
# CAL POLY SUPERMILEAGE DRIVETRAIN

## Final Design Report

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# 0.0 Executive Summary

The following report details the full design process used by the Cal Poly Supermileage Drivetrain (CPSMD) Senior Project team in designing a new drivetrain system for the Cal Poly Supermileage Vehicle (SMV) Team. Included is the development stages of each component, the manufacturing of our parts, the final solution's assembly, the testing of the project, and the results from the 2018 Shell Eco-Marathon Competition. After collecting data from several tests, only the final drivetrain efficiency and final sprocket alignment remain unknown. Otherwise, the project came in under budget, within our weight tolerance, and met all the other design objectives. The team was successfully able to produce a robust and reliable drivetrain system for the 2018 Supermileage car that resulted in a 4$^{th}$ place finish at competition. The learnings in this report provide the SMV team with a repository of information on how to build a reliable single staged drivetrain with a large gear ratio.

# 1.0 Introduction

The Cal Poly SMV Club has the goal of designing and building an extremely high efficiency prototype vehicle which competes against other schools from North and South America for fuel efficiency in the annual Shell Eco-Marathon Competition. To help improve upon the vehicle's efficiency, this project aims to redesign and manufacture a new drivetrain system which will be implemented in the 2018 chassis. The Cal Poly team has always ranked high in the competition; however, it has great potential for improvement. The 2017 car was able to achieve 1500.7 mpg, but was estimated to be capable of reaching mileage in the mid 2000's. Thus, the motivation behind this project is to improve the projected mpg of the car and to more closely match this optimal value. To achieve a full breadth of knowledge in the subject, research into current and past drivetrain designs of Cal Poly's Supermileage Team as well as other high achieving teams was performed. In addition, we conducted research into similar power transferring systems and each of their components. This project is to be completed by the Cal Poly Supermileage Drivetrain Team, comprised of Cal Poly undergraduate students Justin Miller, Heather Fields, and Michael Bolton. The stakeholders for this project are the Cal Poly Supermileage Vehicle Club, Joseph Mello the club advisor, and John Fabijanic the project advisor.

# 2.0 Background

To gain a better understanding of the project's scope, a significant amount of research has been continuously performed. As the project progressed, this depository of knowledge continued to grow as did our understanding of the task at hand. The most up to date collection of this research is included within the sections that follow. The full list of resources referenced can be seen in Appendix A.

## 2.1 Past and Current Supermileage Drivetrain Designs

### 2.1.1 2018 Shell Eco-Marathon Rulebook Adherence

Designing a drivetrain for the Supermileage vehicle that is to compete in the 2018 Shell Eco-Marathon subjects it to all of the official rules of the competition. General rules that the design must adhere to are as follows:

- Any cover of the energy compartment (engine/motor/transmission/battery, etc.) should be easy to open for quick inspection access
- All parts of the drivetrain, including fuel tank, hydrogen system components, etc. must be within the confines of the body cover.

Rules pertaining specifically to the clutch and transmission include the following:

- All vehicle propulsion must be achieved only through the friction between wheels and the road.
- For centrifugal/automatic clutches, the starter motor speed must always be below engagement speed of the clutch.
- All vehicles with internal combustion engines must be equipped with a clutch system

Other components that are affected by the rules of the competition are the chain or belt which are subject to the following rules:

- Mandatory installation of effective transmission or belt guards to protect against the event of the chain or belt breaking.
- Must be made of metal or composite material rigid enough to withstand a break.

### 2.1.2 Former Cal Poly Drivetrain Designs

The various Supermileage Teams from Cal Poly have tried several different drivetrain solutions. In 2015, Ventus I attempted a single staged 13:1 gear reduction with a chain tensioner. The design was highly unreliable and prone to chain throw. Alignment between the engine sprocket and drive sprocket was poor. Additionally, the overall robustness and alignment of the rear assembly with the rest of the car was low due to its large number of components. As a result of this low reliability, Ventus I was limited to only a few completed runs, but achieved 988 miles per gallon. 2016 saw the introduction of Ventus II with a two-stage gear reduction and a single piece modular engine plate. Figure 1 below shows a comparison of Ventus I's rear end to Ventus II's.

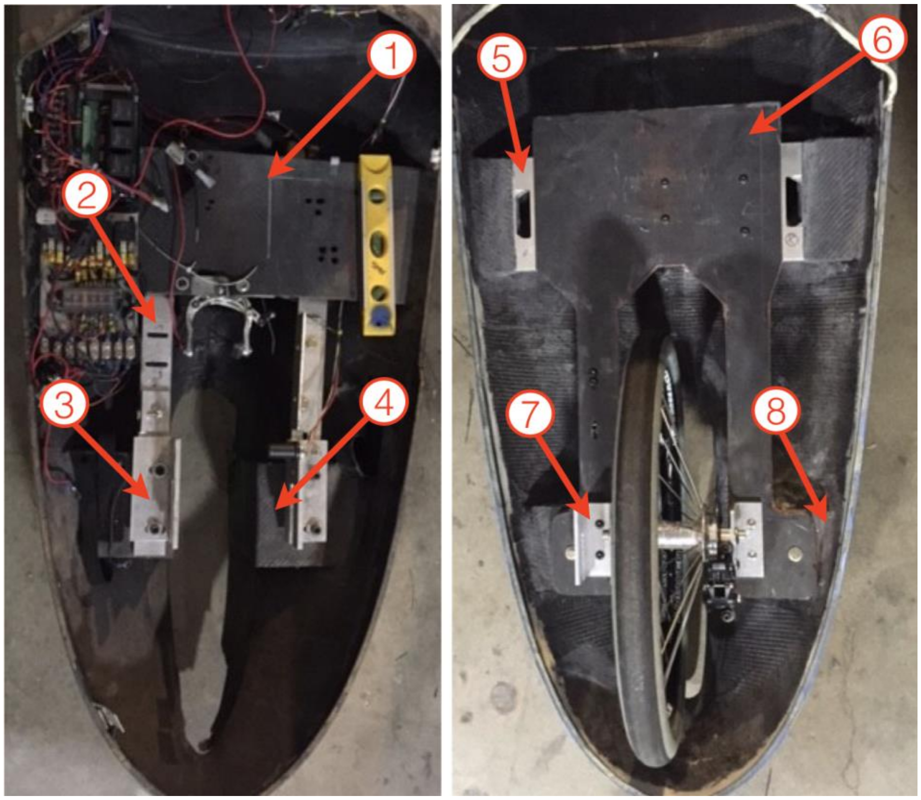



*Figure 1:* Shown on the left is the 2015 car's rear end made up of multiple pieces. Parts 1 through 4 each have to be individually located and fastened. On the right is the improve 2016/2017 modular engine plate. Here, parts 6 and 7 are immediately located by part 6, the bracket. everything is then fully fastened to the chassis by part 8.

On this engine plate, the engine mount, jackshaft, ECU, and rear dropouts were all mounted. This plate, in combination with an alignment jig, was used to align the rear and front wheels improving accuracy and ease of installation. Despite the improvement of front to rear wheel alignment, Ventus II still struggled with sprocket alignment between the engine and the drive sprocket. The introduction of a two-

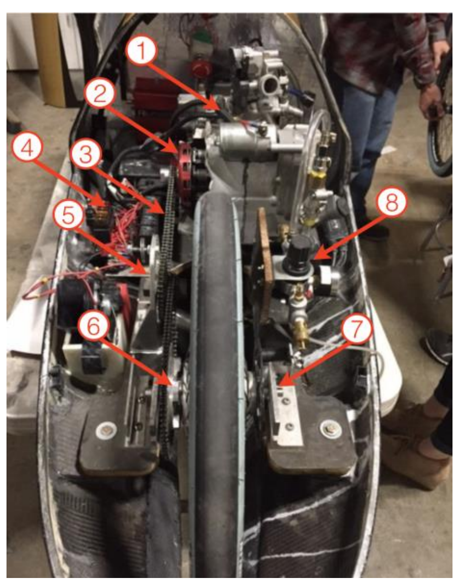



*Figure 2. Shown here is the modular engine plate with all drivetrain and ECU components installed. The entire assembly can be removed together in order to service the chassis or test the system independently of the rest of the car. The parts pictured include; 1) Engine, 2) Vortex Clutch, 3) #25 Chain, 4) Electronics, 5) Jackshaft, 6) Rear Hub and Freewheel, 7) Rear Axle Dropout, 8) Fuel System*

stage gear reduction compensated for this shortcoming and greatly improved reliability. Figure 2 shows the two-stage system and how all powertrain components were mounted to a single, removable plate.

Ventus II completed the 2016 competition without ever throwing the chain and achieved a score of 1,215 miles per gallon. For the 2017 competition, further refinement of Ventus II's drivetrain was achieved through testing and this car was dubbed Ventus II RS. Qualitative optimal chain tension was found by trial and error while the last reliability issues in the overall powertrain, such as the fuel system, were resolved. In the thesis paper, *Optimizing Control of Shell Eco-Marathon Prototype Vehicle to Minimize Fuel Consumption* by Chad Bickel, the efficiency for this drivetrain set up is estimated to be about 62.1%. This number was derived from comparing the theoretical acceleration of the vehicle based on the engine's torque output and measured acceleration values recorded by RaceCapture [1]. Ventus II RS was able to achieve 1,500 miles per gallon.

### 2.1.3 Universitè Laval

The Universitè Laval has now won back to back Shell Eco-Marathons in 2016 and 2017. They are consistently one of the top two teams at competition. Much of Laval's efficiency comes from their custom built engine, but it is in mating this engine to a high quality drivetrain that allows them to reach continued success. Every year, Laval brings essentially the same car to competition, opting to make their improvements via testing and refinement rather than large design changes. As such, the basic design of their drivetrain from 2016 to 2017 was practically unchanged. The drivetrain features a single-stage reduction of 12:1 with a chain tensioner, a custom machined left hand drive hub, and an off the shelf freewheel. Laval uses the same #25 chain that Cal Poly uses but their setup features the Cheetah brand centrifugal clutch, which is further discussed in section 2.2. Laval's drivetrain design uses mounting surfaces integrated into the chassis rather than a modular or removable design. Based on the estimated data for Cal Poly's efficiency and considering that Laval's drivetrain is a single stage, their efficiency is estimated to be about 80%. For the 2017 competition, Laval achieved 2,700 miles per gallon.

### 2.1.4 University of Toronto

The University of Toronto was the 2015 Shell-Eco Marathon champion and from the beginning of their participation in Supermileage, they have consistently been a top three ranking team. Toronto also makes use of a custom built engine that utilizes a right hand drive output shaft. Their drivetrain consists of a single-stage, 14:1 reduction without the use of a chain tensioner. The use of a # 35 chain allows for a thicker sprocket with less tendency to deflect under comparable loads. The right hand drive engine allows for a larger variety of higher quality bike parts such as hubs and freewheels; despite this, for the last two years Toronto has attempted to make use of a custom made clutched hub with little success. The hub has difficulty engaging when the car transitions from coasting to being engine driven. Toronto's only successful competition runs in 2016 and 2017 were accomplished with a traditional right hand drive hub and freewheel. The University of Toronto's drivetrain efficiency can be estimated in the same way as Laval's but at 78% since they use a thicker chain and sprocket with greater inertia. With a young team new to competition, the University of Toronto achieved 1,431 miles per gallon in 2017, but in the past has reached over 2,000 mpg.

### 2.2 Clutch

The first element of an internal combustion powered drivetrain is the clutch. The clutch allows for the engagement or disengagement of two rotating bodies. Besides being required in the Shell Eco-Marathon Rulebook, it is important to incorporate a clutch at the engine's output shaft to allow the engine to remain unloaded during start-up and to allow the engine speed to reach a certain number of revolutions per minute (RPM) before engagement. This is because an engine's output torque is a function of RPM and for lower RPM's, the torque is too small to overcome the drivetrain resistance. Such a scenario would result in stalling the engine. To compensate for this, there is a certain amount of slippage that occurs until the engine speed reaches an optimum point where the clutch fully engages. Then the engine begins to provide substantial drive torque which powers the drivetrain. Slippage is a required feature of a clutch but also inherently introduces losses. Thus, it is important to take great consideration in controlling just how much slippage occurs and finding an optimum engagement speed. Between 2014 and 2015, much work was done to find a high performance clutch that could engage at what data suggested was the optimal engagement speed. The result of these efforts was the incorporation of a go-kart racing, centrifugal clutch by Vortex shown in Figure 3.


*Figure 3:* Vortex centrifugal clutch used by the Cal Poly Supermileage Vehicle.

Data taken in *Optimizing Control of Shell Eco-Marathon Prototype Vehicle to Minimize Fuel Consumption*, suggests that the Vortex Clutch satisfies our optimum engagement speed. Figure 4 shows a plot taken from the thesis supporting this claim.

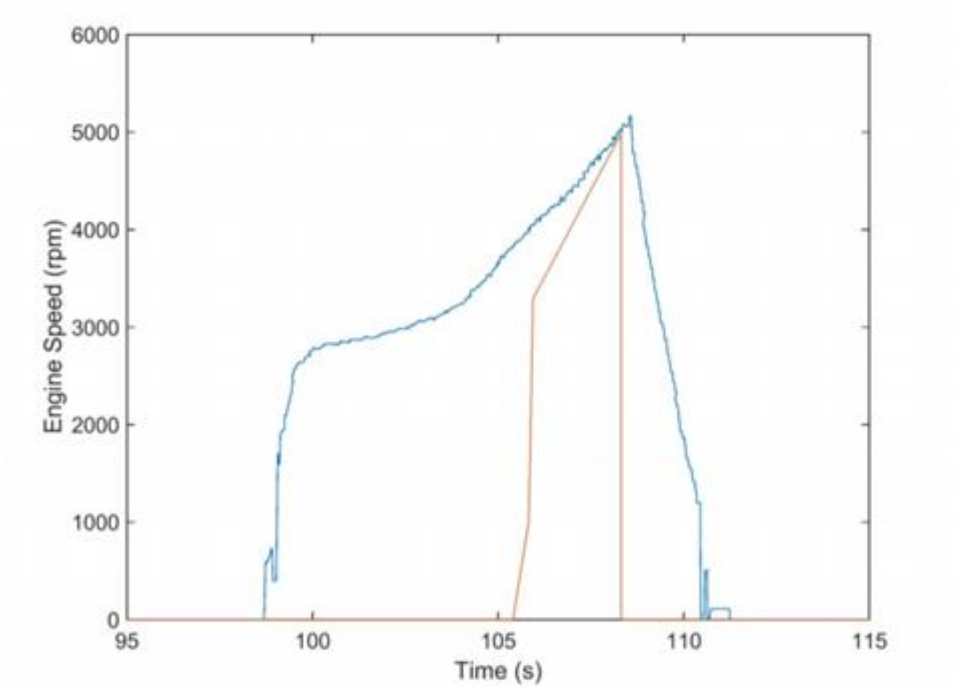



*Figure 4:* Plots of engine speed versus time. Actual engine speed is represented by the blue line and simulated ideal engine speed is represented by the red line.

The linear shape of the plot from 105 seconds to nearly 110 seconds represents a fully engaged clutch. Both the simulated engine speed and the actual engine speed start this full engagement just above 3000 RPM. With the gear reduction used, the minimum vehicle speed that is reached during competition is 17.92 mph and corresponds to a clutch housing speed of 3329 RPM. Since the Vortex clutch's torque is equal to the engine torque at 3021 RPM, there is no slippage while the vehicle is being driven [1]. Teams such as Toronto and Laval use a go-kart racing clutch, known as the Cheetah. It is of the same design and architecture as the Vortex, but produced by a different company. The Cheetah clutch is less expensive than the Vortex; however, with the Supermileage Team's sponsorship by SMC Clutches, the cost difference is mitigated.

## 2.3 Power Transmission Methods

"Power transmission is the movement of energy from its place of generation to a location where it is applied to perform useful work" [2]. Conventional powertrains come in 3 standard forms; chain drive, belt drive, and gear drive. Each form performs the same task, power transmission, in different ways. Each form also comes with its advantages and disadvantages that must be considered before deciding which to use. For our purposes. the chain and belt drive systems such as those shown in Figure 5 are in consideration, neglecting gear drive due to the large length of space needed to transmit power across in the vehicle. A gearbox system to fit our needs would end up being heavy and would require a large number of components.

*Figure 5:* Two power transmission methods in consideration. Belt drive system (Right), Chain drive system (Left).

The most commonly used method of power transmission in low-power energy transmission similar to ours is the rolling chain system. Most often, the power is conveyed by a roller chain, known as the drive chain or transmission chain, passing over a sprocket gear, with the teeth of the gear meshing with the holes in the links of the chain. Chain drive systems usually have an efficiency of roughly 98.5% if aligned correctly. The only significant downfall with using a chain drive is the need to use lubricant during operation, which is not always the case with other forms of power transmission

In cases where a chain drive system is not implemented, a belt drive usually takes its place. A belt drive consists of similar components to that of a chain drive, except the rolling chain is replaced with a toothed belt, often called a timing belt. The belt sprockets, though shaped to allow the belt to correctly fit, are in general the same as the chain sprockets. As with the chain drive system, the belt drive has an efficiency of approximately 98%. Unlike the rolling chain system, no lubrication is needed.

In regard to the advantages of one versus the other, focus was directed at the efficiencies of the two. Through testing done by Friction Facts, a chain driven system is slightly more efficient than an equivalent belt driven system. Their tests indicated that “a conventional chain drive consumes 2.92 watts on average, while the belt eats up 3.93 watts. Although the difference is just 1 watt, this works out as a substantial 34.6 percent” [3].

One particularly efficient belt drive system is Goodyear' Eagle Pd Synchronous Belts and Pulleysas shown in Figure 6. The Eagle belt system features a unique herringbone pattern for the pulley/belt interface. This allows the pulleys to be thinner and the system to be self-tracking removing the need for guide flanges on the pulleys. When compared to traditional V-belts, these Eagles offer a consistent 5% increase in efficiency [4].


*Figure 6 Goodyear Eagle Belt synchronous belt with herringbone pattern*

## 2.4 Sprocket

The 2018 SMV car may employ a chain drive system and consequently sprockets. Merriam Webster defines a sprocket as a “toothed wheel whose teeth engage the links of a chain” [5]. For all intents and purposes, there are six types of chain sprockets: plate, hub on one side, hub on both sides, detachable hub-plate, and shear pin and slip clutch sprockets. The sprockets for the 2017 SMV drivetrain included a large, rear plate sprocket, an intermediate reduction sprocket, and a smaller, hub on one side sprocket as shown in Figure 7 below.

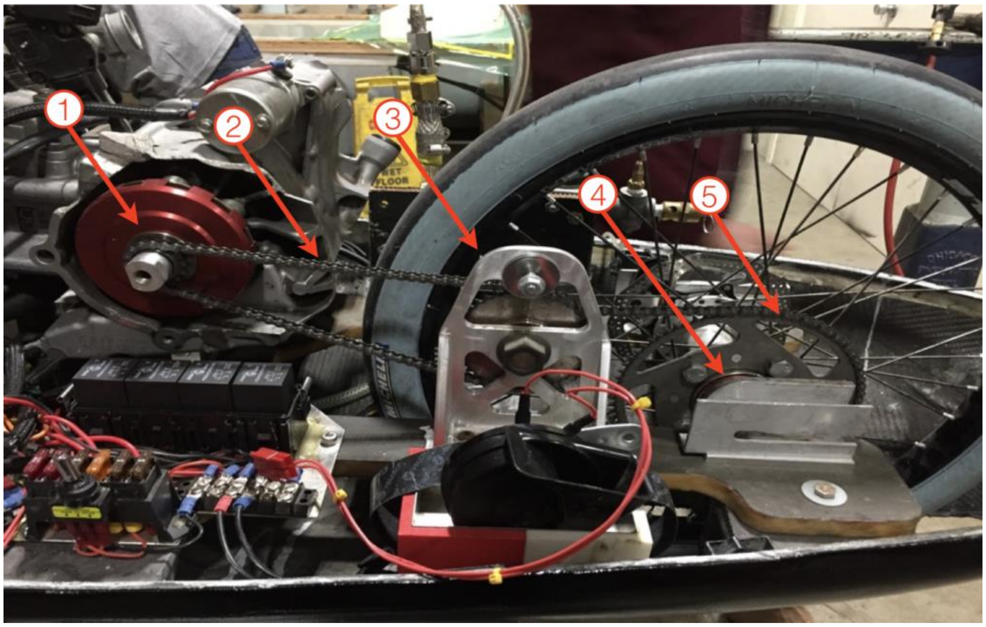



*Figure 7:* The 2017 SMV drivetrain showing the sprockets in the double stage reduction. The parts pictured include; 1) Vortex clutch, 2) #25 Chain 3) Jackshaft System with intermediate sprockets, 4) Freewheel, Hub, and Adapter, 5) Rear sprocket

Sprockets can also be classified into two different classes, commercial and precision. Commercial sprockets are used with slow to moderate speed drives while precision sprockets are required when extreme, high speeds are combined with high loads or when the drive involves fixed centers, critical timing or close clearance with outside interference [6]. The new drive train design for the 2018 SMV would use commercial sprockets as the car will not be experiencing high speeds and loads.

### 2.4.1 Reduction and Stages

In a chain drive system, one sprocket uses the chain to indirectly drive another sprocket. This setup allows for sprocket reduction, using two different sized sprockets, to transfer power and increase or decrease the revolutions per minute derived from the motor. The equation used to determine the needed sprocket reduction is shown in Equation 1 below

$$\text{Sprocket Reduction} = RPM_{input} : RPM_{output} \quad (1)$$

This sprocket reduction equation drives the sprocket reduction ratio through Equation 2 below

$$\text{Sprocket Reduction Ratio} = C_o : C_i \qquad (2)$$

Where

$C_o$ = output sprocket circumference
$C_i$ = input sprocket circumference

The reduction correlates to RPM such that every time the input sprocket makes one full revolution, the output sprocket rotates the fraction determined by the ratio , $\frac{C_i}{C_o}$, thereby reducing the number of revolutions per minute by the inverse of the ratio, $\frac{C_o}{C_i}$ [6].

For our drivetrain design, we have the option of a single stage or a double stage reduction. Using a single stage reduction, the system would be left hand drive based on the current engine's left hand drive (LHD) output shaft. With a double reduction it could be right hand drive. A single reduction will inherently be more efficient since the losses of a chain drive system are not compounded as they are in a two stage system.

In Chad Bickel's thesis, *Optimizing Control of Shell Eco-Marathon Prototype Vehicle to Minimize Fuel Consumption*, it was determined that a final drive ratio of between 14:1 and 16:1 would be optimal to maximize fuel efficiency [1]. Bickel's results are shown in Table 1 below.

*Table 1. Tabulated estimations for fuel economy with respect to final drive ratio.*

| Fuel Economy (MPG) | Speed ($ft/s$) | Final Drive Ratio | Simulation Runs |
|---|---|---|---|
| 1146.2 | 17.92 – 26.86 | 15.17 | 13152 |
| 1146.0 | 17.36 – 27.55 | 14.77 | 7800 |
| 1137.3 | 17.55 – 27.41 | 16.23 | 3197 |
| 1115.2 | 16.96 – 28.69 | 16.44 | 3082 |

### 2.4.2 Sprocket Material

Common materials used for sprockets are carbon steel, with or without hardened teeth, stainless steel, and other special materials such as alloys and titanium. Although it is seldom necessary to use special, high strength materials, a benefit of using certain materials is weight savings, which is useful for our purposes. The 2017 SMV utilizes titanium sprockets that are lighter than the common sprocket materials such as carbon steel and stainless steel. With no standing issues using titanium as the sprocket material, it is a feasible option for the new design. A carbon composite sprocket has been attempted by SMV in the past and is worth a second look.

If a material other than titanium is to be used, hard anodized aluminum would be a viable option. Hard anodizing is a process that creates a hard wearing, corrosion resistant coating on various aluminum and can create a durability that approaches that of hard faced or case hardened steel [7].

## 2.5 Lubrication

A lubricant is "a substance (such as grease) capable of reducing friction, heat, and wear when introduced as a film between solid surfaces" [8]. Choosing the correct lubricant can be the difference between an efficient and reliable system and one that is not. Our potential use of a chain drive system and its requirement for lubrication makes determining the proper lubricants and lubricant method necessary. In the past, this has been overlooked by the drivetrain implementation teams.

In testing done by Friction Facts, chain lubrication has had significant effects on efficiency. Assuming our chain system is similar to that of a bicycle's, which is what the referred to testing was done on, we can use the data collected in these tests. Figure 8 below outlines the data collected through the tests and the overall efficiencies of each lubricant tested is shown, the left being the most efficient with the least power expelled and the right being the worst using the most power during transmission. It is interesting to note that many of the more-efficient lubricants are dry and heavy, the most efficient being Paraffin Wax [8].

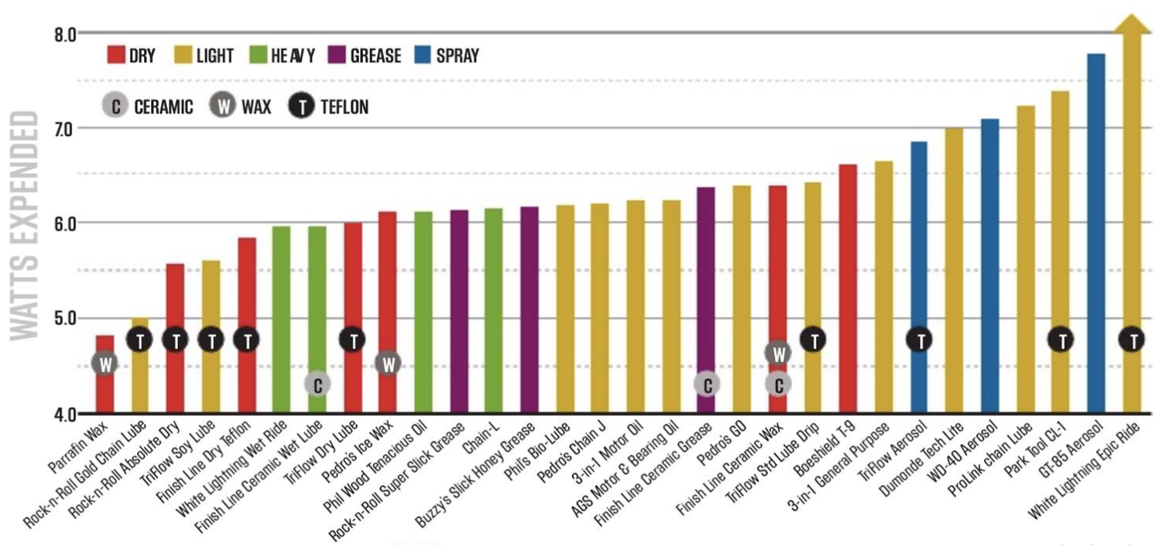



*Figure 8.* Efficiency test done by Friction Facts. Each lube was tested for frictional losses on Friction Facts equipment, which is accurate to ±0.02 Watts.

## 2.6 Hub & Freewheel

The hub and freewheel are the components which directly connect to the rear wheel. The hub is the center portion of the wheel and is connected to the rim with spokes. Bicycle hubs are typically offered as right hand drive but more and more hub manufacturers are offering left hand drive options. A freewheel's purpose is to lock the rotation of the hub and the drivetrain together under power but then allow the hub to continue spinning independently when the drivetrain is not moving. This can be accomplished in many ways, but is most often done so using a one way bearing with pawls.

### 2.6.1 Pawl Design

Conventionally, the freewheel uses a ratcheting system, shown in the Figure 9 below. The ratcheting pieces used to allow the free rotation are called pawls. In past designs, Cal Poly's SMV Club has used a conventional bicycle freewheel which used pawls such as these. The issue with this design is the inherent friction due to the clicking pawls. Pawls can be removed to reduce this friction force, though removing too many can cause issues with power transfer under load. If the conventional pawled design is to be used again, the minimum number of pawls required to transmit the power outputted by our shaft would be necessary to find in order to optimize free coasting efficiency. However, there are other truly frictionless free coasting systems which will be considered as well.

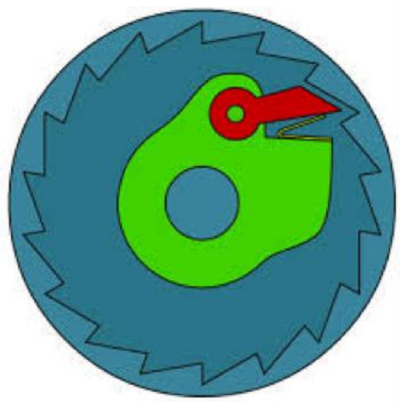


*Figure 9.* Pawled freewheel design

### 2.6.2 Freecoaster Clutched Design

The Freecoaster clutched freewheel is a replacement for the conventional, pawled design mentioned above. As the name suggests, this is a truly frictionless coasting device (excluding the bearings) which uses a conical clutch, illustrated below in Figure 10. This design would theoretically be more robust than the pawled freewheel due to its "single piece" design. The design would also increase free coasting efficiency, which is not necessarily within our scope of objectives but would be an added performance gain that would be beneficial to the car as a whole. Though this design seems to be an automatic answer to higher efficiency, proper analysis and testing into the clutch engagement and strength must be done in order to qualify it for use in our system.

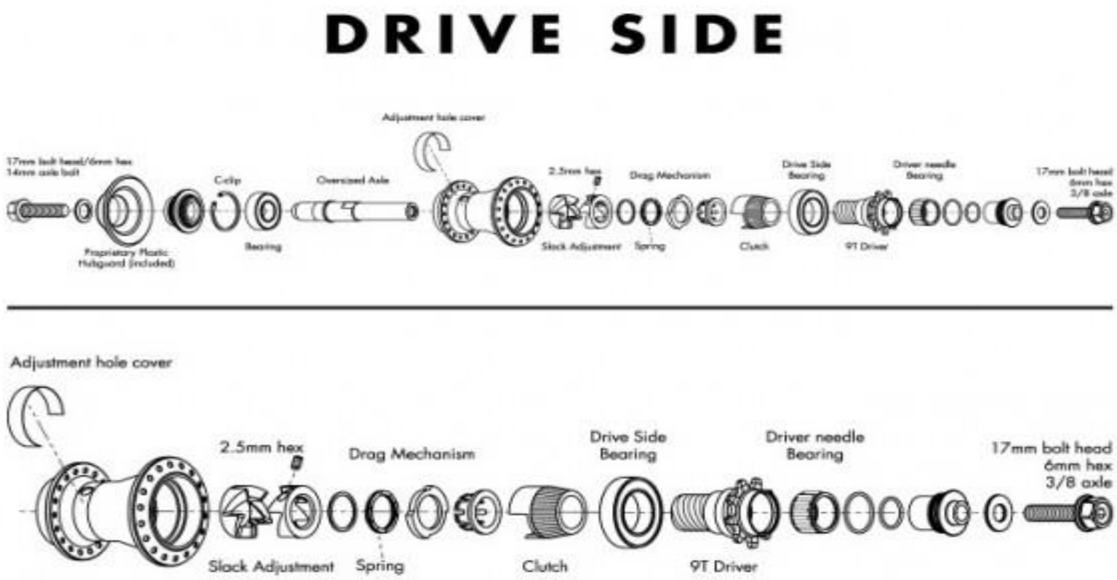



*Figure 10:* Odyssey Clutch Freecoaster Hub part diagram.

### 2.6.3 Availability of LHD Components

Due to our engine's left hand drive output shaft, it is necessary to find LHD components in order to keep to a single staged design. The current hub and freewheel had to be custom manufactured as LHD versions. However, more freewheel and hub designs, as well as the clutch freecoaster hub, are becoming available in LHD. Depending on which is chosen, more research into adapters to allow larger rear sprockets and LHD will be necessary. Considerations for custom manufacturing versus off-the-shelf components will also have to be made.

## 2.7 Alignment and Tolerances

Engineering tolerances are a key factor in ensuring proper alignment in our system. To fully understand the tolerances required for our drivetrain, an in-depth study of proper chain installment and alignment is necessary. Diamond Chain Company has a set of installment instructions which gives the tolerances needed for a traditional chain based off of chain pitch and center to center distance. The two equations used are:

$$\text{Planar Alignment }\left[\frac{\text{in}}{\text{ft}}\right] = \frac{0.00133 * C[in]}{P[in]} \tag{3}$$

$$\text{Center Perpendicular Distance [in]} = 0.045 * P[in] \tag{4}$$

With these tolerances found for our case, we will need to determine the best way to analyze these tolerances on our actual application and determine how we will hold our system to them. Systems such as PROFI Laser Alignment Tool may be applicable, but may not directly work for our application. A modified version of this system seems to be the direction we are headed.

## 2.8 Locating Pins

A method of aligning a work piece on a fixture or aligning two pieces of a fixture is to use locating pins. The combination of a round and diamond pin can be used for precise locating of two holes without binding. Carrlane pins are a brand of precision locating pins that use one round locating pin and one diamond locating pin wherein the diamond pin is relieved to locate in only one axis [9]. An example of these pins can be seen in Figure 19.

## 2.9 CMM Capabilities

CMM, or Coordinate Measuring Machine, is a useful machine which uses a touch probe and encoded gantry system to measure points on physical items with high accuracy. These point measurements can then be used to measure the relation between features and planes to ensure tolerance accuracy after manufacturing [10]. Multiple companies make CMM's, including Renishaw. The machine we have access to on campus has a measuring accuracy of roughly 2 microns. This is extremely accurate and more than acceptable for our use in both reverse engineering our engine mount and confirming the overall alignment of our finished system.

## 2.10 Flatness in Sheet Metal

Depending on the manufacturing process used, sheet metal is inherently not flat. Typical sheet metal is either hot or cold rolled. In the rolling process, "the thickness of the strip must be greater in the center to keep the material tracking properly in the rolling equipment" [11]. This crown causes inconsistent stresses and grain structure throughout the material making the sheet not flat. Companies such as Contrarian Metal Resources use methods such as correction stretching to reform the material into a more evenly stressed flat piece. It is almost impossible to create sheet metal with zero internal, residual stresses, but by making them uniform throughout helps keep the product flat. Sheet metal flatness can be measured by the amount of variance across its plane with unit of length or with flatness specific "I-units." I-units are defined by contrarian as follows: "Flatness in I-units = $(\pi H/2L)2 \times 10^5$. Where H = Height of the deviation and L = Length between deviations. This formula assumes a sinusoidal wave shape" [11].

Flat raw material is not good if machining or post processing imparts stresses that cause the part to warp. Material removal processes such as machining and laser cutting cause the surface of the material to become stressed, creating stress imbalances in the part and causing it to become warped. Ensuring low cutting forces helps reduce the concentration of these stresses and the possibility of warped parts. Such processes as waterjet cutting or wire electrical discharge machining (EDM) can accomplish low stress cutting.

## 2.11 Waterjet Cutting Carbon Fiber

Machining carbon fiber presents different challenges than machining traditional materials, such as metals, does. A viable way to machine carbon fiber includes waterjet cutting. The advantages of using a waterjet to cut carbon fiber include the avoidance of tool clogging and wear, the elimination of needing to change tooling or special tooling, and the avoidance of melting and hazardous fumes that could be involved with other cutting methods. However, with tight tolerances, it would be beneficial to waterjet the part oversized and machine final dimensions [12].

