Advisor of Record Initials: ECC
Project Number: A702

2008 Formula SAE Racecar

A Major Qualifying Project Report
submitted to the Faculty of
WORCESTER POLYTECHNIC INSTITUTE
in partial fulfillment of the requirements for the
Degree of Bachelor of Science
by

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Date: April 24th, 2007

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Acknowledgements

The 2007-2008 Formula SAE team greatly appreciates the support and contributions of everyone throughout the year. We would especially like to thank Brian Barnhill and Matt Douville for their endless assistance in the design, analysis, and fabrication of the entire car; specifically the chassis, intake, and electronics. We would like to recognize Greg Cole for the paddle shifter system and Chris Stefaniak and Doug Cimon for their manufacturing help with the CNC labor. We would also like to thank the businesses that helped us via various donations and discounts given to the team. Thanks go out to Bodycote for donating their services to stress relieve and heat treat various components of the vehicle, Dale Jarrett Racing for donating a gift certificate and Autotech, AED, Tuner Tools and Polaris for product discounts. Finally, we would like to thank WPI and their faculty members who make this project possible: Professor Eben Cobb and Neil Whitehouse for their support and guidance, Barbara Furhman and Pamela St. Louis for their patience and endless logistical support, and Toby Bergstrom and Brendan Powers for their shop space and understanding along with the support of family members, friends and mentors along the way.

Abstract

This project involves the design, analysis, and manufacturing of a Formula SAE approved racecar with the intent of national competition. Through the division of the major vehicle components into the chassis, suspension, drive-train, controls, and aerodynamics, the project team effectively integrates each subsystem while considering performance, dependability, manufacturability, and cost. This project includes a detailed focus on the engineering of each subsystem; namely individual part designs, strength analysis, optimization techniques, process descriptions, and financial details. The project concludes with Formula SAE competition and submission of design and cost reports focused around the concept of real-world production of the vehicle.

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Introduction

Our project involved designing and building a racecar to compete in the Formula SAE competition at Michigan International Speedway. From May 14th-18th we will compete against 121 teams from five continents in a variety of static and dynamic events. For static events we will have a cost and manufacturability event, presentation event and design judging. For dynamic events we will compete in acceleration, skid pad, autocross and endurance events. During design and fabrication we had to take into consideration the events we would be participating in and decide on various sacrifices such as cost to performance or weight to reliability.

Objective

We first set out a few goals that we wanted to achieve as a team. A few of these were to have a much more lightweight car, focus heavily on suspension design, and to have a fairly simple yet competitive car running and tested before competition. In order to achieve all of these goals we first broke down the school year into stages. A-term consisted of mostly design with B-term consisting of analysis and integration of the various subsystem designs. In C-term we focused heavily on fabrication with D-term finishing production, assembling the car and tying up any loose ends followed by testing.

A few of the notable features of our car this year is first off its detachable rear sub-frame. The rear sub-frame is made of 7075 T-651 Aluminum with ANSI 4130 steel as supports. By using the rear sub-frame we are able to leave out a significant amount of ANSI 4130 steel which would otherwise have not been provided any added structural benefit. Another feature is the unique suspension design. We have utilized a monoshock design, meaning we use a single rocker and damper in each the front and rear. This saves both cost and weight on the car and allowed us to afford better quality dampers this year. The control arms and geometry was also all redone to have optimal roll center location and to eliminate rod ends in bending. All of these points will be discussed further in various sections of our paper.

Manufacturability, Cost Report, Marketing

Three aspects of our projects which vary from simple design are the cost, manufacturing and marketing components of the final vehicle. Each of these must be considered while designing the final vehicle. For the cost component of our project we are required to put together a report for competition outlining every piece of the vehicle.

The entire car is first broken down into eight sub components such as brakes, engine & drivetrain, suspension, etc. Then every component in that system is cost. This includes bought parts, manufactured parts and fasteners. For each bought part a receipt is gathered and included in the receipts section of the report. For each manufactured part a process description is required. The process description outlines the raw material used as well as any manufacturing processes carried out on the material such as manual labor, drilled holes, inches of welding, etc. All of these processes and raw materials have designated costs laid out by the FSAE Rules committee.

This years vehicles cost was a total of $17,817.55. After finishing the cost report it became very apparent that there are many parts which specifically could bring down the cost of our vehicle. A few of these components are the rear hubs, brake rotors, muffler and axle components. The rear hubs are currently modified hubs and would be much cheaper if manufactured. The brake rotors would also be fairly easy to machine given time and would save about one hundred dollars. A custom muffler could be manufactured to save a significant amount of money but would need to ensure to still adequately muffle the exhaust. The axle components are very expensive currently and new ways of making these could help.

Aiming for manufacturability on our racecar is always a major concern. This year we managed to find a few innovative ways to make it easier to manufacture our formula SAE car. The main concern for manufacturability is the parts which must be machined either in house or contracted out. This process can be made easier by having many interchangeable parts with as close to an equivalent design as possible. One example of this is our uprights this year, these have the same outline and major features front and rear. This would allow for all the uprights to be cast out of the same basic mold and then have the inside machined based on whether it would be used for the front or rear. The rocker for the monoshock is the exact same design front and rear and the hubs could be made to be cast out of the same molds as well.

For the marketing aspect of our project our vehicle has a much more marketable design this year than years past. This is due to better design, adjustability and also better aesthetic appeal. Our design will appeal to those who are looking for a well designed vehicle since all the important factors of a road racing vehicle have been calculated and can be backed up with statistics and figures. Also the car is very adjustable while still abiding by safe engineering practices such as having the camber adjusted via slots in the uprights vs. using heim joints in bending. Both our frame and body have a pleasing aesthetic look to them this year. The frame is all triangulated and sleek looking with the front end pulled up for ergonomics. By using smaller diameter tubing where possible it actually gives the entire frame an appearance of being very lightweight. By having a separate project focus on the body of our vehicle more time was able to be spent in its design. Since the body is well designed aerodynamically it also comes across as visually appealing as well as functional.

Design, Analysis and Fabrication

In this section our vehicle is broken down into sub-components with each one described from design and analysis through fabrication. The main design point and changes from past years are focused on heavily due to the vast difference is the design process used this year.

Frame

Preliminary Design Considerations

The frame of the chassis performs several vital tasks including providing a rigid connection for the suspension points (left to right, and fore to aft) while providing additional attachment points for other vital systems of the vehicle. In most cases, the chassis also serves as the main mode of protection for the driver during normal operation as well as in the event of a collision. There are many design considerations for a race vehicle chassis, which include: torsional and bending stiffness, ergonomics, weight, yielding strength, and systems packaging.

Suspension design typically assumes an infinitely stiff chassis. This is not the case, however, as typical chassis material will displace under a given load. This deflection, if too great, will adversely affect the roll stiffness of the vehicle. The chassis is all reality and additional spring in the suspension system. The standard engineering criteria for a chassis is that the torsional stiffness be at least ten times greater than the roll stiffness of the suspension. This will prevent the suspension mounting points from twisting and displacing relative to each other and ensure proper suspension geometry and behavior.

Area and Polar moments of inertia also have an effect of chassis stiffness. Area moment of inertia is a used to determine a shapes resistance to bending and deflection while polar moment of inertia is used to predict a shapes resistance to twisting when a torque is applied. The definitions of these properties are show in equations 1 and 2.

Equation 1: Definition of Polar Moment of Inertia

$$J_x = \int r^2 \, dA$$

Jx = the polar moment of inertia about the axis x

dA = an elemental area

r = the radial distance to the element dA from the axis x

The polar moment of inertia is used in the formula that describes torsional stress and angular displacement.

Equation 1.1: Torsional Stress

$$\tau = \frac{Tr}{J_x}$$

T = torque

r = radius

Jx = polar moment of area.

Equation 2: Definition of Area Moment of Inertia

$$I_x = \int y^2 \, \mathrm{d}A$$

Ix = moment of inertia about the axis x

dA = elemental area

y = the perpendicular distance from the axis x to the element dA

Material further from the axis of twist will increase the polar and area moment of inertia, thus increasing the chassis stiffness. Diamond shaped chassis, and integration of structural side pods are methods commonly used to increase the stiffness of the chassis. Care should be taken, however, as adding material can increase the mass of the vehicle. The placement of mass may also have an effect on the center of mass of the vehicle, and could possible adversely impact the performance of the vehicle.

Analysis using Finite Element Methods and software as well as physical torsional testing can be used to verify these criteria. Torsional stiffness may be increased through additional bracing and frame members; however, this may have a detrimental impact on the performance of the vehicle due to that additional mass. Care should be taken to provide as light a chassis as possible, while maintaining optimal torsional stiffness.

The behavior of the chassis subjected to bending loads is another measure of the chassis stiffness. Like the torsional stiffness, bending stiffness is a measure of the spring rate and deflection of the chassis when subject to a bending load. Bending stiffness is typically less important than torsional stiffness; as such deflection does not affect the wheel loads. Additionally, a torsionally stiff frame typically provides adequate resistance to bending.

Yield strength of the selected material is an additional consideration. Typically a chassis sufficiently stiff will not fail due to yielding, however, placement of attachment points should be designed as to prevent yielding at the attachment point. Examples include engine mount locations, and suspension mounting tabs. Care should be taken regarding the placement and lengths of such tabs. The 2007 WPI Formula SAE Chassis provides an example of mounting practices to be avoided (note the length and placement of the tabs in relation to the design loads in Figure 1)

Figure 1: Excessive Tab Length on 2007 WPI Chassis

Packaging of the system components is vital to the frame design. All system components must lie within the frame of the chassis. This will often dictate the location and paths of the chassis members, which may be not allow for the most optimized load path. The suspension, for example, must be attached to the chassis without the use of excessively long mounting tabs. The engine and drive-train system, as well as cockpit design and ergonomics all affect the design of each chassis member. Additionally, access and ease of maintenance are factors which may dictate frame design. The 2008 Formula SAE rules also dictate packaging constraints, specifically regarding the driver and aerodynamic devices. Figure 2 shows the mandated clearance for all drivers to protect the driver's head during a roll over situation.

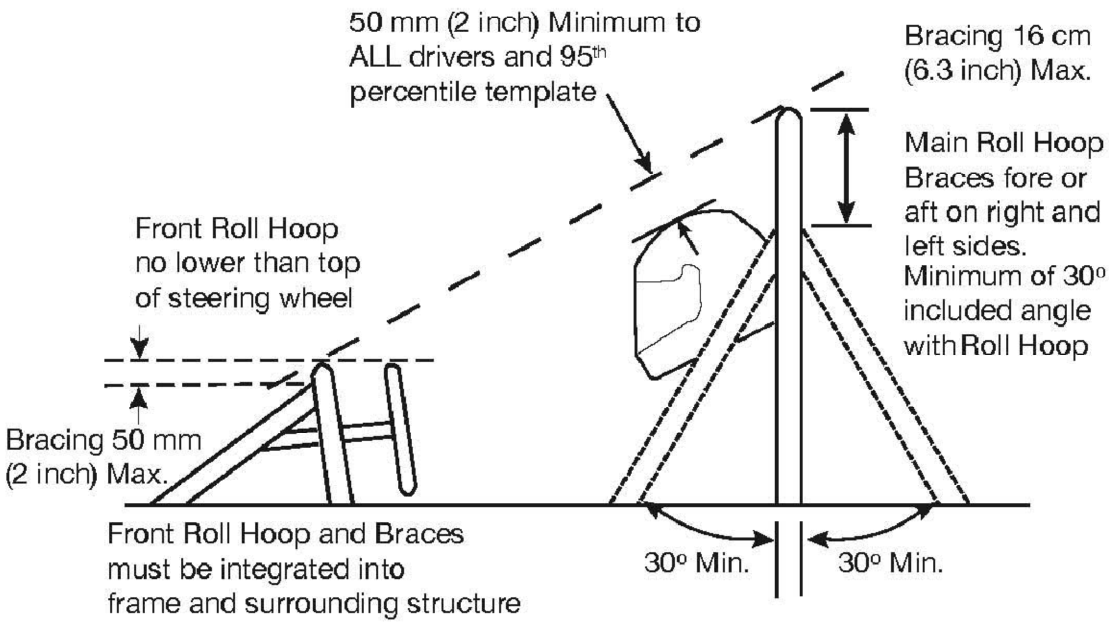

Figure 2: Driver Roll Hoop Clearance from 2008 Formula SAE Rules

The rules also mandate the location of aerodynamic devices in relation to the wheels, and fuel and coolant systems in relation to the driver. Any aerodynamic device may not be further than 18 inches from the front tires. If aerodynamic devices are to be used, the frame must allow adequate space. Additionally, there may be no line of sight from the driver's position to any part of the fuel or cooling system. All these systems must be packaged within the frame, while adhering to the 2008 Formula SAE rules. In addition, systems like the fuel tank which greatly affects the center of mass of the vehicle should be carefully considered in the packaging.

Driver comfort and ergonomics are factors which have previously been secondary in the design of WPI Formula SAE chassis. Driver safety and ease of operation have suffered as a result. The 2008 Formula SAE rules mandate the vehicle accommodate from the 5th percentile female to the 95th percentile male. The vehicle must provide adequate protection during roll-over or impact situations, while also providing free range of movement of the driver and controls, all while packaging the necessary components of the car. Rule section 3.4 (see Appendix A) provides a detailed explanation of required safety harness and cockpit equipment. The safety harness bar must be located as to provide a safe mounting location for the harness, while accommodating all drivers. In addition to the safety rules, Formula SAE rules mandate a maximum driver egress time of 5 seconds. In the past, WPI Formula SAE chasse have not allowed enough leg clearance and/or placed components (i.e. the steering rack) in a location which interferes with the drivers egress, all inhibiting the drivers free movement in and out of the cockpit.

Consideration should be given to load paths at all times during design of the chassis. The load path defines the path which the resultant force are dissipated from a given load (i.e. suspension loads.) Providing proper load will prevent overly complicated structures, providing a strong, but lighter chassis. Free body diagrams of all loads and systems connected to the chassis provides a description of the necessary load path and allows the designer to visualize the load paths of the chassis and sub-systems.

Chassis Type

Selection of the frame type will govern many of the design methods and geometry of the chassis design. The most common race vehicle chassis are: Space-frame, Ladder and Monocoque. The most common chassis in Formula SAE are the space frame and Monocoque chassis. Strength, stiffness, cost, manufacturability, packaging and ergonomics are all factors which effect the selection of chassis type, and will all vary for each type. The chassis type plays a large role in the properties of the chassis, and design goals and compromises should be considered in the selection of the chassis design.

Ladder Chassis

The ladder chassis is arguably the simplest chassis and the easiest to build. A ladder chassis (Figure 3) uses 2 frame members, running fore to aft, with cross bracing to provide support and additional attachment points, closely resembling a ladder, thus the name. A ladder chassis can be constructed quickly, with simple fixtures a jigging, providing a cheap, light and time effective chassis.

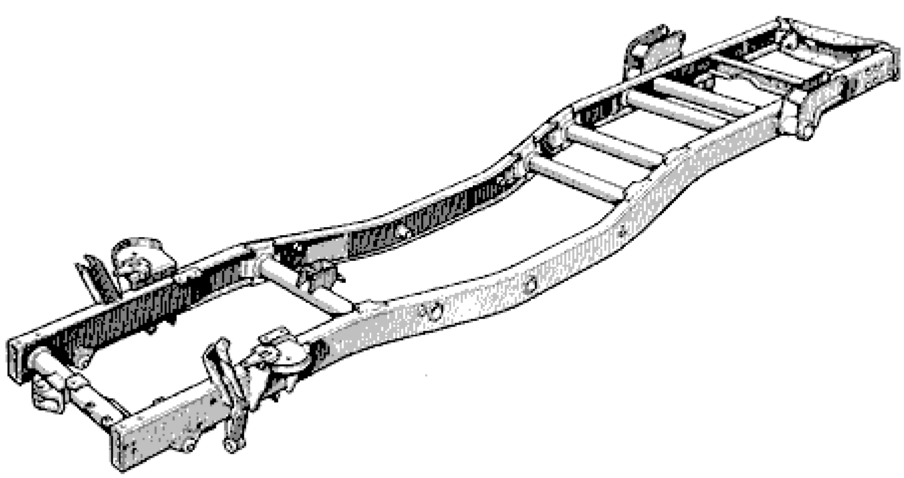

Figure 3: Ladder Chassis

Ladder chasse, are typically only stiff in one direction, not out of plane, since the frame is rectangular which provides less ideal load paths compared to the triangulation usually found in space frame chasse. The frame does not provide an adequate load path for suspension and other vertical loads. Ladder chasse are commonly used in drag racing vehicles, where bending stiffness is more desirable, and the lack of stiffness in torsion can be accommodated. The frame is stiffened through the addition of braces in the suspension points and other load paths; however, this will result in a heavier chassis for the same stiffness in comparison with other methods.

Additionally, the ladder frame does not provide sufficient driver protection. Roll hoops, side impact protection and frontal crash protection are all systems which will have to be added, which will provide little additional strength to the overall chassis while increasing the chassis mass.

Monocoque Chassis

Monocoque is a French term translate as “single shell” and is a chassis design commonly found in high performance formula vehicles, such as Formula 1, Indy, Champ Car, etc, and even in forms such as the steel and aluminum uni-body of today’s production vehicles. A monocoque chassis is constructed from a single skin, or shell, using the body to support all or most of the load. In formula style race vehicle, the material is typically carbon fiber or a similar aramid fabric, infused with resin with a core material for additional support and to prevent separation of the layers.

A monocoque can provide a very light and strong chassis, since the bulk of the chassis is used for stress distribution. Using a material, such as carbon fiber, with a high strength to weight ratio can provide chassis weights significantly less than that of other methods. With the addition of hard points to the monocoque shell other systems can be attached to the chassis. A monocoque chassis is fabricated by first creating a mold and plug system, then layering the chassis material until the desired thickness and strength is achieved.

Figure 4: ETS Monocoque Chassis

Monocoque chassis can be very expensive to fabricate, however, especially for limited production runs. The cost of creating the molds and time for the piece to cure is very high, making the monocoque undesirable for lower production runs which require a quick turnaround time. The material can also be very expensive, over $30 per yard. In addition, a monocoque chassis can make it difficult to access certain components of the vehicle, since the body panels cannot be removed. The Formula SAE rules also specify that the roll hoops and roll hoop supports still be manufactured using steel tubing.

Space Frame

Typically WPI Formula SAE vehicles have used a space frame chassis. Space frames utilize truss like structures and triangulation of chassis members to provide a strong frame. With proper material selection and geometries, space frames can be very light for their strengths, often rivaling monocoque chassis of the same stiffness. A space frame chassis derives its strength through the use of triangles, which are structures, rather than mechanisms. Structures will always be stronger than mechanisms, and this is the backbone behind the design of a space frame chassis.

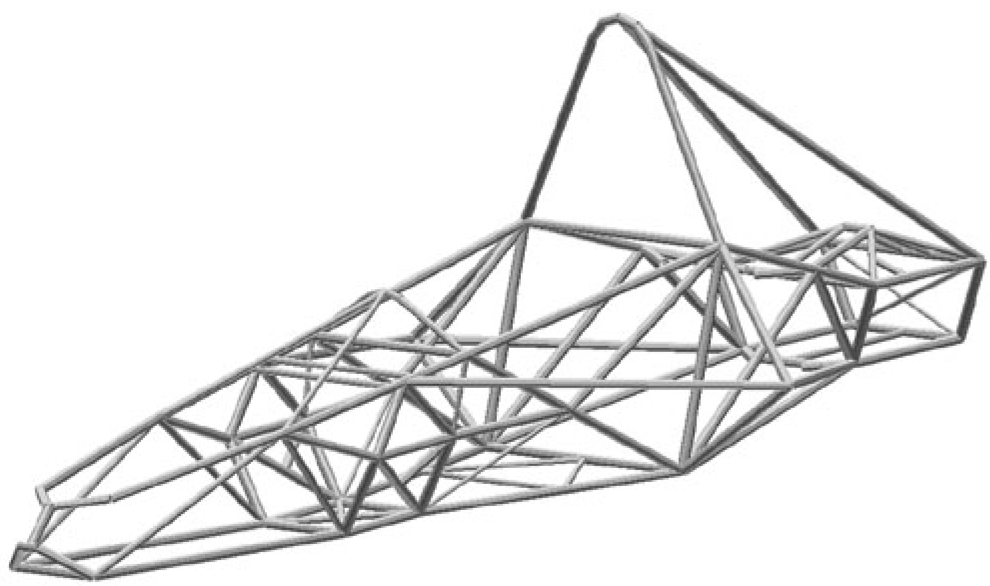

Figure 5: University of Missouri-Rolla's 1996 Formula SAE Space Frame Chassis

Space frame chassis are constructed of metal frame member, typically tubular, but can be square or other cross sections. These frame members are welded and/or bolted to form a single structure. Frame members should be placed to provide proper loading paths for systems such as the suspension and drive-train. These loads can be distributed back through the chassis and to a main support system. The mandated roll hoops and bracing serve as an integral part of the frame strength.

Selection of material is important as well. A high strength, but low weight material is desired. Common space frame materials include AISI 4130 Steel, 1020 steel and certain aluminum alloys. The method of joining and welding also plays a critical role. While some materials may joined using gas-metal arc welding (GMAW, MIG) it is often more common and desirable to use gas tungsten arc welding (GTAW) since it provides cleaner and stronger welds. AISI 4130 steel should always use GTAW procedures.

Space frames can require a significant amount of time to build, as they require accurate fixturing of the chassis members prior to welding. In addition, they require special tooling for bending frame members as well as a skilled welder and welding equipment. Once a fixture is designed, however, it may be reused for a large number of frames. If care is not taken an improperly designed space frame could be very heavy. Welding also introduces stresses at the welded joints. If AISI 4130 steel or aluminum is used, the frame must be heat treated and normalized to regain the lost strength and ensure the frame will not fail at these locations. Packaging of internal components is also a concern, and may dictate the location of certain frame members which may not provide the optimal path for design loads.

Design Methodology

To design the 2008 Formula SAE Chassis, design goals and specifications were selected. These were based on engineering practices as well as analysis of the 2004 and 2007 vehicle frames.

Design Specifications:

Comply with all 2008 Formula SAE Rules

- *Overall mass of the chassis less than 60 lbs (30% reduction from the 2007 Chassis)*
- *Minimum 10 times stiffer than roll stiffness in torsion*
- *Minimize deflection under bending*
- *Reduce the length of the suspension mounting members from the 2007 chassis – lengths 1.5 inches or less.*
- *Increase driver leg space*
- *Remove steering and suspension components from the driver cockpit*
- *Move driver position further aft*
- *Reduce "dead space" in the cockpit, engine and drive-train areas*
- *Provide a Frame design to be completed before the end of the 2007 calendar year*
- *Provide packaging and attachment for all systems (suspension, cooling, engine, drive-train, etc)*
- *Accommodate aerodynamic devices and provide a minimum of 15inches of unobstructed wing space (maximum of three inches from the front of the tires to the front of the frame)*

Chassis Type Selection

With these specifications in consideration, the chassis type was first selected. Weight, strength and cost and manufacturing time were of the main considerations when selecting the chassis type. A space frame chassis was selected because it offers the best compromise between all these factors. A strong and stiff frame can be manufactured within the given deadline and can still maintain our weight goal of under 60 lbs.

The ladder chassis was not selected due to its insufficient stiffness in torsion. While it can be fabricated quickly with a very low cost, it is not suitable for a race vehicle chassis and will prove to be heavy once built to meet the specifications and Formula SAE rules

Cost and time were the major factors preventing a monocoque chassis design. WPI facilities do not easily provide for fabrication of a monocoque chassis. To meet the required stiffness requirements, the potential reduction in weight was not deemed sufficient for the use of such a design. In addition, ease of maintenance and packaging were concerns with a monocoque chassis.

In addition, WPI facilities provide fabrication and welding equipment suited for space frame fabrication. Utilizing the facilities and past experience with space frame chassis will allow the team to quickly and successfully design and build a space frame chassis.

Chassis Material

Material can vary in space frame design and plays a large role in the design, strength and stiffness of the chassis. The 2008 Formula SAE rules mandate specific material requirements or equivalencies. Figure 6 describes the minimum wall thickness and diameter for each frame member if made from steel. There are no additional provisions made for alloy steel over mild steel (i.e. they must have the same minimum wall thickness.) The Main hoop and main hoop bracing must be made of steel.

ITEM or APPLICATION	OUTSIDE DIAMETER x WALL THICKNESS
Main & Front Hoops, Shoulder Harness Mounting Bar	1.0 inch (25.4 mm) x 0.095 inch (2.4 mm) or 25.0 mm x 2.50 mm metric
Side Impact Structure, Front Bulkhead, Roll Hoop Bracing, Driver's Restraint Harness Attachment (except as noted above)	1.0 inch (25.4 mm) x 0.065 inch (1.65 mm) or 25.0 mm x 1.75 mm metric or 25.4 mm x 1.60 mm metric
Front Bulkhead Support	1.0 inch (25.4 mm) x 0.049 inch (1.25 mm) or 25.0 mm x 1.5 mm metric or 26.0 mm x 1.2 mm metric

Figure 6: Material Requirements and Wall Thickness

Aluminum chassis members are allowed by the 2008 Formula SAE rules (see Figure 7 for minimum tube requirements.) If aluminum is used, the structural equivalency calculations must be done in the "as-welded" state, or the chassis must me solution heat treated and age hardened.

MATERIAL & APPLICATION	MINIMUM WALL THICKNESS
Aluminum Tubing	3.0 mm (0.118 inch)

Figure 7: Aluminum Tube Requirements

The strength of aluminum and steel is also a consideration. The tensile strength of 6061 aluminum is 25% that of 1020 steel and 20% of 4130 steel with yield strengths of 16% and 13% respectively, while only weight 35% less than steel. Given these properties of aluminum, it is apparent that an aluminum chassis of the same strength would have a higher mass. Welding aluminum is also very difficult and requires experience and time. Billet or cast subsections or sub-frames however are good candidates for aluminum pieces.

Typical steel alloys used in space frame construction are AISI 4130 steel and AISI 1020 steel. The materials have the same density; however, 4130 is stronger per unit weight. The yield strength of 4130 is 1.4 times greater than 1020 steel and the tensile strength is 1.22 times greater. Fabricating the chassis from 4130 steel will provide a stronger and stiffer chassis for the same weight. AISI 4130 steel require GTAW and subsequent heat treating though, as welding reduces the strength of the joints without it. The cost of AISI 4130 is also higher, almost as much as twice as expensive.

The material selected for the 2008 WPI Formula SAE chassis is AISI 4130 steel using a billet aluminum rear sub frame. This material was selected due to the high strength per unit weight and the ability to weld and manufacture the chassis in house. The chassis will be stress relieved locally by a local sponsor, and the material is available for a discounted rate from additional sponsors.

Fabrication

The fabrication method for the chassis will be GTAW using an inert 4130 filler material. In order to join each chassis member, the ends of the joining member(s) have to be notched and shaped to match the surface to which it is joining. This is required to ensure chassis strength and integrity. Following fabrication, the chassis will be fully stress relived using current heat treating procedures.

Chassis Design

To begin the layout of the vehicle, and thus the chassis, first the wheelbase and track width were selected with attention paid to weight transfer and handling characteristics. After these dimensions were determined the engine placement was determined, and subsequently the driver position. These set the working envelope and general size of the frame.

In the effort to reduce weight and provide a basis for the 2008 chassis previous chassis designs were analyzed to determine a general chassis shape, driver ergonomics and placement.

To determine the major paths of frame members the suspension geometry had to be designed. After calculation of instant centers, roll centers, caster, camber gain, etc the suspension geometry was designed and steering and suspension points calculated and modeled. It is from these points the frame points were based.

Triangulation

With the major points defined the method is a complex series of "connect the dots." In a space frame, the strength of the chassis is derived from proper load paths and reduction of the number of chassis members loaded in bending. This is achieved through proper triangulation of the chassis members. Triangulations is especially critical in the suspension and crash protection areas where the loads should be dissipated through a larger area of the chassis. In the suspension areas, the goal is to reduce the deflection of the suspension connection points, as this deflection could cause inconsistent handling characteristics. Engine mounting and other areas of high loads also benefit from the added strength of proper triangulation.

The suspension loads where connected both fore to aft, as well as left to right to ensure little relative movement. The entire front section of the chassis utilizes a "spider web" system to triangulate all loads from the front bulkhead to the front roll hoop. These loads are then transferred through the triangulated side impact structure and other chassis members to the main hoop and rear frame section.

Engine as Semi-Stressed member

The engine is a significant portion of the overall weight of the vehicle which does not easily offer areas of weight reduction. In past Formula SAE chasse WPI has built the chassis around the engine, and mounted the engine within the chassis envelope. This results in a not only the weight of the engine being added to the chassis, but also the weight of the fame members to hold the engine and connect the rear frame section and suspension points to the rest of the chassis.

Rather than isolate and package the engine, the 2008 WPI chassis design utilizes the engine as a load bearing structure. The front section of the chassis is connected to mounting location of the engine which then supports the rear section of the chassis. Using the engine to support these loads eliminates the need for additional chassis members and reduces the overall weight of the chassis and vehicle. The Formula SAE rules prohibit use of the engine as sole attachment point for the main roll hoop bracing however. This required the addition of frame members to provide triangulation back to the primary chassis structure.

