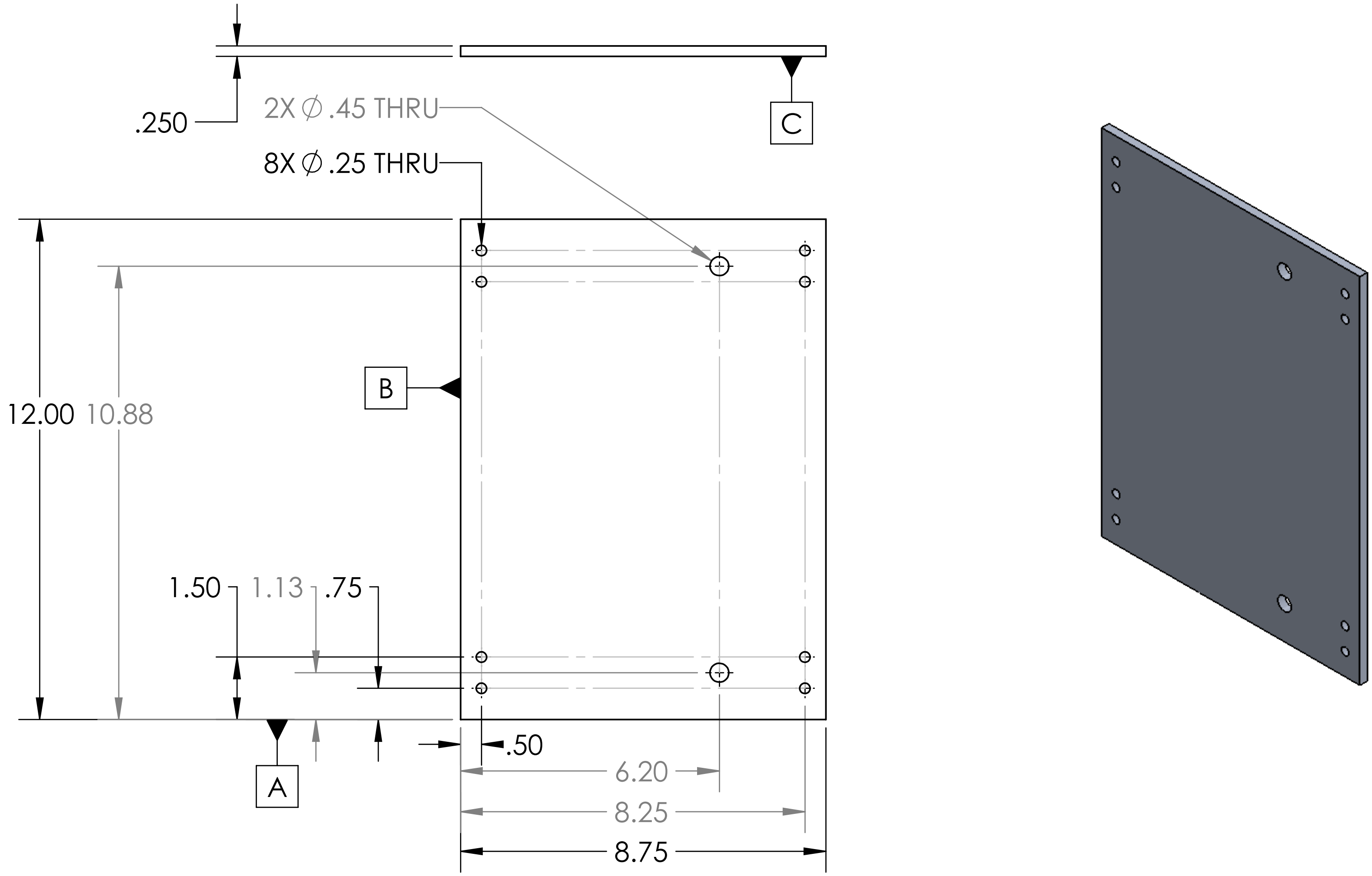
COMMENTS:
TOP PIECE WILL HAVE 2 OPTIONS WITH SAME SPECIFICATION. ONE ACRYLIC, ONE ALUMINUM
Weight IN LBF
LOCATE OIL SEAL OFF OF EDGE CLOSEST TO PINION

TITLE:

HOUSING PLATE FRONT

SIZE	DWG. NO.	REV
A	314	A

SCALE: 1:4 | WEIGHT: 4.34 | SHEET 1 OF 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 1/32 ANGULAR: ± 1° TWO PLACE DECIMAL ±.1 THREE PLACE DECIMAL ±.005	DRAWN	SEID	2-7-17
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL AL 6061-T6	COMMENTS: WEIGHT IN LBF DEBURR THROUGH HOLES		
NEXT ASSY	USED ON: Housing Assembly	FINISH: NONE			
APPLICATION:		DO NOT SCALE DRAWING			

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TITLE:

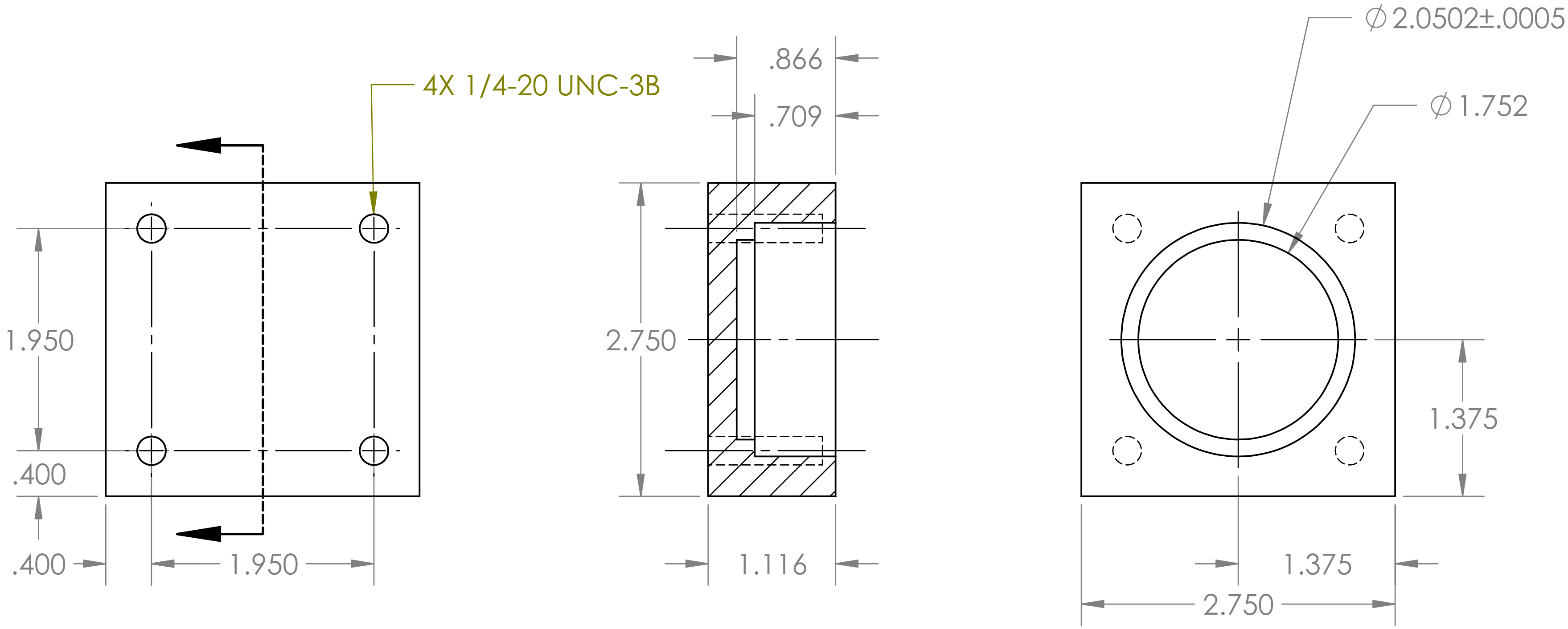
HOUSING BACK

SIZE A | DWG. NO. 315 | REV A

SCALE: 1:4 | WEIGHT: 2.55 | SHEET 1 OF 1

2 1

B B



A A

PROPRIETARY AND CONFIDENTIAL

NEXT ASSY | USED ON: HOUSING ASSEMBLY

APPLICATION:

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR: ± 1°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T6

FINISH
NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	KING	2-9-17
CHECKED	MOORE	2-9-17
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:
TOP PIECE WILL HAVE 2 OPTIONS WITH SAME SPECIFICATION. ONE ACRYLIC, ONE ALUMINUM
Weight IN LBF
LOCATE OIL SEAL OFF OF EDGE CLOSEST TO PINION

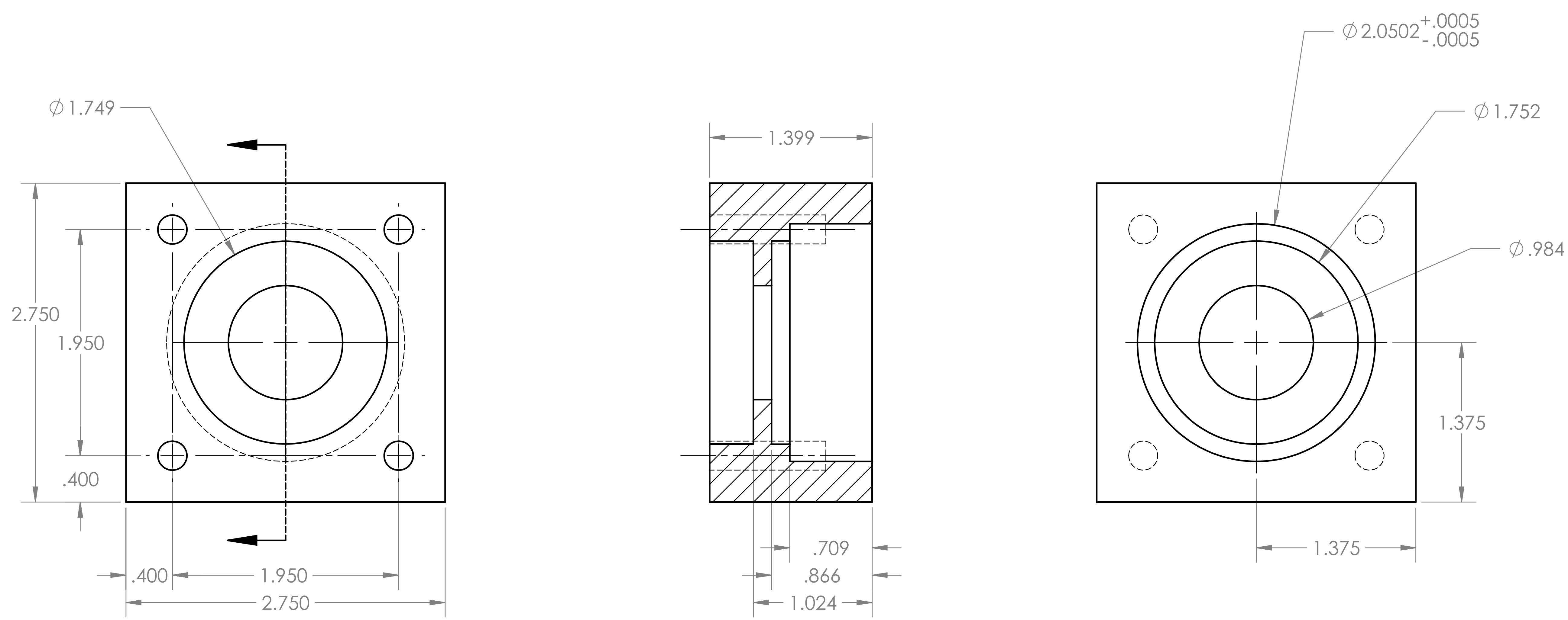
TITLE:

INPUT & INT MOUNT NO SEAL

SIZE **A** | DWG. NO. 401 | REV **A**

SCALE: 1:2 | WEIGHT: 4.34 | SHEET 1 OF 1

2 1



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TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009

MATERIAL: 6061 AL

FINISH: NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JIMMY KING	4/3/17
CHECKED	KEVIN MOORE	4/4/17
ENG APPR.		
MFG APPR.		
Q.A.		