## 2.12 Drivetrain Efficiency Measurements

The efficiency number of 62.1% taken from Bickel's thesis paper was derived using many assumptions. It compares what the acceleration of the car would be if 100% of the torque output from the engine went into the wheel. It was then compared to the measured acceleration of the car by RaceCapture and the percent difference was called "drivetrain efficiency." While a good starting point, this method of estimation is flawed. For one, it neglects to account for the wind speed pushing back against the car and docks this against the drivetrain. In order to get a more accurate measurement of the drivetrain efficiency, it would be best to actually measure the torque output at the wheel and compare this to the torque output at the crankshaft of the motor. A chassis dynamometer (dyno) that measures torque output at the wheel is currently in development for the Supermileage Team. This dyno has a rotational inertia equivalent to the inertia of the Supermileage vehicle and, based on the acceleration of the dyno's flywheels, torque can be calculated. Comparing this torque to that of the engine’s torque at the crankshaft will give a proper measurement of drivetrain efficiency and gives this measurement all along the power curve (ie. it is a transient dyno). In the case that this chassis dyno does not get completed in time for CPSMD's use, a modified bicycle trainer can be used to measure steady state torque at the wheel.

## 3.0 Objectives

The goal of this project is to design and manufacture a more efficient drivetrain for the 2018 Cal Poly Supermileage Vehicle which will compete in the 2018 Shell Eco Marathon Competition. This adheres the team to the important requirements of the Cal Poly SMV club. Quality function deployment, QFD, was used to translate the requirements of the club to effective, measurable engineering requirements or targets tabulated below in Table 2. The accompanying house of quality used during the QFD process can be seen in Appendix B.

### 3.1 Quality Function Deployment

*Table 2: Customer Requirements and Targets*

| Spec # | Parameter Description | Requirement or Target | Tolerances | Risk | Compliance |
|---|---|---|---|---|---|
| 1 | Cost | $1500 | + $500 | L | A |
| 2 | Weight | 4.2 kg | + 1 kg | L | T |
| 3 | Efficiency | 80% | Min | M | A, T |
| 4 | Hub/Sprocket Play | 0.8° | Max | H | A, I, T |
| 5 | Size | 66cm x 38cm x 27cm | Max | M | A, I |
| 6 | Total Removal & Installment Time | 30 minutes | Max | M | I, T |
| 7 | Manufacturing Time | 50 hrs | +50 hrs | M | A |
| 8 | Sprocket/Chain Alignment | 1° - angle tolerance .02" – Center perpendicular distance | Max | H | A, I, T |

Table Key:

Risk - Potential difficulty involved in completing each engineering specification target

- Low (L)
- Medium (M)
- High (H)

Compliance - verification method for each engineering specification target

- Analysis (A)
- Test (T)

- Inspection (I)

### 3.2 Budget and Cost

The Cal Poly Supermileage Vehicle Club has allocated a budget of $1,000- $1,500 for the project. The goal is to effectively utilize the provided funds through thorough cost analysis of every part of the drivetrain and their associated manufacturing.

### 3.3 Efficiency

The efficiency of the design will be quantified in percent power transferred, which is defined as the percent difference between the engine power output at the crank and the power output at the wheel. The goal of the new drivetrain is to exceed the wheel horsepower of the 2017 SMV drivetrain.

### 3.4 Hub/Sprocket Play

The rigidity of the Sprocket Hub assembly is very crucial to the design, weighted as the most important design parameter in the QFD. The rigidity of the drivetrain components directly correlates to the efficiency achieved. Less play also ensures that the alignment of the components always stays within tolerance and is also a key feature of reliability.

### 3.5 Weight

Weight is not considered as highly important as some of the other requirements; however, the goal is to have the new design weigh less than the 2017 design with a considerable weight savings percentage.

### 3.6 Size

The size of the drivetrain will be constrained by the size of the 2018 chassis (66cm x 38cm x 27cm) and the ability to integrate a modular engine plate. The dimensions of the drivetrain will be based off of the existing, removable engine plate and the preliminary designs of the 2018 chassis, as it will be designed in parallel.

### 3.7 Manufacturability

The goal of manufacturability will be to use as many off-the-shelf components as possible; however, the manufacturing of some custom parts will likely be necessary to integrate these components. As a team, CPSMD has experience in 3-D printing, manual machining, CNC machining, welding, and carbon fiber layups. The Cal Poly Supermileage Team also has working relationships with several manufacturing companies, such as Next Intent or WaterJet Central. In the past, these sponsors have been more than happy to lend their time and skills. CPSMD's objective is to keep manufacturability in the forefront of our mind as we design components and develop a drivetrain solution.

# 4.0 Design Development

The current drivetrain system is shown in Figure 7 in section 2.4. The system is chain driven with a #25 chain and achieves a 14:1, two-stage gear reduction. To redesign the system, each component involves an individual process of decision-making. Some components require extensive testing, others use weighted decision matrices, and some use a combination of both. Outlined below are the design concepts for each component and their accompanying decision processes thus far. A collection of sketches from brainstorming and ideation can be found in Appendix C.

## 4.1 Drive system

### 4.1.1 Chain vs. Belt

As discussed in section 2.3 there are two viable options for power transmission, the chain drive and the belt drive. While both options provide similar efficiency at steady state, each has their own advantages and disadvantages. The efficiency of a chain drive system is heavily dependent on the accuracy of sprocket alignment. Since the Eagle belts are self-tracking, they would be much easier to install in the car. A comparison of various other characteristics for belts versus chains was done in a decision matrix that can be found in Appendix D. The result of this matrix gave the edge to a chain drive system.

Perhaps a more compelling reason to use one transmission method over the other comes with comparing their transient energy consumption, that is how much energy it takes to angularly accelerate the sprocket or pulley from zero velocity. Due to the larger mass and larger rotational inertia of pulleys, they would be expected to have a larger rotational kinetic energy than sprockets. In adhering to the Law of Conservation of Energy, one can reason that any energy spent accelerating the sprocket or pulley is not contributing to the propulsion of the vehicle. A rough calculation done in Appendix E showed that a pulley could have 428% greater rotational kinetic energy than would a sprocket of the exact same diameter and material properties. This estimation was conservative. An assumption that an Eagle pulley would have the same diameter as a sprocket to achieve the same 15:1 gear reduction made the pulley undersized. Since the smallest pitch belt offered by Goodyear is 8mm (0.315 inches) and our #25 chain has a pitch of 0.25 inches, the pulley will likely have a larger diameter and an even greater moment of inertia. Since the fuel conversion efficiency and brake specific fuel consumption (BSFC) of an internal combustion engine is only optimal near peak torque, taking a 428% hit on energy consumption during the transient time between startup and peak torque would be highly undesirable. In fairness, this is short period of time; however, in the application of Supermileage where the engine is stopped and started at least 10 times in a single competition run the energy consumption adds up. Therefore, the decision has been made to use a chain drive system where much engineering effort will go into the alignment of the sprockets. A transient dyno, like the chassis dyno introduced in section 2.12, would be the best way to empirically confirm or refute such a conclusion.

The current design uses a # 25 chain; however, a #35 chain is also in consideration for our redesign. With a different pitch size, a different tolerance is allowable. Through our calculations in Appendix F, we show that each would come with it its own benefits and drawbacks in terms of allowable tolerance. The #25 chain would give us a 50% larger shaft/angular tolerance, while the #35 chain would give us a 50% larger sprocket/perpendicular distance tolerance. Currently, our issue is angular alignment. Additionally, our driving sprocket has the ability to be adjusted along the drive shaft ±0.5 inches. With these considerations, we elected to use the #25 chain to give more allowance in the angular misalignment rather than the perpendicular misalignment.

#### 4.1.2 Two-Stage vs. Single-Stage

The current vehicle design utilizes a 14:1, two-stage reduction and our goal is to move to a 15:1, single-stage reduction. This new reduction ratio was the result of refinements done within the thesis *Optimizing Control of Shell Eco-Marathon Prototype Vehicle to Minimize Fuel Consumption* as outlined in section 2.4.1. The debate between a two-stage system and a single-stage system is mainly influenced by two factors: efficiency and reliability. A single-stage system will undoubtedly be more efficient, but a two-stage system allows more room for alignment error. A weighted decision matrix was used to evaluate other relevant characteristics for each system and can be found in Appendix D. This decision matrix gave the edge to a single-stage system. The result of the decision matrix, in conjunction with the ultimate goal of the Supermileage Vehicle Team being efficiency, led to the decision of using a single-staged, 15:1 drivetrain engineered to the greatest accuracy possible.

#### 4.1.2 Clutch Sprocket

The Vortex clutch system is only available off the shelf with #35 drive sprockets of various sizes. The current System uses a 15-tooth pinion sprocket that was custom made by SMC for use with a #25 chain. The requirement of a custom-made sprocket has not been an issue in the past due to SMC's sponsorship and eagerness to help students, but the requirement of a customization must be considered. To run a single-stage, 15:1 gear ratio that fits into the car, a smaller pinion sprocket must be used. Based on limitations of low tooth count pinion sprockets, CPSMD settled on a 13-tooth pinion sprocket and 195 tooth rear sprocket. To run our desired #25 chain with a 13 tooth driving sprocket, we will have to specially request another custom-made sprocket from SMC. If we are unable to custom order this part, we may have to resort to a #35 chain.

### 4.2 Rear Sprocket

#### 4.2.1 Current design and redesign

With such tight tolerance goals, sprocket stiffness was called into question as an area for improvement. Supermileage sprockets have ranged in size and material over the years. When a single-stage system was initially attempted in 2016, a 14-inch diameter, steel sprocket was used for a gear ratio of 14:1. Due to reliability issues, the team moved to a two-stage system with a 7-inch titanium sprocket. It was thought that chain misalignment was causing the sprocket to deflect, so by reducing its size and increasing its strength the tip displacement could be reduced. It is the intent for the 2018 drivetrain to go back to a single stage design but now use a gear ratio of 15:1. Considering these deflection issues, various sprocket concepts featuring thick bodies with thin tooth rings and carbon fiber composites were brainstormed.

#### 4.2.2. Sprocket testing

For CPSMD to effectively employ a single-staged design with an even larger diameter sprocket, the theory of sprocket deflection had to be validated. The required testing involved a calculation of the deflection seen by the sprocket tip as applied by a misaligned chain. Appendix E details these calculations and the whole testing procedure. The results showed that under the most extreme of loading, there was no noticeable deflection in the steel sprocket.

Having removed deflection as an area for improvement in sprocket design, it was then theorized that sprocket flatness might become a limiting factor as alignment becomes more precise. Research suggests that typical sheet metal has a geometrical tolerance for flatness around 0.015 inches. In I-units of flatness, as defined by Contrarian Metal Resources in section 2.10, sheet metal may range from 17-27 I-units [11]. Placing the sprocket on a flat table, a dial indicator was run along its outer edge. This revealed a variation in height of 0.0155 inches along the sprocket face agreeing with industry standards. Since the target for CPSMD is to have a perpendicular sprocket misalignment within 0.02 inches, this lack of flatness in sheet metal proves to be a significant area that requires improvement.

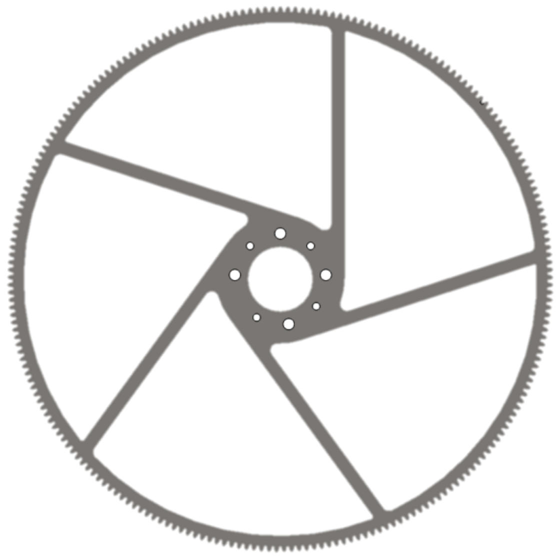


***Figure 11 Rear sprocket concept design made of titanium with 195 teeth***

### 4.2.3 Sprocket Design

Sprocket flatness has been found to be a difficult challenge to confront. Three promising concepts have been developed to combat this challenge. One solution is to use incredibly flat sheet metal. Contrarian Metal Resources has the capability to provide sheet steel and sheet titanium with an advertised 5 I-units of flatness. Considering that waterjet cutting the 14-inch steel sprocket did not decrease its flatness beyond industry standards, it is assumed that after a waterjet process, Contrarian's sheet metal would retain its flatness. However, this would have to be confirmed with supporting measurements before committing to the use of such a sprocket. If need be, wire EDM could provide a low stress process of machining to retain flatness. A limiting factor of this choice is the cost of such flat sheet metal. A line of contact between CPSMD and Contrarian has been established but the price of these sheets has not been determined. If Contrarian's price is outside of CPSMD's budget, we will have to explore other options. It should be noted that Contrarian does have a history of supporting students and may be willing to engage in sponsorship.

Should Contrarian be outside of our price range, another option is to use a steel sprocket which can be ground flat. The sheet steel would first be waterjet cut to shape and then ground down to an acceptable flatness. The capability to grind steel flat is incredibly high and Cal Poly's Formula SAE teams have a working relationship with such a company that possesses this capability.

The last manufacturing option being considered is a post annealing process. A sprocket would first have to be cut to shape and then it would be placed into an oven while between two heavy flat surfaces. The oven would heat the sprocket up to its annealing temperature and the flat faces it is sandwiched between would force the sprocket flat. After removing it from the oven, the sprocket would be slow cooled at room temperature while still between the flat surfaces.

CPSMD's plan moving forward will be to continue our contact with Contrarian in the hopes that we may be able to acquire some of their highly flat sheet metal. In the meantime, research into annealing temperatures, heavy flat surfaces, and ovens large enough to house a 15.5-inch sprocket will be done. Details and contact information about steel grinding companies will also be obtained from the Formula Team so that preparations may be made should all else fail.

## 4.3 Freewheel and Rear Hub

### 4.3.1 Current design

The drivetrain for the last two years made use of a custom-made left hand drive hub by Phil Wood and a freewheel by White Industries. To mate the freewheel to the rear sprocket, an adapter had to be CNC machined out of aluminum. While the hub alone does not offer many areas for improvement, the hub-freewheel assembly certainly does. The freewheel relies on the use of pawls to lock the rotation of the hub and drivetrain under power then unlocks in the fashion discussed in section 2.6.1. It has long been the goal of the Supermileage Team to somehow move away from pawled freewheels in order to improve the rolling resistance of the car. A particularly unattractive facet of this freewheel is the amount of play that is present between the outer and inner race of its housing. Modifications of the freewheel have the potential to dictate hub designs and sprocket integration.

### 4.3.2 Freewheel Testing

To determine if the freewheel play was large enough to be a concern, the same test used to determine sprocket deflection was used and is presented in Appendix D. This test showed that the freewheel alone was responsible for all sprocket movement and allows for 0.26° of total angular movement or 0.13° of movement from the center. With a sprocket of 15.5 inches in diameter, this play translates to a total tip movement of 0.0352 inches or 0.0176 inches of movement from center. Again considering our sprocket misalignment goals, this is a critical area for improvement.

### 4.3.3 Odyssey-clutch hub

The most promising method found to reduce play in the hub-freewheel assembly is to remove the freewheel all together. This can be done using a new style of hub called a freecoaster. The concept of a freeecoaster, as described in section 2.6.2, is to have a clutch mechanism that will disengage the hub from the entire drivetrain when coasting. This would allow all wheels of the car to coast on nothing but bearings solving both the issue of efficiency leaching pawls and the issue of play from the freewheel. One such freecoasting hub is the Odyssey Freecoaster V2 pictured below in Figure 12. At the time of this document, the hub has been purchased and is being shipped to CPSMD. Once the hub arrives, several important steps can take place to determine if the hub will be a suitable replacement for the Phil Wood and White Industries set up. The freecoaster must be tested in a similar way that the freewheel was to quantify improvements. Additionally, a new design for an adapter to integrate the rear sprocket to the Odyssey hub will need to be made. It is expected that this adapter will follow a similar solution to the current adapter, but exact dimensions and any alternative approaches cannot be determined until the hub arrives.

*Figure 12 Odyssey conical clutched hub cut away*

## 4.4 Chain tensioner

The current design does not use a chain tensioner due to the two-stage jackshaft design. Moving towards a single-stage system with larger reduction, proper chain performance and minimization of the risk for chain throwing is very important. A chain tensioner can accomplish these goals by effectively reducing vibration in the chain with the introduction of another node. In doing so, you reduce the possibility of the chain missing the next tooth. However, a chain tensioner adds another source of friction and another rotating mass to the drivetrain. The chain tensioner used for the design would be an off-the-shelf part that is readily compatible with the #25 chain in the system.

In order to decide whether a chain tensioner is worth adding or not, testing will be necessary. One way to measure chain tension is to treat the rear axle as a cantilever beam with one fixed end and one free end with a force gauge attached. Then the chain can be loaded with a set tension and the free end can be reattached to perform a number of runs and measure the chain tension after the runs to see how much the chain elongated or loosened. This process can then be repeated with the chain tensioner and the results can be compared to help quantify how much of a difference a tensioner makes. Figure 13 below delineates the important components of the testing set up.

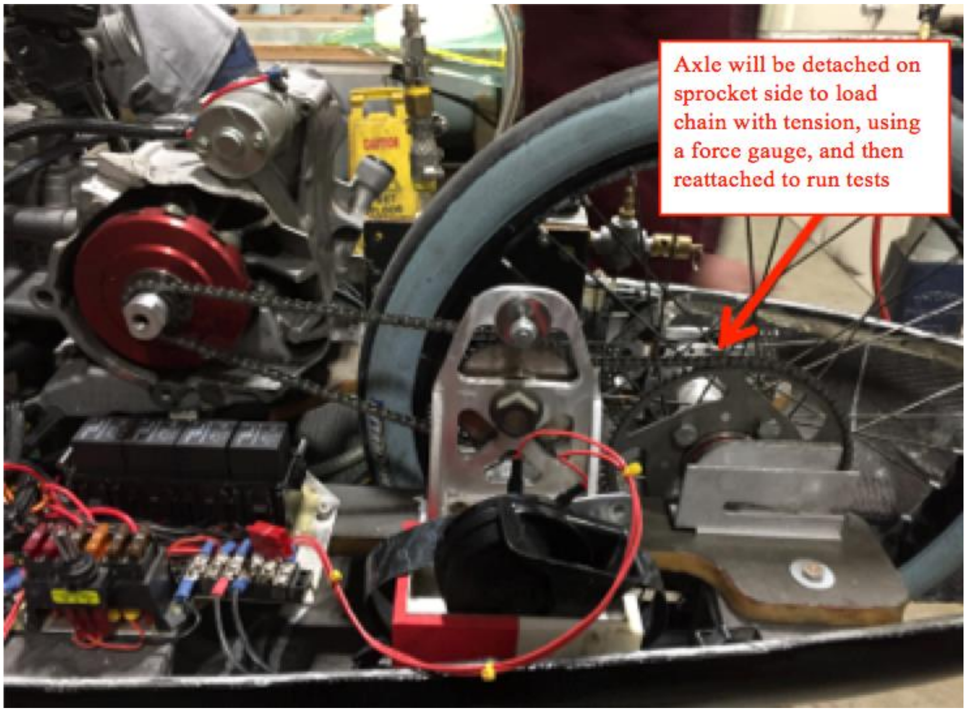



*Figure 13 Chain drive system showing important test components*

Another way to analyze the impact of an added chain tensioner is to analyze its effect on the vibration of the chain. The natural frequency of the tight end, shown in Figure 14 below, of the chain is expressed by equation 5 below where *F* is the tension, *m* is the mass per unit length, *L*, and *k* is the mode number.

$$f_c = \frac{k\sqrt{\frac{F}{m}}}{2L} \tag{5}$$

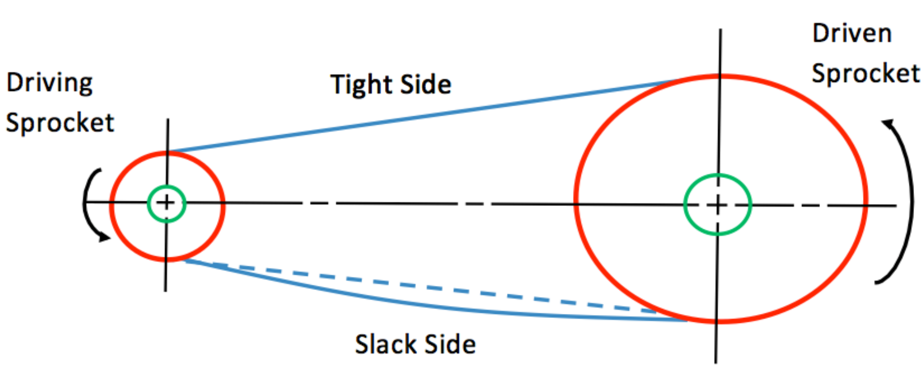



*Figure 14 Chain drive system showing tight and slack side of the chain*

For the tight side of the chain, there is a range of resonant frequencies given by equation 6 below

$$f_{range} = (1.1 \rightarrow 1.2)\frac{k\sqrt{\frac{F_c + F_{cf}}{m}}}{2L} \tag{6}$$

Where $F_c$ is the tight span tension excluding inertial contribution. Using results from the test done with the force gauge, equation 5 and 6 can then be used to see which tensions, with and without a chain tensioner, stay within the range of resonant frequency. This can also help find the optimal tension of the chain for the system by ensuring it keeps the chain resonance within the allowed range. For normal drives, adequate slack should be adjusted to 4% of the chain span and a chain tensioner would be able to easily ensure this. Another reference for proper chain tensioner is shown below in Figure 16. As seen, proper chain tension should keep the slack side of the chain within 2% or ¼" of the sprocket centers.

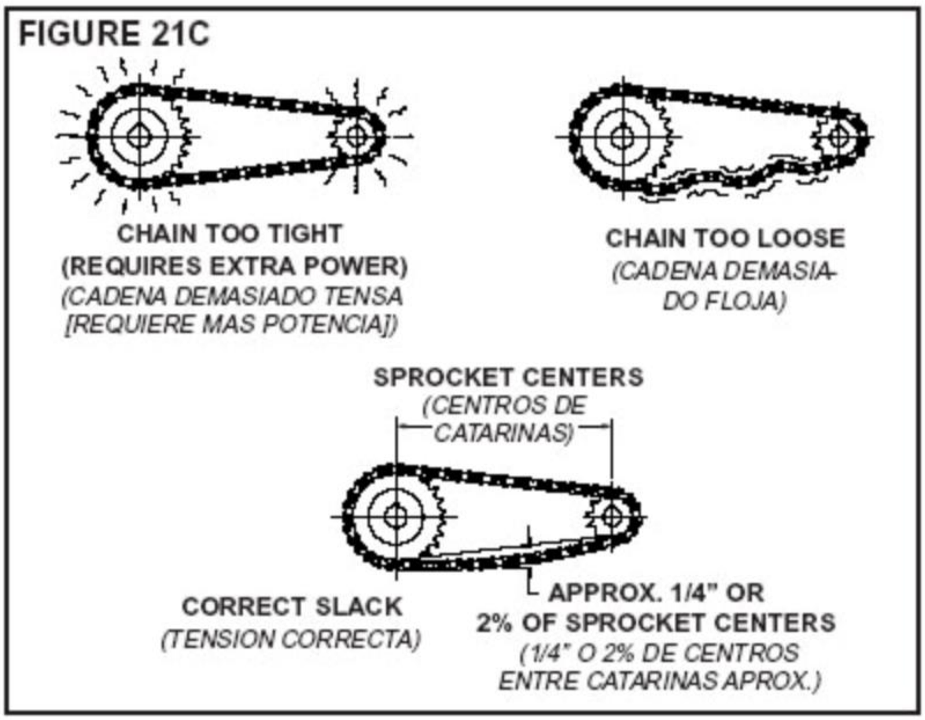



*Figure 15 Diagrams of different chain tension scenarios*

The combination of tension testing, vibration calculations and resonant ranges, along with literature references for chain tension will help in finding the optimal chain tension. In the past, optimal chain tension has been found during competition and driven by the need to go tight to keep the chain from falling off and we would like to have a more quantifiable way of finding it and the processes outline in this section are a good step towards this goal. In addition, our ideas to improve alignment will help allow lower and correct tension to be run.

## 4.5 Drivetrain Plate

The modular drivetrain plate for the 2016-2017 car was an incredibly valuable component because it gave us the ability to quickly and easily remove the entire powertrain assembly from the car. This allowed the powertrain team to work on it separately and allowed the chassis team to easily make emergency repairs. With this in mind, the decision to retain a modular mounting feature in the 2018 car was made. Since the current plate's lifetime has run its course and the 2018 car's engine compartment dimensions have changed, a new plate needs to be made. Working concurrently with the chassis team, a new geometry was developed for the plate that conformed to the new chassis dimensions while still allowing for all powertrain components to be mounted. Figure 16 below shows the old drivetrain plate compared to the newly designed one.

Despite the change in plate geometry, the manufacturing process will remain the same. Stock balsa wood will first be waterjet cut to shape. Separately, carbon fiber face sheets will be laid up on a flat table and once cured, will also be waterjet cut to shape. New locating brackets that position the plate relative to the chassis will be CNC machined out of aluminum. The brackets will be installed into the balsa core, film adhesive will be applied, and the carbon face sheets put into place. This assembly will then be cured in Cal Poly's autoclave using negative pressure. Features for the dropouts can then be machined before finishing the drivetrain plate with an epoxy coating applied to the outer edges.

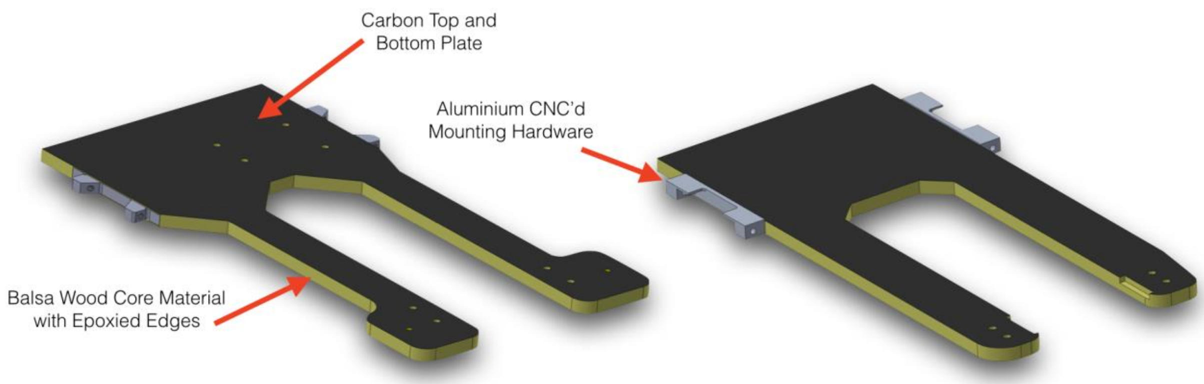



*Figure 16 (left) Current drivetrain plate, (right) Redesigned engine plate*

## 4.6 Engine Mount

### 4.6.1 Current design and redesign

The current engine mount, shown in Figure 17 below, is inadequate for multiple reasons and needs to be redesigned. The design currently holds the engine at a skewed angle, creating misalignment with the chain. Due to our large efficiency goals, any amount of misalignment between the chain and corresponding sprocket is quite bad and has a large impact. Our goal in the redesign of this part is to ensure proper alignment between the driving sprocket and the driven sprocket. For this reason, the redesign of the plate will be heavily dependent upon the method used to align these components, which will be discussed at length below in sections 4.5.2 and 4.5.3. Along with this, the current design is also fairly heavy with unnecessary material in places such as the floor that are not load or force bearing. This can be removed strategically to dramatically cut weight. Lastly, the hole placement securing the engine mount into the drivetrain plate will have to be rearranged due to the changing dimensions of the drivetrain plate. This will be an easy fix, though we will have to ensure proper loading can be achieved to reduce the amount of shear seen by the drilled carbon holes in the plate. These holes will be moved mainly to ensure they are accessible while the engine is mounted. The new design is shown in Figure 18. It will need to be slightly heavier at .92 lbs. versus the current 0.91 lbs. but will have a larger footprint to mount to the drivetrain plate, giving a sturdier base for the engine to sit on.

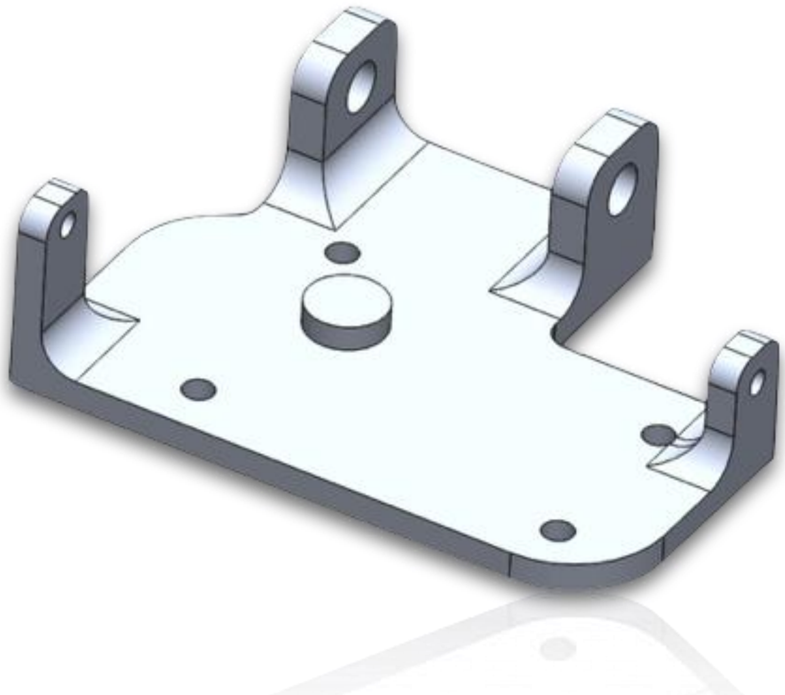


*Figure 17* ***Current engine mount***

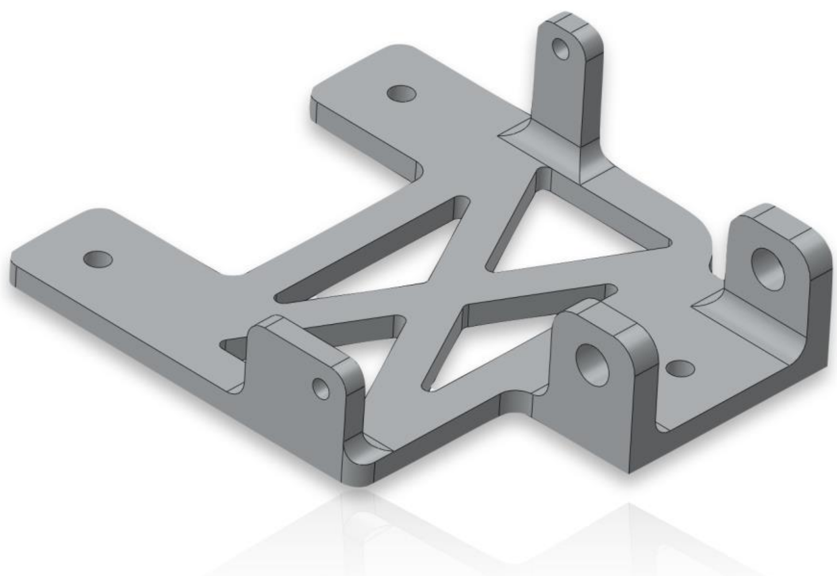


*Figure 18* ***Redesigned drivetrain plate with accessible mounting holes and lightening ribs***

### 4.6.2 Locating pins

With the need to ensure proper alignment in our chain drive system, comes the need to effectively and consistently place the engine mount on our drivetrain plate. Locating pins, like that shown in Figure 19 below, are designed to accurately locate mounting components. If the alignment of the chain drive system can be achieved with high accuracy upon initial installation, these pins will greatly improve the location of the engine for every subsequent install onto the drivetrain plate. Holes for the sheath of the locating pin will be drilled into the composite drivetrain plate in place where the engine mount is likely to go and the sheaths will be inserted. Once the mount is positioned where it is best aligned with the drivetrain, the engine mount will be locked down. This can be achieved through oversized fastener holes or through clamping. Holes for the heads will then be drilled into the metal engine mount using the sheaths as guides. The heads will then be pressed into the engine mount.

To maximize the effectiveness of these locating pins, once the drivetrain is aligned upon the drivetrain plate the engine mount will become a permanent part of the engine block until after competition. Any modifications to the engine's block and crankshaft must be completed by this time. With the use of a chassis dyno, continued engine tuning may occur and the head, cylinder, and piston potentially could still be modified.


*Figure 19 Carrlane locating pins. Head and shank are concentric to within 0.0005".*

### 4.6.3 CMM Alignment

The angular and perpendicular distance offsets between the drive sprocket and the driven sprocket is highly important to the goals of our team. The method in consideration to ensure proper sprocket alignment is to use the Cal Poly IME Department's Coordinate Measuring Machine to accurately relate the two sprocket faces to one another. This concept would be used in one of two ways, outlined below. This will determine the placement of the engine and once the proper placement is established, the locating pins described in section 4.6.2 above will be used to ensure the reliability and repeatability of this alignment.

1. Active Alignment: The first method in question will use the CMM to align the motor while it is placed on the drivetrain plate. The drivetrain plate and all of its installed components will be placed on the CMM. The bushing of two locating bullet pins will be placed into the composite drivetrain plate in places where the engine mount is likely to go. Additionally, oversized fastening holes will be drilled into both the engine mount and the drivetrain plate through CNC. The CMM will then be used to create a datum plane along the rear sprocket. Continual sweeping of the driving sprocket will then be used to determine their alignment. A mallet would be used to lightly readjust the engine's position, then the sprocket will be swept again. Once the mount is positioned where it is best aligned with the drivetrain, the engine mount will be locked down. Holes for the heads of the locating pins will then be drilled into the metal engine mount using the bushings as guides and the heads will be pressed into the

engine mount. The most limiting feature of this method is the ability to properly drill the holes for the pin heads into the engine mount. It is intended that by using the bushings as guides, the pin head holes can easily be hand drilled. Hand drilling involves low precision bits and introduces a path for human error. In theory, this method should allow us to obtain an accuracy of roughly 0.003", which is within our specifications limit.

2. Preemptive Alignment: This method will use the CMM to determine the location of the face of the driving sprocket in relation to the holes in the engine mount. We will machine the engine mount and bolt in the engine. This allows the use of the sprocket side mount face as our machined datum. The CMM will then be used to accurately locate the sprocket face to the same datum face in the mount. We will then update our Solidworks drawings to include the sprocket location in relation to the rest of the motor mount. Once this is determined, we will establish the location of the mounting holes to be machined in the drivetrain plate using the rear axle and sprocket as a datum. All components will be placed on the engine plate except for the engine and engine mount and the plate will be placed in the CNC. Using the touch probe on the CNC, we will touch off on the rear sprocket creating the datum for machining the mounting holes. This will eliminate all tolerance error incurred due to the mounting of the rear dropouts and axle pieces. This will obtain an accuracy of roughly 0.003", which is easily within our allowable specification limits. The main drawback to this method is the fact that once it is performed, any unexpected misalignment cannot be corrected. While it is believed that a full understanding of the associated tolerance stack-up has been achieved, there is always the possibility to overlook something and this method's permanence will not leave room for error.

### 4.6.4 Repeatability Testing

Once a mount is designed and manufactured, we hope to gain an understanding the amount of variance allowed by our design when removing and replacing the engine in the car. To do this, we hope to use the CMM to measure the axial and angular alignment of our two sprockets. With the whole system on the machine, we would repeatedly remove and replace the engine, checking the alignment each time. In doing so, we would acquire the amount of misalignment in our system due to the engine's placement in the engine mount. If the misalignment is found to be larger, we would then need to ensure that the drivetrain mount is designed to be permanently installed so that the alignment of the system holds.

