Formula Hybrid Drivetrain Design

By

Zak McFarland
zak.s.mcfarland@gmail.com

Alex Pruitt
aapruitt@gmail.com

William Domhart
williamedomhart@gmail.com

Advisor: J. Tobias – jtobias@calpoly.edu

Mechanical Engineering Department

California Polytechnic State University

San Luis Obispo

June 2012

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Executive Summary

Our senior project group was tasked with designing and building a drivetrain system for the Cal Poly Society of Automotive Engineers (SAE) Formula Hybrid (FHSAE) team. FHSAE gave customer requirements for performance and geometry for the drivetrain and the FHSAE rulebook has guidelines regarding safety requirements. The team chose to compete in the electric category of the 2012 FHSAE competition. After getting feedback from previous car performance and researching different powertrain options, the senior project team arrived at three conceptual ideas. Using a quality function deployment method, the senior project team chose the concept where two motors independently delivered power to the rear wheels through a chain and sprocket system. Additionally, an electronic control system would handle power delivery to each of the motors to insure proper vehicle operation. The main components of the final drivetrain design consists of: two DC electric motors, two chain/sprocket systems, two motor supports, two gear sprocket/brake rotor mounts, two axle stubs, one axle stubs central support, four constant velocity joints (CVJ's), two half shafts, and bearings. The main components of the final drivetrain controls and data acquisition design consists of: 1 real-time controller with digital and analog inputs/outputs, and 2 motor controllers. Additionally, a 3-axis accelerometer, a 3 axis-gyroscope, a steering wheel position sensor, and four wheel speed sensors were purchased with the intentions to implement a traction control and torque vectoring control system. The control algorithms were developed, but motor controller issues prevented actual implementation of these advanced control systems. The culmination of the physical drivetrain components and the control and data components will provide the FHSAE team with a car that met their performance requirements while leaving growth opportunities to implement and expand upon the control algorithms developed.

Chapter 1: Introduction

SAE Formula Hybrid is a collegiate club that competes in the Formula Hybrid competition. Teams of undergraduate and graduate students design, build, and test a performance hybrid vehicle to compete in a series of static and dynamic events. The static events are the business presentation and the engineering design, and the dynamic events are acceleration-electric power, acceleration-unrestricted, autocross, and endurance. For 2012, the competition is divided into two categories, Hybrid vehicles and Electric vehicles. The competition allows students to stretch the boundaries for what is possible for performance hybrid and electric vehicles by exercising their creativity in the design process.

Cal Poly's SAE Formula Hybrid team is sponsoring a senior project for the design and implementation of a drivetrain for their car for the 2012 competition. The winning team from the 2011 competition was able to complete the 75-m acceleration test in 4.5 seconds, where-as Cal Poly's 2011 car completed the same acceleration test in 7.75 seconds. Competing teams are going to bring faster cars to the 2012 competition, and the team has decided that they would like to have the powertrain redesigned to meet higher performance goals. The design of the powertrain will be in parallel with the design of the rest of the car, and entails working with other subsystem teams to complete the car within the time constraints of competition rules. The drivetrain system is to consist of the electro-mechanical and mechanical components, between and including, the motors and the hubs. This could include, but is not limited to, electric motors, internal combustion engines, a method of power transmission (i.e. transmission, chain and sprockets, etc.), differentials, half-shafts/drive-shafts, and hubs.

The goals of the project are to deliver an electric powertrain that aids Cal Poly's SAE Formula Hybrid team in placing competitively and ultimately winning the 2012 Formula Hybrid competition. Performance targets and design criteria are elaborated on in later sections.

Chapter 2: Background

Last year at the Formula Hybrid competition, most of the teams, including Cal Poly, did not get an opportunity to compete in all of the dynamic events due to issues with the technical inspection process. The two events missed by most of the teams were the 75-meter, electric only, acceleration test and the 75-meter, unrestricted, acceleration test. The fastest time for the 75-meter acceleration test was 4.425 seconds for unlimited and 5.717 seconds for electric only. In the autocross event, Cal Poly performed above many people's expectations, and ended up within less than a tenth of a second of teams that had significantly higher budgets to work with. Finally, during Cal Poly's endurance run, the car had an electrical problem due to the rain, and did not finish the event.

When looking at last year's competition, the team noticed a few shortcomings that they would need to address in order to be competitive at the 2012 competition. Two major aspects they were looking to improve were the reliability and performance of the car as a whole. From looking at designs of the cars at the 2011 competition, the team noticed that most of the other cars were a parallel hybrid configuration. Any specific details of competitor's cars are often kept proprietary, and therefore are not available for comparison. The parallel hybrid design usually leads to more horsepower and torque than the Cal Poly series hybrid has produced. The team this year wants to improve in this area to be able to outperform the fastest cars from the 2011 competition.

According to the 2012 Formula Hybrid rulebook, a new class of electric only "hybrids" was formed. This gave Cal Poly an opportunity to start over and build a new car from the ground up with innovative engineering designs that address the shortcomings of the former car. The 2012 SAE Formula Hybrid rulebook can be found at the Formula Hybrid website (http://www.formula-hybrid.org/).

Requirements / Specifications

The overall goal of this project is to design a drivetrain system for the 2011-2012 Formula Hybrid car that will allow the team to be highly competitive at the 2012 competition. The system is expected to meet or exceed the requirements as listed below.

Table 1: Requirements chart for 2012 SAE Formula Hybrid drivetrain design. Budget is not inclusive of the price of motor controllers. Those costs are absorbed by the electrical subsystem team's budget.

Spec. #	Parameter Description	Requirement or Target (units)	Tolerance	Risk	Compliance
1	Weight	100 lb	Max.	M	A,T
2	Drivetrain Efficiency @ Constant Speed	85%	Min.	M	A,T
3	75-m Acceleration	4.5 seconds	Max.	M	A,T
4	Top Speed	55 mph	Min.	M	A,T
5	Endurance Race Time	22 km in 60 min.	Max.	M	A,T
6	22-km Endurance Race Energy Consumption	5.4 kwh	Max.	M	A,T
7	Component Replacement Time	1 hr.	Max.	L	T
8	Budget	$2,500	Max.	M	A
9	Safety: Chain Guard Fastener	1/4" SAE grade 5	Min.	L	I
10	Safety: Chain Drive Guard Thickness	0.105-in steel	Min.	L	I
11	Safety: Chain Drive Guard Width	3 x Chain width	Min.	L	I
12	Safety: Rotating Component Finger Guard	Mesh, 12mm aperture	Max.	L	I

The development of these specifications comes primarily from the requirements from the rule book, and team experience from previous competitions. Performance goals are based off of the top teams for each event as seen in the Benchmark QFD (Appendix A). Comparing the performance and how well customer needs were met by the benchmarks was difficult given the lack of available information about competitors' cars. Teams are protective of the information about their cars in order to protect their designs. This makes it difficult to compare competitors' cars with our proposed designs other than through known published performance data from previous competitions. Other requirements are derived from the team's design direction, and from team members' observations from previous competitions.

The Benchmark QFD shows that the Texas A&M car was the most competitive car based off of our customer requirements, followed by the BYU car. This correlates with these teams performance in the 2011 competition where Texas A&M and BYU took the 1^{st} and 2^{nd} place respectively.

Method of Approach

The chart below is the proposed plan for completion of the senior project. As we make more progress along this path, we will be continuously updating this to reflect our plan. The timeline for completion is also illustrated in our Gantt chart (Appendix).

For the 2011 Fall quarter, our team expects to identify the problem, brainstorm potential solutions, and present viable options to the SAE Formula Hybrid Team. Once approved, our senior project team will proceed with detailed design of the drivetrain which will carry into the beginning of 2012 Winter quarter. Due to the nature of the project and budget constraints, a physical prototype isn't feasible. The design and packaging of the drivetrain will be conducted in SolidWorks until fabrication and assembly takes place. After assembly, testing will be conducted for all aspects of the competition before the competition in May 2012. We will also compare our test data to the competition data of Cal Poly's car and competitors' cars from the 2011 competition.

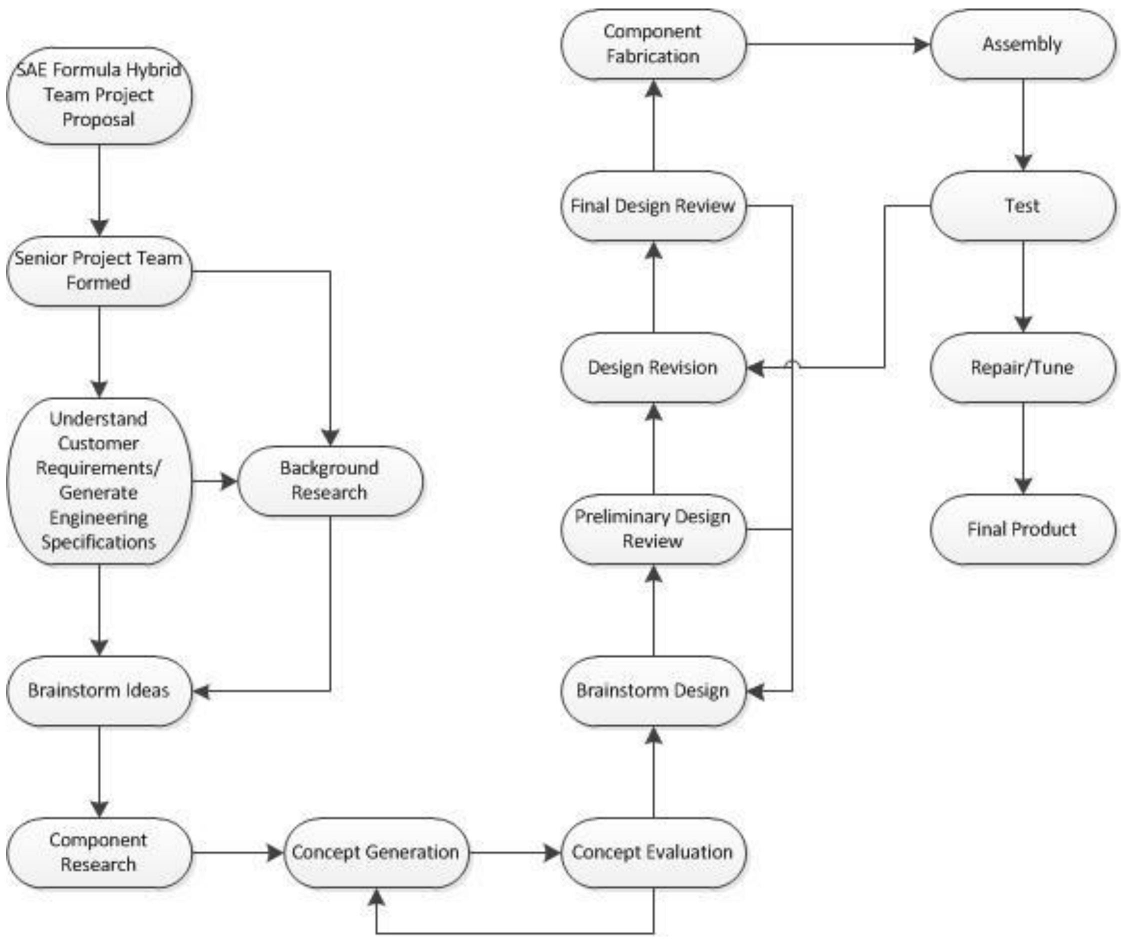


Figure 1: Design Process Flow Chart for 2012 SAE Formula Hybrid drivetrain design.

Chapter 3: Design Development

Conceptual Design 1: Single Motor with Transmission

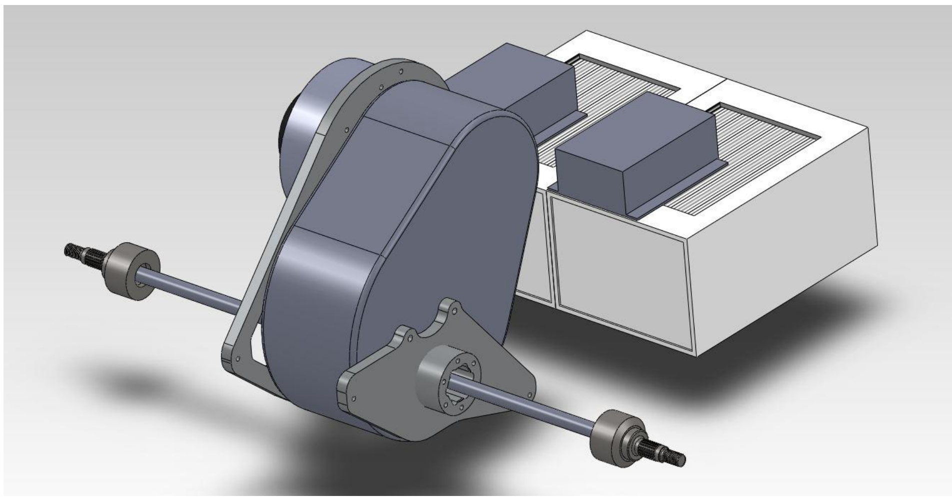


Figure 2: Single Motor with Transmission Conceptual Model (3/4 view).

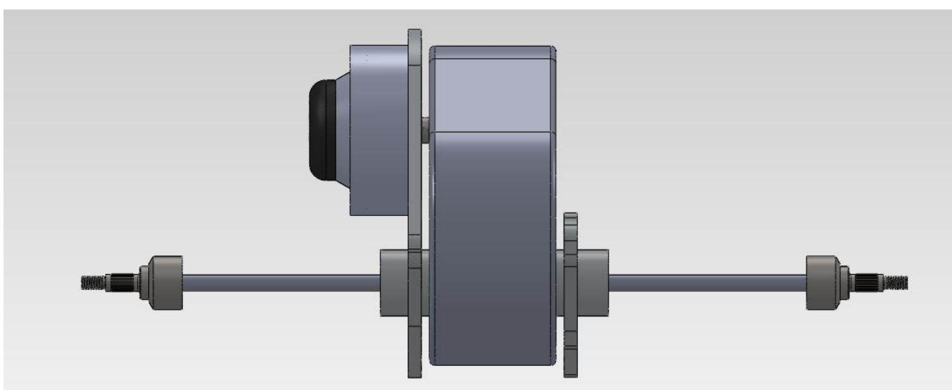


Figure 3: Single Motor with Transmission Conceptual Model (rear view).

The first concept has a drivetrain system consisting of a single DC motor, a transmission, and a limited slip differential driving the rear wheels. Currently there are no electric vehicles using transmissions and this is primarily because of the high amount of torque components in the transmission would see. Therefore, we would have to design our own transmission. We have not included an actual model of a transmission in the Conceptual Design report due to the difficulty and amount of time required to design and draw an accurate model of a transmission. In place of the transmission is a basic box. The transmission would most likely be a 2-speed transmission, and at most, a 3-speed transmission. Until the Formula Hybrid team is able to get batteries, we can't put the motor on a dynamometer to get a torque-speed curve. Therefore, analysis of how many gears and what ratios are optimal can't be worked on.

Conceptual Design 2: Dual Motor with Mechanical Differential

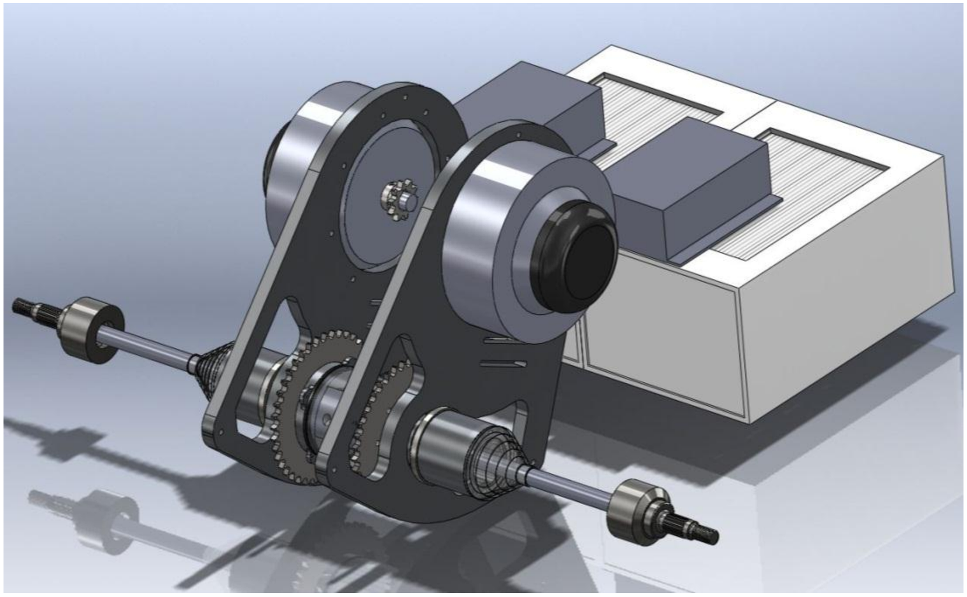


Figure 4: Dual Motors with Different Gearing and Mechanical Differential (3/4 view).

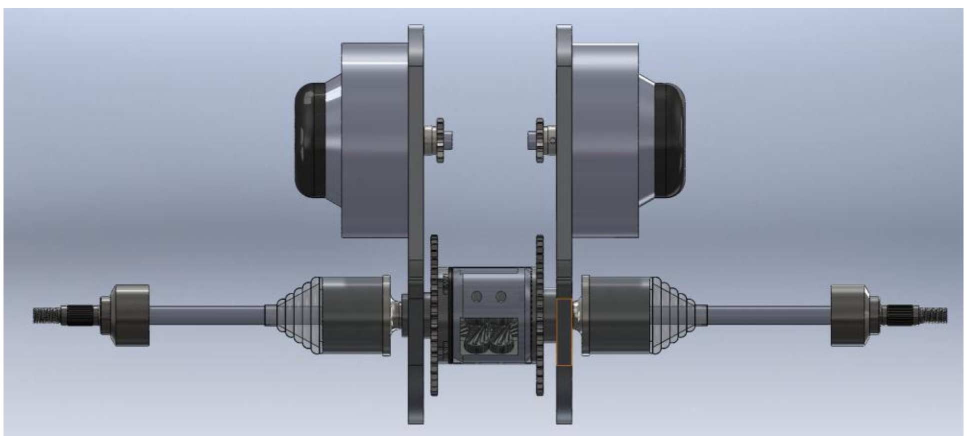


Figure 5: Dual Motors with Different Gearing and Mechanical Differential (rear view).

Note: figures 4/5 show two motors, each with different gearing for the conceptual model. The motors each use a sprocket and chain system to transfer power to the limited slip differential which then transfers power to the drive shafts.

The second concept consists of two DC motors, each connected to a limited slip or Torsen differential using chains and sprockets. The motors will have different sprocket ratios so that both motors are running in different portions of their power bands for a given vehicle speed. There will be a one way bearing or sprag clutch on one of the motors so that it doesn't exceed its maximum rated rpm, causing damage to the motor. This design provides some of the benefits of having two separate gear ratio's in the same design, without the added complexity of a transmission.

Conceptual Design 3: Dual Motor with Electronic Differential

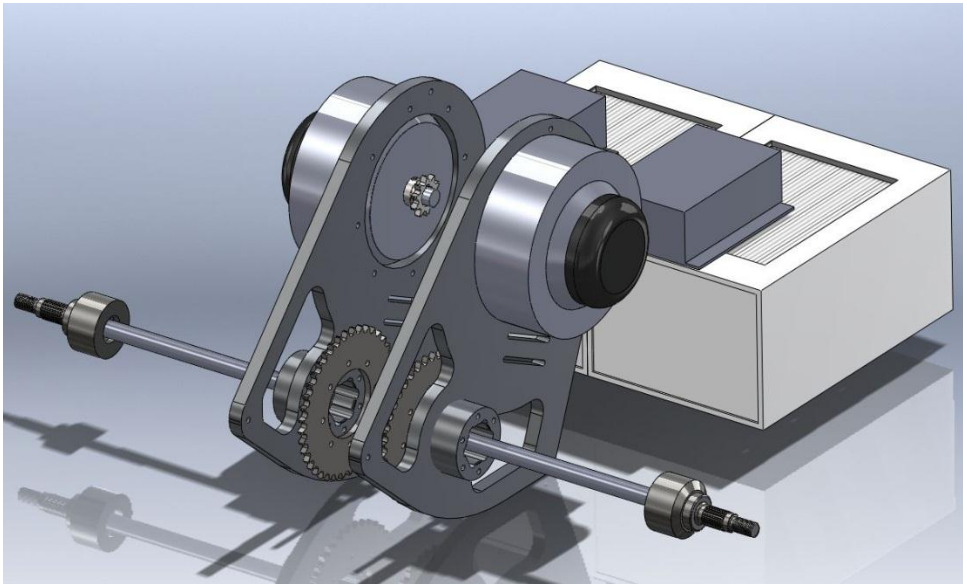


Figure 6: Dual Motors with the Same Gearing and Electronic Differential (3/4 view).

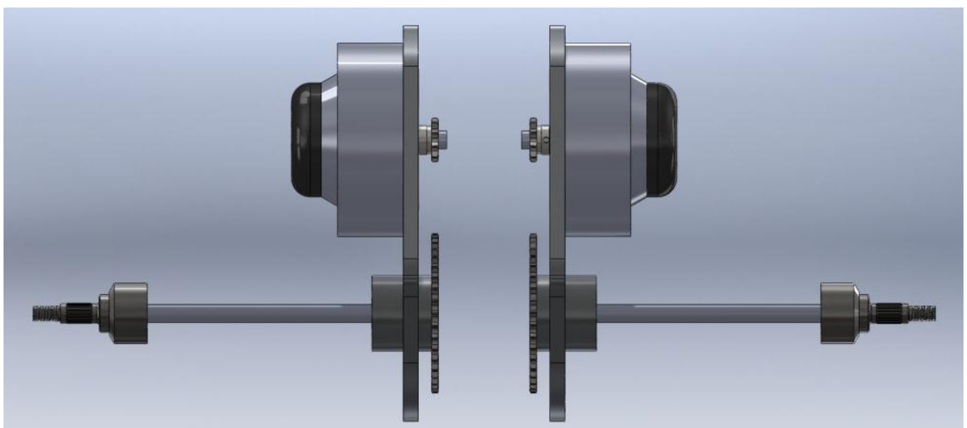


Figure 7: Dual Motors with the Same Gearing and Electronic Differential (rear view).

Note: figures 6/7 show two motors, each driving a respective wheel. The motors each use a sprocket and chain system to transfer power to the respective drive shafts. Power delivery is controlled using a software based differential handle by a cRIO controller (not pictured).

The third conceptual design is a drivetrain system where the two rear wheels are independently powered by their own DC motors. The concept will use an electronic differential with the cRIO controller processing how much power each motor should deliver to its respective tire. This concept will also include implementation of traction control with the cRIO controller also handling the duty of traction control. Outlined below is a proposed traction control system and the necessary components needed to implement this system.

Traction Control for Conceptual Design 3

Traction control was decided as an addition to the Electronic Differential concept because of its ease of implementation in this design and deemed necessary in aiding the transfer of power from the motors to the wheels.

Goals for the traction control system are outlined below:

- Aid in efficient delivery of power to the ground.
- Increase cornering speed of car, therefore lowering lap times.
- Improve driver safety through increased vehicle stability, especially in inclement weather.
- Allow more powerful powertrain to be efficiently implemented.
- Improve powertrain consumption efficiency through reduction of wheel slippage.

The proposed traction control will consist of the following hardware:

- 2 x Kelly KDZ12401 DC motor controllers.
- cRIO 9076 Real Time Controller with digital input/output and analog input/output modules.
- 4 x wheel speed sensors.
- Pedal position sensor (potentiometer).
- 3-axis accelerometer.
- 3-axis gyroscope.
- Steering wheel position sensor.

The basic operation of the traction control system is as follows:

- cRIO controller will compare the average front wheel speed with the wheel speed of each of the rear wheels.
- If the wheel speed of either of the rear wheels is more than 5% faster than the average front wheel speed, the cRIO controller will modulate the respective motor for that wheel to lower its wheel speed to within the threshold.
- In cornering situations, the cRIO will monitor the steering wheel position, 3-axis accelerometer and 3-axis gyroscope and adjust torque bias for the inside and outside drive wheels in order to aide in cornering (yaw control).

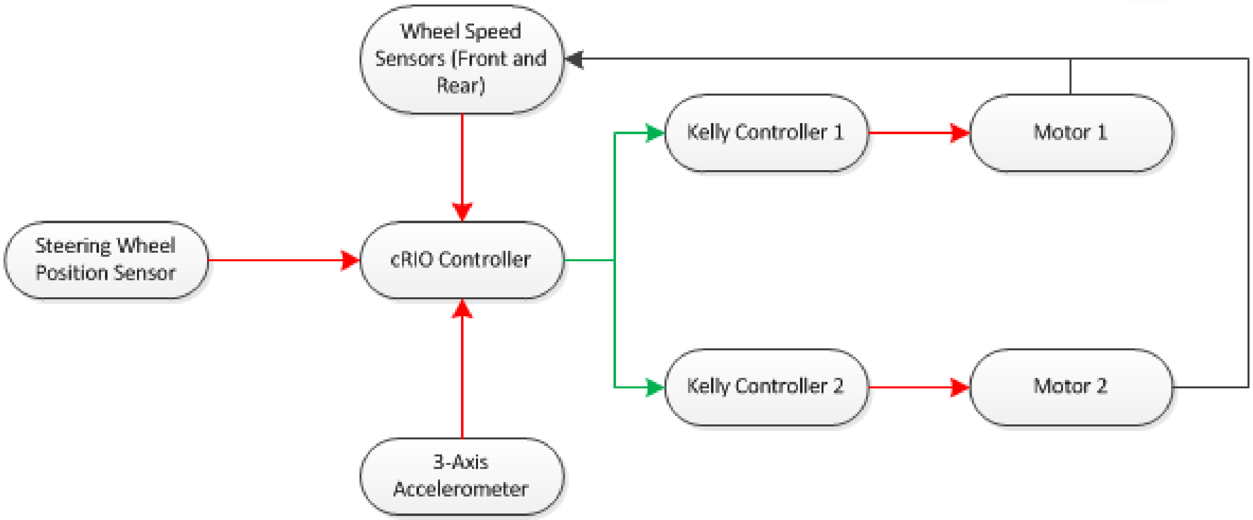


Figure 8: Diagram of Traction Control System.

Note: This is showing basic input/output interactions between cRIO controller and attached peripherals and hardware. Red arrows are for inputs and Green arrows are for outputs.

Conceptual Model Design Comparisons

Drivetrain Design Requirements Comparison

Table 2: Conceptual Design Comparison.

Note: Comparison is using projected weight, projected cost, simulation 75 meter acceleration time, and simulation 22 kilometer endurance race energy consumption.

	2011 Single Motor	**E-Diff 2 Motors**	**Mechanical Diff 2 Motors**	**1 Motor + Transmission**
Weight*	118 lbs	110.5 lbs	135 lbs	128 lbs**
Cost	$900	$3176	$3201	$3462
75m Time***	5.14s	3.84s	3.82s	4.27s (not including shift time)
Energy Consumed*** (in 75m Accl)	232KJ	305KJ	300KJ	275KJ

Drivetrain Decision Pros and Cons

Table 3: Pros and Cons of the three concepts for the 2012 Formula Hybrid drivetrain design.

2 Motors w/ Mech-Diff

Pros

- Allows Different gearing
- Reliable
- Already have most of the parts

Cons

- Less versatile or adjustable (i.e. changing torque bias)
- Harder to Package

2 Motors w/ E-Diff

Pros

- Mechanically Simple
- Adjustable
- Traction Control

Cons

- Heavy dependence on electronics
- Requires time spent to tune

1 Motor w/ Transmission

Pros

- Novel (Design Points)
- Better Power Delivery

Cons

- Added cost to buy or design/manufacture
- Reduced drivetrain efficiency
- More parts to break

Drivetrain Model Design Decision

Once our three leading concepts had been determined, we created two separate simulations to determine the amount of time the conceptual car would take to go 75m and the energy it would use to do so. Based on the comparisons above and the results from our Conceptual Design QFD in Appendix A, we have chosen to design and build an electronic differential based drivetrain. This option has the lightest weight of our three viable options, and will even be lighter than the 2011 car's drivetrain due to the loss of the heavy controller and mechanical differential. Furthermore, the system will be very simple mechanically while still providing the desired level of performance. It can also provide smaller packaging and easier access for maintenance and modification.

The E-diff system will give us the option of creating a more robust traction control system because it allows for each rear wheel to be independently driven. It can accommodate fully tunable traction control, in which only the wheel that is slipping is adjusted or corrected. Additionally, when cornering, the system can adjust the torque bias between the drive wheels in order to aid in handling. Both traction control and yaw control will help improve straight line and cornering performance of the car, and help achieve the overall design requirements.

The idea of building a transmission for a single motor vehicle was originally appealing. However through our research, and through observing Cal Poly's SAE Mini Baja team design and build a transmission for their 2012 car, we came to the conclusion that it would be difficult to build a transmission for less than the cost of a new motor. Also, the time and work involved could be better spent on the rest of the drivetrain instead of only one component of it. Our simulation results show that while adding a transmission would be a substantial improvement over the 2011 car's performance, it would not be able to match the performance of a two motor system, as well as being slower than the winning cars from 2011.

Despite the seemingly simple approach of a differently geared system, our research raised concerns about over-revving the electric motors, meaning we would require a reliable method of disengaging them. While disengaging one of the motors from the primary drivetrain would not be difficult, and could be achieved with a centrifugal clutch, one way bearing or other device, we discovered that the control logic governing the motor behavior during disengaging and reengaging would need to be quite complex to ensure predictable handling. This would potentially need to take into account factors such as wheel position, lateral acceleration, and torque vectoring.

As can be seen from our comparison tables, our simulations show that a two motor system, with each motor geared differently will produce the fastest 75m acceleration time while using slightly less energy than two motors with the same gearing. However the

simulation times do not account for the change in rotating mass due to the addition of the mechanical differential. Given how close the 75m simulation times for these two options are we feel the simulation does not allow us to make an accurate estimation to whether the E-Diff or differently geared system will have better acceleration. When looking at the Conceptual Design QFD (Appendix A), the electronic differential design scored highest among all of our potential concepts as well as meeting or exceeding all of the customer's requirements.

Once the drivetrain system was chosen, we began the design of the actual components of the drivetrain. This process is still under way but will feature machined aluminum mounting plates for motors and sprocket carrier assemblies, and two pairs of steel sprockets for #40 chains.

