PROVE Endurance Car Front Suspension

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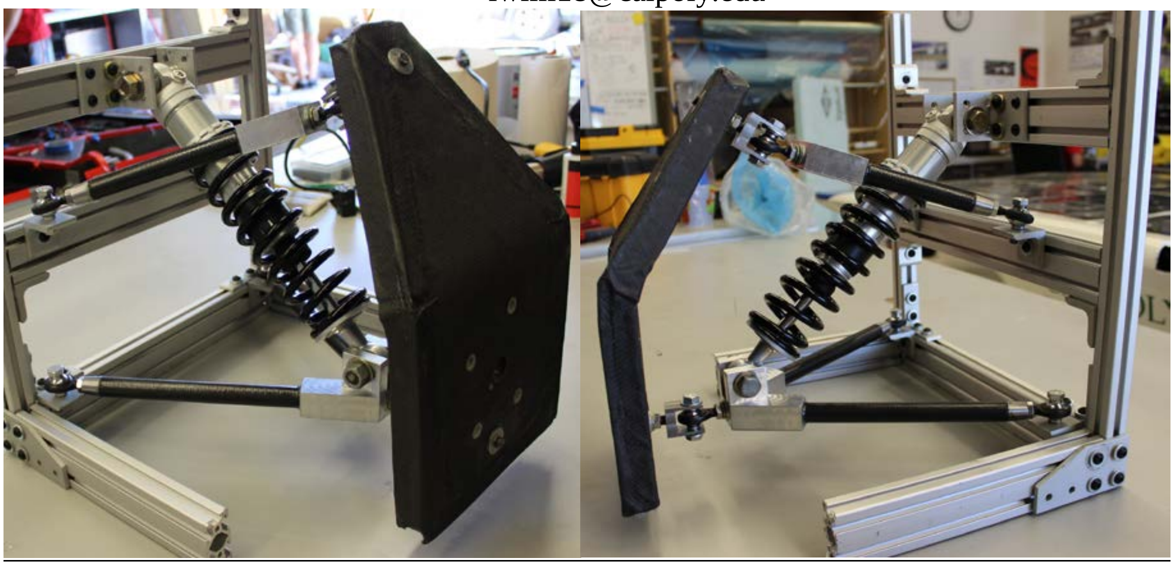

Sponsor: Graham Doig and PROVE Lab

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

This document details the collaborative Mechanical Engineering Senior Project with Cal Poly PROVE Lab on PROVE Lab's Project 2; an electric vehicle designed to travel 1000 miles on a single charge. Logan Simon, Justine Kwan, and Lauren Williams are given the challenge of designing an innovative proof of concept front suspension suspension for this vehicle.

After detailed research of new suspension systems, it was determined that the innovative nature could be in the form of unique manufacturing methods, materials use, or mechanical design. At this point in time, this vehicle is a purely conceptual design with no concrete requirements. Therefore engineering specifications were generated based on vehicles of similar size and function, such as PROVE Lab Project 1, Tesla's Roadster, and the BMW z4. These specification included vehicle weight, speed, vertical travel, system weight, travel speed, and track width.

Since this car is aimed to travel 1000 miles on one charge, efficiency is a big concern for the design. From ideation, the three suspension configurations of interest were MacPherson, double wishbone, and multi-link. A decision matrix was created to evaluate these designs based on design requirements, resulting in the selection of the multi-link configuration. However, after further investigation it was decided that a double wishbone configuration would provide nearly equal performance and be much more manageable of a task on the senior project time frame, compared to multi-link.

The focus of the project then shifted to innovative manufacturing methods. Carbon fiber was chosen as the material to be used due to its lightweight nature, its accessibility through PROVE lab, and its lack of usage in a suspension system thus far. The upright would provide the most weight savings, so it was designed as a carbon fiber sandwich panel. Computer analysis of the system included SolidWorks FEA, Tsai Wu Failure Analysis, and ANSYS composite analysis to verify Tsai Wu. Four destructive tests were performed to validate the analysis and to determine the number of plies to be used for the final part. With all four tests passing the minimum load requirements with a factor of safety above 1, 16 plies per laminate was chosen and with an additional 8 plies around the edges.

The final system proves that a carbon fiber suspension that is structurally sound for maximum loading cases and that cuts weight down to 4.3 pounds is possible. The full non-destructive test will be performed by the PROVE Project 2 team in the future, unassociated with this senior project.

Chapter 1, INTRODUCTION

Cal Poly Prototype Vehicles Laboratory (PROVE Lab) is developing a passenger, electric vehicle with the ultimate goal of 1000 miles on a single charge. This feat is meant to highlight growth, in regards to performance, in the field of electric vehicles. Currently the design of this car is purely conceptual, as the project kickstarted this Fall 2017. The project team, including Lauren Williams, Justine Kwan, and Logan Simon, is to develop a proof of concept for the front suspension system of the car that features an innovative design, manufacturing process, or material use, all while optimizing performance. The customer is Dr. Doig, the sponsor, as well as the PROVE Lab faculty advisor along with the PROVE team itself who will be interfacing with the product for the following years of vehicle development.

Chapter 2, BACKGROUND

The vision for this project is a long-distance electric vehicle with both extreme range and public appeal, summarized by Dr. Doig as "A car that looks like a ferrari, but can go 1000 miles on a single charge." For the front suspension specifically, the system should have a significant element of innovation in areas of system design, manufacturing process, or material use, while being lightweight and providing sufficient performance characteristics for a roadworthy car. Attachment A details the initial sponsor/ customer interview discussing the specifics of this project.

By the end of the year, Dr. Doig expects a rolling chassis "Buggy" with the suspension system integrated for mechanical and electrical driving tests. The car itself is in very early conceptual design, requiring the team to make informed assumptions on the specifications of the car. To generate these specifications the team relied heavily on research of vehicles of similar size, functionality, and application.

2.1 Existing Suspension Designs

<u>Double Wishbone:</u>

Figure 1. Basic Double Wishbone Suspension[1].

- Two Control Arms linked together at upright
- Spring/Damper link to one Control Arm and mount to body
- Small Package

MacPherson Strut:

Figure 2: Basic MacPherson Strut Suspension[1].

- Single Control Arm at bottom, Spring/Damper mounts to top of upright
- Larger package, more dependent on chassis of car

Multi Link:

Figure 3: Basic Multi Link Suspension[1].

- Multiple (usually 4) separate control arms
- Geometry changes with steering

Mercedes Project One Suspension:

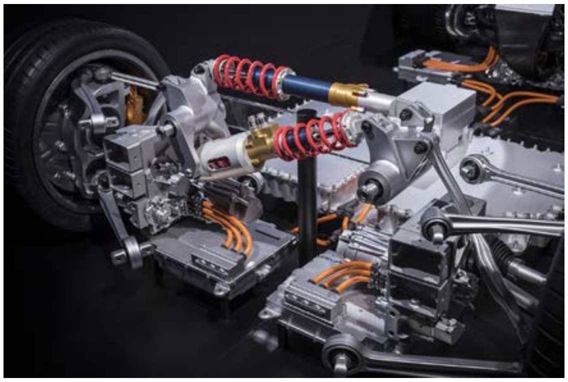

Figure 4. Mercedes Project One Suspension[2].

- Unique variation on Multi Link suspension
- Inboard Springs/Dampers
- Anti Roll capabilities
- Large Package

Divergent 3D’s 3D-Printed Suspension:

Figure 5. Divergent 3D’s Suspension[3].

- Unique variation on Multi-Link Suspension
- 3D Printing allows for optimized organic shapes

PROVE Lab Project 1 Suspension:

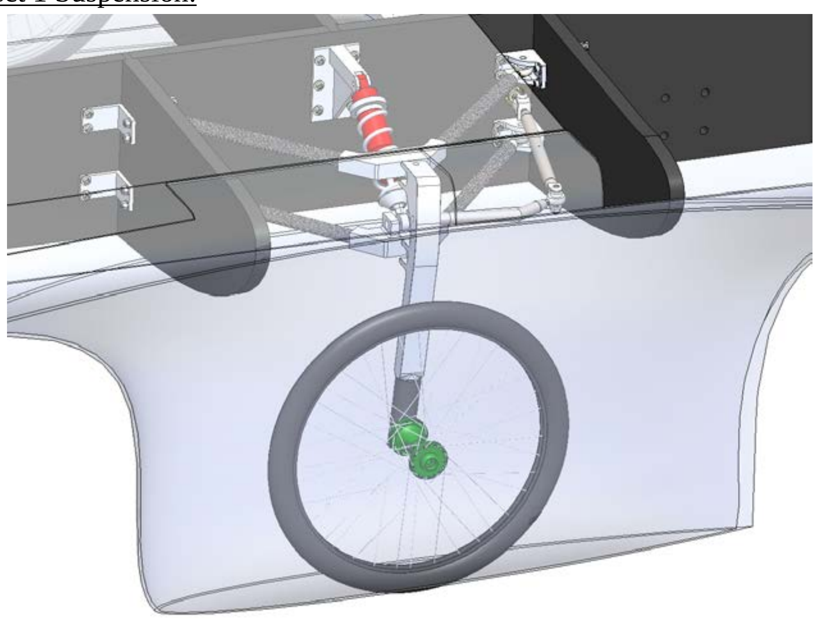

Figure 6. PROVE Lab Solar Car Suspension.

- Double Wishbone Design
- Carbon fiber control arms
- Unusual Packaging

Further details can be found in Appendix A.

2.2 Patent Search:

A detailed patent search on unique suspension systems was done and two patents of note were found. There were not many applicable patents to this challenge which frees the design of limitations. One detailed an independent suspension design with a torsional stiffness control device placed between the damper and strut allowing rotation relative to each other about an axis of articulation with an adjustable torsional stiffness[4]. This effectively improves vehicle stability as far as traction and braking. The second patent researched was a suspension for terrain vehicles with an adjustable track width for variable environmental conditions[5]. Though these designs are not directly correlated to the exact customer needs, it is a good example of an adjustable system that could be looked to for reference.

2.3 Technical Literature:

ASTM Testing Standards

ASTM's composite standards are instrumental in the evaluation and determination of the physical, shear, tensile, flexural, and compressive properties of various forms of composite materials used in structural applications. [6] The ASTM testing standard were referenced throughout the testing procedure of this project. They provide guidelines for the proper fabrication of test samples and correct testing procedure to yield quality results.

How to Make Your Car Handle [7]

How to Make Your Car Handle by Fred Puhn offered standard design criteria for the project such as designing suspension linkages to a 5g bump. This is also the loading case that Formula SAE designs to.

Race Car Vehicle Dynamics [8]

Race Car Vehicle Dynamics by William Milliken was heavily used in the geometry and design phases. It also offered deep theoretical background with equations to calculate loads for multiple loading cases.

Engineering Mechanics of Composite Materials [9]

Engineering Mechanics of Composite Materials by Isaac M. Daniel and Ori Ishai is a textbook used in the Composites class at Cal Poly. It aided in the composite analysis and design process. The equations and theory offered in this textbook was coded in MATLAB.

Chapter 3, OBJECTIVES

3.1 Problem Statement

Current consumer electric cars have a range too short to fully replace gas-powered cars. Cal Poly Prototype Vehicles Laboratory (PROVE Lab) is in need of a proof of concept for the front suspension of their upcoming endurance car project. The design must be innovative in terms of system design, manufacturing, or other aspects without sacrificing performance.

3.2 Boundary Diagram

The main objective of the boundary diagram is to give a physical idea of how the suspension system interacts with the rest of the car and environment. Figure 7 below shows a boundary diagram with the front suspension circled in red. From the figure, it can be see that the system will interface with the chassis, wheels, and steering of the car. The optimal environment is on a track and the car is aimed to go 1000 miles before needing another charge.

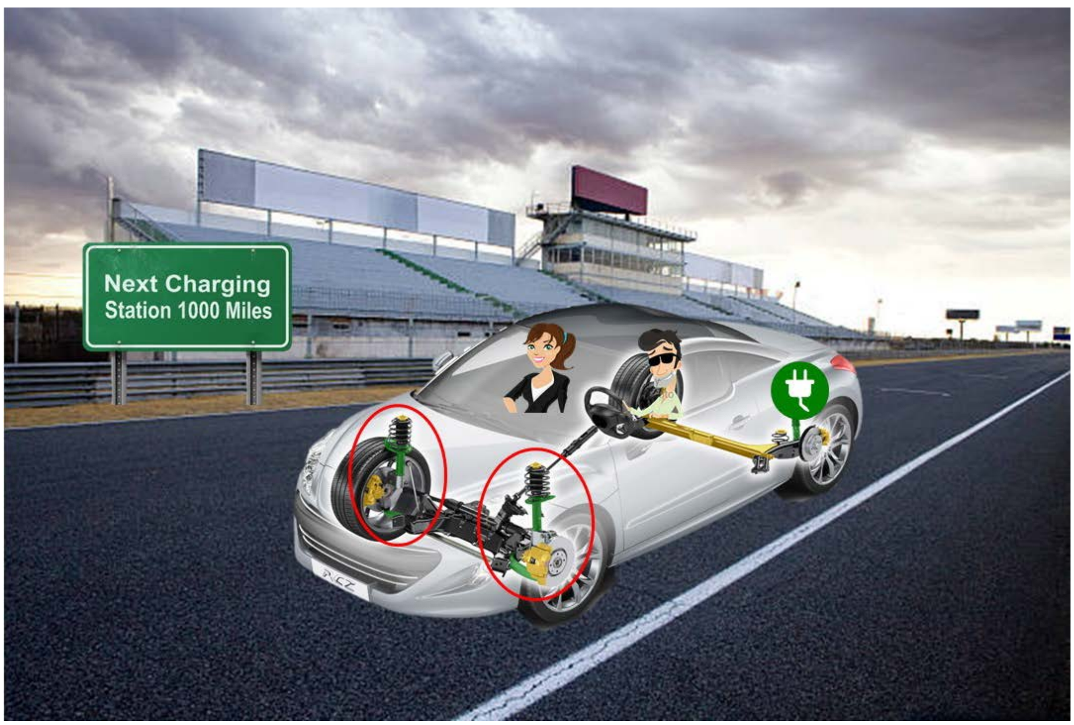

Figure 7. Suspension Boundary Diagram

3.3 Summary of Customer Needs/Wants

After meeting with Dr. Doig and attending the PROVE team meetings, the team was able to capture the vision of what Dr. Doig needs and wants. His key needs include the suspension being able to hold up the weight of an electric vehicle and two people, incorporation of innovative design or manufacturing process, having the size and feel of a real car, and the car being able to travel at highway speeds.

3.4 Quality Function Deployment (QFD) Process
The goal of the QFD is to focus the design by taking Dr. Doig's needs and wants and turning them into measurable engineering specifications. The team began by prioritizing them, giving each requirement a relative importance to each other. The competition was then benchmarked by rating how well competing suspension systems met the customer's requirements. By benchmarking, it became clear which systems met his requirements well in order to build off of those ideas. Engineering requirements were created to match customer requirements. These are all measurable in some aspect. Customer requirements were then related to the engineering specifications. This was an important step in the process because it determined whether the problem had been under or over specified . The last step in the process is setting the engineering targets. This turned the specifications into quantities that could measure, and created targets with the purpose of satisfying the customer's needs and wants. The full QFD can be found in Appendix B.

3.5 Engineering Specifications
The Engineering Specifications Table in Table 1 shows the quantifiable parameters for the project. It assesses the risk of meeting each engineering target and how to complete the compliance check.

Due to the vehicle being in early development, these specifications were derived from similar vehicles including Mazda Miata, BMW Z4, Tesla Roadster, and Tesla Model S. As the vehicle parameters evolve, these should be updated and the system modified accordingly.

Table 1. Engineering Specifications Table. [edited 2/11/17]

Spec #	Parameter	Target	Tolerance	Risk	Compliance
1	Vehicle Weight	2500 lbs	Min	H	A, T
2	Vertical Travel	3-6 in	Range	M	A
3	System Weight	15 lbs	Max	H	A, T
4	Track Width	60 in	+/- 10	H	S
5	Travel Speed	88 mph	+/- 10	H	A, T
6	Wheelbase	100 in	+/- 10	H	S
7	Roll Center	15.5 %	+/- 1%	M	
8	Caster Angle	4 deg	+/- 1 deg	M	
9	Scrub Radius	0.4 in	+/- .050 in	M	
10	Camber Gain	50%	Min	M	
11	Chassis Width	25in	Min	H	
12	Ground Clearance	6in	Min	H	

3.6 Verification of Each Specification

Vehicle weight: The system was structurally designed to support the loads it will experience with this vehicle weight. If the vehicle weight increases significantly (More that 10%) the system will need to be reinforced.

Vertical travel: The system clearances have been validated at its current geometry to accommodate 3in of vertical travel in either direction.

System weight: The uprights and wishbones weigh 4.3 lbs

Track width: The system will be designed to a track width of 60 in and this will be measured upon install of the chassis.

Travel speed: In order to insure that the system will be able to handle highway speeds, the loads resulting from a vehicle of at least 2500 lbs hitting a bump or hole at about 88 mph will be estimated. From there, the geometry will be analyzed based on stress analysis and vertical travel requirements.

Wheelbase: The wheelbase of the vehicle will be 90 in and will be measured upon installation.

3.7 High Risk Specifications

Vehicle weight: One of the main jobs of a suspension system is that it has to be able to hold the weight of the car. This is high risk because if the target of 2500 lbs is not hit, the suspension could fail under premature loads.

Vertical travel: Although the test track will not have many bumps or holes, the suspension should be able to handle them without losing traction with the road. This is why there is a set a vertical travel parameter. The road conditions should not be felt in the cabin, resulting in an uncomfortable ride, or affect performance.

System weight: Because the main objective of the car is to go a very long distance on a single charge, weight is crucial to the efficiency of the car. Therefore, the front suspension should not weigh more than 25 lbs.

Track width: An important customer requirement is that the car be able to hold two people. An an average track width of 60 in will provide enough space for two people to sit in the car.

Travel Speed: Dr. Doig is unimpressed by a car that can go 1000 miles at a low cruising speed. The record to be set will be with the car going at highway speeds, meaning the suspension has to be able to absorb bumps at that speed. Design for low speeds, could cause the system to fail at high speed loads.

Wheelbase: The wheelbase was decided by PROVE Lab’s other subteams. All teams must design to a wheelbase of 100 in.

Chapter 4, INITIAL DESIGN DEVELOPMENT

4.1 Initial Concept Development Process

Concept development began with basic brainstorming methods. Columns headed by design decisions yet to be made were created such as spring type, configuration, and interface. Sticky notes were distributed and each team member was encouraged to write down the first concept they could think of to suite the given categories created. During this process, quantity of ideas generation was emphasized over quality. Therefore, no evaluation or criticism of ideas occurred during this process. The results can be seen in Figure 8.

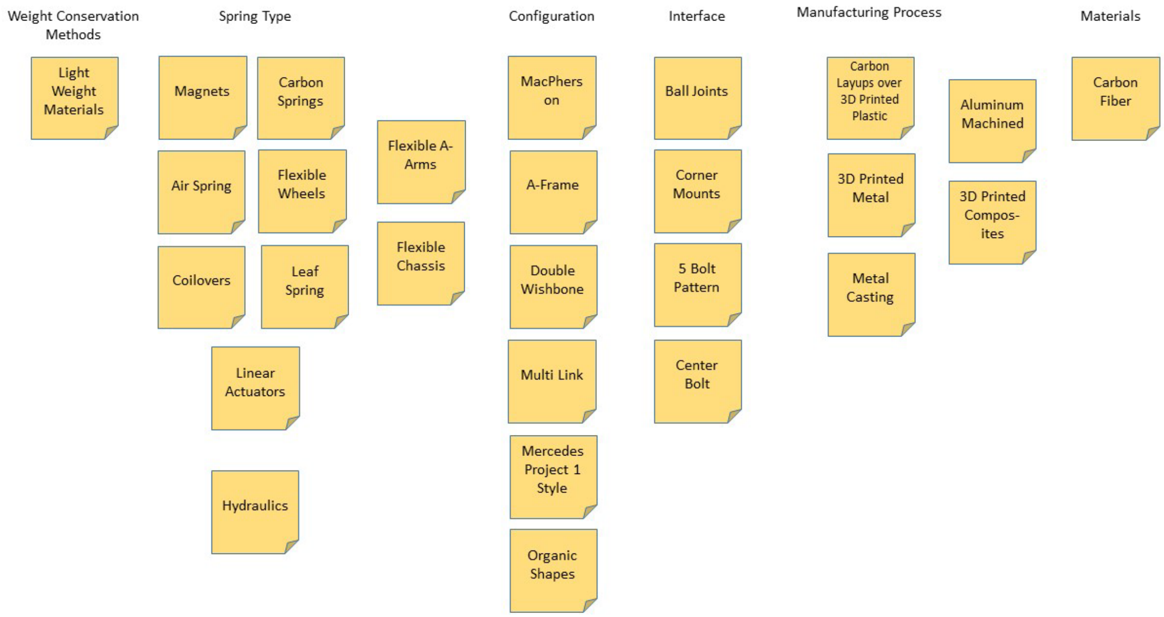

Figure 8. Results of First Brainstorming Session.

The concepts generated and were then reorganized according to applicable design requirements. It was discovered that there was significant overlap between ideas and requirements. For example, in the innovative and lightweight columns solutions involving composites were popular and the ideas were grouped into a more specific category. To highlight this, the groupings were outlined and designated with a name. The groupings can be seen in Figure 9.

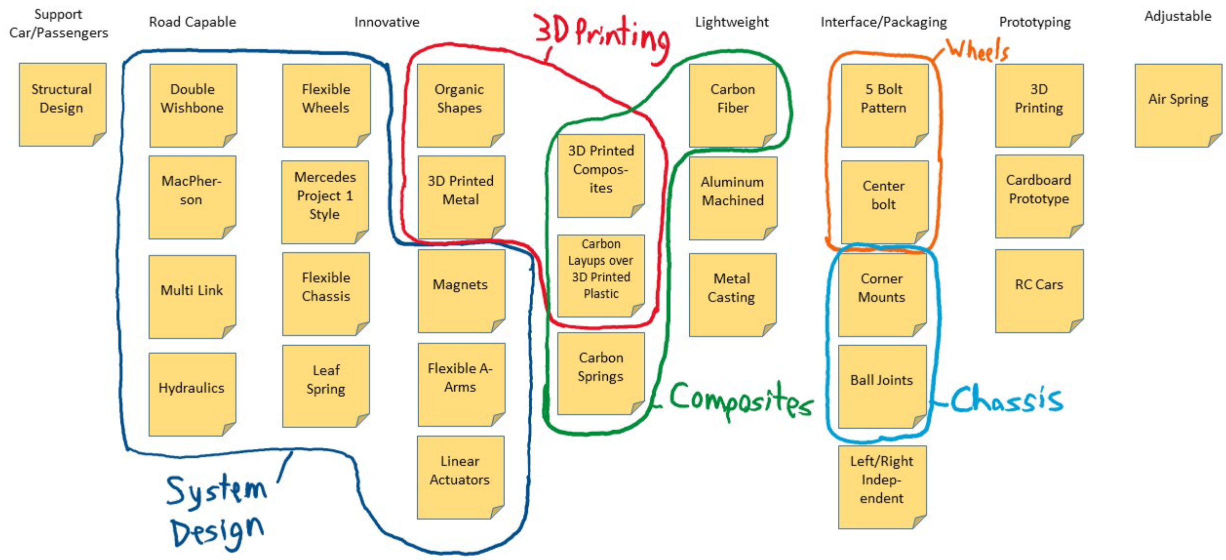

Figure 9. Rearrangement of First Brainstorming Session by Design Requirements.

Lastly, a functional decomposition was conducted in order to break down the functions of a suspension system into smaller categories which could be analyzed in greater detail. First the most general function was defined as follows: the suspension must support the weight of the chassis and two passengers. From here the following sub functions were developed that can be seen heading the three columns below. Sub functions falling under these three categories were then developed until the functions could not be broken down any further. This can be seen in Figure 10.