## 4.7 Rear axle drop-outs

The current rear axle drop-out design is shown below in Figure 20 and employs a c-slot for flexible for/aft rear axle location. This was needed in the current design in order to allow for proper tensioning of the two chain system. In our redesign, we hope to employ a single drive system with a fixed center to center distance dictated by our chain length and proper tension required for optimal performance. Due to this fixed length design, we are able to use a vertically slotted mount. This would ensure proper alignment in our drivetrain with repeated removal and replacement of the rear axle and wheel components, which is inevitable during competition. The vertical slot would also reduce installment time due to the self-aligning characteristics of the design. Using a fixed position design with vertical slot also allows us to design a stronger mount with decreased material as shown below in Figure 21. With a larger cross sectional area throughout the load bearing neck of the mount due to the rib, we are able to significantly increase the ability of the member to resist bending and failure while reducing weight. A visualization of the stresses in the new mount due to the static loading of the car's weight is shown in

Figure 20. Currently, our new design would reduce weight by roughly 40% based on volume, though this will change based on final design geometry.

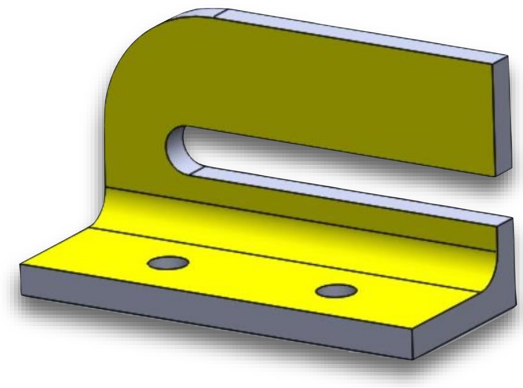


*Figure 20* ***Current rear axle drop-out design with c-slot***

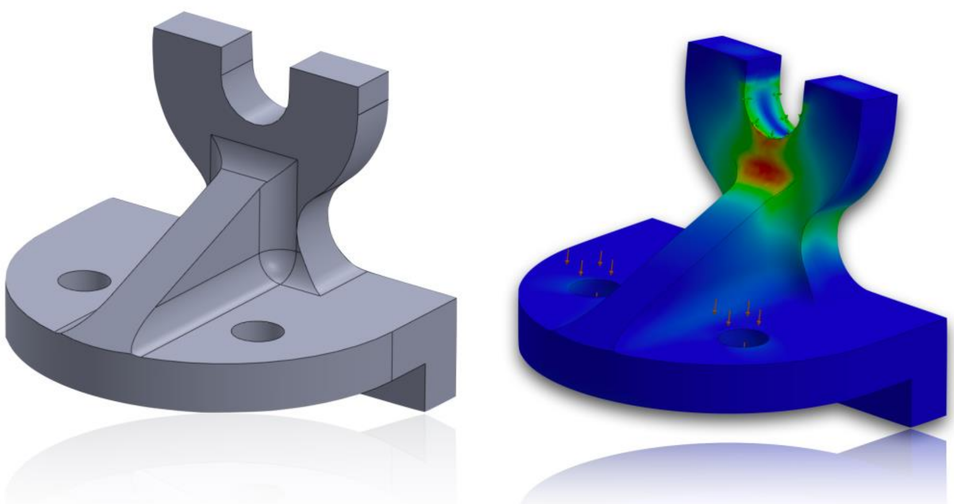


*Figure 21* ***Redesigned systems of the rear axle drop-outs***

## 4.8 Overall Redesign

The current concept for the overall redesign of the Cal Poly Supermileage Vehicle Team's drivetrain is shown in Figure 22 below. Featured is a rough model of the engine, the Vortex clutch, a drivetrain plate conforming to the new chassis size constraints, an engine mount with no hidden fasteners, a 15.5 inch 195 tooth rear sprocket, a #25 chain, the new dropouts, and the wheel. Due to the lack of a proper model for the Odyssey hub, the Phil Wood and White Industries freewheel were used as place holders.

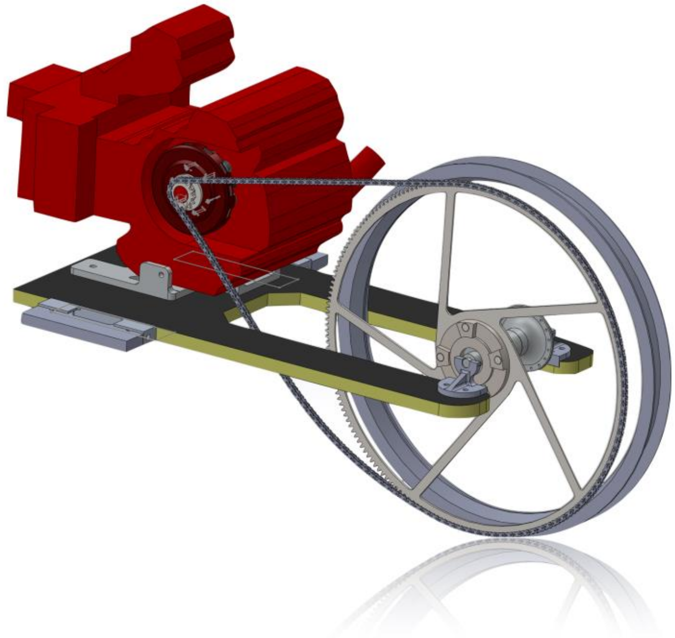


*Figure 22 Full assembly of the redesigned 2018 Cal Poly Supermileage Vehicle drivetrain*

While it is expected that iterations and redesigns will continue to occur over the course of the project, the design concept for several of the drivetrain components, such as the drivetrain plate, the engine mount, and the dropouts have been furthest developed. The design of the rear sprocket has been worked out but the final method of manufacturing has yet to be determined. Another area still up for debate is the method for aligning the front and rear sprockets onto the drivetrain plate. The final design decision about the chain tensioner is a case which is still pending testing data. Each of these design decisions may not be settled upon at this current time, but methods of approach for each are outlined or discussed in their respective sections above. The component that is most currently up in the air is the Odyssey hub. The decision to use the hub and designs of components dependent on its dimensions are on hold until the hub arrives in the mail. Once it arrives and is tested, final design considerations may begin.

# 5.0 Final Design

## 5.1 Overall Final Design

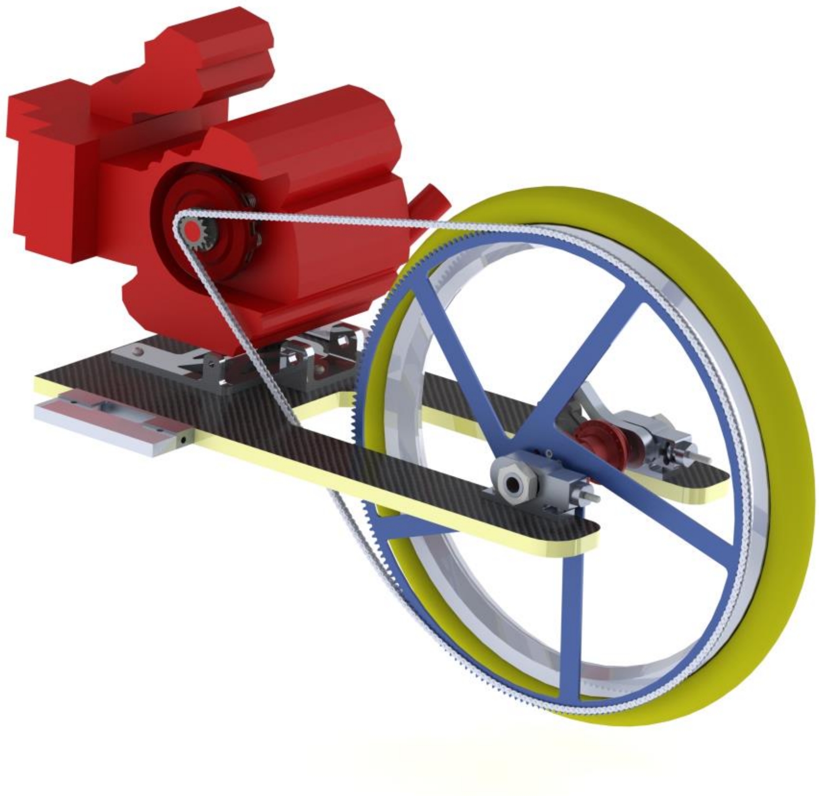


*Figure 22: Cal Poly Supermileage Drivetrain Assembly final design*

The team worked hard from the Preliminary Design review on June 6th, 2017 up until our Critical Design review on October 7th, 2017 to refine our design while also performing necessary analysis. The team was able to come up with a final design, shown in Figure 222 that we are confident will meet all of the specifications outlined in section 3.

## 5.2 Changes From PDR to CDR

Moving from Preliminary Design to Critical Design, alterations were made to both our project scope as well as the overall design in order to better our finished product. Originally, it was not made clear by the Supermileage Vehicle Team that we would need to incorporate the rear brake system into our design. After becoming aware of this, we have now enlarged our scope to cover both the rear brake integration as well as the rest of our drivetrain system and its components. Though this is not a simple task, we have begun and will continue to work closely with the team's brake system team to ensure proper integration into our system. A rough idea of our designs for the brake disk mounting solution are outlined below and this will be an ongoing issue carried forth from here on out.

Another large change in our design moving from our Preliminary Design into our Critical Design is in the rear dropouts. It was proposed to us during our PDR to incorporate the chain tensioning system into our rear dropouts. This required a major overhaul of the dropout design which is outline below in section 5.5. other than this, most of the changes came from smaller tweaks and iterations of our Preliminary Design in order to ensure the best designs moving forwards.

## 5.3 Engine Mount

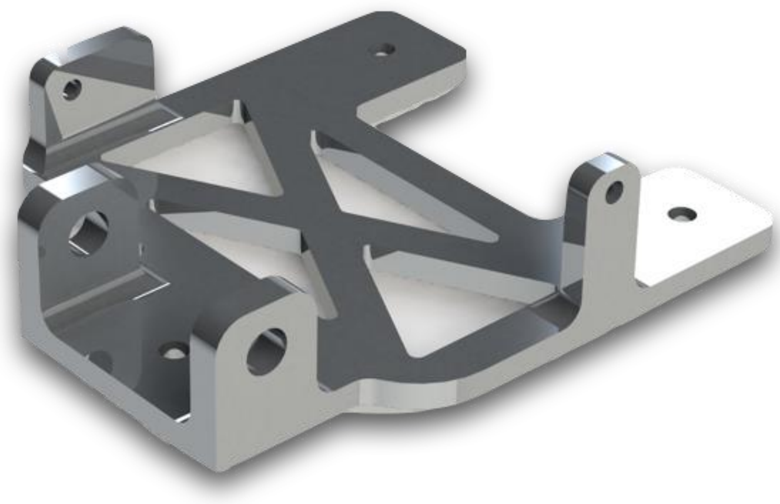


*Figure 23: Engine Mount finished design.*

The engine mount, shown in Figure 23, is a single bracket used to fasten the Yamaha engine onto the drivetrain plate. It will be CNC machined out of a single piece of 6061-T6 aluminum. The design of the mount has not changing much from the prior design as we are using the same model of motor and therefore the mounting geometry has not changed.

We plan to have the engine and engine mount designed to never be removed from one another in an effort to reduce alignment error in our system. The Yamaha engine is not designed for high accuracy installation and therefore is not built with adequate datamable surfaces to use for repeatable alignment and instillation. Therefore, by integrating the engine mount into the engine case we will in turn create the sufficient datamable mounting locations.

With this in mind, the main difference in the new design is the relocation of the fastening holes for the mount to drivetrain plate interface. We have chosen to move these points out from under where the engine will sit as to allow them to be accessible while the engine and mount are together as a single unit.

Stress analysis was done to account for the engine's weight as well as its output torque. We focused on the thinnest mounting upright and used the full weight of the engine and the largest output torque as this would be the worst-case scenario for the engine mount. We found there to be a massive safety factor, SF = 14000, for yielding which is ideal as our part must survive the life of the engine. This is quite large and caused us to look into other materials for the engine mount; however, with consideration for bending deflection, fatigue, and after speaking with the Supermileage team, it was found that with the team's budget and large stock of raw 6061-T6 aluminum blanks in various sizes, we would to stick with the aluminum.

## 5.4 Drivetrain Plate

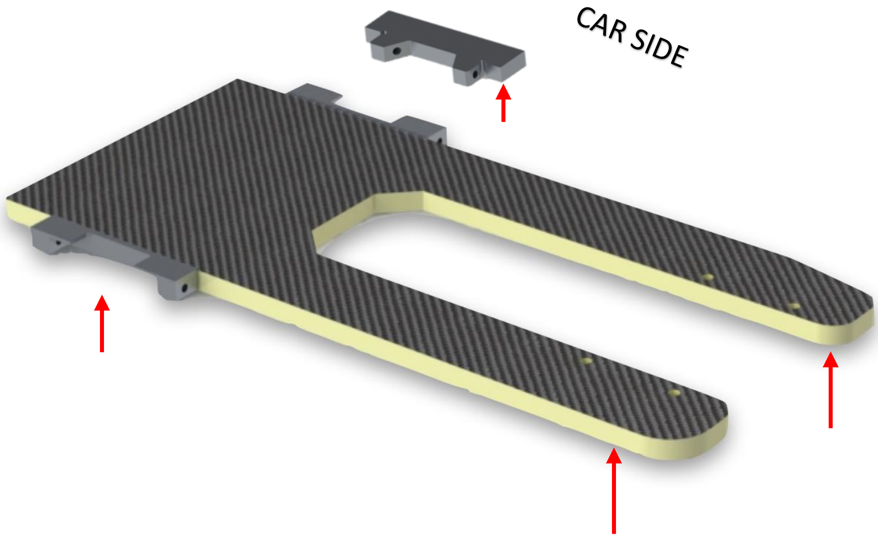



*Figure 24: Drivetrain plate illustrating support locations.*

The Drivetrain Plate provides a platform on which all of the drivetrain components can be attached. Its job is to then secure this drivetrain assembly within the chassis of the car. By having all of the drivetrain components mounted upon this single plate, a certain modularity to the drivetrain system arises. This proves to be a valuable feature because the drivetrain can be assembled and tested independent of the rest of the car. Additionally, in the event of any damage on either the drivetrain or chassis side, it can be removed and allow for easier access to whatever needs attention. The overall profile and geometry of the plate were determined by working in close relation with the new chassis designer so that it, and all things mounted on it, would fit inside of the car.

As indicated by the arrows in Figure 24, the plate is fastened to the car in four locations: at the two CNC machined brackets and at the two ends of the legs. The two CNC machined brackets serve as support and locating features while the legs serve as additional points of support and provide pathways for the road forces to be directly transferred from the wheel to the chassis. A force analysis, found in Appendix K, was conducted on the torsional loads that may be induced upon the plate while cornering. The maximum cornering load resulted in a torsional stress of 58 psi, well below the effective strength of the carbon balsa composite. Once all the components are located relative to each other onto the plate, only the plate needs to then be located inside of the car relative to the front wheels and the car center line. This chassis alignment is accomplished by working with the Supermileage team member responsible for car integration. The plan is for this team member to manufacture a jig which will mate to the rear dropouts and suspend the front of the plate, thus fixing the plate in a position where the car side of the aluminum brackets can be bonded into the car.

The plate itself is made out of a carbon fiber and Divinycell foam composite with localized Garolite pieces. The Gaorlite's purpose is to give added compressive strength at the locations where components will be bolted onto the plate. The carbon fiber face sheets have already been laminated by Tencate in their facilities and will be cut to shape using a water jet. The foam core is to be waterjet to shape and, as mentioned above, the brackets are CNC machined out of aluminum. Once the brackets are made, they will have to be treated with Alumiprep to ensure a successful bond. All of these are then bonded together using a film adhesive and cured under vacuum. The final holes and locating features for the drivetrain components are CNC drilled and reinforced with “top hat” style bushings.

## 5.5 Dropouts

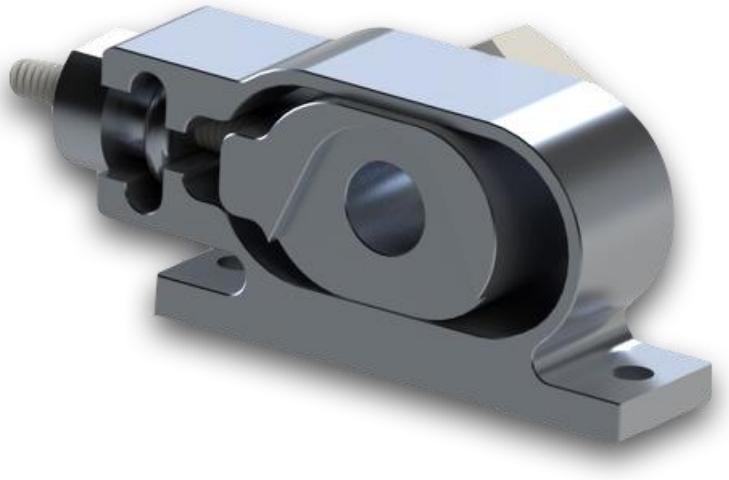


*Figure 25: Rear dropout assembly. Consists of 4 pieces: dropout mount, dropout slide, adjustment screw, locking nut.*

### 5.5.1 Overview

As seen in Figure 25, the rear dropout assembly consists of 4 separate parts, creating a single dropout unit. There will be separate, unique assemblies for the left and right of the vehicle. The main purpose of the dropout assembly is to secure the rear axle assembly onto the drivetrain plate. Our main concern is keeping alignment accurate during repeated removal and installation of rear axle components.

### 5.5.2 Dropout Housing

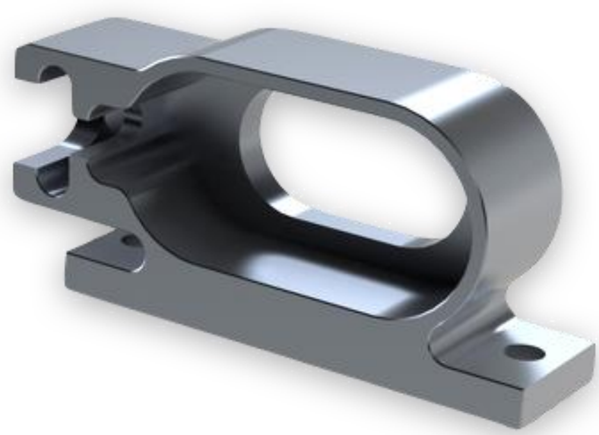


*Figure 26: Rear dropout housing.*

The dropout housing, shown in Figure 26 above, is the nonmoving component of the overall dropout assembly. The housing will be the main structural component of the dropout as well as locate the assembly on the drivetrain plate. For this reason, it is imperative that the housing is made accurately. To do this, we will CNC machine the housing to a ±0.001” tolerance. The lower threaded pegs will be used to both locate the housing on the drivetrain plate as well as secure it to the plate using ¼ - 20 bolts. The main reason for using ¼ - 20 bolts is that the Supermileage team tries their best to centralize the fasteners used to ensure easy maintenance during competition. Most fasteners on the vehicle are ¼ - 20, and therefore we did our best to use the same standard wherever possible.

When using FEA modeling to look into the localized stresses in the dropout mount due to the vehicles weight and cornering forces, we found the stress concentration to be in the right edge of the part. We then did hand calculations to look more in depth into the stresses in this section and found a SF = 3000 for yielding. In this way, we determined the design to be more than acceptable for holding its application loads. With a SF this large, we did look into making the part out of several other materials; however, again due to the availability of the aluminum, its ease of manufacturability, and the very low impact to weight that changing the material would have, we decided to stick with the 6061-T6 aluminum.

### 5.5.3 Dropout Slide


*Figure 27: (Left) Driveside dropout slide. (Right) Brakeside dropout slide.*

The dropout slide, shown in Figure 27, will be used to secure the rear axle to the dropout housing, and in turn securing the axle in the vehicle. As the name suggests, the dropout slide will slide within the dropout housing, allowing for a horizontal, fore/aft, movement of the axle of 75mm. This adjustment length was specifically chosen to allow for a change in chain length of ± 1 chain link length. This allows for an great amount of tension tuning during the installation of the rear axle.

The dropouts slides will be CNC machined out of 7075-T6 series aluminum instead of 6061. The reason behind this decision was due to its higher shear and yield strength. The main reason this is important is that the slide incorporates a threaded boss that the brass locking nut will thread upon to secure the position of the slide within the dropout mount. The intent of using a brass nut is so that it would yield under over torqueing instead of the aluminum threaded boss. This feature is important since the brass nut is an OTS component with the aluminum slide is custom made. We looked into the max torque spec allowed before stripping the slide's boss and (using a safety factor of two) found this to be 130 ftlb for 6061 and 210 ftlb for 7075 compared to the 150 ftlb yield torque for the brass nut, the calculations of which can be found in Appendix K. This confirms that 7075 will be the choice for our dropout slides and should be more than acceptable for securing the slide in the housing as well as protect the slide from possible over-torqueing of the locking nut.

The threaded adjustment stud will be bought off the shelf from McMaster-Carr in a ¼ - 20 size and be threaded into the dropout slide. This will greatly reduce the manufacturing difficulty of the part and allow us to machine the dropout slide ourselves as opposed having to outsource the job. This will shorten manufacturing time and reduce costs dramatically.

### 5.5.4 Adjustment Nut

The adjustment nut will be used to adjust the position of the slide horizontally in the dropout housing in order to tension the chain. The nut will be CNC lathed in house out of 6061-T6 Aluminum. The manufacturing of the part will be quite easy. Because of this, we intend to make several replacement parts incase anything is to happen during installation or testing of the vehicle. Remaking the parts at a later date would prove to be slightly more difficult than doing it now and therefore having extra on hand will prove to be beneficial in the long run.

The adjustment nut will only see load during adjustment of the rear dropout as once the axle location is determined to be correct, a locking nut will be used to secure the slide and the housing together removing any loads seen by the screw. The tensioning loads are very small as we run a very loose chain and therefore the aluminum ¼ - 20 threading should be more than adequate.

### 5.5.5 Locking Nut

As discussed previously, the dropout slide and dropout housing will be secured together using the 7/8 - 14 locking nut. The main reason for using a separate locking nut to lock the slide in the housing as opposed to using the axle's through-bolt is to allow for the removal of the rear wheel while ensuring the mounting location, and therefore the chain tension, doesn't change. This is essential for our design as to meet the Supermileage desired specifications of quick removal of rear wheel and high accuracy of chain alignment, the mounting location of the rear axle must be self-retained.

The locking nut will be bought off the shelf from McMaster-Carr and is chosen in a standard size to allow for quick and easy replacement in the event the part is misplaced or damaged. As stated before, it is usually advised to design the nut of a bolted joint to be made of softer material than the bolt or stud. To do this, we would need to use a material with a yield stress smaller than that of the 7075 aluminum used for the dropout slide. We determined that a brass 7/8$^{th}$ nut would be a feasible option as its shear strength is 34100psi which is less than the 7075's 48000psi allowing it to yield before our dropout slide. After doing similar torque out calculations as was done on the dropout slides, a torque of 150 ftlb with a SF of 2 was found. This should still work well for our application and still have a good chance at withstanding possible over torqueing by inexperienced users during competition.

## 5.6 Sprocket/Hub Assembly

### 5.6.1 Overview

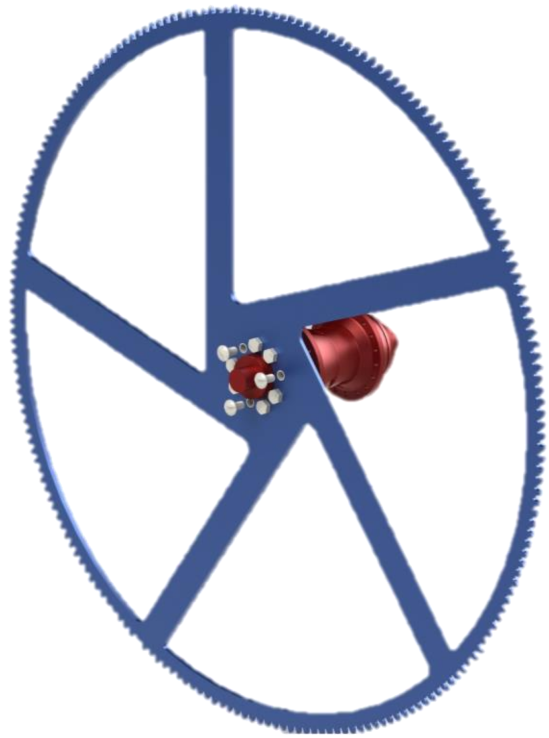


*Figure 28. Sprocket/Hub assembly*

The Sprocket/Hub assembly accomplishes the 14:1 gear reduction and transmits the torque from the engine to the rear wheel. Shown in Figure 28, it features a hub, sprocket, sprocket fastener (hidden), disc brake and adapter (not shown), and the necessary hardware to secure the sprocket to the hub. The main design goal of the Sprocket/Hub assembly is to provide a stiff and rigid assembly which can be located upon.

### 5.6.2 Sprocket

*Figure 29: Rear Sprocket*

The design of the rear sprocket shown in Figure 29 has 210 teeth to give the desired 14:1 reduction with the smaller, driving sprocket of 15 teeth. With sprocket deflection being a very crucial requirement for our overall design, the stiffness and flatness of the rear sprocket design is very crucial. In order, to ensure flatness, the sprocket will be made from steel that has been stretcher leveled. Stretcher leveling is a process of stretching sheet metal beyond its point of yield in order to eliminate any internal stresses in it and therefore prevent any spring back from cutting. Stretcher leveling can achieve 0-5 I-units of flatness, which is a superior way of measuring flatness than the standard "variation from flat" that accounts for the amplitude and frequency of shape deviations in the material. Although stretcher leveling will help eliminate any warping due to cutting, the sprocket will also be cut using the low-stress cutting process of water jet cutting that can be done in house at Cal Poly. Carbon steel was chosen for the material due to its superior specific strength and ration of strength to density, along with the fact that the lighter, titanium was outside of our budget.

The geometries of the sprocket were decided based on a combination of stress analysis, testing, Finite Element Analysis (FEA), and weight savings considerations. The calculations done to decide the widths of the spokes can be seen in Appendix K in which the spokes were conservatively modeled as cantilever beams loaded by the force seen normal to the sprocket face giving spoke widths of 0.81". Performing FEA on the sprockets modeled in SolidWorks that mimicked the aforementioned sprocket deflection testing of section 4.2.2 with weights and dial gauges resulted in a rim thickness of 0.51". Using the maximum, worst-case scenario of sprocket alignment from the requirements, a deflection of 0.147 inches was calculated. The FEA showed that the sprocket would only deflect 0.0043in and this is validated by the previous sprocket deflection testing. FEA results can be seen in Appendix L. In addition to the FEA analysis, stress calculations were done on the teeth of the sprocket and safety factors for wear and bending were calculated to be 10.5 and 7.1 respectively, these calculations can be seen in Appendix K.

### 5.6.3 Hub


*Figure 30. Odyssey Freecoaster Clutch Hub*

Featured above in Figure 30 is the Odyssey Freecoaster Hub that is to be used for the new drivetrain. The hub's job is to fasten to the rear wheel and allow power transfer from the drivetrain to the wheel. As described in Section 4.3.3, the Freecoaster has the unique feature of an internal conical clutch rather than a typical freewheel with pawls. As outlined in Appendix E, it was determined that the most significant source of sprocket movement arose from the play between the inner and outer race of the freewheel. Therefore, the driving decision behind using this new hub was the superior rigidity of the clutch design over a freewheel. Upon initial inspection, this clutched hub appears to have no noticeable amount of play between the driving cog and the axle. By performing a test identical to the one described in Appendix E, a quantitative improvement will be determined for sprocket play reduction. A complementary benefit of using the clutch mechanism is the capability for the drivetrain to be completely decoupled from the hub under coasting conditions. This means that for the significant amount of time where the Supermileage car's engine is not running, and the car is relying on nothing but its own momentum, it will be riding on nothing but bearings.

The use of this new hub did introduce new engineering challenges, namely the fastening of the large 210 tooth sprocket to the hub's cog and the mounting of a brake disc onto the hub's outer shell. However, with consideration given to the expected improvement in sprocket-hub rigidity and the added benefit of coasting without energy drawing pawls, it was decided that the hub's benefits outweighed the challenges it introduced.

### 5.6.4 Sprocket Fastener

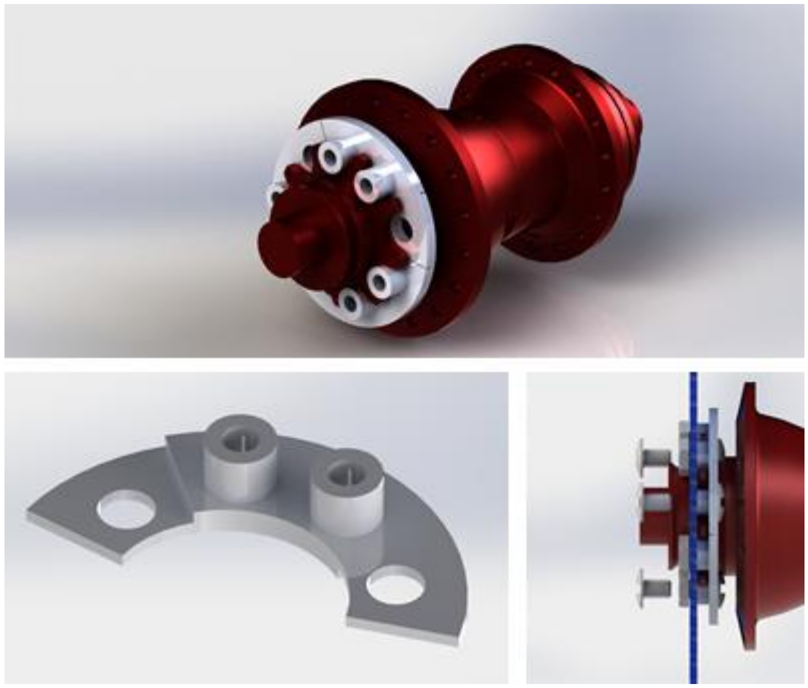


*Figure 31. Sprocket Fastener. The bottom left shows the fastener by itself, while the picture in the bottom right shows a profile view of the assembly*

The first design challenge which the Odyssey hub introduced is fastening the 210-tooth Rear Sprocket to the 9-tooth driver. Because the 9-tooth driver and the internal clutch mechanism are one solid piece, it is impractical to disassemble the whole hub to put on a single piece adapting ring that would secure the Rear Sprocket. Therefore, a three-piece ring was designed that could be installed and removed without the need to dismantle the hub. As shown in the lower left picture of Figure 31, there are two bosses for each of the three pieces to achieve a symmetric fastening profile between the Rear Sprocket and the 9-tooth driver. The bosses have an outer diameter of 8 mm which snugly match the pitch of the 9-tooth driver and feature an M5 internal thread. By threading through the same bosses that fit between the teeth of the driver, the line of action for the resulting force squeezes the Rear Sprocket and the Sprocket Fastener onto the flat faces of the 9-tooth driver. The bottom right picture of Figure 31 shows how this is achieved from a profile view. The three pieces of the Sprocket Fastener are joined together at the through-holes shown by using OTS binding barrels.

The plan is to first use rapid prototyping of the Sprocket Fastener to ensure a proper fit between the bosses and the 9-tooth driver. The three pieces will be 3-D printed and test fit onto the hub, then after any necessary iterations are made, they will be CNC machined out of 6061 aluminum. The machining process will require 2 operations and the use of a simple soft jaw. Consideration for shear forces in the bosses due to the torque of the engine showed that there is 993.3 psi of shear stress. The largest concern for this design is the possibility of stripping the aluminum threads within the bosses by unintentionally over torqueing the M5 bolts. To combat this, helicoil threads can be use in lieu of machined threads.

### 5.6.5 Brake Disc Mount

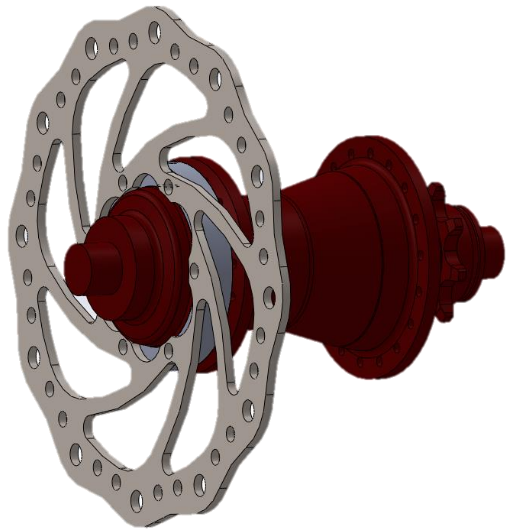


*Figure 32. Rendition of how a brake disc is expected to be mounted to the hub.*

The other design challenge introduced by the Odyssey hub was the incorporation of a brake disc. At the time of this document an exact solution has yet to be finalized, but several possibilities have been ideated. Pictured in 32 above, is a rendition of how a solution is expected to look. The essential requirements of a brake mount are that it can effectively transmit the braking torque to the hub, it can resist axially loading that arises from antisymmetric brake pad squeezing, and it must keep the brake disc concentric with the hub. One of the main difficulties that this hub introduces is a limited amount practical locations to mount the disc. The best location features an inconvenient taper that must be worked around.

Ideally, the brake mount would be removable to permit easy re-lacing of the wheel and hub. In order to accomplish this, some sort of modification to the hub body will most certainly be required. Brainstormed solutions have included machining flats into the taper or turning it down completely. Flat surfaces in combination with pins could achieve the desired effect of axial retainment and torque transmission. However, these are rather difficult solutions to properly achieve when considering how to fixture the hub and how to locate the resulting features. Rapid prototyping has the potential to aid in this process. Two flats could be milled onto opposite sides of the hub and a hole drilled into each flat. Then using an Optical Comparator, the resulting hub geometry can be measured and replicated with a SolidWorks model. From here, a 3-D printed prototype of the brake mount can be made in an effort to match where those holes are located. After several iterations, the brake mount can be made to match the hub geometry and a final version machined out of aluminum. Pins join the brake mount to the hub and the brake disc is fastened to the mount with screws. If all else fails and this solution proves to be ineffective or impractical, a permeant solution can be used where the brake adapter is bonded to the hub. In such a case, the hub would have to be permanently laced and then the brake mount glued on with an epoxy such as Hysol.

## 5.7 Alignment Method

As described in PDR, there were two possible methods for aligning the engine sprocket and the rear sprocket: a preemptive and an active method. It was decided that the better of the two options was the preemptive method due to its lack or iterative process and due to its reliance on CNC drilling rather than hand drilling. Using the CMM, a face on the engine mount (while attached to the engine) will be used as a datum to measure its relationship to the face of the engine sprocket. From this measurement, the Solidworks model relating the engine mount to the engine sprocket will be appropriately updated. With this model, we now know where all locating and fastening features are on the engine mount relative to the engine sprocket are, within the tolerance of the CNC machine used to make the mount. Matching these fastening and locating features on the model of the drivetrain plate allows us to position the engine sprocket in a precise and controlled location. In a separate procedure, the dropouts and hub assembly will be simply installed onto the drivetrain plate. Then the drivetrain plate, with its dropouts and hub assembly, will be fixtured onto a CNC mill. Using the CNC's probe, zeroes in all three planes will determined by touching off on the sprocket's face, the drivetrain plate's surface, and a final location on either the edge of the plate or a dropout. The final location can be arbitrary because its zero merely determines the center to center distance from rear axle to the engine's drive shaft and can be accounted for through appropriate chain length. Once the zero location of the rear sprocket face is determined, the CMM derived engine mount model is used to drill its fastening and locating holes into the drivetrain plate. The holes are to be reinforced with top hat bushings and the locating pins can be installed. The location of the engines sprocket relative to the rear sprocket can be confirmed by probing the two faces. We anticipate that there would be a reasonable amount of difficulty in fixturing the drivetrain plate into the CNC with a 16.7 inch diameter sprocket. To combat this, a smaller sized test sprocket made out of the same flat stock sheet metal can be made and used in its place.