The engine the 2008 WPI vehicle is using is from a Honda CBR 600 F4i motorcycle. In the bike, this engine is also used as an integral part of the chassis. Studying the mounting location and load paths of the F4i motorcycle provided many design ideas for use in the Formula SAE chassis. The front section of the chassis connects to the major mounting locations of the engine, providing adequate support for the engine loads as well as roll hoop bracing. The rear section shares one of the major mounting locations with the front section and is also connected to the swing arm mounts of the engine. The suspension loads are transferred from rear sub frame directly to the main chassis to eliminate additional loading to the engine. This provides a load path very similar to that found on the Honda CBR 600 F4i, from which the engine has been sourced, which can be seen in Figure 8 below

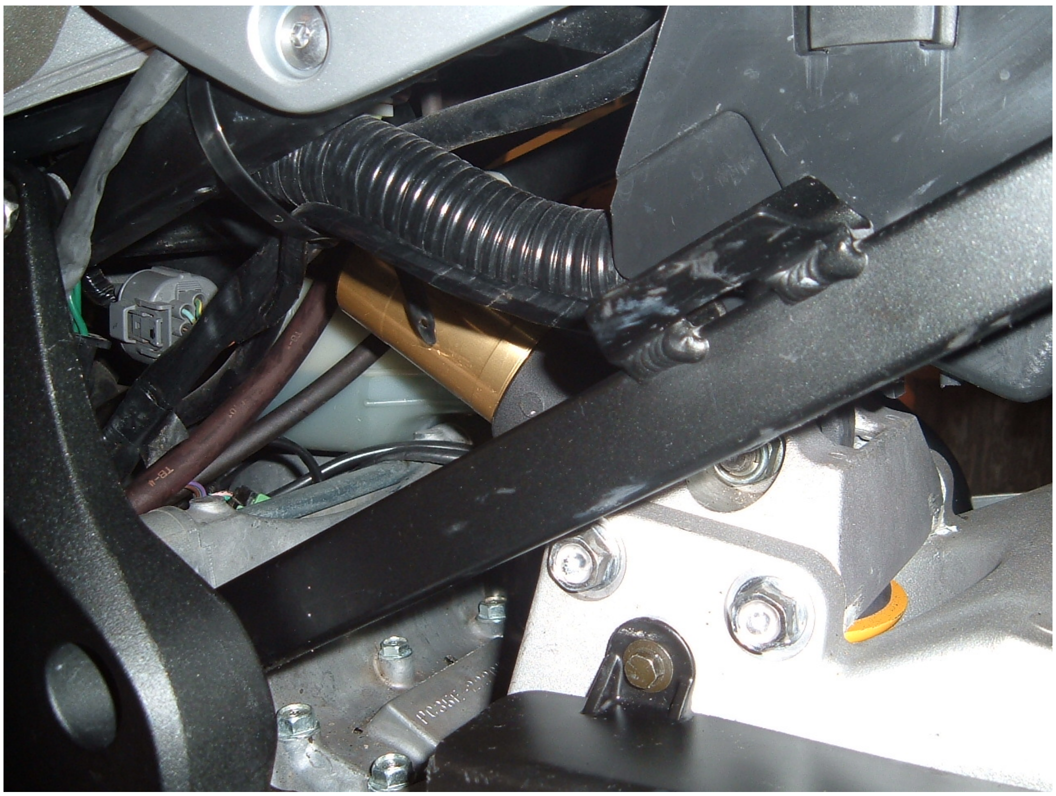

Figure 8: F4i Suspension and Swing Arm Mounting

Sub-Frame

The use of the engine as a semi-stressed member necessitated the design and utilization of a rear sub frame. The rear sub frame will provide attachment locations for the drive train system and rear suspension points and mono-shock system. The sub frame must provide sufficient support for the suspension system without allowing significant deflection.

In the main chassis section aluminum alloys were not used due to the relative strength and stipulations of the 2008 Formula SAE rules. The sub frame, however, does not have to meet minimum wall thicknesses as the main chassis structure does. The relative size also allows a billet or cast section to be used, instead of welded tubular sections. In small properly supported sections aluminum can provide a very light yet strong and stiff structure. For the rear sub frame 6060-T6 Aluminum was selected because of its low density but higher yield and tensile strengths. Aluminum main structures will be used which will be connected using small ASIS 4130 Steel braces.

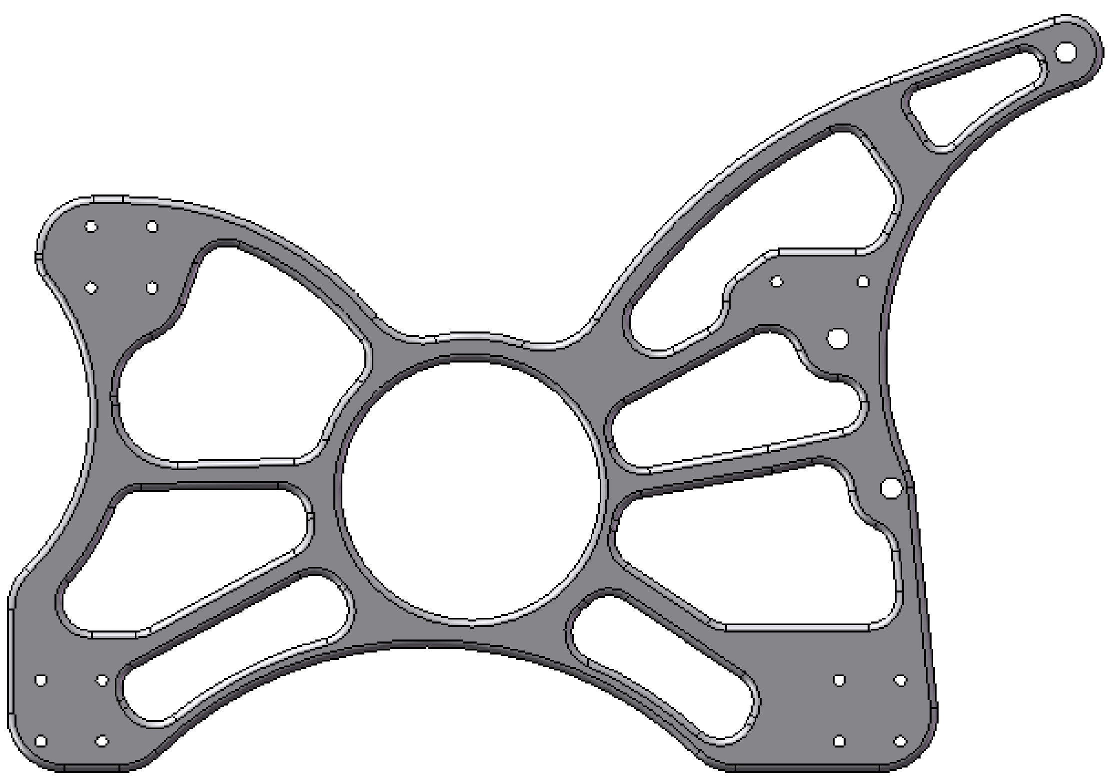

Figure 9: Rear Sub-Frame Aluminum Section

To design the rear sub frame the suspension and mounting points were first modeled, and clearance for push rods and drive shaft created. Using proper triangulation and careful load path planning using free body diagrams, material was removed leaving a thin webbed structure to provide support for the suspension points. These points are brace from left to right using 4130 steel supports. It is to these supports that the mono-shock and drive train systems are connected. The full sub frame assembly can be seen below in Figure 10.

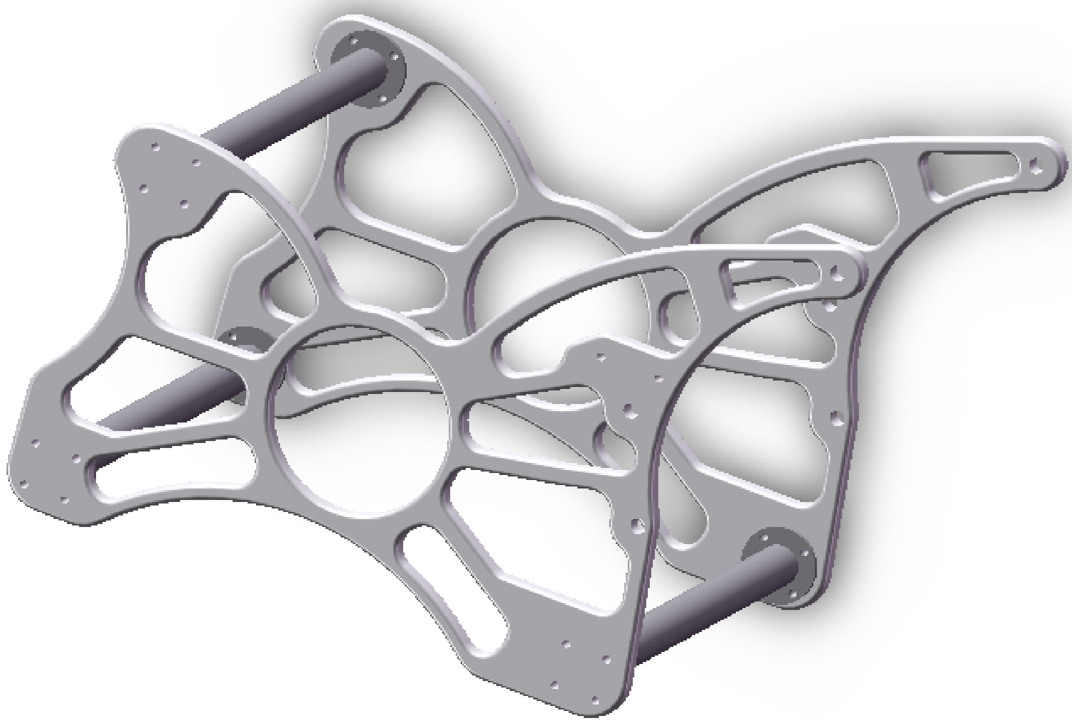

Figure 10: Rear Sub-Frame Assembly

Analysis of Frame

Modeling Techniques

The following section defines the modeling and meshing techniques used for analysis of the chassis and the rear sub-frame. The software used for all modeling was Solid Works 2007. The accompanying Finite Element Software, Cosmos Works was used for all finite element analysis.

Primary Chassis Structure Analysis

Once the preliminary design had been created and the suspension geometry calculated it was possible to sketch the space frame a type of modeling software. The software used for this modeling was Solid Works. Utilizing the 3D sketching feature it was possible to first assign the suspension points as nodes in 3D space. Then, using our anthropomorphic data collected the dimensional restrictions from the Formula SAE rule book and our data collected from research, the chassis was constructed around these points. Copies of the rules regarding the frame characteristics from the Formula SAE rule book can be seen in Appendix A

After the wire-frame model was created, it was necessary to determine how to create a solid model of the sketch in the software. The first idea was to create a series of sweeps along the path of each sketch. This would allow entire frame to be created as one solid body. The reason this method of modeling was selected was that Solid Works/Cosmos Works, which is the Finite Element Analysis (FEA) software for Solid Works, could not mesh a part unless it was a solid model. However, it was not taken into consideration that it would take an extended period of time to run a solid model as large as the space frame in an FEA package. It was then decided that the most suitable analysis for a space frame is to evaluate each member as a beam, not a solid element. This would permit a much faster calculation and allow a large number of iterations to be made in a short amount of time. There are various FEA packages that allow this method of analysis. These programs include Ansys, Pro/Mechanica, Cosmos Works and Abaqus. To avoid exporting the our model into an alternate program Cosmos Works was used. The new version of Cosmos Works 2007 offers a beam analysis option which would allow frame to be analyzed accurately and efficiently.

To prepare the wire-frame for analysis weldments were created for each member. The weldments were created in the form of a structural member. The structural member could be defined by a tube with a certain outer diameter and a certain wall thickness. This allowed a structural member library of the different size tubes to be created The individual lines connecting one point to another were assigned a specific structural member depending on the size of the tube to be used for that chassis member. After each line was defined it was then possible to trim intersecting members to one another to mimic the act of notching the pipe. However, there is one error discovered while using this process. During initial testing in the FEA software the analysis was resulting in the error: "Improper 3rd node at beam element #". This error always occurred when 2 or more beams intersected at the same point or as ComosWorks refers to it, a joint. To remedy this the end point of the sketch defining of one of the members was moved 0.25 inches off of the original point. In turn, this pulled the joint created in the FEA package off of the joint that already existed. The pipe was then extended using the trim/extend feature to properly fit it to the chassis structure. This process eliminated the error and allowed the analysis to continue.

Rear Sub Frame Analysis

The analysis of the rear sub frame was also done using Cosmos Works, but as a solid mesh rather than a beam element analysis. The sub-frame assembly was created, but with the addition of 3 bars at the location to which the sub frame connects to the engine. These braces simulate the load path to the engine and a more realistic analysis of the sub frame.

To create the assembly for the analysis the cross braces and engine braces were first mated to the contact surface on one sub frame and the axes of the corresponding bolt holes aligned. The same steps were then repeated for the other billet sub frame section. Each section was the assigned material properties corresponding to their selected material. The cross braces were defined as AISI 4130 steel, while the aluminum sub frame sections and engine braces were defined as 6061 T6 aluminum.

The mesh for the rear sub frame was created from this model using the meshing procedure included with Cosmos Works. The mesh element size was set to 0.05 inches (the smallest element for which this mesh could run) to achieve the best possible results. The loads and restraints could then be added and the analysis completed.

Finite Element Analysis

Following the preparation detailed above FEA software was run and analysis begun. First beam element mesh was created in a new study and all structural members were treated as beams. However, we it was noted that Cosmos Works did not define all the curved members as beams. These sections had to be manually added for each curved beam. The joints for each intersection of beams in the structure were then calculated in Cosmos Works. This defines the intersection points and load paths for the subsequent analysis. After defining the material as

AISI 4130 steel from the material library it was possible to create a mesh of the beams. The initial meshing, however, revealed a very serious problem. The beam mesh would only calculate linear beams. If the beam was curved it would analyze it as a straight beam from beginning point to end point. It would not follow the path of the sketch. After calculations and additional analysis it was concluded that the error would be acceptable for all of the curved members except for the side rail braces. The reason being was that the other curved beams were short enough where a straight line made a fairly accurate representation. However, in the side rail support that has a much longer curve needed to keep its geometry to ensure an accurate analysis. It was concluded that the side supports be replaced with 2 straight beams which would be attached at the same point as the top of the curve of the beam. The two supports are indicated by the arrows below in Figure 11.

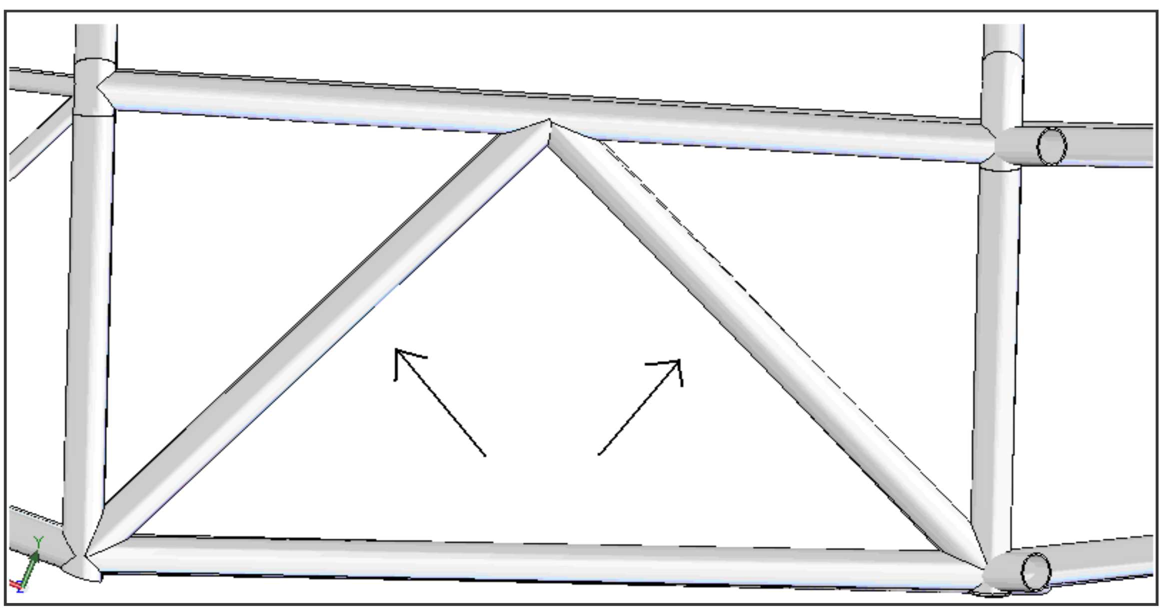

Figure 11: Image of supports altered for analysis

The final fabricated frame will still be using the curved support, however for analyzing purposes the linear beams were used. This will be the same for the two smaller beams attaching the main and front hoop on top of the side rail support. However, even with this change the analysis would not properly complete. It was concluded that if the frame had a significant factor of safety in the bending analysis without those pipes the analysis would be acceptable. However, if there was significant deflection or stress because those beams were omitted then further research must be done to determine the reason for the failure. The meshing was then completed and the analysis could be run. An image of the mesh created can be seen in Figure 12.

<u>**Mesh**</u>

Study name: Bending
Mesh type: Beam mesh
Total nodes: 1264
Total elements: 1203
Time to complete mesh (hh:mm:ss): 00:00:12
Computer name: DSPC3

<u>**Material**</u>

AISI 4130 Steel normalized at 870° C
Yield Strength – 66,717 psi
Tensile Strength – 106,022 psi

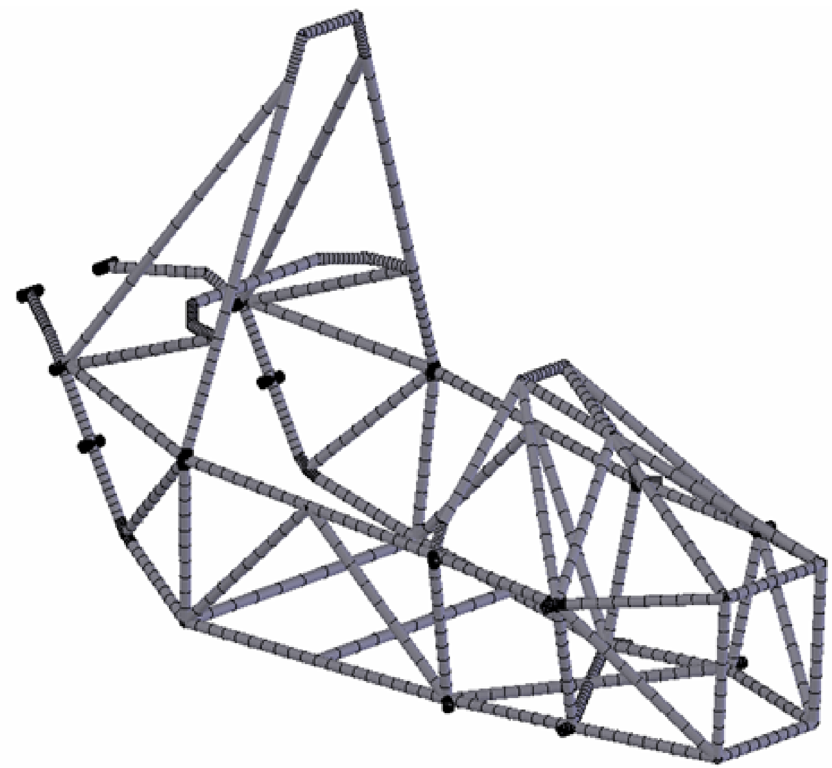

Figure 12: Beam Mesh

Results

A note regarding the following results: the stresses shown in all plots and given in this report are worst case stress, calculated by Solid Works. This stress is the highest value of the 8 stresses calculated for each beam element, and is the recommended stress plot for beam element analysis, as von misses and principle stresses are not shown since it is not a solid body.

Bending Analysis

The first analysis that was preformed was a bending study to ensure that the frame would no deflect between the suspension points. It was decided to consider the engine as a rigid structure. This enabled the chassis to be constrained at the points where the engine attaches to the frame and the front suspension pickup points. Two forces of 325 lbs to simulate a 600 lb total weight including the driver were then applied to the frame where the weight of the driver's weight would be concentrated. The forces and constraints can be seen below in Figure 13.

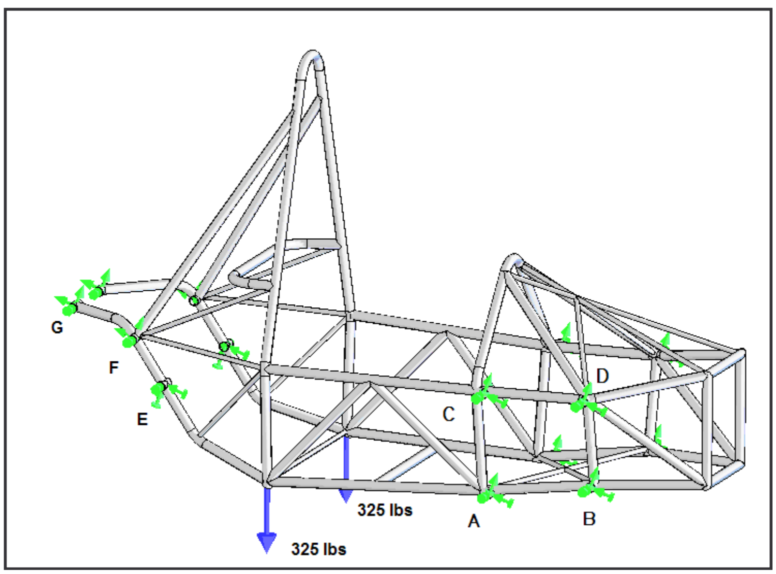

Figure 13: Constraints for Bending Analysis

The loads are applied at the two arrows, labeled with the 325 lb force in the -Z direction. The constraints were applied at points A, B, C, D, E, F and G. The constraint of immovable was defined for these points. This constraint limits translations and rotation in X, Y and Z directions. This simulates the vehicle sitting on the ground with the entire bending load between the suspension points.

To determine the displacement and stress of the chassis plots of total displacement worst case stress were created. The two can be seen below in Figure 14 and Figure 15, respectively. These plots are created within Solid Works/Cosmos Works using the analysis and defined material properties.

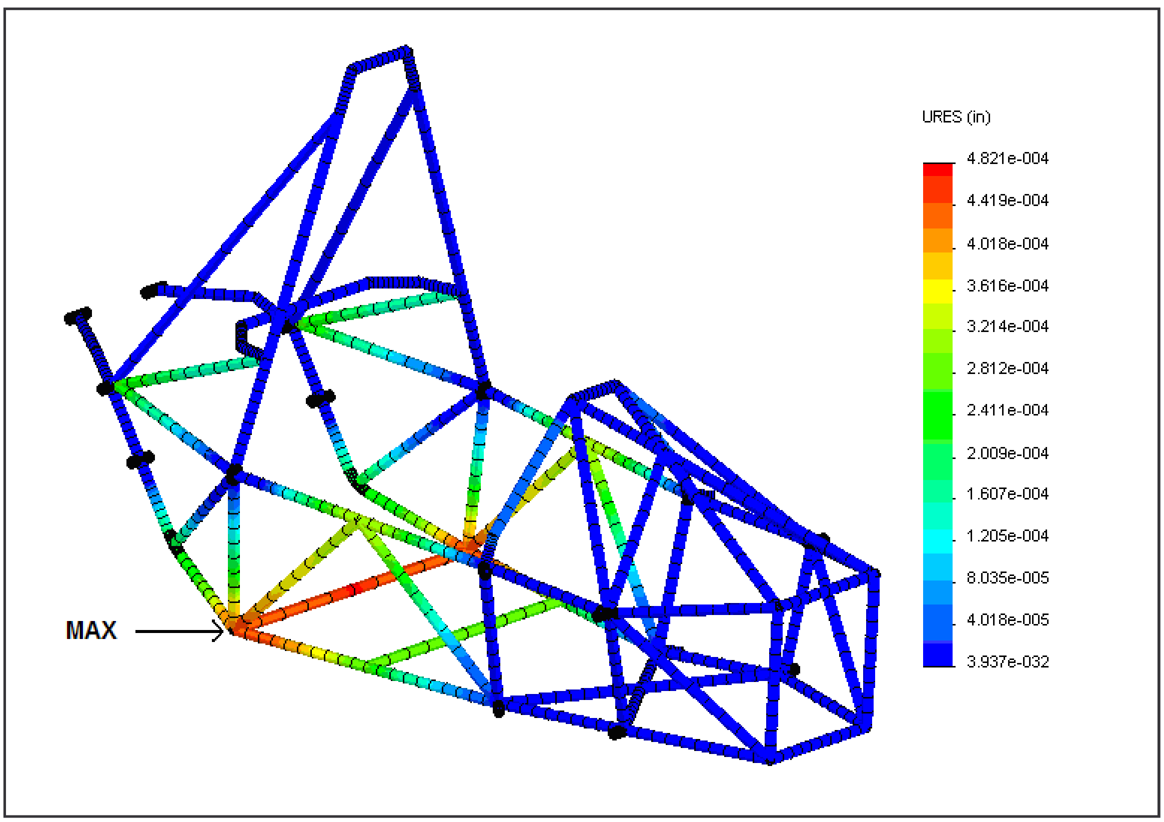

Figure 14: Displacement plot for Bending Analysis

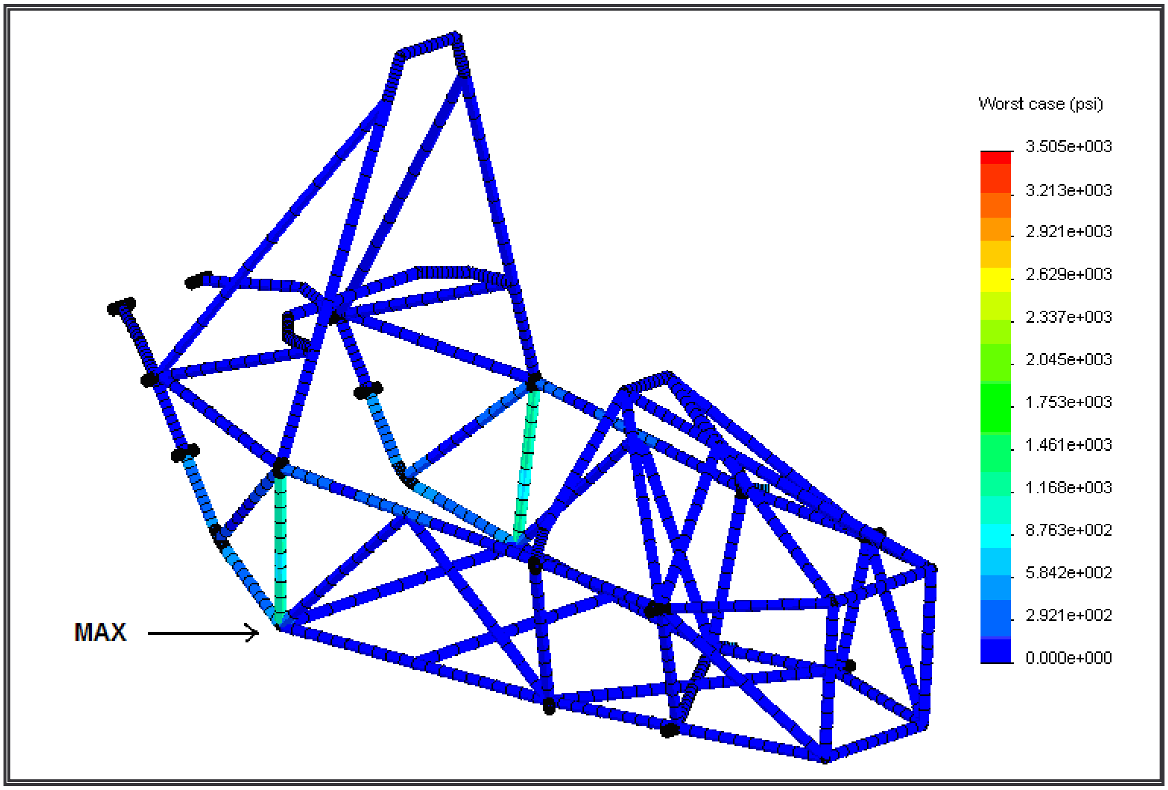

Figure 15: Maximum Stress Plot for Bending Analysis

The maximum displacement that occurs in the chassis when loaded in bending is .0004821 inches and the maximum worst case stress that occurs is 3,505 psi. The displacement if sufficiently small as to not create any significant changed to the dynamic behavior of the vehicle and suspension. Using the equation for a spring constant, k = F/x and the spring constant of the frame in bending is 1.498 x 10^7 lbs/ft. The stress is well below the yield strength of the material without the bars located on the top of the side rails that were neglected in the analysis due to modeling problems. It was therefore determine that no further testing is necessary for this case. However, the two supports will be added into the final model and fabricated chassis.

Torsional Analysis

The next analysis preformed was to ensure that the frame did not have significant rotational deflection when loaded in a cornering situation. To do this the rear of the frame was constrained at the engine mounts the same as in the bending study at locations A, B C, D, E, F, and G. However, for the front we constrained only four pick up points located on one side of the frame. The four pick up points on the opposing side were each subjected to a force of 150 lbs at each point. This loading simulates a 600lb load, distributed among all four points. This would be a worst case situation if the entire weight of the car with fluids and driver was located on the four points. An image of the constraints and loads can be seen below in Figure 16.

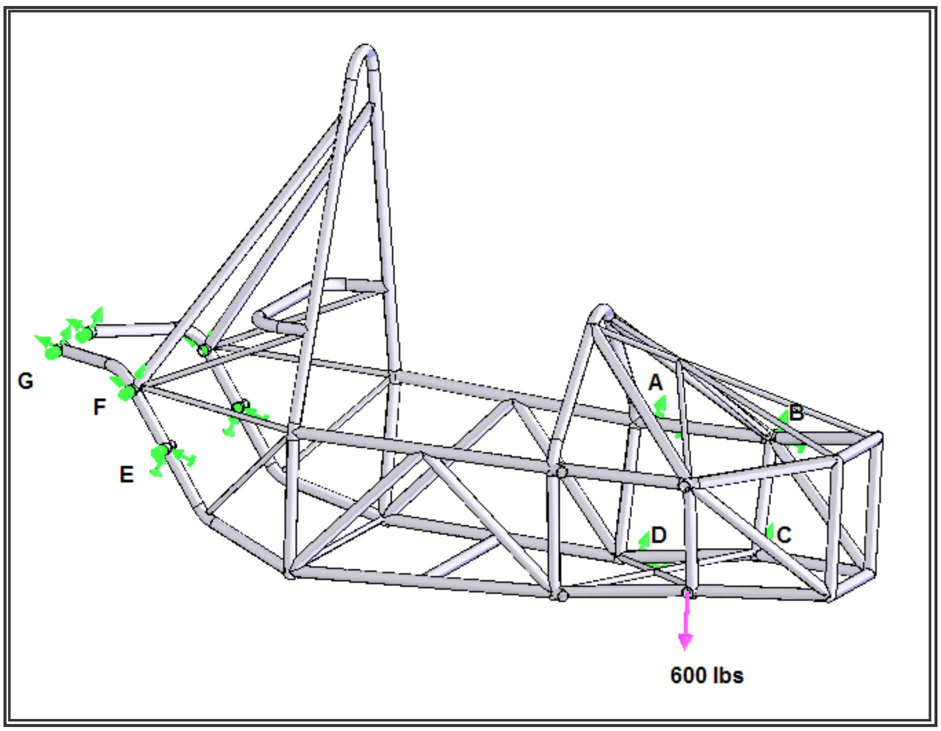

Figure 16: Constraints for Torsional Bending Analysis

The rotational displacement and the worst case stress seen by the chassis. The plots of the two can be seen below in Figure 17 and Figure 18, respectively.