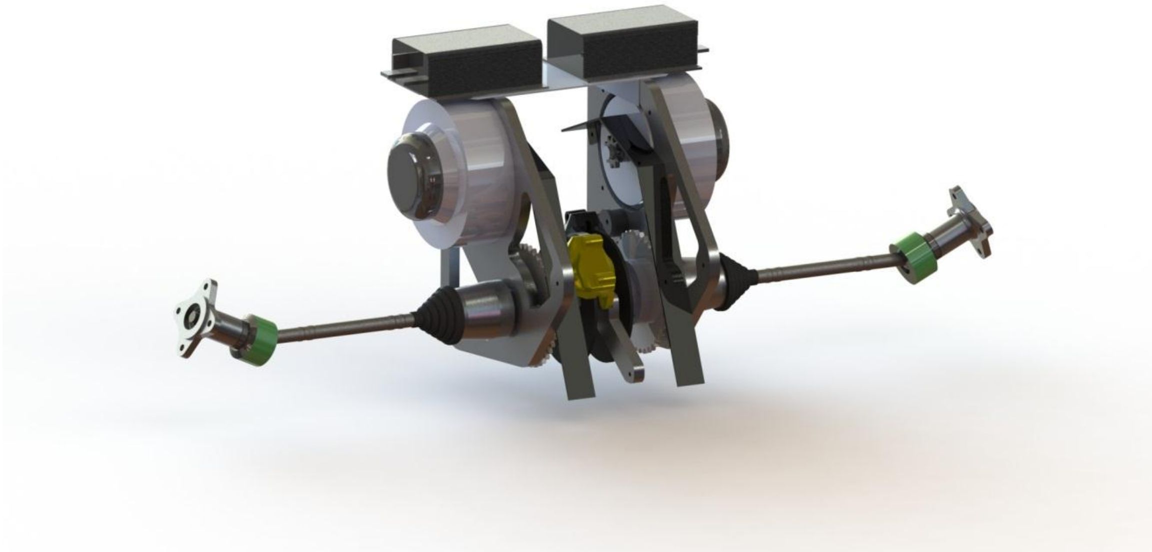


Figure 9: Final Drivetrain Assembly Design (3/4 view).

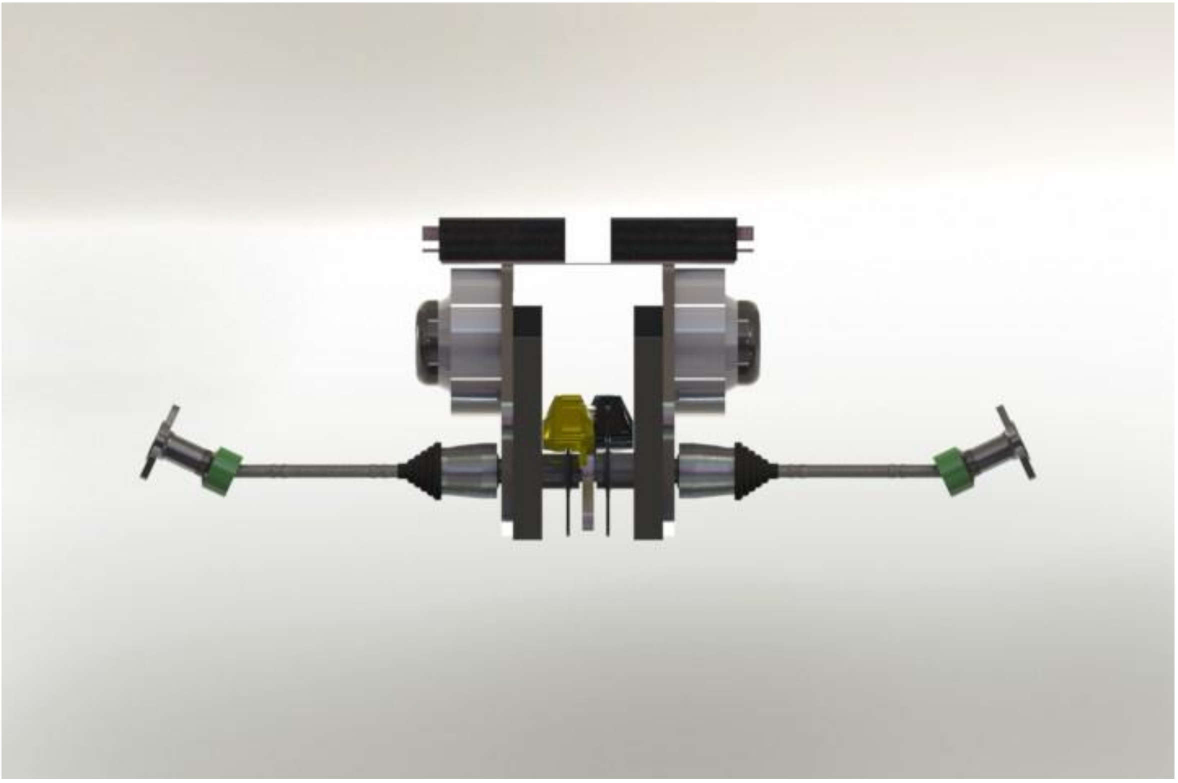


Figure 10: Final Drivetrain Assembly Design (rear view).


Figure 11: Final Drivetrain Assembly Design (partially exploded view).

Chapter 4: Description of the Final Design

Once the decision was made to proceed with the electronic differential, we began designing the final parts and layout. This includes placement of the two motors, chains, sprockets, and brake and rotating assemblies.

The overall layout, as shown in Figure 12 below, is similar to the conceptual design we had previously. Some changes include locating the brakes inboard, as well as revising the uprights to correct the factor of safety to acceptable levels. For all of the parts, we decided to go with aluminum to save weight. The uprights are made of 6061, while the shafts are 7075, and the sprocket/brake mount is 6061.

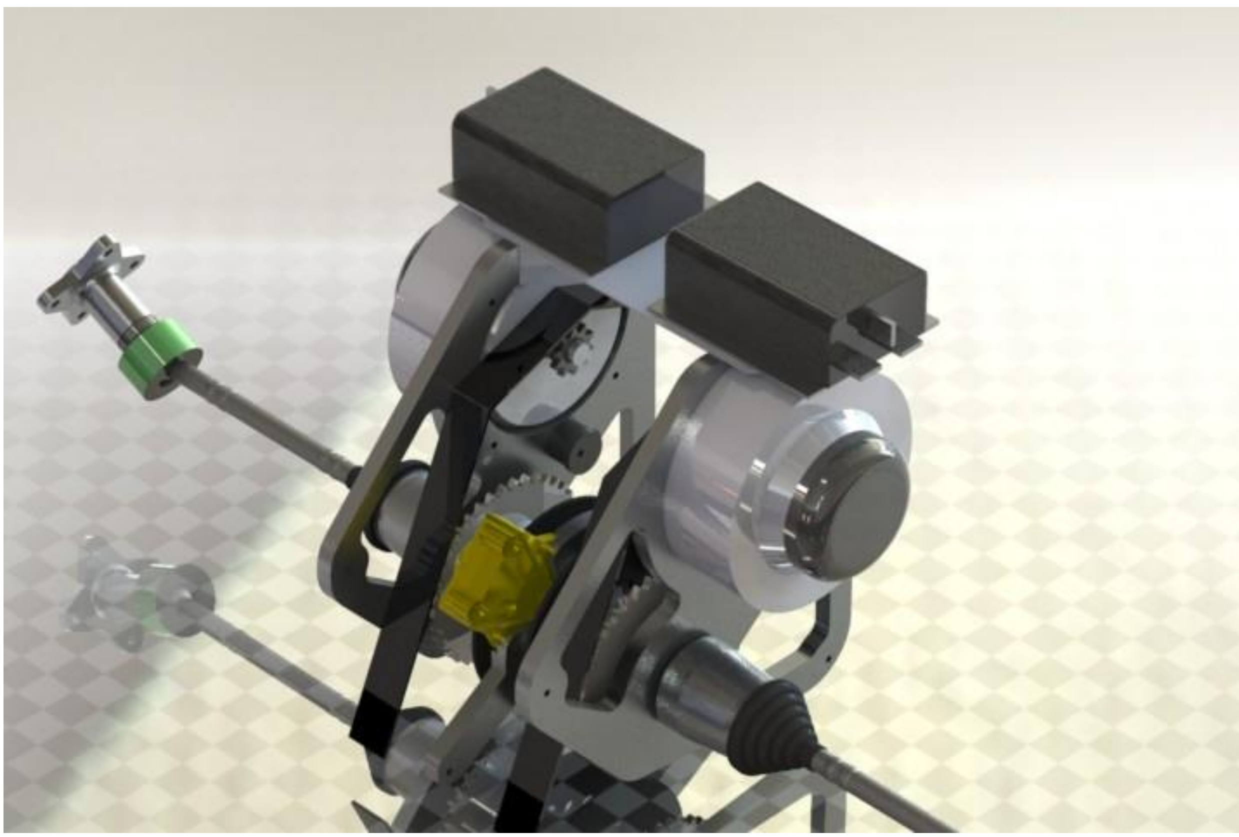


Figure 12: Closer View of Final Drivetrain Assembly (3/4 view).

One of the greatest challenges in the final design iterations was insuring that the final product could not only be assembled in the vehicle, but also quickly disassembled for repair, inspection, and modification. The design above, with the inboard brake rotors allows for the CV housings and axle stubs to be pulled out through the chassis.   This then allows the sprocket and

brake carriage to drop out the bottom of the car, avoiding any disassembly or removal of the brake rotor mounts or the caliper.

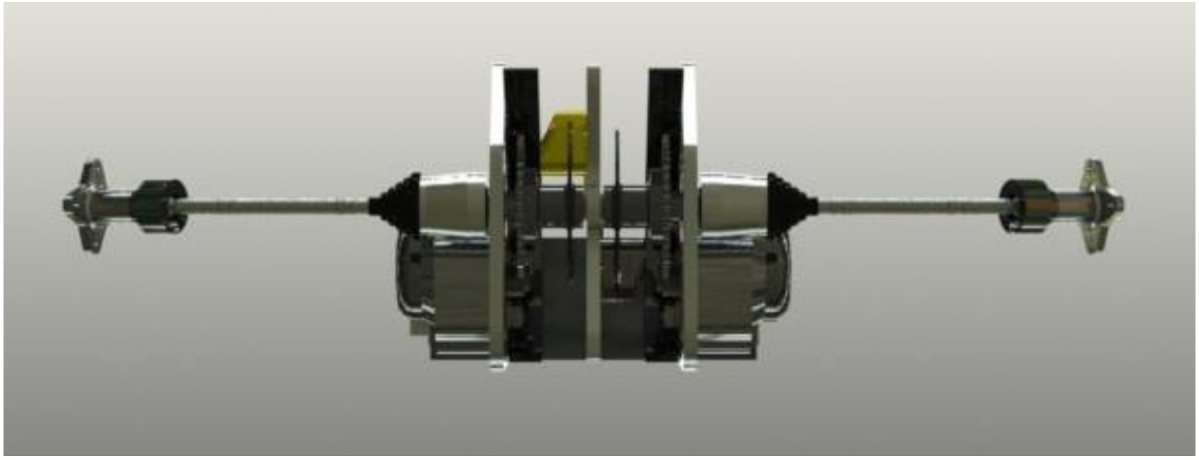


Figure 13: Final Drivetrain Assembly (bottom view)

From the analysis done, we achieved a factor of safety of 1.2 for the uprights, as well as a factor of safety for fatigue and failure of 1.6 for the shafts. The details of these can be found in Appendix F.

This design, because of its weight goals, will be made almost exclusively from aluminum, which results in much more expensive raw materials. The plates being the most expensive items, followed by the material for the axle stubs due to its high quality (7075) even our analysis shows that 6061 aluminum would also withstand the fatigue and loading on the axel, the axle stub is a very critical part which most teams make from hardened steel and a higher factor of safety was desired.

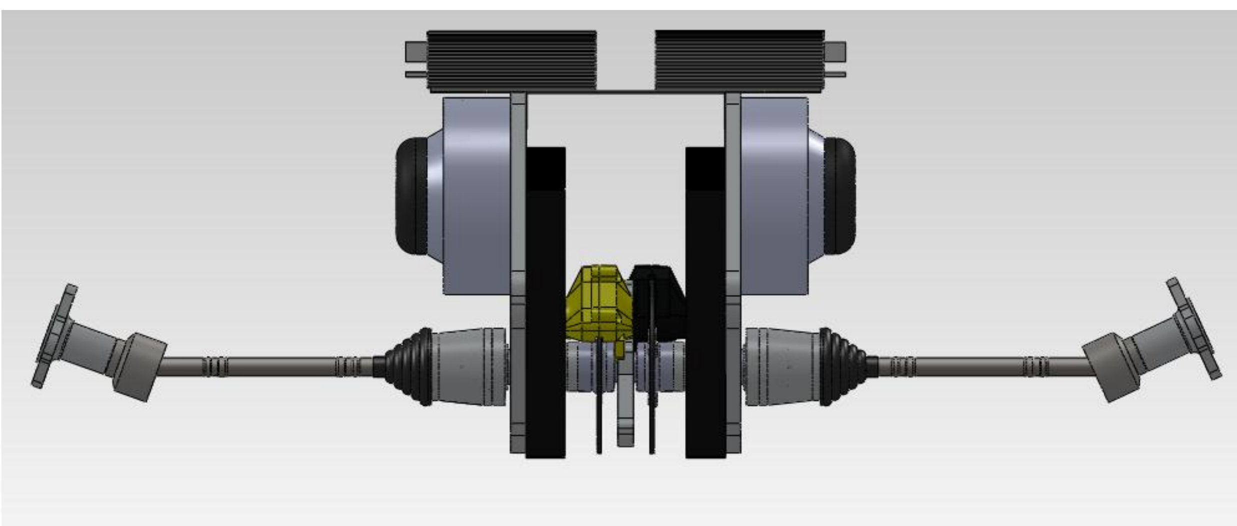


Figure 14. Final Drivetrain Layout (rear view)

Chapter 5: Manufacturing

The Final Design required a significant amount of newly manufactured parts for the final product. The following is list of all newly manufactured parts.

- Motor Mounting Plates
- Central Support
- Axle Stubs
- Brake Carriers
- Sprocket Carriers
- Constant Velocity Joint (CVJ) plates
- Constant Velocity Joint housings
- Half Shafts
- Brake Rotors

Figure 15. Finished and disassembled drivetrain components prior to installation on the car.

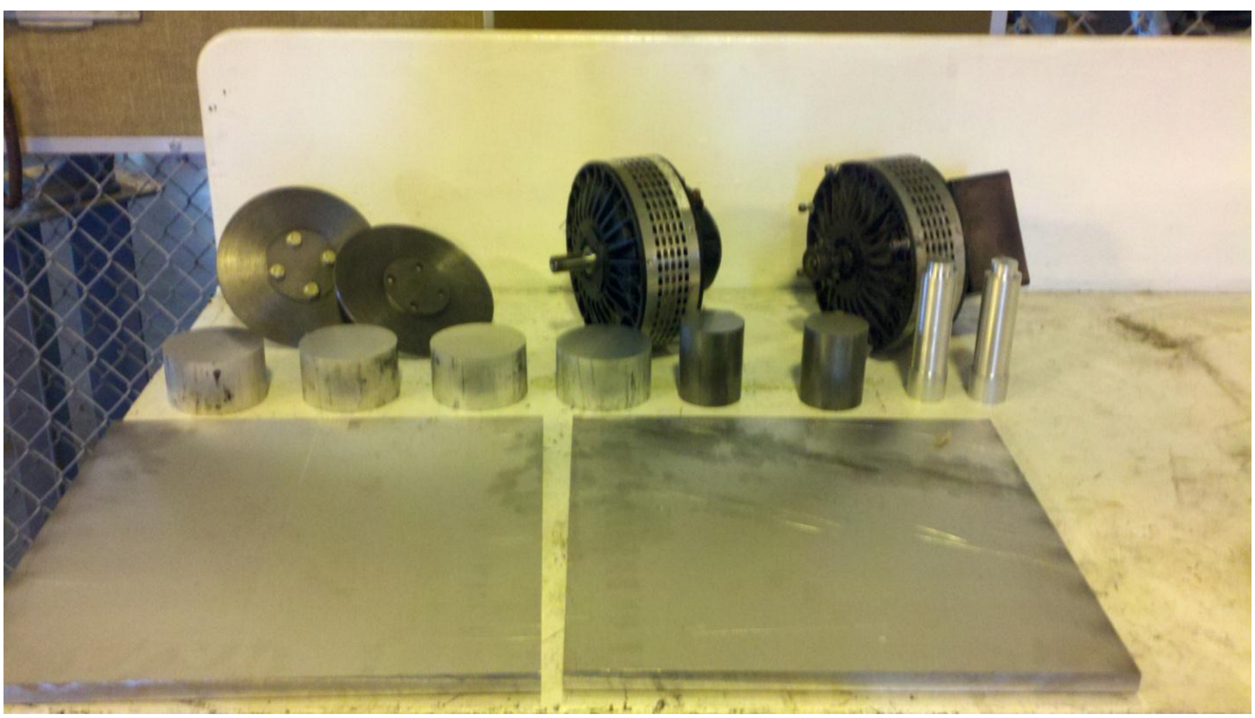


Figure 16. Raw materials prior to machining parts for the drivetrain subsystem.

Motor Mounts

The two motor mounts which form the backbone of the design are an evolution of the mounting system used on the 2011 vehicle, but also offer a number of substantial improvements over the old design. The new mounts each shed 2 pounds from the old mounting plate, while making the entire system more compact, allowing the entire assembly to fit within the rear suspension bay of the chassis. This allowed the team to further improve packaging in the final product.

The raw materials were purchased from SpeedyMetals.com who generously provided them to us at cost. We selected to machine the mounts from 6061 T6 5/8" think aluminum plate as a compromise between cost and weight.

The final parts were CNC'd on the Haas VF2 in the Mustang 60 machine shop, as it was the only machine which offered the necessary travel to machine all outside edges of the without removing it from the machine. This was an important factor because the outside geometry of the part necessary to fit in the chassis is very difficult to indicate and would otherwise require further tooling to be machined just to hold the part.

Figure 17. Left Motor mount being machined on the Haas VF-2 vertical CNC mill.

In order to restrain the plate the while machining the outer edges, the interior features were cut first and then toe clamps were used to hold the part down while the outer edges were cut.

Central Support

The central support was a difficult part to manufacture although it was machined in much the same way as the above motor mounts, but it had be machined on opposite faces, without a designed edge to indicate the part from. In order to compensate for this fact a 90 degree corner was machined into the black piece of metal in a place that could be removed in the final process. This corner allowed the two bearing surfaces to be aligned despite being on opposite sides of the part.

Figure 18. Above: Center support being machined on the Haas VF-2 vertical CNC mill.

Below: SolidWorks model of center support (isometric view).

Axle Stubs

The Axle Stubs proved to be one of the most challenging parts of the drive train. The primary question in designing them was how to transfer the motor and braking torque from the inside the motor mounting plates to the wheel side, while still being able to assemble and work on the drivetrain without dismantling the entire assembly. Furthermore, this had to be accomplished without exceeding our budget constraints. After researching many options, we settled on using a three sided geometric spline. This type of spline has been used effectively by the Formula Hybrid team before, and detailed analysis and standardized geometry's were available in the 27th edition of the *Machinery's Handbook*. Involute splines were considered, but we creating the female spline would have required purchasing premade parts or outsourcing the machining process, both of which were very expensive options for the dimensions required. Bolted Joints were also considered but would not have fit within the tight spacing requirements. The Geometric Spline option allowed us make both the male and female components in house on standard CNC Mills and lathes. The Shafts were made from 2024 T3 Aluminum, for its superior strength and machinability.

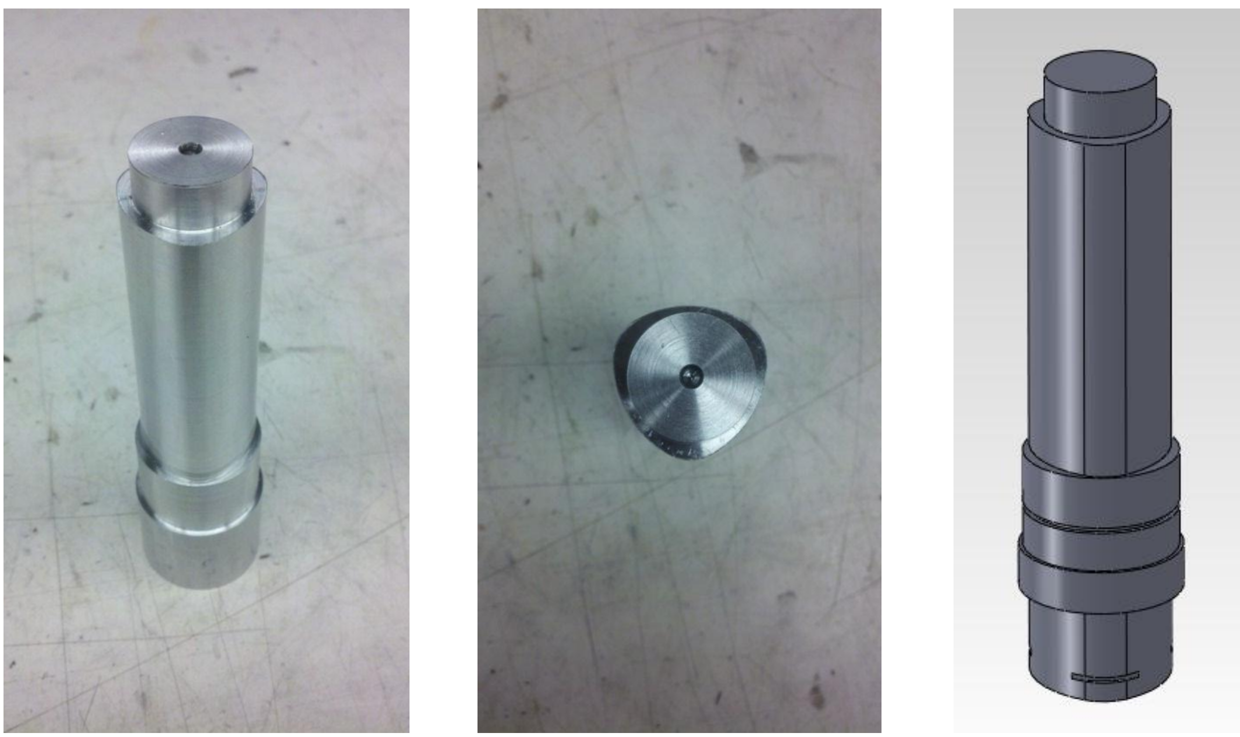


Figure 19. Left: Partially CNC machined axle stub (front-top view).
Middle: Top view of partially CNC machined axle stub.
Top: SolidWorks model of axle stub.

These Shafts were first manually turned to create the bearing surfaces then CNC'd upright in a mill using soft jaws specifically cut to hold the shaft. Machining the splines took multiple passes

at low feed rates due to the length of the shaft and tool becoming subject to bending. Once both spline portions were cut, the axle stubs were turned in the lathe again to cut the C clip grooves.

Sprocket Carriers

These parts were fairly straight forward in both design and manufacturing once the axle stubs had been finalized. They are made from 6061 T6 Aluminum and allow the Sprockets to transfer torque to the shaft. These have a hub and flange portion. The flange portion seats the sprocket and has the bolt holes for mounting, furthermore this section supports the bulk of the torsional load and hoop stresses induced by the splined axle stubs.

The second portion is the hub, which supports some of the hoop and torsional loads but primarily serves to keep the sprocket aligned with the motor sprocket and prevent axial play in the system.

The sprocket carriers were machined vertically in the CNC mill, being held in the large 3 jaw rotary vice. This vice allowed the sprocket carriers and brake carriers below to be held securely in the mill. Once the female spline, hub, and bolt holes were cut, the parts fit tested using the axle stubs which had now been completed, and adjustments to be made without removing the work piece.

Brake Carriers

The Brake Carriers are identical to the sprocket carriers in their construction and merely have different dimensions. The flange of the brake carrier has four half circles milled out of the edge, 90 degrees apart. These serve as the mating point between the carrier and the brake button. The outside of the flange is a mating surface for the brake rotor itself.

The manufacturing process was also identical, merely with different dimensions.

CV housings and CV plates

The CVJ housings and plate are bolted together to contain and transmit torque to CVJ on the half shaft. The CV plate is 6061 Aluminum that has the female polygonal spline CNC'ed into it and contains bolt holes to mount the CV housing. The CV housing is 7075 Aluminum, which provides improved wear and durability compared to the 6061 housings used last year. They are still expected to wear out but given the short life of the vehicle, the wear that occurs is not enough to impact performance and the design allows for weight and cost savings over the more typical steel or steel insert type designs.

The Housings were machined manually on a mill using a specially ordered 13.5 mm ream to ensure a high quality bearing surface, and mounting holes were drilled and tapped using a bottoming 1/4-20 tap. With the milling processes completed the insides were turned and bored on the lathe to the correct inside diameter, and finally an outside profile and boot seat were added.

Half-Shafts

Our original designs had called for the use of production ATV half shafts that the formula hybrid team has been employing as part of the camber system for some time. However due to suspension changes, it was determined that between the new drivetrain design and several changes in suspension geometry the old half shafts would be too long.

Figure 20. Half shafts air cooling in furnace after heat treatment at 1600 °F for one hour.

Luckily, the tools and materials were available to create new half shafts, while still using the factory made CVJ's and without redesigning and remaking our CVJ housings. The factory CVJ uses a 17 tooth spline on a .75in diameter; fortunately the SAE Baja team uses the same spline and allowed us to use their spline cutter. We obtained .75in round stock of 4130 annealed steel, and used the fine toothed horizontal ban saw to cut two lengths to 11.75 in, 1.25in shorter than the original 13in lengths.

Once the stock had been cut to length, we used the 4th axis in the Haas TM1 to cut, make several test splines out of scrape aluminum to insure the fit would be precise. Once the CNC was dialed in the steel splines were cut. The shafts were then taken to the manual lathe to cut snap ring and locking ring grooves into the splined section using a specially ground parting tool. Finally the completed pieces were coated in anti-scaling agent and normalized in the furnace at 1600 ®F for 1hour and allowed to air cool.

Brake Rotors

The brake rotors were technically under the responsibility of a member of the FHSAE team, but due to their need to be integrated with the drivetrain their physical design and manufacturing were part of our responsibility. The team specified that the rotors were to be ductile iron, solid, between .16 and .19 in thick, and to be 7.12in in outside diameter. They also specified that they should be mounted using a floating system.

Using this knowledge we designed rotors to mate with rest of drivetrain via four brake buttons. This would allow the rotor to have a small amount of axial play to insure maximum braking. The inside surface would mate to the 3in outside diameter of the brake carrier on either side of the buttons. The rest of the inside area was offset from the carrier to reduce the weight and ease assembly.

Manufacturing the brake rotors was a very involved process. Starting with the 8in diameter .25+ inch thick blanks as purchased by the team, four evenly spaced holes were drilled in the center and used to mount the disk to a special tool holder so that it could be turned in the lathe. The blanks were then turned to size and had their outer portion faced to a .2in thickness. Once this was done the disks were toe clamped down in the CNC mill on top of a piece of Lexan. This allowed the center portion of the disk to be cut out to its final shape.

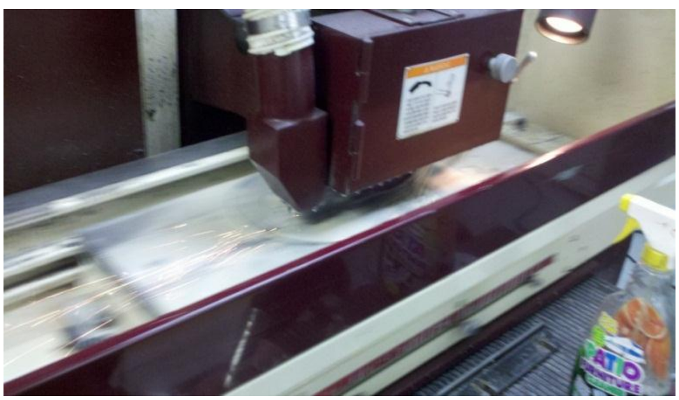


Figure 21. Top: Brake rotors being CNC ground flat.
Bottom: The left rotor has been machined and ground flat and the right rotor has just been heat treated and is ready to get machined and ground flat.

Once the disks were to the correct shape they were stress relieved in the furnace. To do this the disks were clamped between two 7/8 in thick steel plates and bolted down to prevent any warping from machining induced stresses being relived. They were then placed in the furnace, two at a time, for 6 hours at 450®F, and allowed to cool overnight.

Finally once the heat treating process was completed, they were taken to ZBE Inc. in Carpinteria, CA, where we were allowed to use their CNC surface grinder to make the finishing passes on the rotors, ensuring that they were flat, smooth and to the correct thickness.

Chapter 6: Design Verification Plan (Testing)

An extensive design verification plan was developed with the intention of benchmarking the car in preparation for the 2012 FHSAE competition at the beginning of May 2012. Motor controllers issues denied our team the opportunity to complete the outlined testing procedures for dynamic performance of the car and drivetrain. However, we were able to weigh our drivetrain before installing it. The total weight came out to 86 lb, which is significantly less than our goal of 100 lb. Outlined below are the summaries of the tests we proposed. Appendix C contains our detailed design verification plan and report with results.

Component Testing & Evaluation:

Drivetrain Component Fitment

The drivetrain components need to be measured to ensure that they were machined properly before assembly can occur. This will be done on a part by part basis and will require calipers to measure critical dimensions. Additionally, mated parts will be needed to ensure proper mating between these parts.

Drivetrain Weight

The design requirement for drivetrain weight requires that the drivetrain weigh less than 100 pounds. To measure this, all parts, once they are machined, will be weighed individually to within 1 lb and the summed to determine the overall weight of the drivetrain. Drivetrain weight will consist of the weight of the motors, motor supports, central support, central support bearings, pinion sprockets, gear sprockets, chains, gear sprocket/brake disc carriers, axle stubs, CV joints/housings, bearings, and drive wheel hubs.

Motor & Controller Operation Bench-top Testing

Before wiring the Motor and Controller within the car, it is important to test the operation of the motor, controller and battery assembly to ensure that they are all working. This is easier to accomplish when this assembly is outside of the car and provides a blueprint for proper wiring and assembly within the car.

Sensor Operation

Before testing of the traction control can be done, the individual sensors should be tested to ensure that the sensors are outputting correctly. Sensors to be tested are the throttle potentiometer, wheel speed sensors, steering wheel position sensor, 3-axis accelerometer, and yaw gyroscope.

Subsystem Assembly:

Mount Drivetrain Assembly on Car:

Drivetrain components will be assembled on the car after components have been tested for proper fitment and operation.

Wiring for Motors, cRIO Controller, and Kelly Motor Controllers

Wire motors to motor controllers, wire motor controllers to cRIO controller, and wire sensors and inputs to cRIO controller.

Traction Control Sensor Assembly on Car:

Traction Control sensors will be mounted and wired to their appropriate locations after being tested for correct signal outputs.

Subsystem Testing & Evaluation:

75-meter Acceleration Time

The design requirement for 75-meter acceleration time requires that the car accelerate from rest a distance of 75 meters in no more than 5 seconds with the goal of achieving a 4-second acceleration time. This will be tested by measuring a distance of 75-meters in a parking lot, securing the facility for safety measures, and performing a minimum of three tests to confirm consistent 75-meter acceleration results. A person will at the 75-meter distance mark will record the time required for the front of the car to cross the 75-meter distance mark.

Top Speed

The design requirement for the maximum speed of the car requires that the car reach a maximum speed of 65 mph. This will be tested by securing a 100-meter acceleration distance with a 50-meter braking zone to allow the car to have ample distance to reach maximum speed and safely come to rest. The speed of the car will be measured via a GPS application, Android-Speedometer, an application for smartphones running the Google Android operating system. The application will be calibrated versus the speedometer of the 3 senior project group members' automobiles' speedometers. Afterwards, three trial runs will be used to determine the maximum speed of the car.

Energy Consumption and Race Time (22-kilometer & 60-minute Endurance Race)

The design requirement for the energy consumption of the car requires that the car consume no more than 5.4 kWh of energy or the energy storage capacity of the batteries, whichever is less. This will be tested by developing a test track similar to that of the competition track and data-logging the energy consumption over the course of approximately 5-kilometer of distance traveled. This data will then be extrapolated over a distance of 22-kilometers to ensure that the vehicle is capable of meeting the design criteria. Additionally, lap times will be recorded and extrapolated to ensure that the car will finish the endurance race in the allotted time (60 minutes).

