Formula Hybrid Drivetrain Design

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June 2012

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Table of Contents

Table of Contents ........................................................................................................................................ ii
List of Figures ................................................................................................................................................ v
List of Tables ................................................................................................................................................ vi
Executive Summary .................................................................................................................................. vii
Chapter 1: Introduction .............................................................................................................................. 1
Chapter 2: Background ............................................................................................................................... 1
  Requirements / Specifications .................................................................................................................. 2
  Method of Approach ............................................................................................................................. 4
Chapter 3: Design Development .................................................................................................................. 5
  Conceptual Design 1: Single Motor with Transmission ........................................................................... 5
  Conceptual Design 2: Dual Motor with Mechanical Differential ............................................................. 6
  Conceptual Design 3: Dual Motor with Electronic Differential .............................................................. 7
  Traction Control for Conceptual Design 3 ............................................................................................... 8
  Conceptual Model Design Comparisons ............................................................................................... 10
    Drivetrain Design Requirements Comparison ................................................................................... 10
    Drivetrain Decision Pros and Cons ............................................................................................... 10
    Drivetrain Model Design Decision ............................................................................................... 11
Chapter 4: Description of the Final Design ............................................................................................... 14
Chapter 5: Manufacturing .......................................................................................................................... 16
  Motor Mounts .................................................................................................................................... 17
  Central Support .................................................................................................................................. 18
  Axle Stubs ......................................................................................................................................... 19
  Sprocket Carriers .............................................................................................................................. 20
  Brake Carriers .................................................................................................................................. 20
  CV housings and CV plates ............................................................................................................... 21
  Half-Shafts ...................................................................................................................................... 22
  Brake Rotors ................................................................................................................................... 23
Chapter 6: Design Verification Plan (Testing) .......................................................................................... 24
  Component Testing & Evaluation: ....................................................................................................... 24
Subsy...
Mechanical Systems.................................................................................................................54
Traction Control/Data Acquisition ............................................................................................55

Appendix E – Vendor Component Specifications/Data Sheets ................................................56
Mechanical Systems Data Sheets.............................................................................................56
Traction Control/Data Acquisition Data Sheets .....................................................................67

Appendix F – Detailed Supporting Analyses and Equations ..................................................80
Motor Mount FEA Analysis ........................................................................................................80
Axle Stub ..................................................................................................................................83
Female Geometric Spline ...........................................................................................................84
Design Life Calculations ...........................................................................................................86
Drivetrain Subsystem Load Calculations ................................................................................87
Chain Analysis: Fatigue and Ultimate Tensile Strength ..........................................................90
Constant Velocity (CV) Joint Housing ......................................................................................91
Bearing Selection Calculations ................................................................................................96

Appendix G – Gantt Charts .......................................................................................................99
Gantt Chart Fall Quarter ...........................................................................................................99
Gantt Chart Winter Quarter ......................................................................................................100
Gantt Chart Spring Quarter .....................................................................................................101
List of Figures

Figure 1: Design Process Flow Chart for 2012 SAE Formula Hybrid drivetrain design ................. 4
Figure 2: Single Motor with Transmission Conceptual Model (3/4 view) ..................................... 5
Figure 3: Single Motor with Transmission Conceptual Model (rear view) ................................. 5
Figure 4: Dual Motors with Different Gearing and Mechanical Differential (3/4 view) .......... 6
Figure 5: Dual Motors with Different Gearing and Mechanical Differential (rear view) ............ 6
Figure 6: Dual Motors with the Same Gearing and Electronic Differential (3/4 view) ............ 7
Figure 7: Dual Motors with the Same Gearing and Electronic Differential (rear view) .......... 7
Figure 8: Diagram of Traction Control System ........................................................................ 9
Figure 9: Final Drivetrain Assembly Design (3/4 view) ............................................................. 12
Figure 10: Final Drivetrain Assembly Design (rear view) .......................................................... 13
Figure 11: Final Drivetrain Assembly Design (partially exploded view) .................................... 13
Figure 12: Closer View of Final Drivetrain Assembly (3/4 view) ............................................... 14
Figure 13: Final Drivetrain Assembly (bottom view) ................................................................. 15
Figure 14: Final Drivetrain Layout (rear view) ........................................................................ 15
Figure 15: Finished and disassembled drivetrain components prior to installation on the car ... 16
Figure 16: Raw materials prior to machining parts for the drivetrain subsystem. ...................... 16
Figure 17: Left Motor mount being machined on the Haas VF-2 vertical CNC mill .................... 17
Figure 18: Above: Center support being machined on the Haas VF-2 vertical CNC mill ............ 18
Figure 19: Left: Partially CNC machined axle stub (front-top view) ........................................... 19
Figure 20: Half shafts air cooling in furnace after heat treatment at 1600 °F for one hour ........ 22
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 .......................................................................................................................... 23
Figure 22: Benchmark Competitors QFD ............................................................................. 31
Figure 23: Conceptual Design QFD ....................................................................................... 32
Figure 24: Basic electrical schematic for powering the drivetrain subsystem overlayed on top of a model of the Formula Hybrid car ........................................................................ 33
Figure 25: Motor Support FEA Forces Applied ................................................................. 80
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 ........................................................................ 81
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 .................................. 82
Figure 28: Polygon Shaft Geometry reproduced from Machinery’s Handbook: 28th Edition .... 85
Figure 29: Gantt Chart for Formula Hybrid team for Fall Quarter ......................................... 99
Figure 30: Gantt Chart for Formula Hybrid team for Winter Quarter .................................... 100
Figure 31. Gantt Chart for Formula Hybrid team for Spring 2012 ........................................ 101
List of Tables

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. .......................................................................................................................................................... 2

Table 2: Conceptual Design Comparison. .......................................................................................................................... 10

Table 3: Pros and Cons of the three concepts for the 2012 Formula Hybrid drivetrain design... 10

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).................................................................................................................. 83

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. ................................................................. 84
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, whereas 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.

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

![Design Process Flow Chart](image_url)

*Figure 1: Design Process Flow Chart for 2012 SAE Formula Hybrid drivetrain design.*
Chapter 3: Design Development

*Conceptual Design 1: Single Motor with Transmission*

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**

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.

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.
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.

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

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).](image)

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.

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 axle, the axle stub is a very critical part which most teams make from hardened steel and a higher factor of safety was desired.
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 part 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.

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.

Figure 17. Left Motor mount being machined on the Haas VF-2 vertical CNC mill.
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](image-url)
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 straightforward 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.

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.

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

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


Appendices

Appendix A – Quality Function Deployments (QFDs)
- Benchmark Competitors QFD
- Conceptual Design QFD

Appendix B – Drivetrain Subsystem Drawing Package
- Electrical System and Wiring Overview
- Assembly Drawings/Bill Of Materials
- Detailed Part Drawings

Appendix C – Design Verification Plan and Report
- Design Verification Report
- Design Verification Plan
- Component Testing & Evaluation
- Subsystem Assembly
- Subsystem Testing & Evaluation

Appendix D – List of Vendors, Contact Information, and Pricing
- Mechanical Systems
- Traction Control / Data Acquisition

Appendix E – Vendor Component Specifications and Data Sheets
- Mechanical Systems Materials
- Traction Control / Data Acquisition

Appendix F – Detailed Supporting Analyses and Equations

Appendix G – Gantt Charts
- Gantt Chart Fall Quarter
- Gantt Chart Winter Quarter
- Gantt Chart Spring Quarter
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.