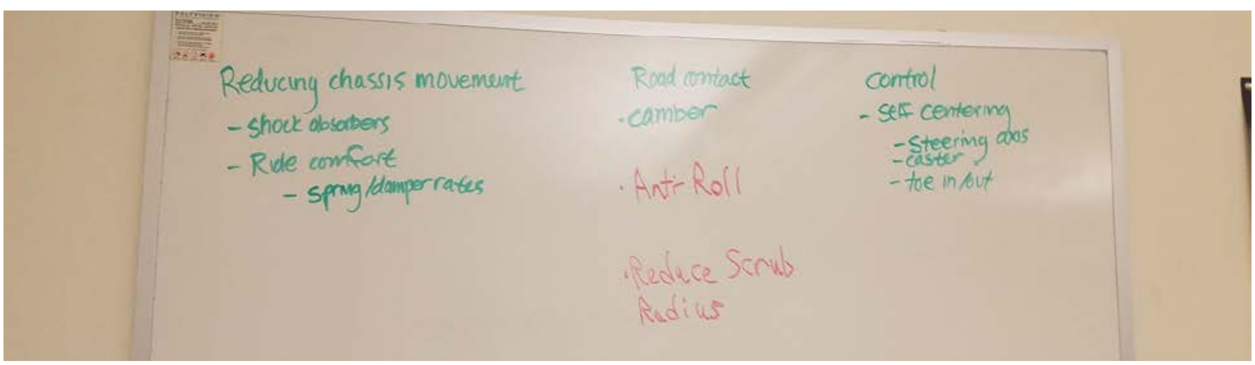

Figure 10. Results of Functional Decomposition.

From these ideation sessions, the three concepts that would be considered for the final design were selected. These suspension configurations included MacPherson, double wishbone, and multi-link. Each design was then evaluated and compared to one another in the following step in the design process, which is the concept selection process.

4.2 Initial Concept Selection Process & Results
The full decision matrix of the front suspension system can be found in Appendix C. The MacPherson suspension system was chosen as the datum for the decision matrix because it is the most common front suspension and the most simple in terms of design. The following criteria was used to compare the concepts: lightweight, performance, adjustable, packaging, innovative, simplicity, and manufacturability.

The lightweight criteria is defined as the weight of the final system. This is affected by the number of components the type of suspension has. Both the double wishbone and multi-link are heavier than the MacPherson based off of quantity of parts and research of average weights for each system.

Performance encompasses road contact, self alignment, ride comfort, shock force absorption, etc. The multi-link scored the highest in this category because it allows the steering axis to change when cornering to match the moving contact point in roll. This minimizes the scrub radius and keeps it constant through corners.

An adjustable systems means that it can be adjusted according to other specifications of the car made at a later date. The design should not be constrained to a certain configuration. The multi-link is advantageous over the MacPherson and double wishbone because one parameter in the suspension can be altered without influencing the rest of the assembly. This is one of the biggest advantages of a multi-link system.

Because the car is still in a concept phase and very subject to change, the packaging of the system is important. The interface points should have minimal interaction with other systems. The MacPherson needs a vertical strut which constrains the chassis design. The double wishbone and multi-link are both systems that allow for an angled strut placement.

The innovative criteria is about the creativity of the design. The sponsor made it very clear that the last thing that he wanted was an off the shelf, everyday suspension system. The MacPherson was chosen as the datum because it is the most typical suspension on economic cars. Although the double wishbone has a little more room for innovation, the multi-link is by far the most innovative. There are many ways a multi-link can be designed with the least amount of constraints.

The simplicity criteria includes the design time and engineering effort that the system will consume. The MacPherson would most likely take relatively little effort to design because it is already a very established design. The double wishbone is also a very established design and is not very complex, as the concept of how it moves is simple. The multi-link however, is by far the most complex system and will require the most time and effort in designing. Because the multi-link is so adjustable, there are many ways to design the system based off of the functionality.

Manufacturability is how easy the parts will be to manufacture. Some questions that arise when dealing with manufacturability are: Will the manufacturability restrain the design? How many parts need to be manufactured? By what means will they be manufactured (CNC, manual, printed, etc)? The MacPherson has one main frame. Based off of PROVE’s Project 1 car, the double wishbone is very difficult to manufacture because of the odd angles that both wishbone frames had to be at. The multi-link’s most difficult part to manufacture would be the upright, which is still easier than the wishbones.

4.3 Initial Concept Models/Prototype

A concrete concept model of the multi link configuration was created. This model included a 3D printed upright with four individual links attached; two adjustable tie rods and two PVC pipe. The system was fastened to wood in order to simulate the interface with the chassis. Images of this model can be seen in Figure 14. This functional model gave the team the opportunity to get hands on with the multi link design. Prior to the model, the movement of this design was not fully understood. Being able to physically manipulate this model with movement that a real suspension would experience on the road hitting a bump or cornering, proved to be vital. Most importantly this model demonstrated the unique cornering capabilities of this design, which will be discussed in greater detail in the following sections.

Figure 14. Final Multi Link Concept Model.

4.4 Initial Design

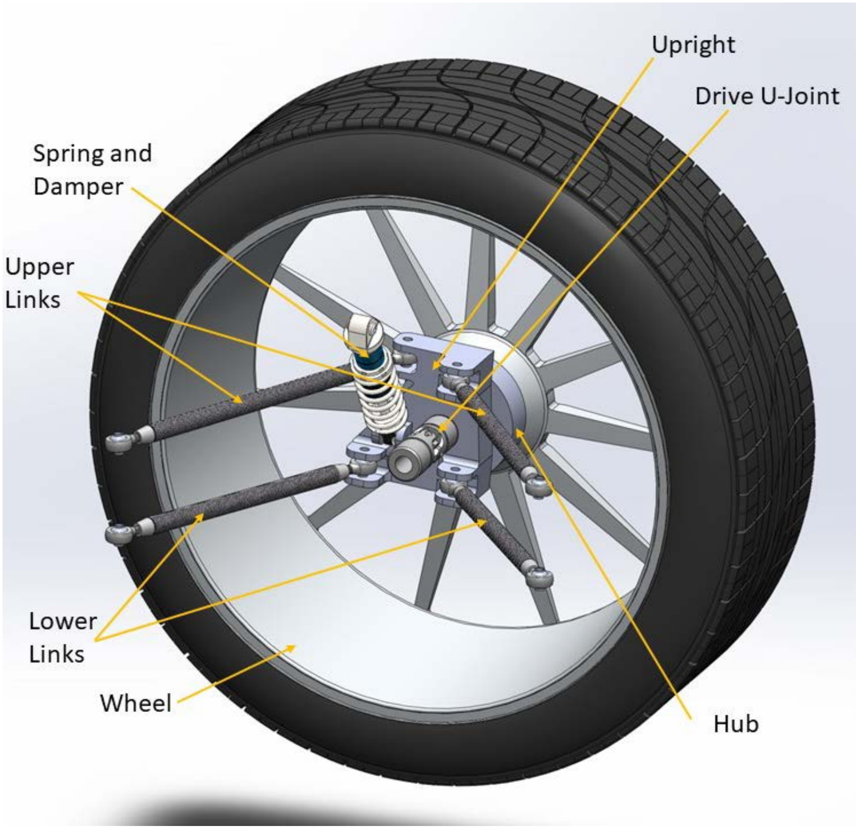

Figure 15. Concept CAD

The concept chosen is a multi-link system. The initial design drawings can be found in Appendix D. Although this system isn't the most lightweight and is very complex, its advantages outweigh its few downsides. This design is advantageous in several areas which made it a clear winner in the selection analysis. This suspension design is the most innovative, as it is mostly used in high-performance cars due to its involved geometry. The geometry and simple linkages give freedom for creativity and innovation. The adjustability of a multi-link improves the performance of this system. It features independent static adjustability and performance advantages such as being able to adapt dynamically. The packaging of a multi-link is minimally invasive to the other subsystems. It also allows for many shock and damper configurations. In addition, this system is reasonable for manufacturing as it makes use of simple linkages.

The remaining tasks in finalizing this design include geometry calculations and material selection. The concept behind the geometry is explained in section 4.5, but further research on tire deflection and tire wear will be necessary to finalize the geometry. Further analysis is also needed to determine the type of material used. If composite materials are selected, sufficient testing will need to be performed on the materials used in the system to determine their properties.

4.5 Initial Design Functionality

Multi-link suspensions have the advantage of being able to independently change parameters under dynamic conditions. One example of this is with the steering axis. When a car enters a turn, it experiences roll, which moves the center of the contact patch of the tire either inward or outward seen in Figure 16. Since a minimized scrub radius is desired because it reduces tire wear and the energy lost when turning, it is advantageous to achieve a constant scrub radius in cornering.

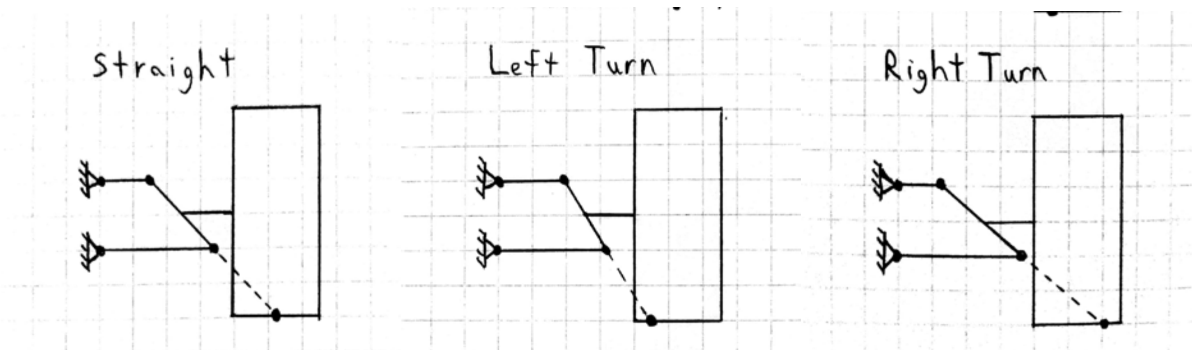

Figure 16. Movement of the Steering Axis to Maintain Constant Scrub Radius in Cornering

A multi link suspension achieves this by the use of four bar linkages. The linkages can be designed such that when the wheel is turned, the rotation point moves in or out. To achieve the desired steering axis change, the upper and lower linkages should move in opposite directions in cornering. Defining the final geometry will require further analysis in tire deflection in cornering and body roll, as discussed in greater detail in the next section.

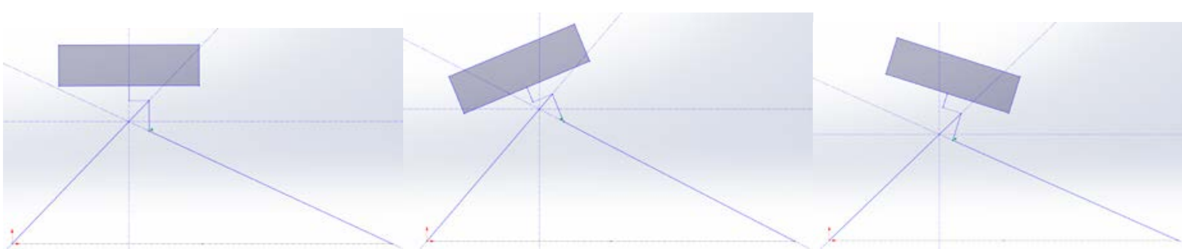
Figure 17. Moving of the Pivot Point When the Wheel is Turned

4.6 Risks and Unknowns with Initial Design

Although the multi-link suspension has many benefits, it also poses risks and unknowns. At this point in the project, the biggest unknown is the geometry of a multi-link. This system has far more complicated geometry than that of the two other suspension designs analyzed. While it offers more room to be creative, this also means that there is less literature to reference. The shock choice and placement will also rely on the geometry.

As the group is looking into innovative materials and manufacturing processes, this raises an unknown about how the finite element analysis will be performed. The system will need a lot of structural testing, but the unknown is whether time should be spent doing FEA or destructive testing. The complexity of the FEA may require the team to rely more on experimentation to reach the desired parameters because there is no standard test procedure. Prototyping is expected to be a significant part of the design process moving forward.

Chapter 5, FINAL DESIGN

Following PDR, the team did further research into the multi-link design, visiting car shows and talking to others who have attempted this suspension configuration. It became clear that this design was far more complicated than expected and could not be completed within the senior project time frame. Instead the focus of the project was turned to innovative manufacturing with a simpler, tried and true, suspension design: double wishbone. This configuration scored second after the multi-link in the decision matrix discussed previously and was expected to provide equal performance capabilities. The incorporation of composites into the design was highly encouraged by the project's sponsor. It was decided to focus on one component in the system with the most potential for composite use, as well as the most benefit. The upright is a large, fairly heavy component in the system and would allow for the most weight reduction in the suspension assembly. Following consultation with teachers and students familiar with composites, it was determined that the upright could be made from carbon sandwich panels. Moving forward, this upright will be the focus of analysis, testing, and manufacture. The development and selection of other components such as the wishbone joints and rods, mounting brackets, ball joints, etc. will also be detailed in this report, but in less detail.

5.1 Overall description & layout

The final design consists of the following components shown and labeled in Figure 18. This includes an upright manufactured as a carbon fiber sandwich panel, with potted inserts for fixturing components to it. The hub is mounted in a four bolt pattern. The wishbone joints are machined aluminum and attached with G-Flex adhesive to the carbon rods. Threaded inserts are fitted and adhered inside the opposite end of the carbon rods to allow the ball joints to thread inside. This allows the rod lengths to be slightly adjustable. The strut, is mounted to the lower wishbone and was donated by Öhlins.

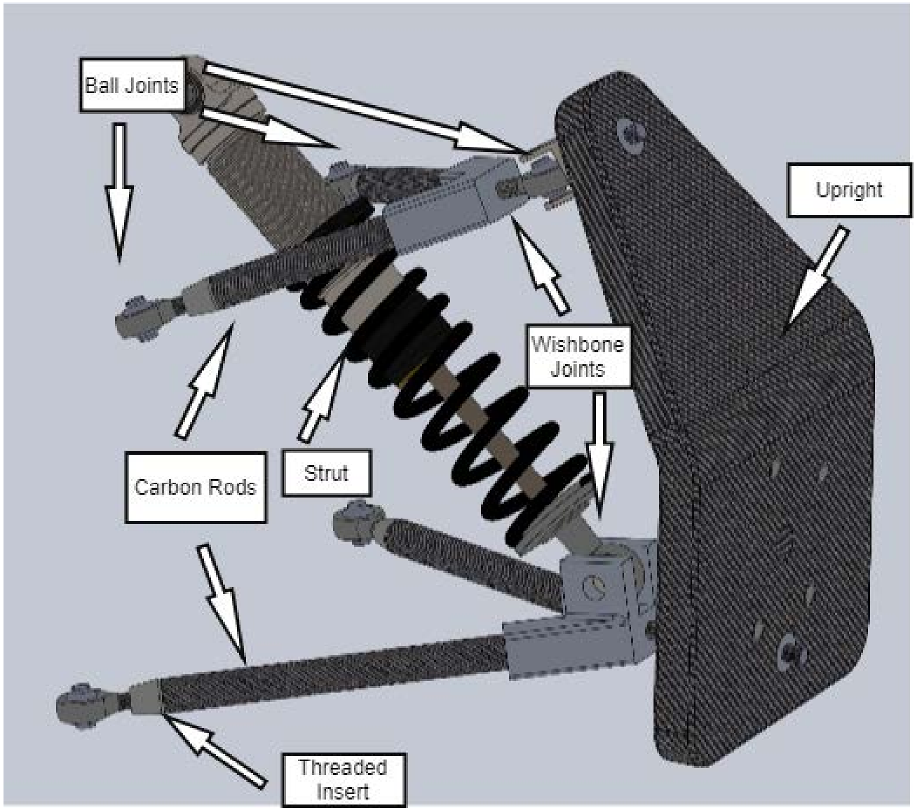

Figure 18. Final Design Label Solid Model

5.2 Detailed design description
This section will discuss the details of each component of the final design. Part drawing of manufactured pieces can be referenced in Appendix E.

<u>Upright Assembly</u>
The most crucial component in this suspension is the upright assembly shown in Figure 19. The geometry of this part was determined by the overall suspension geometry needed to satisfy desired performance objectives, including roll center, camber, caster, etc. Also, the upright was laid carbon fiber, so the geometry had to be a simple shape to allow for easier manufacturing. As mentioned previously, the upright will be constructed from carbon fiber sandwich panels composed of the following: nomex flex core and Hexcel. A quasi-isotropic laminate orientation was used. The number of plies and amount of core within the upright were manipulated to strengthen the part in certain areas according to the loading it will experience in that location. Potted inserts are adhered into the sandwich panel with epoxy allow for fastening points, as threaded holes cannot be machined into the sandwich panel. The upper wishbone joint mounts at the top potted insert and the lower wishbone assembly at the bottom center insert. The brake mounts with the two potted inserts aligned vertically on the far right. Lastly, the steering knuckle mounts to the insert in the far left, bottom corner of the upright. Mounting holes around the hub mount will be made using G10 tubing.

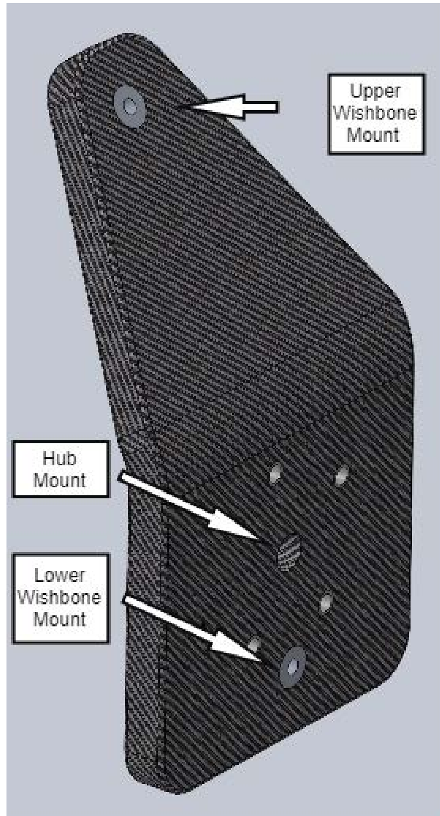

Figure 19. Model of the Upright Assembly Final Design

<u>Wishbone Assembly</u>

The wishbone assembly seen in Figure 20 is composed of four parts: wishbone joint, carbon rods, threaded insert, and ball joints. Both wishbone joints angle are set at 60°, an angle convenient for packaging and manufacturing. The lower wishbone joint has an additional strut mount feature. These wishbones are machined from 6061 Aluminum. The large hole on the arms of the wishbone joint will be fitted around the carbon rod and adhered. Carbon rods were supplied from Rockwest Composites. A threaded insert was adhered inside the opposite end of the carbon rod to insert the ball joints. The length of the rod can then be variably adjusted at this joint.

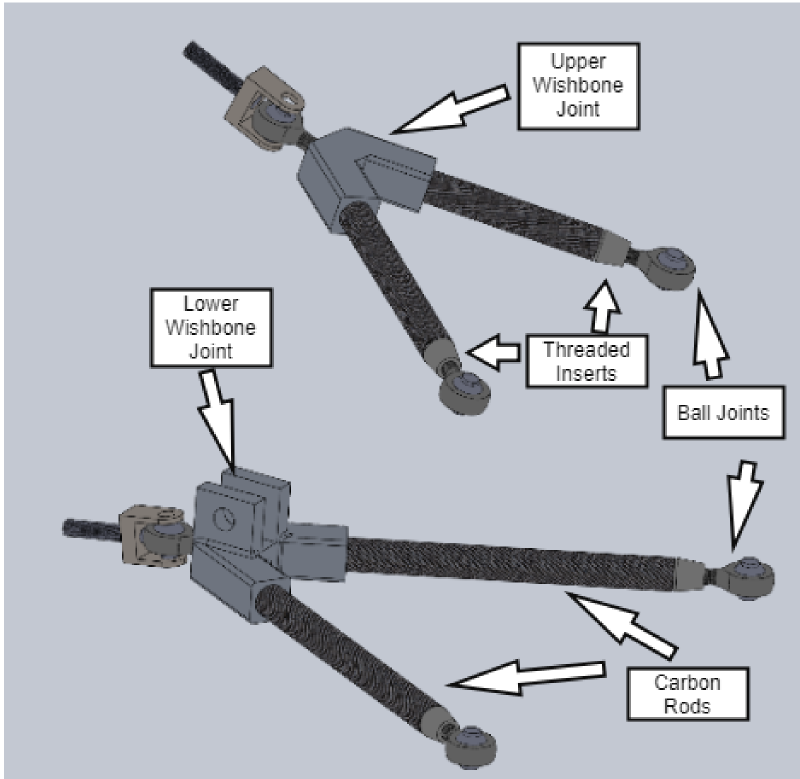

Figure 20. Model of the Final Wishbone Assembly

<u>Hub</u>

The hub selected for this initial design is that of a 1998 Toyota Camry. The four hole mounting scheme integrates well into the design of the upright. As the hub is very heavy however, it is recommended that the team develops a custom hub.

<u>Strut</u>

Öhlins donated a HD 022 strut to our project. This shock does not have externally adjustable damping, but can be adjusted by shim stacking if desired. It will be effective for at least the first iteration of the car. Once the spring rate is finalized, a carbon fiber spring is recommended to reduce weight.

5.3 Analysis description & results

<u>Loading</u>

For the quantification of the suspension loads, the three load cases that were analyzed were bump, cornering, and braking.

Bump: From Fred Puhn's *How to Make Your Car Handle* [7], the industry standard for a bump load when designing suspension linkages is 5G. A quarter-car simulation model in MATLAB was used to verify this. The simulation over a typical road profile resulted in a max load of approximately 4G, verifying the industry standard as a conservative load condition that will be used for the design.

Cornering: The track that the car will be tested on has a one mile radius turn, so the car is not expected to be subjected to high cornering loads. However, the design was based off of typical road conditions. Two cornering loads were considered: the standard highway on ramp which was about .3G and the max cornering load before the car will start to slide which was .9G. These accelerations were related to the weight transfer to the outer wheels and the load was calculated at the contact patch and it was assumed that there was an initial 50:50 weight in the front and rear of the car. The tip load was not taken into consideration because for an electric car, the center of gravity is low and tip is not expected.

Braking: Limited by tire adhesion, the maximum deceleration was calculated to be 12 ft/s^2. Using this, the max moment applied to the upright from the brakes was 400 lb-in. See Appendix F: Brake Calculations.

Individual Component Loading:
The loads on individual components were solved for using statics. The calculations are detailed in Appendix F: Component Loading.

Upright Stress Analysis:
To quantify the approximate internal stresses in the parts, the parts were modeled as steel and had the individual component load cases applied. For bump and cornering, the upper wishbone joint was modeled as a roller and the bottom as a pin. This provided a starting point for the load cases in the carbon fiber. These load cases were then inputted into Tsai-Wu failure criterion and checked for factors of safety, designing for a minimum factor of safety 2.5. Our hand calcs for these gave us a starting point for number of plies, but we used our testing data for structural verification.

Wishbone Stress Analysis:
For the metal parts in the wishbones, FEA was done by applying the calculated loads for each case. The team designed for a minimum factor of safety of 1.5. At the most extreme load case of 5g bump, the max stress in the aluminum 27 ksi, which with a yield strength of 40 ksi, this provides a factor of safety of 1.5 From this same FEA model the max stress in the carbon tubes was calculated to 46.3 ksi, which the carbon tubes can easily withstand given their yield strength of 670 ksi.