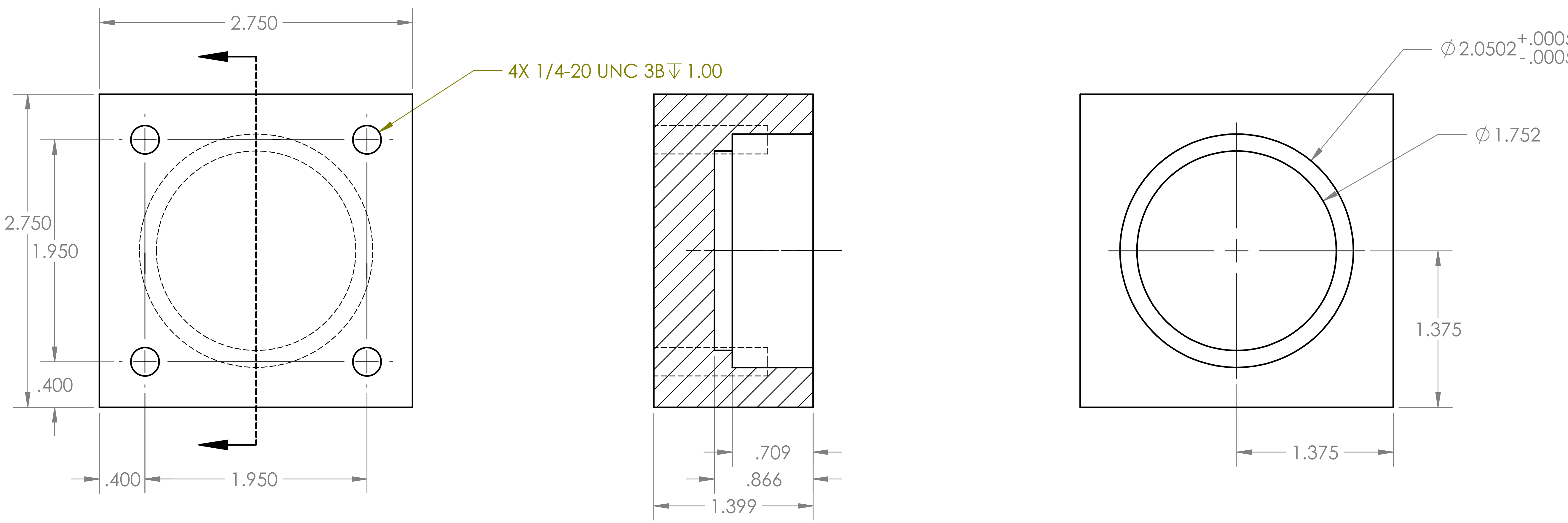
COMMENTS:

TITLE:

INPUT BEARING MOUNT W/ SEAL

SIZE	DWG. NO.	REV
B	402	A

SCALE: 1:1 WEIGHT: SHEET 1 OF 1



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UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009

MATERIAL: 6061 AL

FINISH: NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JIMMY KING	4/3/17
CHECKED	KEVIN MOORE	4/4/17
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

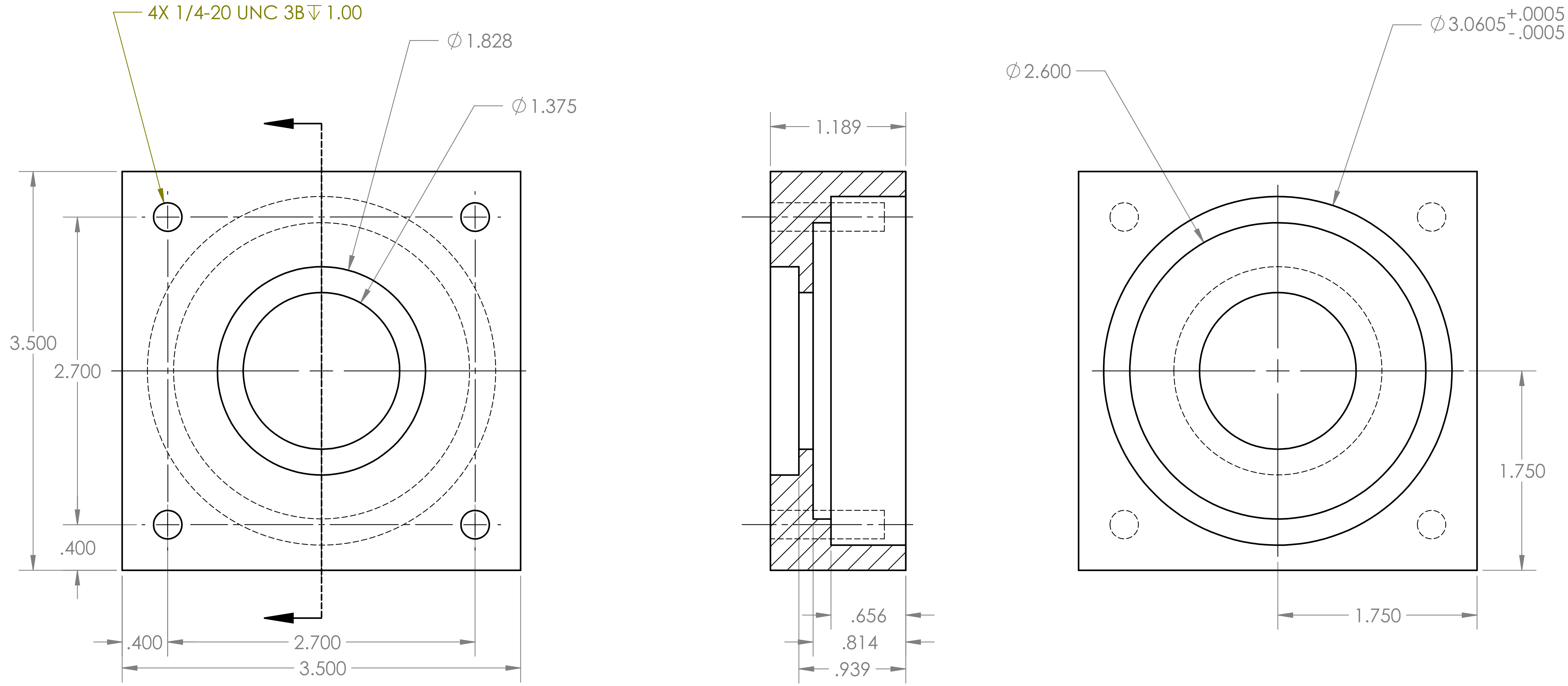
INT. BEARING MOUNT SEAL SIDE

SIZE	DWG. NO.	REV
B	403	A

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

4 3 2 1

B B



A A

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/32
ANGULAR ± 1°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5-2009

MATERIAL 6061 AL

FINISH NONE

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JIMMY KING	4/3/17
CHECKED	KEVIN MOORE	4/4/17
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
OUTPUT BEARING MOUNT

SIZE	DWG. NO.	REV
B	404	A

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

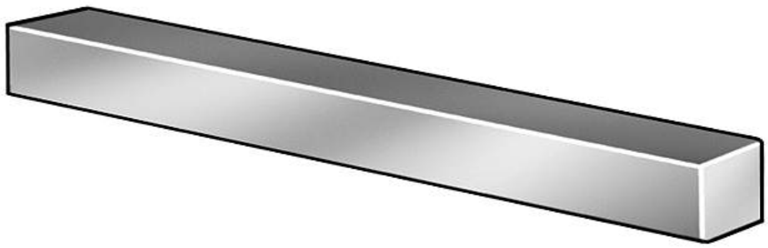
4 3 2 1

E.20

Table of Contents

Grainger

-12" Low Carbon Steel Undersized Key Stock with Zinc Finish 3/8 X 3/8



PRODUCT DETAILS

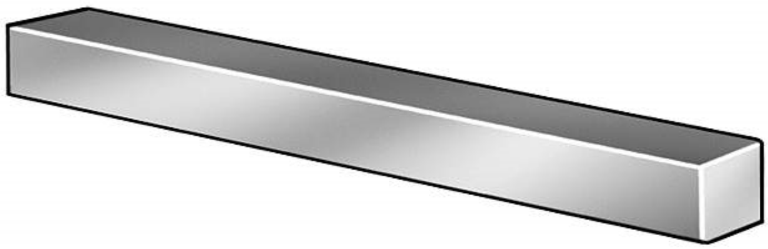
Form, forge, or machine a key with this undersized key stock. It is marginally smaller than the stated size for a looser fit. It does not require filing. It features low carbon steel construction with a zinc finish to resist corrosion.

TECHNICAL SPECS

Item	**Key Stock**	Thickness Tolerance	**-0.002"**
Type	**Undersized**	Length Tolerance	**-0.125"**
Material	**Low Carbon Steel**	Tensile Strength (PSI)	**64,000**
Finish	**Zinc**	Rockwell Hardness	**B85**
Length	**12"**	Features	**Magnetic**
Size	**3/8 x 3/8"**	Application	**Make Your Own Key**

Grainger

-12" Low Carbon Steel Undersized Key Stock with Zinc Finish ¼ X ¼



PRODUCT DETAILS

Stock is marginally smaller than the stated size, for a looser fit; does not usually require filing. Can be cut to size.