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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.

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

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Cal Poly Mechanical Engineering

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Drawn By: Zachary McFarland

Scale: 1:4

Date: 02/01/12

Title: Core Drivetrain

Units of Tech

Net Asb.

SAE Formula Hybrid
Brake Rotor Carrier Assembly
Detailed Part Drawings

Left Motor Mount Drawing
Right Motor Mount

All Radii are .150" unless otherwise noted
All Holes 1/4" unless otherwise noted
Gear Sprocket Carrier Drawing

Cal Poly Mechanical Engineering

SAE Formula Hybrid

Drawn By: Zachary McFetridge

Title: Sprocket Carrier

Date: 02/20/12

Scale: 1:1

Units: Inch

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Page #:

Rev B

See Spline
Geometry

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Ø 2.750

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0.400

0.500

R 2.000

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Brake Rotor Carrier Drawing
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Geometric Spline Geometry

Spline Geometry from Mechanical Engineering Handbook
Three Sided Design where Da = 1.380m

R = 3.65m

3

120°
CV Housing Female Spline Plate
Chain Guard
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**Notes:**
- All tests are performed under standard atmospheric conditions.
- Test results are reviewed and approved by the team.
- Any discrepancies are investigated and addressed.
- Final test report is submitted to the design review board.

**Test Plan:**
- Components are assembled and tested individually.
- Integration testing is conducted to ensure proper functioning.
- Performance benchmarks are set and exceeded.

**Reporting:**
- Test results are recorded and analyzed.
- Test reports are prepared and distributed.
- Feedback is provided to the design team for improvements.

**Engineer:**
- Component/Assembly: Drivetrain
- Report Date: 6/1/2012
- Sponsor: FH SAE Drivetrain

---

The design validation plan and report for the ME429 project include a detailed testing plan and results. Each test is documented with specifications, criteria, and responsible parties, ensuring thorough evaluation of the drivetrain components. The results are reported in a standardized format, allowing for easy review and analysis.

---

**Design Verification Report**

Appendix C - Design Validation Plan and Report
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.
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.
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.
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/](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

  - 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

  - 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

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 good toughness characteristics. Applications range from transportation components to machinery and equipment applications to general manufacturing and consumer products.

Alcoa produces 6061 for use in standard and custom shapes: rod, bar, tube, and sheet products, and seamless and structural pipe and tube.

Alloy 6061 has excellent corrosion resistance to atmospheric conditions and good corrosion resistance to seawater. This alloy offers good finishing characteristics and response well to anodizing, however, some anodizing appearances are critical. The use of alloy 6063. The most common anodizing methods include buff, clear, and color and dye, and natural.

Alloy 6061 is easily worked 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 the -70° condition can be reduced in the weld region. Selection of an appropriate filler alloy will depend on the desired weld characteristics.

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 (T6 temper), 6061 has good formability for bending. After artificial aging (precipitation-hardening, T6, or T7) it is capable of developing its ultimate properties.

Standard Temper Definitions

- O: As supplied. There is no stress relief or heat treatment; this is the weakest temper.
- T: Heat treated. Various tempers are available to achieve the desired strength levels.
- T6: Solution heat-treated and naturally aged. (See Notes C & D.)
- T69: Solution heat-treated and artificially aged. (See Notes C & D.)
- T8: Solution heat-treated and artificially aged. (See Notes C & D.)

Alcoa Special Tempers

T4C: For 6061 extrusions requiring maximum formability in the annealed condition and successfully aged to T6. May not meet minimum mechanical properties, but is within the minimum property range. This temper is suitable for applications where maximum formability is required. T4C is not recommended for applications requiring maximum formability in the naturally aged condition. (See Note A.)

T22: T22 is a precipitation-hardening temper that is well suited for applications requiring high strength and good formability. This temper is suitable for applications requiring high strengths in the naturally aged condition. (See Note A.)

T521 & T532: For 6061 extrusions requiring maximum formability and good strength. These tempers are well suited for applications requiring maximum formability and good strength. (See Note A.)

T691: Alcoa's M temper is designed for special applications requiring improved machinability and higher mechanical properties than standard T6 or T691. Minimum mechanical properties of 42-46 ksi tensile, 35-50 ksi yield, and 10 ksi elongation are guaranteed. This temper is available in corrosion-resistant material and when specified, for extrusions and sections. (See Notes C & D.)

T561: Alcoa's M temper is designed for severe forming applications requiring improved machinability and higher mechanical properties than standard T5 or T691. Minimum mechanical properties of 42-46 ksi tensile, 35-50 ksi yield, and 10ksi elongation are guaranteed. (See Notes C & D.)

T562: For 6061 extrusions requiring maximum formability and good strength. This temper is well suited for applications requiring maximum formability and good strength. (See Note A.)

For further details of specifications, see Alcoa's Spec Sheet 6061A. Alcoa's specification will conform to Alcoa's, ASTM, and ACP specifications. All Alcoa products are produced to Alcoa's specification. All Alcoa products are produced to Alcoa's specification.
### Alloy 6061 Chemical Analysis

<table>
<thead>
<tr>
<th>Percent Weight</th>
<th>Fe</th>
<th>Cu</th>
<th>Mn</th>
<th>Si</th>
<th>Mg</th>
<th>Zn</th>
<th>Cr</th>
<th>Ni</th>
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<td>0.08</td>
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Average coefficient of thermal expansion (6x10^-6/°F) = 13.1 x 10^-6 in/in per °F.

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

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<th>Tempor</th>
<th>Specified Section or Wall Thickness (inches)</th>
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<th>Elevation</th>
<th>Typical Reliability</th>
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<td>Percent</td>
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*The mechanical property limits or standard tempers are taken from the "Standard Section" of the Aluminum Association's "Aluminum Data Book" and are subject to change. The table includes only material that is standard and corresponds to the applicable national or international standard. All data is based on tests conducted in accordance with standard test methods. Properties of other material or special tempers may be different. For further details, see Aluminum Association's "Aluminum Standards and Data Book.