## 5.8 Manufacturing and Assembly Plan

Manufacturing and cam work has begun for the drivetrain plate and full prototypes have been built for the rear dropout assemblies. The cam work for the parts that will be CNC machined is expected to be finished by the end of the quarter so that machining and assembly can begin without impedance next quarter. The individual manufacturing processes for each part has been described above in their individual sections.

The path forward for manufacturing is determined by a critical path in the Gantt char that takes into account the direct relationships between the manufacturing of each part. The initial step of the plan is to procure all purchased parts and materials needed to manufacture the parts and build the assembly. The Odyssey hub has been purchased and received and all other parts have been quoted. All fasteners and the majority of materials will be purchased from McMaster-Carr with the exception of the balsa wood which will come from National Balsa or a similar balsa wood supplier. All quotes for the parts and materials can be seen in Appendix J.

Once the materials have been received, manufacturing the drive train plate will be the number one priority in order to lock in place where the other components will go. Next, manufacturing of the engine mount will be a main focus as it is crucial for the CMM alignment process and testing. An IME faculty member, Trian Georgeou, has agreed to work with Justin to manufacture the parts that will be CNC machined. This seriously decreases the waiting time and overall cost of our project and allows us to directly communicate with the manufacturer before, during, and after the manufacturing process.

Once all parts have been manufactured or obtained, part assembly will begin and upon completion will lead into car integration. The final drivetrain design will then be assembled into the car and integrated into the new 2018 chassis.

Regarding the responsible individuals for the manufacturing process, Heather is in charge of part and material procurement and maintaining a line of communication between the manufacturers, Justin is in charge of machining the CNC machined parts and Mike is in charge of manufacturing the drivetrain plate as well as communication for integration of the parts.

## 5.9 Weight Analysis

Along with high reliability and efficiency, light weight is an important specification of the project and defined as our design meeting and exceeding a target weight of 4.2 + 1kg. The projected weight of the final design is 3.85 kg, below our target weight. A breakdown of the weight of the assemblies and their corresponding parts can be seen in Table 3 below.

*Table 3: Part weight overview.*

| Component | Description | Weight [kg] |
|---|---|---|
| Engine Mount Assembly | Includes engine mount, fasteners, and locating pins. | .44 |
| Drive Train Plate Assembly | Includes balsa core, carbon sheet, and mounting hardware. | 1.4 |
| Rear Dropouts Assembly | Includes rear dropout housing, slide, locking nut, and adjustment screw. | 0.47 |
| Rear Hub Assembly | Includes hub, hub adapter, fasteners, and rear sprocket | 1.5 |
| Chain | Includes #25 chain | 0.04 |
| | Total: | 3.85 |

## 5.10 Cost Analysis

The budget allocated by the Cal Poly Supermileage team is $1,500 and is not foreseen to be exceeded by the project. Table 4 below tabulates the general cost of each component of the drivetrain assembly and a more detailed cost analysis can be found in the Bill of Materials in Appendix J.

*Table 4: Part cost overview.*

| Component | Description | Cost [$] |
|---|---|---|
| Engine Mount Assembly | Includes engine mount, fasteners, and locating pins. | 174.70 |
| Drive Train Plate Assembly | Includes balsa core, carbon sheet, and mounting hardware. | 29.82 |
| Rear Dropouts Assembly | Includes rear dropout housing, slide, locking nut, and adjustment screw. | 11.42 |
| Rear Hub Assembly | Includes hub, hub adapter, fasteners, and rear sprocket | 545.26 |
| Chain | Includes #25 chain | 35.98 |
| | Total: | 796.90 |

It is shown that the cost of raw materials and manufacturing comes to a total of $796.90, way below our budget of $1,500. A lot of cost savings were due to being able to manufacture a majority of the parts in house at Cal Poly, as well as from having some stock material readily available from the SMV club.

## 5.11 Maintenance Considerations

The final, main components of the drivetrain were designed with high factors of safety and are not likely to fail or need maintenance on directly due to factors of the 2018 competition and are expected to be reused in future year's designs as possible. The main maintenance foreseen for the design, is in the fasteners in the individual part assemblies and they can easily be replaced with parts at ready.

If any unexpected maintenance was to be needed, all CNC machined parts would be able to be replaced using the made-available G code and all of the parts are modeled in SolidWorks and can be remanufactured if needed.

## 5.12 Safety Considerations

The manufacturing of the final parts does not create any potential safety hazards and neither does the autonomous final design itself. The main, potential hazard associated with the design is if a critical part were to fail and jeopardize other components of the Supermileage car, possibly endangering the driver. In addition, if the chain were to throw, it could potentially injure someone. Although these potential hazards exist, the risk of them is fairly low as they are unlikely to happen.

In addition to identifying potential hazards with the design manufacturing and final design performance, a hazard check list was completed and can be seen in Appendix N. The check list revealed that there are no potential hazards that would require outside hazard assessment. Furthermore, we will thoroughly test our prototypes to reveal any hazards not foreseen during design and construction.

## 5.13 Design Verification Plan

### 5.13.1 Design Verification Plan and Test Descriptions

A design verification plan was developed to ensure that the drivetrain prototype will meet all specifications and requirements. The plan outlines the various tests that will be performed, the requirement it is testing fulfillment of, the individual responsible for performing the test and analyzing the results of, as well as the timeline of the testing. A breakout of the test plan portion of the design verification plan and report of Appendix M is shown below in Figure 333.

**TEST PLAN**

| Item No | Specification or Clause Reference | Test Description | Acceptance Criteria | Test Responsib | Test Stage | SAMPLES | | TIMING | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | Quantity | Type | Start date | Finish date |
| 1 | 4 | Sprocket/hub diffliection test | < 0.2° | Justin | DV | 1 | DV | TBD | TBD |
| 2 | 2 | Drivetrain weight test/ weight savings | < 4.2 kg | Heather | DV | 1 | DV | TBD | TBD |
| 3 | 3 | Inertia dynomometer efficiency testing | > 80 % | Mike | DV | 5 | DV | TBD | TBD |
| 4 | 8 | Installation alignment repeatability testing - angle tolerance between sprockets | < 1° | Justin | DV | 12 | DV | TBD | TBD |
| 5 | 8 | Installation alignment repeatability testing - horizontal center to center distance between sprockets | < 0.02" | Justin | DV | 12 | DV | TBD | TBD |
| 6 | 6 | Insallation time testing | < 30 minutes | Mike | DV | 12 | DV | TBD | TBD |

*Figure 33: Design verification plan for testing of the drivetrain design*

The first requirement tested is sprocket deflection and the testing procedure was described above in Section 4.2.2. and elaborated on in Appendix A and will be performed by Justin. The requirement of weight will be verified by weighing the entire drivetrain assembly and ultimately calculating weight savings and will be performed by Heather. Efficiency will be tested using an inertia dynamometer that is predicted to be available during time of testing and will be used by Mike. Sprocket and chain alignment will be tested using two separate installation alignment repeatability tests lead by Justin, one for angular tolerance between sprockets and one for horizontal, center-to-center distance between sprockets. These repeatability tests will involve measuring the two different alignments with a dial indicator and length measuring tool before and after installation to ensure that they stay within the specified ranges with installment and removal of the rear axle. Finally, the requirement of total removal and installation time will be tested by having each member of the team install the drivetrain for time and Mike will then ensure that this is within our requirement of <30 minutes and subsequently determine who will be able to repeatedly meet this requirement during competition.

### 5.13.2 Specification Verification Checklist (DVPR)

After all of the tests included in the design verification plan have been performed, the results will be used to prove that all of the specifications have been met and will be reported in the test report section of the design verification and report of Appendix M.

# 6.0 Management Plan

## 6.1 Assigned Team Roles

Roles encompassing certain responsibilities have been assigned to each team member in order to delegate workloads efficiently. The primary, assigned roles are as follows:

### 6.1.2 Justin Miller

Justin will be the main line of communication with the sponsor and all third parties. He will facilitate meetings with the sponsor and inform them of all pertinent information as needed. He is also responsible for keeping all group members up to date with communications with the sponsor and facilitating all general communication between the team.

Justin will also act as the manufacturing lead and be responsible for scheduling the manufacturing of the parts that will be manufactured in house. This will include scheduling the CNC machine in advance to ensure that the manufacturing of the parts will be able to start at the beginning of next quarter. This also means that Justin will make sure that the CAM work will be completed by the end of this quarter as to facilitate the start of all manufacturing.

### 6.1.3 Mike Bolton

Mike will be in charge of managing the team's $1,500 budget and ensuring funds are allocated efficiently. This means making sure that all necessary parts and materials will be able to be purchased within the limitations of our budget, keeping in mind manufacturing and testing costs.

Mike will also act as club liaison to ensure cohesive communication with the club and provide access to the 2017 SMV as well as connections for parts and other club benefits. He will also be responsible for ensuring the new drivetrain system will fit into the new 2018 chassis design.

Mike will also be in charge of manufacturing the drive train plate and ensuring integration of all other components onto the plate and with each other. This means ensuring that the manufacturing of the plate will be at the forefront of the manufacturing plan and communicating with Justin as manufacturing lead to schedule accordingly.

### 6.1.4 Heather Fields

Heather will be in charge of organizing, recording, and storing all pertinent information involved with the project, i.e. research, meeting minutes, documentation. The group has agreed to use Google Drive as storage and sharing means. She will also be in charge of taking meeting minutes and posting them to the drive.

Heather will also be in charge of procuring all pertinent parts and materials. She is responsible for quoting and purchasing the parts and necessary materials that are needed to manufacture all of the parts that will be manufactured in house. She is also responsible for ensuring that the materials and parts will be procured before the beginning of next quarter so that all manufacturing can begin immediately. This process includes updating the created bill of materials, pushing the already received quotes through to purchase orders and keeping track of all parts and materials during and upon delivery.

## 6.2 Project scheduling

A Gantt chart was utilized to create a schedule for the project and ensure that deadlines would be met, this chart can be seen in Appendix H. During the design phase, direct responsible individuals (DRI) were assigned to each major component of the drivetrain assembly. Mike was assigned as the DRI for the drivetrain plate and hub, Justin was assigned to the engine mount and rear dropouts and Heather was given the rear sprocket and the chain. This made it possible to design the components in parallel and streamline the path to the final component designs. Currently, the team is on track with the Gantt chart and has successfully completed the preliminary and critical design phases of each part and will move into the part procurement, manufacturing and eventual testing phase of the chart.

On June 6th, the team presented their design to date during the preliminary design review with the senior project class and advisor. Following the PDR, the team incorporated suggested changes and researched new ideas. During this phase, the DRI's for each part brainstormed design ideas, performed stress analysis for their parts, built SolidWorks models, and researched materials, manufacturing methods, and specific design criteria for their parts.

Once designs were finalized, the team presented their final designs during the critical design review with the senior project class and advisor as well as the president of SMV on October 19th.
During this review, feedback was given to implement changes as necessary before moving into manufacturing of the final designs.

From November 7th to January 8th, the team will be ordering the required materials and parts as well as ensuring that all CAM work is being finished by the end of the quarter as to moved forward with manufacturing next quarter. Once manufacturing is complete, the team will perform testing and verification as outlined in section 5.12. After it has been verified and reported that all specifications have been met or after reworking designs until all specifications are met, the team will update the critical design report to create the final project report. The final project report will document the entire senior project process and outline all changes made during the construction or after testing of our final design. This report will be an extension of the critical design report with added portions for manufacturing, testing, and final conclusions of the project.

Moving forward after the final project report, the team will focus on creating the poster that will be presented at the Senior Design Expo and delivering the poster, final report, and final prototype to our sponsor.

## 6.3 Outstanding Project Tasks

The team has thus far completed the critical design review and report of the final designs for the drivetrain assembly. The path forward is highly focused on manufacturing and following the manufacturing plan of section of 5.7 and Gantt Chart of Appendix H. As aforementioned, finishing the CAM work for all of the parts that are to be manufactured in house is a very high priority along with ordering all parts and materials for manufacturing the finals designed parts. Once manufacturing, assembling, and test verification process has been completed, the drivetrain assembly will be assembled into the car and integrated into the new 2018 chassis.

Outstanding tasks following the manufacturing, testing, and verification processes include finishing the final report project, creating the Senior Design Expo poster, and delivering them to our sponsor.

# 7.0 Product Realization & Manufacturing

## 7.1 Overall Manufacturing Plan

The main goal during manufacturing was to ensure that each subassembly of the drivetrain system was ready and available in time to meet the SMV team's milestones. With consideration to our own critical path for the project, we were able to prioritize the components that needed to be completed first. The drivetrain plate and the dropouts were the first components to be manufactured so that the SMV team could bond the drivetrain-chassis interfaces into the car and align the front and rear wheels. The sprocket was then made and fitment with the rear hub was checked before the sprocket fasteners were manufactured. While the sprocket and hub were being assembled together with the dropouts, the engine mount was being machined. After all components were finished being manufactured, the engine mount was installed onto the engine and dimensions of the two together were taken using metrology techniques. Once these measurements had been taken, we were able to drill the final fastening and locating features into the drivetrain plate and put together the fully completed drivetrain. Engineering drawings for each of the following parts and the assembly can be found in Appendix I.

## 7.2 Dropouts

The dropouts are composed of 3 main pieces, the housing, the slide, and the adjustment screw. Due to the fact that all 3 pieces were designed by us and were one off parts, all three had to be machined by us as well.

### 7.2.1 Left and Right Dropout Mounts

The left and right dropout mounts are mirror images of each other. This made machining them much easier. All machining was done on a Haas VF2 CNC machine and CAM work was all done using HSMWorks.

All operations were first planned out for the left dropout mount and the mount was then machined. Once we ensured that all operations were as we planned and that the part was within spec, we copied the operations onto the mirror of the geometry and reran the part to get the right dropout mount.

The first operation was done to get the bottom surface and mounting posts of the dropout mount. A 5-flute, 3 inch carbide face mill was first used to face 0.01" of stock material from the top of the part to give a nice and flat surface. A size A drill and 1/4 -20 form tap were then used to create the threads for the post. A ¾" flat endmill was then used to machine out the post geometry as well as machine a small corner of the piece to use as a datum in the second operation to ensure proper geometric relationship overall.

The second operation focused on the main profile and inside pockets of the part. This ended up taking 26 individual toolpaths and to ensure the part was machined with the proper geometric tolerances to fit the internal slide. For brevity, not all toolpaths will be covered but a general overview will be given. The part was first faced using the same 5-flute, 3 inch carbide face mill from the first operation. A 1/2” flat endmill was then used to create the larger pocket and slot. A 3/8” and ¼” flat endmill were then used to clean up the sharper corners of the pocket. A ¼” ball endmill was then used to machine the space in which the adjustment screw would eventually be placed. Lastly, the ¾” flat endmill was used to hog out the outer geometry of the part as well as machine the lower flat surface of the part.

The third and final operation was the simplest. The 5-flute, 3 inch carbide face mill was used to remove the 0.100” stock that was used to hold the part in the second operation, revealing the through slot and the outer part geometry. All in all, the part took roughly 2 hours of machining time. A picture of the end result from each operation is shown below along with the final part.

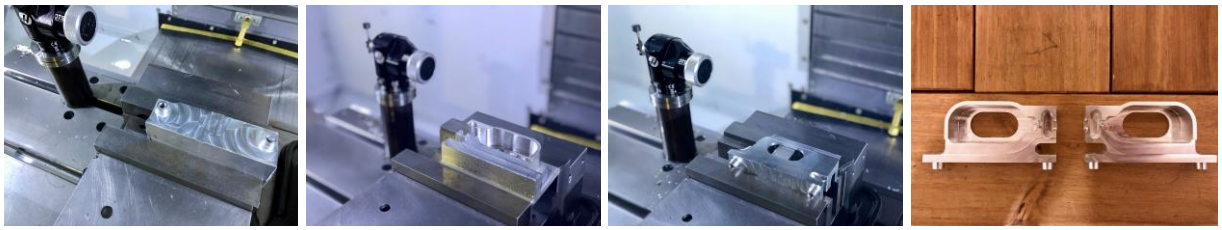


*Figure 34:Dropout mount geometry after each machining operation final left and right dropout mounts.*

### 7.2.2 Left and Right Dropout Slides

Again, the left and right dropout slides are mirror images of each other. This made machining them much easier. The only difference here was that the right dropout slide was designed with an extra 0.5” spacer in which we were going to attempt to mount a disc brake caliper to.  This never ended up happening due to difficulties in attachment. However, the spacer barely changed the geometry of the part and therefore the toolpaths were still interchangeable between the two parts, again making machining them much easier. The dropout slides sitting in their mount can be seen in Figure 35.

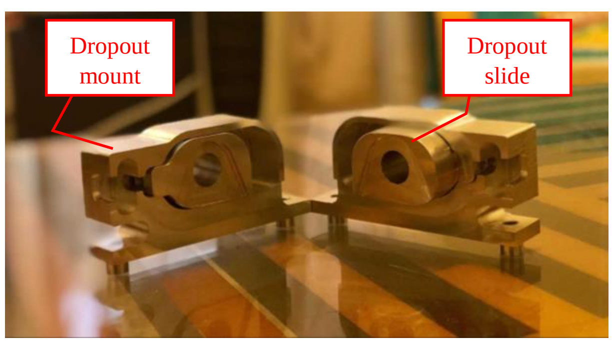



*Figure 35. Dropout mounts and slides.*

The dropout slides were machined with the same Haas VF2 mill and endmills as the dropout mounts, thereby making machine setup much simpler. We first machined the body of the slides which would be located inside the mount as well as locate the location of the rear hub. This was the most important step as getting this step correct ensured the location of our rear wheel and hub assembly would be exactly where we desired it to be. We then flipped the part over and machined the threaded post. This was quite simple, though we continually had issues keeping the through axil hole and post concentric. Luckily, this was not an issue for our part, it just meant the housing slots had to be large enough to allow for this machining tolerance, which we had taken care of during the designing phase of the project.

Lastly, the part was put on end and the last threaded hole was placed for the ¼-20 rod to be placed. Again, this was a very simple process and only took a minute after machine setup. A picture of the end result from each operation is shown below along with the final part.

### 7.2.3 Adjustment Screw

The adjustment screw was designed to be machined in a single operation on the CNC lathe. As with the dropout slides, the G-Code was created using HSMWorks. The toolpath used a CNMG 432 insert for most of the work and a grooving tool to finish off the groove in the part. Lastly, the part was tapped using a ¼-20 hand tap and knurled using a knurling tool to give it that final touch. All in all, this was a fairly simple part to accomplish and multiple backups were made incase any were lost or damaged at competition. The adjustment screws are shown in Figure 36 below.

*Figure 36. Adjustment screws. By turning these, the slides will move back and forth within their mounts.*

## 7.3 Drivetrain Plate

The first items completed were the aluminum brackets that would secure the drivetrain plate into the car. These were CNC machined by SMV team member Pedro Mogollon and were then roughed up with sandpaper before being given an Alumiprep bath. The Alumiprep was used to remove the oxide layer on the aluminum then treat it to better bond with adhesives. This process left a gold finish on the brackets which can be seen in Figure 37 below.

*Figure 37. Drivetrain plate brackets after being given an Alumiprep bath.*

The Divinycell foam core and carbon face sheets from Tencate were both cut using the on-campus waterjet behind the Architecture Engineering Machine Shop. While the carbon sheets only needed to resemble the outer profile of the plate, the foam core included internal features which the waterjet was easily able to accommodate. The Garolite inserts were machined using the Aero Hangar's manual mills and Figure 38 below shows the collection of these finished parts.

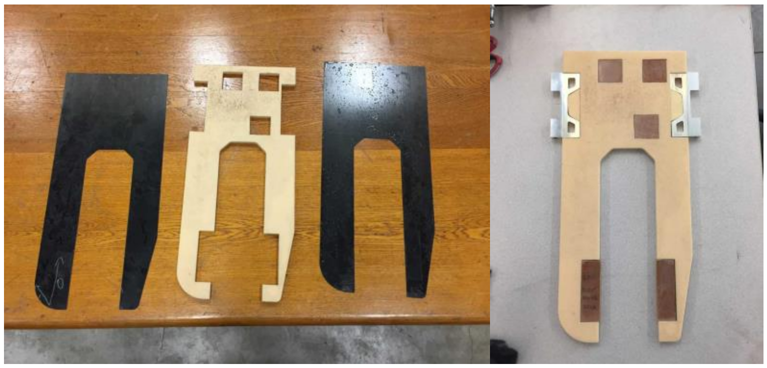


*Figure 38. Individual drivetrain plate parts being test fitted.*

Once all components were ready, 3M brand film adhesive was placed in between the core and carbon sheets, the collection was inserted into a vacuum bag, and placed into the Composites Lab Autoclave to cure. Unfortunately, proper attention was not given to the temperature limits of the Divinycell foam and the heat from the Autoclave caused it to melt. This resulted in the loss of the part and only the aluminum brackets were able to be recovered. Despite the setback, we were able to quickly remake the foam core and Garolite pieces. After discussion with the SMV team's Composites Lead, it was decided that a wet-layup method would be used instead of the film adhesive with the intention of expediting the completion of the part. This process included cutting carbon fiber fabrics to shape, wetting them with epoxy, and using them to wrap the foam, Garolite, and brackets. This collection was then vacuumed onto a glass table and allowed to cure overnight. Figure 39 shows the assembly as it was being cured.

*Figure 39. Drivetrain Plate being cured using a glass table.*

Although the resulting product came in heavier than was desired, due to the use of epoxy rather than film adhesive, the glass table method was able to adequately achieve a flat mounting surface for the drivetrain to be assembled upon.

## 7.4 Sprocket/Hub Assembly

Material for the sprocket stock was purchased from Douglas Steel Supply and had to be picked up from their warehouse in the Los Angeles County. Douglas Steel Supply was chosen for their stretcher leveling technique which could produce exceptionally flat sheet metal. Given the special quality of this sheet metal, plenty of stock was purchased so that the club will have material for years to come. Once acquired, the sheet steel was cut to shape using the same waterjet as before. A rapid prototype of the sprocket had previously been 3D printed to check fitment and to refine the design, but once the actual sprocket was cut its fitment was again checked and confirmed to match the hub's features. As an added benefit, the sprocket was given a graphite coating to reduce friction with the chain.

The sprocket fasteners received a slight redesign before being manufactured. The final design for these fasteners can be found in Figure 40 below. Each of the three fasteners feature a solid cylinder sized to precisely match the pitch of the driving cog attached to the hub. The remaining two cylinders are threaded through-holes which were sized to be a clearance fit in both the hub's cog and in the hole pattern on the sprocket.

This greatly reduced the complexity that would have been required in machining exact fit components. The sprocket is secured to the hub using M8 bolts and torque is transmitted through the solid cylinders. Figure 40 below shows how the sprocket fasteners fit onto the hub and how they secured the sprocket.

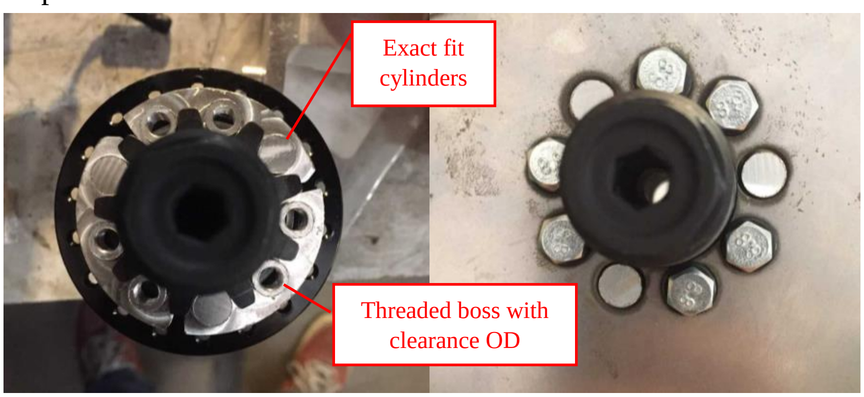



*Figure 40. Sprocket fasteners on the hub (left) and the test sprocket secured onto the hub (right).*

## 7.5 Engine Mount

Due to time constraints, the engine mount was machined by a Cal Poly Mustang '60 Shop Technician. The 7.5" x 6.5" x 2" aluminum stock was purchased from onlinemetals.com. As we had originally planned on machining this ourselves, a CNC operation plan was already started as to which operations would be done and in which order to obtain the tolerances and geometries desired. Therefore, it was easy to convey our ideas to the tech and allow him to complete the job much quicker. Several views of the mount installed on the engine are shown below in Figure 41.

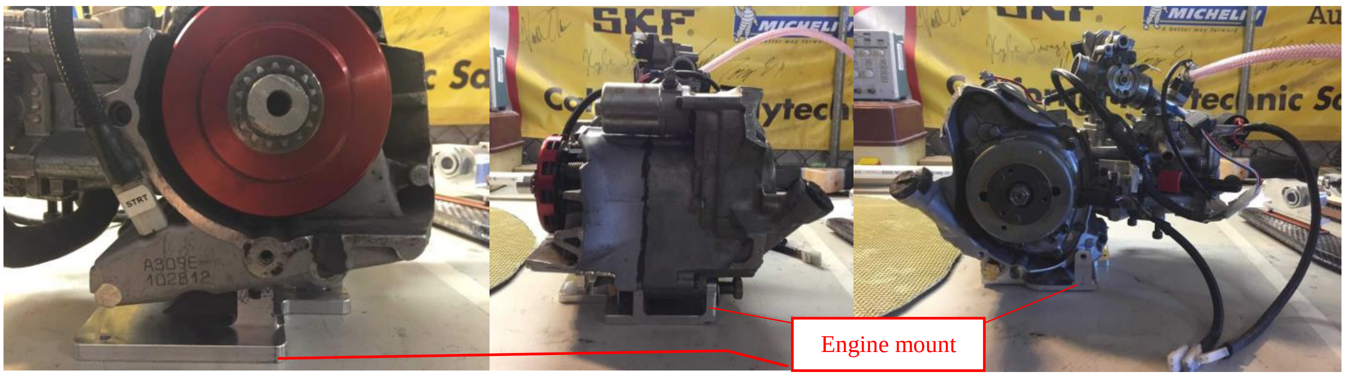



*Figure 41. Engine mount installed on the engine.*

## 7.6 Final Assembly

After all of the parts were manufactured, it was time to assemble the parts together onto the drivetrain plate. We first drilled the holes for the rear dropouts by securing the them together with the rear hub and using a transfer punch to locate the holes. Once the rear dropouts were installed, we worked with the SMV team to use a jig which positioned the plate in the car and then the plate-chassis interfaces were bonded in. The result of this is pictured in Figure 42 below.

*Figure 42. Drivetrain plate with hub/dropouts installed into the car.*

Once the mounts were bonded into the vehicle, we could continue with mounting the engine onto the plate. The first step in this process was to determine the relationship between the engine mount and the driving sprocket on the engine's clutch. This told us where we needed to place the locating pins and mounting holes relative to each other onto the drivetrain plate. This was done using a CMM (Coordinate Measurement Machine) which uses a wand like piece to touch the part to create a 3D point cloud. This process is shown in Figure 43 below.

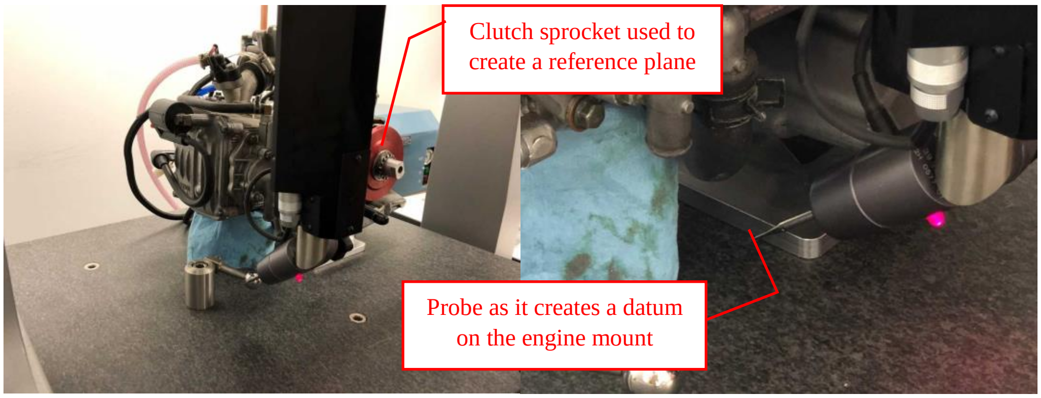



*Figure 43. Full picture of engine and CMM (left) and the datum created (right).*

This was then used to create a datum on the flat surface of the mount, ultimately giving us the angular and perpendicular distance measurements relating to the clutch sprocket. After this, an Optical Comparator was used to determine the final dimensions of the fastening and locating holes on the engine mount relative to that created datum. The dropouts were then fastened onto the drivetrain plate along with a test sprocket. The whole assembly was put onto a manual mill and

*Figure 44. Sprocket being trammed in and preparing to drill the holes for the engine mount. Note that the probe feature here was not the one used. The proper probe that was used is the same test indicator found in Appendix E.*

secured with toe clamps. In the fashion shown in Figure 44, the test sprocket was trammed in just like a vice would be.

By touching off on the test sprocket and calling that reference plane "zero," the mill's digital read outs were used to drill the fastening and locating holes in accordance with measurements taken in the metrology lab. The result of these efforts placed the clutch sprocket in the exact same plane as the rear sprocket. The fully assembled drivetrain is shown here in Figure 45.

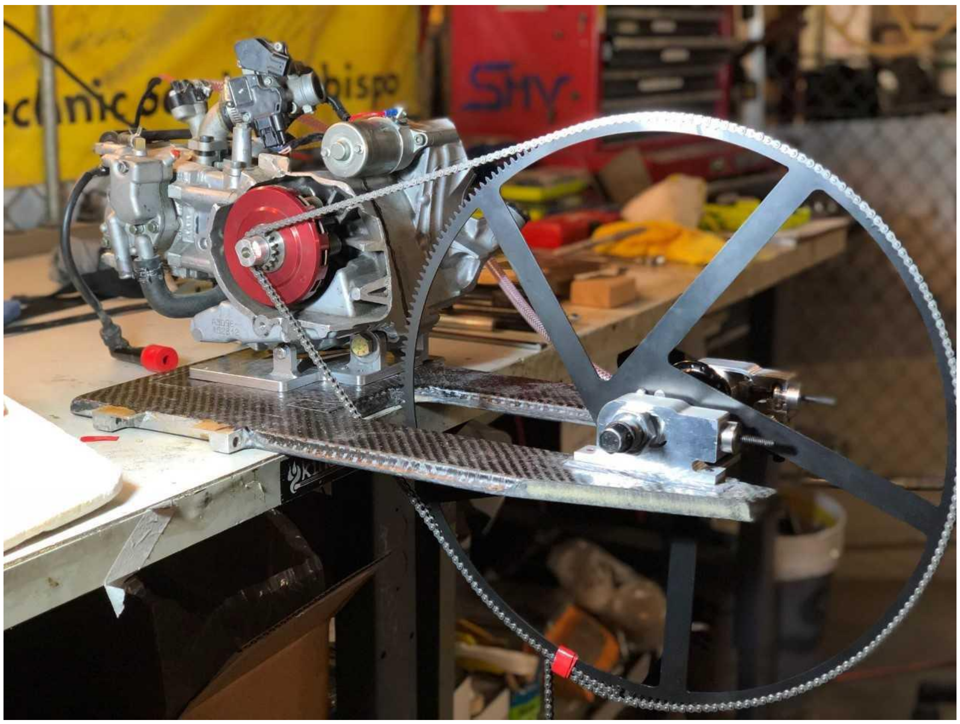


*Figure 45. Fully assembled drivetrain.*

Hidden from view are the round and diamond headed locating pins that allow the engine to be removed from the drivetrain plate and subsequently reinstalled while maintaining the same alignment every time. These pins were bonded into the drivetrain plate and are exact fits with the reamed holes on the engine mount.

# 8.0 Design Verification & Testing

## 8.1 Overview

As introduced and described in section 5.13, several tests were devised to verify our design, ultimately leading up to the definitive test of competition. While most of these tests were successfully carried out, the efficiency test and alignment test could not be completed. The SMV team had intended to provide our Drivetrain Team with a functioning chassis dyno in order to measure drivetrain efficiency; however, this dyno was not completed in time leaving us with no way to determine the final ratio of wheel horsepower to crank horsepower. Additionally, an attempt to measure final alignment between the clutch sprocket and the rear sprocket was made, but due to size restrictions on the CMM it could not be completed. The following sections discuss the tests that were able to be carried out and presents their results.

## 8.2 Sprocket Flatness

While not specifically mentioned in the design verification plan from section 5.13, a measurement of sprocket flatness was done to verify our hypothesis that waterjet cutting would not distort the sprocket's flatness. The sprocket was placed of a granite micro-flat table and a dial indicator was run around the outer edge to measure the amount of deviation. The test showed about 0.005" of deviation. When compared to measurements of the old sprocket at 0.0155" of deviation, this is a 68% decrease.

## 8.3 Sprocket-Hub Deflection/Rigidity

Using the same test as described in Appendix E, the amount of sprocket tip movement when attached to the odyssey hub was measured. This test showed 0.55° of play. That measurement places us within our design objective of 0.8° of play and is a 41% improvement over the 0.93° of play in the old system. Figure 46 below illustrates the test that was conducted.

*Figure 46. Sprocket-Hub play test.*

## 8.4 Removal/Installation Time

One design goal for this project was to maintain the user friendliness of the previous design. As a metric for this, the amount of time it takes for a trained individual to install and remove the whole system was measured. We found that on average, a trained SMV team member could remove the entire drivetrain from the car in roughly 13 minutes. Similarly, it was measured that a trained individual could assemble and install the whole drivetrain in about 16 minutes. These numbers sufficiently pass our design objective of 30 minutes total for combined removal and installation time. It should be noted that these times were only achievable after 3 practice sessions with the individual. Most of the difficulty came from trying to reach the various electrical connectors between the engine and the ECU as well as issues with disconnecting the brake cable.

## 8.5 Weight Test

Though not of the utmost importance, the team intended to deliver a drivetrain whose weight was on par with the old design. The new single stage drivetrain came in weighing 4.74 kg. This missed our nominal goal of 4.2 kg but was within our specified tolerance limit of 5.2 kg.

## 8.5 Cost

The new drivetrain was able to be completed under budget by $185 with a total cost of $1,315. This cost is broken down in Table 5 that follows.

*Table 5. Cost of Drivetrain*

| Item | Cost |
|---|---|
| Prototyping | $138.84 |
| Engine Mount Stock | $104.61 |
| Drivetrain Plate Stock | $108.09 |
| Sprocket Stock | $563.93 |
| Engine Mount Machining | $230 |
| Misc. Hardware | $169.53 |
| TOTAL | $1,315 |

The main cause of an increase of actual cost versus the cost analysis conducted in section 5.10 is the use of rapid prototyping to aid our designs and the need to outsource the machining of the engine mount.