TECHNICAL SPECS

Item	**Key Stock**	Thickness Tolerance	**-0.002"**
Type	**Undersized**	Length Tolerance	**-0.125"**
Material	**Low Carbon Steel**	Tensile Strength (PSI)	**64,000**
Finish	**Zinc**	Rockwell Hardness	**B85**
Length	**12"**	Features	**Magnetic**
Size	**1/4 x 1/4"**	Application	**Make Your Own Key**

Grainger

-Retaining Ring, Ext, Dia 1 9/16 In

TECHNICAL SPECS

Item	**Heavy Duty Retaining Ring**	Fits Groove Width	**0.120"**
Type	**External**	Groove Width Tolerance	**+0.005"**
Material	**Carbon Steel**	Fits Groove Depth	**0.047"**
Finish	**Black Phosphate**	Clearance Dia. (Expanded)	**2.140"**
For Shaft Dia.	**1-9/16"**	Clearance Dia. (Released)	**2.050"**
Free I.D.	**1.437"**	Hole Dia.	**0.109"**
Thickness	**0.109"**	Tensile Strength (PSI)	**150,000**
Thickness Tolerance	**+/-0.003"**	Rockwell Hardness	**RC47-52**
Fits Groove Dia.	**1.468"**	Meets/Exceeds	**ASME**
Groove Dia. Tolerance	**+0.003/-0.004"**	Package Quantity	**5**

Grainger

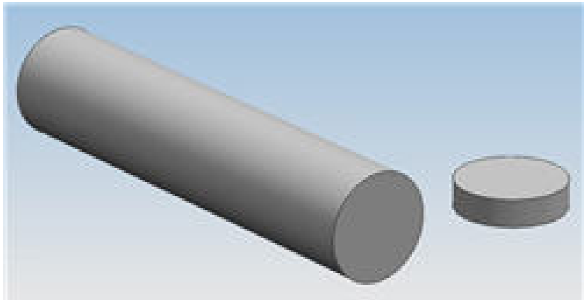
-Retaining Ring, Ext, Dia 1 1/4 In

TECHNICAL SPECS

Item	Standard Retaining Ring
Type	External
Material	Stainless Steel
Finish	Plain
For Shaft Dia.	1-1/4"
Free I.D.	1.156"
Thickness	0.050"
Thickness Tolerance	+/-0.002"
Fits Groove Dia.	1.176"
Groove Dia. Tolerance	+/-0.004"
Fits Groove Width	0.056"
Groove Width Tolerance	+0.004"
Fits Groove Depth	0.037"
Clearance Dia. (Expanded)	1.690"
Clearance Dia. (Released)	1.620"
Hole Dia.	0.078"
Tensile Strength (PSI)	150,000
Rockwell Hardness	RC44-51
Meets/Exceeds	ASME
Package Quantity	1

Speedy Metals

-1-3/4" Rd 4140 Hot Rolled, Heat Treated



ANALYSIS

Carbon (C)	Manganese (Mn)	Silicon (Si)	Chromium (Cr)	Molybdenum (Mo)	Phosphorus (P) Max	Sulfur (S) Max
0.4-0.45	0.7-1	0.205-0.35	0.8-1.1	0.15-0.25	0.04	0.04

4140 cold roll bars conforms to ASTM A-193 Grade B7, ASTM A434 Grades BC and BD

MECHANICAL PROPERTIES

Tensile Strength (PSI)	Yield Strength (PSI)	Reduction of Area	Elongation in 2"	Brinell Hardness
140,000	90,000	45	20	285

The above values are average and may be considered as representative of 4140 HRHT

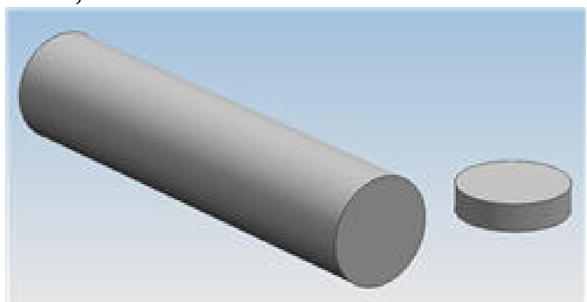
TOLERANCES

Rounds

Size	Over	Under	Out of Round
To 5/16" incl	.005	.005	.008
Over 5/16 to 7/16" incl	.006	.006	.009
Over 7/16 to 5/8" incl	.007	.007	.010
Over 5/8 to 7/8" incl	.008	.008	.013
Over 7/8 to 1-1/8" incl	.010	.010	.015
Over 1-1/8 to 1-1/4" incl	.011	.011	.016
Over 1-1/4 to 1-3/8" incl	.012	.012	.018
Over 1-3/8 to 1-1/2" incl	.014	.014	.021
Over 1-1/2 to 2" incl	.015	.015	.023
Over 2 to 2-1/2" incl	.031	-	.023
Over 2-1/2 to 3-1/2" incl	.047	-	.035
Over 3-1/2 to 4-1/2" incl	.063	-	.046
Over 4-1/2 to 5-1/2" incl	.078	-	.058
Over 5-1/2 to 6-1/2" incl	.125	-	.070
Over 6-1/2 to 8-1/2" incl	.156	-	.085
Over 8-1/2 to 9-1/2" incl	.188	-	.100
Over 9-1/2 to 10" incl	.250	-	.120

Speedy Metals

-2” Rd 4140 Hot Rolled, Heat Treated



ANALYSIS

Carbon (C)	Manganese (Mn)	Silicon (Si)	Chromium (Cr)	Molybdenum (Mo)	Phosphorus (P) Max	Sulfur (S) Max
0.4-0.45	0.7-1	0.205-0.35	0.8-1.1	0.15-0.25	0.04	0.04

4140 cold roll bars conforms to ASTM A-193 Grade B7, ASTM A434 Grades BC and BD

MECHANICAL PROPERTIES

Tensile Strength (PSI)	Yield Strength (PSI)	Reduction of Area	Elongation in 2"	Brinell Hardness
140,000	90,000	45	20	285

The above values are average and may be considered as representative of 4140 HRHT

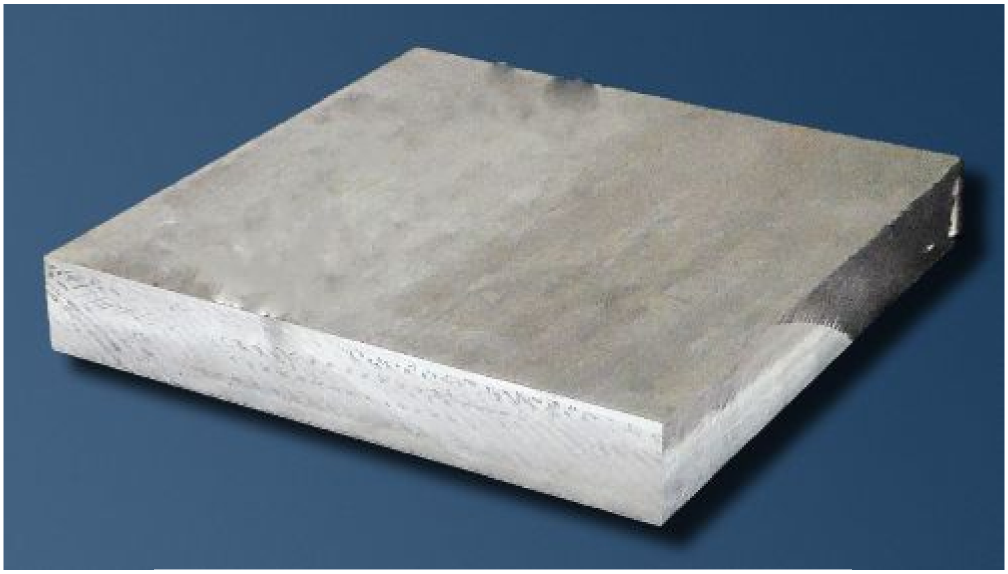
TOLERANCES

Rounds

Size	Over	Under	Out of Round
To 5/16" incl	.005	.005	.008
Over 5/16 to 7/16" incl	.006	.006	.009
Over 7/16 to 5/8" incl	.007	.007	.010
Over 5/8 to 7/8" incl	.008	.008	.013
Over 7/8 to 1-1/8" incl	.010	.010	.015
Over 1-1/8 to 1-1/4" incl	.011	.011	.016
Over 1-1/4 to 1-3/8" incl	.012	.012	.018
Over 1-3/8 to 1-1/2" incl	.014	.014	.021
Over 1-1/2 to 2" incl	.015	.015	.023
Over 2 to 2-1/2" incl	.031	-	.023
Over 2-1/2 to 3-1/2" incl	.047	-	.035
Over 3-1/2 to 4-1/2" incl	.063	-	.046
Over 4-1/2 to 5-1/2" incl	.078	-	.058
Over 5-1/2 to 6-1/2" incl	.125	-	.070
Over 6-1/2 to 8-1/2" incl	.156	-	.085
Over 8-1/2 to 9-1/2" incl	.188	-	.100
Over 9-1/2 to 10" incl	.250	-	.120

Midwest Steel and Aluminum

-6061-T651 Aluminum Plate



6061-T651 Aluminum plate has a tensile strength range of **42-45 KSI**, and conforms to **AMS QQ-A-250/11, ASME SB209,** and **ASTM B209**

Mill Thickness Tolerance ranges from -0 to +.014/0.130 depending on thickness. Call for specific tolerance if needed.

6061 COMPOSITION	
ALUMINUM	& alloying elements:
SILICON	0.4-0.8%
MAGNESIUM	0.8-1.2%
IRON	0.7% (max)
ZINC	0.25% (max)
COPPER	0.15%-0.4%
MANGANESE	0.15% (max)
TITANIUM	0.15% (max)
CHROMIUM	0.04-0.35%
OTHER:	0.05% (max) per Other total: 0.15%

6061-T651 CHARACTERISTICS	
CORROSION RESISTANCE	B
STRESS-CORROSION CRACKING RESISTANCE	A
WORKABILITY	C
MACHINABILITY	C
BRAZEABILITY	A
WELDABILITY: GAS	A
WELDABILITY: ARC	A
WELDABILITY: RESISTANCE SPOT & SEAM	A

Home Depot

-Optix 36 in. X 48 in. X .093 in. Acrylic Sheet

Dimensions

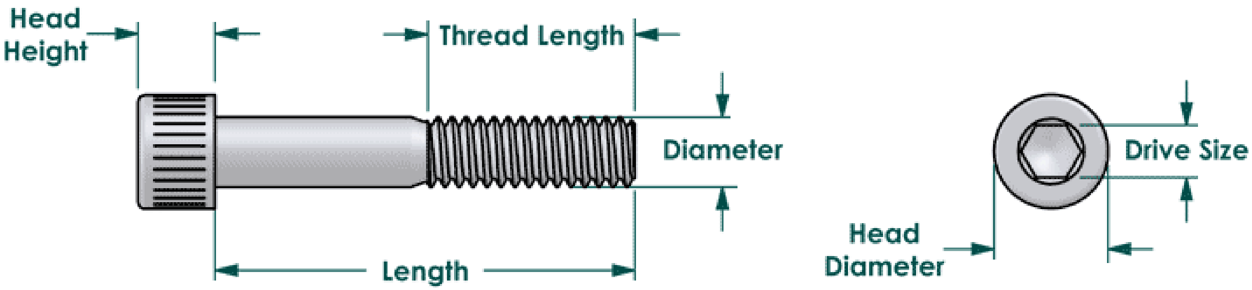
Product Depth (in.)	36	Product Thickness (in.)	0.093
Product Height (in.)	0.093	Product Width (in.)	36
Product Length (in.)	48		

Details

Bullet Proof	No	Returnable	90-Day
Mirrored	No	Shatter Resistant	No
Non-Glare	No	Surface Type	Clear
Product Weight (lb.)	8.2lb		