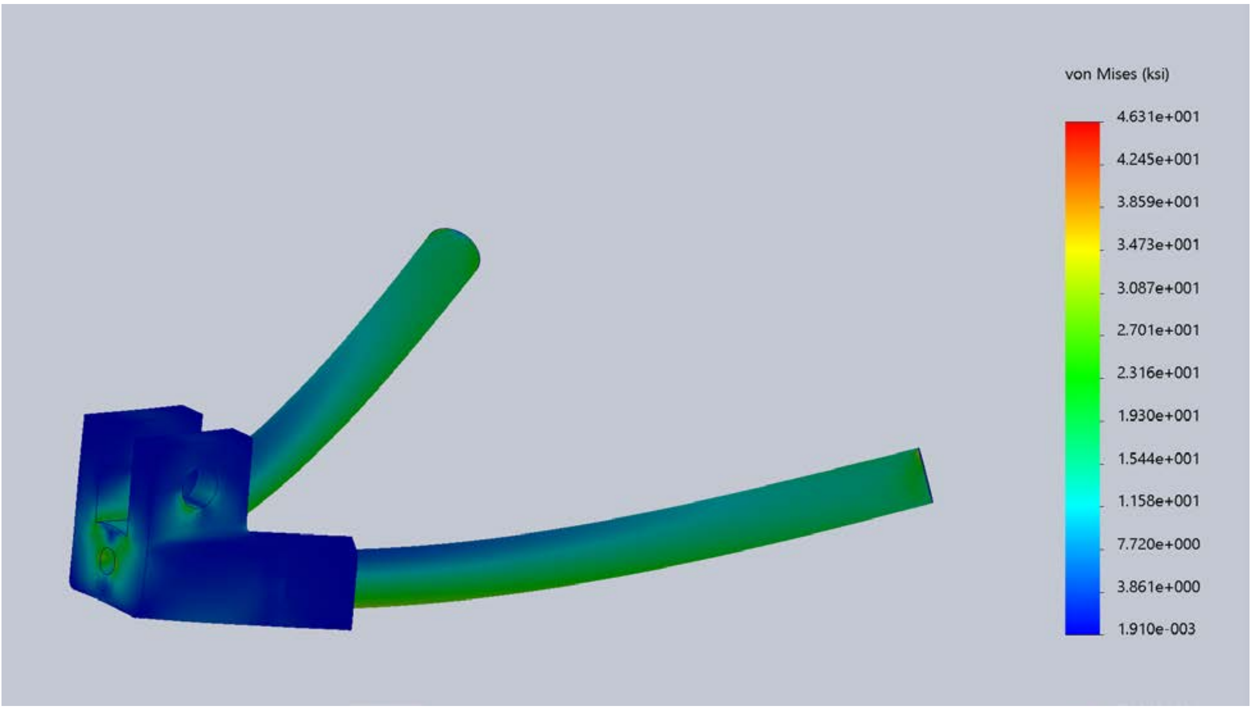

Figure 21. FEA of Lower Wishbone Joint in Cornering

5.4 Cost Analysis

The cost analysis includes both purchased parts and materials that will be used in manufacturing parts. These costs were estimated based on research into multiple suppliers. Excluded from the recorded costs are supplies donated by current PROVE sponsors. These include carbon tubes, prepreg, core, and adhesives. A table of the full cost breakdown can be see below in Table 2.

Table 2. Cost Breakdown

Item	Cost per unit	Quantity	Total Cost
Carbon Tube	$132	1	$132
Ball Joints	$10	12	$120
Threaded Inserts	$5.30	10	$53
Fasteners	-	-	$50
6061 Aluminum Round Rod	$32	1	$32
6061 Extruded Aluminum Rectangle	$41	1	$41
6061 Extruded Aluminum Rectangle	$13	1	$13
1018 Steel	$23	1	$23
Threaded Steel Rod	$9	1	$9
Garolite Tubes	$47	1	$47
Cost of Prototype			$520

The greater majority of the cost is the coilovers, followed by the ball joints, aluminum, and hubs. This total falls well within the range discussed with the sponsor.

5.5 Explanation of material, geometry, & component choices

Materials

With weight as a primary concern, carbon fiber was used as much as practical. The upright was selected to be a carbon fiber sandwich panel due to it requirement to support multiple load conditions. Sandwich panels can be designed to support specific loading conditions by variation of core/laminate ratios. Due to the complex loading, a quasi-isotropic laminate was required. PROVE lab had a 0°/90° weave available. Therefore, to be quasi-isotropic, the laminates were constructed with alternating 0° and 45° plies. A depiction can be seen in Figure 22.

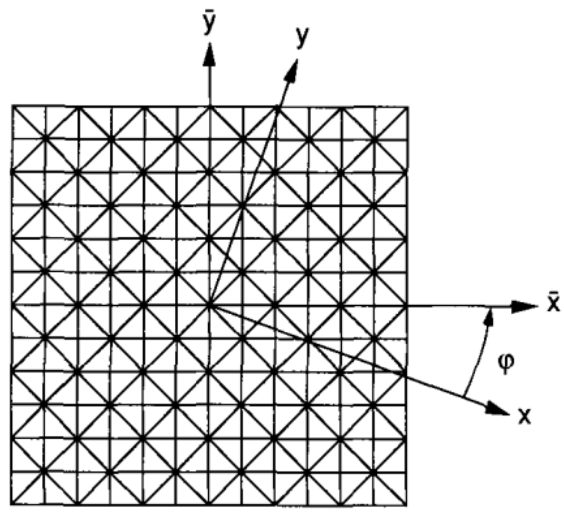

Figure 22. Quasi-isotropic Laminate [10]

The wishbones were selected to be a combination of carbon fiber tubes and machined aluminum. Using the carbon tubes for the wishbones was fairly simple given known material properties and previous use on the PROVE solar car front suspension. Making the wishbone joints out of carbon fiber was investigated, but was deemed not worth the risk/design time due to their loading cases being very non-ideal for composites and the weight savings being minimal due to their small size.

Components

The struts were donated so will be used for at least the first iteration. These are very high quality so should be used if they can achieve the desired system parameters.

The ball joints and threaded inserts were selected due to their successful use on the solar car, availability, and adequate strength.

Geometry

The defining parameters for geometry were track width, ride height, ground clearance, chassis width, camber, caster angle, and scrub radius. These vehicle parameters are preliminary due to the early development phase of the car, so the geometry will need to be updated once the vehicle parameters are finalized. To get the approximate geometry, the team used Racing Aspirations, a 2D online geometry calculator. See Figure 23. This was an iterative process, but eventually provided a functional approximate 2D model.

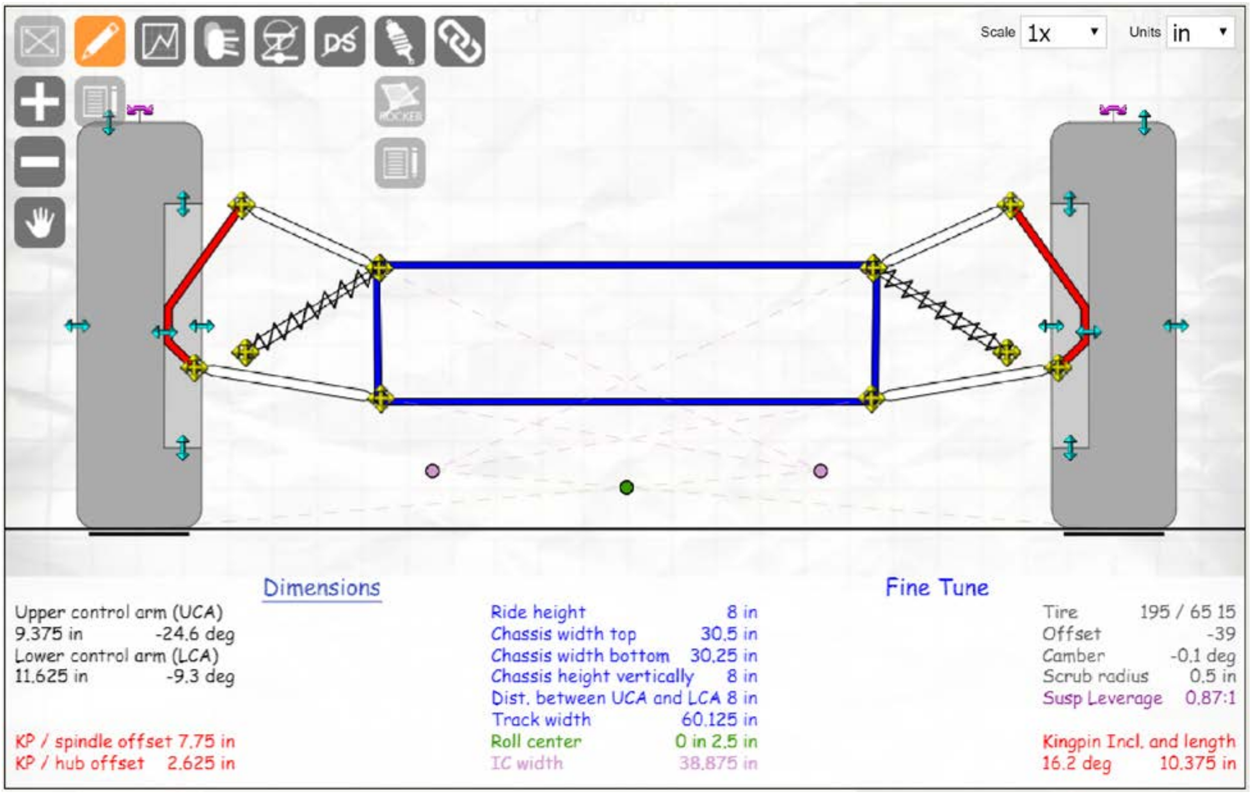

Figure 23. Racing Aspirations Model

The team then moved to a 3D model in Lotus Shark. This required the team to incorporate the caster angle and select wishbone angles. 60 degrees was selected for the wishbone angle as it fit well with the packaging of the chassis. In Lotus Shark the geometry was tested for all the parameters and tweaked them to match more precisely. See Figure 24.

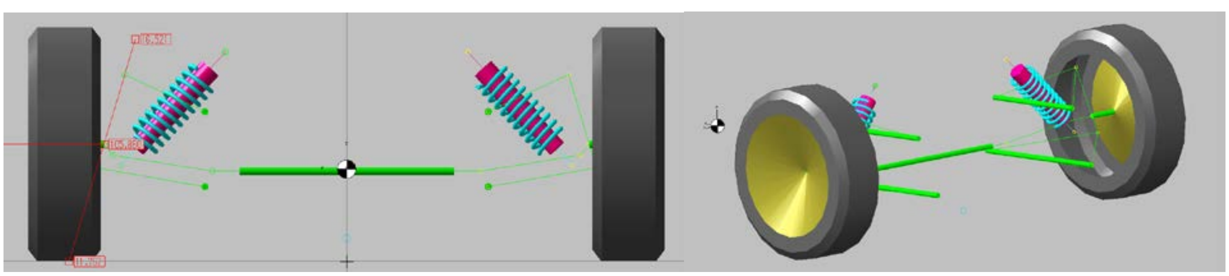
Figure 24. Lotus Shark Model

Following Lotus Shark, the team incorporated the geometry into a 3D line sketch in Solidworks to base the model off of. This verifies that the system matches the intended geometry. See Figure 25.

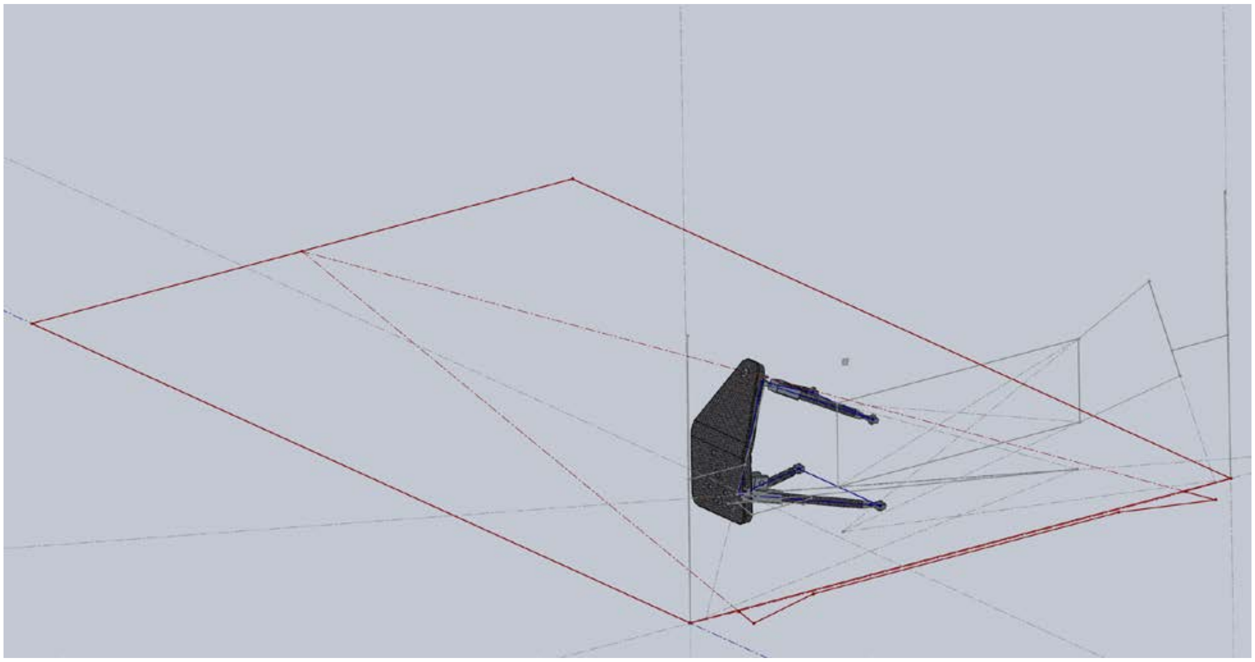
Figure 25. System Assembly with Line Sketch

5.6 Safety, maintenance, & repair considerations
The largest risk in our system is a catastrophic structural failure. A failure could severely damage the car and could potentially injure the driver. For this reason, destructive testing was be done (see Chapter 7).

There are also risks in pinching during assembly, so care should be taken with pinch points. See Appendix G for FMEA

Errors in handling involving spring and damper rates may be encountered but will be able to be adjusted for with the coilovers.

Several of the components, including the strut and ball joints will show wear over time, but given the relatively low milage this car will experience, replacement will not be necessary.

See Appendix H for Operator’s Manual for further instruction on maintenance or repairs.

Chapter 6, MANUFACTURING PLAN
6.1 Procurement
PROVE lab has current sponsors that donated a significant amount of the materials needed for manufacture. The carbon fiber weave was donated by Hexcel. The nomex core was donated by Westerly Marine. The laminating resin and hardener were donated by Pro-Set. These three resources will be vital for the manufacture of the sandwich panel upright. The struts were also kindly donated to the project. Lastly, all epoxy will be met with the ProSet sponsorship.

Other materials needed for manufacturing were purchased. Aluminum needed for wishbone joint and potted insert manufacture as well as steel for clevis manufacture were be purchased from OnlineMetals.com.

Purchased parts have been sourced from a variety of vendors. Threaded inserts, threaded rod, fasteners, ball joints, and G10 tubing were purchased from McMaster Carr. Carbon tubing was purchased from Rockwest Composites.

Links to vendors and further purchasing details can be found in Appendices I and J.

6.2 Manufacturing

<u>Wishbones</u>

The wishbone joints were machined out of aluminum. The profile and major features were done with a CNC machine and the holes, including the tube mount holes were machined by hand. Machining tube mount holes required an angle vice and a boring bar to achieve the correct inner diameter. The tubes were marked and cut to length on a tile saw.

<u>Potted Inserts/Clevises</u>

The potted inserts and clevises were machined by hand. The clevises were done entirely on a mill, while the potted inserts were started on a lathe and finished on a mill. Images of the potted insert processes can be seen in Figure 26.

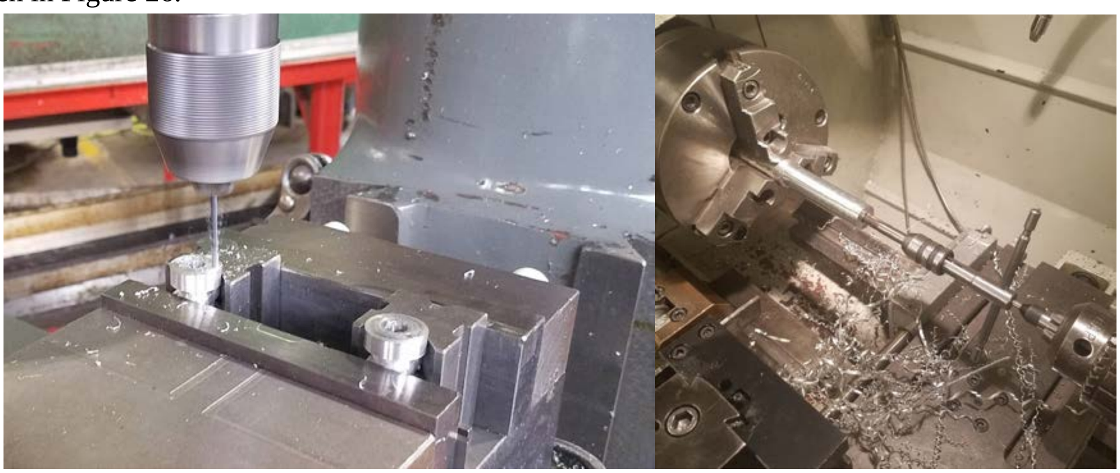

Figure 26. Potted Insert Manufacturing

<u>Upright</u>

Layup Procedure:

The upright was made as a sandwich panel carbon fiber layup. The sandwich panels will be manufactured from bottom to top. Due to material availability we were limited to wet layup with standard carbon weave, however, the use of pre preg carbon and unidirectional layers would be advisable for future iterations. This panels were manufactured by first laminating the carbon fiber sheets with resin, applying the, the nomex flex-core, and then laying the top carbon fiber laminate on top. Figure 27 below shows a visual of how the sandwich panel was constructed.

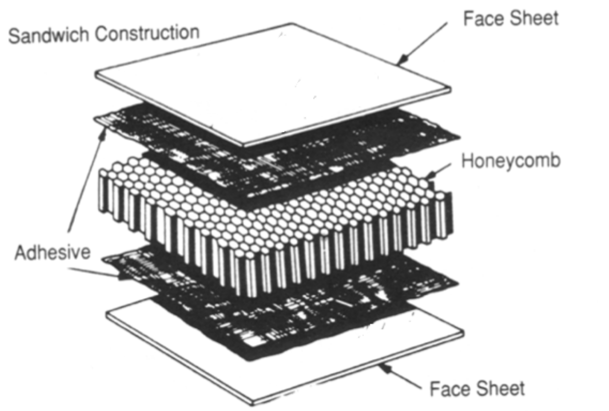

Figure 27. Sandwich Panel Construction for Upright

The number of plies was determined from the destructive testing results. It was shown that 16 plies laminate with 8 ply edge wrap was necessary to support our loading conditions. To achieve a quasi-isotropic laminate, alternating 45 and 0 degree orientations were used. After the sandwich panel was laid up, it was put in a compressive vacuum bag, as seen in Figure 28, to ensure that there is proper adhesion between all of the layers.

To get the proper angle for the upright, a bent sheet metal tool was made on the sheet metal bender. From a manufacturing standpoint, this was functionally effective, but it would be ideal to create a machined foam mold for future iterations to maintain higher angular precision. The nomex core was of the flex-core type so could fit the curve without modification, but the orientation of the core was important as it is only flexible in a single direction. The final profile of the sandwich panel can be seen in Figure 29.

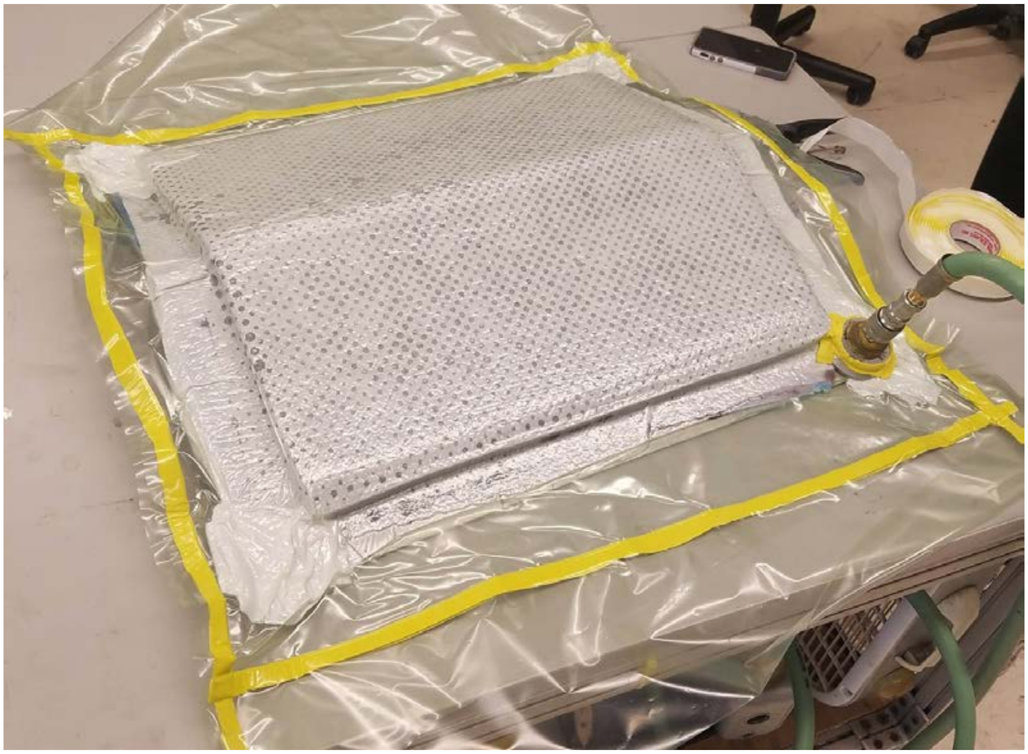

Figure 28. Upright Curing in Vacuum Bag

Figure 29. Profile of Final Upright

Post Processing:

Once cured and unbagged, the exterior shape of the upright was marked on the panel and it was cut to shape on a tile saw.

Edge Wrapping:

In order to prevent delamination of the sandwich panels, an 8 ply edge wrap was necessary around the entire part. This process is difficult so should be practiced if the team remakes the uprights. Strips of carbon 0.5in longer than the circumference of the part were cut into two strips of each width, 2in, 2.25in, 2.5in, and 2.75in. Starting with the 2in strips, the two strips of the same width were laminated together.

Once laminated, the combined strip was wrapped around outside of the upright and relief cuts were cut with shears at each corner. To accelerate adhesion a heat gun was used until the resin was slightly tacky. Once the edges stuck on one entire face, the process was repeated. Once all layers were on, the upright was placed into its mold and the other side edges were pressed down. Once all adhered, the part was placed into a full vacuum bag with care taken to keep the edges adhered.

Potting the Inserts:
The holes for the potted inserts and hub were marked with a paper template and drilled by hand on a drill press with a series of four drill bits increased in size by approximately ¼" each step, however a higher precision jig for drilling may be necessary for future iterations as the suspension geometry becomes more defined. The potted inserts were be placed in their respective holes and be potted with G-Flex epoxy using a syringe.

6.3 Assembly

For a detailed account of assembly construction, see the Operator's manual in Appendix H.

Chapter 7, DESIGN VERIFICATION

Although calculations and FEA are important steps in the design validation, the most crucial part of validating this project's design is testing. Because manufacturing carbon fiber composites is so variable due to it being a very hands on and human labor intensive process, it is difficult to accurately predict how the carbon parts will handle loads. Potential defects in the manufacturing process include but are not limited to cure parameters, fiber misalignment, service impacts, preparation of the fibers, and delamination of core and skin. Defects in the manufacturing process will lead to strength properties that vary from their published or analyzed values. Therefore, the testing phase of the project is crucial part of the design verification.