## 8.6 System Alignment

As mentioned above, a proper alignment measurement with the CMM was not possible due to the fact that the fully assembled drivetrain with the wheel installed could not fit on the IME Department's CMM. However, one of the compliance methods listed in Table 1 is through

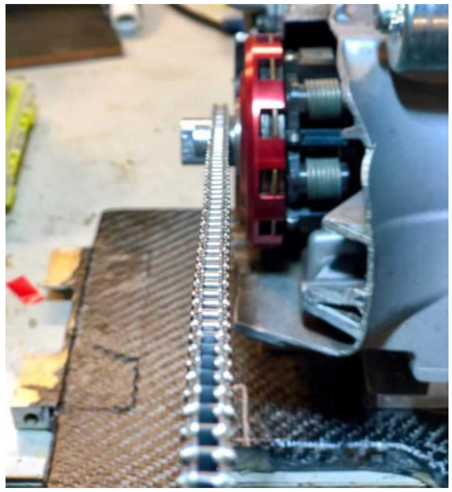


*Figure 47. Visual inspection of chain alignment.*

inspection. As can be seen in Figure 47, the new drivetrain appears to have achieved a great deal of alignment. Though this is a purely qualitative analysis, comparisons to the previous system and the performance of the new system suggest that worthwhile improvements have been made.

## 8.7 Alignment Repeatability

Without the use of the CMM, alignment repeatability was again unable to be directly measured. Despite this, after several removal and installations of the engine, chain alignment from a visual standpoint seemed to be unaffected. More importantly, after continuously removing and installing the components that make up our drivetrain no detriment to reliability was seen since chain throw never occurred.

# 8.0 Conclusion

## 9.1 Preliminary Design Efforts

For nine weeks, the Cal Poly Supermileage Drivetrain team worked to gain a thorough understanding of the problem at hand and worked to develop the Supermileage team's needs into a series of specifications. These efforts culminated in a collection of preliminary designs and a strong sense of direction heading forward. A single staged chain drive was chosen with a gear reduction that could fall anywhere between 14:1 or 16:1. A method to have a modular drivetrain assembly would be retained and an improved system for finding datums upon the engine would be developed. Through a series of tests and measurements, areas for major improvement of sprocket/hub stiffness and rigidity were determined and solutions were developed. A hub with a clutch mechanism was found and ways to improve upon sheet metal flatness for the sprocket were heavily researched. Early designs for the dropouts were ideated and the most promising ones chosen for review. Two competing methods to achieve precise and repeatable sprocket alignment were developed and presented for review. At the end of this preliminary stage, a chain tensioning method had yet to be determined and, while design challenges related to the new hub were anticipated, they had yet to be determined.

## 9.2 Critical Design Efforts

After a review of the preliminary designs developed by CPSMD, the team moved into a seven-week-long critical design phase. Due to packaging considerations and simplicity, a 14:1 gear reduction was chosen. The final geometries for the drivetrain plate, sprocket, and engine mount were settled upon and appropriate stress analyses were completed. An overhaul of the dropout design was completed and the resulting design achieved superior axle location, included a brake caliper mount, and achieved a chain tensioning method. A source of flat stock metal for the sprocket was found and price quoted. The new hub arrived and design challenges identified. A method for fastening the rear sprocket onto the hub was designed and initial ideation for solutions to mounting a brake disc onto the hub were conducted. The preemptive alignment method for the engine and rear sprocket was chosen and the process for carrying this out further detailed. A parts list including necessary hardware was compiled and detailed drawings or spec sheets were created. With a manufacturing and verification plan laid out, the team planed to move forward and order the necessary stock and hardware, build prototypes, conducted testing, and began the manufacturing/measurement process of the engine mount. In parallel, high priority was assigned to efforts for developing a solution to mounting the brake disc.

## 9.3 Final Design

The final design for the 2018 Supermileage Drivetrain saw only small tweaks from our CDR. Most notably, was the decision to abandon a disc brake and the successful application of a rim brake. It is fair to say that the use of a rim brake was not an ideal solution but a thoroughly adequate one. The only other component that saw a redesign was the sprocket fastener which featured particularly chosen exact fit and clearance fit features that greatly simplified their manufacturing and assembly.

All in all, the CPSMD Team was able to deliver a functioning drivetrain to the SMV team on time and under budget. At competition, two full 10-mile runs were able to be completed. Three more attempts were made, but due to mental errors on the SMV team's behalf and a steering system failure they were not completed. In each of the attempts, (successful or otherwise) the drivetrain performed reliably without ever throwing a chain. Through the work of the CPSMD Senior Project team, we were able to produce a Supermileage drivetrain system with a single staged 14:1 gear reduction which had unmatched reliability when compared to years past. It is believed that no single design decision is responsible for this success, but rather it is the culmination of precisely manufactured and aligned components that was able to produce this result.

## 9.4 Lessons Learned and Areas with Potential for Improvement

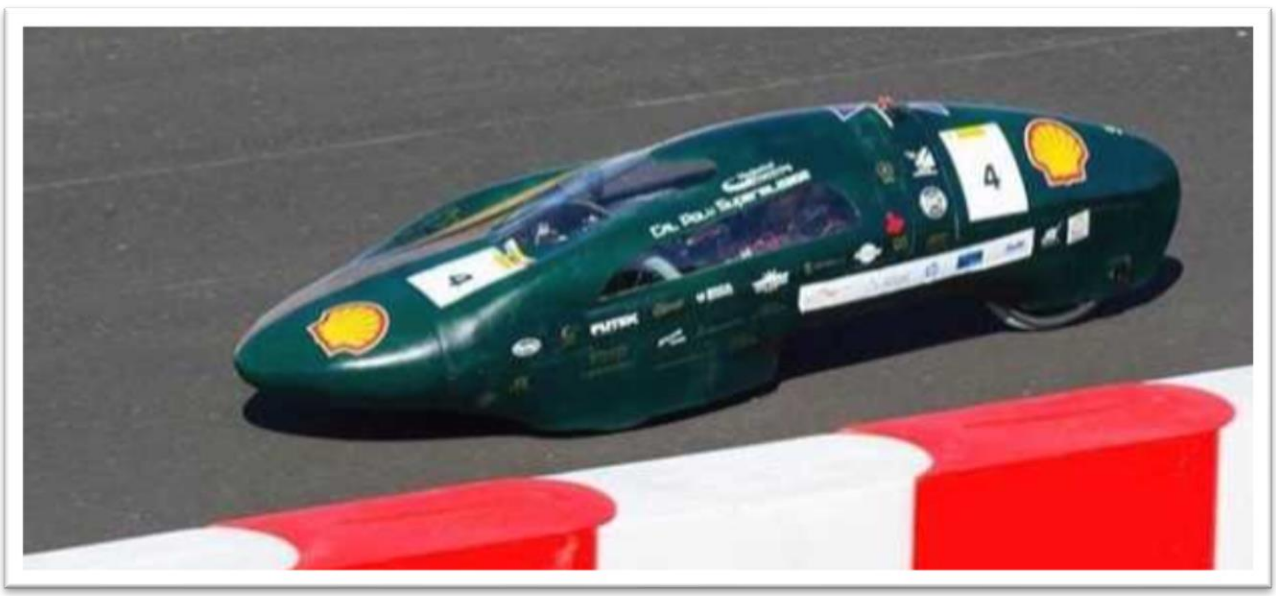
Throughout the manufacturing process, several areas with potential for improvement began to arise. The first lesson learned included the mishap with melting the drivetrain plate core material. In the past, balsa wood was used as core and curing temperature was not an issue. While switching to Divinycell makes acquiring the core material easier, care must be taken to not exceed its temperature limit. Since the film adhesive does not require to be cured at temperature (the oven had been used in an attempt to speed up the cure process), the original process of carbon face sheets and 3M film could still be accomplished. However, with the success of the wet layup, one might elect to use this simplified process. It should be noted that the larger amount of epoxy used resulted in a heavier part when compared to the film adhesive approach. Also, when choosing the wet layup method, attention must be given as to which side is facing the glass surface of the table. In our case, the bottom of the drivetrain plate was accidentally vacuumed to the glass rather than the top. This required a good deal of sanding and epoxy to fill in the roughness that resulted from the layup.

Along these same lines, there were many lessons to be learned during the machining phase of the build. As these were the first parts completely designed and CNC machined by ourselves alone, there were many hurtles to surpass before achieving parts we deemed acceptable for our high standards of accuracy. Most of these lessons were learned at the beginning, with the machining of the dropout slides. As the slides were the first parts to be CAM'd and machined, we were able to get insight into fixturing techniques to give us better surface finish, as well as tool path sequences that would allow for the use of larger tools, ultimately reducing chatter and giving us better surface finishes and tighter tolerances. This process was not without its multiple broken endmills and taps as well as a few more scrapped pieces of stock metal than we would like to admit, though in the end we were able to take away machining skills only learned through trial by fire.

While the process of waterjet cutting the sprocket was very successful in achieving its shape and retaining its flatness, it was found the finished sprocket is very susceptible to damage. Simply by allowing SMV team members to carelessly handle the sprocket, it was quickly bent. Thankfully, with the excess stock sheet metal, multiple replacement sprockets were made and could be quickly swapped out when damaged. In the future, it may beneficial to reinforce the sprocket with a carbon layup to improve its resilience to bending when dropped or stepped on.

After CDR, a great deal of effort was given towards the rear braking system. Many attempts to come up with a way to mount a brake disk to the hub were made, but ultimately, they all were abandoned. While this project was unable to devise a solution in time that would allow a disc brake to be added to the Odyssey hub, a rim brake was able to be successfully installed. The rim brake required some slight modifications and is a real hassle to work around, but it was able to consistently pass the braking tests at competition. There is still some possibility for a disc brake solution (or some other creative braking solution) to be devised; however, future iterations may desire to look into disc brake compatible hubs. It is the opinion of this senior project team that the improvements seen in sprocket-hub rigidity and freecoasting from the Odyssey hub greatly outweighed the hassle of using rim brakes.

2018 Supermileage Car, *DeLamina*: 4$^{th}$ place; 1,292 miles per gallon



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# Appendix B: Quality Function Deployment

SMDQFD.xls - Sheet1

Engineering Requirements (HOWS)

Cal Poly Supermileage Team

Customer Requirements (Step #2)

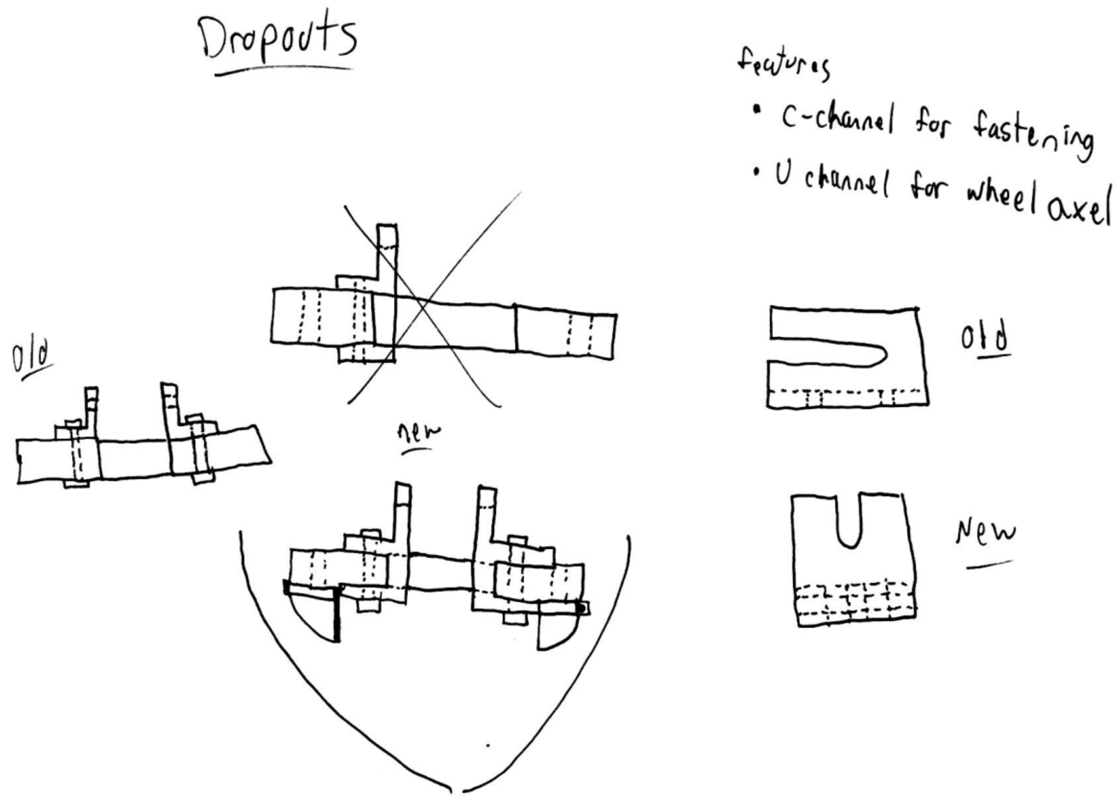
| Supermilage Requirements (Whats) | Weighting (Total 100) | Number of Parts (#) | Weight (kg) | Efficiency (%) | Meets Budget ($) | Lifetime (cylces) | Deflection (in) | Left or Right Hand Drive (L/R) | Size LxW (in) | Time to Install (s) | Benchmarks: Current | Toronto | Laval |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Functional Performance | | | | | | | | | | | | | |
| High stiffness | 17 | | 9 | 9 | 3 | 9 | 9 | 3 | 9 | | 3 | 4 | 4 |
| Lightweight | 8 | 9 | 9 | 1 | 3 | 3 | 9 | 3 | 9 | | 2 | 3 | 3 |
| Fits Chassis | 15 | 3 | 1 | 1 | | | | 3 | 9 | | 5 | 5 | 5 |
| Ease of access (maintenance) | 10 | 3 | 1 | 1 | 1 | 3 | | 3 | 3 | 9 | 4 | 4 | 4 |
| Ease of installation | 10 | 9 | 1 | 1 | 1 | 1 | | 1 | 3 | 9 | 3 | 3 | 3 |
| Alignment/Tension repeatability | 15 | 9 | 1 | 9 | 1 | 1 | 3 | 9 | | 3 | 2 | 3 | 4 |
| Reliability | 14 | 9 | 1 | | 1 | 9 | 9 | 9 | | | 5 | 2 | 5 |
| Part availablity | 11 | 9 | 3 | 3 | 9 | 1 | 3 | 9 | | | 1 | 4 | 2 |
| | 100 | | | | | | | | | | | | |

Targets

| | # | kg | % | $ | cylces | in | L/R | in x in | s |
|---|---|---|---|---|---|---|---|---|---|
| Units | # | kg | % | $ | cylces | in | L/R | in x in | s |
| Laval | 8 | 4.7 | 80 | $900 | infi | Unk | L | Unk | 3600 |
| Current | 11 | 5.293 | 62.1 | $700 | infi | 0.0352 | L | 14 x 8 | 5400 |
| Toronto | 7 | 5.4 | 78 | $900 | infi | Unk | R | Unk | 3600 |

9 Strong Correlation
3 Medium Correlation
1 Small Correlation
Blank No Correlation

## Appendix C: Various Component Ideation



# Appendix D: Weighted Decision Matrix

| Design Criteria / Weight Factor / Alternatives | Cost | Shelf Life | Speed Application | Life | Stretch | Alignment | Efficiency | Overall Satisfaction |
|---|---|---|---|---|---|---|---|---|
| | 0.1 | 0.1 | 0.1 | 0.2 | 0.1 | 0.3 | 0.3 | |
| Chain drive system | 50 / 5 | 90 / 9 | 75 / 7.5 | 50 / 5 | 90 / 9 | 90 / 27 | 90 / 27 | 98.5 |
| Belt drive system | 50 / 5 | 50 / 5 | 90 / 9 | 50 / 5 | 75 / 7.5 | 75 / 22.5 | 75 / 22.5 | 76.5 |

| Design Criteria / Weight Factor / Alternatives | Cost | Size Restrictions | Alignment | Components | Efficiency | Weight | Overall Satisfaction |
|---|---|---|---|---|---|---|---|
| | 0.1 | 0.2 | 0.2 | 0.1 | 0.3 | 0.1 | |
| Single-stage reduction | 75 / 7.5 | 50 / 10 | 75 / 15 | 90 / 9 | 90 / 27 | 90 / 9 | 68.5 |
| Two-stage reduction | 50 / 5 | 75 / 15 | 75 / 15 | 75 / 7.5 | 75 / 7.5 | 75 / 7.5 | 57.5 |

## Appendix E: Sprocket and Hub Assembly Deflection Testing

The testing procedure for sprocket and freewheel deflection was developed by reasoning that if a chain is not perfectly parallel to the sprocket, it is the normal component of the driving force that causes bending. In order to simulate this scenario, the wheel-hub-sprocket assembly was placed on a surface plate and a test mass was placed on the outer edge of the sprocket. Initial analysis of the current drivetrain was achieved through calculations to determine a reasonable range for what this test mass could be. Once that was done, a weight which was available in the Hangar was chosen and weighed to ensure it was within this range. An extreme case of 16° of chain misalignment was backed out using the test mass available and the force derived from the engine's peak torque. Figures #'s below show the test rig setup.

*Figure #. Testing of the freewheel play.*

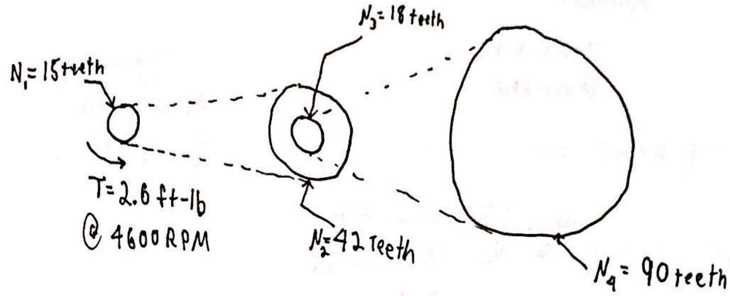
*Figure #. Close up of the inner and outer race of the freewheel.*


*Figure #. Measurements of sprocket edge movement.*

Despite having used the jack stands and the test indicator to ensure that the wheel was level, the sprocket itself is not exactly balanced so it could have been resting at an angle with respect to the wheel's plane. To overcome this, the test mass was first placed on one edge of the sprocket, measurements were made, the test mass was moved to the opposite edge of the sprocket, and the measurements were made again. This process effectively swept through the full amount of deflection in both directions from the center and accounted for any initial wobble there may have been. By making measurements of how much the outer race of the freewheel moved up or down and then making measurements of how much the outer edge of the sprocket moved up or down we could compare the two. Knowing the distance from the center and these heights, trigonometry determined an angular amount of play in the assembly. The difference between the angle of play at the freewheel and the angle of play at the sprocket tip tells you how much deflection each one is responsible for. The 14-inch steel sprocket was chosen since it would be closer in size to the 15:1 sprocket and the steel would be more susceptible to bending than would titanium. The results of this test showed that the angle of play at the freewheel and the angle of play at the sprocket tip was exactly the same. This suggests that there is no measurable amount of deflection within the sprocket material under this loading condition and all play within the wheel assembly could be attributed to the freewheel.

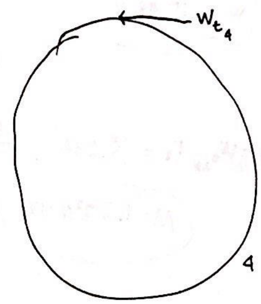
# SPROCKET/HUB ASSEM. DEFLECTION TESTING

CURRENT D.T.

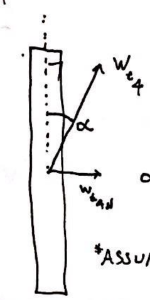



G.R.= 14

$r_4 = \cancel{3.71\,in}\ 3.58\,in$




SIDE VIEW




TOP VIEW

$\alpha$ = ANGLE BETWEEN DRIVELINE & SPROCKET

*ASSUME $\alpha < 20°$

*EVEN 20° IS EXTREME

FROM SHIGLEY'S

$$T = \frac{d}{2} W_t$$

$$W_{t_4} = \frac{T}{r_4}$$

$$W_{t_4} = \frac{2.6\ ft \cdot lbf}{(\cancel{3.71\,in}\ 3.58)\left(\frac{1\,ft}{12\,in}\right)} = \cancel{8.41}\ 8.72\ lbf$$

$$W_{t_4} = \cancel{37.4}\ 38.8\ N$$

$$W_{t_{4N}} = W_{t_4} \sin\alpha$$

WORST CASE $\quad W_{t_{4N}} = \cancel{37.4}\ 38.8 \sin(20)$

$$W_{t_{4N}} = \cancel{12.69\,N}\ 13.27$$

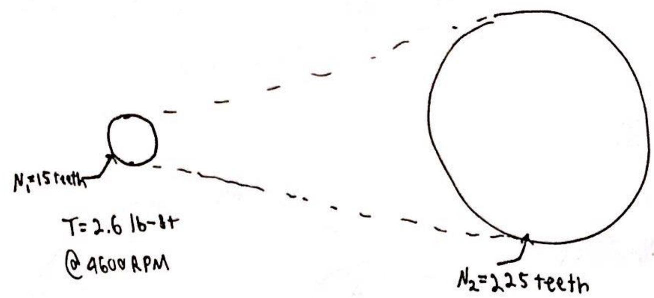
NORMAL FORCE THAT IS TRYING TO BEND/ DEFLECT SPROCKET. THE FORCE LEADING TO CHAIN DE-RAILMENT FOR 2-STAGE 14:1 G.R.

$M \equiv$ MOMENT APPLIED NORMAL TO SPROCKET CAUSING DEFLECTION

$$M = W_{t_{4N}} r_4 = (\cancel{12.69}\ 13.27\,N)(\cancel{3.71}\ 3.58\,in)\left(\frac{0.0254\,m}{1\,in}\right) = \cancel{1.1958}\ 1.21\ N\text{-}m$$

$$M = 1.20\ N\text{-}m$$

15:1
SINGLE-STAGE




$d_2$ = 17.90551 in  * Diameter of wheel including unloaded tire is 20 in

$$W_{t_2} = \frac{2T}{d_2} = \frac{2(2.6\ \text{lb}\cdot\text{ft})}{(17.90551\ \text{in})\left(\frac{1\ \text{ft}}{12\ \text{in}}\right)}$$

$$W_{t_2} = 3.48\ \text{lbf}$$

$$W_{t_2} = 15.48\ N$$

$$W_{t_{2N}} = 15.48 \sin 20$$

$$W_{t_{2N}} = 5.29\ N$$

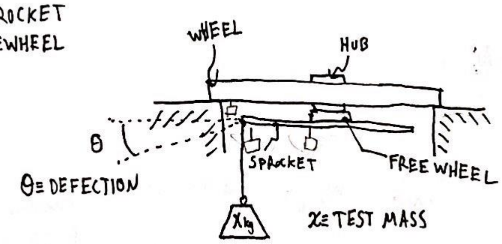
$$M = W_{t_{2N}}\, r_2 = (5.29N)\left(\frac{17.90551\ \text{in}}{2}\right)\left(\frac{0.0254\ \text{m}}{1\ \text{in}}\right)$$

$$M = 1.203\ N\text{-m}$$

SAME MOMENT ACTING AS WITH 2-STAGE. AS EXPECTED

## SPROCKET MOMENT DEFLECTION TEST RIG

- WANT TO TEST SPROCKET DEFLECTION & FREEWHEEL PLAY.




$$M = \chi g r$$

$$\chi = \frac{M}{gr}$$

2-STAGE: $$\chi_1 = \frac{1.21\ N\text{-m}}{9.81\,\text{m/s}^2\,(3.58)(0.0254)}$$

1-STAGE: $$\chi_2 = \frac{1.20\ N\text{-m}}{9.81\,\text{m/s}^2\,(8.95\,\text{in})(0.0254)}$$

*USE DIAL INDICATOR AT SPROCKET TIP, FREEWHEEL, AND WHEEL TO FIND SOURCE OF DEFLECTION

$$\chi_1 = 1.36\ \text{kg}$$

$$\chi_2 = 0.538\ \text{Kg}$$

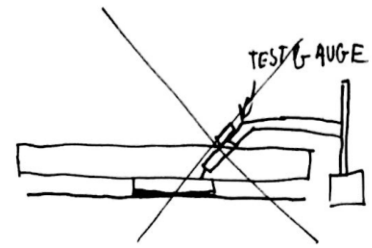
DESIGN GOAL IS TO MEASURE $\theta_2 < \theta_1$ USING MASSES OF 0.538kg to MEASURE $\theta_2$ and 1.36kg TO MEASURE $\theta_1$. THE DIFFERENT MASSES APPROPRIATELY REPRESENT THE DEFLECTING FORCE NORMAL TO THE SPROCKET FACE THAT EACH SETUP WOULD SEE.

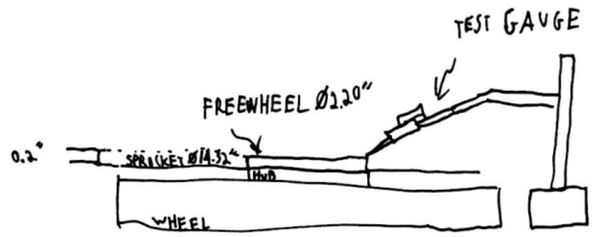
# FREEWHEEL TEST

TEST MASS $m = 0.436\,kg$

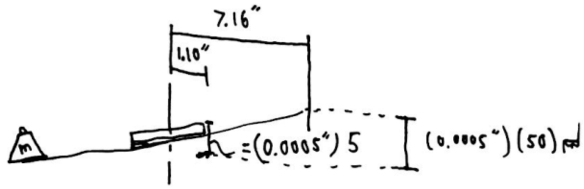
TESTED STEEL SPROCKET $\varnothing$ 14.32"

FREEWHEEL HAS $\varnothing$ 2.20"

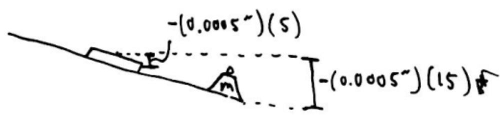


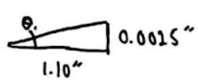

A.)




B.)

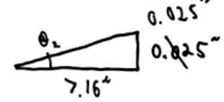



A.)




$\frac{O}{A} = \tan\theta$

$\theta_1 = 0.13°$



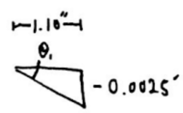

0.025" 0.~~0~~25"

$\theta_2 = 0.~~2~~0°$ 0.20°

$\theta_2 - \theta_1 = \Delta\theta$

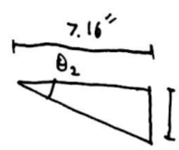
$\boxed{\Delta\theta_A = 0.07°}$

B.)




$\tan\theta = \frac{O}{A}$

$\theta_1 = -0.13°$




-0.0075" -0.~~00~~75"

$\theta_2 = -0.~~0~~60°$ -0.060°

$\theta_2 - \theta_1 = \Delta\theta$

$\boxed{\Delta\theta_B = +0.07°}$

~~$\Delta\theta_A = \Delta\theta_B$~~

$$\Delta\theta_1 = \theta_{1A} - \theta_{1B}$$
$$= 0.13° - (-0.13°)$$
$$\boxed{\Delta\theta_1 = 0.26°}$$

TOTAL WOBBLE IN FREEWHEEL

$$\Delta\theta_2 = \theta_{2A} - \theta_{2B}$$
$$= 0.20° - (-0.060°)$$
$$\boxed{\Delta\theta_2 = 0.26°}$$

$$\Delta\theta_1 = \Delta\theta_2$$

∴ MATERIAL IN SPROCKET DOES NOT DEFLECT UNDER A LOAD OF $(0.436)\,9.81 = 4.28\,N$. ALL PLAY IS DUE TO THE FREEWHEEL

TH ref. (pg. 20) $W_{t_2} = 15.48$ $\quad W_{t_{2N}} = 5.29\,N$

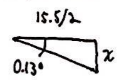
$$\sin^{-1}\left(\frac{5.29}{15.48}\right) = 16.0° = \theta$$

USING THE 0.436 kg TEST MASS AND AN APPLIED TORQUE FROM MOTOR OF 2.6 lb-ft; AN ASSUMED SPROCKET MISALIGNMENT OF 16° RESULTS.

---

TOTAL TIP MOVEMENT OF A Ø14.32" SPROCKET w/ THIS FREEWHEEL IS 0.0325"

FOR A Ø15.5" SPROCKET:




$x = 0.0176$

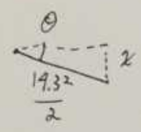
TIP DEFLECTION FROM MAX. TO MIN.

$$\boxed{\text{TOTAL } 0.0352''}$$

OLD HUB-SPR.

| □ ρ | □ρ | Δ | Total Tip Trave [in], $x$ | $\theta$ |
|---|---|---|---|---|
| ~~1~~ | ~~1rev 47~~ | | | |
| 36 | 1rev 47 | 1rev 11 | 0.111 | 0.888° |
| 30 | 1rev 48 | 1rev 18 | 0.118 | 0.944° |
| 25 | 1rev 46 | 1rev 21 | 0.121 | 0.968° |
| 33 | 1rev48 | 1rev 15 | 0.115 | 0.920° |

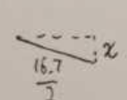
$\boxed{\text{AVG } 0.930^\circ}$




NEW HUB-SPR.

| Δ ρ | Δρ | Δ | $x$ | $\theta$ |
|---|---|---|---|---|
| 0 | 85 | 85 | 0.085 | 0.583° |
| 11 | 89 | 78 | 0.078 | 0.535° |
| 10 | 86 | 76 | 0.076 | 0.522° |
| 9 | 90 | 81 | 0.081 | 0.556° |
| 10 | 89 | 79 | 0.079 | 0.542° |

$\boxed{\text{AVG } 0.5476^\circ}$

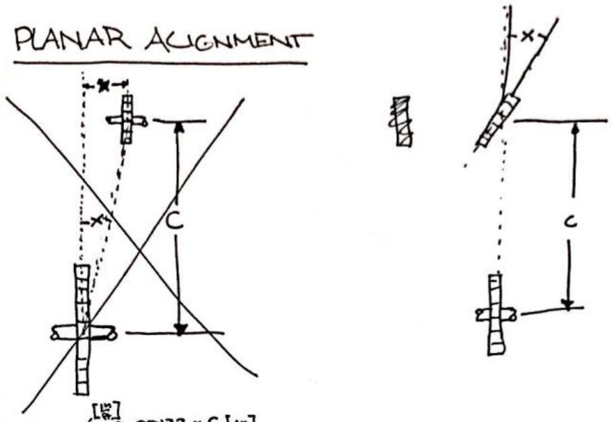



$$\% \text{ Decrease} = \frac{0.93 - 0.5476}{\cancel{0.5476}\ 0.93} \times 100\% = \cancel{30\%}\ 41\%$$

## Appendix F: Chain Alignment Specification Calculations

SPROCKET ALIGNMENT CALCS ... TAKE 2

PLANAR ALIGNMENT




$$X\left[\tfrac{in}{ft}\right] = \frac{0.00133 \times C[in]}{P[in]}$$

$$X_{\#25} = \frac{0.00133 \times (13.5in)}{1/4\,in}$$

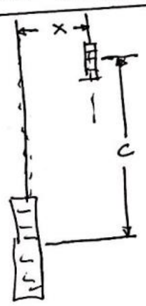
NOTE: "C" comes from current DESIGN, NOT NEW DESIGN

$$\boxed{X_{\#25} = 0.07182\ \tfrac{in}{ft}} = .35^\circ$$

$$X_{\#35} = \frac{0.00133 \times (13.5in)}{0.375in}$$

$$\boxed{X_{\#35} = 0.04788\ \tfrac{in}{ft}} = .24^\circ$$

AXIAL ALIGNMENT




$$X[in] = 0.045\,P[in]$$

$$X_{\#25} = 0.045(.25in)$$

$$\boxed{X_{25} = 0.01125in}$$

$$X_{35} = 0.045(.375in)$$

$$\boxed{X_{35} = 0.016875in}$$

CONCLUSIONS:

## Appendix G: Rotational Kinetic Energy Comparison

ROTATIONAL KINETIC ENERGY

$\omega = 300$ rpm

$I_{sprocket} = 8839064.69\ g \cdot mm^2$  $I_{pulley} = 46698093.64\ g\,mm^2$

*DERIVED FROM SOLIDWORKS USING EXACT SAME MATERIAL PROPETIES.

$$KE = \frac{1}{2} I \omega^2$$

$$KE_{sprocket} = \frac{1}{2}\, 8839064.69 \left(\frac{300\,rev}{min} \cdot \frac{1\,rad}{2\pi} \cdot \frac{1\,min}{60}\right)^2$$

$$KE_{sp} = 2798701.552\ \frac{g\,mm^2}{s^2} \left(\frac{1\,kg}{1000\,g}\right)\left(\frac{1\,m}{1000\,mm}\right)^2$$

$$\boxed{KE_{sp} = 2.799\,E\text{-}3\ J}$$

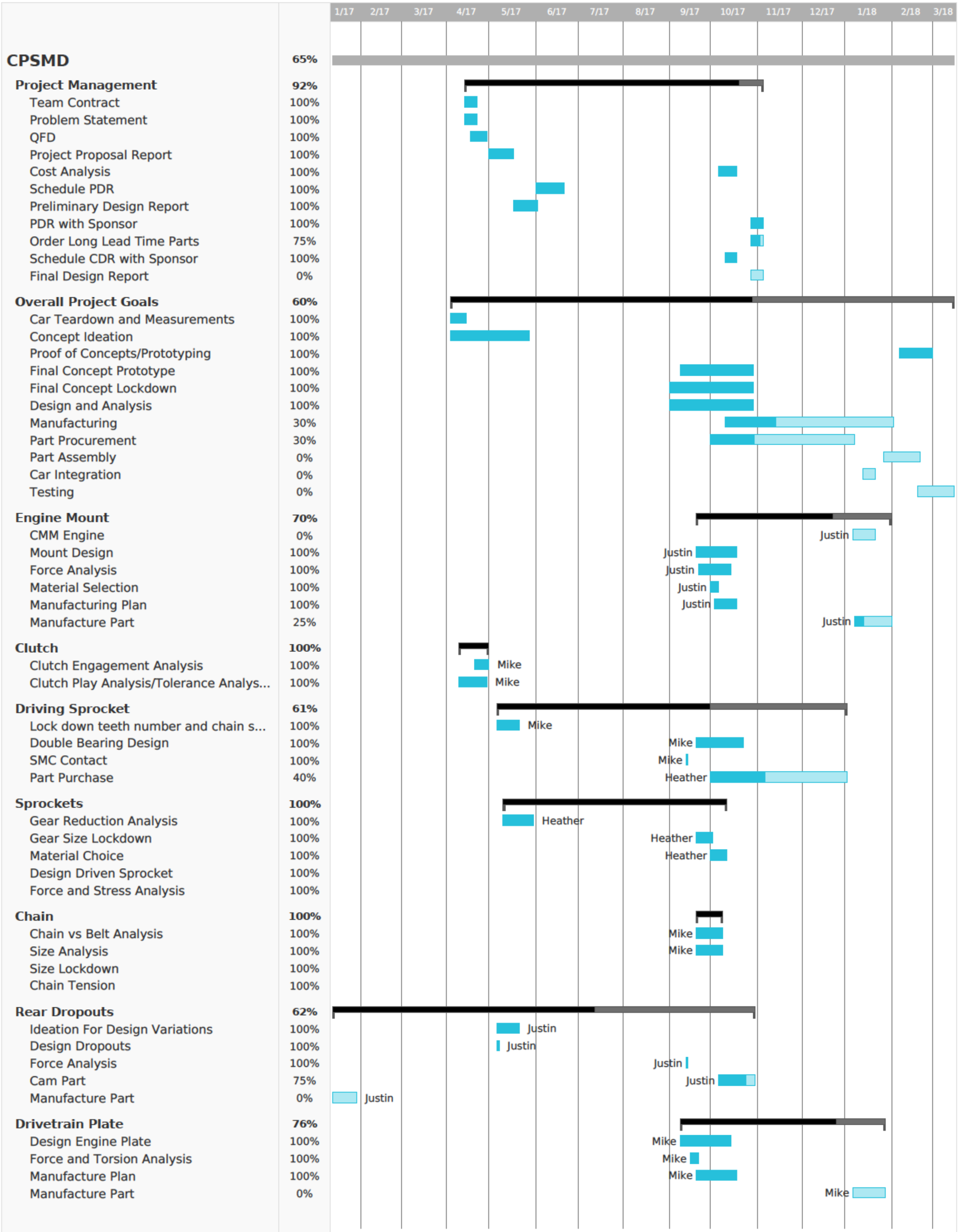
$$KE_{pulley} = \frac{1}{2}\, 46698093.64 \left(\frac{1}{1000}\right)^3 \left(\frac{300}{2\pi\, 60}\right)^2$$