Bolt Depot

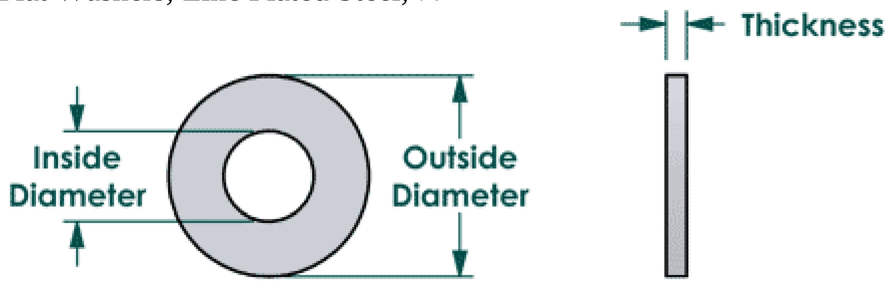
-Socket Cap, Alloy Steel Black Oxide Finish, ¼”-20 X 1/1/2”



Product details	
Bolt Depot Product #:	5029
Units:	US
Category:	Socket products
Dimensional standard:	ASME B18.3
Head style:	Socket cap
Drive type:	Allen
Material:	Steel
Grade:	ASTM A574
Finish:	Black oxide
Thread direction:	Right hand
Thread density:	Coarse
Diameter:	1/4"
Thread count:	20
Length:	1-1/2"
Head diameter Max:	0.375"
Head diameter Min:	0.365"
Head height Max:	0.375"
Head height Min:	0.244"
Drive size:	3/16"
Fully threaded:	No
Thread length Min:	1"
Body length Min:	0.25"

Bolt Depot

-SAE Flat Washers, Zinc Plated Steel, ¼”

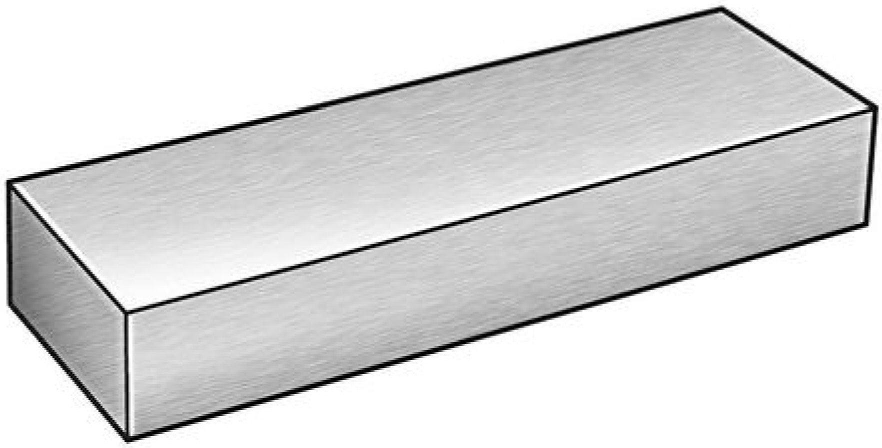


Product details

Bolt Depot Product #:	2976
Units:	US
Category:	Washers
Inside diameter:	9/32"
Subcategory:	SAE flat washers
Material:	Steel
Plating:	Zinc
Size:	1/4"
Outside diameter:	5/8"
Thickness Min:	0.051"
Thickness Max:	0.080"

Zoro

-Flat Stock, Al, 6061, 1 X 2 In, 6Ft



Technical Specifications	
Zoro #: G3495283 | **Mfr #:** 2EZX8	
Standards: ASTM B221, AMS-QQ-A-200	**Temper**: Extruded
Finish: Unpolished	**Material**: Aluminum
Item: Flat Stock	**Typical Yield Strength (PSI)**: 40, 000
Type: Corrosion Resistant	**Typical Tensile Strength (PSI)**: 45, 000
Alloy Type: 6061	**Hardness Scale**: BHN
Typical Hardness: 95	**Width Tolerance**: +/-0.024"
Thickness Tolerance: +/-0.012"	**Length Tolerance**: +/-6.000"
Thickness: 1.000"	**Width**: 2"
Length: 72"	**Country of Origin (subject to change)**: United States

EmedCo

- Loctite - 242® Threadlocker, Medium Strength

Base Type	Acrylic
Color	Blue
Container Size	10 ml
Container Type	Bottle
Cure Time	24h
Curing Method	Anaerobic
Curing Temperature	77°f
Standard	ABS / CFIA / MIL-S-46163A / NSF P1 / NSF/ANSI 61
Maximum Application Temperature	300°f
Maximum Viscosity	5000 cP
Minimum Application Temperature	-65°f
Minimum Viscosity	1200 cP
Item Type	Thread and Flange Sealant
Resistance Properties	Acid-Resistant / Chemical-Resistant
Manufacturer Name	Loctite
Manufacturer Part Number	24221

autohausAZ

- Loctite - RTV 5920 Copper Silicone

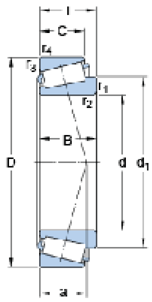
Premium silicone for 4-cylinder, turbocharged or high-performance engines. Sensor-safe, low odor, non corrosive, low volatility, non-conductive. Superior adhesion and oil resistance. Temperature range -75 degree F to 700 degree F intermittent; resists auto and shop fluids and vibration. **Suggested Applications:** Exhaust manifolds/headers, valve covers, oil pans, timing covers, water pumps, thermostat housings.

SKF

- 32305 J2 Tapered Roller Bearing

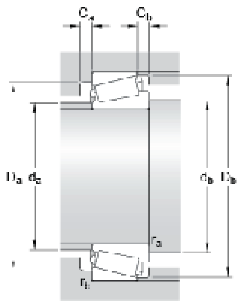
32305 J2

Dimensions



d		25	mm
D		62	mm
T		25.25	mm
d_1	≈	41.7	mm
B		24	mm
C		20	mm
$r_{1,2}$	min.	1.5	mm
$r_{3,4}$	min.	1.5	mm
a		15	mm

Abutment dimensions



d_a	max.	33	mm
d_b	min.	33	mm
D_a	min.	52	mm
D_a	max.	55	mm
D_b	min.	57	mm
C_a	min.	3	mm
C_b	min.	5	mm
r_a	max.	1.5	mm
r_b	max.	1.5	mm

Calculation data

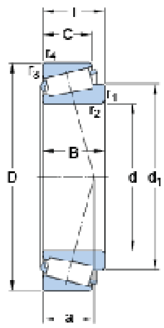
Basic dynamic load rating	C	60.5	kN
Basic static load rating	C_0	63	kN
Fatigue load limit	P_u	7.1	kN
Reference speed		8000	r/min
Limiting speed		12000	r/min
Calculation factor	e	0.3	
Calculation factor	Y	2	
Calculation factor	Y_0	1.1	

SKF

- 33206/Q Tapered Roller Bearing

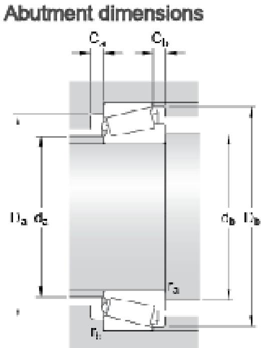
33206/Q

Dimensions



d		30	mm
D		62	mm
T		25	mm
d_1	≈	45.85	mm
B		25	mm
C		19.5	mm
$r_{1,2}$	min.	1	mm
$r_{3,4}$	min.	1	mm
a		15.747	mm

Abutment dimensions



d_a	max.	36	mm
d_b	min.	36	mm
D_a	min.	53	mm
D_a	max.	56	mm
D_b	min.	59	mm
C_a	min.	5	mm
C_b	min.	5.5	mm
r_a	max.	1	mm
r_b	max.	1	mm

Calculation data

Basic dynamic load rating	C	64.4	kN
Basic static load rating	C_0	76.5	kN
Fatigue load limit	P_u	8.5	kN
Reference speed		7500	r/min
Limiting speed		11000	r/min
Calculation factor	e	0.35	
Calculation factor	Y	1.7	
Calculation factor	Y_0	0.9	