Chapter 7: Project Management Plan

The roles of the team members are as follows:

Alex Pruitt – Testing Manager, Formula Hybrid Technical Director

Zak McFarland – Project Manager, Formula Hybrid Team Lead

William Domhart – Content & Media Manager/Traction Control/DAQ Manager

Apart from the individual tasks listed above, each member will be responsible for doing background research, brainstorming ideas and solutions to problems that arise, participating in design and analysis of components, participate in fabrication and manufacturing the final product, and testing once the project is complete. Each member is expected to fully document their individual progress and ideation, as well as the group's progress towards the goal. A proposed timeline for completion dates can be found in our Gantt chart (Appendix).

Chapter 8: Conclusions and Recommendations

The drivetrain subsystem for the 2012 FHSAE car was a general success. Despite not being able to verify most of the performance parameters outlined in the DVP&R, the drivetrain exceeded the expectations of the FHSAE team and performed well when not hampered by motor controller issues. Motor controller issues have been pervasive throughout the entire testing period and New Hampshire competition. In retrospect, the drivetrain budget limited the ability to purchase more reliable motor controllers and was an oversight of the FHSAE and senior project team. However, the mechanical components of the drivetrain are thoughtfully designed for longevity, ease of replacement of worn parts, and the weight of the drivetrain system was reduced from last year's design while doubling the power output. Most importantly, the drivetrain is a stable platform for the implementation of traction control and torque vectoring systems, and mechanically, will undergo little, if any, change in next year's car. The team is very confident with the power output, despite not having quantified results of the acceleration, energy consumption, or endurance race time. The team will pursue attaining these numbers after motor controller issues have been taken care of and a suitable venue for testing has been booked.

While we are overall satisfied with the development of the drivetrain, we would like to see the following areas addressed by the future drivetrain team:

- Purchase and implementation of better quality motor controllers.
- Design of sensor mounting for traction control/torque vectoring sensors.
- Implementation and further development of the traction control/torque vectoring algorithms developed.
- Race track testing to quantify overall vehicle performance and drivetrain performance.
- Consideration of adopting an idler-sprocket -style chain tensioner set up.

References

Machinery's Handbook 28th edition. Erik Oberg, Franklin D. Jones, Holbrook L. Horton, Henry H. Ryffel. Industrial Press. 2008. New York, New York.

Society of Automotive Engineers: Formula Hybrid Rulebook 2012. 2011. http://formula-hybrid.org/rules.php

Shigley's Mechanical Engineering Design 9th Edition. Richard G. Budynas, Keith Nisbett. McGraw Hill. 2008. New York, New York.

Appendices

Appendix A- Quality Function Deployments

Benchmark Competitors QFD

Note: QFD for SAE Formula Hybrid Drivetrain Design with benchmarking from 2011 competitors. Some comparisons were unable to be made because competing teams' car information is typically proprietary. This is denoted by "??." Additionally, until the Critical Design Review (CDR) is complete, the budget for the 2012 car will not be known. Blocks highlighted yellow under benchmarks indicates the benchmark that best suits the corresponding customer.

SAE Formula Hybrid Drivetrain Design

Engineering Requirements (HOWS) / Benchmarks

Customer (Step #1) Requirements (Whats) — Customer Requirements (Step #2)	FHSAE Team/Driver	FHSAE Judges	FHSAE Sponsors	Maximum Speed (mph)	Maximum Acceleration (g's)	Drivetrain Weight (lbf)	Const. Speed Drivetrain Efficiency (% eff.)	Complete 22-km endurance race (minutes)	Energy Consumption in 22-km endurance race (kWh)	Component replacement time (minutes)	Drivetrain Budget ($)	Steel Chain Guard Thickness (in)	Chain Guard Fastener Specification (SAE Grade 5)	Rotating Machinary Maximum Aperturer Size (in)	Component Geometry and fit (inches)	2011 Cal Poly FHSAE Car	2011 Texas A&M FHSAE Car	2011 BYU FHSAE Car
Performance																		
High Top Speed	12	10	15	9	3	1		3								2	5	4
Fast Acceleration	15	10	15	3	9	1		3								3	5	4
Lightweight	11	10	12		3	9		3	3						1	4	5	3
Energy Efficient	9	10	12			3	3	3	9							4	3	4
Long Range Capable	11	10	12			3	3	9	9							3	3	4
Power Deliverability	9	10	5				9	1								3	4	4
Human Factors																		
Easy to maintain	9	10	4							9						4	3	3
Safe	8	10	15									9	9	9		4	4	4
Interface with Car																		
Compact	8	10	5			1							3		9	3	3	3
Other																		
Cheap	8	10	5								9					5	2	3
SUM	100	100	100															
FHSAE Team Driver Total																341	385	364
FHSAE Judge Total																350	370	360
FHSAE Sponsors Total																338	399	374
Units				mph	g's	lbf	%	min	kWh	min	$	in	in	in	in			
Targets				65	0.48	100	85	60	5.4	60	2500	0.105	0.25	0.5	??			
2011 Cal Poly FHSAE Car				55	0.24	60	??	35.4	??	??	400	0.105	0.25	0.5	??			
2011 Texas A&M FHSAE Car				??	0.4	??	??	DNF	??	??	??	0.105	0.25	0.5	??			
2011 BYU FHSAE Car				??	??	??	??	31.25	??	??	??	0.105	0.25	0.5	??			

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

Figure 22: Benchmark Competitors QFD.

Conceptual Design QFD

Note: QFD for SAE Formula Hybrid Drivetrain Design with benchmarking from 2012 Design Concepts. Some comparisons were unable to be made because of unknown information. This is denoted by "??." Additionally, until the Critical Design Review (CDR) is complete, the budget for the 2012 car will not be known. Blocks highlighted yellow under benchmarks indicates the benchmark that best suits the corresponding customer.

SAE Formula Hybrid Drivetrain Design

Customer (Step #1) Requirements (Whats) / Customer Requirements (Step #2)	FHSAE Team/Driver	FHSAE Judges	FHSAE Sponsors	Maximum Speed (mph)	Maximum Acceleration (g's)	Drivetrain Weight (lbf)	Const. Speed Drivetrain Efficiency (% eff.)	Complete 22-km endurance race (minutes)	Energy Consumption in 22-km endurance race (kWh)	Component replacement time (minutes)	Drivetrain Budget ($)	Steel Chain Guard Thickness (in)	Chain Guard Fastener Specification (SAE Grade 5)	Rotating Machinary Maximum Aperturer Size (in)	Component Geometry and fit (inches)	Concepts: 2 Motors w/ Different Gearing & Mechanical Differential	Concepts: 2 Motors w/ Electronic Differential	Concepts: 1 Motor w/ Transmission
Performance																		
High Top Speed	12	10	15	9	3	1		3								5	5	5
Fast Acceleration	15	10	15	3	9	1		3								5	5	3
Lightweight	11	10	12		3	9		3	3						1	3	5	4
Energy Efficient	9	10	12			3	3	3	9							4	4	5
Long Range Capable	11	10	12			3	3	9	9							4	4	5
Power Deliverability	9	10	5				9	1								4	4	4
Human Factors																		
Easy to maintain	9	10	4							9						3	5	3
Safe	8	10	15									9	9	9		4	5	4
Interface with Car																		
Compact	8	10	5			1							3		9	3	5	3
Other																		
Cheap	8	10	5								9					5	5	4
SUM	100	100	100															
FHSAE Team Driver Total																407	471	400
FHSAE Judge Total																400	470	400
FHSAE Sponsors Total																414	471	415
Units				mph	g's	lbf	%	min	kWh	min	$	in	in	in	in			
Targets				65	0.96	100	85	60	5.4	60	2500	0.105	0.25	0.5	??			
2 Motors w/ Different Gearing (Mech. Diff)				??	1.05	135	??	??	??	??	3201	0.105	0.25	0.5	??			
2 Motors w/ Electronic Differential				??	1.04	110.5	??	??	??	??	3176	0.105	0.25	0.5	??			
1 Motor with Transmission				??	0.84	128	??	??	??	??	3462	0.105	0.25	0.5	??			

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

Figure 23: Conceptual Design QFD.

Appendix B – Drawing Packet

Electrical System and Wiring Overview

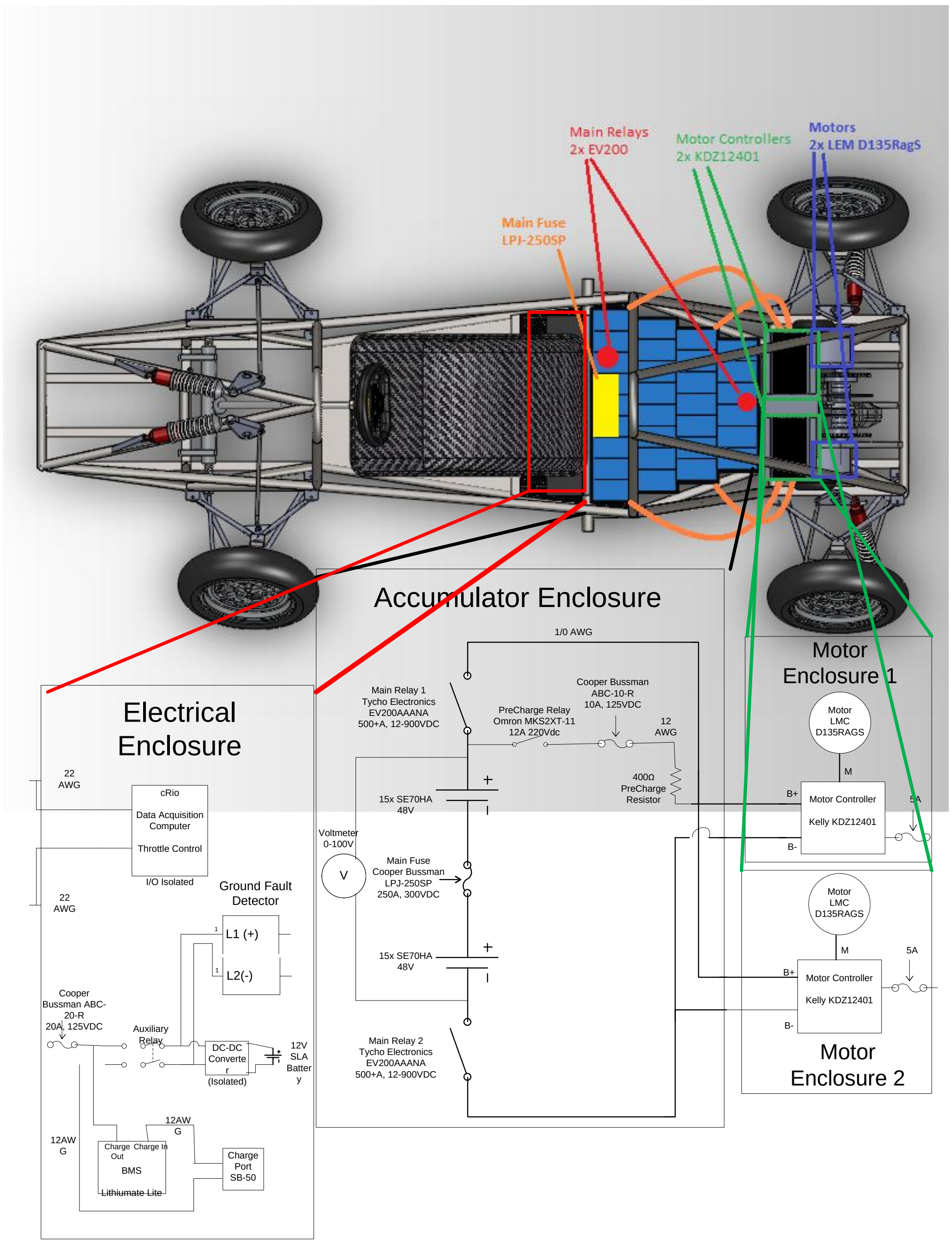
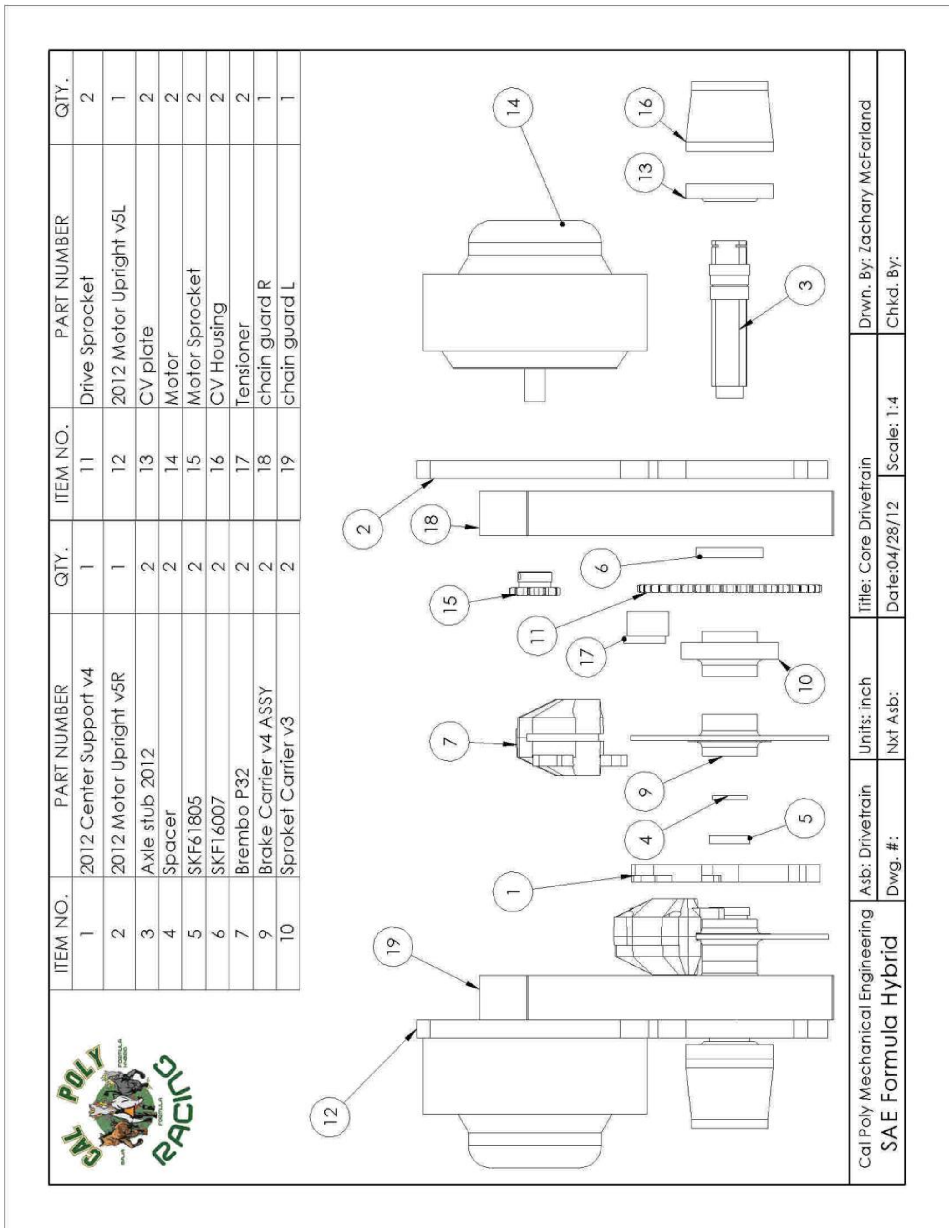


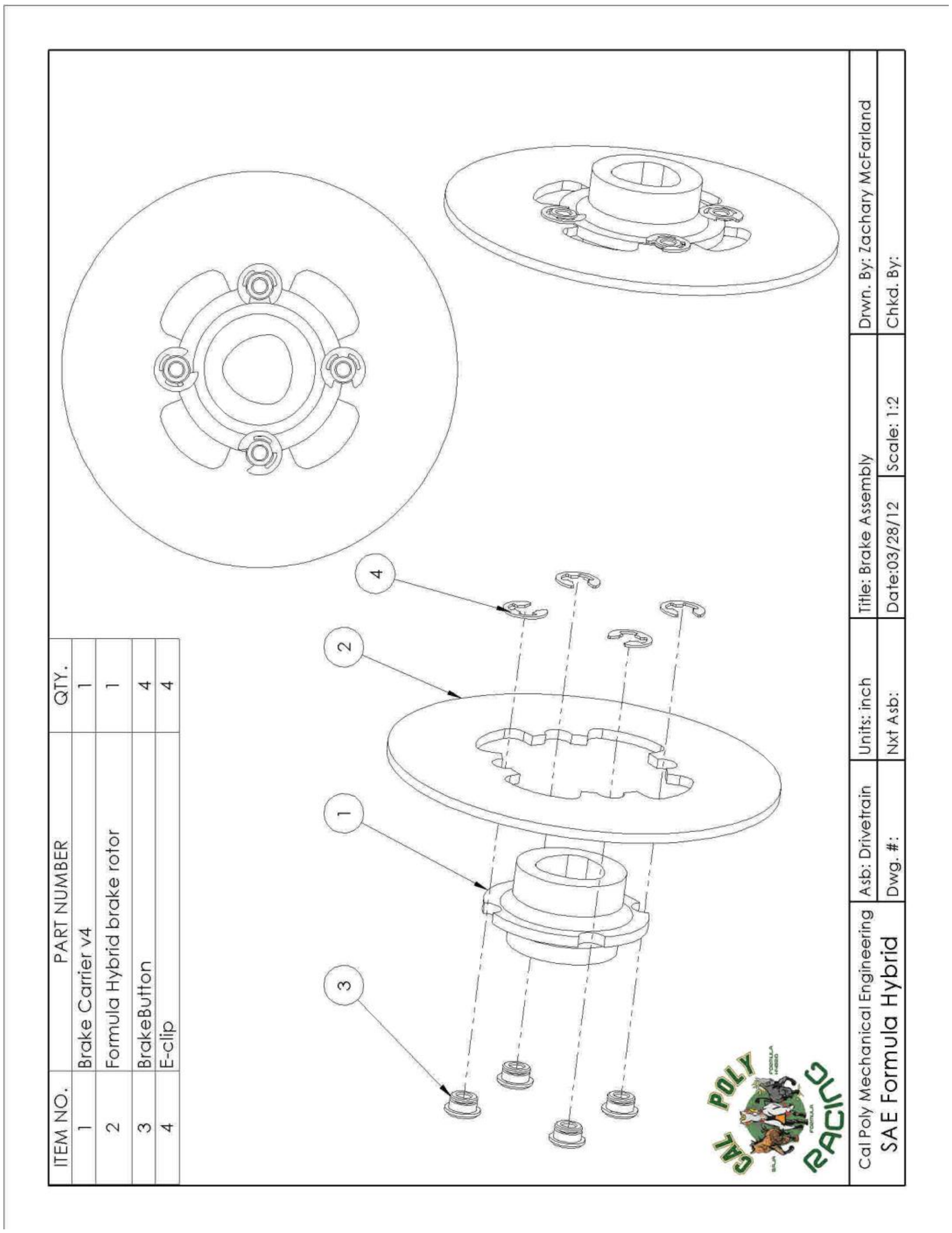
Figure 24. Basic electrical schematic for powering the drivetrain subsystem overlayed on top of a model of the Formula Hybrid car.