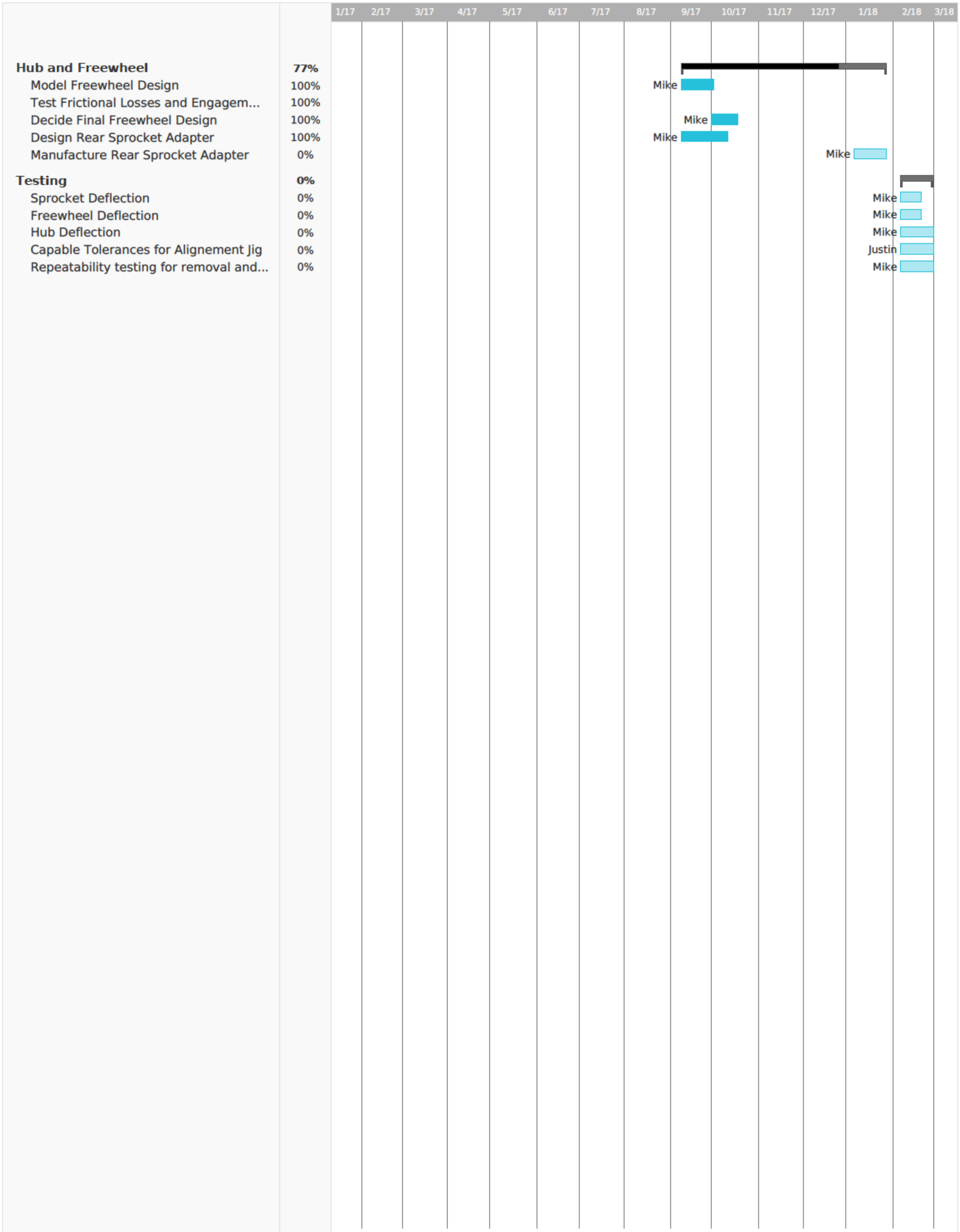
$$\boxed{KE_{pulley} = 1.479\,E\text{-}2\ J}$$

$$\text{PERCENT INCREASE} = \frac{1.479E\text{-}2 - 2.799E\text{-}3}{2.799E\text{-}3} \times 100$$

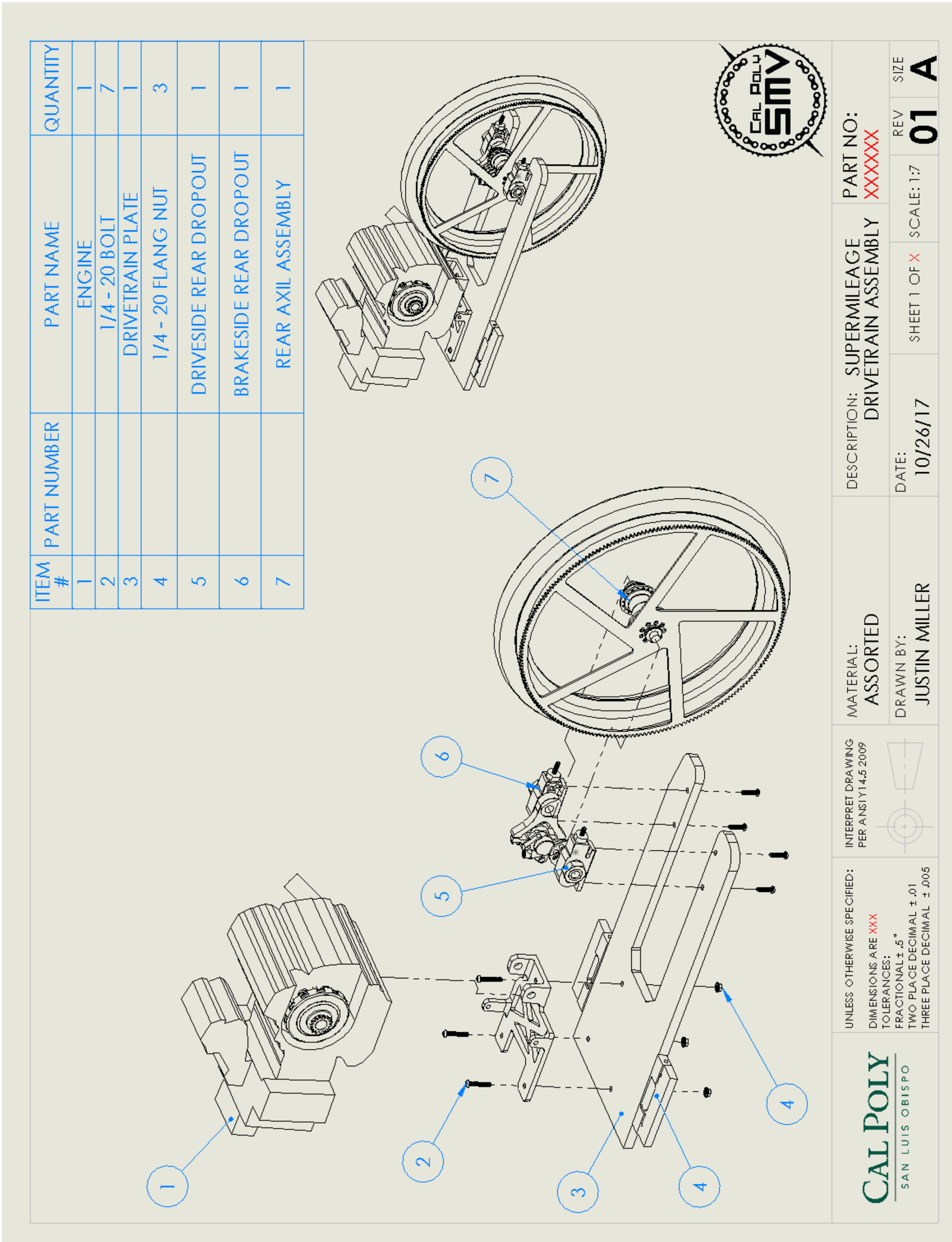
$$= \boxed{\text{428.3\% INCREASE IN ROTATIONAL KE BETWEEN SPROCKET AND PULLEY.}}$$

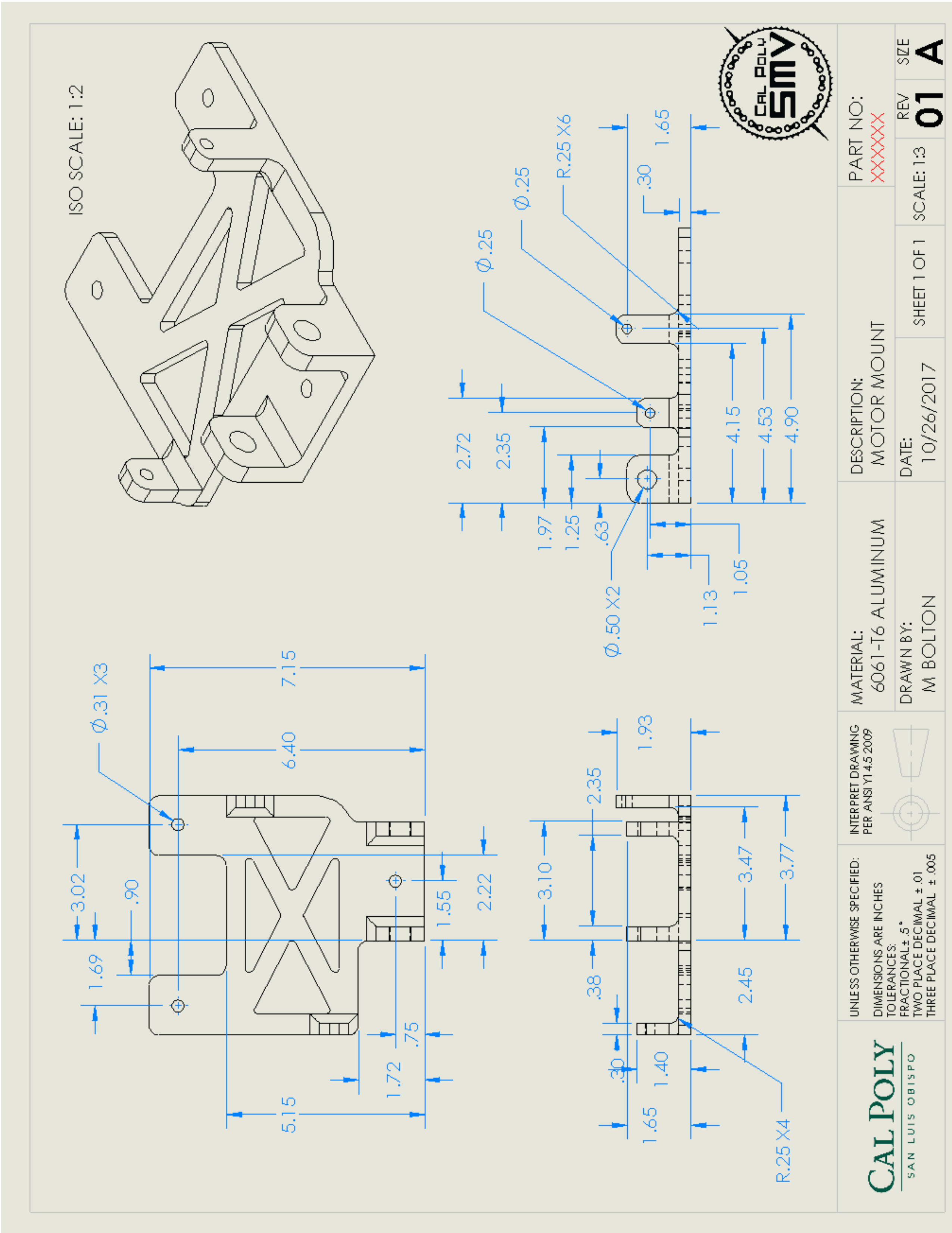
# Appendix H: Gantt Chart

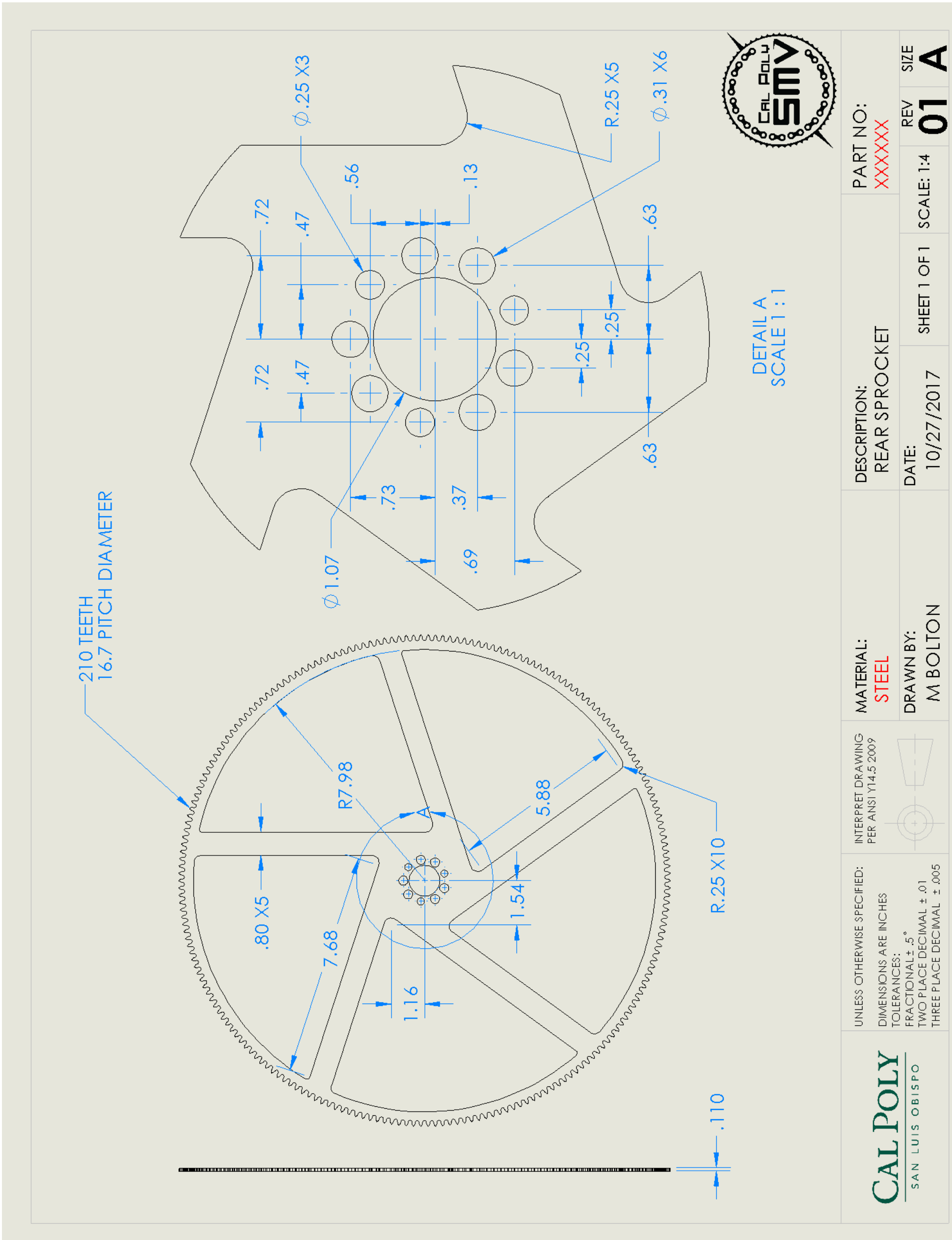


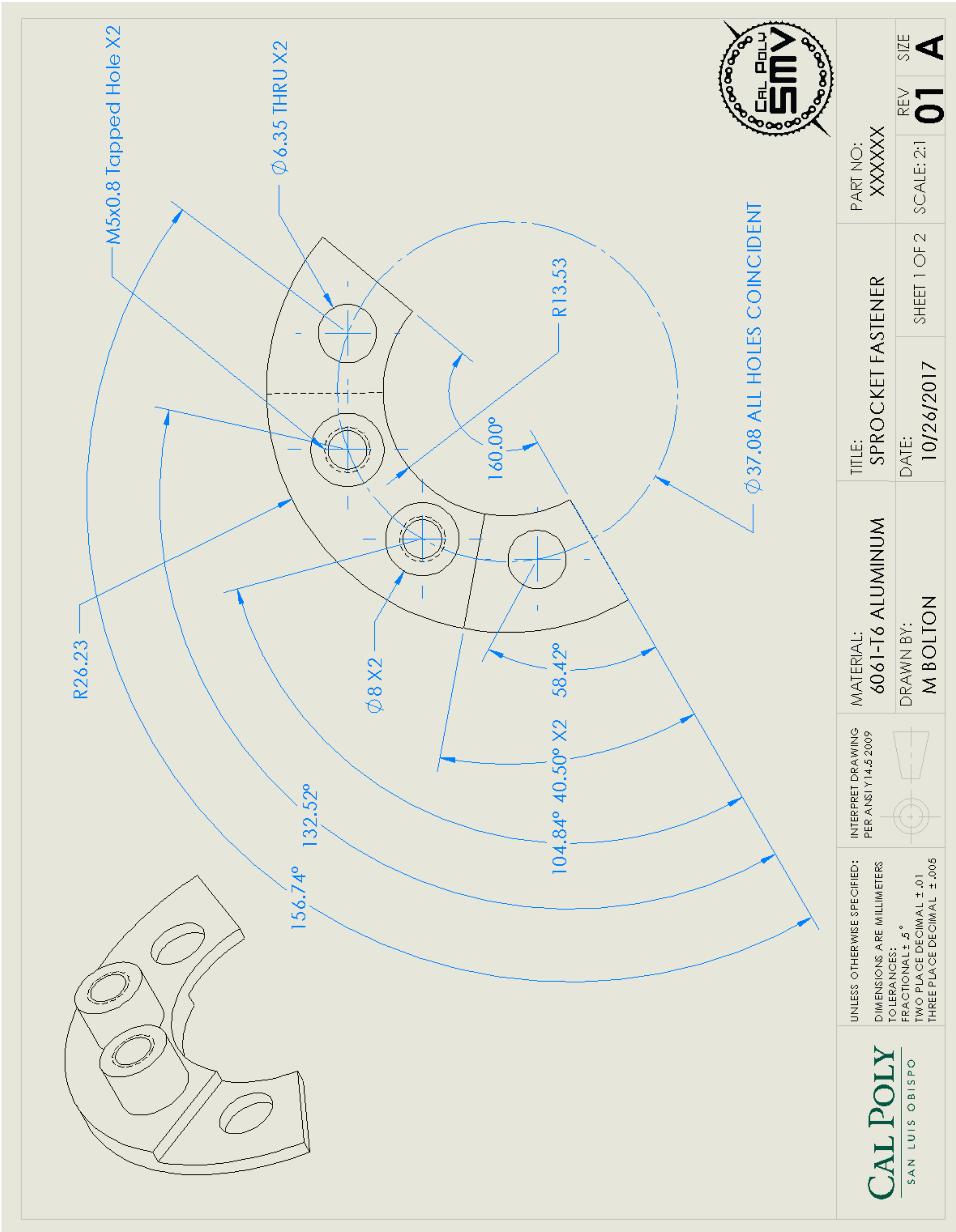

1/17
2/17
3/17
4/17
5/17
6/17
7/17
8/17
9/17
10/17
11/17
12/17
1/18
2/18
3/18
Hub and Freewheel 77%
Model Freewheel Design 100%
Test Frictional Losses and Engagem... 100%
Decide Final Freewheel Design 100%
Design Rear Sprocket Adapter 100%
Manufacture Rear Sprocket Adapter 0%
Testing 0%
Sprocket Deflection 0%
Freewheel Deflection 0%
Hub Deflection 0%
Capable Tolerances for Alignement Jig 0%
Repeatability testing for removal and... 0%
Mike
Mike
Mike
Mike
Mike
Mike
Mike
Justin
Mike

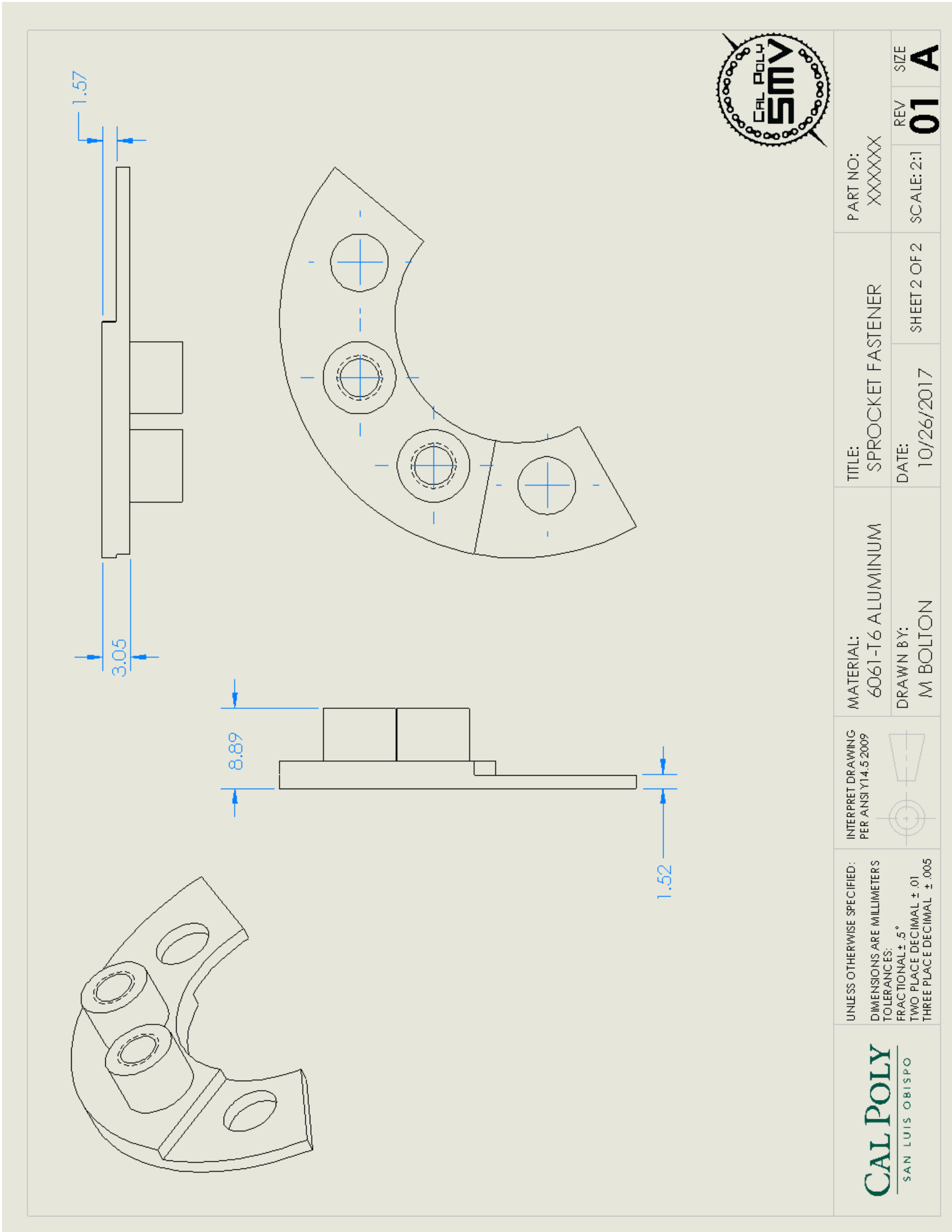
## Appendix I: Component Drawings

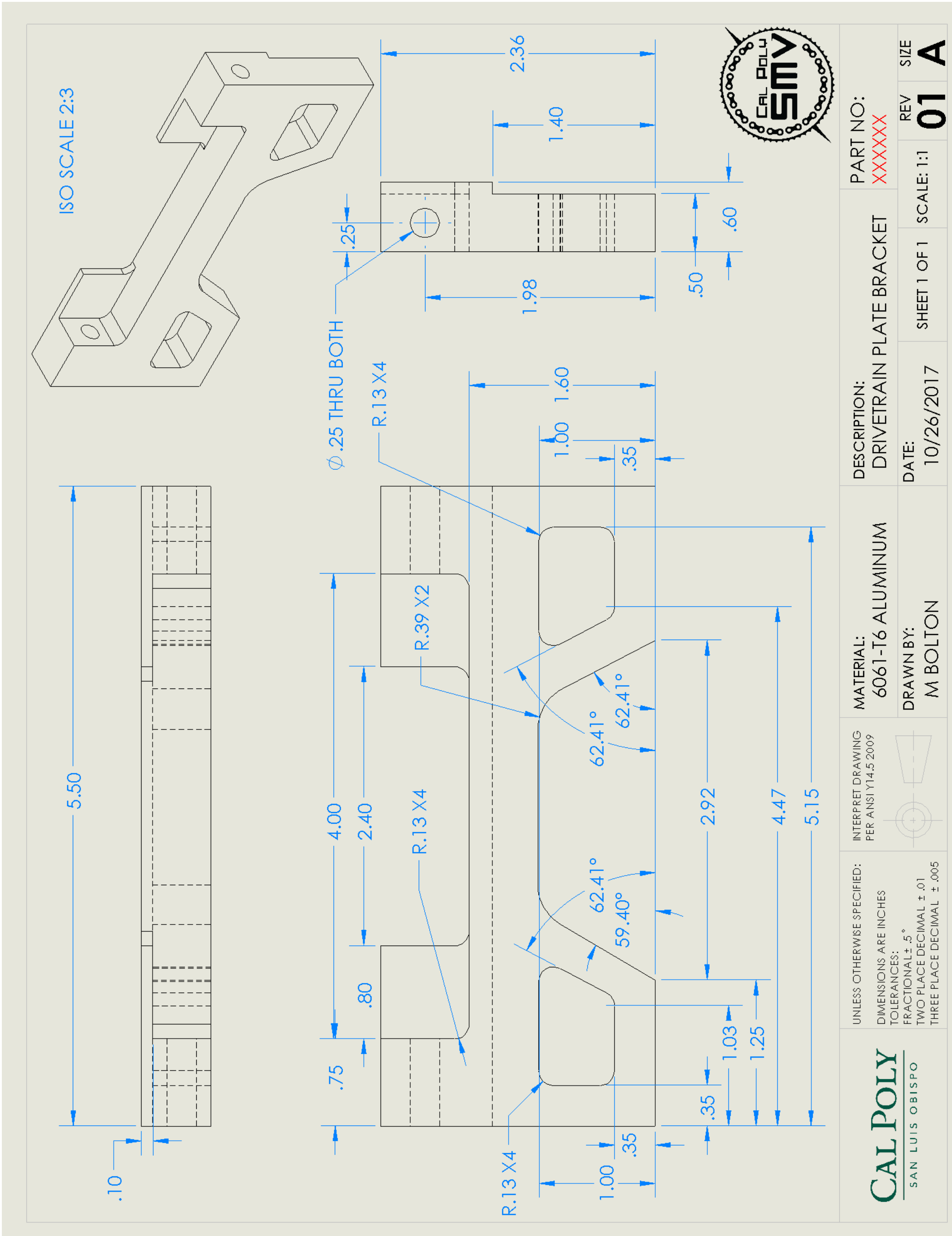


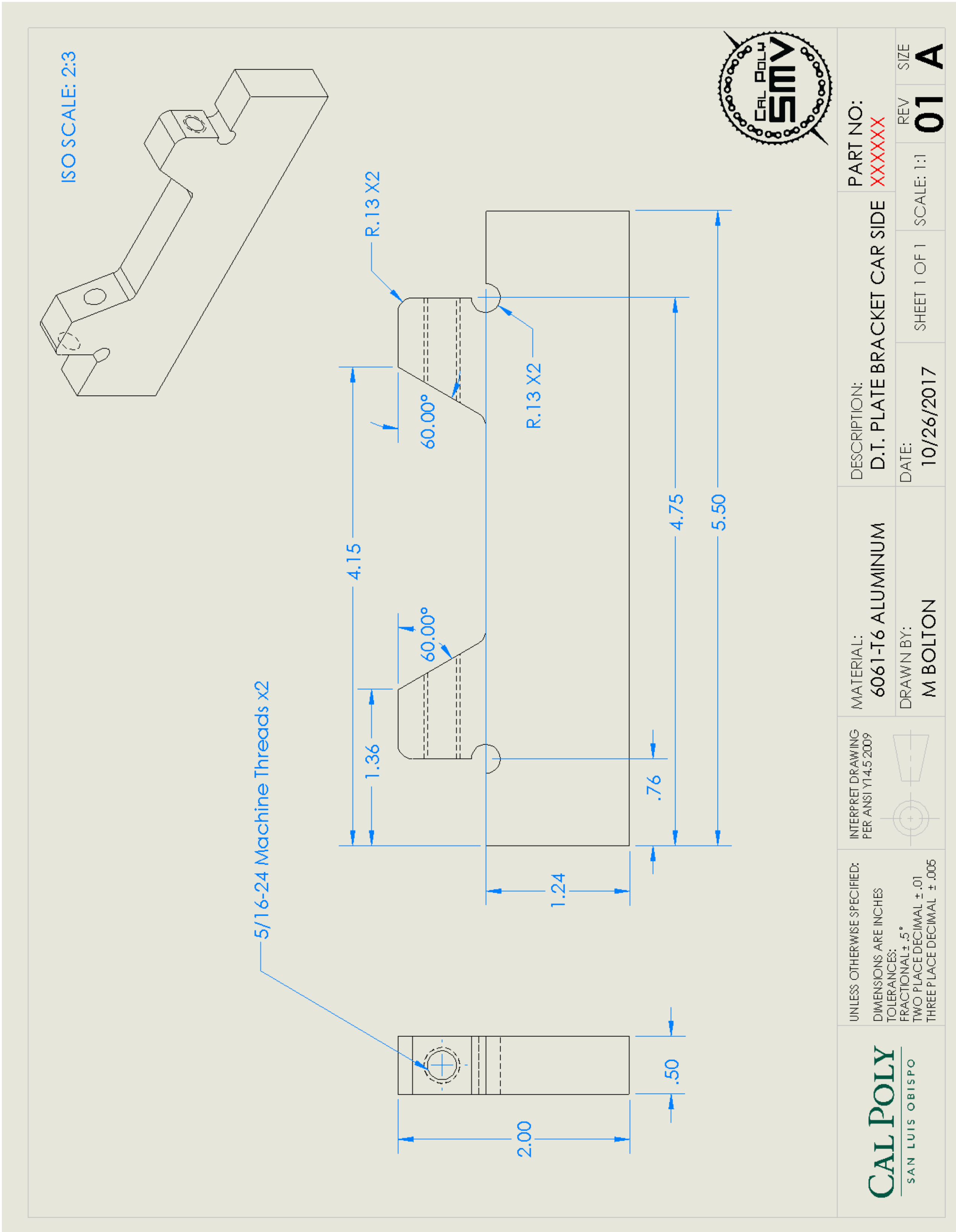

ISO SCALE: 1:2
Ø.31 X3
3.02
1.69
.90
7.15
6.40
5.15
1.72
.75
1.55
2.22
2.35
3.10
.38
.30
2.45
1.65
1.40
1.93
3.47
3.77
R.25 X4
Ø.25
Ø.25
R.25 X6
.30
1.65
2.72
2.35
1.97
1.25
.63
Ø.50 X2
1.13
1.05
4.15
4.53
4.90
CAL POLY SMV
CAL POLY
SAN LUIS OBISPO
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING
PER ANSI Y14.5 2009
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
DESCRIPTION:
MOTOR MOUNT
DATE:
10/26/2017
SHEET 1 OF 1
PART NO:
XXXXX
SCALE: 1:3
REV
01
SIZE
A


210 TEETH
16.7 PITCH DIAMETER
R7.98
5.88
.80 X5
7.68
1.16
1.54
A
R.25 X10
.110
Ø.25 X3
R.25 X5
Ø.31 X6
Ø1.07
.72
.47
.56
.13
.63
.25
.73
.37
.69
DETAIL A
SCALE 1 : 1
CAL POLY
SAN LUIS OBISPO
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
STEEL
DRAWN BY:
M BOLTON
DESCRIPTION:
REAR SPROCKET
DATE:
10/27/2017
SHEET 1 OF 1
SCALE: 1:4
PART NO:
XXXXXX
REV
01
SIZE
A
CAL POLY SMV

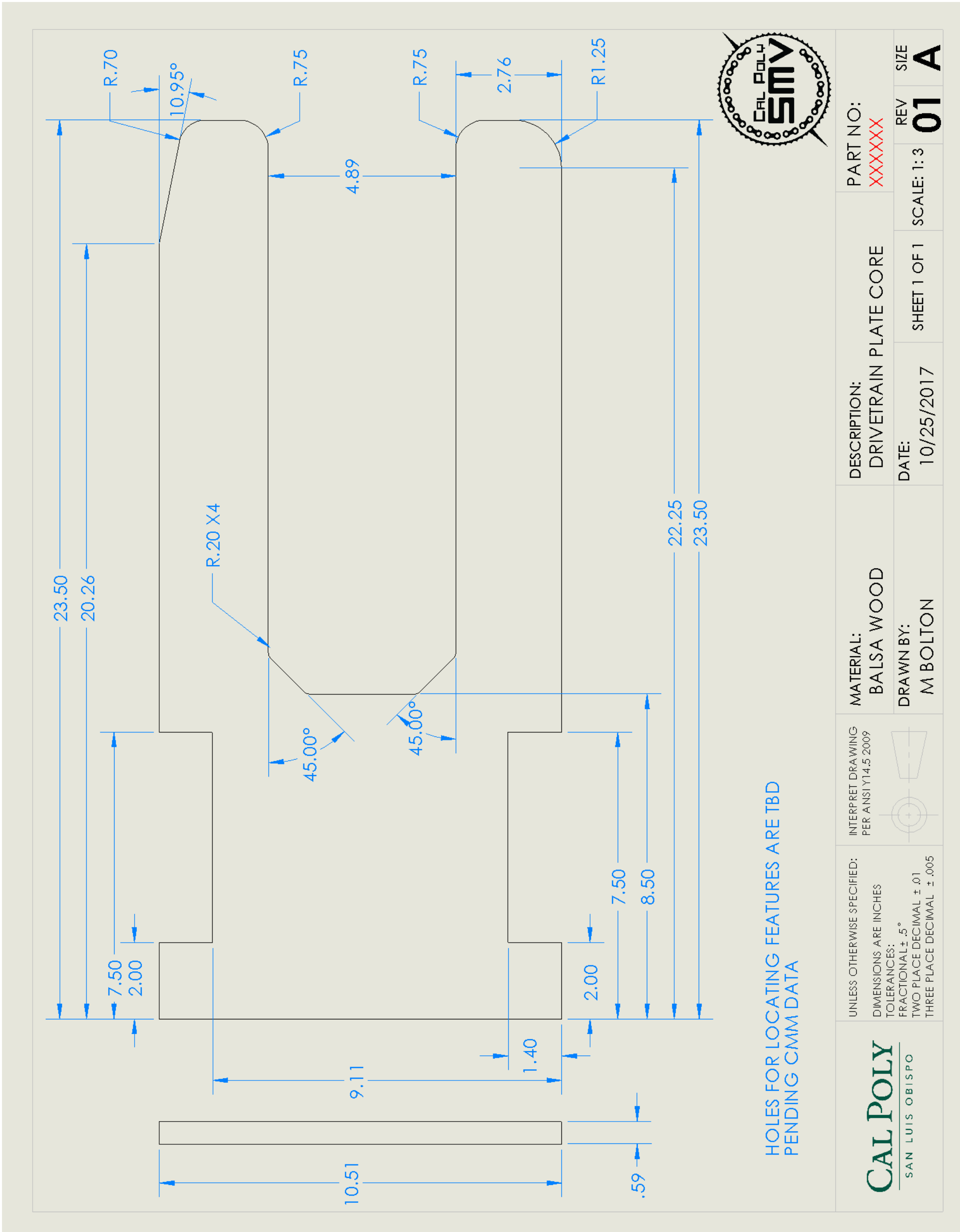

M5x0.8 Tapped Hole X2
Ø6.35 THRU X2
R13.53
160.00°
Ø37.08 ALL HOLES COINCIDENT
R26.23
Ø8 X2
58.42°
104.84° 40.50° X2
132.52°
156.74°
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE MILLIMETERS
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
TITLE:
SPROCKET FASTENER
DATE:
10/26/2017
SHEET 1 OF 2
PART NO:
XXXXXX
SCALE: 2:1
REV
01
SIZE
A
CAL POLY
SAN LUIS OBISPO
CAL POLY SMV