The test plan is to apply the maximum loading cases to test panels test panels for four different loading cases with different core to laminate ratios to find the most efficient ratio to handle the applied loads. The potted inserts were made to accomodate the maximum of 20 plies, which could also be used for any other test.

7.1 Destructive Testing

Four destructive tests were performed: Bearing test of potted insert, pullout test of potted insert, tensile test of structural adhesive, and four point bending of the sandwich panel. All testing will be done with an Instron Universal Testing System, which is available through the mechanical engineering composites lab. The Instron Universal Testing System is capable of static testing, including tensile and compressive applications. ASTM testing standards were used for applicable tests, however, some tests were conducted for the suspension's specific loading case without referring to a standard. The tests were designed to be able to use premade testing fixtures available in the composites lab to avoid having to manufacture test fixtures.

<u>Bearing Test of Potted Insert</u>
The bearing test is intended to determine the bearing strength of the potted insert when in the sandwich

panel. Because the sandwich panel was being tested with potted inserts, there was no ASTM standard to follow. A cantilever bearing test was conducted because that is the true loading case that the panel will see. The test was performed by an Instron test machine and the load was applied at a constant rate of .001”/second to provide enough data points to get a smooth graph. The test setup can be seen in Figure 30.

Figure30. Bearing Test Setup for 12 Ply Sandwich Panel

Figure 31 is a load vs. position graph for a 12 ply bearing test. The case designed to was a 5g bump load which produced a 3500lb which, from the plot, is out of the linear region. A 5g bump will most likely never be experienced, where as a 3g bump that produces a 2100lb force is more likely. With this load, the 12 ply will suffice. The mode of failure was in bearing. Only one bearing test was performed because the cantilever setup broke the test fixture.

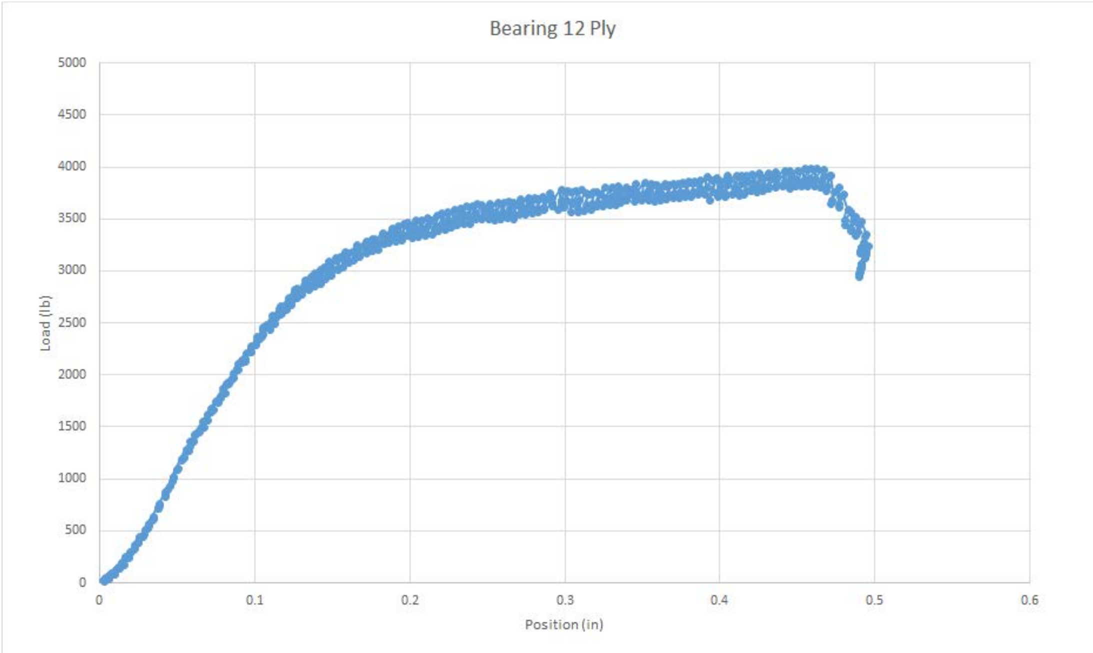

Figure 31. Load vs. Position Plot for 12 Ply Bearing Test

<u>Pull-out Test of Potted Insert</u>

The pull-out test is intended to determine the strength of the potted insert in the sandwich panel due to a normal load to the plane of the panel, also known as a pull-out load. A bolt was placed in the potted insert through the whole thickness of the sandwich panel and through both fixtures. The set up can be seen in Figure 32. A tensile force was applied at a constant rate of .002"/second until failure. The test coupons were made to fit the size of the premade fixtures.


Figure 32. Pullout Test Setup

The results of the pullout test are seen in Figure 33. The max pullout load that will be experienced is 1500lb. From the plot, the max load that the panel could handle exceeds the actual load that the car will experience. The mode of failure was epoxy shear. Three tests were performed with 12, 16, and 20 plies.

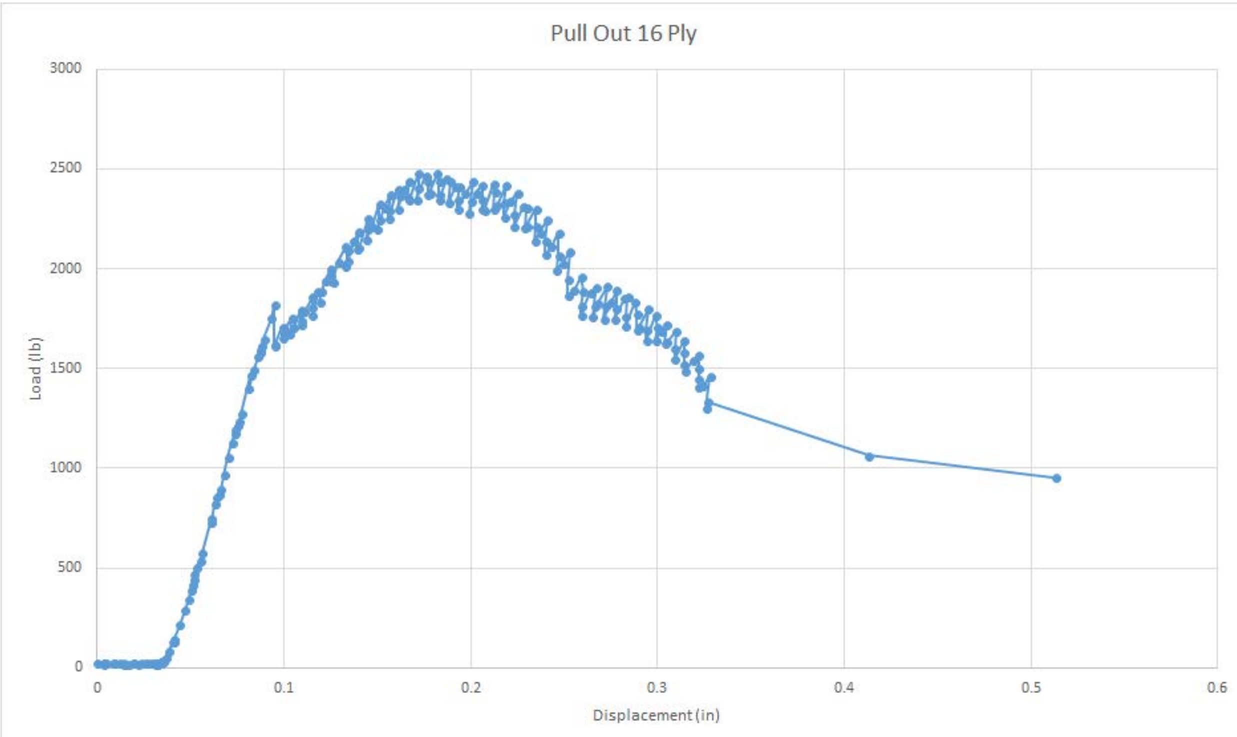

Figure 33. Pullout Test Results for 16 Ply Panel

<u>Tensile Test of Structural Adhesive</u>

A difficult test to model using software is the tensile test aimed to test the shear strength of the G-Flex adhesive. This is largely due to the adhesive bonding the aluminum wishbone joint to carbon tubes. The test will determine whether the aluminum, adhesive, or carbon fails first and provide a complete tensile profile. A mockup of the A arm was made and fixtured to the Instron machine as seen in Figure 34. The tensile load was applied at .001"/second.

Figure 34. Tensile Test Setup

The results from the tensile test can be seen in Figure 35. The worst performing specimen failed at 1800 lbs. The max tensile load that these joints will experience is 800 lbs, and therefore will be strong enough. Three tests were performed. The mode of failure was the epoxy interface between the carbon tube and the threaded rod end. This may have been due to surface preparation or an inadequate cure time. As an added safety precaution, we took extra care with surface preparation in the final product.

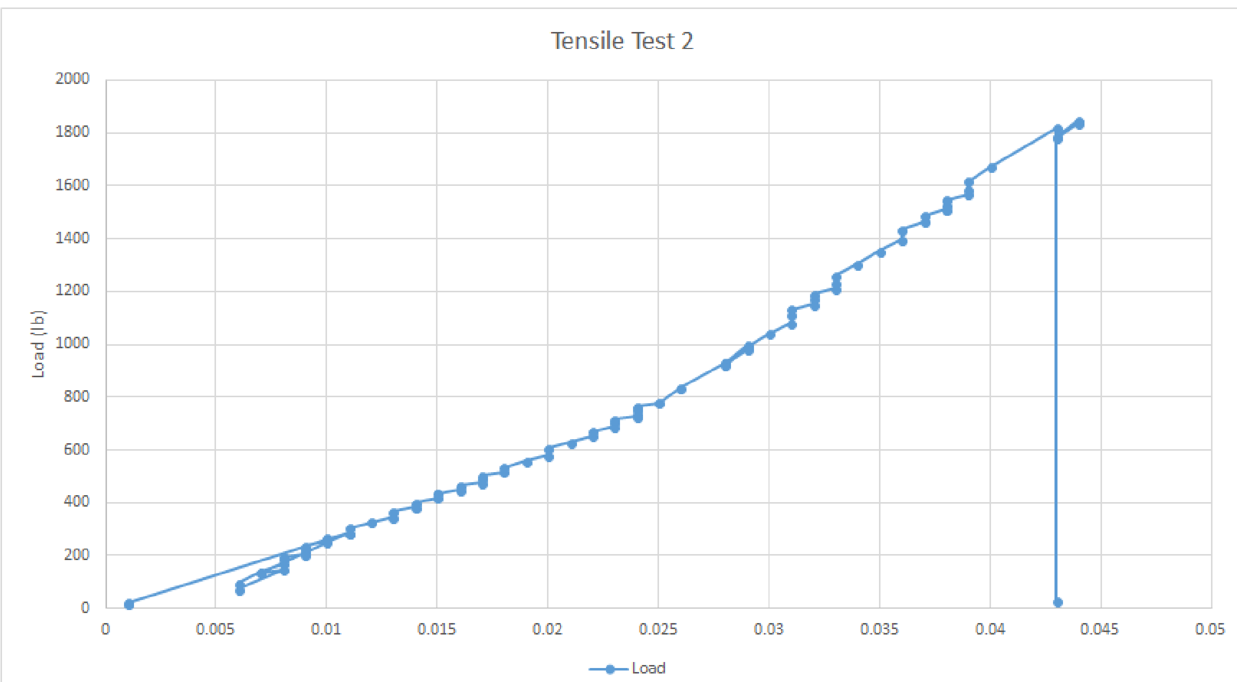

Figure 35. Tensile Test Results

4 Point Bending of Sandwich Panel

The aim of the 4 point bending test was to help determine how many plies the sandwich panel will need. It is known as a flexural test. The panel will be supported by a roller on each end of the beam and then be subjected to a concentrated area by the crosshead in the center. The compressive force was applied at .004”/second. The max bending load that the upright will see is 1000lbs. The experimental setup can be seen in Figure 36.

Figure 36. Four Point Bending Test Setup

The results from the four point bending test can be seen in Figure 37. The initial four point bending test showed delamination of the laminate from the core for all three tests done at a load less than the 1000lb max load that the upright will see. Therefore, the edges were wrapped with an additional eight plies of carbon fiber. After wrapping the edges, the sandwich panel handled a load of 1400lbs, which meets the requirements.

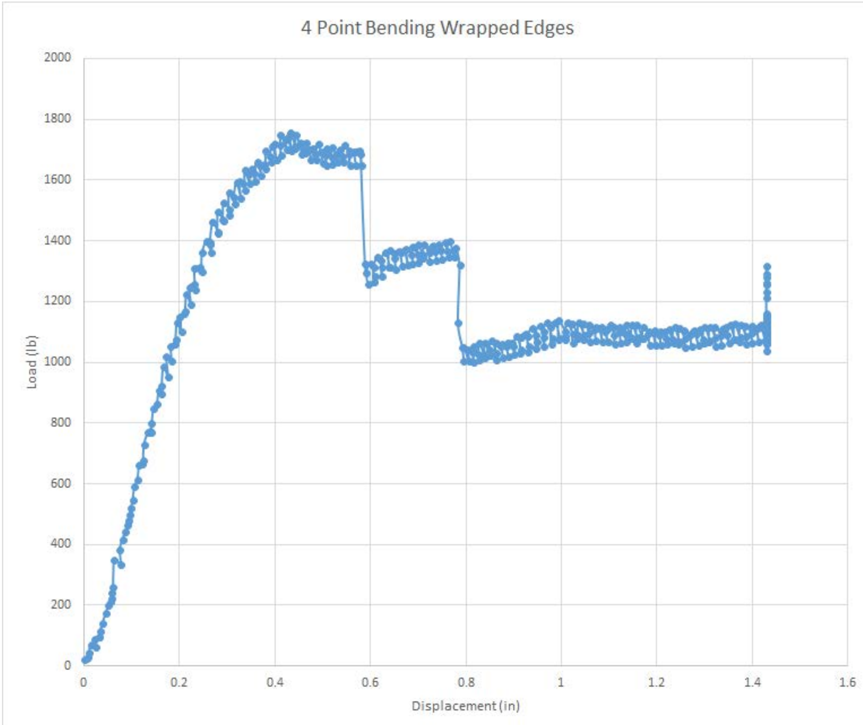

Figure 37. Four Point Bending Test Results

Non-destructive Testing
The non-destructive test will serve as the final strength verification. As mentioned earlier, carbon fiber layups are very manual labor intensive and is therefore subject to manufacturing variation. The non-destructive test will not be performed by the senior project team. PROVE lab will drive the car once the subteams finish the other systems and this will serve as the test.

Design Verification Plan and Report
The DVP&R documents all of the tests to be completed and test results. This is to ensure that the same mistakes are not repeated and the test results will make recommendations for changes if the parts fail its adequate criteria. All tests were completed except for the non destructive test that the team will do later in the project. The full DVP&R can be found in Appendix K.

Chapter 8, PROJECT MANAGEMENT

To best describe the overall design process in solving the problem, a flowchart was made; Figure 28. This outlines the primary flow of tasks required for the completion of the project.

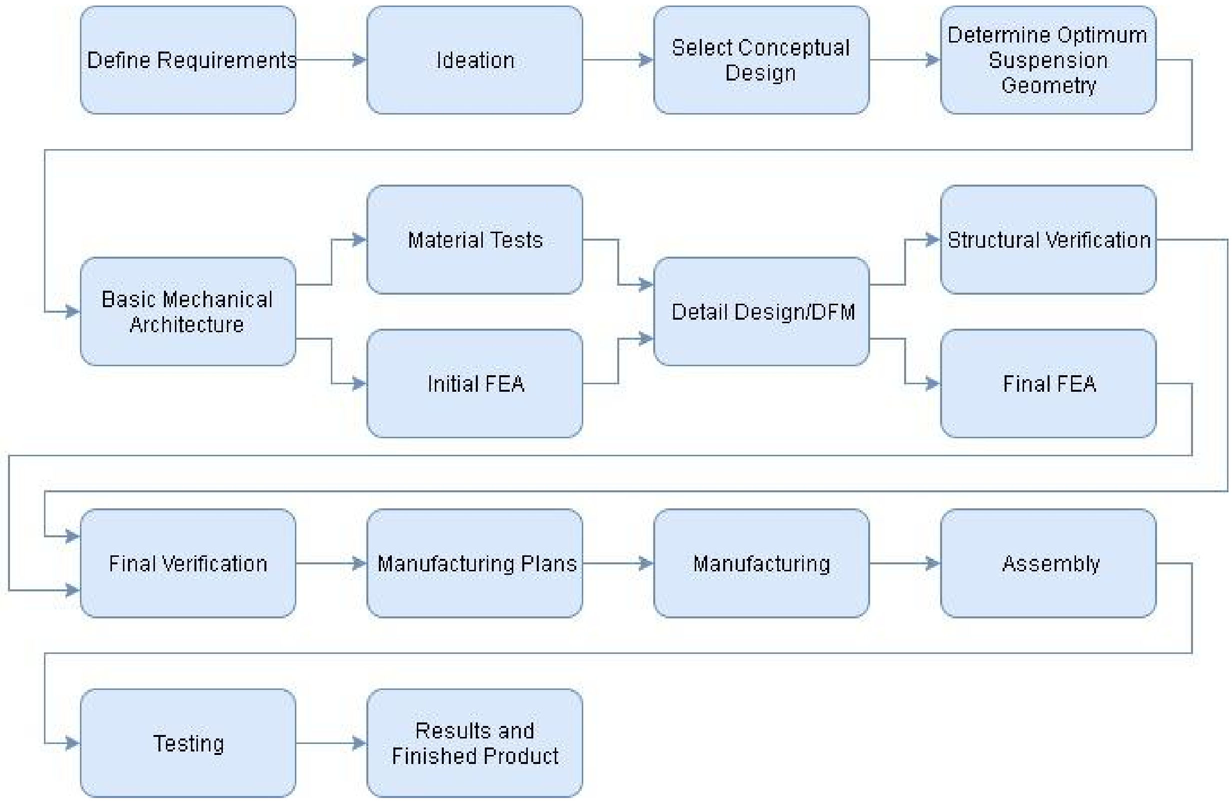

Figure 28. Basic Flow of Project Tasks

Solidworks will be utilized mainly for mechanical component design, but will most likely also use Autodesk Fusion 360 for surface modeling and CAM. ANSYS was used for composite analysis. Through PROVE, Lotus Shark was used for suspension system analysis.

Table 3. Key Deliverables by Quarter

Quarter	Deliverable	Due Date
Fall	Statement of Work	10/12/17
	Concept Models Prototypes	10/24/17
	Failure Modes & Effects Analysis	11/2/17
	Concept Prototypes	11/7/17
	PDR Report	11/9/17
	PDR Presentation	11/14/17
Winter	Interim Design Review	1/16/18
	Structural Prototype	1/23/18
	Detailed CAD/ Manufacturing Plan	1/30/18
	CDR Presentation	2/6/18
	CDR Report	2/8/18
	Risk Assessment	2/13/18
	Safety Review	2/15/18
	Manufacturing & Test Review (M&T)	3/13/18
Spring	Hardware/ Safety Demo	4/26/18
	Operators' Manual	5/22/18
	Expo Poster	5/22/18
	FDR Report	5/31/18
	Final Prototype at Expo	6/1/18

Table 3 above has the key deliverables for each quarter as regards to this project. The full Gantt chart with the subsequent steps and key deliverables can be found in Appendix L.

<u>Deviations from Project Plan</u>

Manufacturing specimens and preparing for material testing took much longer than initially planned. This put the team two weeks behind schedule spring quarter. As a result of this and the lack of a sufficient chassis to test the system on, non-destructive testing on the final part was not conducted as planned. It should be noted that this must be conducted at a later date by PROVE members following this senior project team.

Chapter 9, CONCLUSION & RECOMMENDATIONS

Finished Product

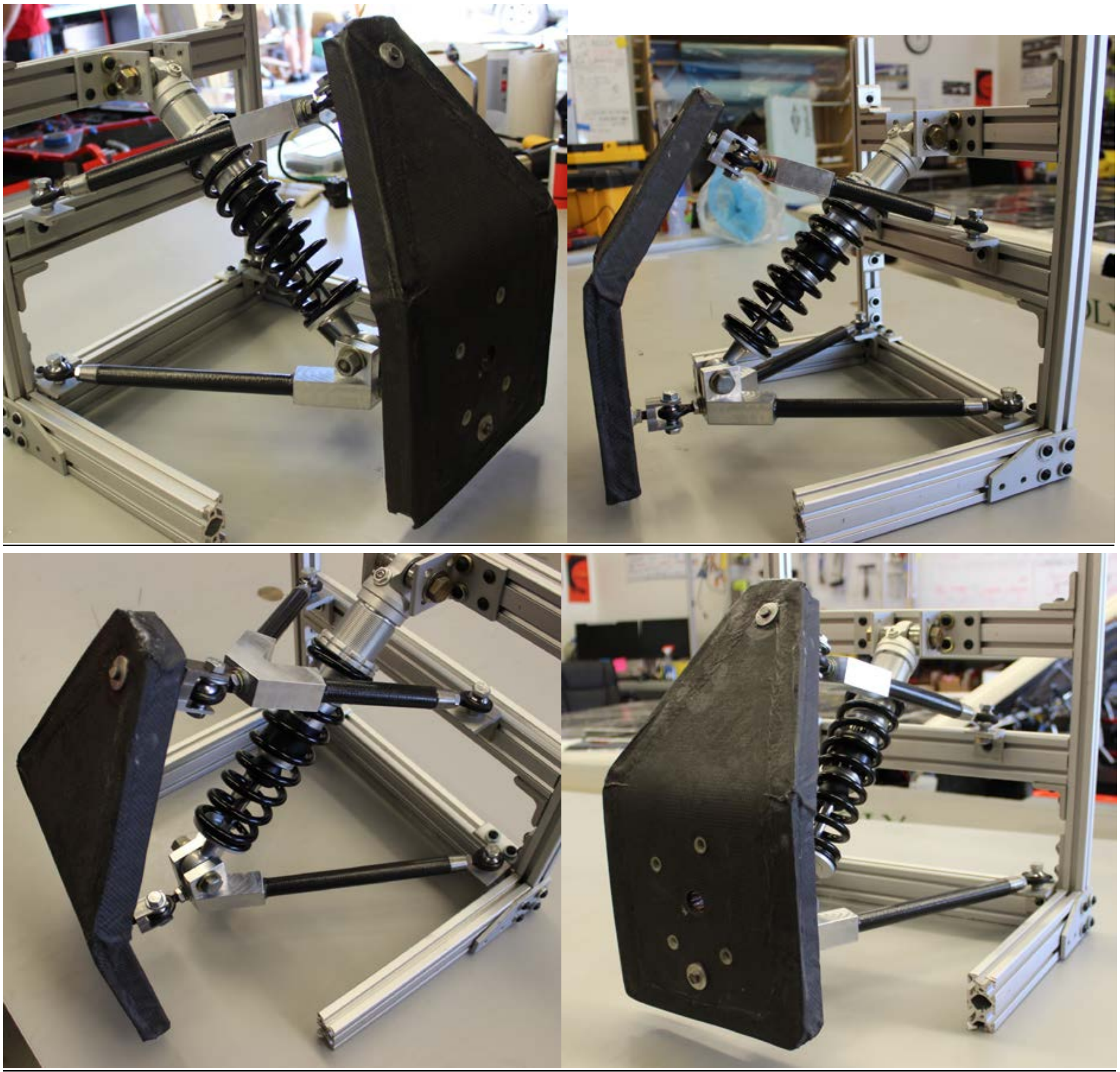

Figure 39. Images of Final Front Suspension Design

Recommendations

As the car's development progresses, the team must update the system parameters accordingly. They are granted full permission and discretion for future developments of this system. A custom hub is highly recommended to reduce weight. If rebuilt, prepreg carbon is highly advised due to its more uniform properties and easier manufacturing. Also, a machined foam is recommended for the bend shape to ensure greater precision. For drilling the holes, a rigid jig is advised for higher precision.