Assembly Drawings/Bill of Materials

Exploded Drivetrain Assembly

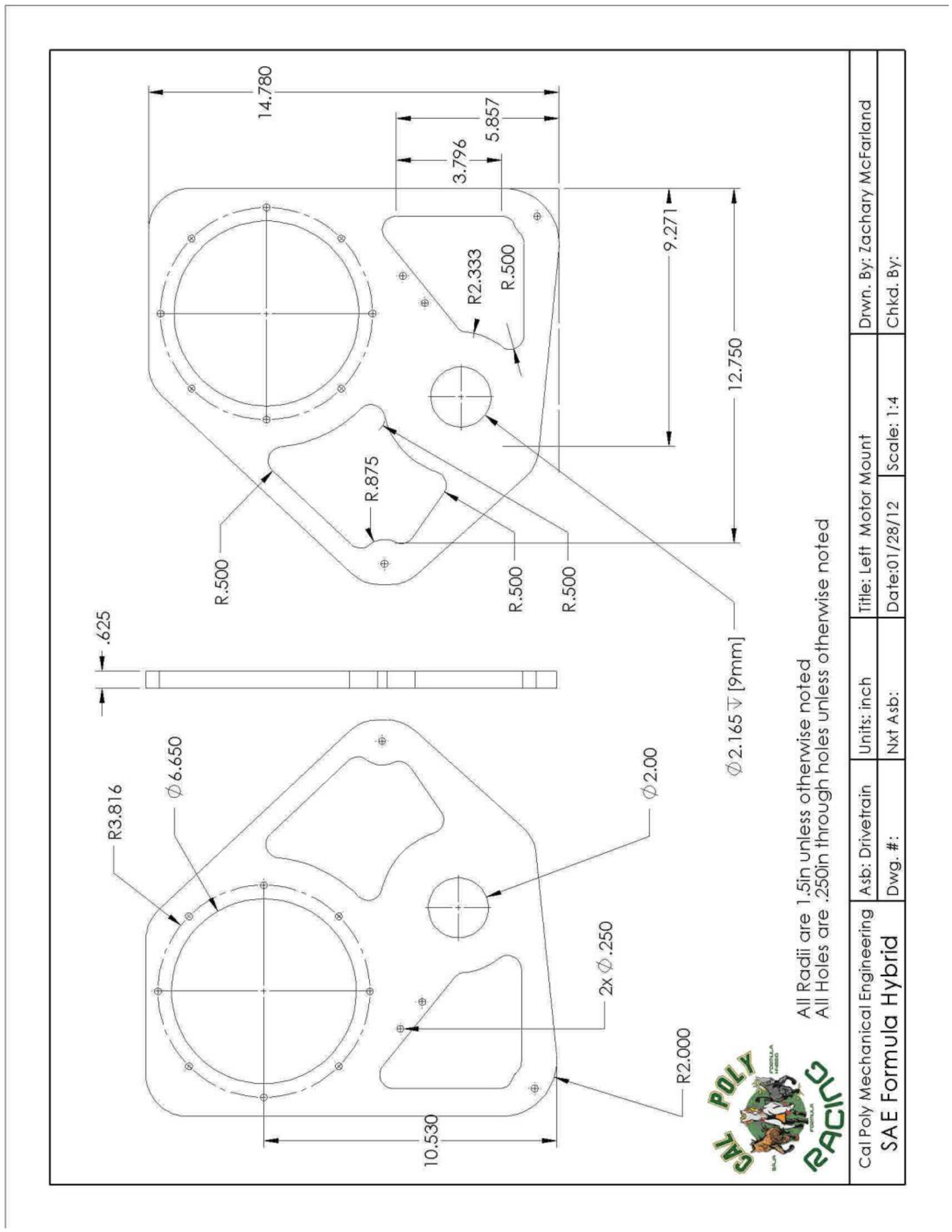


Brake Rotor Carrier Assembly

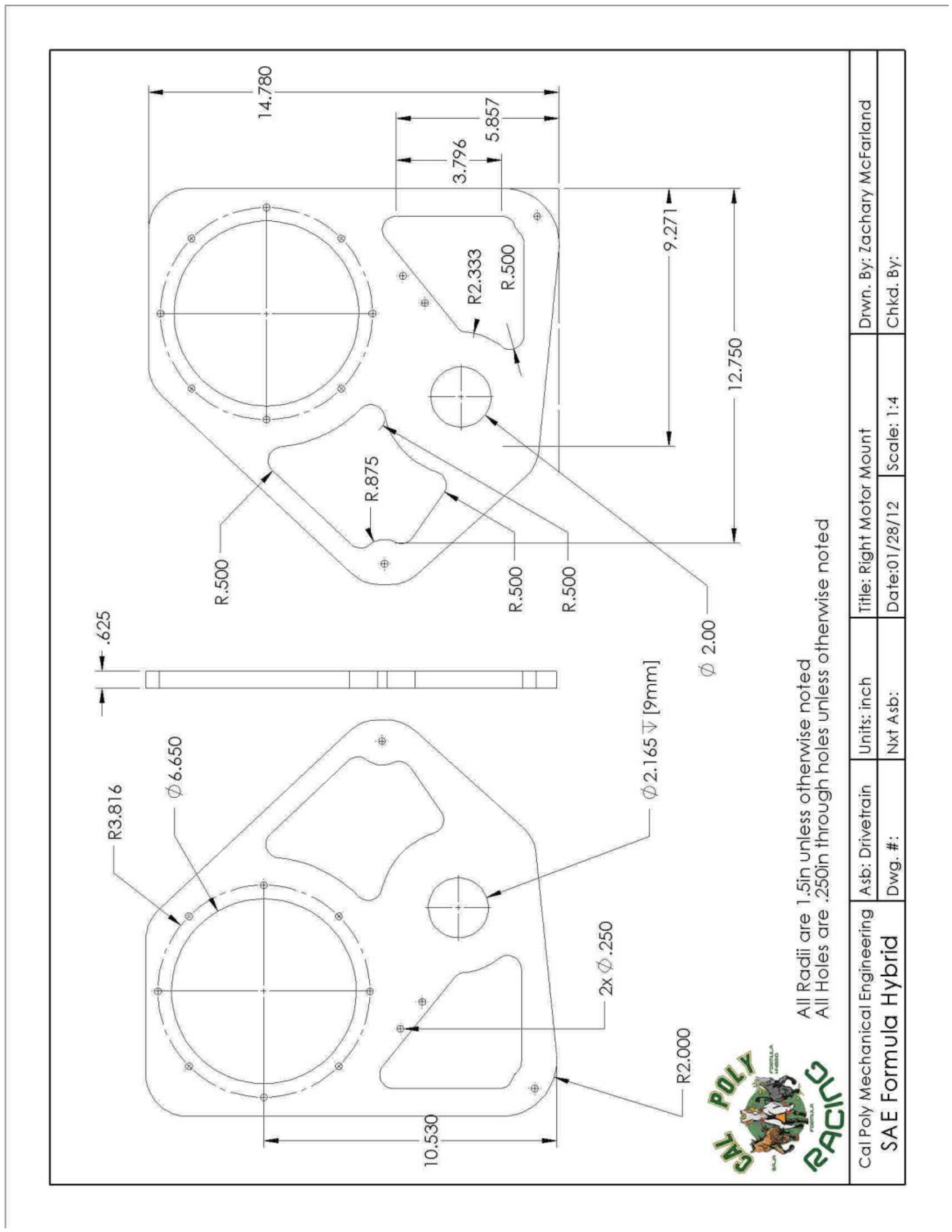


Detailed Part Drawings

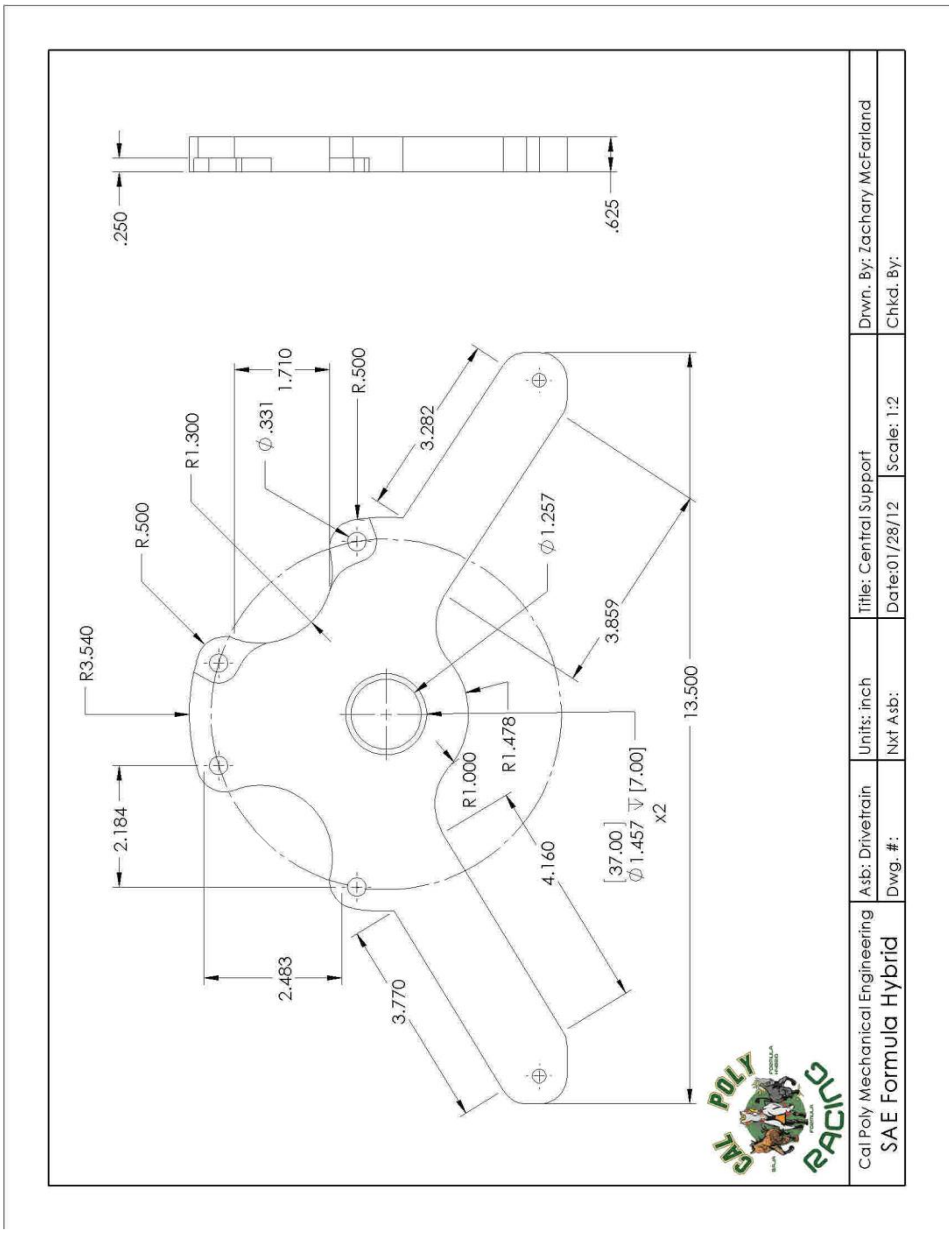
Left Motor Mount Drawing



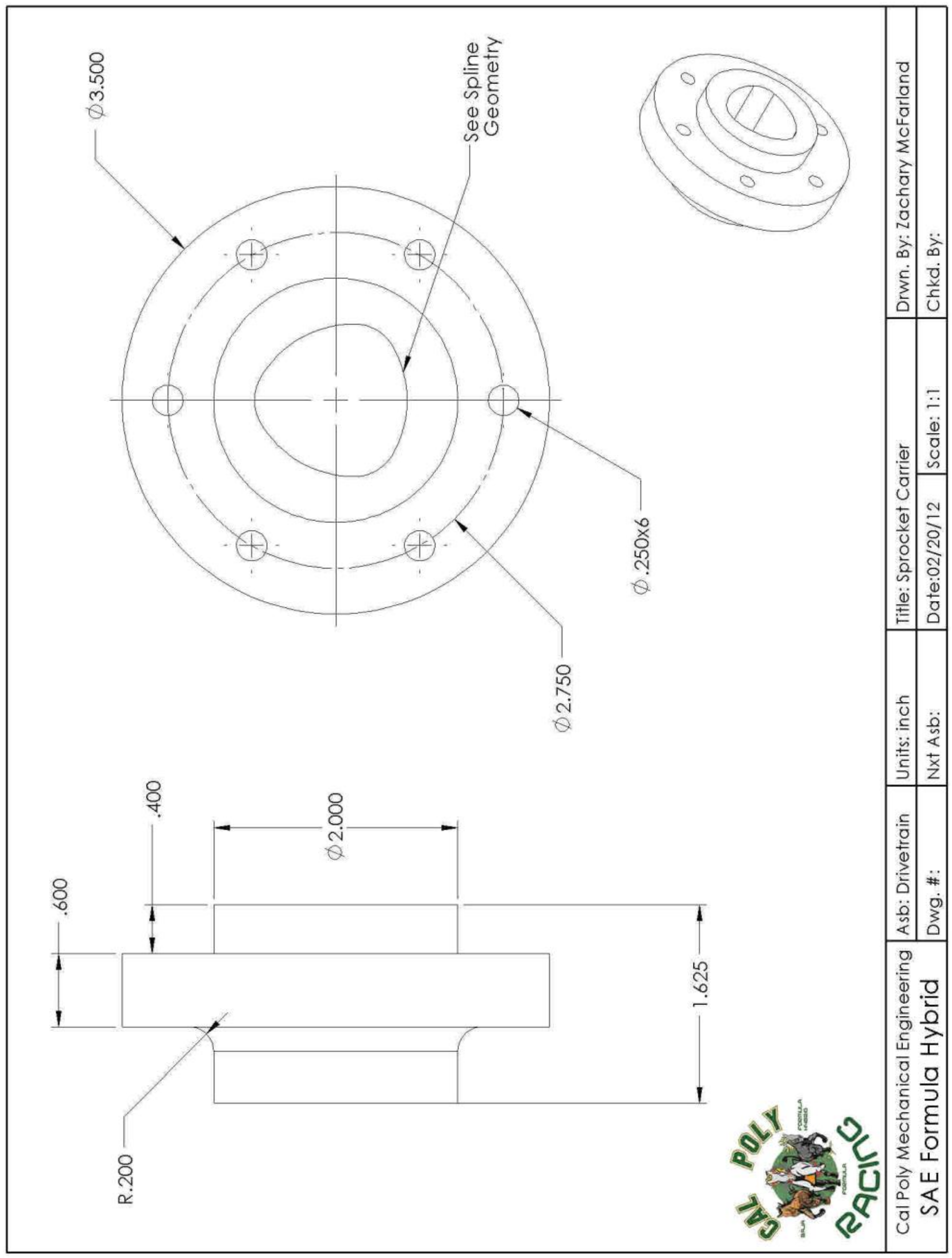
Right Motor Mount



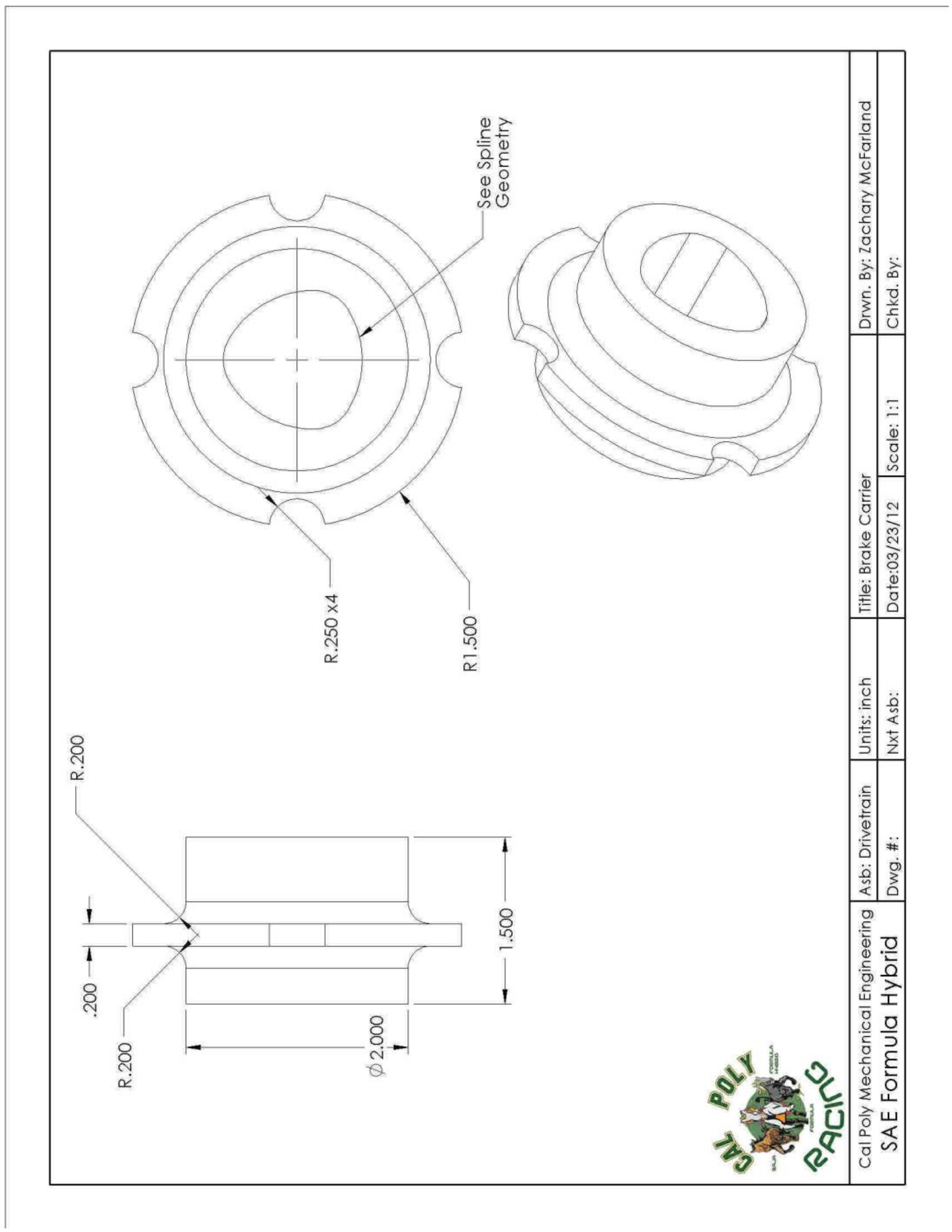
Central Support Drawing



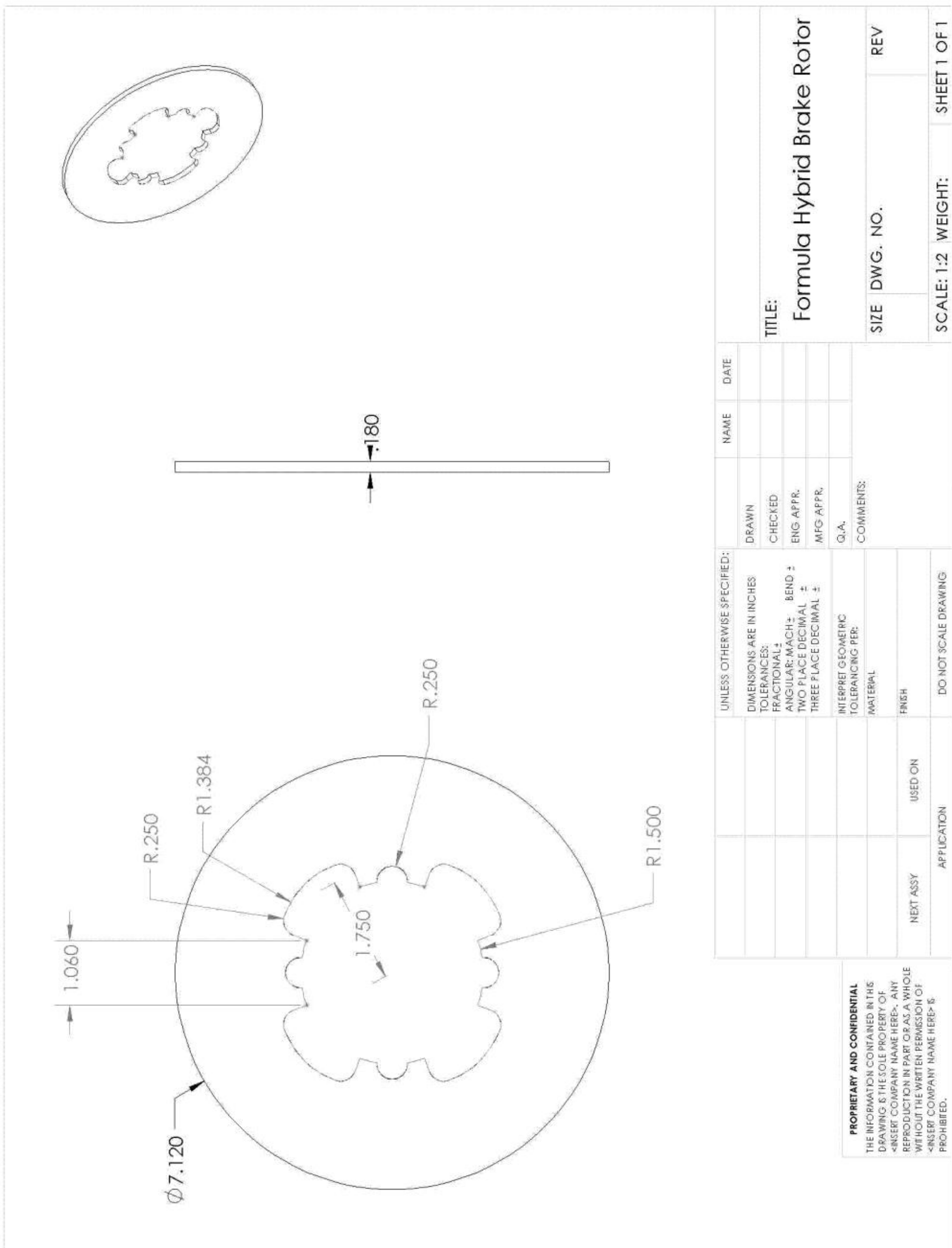
Gear Sprocket Carrier Drawing



Brake Rotor Carrier Drawing



Brake Rotor


Ø7.120
1.060
R.250
R1.384
R.250
1.750
R1.500
.180
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
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FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
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NAME
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Formula Hybrid Brake Rotor
SIZE
DWG. NO.
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WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.
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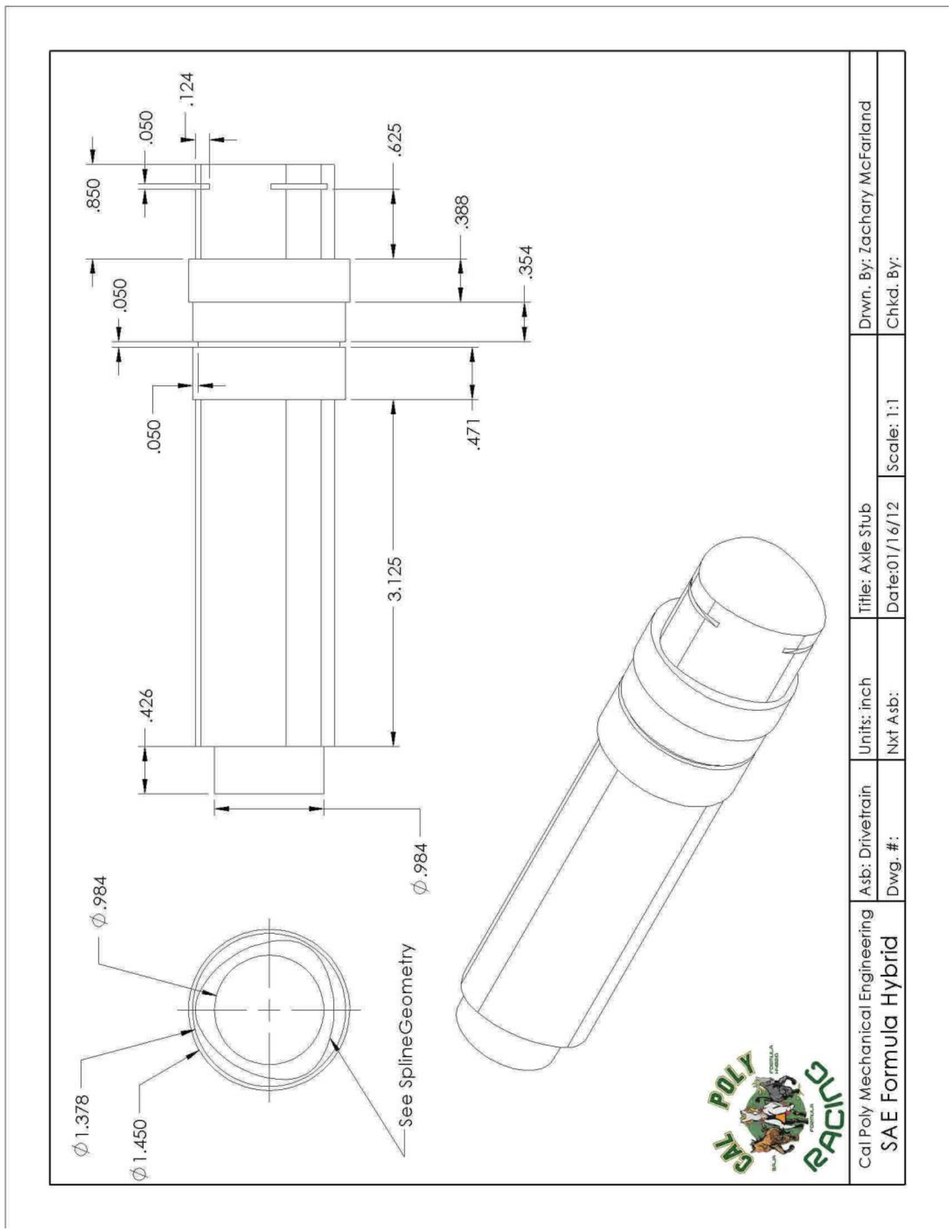
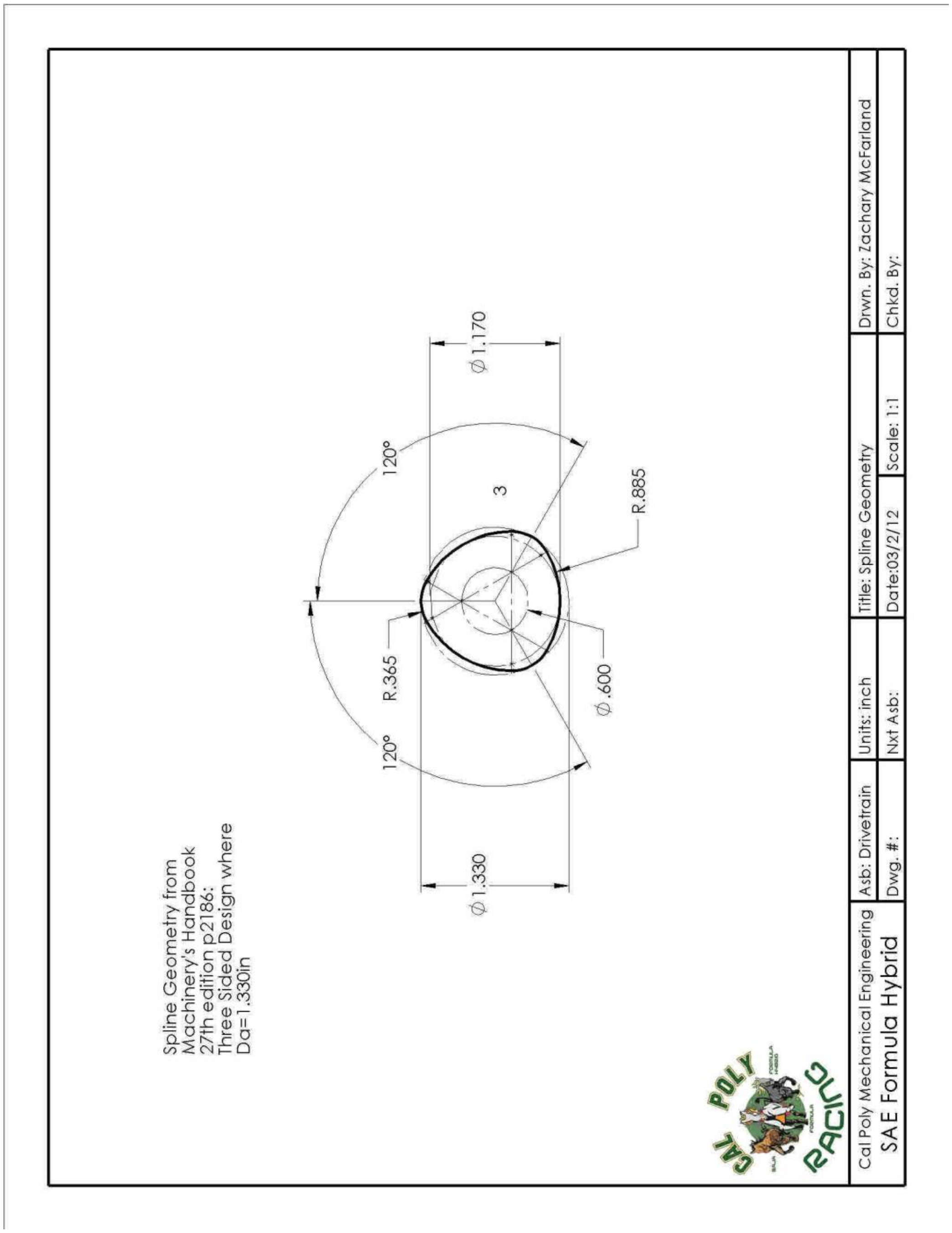
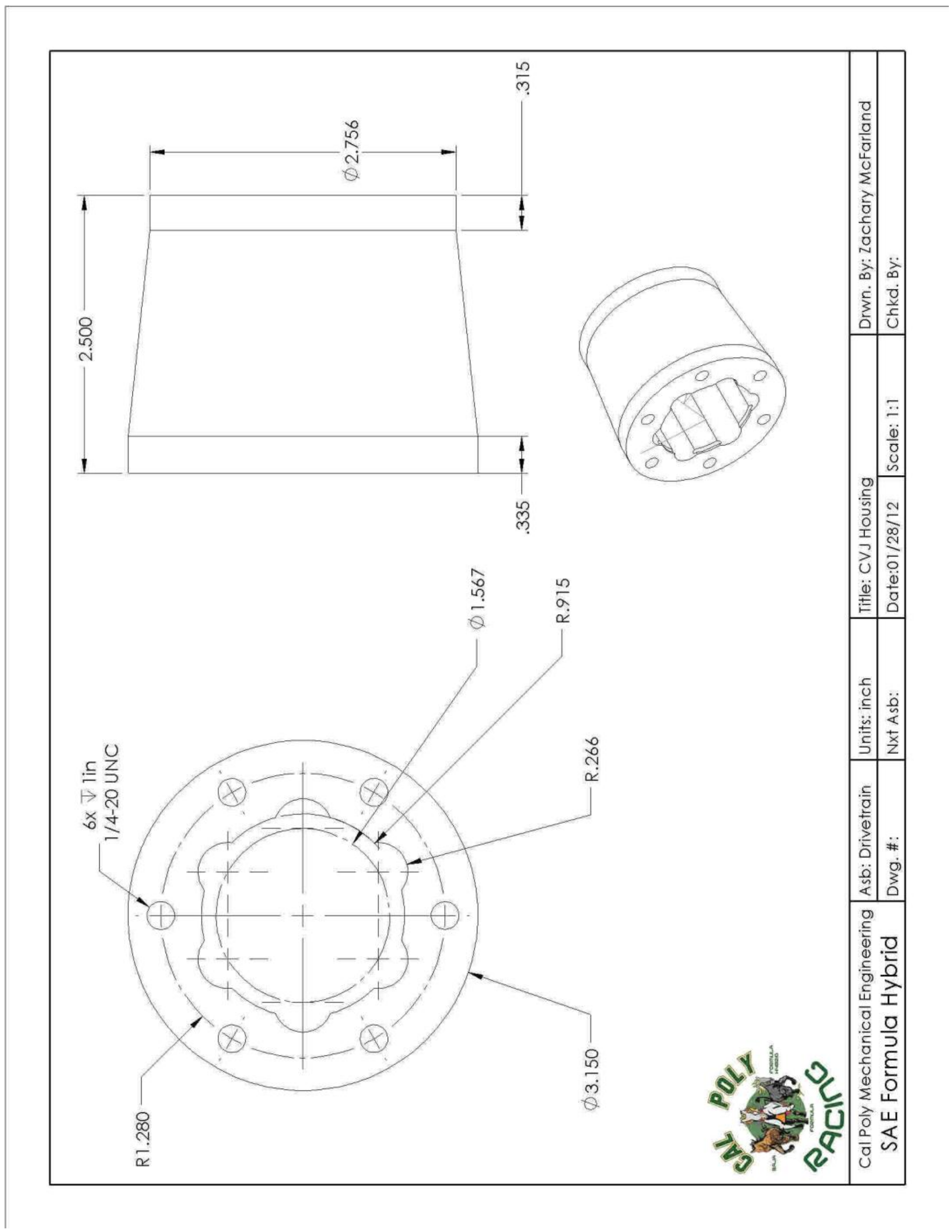
Axle Stub Drawing



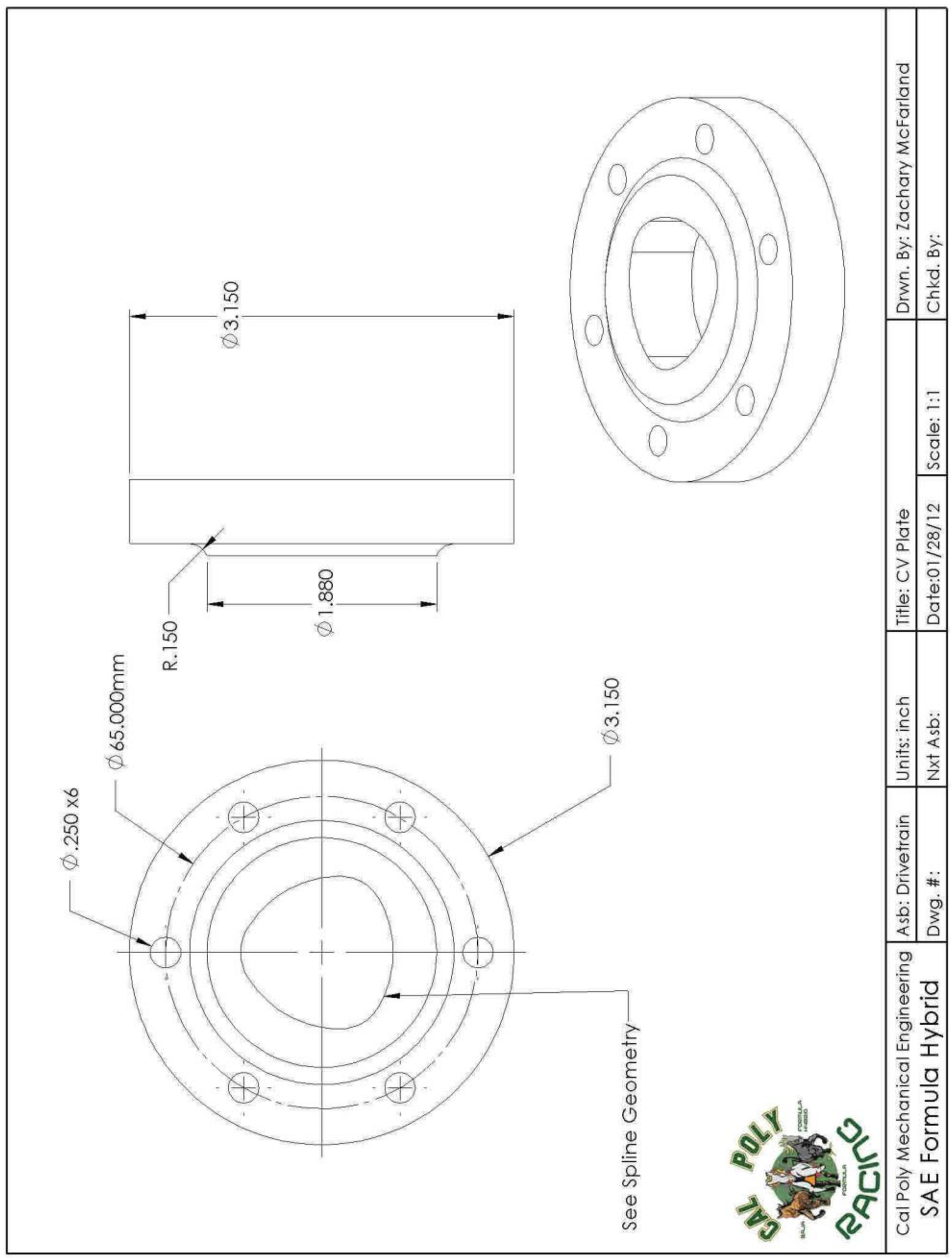
Geometric Spline Geometry



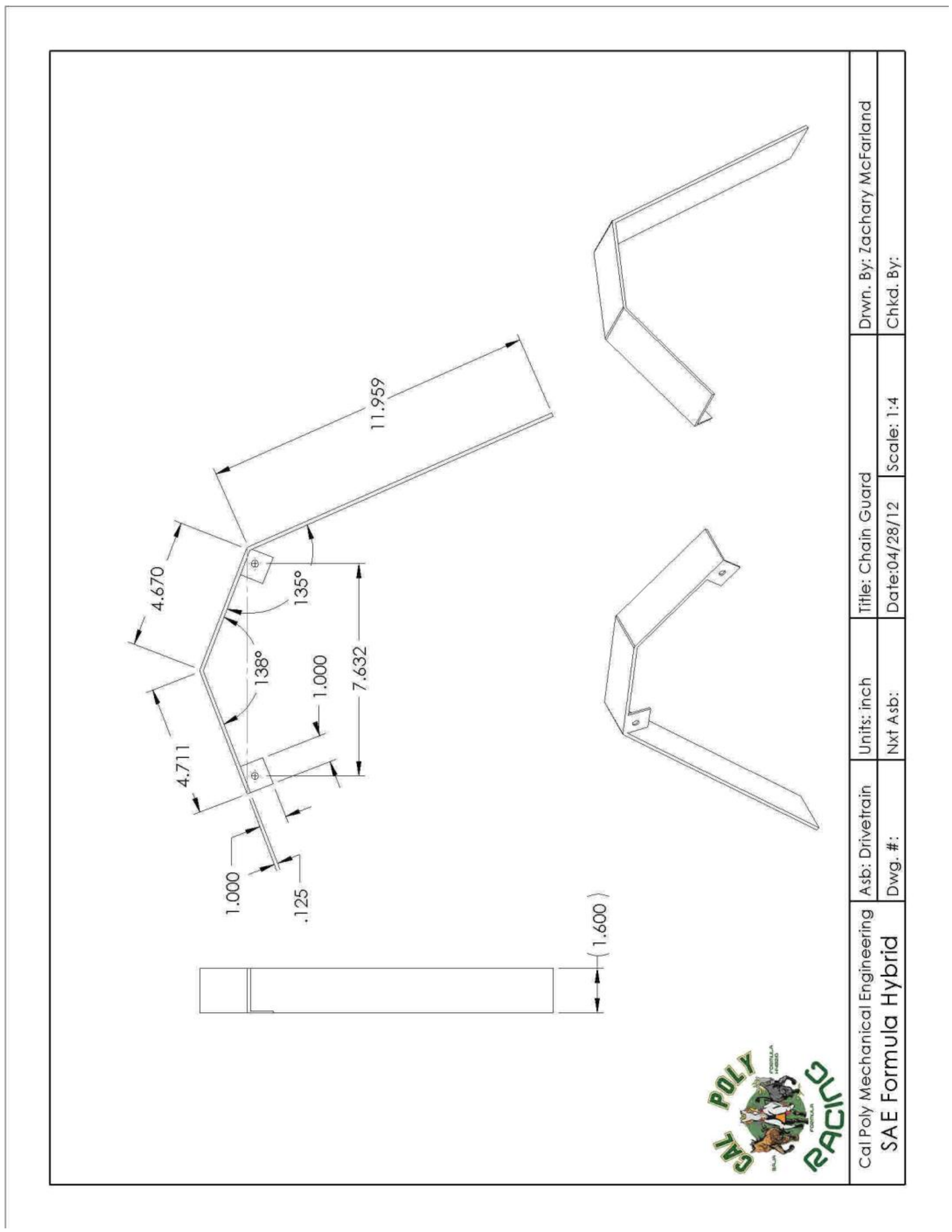
CV Housing Drawing



CV Housing Female Spline Plate

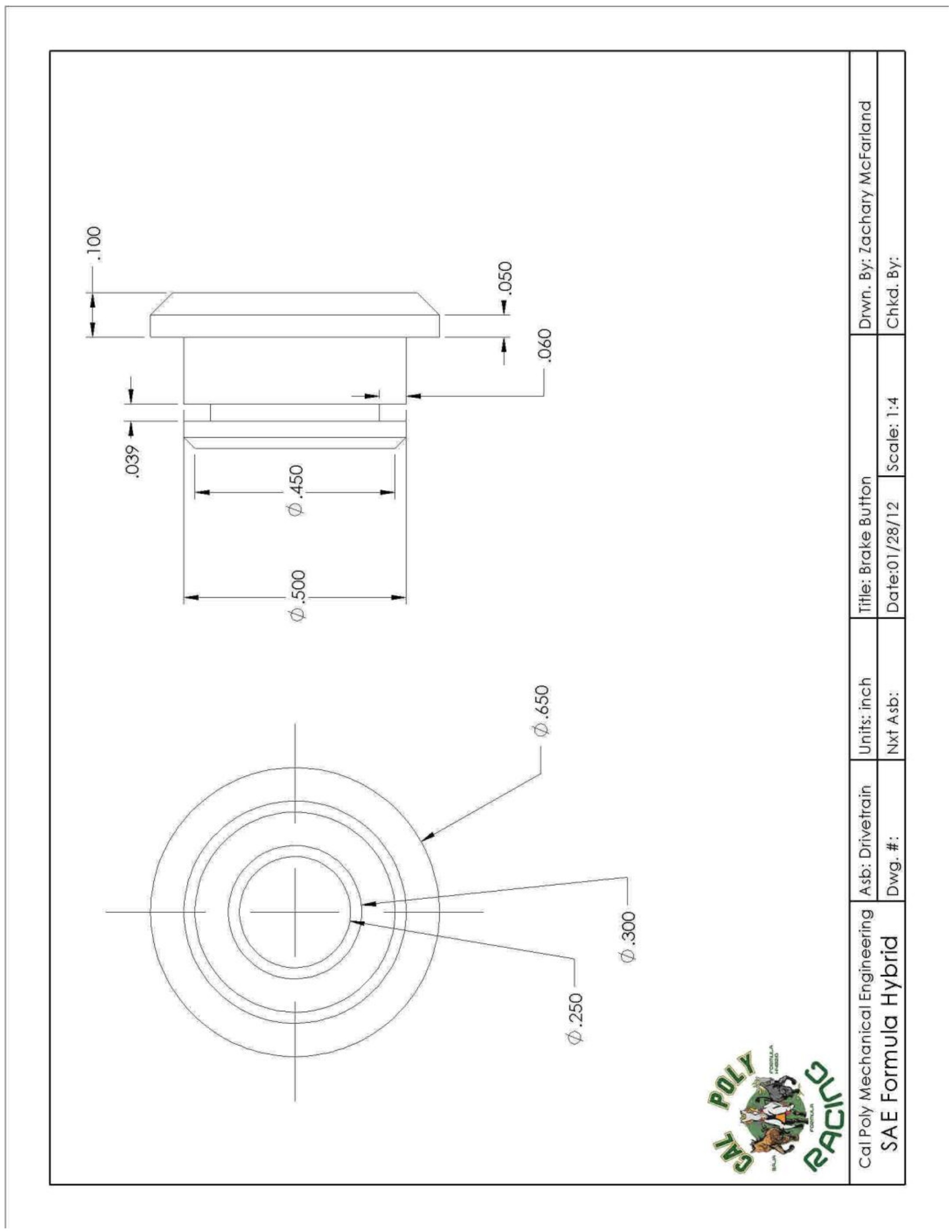


Chain Guard



11.959
4.670
135°
138°
1.000
7.632
4.711
1.000
.125
1.600
Cal Poly Mechanical Engineering
SAE Formula Hybrid
Asb: Drivetrain
Dwg. #:
Units: inch
Nxt Asb:
Title: Chain Guard
Date:04/28/12
Scale: 1:4
Drwn. By: Zachary McFarland
Chkd. By:

Brake Rotor Button Drawing



Appendix C – Design Validation Plan and Report

Design Verification Report

ME429 Design Validation Plan & Report													
	Report Date	6/1/2012	Sponsor	FH SAE					Component/Assembly		Drivetrain	REPORTING ENGINEER:	
TEST PLAN										TEST REPORT			
						SAMPLES		TIMING		TEST RESULTS			
Item No	Specification or Clause Reference	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	Quantity	Type	Start date	Finish date	Test Result	Quantity Pass	Quantity Fail	NOTES
1	Drivetrain Weight	Measure with scale	Weight < 100 lbf	Will Domhart	PV	1	C	3/17/2012	3/17/2012	86 lb	X		
2	Drivetrain Assembly Fitment	Assemble and make sure it rotates	Pass	Will Domhart	PV	1	C	3/17/2012	3/17/2012	X	X		
3	Wheel Speed Sensors	Validate with cRIO	Pass all 4	Will Domhart	PV	4	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
4	Accelerometer	Validate with cRIO	Pass	Will Domhart	PV	1	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
5	Gyrometer	Validate with cRIO	Pass	Will Domhart	PV	1	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
6	Steering Wheel Position Sensor	Validate with cRIO	Pass	Will Domhart	PV	1	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
7	75-m Acceleration Time	Timed distance starting from rest	Time < 5.0 sec	Will Domhart	PV	3	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
8	Top Speed	Measure with GPS	Velocity >= 65 mph	Will Domhart	PV	3	C	3/17/2012	3/17/2012	-	-	-	Motor Controller Issues
9	Energy Consumption (22 km Endurance Race)	Measure power consumption over 1-km with Battery Monitoring System and interpolate over 22-km distance.	Power Consumpiton <= 5.4 kWh	Will Domhart	PV	3	C	3/18/2012	3/25/2012	-	-	-	Motor Controller Issues
10	22-km Race Time	Measure distance vs time over 1-km and interpolate over 22-km distance.	Time < 60 minutes	Will Domhart	PV	3	C	3/18/2012	3/25/2012	-	-	-	Motor Controller Issues

Design Verification Plan: Step – by – Step

Below is the detailed outline for how to test and validate the 2012 Formula Hybrid Drivetrain Design. We have outlined a step by step procedure to test and record data collected in order to verify that the design meets the requirements of the project.