E-Drive Budget

Part #	Part Name	Description / Location	Source	# pcs	Pc $	Tax $	Ship $	Total $	How ordered?	Date Ordered	Date Received	Reimb. Submitted	Where Stored?
ST500-00	Drivetrain Assembly					$ -		$ -					
ST501-00	Input Shaft Assembly					$ -		$ -					
ST501-01	Input Shaft (1 3/4 X 18 in.)	Drivetrain Input Shaft off the motor	Speedy Metals	1	$ 38.60		$ 19.54	$ 58.14	Sponsor Credit Card	3/16	4/10		Bonderson Cage
ST501-02	RH Cut, 2.5" Helical Gear	Pinion Gear	Rush Gears	1	$ 1,068.75			$ 1,068.75	Sponsor Credit Card	3/1	4/7		Bonderson Cage
ST501-03	Retaining Ring (1 pc.)	Holds Pinion Gear in place	Grainger	1	$ 4.39	$ 0.52	$ 2.62	$ 7.53	Sponsor Credit Card	3/16	4/6		Bonderson Cage
ST501-04	Key Way Stock (12 in.)	Keyway to mate pinion and input shaft	Grainger	1	$ 2.60	$ 0.52	$ 2.62	$ 5.74	Sponsor Credit Card	3/16	4/6		Bonderson Cage
ST501-05	Tapered Roller Bearing - SKF 33205/Q	Carries axial load from pinion gear	123Bearing	2	$ 22.96	$ 3.67	$ 2.39	$ 51.98	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST501-06/07	Stock Aluminum Block 2.75" X 2.75" X 6"	Input and intermediate shaft bearing mount	Metals Depot	2	$ 21.69		$ -	$ 43.38	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST501-08	Input Shaft Seal - SKF 9803	Seal the input shaft and keep debri out of the housing	KSC direct	1	$ 20.25	$ -	$ 4.69	$ 24.94	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST502-00	Intermediate Shaft Assembly					$ -		$ -					
ST502-01	Intermediate Shaft (2 X 12 in.)	Shaft to support intermediate gears	Speedy Metals	1	$ 32.70		$ 19.54	$ 52.24	Sponsor Credit Card	3/16	4/10		Bonderson Cage
ST502-02	LH Cut, 2.83" Helical Gear	Secondary Gear	Rush Gears	1	$ 1,116.25			$ 1,116.25	Sponsor Credit Card	3/1	4/7		Bonderson Cage
ST502-03	LH Cut, 6" Helical Gear	Primary Gear	Rush Gears	1	$ 1,306.25			$ 1,306.25	Sponsor Credit Card	3/1	4/7		Bonderson Cage
ST502-04	Retaining Ring (5 pcs.)	Holds intermediate gears in place on unshouldered side	Grainger	1	$ 17.01	$ 0.52	$ 2.62	$ 20.15	Sponsor Credit Card	3/16	4/6		Bonderson Cage
ST502-05	Key Way Stock (12 in.)	Keyway to mate intermediate gears and intermediate shaft	Grainger	1	$ 3.05	$ 0.52	$ 2.62	$ 6.19	Sponsor Credit Card	3/16	4/6		Bonderson Cage
ST502-06	Tapered Roller Bearing - SKF 33205/Q	Carries axial load from intermediate gears	123Bearing	2	$ 22.96	$ 3.67	$ 3.05	$ 52.64	Sponsor Credit Card	3/16	3/24		Bonderson Cage
T501-06/ST502-	Stock Aluminum Block 2.75" X 2.75" X 6"	Input and intermediate shaft bearing mount	Metals Depot	2	$ 21.69		$ -	$ 43.38	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST503-00	Differential Assembly					$ -		$ -					
ST503-01	31 Spline Gear Rebuild Set	IRS parts for differential	EastCoastGearSupply	1	$ 145.00	$ 11.60		$ 156.60	Sponsor Credit Card	4/13			Bonderson Cage
ST503-02	Ford 8.8 Carrier Bearing	Interchange parts on current LSD differential	EastCoastGearSupply	2	$ 24.00			$ 48.00	Sponsor Credit Card	3/16	4/7		Bonderson Cage
ST503-03	RH Cut, 9.5" Helical Ring Gear	Helical Ring gear on differential	Rush Gears	1	$ 1,591.25			$ 1,591.25	Sponsor Credit Card	3/1	4/7		Bonderson Cage
ST503-04	7/16"-20 x 1-3/4" Grade 8 Ring Gear Mounting Bolts	Bolt ring gear onto differential according to the current bolt pattern	Fastenal	10	$ 0.46	$ 0.24		$ 4.84	Teammember Credit Card	5/1	5/1		Bonderson Cage
ST503-05	31 Spline Axle Shaft	Output shafts from the differential	4 Wheel Parts	2	$ 112.99	$ 18.08		$ 244.06	Sponsor Credit Card	4/13	5/16		Bonderson Cage
ST503-06	Stock Aluminum Block 3.5" X 3.5" X 6"	Differential bearing mount	Metals Depot	2	$ 42.67			$ 68.12	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST503-07	7/16"-20 Grade 8 Ring Gear Mounting Nuts	Nuts to hold bolts to ring gear on diffferential	Fastenal	10	$ 0.14	$ 0.24		$ 1.64	Teammember Credit Card	5/1	5/1		Bonderson Cage
ST504-00	Housing Assembly							$ -					
ST504-01	Plates (0.25x12x19.25)	Left side plate for the housing	MidWest Steel And Aluminum	1	$20.67		$8.14	$ 28.81	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-02	Plates (0.25x12x19.25)	Right side plate for the housing	MidWest Steel And Aluminum	1	$20.67		$8.14	$ 28.81	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-03	Plates (0.25x9.25x19.25)	Top plate for the housing	MidWest Steel And Aluminum	1	$17.17		$8.14	$ 25.31	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-04	Plates (0.25x9.25x19.25)	Bottom plate for the housing	MidWest Steel And Aluminum	1	$17.17		$8.14	$ 25.31	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-05	Plates (0.25x8.75x12)	Back end plate for the housing	MidWest Steel And Aluminum	1	$11.84		$8.14	$ 19.98	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-06	Plates (0.25x8.75x12)	Front end plate for the housing	MidWest Steel And Aluminum	1	$11.84		$8.14	$ 19.98	Sponsor Credit Card	3/16	3/30		Bonderson Cage
ST504-07	Sheet	Clear Acrylic top for housing display	Tap Plastics	1	$ 10.28	$ 0.82	$ 10.28	$ 21.38	Sponsor Credit Card	4/23	5/16		Bonderson Cage
ST504-08	Flat Stock, Steel, 4140	Corner blocks and re-enforcement on housing	Zoro	1	$73.61	$ 5.34		$ 78.95	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST504-09	Output Shaft Seals - SKF 13510	Seal oil in the housing	KSC direct	2	$ 15.71	$ -	$ 4.69	$ 36.11	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST504-10	Housing Bolts [30]	Bolts the housing together	Bolt Depot	1	$ 4.06	$ 0.32	5.975	$ 10.36	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST504-11	Oil Drain Plug (1/2-20 Thread)	On the bottom of the housing to drain out oil	AutoZone - In Store	1	$ 4.99	$ 0.40		$ 5.39	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST504-12	Oil Fill Plug (1/2-20 Thread)	On top portion of the housing to fill oil	AutoZone - In Store	1	$ 4.99	$ 0.40		$ 5.39	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST504-13	Permatex ThreadLocker	To seal in all the screws used in the housing	AutoZone - In Store	1	$ 22.99	$ 1.84		$ 24.83	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST504-14	Permatex Gasket Sealant	To seal all the seam lines between housing plates	AutoZone - In Store	2	$ 7.49	$ 0.53		$ 15.51	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST504-15	Flat Stock, Steel, 4140	Motor mounting strip	Zoro	3	$73.61	$ 16.01		$ 236.84	Sponsor Credit Card	3/16	3/30		Bonderson Cage
Misc. Parts													
ST505-01	Motor and Input Shaft Coupler	To couple the output shaft of the motor to the input shaft with the pinion	Vaxo Systems	1	$ 106.00	$ 8.48	$ 10.48	$ 124.96	Sponsor Credit Card	4/13	4/25		Bonderson Cage
ST505-02	Delo Syn-Gear XDM SAE 75W-90 Gearbox Oil	Oil bath for house	AutoZone - In Store	9	$ 5.99	$ 4.18		$ 58.09	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST505-03	Mounting Block Bolts	1/4-20 x 1-1/8 in Stainless Steel Hex Bolt	Bolt Depot	1	$ 19.08	$ 1.53	5.975	$ 26.58	Sponsor Credit Card	3/16	3/24		Bonderson Cage
ST505-04	Multi-Purpose Funnel	To fill housing with oil	AutoZone - In Store	1	$ 1.39	$ 0.11		$ 1.50	Teammember Credit Card	4/15	4/15		Bonderson Cage
ST505-05	Quiksteel Epoxy Compound	Filling extra through holes	AutoZone - In Store	1	$ 6.39	$ 0.51		$ 6.90	Teammember Credit Card	4/28	4/28		Bonderson Cage
ST505-06	Carbide Drill Bit	To drill through helical ring gear	One Way Industrial Supply - In Store	1	$ 50.17	$ 4.35	$ 6.00	$ 60.52	Teammember Credit Card	4/26	4/26		Bonderson Cage
ST505-07	Plywood Round	Bolt pattern template for ring gear manufacturing	Ace Hardware	1	$ 9.99	$ 0.87		$ 10.86	Teammember Credit Card	4/25	4/25		Bonderson Cage
ST505-08	Paint Thinner	Remove sharpie and marks from housing plate	AutoZone - In Store	1	$ 8.49	$ 0.66		$ 9.15	Teammember Credit Card	4/27	4/27		Bonderson Cage
ST505-09	Plastic Spreader	To spread the gasket sealant on the housing plates	AutoZone - In Store	1	$ 3.99	$ 0.31		$ 4.30	Teammember Credit Card	4/15	4/15		Bonderson Cage

Total Cost: $ 6,764.62

Attn: CHARISSA SEID Quotation# 674690

CALL: 800-523-2576
FAX: 800-635-6273

RUSH GEARS inc.

550 Virginia Drive
Fort Washington, PA 19034

TERMS & ORDER FORM

Cancellations: >>> We do not allow cancellations after an order has been placed. <<<

In order to offer our Rush Deliveries, we begin work immediately upon receipt of your order. Within minutes material is procured by purchasing, manufacturing drawings are created by engineering and manufacturing time is reserved by production for each operation. In most cases we are machining your gears within hours of receiving your order.

Distributors: Please make sure your customer is aware of this policy before accepting an order.

Returns: We do not allow returns on either standard or special items, unless we are in error.

All custom items are Made-To-Order and standard catalog items are Finished-To-Order, therefore we can not accept returns. If parts are wrong or defective, we will replace at no charge.

Distributors: Please make sure your customer is aware of this policy before accepting an order.

Samples supplied by Customer: These terms apply when drawings are not available and we are making new Gear(s) from dimensions taken from original gear(s).

We will make the best attempt to determine the approximate original specifications/properties of an original gear. We have no way of determining what the exact original material/specifications were or what the exact original manufacturing/heat treating processes were. It is the responsibility of the customer to make us aware in writing of dimensions or tolerances that are critical to their application.

We are offering a substitute product, not an original replacement. It is up to the customer to determine whether or not our substituted gear(s) will work in his specific application. If alterations are required for proper fit, we will make them at no additional charge. We can not be held responsible for costs of down time if alterations need to be made.

Distributors: Please make sure your customer is aware of this policy before accepting an order.

Application / Warranty: Rush Gears products are intended for commercial, industrial uses, they are not designed for use in automotive or aircraft drive train units.

Rush Gears products are not warranted or recommended for any specific customer application. It is the customer's responsibility to calculate and determine the proper gear selection, horsepower ratings and safety factors for his specific application. All Rush Gears products are unconditionally guaranteed against manufacturing defects. Any item found to be defective will be replaced provided we are notified within 60 days of shipment. Our liability for defects shall not exceed its replacement cost to us.

Distributors: Please make sure your customer is aware of this policy before accepting an order.

Matrix Quote: If used correctly, our Matrix options can save you or your customer $$$$!!

Example Quote, (we will offer fastest mfg. time possible)

	Standard	Rush	Urgent	Critical	Crisis
Production Time Options (Applies to business days. Weekend / holiday service available)	2 weeks	1 week	48 hrs	24 hrs	Same Day
Rush Gear price for each Option to make (1) Gear	$500.00	$750.00	$1000.00	$1500.00	$3000.00
Your Cost of Down Time if you are losing only $1000.00/day	+$10000.00	+$5000.00	+$3000.00	+$2000.00	+$1000.00
Your total Cost to wait for each Option	=$10500.00	=$5750.00	=$4000.00	=$3500.00	=$4000.00

In this example, your least expensive option would be to order a 24 hour Production Time
(You should consider other effects of lost production besides only cost when making your decision, such as waiting customers, idle employees or the effects on future business or scheduling)
In addition, we recommend that you order backup gears at the Standard Production Time.

Order Form: *WE ACCEPT CREDIT CARDS!*

Circle the following information on our QUOTE and fax it back with this Order Form:

1. Part numbers you wish to order
2. Quantity of each (write in if not listed)
3. Production Time option for each

Ship Via:

- ☐ UPS GroundTrac
- ☐ UPS ORANGE 3rd Day Air
- ☐ UPS BLUE 2nd Day Air
- ☐ UPS RED Next Day Air
- ☐ UPS RED Early AM
- ☐ UPS RED Saturday delivery
- ☐ Federal Express Overnite
  FedEx acct#______________
- ☐ Other____________________

Ship To: (other than shown on quote)

Company____________________
Street____________________
City____________________
State__________Zip__________
Tag#____________________

Payment Type:

☐ Credit Card
Type____________________
Card#____________________
Exp Date____________________
Print Name____________________
Billing Address____________________
____
Signature____________________

☐ Bill me (fill in billing address above) (terms are 2% 10 days / 1% 30 days with approved credit)
PO#____________________

☐ C.O.D.
Your Ref#____________________

RUSH GEARS inc.
550 Virginia Dr, Fort Wash. PA 19034
PHONE: 800-523-2576 : FAX: 800-635-6273
www.rushgears.com : sales@rushgears.com

QUOTE# 674690
DATE: 03/01/17
ACCT#: 946091
INQUIRY#: PHONE
QUOTED BY: Stewart Mc Gann
TERMS: 5% Pre-Payment Discount

Rush To >>> **CHARISSA SEID**
SHARPELL TECHNOLOGIES
288 CRAIG WAY
SAN LUIS OBISPO CA 93405

WE ACCEPT CREDIT CARDS! MasterCard VISA

COMMENTS: New Pre-Pay with Order: Take an additional 5% off quoted prices!