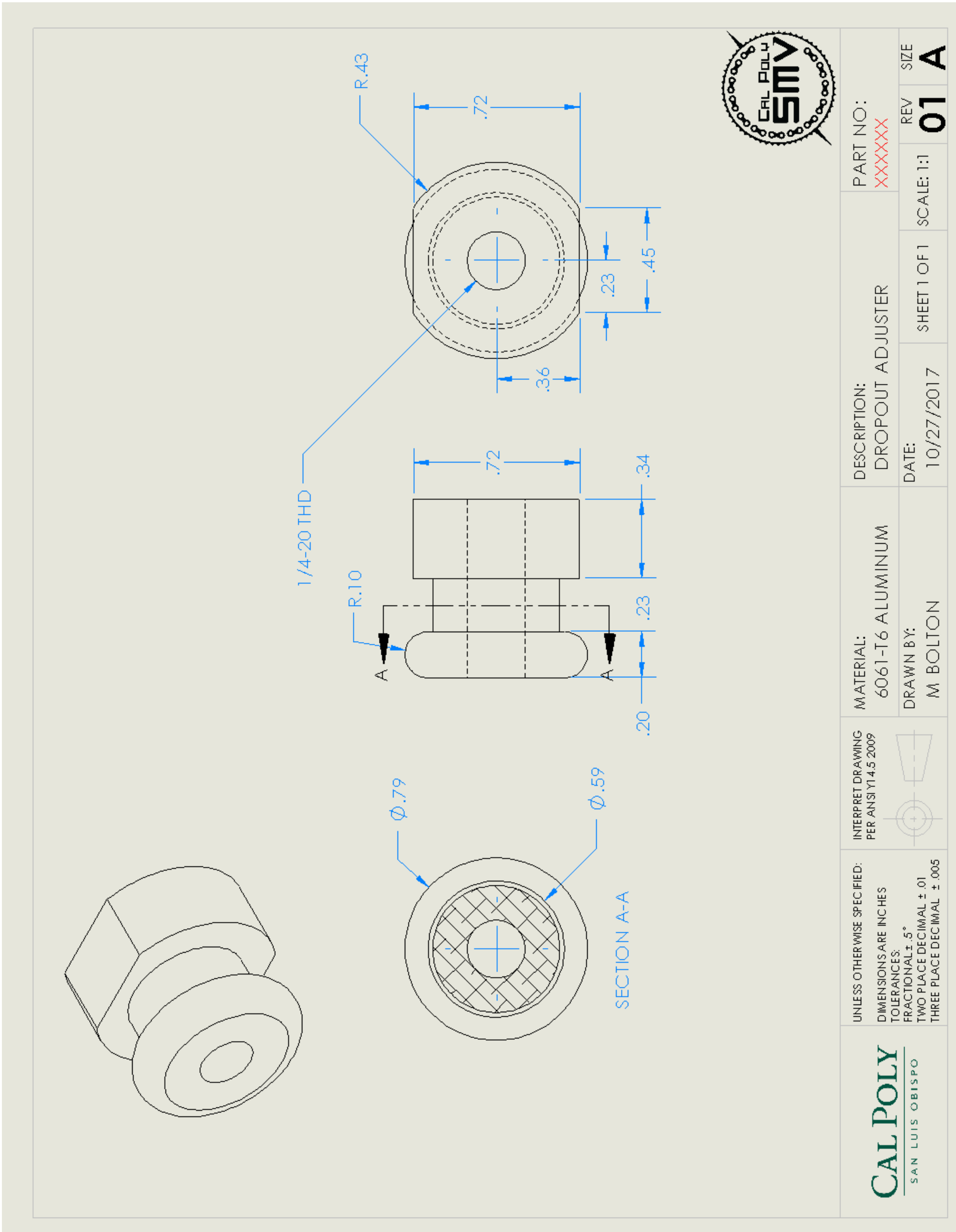

1.57
3.05
8.89
1.52
CAL POLY
SAN LUIS OBISPO
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE MILLIMETERS
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
TITLE:
SPROCKET FASTENER
DATE:
10/26/2017
SHEET 2 OF 2
PART NO:
XXXXXX
SCALE: 2:1
REV
01
SIZE
A


ISO SCALE 2:3
DESCRIPTION:
DRIVETRAIN PLATE BRACKET
PART NO:
XXXXXX
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
DATE:
10/26/2017
SHEET 1 OF 1
SCALE: 1:1
REV
01
SIZE
A
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING
PER ANSI Y14.5 2009
CAL POLY
SAN LUIS OBISPO
CAL POLY SMV
Ø .25 THRU BOTH
R.13 X4
R.39 X2
62.41°
59.40°
5.50
.10
4.00
2.40
.80
.75
2.92
4.47
5.15
1.03
1.25
.35
1.00
1.60
2.36
1.40
1.98
.25
.50
.60

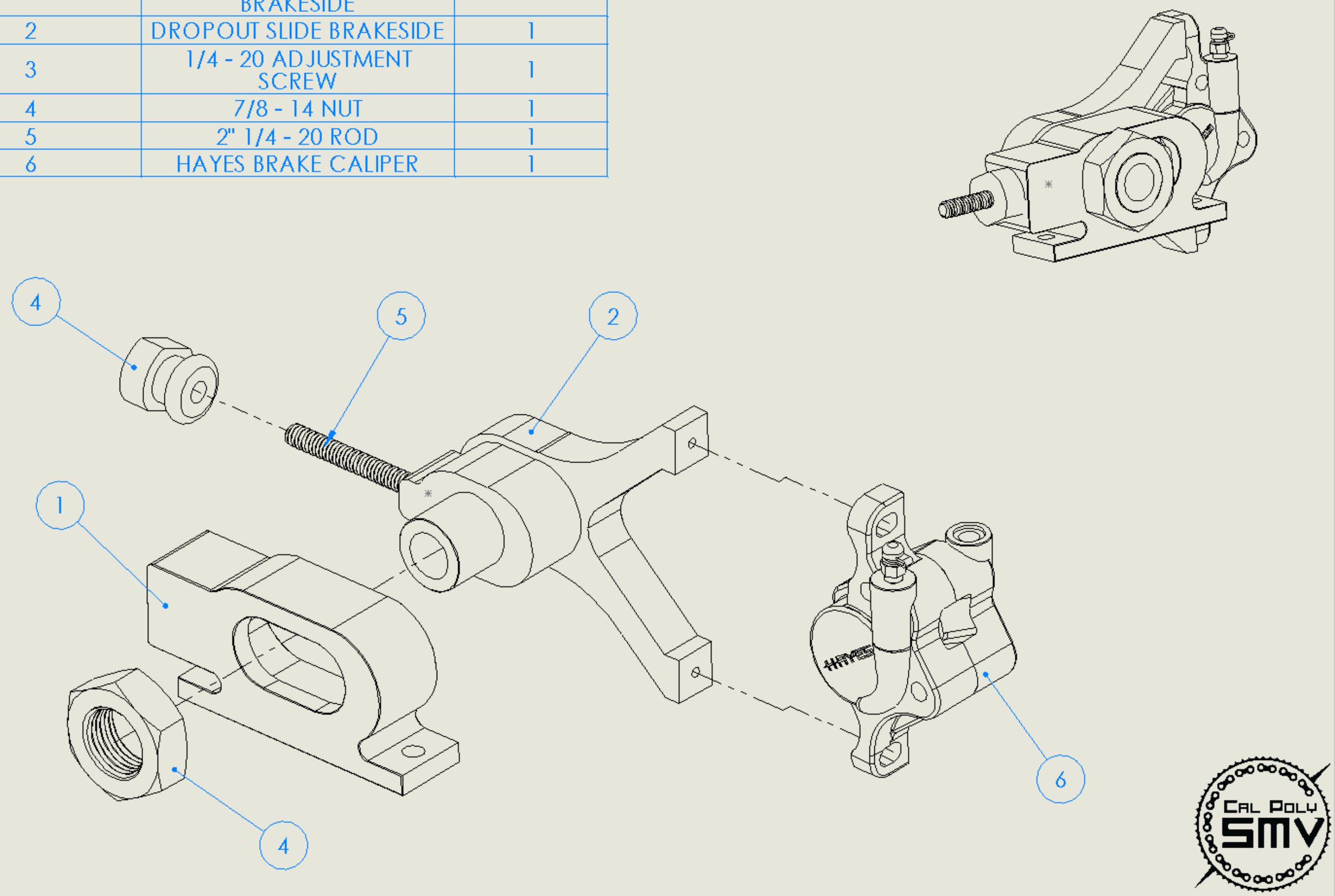


ISO SCALE: 2:3
R.13 X2
R.13 X2
5/16-24 Machine Threads x2
60.00°
60.00°
4.15
1.36
4.75
5.50
.76
1.24
2.00
.50
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
DESCRIPTION:
D.T. PLATE BRACKET CAR SIDE
DATE:
10/26/2017
SHEET 1 OF 1
SCALE: 1:1
PART NO:
XXXXXX
REV
01
SIZE
A
CAL POLY
SAN LUIS OBISPO
CAL POLY SMV


R.70
10.95°
R.75
R.75
2.76
R1.25
4.89
23.50
20.26
R.20 X4
22.25
23.50
45.00°
45.00°
7.50
2.00
7.50
8.50
2.00
1.40
9.11
10.51
.59
HOLES FOR LOCATING FEATURES ARE TBD
PENDING CMM DATA
CAL POLY
SAN LUIS OBISPO
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING
PER ANSI Y14.5 2009
MATERIAL:
BALSA WOOD
DRAWN BY:
M BOLTON
DESCRIPTION:
DRIVETRAIN PLATE CORE
DATE:
10/25/2017
SHEET 1 OF 1
PART NO:
XXXXXX
SCALE: 1:3
REV
01
SIZE
A
CAL POLY
SMV


R.43
.72
.45
.23
.36
1/4-20 THD
.72
.34
R.10
.23
A
A
.20
Ø.79
Ø.59
SECTION A-A
PART NO:
XXXXXX
DESCRIPTION:
DROPOUT ADJUSTER
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
DATE:
10/27/2017
SHEET 1 OF 1
SCALE: 1:1
REV
01
SIZE
A
INTERPRET DRAWING PER ANSI Y14.5 2009
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
CAL POLY
SAN LUIS OBISPO

| ITEM # | PART NAME | QUANTITY |
|---|---|---|
| 1 | DROPOUT HOUSING BRAKESIDE | 1 |
| 2 | DROPOUT SLIDE BRAKESIDE | 1 |
| 3 | 1/4 - 20 ADJUSTMENT SCREW | 1 |
| 4 | 7/8 - 14 NUT | 1 |
| 5 | 2" 1/4 - 20 ROD | 1 |
| 6 | HAYES BRAKE CALIPER | 1 |



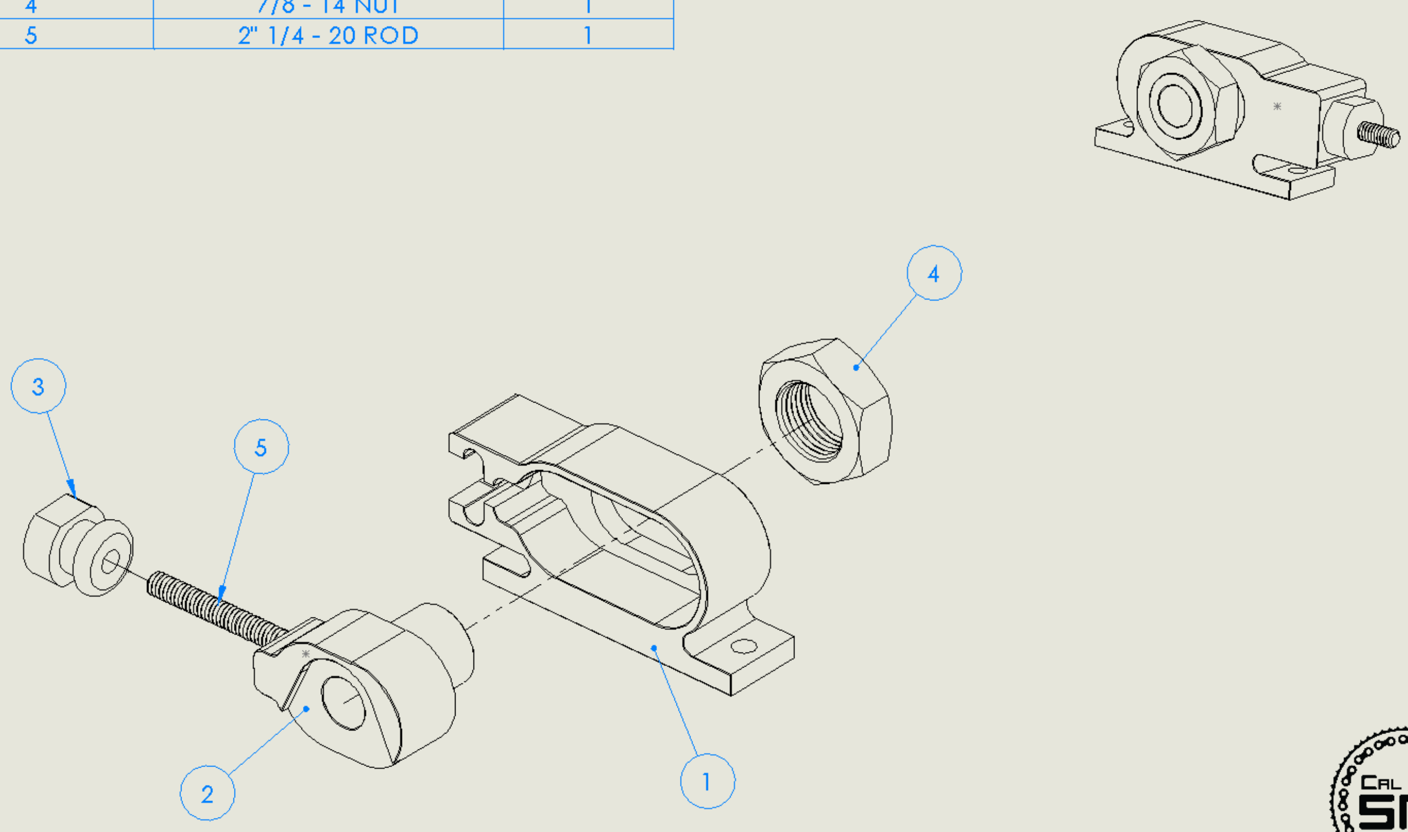

CAL POLY
SAN LUIS OBISPO

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE XXX
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

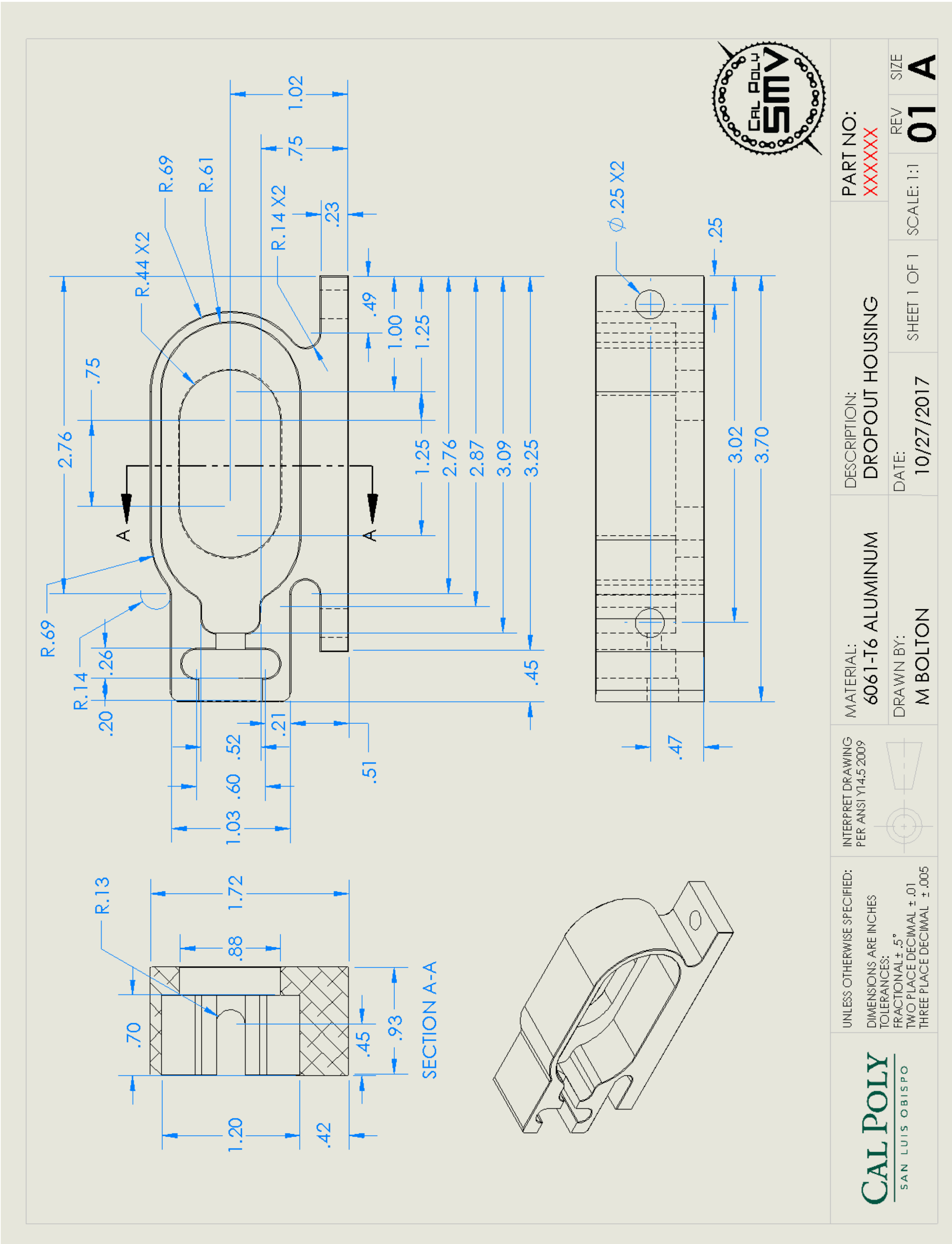
INTERPRET DRAWING PER ANSI Y14.5 2009

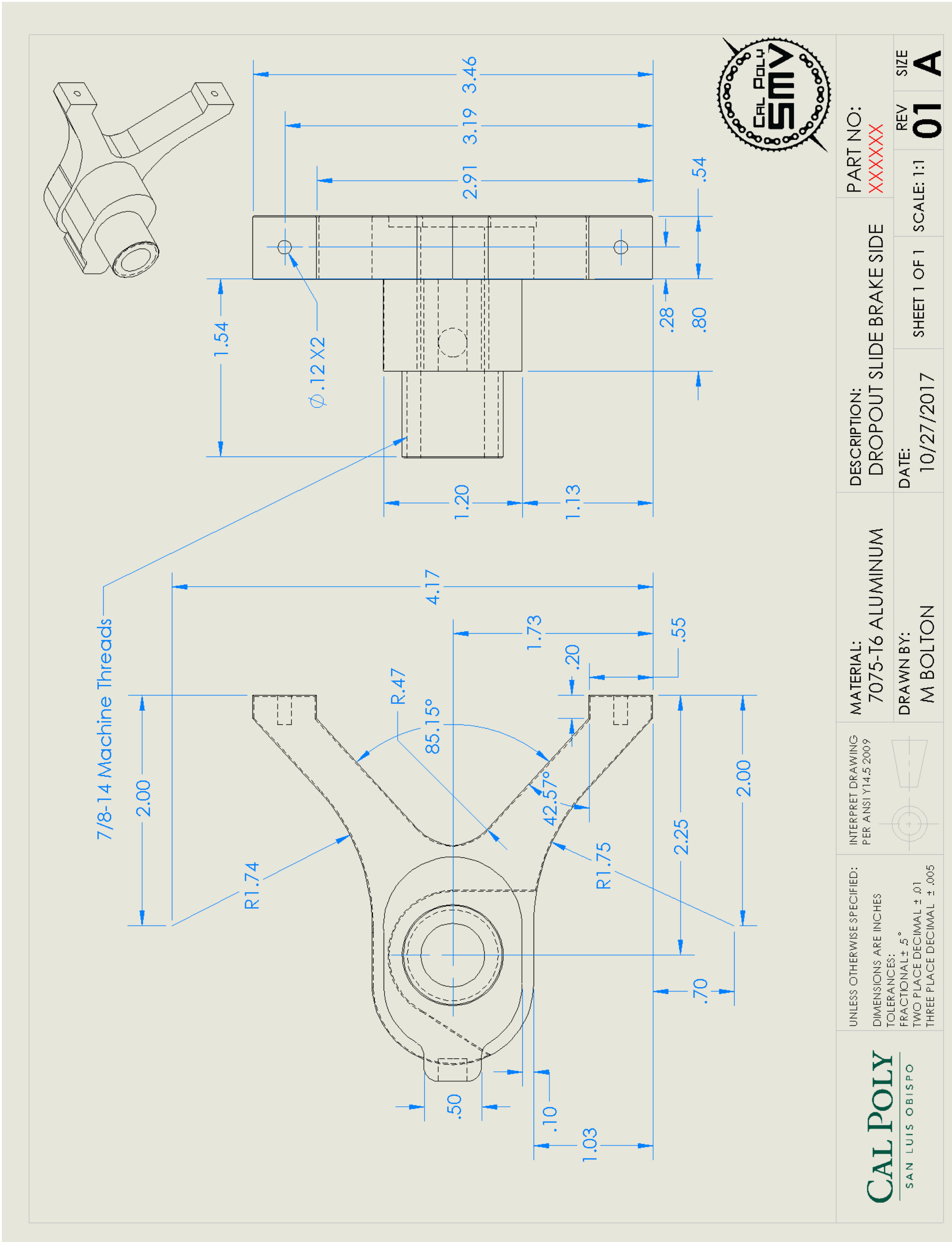
MATERIAL: 6061-T6 ALUMINUM
DRAWN BY: JUSTIN MILLER
DESCRIPTION: DROPOUT ASSEMBLY BRAKESIDE
DATE: 10/27/17
SHEET 1 OF 1
SCALE: 2:3
PART NO: XXXXXX
REV 01
SIZE A

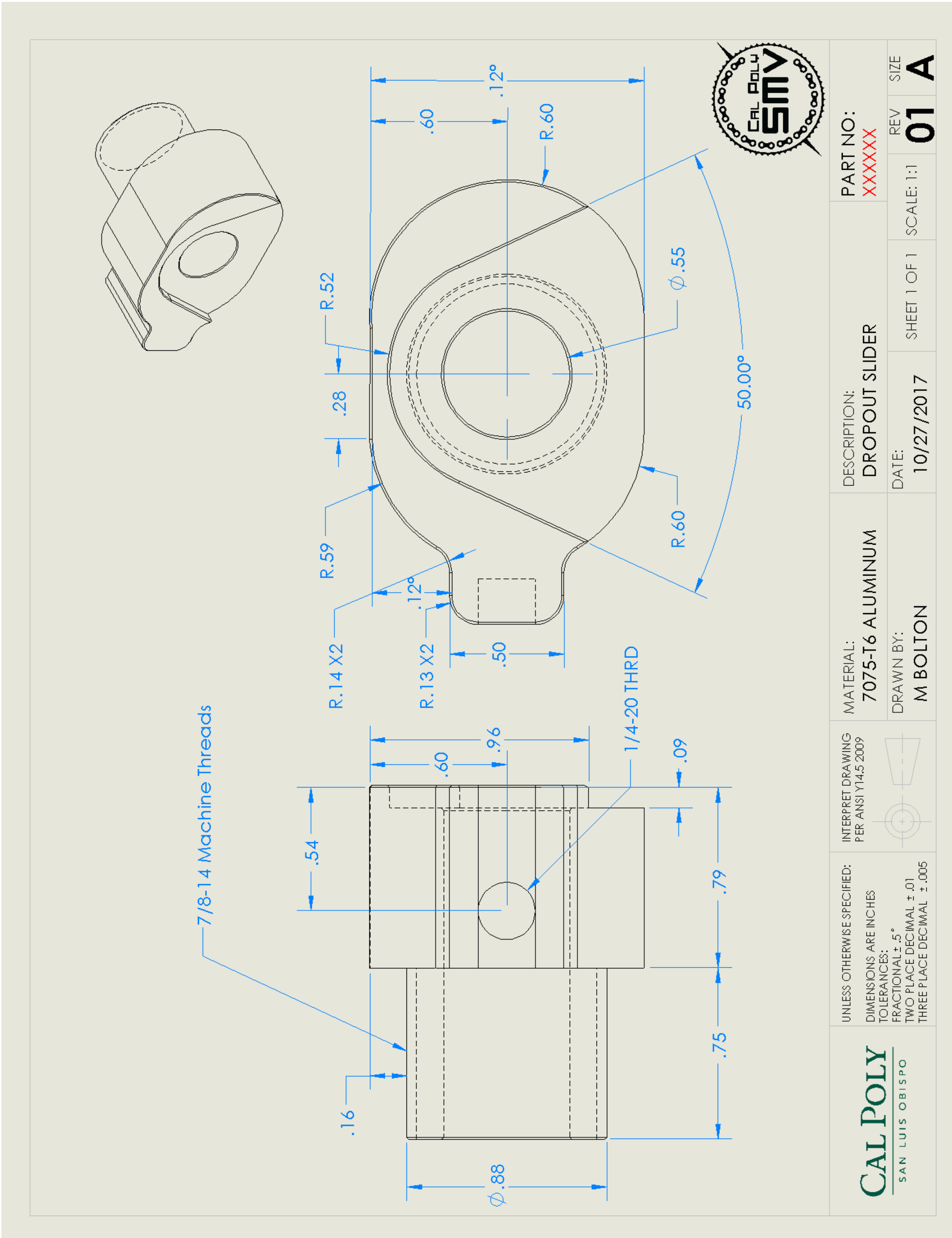
| ITEM # | PART NAME | QUANTITY |
|---|---|---|
| 1 | DROPOUT HOUSING DRIVESIDE | 1 |
| 2 | DROPOUT SLIDE DRIVESIDE | 1 |
| 3 | 1/4 - 20 ADJUSTMENT SCREW | 1 |
| 4 | 7/8 - 14 NUT | 1 |
| 5 | 2" 1/4 - 20 ROD | 1 |




| | | | | | |
|---|---|---|---|---|---|
| CAL POLY SAN LUIS OBISPO | UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE XXX TOLERANCES: FRACTIONAL ± .5° TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005 | INTERPRET DRAWING PER ANSI Y14.5 2009 | MATERIAL: 6061-T6 ALUMINUM | DESCRIPTION: DROPOUT ASSEMBLY DRIVESIDE | PART NO: XXXXXX |
| | | | DRAWN BY: JUSTIN MILLER | DATE: 10/26/17 | SHEET 1 OF 1 · SCALE: 2:3 · REV 01 · SIZE A |


SECTION A-A
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
6061-T6 ALUMINUM
DRAWN BY:
M BOLTON
DESCRIPTION:
DROPOUT HOUSING
DATE:
10/27/2017
SHEET 1 OF 1
PART NO:
XXXXXX
SCALE: 1:1
REV
01
SIZE
A
CAL POLY
SAN LUIS OBISPO


7/8-14 Machine Threads
2.00
4.17
1.73
.55
.20
R.47
85.15°
42.57°
R1.74
R1.75
2.25
2.00
.70
.50
.10
1.03
1.54
⌀.12 X2
2.91
3.19
3.46
.54
.28
.80
1.20
1.13
CAL POLY
SAN LUIS OBISPO
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING
PER ANSI Y14.5 2009
MATERIAL:
7075-T6 ALUMINUM
DRAWN BY:
M BOLTON
DESCRIPTION:
DROPOUT SLIDE BRAKE SIDE
DATE:
10/27/2017
SHEET 1 OF 1
SCALE: 1:1
PART NO:
XXXXXX
REV
01
SIZE
A


7/8-14 Machine Threads
1/4-20 THRD
R.52
R.59
R.60
R.14 X2
R.13 X2
.12°
.60
.28
.50
.96
.54
.09
.79
.75
.16
$\phi$.88
$\phi$.55
50.00°
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE INCHES
TOLERANCES:
FRACTIONAL ± .5°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005
INTERPRET DRAWING PER ANSI Y14.5 2009
MATERIAL:
7075-T6 ALUMINUM
DRAWN BY:
M BOLTON
DESCRIPTION:
DROPOUT SLIDER
DATE:
10/27/2017
SHEET 1 OF 1
SCALE: 1:1
PART NO:
XXXXXX
REV
01
SIZE
A
CAL POLY
SAN LUIS OBISPO

# Appendix J: Bill Of Materials & Part Specifications Sheets

**Indented Bill of Material (BOM)**

**2018 Supermileage Drive Train**

| Part Number | Description | MFG Notes | Vendor | Part # | Qty Needed | Cost | Line Cost | Notes | Matl |
|---|---|---|---|---|---|---|---|---|---|
| P02000 | Full Assembly | | | | | | | | |
| P02100 | Engine Mount Assembly | | | | | | | | |
| P02110 | Engine Mount | CNC'd from 6061 aluminum stock | McMaster Carr | 9246k15 | 1 | $ 108.49 | $ 108.49 | | Al 6061-T6 |
| P02120 | 1/4"-20 x 1-3/4" SHCS | Purchased | McMaster Carr | 90044A124 | 5 | $ 9.71 | $ 9.71 | | Alloy Steel |
| P02130 | OTS locating pins | Purchased | McMaster Carr | 31375A28 | 5 | $ 11.30 | $ 56.50 | | Steel |
| P02130 | 1/4"-20 x 7/16 Flange Nut | Purchased | McMaster Carr | 99904A101 | 1 | $ 4.38 | $ 4.38 | | Steel |
| P02200 | Drive Train Plate Assembly | | | | | | | | |
| P02210 | Balsa Core | Custom cut by a balsa wood company | Custom Balsa | | 1 | $ 19.70 | $ 19.70 | | Balsa Wood |
| P02220 | Carbon Sheet | Custom cut per drawing | | | | $ 10.12 | $ 10.12 | | Carbon Fiber |
| P02230 | Mounting Hardware | CNC'd from 6061 aluminum stock from SMV | McMaster Carr | | | | | | Al 6061-T6 |
| P02300 | Brakeside Rear Drop Out Assembly | | | | | | | | |
| P02310 | Brakeside Drop Out Housing | CNC'd from 2"x2"x6" 6061 aluminum stock | Cal Poly SMV | | --- | --- | already supplied by | Al 6061-T6 | Al 6061-T6 |
| P02320 | Brakeside Drop Out Slide | CNC'd from 2"x2"x6" 7075 aluminum stock | Cal Poly SMV | | -- | --- | already supplied by | Aluminum 7075 | Aluminum 7075 |
| P02330 | 7/8"-1/4 Nut | Purchased | McMaster Carr | 9185 | 1 | $ 0.92 | $ 0.92 | | Alloy Steel |
| P02340 | 1/4"-20x1 1/2"" SHCS | Purchased | McMaster Carr | 90044A127 | 1 | 10.22 | $ 10.22 | | Steel |
| P02400 | Brakeside Rear Drop Out Assembly | | | | | | | | |
| P02410 | Brakeside Drop Out Housing | CNC'd from 2"x2"x6" 6061 aluminum stock | Cal Poly SMV | | --- | --- | already supplied by | Al 6061-T6 | Al 6061-T6 |
| P02420 | Brakeside Drop Out Slide | CNC'd from 2"x2"x6" 7075 aluminum stock | Cal Poly SMV | | -- | --- | already supplied by | Aluminum 7075 | Aluminum 7075 |
| P02430 | 7/8"-1/4 Nut | Purchased | McMaster Carr | 9185 | 1 | $ 0.92 | $ 0.92 | | Alloy Steel |
| P02440 | 1/4"-20x1 1/2"" SHCS | Purchased | McMaster Carr | 90044A127 | 1 | 10.22 | $ 10.22 | | Steel |
| | | | | | | | | | Alloy Steel |
| P02500 | Rear Hub Assembly | | | | | | | | |
| P02510 | Odyssey Hub | Purchased | Odyssey | | 1 | $ 180.00 | $ 180.00 | | Al 6061-T6 |
| P02520 | Hub Adapter | CNC'd from 6061 aluminum stock | McMaster Carr | --- | --- | Purchased above | Purchased above | | Al 6061-T6 |
| P02530 | M5 bolts | Purchased | McMaster Carr | | 6 | $ 10.00 | $ 10.00 | | Steel |
| P02540 | Rear Sprocket | Water jet cut from Carbon Steel | McMaster Carr | 6544K68 | | $ 80.26 | $ 355.26 | Additional cost for stretcher leveling | Carbon Steel |
| - | # 25 Chain | | | | | | | | |
| P02600 | #25 Chain | Purchased | McMaster Carr | 6261K171 | 1 | $ 35.98 | $ 35.98 | | |
| | | | | | Total part cost | | $ 796.90 | | |
| | | | | Engine Mount Assembly Total Cost | | | $ 174.70 | | |
| | | | | Drive Train Plate Assembly Total cost | | | $ 29.82 | | |
| | | | | Rear Hub Assembly Total Cost | | | $ 545.26 | | |
| | | | | Rear Drop Outs Assembly Total cost | | | $ 11.14 | | |
| | | | | # 25 Chain Total Cost | | | $ 35.98 | | |

# HUBS / CLUTCH V2 FREECOASTER

## CLUTCH V2 FREECOASTER

The Clutch Hub is a traditional clutch-based coaster hub design that eliminates all of the problematic elements on ordinary hubs and only keeps the parts that work. A new, super-strong FULL 14mm axle, bearing configuration, and drag mechanism design stops the need for constant repairs. Equipped with external slack adjustment, which can be done with a 2.5mm hex key through a hub shell access hole (can be adjusted without removing the wheel), and an improved plastic hub guard spec.

Same Clutch coaster hub you've come to love, but with a few upgrades.