Component Testing & Evaluation:

Drivetrain Component Fitment

Ensure that axle stub & CV housing, sprocket & sprocket carrier, brake rotor & brake carrier, bearings & center support, bearings and motor supports, brake carrier & axle stub, and sprocket carrier & axle stub assemble and mate correctly:

1. Properly deburr, clean and lubricate parts to be mated.
2. Measure mated surfaces and compare critical dimensions versus design dimensions in the component drawings.
3. Carefully assembly mated parts ensuring not to force parts together if they don't fit together quite right. If parts aren't assembling together correctly, recheck measurements.
4. If necessary, make necessary adjustments to parts in order for parts to fit together by machining, shimming or other methods necessary.

Drivetrain Weight

To be conducted for each of the following components: (2) Axle stubs, (2) Motor mounts, (1) Center support, (4) Axle Stub bearings, (2) Brake Carriers, (2) Sprocket Carriers, (2) CV Housings, (2) Pinion Sprockets, (2) Gear Sprockets, (2) RAG/s Motors

1. Turn on scale and zero the scale.
2. Measure parts, one at a time and record the mass of each part in a spreadsheet.

Motor & Controller Operation Bench-top Testing

1. Wire Motors, controllers, and cRIO, and test operation of motors
2. Connect Throttle potentiometer signal wire to the cRIO Controller throttle input pin.
3. Connect cRIO Throttle Output wires to Throttle Input terminals on respective Kelly Motor Controllers.
4. Connect Brake Signal wire from Brake pedal sensor to cRIO Controller Brake Signal input pin.
5. Connect Brake signal wires from cRIO Controller to respective Kelly Motor Controller Brake signal input wires.
6. Connect Positive terminal of batteries to respective positive input terminals on Kelly Motor Controllers
7. Connect Ground terminal of batteries to respective ground input terminals on Kelly Motor Controllers.

8. Connect Positive output on Kelly Motor Controllers to positive input terminals on respective RAG/S motors.
9. Connect Ground output on Kelly Motor Controllers to ground input terminals on respective RAG/S motors.

Sensor Operation

To be conducted for the following sensors: (1) Throttle Pedal Potentiometer, (1) Steering Wheel Position Sensor, (4) Wheel Speed Sensors, (1) Yaw Gyroscope, and (1) 3-axis Accelerometer:

1. Mount Wheel Speed sensors to each wheel upright.
2. Wire output signal from each wheel speed sensor to the correct input pin on the cRIO.
3. Supply Power and Ground to Wheel Speed sensors.
4. Mount Accelerometer and Gyro near center of mass of the car.
5. Wire output signal from accelerometer sensor to the correct input pin on the cRIO.
6. Wire output signal from Gyro sensor to the correct input pin on the cRIO.
7. Wire output signal from Throttle Potentiometer to the correct input pin on the cRIO.
8. Supply Power and Ground to the Throttle Potentiometer.
9. Supply Power and Ground to the Accelerometer.
10. Supply Power and Ground to the Gyro.
11. Mount Steering Wheel Position Sensor on steering column.
12. Wire output signal from Steering Wheel Position sensor to the correct input pin on the cRIO.
13. Supply Power and Ground to the Steering Wheel Position Sensor.
14. Power up low voltage sensor system.
15. Connect cRIO to computer.
16. Spin the Front Left Wheel and see if the computer registers a change when the sensor triggers.
17. Record Data in the attached table.
18. Repeat Steps 14 and 15 for the Front Right, Left Rear, and Right Rear wheels.
19. Rotate the car in order to activate the Gyro and Accelerometer. Verify both sensors are reading the respective change in state of the chassis.
20. Record Data in the attached table.
21. Rotate Steering Wheel and see if the computer registers a change in the rotation rate and accelerations of the chassis.
22. Record Data in the attached table.
23. Rotate Steering Wheel and see if the computer registers a change with the position of the steering wheel.
24. Displace Throttle pedal and see if the computer registers a change in pedal position.
25. Record Data in the attached table.

Subsystem Assembly:

Mount Drivetrain Assembly on Car

1. Press Center support axle stub bearings into each side of the Center Support.
2. Bolt Center Support to rear frame.
3. Press axle stub bearing into each of the Motor supports
4. Bolt Right and Left Motor Supports to rear frame.
5. Bolt motors to motor supports using 8 bolt fasteners. Orient output shafts inboard.
6. Mount pinion sprockets on respective motor output shafts with key and setscrew.
7. Bolt Drive Sprockets to right and left sprocket carriers
8. Secure Brake rotors to right and left brake carriers with button and C-clips.
9. Hold Right side sprocket carrier while sliding axle stub from outboard to inboard through the right motor support and into sprocket carrier.
10. Continue holding sprocket carrier and axle stub while positioning Brake carrier. Slide axle stub through brake carrier and chain and into bearing in center support.
11. Attach CV housing to axle stub on outboard side of motor support and secure with C-Clip.
12. Attach Right side half shaft to right side suspension upright.
13. Slide inboard side of Half-shaft into CV housing and bolt upright into place on the A-arms.
14. Bolt right side brake caliper to right side brake carrier.
15. Mount right side chain tensioner to motor upright.
16. Engage chain to pinion, drive, and tensioner sprockets. Tension chain to 1/8" slack between pinion and drive sprockets.
17. Repeat steps 9-16 for the left side.

Wire Motors, cRIO Controller, and Kelly Motor Controllers on Car

Wire motors and controllers as described in *Motor & Controllers Bench-top Testing*.

Traction Control Sensor Assembly on Car

1. Secure front wheel speed sensors to front left and right suspension uprights. Connect power, ground, and signal wires between leads on respective sensors and the corresponding input pins on the cRIO controller.
2. Secure rear wheel speed sensors to right and left motor mounts. Connect power, ground, and signal wires between leads on respective sensors and the corresponding input pins on the cRIO controller.
3. Mount steering wheel position sensor.
4. Mount Gyroscope & Accelerometer breakout boards in compartment with cRIO controller. Connect power, ground and signal wires between leads on respective sensors and the corresponding input pins on the cRIO controller.

Subsystem Testing & Evaluation:

Motor Bias

1. Prepare the car for driving.
2. Mark a straight line on the ground.
3. Line up the center of the car on the line.
4. Accelerate the car slowly without using the steering to correct the path.
5. Notice if the car drifts left or right.
6. Modify the Motor Bias's to correct for the drift.

Traction Control

1. Attempt to spin the tires.
2. If the tires are spinning, Adjust the Traction Control Constants.
3. If the tires are not spinning, but the car is slow to accelerate, adjust the Traction Control gains.
4. If the tires are not spinning, but the car is normal to accelerate, do not do anything.

Yaw Control

1. Attempt to corner the car too fast.
2. If the car has understeer in the corner, Adjust the Yaw Control Constants.
3. If the car has oversteer in the corner, Adjust the Yaw Control Constants.
4. If the car has neutral steer in the corners, do not do anything.

75-m Acceleration Test

1. Fully Charge the Batteries.
2. Measure out a 75m straight path, with start and finish cones.
3. Place two people with stopwatch timers at finish.
4. Line the car up at the start line.
5. Measure the time it takes for the car to complete the 75 meter run.
6. Repeat three times to make sure it's consistent.
7. Record the time in the attached table.

Top Speed Test

1. Prepare a Longer Straight Course in order to test top speed.
2. Mount a GPS to the car that has peak speed recall.
3. Accelerate the car to top speed.
4. Measure the wheel speed from the cRIO and calculate the vehicle speed.
5. Compare with the GPS data for peak speed.
6. Record Data in the attached table.

Endurance Test

1. Fully Recharge the Batteries.
2. Map out a circuit track that has a start / finish line at the same spot.
3. Measure how long the track is in kilometers.
4. Compute the number of laps in order to reach approximately 5 km total distance. (Get as close as you can but keep an even number of laps, no partial laps)
5. Line up the car at the starting line.
6. Run the course at a modest, endurance racing pace.
7. Stop when the driver completes the number of laps from part 3.d.
8. Look at the BMS and compute the energy consumption used for 5km and extrapolate to the whole 22km race length.
9. Record Data in the attached table.
10. Measure the time it took to complete the 5km distance and extrapolate to find the 22km race time.
11. Record Data in the attached table.

Appendix D – List of Vendors, Contact Information, and Pricing

Mechanical Systems

- Lynch Motor Company http://www.lemcoltd.com/
    - (1) x Lynch D135 RAG/S DC Motor
    - $1600
- Kelly Controllers http://www.kellycontrollers.com/
    - (2) x Kelly KDZ motor controllers
    - $600/ea $1200/total
- McMaster-Carr http://www.mcmaster.com/
    - Axle Stub Material
    - 7075 Aluminum
    - (1) x 3-in x 12-in (diameter x length)
    - $125/ea
- McMaster-Carr http://www.mcmaster.com/
    - Motor and center mount material
    - 6061 Aluminum
    - (3) x 18-in x 24-in x 5/8-in (length x width x thickness)
    - $350/total
- Go Kart Galaxy http://www.gokartgalaxy.com/
    - (2) x 41 tooth gear sprockets
    - $40/ea $80/total
- McMaster-Carr http://www.mcmaster.com/
    - (1) x 10 tooth pinion sprocket
    - $13/ea
- McMaster-Carr http://www.mcmaster.com/
    - Gear sprocket/brake rotor carriage material
    - 6061 Aluminum
    - (1) x 4-in x 6-in (diameter x length)
    - $80/ea
- McMaster-Carr http://www.mcmaster.com/
    - Bearings
    - (2) x SKF 6006
    - (2) x SKF 61805
    - $100/total
- McMaster-Carr http://www.mcmaster.com/
    - Chains
    - 4-ft of series 40 chain
    - $15/total

Traction Control/Data Acquisition

- Digi-Key http://www.digikey.com/
  - Supplier Part Number: MA3
  - (1) x Miniature Absolute Magnetic Shaft Encoder (Steering Wheel Position sensor)
  - 10-bit analog output signal
  - $39.60/ea
- Sparkfun http://www.sparkfun.com/
  - Supplier Part Number: SEN-09812
  - (1) x 6 Degree-of-Freedom Sensor Board
  - 10-bit analog output signal
  - 3-axis accelerometer (±1.5 or ±6 gees maximum acceleration)
  - Three 1-axis gyroscopes for yaw, pitch, and roll (300°/s maximum)
  - $124.95/ea
- US Digital http://www.usdigital.com/
  - Supplier Part Number: 480-2826-5-ND
  - (4) x Hall Effect, Ferrous Gear Tooth Detectors
  - Detects change in magnetic field caused by ferrous metal gear/sprocket tooth passing by front of sensor.
  - $18.24/ea 72.73/total

Appendix E – Vendor Component Specifications/Data Sheets

Mechanical Systems Data Sheets

Aluminum 6061 Data Sheet

ALCOA

Alcoa Engineered Products

ALLOY 6061

Understanding Extruded Aluminum Alloys

Alloy 6061 is one of the most widely used alloys in the 6000 Series. This standard structural alloy, one of the most versatile of the heat-treatable alloys, is popular for medium to high strength requirements and has good toughness characteristics. Applications range from transportation components to machinery and equipment applications to recreation products and consumer durables. Alcoa produces 6061 for use in standard and custom shapes, rod and bar products, and seamless and structural pipe and tube.

Alloy 6061 has excellent corrosion resistance to atmospheric conditions and good corrosion resistance to sea water. This alloy also offers good finishing characteristics and responds well to anodizing; however, where cosmetic appearance is critical, consider the use of alloy 6063. The most common anodizing methods include clear, clear and color dye, and hardcoat.

Alloy 6061 is easily welded and joined by various commercial methods. (Caution: direct contact by dissimilar metals can cause galvanic corrosion.) Since 6061 is a heat-treatable alloy, strength in its -T6 condition can be reduced in the weld region. Selection of an appropriate filler alloy will depend on the desired weld characteristics. Consult the Material Safety Data Sheet (MSDS) for proper safety and handling precautions when using alloy 6061.

For screw machine applications, alloy 6061 has adequate machinability characteristics in the heat-treated -T6/-T6511 condition. With -T6/-T6511 mechanical properties, chips from machining (particularly turning and drilling) are difficult to break. Chip breakers are recommended, and special machining techniques (i.e. peck drilling) can improve chip formation. To enhance the machinability of its ECON-O-ROD®, ACC-U-ROD®, ECON-O-PLATE® and ACC-U-PLATE® products, Alcoa has developed a unique chemistry for alloy 6061, which conforms to industry specifications.

For minor bending applications, special forming tempers -T6S2, -T6S15, -T6S9 or -T6S10 may be sufficient to facilitate bending (dependent upon bend radius and degree of bend). When more severe bends are required, a softer temper condition such as -T1, -T4 or even -0 may be necessary to prevent cracking.

Alcoa offers alloy 6061 in a variety of standard tempers, as well as special tempers developed for unique applications. These are summarized below:

6061 Temper Designations and Definitions

Alcoa produces 6061 alloy with a wide selection of standard and special tempers. In the annealed condition (-O temper), 6061 is extremely ductile and well suited for severe forming applications. When solution heat-treated and naturally aged (-T4 condition), 6061 has good formability for bending. After artificial aging (precipitation heat-treating), 6061-T4 is capable of developing -T6 properties.

Standard Tempers	Standard Temper Definitions*
F	As fabricated. There is no special control over thermal conditions and there are no mechanical property limits.
O	Annealed. Applies to products that are annealed to obtain the lowest strength temper.
T1	Cooled from an elevated temperature shaping process and naturally aged. (See Note B.)
T4, T4511	Solution heat-treated and naturally aged. (See Notes C & D.)
T51	Cooled from an elevated temperature shaping process and artificially aged. (See Note B.)
T6, T6511	Solution heat-treated and artificially aged. (See Notes C & D.)
Alcoa Special Tempers**	**Alcoa Special Temper Definitions**
T4S6	For 6061 extrusions requiring maximum formability in the unaged condition and subsequently aged to -T6. May not meet -T4 minimum mechanical properties, but will meet -T6 minimum when properly aged. Test reports state -T6 properties to demonstrate heat treat capabilities, but extrusions are supplied unaged. (See Note A.)
T6S2, T6S15	For 6061 extrusions requiring good formability; meets standard 6061-T6 minimum properties.
T6S9, T6S10	For 6061 extrusions requiring improved forming characteristics not obtainable with -T6S2 and -T6S15 tempers. Lower minimum properties of 35.0 ksi tensile & 30.0 ksi yield guaranteed to enhance formability. (See Note A.)
T6S4	Applies to 6061 extrusions requiring maximum hardness for strength and good machinability. Same minimum tensile and yield strengths as standard -T6, but with lower minimum elongation of 6%. (See Note A.)
T6H, T6511H	Alcoa's "H" temper is offered for special applications requiring improved machinability and higher minimum mechanical properties than standard -T6 or -T6511. Minimum properties of 42 ksi tensile, 38 ksi yield, and 10.0% elongation are guaranteed. "H" temper is available for rod, bar, and certain solid profiles with a principle thickness of .500" or greater. (See Notes C & D.)
T6G, T6511G	Alcoa's "G" temper is available for applications requiring a uniform grain structure to enhance anodized appearance for rod and bar sizes with a thickness of 2.00" or greater. A minimal peripheral grain band may still be present, but it is greatly reduced compared to standard -T6/-T6511. Minimum mechanical properties are same as "H" tempers. (See Notes C & D.)
T6X, T6511X	Alcoa's "X" temper is available for special applications requiring a uniform recrystallized grain structure in extrusions less than 2" thickness to enhance anodizing appearance. Other benefits include improved machinability, same mechanical properties as 6061-T6/T6511. (See Notes C & D.)
T5S26	For 6061 press-quenched and over-aged extrusions requiring improved stamping characteristics. Minimum mechanical properties are 26.0 ksi tensile, 16 ksi yield, 16% elongation. (See Note A.)

*For further details of definitions, see Aluminum Association's Aluminum Standards and Data manual and Tempers for Aluminum and Aluminum Alloy Products. **Note A:** The specified special temper will not conform to Military, Federal, ASTM, ASME and AMS specifications. **Note B:** Applies to products that are not cold worked after cooling from an elevated temperature shaping process, or in which the effect of cold work in flattening or straightening may not be recognized in mechanical properties. **Note C:** Applies to products that are not cold worked after solution heat-treatment, or in which the effect of cold work in flattening or straightening may not be recognized in mechanical properties. **Note D:** Temper -T4511 and -T6511 apply to products that are stress-relieved by stretching.

**Alcoa Special Temper designations are unregistered tempers for reference only and provided for customer use to identify unique processing, material, or end use application characteristics.

Alloy 6061 Chemical Analysis

Liquidus Temperature: 1206°F **Solidus Temperature:** 1080°F **Density:** 0.098 lb./in.^{3}

Percent Weight	Si	Fe	Cu	Mn	Mg	Cr	Zn	Ti	Others Each	Others Total	Aluminum
Minimum	.40	—	.15	—	.8	.04	—	—	—	—	
Maximum	.8	.7	.40	.15	1.2	.35	.25	.15	.05	.15	Remainder

Average Coefficient of Thermal Expansion (68° to 212°F) = 13.1 x 10^{-6} (inch per inch per °F)

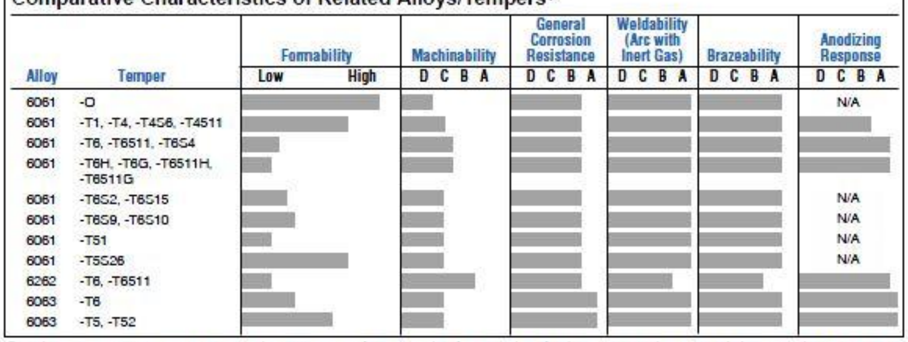
Alloy 6061 Mechanical Property Limits for Rod, Bar, Tube, Pipe and Standard Shapes

Temper	Specified Section or Wall Thickness[2] (inches)	Tensile Strength (ksi) Ultimate Min.	Ultimate Max.	Yield (0.2% offset) Min.	Yield (0.2% offset) Max.	Elongation[3] Percent Min. in 2 inch or 4D[5]	Typical Brinell Hardness (500 kg load/ 10 mm ball)	Typical Ultimate Shearing Strength (ksi)
Standard Tempers[1]								
O	All	—	22.0	—	16.0	16	30	12
T1	Up thru 0.625	26.0	—	14.0	—	16	—	—
T4, T4511[4]	All	26.0	—	16.0	—	16	65	24
T51	Up thru 0.625	35.0	—	30.0	—	8	—	—
T6, T6511[4]	Up thru 0.249	38.0	—	35.0	—	8	95	30
	0.250 and over	38.0	—	35.0	—	10	95	30
Alcoa Special Tempers*								
T6S2, T6S15	Up thru 0.249	38.0	—	35.0	—	8	95	30
	0.250 and over	38.0	—	35.0	—	10	95	30
T6S9, T6S10	Up thru 0.249	35.0	—	30.0	—	8	—	—
	0.250 and over	35.0	—	30.0	—	10	—	—
T6S4	All	38.0	—	35.0	—	6	95	—
T6H, T6511H	1.000 and over	42.0	—	38.0	—	10	95	30
T6G, T6511G	3.000 and over	42.0	—	38.0	—	10	95	30
T6X, T6511X	.250 thru 1.999	38.0	—	35.0	—	10	95	30
T5S26	All	26.0	—	16.0	—	16	65	24

① The mechanical property limits for standard tempers are listed in the "standards section" of the Aluminum Association's Aluminum Standards and Data manual. ② The thickness of the cross section from which the tension test specimen is taken determines the applicable mechanical properties. ③ For material of such dimensions that a standard test specimen cannot be taken, or for shapes thinner than 0.062", the test for elongation is not required. ④ For stress-relieved tempers, the characteristics and properties other than those specified may differ somewhat from the corresponding characteristics and properties of material in the basic temper. ⑤ D = Specimen diameter.

*Alcoa Special Temper designations are unregistered tempers for reference only and provided for customer use to identify unique processing, material, or end use application characteristics.

Comparative Characteristics of Related Alloys/Tempers[1]



① Rating: A=Excellent B=Good C=Fair D=Poor For further details of explanation of ratings, see Aluminum Association's Aluminum Standards and Data manual.

Alcoa Distribution and Industrial Products

53 Pottsville Street
Cressona, PA 17929
Phone: 800-233-3165
FAX: 800-252-4646

Aluminum 7075 Data Sheet

ALCOA

Alcoa Global Cold Finished Products

ALLOY 7075

Understanding Cold Finished Aluminum Alloys

Alloy 7075, a cold finished aluminum wrought product, has the highest strength of all aluminum screw machine alloys. The -T6 and -T651 tempers have a typical tensile strength of 83 ksi, which is higher than many mild steels.

Due to its very high strength, alloy 7075 is used for highly stressed structural parts. Applications include aircraft fittings, gears and shafts, fuse parts, meter shafts and gears, missile parts, regulating valve parts, worm gears, keys, and various other commercial aircraft, aerospace and defense equipment. Rod and bar product forms can be machined on multi-spindle and CNC machining equipment.

Machining

Alloy 7075 offers good machinability when machined using single-point or multi-spindle carbide tools on screw machines. The use of a chip breaker is recommended. The alloy is rated “B” on the Aluminum Association machinability rating system, giving curled or easily broken chips with good to excellent surface finish.

Corrosion

Alloy 7075 has moderate corrosion resistance. The overaged -T73 and -T7351 tempers offers good stress-corrosion cracking resistance as compared to the -T6 and -T651 tempers. (Caution: direct contact by dissimilar metals can cause galvanic corrosion.)

Anodizing

The anodizing response rating for 7075 alloy is good using commercially available methods. The alloy can be both hard and clear-coat anodized.

The properties listed in this Alloy Data Sheet represent the best current information for this alloy. In each specific application, the user is expected to evaluate and test the alloy, temper and finishing method. Consult the Material Safety Data Sheet (MSDS) for proper safety and handling precautions when using alloy 7075.

7075 Temper Designations and Definitions

Standard Tempers	Standard Temper Definitions*
T6, T651	Solution heat-treated and then artificially aged. Applies to products that are not cold worked after solution heat-treatment, or in which the effect of cold work in flattening or straightening may not be recognized in mechanical properties. Temper -T651 applies to products that are stress-relieved by stretching.
T73, T7351	Solution heat-treated and overaged/stabilized. Applies to wrought products that are artificially aged after solution heat-treatment to carry them beyond a point of maximum strength to provide control of some significant characteristic other than mechanical properties. Applies to cast products that are artificially aged after solution heat-treatment to provide dimensional and strength stability.

*For further details of definitions, see Aluminum Association's Aluminum Standards and Data manual and Tempers for Aluminum and Aluminum Alloy Products.

Alloy 7075 Chemical Analysis Liquidus Temperature: 1175°F Solidus Temperature: 890°F Density: 0.101 lb./in.^{3}

Percent Weight	Si	Fe	Cu	Mn	Mg	Cr	Zn	Ti	Others Each	Others Total	Aluminum
Minimum	—	—	1.2	—	2.1	.18	5.1	—	—	—	
Maximum	.40	.50	2.0	.30	2.9	.28	6.1	.20	.05	.15	Remainder

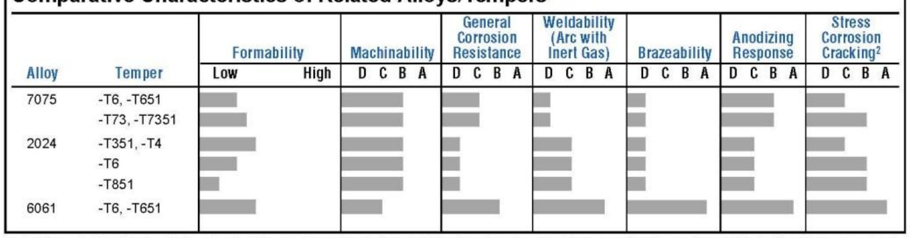
Average Coefficient of Thermal Expansion (68° to 212°F) = 13.1 x 10^{-6} (inch per inch per °F)

Alloy 7075 Global Cold Finished Products Capabilities and Mechanical Property Limits

Temper	Specified Section or Wall Thickness [2] (inches)	Tensile Strength (ksi) Ultimate Min.	Tensile Strength (ksi) Ultimate Max.	Tensile Strength (ksi) Yield (0.2% offset) Min.	Tensile Strength (ksi) Yield (0.2% offset) Max.	Elongation[3] Percent Min. in 2 inch or 4D[4]	Typical Brinell Hardness (500 kg load/ 10 mm ball)	Typical Ultimate Shearing Strength (ksi)	Typical Electrical Conductivity (%IACS)
Standard Tempers[1]									
T6	Up thru 4.000[5]	77.0	—	66.0	—	7	150	48	33
T651	Up thru 4.000[5]	77.0	—	66.0	—	7	150	48	33
	4.001 - 6.000	75.0	—	64.0	—	7	150	48	33
	6.001 - 7.000	73.0	—	62.0	—	7	150	48	33
T73, T7351	Up thru 4.000	68.0	—	56.0	—	10	130	—	40[6]
	4.001 - 5.000	66.0	—	55.0	—	8	130	—	40[6]
	5.001 - 6.000	64.0	—	52.0	—	8	130	—	40[6]

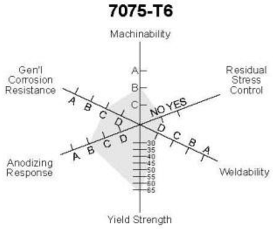
① The mechanical property limits for standard tempers are listed in the "standards section" of the Aluminum Association's Aluminum Standards and Data manual and Tempers for Aluminum and Aluminum Alloy Products. ② The thickness of the cross section from which the tension test specimen is taken determines the applicable mechanical properties. ③ For material of such dimensions that a standard test specimen cannot be obtained, or for shapes thinner than 0.062", the test for elongation is not required. ④ D = Specimen diameter. ⑤ Mechanical properties applicable to rod for thickness shown. For square, hexagonal and octagonal bar, the maximum thickness is 3.5" and for rectangular bar the maximum thickness is 3". ⑥ Minimum value when lot acceptance required for stress corrosion and exfoliation corrosion; 38.0-39.9 per standard requirements and yield strength does not exceed minimum by more than 11.9 ksi.

Comparative Characteristics of Related Alloys/Tempers[1]

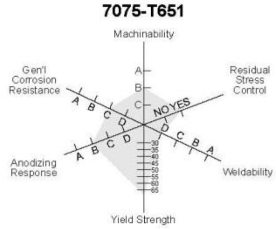


① Rating: A=Excellent B=Good C=Fair D=Poor For further details of explanation of ratings, see Aluminum Association's Aluminum Standards and Data manual. ② Ratings A, B and C are relative ratings based on stress applied transversely with respect to the direction of fabrication after controlled exposure to sodium chloride solution by alternate immersion: A - No known instances of failure in service or laboratory tests. B - No known instances of failure in service, laboratory failures only. C - Service and laboratory failures under special conditions.