Part# 329-2669

QTY	Net each	Total	>Standard< Prod. Time
1	$3350.00 ea	$3350.00	3 WEEKS
2	$1675.00 ea	$3350.00	3 WEEKS
4	$1507.50 ea	$6030.00	3 WEEKS
6	$1340.00 ea	$8040.00	3 WEEKS
10	$1172.50 ea	$11725.00	3 WEEKS

Description: MADE TO ORDER, 8620 CASE HARDENED (58-62 RC) STEEL, HELICAL GEAR, 6.000 DP, 20.0 PA, 57 HOBBED TEETH, 1.250" FACE, 5.125" BORE, A TYPE, 9.5000" PD , 9.789" OD:, 30 DEGREE, RIGHT HAND

Expedited Production Times: OEM prices (net each piece)

QTY	2 WEEKS	1-WEEK		
1	$5025.00 ea	$6700.00 ea		
2	$2512.50 ea	$3350.00 ea		
4	$2261.25 ea	$3015.00 ea		
6	$2010.00 ea	$2680.00 ea		
10	$1758.75 ea	$2345.00 ea		

Orders may not be Canceled (see Page#1 for full terms)

Part# 329-2678

QTY	Net each	Total	>Standard< Prod. Time
1	$2750.00 ea	$2750.00	3 WEEKS
2	$1375.00 ea	$2750.00	3 WEEKS
4	$1237.50 ea	$4950.00	3 WEEKS
6	$1100.00 ea	$6600.00	3 WEEKS
10	$962.50 ea	$9625.00	3 WEEKS

Description: MADE TO ORDER, 8620 CASE HARDENED (58-62 RC) STEEL, HELICAL GEAR, 6.000 DP, 20.0 PA, 36 HOBBED TEETH, 1.000" FACE, 1.563" BORE, A TYPE, 0.375 X .1875" KEYWAY, 6.0000" PD , 6.289" OD:, 30 DEGREE, LEFT HAND

Expedited Production Times: OEM prices (net each piece)

QTY	2 WEEKS	1-WEEK		
1	$4125.00 ea	$5500.00 ea		
2	$2062.50 ea	$2750.00 ea		
4	$1856.25 ea	$2475.00 ea		
6	$1650.00 ea	$2200.00 ea		
10	$1443.75 ea	$1925.00 ea		

Orders may not be Canceled (see Page#1 for full terms)

Part# 329-2684

QTY	Net each	Total	>Standard< Prod. Time
1	$2350.00 ea	$2350.00	3 WEEKS
2	$1175.00 ea	$2350.00	3 WEEKS
4	$1057.50 ea	$4230.00	3 WEEKS
6	$940.00 ea	$5640.00	3 WEEKS
10	$822.50 ea	$8225.00	3 WEEKS

Description: MADE TO ORDER, 8620 CASE HARDENED (58-62 RC) STEEL, HELICAL GEAR, 6.000 DP, 20.0 PA, 17 HOBBED TEETH, 1.250" FACE, 1.563" BORE, A TYPE, .375 X .1875" KEYWAY, 2.8333" PD , 3.122" OD:, 30 DEGREE, LEFT HAND

Expedited Production Times: OEM prices (net each piece)

QTY	2 WEEKS	1-WEEK		
1	$3525.00 ea	$4700.00 ea		
2	$1762.50 ea	$2350.00 ea		
4	$1586.25 ea	$2115.00 ea		
6	$1410.00 ea	$1880.00 ea		
10	$1233.75 ea	$1645.00 ea		

F.18

RUSH GEARS inc.

550 Virginia Dr, Fort Wash. PA 19034
PHONE: 800-523-2576 : FAX: 800-635-6273
www.rushgears.com : sales@rushgears.com

QUOTE# 674690
DATE: 03/01/17
ACCT#: 946091
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QUOTED BY: Stewart Mc Gann
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Rush To >>> **CHARISSA SEID**
SHARPELL TECHNOLOGIES
288 CRAIG WAY
SAN LUIS OBISPO CA 93405

WE ACCEPT CREDIT CARDS!

COMMENTS: New Pre-Pay with Order: Take an additional 5% off quoted prices!

Part# 329-2700

QTY	Net each	Total	>Standard< Prod. Time
1	$2250.00 ea	$2250.00	3 WEEKS
2	$1125.00 ea	$2250.00	3 WEEKS
4	$1012.50 ea	$4050.00	3 WEEKS
6	$900.00 ea	$5400.00	3 WEEKS
10	$787.50 ea	$7875.00	3 WEEKS

Description: MADE TO ORDER, 8620 CASE HARDENED (58-62 RC) STEEL, HELICAL GEAR, 6.000 DP, 20.0 PA, 15 HOBBED TEETH, 1.000" FACE, 1.250" BORE, A TYPE, .25 X .125" KEYWAY, 2.5000" PD , 2.789" OD:, 30 DEGREE, RIGHT HAND

Expedited Production Times: OEM prices (net each piece)

QTY	2 WEEKS	1-WEEK		
1	$3375.00 ea	$4500.00 ea		
2	$1687.50 ea	$2250.00 ea		
4	$1518.75 ea	$2025.00 ea		
6	$1350.00 ea	$1800.00 ea		
10	$1181.25 ea	$1575.00 ea		

Orders may not be Canceled (see Page#1 for full terms)

Part#

QTY	Net each	Total	>Standard< Prod. Time

Description:

Expedited Production Times: OEM prices (net each piece)

QTY				
1				
2				
4				
6				
10				

Orders may not be Canceled (see Page#1 for full terms)

Part#

QTY	Net each	Total	>Standard< Prod. Time

Description:

Expedited Production Times: OEM prices (net each piece)

QTY				
1				
2				
4				
6				
10				

Terms and Conditions

1. DEFINITIONS
a) SELLER: As used in this QUOTATION and or SALES ORDER, means Rush Gears Inc., Globe Transmission Corp., Globe Gears or any of its subsidiaries or divisions.
b) FURNISH COMPLETE: SELLER will furnish parts, gears, machinery or apparatus complete, including material, with all the machining, cutting, heat treat, and/or assembly operations being performed in accordance with specifications stated herein or on drawings.
c) MACHINING AND/OR CUTTING ONLY AND/OR HEAT TREAT ONLY AND/OR GEAR GRIND ONLY: Buyer furnishes to Seller material and/or machined blanks ready for the operations as specified herein only.

2. ACCEPTANCE, GOVERNING PROVISIONS, AND CANCELLATION
No orders for products or services of Seller shall be binding upon the Seller unless accepted in writing by an authorized official at its Home Office. Any such order shall be subject to these terms and conditions of sale and acceptance of an order by the Seller shall be expressly conditioned on assent to such terms and conditions. No modifications to these terms and conditions or other conditions will be recognized by Seller unless specifically agreed to in writing and failure of Seller to object to provisions contained in any purchase order or other communications from a Buyer shall not be construed as a waiver of these conditions or an acceptance of any such provisions. Receipt of Purchase Order from Buyer for products and services contained herein represents acceptance of these terms and conditions. Any contract for sale and these conditions and terms shall be governed by and construed according to the laws of the State of PENNSYLVANIA. **No order accepted by Seller may be altered or modified by the Buyer unless agreed to in writing by the Seller; and no such order may be canceled or terminated at any time after receipt of such order in writing by Seller. Order(s) placed by Buyer before 3:00pm ET are considered received the same business day. Order(s) placed by Buyer after 3:00pm ET are considered received the next business day.**

3. DELIVERY
Delivery shall be F.O.B. Seller's plant. Delivery of products to a carrier at Seller's plant or other shipping point shall constitute delivery to Buyer and title shall pass at that time, regardless of freight payment. All risks of loss or damage in transit shall be borne by Buyer. Delivery promises are based on Seller's best judgment and Seller will attempt to fill orders at the agreed time. However, Seller shall not be liable for any damage claimed to result from any delay in delivery due to any cause whatsoever. Delivery times do not include business holidays.

4. TERMS OF PAYMENT
All invoices are due and payable net thirty (30) days from date of invoice from Buyers with approved credit. Delays in transportation shall not extend terms of payment. Seller reserves the right to collect payment in part or in full as a condition of acceptance of an order from Buyer. Should the Buyer's financial responsibility become unsatisfactory to the Seller, cash payment or satisfactory security may be demanded by the Seller and in default of such cash payment or satisfactory security, deliveries herein may be discontinued at the option of the Seller and a charge rendered covering the value of any partially finished articles that are being manufactured on this order or contract. Seller retains all other remedies it may have as a result of Buyer's unsatisfactory financial responsibility.

5. TAXES AND OTHER CHARGES
Any manufacturer's tax, retailer's occupation tax, use tax, sales tax, excise tax, duty, custom, inspection or testing fee, or any other tax, fee or charge of any nature whatsoever, imposed by any governmental authority, on or measured by any transaction between Seller and the Buyer, shall be paid by the Buyer in addition to the prices quoted or invoiced.

6. WARRANTY
Seller's products are not warranted or recommended for any specific customer application. It is the Buyer's responsibility to calculate and determine the proper gear selection, horsepower ratings and safety factors for any specific application. Seller's products are intended for commercial industrial uses. They are not to be used in automotive, marine or aircraft drive train or propulsion systems. Seller warrants its products to be free from defects in materials and workmanship for a period of sixty days from date of shipment by Seller. If within such period any such product shall be proved to Seller's satisfaction to be so defective, such products shall be repaired or replaced at Seller's option. Seller's obligation upon such warranty shall be limited to such repair and replacement and shall be conditioned upon Seller's receiving written notice of any alleged defect within 10 days after its discovery, but not more than 60 days after receipt, and at Seller's option, return of such products or parts to Seller F.O.B. its factory. This warranty shall not apply to products or parts not manufactured by Seller or to products or parts which shall have been repaired or altered by others than Seller so as, in its judgment, adversely to affect the same, or which shall have been subject to negligence, accident, damaged by circumstances beyond Seller's control, or improper operation, maintenance or storage or to other than normal use of service. With respect to products and parts not manufactured by Seller, the warranty obligations of Seller shall in all respects conform and be limited to the warranty actually extended to Seller by the supplier. THE FOREGOING WARRANTY IS EXCLUSIVE AND IN LIEU OF ALL OTHER EXPRESS AND IMPLIED WARRANTIES WHATSOEVER, INCLUDING BUT NOT LIMITED TO IMPLIED WARRANTIES OF MERCHANTABILTY AND FITNESS FOR A PARTICULAR PURPOSE. Seller shall not be subject to any other obligations or liabilities whatsoever with respect to products or any undertakings, acts or omissions relating thereto. The standards of AGMA will be used, where applicable, in the manufacture of gears, unless an express agreement to the contrary is reached between Buyer and Seller.
Seller guarantee(s) that all goods and every part and ingredient thereof sold to Buyer are produced in accordance with the Fair Labor Standards Act of 1938 and all amendments thereto.