### Comparative Characteristics of Related Alloys/Temper

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<th>Alloy</th>
<th>Temper</th>
<th>Forging</th>
<th>Machinability</th>
<th>General Corrosion Resistance</th>
<th>Weldability</th>
<th>Brazability</th>
<th>Anodizing Response</th>
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<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
</tr>
<tr>
<td>6061</td>
<td>T6G400</td>
<td>Low</td>
<td>Low</td>
<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
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<tr>
<td>6061</td>
<td>T6G500</td>
<td>Low</td>
<td>Low</td>
<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
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<td>6061</td>
<td>T6G600</td>
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<tr>
<td>6061</td>
<td>T6G1000</td>
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<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
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<tr>
<td>6061</td>
<td>T6G1200</td>
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<td>D</td>
<td>C</td>
<td>B</td>
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<tr>
<td>6061</td>
<td>T6G1500</td>
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<td>C</td>
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<td>D</td>
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<td>C</td>
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<td>T6G7000</td>
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<td>D</td>
<td>C</td>
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<td>T6G8000</td>
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<td>Low</td>
<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
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<td>T6G9000</td>
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<td>Low</td>
<td>D</td>
<td>C</td>
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<td>6061</td>
<td>T6G10000</td>
<td>Low</td>
<td>Low</td>
<td>D</td>
<td>C</td>
<td>B</td>
<td>A</td>
</tr>
</tbody>
</table>

Alcoa Distribution and Industrial Products

52 Plattsville Street
Cressona, PA 17920
Phone: 800-232-2165
FAX: 800-232-2165

57 | Page
Aluminum 7075 Data Sheet

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 88 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 offer 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.

<table>
<thead>
<tr>
<th>7075 Temper Designations and Definitions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Standard Tempers</td>
</tr>
<tr>
<td>T6, T651</td>
</tr>
<tr>
<td>T73, T7351</td>
</tr>
</tbody>
</table>

*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

<table>
<thead>
<tr>
<th>Percent Weight</th>
<th>Elements</th>
<th>Liquidus Temperature: 1175°F</th>
<th>Solidus Temperature: 890°F</th>
<th>Density: 0.101 lb./in.³</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum</td>
<td>Si</td>
<td>0.6</td>
<td>0.3</td>
<td>0.0</td>
</tr>
<tr>
<td>Maximum</td>
<td>Si</td>
<td>0.6</td>
<td>0.3</td>
<td>0.0</td>
</tr>
</tbody>
</table>

Average Coefficient of Thermal Expansion (60°F to 212°F) = 3.3 x 10⁻⁶ inch per inch per °F.
### Alloy 7075 Global Cold Finished Products Capabilities and Mechanical Property Limits

<table>
<thead>
<tr>
<th>Temper</th>
<th>Specified Section on Wall Thickness (inches)</th>
<th>Tensile Strength (ksi)</th>
<th>Elongation (in %)</th>
<th>Ultimate Yield (0.2% offset)</th>
<th>Percent Min 2 inch or 40(^\circ)</th>
<th>Typical Bending Radii</th>
<th>Typical Ultimate Shearing Strength</th>
<th>Typical Electrical Conductivity</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Min.</td>
<td>Max.</td>
<td>Min.</td>
<td>Max.</td>
<td>Min.</td>
<td>Max.</td>
<td>(560 lb/10 mm ball)</td>
</tr>
<tr>
<td>T6</td>
<td>Up thru 4.000(^1)</td>
<td>77.0</td>
<td>86.0</td>
<td>7</td>
<td>150</td>
<td>48</td>
<td>33</td>
<td>4000</td>
</tr>
<tr>
<td>7051</td>
<td></td>
<td>77.0</td>
<td>86.0</td>
<td>7</td>
<td>150</td>
<td>48</td>
<td>33</td>
<td>4000</td>
</tr>
<tr>
<td></td>
<td>Up thru 4.000(^1)</td>
<td>75.0</td>
<td>84.0</td>
<td>7</td>
<td>150</td>
<td>48</td>
<td>33</td>
<td>4000</td>
</tr>
<tr>
<td></td>
<td>6.001-7.000</td>
<td>73.0</td>
<td>82.0</td>
<td>7</td>
<td>150</td>
<td>48</td>
<td>33</td>
<td>4000</td>
</tr>
<tr>
<td>7073</td>
<td>Up thru 4.000(^1)</td>
<td>66.0</td>
<td>75.0</td>
<td>10</td>
<td>150</td>
<td>40</td>
<td>21</td>
<td>4000</td>
</tr>
<tr>
<td></td>
<td>6.001-5.000</td>
<td>66.0</td>
<td>75.0</td>
<td>10</td>
<td>150</td>
<td>40</td>
<td>21</td>
<td>4000</td>
</tr>
<tr>
<td></td>
<td>5.001-6.000</td>
<td>64.0</td>
<td>72.0</td>
<td>8</td>
<td>130</td>
<td>40</td>
<td>21</td>
<td>4000</td>
</tr>
</tbody>
</table>

\(^1\) The mechanical property limits for standard tempers are listed in the “Standard Section” of the Aluminum Association’s “Aluminum Standards and Data” manual and “Tuppers for Aluminum and Aluminum Alloys.” The thickness of the section from which the tension test specimen is taken determines the applicable mechanical properties. \(^2\) For material of such dimensions that a standard test specimen cannot be obtained, or for shapes thinner than 0.005”, the test for elongation is not required. \(^3\) Specimen diameter. \(^4\) Mechanical properties applicable to all thickness shown. For square, hexagonal, and octagonal bar, the maximum thickness is 3.5” and for rectangular bar the maximum thickness is 2”. \(^5\) Minimum value when lot acceptance is required for stress corrosion and exfoliation corrosion; 30.0-35.9 per standard requirements and yield strength does not exceed minimum by more than 11.9 kpsi.

### Comparative Characteristics of Related Alloys/Tempers

<table>
<thead>
<tr>
<th>Alloy</th>
<th>Temper</th>
<th>Formability</th>
<th>Machinability</th>
<th>General Corrosion Resistance</th>
<th>Weldability (Arc with 3/32” Rod)</th>
<th>Fatigue</th>
<th>Anodizing Response</th>
<th>Stress Corrosion Cracking</th>
</tr>
</thead>
<tbody>
<tr>
<td>7075</td>
<td>-T6</td>
<td>-T651</td>
<td>-T6, -T651</td>
<td>-T73, -T7351</td>
<td>-2500</td>
<td>-2500</td>
<td>-2500</td>
<td>-2500</td>
</tr>
<tr>
<td>2024</td>
<td>-T501, -T4</td>
<td>-T6</td>
<td>-T651</td>
<td>-T651</td>
<td>-T6</td>
<td>-2500</td>
<td>-2500</td>
<td>-2500</td>
</tr>
<tr>
<td>6061</td>
<td>-T6</td>
<td>-T651</td>
<td>-T651</td>
<td>-T651</td>
<td>-T6</td>
<td>-2500</td>
<td>-2500</td>
<td>-2500</td>
</tr>
</tbody>
</table>

\(^1\) For further details of explanation of ratings, see Aluminum Association’s “Aluminum Standards and Data” manual. \(^2\) Ratings A, B, C, and D 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 failures only, B - No known instances of failure in service, laboratory failures only. C - Service and laboratory failures under special conditions.