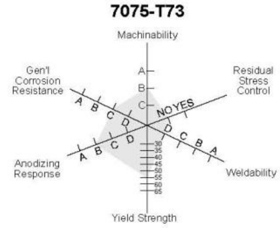
7075-T6



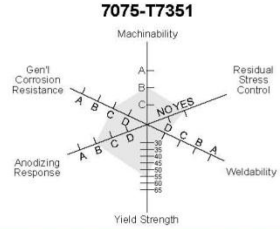
7075-T651



7075-T73



7075-T7351



ALCOA
Global Cold Finished Products
Park Avenue East, PO Box 150
Massena, NY 13662-0150 USA
Phone: 1-877-265-3346
Fax: 315/764-4416
www.alcoa.com/gcfp

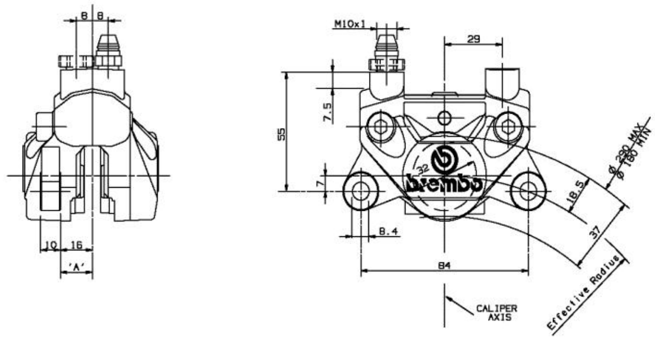
Brembo Rear Brake Calipers

brembo

FIXED CALIPER

P 32 F

20.5161.71/.81

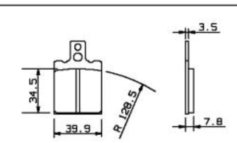


DESCRIPTION

Fixed caliper with 2 opposed pistons,
realized by semicalipers joined by bolts.
Pads are supported by one pin.
Construction Material: Aluminium Alloy.
Operating fluid: brake fluid DOT4.

N.B.: drawing shows caliper Code 20.5161.71
Caliper Code 20.5161.81 specular.

PADS SHAPE AND DIMENSIONS



TECHNICAL CHARACTERISTICS

Pistons number and diameter: 2	32 mm	Average pad surface (1 pad):	13.3 cm²
MAX working pressure:	100 bar	Weight (with not sintered pad):	0.540 Kg
A) Disc position (TOLERANCE): Disc thickness 5.0 mm	±0.1 mm		
Disc thickness 4.6/4.0 mm	±0.3 mm		

CONFIGURATION

CODE CALIPER / CHARACT.	PADS FRICTION MATERIAL	BODY COLOR	MOUNT. SIDE	USABLE DISCS THICKNESS	BRAKE FLUID FOR PADS WEAR
20.5161.71/.81	FRENDO 222	BLACK	SX DX	5.0 mm 4.6 mm 4.0 mm	6.60 cm³

AVAILABLE SPARE PARTS

CODE CALIPER / SPARE PARTS	PADS	BLEEDER SCREW	DUST COVER	PIN
20.5161.71/.81	107.2686.41	105.3246.11	105.1502.10	120.2800.11

Indicating informations subject to modification.
For further specifications contact Technical Section.

LAST UPDATE: 09/09/1994

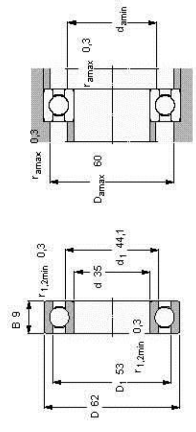
Deep Groove Ball Bearing, 16007

SKF

Deep groove ball bearings, single row, unsealed
Product information

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				* - SKF Explorer bearing
mm			kN		kN	r/min		kg	-
35	62	9	13	8,15	0,375	24000	15000	0,11	16007 *



Calculation factors
k_r 0,02
f_0 14

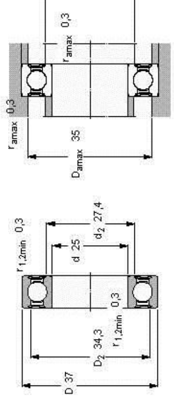
Deep Groove Ball Bearing, 61805

Deep groove ball bearings, single row, seal on both sides

Product information

Tolerances , see also text
Radial internal clearance , see also text
Recommended fits
Shaft and housing tolerances

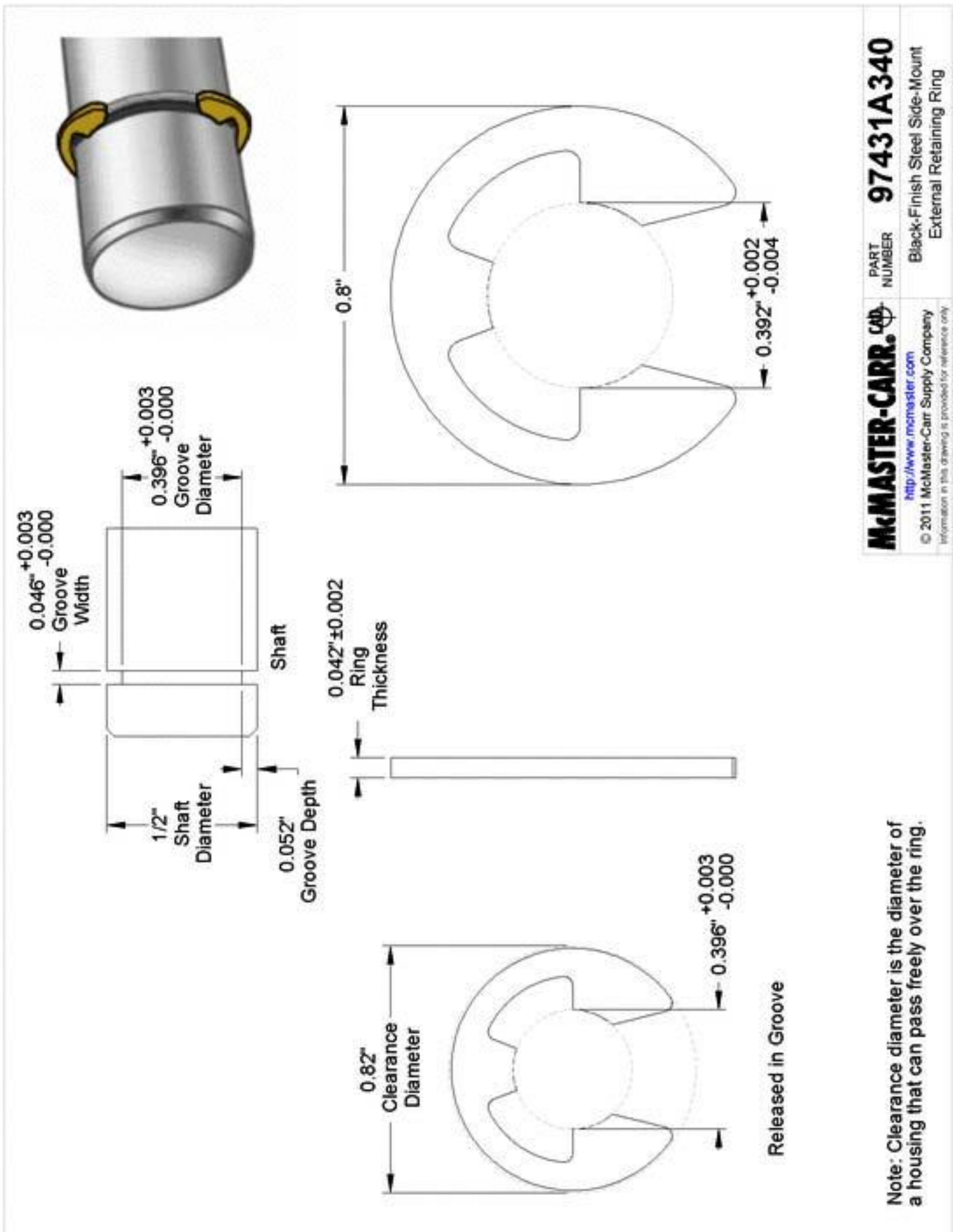
Principal dimensions			Basic load ratings		Fatigue load limit	Speed ratings		Mass	Designation
			dynamic	static		Reference speed	Limiting speed		
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	-
25	37	7	4,36	2,6	0,125	-	11000	0,022	61805-2RS1



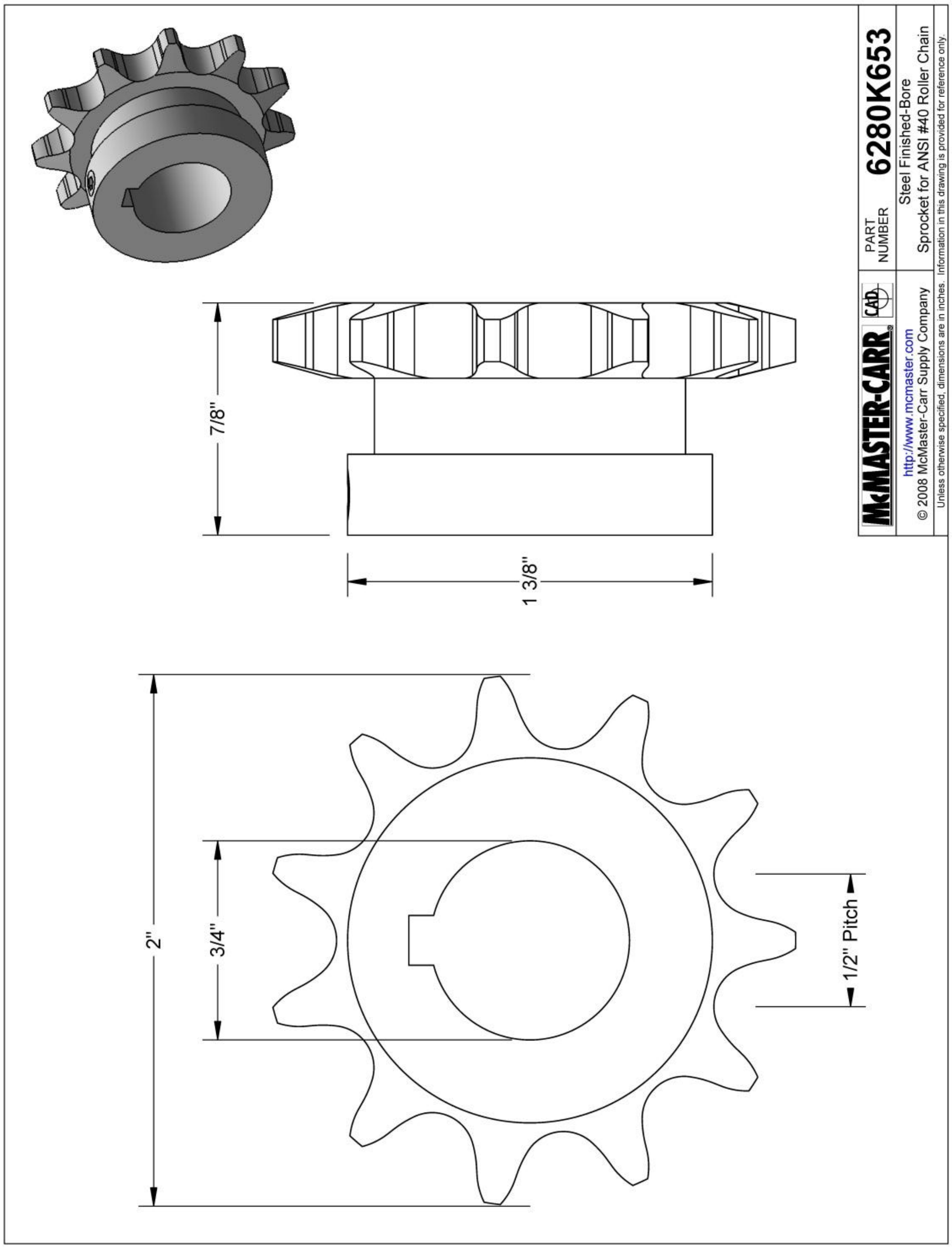
Calculation factors

k_r 0,015
f_0 14

Brake Rotor E-Clip



Pinion Sprocket



LMC D135Rag/S DC Electric Motor

L.M.C.
THE LYNCH MOTOR COMPANY

130 Table

Motor	No Load Current A	Torque Constant Nm/A	Speed Constant Rpm/V	Armature Resistance DC mΩ	Armature Inductance @ 15kHz μH	Armature Inertia Kgm^2	Peak Power kW	Peak Efficiency %	Peak Current A	Rated Power kW	Rated Speed Rpm	Rated Voltage V	Rated Current A	Rated Torque Nm
95	6	0.0631	138	32.5	14	0.0116	3	82	100	2.27	4968	36	75	4.35
95S	6	0.0631	138	32.5	14	0.0117	4	87	100	3.02	6624	48	75	4.35

170 Table

Motor	No Load Current A	Torque Constant Nm/A	Speed Constant Rpm/V	Armature Resistance DC mΩ	Armature Inductance @ 15kHz μH	Armature Inertia Kgm^2	Peak Power kW	Peak Efficiency %	Peak Current A	Rated Power kW	Rated Speed Rpm	Rated Voltage V	Rated Current A	Rated Torque Nm
126	18	0.055	140	185	8	0.0234	7	76	400	4.30	3360	24	240	12.2
127	5	0.12	68	24	23	0.0236	16	88	400	5.54	3264	48	140	16.2
D127	4	0.134	62	21	15	0.0236	21	88	400	7.10	3720	60	140	18.2

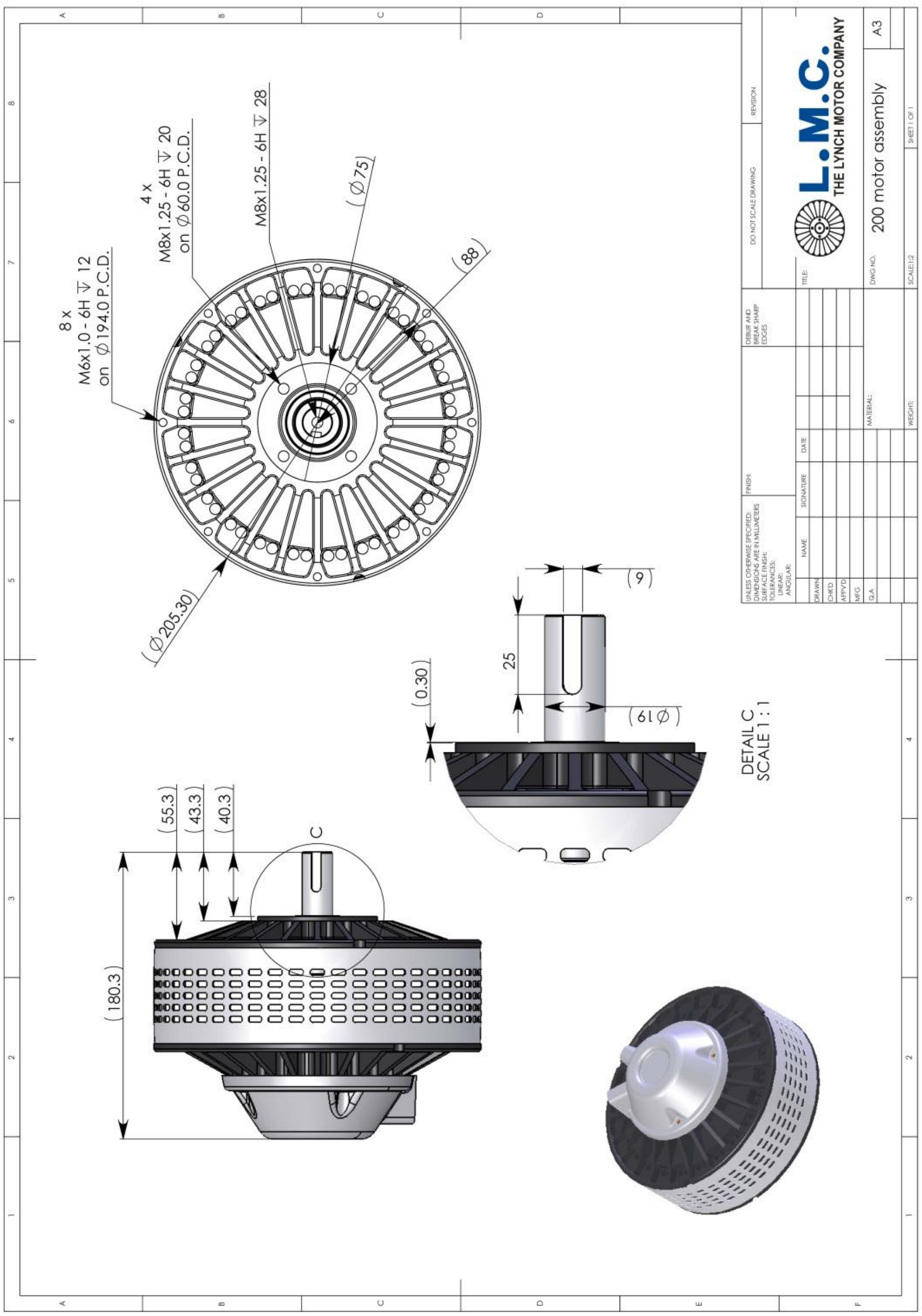
200 Table

Motor	No Load Current A	Torque Constant Nm/A	Speed Constant Rpm/V	Armature Resistance DC mΩ	Armature Inductance @ 15kHz μH	Armature Inertia Kgm^2	Peak Power kW	Peak Efficiency %	Peak Current A	Rated Power kW	Rated Speed Rpm	Rated Voltage V	Rated Current A	Rated Torque Nm
95	6	0.113	76	21.5	22	0.0238	18	89	400	6	3200	48	175	19
126	10	0.0737	105	175	6	0.0234	7.59	83	400	5.06	2520	24	270	19.2
127	5	0.15	54	22.5	23	0.0236	16.08	88	400	8.55	2592	48	215	31.5
D95B	6	0.14	66	20.5	11	0.0238	28.50	91	400	15.00	6000	72	200	30.0
D126	5	0.0748	100	138	5	0.0234	11.14	81	400	6.91	3600	36	250	18.3
D127	4	0.17	50	17.5	13	0.0236	25.38	90	400	12.56	3600	72	200	33.3
D135	3.5	0.185	45	16.75	16	0.0236	29.04	90	400	14.39	3780	84	200	36.4
D135RAG	7.36	0.207	42	16.95	16	0.0238	34.32	91	400	16.84	4032	96	200	39.88
D135RAGS	7.45	0.21	40	16.95	16	0.0238	36.00	91	400	18.00	4400	110	200	42.00

Any model of the LEM-200 can be made up into the 2X2 version this is 2 motors married together on a single shaft see installation drawing for details

New V-twin model available for increased power, larger motor in development for mid 2012

Torque Output of Motor; J [Nm] = Kt [Nm/A] * (Current [A] - No Load Current [A])


8 x
M6x1.0 - 6H ↧ 12
on ∅194.0 P.C.D.
4 x
M8x1.25 - 6H ↧ 20
on ∅60.0 P.C.D.
M8x1.25 - 6H ↧ 28
(∅75)
(88)
(∅205.30)
(9)
25
(0.30)
(∅19)
DETAIL C
SCALE 1 : 1
(55.3)
(43.3)
(40.3)
(180.3)
C
L.M.C.
THE LYNCH MOTOR COMPANY
200 motor assembly
A3

Traction Control/Data Acquisition Data Sheets

Miniature Absolute Magnetic Shaft Encoder Data Sheet

US DIGITAL **MA3** **Miniature Absolute Magnetic Shaft Encoder**

RoHS COMPLIANT

Description

The **MA3** is a miniature rotary absolute shaft encoder that reports the shaft position over 360 ° with no stops or gaps. The **MA3** is available with an analog or a pulse width modulated (PWM) digital output.

Analog output provides an analog voltage that is proportional to the absolute shaft position. Analog output is only available in 10-bit resolution.

PWM output provides a pulse width duty cycle that is proportional to the absolute shaft position. PWM output is available in 10-bit and 12-bit resolutions. While the accuracy is the same for both encoders, the 12-bit version provides higher resolution.

Three shaft torque versions are available. The standard torque version has a sleeve bushing lubricated with a viscous motion control gel to provide torque and feel that is ideal for front panel human interface applications.

The no torque added option has a sleeve bushing and a low viscosity lubricant (that does not intentionally add torque) for low RPM applications where a small amount of torque is acceptable.

The ball bearing version uses miniature precision ball bearings that are suitable for high speed and ultra low torque applications. The shaft diameter for ball bearing version option is 1/8" rather than 1/4".

Connecting to the **MA3** is simple. The 3-pin high retention snap-in 1.25mm pitch polarized connector provides for +5V, output, and ground.

Features

- Patent pending
- Miniature size (0.48" diameter)
- Non-contacting magnetic single chip sensing technology
- -40C to 125C. operating temperature range
- 10-bit Analog output - 2.6 kHz sampling rate
- 10-bit PWM output - 1024 positions per revolution, 1 kHz
- 12-bit PWM output - 4096 positions per revolution, 250 Hz

Mechanical Drawing

US DIGITAL
1400 NE 136th Avenue
Vancouver, Washington 98684, USA
info@usdigital.com
www.usdigital.com
Local: 360.260.2468
Toll-free: 800.736.0194

Rev. 120202102237

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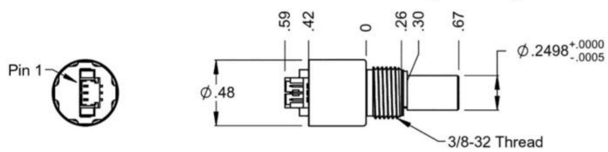
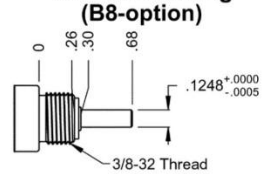
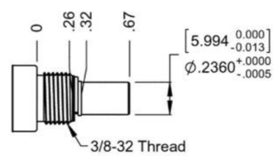
1/4" Sleeve Bushing (Default)



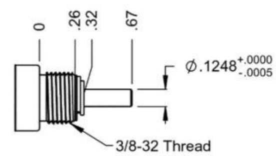
1/8" Ball Bearing (B8-option)



6mm Sleeve Bushing (M6-option)



1/8" Sleeve Bushing (8-option)



Mechanical

Specification	Sleeve Bushing	Ball Bearing
Moment of Inertia	4.1 x 10^-6 oz-in-s²	4.1 x 10^-6 oz-in-s²
Angular Accuracy	<0.5 deg. @ 25C	<0.5 deg. @ 25C
Angular Accuracy Over Temperature	<0.9 deg. @ -40C to 125C	<0.9 deg. @ -40C to 125C
Shaft Speed	100 RPM max. continuous	15,000 RPM max. continuous
Acceleration	10,000 rad/sec²	250,000 rad/sec²
Vibration	20G. 5Hz to 2kHz	20G. 5Hz to 2kHz
Shaft Torque	0.5 ± 0.2 in. oz. (**D** - torque option) 0.3 in. oz. max. (**N** - torque option)	0.05 in. oz. max.
Shaft Loading	2 lbs. max. dynamic* 20 lbs. max. static	1 lb. max.
Bearing Life	> 1,000,000 revolutions	$L_{10} = (18.3/F_r)^3$? Where L_{10} = bearing life in millions of revs, and F_r = radial shaft loading in pounds
Weight	0.46 oz.	0.37 oz.
Shaft Runout	0.0015 T.I.R. max.	0.0015 T.I.R. max.

* When a pulley, gear, or friction wheel drives the shaft, the Ball Bearing option is recommended instead of the Sleeve Bushing. The c hip that decodes position uses sampled data; note that there will be fewer readings per revolution as the speed increases. The formula f or number of readings per revolution is given by:

n = (60 / (rpm * 96 usec))

? only valid with negligible axial shaft loading

Environmental

Parameter	Value
Operating Temperature	-40C to +125C
Storage Temperature	-55C to +125C
ESD	± 2 kV max.
Humidity Non-condensing	5% to 85%

Mounting

Parameter	Value
Hole Diameter	0.375" +0.005 / -0.0
Panel Thickness	0.125" max.
Panel Nut Max. Torque	20 in.-lbs.

Materials

Parameter	Dimension
Shaft	Stainless
Bushing	Brass

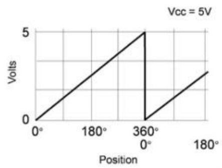
Magnetic Field Crosstalk

The **MA3** absolute encoder contains a small internal magnet, mounted on the end of the shaft that generates a weak magnetic field extending outside the housing of each encoder. If two **MA3** units are to be installed closer than 1 inch apart (measured between the center of both shafts), a magnetic shield, such as a small steel plate should be installed in between to prevent one encoder from causing small changes in reported position through magnetic field cross-talk.

Electrical

Parameter	Min.	Typ.	Max.	Units
Power Supply	4.5	5.0	5.5	Volts
Supply Current	-	16	20	mA
Power-up Time	-	-	50	mS

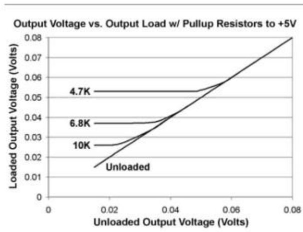
Analog Output Operation



Analog output is only available in 10-bit resolution. The analog output voltage is ratiometric to the power supply voltage and will typically swing within 15 millivolts of the power supply rails with no output load. This non-linearity near the rails increases with increasing output loads. For this reason, the output load impedance should be ≥ 4.7kΩ and less than 100pF. The graphs below show the typical output levels for various output loads when powered by a 5V supply.

Parameter	Min.	Typ.	Max.	Units
Position Sampling Rate	2.35	2.61	2.87	kHz
Propagation Delay	-	-	384	?S
Analog Output Voltage Maximum (1)	-	4.987	-	Volts
Analog Output Voltage Minimum (1)	-	0.015	-	Volts
Output Short Circuit Sink Current (2)	-	32	50	mA
Output Short Circuit Source Current (2)	-	36	66	mA
Output Noise (2)	160	220	490	μ Vrms
Output Transition Noise (3)	-	0.03	-	Deg. RMS

(1) With no output load. See graphs below.
(2) Continuous short to +5V or ground will not damage the**MA3**.
(3) Transition noise is the jitter in the transition between two adjacent position steps.



US DIGITAL
1400 NE 136th Avenue
Vancouver, Washington 98684, USA
info@usdigital.com
www.usdigital.com
Local: 360.260.2468
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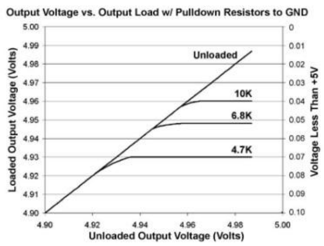
Rev. 120202102237



PWM Output Operation

The magnetic sensor chip in the **MA3** has an on-chip RC oscillator which is factory trimmed to 5% accuracy at room temperature (10% over full temperature range). This tolerance influences the sampling rate and pulse period of the PWM output. If only the PWM pulse w idth ton and the nominal pulse period is used to measure the angle, the resulting value also has this timing tolerance. However, this to lerance can be cancelled by measuring both ton and toff and calculating the angle from the duty cycle. Angular accuracy including non-linearity is within 0.5 deg. at 25C, but may increase to 0.9 deg. at high temperatures.

Parameter	Min.	Typ.	Max.	Units
PWM Frequency (-40C to 125C)				
10-bit	0.877	0.975	1.072	kHz
12-bit	220	244	268	Hz
Minimum Pulse Width				
10-bit	0.95	1.00	1.05	?S
12-bit	0.95	1.00	1.05	?S
Maximum Pulse Width				
10-bit	974	1025	1076	?S
12-bit	3892	4097	4302	?S
Internal Sampling Rate				
10-bit	9.38	10.42	11.46	kHz
12-bit	2.35	2.61	2.87	kHz
Propagation				
10-bit	-	-	48	?S
12-bit	-	-	384	?S
Output Transition Noise, 12-bit version (1)		0.03		Deg. RMS
Output Transition Noise, 10-bit version (1)		0.12		Deg. RMS
Output High Voltage (V OH: @4mA Source) (2)	Vcc -0.5	-	-	V
Output Low Voltage (V OL: @4mA Sink) (2)	-	-	0.4	V

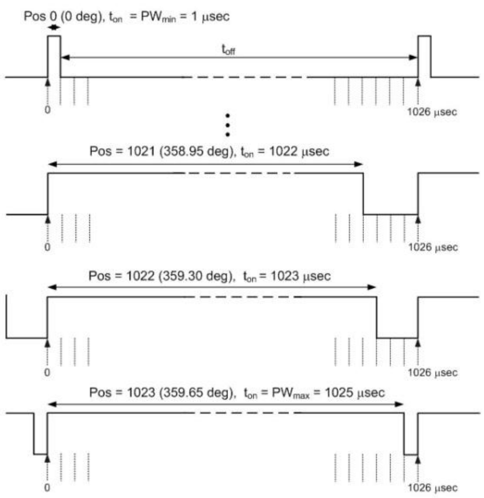
(1) Transition noise is the jitter in the transition between two adjacent position steps.
(2) Continuous short to +5V or ground will not damage the**MA3**.

10-bit PWM:

$x = ((t_{on} * 1026) / (t_{on} + t_{off})) - 1$

If $x <= 1022$, then Position = x

If $x = 1024$, then Position = 1023



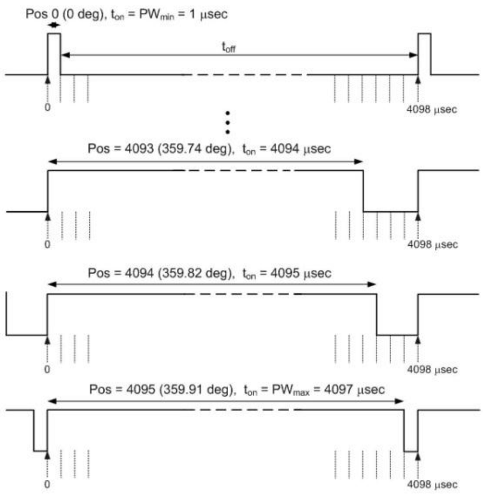
12-bit PWM:

$x = ((t_{on} * 4098) / (t_{on} + t_{off})) - 1$

If $x <= 4094$, then Position = x

If $x = 4096$, then Position = 4095

MA3 Miniature Absolute Magnetic Shaft Encoder



Pin-outs

Analog Output (MA3-A):

Pin	Name	Description
1	5	+5VDC power
2	A	Analog output
3	G	Ground

PWM Output (MA3-P10, MA3-P12):

Pin	Name	Description
1	5	+5VDC power
2	A	PWM output
3	G	Ground

US DIGITAL
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Vancouver, Washington 98684, USA
info@usdigital.com
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Rev. 120202102237

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Ordering Information

MA3 - [ ] - [ ] - [ ]

Interface	Shaft Diameter	Torque
A10 = *10-Bit Analog*	125 = *1/8"*	D = *Sleeve Bushing, Most Drag*
P10 = *10-Bit PWM*	236 = *6mm*	N = *Sleeve Bushing, Somewhat Lighter Drag*
P12 = *12-Bit PWM*	250 = *1/4"*	B = *Ball Bearing, Free Spinning (Least Drag)*

Rules

- Torque must be something other than B when Shaft Diameter is something other than 125

Notes

- Cables and connectors are not included and must be ordered separately.
- US Digital warrants its products against defects in materials and workmanship for two years. See complete warranty for details.