7. CLAIMS
Expenses incurred in connection with claims for which the Seller is not liable may be charged to the purchaser. No claim for correction will be allowed except for work done with the written consent of the Seller. Defects that do not impair service shall not be a cause for rejection. The Seller shall not be liable under any circumstances, and anything to the contrary herein contained notwithstanding, for any direct, indirect consequential, contingent or incidental damages whatsoever arising from or resulting from the failure or improper functioning of any of its products.
Claims for shortages or other errors must be made in writing to Seller within 10 days after receipt of shipment and failure to give such notice shall constitute unqualified acceptance and waiver of all such claims by purchaser.
The Buyer will defend, at his own expense, and hold Seller harmless against any suit that may be brought against Seller by reason of the manufacture or sale of parts made to the Buyer's specifications.
No claim will be allowed for material mutilated by the Buyer or damaged in transit.
Where the Buyer furnishes the material, and it proves defective or involves expense not contemplated by the contract, the Seller will invoice all expenses involved.
When work of any kind is performed by Seller on material supplied by the Buyer, Seller shall not be liable for any cost of the material or other damages in event of spoilage or rejection for whatsoever cause or reason. The Seller shall not be liable for loss of patterns, tooling, or merchandise by reason of circumstances beyond Seller's control.

8. ALTERATIONS
No alterations in specifications, either for total quantity, delivery, mechanical, chemical or other details may be made without written consent of an authorized official of Seller and readjustment of price.

9. PRICING POLICY
Prices quoted are for acceptance within 30 days. Prices are based on running the full quantity for shipment at one time and to one destination unless otherwise agreed to in writing.

10. ERRORS AND VARIANCES
All clerical errors in Seller's quotations, acknowledgments and invoices are subject to correction.

11. OVERRUNS -- UNDERRUNS
All quotations are based on customer accepting overruns or under runs, not exceeding 10% of quantity ordered, to be paid for or allowed pro rata.

12. PACKING
All prices listed provide for packing in accordance with the Company standard specifications.

13. DEVELOPMENT, DRAWING, PATTERN AND/OR TOOL CHARGES
Development, drawing, pattern and/or tool charges quoted in a proposal represent the Buyer's proportionate cost thereof and it is expressly understood that such drawings, patterns, and/or tools remain the property of the Seller, unless otherwise agreed in writing.

14. PATENTS, ETC.
Seller will have no responsibility whatsoever with respect to patent infringement if the infringing products shall have been made to the specifications of the Buyer or a third party or if such alleged infringement shall consist of the use of Seller's products for purposes other than those for which the same shall have been sold by Seller and Buyer shall indemnify Seller against all claims arising out of alleged infringement of patents, designs, copyrights, or trademarks with respect to any goods manufactured to Buyer's specifications.

15. SAMPLES SUPPLIED BY BUYER
These terms apply when drawings are not available and Seller is producing new product(s) from dimensions taken from original gear(s). Seller will make the best attempt to determine the approximate original specifications and or properties of the original product. Seller has no way of determining what the exact original material and or specifications were or what the exact original manufacturing and or heat treating processes were. It is the responsibility of the Buyer to notify Seller in writing of dimensions and or tolerances that are critical to the Buyer's intended use of the product(s).

16. ADDITIONAL COSTS
Prices quoted by Seller are based on Seller's best attempt to estimate all costs needed to produce products that are made to order from customer's specifications or samples. However, from time to time an unforeseen need for special tooling and or processes may be required in order to produce made to order products that conform to the customer's specifications or samples. In the event of this occurrence Seller reserves the right to charge the customer in addition to the originally quoted price for any additional costs incurred in order to conform to the customer's specifications or samples. Furthermore, Seller shall not be liable for any damage or costs claimed to result from any delay in delivery due to any cause whatsoever.
Revised 12/12/2013

Indented Bill of Material (BOM)

E-Drive Drivetrain Assembly (Final Parts Only)

Assy Level	Part Number	Description Lvl0	Lvl1	Lvl2	Material	Weight (lbs)	Vendor	Qty	Weight SubTotal	Cost	Shipping	Tax $	Total Cost
0	ST5-00	Drivetrain Assembly											
1	ST5-01		Input Shaft Assembly										$ 1,260.45
2	ST501-01			Input Shaft (1 3/4 X 12 in.)	4140 HT		Speedy Metals	1	0	$ 38.60	$ 19.54		$ 58.14
2	ST501-02			RH Cut, 2.5" Helical Gear	8620		8620 Steel	1	0	$ 1,068.75			$ 1,068.75
2	ST501-03			Retaining Ring (1 pc.)	Stainless Steel		Grainger	1	0	$ 4.39	$ 2.62	$ 0.52	$ 7.53
2	ST501-04			Key 1	Low Carbon Steel		Grainger	1	0	$ 2.60	$ 2.62	$ 0.52	$ 5.74
2	ST501-05			Tapered Roller Bearing - SKF 33205/Q	High Carbon Chrome Steel		123Bearing	2	0	$ 22.96	$ 2.39	$ 3.67	$ 51.98
2	ST501-06			Bearing Block Input Shaft No Seal	6061 AL	0.19	Metals Depot	1	0	$ 21.69			$ 21.69
2	ST501-07			Bearing Block Input Shaft W/ Seal	6061 AL	0.19	Metals Depot	1	0	$ 21.69			$ 21.69
	ST501-08			Input Shaft Seal - SKF 9803			KSC direct	1	0	$ 20.25	$ 4.69		$ 24.94
1	ST502-00		Intermediate Shaft Assembly										$ 2,599.73
2	ST502-01			Intermediate Shaft (2 X 12 in.)	4140 HT		Speedy Metals	1	0	$ 32.70	$ 19.54		$ 52.24
2	ST502-02			LH Cut, 2.83" Helical Gear	8620	1.79	8620 Steel	1	1.79	$ 1,116.25			$ 1,116.25
2	ST502-03			LH Cut, 6" Helical Gear	8620	9.58	8620 Steel	1	9.58	$ 1,306.25			$ 1,306.25
2	ST502-04			Retaining Ring (5 pcs.)	Carbon Steel		Grainger	1	0	$17.01	$ 2.62	$ 0.52	$ 20.15
2	ST502-05			Key 2	Low Carbon Steel		Grainger	2	0	$ 3.05	$ 2.62	$ 0.52	$ 9.24
2	ST502-06			Tapered Roller Bearing - SKF 33206/Q	High Carbon Chrome Steel		123Bearing	2	0	$ 22.76	$ 3.05	$ 3.67	$ 52.24
2	ST502-07			Bearing Block Intermediate Shaft No Seal	6061 AL		Metals Depot	1	0	$ 21.68			$ 21.68
2	ST502-08			Bearing Block Intermediate Shaft W/ Seal	6061 AL		Metals Depot	1	0	$ 21.68			$ 21.68
1	ST503-00		Differential Assembly										$ 2,114.50
2	ST503-01			31 Spline Gear Rebuild Set	Steel		EastCoastGearSupply	1	0	$ 145.00		$ 11.60	$ 156.60
2	ST503-02			Ford 8.8 Carrier Bearing	High Carbon Chrome Steel		EastCoastGearSupply	2	0	$ 24.00			$ 48.00
2	ST503-03			RH Cut, 9.5" Helical Ring Gear	4140	13.24	8620 Steel	1	13.24	$ 1,591.25			$ 1,591.25
2	ST503-04			Ring Gear Mounting Bolts	Steel		Fastenal	10	0	$ 0.46		$ 0.24	$ 4.84
2	ST503-05			31 Spline Axle Shaft	Steel		4 Wheel Parts	2	0	$ 112.99		$ 18.08	$ 244.06
2	ST503-06			Bearing Block Differential	6061 AL		Metals Depot	1		$ 42.67			$ 34.06
2	ST503-07			Bearing Block Differential	6061 AL		Metals Depot	1		$ 42.67		$ 0.24	$ 34.06
2	ST503-08			7/16"-20 Grade 8 Ring Gear Mounting Nuts	Steel		Fastenal	10					$ 1.64
1	ST504-00		Housing Assembly										$ 539.91
2	ST504-01			Plates (0.25x12x19.25)	6061 AL		MidWest Steel And Aluminum	1	0	$20.67	$8.14		$ 28.81
2	ST504-02			Plates (0.25x12x19.25)	6061 AL		MidWest Steel And Aluminum	1	0	$20.67	$8.14		$ 28.81
2	ST504-03			Plates (0.25x9.25x19.25)	6061 AL	4.34	MidWest Steel And Aluminum	1	4.34	$17.17	$8.14		$ 25.31
2	ST504-04			Plates (0.25x9.25x19.25)	6061 AL	2.55	MidWest Steel And Aluminum	1	2.55	$17.17	$8.14		$ 25.31
2	ST504-05			Plates (0.25x8.75x12)	6061 AL	2.55	MidWest Steel And Aluminum	1	2.55	$11.84	$8.14		$ 19.98
2	ST504-06			Plates (0.25x8.75x12)	6061 AL		MidWest Steel And Aluminum	1		$11.84	$8.14		$ 19.98
2	ST504-07			Clear Housing Top	Acrylic		Tap Plastics	1	0	$ 10.28	$7.58	$ 0.82	$ 18.68
2	ST504-08			Corner Blocks	6061 AL	0.28	Zoro	1	0.28	$73.61		$ 5.34	$ 78.95
2	ST504-09			Output Shaft Seals - SKF 13510			KSC direct	2	0	$ 15.71	$ 4.69	$ -	$ 36.11
2	ST504-10			Housing Bolts	Hex bolts, Zinc plated grade 5 steel, 1/4'		Bolt Depot	1	0	$ 4.06	5.975	$ 0.32	$ 10.36
2	ST504-11			Oil Drain Plug (1/2-20 Thread)	Zinc Plated Steel		AutoZone - In Store	1		$ 4.99		$ 0.40	$ 5.39
2	ST504-12			Oil Fill Plug (1/2-20 Thread)			AutoZone - In Store	1	0	$ 4.99		$ 0.40	$ 5.39
2	ST504-15			Flat Stock, Steel, 4140	4140		Zoro	3		$73.61		$ 16.01	$ 236.84

N/A	
Estimate	

BUDGET	$	10,000.00
PROTOTYPE TOTA	$	6,514.60
REMAINING DIFFE	$	3,485.40
Weight total (lbf)		89