---

**ALCOA**

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Phone: 1-877-265-3346
Fax: 315/766-4416
www.alcoa.com/gcfp

---

© 2004 ALCOA
Brembo Rear Brake Calipers

**TECHNICAL CHARACTERISTICS**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
<td>Plates number and diameter</td>
<td>32 mm</td>
</tr>
<tr>
<td>Max working pressure</td>
<td>10 bar</td>
</tr>
<tr>
<td>A) Disc position (tolerance)</td>
<td></td>
</tr>
<tr>
<td>Disc thickness</td>
<td>5.0 mm</td>
</tr>
<tr>
<td>Disc thickness (Tolerance)</td>
<td>±0.1 mm</td>
</tr>
<tr>
<td>B) Weight</td>
<td>0.540 Kg</td>
</tr>
<tr>
<td>(with no sintered pad)</td>
<td></td>
</tr>
<tr>
<td>Configuration</td>
<td></td>
</tr>
</tbody>
</table>

| Brake Fluid for working mean | 8.50 cm |

<table>
<thead>
<tr>
<th>AVAILABLE SPARE PARTS</th>
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<tr>
<td>CALIPER</td>
<td>20.5181.71</td>
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<tr>
<td>BRAKE FLUID</td>
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<tr>
<td>FRENDO</td>
<td></td>
</tr>
<tr>
<td>105.3246.11</td>
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</tr>
<tr>
<td>105.1502.10</td>
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<tr>
<td>120.2800.11</td>
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</table>

Indicating informations subject to modification. For further specifications contact Technical Section. Last Update: 09/09/1994
Deep Groove Ball Bearing, 16007

<table>
<thead>
<tr>
<th>Model</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>ID</td>
<td>16007</td>
</tr>
<tr>
<td>Name</td>
<td>Deep Groove Ball Bearing, single row, unsealed</td>
</tr>
</tbody>
</table>

**Technical Specifications**
- Designation: 16007
- Type: Deep Groove Ball Bearing
- Single Row
- Unsealed

**Dimensions**

<table>
<thead>
<tr>
<th>Model</th>
<th>ID</th>
<th>Name</th>
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</thead>
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<tr>
<td>16007</td>
<td>Deep Groove Ball Bearing, single row, unsealed</td>
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</tr>
</tbody>
</table>

**Technical Details**

- **ID**: 16007
- **Type**: Deep Groove Ball Bearing
- **Designation**: 16007
- **Technical Specifications**: Single Row, Unsealed

---

**Calculation Notes**

- For further technical details and calculations, refer to the manufacturer's official documentation.

---

**Image Description**

The image includes a diagram of the deep groove ball bearing, showing its internal components and dimensions. The diagram is essential for understanding the bearing's design and for accurate assembly and maintenance.

---

**Additional Information**

Deep groove ball bearings are widely used in various applications due to their high radial and axial load capacity, and their ability to accommodate misalignment. They are commonly used in automotive, aircraft, and machinery industries.
Deep Groove Ball Bearing, 61805
Brake Rotor E-Clip

Note: Clearance diameter is the diameter of a housing that cap pass freely over the ring.
Pinion Sprocket
## LMC D135 Rag/S DC Electric Motor

### 130 Table

<table>
<thead>
<tr>
<th>Motor</th>
<th>Amps</th>
<th>Torque</th>
<th>Speed</th>
<th>Current</th>
<th>Resistance DC</th>
<th>Efficiency</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Current</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>3.805</td>
<td>159</td>
<td>13</td>
<td>14</td>
<td>0.379</td>
<td>0.85</td>
<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>4.35</td>
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</table>

### 170 Table

<table>
<thead>
<tr>
<th>Motor</th>
<th>Amps</th>
<th>Torque</th>
<th>Speed</th>
<th>Current</th>
<th>Resistance DC</th>
<th>Efficiency</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Current</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
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<td>159</td>
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<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>4.35</td>
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### 200 Table

<table>
<thead>
<tr>
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<th>Amps</th>
<th>Torque</th>
<th>Speed</th>
<th>Current</th>
<th>Resistance DC</th>
<th>Efficiency</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Power</th>
<th>Current</th>
<th>Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>3.805</td>
<td>159</td>
<td>13</td>
<td>14</td>
<td>0.379</td>
<td>0.85</td>
<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>13</td>
<td>0.37</td>
<td>4.35</td>
</tr>
</tbody>
</table>

Any model of the LMC-D135 can be made up to the 240 version this is 2 motors mounted together as a single shaft use incalation drawing for details

The 130 model available for increased power, larger motor in development for 2012

Torque @ full load (J): B | N, (J): T (J), (J), (J): (J) (J) 

LMC reserves the right to change this information without prior notice.
Traction Control/Data Acquisition Data Sheets

Miniature Absolute Magnetic Shaft Encoder Data Sheet

**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 +6V, 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, 280 Hz

**Mechanical Drawing**

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www.usdigital.com
Local: 360.290.2468
Toll-free: 800.336.0194

Rev. 120202102237
**Mechanical**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Sleeve Bushing</th>
<th>Ball Bearing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moment of Inertia</td>
<td>4.1 x 10^-6 oz-in-s^2</td>
<td>4.1 x 10^-6 oz-in-s^2</td>
</tr>
<tr>
<td>Angular Accuracy</td>
<td>&lt;0.5 deg @ 25C</td>
<td>&lt;0.5 deg @ 25C</td>
</tr>
<tr>
<td>Angular Accuracy Over Temperature</td>
<td>&lt;0.9 deg @ -40C to 125C</td>
<td>&lt;0.9 deg @ -40C to 125C</td>
</tr>
<tr>
<td>Shaft Speed</td>
<td>100 RPM max. continuous</td>
<td>15,000 RPM max. continuous</td>
</tr>
<tr>
<td>Acceleration</td>
<td>10,000 rad/sec^2</td>
<td>290,000 rad/sec^2</td>
</tr>
<tr>
<td>Vibration</td>
<td>20G. 5Hz to 2kHz</td>
<td>20G. 5Hz to 2kHz</td>
</tr>
<tr>
<td>Shaft Torque</td>
<td>0.5 ± 0.2 in. oz. (D - torque option)</td>
<td>0.05 in. oz. max.</td>
</tr>
<tr>
<td></td>
<td>0.3 in. max. (N - torque option)</td>
<td></td>
</tr>
<tr>
<td>Shaft Loading</td>
<td>2 lbs. max. dynamic*</td>
<td>1 lb. max.</td>
</tr>
<tr>
<td></td>
<td>20 lbs. max. static</td>
<td></td>
</tr>
<tr>
<td>Bearing Life</td>
<td>&gt; 1,000,000 revolutions</td>
<td>(L_{10} = (18.3F_i)^{-2})</td>
</tr>
<tr>
<td></td>
<td>Where (L_{10}) = bearing life in millions of revs, and (F_i) = radial shaft loading in pounds</td>
<td></td>
</tr>
<tr>
<td>Weight</td>
<td>0.40 oz.</td>
<td>0.37 oz.</td>
</tr>
<tr>
<td>Shaft Runout</td>
<td>0.0015 T.I.R. max.</td>
<td>0.0015 T.I.R. max.</td>
</tr>
</tbody>
</table>

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

---

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www.usdigital.com  
Local: 360.290.2468  
Toll-free: 800.736.0194

Rev. 12002010237
\[ n = \frac{60 \times \text{rpm} \times \text{us}}{96} \]

*only valid with negligible axial shaft loading*

### Environmental

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operating Temperature</td>
<td>-40°C to +125°C</td>
</tr>
<tr>
<td>Storage Temperature</td>
<td>-55°C to +125°C</td>
</tr>
<tr>
<td>ESD</td>
<td>± 2 kV max.</td>
</tr>
<tr>
<td>Humidity Non-condensing</td>
<td>5% to 85%</td>
</tr>
</tbody>
</table>