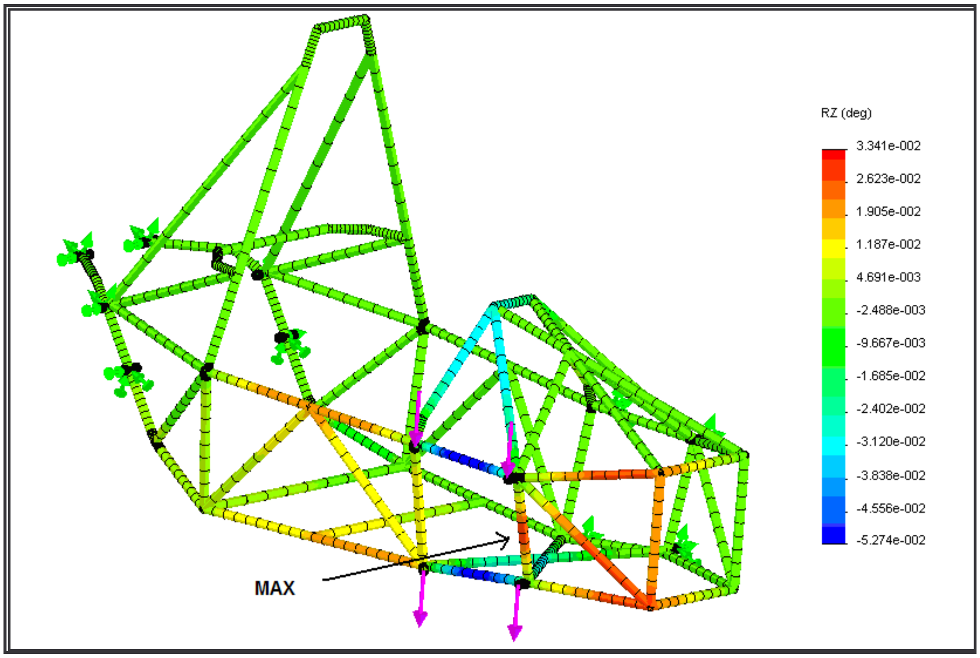

Figure 17: Displacement Plot for Torsional Bending Analysis

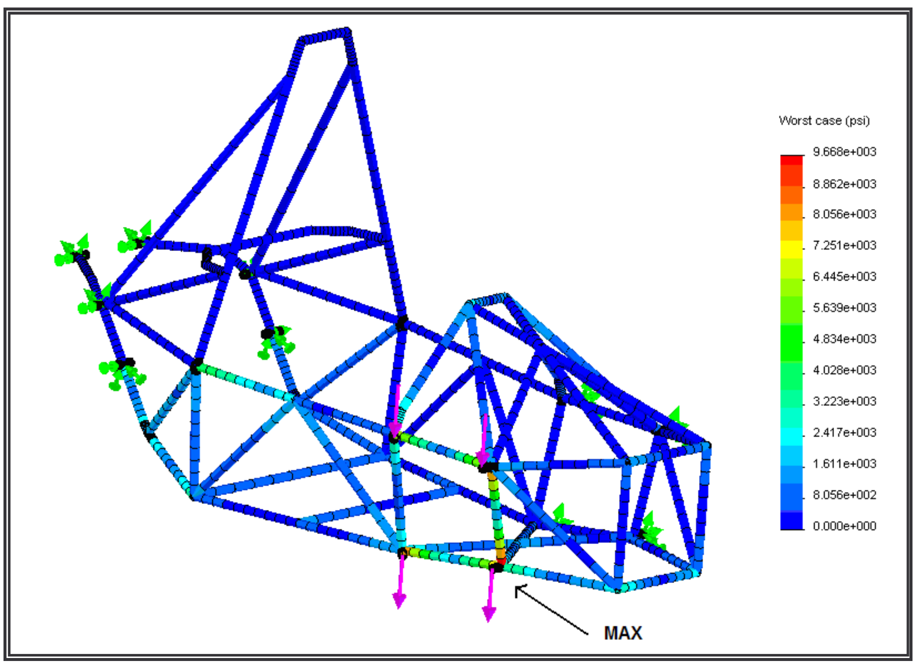

Figure 18: Maximum Stress Plot for Torsional Bending Analysis

The overall displacement recorded was .085 degrees and the Worst Case Stress was 9,668 psi. With a maximum stress value of 9,668 psi we have a factor of safety of 6.9 given the yield strength of AISI 4130 steel of 66700 psi . This is well within our range. Calculating the k value of the chassis in torsion using the equation k = F/x, where F is the applied force and x is the displacement in degrees, a stiffness value of 7058 lbs/degree. This value, however, is not accurately representative of the k value of the entire chassis, as this does not take into account deflection of the sub frame, and assumes a perfectly rigid rear section.

Rollover Analysis

Force Parallel With Main Roll Hoop

Our next test was to simulate the car rolling over directly on the top of the main roll hoop. We constrained the frame exactly as we did for the bending analysis, defining points A, B, C, D and the corresponding points on the other side of the chassis as immovable. A 600 lbs load was then applied directly on top of the main roll hoop to simulate the weight of the car on it. This load is shown in Figure 19 as the two pink arrows, each labeled with 300 lbs, simulating the 600 lb load.

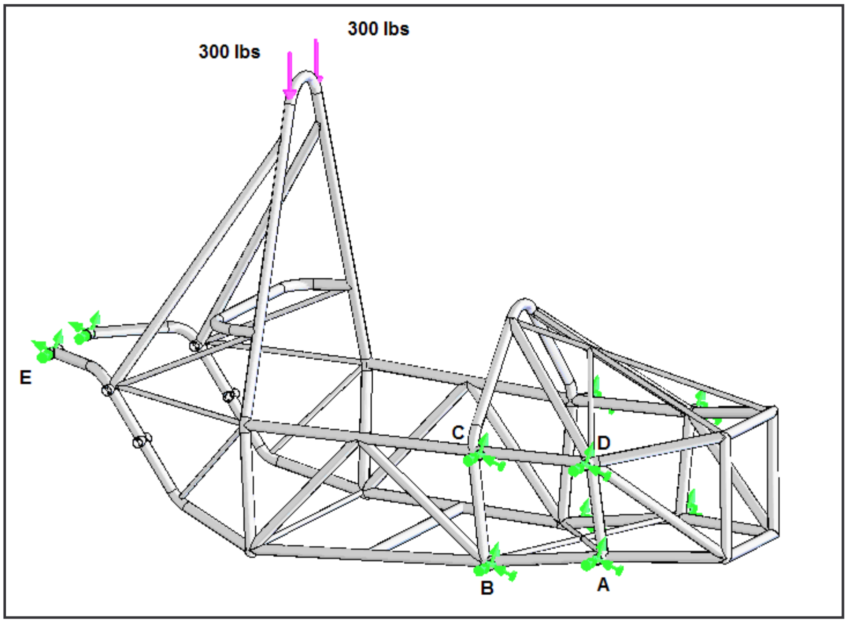

Figure 19: Constraints and Loads for Roll over Analysis

We then plotted the maximum displacement and worst case stress which can be seen below in Figure 20 and Figure 21, respectively.

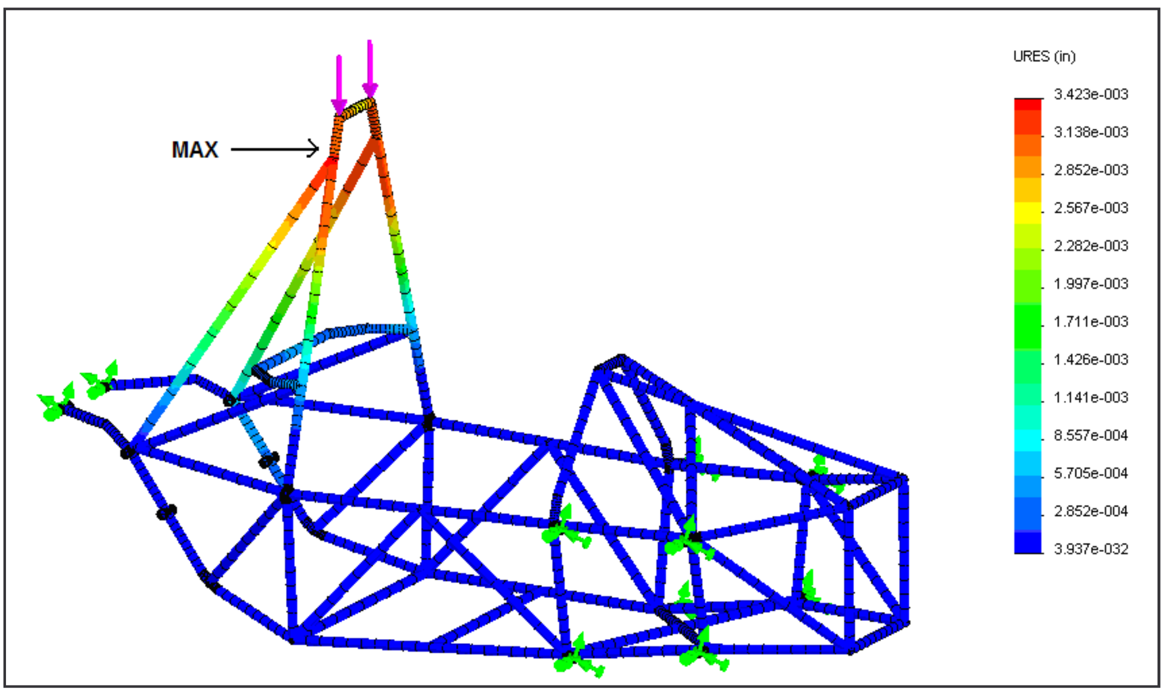

Figure 20: Displacement Plot for Rollover Analysis

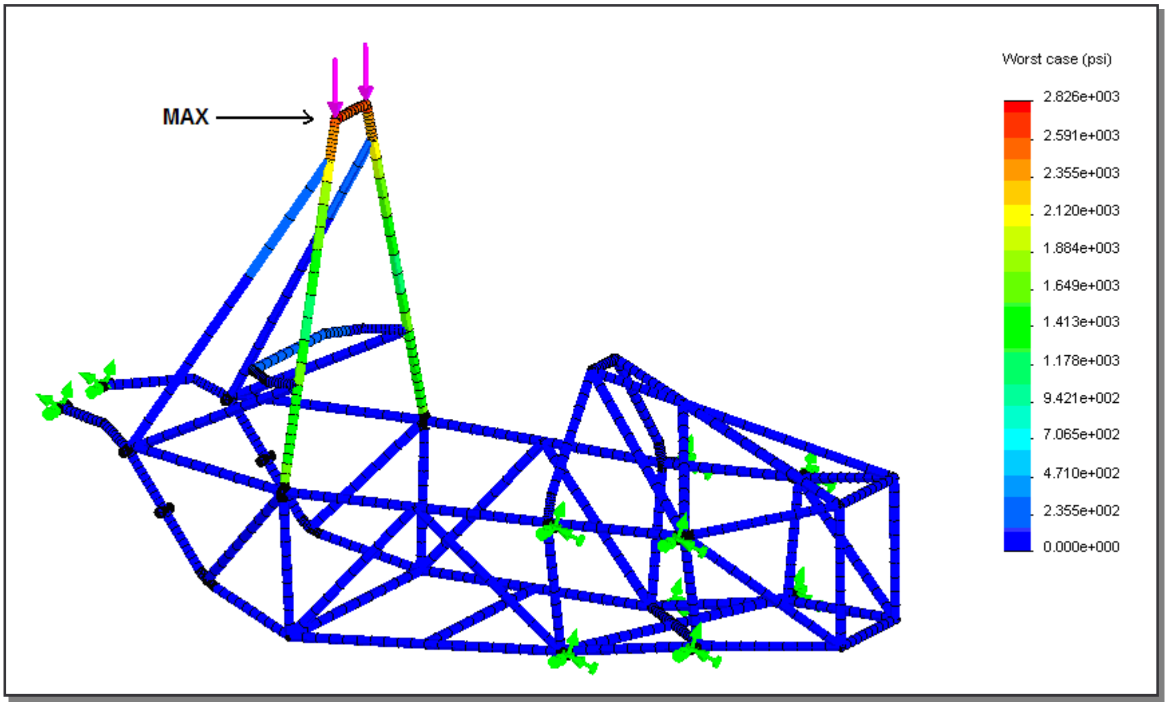

Figure 21: Maximum Stress Plot for Rollover Analysis

The plot shows minimal displacement and a very small maximum stress of only 2,826 psi. This situation is very unrealistic but if it did happen the frame would be able to withstand the forces and not fold on itself.

Force Normal to Main Roll Hoop

The final test was to see if the frame could withstand the force of 600 lbs applied normal to the Main Roll Hoop. This again is a very unrealistic situation. However, to determine maximum driver safety, it was deemed necessary to see how the frame would distribute the forces throughout its members. The chassis was constrained the same as in the previous rollover analysis above. An image portraying how the frame is loaded and constrained can be seen below in Figure 22. You can see points A, B,C, D and E and the corresponding symmetric points, as well as the 600lb load, split between the two sides of the main roll hoop.

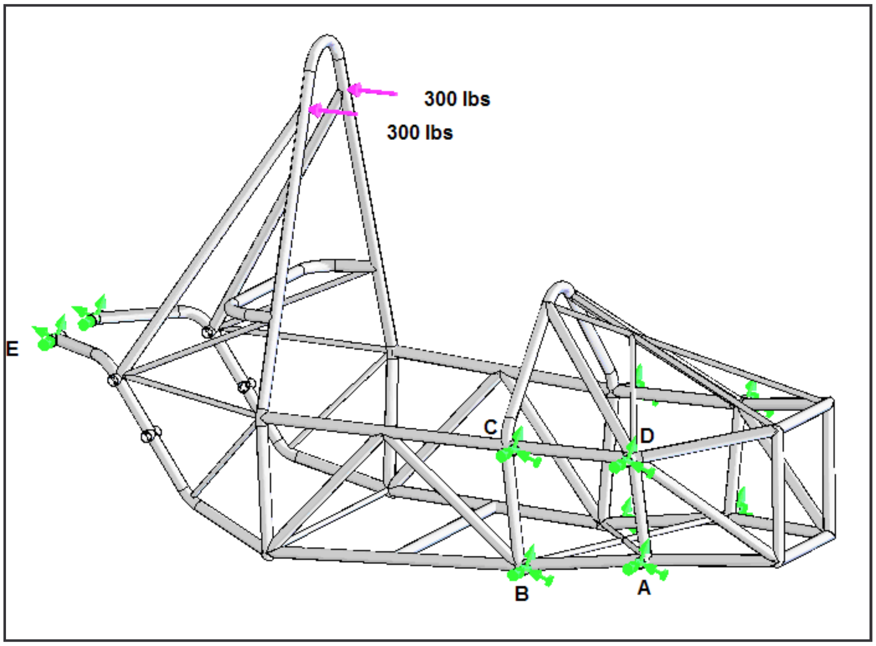

Figure 22: Constraints for Rollover Analysis

The total Displacement and the Worst Case Stress were then plotted which can be seen below in Figure 23 and Figure 24, respectively.

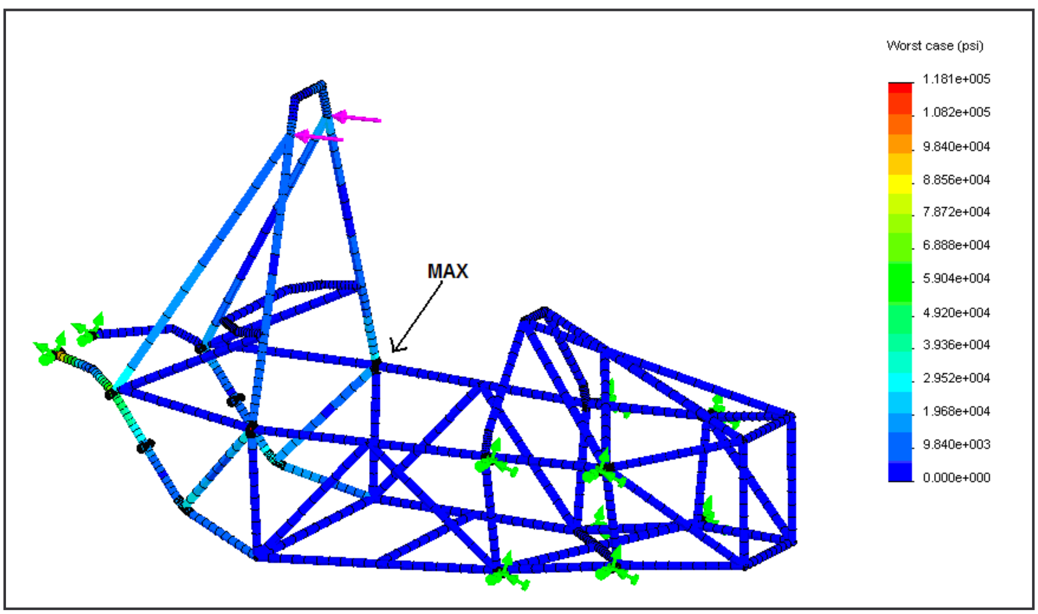

Figure 23: Displacement Plot for Rollover Analysis

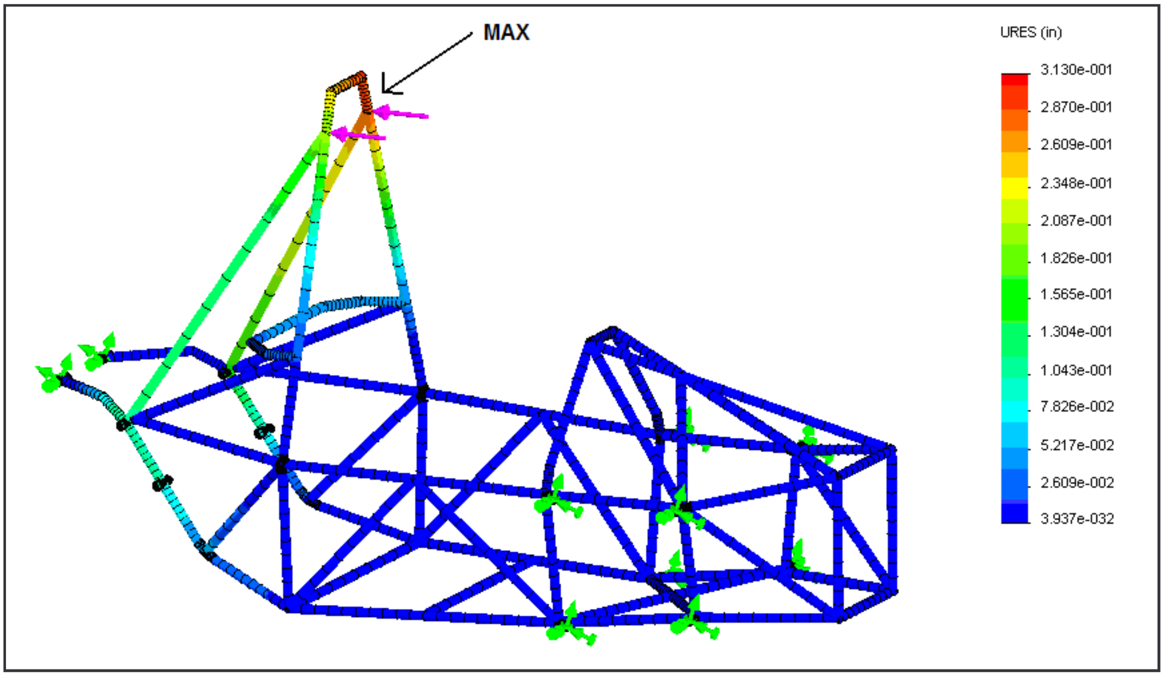

Figure 24: Maximum Stress Plot for Rollover Analysis

We found that there was a large significant displacement of .313 inches at the location shown in Figure 24. This means that the frame would flex and bend if this occurred. It may make the frame unusable after the crash; however it will be suitable to protect the driver.

Sub Frame Analysis

The analysis of the Sub Frame was done using Cosmos works. A full assembly of the sub frame was create, and the appropriate materials assigned. For the Sub frame aluminum sections, 6061-t6 aluminum is assigned, and the cross braces are AISI 4130 steel (Annealed.) In addition to the regular assembly, three additional bars were added where the sub frame attaches to the engine. These bars simulate the bracing added from the engine.

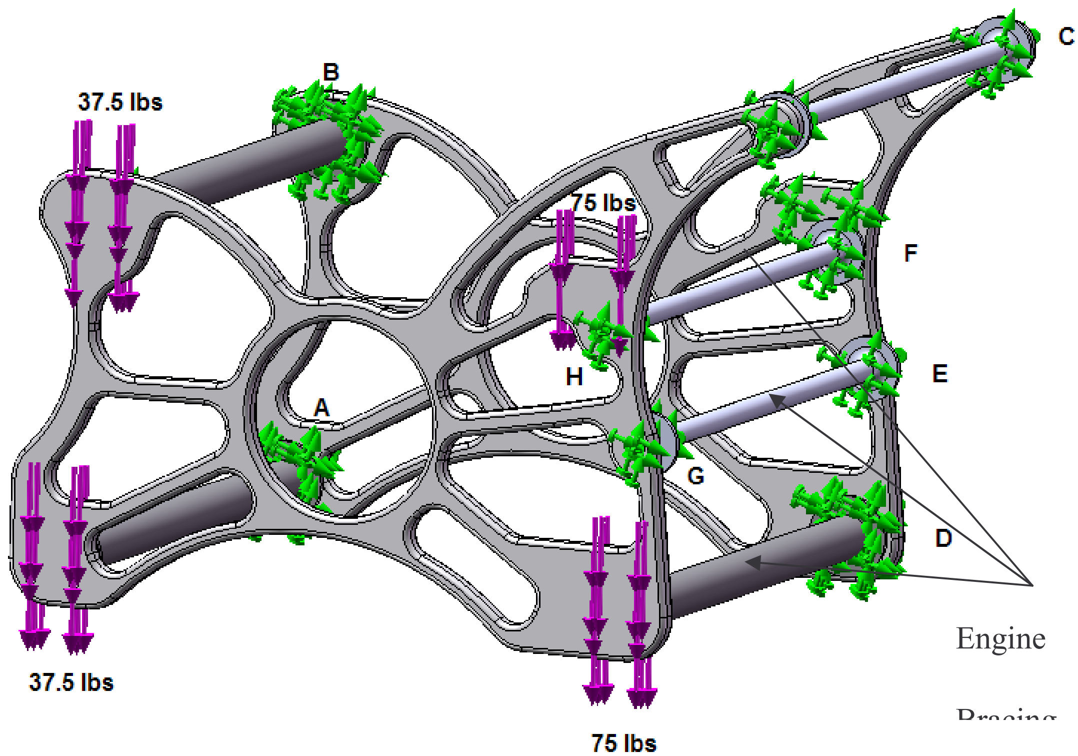

Figure 25: Sub frame Constraints

The sub frame was constrained at the engine mounting locations, C, E and F and the corresponding points on the opposite side. It was also constrained as suspension points on one side; point A, B, D and F. In addition, the faces of the engine bracing bars were constrained, as the engine is treated as an immovable object as it will deflect very little with the applied loads. At the suspension mounting locations on the other side a total load of 600 lbs was applied, divided among the bolt holes for the suspension mounting points (150 lbs at each suspension point.) These loads can be seen at the purple arrows. The stress and distribution plots can be seen below, in Figure 26 and Figure 27, respectively.

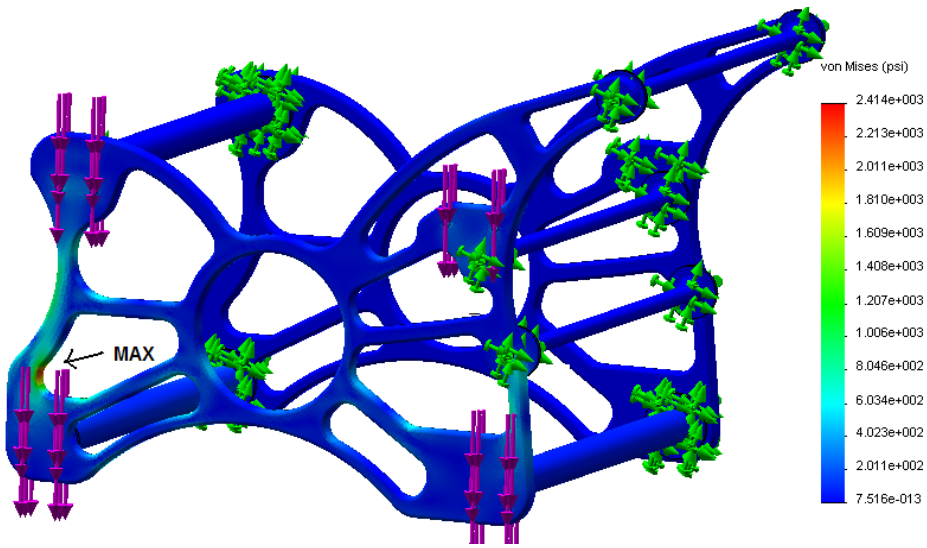

Figure 26: Sub Frame Stress Plot

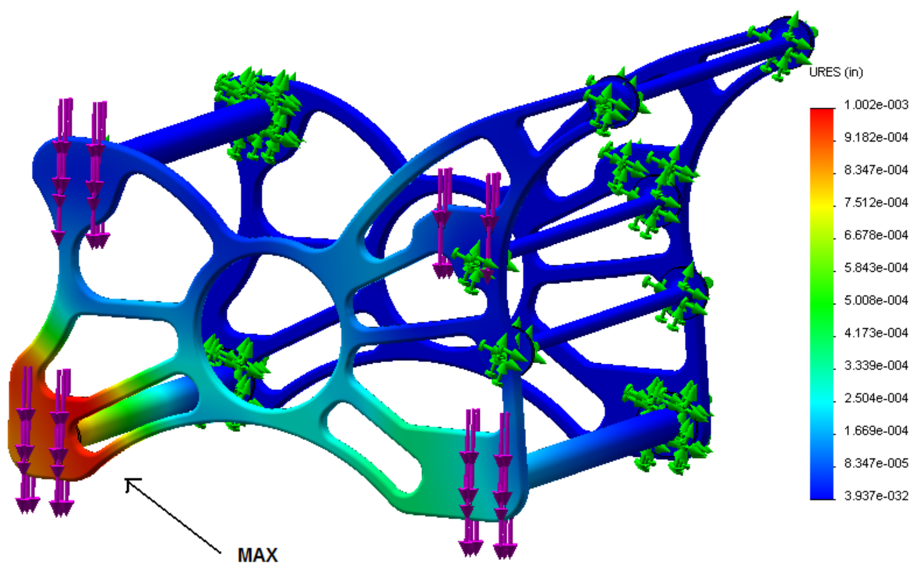

Figure 27: Sub Frame Displacement Plot

The maximum von misses stress is 2414 psi and the yield strength of 6061 T6 aluminum is 40,000 psi. This results in a factor of safety of 16.57. The total displacement of the sub frame is 0.0012 inches and is located at the rear suspension point. Cosmos Works does not allow the angular displacement to be calculated, however, so a proper correlation to the front section of the chassis is not possible using cosmos works. The total weight of the final sub frame is 8.5 lbs.

Final Design

The final design of the chassis with the engine can be seen below in Figure 28, Figure 29 and Figure 30, respectively.