Vendor List and Contact Information

1.	123Bearing	(646) 712 9672	123Bearing.com
2.	4 Wheel Parts	(877) 260 5452	4WheelParts.com
3.	Ace Hardware	(805) 543 2191	AceHardware.com
4.	AutoZone	(805) 783 1212	AutoZone.com
5.	Bolt Depot	(866) 337 9888	BoltDepot.com
6.	Borah’s Awards	(805) 543 6514	BorahsAwards.com
7.	East Coast Gear Supply	(919) 672 2705	EastCoastGearSupply.com
8.	Fastenal	(805) 786 4505	Fastenal.com
9.	Grainger	(800) 472 4643	Grainger.com
10.	Home Depot	(800) 466 3337	HomeDepot.com
11.	KSC Direct	(800) 918 8939	KSCDirect.com
12.	Metals Depot	(859) 745 2650	MetalsDepot.com
13.	Midwest Steel and Al	(888) 744 6868	MidwestSteelSupply.com
14.	O’Reilly Auto Parts	(805) 546 9644	OreillyAuto.com
15.	One Way Industrial Supply	(805) 596 0645	OneWayInd.com
16.	Rush Gears	(800) 523 2576	RushGears.com
17.	Speedy Metals	(866) 938 6061	SpeedyMetals.com
18.	Tap Plastics	(888) 827 7330	TapPlastics.com
19.	Vaxo Systems	(928) 458 5176	VaxoSystems.com
10.	Zoro	(855) 289 9676	Zoro.com

Table of Contents

E-Drive DVP&R

Project:	Drivetrain		Team:	E-Drive		Sponsor:	Sharpell Technologies				
TEST PLAN										**TEST REPORT**	
Test #	Specification	Test Description	Test Location	Test Equipment	Parts Needed	Acceptance Criteria	Test Responsibility	TIMING		Numerical Results	Notes on Testing
								Start date	Finish date		
1	Zero Oil Leakage From Housing	The drivetrian housing will be filled with the appropriate level of oil, and will be placed in a container to collect oil that may possibly leak. The drivetrain will be left to sit in the container for 24 hours. The amount of oil in the container will be recorded if a leak exists. The drivetrains seals will be inspected and improved and the test will be repeated, if necessary.	Off Campus	N/A	Final Drivetrain Prototype Shallow Container Oil	0 mL of Oil Leakage	Charissa	5/11/17	5/11/17	Passed	2 drops of oil leaked from two bolt locations. Fixed by wrapping bolts with PTFE tape (teflon tape)
2	Verify Speed Reduction	Rotate the input shaft 8.04 revolutions and verify that the output shafts have completed	Bonderson	N/A	Final Drivetrain Prototype	8.04:1 Reduction	Kevin	5/11/17	5/11/17	Passed	Visually inspected an 8.04:1 speed reduction ratio.
3	Verify Torque Increase	Attach digital torque wrenches to input and output shafts. Holding the output shafts' torque wrenches in a fixed position, apply a load with the input shaft's torque wrench. Verify that the output shafts' digital readout sums up to 8.04 times the input shaft torque input.	Bonderson	3 Digital Torque Wrenches	Final Drivetrain Prototype Axle Shafts with a nut/bolt attached at the end Input Shaft with a nut/bolt attached at the end	8.04:1 Increase	Kevin	5/16/17	5/23/17	Torque Increase Ratio of 7.36:1	Torque did not equally split between the two output shafts, so the procedure was modified to measure the torque at only one output shaft.

Input Torque [lb-ft]	Uncertainty (Bias) [lb-ft]	Uncertainty (Precision) [lb-ft]	Total Uncertainty [lb-ft]	Avg. Uncertainty [lb-ft]
5.1	0.10	0.05	0.11	
5.0	0.10	0.05	0.11	
5.1	0.10	0.05	0.11	0.11
5.1	0.10	0.05	0.11	
5.0	0.10	0.05	0.11	
7.8	0.16	0.05	0.16	
7.5	0.15	0.05	0.16	
7.5	0.15	0.05	0.16	0.16
7.5	0.15	0.05	0.16	
7.5	0.15	0.05	0.16	
10.0	0.20	0.05	0.21	
10.0	0.20	0.05	0.21	
10.1	0.20	0.05	0.21	0.21
10.0	0.20	0.05	0.21	
10.0	0.20	0.05	0.21	
12.5	0.25	0.05	0.25	
12.5	0.25	0.05	0.25	
12.5	0.25	0.05	0.25	0.25
12.5	0.25	0.05	0.25	
12.5	0.25	0.05	0.25	

Output Torque [lb-ft]	Uncertainty (Bias) [lb-ft]	Uncertainty (Precision) [lb-ft]	Total Uncertainty [lb-ft]	Avg. Uncertainty [lb-ft]
35.3	0.7	0.05	0.71	
36.1	0.7	0.05	0.72	
36.0	0.7	0.05	0.72	0.73
38.4	0.8	0.05	0.77	
37.3	0.7	0.05	0.75	
55.8	1.1	0.05	1.12	
53.6	1.1	0.05	1.07	
53.2	1.1	0.05	1.07	1.08
54.0	1.1	0.05	1.08	
53.7	1.1	0.05	1.08	
71.6	1.4	0.05	1.43	
71.1	1.4	0.05	1.42	
75.2	1.5	0.05	1.50	1.43
71.2	1.4	0.05	1.42	
68.4	1.4	0.05	1.37	
94.9	1.9	0.05	1.90	
86.7	1.7	0.05	1.73	
88.7	1.8	0.05	1.77	1.83
94.4	1.9	0.05	1.89	
93.3	1.9	0.05	1.87	

Overall Ratio
6.92
7.22
7.06
7.53
7.46
7.15
7.15
7.09
7.20
7.16
7.16
7.11
7.45
7.12
6.84
7.59
6.94
7.10
7.55
7.46

Average Input	Average Output
5.1	36.6
7.6	54.1
10.0	71.5
12.5	91.6

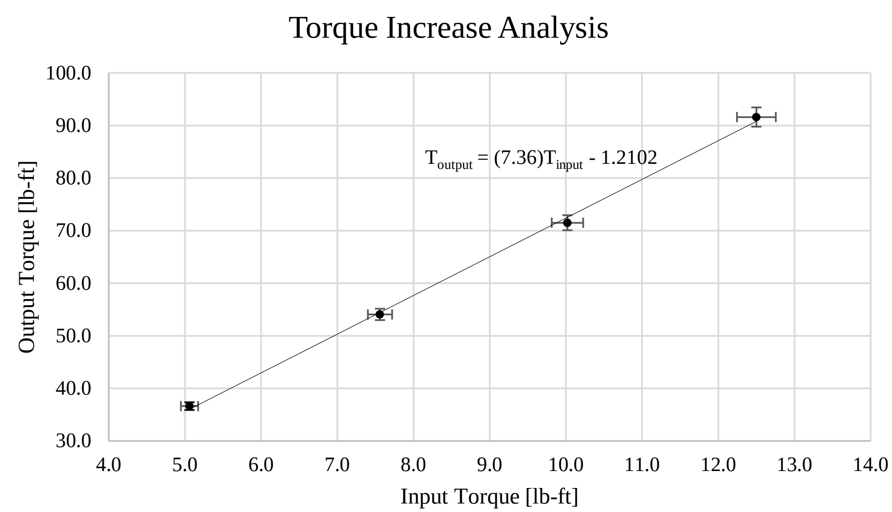
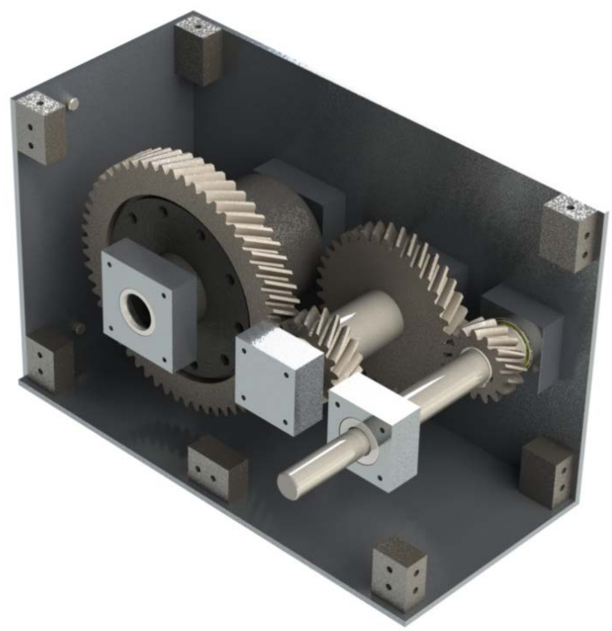


Table of Contents

ST5 Prototype Drivetrain Operator's Manual

The main interaction the user will have with the drivetrain is during the initial installation of the drivetrain into the vehicle. After it is installed into the vehicle, it will be operated based on the driver's input and regulation of the accelerator which is an indirect input to the system. As the motor controller system receives signals based on the input into the accelerator, the drivetrain will directly experience a response due to its coupling with the REMY HVH250-115 motor.



There will be additional user interaction, however, when conducting general maintenance. The required servicing is described below.

Service & Maintenance
The drivetrain requires regular servicing to ensure safe operation and longevity of the drivetrain.

The oil should be replaced every 25,000 miles. Oil is necessary to lubricate and protect the gears, bearings, seals, and other drivetrain components from wear caused by friction. Additionally, it helps to reduce heat caused by friction, and will keep the drivetrain temperature down, decreasing the likelihood of parts warping. With extensive usage, the oil used in the drivetrain will begin to break down and will not protect the drivetrain's components as well. Due to this, the recommended oil change should be conducted every 25,000 miles as previously stated. To replace the oil, refer to the following instructions:

1. Place a bucket or catch can beneath the drivetrain drain plug, which is located on the rear housing plate.
2. Remove the fill plug on the top of the rear housing plate.

3. Remove the drain plug on the bottom of the rear housing plate.
    a. Recommended: Place a tray or oil pan prior to removing the bottom drain plug to catch the oil as it drains from the housing.
4. Let all oil drain completely from the drivetrain housing.
    a. Recommended: Strain used oil to check for any debris for signs of gear/bearing wear.
5. Re-install the drain plug.
6. Pour in 9 quarts of unused oil into the fill plug hole.
7. Re-install the fill plug.
    a. Tighten with a 3/4” socket head wrench or similar tool.
8. Dispose of used oil according to local law.

If there are any oil shavings in the strained oil, please contact your local Sharpell Technologies representative. Depending on the severity, the drivetrain may need to be removed from the vehicle and then inspected for component wear. If necessary, components may need to be replaced.

Intermittently inspect the drivetrain housing seals, to ensure that no debris enters the enclosed system. Dirt and/or other debris can increase the wear on the gears and bearings and can potentially cause premature failure. If there is a faulty seal, please contact your local Sharpell Technologies representative.

Warnings & Cautions

Do not put anything near the input and output shafts, as they will be rotating. Any interference can cause harm to the drivetrain system as well as physical harm to humans if contact is made. This includes arms, hands, legs, feet, loose clothing, and hair.

The drivetrain should be installed in a location where the input and output shafts are protected and isolated from unwanted contact.

If any unusual noise is emitted from the car while driving or excessive vibration is felt in the car contact your local Sharpell Technologies representative as serious malfunction due to misalignment of shafts or gears, bearing failure or warping, interference between gear teeth, or shaft warping may be the cause.

If the car is unresponsive or a wheel is continuously slipping (meaning no torque is translating to one side) this is most likely the result of a differential failure, due to the spider gears or the S spring. Please contact your local Sharpell Technologies for a professional diagnose and guidance for further action for the issues described above or any other sign of part failure or damage.