### Mounting

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hole Diameter</td>
<td>0.375&quot; ± 0.005&quot; -0.0</td>
</tr>
<tr>
<td>Panel Thickness</td>
<td>0.125&quot; max.</td>
</tr>
<tr>
<td>Panel Nut Max. Torque</td>
<td>20 in.-lbs.</td>
</tr>
</tbody>
</table>

### Materials

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Dimension</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft</td>
<td>Stainless</td>
</tr>
<tr>
<td>Bushing</td>
<td>Brass</td>
</tr>
</tbody>
</table>

### 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

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min.</th>
<th>Typ.</th>
<th>Max.</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Supply</td>
<td>4.5</td>
<td>5.0</td>
<td>5.5</td>
<td>Volts</td>
</tr>
<tr>
<td>Supply Current</td>
<td>-</td>
<td>16</td>
<td>20</td>
<td>mA</td>
</tr>
<tr>
<td>Power-up Time</td>
<td>-</td>
<td>-</td>
<td>50</td>
<td>μS</td>
</tr>
</tbody>
</table>

1400 NE 136th Avenue  info@usdigital.com  Local: 360.200.2468
Vancouver, Washington 98684, USA  www.usdigital.com  Toll-free: 800.736.6194

Rev: 120202102237
**Analog Output Operation**

Analog output is only available in 10-bit resolution. The analog output voltage is ratio-metric 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 \( \geq 4.7k\Omega \) and less than 100pF.

The graphs below show the typical output levels for various output loads when powered by a 5V supply.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min.</th>
<th>Typ.</th>
<th>Max.</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Position Sampling Rate</td>
<td>2.35</td>
<td>2.61</td>
<td>2.87</td>
<td>kHz</td>
</tr>
<tr>
<td>Propagation Delay</td>
<td>-</td>
<td>-</td>
<td>364</td>
<td>?S</td>
</tr>
<tr>
<td>Analog Output Voltage Maximum (1)</td>
<td>-</td>
<td>4.667</td>
<td>-</td>
<td>Volts</td>
</tr>
<tr>
<td>Analog Output Voltage Minimum (1)</td>
<td>-</td>
<td>0.015</td>
<td>-</td>
<td>Volts</td>
</tr>
<tr>
<td>Output Short Circuit Sink Current (2)</td>
<td>-</td>
<td>32</td>
<td>50</td>
<td>mA</td>
</tr>
<tr>
<td>Output Short Circuit Source Current (2)</td>
<td>-</td>
<td>36</td>
<td>68</td>
<td>mA</td>
</tr>
<tr>
<td>Output Noise (2)</td>
<td>100</td>
<td>220</td>
<td>490</td>
<td>?Vrms</td>
</tr>
<tr>
<td>Output Transition Noise (3)</td>
<td>-</td>
<td>0.03</td>
<td>-</td>
<td>Deg. RMS</td>
</tr>
</tbody>
</table>

(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.
**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 width is known and the nominal pulse period is used to measure the angle, the resulting value also has this timing tolerance. However, this tolerance can be cancelled by measuring both the pulse and calculating the angle from the duty cycle. Angular accuracy including non-linearity is within 0.5 deg. at 25°C, but may increase to 0.9 deg. at high temperatures.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Min.</th>
<th>Typ.</th>
<th>Max.</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>PWM Frequency (40°C to 125°C)</td>
<td>0.577</td>
<td>0.975</td>
<td>1.072</td>
<td>kHz</td>
</tr>
<tr>
<td></td>
<td>220</td>
<td>244</td>
<td>268</td>
<td>Hz</td>
</tr>
<tr>
<td>Minimum Pulse Width</td>
<td>0.06</td>
<td>1.00</td>
<td>1.05</td>
<td>ms</td>
</tr>
<tr>
<td></td>
<td>0.05</td>
<td>1.00</td>
<td>1.05</td>
<td>ms</td>
</tr>
<tr>
<td>Maximum Pulse Width</td>
<td>674</td>
<td>1025</td>
<td>1076</td>
<td>ms</td>
</tr>
<tr>
<td></td>
<td>3882</td>
<td>4387</td>
<td>4902</td>
<td>ms</td>
</tr>
<tr>
<td>Internal Sampling Rate</td>
<td>9.38</td>
<td>10.42</td>
<td>11.46</td>
<td>kHz</td>
</tr>
<tr>
<td></td>
<td>2.35</td>
<td>2.61</td>
<td>2.87</td>
<td>kHz</td>
</tr>
<tr>
<td>Propagation</td>
<td>-</td>
<td>-</td>
<td>48</td>
<td>ms</td>
</tr>
<tr>
<td></td>
<td>-</td>
<td>-</td>
<td>384</td>
<td>ms</td>
</tr>
<tr>
<td>Output Transition Noise, 12-bit version (1)</td>
<td>0.03</td>
<td>Deg. RMS</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Output Transition Noise, 10-bit version (1)</td>
<td>0.12</td>
<td>Deg. RMS</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Output High Voltage (V OH, @4mA Source) (2)</td>
<td>Vcc-0.8</td>
<td>-</td>
<td>-</td>
<td>V</td>
</tr>
<tr>
<td>Output Low Voltage (V OL, @4mA Sink) (2)</td>
<td>-</td>
<td>-</td>
<td>0.4</td>
<td>V</td>
</tr>
</tbody>
</table>

(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:

\[ x = \left( \frac{\text{bin} \times 1024}{1 + \text{t \_ off}} \right) - 1 \]

If \( x \leq 1022 \), then Position = \( x \)
If \( x = 1024 \), then Position = 1023

12-bit PWM:

\[ x = \left( \frac{\text{bin} \times 4098}{1 + \text{t \_ off}} \right) - 1 \]

If \( x \leq 4064 \), then Position = \( x \)
If \( x = 4096 \), then Position = 4095
Pin-outs

Analog Output (MA3-A):

<table>
<thead>
<tr>
<th>Pin</th>
<th>Name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>S</td>
<td>+5VDC power</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>Analog output</td>
</tr>
<tr>
<td>3</td>
<td>G</td>
<td>Ground</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Pin</th>
<th>Name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>S</td>
<td>+5VDC power</td>
</tr>
<tr>
<td>2</td>
<td>A</td>
<td>PWM output</td>
</tr>
<tr>
<td>3</td>
<td>G</td>
<td>Ground</td>
</tr>
</tbody>
</table>
**Ordering Information**

<table>
<thead>
<tr>
<th>Interface</th>
<th>Shaft Diameter</th>
<th>Torque</th>
<th>Rules</th>
</tr>
</thead>
<tbody>
<tr>
<td>A10 = 10-Bit Analog</td>
<td>125 = 1/8&quot;</td>
<td>D = Sleeve Bushing, Most Drag</td>
<td>Torque must be something other than B when Shaft Diameter is something other than 125</td>
</tr>
<tr>
<td>P10 = 10-Bit PWM</td>
<td>236 = 6mm</td>
<td>N = Sleeve Bushing, Somewhat Lighter Drag</td>
<td></td>
</tr>
<tr>
<td>P12 = 12-Bit PWM</td>
<td>250 = 1/4&quot;</td>
<td>B = Ball Bearing, Free Spinning (Least Drag)</td>
<td></td>
</tr>
</tbody>
</table>

**Base Pricing**

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Price</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$30.60</td>
</tr>
<tr>
<td>10</td>
<td>$34.69</td>
</tr>
<tr>
<td>50</td>
<td>$30.56</td>
</tr>
<tr>
<td>100</td>
<td>$20.29</td>
</tr>
</tbody>
</table>

* 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)
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 complies with the free AVR GCC compiler.