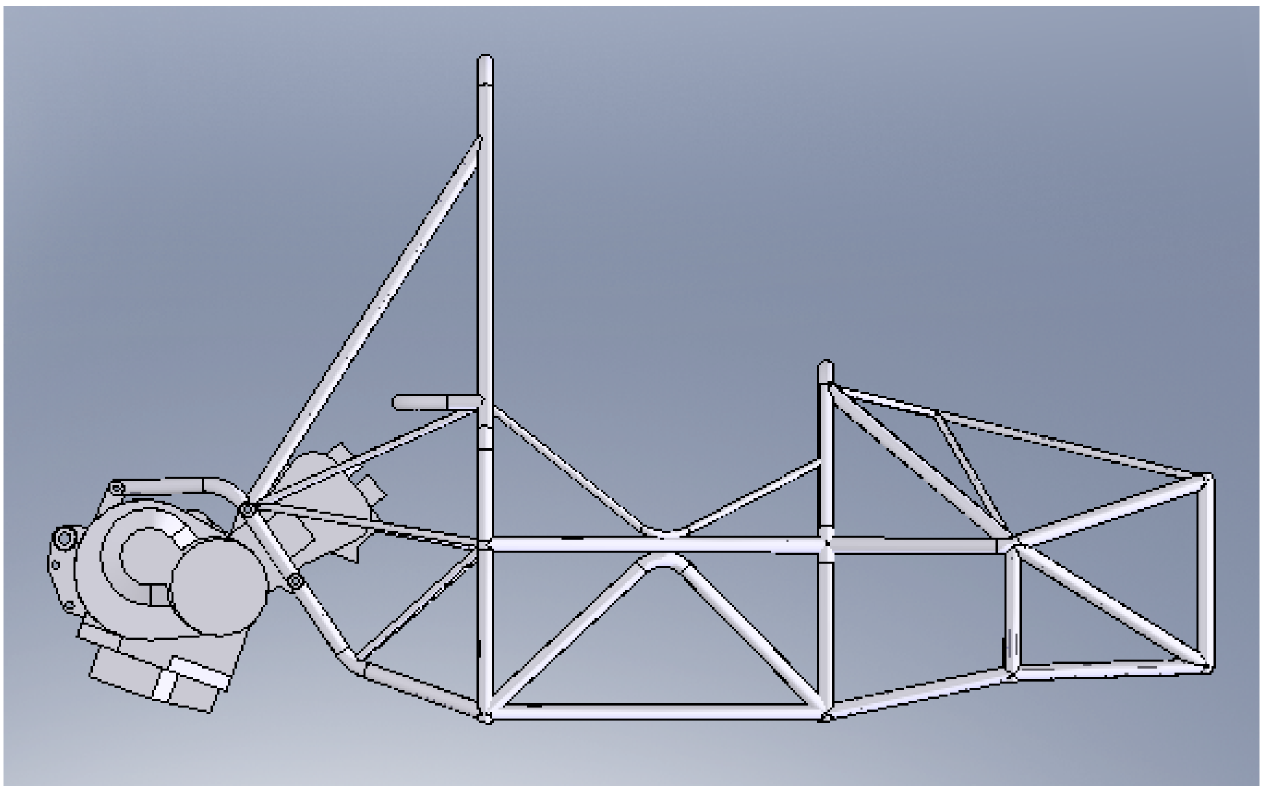

Figure 28: Right Side View of Final Design

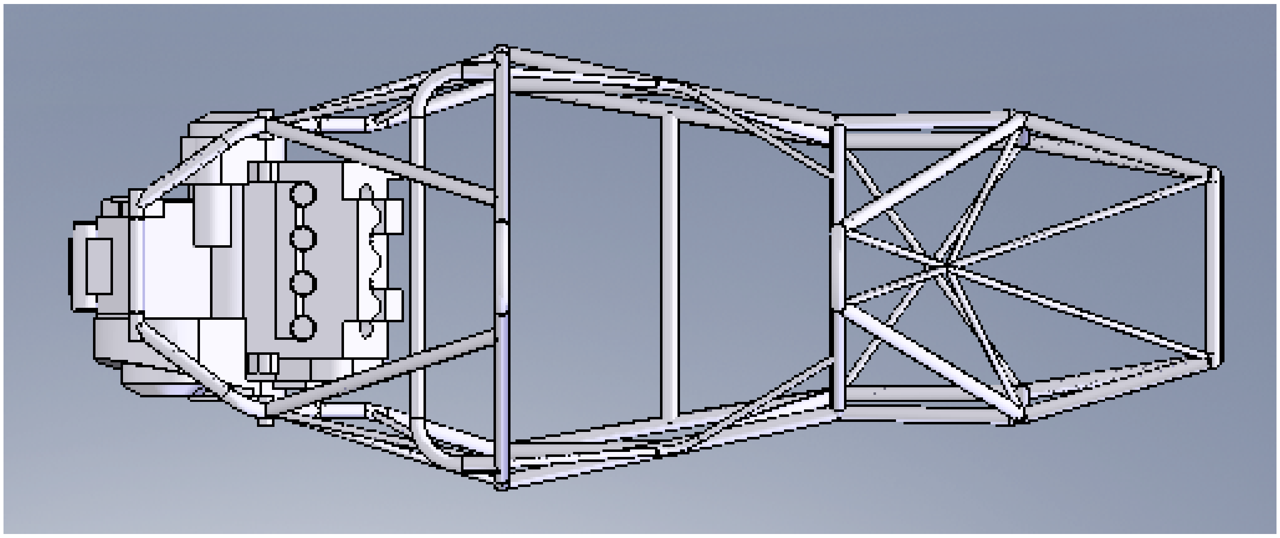

Figure 29: Top View of Final Design

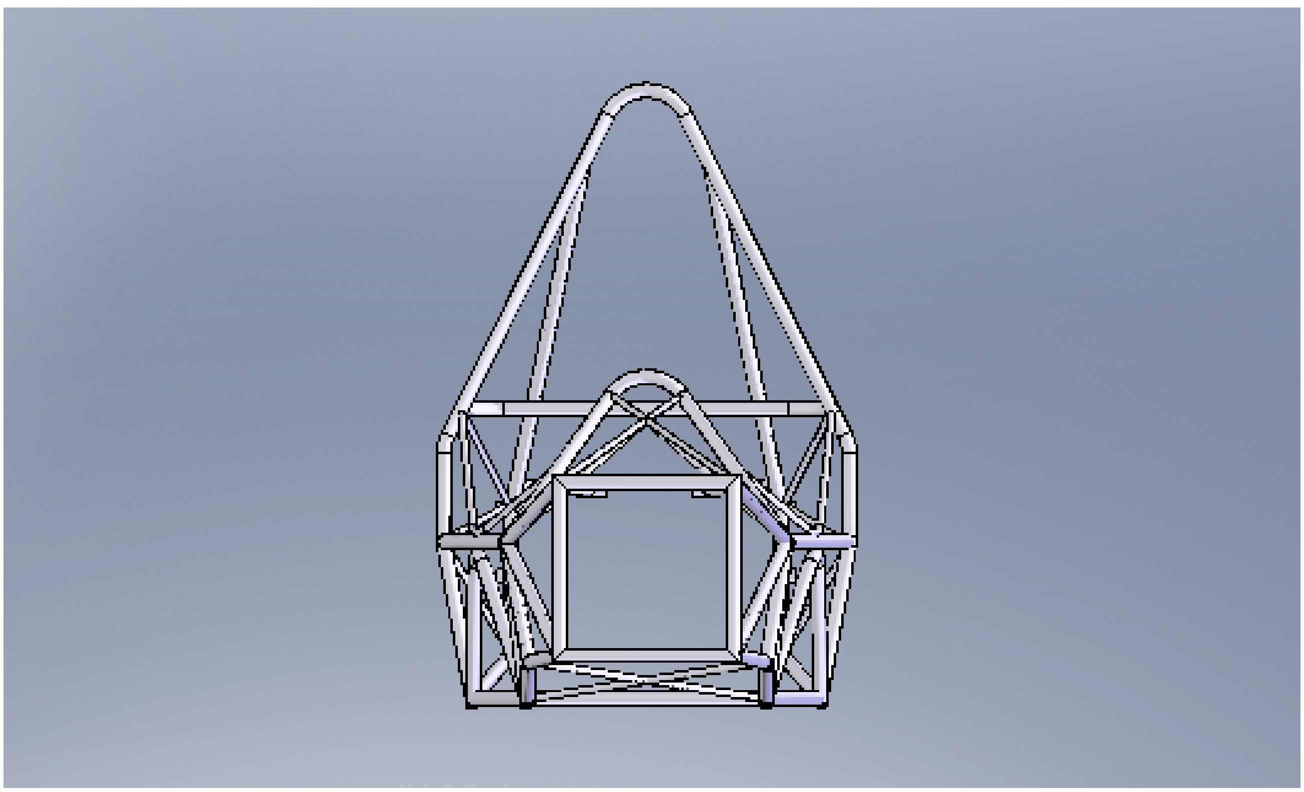

Figure 30: Front View of Final Design

Fabrication

Tubing

There are four different sizes of tubes used in the chassis. All of the tubes were made of AISI 4130 steel. The drawings for each piece that needs to be bent can be seen in Appendix C. In Figure 31, you can see a colored version of the frame. Each color represents a different size of tubing. We then calculated the amount of each tube that we would need to order. This data can be seen in Table 1.

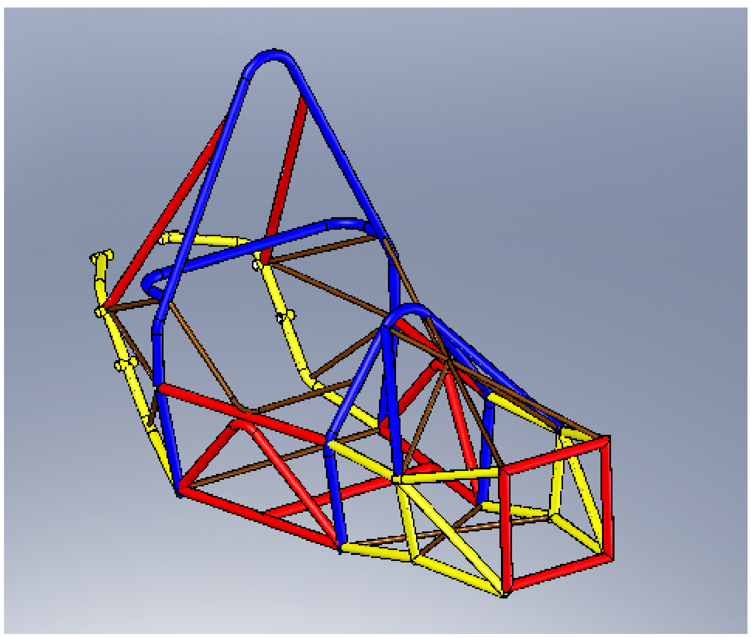

Figure 31: Frame color coded according to tube geometry

	Φ1 - .049 in	Φ1 - .065 in	Φ1 - .095 in	Φ.5 - .039 in
Color on Frame				
Length (ft)	19	24	19	25

Table 1: Length of Pipe

Jigging and Welding

The chassis took a few weeks of work to put together however the result was a well made frame. First tubes were cut and those requiring bends were bent. After they were all cut the tubes were all sandblasted to prep for welding and later priming. The tubes were then notched individually for fitment to the frame. Hole saws, end mills and grinders were used to notch pipes depending on the situation.

First the main roll hoop and front roll hoop were setup on the frame jig along with the front square. These were held in place with a combination of strings and clamps and measurements were done with a variety of tape measures and angle finders. Once these integral parts were setup we notched a few tubes joining them and tack welded them in place to hold them up. We then notched all of the other pipes required for the front half of the chassis. Things to be careful for are pipes which cannot be fit in their place after other tubes are placed. This can lead to large gaps and poor welding.

Once the front half of the chassis was finished then we moved to the rear of the main roll hoop. We hoisted the engine onto the frame jig and used it to accurately place the remaining tubing. This included the engine mounts and small bars for triangulation. Once finished we finish welded the chassis and began welding on other mounting tabs. A few examples of these tabs were the body, seat, floor pan, and suspension mounting points. The suspension mounting points had to be jigged up in order to incorporate anti-dive and anti-squat as well as account for the non-adjustable control arms.

Once all of this was finished the chassis was sent to Bodycoat for stress relief to ensure all the strength was maintained after welding.

The rear sub-frame was water-jet cut from 7075 Aluminum and the support braces were made from ANSI 4130 Steel. The aluminum end pieces and steel braces were all assembled for welding in order to avoid any deformation.

Suspension

For the 2007-2008 vehicle, we made a decision to redesign the suspension from the ground up in order to have a well balanced car. A racecar has two main considerations when determining how competitive it will be and that is power to weight ratio and handling. Power to weight ratio does not differ quite as much since power is limited by the restrictor and weight can only vary so much. So for this reason the next most important aspect to consider is the handling of a vehicle. In our effort to maximize handling we will start with our suspension geometry and kinematics design.

Kinematics

Designing the kinematics for the cars suspension required many steps. The first step performed was in depth research on vehicle suspension. Since our past few teams have not done so well at competition my research focused on mainly outside sources such as books from Allan Staniforth and Carroll Smith. The team also used the 2007 Baja Suspension MQP for a few of the more complex equations since it made them a little easier to understand at times.

Once the basic kinematics were understood, the team started figuring out what the target track width and wheelbase were going to be. With our target track width of 42 inches we designed the control arms around that and with an 18 inch wide frame at the suspension mounting points came up with lower control arms being 12 inches long. Next was the most important part, deciding the angle and length of the upper control arms since that determines the movement of the roll center during dynamic movement.

The roll center is the most important aspect of suspension deign. This is the point which all the jacking forces of the vehicle are centered around during cornering hence impacting the rolling moment on the car. The roll center ideally should be as close to the ground as possible to have as little weight transfer as possible. The FSAE rules require that the suspension linkage be able to have 1 inch of droop and 1 inch of bump however, does not necessitate that the vehicle must ever see such movement during use. For this reason we assumed maximum suspension travel to be ¾ of an inch in either bump or droop. While the suspension is capable of moving the required inch, the forces required to do so exceed those of which the racecar will ever be subject to while being raced. By assuming a ¾ inch movement this allows us to bring our roll center closer to the ground without crossing the ground plane during normal movement.

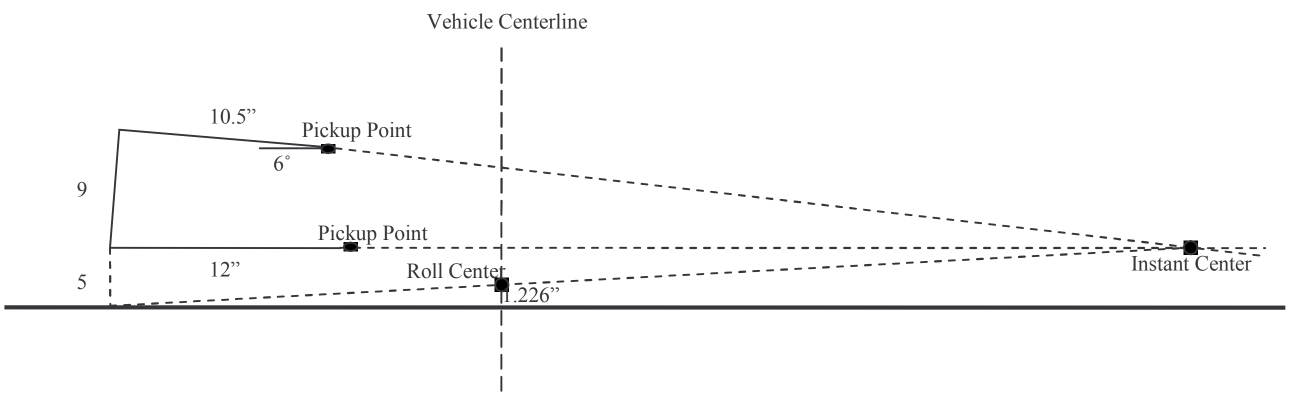

Figure 32: Roll Center Location

This diagram shows exactly how the roll center is calculated. As you can see with various movement of the control arms it is possible to have the roll center move below the ground plane. When the roll center crosses the ground plane this effectively changes the jacking forces of the suspension and causes the car to become unstable. This is why a vehicles suspension should be designed to always keep the roll center either above or below the ground and not cross the ground plane.

With the upper arms being 10.5 inches long and at an angle of 6 degrees this puts the roll center only a tenth of an inch above the ground in full bump. Once confirming this we analyzed the camber gain throughout the suspension travel and found it to be .72 degrees per inch of travel which is also an acceptable value. We then moved on to solid modeling and analysis to design the actual make of these control arms.

Weight Transfer

After doing research on various suspension components it was apparent that the weight transfer of the vehicle is an important factor in how the car will handle. Allan Staniforth has a great walk through of how to calculate a vehicles weight transfer in his book, Competition Car Suspension: Design, Construction, Tuning, under steady state cornering, deceleration and acceleration. This begins with a variety of measured values such as track width, vehicle weight, roll center height and roll stiffness.

These measured values are then used in a variety of equations which goes from one calculated value to another. In the end these values are used to calculate the expected forces seen at each tire for a given number of g forces in each of the three situations. These calculations can be seen in the Weight Transfer Calculation sheet located in the appendices. On this sheet all green blocks can be changed for various vehicles and the values on the right are the outputs for each situation.

Control Arms

Once the kinematic analysis of the suspension was performed, the suspension points of the upper and lower control arms were calculated. The integrated 15% anti-dive, 5% anti-squat, 5 degrees of caster, and 1 degree of camber (front suspension) were all controlled through the kinematics of the suspension and adjustability of the uprights. Therefore, the control arms were made not adjustable in order to ensure there were no rod ends in bending. Previous years placed heim joints at the inboard and outboard side of the control arms in order to adjust the camber of the vehicle. However, this is not proper engineering practice and in order to adjust the camber, the entire wheel, hub, and upright had to be taken off. This year, the camber is completely adjustable through the upright design and the control arms have solid inboard and outboard joints. The inboard side of the 2007-2008 control arms is composed of Delrin bushings with brass inserts; all press fit into a .625 OD 4130 steel tube. The outboard sides of the control arms are stainless steel spherical with joints Teflon liners that have a misalignment angle of 9 degrees. These joints were also press fit into 4130 steel of one inch diameter.

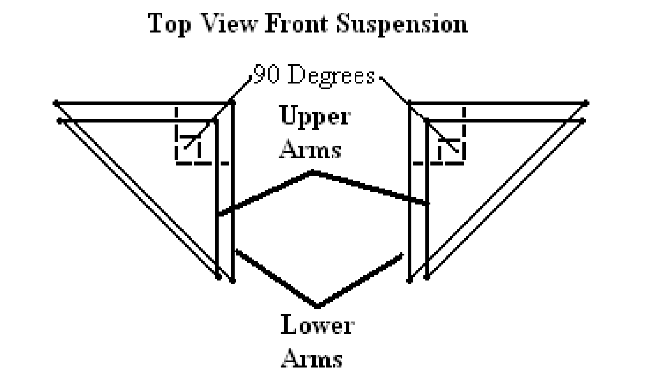

Figure 33: Original Front Control Arm Setup

Several iterations were performed to decide the appropriate geometry of the control arms. The initial designs called for right angle (with respect to the frame) double wishbones in the front and 60 degrees (outboard measured angle) in the rear. This setup was chosen to evenly displace as much of the stress under braking, cornering, and acceleration through the control arms and reduce the deflection. This control arm setup also

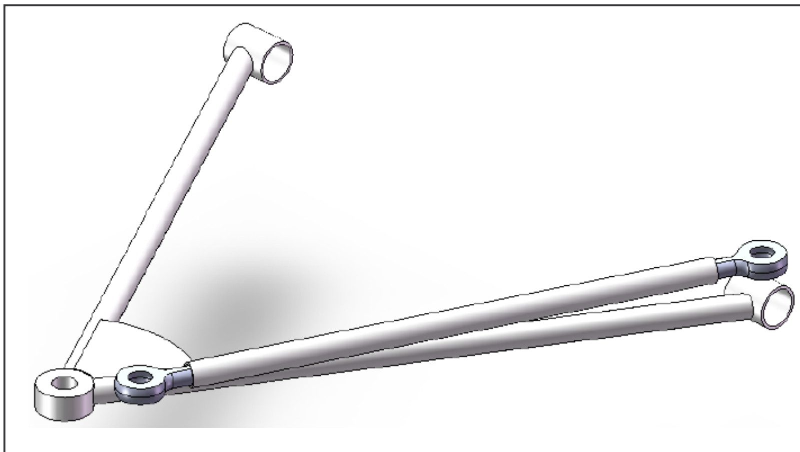
Figure 34: Final Rear Lower Control Arm

allowed for a smaller overall wheelbase of approximately 61 inches.

Due to constraints specified by the aerodynamics group, the front control arms had to be adjusted in order to fit the front wing on the vehicle. Therefore, the wheelbase of the vehicle increased to 68 inches and the front control arms had to be changed to form a 50 degree angle at the outboard side. The rear control arms remained constant at 60 degrees at the outboard side. In order to adjust the tow of the rear suspension, the tow link was made a separate member from the control arms, having two heim joints of opposite threading at each end. This setup was chosen to make the process of adjusting the tow easy and quick, only requiring a wrench to turn the toe link for increased or reduced tow.

After all tolerances were checked, the finalization process of the control arms began. This involved several iterations of various 4130 tube sizing to determine the optimize size (inner and outer diameters) of the control arms. The front suspension consists of a pull rod setup that does not meet the control arms as traditional push rod setups do. The pull rods in the front mount directly to the upright; therefore, much of the stress is removed from the control arms. The rear assumes the traditional push rod setup and therefore the rear lower control arm would experience more load, resulting in a larger wall thickness. Steel tubing was used for this years control arms rather than aluminum because of the durability and manufacturability of 4130. Aluminum control arms would require more manufacturing time and also be susceptible to critical failure under the constant stress experienced during dynamic events in competition.

The optimization process involved looking at various 4130 steel tubing outer and inner diameters as well as geometries. At one point, streamline tubing was considered to aid in creating a downward force on the suspension for increased grip and a lower center of gravity. This option was discarded because the high cost of tubing was not worth the minimal aerodynamic benefits. Originally, .625 inch outer diameter tubular steel was selected for the control arms with a wall thickness of .049 inches for all control arms except the rear lower having .058 inches. After performing several iterations of FEA using COSMOS, the team decided to reduce the tubing size for weight savings. The final control arm design resulted in half inch tubing throughout the suspension with a wall thickness of .049 inches on all control arms except the rear lower having .058 inches. The analysis of the control arms was performed using the weight transfer sheet calculations for turning, braking, and acceleration. The control arms were subject to 300 lbs of load for the worst case scenario. This load is an overestimation of the actual load that an individual control arm would experience during any of these scenarios. However, this load was used to ensure that each control arm would withstand extreme loading. The end result proved that the maximum deflection was 0.006 inches with a

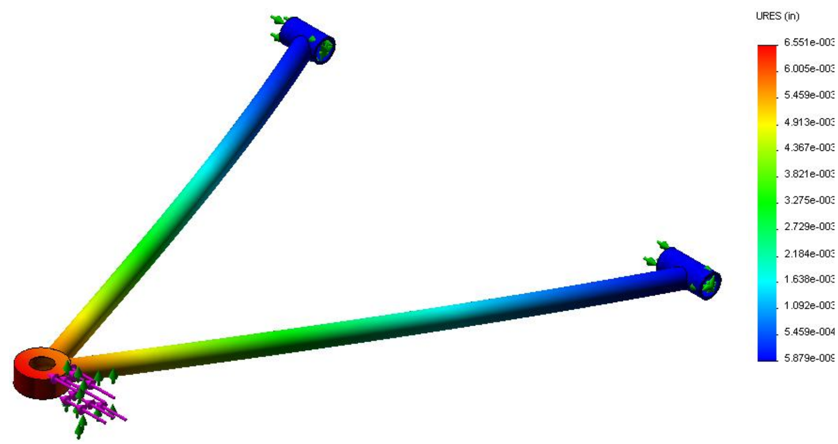

Figure 35: Maximum Deflection of Front Control Arm

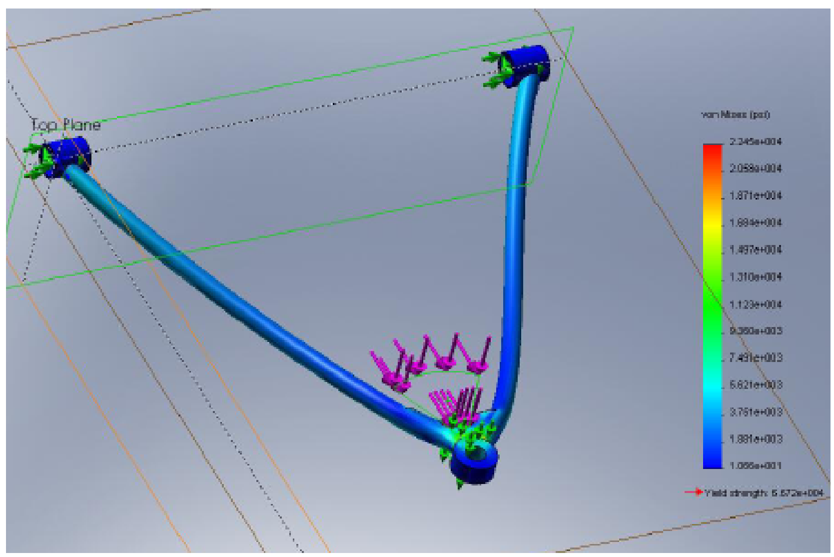

Figure 36: Von Mises Stress of Rear Lower Control Arm

safety factor of 7. As mentioned, several iterations were performed with the overall goal to optimize the control arms for weight. Previous years never performed the appropriate optimization of control arms, which resulted in a suspension that could be used to pick up the entire car, which should never happen. The end result of this year optimization process were control arms that individually weigh under a pound each. Further reduction of size was not pursued because the reduction of safety factor and increased deflection were not worth the fractions of weight reduced.

After finalizing the optimization of the control arms, the proper 4130 steel tubing was ordered from AED motorsports. In order to ensure that the precision of the geometry of the control arms was reflected in the fabrication process, a suspension jig was created. The jig was created using an 11x11x0.75 inch 6061 aluminum stock as the base plate from which the exact inboard and outboard points of each control arm were CNC drilled. (The suspension jig not only was used to weld the control arms, but also to correctly place the suspension tabs on the frame) Two aluminum blocks were also CNC machined to hold the inboard 0.625 inch OD tubes and two, one inch diameter cylinders were turned to hold the outboard members (one for the upper control arms and one for the lower). The upper control arms, being at

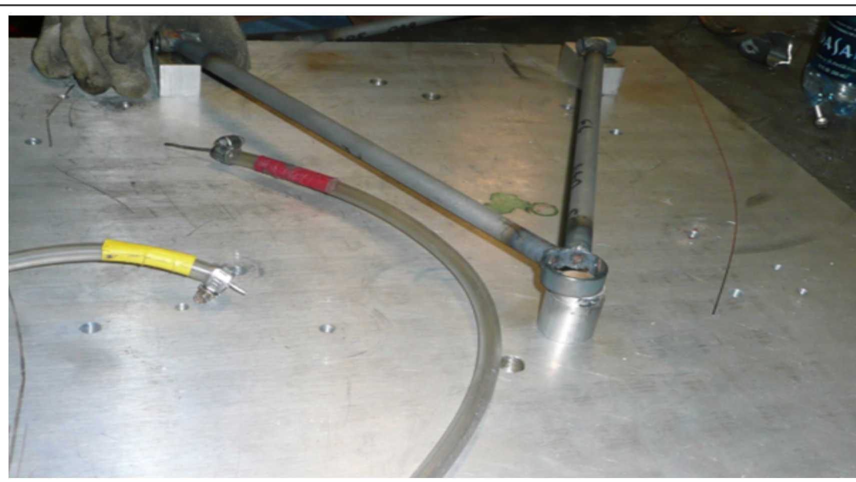

Figure 37: Control Arm Fabrication Jig

a 6 degree incline, were fabricated to maintain no offset of the spherical bearing at suspension rest. This means that the spherical bearings do not sit in misalignment and all 9 degrees are available for use in suspension travel. Once the jig was completely setup, each precut 4130 tube was placed in its respective setup and TIG welded, while purged with argon. The rear lower control arms received a cross member plate located at the outboard joining of each arm and two vertical 1/16th inch steel tabs were welded in place to provide a connection point for the pushrods. Once completely fabricated, the control arms were sent for heat treatment at 800 degrees Celsius and were oil quenched to increase the overall strength of the material. There was little to no deflection of the control arms during heat treatment. After heat treating, the inboard and outboard bearings/bushings were press fit and the control arms were prepped, painted, and installed on the frame.

Monoshock/Rockers

At the heart of the WPI Formula SAE monoshock suspension is a device called a rocker. Two such pieces exist on the car—one in the front suspension and one in the rear. These rockers are essentially bellcranks, which are commonly found on race vehicles. A bellcrank serves as a key facilitator when spatial constraints limit the positioning of dampers. Bellcranks effectively translate force from one direction to another by pivoting about a central pin joint. This way, instead of being directly connected to the control arms of a vehicle to achieve ideal *transmission angles*, dampers can be mounted in locations more convenient for packaging and/or weight reasons. For example, on this year's car, the front suspension is mounted below the floor panel of the cockpit, which improves ergonomics and lowers the center of gravity (CG) of the vehicle.

The transmission angle between two rigid links refers to the angular deviation from the direct application of force. For example, consider pushing a shopping cart through the grocery store. There is a noticeable transmission angle between a person's arms and the direction the cart is moving. We exert force not only in the direction parallel to the floor, but also downward. The downward component of force is essentially wasted, as no work is performed in this direction. If we pushed on shopping carts only in the precise direction of motion, no force would be wasted, the transmission angle would be zero, and the action would be much more efficient.

In the automotive realm, available space is rarely sufficient to achieve this feat. Furthermore, compromises must be made when mounting dampers. If they were to be placed normal to the control arms of a vehicle, during bump a phenomenon called 'falling rate' would occur. Simply put, this means that the control arm would see less resistance from the spring on the damper as it rotates toward it (when the wheel attached to the control arm is going over a bump). This is due to an increasing transmission angle between the damper and the force normal to the tire pushing on the wheel. What happens is that the spring is "less able" to resist the upward rotation of the control arm. This can be counteracted by angling the damper in its static position, so that during initial bump, the damper becomes more perpendicular with the control arm (and therefore transmission angle decreases). This achieves what is known as a 'rising rate'. The effective spring constant in the damper would rise in the initial bump phase. However, after the position where the damper and control arm are perpendicular, spring rate falls again as the control arm continues through its rotation. Thus, a situation of compromise emerges. Suspension designers must decide how to orient dampers to balance rising rate and falling rate, while still minimizing transmission angles.

The example outlined above also applies to remotely mounted dampers that use bellcranks for force translation. Race cars often use two types of suspension that incorporate bellcranks: push rod and pull rod. These setups are reciprocals of one another. In a push rod suspension, rigid links are pinned near the wheels, closer to the ground. These links then slope upward to bellcranks and dampers that are typically placed closer to the top of the car, so that bump induces compression in the links—the push rods—and compresses the dampers. Pull rod suspensions use the opposite orientation. Links are pinned higher at the wheels and slope downward to the bellcrank/damper assemblies, so that bump induces tension in the pull rods as the dampers are compressed. Both of these setups have their advantages and disadvantages, but deciding on either type usually involves packaging and material properties. If the suspension links are made out of a material that is generally stronger in cyclic loading in compression, then a push rod setup is desirable because the rods will see the most force in compression (bump). Similarly, if space is not available near the top of the vehicle, a pull rod design might be necessary. The 2008 WPI FSAE car uses both types of suspension, for reasons explained later in this section.

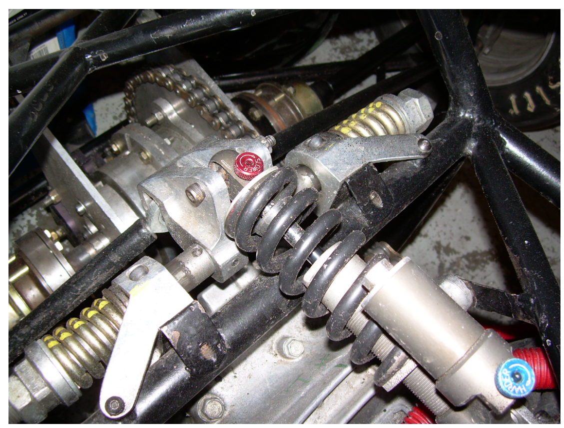

Figure 38: The rear monoshock suspension setup on the 2004 WPI FSAE car.

A monoshock suspension is exactly as advertised: it uses one damper for the rear wheels and one for the front wheels, while maintaining front and rear independent suspension. These traits seem to contradict one another, but a closer look at the kinematics of the setup reveals that it is possible in a race setting with a relatively flat track surface. FSAE is, therefore, a perfect candidate for the use of a monoshock setup.

In this arrangement, pull or push rods connect to a common rocker (or bellcrank). The rocker is then either allowed to slide along a shaft transverse to the vehicle or attached to a shaft that slides back and forth in the same manner. This is the main portion of the mechanism that allows the wheels to remain virtually independent of one another. An image is included in Figure 38 for visualization of the setup. Consider a situation in which the right wheel undergoes bump. A force is exerted normal to the contact patch of the tire, and is transmitted through a push rod to the rocker. The rocker sees three components of force (left-right, up-down, and backward-forward). Backward-forward (longitudinal) and up-down (vertical) forces are resisted by shear reaction forces in the rocker shaft. Left-right (transverse) forces, however, are resisted by springs on the rocker shaft.

If the rocker shaft moves with the rocker, roll springs should be pinned between the moving shaft and a stationary attachment point. If the rocker moves instead and the shaft remains stationary, springs must be placed between stationary attachment points and the sides of the rocker. Side-to-side motion shifts the attachment point of the suspension links transversely across the car. When a wheel on one side of the car encounters a bump, this attachment point will be pushed away from the centerline of the car, toward the other side of the vehicle. Meanwhile, the aforementioned springs will resist this movement and the assembly will act as a virtual stabilizer bar. During this translation, the push rod connection point also shifts with the upward rotation of the rocker. The net motion that results follows the range of motion of the end of the opposing push rod quite closely. This means that very little force is exerted on that push rod. It is merely rotated with the rotation and sliding movements of the rocker.

By default, barring loose joints and other play in the system, any force on this opposing push rod will result in paired wheel movement. If roll springs are made too stiff, or conversely, if damper springs are made too firm, there will either be too much sliding of the rocker or too much rotation, which will cause the wheels to react to bumps in unison. This has a dramatically detrimental effect to the performance of the car, which enforces the importance of tuning the suspension. Despite this sensitivity, the effect will be less noticeable on relatively flat tracks similar to the ones the car will be driven on during the FSAE competition. Banked sections exist and there is no such thing as a perfectly smooth surface, but the degree of both of these potential deterrents to monoshock during competition will be insignificant enough to keep a monoshock vehicle's wheel movements independent of one another.

As demonstrated, in the monoshock system, the rocker and shaft not only serve as force transmission and support tools, but also as part of the anti-roll system. Therefore, the functional density of the system is high. Only one damper is used, which detracts from the weight of the car, components serve multiple purposes. In addition, using a monoshock setup increases cost effectiveness. Dampers are relatively expensive pieces of equipment. On the 2008 car, not needing to buy four dampers allowed the team to instead purchase two dampers of much greater performance and quality.