Base Pricing

Quantity	Price
1	$39.60
10	$34.69
50	$30.56
100	$26.28

- Add 17% per unit for **Interface** of 12-Bit PWM
- Add $1.00 per unit for **Shaft Diameter** of 6mm
- Add $5.80 per unit for **Torque** of Ball Bearing, Free Spinning (Least Drag)

Sparkfun Breakout Board Datasheet

sparkfun ELECTRONICS

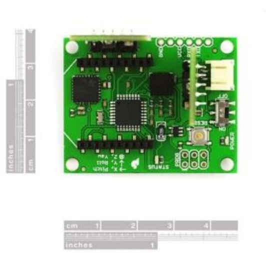
WEBSITE: sparkfun.com
6175 LONGBOW DRIVE, SUITE 200 BOULDER, COLORADO USA
ZIP CODE: 80301
[303] 284.0979 [GENERAL] P
443.0048 F

Atomic IMU - 6 Degrees of Freedom - XBee Ready

2009.3.25

Overview

Need a jump start on your robotics project, but strapped for cash? Is your junk food budget cutting into your autonomous monkey-shaving-robot plans? Is your quest for world domination constantly thwarted, week after week, because you always end up 25 bucks short of a 6DOF v4?



The 6DOF Atomic is a stripped-down IMU unit, designed to give good performance at a low price. The unit can run as a hard-wired UART interface (0-3.3V, 115200bps), or optionally with an XBee TM RF module, and is powered from a single LiPo (Lithium Polymer) cell.

The processor is an Atmel ATMega168TM running at 10MHz with 6 dedicated 10-bit ADC channels reading the sensors. Source code for the 6DOF Atomic is freely available and compiles with the free AVR GCC compiler.

The 6-DOF Atomic uses these sensors:

- Freescale MMA7260Q TM triple-axis accelerometer, settable to 1.5g, 2g, 4g or 6g sensitivity
- 3 ST Microelectronics LISY300AL TM single-axis, 300°/s gyros

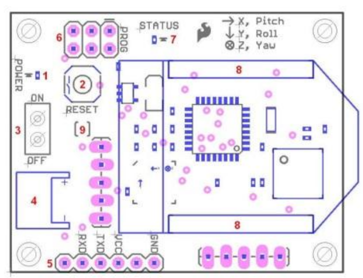
Electrical Specifications:

- Input voltage: 3.4V to 10V DC
- Current consumption: 24mA (75mA with XBee)
- Sensor bandwidth and resolution:
- LISY300AL Gyros: 88Hz, 0.977°/tick (ADC count)
- MMA7260Q Accelerometer:
- 350Hz, X and Y axes
- 150Hz, Z axis
- 0.00403g/tick @ 1.5g
- 0.00537g/tick @ 2g
- 0.0107g/tick @ 4g
- 0.0161g/tick @ 6g

The MMA7260Q accelerometer and the LISY300AL gyros have each been set up per their manufacturer's recommendations, i.e. internal clock suppression filters on their outputs.

These sensors are also internally temperature compensated. For a full description of the sensor specifications, please see the respective manufacturer's data sheets (available at http://www.sparkfun.com/).

Hardware Overview



1) Power indicator LED
2) Reset switch for the ATMega168
3) Power Switch
4) JST power connector for single Lipo cell
5) Serial port, 0-3.3V, 115200/8/1/N. The unit can optionally be powered with 3.4V to 10V from this port by closing the solder jumper marked "9" in Figure 1.
6) Programming port for ATMega168

 1

6dofAtomic_DS_090325 ADC

Atomic IMU - 6 Degrees of Freedom - XBee Ready

2009.3.25

7) Status LED, blinks at the sample rate when unit is sampling, off otherwise
8) Sockets for optional XBee RF module
9) Solder jumper, connects VCC on the serial port to the battery input

Setup

When you first power up the 6DOF Atomic, you will see the power indicator LED light up and status LED blink 5 times quickly. During normal sampling operations, the status LED will toggle on or off every 64 sampling cycles. When in the configuration menu, the status LED will be on continuously. If the device is not in the configuration menu and not sampling, the 6DOF Atomic is in its idle state and the status LED will be off (see the section on the configuration menu auto run mode for more details on this).

The 6DOF Atomic can run either as a hard-wired device or optionally with a pair of XBee RF modules. The unit uses the same UART for both operations so no firmware configuration is required to run in either mode. However, care must be taken not to connect both a hard line and an XBee at the same time as that will result in a UART conflict and may possibly damage the unit.

Hard line connection

If the user chooses a hard-wired connection to the Atomic, the serial UART lines can be accessed on the serial port header (marked "5" in Figure 1). TX, RX and ground are all that's required. The logic is 0 – 3.3V, but the lines are 5V tolerant. DO NOT connect RS-232 to the UART lines as you will most likely damage the unit.

The serial port header is lined up to match the SparkFun FTDI Basic Breakout – 5V (sku: DEV-09115). It provides the quickest way to get up and running with the 6DOF Atomic. The user only needs to install a right-angled male header into the serial port and plug them together.

XBee connection

To establish a connection over XBee modules, the user must first purchase the necessary hardware, such as two XBee 1mW Chip Antenna modules and an XBee Explorer USB (sku: WRL-08664 and WRL-08687). Then you only need to follow the beginning part of the Wireless XBee/AVR Bootloading tutorial (setting up the XBee modules, http://www.sparkfun.com/commerce/tutorial_info.php?tutorials_id=122), substituting 115200 baud for 19200 baud. Once the setup is complete, the XBee modules will operate transparently.

It is worth noting, however, that stopping the 6DOF Atomic over the XBee link at frequencies over 150Hz can be problematic. With so much data streaming over the link it becomes difficult to get the stop signal back to the 6DOF. The only way to stop in this situation is to hit the reset button or cycle power.

6DOF Atomic Mixer demo application

For a demonstration of the 6DOF Atomic's operation, the user can run the 6DOF Atomic Mixer program. With the 6DOF powered up and connected in its idle state, close any terminal programs open the 6DOF's serial port and start the mixer application. Select the port number to which the 6DOF (or XBee base unit) is connected, set the frequency and sensitivity, hit the start button and you're off and running.

The mixer program requires that all channels are active. If you have any trouble getting the program to work correctly, check that all channels are active in the configuration menu.

Using a Terminal and the Configuration Menu

Once the initial novelty of the mixer program wears off, the user may want to do something slightly more useful with the Atomic, like attach it to something and log a file with a terminal program. What follows is a brief description of the configuration menu as well as a functional description of the 6DOF Atomic's operation.

Operation from the Idle State

Upon reset and in its default configuration, the Atomic will check to see if it has been configured for "auto run" mode (more on that later). If it's not in auto run mode, it goes into an idle state waiting for input. Normally this idle state serves as the start up state for the mixer application. In this state, the following inputs have the following effects:

1) "%", ASCII 37, sets the accelerometer to 1.5g sensitivity
2) "&", ASCII 38, sets the accelerometer sensitivity to 2g

Atomic IMU - 6 Degrees of Freedom - XBee Ready

2009.3.25

3) " ' " (apostrophe), ASCII 39, sets the accelerometer sensitivity to 4g
4) " (", ASCII 40, sets the accelerometer sensitivity to 6g
5) ") ", ASCII 41, sets the sample frequency to 50Hz
6) " * ", ASCII 42, sets the sample frequency to 100Hz
7) "+", ASCII 43, sets the sample frequency to 150Hz
8) " , ", ASCII 44, sets the sample frequency to 200Hz
9) " - ", ASCII 45, sets the sample frequency to 250Hz
10) "#", ASCII 35, starts the unit running in binary mode with all channels active
11) " " (space), ASCII 32, stops the unit and returns it to the idle state (issuing another ASCII 32 will bring up the configuration menu)

Operation from this idle state will always be in binary output mode, but the user may select which channels are active. Also, configuration from this state as done in the mixer application will not be saved in memory, whereas settings from the actual configuration menu will be saved to memory for future use. At the same time, all but the active channel settings saved in memory have no effect on operation from the idle state.

It should be noted that the primary purpose of this idle state and mode of operation is to more easily interact with the Atomic mixer demonstration application, but there's no reason that a user's own application couldn't use it for a quick setup.

Operation from the Configuration Menu

6DOF Atomic setup, version 1.0
================================

1) View/edit active channel list
2) Change output mode, currently binary
3) Set Auto run mode, currently off
4) Set accelerometer sensitivity, currently 1.5g
5) Set output frequency, currently 100
6) Save settings and run unit

Active Channel List

Pressing "1" will bring up the active channel list:

1) Accel X = on
2) Accel Y = on
3) Accel Z = on
4) Pitch = on
5) Roll = on
6) Yaw = on

Press the number of the channel you wish to change, or press x to exit

To change a channel from active to inactive (or the reverse), just press the number of the channel you wish to change. It's a toggling function; pressing a number will bring up the full list again, but with the channel you wished to change in its opposite state. Press a few numbers and get a feel for it. Pressing "x" gets you back to the main menu.

Output Mode

Pressing "2" from the main menu will toggle the output mode from binary to ASCII and back again. What are these output modes, you ask?
In both output modes, the data from all active channels is framed by an "A" (ASCII 65) at the start and a "Z" (ASCII 90) at the end. Also in both modes, each channel is reported in exactly the sequence shown in the active channel list, with the addition of a sample count that immediately follows the "A" and precedes the first active measurement, which is to say:

1) Count
2) Accel X
3) Accel Y
4) Accel Z
5) Pitch
6) Roll
7) Yaw

The count is two bytes that comes as MSB-LSB, and will range from 0 to 32767. If any of the channels are selected as inactive, that data is omitted from the frame and subsequent data moves up in the report sequence.

In binary mode, each active channel report comes as 2 bytes: MSB and LSB, in that sequence, and they will always be between 0 and 1023 because we're reading from 10-bit ADC's. The width of the data

Atomic IMU - 6 Degrees of Freedom - XBee Ready

2009.3.25

frame in binary mode will be 4 bytes ("A", "Z", and count are always present) plus 2 bytes for each active measurement. So for all active channels the data frame will be 16 bytes wide.

In ASCII mode, the count and active measurements are reported in ASCII so it's easier to read with a terminal program, plus all measurements and the count are delimited with TAB characters (ASCII 9) as well as a carriage return and line feed at the end of the data frame. This makes data capture and importation into a spreadsheet a relatively simple matter.

Auto Run Mode

Pressing "3" from the main menu will toggle the auto run setting. If you intend to use the 6DOF Atomic in ASCII mode, set this to "on". If the auto run feature is off, the unit will always run from it's primary idle state, which means that it will always wait for a "#" to begin sampling and it will always run in binary mode.

One feature of auto run mode is that if the setting is active the Atomic will begin sampling immediately upon power up. Pressing the spacebar will bring up the configuration menu again.

Setting the Accelerometer Sensitivity

Pressing "4" from the main menu will bring up the following submenu:

Set to:

1) 1.5g
2) 2g
3) 4g
4) 6g

Just press the number which corresponds to your choice and the unit will revert to the main menu with the sensitivity changed.

Setting the Output Frequency

Pressing "5" from the main menu will allow you to change the sample frequency. Simply press "i" to increase or "d" to decrease, or "x" to revert to the main menu.

The minimum frequency setting is 10Hz, and there is no maximum setting. This allows the user to experiment with smaller data frames and higher sampling rates.

Save Settings and Run Unit

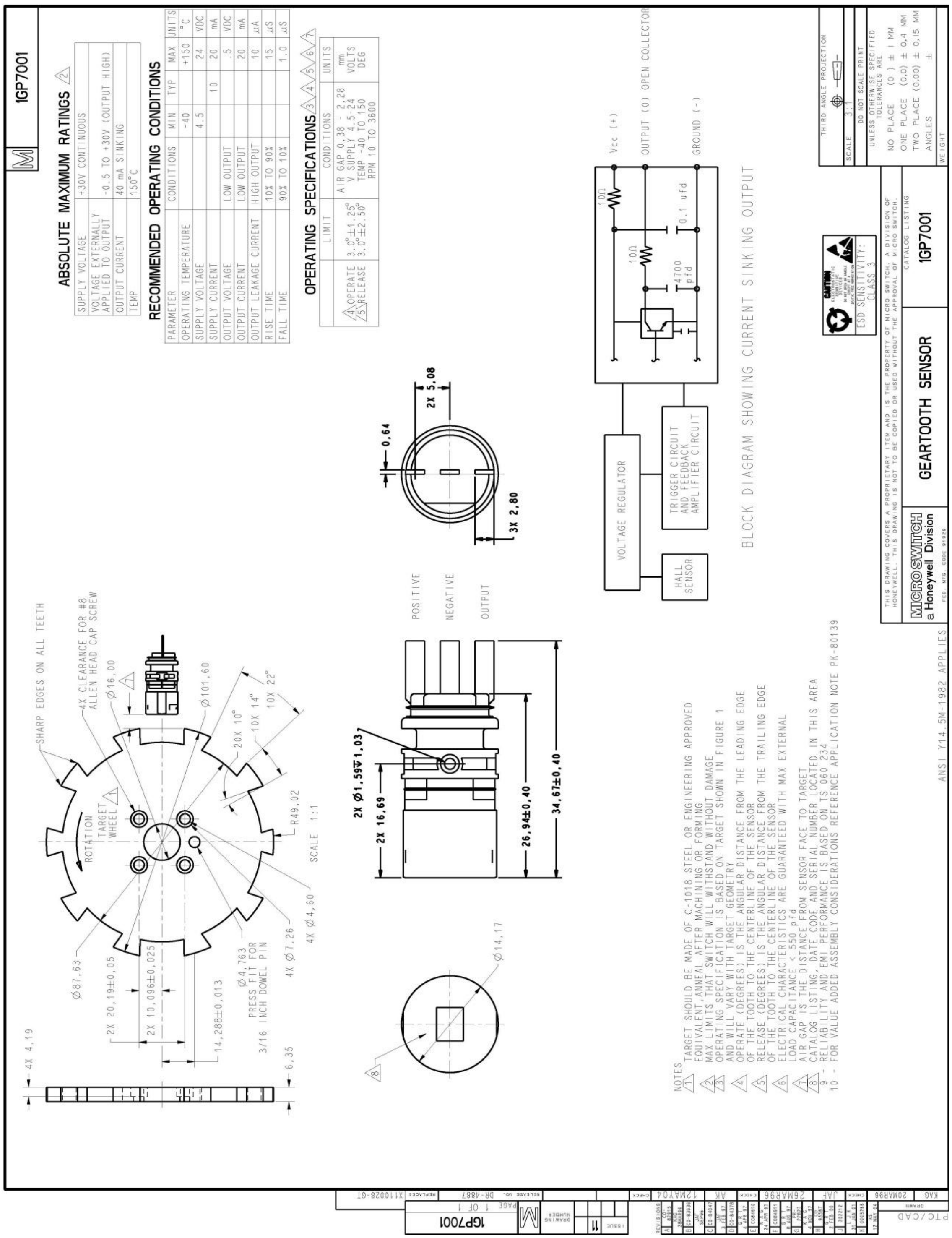
Pressing "9" from the main menu will save the current settings to flash and exit the configuration menu. If the auto run feature has been activated the unit will begin running immediately. If it has not been set, the unit will revert to the initial idle state and wait for additional input.

Bandwidth Considerations and Firmware

The 6DOF Atomic does not have any filtering in firmware, though there is enough memory left in the ATMega168 flash program space to implement filtering. The internally set output bandwidth of the MMA7260Q accelerometer is 350Hz for the X and Y axis, and 150Hz for the Z axis. There are also additional single-poll low-pass filters to reduce switching noise from the sensor with polls set at 1591Hz (recommended by Freescale). The internally set output bandwidth for the LISY300AL gyro sensors is 88Hz. Of course, it's a good idea for the user to consider these numbers when developing an application to ensure that the proper filtering is in place for whatever sampling rate is selected.

All source code and schematics for the 6DOF Atomic are freely available from the Atomic's product description page on http://www.sparkfun.com. Source code is compiled with GCC using WinAVR (http://winavr.sourceforge.net/). Programming is typically done with a AVR-PG1B (sku: PGM-00014, $12.95) along with our programming adapter (sku: BOB-08508).

Ferrous Gear Tooth Detector Datasheet



Appendix F – Detailed Supporting Analyses and Equations

Motor Mount FEA Analysis

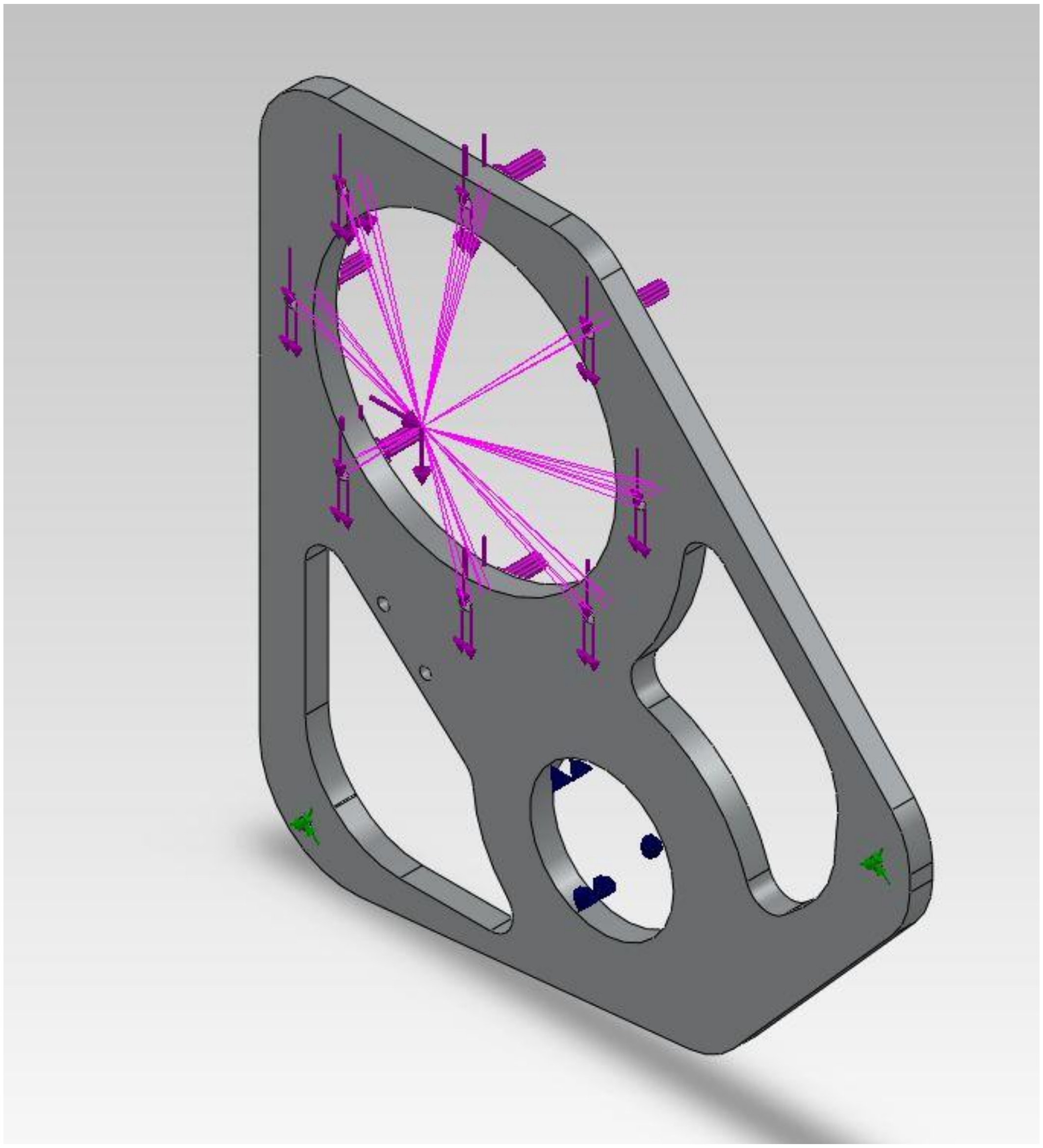


Figure 25: Motor Support FEA Forces Applied.

Note: The forces used for FEA analysis in Solidworks included a vertical weight force of 52 lbs (2g acceleration), a lateral force of 52 lbs (2g acceleration), and a remote chain tension of 1000 lbs (707.1 lbs vertical and 707.1 lbs horizontal).

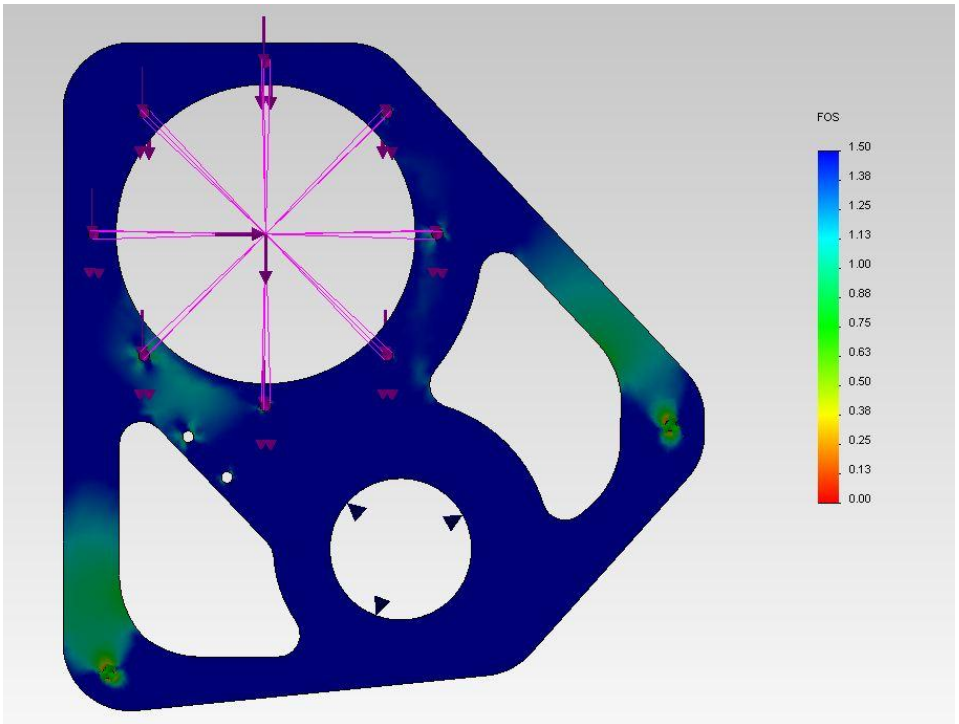


Figure 26: Motor Mount FEA analysis with Factor Of Safety (0.500" thick). Motor Mount fails at the bolt holes for mounting to the chassis.

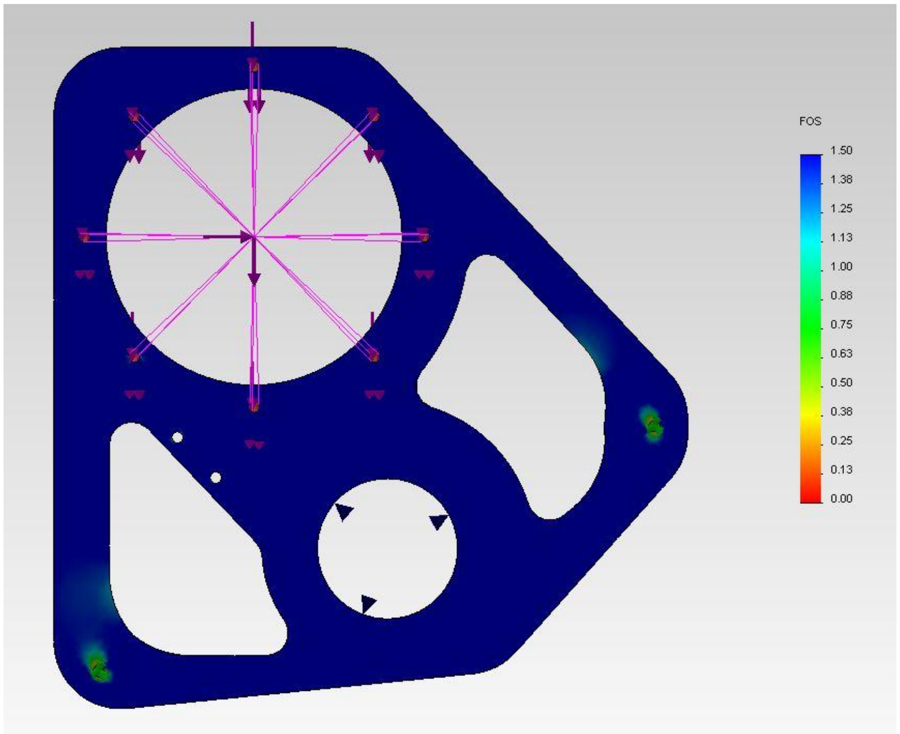


Figure 27: Motor Mount FEA analysis with Factor Of Safety (0.625" thick). The increase in thickness from 0.500" to 0.625" changes the Factor of Safety dramatically.

Note: Throughout the analysis of any parts using the FEA simulation tool in SolidWorks, when it encounters a bolt hole, it concentrates all of the stresses there. Thus, the larger the hole, the higher the minimum factor of safety. For our analysis, we ignored the stresses at the bolt holes, and just looked at the part as a whole.

When looking at the two different options of thicknesses of the plates, we notice that the thinner plate has a lower factor of safety when loaded than the thicker one. The weight difference is relatively small at approximately 1.14 pounds lighter for the 0.5" thick plate versus the 0.625" one. For this little weight savings, the team felt that it would be better to be cautious and use the larger one for our design.

Axle Stub

Fatigue Analysis

Table 4. Excel table used to compute Goodman shaft fatigue for the axle stubs. Various materials were compared and the Endurance limit values for aluminum were found in tables from the Department of Defense document, "Metallic Materials and Elements for Aerospace Vehicle Structures," (MIL-HDBK-5J, 31 January 2003).

Goodman Shaft Fatigue

							Calcs/Equations/ tables
Acc		Brake		d	1		Variables
Loads	lbins	Loads	lbins				Titles
Mma	0	Mma	0	kb	0.879		Results
Tma	744	Tma	1200	kc	1		
Maa	717	Maa	303	kd	1		
Taa	0	Taa	0	ke	0.753	99.90%	

Accel	Materials	Sut	Se'	ka	Se	Kf	Kfs		mean	alt		n
	6061	45000	20000	0.984606	14082.59	1	1	5.092958	0.028637	0.101828	0.66445	1.505004
	7075	78000	25000	0.851057	14082.59	1	1	5.092958	0.016521	0.101828	0.602746	1.659073
	1020	61000	33550	0.908346	14082.59	1	1	5.092958	0.021125	0.101828	0.626195	1.596945
	4130	97000	53350	0.803284	14082.59	1	1	5.092958	0.013285	0.101828	0.586265	1.705713

Brake	Materials	Sut	Se'	ka	Se	Kf	Kfs		mean	alt		n
	6061	45000	20000	0.984606	14082.59	1	1	5.092958	0.053333	0.043032	0.490784	2.037557
	7075	78000	25000	0.851057	14082.59	1	1	5.092958	0.030769	0.043032	0.375866	2.660524
	1020	61000	33550	0.908346	14082.59	1	1	5.092958	0.039344	0.043032	0.419538	2.383574
	4130	97000	53350	0.803284	14082.59	1	1	5.092958	0.024742	0.043032	0.345171	2.897117

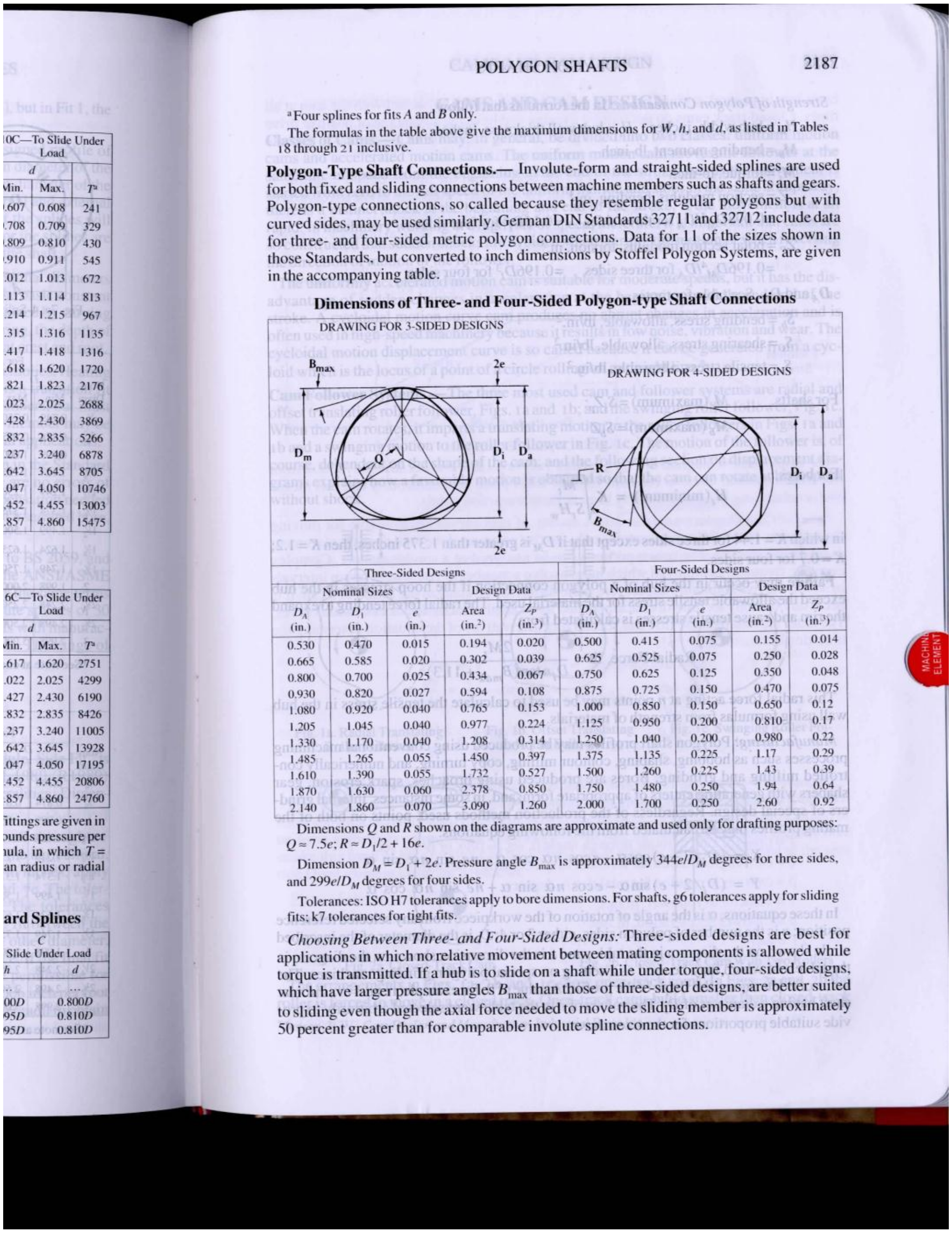
Female Geometric Spline

Tensile Stress Failure Analysis

Table 5. Tensile Failure Analysis Table for Female Geometric Spline on Sprocket Carrier, Brake Carrier, and CV Housing Plate. See next page for related geometry.

Material	Aluminum 6061 T6		
Modulus Z Polar (in^3)	0.3598		
Spline Diameter (in)	1.33		
D_m (in)	1.25		
	Sprocket Carrier	Brake Carrier	CV Housing Female Plate
Actual Torque Transmitted (in-lb_f)	2705.5	1380.4	2705.5
Hub			
Spline Length (in)	1.125	1.3	0.125
Wall Thickness (in)	0.33	0.33	0.33
Flange			
Spline Length (in)	0.5	0.2	0.5
Wall Thickness (in)	1.085	1.085	1.085
Allowable Stress (psi)	13000	13000	13000
Allowable Torque Transmitted			
Hub (in-lb_f)	768.1	887.5	3690.2
Flange (in-lb_f)	3690.2	1476.1	85.3
Total (in-lb_f)	4458.2	2363.6	3775.5
Factor of Safety	1.65	1.71	1.40

Polygon Shaft Geometry and Nomenclature



[a] Four splines for fits A and B only.