Conclusion

The team produced an innovative suspension system for Dr. Doig and PROVE Lab for their endurance car project. The system exceeded its weight target by weighing 4.3 lbs per side, excluding dampers. Due to the car being in a conceptual development stage, this system defined some constraints on the rest of the car's drivetrain including mounting points, and track width. If needed, the team is free to modify or redo any part of this system as needed. Although constraints will be defined, no design of any other system other than the front suspension was done by this team. Double wishbone was the selected configuration of the suspension, as decided with a decision matrix weighted towards the most important design requirements. Manufacturing methods utilizing composites made up the innovation factor required for the design. The focus of the design was the upright made from carbon fiber sandwich panels. Loads in cornering, braking, and bump have been integrated into FEA to give the team a fundamental understanding of the stresses seen by the upright. The wishbone joints were also analyzed in SolidWorks FEA. This analysis will serve useful when designing the sandwich panel structure. The destructive testing showed that the suspension will be sufficient in supporting its applied loads.

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APPENDICES

Appendix A- Preliminary Analyses and Benchmarking
Appendix B- QFD House of Quality
Appendix C- Decision Matrix
Appendix D- Concept Layout Drawing(s)
Appendix E- Complete Drawings Package
Appendix F- Final analyses
Appendix G- Safety Hazard Checklist, FMEA, Risk Assessment
Appendix H- Operator’s Manual
Appendix I- Purchased Parts Details
Appendix J- Budget/Procurement List
Appendix K- DVP&R
Appendix L- Gantt Chart

Tesla Model S
We used a Tesla Model S suspension system as a benchmark due to its electric drivetrain that the car will likely resemble. We found that the front suspension is a double wishbone system with vertical coilover struts. It currently does not feature any composite parts. The system is compactly packaged in the horizontal direction with a larger vertical proportion. The system is fairly conventional in overall design.

PROVE Project 1 Suspension
We used PROVE Lab's solar car suspension system as a benchmark due to its use of composites and our easy access to the system and the team's design process. The suspension is a double wishbone system with coilover struts angled at a shallow angle of 31 degrees. This system is very tightly packaged within the aero shell of the car with a flat and wide proportion for the wishbones and a long hanging down upright with the wheel positioned about 1ft below the lower wishbone. The wishbones are both made of carbon fiber tube from Rockwest Composites adhered to aluminum CNC'd parts with G-Flex adhesive.

BMW Z4 suspension
We used a BMW Z4 as a benchmark due to it being approximately the size of the car and due to the fact that Justine has one. The Z4 has a MacPherson suspension system on the front with vertical coilover struts. It has a track width of around 60in, which we are using as an approximate size constraint for our system. The packaging of the system is fairly standard for a MacPherson with the vertical proportion being the largest. The system is made of steel.

Appendix B - QFD House of Quality

PROVE Lab Suspension

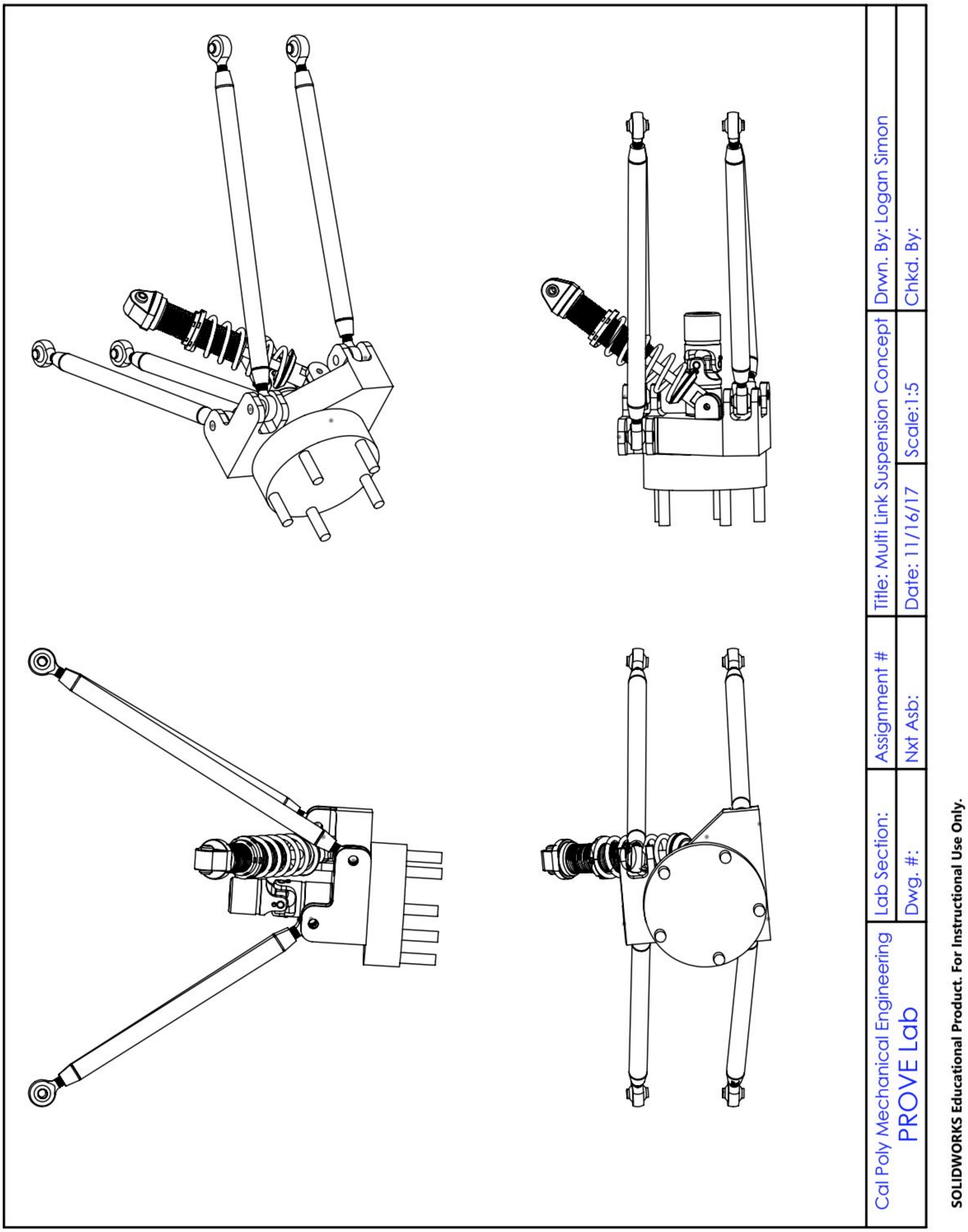
Engineering Requirements (HOWS)

Customer (Step #1) Requirements (Whats)	Weighting (1 to 5)	Target Vehicle Weight	Vertical Travel	Innovative Design	Manufacturing Process	Max Weight	Unconventional Materials	Define Max Vehicle Weight	Define Mounting Points	Define Steering Interface	Recommend Rear Suspension	Prototypes made	Adjustable Damping Ratio	Alignment Features	Track Width	Minimal dependency on other systems	Freeway Speed				Benchmark: Tesla Model S	PROVE Project 1	BMW Z4
Support car and 2 passengers	5	9																			5	1	5
Capable of driving on the road	3		6														3				5	1	5
Innovative design or manufacturing process	5			9	9		6					3									2	5	2
Lightweight	4					9	6														2	5	3
Provide design constraints for other subteams	3							9	9	9	3					6						5	
Small scale prototyping	4											9										2	
Adjustability	4												9	9							3	3	3
Sizing of a real car	5														9						5	1	5
Sustainable vehicle life cylce	1				6		3														3	2	3
Packaging	4								6	6					9	9					3	5	2
Travel Speed	5																9				5	3	5
Units		lb	in			lbs									in		mph						
Targets		3500	3 to 6	Yes/No	Yes/No	<25	Yes/No	Yes/No	Yes/No	Yes/No	Yes/No	Yes/No	Yes/No	Yes/No	60	Yes/No	60 to 70						
Benchmark #1																							
Benchmark #2																							
Importance Scoring		45	18	45	51	36	57	27	51	51	9	51	36	36	81	54	54	0	0	0			
Importance Rating (%)		56	22	56	63	44	70	33	63	63	11	63	44	44	100	67	67	0	0	0			

● = 9 Srong Correlation
○ = 6 Medium Correlation
Δ = 3 Small Correlation
Blank No Correlation

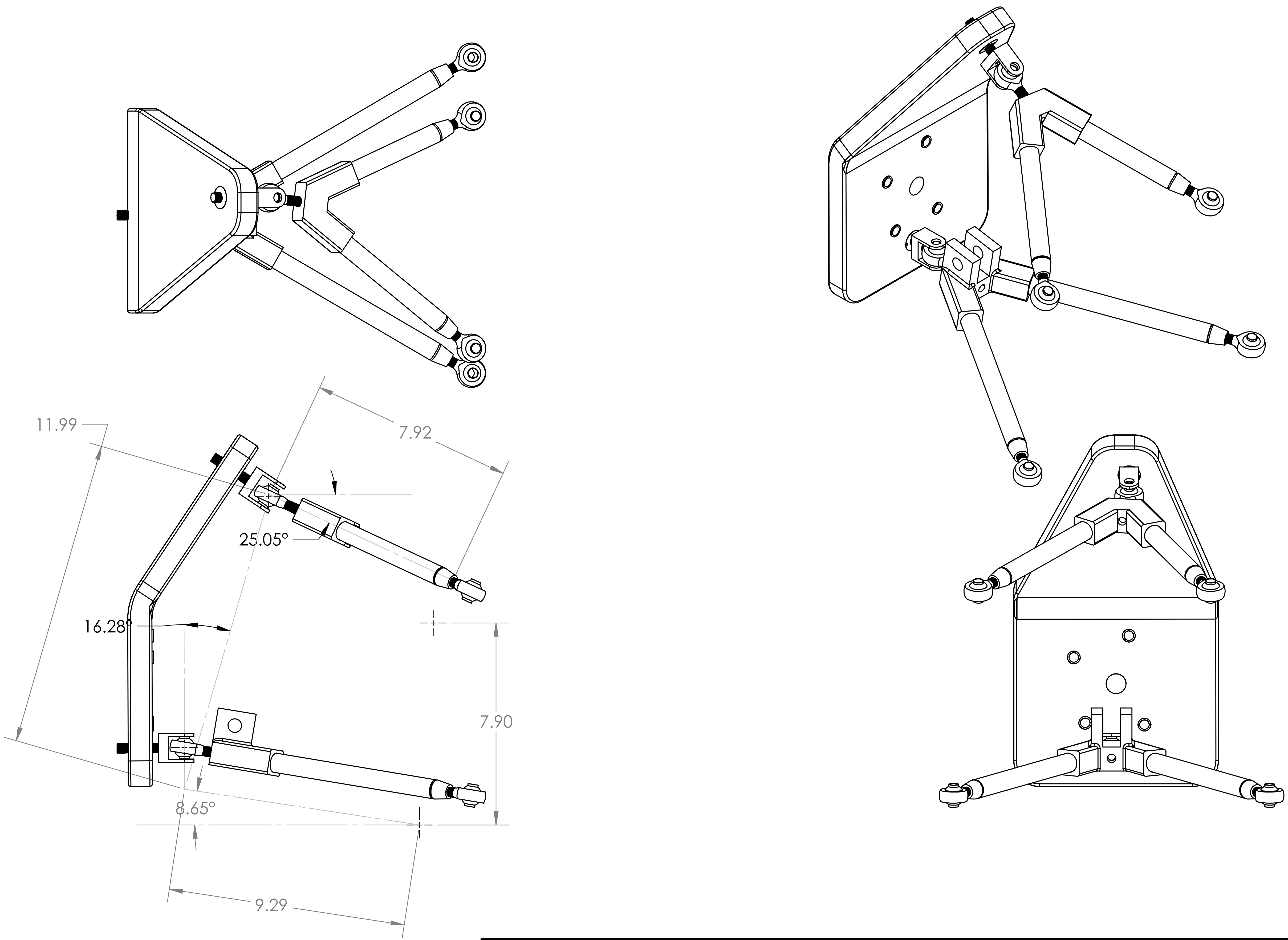
Linkage Style							
		Concepts					
Criteria	Weight	MacPherson		Double Wishbone		Multi-Link	
Lightweight	3	0	0	-1	-3	-1	-3
Performance	2	0	0	1	2	2	4
Adjustable	2	0	0	1	2	2	4
Packaging	2	0	0	1	2	1	2
Innovative	3	0	0	1	3	2	6
Simplicity	1	0	0	-1	-1	-2	-2
Manufacturability	2	0	0	-1.5	-3	-0.5	-1
Sum			0		2		10

Appendix D- Concept Layout Drawings

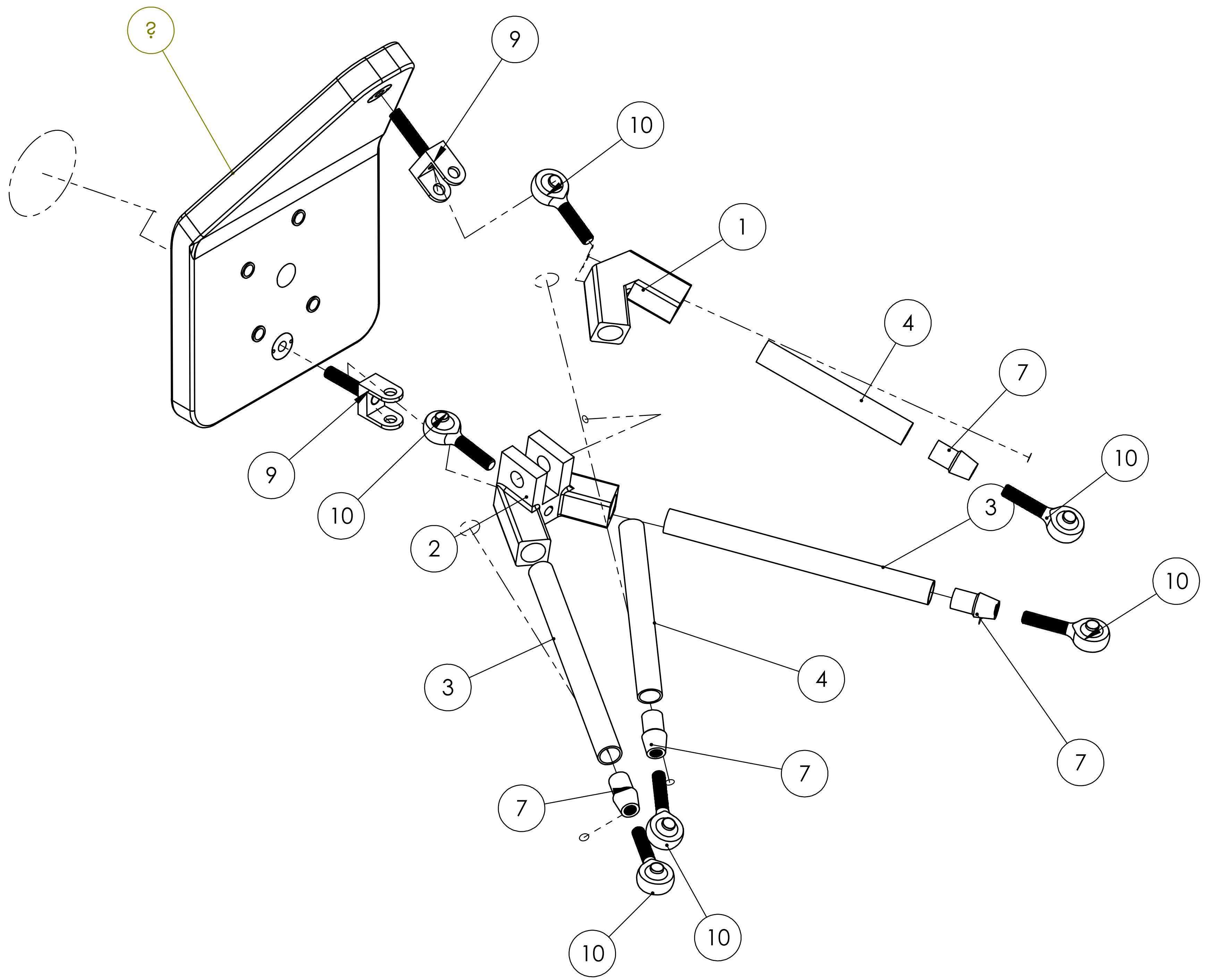


Appendix E- Complete Drawings Package

			Indented Bill of Material (BOM)									
			Front Suspension Assembly									
Assy Level	Part Number	Sku Number	Description			Matl	Vendor	Stock	Qty	Cost	Ttl Cost	Link
			Lvl0	Lvl1	Lvl2							
0	100000	------	Front Suspension Assembly			------	------		------	------	------	------
1	101001	------		Upright		------	------		2	------	------	------
2	102001	V61401011A			HexForce Weave	Carbon Fiber	Hexcel	------	------	------	(Donated)	------
2	102002	------			Flex Core	Nomex	Hexcel	------	------	------	(Donated)	------
2	102003	8668K14			G10 Tubing	Garolite	McMaster-Carr	------	1	$47.00	$47	https://w
2	102004	------			Laminating Resin/Harde	------	Pro-Set	------	------	------	(Donated)	------
2	102005	18037			Potted Inserts	Aluminum Rod	Online Metals	0.5x12	1	$32.00	$ 32.00	
1	101002	------		Wishbone Assembly		------	------	------	4	------	------	------
2	102006	12769			Upper Wishbone Joints	Aluminum	Online Metals	2.5x4x2.5	1	$12.88	$13	https://w
2	102007	14720			Lower Wishbone Joints	Aluminum	Online Metals	2.5x3x10-12	1	$40.86	$41	https://w
2	102008	45558-UHM			Carbon Tubes	High Modulus Carbon	Rock West Composites	.75x72	1	$131.99	$131.99	https://w
2	102009	94640A115			Threaded Insert	4130 Alloy Steel	McMaster-Carr	------	10	$5.30	$53.00	https://w
2	102010	6960T6			Ball Joints	Black-Oxide Alloy Steel	McMaster-Carr	------	12	$9.92	$119.04	https://w
2	102011	------			Epoxy Resin/Hardener	------	Pro-Set	------	------	------	(Donated)	------
1	101003	------		Clevis		------	------	------	------	------	------	------
2	102012	9902			Body	Steel	Online Metals	1.25x2x10-12	1	$25.75	$25.75	https://w
2	102013	90322A134			Threaded Rod	Steel	McMaster Car	3/8-24	1	$9.00	$9.00	https://w
1	101004	HD022		Strut		------	Ohlins	------	2	------	(Donated)	------
1	101005	------		Fasteners		Steel	McMaster Car	3/8-24	------	------	$50	https://w
								Purchased Parts Total:			$ 521.52	

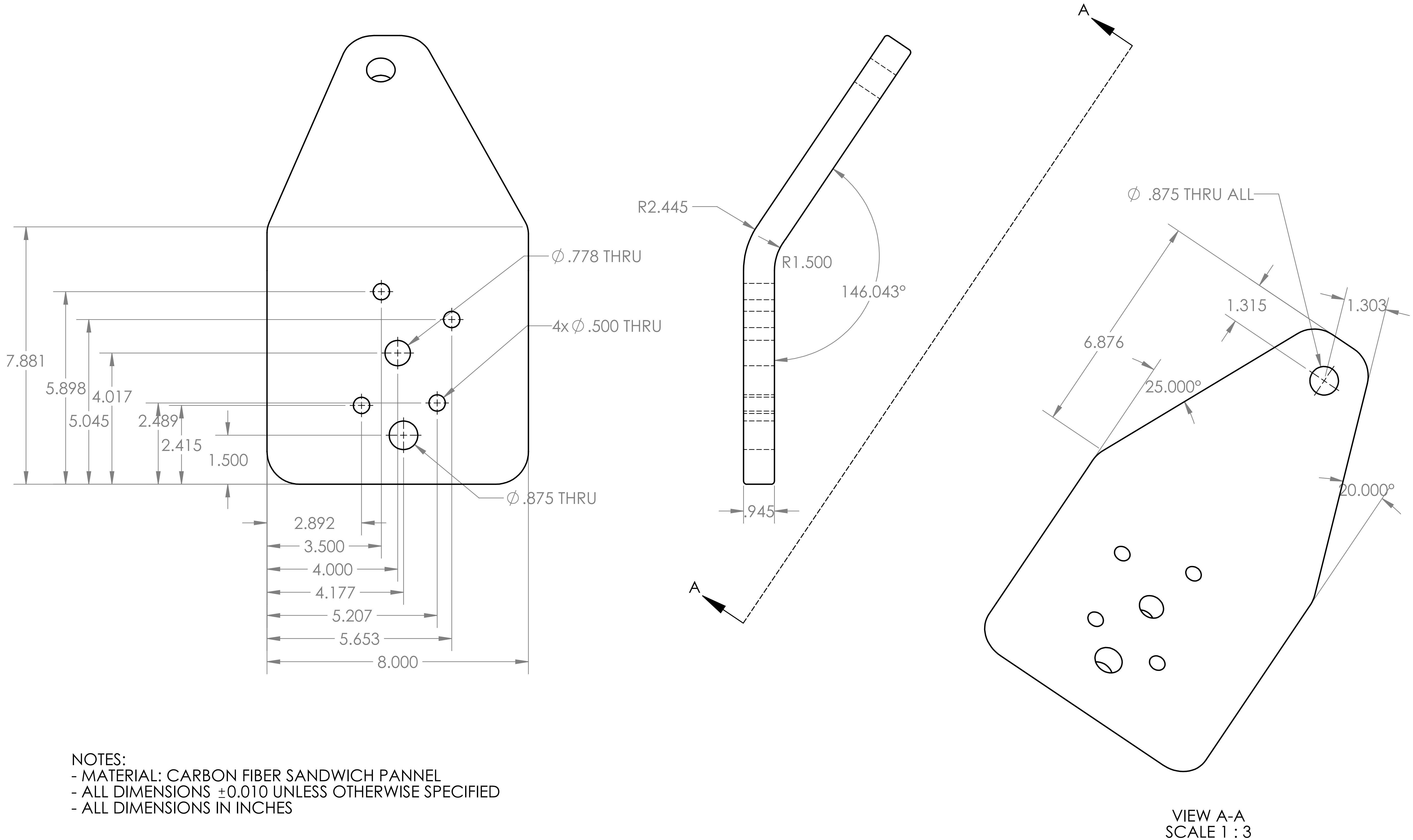


Cal Poly PROVE Lab	Subsystem: Front Suspension	Title: Front Suspension Assembly	Drwn. By: Logan Simon
	Dwg. #: / Nxt Asb:	Date: 2/1/18 / Scale: 1:4	Chkd. By:



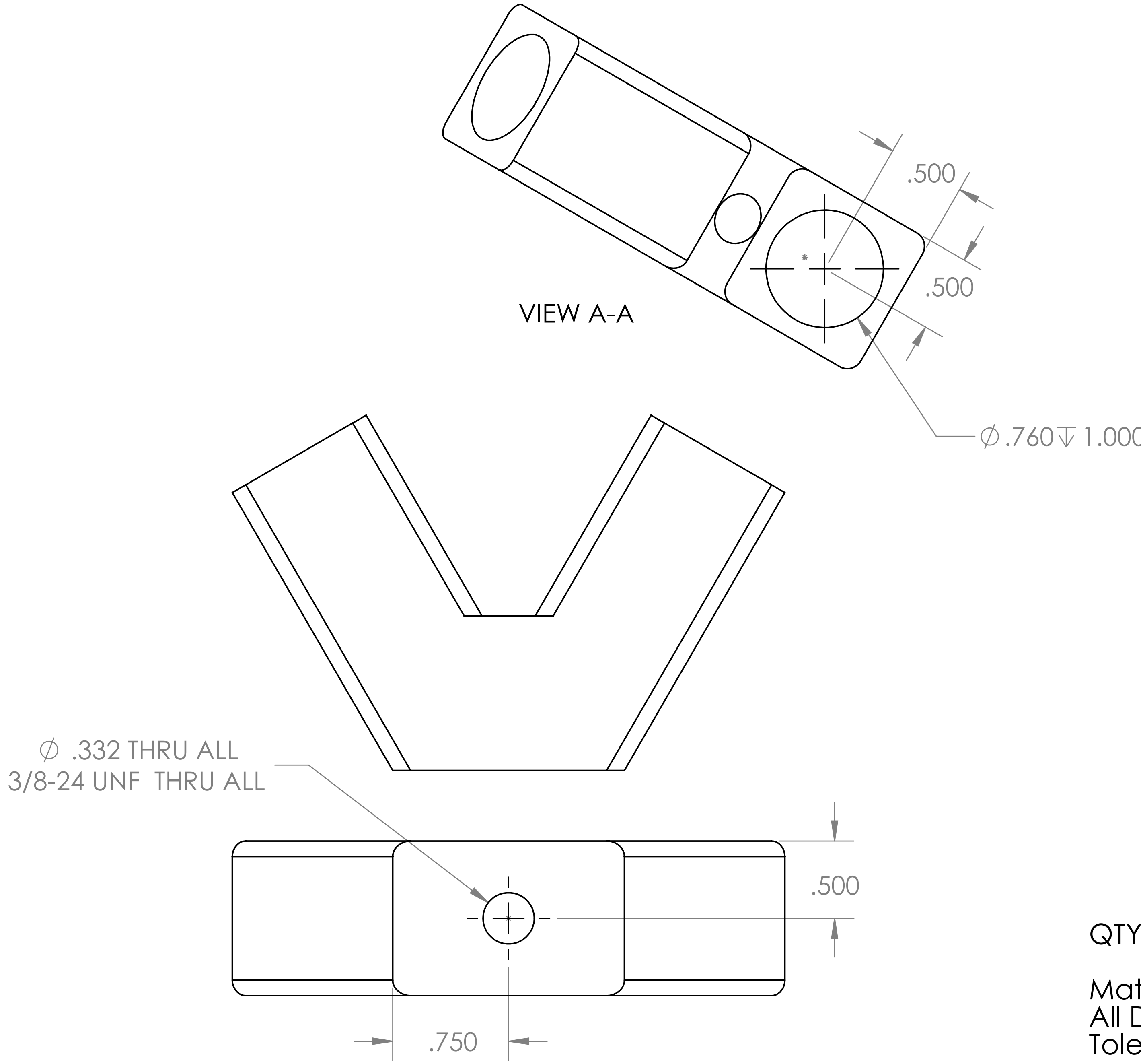
ITEM NO.	DESCRIPTION	QTY.
1	Upper Wishbone Joint	1
2	Lower Wishbone Joint	1
3	Lower Wishbone Rod	2
4	Upper Wishbone Rod	2
6	Upright Assembly	1
7	Threaded Insert	4
9	Clevis	2
10	Ball Joint	6

Cal Poly PROVE Lab	Subsystem: Front Suspension		Title: Front Suspension Assy Expl	Drwn. By: Logan Simon
	Dwg. #:	Nxt Asb:	Date: 6/8/18	Scale: 1:4	Chkd. By:



NOTES:
- MATERIAL: CARBON FIBER SANDWICH PANNEL
- ALL DIMENSIONS ±0.010 UNLESS OTHERWISE SPECIFIED
- ALL DIMENSIONS IN INCHES

Cal Poly Mechanical Engineering	Lab Section:06	Assignment #	Title: UPRIGHT		Drwn. By: JUSTINE KWAN
ME 429 - WINTER 2018	Dwg. #:	Nxt Asb:	Date: 2/1/18	Scale: 1:3	Chkd. By: ME STAFF



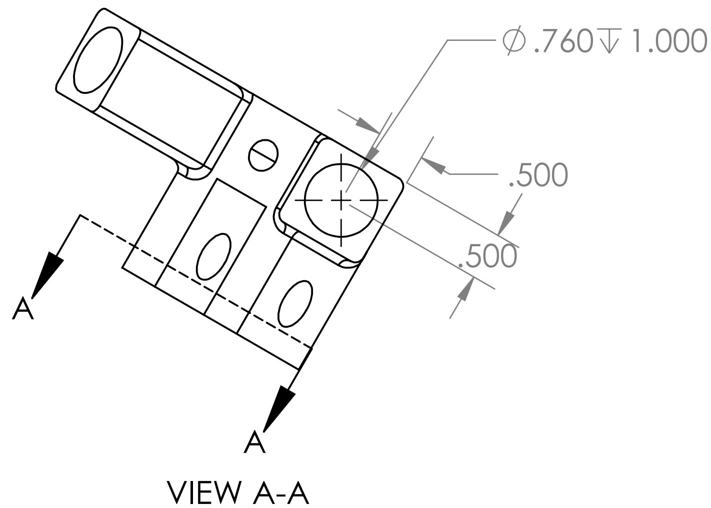
QTY:2

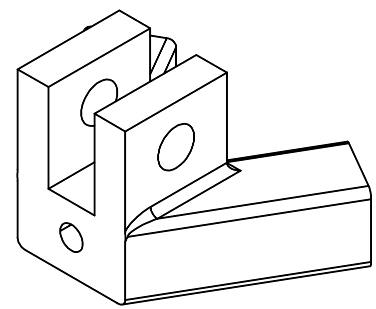
Material: Aluminum
All Dimensions in Inches
Tolerance: ±0.005in

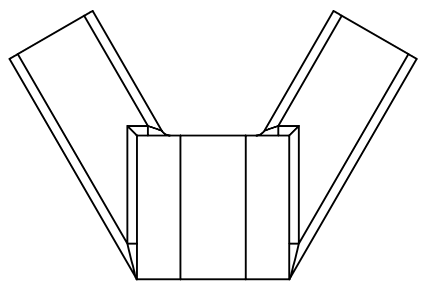
Cal Poly Engineering PROVE Lab	Subsystem:		Title: Upper Wishbone Joint		Drwn. By: Logan Simon
	Dwg. #:	Nxt Asb:	Date: 6/8/2018	Scale:1:1	Chkd. By:

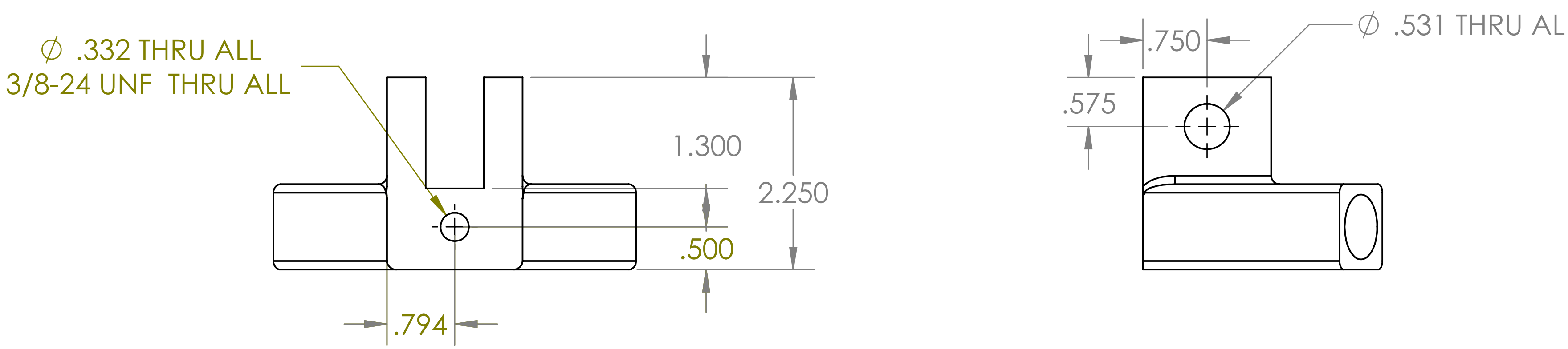
QTY:2

Material: Aluminum
All Dimensions in Inches
Tolerance: ±0.005in

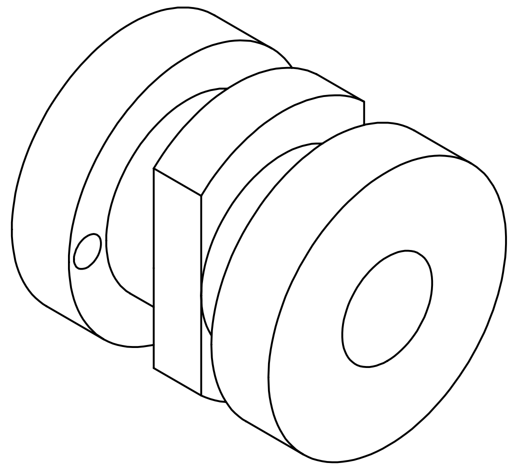




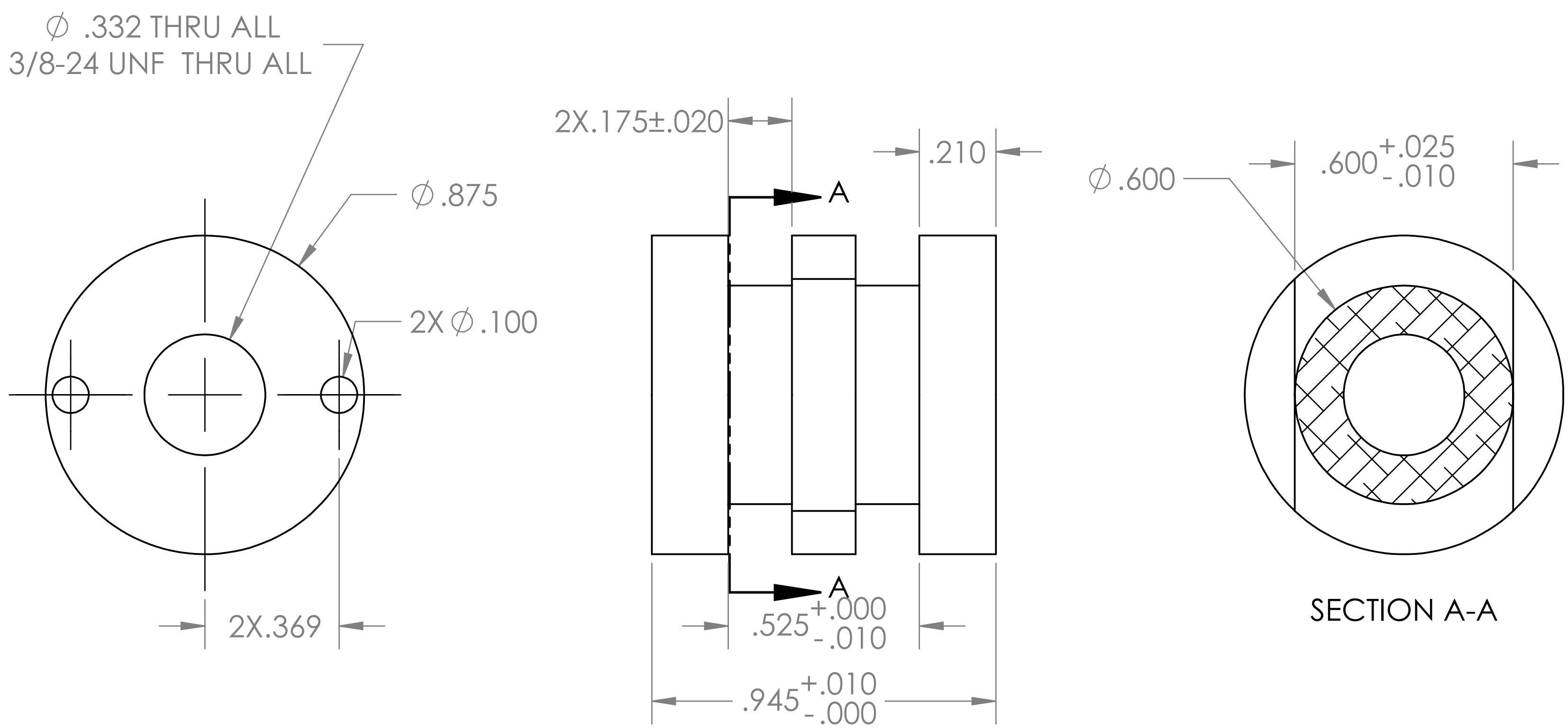




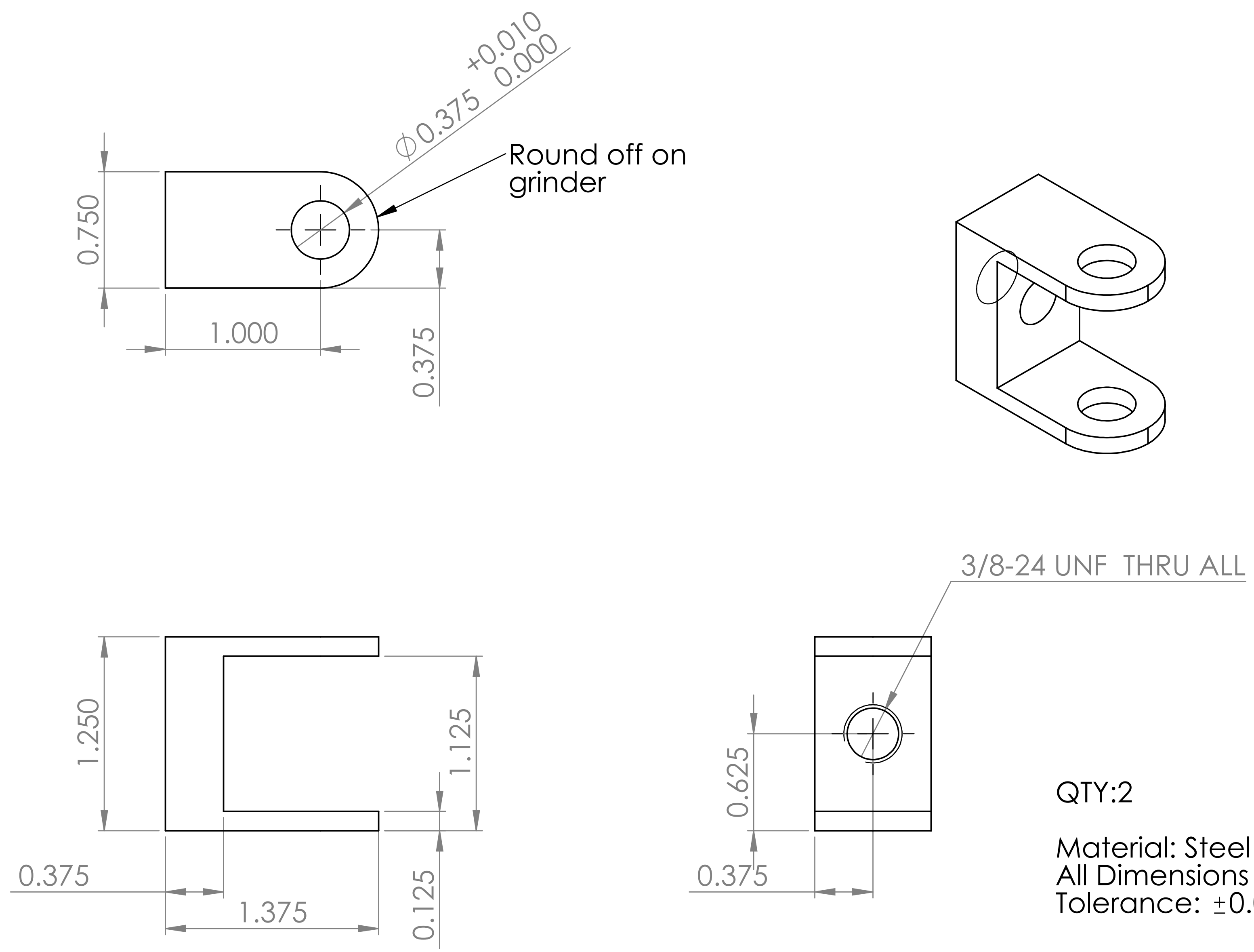
Cal Poly Engineering PROVE Lab	Subsystem:		Title: Lower Wishbone Joint	Drwn. By: Logan Simon
	Dwg. #:	Nxt Asb:	Date: 6/8/2018 · Scale:1:2	Chkd. By:



NOTES:
ALL DIMS ±0.005IN
QTY: 9



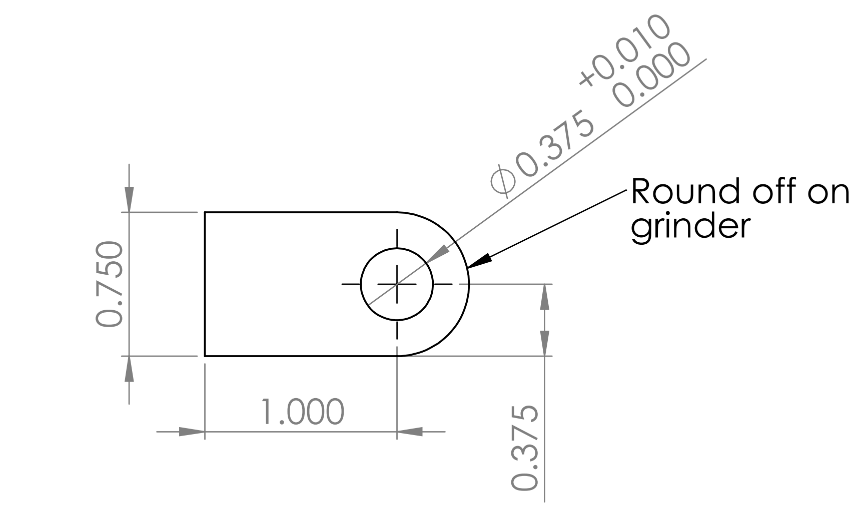
Cal Poly Engineering	Subsystem:		Title: Potted Insert 0375-24 20 Ply	Drwn. By: Logan Simon
PROVE Lab	Dwg. #:	Nxt Asb:	Date: 6/8/2018 · Scale:2:1	Chkd. By:

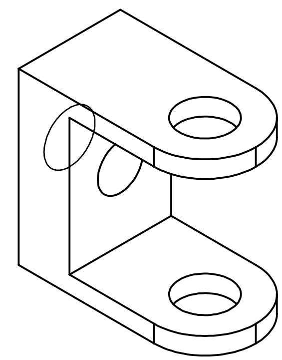


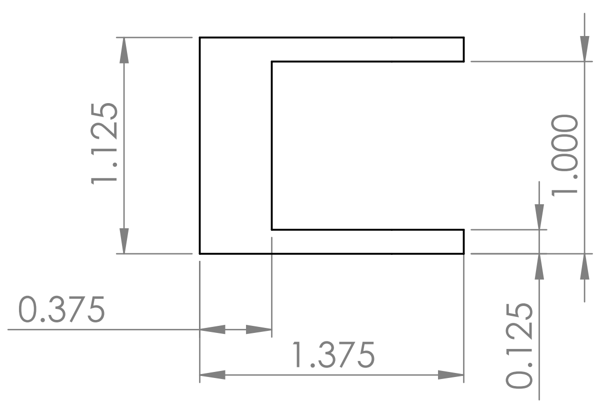
QTY:2

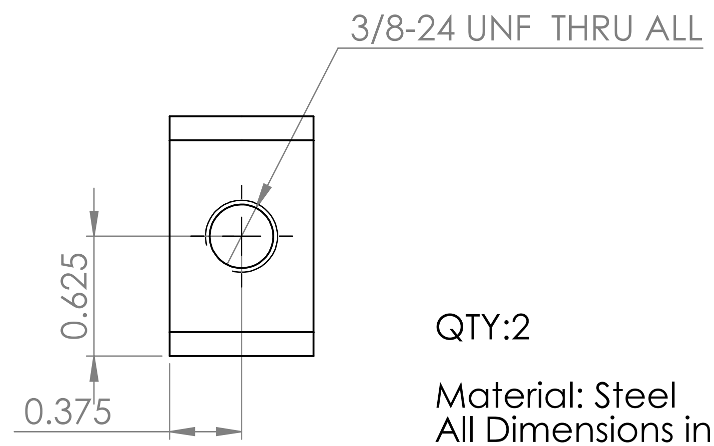
Material: Steel
All Dimensions in Inches
Tolerance: ±0.005in

Cal Poly Engineering	Subsystem:		Title: Clevis End Top	Drwn. By: Logan Simon
PROVE Lab	Dwg. #:	Nxt Asb:	Date: 6/8/2018	Scale:1:1	Chkd. By:









QTY:2

Material: Steel
All Dimensions in Inches
Tolerance: ±0.005in

Cal Poly Engineering PROVE Lab	Subsystem:		Title: Clevis End Bottom		Drwn. By: Logan Simon
	Dwg. #:	Nxt Asb:	Date: 6/8/2018	Scale:1:1	Chkd. By:

```
%%PROVE Fsus Spring Calcs
clc, clear all, close all

%Given
d1=9.56; %in
d2=14.76; %in
A=60; %degrees
SW=750; %lbs
Sm=SW/386.4; %Snails
SF=90; %cpm
BL=100; %lbs
ds=.74; %in
bs = 50;
C=8;
k=800
kt=1600


%Preliminary Values
ACF=sind(A);

%Static Calcs

% k=(BL*d2)/(ds*ACF*d1)
% % k=150
PL=(SW*d2)/(k*ACF*d1)

%Dynamic Calcs

MR=d1/d2;

WR=(MR^2)*k*ACF;
SF=187.8*sqrt(WR/SW)/60
omega=(1/(2*pi))*sqrt(((WR*kt)/(WR+kt))/Sm)

% WR=((SF/187.8)^2)*(SW)
% k=(WR)/((MR^2)*(ACF))
%
Ks=k*sind(31);

%Fs=(SW*d2)/(ACF*d1)

%Damping
Ccr=2*sqrt(WR*Sm);

dr=bs/Ccr


k =

   800
```

```
kt =

        1600

PL =

    1.6714

SF =

    1.9485

omega =

    1.7916

dr =

    1.0526
```

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Table of Contents

Force Calculations

```
clc, clear all, close all

% System Parameters

A=2.5103; %in
L_LW=11.63; %in
theta_LW=9.95; %deg
L_UW=9.38; %in
theta_UW=24.95; %deg
h=12.4902; %in
L_SA=10.375; %in
theta_SA=16.52; %deg
B=3.3086; %in
S1=1.725;
S2=1.125;
theta_S=60; %deg
```

Cornering Load

```
Fy_c=916; %lbf
Fz_c=824.4; %lbf

%Upright

syms LWy_c LWz_c UW_c

eq1c= -LWy_c - UW_c*cosd(theta_UW) == Fy_c;
eq2c= -LWz_c + UW_c*sind(theta_UW) == Fz_c;
eq3c= -(L_SA*cosd(theta_SA))*LWy_c + (L_SA*sind(theta_SA))*LWz_c ==
 Fy_c*(h-A) - Fz_c*(B+L_SA*sind(theta_SA));

sol = solve([eq1c, eq2c, eq3c], [LWy_c, LWz_c, UW_c]);
syms off
LWy_c= double(sol.LWy_c) %lbf
LWz_c= double(sol.LWz_c) %lbf
UW_c= double(sol.UW_c) %lbf

% Lower Wishbone

syms Fs_c RLy_c RLz_c
eq4c= Fs_c*cosd(theta_S+theta_LW)*(L_LW-S1) ==
 LWz_c*(L_LW*cosd(theta_LW)) - LWy_c*(L_LW*sind(theta_LW));
```

```
eq5c= RLy_c == LWy_c - Fs_c*cosd(theta_S);
eq6c= RLz_c == LWz_c - Fs_c*sind(theta_S);

sol = solve([eq4c, eq5c, eq6c], [Fs_c, RLy_c, RLz_c]);
syms off
Fs_c=double(sol.Fs_c) %lbf
RLy_c=double(sol.RLy_c) %lbf
RLZ_c=double(sol.RLz_c) %lbf
```

```
LWy_c =

 -677.7140


LWz_c =

 -935.2616


UW_c =

 -262.8127


Fs_c =

  -2.7538e+03


RLy_c =

  699.2099


RLZ_c =

   1.4496e+03
```