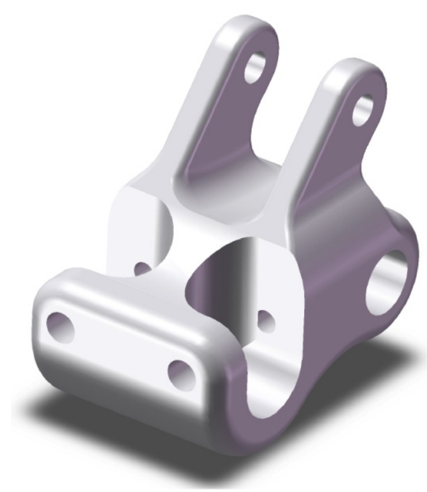

Figure 39: Preliminary rocker design

The appearance and form of the front and rear rockers on the 2008 WPI FSAE vehicle's monoshock suspension underwent several small iterations, but made one major transition between designs. The final stage of the first design is depicted in Figure 39. The primary reason for this modification was ease of manufacture. The initial shape would have been impossible to fixture in a CNC mill, which was the best available method of manufacture for the rockers. Therefore, the basic rocker shape was simplified, taking visual cues from the 2007 vehicle's rockers, while being structurally tailored to fit this year's car.

Initial layout of the rocker was dependent on three attachment areas:

- Pull/push rods
- Rocker shaft
- Damper

Force would be translated at virtually 90 degrees from input to output, so rod and damper attachments needed to be located at opposite ends of the rocker, with radii extending to them from the center of the rocker shaft at close to 90 degrees between one another. This translated to an "L" shape for the rockers. The 90 degree angle in the L shape was later modified to meet the packaging demands of the front suspension. Placing the steering rack, rocker assembly, and damper in close proximity to each other resulted in the angling of the damper relative to the ground (at 15 degrees). As a result, the angle between the damper points and rod points was increased to 105 degrees, allowing the rocker to remain horizontal when the car is in a static position.

The suspension was designed with both bump and droop in mind. Rockers were oriented so that transmission angles are near zero in stasis. Essentially, when the vehicle traverses a bump in the track and the dampers compress, the push/pull rods will experience decreasing force until maximum deflection. After maximum deflection, while returning to the suspension's initial position, they will experience a rising rate. The same goes for droop. Effectively, this balances the car's handling between absorbing bumps and returning to normal after bumps.

Finally, two principles known as *motion ratio* and *wheel rate* were considered during rocker design. Consider once more an L-shaped rocker. If input force on one arm of the L has the same transmission angle and orientation as the output force from the other arm, the amount of motion at input will be directly proportional to the resulting motion at output, based solely on the ratio of the length of the arms. This can be better expressed using the diagrams in **Error! Reference source not found.**.

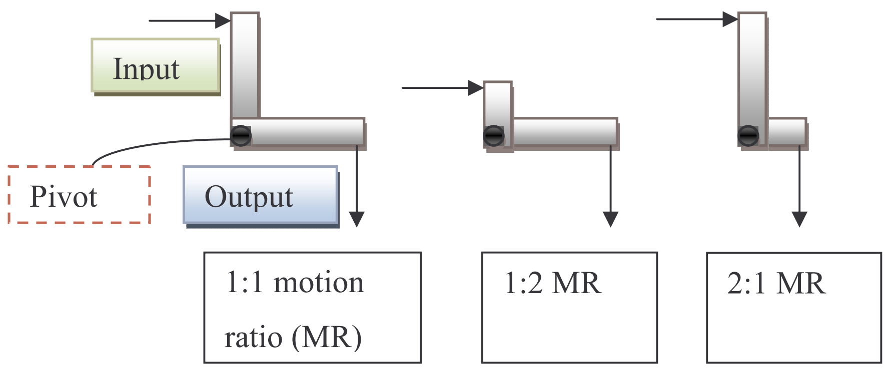

Figure 40: Visual aid of motion ratios.

The preliminary rocker design utilized close to a 1:1 MR, which would necessitate using springs with a constant of 1200 lbf/inch. The front rocker was modified to have an MR of 1.33:2, after the original design was proven to compress the spring too easily. This effectively made suspension movement more difficult by decreasing the mechanical advantage introduced to the wheels by the rocker. Springs were downgraded to 650 lbf/inch. Wheel rate was then determined by the following equation:

$$\textit{wheel rate} = (MR)^2 \times (\textit{spring rate})$$

The wheel rate for the front wheels (independently) was calculated to be 3.2 Hz. The rear wheels achieved a rate of 2.8 Hz. The motion ratio of the rear rocker was maintained at 2:2.25.

The newly modified rockers were analyzed using FEA in COSMOSWorks. Their material was to be a choice of aluminum alloy—either 6061 or 7075. 6061 T6 aluminum was chosen for its light weight and availability. Testing confirmed safety factors of 1.71 (front rocker) and 1.34 (rear rocker) with an excessive 600-lbf load. Displacement of both rockers was minimal, with the greatest seen in the rear rocker at one hundredth of an inch. The front rocker deflected by less than six thousandths of an inch.

Uprights and Hubs

Upright design for the 2008 vehicle was initiated in a similar fashion to previous years' designs, after having observed several of these on and off of their respective vehicles. In particular, special attention was paid to the uprights on the 2004 and 2007 cars. The 2004 uprights exhibit minimalistic structure, in which purpose is immediately evident. The 2007 uprights, specifically in the front, were very abstract in shape and design. In order to determine which would be the best initial form for this year's pieces, both types of structures were modeled using computer aided design (CAD) and analyzed by employing the use of a process called finite element analysis, or FEA.

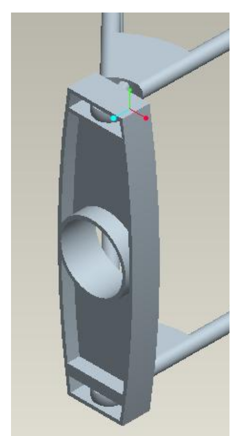
Figure 41: Initial Upright Design

SolidWorks 2007, after a brief beginning with ProEngineer Wildfire 2.0, became the medium of choice for building the virtual uprights. Construction began with a convex, elongated shape approximately one inch thick and eleven inches tall. The initial shape is shown in Figure 41. The piece was given a hollow center, with the idea of adding in gusseted supports to connect the various connection points in the upright and distribute the stresses appropriately. This very preliminary design was submitted to experienced veterans, resulting in mixed reviews. No FEA had yet been conducted.

A secondary design was modeled in SolidWorks that took a form not unlike that of the 2007 front upright, the latter of which is depicted in Figure 42. Images of the 2008 revision are not available, but its shape followed the same principal. The design began with boundary conditions. Essentially, the upright must support up to six connections: brake caliper, two control arms, tie rod, pull or push rod, and wheel shaft. The positioning of most of these entities was already known, based on previous suspension design work. The following set of statements best describes the configuration constraints already in place during the designing of the uprights:

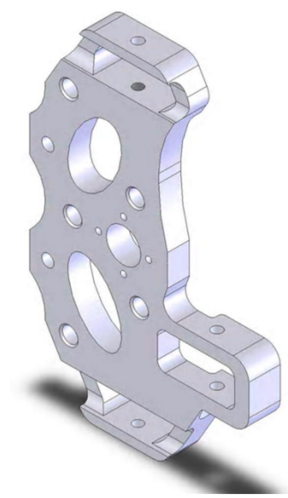

Figure 42: 2007 Front Upright

1. Upper and lower control arm connections must be nine inches away from one another and must be arranged vertically.
2. In the front, a point in the same rotational plane (with respect to the wheel), but not collinear with the control arm connection sites must be designated as the site of steering connection. In the rear, a similar location must be determined for a toe link, to prevent the rear wheels from turning and still allow suspension movement.
3. There must be a central location from which the control arm connection points are equidistant, and that also lies in the same rotational plane as those points and the steering link point, for the placement of the wheel shaft.
4. A location must be established for the mounting of the pull/push rods, ideally in a collinear manner with the wheel axis and control arm mounts.

5. Finally, a site on either side of the upright (toward the front or rear of the vehicle) must be established for connecting a brake caliper.

With these constraints in place, the setup for the control arms and wheel shaft/hub mounting was already fully established.

During the period when the uprights were being created, work had been completed on the frame design for the vehicle, images of which are available in the according section of this report. Control arm design was also near completion, so locations of the frame-side control arm mounts were known. Based on where these fell in relation to the remainder of the cockpit (especially the area for the driver's legs), the decision was made to use 'rear-mounted' steering. This simply means that steering would need to be linked to the uprights behind the axes of the wheels. To avoid concentrating stress on one side of the upright, brakes would mount in front of the wheel axis.

Because of where the lower control arms were set to mount against the frame of the car, it was clear that the steering rack would be positioned higher than the lower control arms. In order to avoid 'bump steer', a phenomenon discussed in the steering system design section, steering link mounting points were positioned higher than the lower control arm mounts (by one inch), and two inches aft of the figurative central axis linking the control arm mounts. This provided enough torque for turning capability, while providing the necessary clearance between the upright and the wheel.

A preliminary design arose from these parameters as a model in SolidWorks. However, upon being inspected by those more experienced in manufacturing, it was deemed virtually impossible to create. One member of Tech Racing took the original, less abstract model, and modified its internal structure to great success. This model was further modified to produce a design that would use a sliding upper control arm mount to adjust camber. The outer border of the upright was kept at a one-inch thickness to minimize deflection under loading, while material was removed, or 'faced down' in the central part of the upright, on the back and front surfaces.


Figure 43: Upright with Detachable Camber Adjustment

The team made the decision to utilize 7.75-inch diameter brake rotors, and also to incorporate brake caliper mounting tabs into the upright for modularity. This decision would have little to no impact on the machining times of the parts, and furthermore reduced hardware requirements. The same idea applies to the incorporation of the steering link mount into the upright. Unlike some of Worcester Polytechnic Institute's previous FSAE cars, the pull rods in the front suspension mount to the uprights. Because of the size of the pieces and space limitations, a mounted tab was the chosen solution for the attachment of the pull rods. Holes needed to be added to the upright for those two attachment bolts, as well as a bolt hole for the steering link, a bolt hole for the lower control arm, three bolt holes for the upper control arm, four bolt holes for the wheel shaft, and two bolt holes for mounting the brake caliper. The resulting iteration that emerged from these decisions is depicted in Figure 43.

Despite the mechanically sound nature of the sliding upper control arm mount, in this form it would require the manufacture of additional pieces. The idea was not abandoned, but rather restructured to use a different sliding mechanism, integrated into the upright. This alleviated the need for additional pieces. A slot was cut in the top of the model to allow a 5/16-

inch diameter bolt to slide back and forth in the upright, resulting in a two-degree range of camber adjustability. The final design is depicted in Figure 44.

Figure 44: Final 2008 Upright Design

Finite element analysis needed to be performed on the front upright to determine its structural integrity, but in order for this to occur, a list of potential materials had to be constructed. Aluminum is and has generally been the metal of choice for uprights, due to its high specific strength (strength per weight). The Al-6061-T6 and Al-7075-T6 alloys were selected as candidates, due to their availability. Initial FEA using the 6000 series alloy produced severe deflection, on the order of tenths of an inch. Secondary attempts using 7075 aluminum yielded much more satisfactory deflection values, closer to thousandths of an inch.

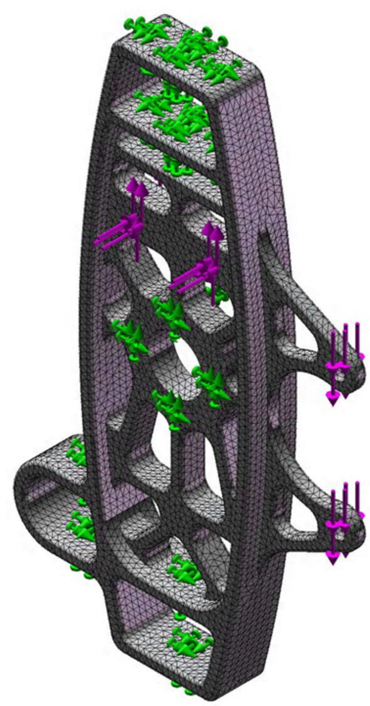

Figure 45: Front upright meshed for FEA

FEA was conducted using COSMOSWorks. Force placement was derived using visualization and other techniques, and will be explained shortly, while force values were based upon calculations. A member of Tech Racing created a weight transfer worksheet in Microsoft Excel, a process better explained in the suspension design section of the report, based upon a 600-lb car with an approximated center of gravity. The worst-case scenarios were taken as the vehicle cornering with normal (centrifugal) acceleration reaching nearly twice that of gravity while it is braking at two times the acceleration of gravity (front uprights), and the vehicle accelerating at full throttle and maximum tire grip (rear uprights). In reality, the tires selected would not hold up to this abuse, so the vehicle would never see such an extreme load. Nevertheless, the net load on the outer front wheel in the former situation becomes nearly equal to that of the entire car. Calculations for the values of all the forces used in FEA are available in Appendix A.

Force was applied to the brake caliper mounting holes of the front upright in the direction of brake rotor rotation at that location (downward). Braking bias was assumed to be 80% front, 20% rear. The resulting brake force on the upright was calculated to be approximately 480 lbf (pounds force). The wheel shaft considered a stationary object and was fixed in place. Control arms were considered to be "locked" in place at maximum bump, so their mounts were also fixed in SolidWorks. However, force was applied to the pull rod mount, as a response to the compressive force exerted upon the pull rod by the front shock absorber caused by the 'dive' the car experiences. To prove the point that the upright would not even yield with the entire weight of the car pressing on the push rod mount, 600 lbf was applied at a 30 degree angle from horizontal (the approximate orientation of the pull rods). The force is an obvious approximation, as this situation would never occur in reality. The meshed upright and applied forces and restraints are shown in Figure 45. The result of this FEA study on the front upright yielded a safety factor of 3.8, even with extreme applied forces. The stress and displacement distributions are shown in Figure 46 and Figure 47.

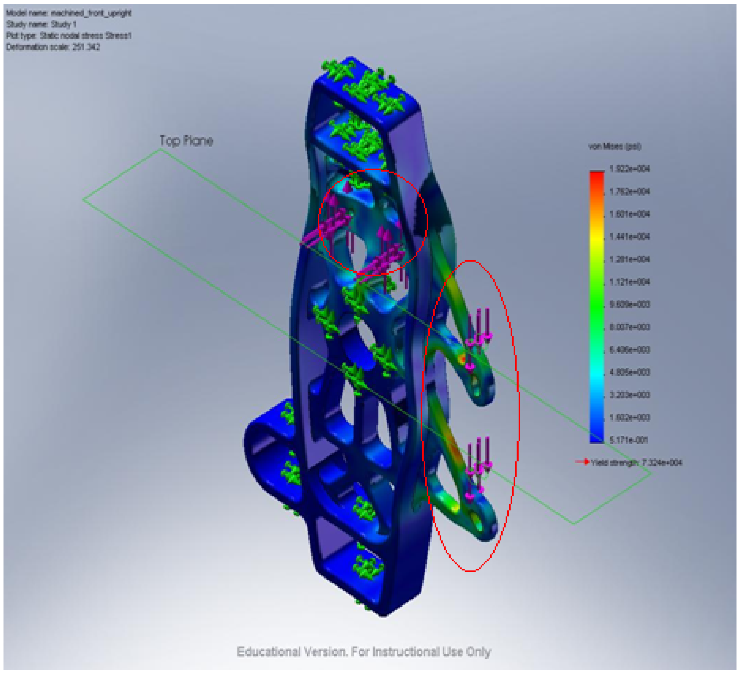

Figure 46: The front upright with stress distribution

Areas of high stress concentration are circled. The scale of deformation is highly exaggerated. The maximum normal stress achieved was 19.2 ksi.

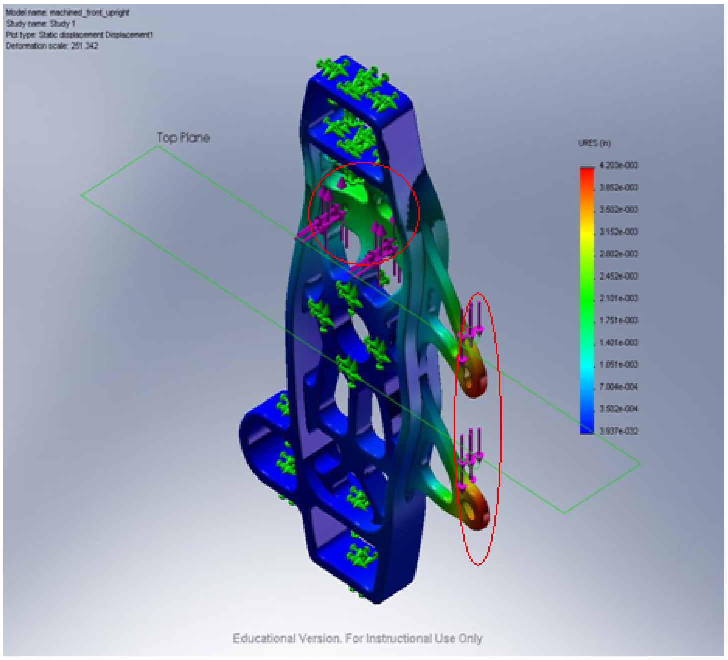

Figure 47: The front upright with displacement distribution

The scale of deformation is highly exaggerated. The maximum displacement achieved was 4 thousandths of an inch.

Design and analysis of the rear upright followed a similar procedure. There were three key differences when considering the rear upright versus the front:

- Lack of turning requirement
- Drive-train accommodation
- Lack of push rod attachment

In regard to the last member of the list, the team spent some time considering the geometry (and especially transmission angles) of the rear suspension in both situations, eventually deciding to mount the push rods at the ends of the lower control arms on the outboard side. This decision is also mentioned in the suspension design section of the report. Part of the reason for this was to get the minimum possible *transmission angle* between the push rods and the rear rocker. The rear suspension setup is better described in its proprietary section, but let it be said that the more horizontal the push rods are oriented, the more poorly the rear suspension will function. Because of this, mounting them as low as possible on the outboard side (near the wheels of the car) was beneficial to overall suspension performance. This alleviated some space on the rear uprights, while also eliminating a significant amount of force. Material was able to be removed from the model as a result.

Another consideration, drive-train accommodation, simply involved a modification of the central hole and its surrounding structure, in order to house bearing races for the drive shafts. Later FEA proved that the internal gussets of the front upright model would hold up well even if torque from the engine were to be transmitted to the bearing races.

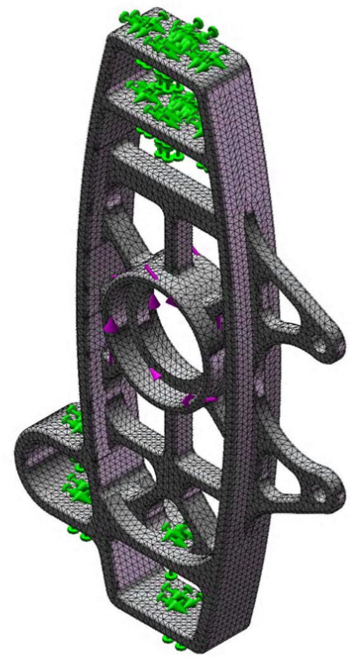

Figure 48: The final rear upright, with a mesh, constraints (green) and torque (purple)

Finally, one of the most ingenious decisions for the design of the rear uprights involves the use of the steering link mounts for toe links in the rear. In fact, this feature allows the front and rear uprights to be cast from the same initial mold, as they share the same basic shape. The uprights require only subtle differences in the machining of their internal structure to be specialized for their specific position (front or rear) on the car. Uprights are also symmetric down the centerline of the vehicle. All of this is hugely beneficial to manufacturing. The finalized design for the rear upright (meshed and with forces applied) is depicted in Figure 48. The same basic constraints were applied as in the FEA of the front upright model, but this time a torque of 800 lbf-in was applied at the center hub to test the capability of the upright in handling loads produced by the driveshafts. The resulting safety factor was 8, confirming the structural integrity of the rear uprights. The stress and displacement plots from this analysis are available in Figure 49 and Figure 50.

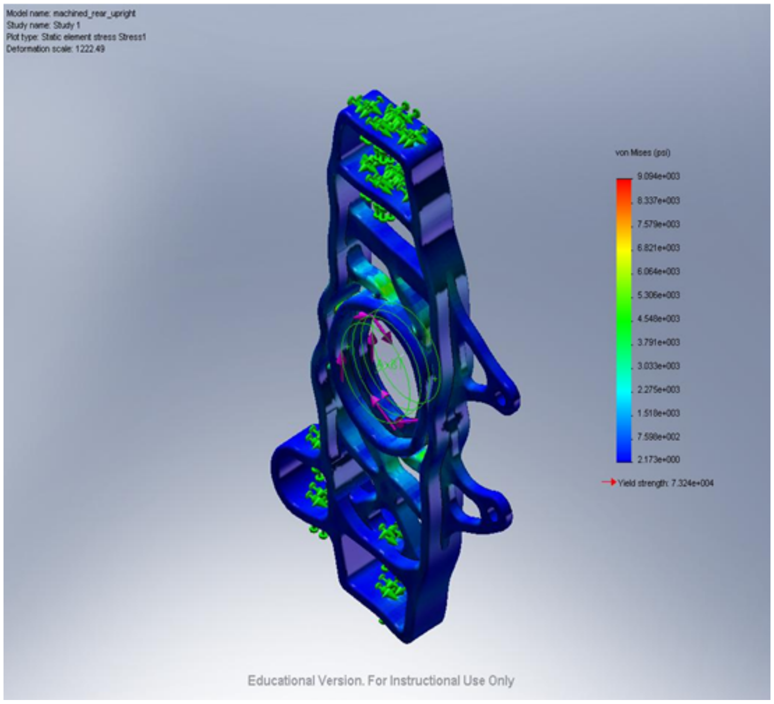

Figure 49: The rear upright with stress distribution

Areas of high stress concentration are circled. The scale of deformation is highly exaggerated. The maximum normal stress achieved was 9.1 ksi.

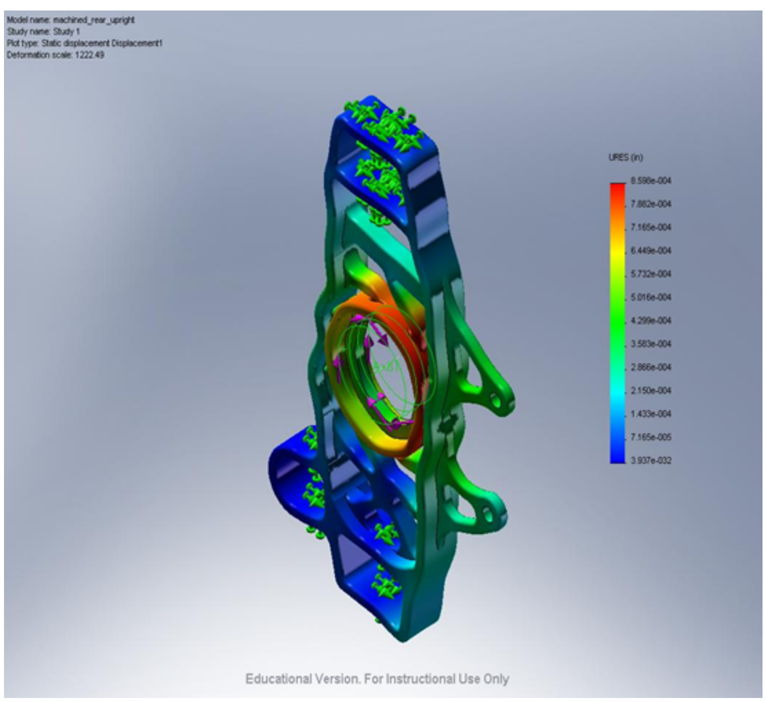

Figure 50: The rear upright with displacement distribution

The scale of deformation is highly exaggerated. The maximum displacement achieved was 8.6 thousandths of an inch.

Next, the design and analysis of the wheel hubs will be discussed in detail. Further information on the uprights is available in the appropriate manufacturing section of this report.

The main design considerations for the hubs are ensuring that they match the bolt patterns of both the brake rotor and the wheel which mounts to it. Also the offset is determined by the wheel hub which is very important especially in the front of the vehicle. In order to accommodate steering the front hubs had a little more offset than desired but was within acceptable limits.

The front hubs were machined using the VF4 CNC Machine out of a cylinder of 6061 aluminum. This was good both for design and cost. The rear hubs are altered ATV hubs which were readily available. This was good for this year's team due to limited time and materials however as far as cost and design goes it would be much better to machine the rear hubs as well.

Wheels and Tires

The wheels and tires are one of the most expensive sections of our vehicle which we cannot seem to lower the cost for at all. We have been using lightweight Keizer three piece aluminum wheels for the past few years. We opted to use these wheels this year mainly because it is what we had and we knew our budget was already tight. For future years it may be worthwhile to look into other available wheels and see what options there are to cut some costs here. Different wheel offsets could also be looked at to work with the suspension geometry.

The tires we are using are Hoosier 20.5x6-13. These tires have treated us very well and are well known to be a quality tire. During testing we will be evaluating some of the handling characteristics to compare with the tire consortium data which we got this year. The tire consortium data will be a valuable resource for the team next year.

Steering System

Our vehicle this year uses a 12:1 ratio rack and pinion type steering. This is the most typical type of steering system in a vehicle. The steering shaft comes into the steering box perpendicular to the steering axis and has a circular gear called the pinion attached. This pinion engages a flat bar called the rack and turns the circular motion into a linear motion which the rack sees. The rack then has rack ends to extend the steering points to a position which is optimal for the car.

For our vehicle this made each steering rack end about 7 inches long in order to bring the steering point in the same plane as our suspension points. This is in order to minimize bump steer which will be addressed later in this section. Our steering shaft is made of three quarter inch ANSI 4130 steel with a single universal joint. The rack placement is as low as possible but ended up being a little higher than ideal due to the undercar packaging of the suspension components. Our tie rods or steering arms are 12 inches long and extend from the rack ends to our uprights.

In the back of the car the same spot on the upright is used by the toe link. The toe link serves the same purpose as the tie rod except is stationary and only used to adjust the static toe of the rear wheels. Changing the static toe has an effect on corner entry of the vehicle and can make the car either more or less stable.

Ackermann Angle

Ackermann steering is the effect cause by the angle of the steering arm in comparison to the mounting points on the upright. Essentially Ackermann causes the inside tire to turn slightly more in order to better travel the smaller turn radius when compared to the outside tire. On old vehicles and street cars this is good because it eliminates dragging of the inside tire during cornering. However on a race car you will often be pushing the limits of the vehicle and tires which causes a high slip angle. The slip angle is the difference in angle of the direction the tire is pointing and the direction that it is actually traveling. Too much slip angle is a negative factor and so Ackermann actually increases slip angle on a vehicle.

So for our racecar we made the decision that Ackermann was not going to be a priority. We knew the car would have some which makes it easier to push around the pits but did not particularly want too much in order to maintain lower slip angles on the inside tire. Another factor is that the inside tire will not see as much cornering force anyways which is why we did not look into anti-Ackermann design. Anti-Ackermann is when the inside tire turns less and hence makes up for some of the added slip angle on the inside tire. Due to packaging this was not an option. Our final design includes enough Ackermann so that while turning on a circle with a 50 foot radius the inside tire will steer 1 degree further than the outside tire.

Steering Damper

In small, lightweight racecars such as ours the steering system can tend to get very twitchy especially at high speeds. Castor is built into our front suspension geometry in order to help with this steering stability however with the stiff suspension small road imperfections can be amplified through the steering system to cause uncomfortable or uneasy feedback though the steering column. In order to eliminate some of this we have used the steering damper off of a motorcycle. The body of the damper is fastened to a non-moving part of the frame while the shaft is attached to the steering rack end. When the steering system is moved it moves the fluid through this damper and provides some resistance to the steering movement. This resistance can take out many small variations in the movement of the steering. This is why we have included a steering damper on this years steering system.

Bump Steer

Bump steer is based on a phenomenon that occurs when three pinned-pinned links are pivoting together and attached to the same links at either side. Ideally, these links should be parallel and of equal or proportional length. If this is not the case, there are implications detrimental to the functionality of the system. There are several combinations of parallelism and position that determine how rotation of the three links will be affected. Three of these combinations are shown below:

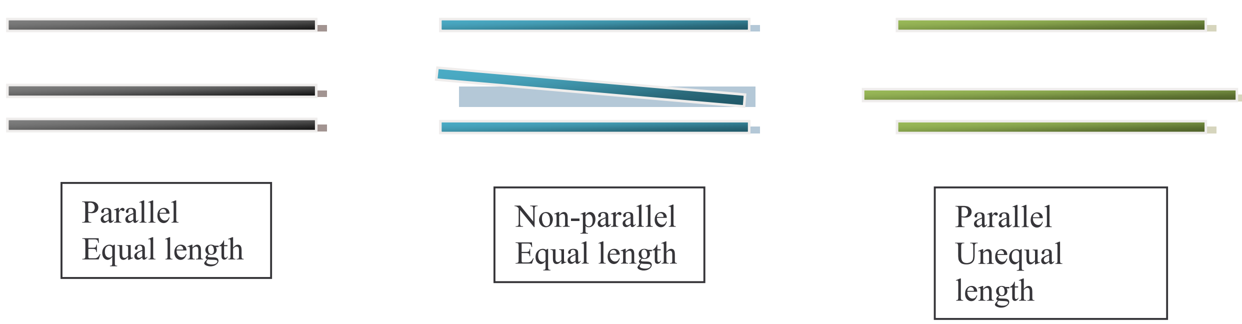

Figure 51: Examples of Various Links

Imagine that the ends of these "links" are pinned in place. Each set of three pin joints (at the ends of the links) is joined by a single solid body. Clearly, the most predictable system is the first. All three links are of equal length, so when the solid link at one end of them is held in place and they are rotated via movement of the solid body at the other end, the arcs traced by each of the three links will have the same radius. Because they are also parallel, the solid links at the ends will remain parallel and the system has an extensive range of motion:

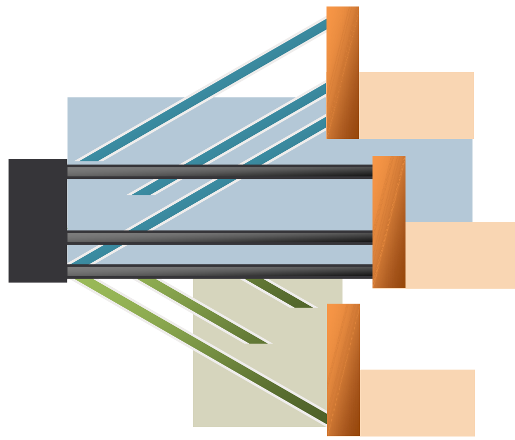

Figure 52: Bump Steer Diagram

This system is the ideal setup for the control arms and steering attachment on a car. Because of the lengths and position of the control arms on the 2008 WPI FSAE car, this is impossible to achieve. The control arms are both non-parallel and of unequal length. Fitting in a tie rod arbitrarily would thus have induced bump steer.