The formulas in the table above give the maximum dimensions for W, h, and d, as listed in Tables 18 through 21 inclusive.

Polygon-Type Shaft Connections.— Involute-form and straight-sided splines are used for both fixed and sliding connections between machine members such as shafts and gears. Polygon-type connections, so called because they resemble regular polygons but with curved sides, may be used similarly. German DIN Standards 32711 and 32712 include data for three- and four-sided metric polygon connections. Data for 11 of the sizes shown in those Standards, but converted to inch dimensions by Stoffel Polygon Systems, are given in the accompanying table.

Dimensions of Three- and Four-Sided Polygon-type Shaft Connections

Three-Sided Designs					Four-Sided Designs				
Nominal Sizes			Design Data		Nominal Sizes			Design Data	
D_A (in.)	D_1 (in.)	e (in.)	Area (in.^{2})	Z_P (in.^{3})	D_A (in.)	D_1 (in.)	e (in.)	Area (in.^{2})	Z_P (in.^{3})
0.530	0.470	0.015	0.194	0.020	0.500	0.415	0.075	0.155	0.014
0.665	0.585	0.020	0.302	0.039	0.625	0.525	0.075	0.250	0.028
0.800	0.700	0.025	0.434	0.067	0.750	0.625	0.125	0.350	0.048
0.930	0.820	0.027	0.594	0.108	0.875	0.725	0.150	0.470	0.075
1.080	0.920	0.040	0.765	0.153	1.000	0.850	0.150	0.650	0.12
1.205	1.045	0.040	0.977	0.224	1.125	0.950	0.200	0.810	0.17
1.330	1.170	0.040	1.208	0.314	1.250	1.040	0.200	0.980	0.22
1.485	1.265	0.055	1.450	0.397	1.375	1.135	0.225	1.17	0.29
1.610	1.390	0.055	1.732	0.527	1.500	1.260	0.225	1.43	0.39
1.870	1.630	0.060	2.378	0.850	1.750	1.480	0.250	1.94	0.64
2.140	1.860	0.070	3.090	1.260	2.000	1.700	0.250	2.60	0.92

Dimensions Q and R shown on the diagrams are approximate and used only for drafting purposes: $Q \approx 7.5e$; $R \approx D_1/2 + 16e$.

Dimension $D_M = D_1 + 2e$. Pressure angle B_{max} is approximately $344e/D_M$ degrees for three sides, and $299e/D_M$ degrees for four sides.

Tolerances: ISO H7 tolerances apply to bore dimensions. For shafts, g6 tolerances apply for sliding fits; k7 tolerances for tight fits.

Choosing Between Three- and Four-Sided Designs: Three-sided designs are best for applications in which no relative movement between mating components is allowed while torque is transmitted. If a hub is to slide on a shaft while under torque, four-sided designs, which have larger pressure angles B_{max} than those of three-sided designs, are better suited to sliding even though the axial force needed to move the sliding member is approximately 50 percent greater than for comparable involute spline connections.

Figure 28. Polygon Shaft Geometry reproduced from Machinery's Handbook: 28th Edition.

Design Life Calculations

Design Life Calculations

- Finite Life for bearings, aluminum components.
- Assuming 200 mile distance life.
  - Longest dynamic event for FHSAE is 22-km endurance race.
  - Car components typically redesigned biennially (every two years).
  - Good allowance for substantial testing of car without fear of drivetrain component failure.

Tire Size: $\underbrace{100}_{①}/\underbrace{85}_{②}\underbrace{R10}_{③}$ (PMT MS10024 family of tires)

① 100 ≡ tire section width (mm)
② 85 ≡ tire aspect ratio as a function of section width
③ R10 ≡ wheel diameter (in)

d_T ≡ tire outside diameter

$$d_T = (2)(0.85)(100\,mm)(1\,in/25.4\,mm) + (10\,in)$$

$$\underline{d_T = 16.693\ in}$$

Cycles

Cycles = distance / tire circumference

$$Cycles = \frac{200\,mi}{1}\cdot\frac{5280\,ft}{1\,mi}\cdot\frac{12\,in}{1\,ft}\cdot\frac{2}{16.693\,in}\cdot\frac{1\,rev}{2\pi}$$

Cycles = 241,635 revolutions → Cycles ≈ 2.5E5 revolutions

Assuming an overly conservative average speed of 10 mph, life in hours is calculated below:

$$t = dist/speed \Rightarrow t = 200\,mi/10\,mph \Rightarrow t = 20\ hours$$

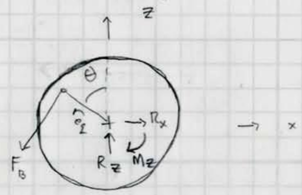
20 hours of life is still fairly low, so we are assuming a 100 hour design life for chains and other applications

$$\underline{h = 100\ hours}$$

Drivetrain Subsystem Load Calculations

Load Calculations

Under Braking:



$\theta = \tan^{-1}(1.64/1.44) \Rightarrow \theta = 48.7°$

$F_B = 471.13\ lb_f$

$r_{eq} = 2.93\ in$

$F_B \equiv$ Max braking force per brake rotor.

$r_{eq} \equiv$ Effective radius that the braking force acts upon the rotor.

$R_x \equiv$ Reaction force in the x.

$R_z \equiv$ Reaction force in the z.

$M_y \equiv$ Reaction moment in the y.

$\Sigma F_x = 0 = R_x - F_B \sin\theta \Rightarrow R_x = F_B \sin\theta \Rightarrow R_x = (471.13\ lb_f)\sin(48.7°)$

$\underline{R_x = 353.94\ lb_f}$

$\Sigma F_z = 0 = R_z - F_B \cos\theta \Rightarrow R_z = F_B \cos\theta \Rightarrow R_z = (471.13\ lb_f)\cos(48.7°)$

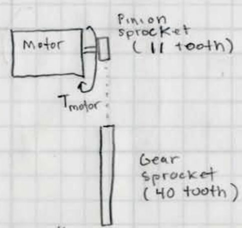
$\underline{R_z = 310.95\ lb_f}$

$\Sigma M = 0 = F_B r_{eq} - M_y \Rightarrow M_y = F_B r_{eq} \Rightarrow M_y = (471.13\ lb_f)(2.93\ in)$

$\underline{M_y = 1380.41\ in\text{-}lb_f}$

Load Calculations

Under Acceleration:



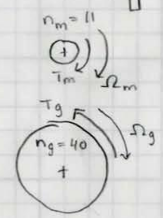
$T_{motor} \equiv$ Peak Torque output

$T_{motor} = K_t (I_{peak} - I_{No,Load})$

$T_{motor} = (0.21 \frac{N \cdot m}{A})(400\,A - 7.45\,A)$

$T_{motor} = (82.44\, N \cdot \cancel{m})(\frac{1\, ft \cdot lb_f}{1.356\, \cancel{N \cdot m}})$

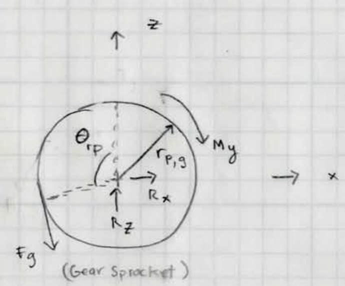
$T_{motor} = 61.95\, ft \cdot lb_f$



$\Sigma M = 0 = \frac{1}{n_g} T_g - \frac{1}{n_m} T_m$

$\Rightarrow T_g = {}^{n_g}/_{n_m}\, T_m \Rightarrow T_g = ({}^{40}/_{11})(61.95\, ft \cdot lb_f)$

$\underline{T_g = 225.45\, ft \cdot lb_f}$



$P = 0.5$ in , $N = 40$ teeth

Pitch Diameter:

$D_p = P / \sin(180° \div N)$

$D_p = \frac{0.5\text{ in}}{\sin(180°/40)}$

$D_p = 6.373$ in.

$r_{p,g} = D_p/2 \Rightarrow r_p = (6.373\text{ in})/2$

$r_{p,g} = 3.187$ in

$F_g \equiv$ Gear Sprocket Force from chain or Chain Tension

$r_{p,g} \equiv$ pitch radius of gear sprocket

$R_x \equiv$ Reaction force in the x.

$R_z \equiv$ Reaction force in the z.

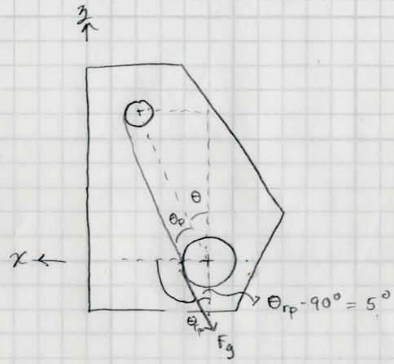
$M_y \equiv$ Reaction moment in the y.

$F_g = T_g / r_p \Rightarrow F_g = (225.45\, ft \cdot lb_f)(12\, in/ft)/(3.187\, in)$

$\Rightarrow F_g = 848.89\, lb_f$

Load Calculations

Under Acceleration (cont.)



$\theta_{rp} = \theta + \theta_D$

$\theta_{rp} = 23.2° + 71.8°$

$\underline{\theta_{rp} = 95°}$

$\Sigma F_x = 0 = R_x + F_g \sin 5° \Rightarrow R_x = -F_g \sin 5°$

$R_x = -(848.89\ lb_f)(\sin 5°) \Rightarrow \underline{R_x = -73.98\ lb_f}$

$\Sigma F_z = 0 = R_z - F_g \cos 5° \Rightarrow R_z = F_g \cos 5°$

$R_z = (848.89\ lb_f)(\sin 5°) \Rightarrow \underline{R_z = 845.66\ lb_f}$

$\Sigma M = 0 = F_g r_{p,g} - M_y \Rightarrow M_y = F_g r_{p,g}$

$M_y = (848.89\ lb_f)(3.187\ in) \Rightarrow M_y = \underline{2705.41\ in\text{-}lb_f}$

☆ Critical Loading Condition

$\underline{T_{accel} = 2705.41\ in\text{-}lb_f}$

Chain Analysis: Fatigue and Ultimate Tensile Strength

Chain Analysis:

Fatigue Strength:

Link Plate Limited:

$$H_1 = 0.004 N_1^{1.08} n_1^{0.9} p^{(3-0.07p)} \text{ hp}$$

$N_1 \equiv$ # of teeth in pinion sprocket = 11 teeth
$n_1 \equiv$ sprocket speed (rpm)

From manufacturing specs:

$$n = (\text{voltage})(40 \text{ RPM}/\text{volt}) \leq 5000 \text{ rpm}$$
$$n = (96 \text{ volt})(40 \text{ RPM}/\text{volt})$$
$$n = 3840 \text{ rpm}$$

$p \equiv$ chain pitch = 0.5 in

$$H_1 = (0.004)(11)^{1.08}(3840)^{0.9}(0.5)^{[3-(0.07)(0.5)]} \text{ hp}$$

$H_1 = 11.48$ hp *(highly conservative, based on $h = 15000$ hours where as our design life is $h = 100$ hours)*

Roller Limited:

$$H_2 = 1000\left[K_r\left(\frac{N_1}{n_1}\right)^{1.5} p^{0.8}\left(\frac{L_p}{100}\right)^{0.4}\left(\frac{15000}{h}\right)^{0.4}\right]$$

$L_p \equiv$ chain length in pitches

$$L_p = \frac{2C}{p} + \frac{N_1 + N_2}{2} + \frac{(N_2 - N_1)^2}{4\pi^2 C/p}$$

$C \equiv$ center to center distance = 7.616 in (see Under Accel. Calcs.)

$p = 0.5$ in

$N_1 = 11$ teeth ; $N_2 = 40$ teeth

$$L_p = \frac{(2)(7.616 \text{ in})}{0.5 \text{ in}} + \frac{(11+40)}{2} + \frac{(40-11)^2}{(4)(\pi^2)(7.616 \text{ in})/(0.5 \text{ in})}$$

$L_p = 57.36$ pitches ~ 58 pitches

$h = 100$ hrs

$K_R = 17$, (for chains 40-240, using 40 chain)

Chain Analysis:

Fatigue strength (cont.):

Roller Limited (cont.):

$$H_2 = 1000\left[(17)\left(\frac{11}{3840}\right)^{1.5}(0.5)^{0.8}\left(\frac{58}{100}\right)^{0.4}\left(\frac{15000}{100}\right)^{0.4}\right]$$

$H_2 = 8.93$ hp ☆ (see comment regarding H_1)

Maximum Tensile strength:

Maximum chain tension from motor: $T_{max} = 848.89$ lb_f
(See Load calculations: Under Acceleration)

Minimum tensile force, ANSI 40 chain: 3130 lb_f

$N = {}^{3130}/_{848.89} \Rightarrow \underline{\underline{N = 3.7}}$

☆ Based off of our low service life and the overly conservative nature of the fatigue strength calculations, we used maximum tensile strength as our design criteria ☆

$\therefore$ $\underline{\underline{\text{Chain: ANSI 40}}}$

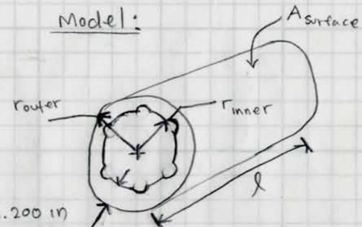
Constant Velocity (CV) Joint Housing

Von Mises Stress

CV Housing Design Analysis: C.V. ≡ Constant Velocity

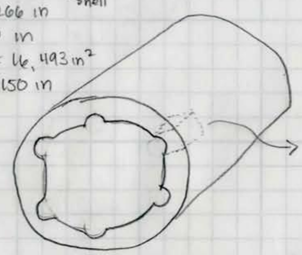
Von Mises Stress:

Model:

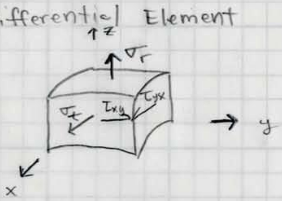


$r_{outer} = 1.200$ in
$r_{inner} = 1.050$ in
$r_{ball} = 0.266$ in
$\ell = 2.50$ in
$A_{surface} = 16.493 \text{ in}^2$
$t_{shell} = 0.150$ in

r_{inner} ≡ inner C.V. housing radius
r_{outer} ≡ outer C.V. housing radius
t_{shell} ≡ C.V. housing shell thickness
r_{bb} ≡ ball bearing radius

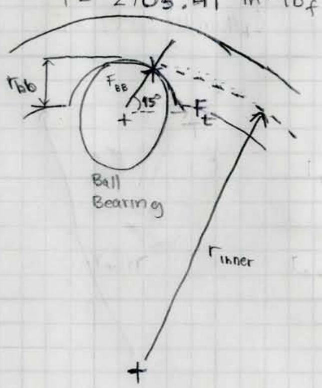


Differential Element



CV Housing transmits torque from axle stub to ball bearings / bearing cage in CV joint.

Torque ≡ Max torque transmitted through axle stubs.
T = 2705.41 in-lb_f (see Load calculations: Under Acceleration).



F_t ≡ force transmitted per ball bearing

$$F_t = \left(\frac{T}{r_{inner} - r_{bb}}\right) / 6 \text{ ball bearings}$$

$$F_t = \left(\frac{2705.41 \text{ in-lb}_f}{1.050 \text{ in} - 0.266 \text{ in}}\right) / 6$$

$$F_t = 574.8 \text{ lb}_f$$

F_{BB} ≡ force transmitted from ball bearing to CV housing.

$$F_{BB} = F_t / \cos(45°) \Rightarrow F_{BB} = 812.8 \text{ lb}_f$$

CV Housing Design Analysis:

Von Mises stress (cont.)

Material: 6061-T6 Aluminum
S_y : 40000 psi

σ_t Calculations:

$$P_{i,avg} = \frac{6 * F_{BB}}{A_{surface}} \Rightarrow P_{i,avg} = \frac{(6)(812.8\ lb_f)}{(16.493\ in^2)}$$

$P_i = 295.7$ psi

$$\sigma_t = \frac{r_i^2 P_i \left(1 + {r_o^2}/{r_i^2}\right)}{r_o^2 - r_i^2} \Rightarrow \sigma_t = \frac{(1.050\ in)^2(295.7\ psi)\left[1 + \frac{(1.200\ in)^2}{(1.050\ in)^2}\right]}{\left[(1.200\ in)^2 - (1.050\ in)^2\right]}$$

$\sigma_t = 2227.6$ psi

σ_r Calculations:

$P_{i,avg} = 295.7$ psi

$$\sigma_r = \frac{r_i^2 P_i \left(1 - {r_o^2}/{r_i^2}\right)}{r_o^2 - r_i^2} \Rightarrow \sigma_r = \frac{(1.050\ in^2)(295.7\ psi)\left[1 + \frac{(1.200\ in)^2}{(1.050\ in)^2}\right]}{\left[(1.200\ in)^2 - (1.050\ in)^2\right]}$$

$\sigma_r = -295.7$ psi

τ_{xy} Calculations:

$F_t = 574.8\ lb_f$

$K_{fs} = 2$ (conservative)

$$\tau_{xy} = \left[\frac{F_t\ K_{fs}}{A_{surface}}\right] * 6 \Rightarrow \tau_{xy} = \left[\frac{(574.8\ lb_f)(2)}{16.493\ in^2}\right] * 6$$

$\tau_{xy} = 418.2$ psi

3-0235 — 50 SHEETS — 5 SQUARES
3-0236 — 100 SHEETS — 5 SQUARES
3-0237 — 200 SHEETS — 5 SQUARES
3-0137 — 200 SHEETS — FILLER

COMET

CV Housing Design Analysis:

Von Mises Stress (cont.)

$$\sigma' = \left(\sigma_t^2 - \sigma_t \sigma_r + \sigma_r^2 + 3 * \tau_{xy}^2\right)^{1/2}$$

$$\sigma' = \left[(2227.6\ psi)^2 - (2227.6\ psi)(-295.7\ psi) + (-295.7\ psi)^2 + 3(418.2\ psi)^2\right]^{1/2}$$

$$\sigma' = 2496.6\ psi$$

$$N = S_y / \sigma' \Rightarrow N = (40{,}000\ psi)/(2496.6\ psi)$$

$$\underline{\underline{N = 16.0}}$$

Hertz Contact Stress

CV Housing Design Analysis:

Hertzian Contact Stress

$d_{BB} \equiv$ diameter of ball bearing

$r_{housing} \equiv$ radius of housing inner surface in contact w/ ball bearing

$d_{housing} = 2 * r_{housing}$

$d_{bb} = 0.531$ in

$d_{housing} = 0.536$ in

Material Properties:

Ball Bearing

Material: 52100 Chrome Alloy Steel

E: 30.5 E6 psi

ν: 0.3

CV Housing

Material: 6061-T6 Aluminum

E: 10.2 E6 psi

ν: 0.35

S_y: 40,000 psi

Loads:

$F_{bb} = 812.8\ lb_f$ (See Von Mises Stress Analysis)

Analysis:

Contact Radius:

$$\left\{ (3F_{bb}/8)\left[\frac{1-\nu_{Al}^2}{E_{Al}} + \frac{1-\nu_{bb}^2}{E_{bb}}\right] \Big/ \left[\frac{1}{d_{bb}} - \frac{1}{d_{housing}}\right] \right\}^{1/3}$$

Contact Radius = 0.124 in

$$P_{max} = \frac{3F_{bb}}{2\pi r_{contact}^2} \Rightarrow P_{max} = \frac{(3)(812.8\ lb_f)}{(2)(\pi)(0.124\ in)^2}$$

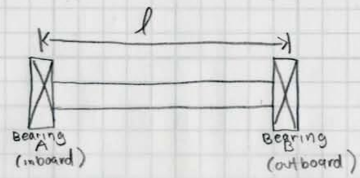
$P_{max} = 25,352$ psi

$N = S_y / P_{max} \Rightarrow$ $\underline{\underline{N = 1.58}}$

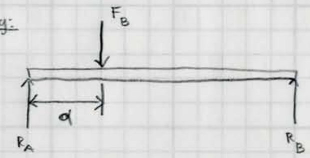
Bearing Selection Calculations

Bearing Selection Calculations:

Bearing A: In center support
Bearing B: In motor support



Braking:



$\ell = 3.775$ in
$a = 0.680$ in
$F_B = 471.13\ lb_f$

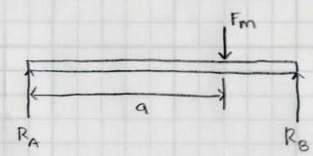
$\Sigma M_A = 0 \Rightarrow R_B \ell - F_B a = 0 \Rightarrow R_B = \frac{a}{\ell} F_B$

$R_B = (0.680\text{ in}/3.775\text{ in})(471.13\ lb_f) \Rightarrow \underline{R_B = 84.87\ lb_f}$

$\Sigma F_y = 0 \Rightarrow R_A + R_B - F_B = 0 \Rightarrow R_A = F_B - R_B$

$R_A = (471.13 - 84.87)\ lb_f \Rightarrow \underline{R_A = 386.26\ lb_f}$

Acceleration:



$a = 2.875$ in
$F_m = 848.89\ lb_f$

$\Sigma M_A = 0 \Rightarrow R_B \ell - F_m a = 0 \Rightarrow R_B = \frac{a}{\ell} F_B$

$R_B = (2.875\text{ in}/3.775\text{ in})(848.89\ lb_f) \Rightarrow \underline{R_B = 646.51\ lb_f}$

$\Sigma F_y = 0 \Rightarrow R_A + R_B - F_m = 0 \Rightarrow R_A = F_B - R_B$

$R_A = (848.89 - 646.51)\ lb_f \Rightarrow \underline{R_A = 202.38\ lb_f}$

For Bearing A: Max Radial Load = 386.26 lb_f
For Bearing B: Max Radial Load = 646.51 lb_f

Bearing selection Calculations:

Bearing A:

$F_a = 0$ (CV joints can plunge ∴ axial load negligible)

$F_r = 386.26\ lb_f$

Bore: 25 mm (SKF Bearing 61805, $C_0 = 2.6\ kN$, $C_{10} = 4.36\ kN$)

$C_0 = 2.6\ kN$

$F_a/C_0 = 0/2.6\ kN \Rightarrow e = 0.19$

$F_a/VF_R = 0/(1.2)(386.26\ lb_f) \Rightarrow X_1 = 1,\ Y_1 = 0$

$F_e = X_1 V F_R + \cancel{Y_1 F_a}^{\,0} \Rightarrow F_e = (1)(1.2)(386.26\ lb_f)\left(\frac{4.448\ kN}{1000\ lb_f}\right)$

$F_e = 2.06\ kN < C_0$ ✓

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}}\right]^{1/a}$$

$R_D = 0.99$

$x_D = L_D/L_R = \frac{241{,}635\ rev}{1E6\ rev} \Rightarrow x_D = 0.242$

$x_0 = 0.02$

$(\theta - x_0) = 4.439$

$a = 3$

$b = 1.483$

$a_f = 1.5$ (moderate impact)

$$C_{10} = (1.5)(386.26\ lb_f)\left(\frac{4.448\ kN}{1000\ lb_f}\right)\left[\frac{0.242}{0.02 + (4.439)(1-0.99)^{1/1.483}}\right]^{1/3}$$

$C_{10} = 1.03\ kN < 4.36\ kN$ (SKF rated) ✓

3-0235 — 50 SHEETS — 5 SQUARES
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3-0237 — 200 SHEETS — 5 SQUARES
3-0137 — 200 SHEETS — FILLER

COMET

Bearing Selection Calculations:

Bearing B:

$F_a = 0$ (CV joints can plunge ∴ axial load negligible)

$F_r = 646.51 \text{ lb}_f \left| \frac{4.448 \text{ kN}}{1000 \text{ lb}_f} \right. \Rightarrow F_r = 2.87 \text{ kN}$

Bore: 35 mm (SKF Bearing 16007, $C_0 = 8.15 \text{ kN}$, $C_{10} = 13 \text{ kN}$)

$F_a / C_0 = 0 / 2.87 \text{ kN} \Rightarrow e = 0.19$

$F_a / VF_R = \frac{0}{(1.2)(2.87 \text{ kN})} \Rightarrow X_1 = 1, \ Y_1 = 0$

$F_e = X_1 V_1 F_R + \cancel{Y_1 F_a}^{\,0} \Rightarrow F_e = (1)(1.2)(2.87 \text{ kN})$

$F_e = 3.44 \text{ kN} < C_0$ ✓

$$C_{10} = a_f F_D \left[\frac{x_D}{x_0 + (\theta - x_0)(1 - R_D)^{1/b}} \right]^{1/a}$$

$$C_{10} = (1.5)(2.87 \text{ kN}) \left[\frac{0.242}{0.02 + (4.439)(1 - 0.99)^{1/1.483}} \right]^{1/3}$$

$C_{10} = 1.73 \text{ kN} < 13 \text{ kN}$ (SKF C_{10} rating) ✓

Appendix G – Gantt Charts

Gantt Chart Fall Quarter

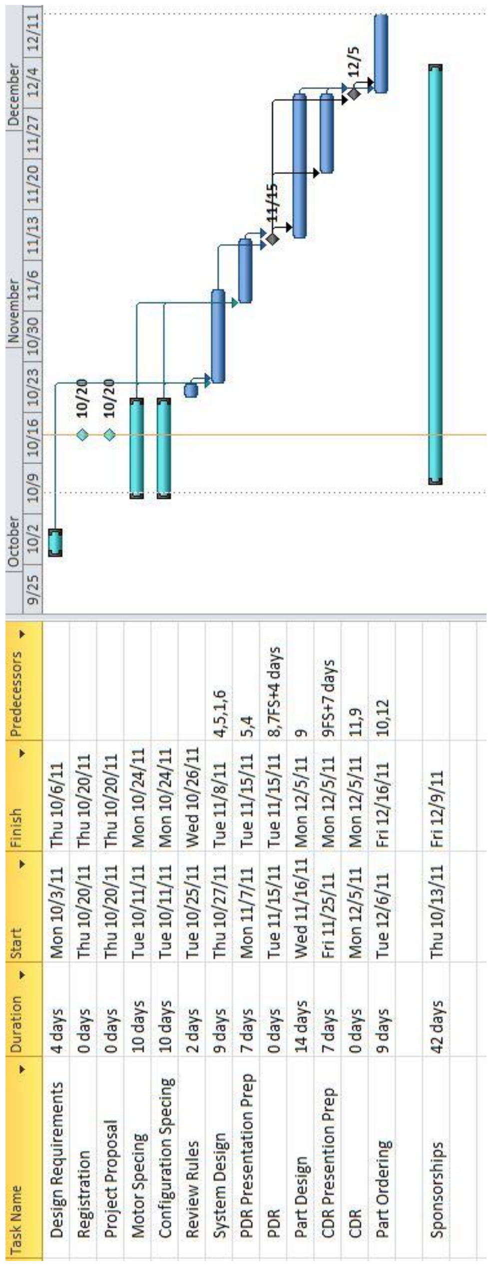

Figure 29: Gantt Chart for Formula Hybrid team for Fall Quarter.

Note: GANTT Chart for Fall 2011 with 2012 SAE Formula Hybrid drivetrain design timeline with tasks and milestones.

Gantt Chart Winter Quarter

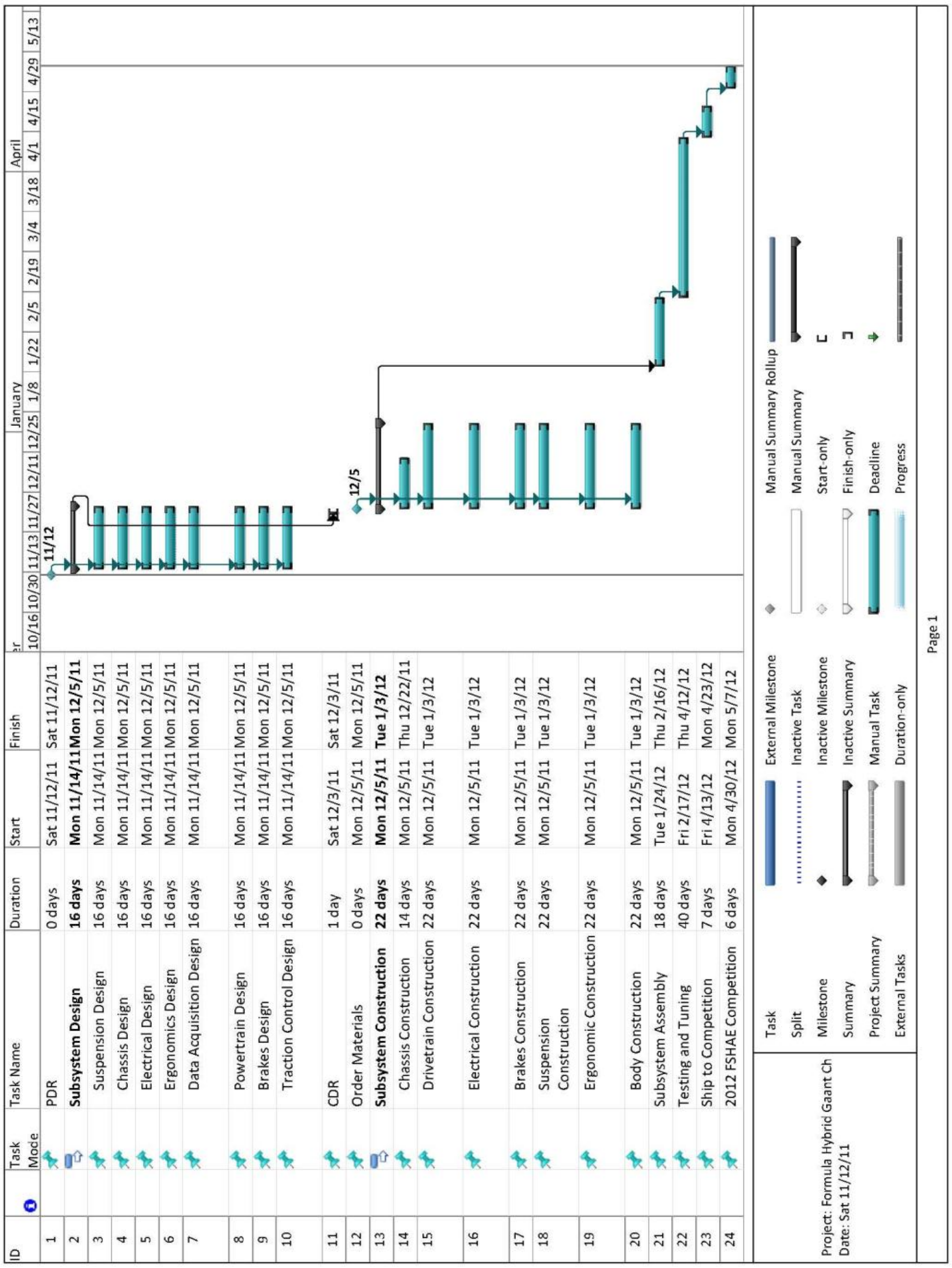

Figure 30: Gantt Chart for Formula Hybrid team for Winter Quarter.

Note: GANTT Chart for Winter 2012 with 2012 SAE Formula Hybrid Team design timeline with tasks and milestones.

Gantt Chart Spring Quarter

Figure 31. Gantt Chart for Formula Hybrid team for Spring 2012.