The 6-DOF Atomic uses these sensors:
- Freescale MMA7260Q 7 tri-axial accelerometer, selectable to 1g, 3g, 4g or 6g sensitivity
- 3 ST Microelectronics LISY300AL 7 single-axis, 300/64 gyro

Electrical Specifications:
- Input voltage: 3.4V to 10V DC
- Current consumption: 24mA (115mA with XBee)
- Sensor bandwidth and resolution:

- LISY300AL Gyros: 68Hz, 5.9777°/tick (ADC count)
- MMA7260Q Accelerometer:
  - 350Hz, X and Y axes
  - 150Hz, Z axis
  - 0.00403°/tick @ 1g
  - 0.0067°/tick @ 2g
  - 0.0107°/tick @ 4g
  - 0.0161°/tick @ 8g

The MMA7260Q accelerometer and the LISY300AL gyro's 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).
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 R8 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 “S” in Figure 1). TX, RX,
and ground are all thats 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-09864 and
WRL-08667). 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/tutorials/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 to 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) “A”, ASCII 37, sets the accelerometer to
   1.5g sensitivity
2) “B”, ASCII 26, sets the accelerometer
   sensitivity to 2g
Atomic IMU - 6 Degrees of Freedom - Xbee Ready

2009.3.25

3) ";" (asterisk), ASCII 39, sets the accelerometer sensitivity to 4g
4) "*", ASCII 44, sets the accelerometer sensitivity to 5g
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 is to make it easier to interact with the Atomic mixer application, and there is no reason that a user's own application couldn't use it for a quick setup.

Operation from the Configuration Menu

5DOF 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 "9" 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 its primary idle state, which means that it will always wait for a "9" 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 "0" to increase or "9" 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 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 MMA7361Q accelerometer is 300Hz for the X and Y axis, and 150Hz for the Z axis. There are also additional single-pole low-pass filters to reduce switching noise from the sensor with a filter set at 159Hz (recommended by Freescale). The internally set output bandwidth for the LUSB900AL 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.

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**

<table>
<thead>
<tr>
<th></th>
<th>Acc</th>
<th>Brake</th>
<th>d</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Loads</td>
<td>Mma</td>
<td>Mma</td>
<td></td>
<td></td>
</tr>
<tr>
<td>lbins</td>
<td>kb</td>
<td>kc</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0.879</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tma</td>
<td>Tma</td>
<td>kc</td>
<td></td>
<td></td>
</tr>
<tr>
<td>744</td>
<td>1200</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Maa</td>
<td>Maa</td>
<td>kd</td>
<td></td>
<td></td>
</tr>
<tr>
<td>717</td>
<td>303</td>
<td>1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Taa</td>
<td>Taa</td>
<td>ke</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>0.753</td>
<td>99.90%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Calcs/Equations/ tables**

<table>
<thead>
<tr>
<th>Variables</th>
<th>Titles</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acc</td>
<td>Brake</td>
<td>d</td>
</tr>
<tr>
<td>Loads</td>
<td>Mma</td>
<td>kb</td>
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<tr>
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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.

<table>
<thead>
<tr>
<th>Material</th>
<th>Aluminum 6061 T6</th>
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<tbody>
<tr>
<td>Modulus Z Polar ($in^3$)</td>
<td>0.3598</td>
</tr>
<tr>
<td>Spline Diameter (in)</td>
<td>1.33</td>
</tr>
<tr>
<td>$D_m$ (in)</td>
<td>1.25</td>
</tr>
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</table>

<table>
<thead>
<tr>
<th></th>
<th>Sprocket Carrier</th>
<th>Brake Carrier</th>
<th>CV Housing Female Plate</th>
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<tbody>
<tr>
<td>Actual Torque Transmitted (in-lbf)</td>
<td>2705.5</td>
<td>1380.4</td>
<td>2705.5</td>
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<tr>
<td>Hub</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Spline Length (in)</td>
<td>1.125</td>
<td>1.3</td>
<td>0.125</td>
</tr>
<tr>
<td>Wall Thickness (in)</td>
<td>0.33</td>
<td>0.33</td>
<td>0.33</td>
</tr>
<tr>
<td>Flange</td>
<td></td>
<td></td>
<td></td>
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<td>Spline Length (in)</td>
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<td>Wall Thickness (in)</td>
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<td>Allowable Stress (psi)</td>
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<tr>
<td>Allowable Torque Transmitted</td>
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<td>Hub (in-lbf)</td>
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<td>Flange (in-lbf)</td>
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<td>Total (in-lbf)</td>
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<td>Factor of Safety</td>
<td>1.65</td>
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Polygon Shaft Geometry and Nomenclature

Figure 28. Polygon Shaft Geometry reproduced from Machinery’s Handbook: 28th Edition.
Design Life Calculations

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

**Tire Sizing:**

1. Tire section width (mm)
2. Tire aspect ratio as a function of section width
3. Wheel diameter (in)

\[ d_t = \text{tire outside diameter} \]
\[ d_s = (2)(0.85)(100 mm)(1.75/25.4 mm) + (10 in) \]
\[ d_s = 16.693 \text{ in} \]

**Cycles:**

\[ \text{Cycles} = \frac{\text{distance}}{\text{tire circumference}} \]
\[ \text{Cycles} = \frac{200}{(2)(0.85)(100 mm)(1.75/25.4 mm) + (10 in)} \]
\[ \text{Cycles} = 241.635 \text{ revolutions} \]

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

\[ t = \frac{\text{distance}}{\text{speed}} = \frac{200 \text{ mi}}{10 \text{ mph}} = 20 \text{ hours} \]

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

\[ n = 100 \text{ hours} \]
Drivetrain Subsystem Load Calculations

Load Calculations:

Under Braking:

\[ \theta = \tan^{-1} \left( \frac{1.12}{1.14} \right) \rightarrow \theta = 48.7^\circ \]

\[
F_b = 471.13 \text{ lb} \quad r_{br} = 2.93 \text{ in}
\]

\[ F_b = \text{Max braking force per brake rotor}, \]
\[ r_{br} = \text{Effective radius that the braking force acts upon the rotor}, \]
\[ R_x = \text{Reaction force in the x}, \]
\[ R_y = \text{Reaction force in the y}, \]
\[ M_z = \text{Reaction moment in the z}, \]

\[ \Sigma F_x = 0 \Rightarrow R_x = -F_b \sin \theta \quad \Rightarrow R_x = -F_b \sin 48.7^\circ \]
\[ R_x = 355.44 \text{ lb} \]

\[ \Sigma F_y = 0 \Rightarrow R_y = -F_b \cos \theta \quad \Rightarrow R_y = -F_b \cos 48.7^\circ \]
\[ R_y = 310.95 \text{ lb} \]

\[ \Sigma M = 0 \Rightarrow F_b r_{br} - M_y \Rightarrow M_y = F_b r_{br} = M_y = (471.13 \text{ lb})(2.93 \text{ in}) \]
\[ M_y = 1350.417 \text{ lb-in} \]
Load Calculations