## FEATURES

- Maximum strength axle design
- Super-durable bearing arrangement throughout
- 2.5mm hex key slack adjustment through hub shell access hole. No wheel removal required.
- Super-durable drag mechanism design
- Crankflip ready
- Protected by US Patent 9,469,157
- Protected by Taiwan Patents M487850, M487851, M502583

## IMPROVEMENTS OVER THE CLUTCH V1

- 14mm chromoly axle bolts on both the drive and non-drive side for greater reliability
- Improved clutch engagement
- Improved manufacturing precision throughout
- More slack than the V1 hub
- All-new durable proprietary plastic spec for the removable hub guard

## SPECS

- 36-hole, 2014-T6 aluminum shell
- 9-T Driver
- Left or right hand drive
- 22 oz / 623g

## GUARD OPTIONS

- Plastic hub guard included (but optional to use)
- Alloy hub guard also available

## CURRENT COLORS

- **Black**
- **Polished**

## BUY NOW

Support your local shop or favorite mailorder to upgrade your whip. Find a dealer near you →

**$189.99**

RHD - Black

1 Add to Cart

## Black-Oxide Alloy Steel Socket Head Screw

**1/4"-20 Thread Size, 2-1/2" Long, Fully Threaded**

In stock
$10.22 per pack of 25
90044A127

| | |
|---|---|
| Thread Size | 1/4"-20 |
| Length | 2 1/2" |
| Threading | Fully Threaded |
| Head Diameter | 0.375" |
| Head Height | 0.25" |
| Drive Size | 3/16" |
| Material | Black-Oxide Alloy Steel |
| Hardness | Rockwell C37 |
| Tensile Strength | 170,000 psi |
| Screw Size Decimal Equivalent | 0.250" |
| Thread Type | UNC |
| Thread Spacing | Coarse |
| Thread Fit | Class 3A |
| Thread Direction | Right Hand |
| Head Type | Socket |
| Socket Head Profile | Standard |
| Drive Style | Hex |
| Specifications Met | ASTM A574 |
| System of Measurement | Inch |
| RoHS | Compliant |

These screws are made from an alloy steel that's stronger than Grade 8 steel. Length is measured from under the head.

Black-oxide steel screws are mildly corrosion resistant in dry environments.

## Black-Oxide Alloy Steel Socket Head Screw

**1/4"-20 Thread Size, 1-3/4" Long, Fully Threaded**

In stock
$9.71 per pack of 50
90044A124

| | |
|---|---|
| Thread Size | 1/4"-20 |
| Length | 1 3/4" |
| Threading | Fully Threaded |
| Head Diameter | 0.375" |
| Head Height | 0.25" |
| Drive Size | 3/16" |
| Material | Black-Oxide Alloy Steel |
| Hardness | Rockwell C37 |
| Tensile Strength | 170,000 psi |
| Screw Size Decimal Equivalent | 0.250" |
| Thread Type | UNC |
| Thread Spacing | Coarse |
| Thread Fit | Class 3A |
| Thread Direction | Right Hand |
| Head Type | Socket |
| Socket Head Profile | Standard |
| Drive Style | Hex |
| Specifications Met | ASTM A574 |
| System of Measurement | Inch |
| RoHS | Compliant |

These screws are made from an alloy steel that's stronger than Grade 8 steel. Length is measured from under the head.

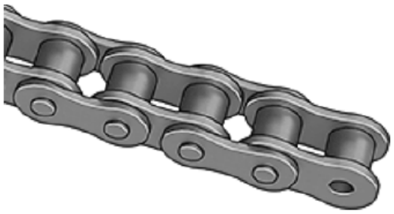
Black-oxide steel screws are mildly corrosion resistant in dry environments.

McMASTER-CARR.

# Roller Chain

**Single Strand, ANSI Number 25, 1/4" Pitch**

In stock
$35.98 Each
6261K171



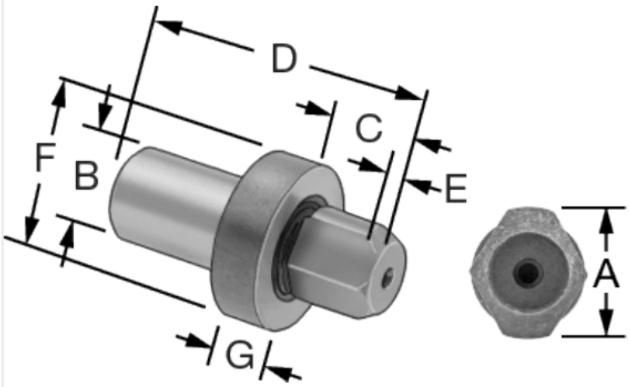
| | |
|---|---|
| Strand Type | Single |
| Pitch Type | Single |
| Material | Steel |
| Roller Chain Standard | ANSI |
| Roller Chain Trade Size | 25 |
| Roller Chain Trade No. | 25 |
| Pitch | 1/4" |
| Roller Diameter | 0.130" |
| Roller Width | 1/8" |
| Working Load | 85 lbs. |
| Component | Chain |
| Includes | One Connecting Link |
| Lengths | 1-20 ft., 50 ft., 100 ft. |
| Length | 7 ft. |

McMASTER-CARR®

## Press-Fit Shoulder-Style Locating Pin

**1/4" Diameter Diamond Head**

In stock
$11.30 Each
31375A28




| | |
|---|---|
| Head Diameter (A) | 1/4" |
| Shank Diameter (B) | 5/16" |
| Head Height (C) | 7/16" |
| Overall Height (D) | 1 3/16" |
| Head End Height (E) | 3/32" |
| Shoulder Diameter (F) | 5/8" |
| Shoulder Height (G) | 17/64" |
| Tolerance | |
| Head Diameter | -0.001" to -0.0005" |
| Shank Diameter | 0.0005" to 0.001" |
| Locating Pin Style | Shoulder |
| Head Style | Diamond |
| Mount Type | Press Fit |
| Features | Tapered End |
| Material | Steel |
| Hardness | Rockwell C60 |
| RoHS | Not Compliant |

The shoulder provides a consistent height reference point and keeps the pins from being pressed through the fixture. Use for tight-tolerance positioning. The tapered end allows easy installation.

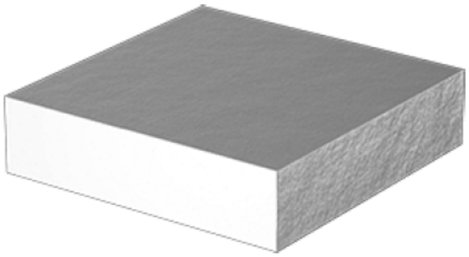
Diamond head minimizes contact with the workpiece to reduce sticking and jamming.

McMASTER-CARR®

# 6061 Aluminum

**2" Thick, 8" x 8"**

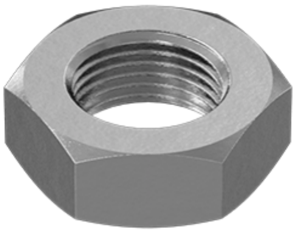
In stock
$108.49 Each
9246K15



| | |
|---|---|
| Material | 6061 Aluminum |
| Cross Section Shape | Rectangle |
| Construction | Solid |
| Appearance | Plain |
| Thickness | 2" |
| Thickness Tolerance | -0.024" to 0.024" |
| Tolerance Rating | Standard |
| Width | 8" |
| Width Tolerance | -0.084" to 0.084" |
| Length | 8" |
| Length Tolerance | -1/16" to 1/16" |
| Yield Strength | 35,000 psi |
| Fabrication | Heat Treated |
| Temper | T6511 |
| Temper Rating | Hardened |
| Hardness | Brinell 95 |
| Hardness Rating | Soft |
| Heat Treatable | Yes |
| Temperature Range | -320° to 300° F |
| Specifications Met | ASTM B221 |
| Flatness Tolerance | 0.010" per in. |
| Magnetic Properties | Nonmagnetic |
| Density | 0.1 lbs./cu. in. |
| Surface Resistivity | 25 Ohm-Cir Mil/ft |
| Melting Point Temperature | 1080° F |
| Modulus of Elasticity | 10.0 ksi × $10^3$ |
| Thermal Conductivity | 1,160 Btu/hr. × in./sq. ft./°F @ 75° F |
| Elongation | 12.5% |
| Material Composition | |
| Aluminum | 95.6-98.6% |
| Chromium | 0.04-0.35% |
| Copper | 0.05-0.4% |
| Iron | 0-0.7% |
| Magnesium | 0.8-1.2% |
| Manganese | 0-0.15% |
| Nickel | 0-0.05% |

# Brass Thin Hex Nut

## 7/8"-14 Thread Size



Packs of 1

ADD TO ORDER

In stock
$3.77 per pack of 1
92174A668

| | |
|---|---|
| Material | Brass |
| Thread Size | 7/8"-14 |
| Thread Type | UNF |
| Thread Spacing | Fine |
| Thread Fit | Class 2B |
| Thread Direction | Right Hand |
| Width | 1 5/16" |
| Height | 31/64" |
| Drive Style | External Hex |
| Nut Type | Hex |
| Hex Nut Profile | Thin |
| System of Measurement | Inch |
| RoHS | Not Compliant |

These nuts are corrosion resistant in wet environments, electrically conductive, and nonmagnetic. Also known as jam nuts, they are about half the height of standard hex nuts. Use them in low-clearance applications or jam one against another nut to hold it in place.

# 18-8 Stainless Steel Hex Head Screw

**M5 x 0.8 mm Thread, 6 mm Long**

Packs of 50

ADD TO ORDER

In stock
$7.71 per pack of 50
91287A081

| | |
|---|---|
| Thread Size | M5 |
| Thread Pitch | 0.8 mm |
| Length | 6 mm |
| Threading | Fully Threaded |
| Head Width | 8 mm |
| Head Height | 3.5 mm |
| Material | 18-8 Stainless Steel |
| Hardness | Not Rated |
| Tensile Strength | 100,000 psi |
| Thread Type | Metric |
| Thread Spacing | Coarse |
| Thread Fit | Class 6g |
| Thread Direction | Right Hand |
| Head Type | Hex |
| Hex Head Profile | Standard |
| Drive Style | External Hex |
| Specifications Met | DIN 933 |
| System of Measurement | Metric |
| RoHS | Compliant |

These screws have good chemical resistance and may be mildly magnetic. 18-8 stainless steel screws in metric sizes are also known as A2 stainless steel screws. Length is measured from under the head.

## Multipurpose 6061 Aluminum

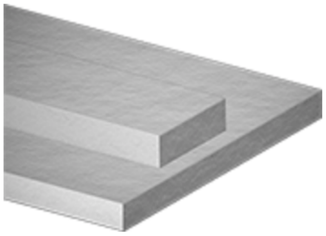
**Rectangular Bar, 1-1/4" x 2-1/4"**

| | |
|---|---|
| (Web) System of Measurement | Inch |
| Material | 6061 Aluminum |
| Cross Section Shape | Rectangle |
| Construction | Solid |
| Thickness | 1 1/4" |
| Thickness Tolerance | -0.012" to 0.012" |
| Tolerance Rating | Standard |
| Width | 2 1/4" |
| Width Tolerance | -0.034" to 0.034" |
| Yield Strength | 35,000 psi |
| Fabrication | Heat Treated |
| Temper | T6511 |
| Temper Rating | Hardened |
| Hardness | Brinell 95 |
| Hardness Rating | Soft |
| Heat Treatable | Yes |
| Appearance | Plain |
| Temperature Range | -320° to 300° F |
| Specifications Met | ASTM B221 |
| Straightness Tolerance | Not Rated |
| Magnetic Properties | Nonmagnetic |
| Density | 0.1 lbs./cu. in. |
| Surface Resistivity | 25 Ohm-Cir Mil/ft |
| Melting Point Temperature | 1080° F |

| | |
|---|---|
| Modulus of Elasticity | 10.0 ksi × $10^3$ |
| Thermal Conductivity | 1,160 Btu/hr. × in./sq. ft./°F @ 75° F |
| Elongation | 12.5% |
| Material Composition | |
| Aluminum | 95.1-98.2% |
| Chromium | 0.4-0.8% |
| Copper | 0.05-0.4% |
| Iron | 0-0.7% |
| Magnesium | 0.8-1.2% |
| Manganese | 0-0.15% |
| Nickel | 0-0.05% |
| Silicon | 0.4-0.8% |
| Titanium | 0-0.15% |
| Zinc | 0-0.25% |
| Zirconium | 0-0.25% |
| Other | 0.15% |
| Length Tolerance | -1" to 1" |
| Length | 1/2 ft., 1 ft., 2 ft., 3 ft., 6 ft. |
| RoHS | Not Compliant |

Note: See Bill of Materials for specific aluminum stock needed for each part.

## Hard High-Strength 7075 Aluminum Sheets and Bars



Originally developed for aircraft frames, uses for 7075 aluminum now include keys, gears, and other high-stress parts. It is harder and has a higher yield strength than 2024. 7075 is nonmagnetic and heat treatable.

### Sheets

- Yield Strength: 61,000 psi
- Hardness: Brinell 150 (Medium)
- Temper:
  0.032" thick. to 0.190" thick.: T6 (Hardened)
  1/4" thick. and thicker: T651 (Hardened)
- Fabrication: Heat Treated
- Specifications Met: ASTM B209

| | |
|---|---|
| Material | 7075 Aluminum |
| Cross Section Shape | Rectangle |
| Construction | Solid |
| Appearance | Plain |
| Thickness | 0.032" |
| Thickness Tolerance | -0.002" to 0.002" |
| Tolerance Rating | Standard |
| Width | 6" |
| Width Tolerance | -1/16" to 1/16" |
| Length | 6" |
| Length Tolerance | -1/16" to 1/16" |
| Yield Strength | 61,000 psi |
| Fabrication | Heat Treated |
| Temper | T6 |
| Temper Rating | Hardened |
| Hardness | Brinell 150 |
| Hardness Rating | Medium |
| Heat Treatable | Yes |
| Temperature Range | -320° to 210° F |
| Specifications Met | ASTM B209 |
| Flatness Tolerance | Not Rated |
| Magnetic Properties | Nonmagnetic |
| Density | 0.1 lbs./cu. in. |
| Surface Resistivity | 30 Ohm-Cir Mil/ft |
| Melting Point Temperature | 890° F |
| Modulus of Elasticity | 10.4 ksi × $10^3$ |
| Thermal Conductivity | 900 Btu/hr. × in./sq. ft./°F @ 75° F |
| Elongation | 11% |

Note: See Bill of Materials for specific aluminum stock needed for each part.

## Appendix K: Force Calculations

$$\tau_{BOLT} = \frac{F}{A} = \frac{4F}{\pi \partial^2_{bolt}}$$

$$F_{shear} = \tau \pi \partial^2 \left(\frac{1}{4}\right)$$

$$= (\tau)\,\pi\,(7/8'')^2(1/4)$$

6061-T6

$$F_{shear} = (30000)\,\pi\,(7/8'')^2(1/4)$$

$$\underline{F_{shear} = 18 \times 10^3\ \text{lb}}$$

7075

$$F_{shear} = (48000)(\pi)(7/8'')^2(1/4)$$

$$\underline{F_{shear} = 28.8 \times 10^3\ \text{lb}}$$

$$\text{FASTENING TORQUE} = 0.2\,(F_{shear})(\partial)(1/SF)$$

$$T_{6061} = 0.2\,(18\times10^3\ \text{lb})(7/8'')(1\,\text{ft}/12\,\text{in})(1/2)$$

$$\underline{= 130\ \text{ft/lb}}$$

$$T_{7075} = 0.2\,(28.8\times10^3\ \text{lb})(7/8'')(1\,\text{ft}/12\,\text{in})(1/2)$$

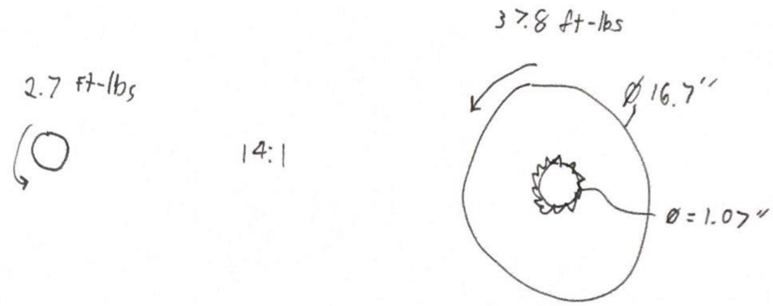
$$\underline{= 210\ \text{ft/lb}}$$

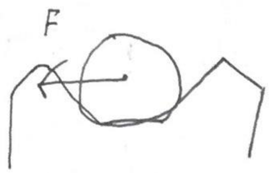
Brass BOLT

$$F_{shear} = 20\times10^3\ \text{lb}$$

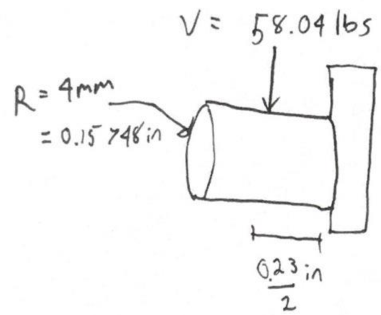
$$\underline{\tau = 150\ \text{ft lb}}$$

SPROCKET Fastener




$$F = 37.8 \text{ ft-lbs} \; \frac{1}{\left(\frac{16.7}{2} - \frac{1.07}{2}\right)\left(\frac{1}{12}\right)} = 58.04 \text{ lbs}$$



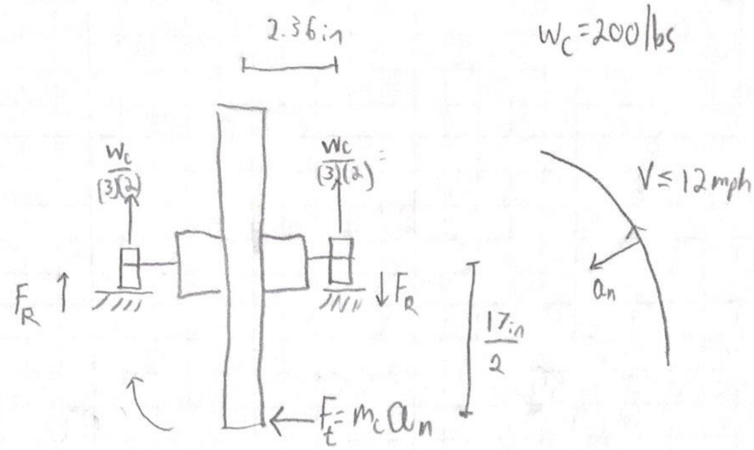

$$\tau_{max} = \frac{4V}{3A} = \frac{4\,(58.04 \text{ lbs})}{3\left(\pi\,(0.15748 \text{ in})^2\right)}$$

$$\boxed{\tau_{max} = 993.3 \text{ psi}}$$

$$\boxed{G_{AL} = 3.9E6 \text{ psi}}$$

FS

$W_c = 200\,\text{lbs}$




$$F_R = F_t\left(\frac{17}{2}\right)\left(\frac{1}{2.36}\right) = m_c a_n \left(\frac{17}{2}\right)\left(\frac{1}{2.36}\right)$$

$$= \frac{W_c}{g}\frac{V^2}{R}\left(\frac{17}{2}\right)\left(\frac{1}{2.36}\right)$$

min required Turning radius

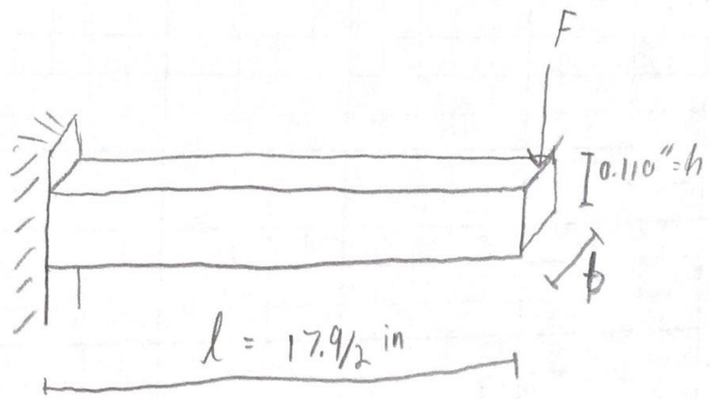
$$= \frac{200\,\text{lbs}}{32.2\,\text{ft/s}^2}\;\frac{\left(12\,\frac{\text{mile}}{\text{hr}}\right)^2}{8\,\text{m}}\left(\frac{17\,\text{in}}{2}\right)\left(\frac{1}{2.36\,\text{in}}\right)$$

$$= \frac{200\,\text{lbf}}{32.2\,\text{ft/s}^2}\;\frac{\left(12\,\frac{\text{m}}{\text{hr}}\left(\frac{5280\,\text{ft}}{1\,\text{mile}}\right)\left(\frac{1\,\text{hr}}{3600\,\text{s}}\right)\right)^2}{8\,\text{m}\left(\frac{3.28\,\text{ft}}{\text{m}}\right)}\left(\frac{17}{2}\right)\left(\frac{1}{2.36}\right)$$

$$F_R = 264.1\,\text{lbs}$$

$$\sigma_t = \frac{F_R + \frac{W_c}{(3)(2)}}{A} = \frac{264.1 + \frac{200}{(3)(2)}}{(0.1)(0.95)} = \boxed{3130.7\,\text{psi} = \sigma_{tension}}$$

SPROCKET SPOKE WIDTH




$F = 5.29\,N$

$$y_{max} = -\frac{F l^3}{3EI}$$

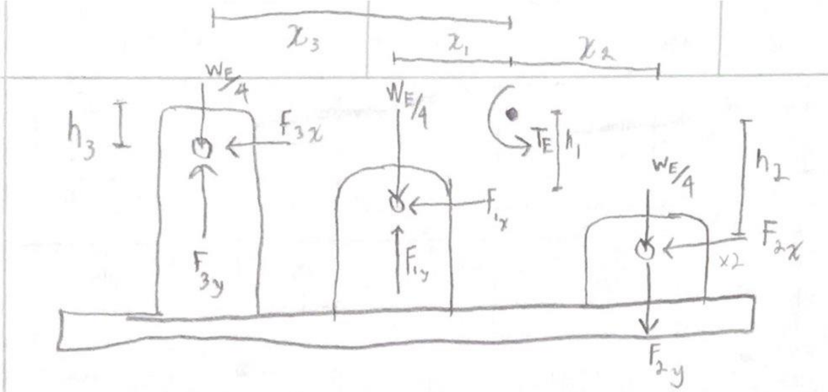
max desired Sprocket Tip deflection $\leq 0.01"$

$$I = -\frac{F l^3}{3 E y_{max}}$$

$$\frac{1}{12} b h^3$$

$$b = \frac{-12 F l^3}{3 h^3 E y_{max}}$$

$$b = \frac{-12\,(5.29N)\left(\frac{0.224809\,lbf}{1N}\right)\left(\frac{17.9}{2}\,in\right)^3}{3\,(0.110\,in)^3\,(15E6\,psi)(0.01\,in)}$$




$T_E = 2.7$ ft-lbs.

$T_E = 32.4$ in-lbs

$W_E = 20$ lbs

1:

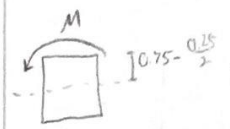
$$F_{1x} = \frac{T_E}{4}\left(\frac{1}{h_1}\right) = \frac{32.4^{\text{in-lb}}}{4}\left(\frac{1}{3.33\text{in}}\right) = 2.43\text{ lbf} = F_{1x} \leftarrow$$

$$F_{1y} = \frac{T_E}{4}\left(\frac{1}{x_1}\right) = \frac{32.4\text{ in lb}}{4}\left(\frac{1}{3.93\text{in}}\right) = 2.06\text{ lbf} = F_{1y} \uparrow$$

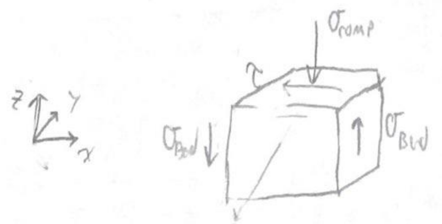
$$F_{1ynet} = F_{1y} + \frac{W_E}{4} = 2.06 - \frac{20}{4} = -2.94\text{ lbf} = F_{1ynet}$$ compression

$$\sigma_{comp} = \frac{F}{A} = \frac{2.94}{(0.75)(0.3)} = \boxed{13.1\text{ psi} = \sigma_{comp}}$$

$$\tau_{shear} = \frac{V}{A} = \frac{2.43}{(0.3)(0.5)} = \boxed{16.2\text{ psi} = \tau_{shear}}$$ through bolt hole




$$\sigma_{Bnd} = \frac{Mc}{I} = \frac{(2.43)(0.75)(0.25)}{\frac{1}{12}(0.75)(0.5)^3} = \boxed{74.6\text{ psi} = \sigma_{Bnd}}$$




$$FoS = \frac{E}{\sigma} = \frac{10E6}{74.6 + 13.1} = 114{,}000$$

Sprocket Tooth Wear

Gear Contact Stress

$$\sigma_c = C_p\left(W^t K_o K_v K_s \frac{K_m}{d_p F} \frac{C_f}{I}\right)^{1/2}$$

$$W^t = \frac{33000(2.28hp)}{V} = \frac{33000(2.28hp)}{\frac{\pi d n}{12}} = \frac{33000(2.28hp)}{\frac{\pi(8.35\,in)(4600\,rpm)}{12}} = 7.49\,lbf$$

$K_o = 1.25$ light shock

$$K_v = \left(\frac{A+\sqrt{V}}{A}\right)^B = \left(\frac{50+56(1-0.8)+\sqrt{4600}}{50+56(1-0.8)}\right)^{0.25(12-6)^{2/3}} = 1.8$$

$K_s = 1$

$K_m = 1$

$$d_p = \frac{N_p}{P_d} = \frac{225}{16.7} = 13.5$$

$F = 0.11$ in

$C_f = 1$

$$I = \frac{\cos\phi_t \sin\phi_t}{2m_N}\,\frac{m_G}{m_G+1} = \frac{\cos 20 \sin 20}{2(1)}\,\frac{\frac{d_G}{d_p}}{\frac{d_G}{d_p}+1} = 0.161\,\frac{\frac{16.7}{1.2}}{\frac{16.7}{1.2}+1} = 0.15$$

$$\sigma_c = 2300\,psi\left(7.49\,lbf(1.25)(1.8)(1)\left(\frac{1}{13.5\;0.11\,in}\right)\left(\frac{1}{0.15}\right)\right)^{1/2}$$

$\sigma_c = 20{,}005$ psi

Gear Contact strength

$$\sigma_{c,all} = \frac{S_c Z_N C_H}{S_H K_T K_R} = \frac{190{,}000(0.94)(1)}{SH\;(1)(0.85)} = \frac{210{,}118}{SH}$$

$$SH = \frac{210{,}118}{20{,}005}$$

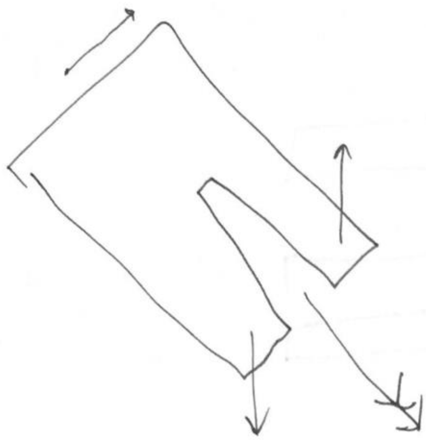
$$\boxed{S_H = 10.5}$$

Gear Bending stress

$$\sigma_{all} = \frac{S_t}{S_F}\,\frac{Y_N}{K_T K_R} = \frac{55{,}000(0.94)}{S_F\;(1)(0.85)}$$

$$S_F = \frac{60{,}823}{W^t K_o K_v K_s \frac{P_d}{F}\,\frac{K_m K_b}{J}} = \frac{60{,}823}{7.49\,lbf(1.25)(1.8)(1)\left(\frac{16.7}{.11}\right)\left(\frac{1(1)}{0.3}\right)}$$

$$\boxed{S_F = 7.1}$$



$T = 51.9 \text{ ft-lb}$

$$\text{Shear } E_{\text{balsa-carbon}} = \frac{0.04}{(0.04)(0.59)}\left(\overset{725188}{\cancel{33E6}}\right) + \left(1 - \frac{0.04}{(0.04)(0.59)}\right)\left(\overset{0.033E6}{\cancel{538000}}\right)$$

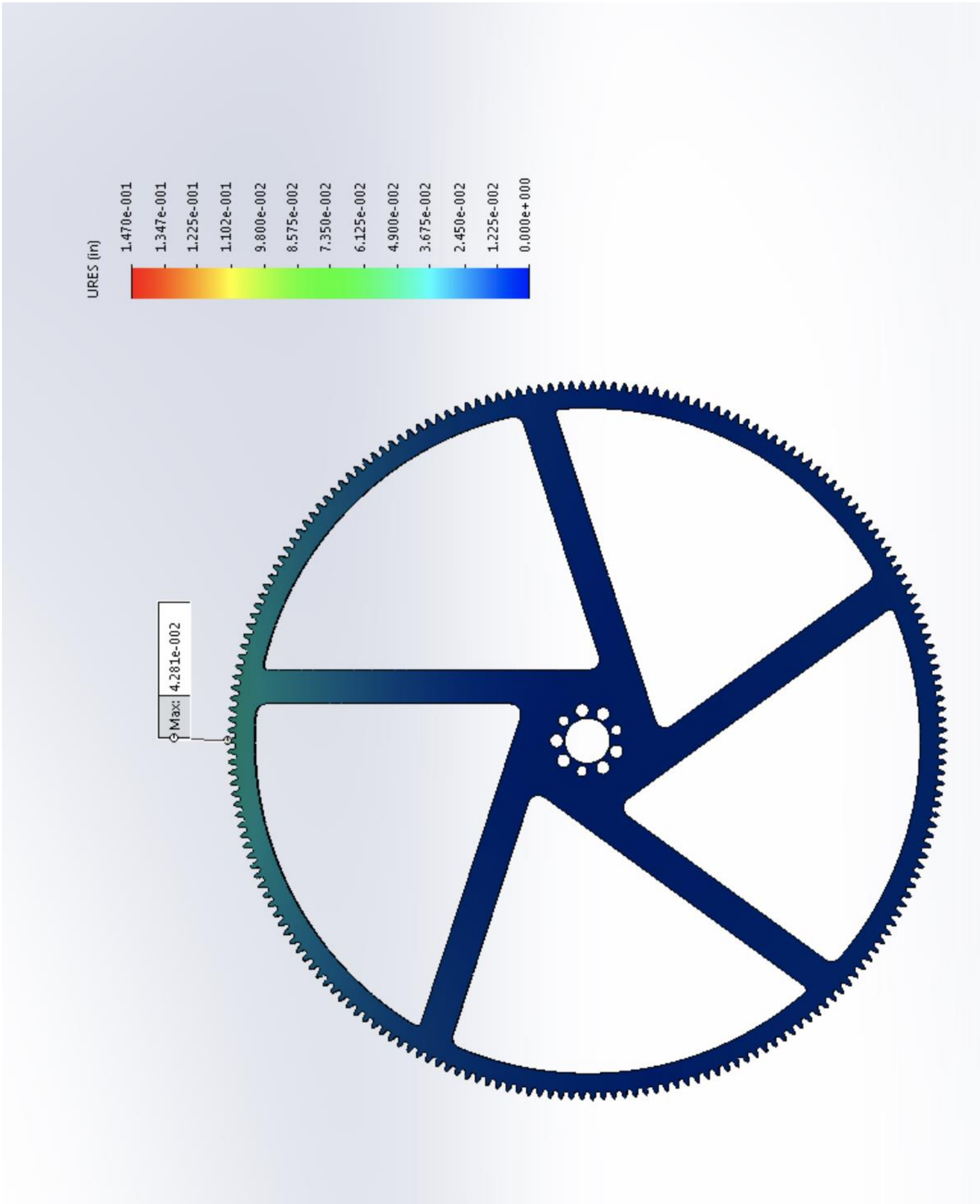
$$E = \overset{1206200}{\cancel{55.9E6}}\text{ psi}$$

$$\tau_{max} = \frac{T r_o}{J} = \frac{(51.9 \text{ ft-lb})\left(\frac{10.51}{2}\text{ in}\right)}{\cancel{38.98 \text{ lb}\cdot\text{in}^2} \quad \frac{(10.51)(0.59)}{12}\left(10.51^2 + 0.59^2\right)\text{ in}^4\left(\frac{1\text{ ft}}{12\text{ in}}\right)^3}$$

$$\tau_{max} = 8230.7\ \frac{\text{lb}}{\text{ft}^2}\left(\frac{1\text{ ft}}{12\text{ in}}\right)^2$$

$$\boxed{\tau_{max} = 57.6 \text{ psi}}$$

# Appendix L: Sprocket FEA



# Appendix M: Design Verification Plan and Report

| Report Date | | | Sponsor: Fabijanic | | | | | | Component/Assembly | | REPORTING ENGINEER: | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| TEST PLAN | | | | | | | | | | TEST REPORT | | | |
| Item No | Specification or Clause Reference | Test Description | Acceptance Criteria | Test Respons | Test Stage | SAMPLES | | TIMING | | TEST RESULTS | | | NOTES |
| | | | | | | Quantity | Type | Start date | Finish date | Test Result | Quantity Pass | Quantity Fail | |
| 1 | 4 | Sprocket/hub diffection test | < 0.2° | Justin | DV | 1 | DV | TBD | TBD | | | | |
| 2 | 2 | Drivetrain weight test/ weight | < 4.2 kg | Heather | DV | 1 | DV | TBD | TBD | | | | |
| 3 | 3 | Inertia dynamometer efficiency | > 80 % | Mike | DV | 5 | DV | TBD | TBD | | | | |
| 4 | 8 | Installation alignment repeatability testing<br>- angle tolerance between sprockets | < 1° | Justin | DV | 12 | DV | TBD | TBD | | | | |
| 5 | 8 | Installation alignment repeatability testing<br>- horizontal center to center distance between sprockets | < 0.02" | Justin | DV | 12 | DV | TBD | TBD | | | | |
| 6 | 6 | Insallation time testing | < 30 minutes | Mike | DV | 12 | DV | TBD | TBD | | | | |
| 7 | | | | | | | | | | | | | |

# Appendix N: Hazard Checklist

ME428/429/430 Senior Design Project 2015-2016

**SENIOR PROJECT CRITICAL DESIGN HAZARD IDENTIFICATION CHECKLIST**

Team: Cal Poly Supermileage Drive train Advisor: Fabijanic

| Y | N | |
|---|---|---|
| ☐ | ☒ | Do any parts 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 adequately guarded? |
| ☐ | ☒ | Does any part of the design undergo high accelerations/decelerations that are exposed to the user? |
| ☐ | ☒ | Does the system have any large moving masses or large forces that can contact the user? |
| ☐ | ☒ | Does the system produce a projectile? |
| ☐ | ☒ | Can the system to fall under gravity creating injury? |
| ☐ | ☒ | Is the user exposed to overhanging weights as part of the design? |
| ☐ | ☒ | Does the system have any sharp edges exposed? |
| ☐ | ☒ | Are there any ungrounded electrical systems in the design? |
| ☐ | ☒ | Are there any large capacity batteries or is there electrical voltage in the system above 40 V either AC or DC? |
| ☐ | ☒ | Is there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids when the system is either on or off? |
| ☐ | ☒ | Are there any explosive or flammable liquids, gases, dust, or fuel in the system? |
| ☐ | ☒ | Is the user of the design required to exert any abnormal effort and/or assume a an abnormal physical posture during the use of the design? |
| ☐ | ☒ | Are there any materials known to be hazardous to humans involved in either the design or the manufacturing of the design? |
| ☐ | ☒ | Will the system generate high levels of noise? |
| ☐ | ☒ | Will the product be subjected to extreme environmental conditions such as fog, humidity, cold, high temperatures ,etc. that could create an unsafe condition? |
| ☐ | ☒ | Is it easy to use the system unsafely? |
| ☐ | ☒ | Are there any other potential hazards not listed above? If yes, please explain on the back of this checklist. |

For any "Y" responses, add a complete description on the reverse side. DO NOT fill in the corrective actions or dates until you meet with the mechanical and electrical technicians.

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