Bump steer is a phenomenon that occurs in a vehicle as a result of an imperfect arrangement of the three links previously discussed. In a car, these are the upper control arm, lower arm, and the tie rod. Bump steer can occur with the introduction of the third member (the tie rod). Before this link is added to the system, it is a simple fourbar mechanism. Once the tie rod is added, it becomes a fivebar. The only way to have free rotation when the ends of these three links are joined by solid bodies such as the uprights and the frame of the car is to coordinate the centers of rotation. If the motion path of the tie rod does not coincide with the attachment point at the upright when the wheel goes through its two-inch range of suspension movement, it will push and pull on its pins.

This pushing and pulling will result in forces that turn the wheel left and right as it moves up and down. This is called *bump* steer. The upright has two degrees of freedom (left-right rotation being the second). If rotation of the upright in the first degree of freedom is imperfect, it will translate to additional rotation in the second degree. If the front wheels were locked in toe and not allowed to turn at all, there would have to be material yielding or flexing of some nature in order for bump and droop of the suspension to take place. Essentially, all degrees of freedom of the fivebar mechanism would be removed and it would become an immobile structure. Instead, resulting compressive and tensile force generated in the tie rod pushes the upright and rotates the wheel left and right.

In order to place the steering rack and determine the appropriate lengths for the tie rods on the 2008 car, boundary conditions were first established. The locations of the tie rod mounts on the uprights had already been chosen. The geometry of the control arms was also known. FSAE constraints dictate that suspension movement must allow for one inch of bump and one inch of droop, so because the control arm lengths and positions were already selected, a motion path could be generated for the entire suspension movement.

A drawing was created in ProEngineer (and later in SolidWorks) to represent the layout of one side of the front suspension. Lines were drawn to correspond to the control arms as seen from the side. The lengths were made constant, as were the distances between the end points. An additional line was drawn to represent the upright. Next, rotated control arms and uprights were drawn at the points of maximum bump and droop. Points were created on the uprights at the locations of the tie rod mount. An arc was drawn through those three points and its radius determined the length (as seen from the front/rear of the vehicle) of the tie rod. The location of the center of the arc determined the vertical position of the steering rack and the position of the inner pivot for the tie rod. As can be seen in **Error! Reference source not found.**, the highlighted line is the side profile of the steering link as it was determined in the drawing. The inner pivot point coincides with the plane of the four inner control arm pivot points (seen as two from the side). In reality, the inner control arm pivots are not parallel with the ground for anti-dive reasons, but the variance is negligible.

It was found that for optimum bump steer prevention, the steering rack should be placed as close as possible to the tops of the lower frame rails (ideally 0.86 inches center to center). This was adhered to as much as possible. Given the diameters of the steering rack and lower frame rails, perfect elimination of bump steer was not possible with this year's car. The longitudinal rack placement had been set, so using the length of the line representing the side profile of the tie rod as determined above, a simple trigonometric relation was used to find the tie rod's true length.

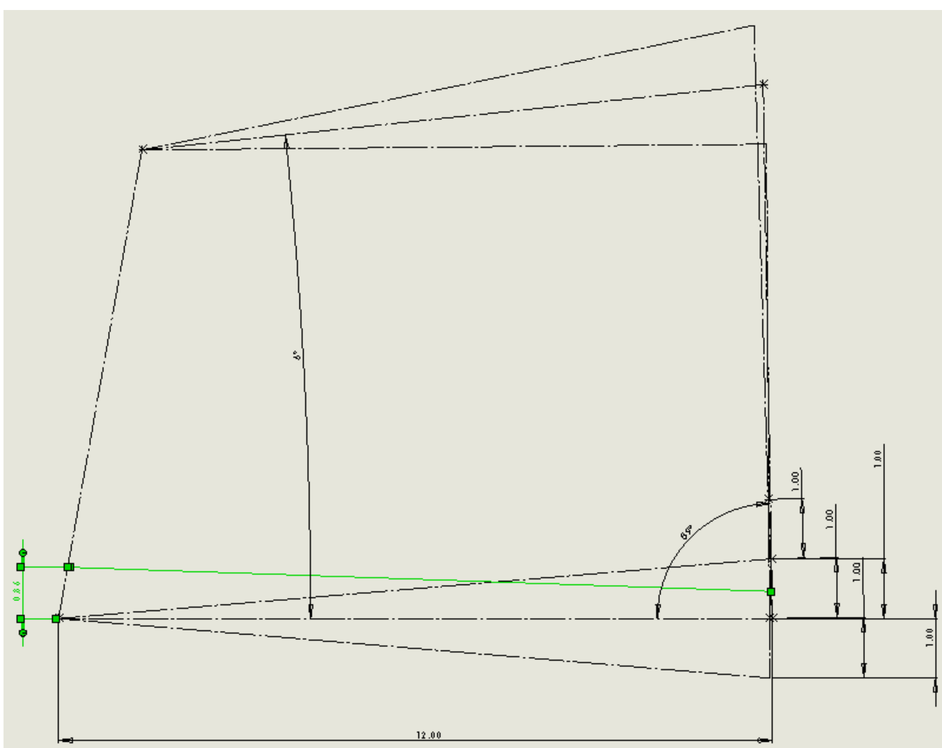

Figure 53: Calculation of vertical rack placement and tie rod length

Cockpit

The cockpit area of our car has a good amount of consideration taken in regards to ergonomics of the vehicle. The main point which shows this is how much the front of the vehicle is pulled up to put the driver in a more reclined position. This then puts the steering and suspension components all underneath the drivers legs.

The cockpit also has a few safety features most of which is required by Formula SAE rules. First we have a current up to date 5 point harness which serves to keep the driver firmly planted while driving. Next is the analysis which was done during frame design in which case many scenarios were considered involving a crash and driver protection. The cockpit area would remain fully intact in the event of any sort of crash whether that be side impact, frontal impact or rollover. Finally the driver is also protected by a firewall which is attached to the seat. This firewall is afire resistant barrier between the engine and fuel and the driver.

Finally our dashboard this year has been kept fairly simple in order to take out a lot of driver distractions. The dashboard is made of carbon fiber with a balsa core and only contains three LED warning lights as well as the necessary switches to control vehicle starting and miscellaneous functions.

Seat

One of the major changes made for the 2007-2008 FSAE racecar was the transition from an aluminum sheet metal to carbon fiber seat. Since one of the major goals of this year's vehicle was weight reduction, as many components were made from carbon fiber as possible. The seat was designed to encompass the entire width of the main roll hoop in order to ensure the 95th percentile male would have sufficient shoulder and waist room. The edges of the seat were designed to follow to flow of the frame, therefore, no addition barriers between the outside of the vehicle and driver had to be made. The back panel of the seat was designed to vertically drop approximately 8 inches from the harness bar and then angle at 12 degrees from vertical to fill the remaining space until reaching the ½ inch cross member at the base of the main roll hoop.

The fabrication process included several steps. Initially a cardboard mock up of the seat was placed in the frame to make sure all drivers would fit comfortably. Once the cardboard model was complete, a fiberglass resin mold was created over the cardboard to create a flowing mold. Finally, the carbon fiber was laid over the fiberglass mold. Inside the base of the seat was placed a 1/8th inch aluminum plate to ensure the driver would not depress the carbon fiber during entry and exit of the vehicle. Initially, only a single layer was placed on the sides of the seat and two layers for the base and back. However, do to the complex geometry of the seat; the vacuum was unable to function properly, which resulted in a fragile seat. Therefore, additional layers were wet laid with balsa inserts for the rear to ensure stability. A five point harness will be used for the 2007-2008 vehicle, therefore slots were cut in accordance with the FSAE regulations. In addition, holes were placed in the back support of the seat for the mounting of the fuel tank. A final layer of carbon fiber was placed along the edges and safety harness slots for safety and stability and a marine foam insert was formed using one of

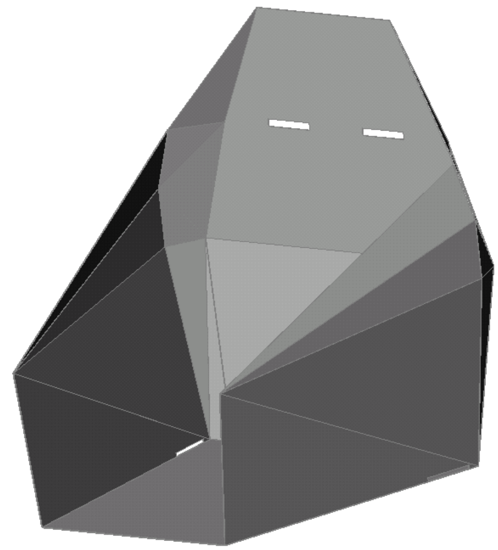

Figure 54: CAD model of seat

the team members as the model. Once the foam hardened, each member of the team sat in the foam insert to ensure ergonomics. After shaping and removing the excess, the foam was wrapped in cloth and placed in the seat. For personalization, some members of the team will require additional back padding to comfortably reach the steering wheel and pedal box. In the end, the weight savings of the carbon fiber over the traditional aluminum sheet metal was 4.73 pounds.

Floor Panel

For similar weight reduction goals as the seat, a carbon fiber floor panel was created to mount the pedal box and protect the driver from moving suspension and steering components. Since the front suspension uses pull rods and the steering rack was placed along the base rails of the frame, the ideal floor panel required a rise in the area of the driver's calves. Ergonomically, this rise meant that the driver would not have to hold his or her legs in a bent position; they would rest naturally. The fabrication of the carbon fiber floor panel was a two step process involving a Styrofoam mold. The mold was hand carved to fit the pull rods, steering damper, rocker, and steering rack underneath, while still allowing ease of entrance and exit for the driver. Several ergonomic and tolerance checks were performed once the mold was complete. Once finalized, two carbon fiber layers (with balsa inserts) were laid and vacuum bagged to reduce the amount of resin for

Figure 55: Carbon Fiber Floor Panel

rigidity. Do to the violent nature of driver foot movement, an addition 1/16th inch aluminum plate was placed under the carbon fiber floor panel in the location of the driver's heals. Once dry, holes were placed for pedal box mounting and a small rectangle was cut for adjustment of the Ohlin dampers. Once finalized, the floor panel assembly weighs less than 2 pounds.

Dashboard

Due to limited space, the 2007-2008 vehicle dashboard was designed to only contain the essential controls and indicators. The team decided to limit the size of the dashboard to the constraints of the 4130 steel steering wheel supports that extended toward the driver from the front roll hoop. For weight savings and cosmetic appeal, a carbon fiber dashboard was fabricated using a cardboard model as a size reference. For rigidity, an insert of balsa was placed in between the two layers of carbon fiber. One installed the holes and slots for the switches and indicators were cut. The switches include: Power (On/Off), Start, Launch Control Activate (On/Off), Fan Control (On/Auto), Auto Shift (On/Off), and Neutral Find. The indicators include: Neutral, Oil Pressure Warning, Water Temperature Warning, and a Shift Light.

Figure 56: Dashboard Switches and Indicators

Paddle Shifter

Due to the fact that the car is always changing and being rebuilt, it was decided that it would be useful to design an electro-pneumatic shifter linkage system as one complete module. By having one package that only requires pressure and signal inputs, the device can be mounted at different positions on different cars and still be effective.

Exhaustive research has been performed on the available solutions to this problem. Balancing product weight, cost, materials, operating temperature range, operating pressure range, and versatility a proposed final selection of parts has been identified. The breakdown of these costs, both monetary and weight, have been shown in the chart below:

Cost Analysis For Electro-Pnuematic Shifter						
					Total Cost of Device	$133.40
Bimba 02-DXP					**Total Weight of Device**	12.24
	Base Price	20.95				
	Cost per inch of stroke	1.75				
	Inches of Stroke	5				
	Cost per spring	3				
	Number of Springs	2				
	Base Weight	1.44				
	Weight per Inch	0.32				
	Cost Total		35.7			
	Weight Total		3.04			
Ingersoll Rand Miniature Solenoid Valve						
	Price	42.95				
	Number of Valves	2				
	Weight	4.1				
	Cost Total		45.9			
	Weight Total		8.2			

1/8 NPT Pressure Fitting						
	Price	2.95				
	Number of Fittings	4				
	Weight	0.25				
	Cost Total		11.8			
	Weight Total		1			

Table 2: Shifter Weight and Cost

All weights in the chart above are in ounces. The linkage incorporates two solenoid valves, one Bimba aluminum actuator piston, with two return springs to recenter the piston after it completes and actuation, 1/8" NPT gas fittings, two custom fabricated brackets and bolts. An

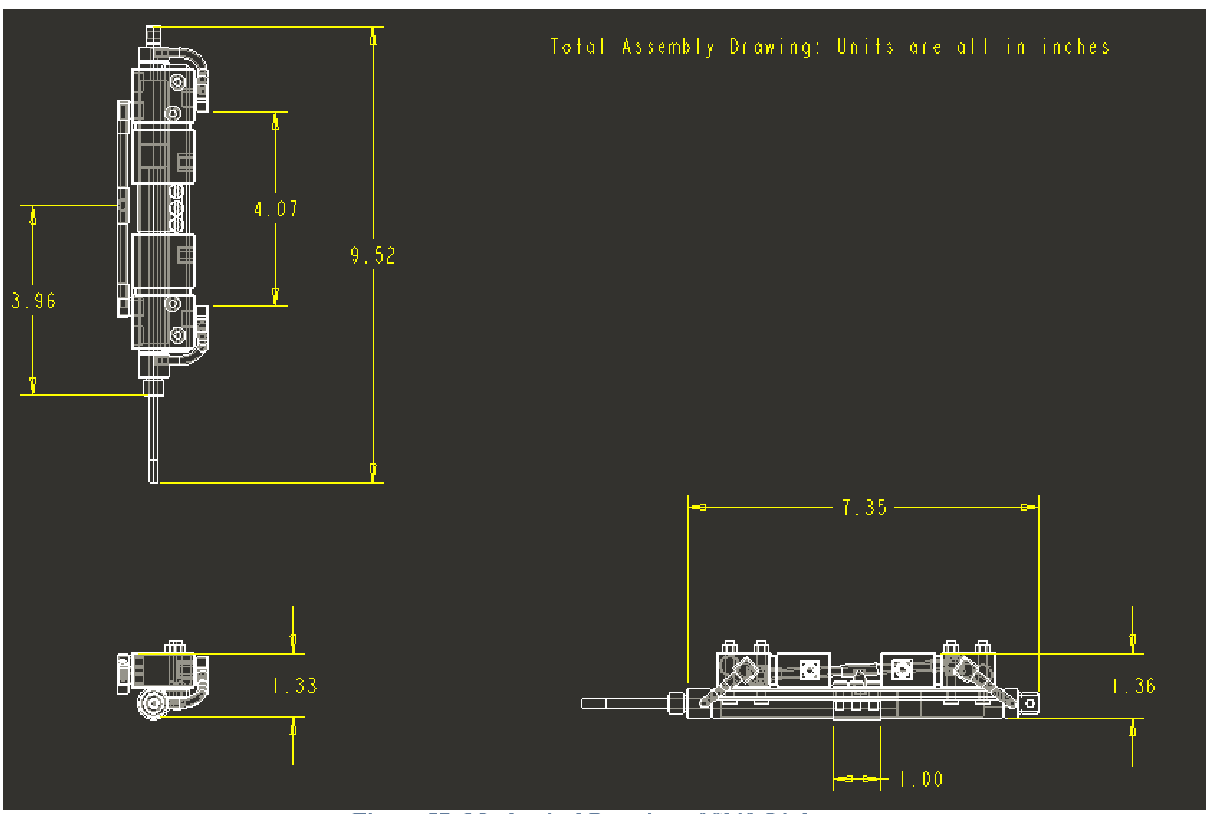

Figure 57: Mechanical Drawing of Shift Linkage

additional bracket will be required to anchor the non actuating end of the Bimba piston, but this could not be designed until the frame returns from heat treatment. This system, due to its modular design, requires only to be anchored to the shift arm on the transmission, and to one other point on the frame. This effectively creates a single four bar linkage, comprised of transmission shift arm being the output, the piston rod being the coupler, the cylinder being the crank, and the frame and transmission being the ground. Once the optimal grounding point has been identified, the resting angle of the transmission shift arm can be adjusted so the complete linkage provides and optimal transmission angle between the pneumatic cylinder and the shift arm. This will limit horizontal stresses on the pneumatic cylinder increasing its operating life and decreasing its air consumption. Below are some 3D renderings of the completed mechanism as well as a mechanical drawing used to show scale of the whole assembly and the individual parts.

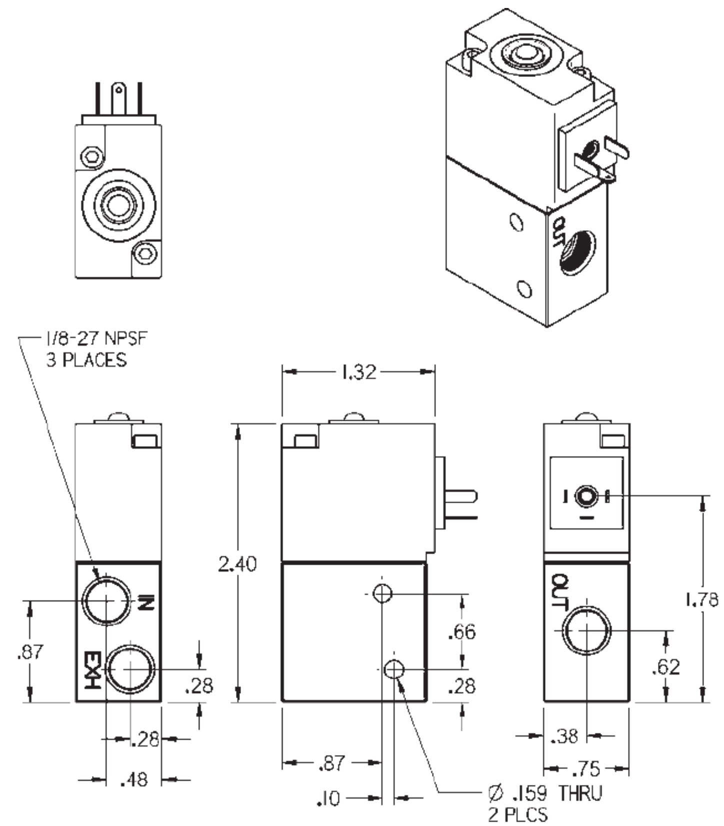

Figure 59: Mechanical Drawing of Solenoid Valve

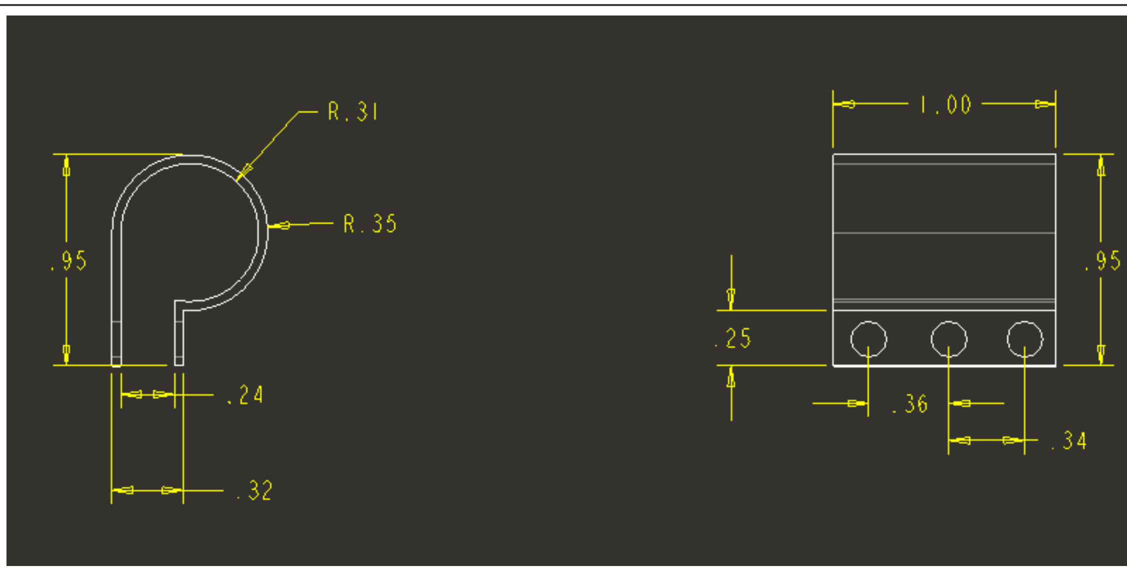

Figure 58: : Mechanical Drawing of Piston Clamp

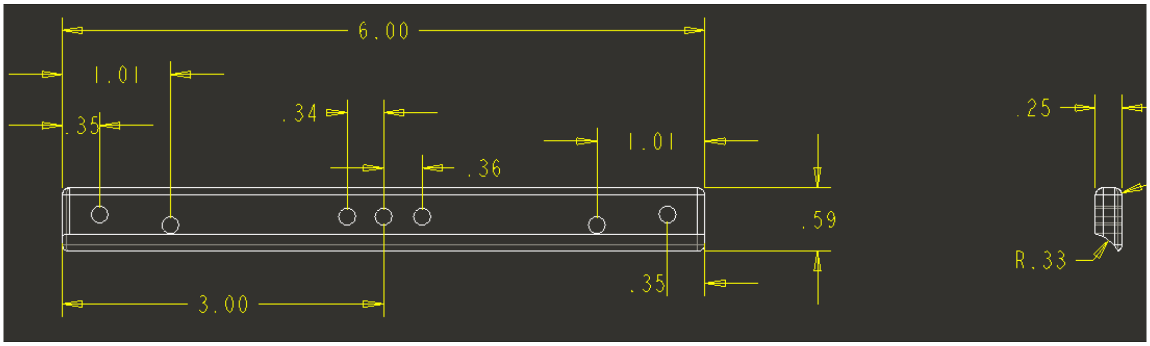

Figure 60: Mechanical Drawing of Valve Bracket

The remaining parts are completely off the shelf standard, and very small in comparison to the rest of the parts so their dimensions have not been shown. The physical control aspect of this design has been brought to the limit of my abilities to work on it without getting input, or having the frame to finish design on the mounting brackets. The electronics are still in the development process but are coming along nicely. I have logged all of my hours on this project if you want to look at them. I finished the final details on this part first because the electronics do not effect other design aspects of the vehicle, and it seemed more important to finish the actual linkage first. Please let me know how you want me to proceed from here. Also included in this email is a zip file of all Pro-E files used to produce

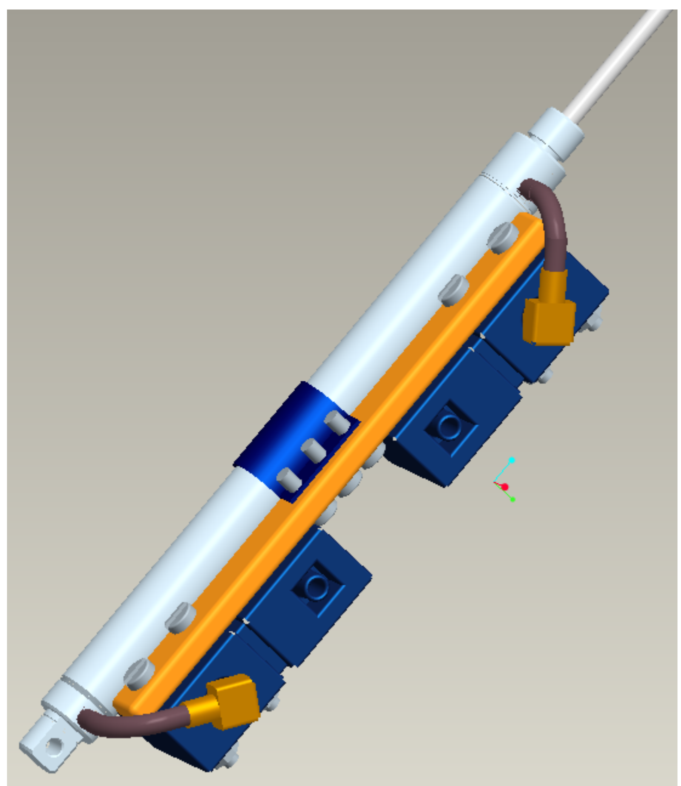

Figure 61: 3D Rendering Showing All Attachments on Bracket

these renderings, though the WPI email server has been having trouble with larger files, so it may not be.

Electropnuematic Shifter Electronics

Once the physical mechanical portion of this shifter had been designed, the electronic portion was the final major hurdle. Many considerations had to be taken into account, the most important of which was resilience. The shifter is not a major factor in the performance of the vehicle, however, if it should fail the results would be disastrous. In order to prevent this, every care has been taken to design the control circuitry to be able to endure every kind of abuse that can be thought of.

Overall Design Goals

The following task specifications were developed for the control circuitry of the shifter:

1) Solid-state circuitry design: By not employing relays or other mechanical circuitry devices, we have insured that vibration will be able to cause damage to the fewest number of components of this device.
2) Non-programmable architecture: It was also immediately determined that a micro controller would be a poor choice for the logic control in this circuit. This is due to the fact that if there was a problem with the programming, or if the programming got erased, it would be VERY difficult to reprogram the shifter during competition
3) Maximum adjustability with minimum adjustments: the team needs to be able to adjust shift time, shift travel, and shift pressure, for the up shift, and downshift. In addition to this for the neutral find mechanism, the team needs to be able to adjust downshift time, shift delay, and upshift time (which would be shifting from first gear to neutral), with the fewest number of inputs.

4) Circuitry must be robust to voltage surges, varying voltage levels, non constant activation currents, high temperature differentials, high vibration and impact forces, varying activation times, and other mechanical and electrical stresses.
5) Circuit must be easily and rapidly swappable. It is recognized that even with our best efforts, it would be impossible to predict all conditions that could damage or destroy the circuit. This is especially true for non desirable occurrences such as car to car impact, or a foreign object on the track. To prepare for these eventualities, three copies of the control circuit are going to be constructed, and will be designed to be replaced on the track.

Implementation of Task Specifications

All of the previously listed task specifications are vital, but the first and foremost is functionality under all predictable electrical stresses. If the circuit will not function under varying electrical conditions, it will not function period. In order to achieve this analog timing circuitry was implemented. Of the varying components investigated, the 555 timer was selected. In both of the timers most commonly used operational modes, monostable and astable, the timing period is determined independently of the input voltage and current. This will allow the shifter to function normally during power surges or drains that are common in the operation of vehicles electrical systems. In addition to this, both of these operational modes can be triggered through grounding the input pin, rather supplying a high voltage load. This is desirable because we do not need to supply power to the activation switches, in fact, we can run a single wire from the switch to the control circuit and simply ground the other side of the switch to the frame. In order to ensure consistent operation, two wires per switch will be utilized, to prevent the possible intermittency of frame grounding.

After electrical reliability has been addressed, the next most important task specification is mechanical robustness. In order to minimize the impact and vibration transmitted to the circuitry, a floating rubber mount was utilized, that is very similar to the antiskip mechanisms in automotive CD players. Four soft rubber grommets were mounted on the corners of the circuit, and were attached to the mounting rail using large diameter bolts with high surface area heads. The adjustment controls for the shifter have been selected as 12 turn potentiometers, with high levels of friction between the adjustment screw and the body. By using a 12 turn potentiometer, we give ourselves a high degree of control accuracy as well as minimizing the percentage error from the adjustment screw turning from vibration. Finally, by selecting all components to be rated at 200 F.