Under Acceleration:

\[ T_{\text{motor}} = \frac{F_{\text{peak}} \times J_{\text{load}}}{K_{T}} \]

\[ T_{\text{water}} = \left(0.21 \frac{\text{ft-lb}}{\text{rpm}}\right) (400 \text{ rpm} - 745.4) \]

\[ T_{\text{water}} = (82.44 \text{ Nm}) \left(1 \frac{\text{ ft-lb}}{1.33 \text{ Nm}}\right) \]

\[ T_{\text{water}} = 61.95 \text{ ft-lb} \]

\[ \sum M = 0 = \frac{1}{2} r_3^2 T_3 - \frac{1}{2} r_m^2 T_m \]

\[ \Rightarrow \frac{T_3}{r_3^2} = \frac{T_m}{r_m^2} \Rightarrow T_3 = \left(\frac{r_m^2}{r_3^2}\right) \times \left(\frac{61.95}{16.45}\right) \]

\[ T_3 = 225.45 \text{ ft-lb} \]

\[ P = 0.5 \text{ m}, \ N = 40 \text{ teeth} \]

Pitch Diameter:

\[ D_p = \frac{P}{\sin \left(180^\circ/40\right)} \]

\[ D_p = \frac{0.5}{\sin \left(180^\circ/40\right)} \]

\[ D_p = 8.573 \text{ in.} \]

\[ r_{p3} = D_p/2 = 4.286 \text{ in.} \]

\[ r_{p3} = 3.187 \text{ in.} \]

**Symbols:**
- \( F_g \): Gear Sprocket Force from chain or chain tension
- \( r_{p3} \): Pitch radius of gear sprocket
- \( R_a \): Reaction force at the \( z \)
- \( R_y \): Reaction force in the \( y \)
- \( M_y \): Reaction moment in the \( y \)

\[ F_g = \frac{T_3}{r_{p3}} \Rightarrow F_g = \left(225.45 \text{ ft-lb}\right) \left(12 \text{ in./ft}\right) / (3.187 \text{ in.}) \]

\[ F_g = 848.89 \text{ lb} \]
Load Calculations

Under Acceleration (cont.)

\[ \Theta_{rp} = \Theta + \epsilon_r \]
\[ \Theta_{rp} = 23.2^\circ + 71.8^\circ \]
\[ \Theta_{rp} > 95^\circ \]

\[ \Sigma F_x = 0 = R_x + F_3 \sin 5^\circ \Rightarrow R_x = -F_3 \sin 5^\circ \]
\[ R_x = -\left(848.89 \text{ lb}_3\right)\left(\sin 5^\circ\right) \Rightarrow R_x = -73.98 \text{ lb}_3 \]

\[ \Sigma F_y = 0 = R_y - F_3 \cos 5^\circ \Rightarrow R_y = F_3 \cos 5^\circ \]
\[ R_y = \left(848.89 \text{ lb}_3\right)\left(\cos 5^\circ\right) \Rightarrow R_y = 845.46 \text{ lb}_3 \]

\[ \Sigma M = 0 = F_3 r_{3g} - M_3 \Rightarrow M_3 = -F_3 r_{3g} \]
\[ M_3 = \left(848.89 \text{ lb}_3\right)\left(3.147 \text{ in}\right) \Rightarrow M_3 = \frac{2705.41}{\text{in-lb}} \]

**Critical Loading Condition**

\[ T_{acc} = 2705.41 \text{ in-lb} \]
Chain Analysis: Fatigue and Ultimate Tensile Strength

Chain Analysis:

Fatigue Strength:

Link Plate Limited:

\[ H_f = 0.004N_{p0} \cdot \frac{\rho}{n_t} \cdot \left(3.07P\right) \cdot \frac{\text{rpm}}{3.07} \]

- \( N_{p0} \): 
  - 40 teeth in pinion sprocket = 11 teeth
  - 20 teeth in sprocket speed (rpm)

From manufacturing specs:

- \( n = (\text{rpm}) \leq 5000 \text{ rpm} \)
- \( n = (40 \text{ volts}) \cdot (40 \text{ RPM/volt}) \)
- \( n = 3840 \text{ rpm} \)
- \( p = \text{chain pitch} = 0.5 \text{ in} \)

\[ H_f = \frac{1000}{(1.1)^{1.48}} \cdot \left(3840\right)^{0.9} \cdot (0.5) \cdot \left[3 - 0.07\right] \text{ in} \]

- \( H_f = 11.48 \text{ hp} \) (highly conservative, based on \( n = 15000 \text{ hours} \) where as our design life is \( n = 100 \text{ hours} \))

Roller Limited:

\[ H_r = 1000 \cdot \left[ K_r \cdot (N) \right]^{1.5} \cdot \left( \frac{L_p}{190} \right) \cdot \left( \frac{15000}{n} \right) \cdot \left( \frac{0.4}{n} \right) \]

- \( L_p = \text{chain length in pitches} \)
- \( L_p = \frac{2}{P} \cdot \left[ N_1 + N_2 \cdot \left( N_3 - N_2 \right) \right] \cdot \frac{90^\circ \text{C}}{P} \)

- \( C \): center to center distance = 7.616 in (see Under Accel. Calc.s., \( P = 0.5 \text{ in} \)

- \( N_1 = 11 \text{ teeth} \); \( N_2 = 40 \text{ teeth} \)

\[ L_p = \frac{(2)(7.616 \text{ in}) + (11 + 40) \cdot (40 - 11)^2}{0.5 \text{ in}} \cdot \frac{90^\circ \text{C}}{P} \]

- \( L_p = 57.26 \text{ pitches} \) ~ \( 58 \text{ pitches} \)

- \( n = 100 \text{ hrs} \)

- \( K_r = 17 \) (Per chains \( 40 - 240 \), using 40 chain)
Chain Analysis:

Fatigue Strength (cont.):

Roller Limited (cont.):

\[
H_2 = 1000 \left( \frac{N}{3240} \right)^{1.5} (0.5)^{0.2} \left( \frac{32}{100} \right)^{0.4} \left( \frac{15000}{100} \right)^{0.4}
\]

\[H_2 = 8.93 \text{ hp} \]

Maximum Tensile Strength:

Maximum chain tension from motor: \( T_{\text{max}} = 248.89 \text{ lb} \) (See load calculations under acceleration.)

Minimum tensile force, ANSI 40 chain: 3130 lb

\[N = \frac{3130}{848.89} \Rightarrow N = 3.7\]

\[\text{Based on our low service life and the overly conservative nature of the fatigue strength calculations, we used maximum tensile strength as our design criteria.}\]

\[\therefore \text{Chain: ANSI 40}\]
Constant Velocity (CV) Joint Housing

Von Mises Stress
CV Housing Design Analysis:

von Mises Stress (cont.)