5g Bump Load

```
Fy_b=0; %lbf
Fz_b=3125; %lbf

syms LWy_b LWz_b UW_b

eq1b= -LWy_b - UW_b*cosd(theta_UW) == Fy_b;
eq2b= -LWz_b + UW_b*sind(theta_UW) == Fz_b;
eq3b= -(L_SA*cosd(theta_SA))*LWy_b + (L_SA*sind(theta_SA))*LWz_b ==
 Fy_b*(h-A) - Fz_b*(B+L_SA*sind(theta_SA));

sol = solve([eq1b, eq2b, eq3b], [LWy_b, LWz_b, UW_b]);
```

```
syms off
LWy_b= double(sol.LWy_b) %lbf
LWz_b= double(sol.LWz_b) %lbf
UW_b= double(sol.UW_b) %lbf

% Lower Wishbone

syms Fs_b RLy_b RLz_b
eq4b= Fs_b*cosd(theta_S+theta_LW)*(L_LW-S1) ==
 LWz_b*(L_LW*cosd(theta_LW)) - LWy_b*(L_LW*sind(theta_LW));
eq5b= RLy_b == LWy_b - Fs_b*cosd(theta_S);
eq6b= RLz_b == LWz_b - Fs_b*sind(theta_S);

sol = solve([eq4b, eq5b, eq6b], [Fs_b, RLy_b, RLz_b]);
syms off
Fs_b=double(sol.Fs_b) %lbf
RLy_b=double(sol.RLy_b) %lbf
RLZ_b=double(sol.RLz_b) %lbf
```

```
LWy_b =

  913.4319


LWz_b =

  -3.5500e+03


UW_b =

  -1.0075e+03


Fs_b =

  -1.2516e+04


RLy_b =

   7.1712e+03


RLZ_b =

   7.2888e+03
```

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Table of Contents

PROVE P2 Suspension Brake Loads

```
clc, clear all, close all
%Vehicle Parameters

mumax=0.9; % Max adhesion coeff
L=100/12; %Wheelbase (ft)
Lr=50/12; %Rear to CG (ft)
Lf=L-Lr; %Front to CG (ft)
g=32.2; %Gravitational accel(ft/s^2)
h=15.5/12; %Height of CG (ft)
fw=0.011; %Wheel drag coeff
Wcar=2500; %;lb
mcar=Wcar/g;

%Acceleration Limited by adhesion
amaxfwd=g*(((mumax*(Lr/L))-fw)/(1+(mumax*(h/L)))); %ft/s^2
amaxrwd=g*(((mumax*(Lf/L))-fw)/(1-(mumax*(h/L)))); %ft/s^2
amaxrwd=0.7*g
%Rated Deceleration
z=amaxrwd/g;
zmax=mumax;
```

```
amaxrwd =

   22.5400
```

Brake Ratios

```
% Ratios
k = Lr/L;
zeta = h/L;

% Calculation
a = (k^2)/(4*(zeta^2)); % see notes 1st term
b = 1/zeta; % see notes 2nd term
c = 1/Wcar; % see notes 3rd term
d = k/(2*zeta); % see notes 4th term

BR = (Wcar*(sqrt(a + (b*c)) - d - c)) %Fr/Ff
```

```
BR =

    0.9995
```

Brake Forces

```
Fbf=(mcar*amaxrwd)/(BR+1) %ft-lbf
Fbr=Fbf*BR; %ft-lbf
```

```
Fbf =

  875.2169
```

Brake Torque

```
Rrotor=5.5/12; %ft

Tbf=Fbf*Rrotor %ft-lb
Tbr=Fbr*Rrotor %ft-lb
```

```
Tbf =

  401.1411


Tbr =

  400.9422
```

Stopping Speed

```
T60_0=60/(amaxrwd*(3600/5280)) %seconds
```

```
T60_0 =

    3.9042
```

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```
%%PROVE Fsus Spring Calcs
clc, clear all, close all

%Given
d1=9.56; %in
d2=14.76; %in
A=60; %degrees
SW=750; %lbs
Sm=SW/386.4; %Snails
SF=90; %cpm
BL=100; %lbs
ds=.74; %in
bs = 50;
C=8;
k=800
kt=1600


%Preliminary Values
ACF=sind(A);

%Static Calcs

% k=(BL*d2)/(ds*ACF*d1)
% % k=150
PL=(SW*d2)/(k*ACF*d1)

%Dynamic Calcs

MR=d1/d2;

WR=(MR^2)*k*ACF;
SF=187.8*sqrt(WR/SW)/60
omega=(1/(2*pi))*sqrt(((WR*kt)/(WR+kt))/Sm)

% WR=((SF/187.8)^2)*(SW)
% k=(WR)/((MR^2)*(ACF))
%
Ks=k*sind(31);

%Fs=(SW*d2)/(ACF*d1)

%Damping
Ccr=2*sqrt(WR*Sm);

dr=bs/Ccr
```

k =

800

```
kt =

        1600

PL =

    1.6714

SF =

    1.9485

omega =

    1.7916

dr =

    1.0526
```

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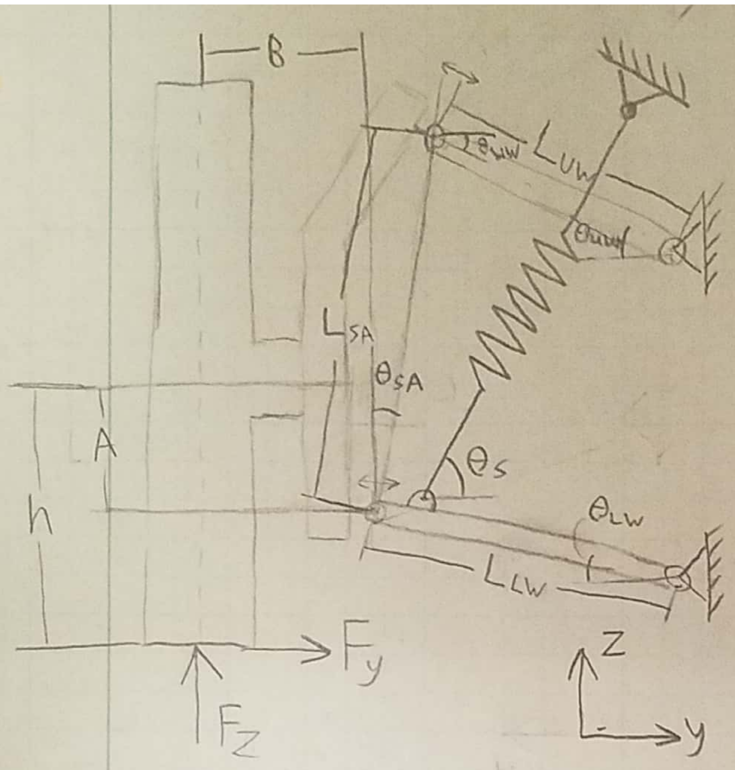
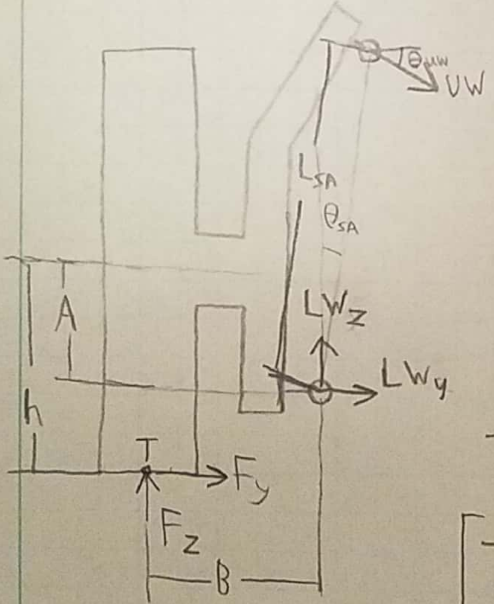


Upright/Hub



$$\sum F_y = 0 = F_y + LW_y + UW\cos\theta_{uw}$$
$$F_y = -LW_y - UW\cos\theta_{uw}$$

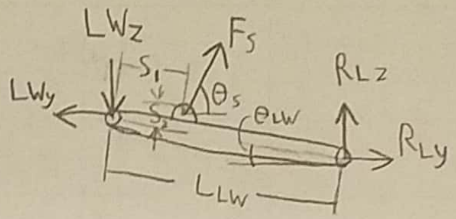
$$\sum F_z = 0 = F_z + LW_z - UW\sin\theta_{uw}$$
$$F_z = -LW_z + UW\sin\theta_{uw}$$

$$\sum M_{uw} = 0 = LW_y \cdot (L_{SA}\cos\theta_{SA}) - LW_z (L_{SA}\sin\theta_{SA}) - F_z(B + L_{SA}\sin\theta_{SA}) + F_y(h-A)$$

$$F_y(h-A) = -(L_{SA}\cos\theta_{SA}) \cdot LW_y + (L_{SA}\sin\theta_{SA}) \cdot LW_z - F_z(B + L_{SA}\sin\theta_{SA})$$

$$\begin{bmatrix} -1 & 0 & -\cos\theta_{uw} \\ 0 & -1 & \sin\theta_{uw} \\ -L_{SA}\cos\theta_{SA} & L_{SA}\sin\theta_{SA} & 0 \end{bmatrix} \cdot \begin{bmatrix} LW_y \\ LW_z \\ UW \end{bmatrix} = \begin{bmatrix} F_y \\ F_z \\ F_y(h-A) - F_z(B + L_{SA}\sin\theta_{SA}) \end{bmatrix}$$

Lower Wishbone:



$$\sum M_R = LW_z(L_{LW} \cdot \cos\theta_{LW}) - LW_y(L_{LW} \cdot \sin\theta_{LW}) - F_S \cdot \cos(\theta_s + \theta_{LW}) \cdot (L_{LW} - S_1) = 0$$

$$\sum F_y = 0 = -LW_y + F_s\cos\theta_s + R_{Ly}$$

$$R_{Ly} = LW_y - F_s\cos\theta_s$$

$$\sum F_z = 0 = -LW_z + F_s\sin\theta_s + R_{Lz}$$

$$R_{Lz} = LW_z - F_s\sin\theta_s$$

Composite Stress Analysis

```
clear all, clc, close all

%Material Properties

%Inputs to ABD

np=10 %NUMBER OF PLIES

epsilon_ult_1t=0.015;
% epsilon_ult_1c=1
epsilon_ult_2t=0.006;
% epsilon_ult_2c=1
% gamma_ult_6=1

E1=transpose(linspace(21.3e6, 21.3e6, np)) ; %ELASTIC MODULUS,
 LONGITUDINAL, VECTOR LENGTH n %psi

E2=transpose(linspace(1.5e6, 1.5e6, np))  ; %ELASTIC MODULUS,
 TRANSVERSE, VECTOR LENGTH n

G12=transpose(linspace(1e6, 1e6, np)) ; %SHEAR MODULUS, VECTOR LENGTH
 n

nu12=transpose(linspace(.27, .27, np)) ; %MAJOR POISSON'S RATIO,
 VECTOR LENGTH n
theta=[0; 45; 90; -45; 0; 45; 90; 0; -45; 90]; %PLY ANGLES, BOTTOM-TO-
TOP, GIVEN AS VECTOR WITH LENGTH n
materials=['same']

h=.100; %in
k=np;
z=z_even(h,k) ;  %LAMINATE FACES DISTANCES FROM MIDPLANE, GIVEN AS A
 VECTOR WITH LENGTH EQUAL TO THE NUMBER OF PLIES + 1

[A, B, D]=laminate_ABD(z,np,E1,E2,G12,nu12,theta);

%Compliance
[a, b, c, d]=laminate_compliance(A,B,D);

% Properties
[Ex, Ey, Gxy, nuxy, nuyx, etasx, etaxs, etays, etasy] =
 laminate_properties(h, a);

% Mechanical Loading
N=[0; 1000; 0];
M=[0; 0; 0];

scinv=[a b; c d]*[N; M];
epsilon0=scinv([1 2 3]);
kappa=scinv([4 5 6]);
```

```
% Layer Strains xy

epsilon_xy=epsilon0+(z.*kappa);

thetak=0;
m=cos(thetak);
n=sin(thetak);
T=[m^2 n^2 2*m*n; n^2 m^2 -2*m*n; -m*n m*n ((m^2)-(n^2))];

% Layer Strains 12
epsilon_12=T*epsilon_xy;

nu21=nu12.*(E2./E1); %4.49
Q12=(nu12.*E2)./(1-(nu12.*nu21)); %4.56
Q11=E1./(1-(nu12.*nu21)); %4.56
Q22=E2./(1-(nu12.*nu21)); %4.56
Q66=G12; %4.56

% sigma_12=linspace(1, 1, np)
for k=1:np
sigma_12(:,k)=[Q11(k) Q12(k) 0; Q12(k) Q22(k) 0; 0 0
 Q66(k)]*epsilon_12(:,k);
end

% Tsai-Wu Lamina Strengths

F1t=E1*epsilon_ult_1t;
% F1c=-E1*epsilon_ult_1c
F1c=transpose(linspace(222000, 222000, np));
F2t=E2*epsilon_ult_2t;
% F2c=-E2*epsilon_ult_2c
F2c=(F1c/F1t)*F2t;
%F6=gamma_ult_6*G12
F6=transpose(linspace(15000, 15000, np));

% Tsai-Wu Coefficients
for k=1:np
[f11(k), f22(k), f1(k), f2(k), f66(k), f12(k)] =
 Tsai_Wu_2D(F1t(k),F2t(k),F1c(k),F2c(k),F6(k));
end

% Tsai-Wu Factors of Safety
sigma_1=sigma_12(1,:);
sigma_2=sigma_12(2,:);
tau_6=sigma_12(3,:);

for k=1:np
    [FS_a(k), FS_b(k)] =
 Tsai_Wu_2D_Failure(sigma_1(k),sigma_2(k),tau_6(k),f1(k),f2(k),f11(k),f22(k),f66(k
end

FS_a_lam=min(FS_a)
FS_b_lam=min(FS_b)
```

```
np =

    10

materials =

    'same'