Circuit Diagrams

Up shift and Downshift circuits:

The upshift and downshift circuits were far simpler to design. The design started with a basic monostable 555 timer circuit. This circuit diagram is shown below:

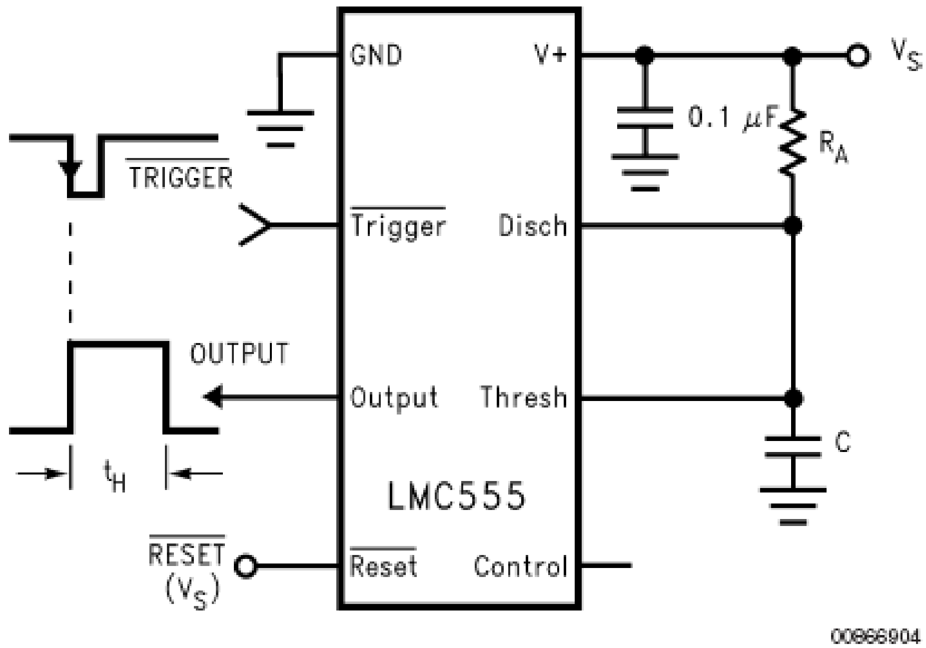

Figure 62: Upshift and Downshift circuit

This type of circuit is currently called a one shot, and is taken directly out of the 555 timer data sheet. This circuit produces and output pulse that goes from logic low to logic high for a period of time t_h, where t_h is defined by the following equation:

$$t_h = 1.1 * R_A * C$$

Where the length of the input pulse is less than 1/3 of V_S. Due to the fact that driving conditions are unpredictable, and that 12 volts is the maximum voltage available for V_s, it became apparent that it would be most desirable to design a circuit that would operate independent of the duration of time the shift button was pressed. In order to accomplish this, a capacitive coupler was utilized. This circuit is shown below:

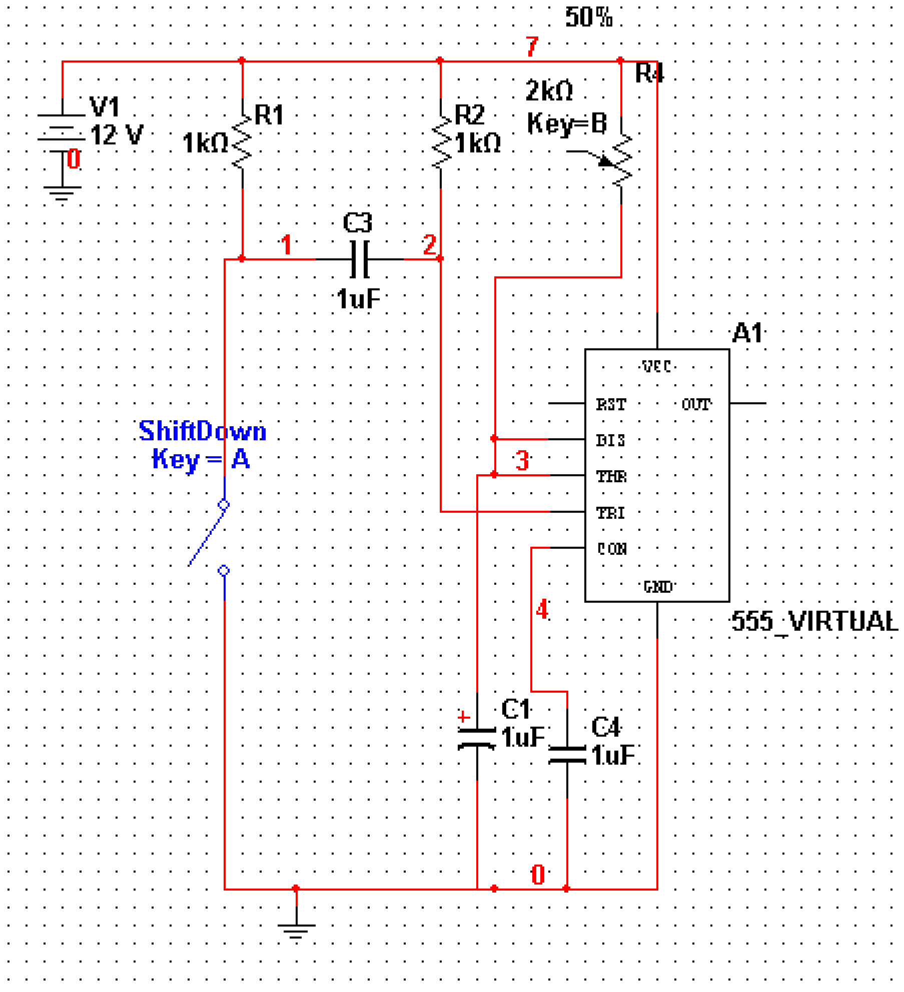

Figure 63: Capacitor Circuit

By doing this, when the side of the capacitor opposite the timer is grounded, the capacitor discharges to ground, then immediately begins to charge again. Once the capacitor recharges to V_S, the voltage on the timer side of the capacitor remains the same. By selecting a suitably small value for the coupling capacitor, C3, the monostable timing circuit can be triggered reliably with any duration button press. In the circuit diagram above, the resistor labeled Key=B refers to the potentiometer used to adjust the timing of shifter, and C1 is the timing capacitor. The next step in developing this part of the circuitry is to design the power transfer device to actually activate the solenoid valves. Since the solenoid can sink far more current than the 555 timer can source, a

MOSFET was implemented to act as a voltage controlled resistor. When the output of the timer is low, the resistance from drain to source is effectively infinite. When the gate voltage is raised high, corresponding to when the output of the timer is high, the drain to source resistance drops to nearly 0. An appropriately sized MOSFET was selected in order to meet the current drawing requirements of the solenoid. The next step to making the circuit robust was to account for capacitor drift, and the possibility of incomplete activations, and the timing capacitors getting charged accidentally during operation or off time. In order to take care of this a power on reset circuit as well as a reset switch was implemented. In this way, the timing capacitor and the entire circuit is reset to zero every time power is initially supplied to the system as well as when the reset button is pushed. This allows the team to reset the shifter device without turning the engine off, and ensures that every time the power of the vehicle is turned on, the shifter will be in the same initial state. The final iteration of the monostable shifter circuit is shown below:

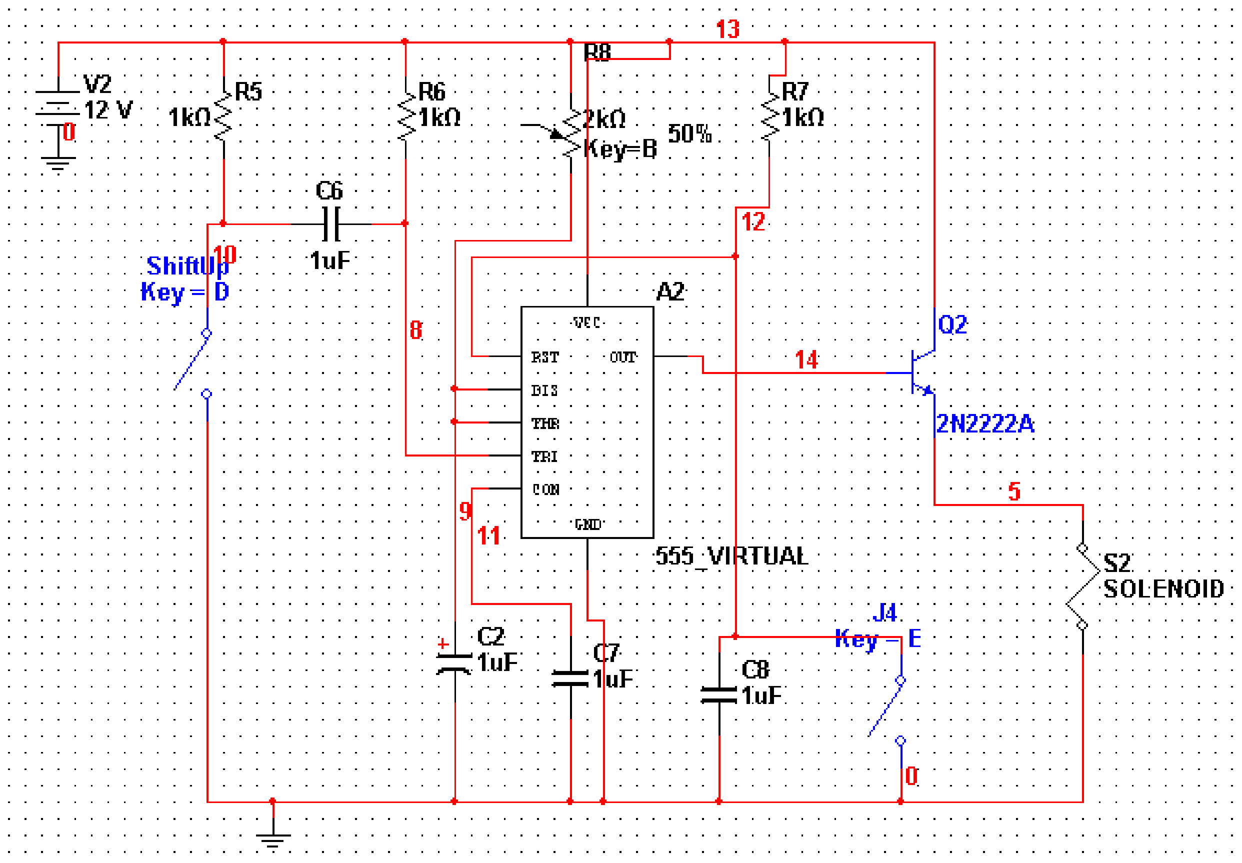

Figure 64: Reset Circuit

The switch labeled Key = E is the reset button and the capacitor C8 is the power on reset device. Final values of all of the components have not been selected, and will be demonstrated in the final report. The solenoid S2, refers to the air valve, and the device marked 2N2222A refers to the power supply MOSFET. In the final circuitry, though there will be a duplicate copy of this circuit to control the downshift, and to activate the neutral find, all three reset buttons will be combined to one momentary pushbutton. The next aspect of the circuitry that will be discussed is the neutral find circuit.

Nuetral Find Circuit

The neutral find circuit is by far the most complicated part of this circuit. This is due to the fact that we must count 5 timed downshifts, stop shifting down, then go one half shift up. In addition to this, we will only have a single activation pulse, and no kind of programmable circuit or timer. It was determined that the best way to accomplish this is through the use of another 555 monostable timer circuit, driving a 555 astable timer circuit, which in turn drives a binary divider, which will drive the solenoids through use of MOSFETS. The first step in designing this circuit is to create a timing diagram for the binary divider. This is shown below:

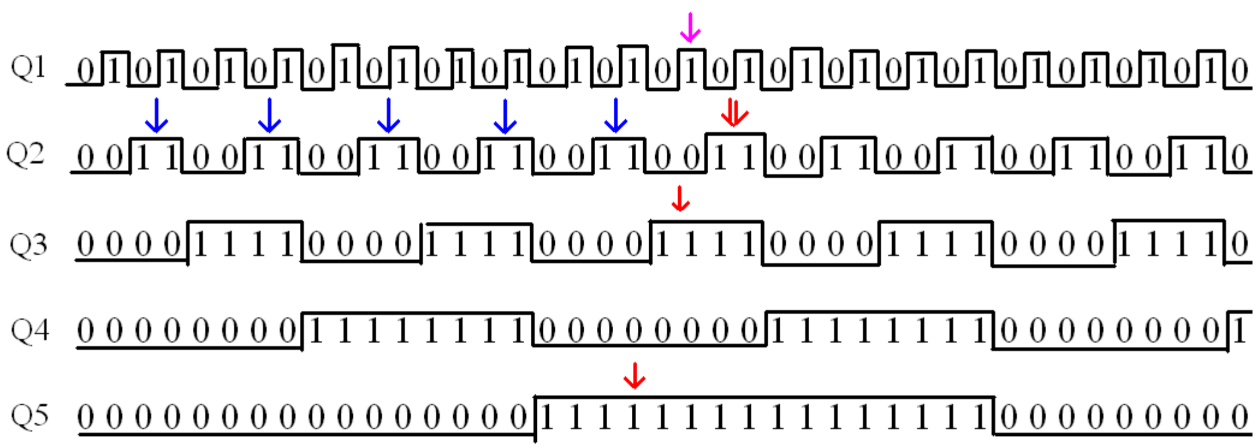

Figure 65: Neutral Find Diagram

In the above diagram, Q1-Q5 indicate 5 outputs of the binary divider. The 5 blue arrows are the 5 downshifts required to ensure that the transmission is in first gear. The two red arrows indicate logic high points that will be connected to a CMOS logic circuit in such a way that is only on, when both Q3 and Q5 are high. In this situation, Q1 will be connected to a power MOSFET that will operate the upshift solenoid. This timing is indicated by the purple arrow. Also, when Q3 and Q5 are on, Q2 will be connected to a MOSFET that will ground the reset switch on the 555 circuit, resetting it, and turning off power to the assembly. The final schematic is shown below:

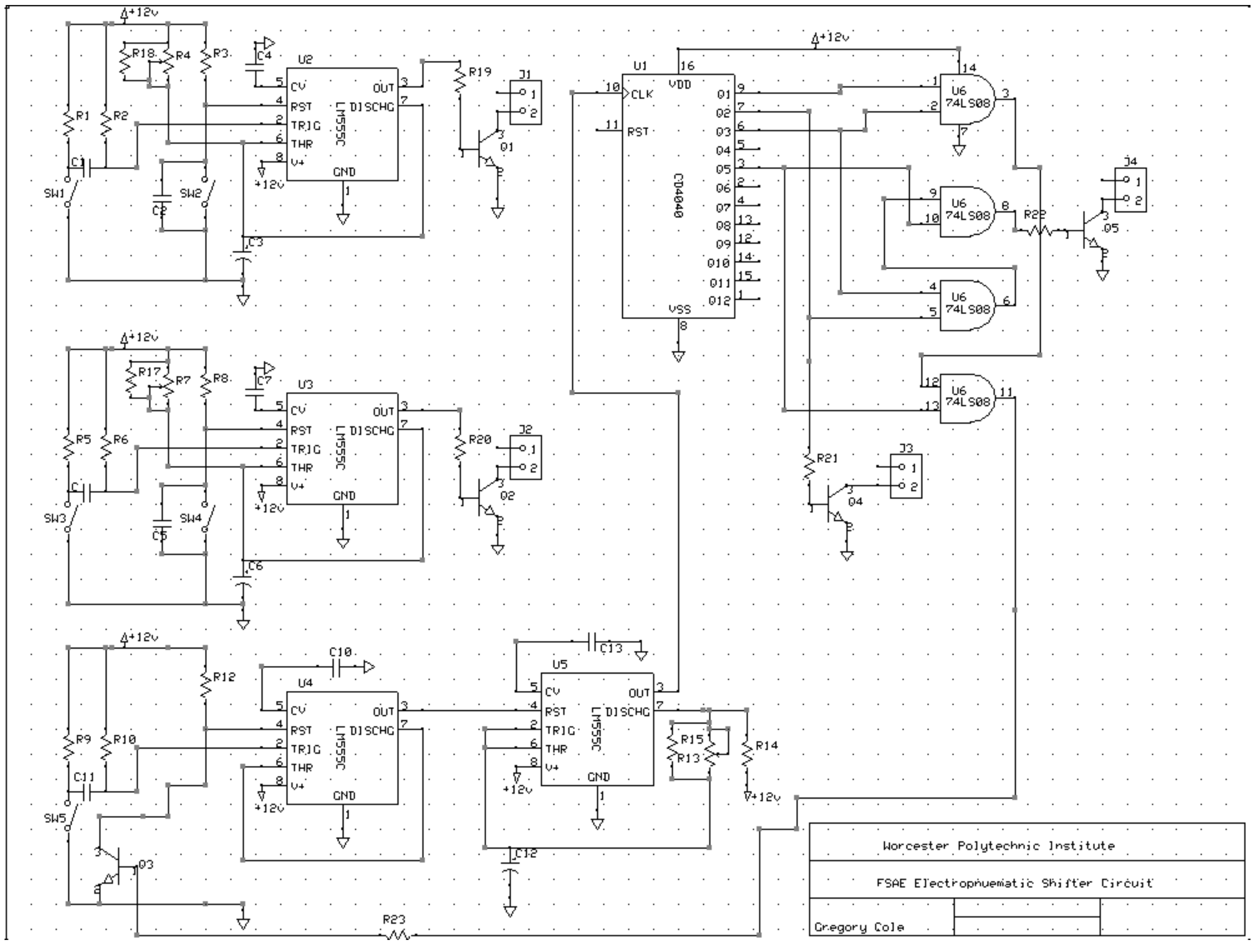

Figure 66: Final Shifter Schematic

Implementation

After the final schematic was produced and tested using a protoboard development tool, a printed circuit board, or PCB, was designed and produced. Images of the gerber files for the production of the PCB are shown below. The first image is the traces on the back panel, the second image are the traces on the front of the panel, the third image shows the component placement and the final image shows all three layers together to illustrate component interaction with the traces.

Crash Protection

The rules for frontal impact attenuator are summarized as follows:

Performance

Calculations and/or test data must show the Impact Attenuator, when mounted on the front of a vehicle with a total mass of 300 kg (661 lbs) and run into a solid, non-yielding impact barrier with a velocity of impact of 7.0 meters/second (23.0 ft/sec), would give an average deceleration of the vehicle not to exceed 20 g.

Requirements

The Impact Attenuator must be:

a. Installed forward of the Front Bulkhead.

b. At least 200 mm (7.8 in) long, with its length oriented along the fore/aft axis of the Frame.

c. At least 100 mm (3.9 in) high and 200 mm (7.8 in) wide for a minimum distance of 200 mm (7.8 in) forward of the Front Bulkhead.

d. Such that it cannot penetrate the Front Bulkhead in the event of an impact. If the Impact Attenuator is foam filled or honeycomb, a 1.5 mm (0.060 in) solid steel or 4.0 mm (0.157 in) solid aluminum metal plate must be integrated into the Impact Attenuator. The metal plate must be the same size as the Front Bulkhead and bolted or welded to the Front Bulkhead.

e. Attached securely and directly to the Front Bulkhead and not by being part of non-structural bodywork. The attachment of the Impact Attenuator must be constructed to provide an adequate load path for transverse and vertical loads in the event of off-center and off-axis impacts. If not integral with the frame, i.e.

welded, a minimum of four (4) 8 mm Grade 8.8 (5/16 inch Grade 5) bolts must attach the Impact Attenuator to the Front Bulkhead.

Crash Protection Calculations

The following information was given in the FSAE handbook as their minimum requirements.

Given:

$$Mass, m \leq 300\ kg$$

$$Acceleration, a = -20g = -190\ \frac{m}{s^2}$$

$$Gravity, g = 9.81\ \frac{m}{s^2}$$

$$Given\ Velocity, v = 7\ \frac{m}{s}$$

$$Initial\ velocity, v_o = 7\ \frac{m}{s}$$

$$Final\ velocity, v_f = 0\ \frac{m}{s}$$

$$Length, l \geq 200\ mm = 0.2\ m$$

$$Width, w \geq 200\ mm = 0.2\ m$$

$$Height, h \geq 100\ mm = 0.1\ m$$

$$Initial\ Distance, s_o = 0\ m$$

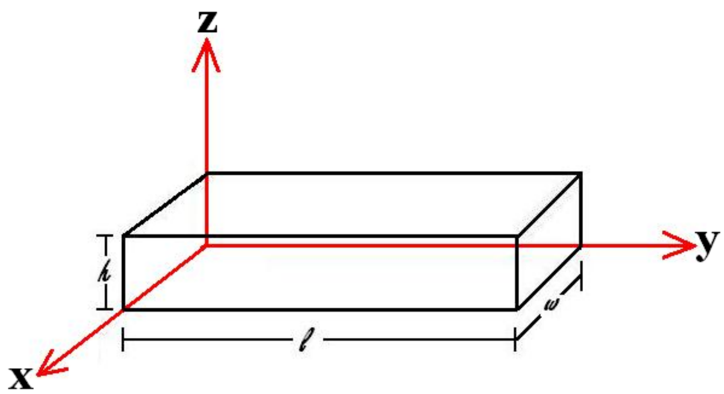

Figure 67: Dimensional direction of Impact Attenuator

Here are the calculations using the information giving above, the minimum volume of the impact attenuator, the kinetic energy produced by the car can be found.

$$Volume, V = l\ x\ w\ x\ h$$

$$= (0.2m)(0.2m)(0.1m)$$

$V = 0.004\ m^3$ ← Minimum volume requirement by rules

$$Kinetic\ Energy, KE = \frac{1}{2}mv^2$$

$$= \frac{1}{2}(300\ kg)(7\frac{m}{s})^2$$

$$KE\ =\ 7350\ Joules$$

The kinetic energy which the Impact Attenuator must be able to be absorb is 7350 Joules.

From the final velocity being zero (the velocity of the car will be zero after collision), knowing the initial velocity and the acceleration given above, the minimum impact time of the car and the minimum deformation distance of the impact attenuator can be found as the following:

Minimum Impact time:

$$v_f = v_o + at$$

$$0 = \left(7\frac{m}{s}\right) + (-190\frac{m}{s^2})t$$

$$t = \frac{7\frac{m}{s}}{190\frac{m}{s^2}}$$

$$t = 0.0368\ seconds$$

Minimum Deformation Distance:

$$s = s_o + v_o t + \frac{1}{2}at^2$$

$$= 0 + \left(7\frac{m}{s}\right)(0.0368\ s) + \frac{1}{2}(-190\ \frac{m}{s^2})(0.0368)^2$$

$$s = 0.128\ m$$

The minimum deformation distance that the Impact Attenuator need to have to stop the car at a velocity of $7\frac{m}{s}$ and a time of $0.0368\ seconds$ will be a distance of $0.128\ m$. The force that the impact attenuator must be able to absorb is found by Newton's second Law, F=ma. From knowing the mass of the car and calculating the acceleration of the car, we are able to find the force.

Force must be able to absorb:

$$F = ma$$

$$= (300\ kg)(-190\frac{m}{s^2})$$

$$F = -57{,}000\ N$$

Here it shown that the force that the Impact Attenuator needs to absorb is 57,000 N. The next step is to find the cross-sectional area of the Impact Attenuator that is able to handle the force as calculated above. To find such area, the yield strength of given material and the force that the impact attenuator must be able to absorb must be known variables.

Here is the equation to find the cross-sectional area:

Maximum Area of Given Material (of Impact Attenuator):

$$\left(Yield\ Strength\ of\ Given\ Material, [\frac{N}{m^2}]\right)(\ Area, [m^2]) = Force, [N]$$

$$\frac{(Force, [N])}{\left(Yield\ Strength\ of\ Given\ Material, [\frac{N}{m^2}]\right)} = Area, [m^2]$$

Based on the dimension of the minimum requirements of the cross-sectional area of the impact attenuator, here are the general terms and equations that need to be calculated:

Actual Acceleration, a_{actual}

Average Acceleration, $a_{average}$

Maximum velocity, v_m, the impact attenuator can take before total deformation

Yield Strength of chosen material, δ_y

Cross-sectional area of impact attenuator, $A(L)$

Numerical numbers of actual length of impact attenuator, s_1, s_2

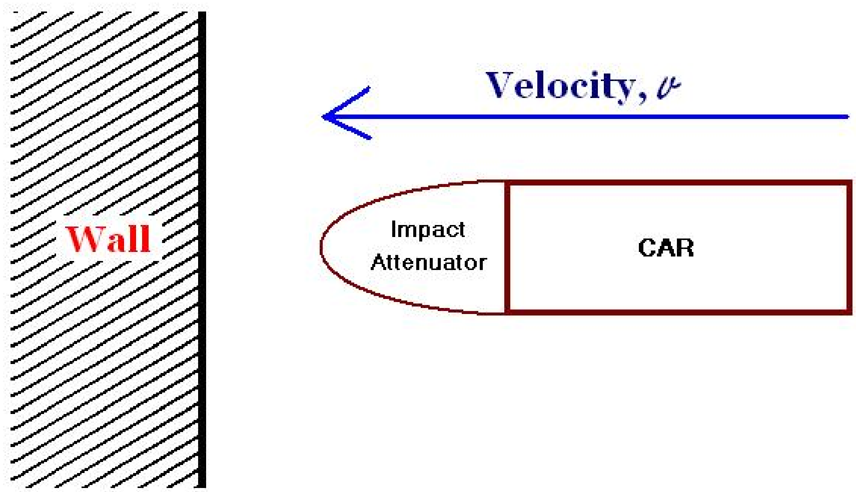

Figure 68: Direction of Car with Impact Attenuator hitting a solid wall

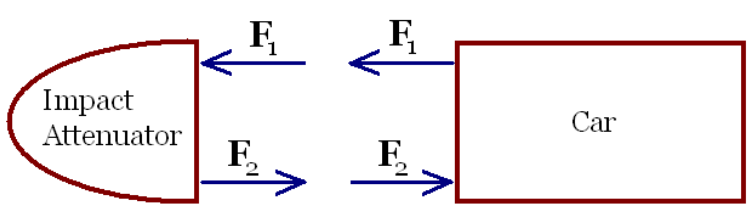

Figure 69: Equal but opposite forces between Car and Impact Attenuator

The following equations are to show the relationship of mass of car, actual acceleration of car, yield strength of given material and the cross-sectional area of the Impact Attenuator.

$$F_1 = \sigma_y * A(L)$$

$$F_2 = m * a_{actual}$$

$$F_1 = F_2$$

$$\sigma_y * A(L) = m * a_{actual}$$

From this relationship, we are able to calculate the actual acceleration the car that is going when the car is hitting the wall and compressed the Impact Attenuator.

$$a_{actual} = \frac{[\sigma_y * A(L)]}{m}$$

Note: a_{actual} is true if force on the impact attenuator is high enough to deform

$$a_{average} = \int_{s^1}^{s_2} \frac{a_{actual}}{s_2 - s_1} \, dL$$

To find maximum velocity the car can hit the wall before the impact attenuator reach maximum deformation we can use the following equation:

$$v_f{}^2 = v_o{}^2 + 2a_{average}(s_2 - s_1)$$

Where $v_f = 0$ because the car velocity will be zero after collision.

$$0 = v_o{}^2 + 2\left(\int_{s^1}^{s_2} \frac{a_{actual}}{s_2 - s_1} \, dL\right)(s_2 - s_1)$$

$$0 = v_o{}^2 - 2\int_{s^1}^{s_2} a_{actual} \, dL$$

$$v_o^{\,2} = 2\int_{s^1}^{s_2} a_{actual}\ \ dL$$

$$v_o = \sqrt{2\int_{s^1}^{s_2} a_{actual}\ \ dL}$$

$$= \sqrt{2\int_{s^1}^{s_2} \frac{\left[\sigma_y * A(L)\right]}{m}\ \ dL}$$

$$v_o = \sqrt{\frac{\left[2\sigma_y\right]}{m}} * \sqrt{\int_{s^1}^{s_2} A(L)\ \ dL}$$

Having the yield strength of material, mass of car, cross-sectional are of impact attenuator and the actual length of nose cone, we can plug it in this equation to find v_o, the maximum velocity the car can go when hitting the wall before total deformation occurs in the impact attenuator.

Note: In this equation, $a_{average}$ is the average acceleration under constant deceleration.

To find the amount of Force versus Time where human can tolerate spine-ward acceleration, we use the minimum impact time versus acceleration:

$$v_f = v_o + at$$

$$0 = \left(7\frac{m}{s}\right) + (-190\frac{m}{s^2})t$$

$$t = \frac{7\frac{m}{s}}{190\frac{m}{s^2}}$$

$$t = 0.0368\ seconds$$

So we find the time by using:

$$(20g\ , t)$$

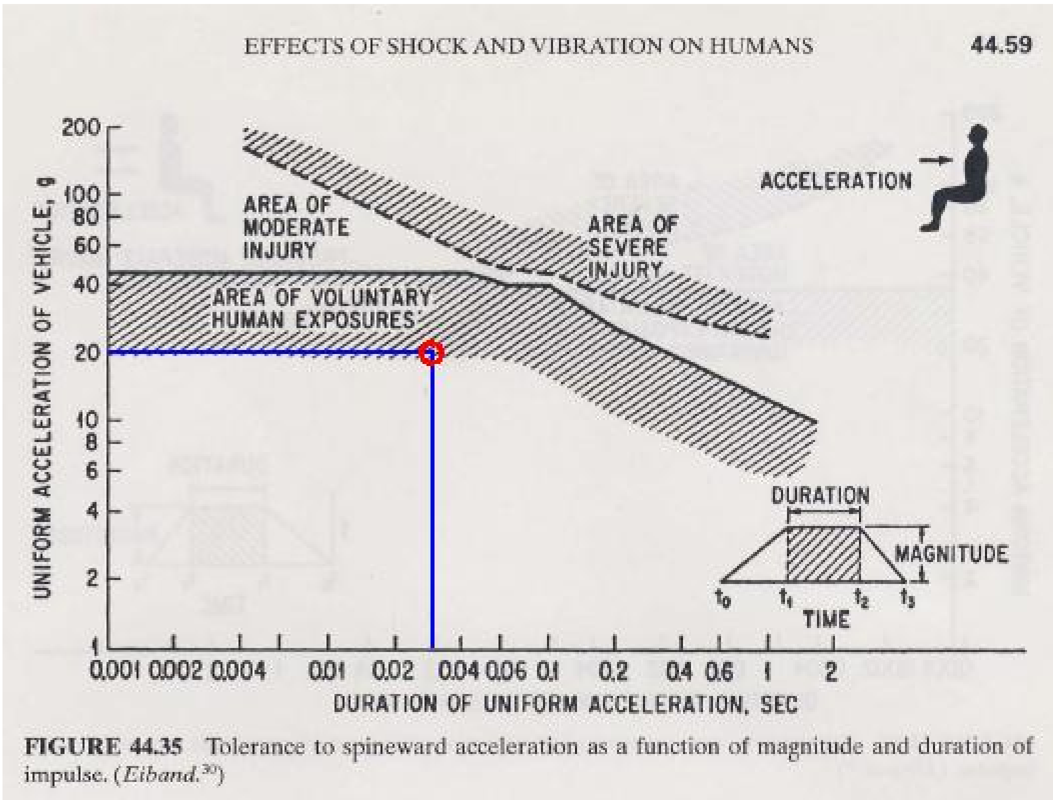

Figure 70: Graph of Effects of Shock and Vibration on Humans

As we can see from the graph above, with acceleration of 20g in 0.0368 seconds, the impact force will be in the Area of Voluntary Human Exposures zone where, in general, it is still a safety zone for human being. Often, the results of damage do not depend on the force itself but rather it is greatly affected by the time. Since the time is less than half a second, the human body will still be safe under this impact force.

Therefore, with yield strength of chosen material,σ_y , and the mass of car, m. The actual acceleration, a_{actual} , of the car can be calculated by:

$$F_1 = \sigma_y * A(L)$$

$$F_2 = m * a_{actual}$$

$$F_1 = F_2$$

$$\sigma_y * A(L) = m * a_{actual}$$

$$a_{actual} = \frac{[\sigma_y * A(L)]}{m}$$

In our case, we will be using West Marine Foam is the impact absorb material. This material is low viscosity, flame-resistant, closed-cell liquid polyurethane foam. At a nominal 1.8 pounds per cubic foot density, it produces a high yield foam while maintain an excellent physical properties. The manufacturer's given compressive strength, σ_y, for this material is 35 psi.