Material: 6061-T6 Aluminum

σ_y: 40,000 psi

σ_y - Calculations:

\[ P_{\text{avg}} = \frac{F_t + P_{L}}{A_{\text{avg}}} \Rightarrow P_{L} = \frac{(5)(812.27 - 16)}{10.443 \text{ in}^2} \]

\[ P_L = 245.7 \text{ psi} \]

\[ \sigma_t = \frac{P_L (1 + \frac{r_i}{r_t})}{\frac{r_t^2 - r_i^2}{r_t^2}} \Rightarrow \sigma_t = \frac{(1,050 \text{ in})(295.7 \text{ psi})}{[14 \text{ in}]^2 - [10 \text{ in}]^2} \]

\[ \sigma_t = 2,227.4 \text{ psi} \]

\[ \sigma_t - \text{Calculations:} \]

\[ P_{L_{\text{avg}}} = 2,95.7 \text{ psi} \]

\[ \sigma_t = \frac{P_L (1 - \frac{r_i}{r_t})}{\frac{r_t^2 - r_i^2}{r_t^2}} \Rightarrow \sigma_t = \frac{(1,050 \text{ in})(295.7 \text{ psi})}{[14 \text{ in}]^2 - [10 \text{ in}]^2} \]

\[ \sigma_t = 2,95.7 \text{ psi} \]

\[ \tau_{xy} - \text{Calculations:} \]

\[ F_t = 574.8 \text{ lb} \]

\[ K_t = 2 \text{ (Conservative)} \]

\[ \tau_{xy} = \frac{F_t K_t}{A_{\text{surface}}} \Rightarrow \tau_{xy} = \frac{(574.8 \text{ lb})(2)}{16.443 \text{ in}^2} \]

\[ \tau_{xy} = 418.2 \text{ psi} \]
CV Housing Design Analysis:

Von Mises Stress (cont.)

\[ \sigma_v = \left( \sigma_x^2 + \sigma_y^2 + \sigma_z^2 + 2\varepsilon_{12}^2 \right)^{1/2} \]

\[ \sigma_v = \left[ (222.7 \text{ psi})_x - (122.7 \text{ psi})_y + (295.7 \text{ psi})_z + \varepsilon_{12}^2 \right]^{1/2} \]

\[ \sigma_v = 2496.6 \text{ psi} \]

\[ N = \frac{\sigma_v}{\sigma_{eq}} = 7 \]

\[ N = \frac{(400,000 \text{ psi})}{(2496.6 \text{ psi})} \]

\[ N = 16,000 \]
Hertz Contact Stress

\[ d_{bb} = 0.501 \text{ in} \]
\[ d_{	ext{housing}} = 0.551 \text{ in} \]

Material Properties:

**Ball Bearing**
- Material: 52100 Chrome Alloy Steel
- \( E \): 30.5 x 10^6 psi
- \( Y \): 0.3

**CV Housing**
- Material: 6061-T6 Aluminum
- \( E \): 10.2 x 10^6 psi
- \( Y \): 0.35
- \( S_y \): 40,000 psi

Loads:
- \( F_{bb} = 812.8 \text{ lb} \) (See Von Mises Stress Analysis)

Analysis:

\[
\frac{1}{2} F_{bb} = \frac{1 - Y^2}{E_{bb}} + \frac{1 - Y^2_{Al}}{E_{Al}}
\]

Contact Radius:

\[
( \frac{3 F_{bb}}{2 Y R_{contact}} ) \Rightarrow P_{max} = \frac{3(812.8 \text{ lb})}{2(0.35)(0.124 \text{ in})^2} = 25,352 \text{ psi}
\]

\[
N = \frac{S_y}{P_{max}} = 7 \quad N = 1.53
\]
Bearing Selection Calculations

For Bearing A: Max Radial Load = 384.24 lb_f
For Bearing B: Max Radial Load = 646.51 lb_f
Bearing Selection Calculations:

**Bearing A:**

\[ F_x = 0 \quad (C/V \text{ joints can plunger}) \]
\[ F_y = 380 \, \text{lb} \]

Bore: 25 mm  (SKF Bearing G13005, \( C_0 = 2.6 \, \text{KN}, C_{10} = 4.36 \, \text{KN} \))

\[ C_o = 2.6 \, \text{KN} \]

\[ \frac{F_x}{C_o} = \frac{0}{2.6 \, \text{KN}} = 0 \quad e = 0.19 \]

\[ \frac{F_y}{F_x} = \frac{0}{380} = 0 \quad X_t = 1 \quad Y_t = 0 \]

\[ F_c = X_t \sqrt{F_x^2 + Y_t^2} \to F_c = 1(1.2)(386.26 \, \text{lb})(\frac{144 \, \text{KN}}{1000 \, \text{lb}}) \]

\[ F_c = 2.06 \, \text{KN} < C_o \quad \checkmark \]

\[ C_{10} = \frac{4}{3} \left[ \frac{X_p}{(X_p + (b - X_p)(1 - F_p)^{1/2})} \right]^{1/3} \]

\[ R_b = 0.99 \]
\[ p = 1440 \, \text{rev} \]
\[ b = 1.439 \]
\[ a = 3 \]
\[ b = 1.439 \quad \text{moderate impact} \]

\[ C_{10} = (1.5)(386.26 \, \text{lb})(\frac{X_{10} \, \text{KN}}{1000 \, \text{lb}}) \left[ 0.02 + (4.439)(1 - 0.99) \right]^{1/3} \]

\[ C_{10} = 1.03 \, \text{KN} < 4.36 \, \text{KN} \quad (SKF \text{ rated}) \quad \checkmark \]
Bearing Selection Calculations:

Bearing B:

\[ F_a = 0 \quad (CV \text{ joints can plunge; axial load negligible}) \]

\[ F_r = 644.51 \text{ lbf} = 44.48 \text{ kN} \Rightarrow F_r = 2.87 \text{ kN} \]

Bore: 35 mm (SKF Bearing 16007, \( C_r = 8.95 \text{ kN}, C_{10} = 13 \text{ kN} \))

\[ \frac{F_r}{C_r} = \frac{2.87 \text{ kN}}{8.95 \text{ kN}} \Rightarrow e = 0.19 \]

\[ \frac{F_r}{C_r} = \frac{0.19}{(1.2)(2.87 \text{ kN})} \Rightarrow Y_1 = 1, Y_2 = 0 \]

\[ F_e = X_i V_i F_r + Y_i F_e \Rightarrow F_e = (1)(1.2)(2.87 \text{ kN}) \]

\[ F_e = 3.44 \text{ kN} < C_r \checkmark \]

\[ C_{10} = 0.4 F_o \left[ \frac{Y_o}{Y_o + (1 - X_o)(1 - P_o) V_b} \right] V_b \]

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

\[ C_{10} = 1.73 \text{ kN} < 13 \text{ kN} \quad (SKF C_{10} \text{ rating}) \checkmark \]
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.
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.