x =

     1

FS_a_lam =

    3.5918

FS_b_lam =

    2.5881
```

Published with MATLAB® R2017b

Destuctive Testing Reslults

Tensile Tests

Number	Max Load (lb)
1	3081.665
2	1846.924
3	1992.187
Average	2306.925333

4 Point Bending Tests

# of Plies	Max Load (lb)
8	600
12	410
16	570
16 Wrapped	1400

Pull Out Tests

# of Plies	Max Load (lb)
12	1500
16	1800
20	2550

Cantilevered Bearing

# of Plies	
12	3500

Appendix G- Safety Hazard Checklist, FMEA, Risk Assessment

DESIGN HAZARD CHECKLIST

Team: PROVE Ultra Range Vehicle Advisor: McFarland Date: 11/9/17

Y	N	
☐	☒	1. Will the system include hazardous revolving, running, rolling, or mixing actions?
☐	☒	2. Will the system include hazardous reciprocating, shearing, punching, pressing, squeezing, drawing, or cutting actions?
☒	☐	3. Will any part of the design undergo high accelerations/decelerations?
☒	☐	4. Will the system have any large (>5 kg) moving masses or large (>250 N) forces?
☐	☒	5. Could the system produce a projectile?
☐	☒	6. Could the system fall (due to gravity), creating injury?
☐	☒	7. Will a user be exposed to overhanging weights as part of the design?
☐	☒	8. Will the system have any burrs, sharp edges, shear points, or pinch points?
☐	☒	9. Will any part of the electrical systems not be grounded?
☐	☒	10. Will there be any large batteries (over 30 V)?
☐	☒	11. Will there be any exposed electrical connections in the system (over 40 V)?
☒	☐	12. Will there be any stored energy in the system such as flywheels, hanging weights or pressurized fluids/gases?
☐	☒	13. Will there be any explosive or flammable liquids, gases, or small particle fuel as part of the system?
☐	☒	14. Will the user be required to exert any abnormal effort or experience any abnormal physical posture during the use of the design?
☒	☐	15. Will there be any materials known to be hazardous to humans involved in either the design or its manufacturing?
☐	☒	16. Could the system generate high levels (>90 dBA) of noise?
☐	☒	17. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, or cold/high temperatures, during normal use?
☐	☒	18. Is it possible for the system to be used in an unsafe manner?
☐	☒	19. For powered systems, is there an emergency stop button?
☐	☒	20. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

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

Description of Hazard	Planned Corrective Action	Planned Date	Actual Date
High Accelerations and Decelerations	Structurally design system to withstand loads from acceleration and deceleration	11/16-1/11	1/20
High masses and loads	Structurally design to withstand loads	11/16-1/11	1/20
Stored energy	Develop procedure for installing and removing spring (Remove preload)	3/8	
Hazardous Materials	Follow safe operating procedures from Prove for handling hazardous materials	3/8	

System / Function	Potential Failure Mode	Potential Effects of the Failure Mode	Severity	Potential Causes of the Failure Mode	Current Preventative Activities	Occurence	Current Detection Activities	Detection	Priority	Recommended Action(s)	Responsibility & Target Completion Date	Action Results: Actions Taken	Action Results: Severity	Action Results: Occurence	Action Results: Criticality
Upright must support the loads applied to the system	Fracture	System collapses Car crashes	10	Material failure Failure at joints Upright is too thin	1. FOS 2. Stress analysis 3. Destructive testing 4. Fatigue strength	2	1. Frequent inspection	5	100						
	Deflection	System in unstable Uncomfortable ride	4	Material is not strong enough Upright is too thin Missallignment	1. Deflection analysis 2. Stress analysis	4	1. Frequent inspection 2. Measurement of critical dimensions	3	48						
Linkage must maintain correct suspension geometry and support applied loads	Breaking	System collapses Car crashes	10	Failure of linkages Failure of Ball joints Failure of Rod Ends	1. FOS 2. Stress analysis 3. Destructive testing 4. Fatigue strength	1	1. Frequent inspection	5	50						
	Bending	Desired handling not achieved Loss of vehicle control Binding Link Interference	9	Poor manufacturing tolerances Deformation of components Flawed geometric design	1. Geometry optimization 2. Stress analysis	1	1. Frequent inspection 2. Measurement of critical dimensions	3	27						
	Cracking	Increased likelyhood of breaking	8	Flawed fatigue analysis	Fatigue strength	3	1. Frequent inspection	5	120						
The spring stores energy from the applied loads to provide proper ride and handling characteristics	Plastic deformation	Damage to the system System is unstable	4	Spring fatigue Material defects Manufacturing flaws Complex loading	1. Fatigue strength 2. FOS	2	1. Frequent inspection 2. Measurement of critical dimensions	3	24						
	Breaking	System Collapses Car crashes	10	Spring fatigue Spring failure Material defects Manufacturing flaws Complex loading	1. Fatigue strength 2. FOS	1	1. Frequent inspection	5	50						
	Uncomfortable ride	User is uncomfortable	5	Improper spring rate Spring wear	Choose spring rate according to loads and other specs.	5	1. Electronic sensors 2. Driver feedback	2	50						
	Poor handling	Car is out of control Car crashes	10	Improper spring rate Spring wear	Choose spring rate according to loads and other specs.	5	1. Electronic sensors 2. Driver feedback	2	100						
Damper must dissipate the energy stored by the spring to provide proper ride and handling characteristics	Uncomfortable ride	User is uncomfortable	5	Improper damping ratio Damper wear	Choose damping rate according to loads and other specs.	5	1. Electronic sensors 2. Driver feedback	2	50						
	Poor handling	Car is out of control Car crashes	10	Improper damping ratio Damper wear	Choose damping rate according to loads and other specs.	5	1. Electronic sensors 2. Driver feedback	2	100						
Joining methods must withstand applied loads	Separation of Joints	System Collapses Car Crashes	10	Bolt failure Improper bolt selection Improper bolt preload Adhesive failure Improper adhesive selection Improper adhesive process	1. Bolt shear analysis 2. Adhesive testing	4	1. Frequent inspection	5	200						

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Control System	Final Assessment Severity Probability	Risk Level	Status / Responsible /Comments /Reference
1-1-1	driver normal operation	mechanical : loss of vehicle control Improper geometry	Catastrophic Remote	Low		Catastrophic		
1-1-2	driver normal operation	ergonomics / human factors : Uncomfortable ride Improper spring/damper ratio	Moderate Unlikely	Low		Moderate		
1-1-3	driver normal operation	noise / vibration : fatigue / material strength Flawed analysis/testing	Catastrophic Unlikely	Medium		Catastrophic		
2-1-1	maintenance technician periodic maintenance	mechanical : pinch point	Moderate Unlikely	Low		Moderate		
2-1-2	maintenance technician periodic maintenance	mechanical : Stored Energy	Serious Unlikely	Medium		Serious		
2-2-1	maintenance technician set-up or changeover	mechanical : pinch point	Moderate Unlikely	Low		Moderate		
2-2-2	maintenance technician set-up or changeover	mechanical : Stored Energy	Serious Unlikely	Medium		Serious		

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Control System	Final Assessment Severity Probability	Risk Level	Status / Responsible /Comments /Reference
2-2-3	maintenance technician set-up or changeover	fluid / pressure : fluid leakage / ejection Seal failure	Minor Unlikely	Negligible		Minor		
2-3-1	maintenance technician parts replacement	mechanical : pinch point	Moderate Unlikely	Low		Moderate		
2-3-2	maintenance technician parts replacement	mechanical : Stored Energy	Serious Unlikely	Medium		Serious		
2-3-3	maintenance technician parts replacement	fluid / pressure : fluid leakage / ejection Seal failure	Minor Unlikely	Negligible		Minor		
3-1-1	installer set up / install machinery	mechanical : pinch point	Moderate Unlikely	Low		Moderate		
3-1-2	installer set up / install machinery	mechanical : Stored Energy	Serious Unlikely	Medium		Serious		
3-1-3	installer set up / install machinery	fluid / pressure : fluid leakage / ejection Seal failure	Minor Unlikely	Negligible		Minor		
3-2-1	installer test machinery	mechanical : Stored Energy	Moderate Unlikely	Low		Moderate		
3-2-2	installer test machinery	noise / vibration : fatigue / material strength Flawed analysis/testing	Serious Unlikely	Medium		Serious		

PROVE ENDURANCE CAR FRONT SUSPENSION
Operators Manual

Written by Lauren Williams, Logan Simon, and Justine Kwan
June 6, 2018

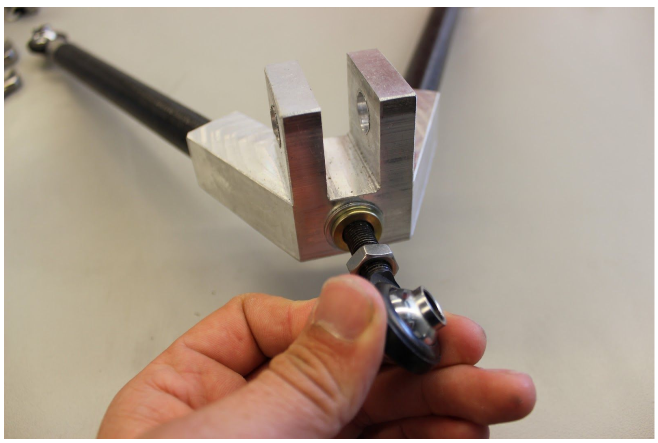
Assembly Instructions

I. The first step is to assemble the wishbone assembly. A list of required parts and instructions for assembly can be found below.

Equipment Required:
- Wishbones (2)
- Ball Joints (6)
- 3/8 -24 Jam nuts (6)
- ⅜ Lock Washers (2)
- ⅜ Washers (2)
- 9/16 Wrench

1. Thread the four ball joints into the threaded inserts inside the carbon tubing of the wishbone joint and lock in place with jam nuts.

2. Thread the remaining two ball joints into the head of the wishbone joint lock in place with jam nuts, washers, and lock washers.



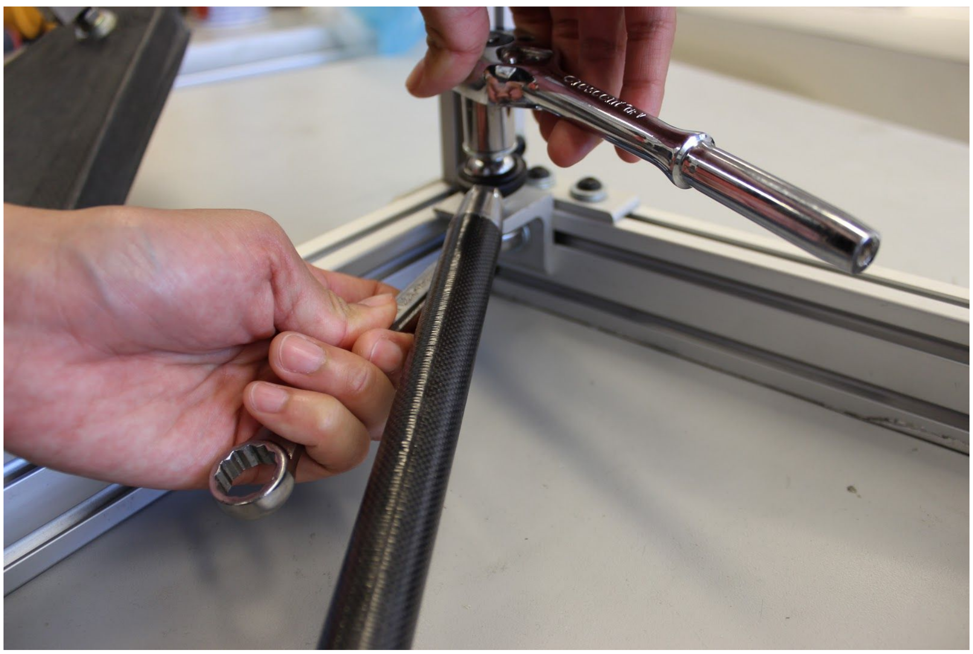
II. Now the lower wishbone can be mounted to the chassis. A list of required parts and instructions for assembly can be found below.

Equipment Required:

- Wishbone Assembly (1)
- Mounting Brackets (2)
- ⅜ Bolts (2)
- 9/16 Socket Wrench
- 9/16 Wrench

1. The ball joints at the end of the wishbones can be twisted to match the width between the chassis mount points. These should be adjusted symmetrically.

2. Bolt the four ball joints at the end of the carbon tubing to the four brackets.



III. The spring and damper can be assembled according to the instructions from the manufacturer. A link to the manual you should reference can be found below.

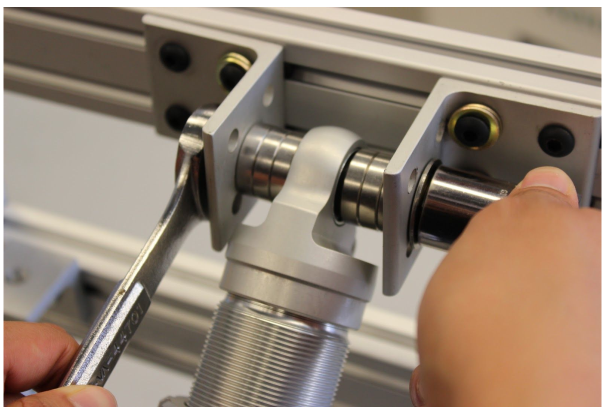
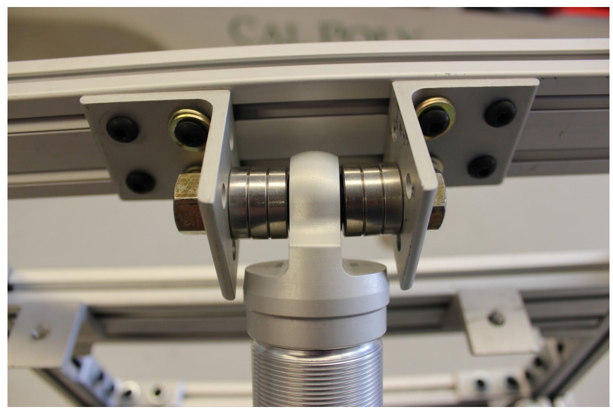
Link : https://www.ohlins.com/app/uploads/world/documents/2015/11/OM_07241-02.pdf

IV. Then, attach the assembled spring and damper to the lower wishbone joint as instructed below.

Equipment Required:

- Lower Wishbone Assembly (1)
- Mounting Bracket (1)
- ½ Bolts (2)
- Lock Nuts (2)
- Spacers (will vary)
- ½ Washers (2)
- ¾ Socket Wrench
- ¾ Wrench
- Spanner Wrenches

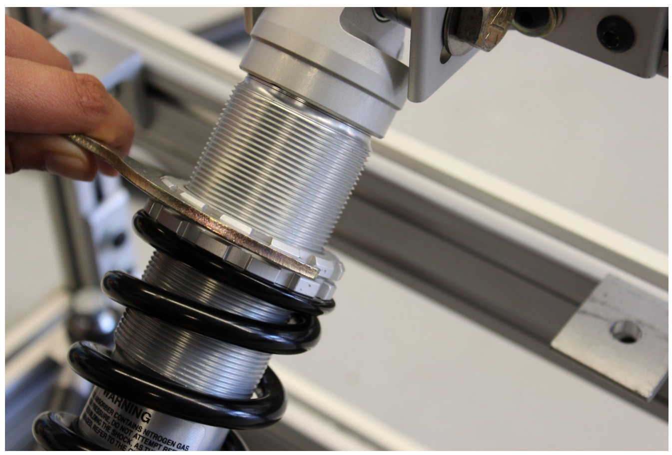
1. Bolt the spring and damper to the bracket using the appropriate amount of spacers needed.

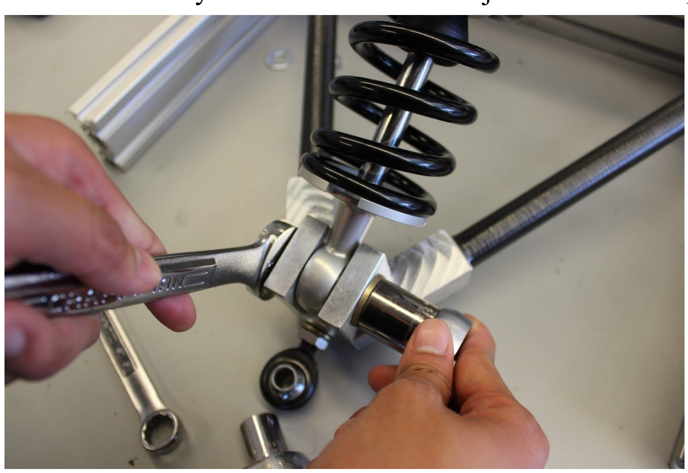
2. Bolt the bracket to the chassis (exact process to be defined).
3. Add the required preload.
    a. Loosen top ring using spanner wrench.

b. Tighten bottom ring to the desired preload.

c. Retighten top ring.



4. Mount the strut assembly to the lower wishbone joint with its bolt, lock nut, and washers.

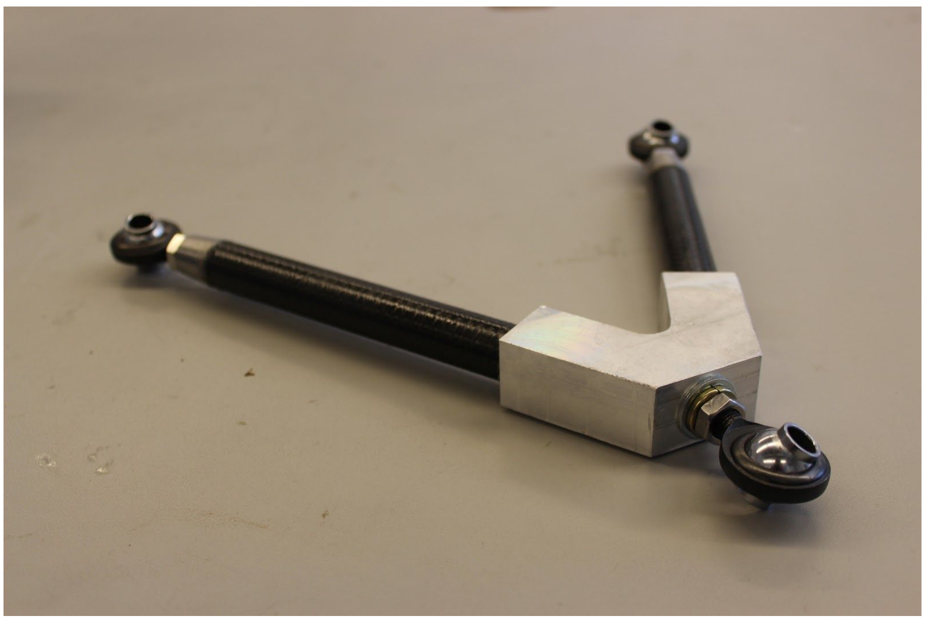


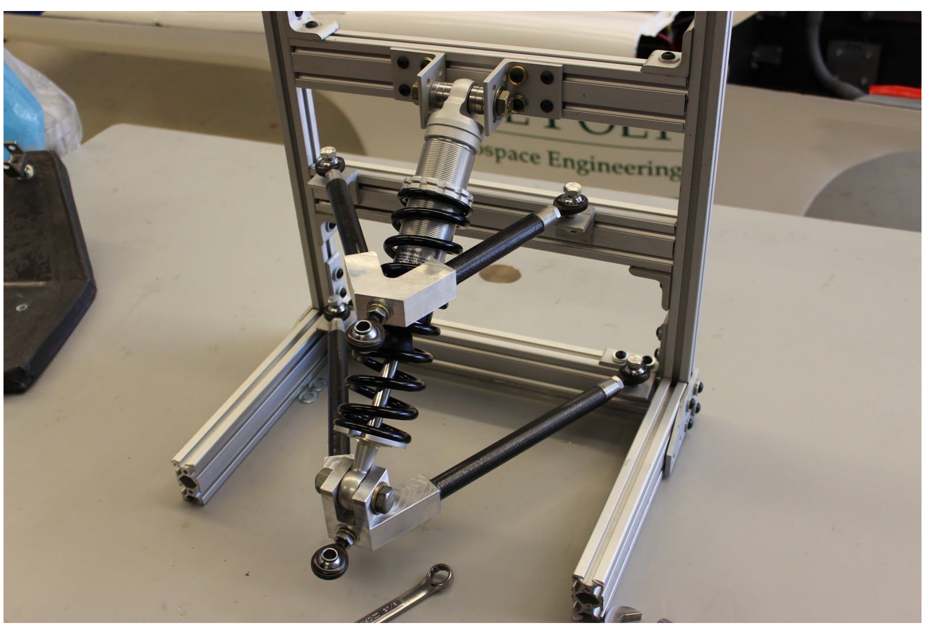
V. Now the upper wishbone can be mounted to the chassis. A list of required parts and instructions for assembly can be found below.

Equipment Required:

- Wishbone Assembly (1)
- Mounting Brackets (2)
- ⅜ Bolts (2)
- 9/16 Socket Wrench
- 9/16 Wrench

1. Repeat process detailed in Step II.



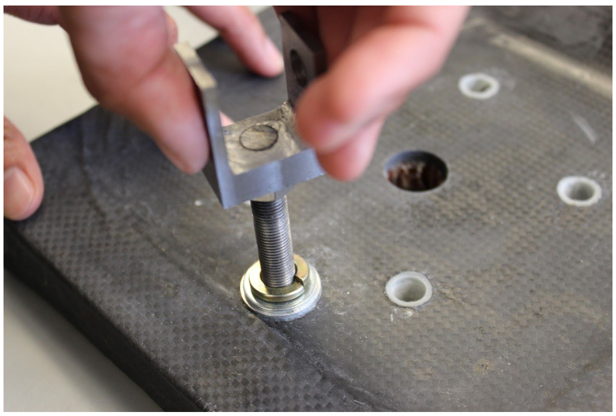


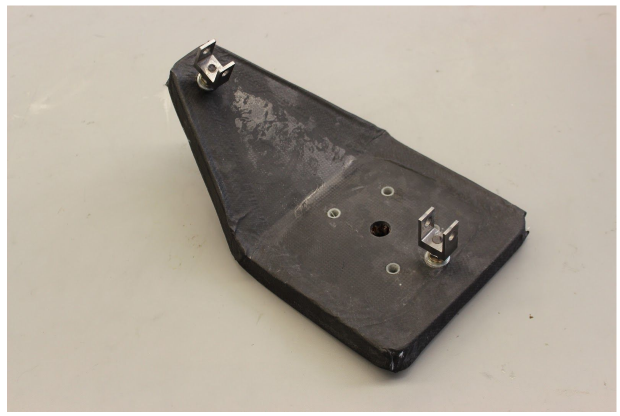
VI. The upright can now be attached to the wishbone joints as follows.

Equipment Required:

- Wishbone Assemblies (2)
- Clevises (2)
- Upright (1)
- ⅜-24 Bolts (4)
- ⅜-24 Jam Nut (2)
- ⅜ Washer (4)
- Lock Washer (2)
- 9/16 Socket Wrench
- 9/16 Wrench

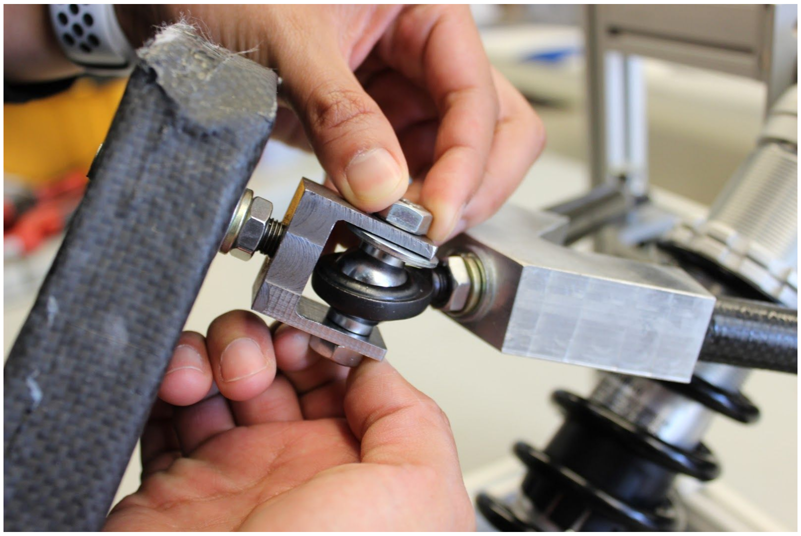
1. Thread the clevises through the upright potted inserts and lock them in place with jam nuts, washers, and lock washers as indicated in the diagram below.





2. Place a bolt through the lower clevises and ball joints first and secure.

3. Repeat for the top clevis and ball joint with two washers added for clearance.



VII. Ensure hardware is tight throughout
VIII. GO BREAK THE RECORD!

Adjustment and Tuning

Adjusting Track Width and Camber

1. Remove bolt between clevis and ball joint.
2. Remove clevis or ball joint.
3. Twist to adjust length needed for Track Width/Camber adjustment.
4. Replace all parts.

Adjusting Shocks

The Shock operators manual should be referenced to tune the shocks. It may be desirable to adjust the damping ratios and spring rates of the car based on the changing vehicle parameters and following drive testing.

Link : https://www.ohlins.com/app/uploads/world/documents/2015/11/OM_07241-02.pdf

Appendix I- Purchased Parts Details

Threaded Inserts- https://www.mcmaster.com/#94640a115/=1bjya6o

Ball Joints- https://www.mcmaster.com/#6960t6/=1bjy9tx

G10 Tubing- https://www.mcmaster.com/#8668k14/=1bjyatc

Carbon Tubes-
https://www.rockwestcomposites.com/downloads/RW-PS-071-Stock-Tube-Tolerance-Sheet.pdf

Appendix J- Budget/Procurement List

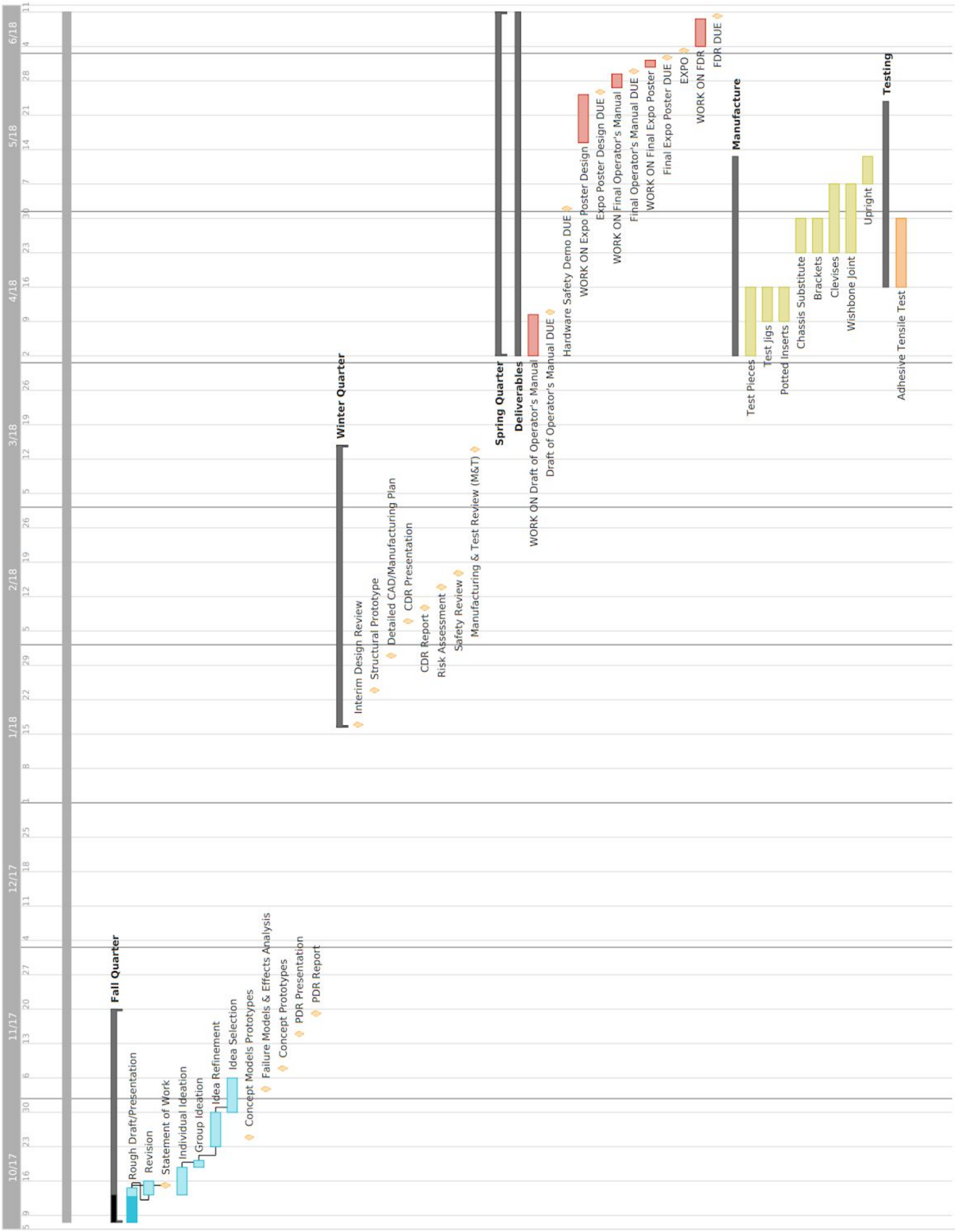
Assembly	Item	Quantity	Part Number	Details	Vendor	Donated	Purchased
Upright	Carbon Weave	-	V61401011A	HexForce 3k weave	Hexcel	X	
	Core	-	-	⅝” Nomex Flex Core	Westerly Marine	X	
	Laminating Resin	-	-	LAM-135	PRO-SET	X	
	Laminating Hardener	-	-	LAM-226	PRO-SET	X	
	G10 Tubing	1	8668K14	½” outer diameter, ⅜” inner diameter 39” length	McMaster Carr		X
	Aluminum (potted inserts)	1	18037	4 ft of 1" aluminum round rod	OnlineMetals		X
Wishbones	Aluminum (lower wishbone joints)	1	12769	Aluminum 6061 2.5”x3”x10”-12”	OnlineMetals		
	Aluminum (upper wishbone joints)	1	14720	Aluminum 6061 2.5”x4”x2.5”	OnlineMetals		
	Carbon Tubing	1	45558-UHM	0.625"ID x 0.750" OD ultra high modulus tubing	RockWest Composites		X
	Tube End Weld Nuts	10	94640A115	⅜”-24 partially threaded	McMaster Carr		X
	Ball Joints	12	6960T6	⅜”-24 Super-swivel Black-oxide alloy steel	McMaster Carr		X
	Epoxy Resin	-	-	M1033	PRO-SET	X	
	Epoxy Hardener	-	-	M2036	PRO-SET	X	
Clevises	Steel	1	9902	Steel 1018 1.25”x2”x10”-12”	OnlineMetals		X
	Threaded Rod	1	90322A134	High-Strength Steel Threaded Rod 3/8"-24 Thread Size, 3 Feet Long	McMaster Carr		X
-	Strut	2	HD022	Harley Davidson FLHRS Road King	Ôhlins	X	
-	Fasteners	-	-	⅜”-24	McMaster Carr		X

Senior Project DVP&R

Date: 5/1/18 | Team: Suspenders | Sponsor: Graham Doig | Description of System: Double Wishbone Suspension System | DVP&R Engineer:

		TEST PLAN									TEST REPORT			
Item No	Specification #	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES TESTED		TIMING			TEST RESULTS			NOTES
						Quantity	Type	Start date	Finish date		Test Result	Quantity Pass	Quantity Fail	
1		Bearing test of potted inserts	N/A- destructive	Justine Kwan	CP	3	C	5/9/2018	5/9/2018					Destructive test: all parts fail.
2		Pullout test of potted inserts	N/A- destructive	Justine Kwan	CP	3	C	5/9/2018	5/9/2018					Destructive test: all parts fail.
3		Tensile test of structural adhesive	N/A- destructive	Justine Kwan	CP	3	C	5/9/2018	5/9/2018					Destructive test: all parts fail.
4		4 point bending test of sandwich panels	N/A- destructive	Justine Kwan	CP	3	C	5/9/2018	5/9/2018					Destructive test: all parts fail.
5		Measure dimensions of machined parts	Within tolerance specified on drawing	Lauren Williams	FP	4	Sub	5/11/2018	5/18/2018		Within tolerance			
6		Measure dimensions of final composite part	Within tolerance specified on drawing	Lauren Williams	FP	1	Sub	5/28/2018	5/28/2018		Within tolerance			
7		Visual inspection of adhesive coverage	Bond must have all around coverage without voids	Lauren Williams	FP	4	Sub	5/31/2018	5/31/2018					Good coverage observed.
8		Test component packaging	All components fit with adequate clearances	Logan Simon	FP	1	Sys	-	-					To be done by PROVE Lab.
9		Load testing on final part	No fail within safety factor load	Logan Simon	FP	1	Sub	-	-					To be done by PROVE Lab.
10														

Appendix L - Gantt Chart



Appendix L - Gantt Chart