From knowing the compressive strength of the material, the mass of the car and the cross-sectional area of the impact attenuator, we are able to find the actual acceleration of the car.

$$\sigma_y = 35psi = 241{,}316.495\ \frac{N}{m^2}$$

$$m = 661\ lbs = 300\ kg$$

$$A = 0.0927\ m^2$$

$$a_{actual} = \frac{\left[(241{,}316.495\frac{N}{m^2}) * (0.0927\ m^2)\right]}{300\ kg}$$

$$a_{actual} \quad = 74.567 \frac{m}{s^2}$$

From knowinga_{actual}, the average acceleration of the car, $a_{average}$, can be calculated by the following equation:

$$a_{average} = \int_{s^1}^{s_2} \frac{a_{actual}}{s_2 - s_1} \; dL$$

Where $s_1$and s_2 is the length of the impact attenuator and since a_{actual} is a constant, therefore $a_{average}$ is defined as following:

$$a_{average} = 74.567 \frac{m}{s^2}$$

From knowing average acceleration, $a_{average}$, the maximum velocity the car can hit the wall before the impact attenuator will reach total deformation, v_m, can be obtained:

$${v_f}^2 = \; {v_o}^2 + 2a_{average}(s_2 - s_1)$$

$$0 \; = \; {v_o}^2 + 2(74.56 \frac{m}{s^2})(0.8128 \, m)$$

$$v_o = \; \sqrt{2(74.56 \frac{m}{s^2})(0.8128 \, m)}$$

$$v_o = \; 7.781 \frac{m}{s}$$

Note: All of these calculations are based on when the impact attenuator are being compressed directly from y-axis

Analysis when Impact Attenuator is being hit at an off-axis angle

When the Impact Attenuator is being hit at an off-axis angle, the total amount of force will be divided into two different directions. Here is shown the impact attenuator is being hit/compressed off-axis of 15° angle

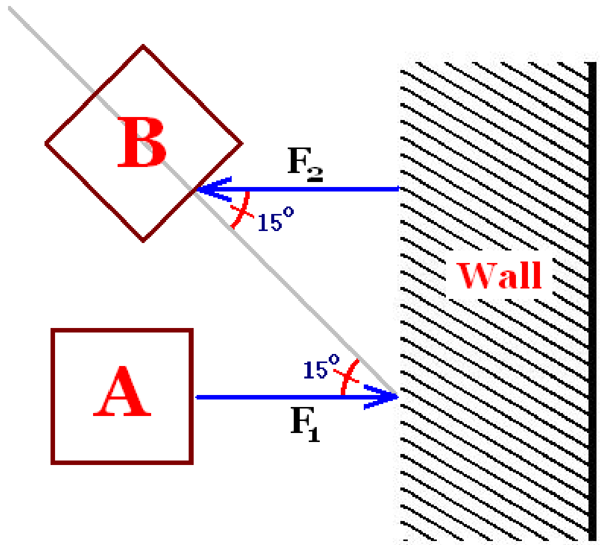

Figure 71: Free Body Diagram showing off axis impact

The force on impact attenuator B can be distributed as follow:

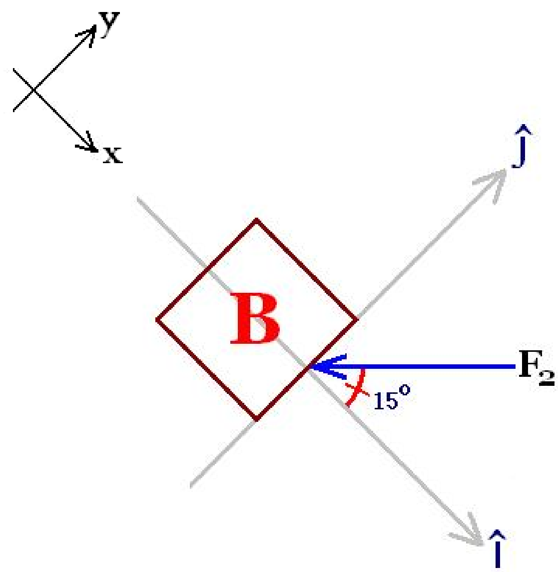

Figure 72: Free Body Diagram showing Force distribution on Impact Attenuator B

Here shown the total force from the impact:

$$F = ma$$

$$= (300\ kg)(-190\frac{m}{s^2})$$

$$F = -57{,}000\ N$$

Here shown the total force from the impact being distributed into two different direction from being hit at an off axis of 15° angle indicated by F_y and F_x.

$$F_y = Fsin15°$$

$$= (-57{,}000N)sin15°$$

$$F_y = -14{,}752.686\ N$$

$$F_x = F cos 15°$$

$$= (-57{,}000 N) cos 15°$$

$$F_x = -55{,}057.772\ N$$

The negative signs only indicate direction of the force. Since the sign is negative, that means the force is compressive.

Analysis of Forces $\boldsymbol{F_y}$ and $\boldsymbol{F_x}$

Force acting from ĵ-direction, $\boldsymbol{F_y}$:

Force $F_y = 14{,}752.686\ N$ will be hitting at cross-sectional area of

$$A_{Fy}(L) = x * y$$

$$= (13\ in) * (14 in)$$

$$= 182\ in^2$$

$$A_{Fy}(L) = 0.117\ m^2$$

Maximum area it needs to have in order to absorb that force:

Since the force that the impact attenuator needs to absorb in this direction will be shear force, the yield strength that we are using here will be the shear strength. The manufacturer's given shear strength, δ_s, for this material is 35 psi.

$$\sigma_s = 33\text{psi} = 227{,}526.981\ \frac{N}{m^2}$$

$$A_{Fy} = \frac{F_y}{\sigma_s *}$$

$$= \frac{14{,}752.686\ N}{227{,}526.981\ \frac{N}{m^2} *}$$

$$A_{Fy} = 0.06484\ m^2$$

***Note:** From the Free Body Diagram of Impact Attenuator B, the force is acting at ĵ-direction, perpendicular from î-direction, therefore the yield strength that is being apply is the Shear Strength, σ_s

Finding the maximum velocity to cause total deformation of the impact attenuator:

$$A_{Fy}(L) = 0.117\ m^2$$

$$\sigma_s = 33\text{psi} = 227{,}526.981\ \frac{N}{m^2}$$

$$m = 661\ lbs = 300\ kg$$

$$a_{actual(Fy)} = \frac{[\sigma_s * A_{Fy}(L)]}{m}$$

$$= \frac{\left[\left(227{,}526.981\ \frac{N}{m^2}\right) * (0.117\ \ m^2)\right]}{300\ kg}$$

$$a_{actual(Fy)} = 88.736 \frac{m}{s^2}$$

$$a_{average(Fy)} = a_{actual(Fy)}$$

$$a_{average(Fy)} = 88.736 \frac{m}{s^2}$$

$$v_{f(Fy)}{}^2 = v_{o(Fy)}{}^2 + 2a_{average(Fy)}(s_2 - s_1)$$

$$0 = v_{o(Fy)}{}^2 + 2a_{average(Fy)}(s_2)$$

$$v_{o(Fy)}{}^2 = 2a_{average(Fy)}(s_2)$$

$$v_{o(Fy)} = \sqrt{2a_{average(Fy)}(s_2)}$$

$$v_{o(Fy)} = 12.010\frac{m}{s}$$

Force acting from î-direction, F_x:

Force $F_x = 55{,}057.772\ N$ will be hitting at cross-sectional area of

$$A_{Fx}(L) = z * x$$

$$= (0.008\ m) * (0.009m)$$

$$A_{Fx}(L) = 0.02\ m^2$$

Maximum area it needs to have in order to absorb that force:

$$\sigma_y = 35\text{psi} = 227{,}526.981\ \frac{N}{m^2}$$

$$A_{Fx} = \frac{F_x}{\sigma_y{}^+}$$

$$= \frac{55{,}057.772N}{241{,}316.495\ \frac{N}{m^2}{}^+}$$

$$A_{Fx} = 0.228\ m^2$$

Note: From the Free Body Diagram of Impact Attenuator B, the force F_x is acting at î-direction, absorbing the compressive force, therefore the yield strength that is being apply is the Compressive Strength, σ_y

Finding the maximum velocity to cause total deformation of the impact attenuator:

To find the maximum velocity that will cause a total deformation of the impact attenuator, we need to use the relationship of the cross-sectional area of the impact attenuator, the yield strength of given material, the mass of car and the actual acceleration of the car when the impact occurs.

$$A_{Fx}(L) = (13\ in)(14\ in) = 182\ in^2 = 0.117\ m^2$$

$$\sigma_y = 35\text{psi} = 227{,}526.981\ \frac{N}{m^2}$$

$$m = 661\ lbs = 300\ kg$$

$$a_{actual(Fx)} \quad = \frac{\left[\sigma_y * A_{Fx}(L)\right]}{m}$$

$$= \frac{\left[\left(241{,}316.495\ \frac{N}{m^2}\right) * (0.117\ m^2)\right]}{300\ kg}$$

$$a_{actual(Fx)} \quad = 94.450\frac{m}{s^2}$$

Since the actual acceleration of the car is a constant, the average acceleration will be same as the actual acceleration of the car.

$$a_{average(Fx)} \quad = 94.450\frac{m}{s^2}$$

Now, we are able to find the velocity of total deformation of the impact attenuator when the impact occurs:

$$v_{f(Fx)}{}^2 = v_{o(Fx)}{}^2 + 2a_{average(Fx)}(s_2 - s_1)$$

$$0 = v_{o(Fx)}{}^2 + 2a_{average(Fx)}(s_2)$$

$$v_{o(Fx)}{}^2 = 2a_{average(Fx)}(s_2)$$

$$v_{o(Fx)} = \sqrt{2a_{average(Fx)}(s_2)}$$

$$= \sqrt{2(94.450\frac{m}{s^2})(0.117\ m^2)}$$

$$v_{o(Fx)} = 4.558\frac{m}{s}$$

Results from *Analysis of Forces F_y and F_x*

Force directly from frontal impact (on y-axis)

$$F = -57{,}000\ N$$

$$v_o = 7.781\frac{m}{s}$$

Force from off-axis at 15° angle (distributed to ĵ-axis and î-axis)

ĵ-axis: $F_y = -14{,}752.686\ N$

$v_{o(Fy)} = 4.926\frac{m}{s}$

î-axis: $F_x = -55{,}057.772\ N$

$$v_{o(Fx)} = 4.558\frac{m}{s}$$

As long as $v_{o(Fy)}$ and $v_{o(Fx)}$ are equal or greater then v_o then the Impact Attenuator will be able to absorb the force. Because $v_{o(Fy)}$ and $v_{o(Fx)}$ need to have a greater in order to make a damage on the Impact Attenuator.

Study of Deformation of West Marine Foam

Test of deformation were performed on the West marine Foam. The test was being performed on a smaller scale of West Marine Foam. Then a solid steel bar were being dropped, free fall, onto the West Marine Foam. When the solid steel bar compressed the foam, the depths of deformation were collected. Here are the following data:

Initial distance between foam and bar, $h_o = 27\ inches$

Cube Dimension, $D_{cube} = 4\ in^3$

Steel Rod:

Radius, $r_{rod} = 1\ in = 0.0254\ m$

Length, $l_{rod} = 15.5\ in = 0.381\ m$

Mass, $m_{rod} = 16.135\ lbs = 7.319\ kg$

Three drops were performed and the depths that the steel rod made in the foam were collected:

$$s_1 = 2.122\ inches = 0.0539\ m$$

$$s_2 = 2.053\ inches = 0.0521\ m$$

$$s_3 = 2.125\ inches = 0.0540\ m$$

Area of Steel Rod:

$$A_{steel} = \pi \frac{1}{4} {d_{rod}}^2$$

$$A_{steel} = \pi \frac{1}{4} (0.0508\ m)^2$$

$$A_{steel} = 0.002\ m$$

Experiment:

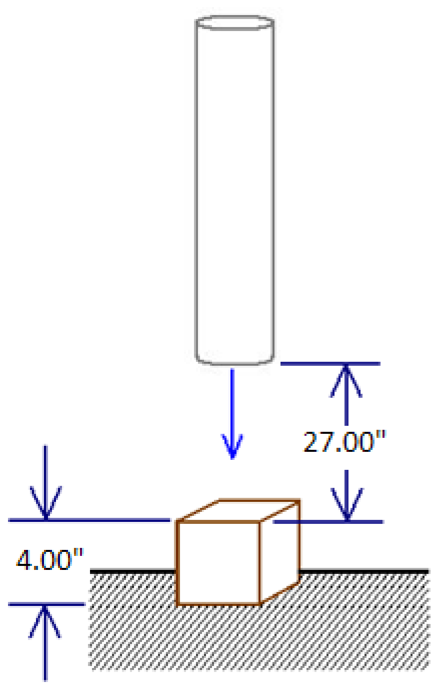

Figure 73: Diagram showing how the Impact Test was performed

$m_{rod} = 16.135\ lbs = 7.319\ kg$

$g = 9.81\frac{m}{s^2}$

$h_o = 27.00\ inches = 0.6858\ m$

Find time of impact:

$$h_o = \frac{1}{2}gt^2$$

$$t = \sqrt{\frac{2h_o}{g}}$$

$$t = \sqrt{\frac{2(0.6858\ \ m)}{(9.81\frac{m}{s^2})}}$$

$$t = 0.3739\ seconds$$

Initial velocity before collision:

$$v_o = gt$$

$$v_o = \left(9.81\frac{m}{s^2}\right)(0.3739\ \ s)$$

$$v_o = 3.668\ \frac{m}{s}$$

Deformation One:

For test results of Deformation One, here are the following data and results.

$$s_1 = 2.122\ inches = 0.0539\ m$$

$$v_f{}^2 - v_o{}^2 = +2a_1 s_1$$

$$0 - v_o{}^2 = +2a_1 s_1$$

$$a_1 = \frac{2gh_o}{2s_1}$$

$$a_1 = \frac{2\left(9.81\frac{m}{s^2}\right)(0.6858\ \ m)}{2(0.0539\ m)}$$

$$a_1 = 124.8180\frac{m}{s^2}$$

Compressive Force, F_{c1}

$$F_{c1} = a_1 m_{rod}$$

$$F_{c1} = \left(129.130\frac{m}{s^2}\right)(7.319\ kg)$$

$$F_{c1} = 945.102\ N$$

For Deformation Two:

For test results of Deformation Two, here are the following data and results.

$$s_2 = 2.053\ inches = 0.0521\ m$$

$$v_f{}^2 - v_o{}^2 = +2a_2 s_2$$

$$0 - v_o{}^2 = +2a_2 s_2$$

$$a_2 = \frac{2gh_o}{2s_2}$$

$$a_2 = \frac{2\left(9.81\frac{m}{s^2}\right)(0.6858\ \ m)}{2(0.0521\ m)}$$

$$a_2 = 129.130\frac{m}{s^2}$$

Compressive Force, F_{c2}

$$F_{c2} = a_2 m_{rod}$$

$$F_{c2} = \left(129.130\frac{m}{s^2}\right)(7.319\ kg)$$

$$F_{c2} = 945.102\ N$$

For Deformation Three:

For test results of Deformation Three, here are the following data and results.

$$s_3 = 2.125\ inches = 0.0540\ m$$

$$v_f^{\ 2} - v_o^{\ 2} = +2a_3 s_3$$

$$0 - v_o^{\ 2} = +2a_3 s_3$$

$$a_3 = \frac{2gh_o}{2s_3}$$

$$a_3 = \frac{2\left(9.81\frac{m}{s^2}\right)(0.6858\ \ m)}{2(0.0540\ m)}$$

$$a_3 = 124.587\frac{m}{s^2}$$

Compressive Force, F_{c3}

$$F_{c3} = a_3 m_{rod}$$

$$F_{c3} = \left(124.587\frac{m}{s^2}\right)(7.319\ kg)$$

$$F_{c3} = 911.852\ N$$

List of Materials:

One aluminum plate

Dimensions: 0.004m x 0.305m x 0.305m

Two aluminum straps

Dimensions: 0.0254m x 0.00008m x 0.381m

Dimensions: 0.0254m x 0.00008m x 0.305m

Four 8mm Grade 8.8 (5/16 inch Grade 5) bolts

Eight rivets

2oz of West Marine Foam

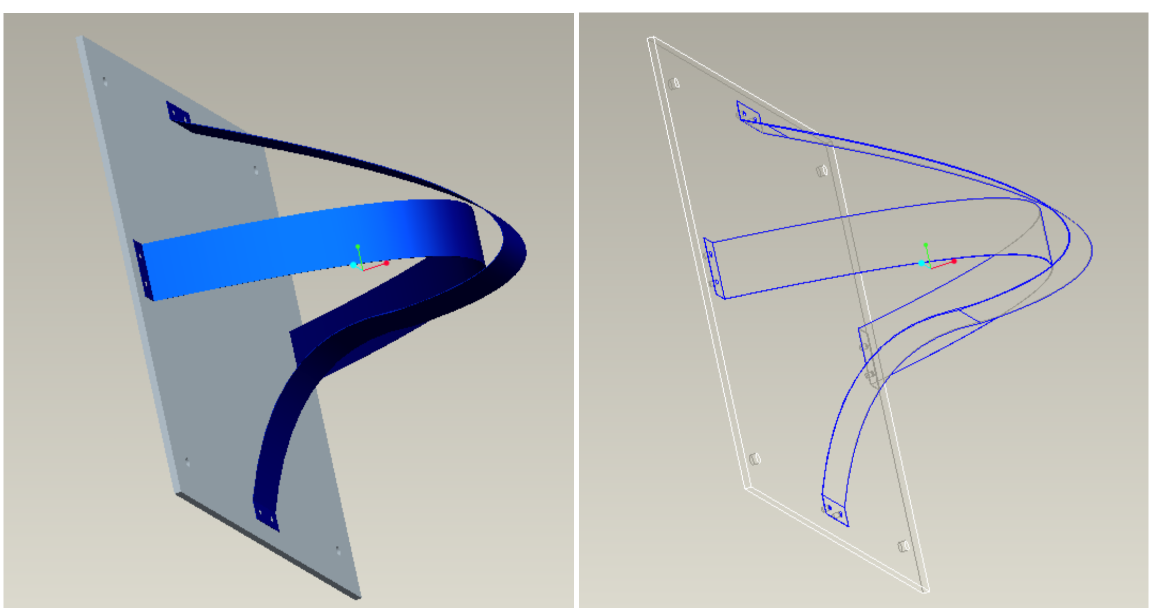

Figure 74: Pro-Engineering CAD model on Impact Attenuator internal structure

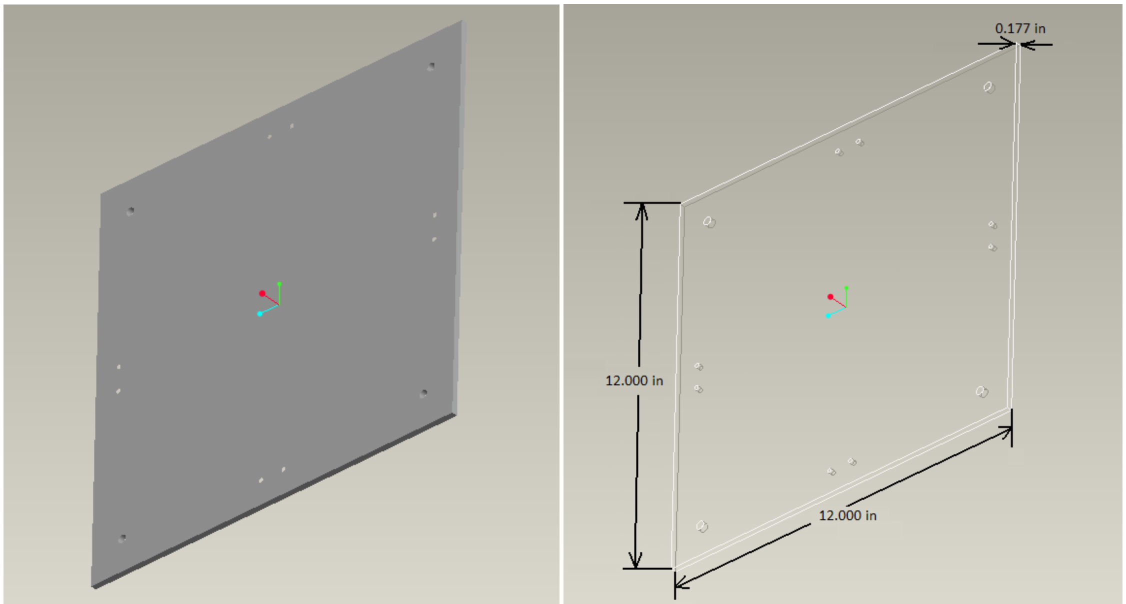

Figure 75: Pro-Engineering CAD model on Aluminum plate showing its dimensions

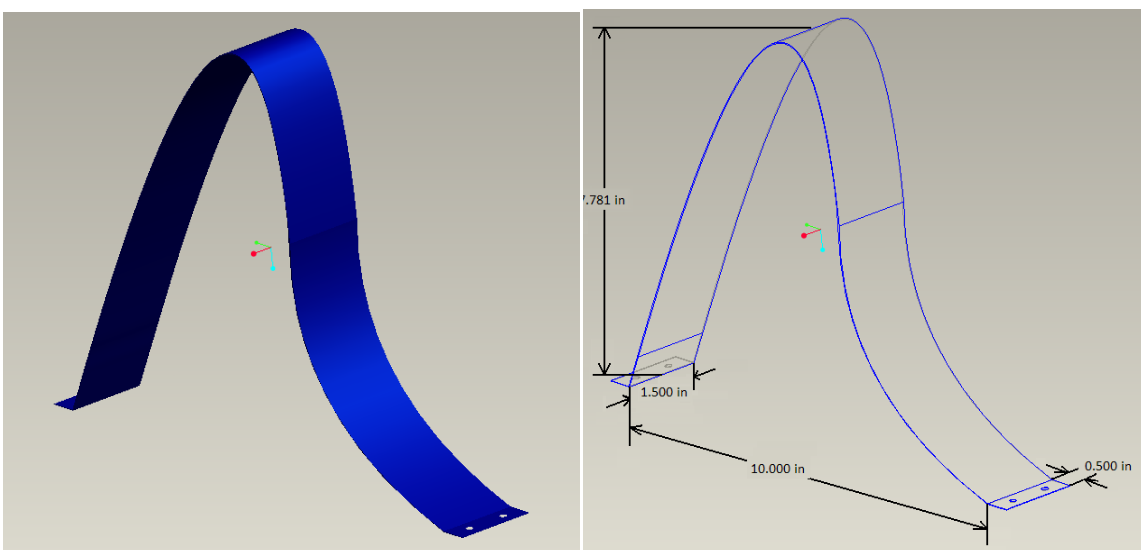

Figure 76: Pro-Engineering CAD model on Aluminum Strap 1 showing its dimensions

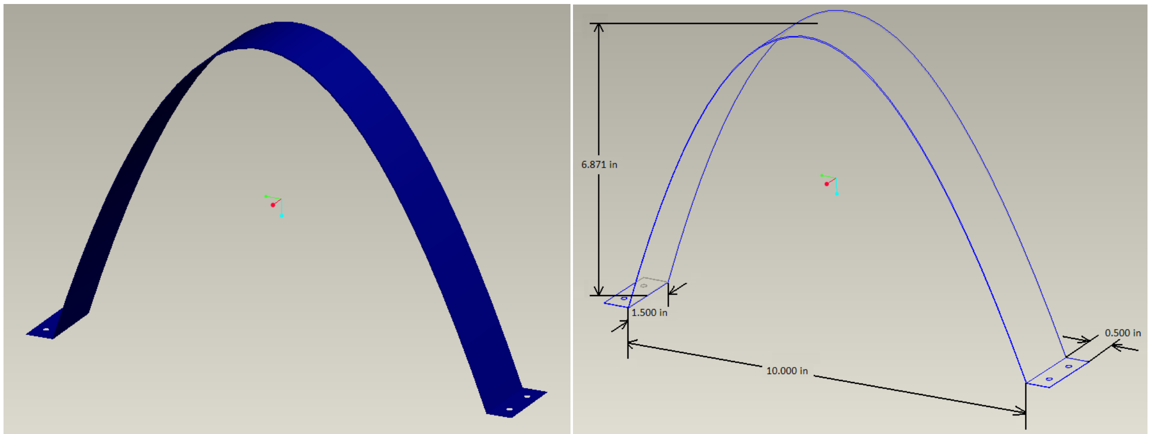

Figure 77: Pro-Engineering CAD model on Aluminum Strap 2 showing its dimensions

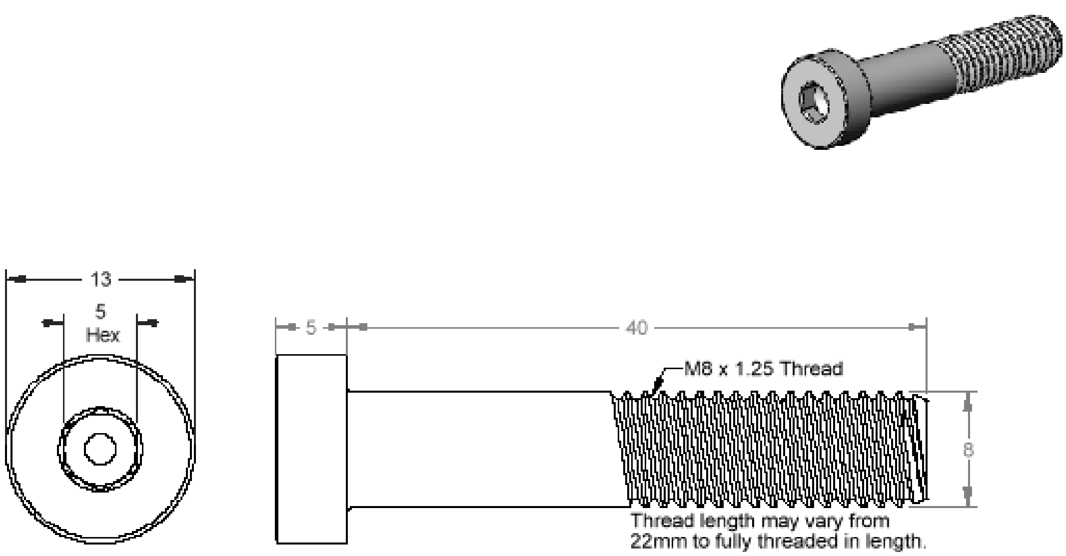

Figure 78: CAD model on 8mm Grade 8.8 showing its dimensions

Figure 79: Photo of rivet being used on the Impact Attenuator structure

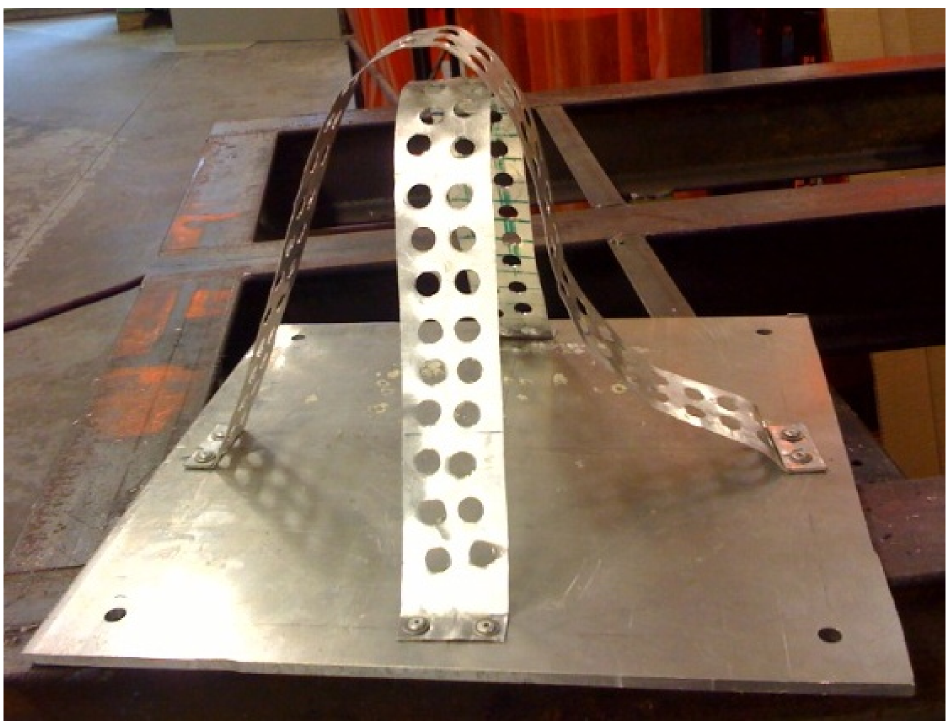

Figure 80: Photo of Impact Attenuator structure

Braking System

We wanted to make our brake system this year as light as possible while maintaining high performance. We noticed that in years past many of the braking systems have been over designed and hence are heavier than necessary. The braking force required is actually fairly low for a formula SAE vehicle so we looked into the smallest brakes possible. We originally considered mountain bike rotors and decided not to go further based on the fact that they could not deal with the heat requirements we would face. We settled with an ATV brake system.

Also in the past many of the cars have run a single differential mounted rear brake. However, this depends on the differential for even distribution and can lead to negative handling characteristics entering corners. Also a single rear brake requires a fairly large rotor and caliper. After considering all of this we decided to run outboard brakes and hence have 4 wheel 7.75 inch disc brakes all around the car.

The majority of the brake system components for this years car was sourced from Polaris ATVs. This was done for simplicity, time and budget reasons. Given more time it would be ideal to manufacture our own rotors. While the Polaris rotors are cheap due to their discount they still impact the cost report due to their retail cost. The calipers also have a high retail cost and are especially expensive due to our 4 wheel braking system this year.

Pedal Box

The 2007-2008 pedal box was a major reduction in size and weight over previous years' models. The kinematics of the pedal box system follows a slider crank motion with several linkages. To accompany varying sized drivers, the pedal box was also designed for adjustment to the front and rear of about 3 inches. Overall this years pedal assembly weighs 4.6 pounds compared to the over 12 pound weight of last years pedal assembly.

Several design iterations were performed for the kinematic setup of each pedal. The three pedal consists of the traditional brake, accelerator, and clutch. This year, smaller and lighter master cylinders were used from a Polaris ATV; enabling the pedal box to be adjustable up to the front bulkhead. The brake and accelerator were designed, using Working Model, to rest at approximately 9 degrees from vertical towards the front of the vehicle for ergonomics. The clutch rests at a greater angle of about 25 degrees from vertical to ensure the driver does not engage the clutch accidentally. For the brake, the master cylinders were placed 2.5 inches above the base plate and are designed to compress 0.75 inches for complete brake lock through 24 degrees of brake pedal rotation. All three pedals rotate about a 0.5 inch 4130 steel tube placed 0.75 inches to center above the base plate. The master cylinders are mounted on two individual 0.5 inch thick 6061 aluminum rectangles measured to height and length and drill pressed for weight reduction. The pivot point of the master cylinder bias bar is 1.6 inches vertically and 0.6 inches aft of the 4130 steel pivot bar center for perfect parallel movement to compression of the master cylinder slider setup. The overall brake pedal height is approximately 7 inches with a 3 inch perpendicular extension for foot placement.

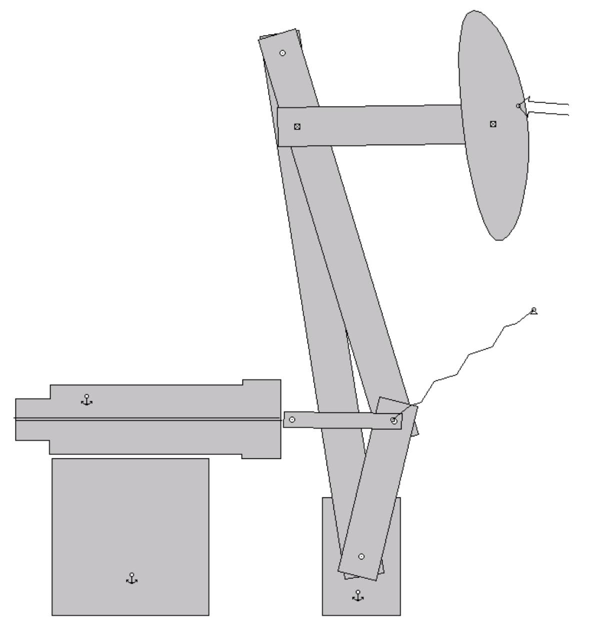

Figure 81: Working Model Kinematic Brake Pedal

The clutch and accelerator are each equipped with two, 3 inch long, 1/16th inch thick aluminum sheet metal linkages that connect the pedals to the cable engaging mechanisms. The clutch and accelerator cables mount vertically down into one inch cubic aluminum blocks that extend vertically 2.5 inches from the base plate. The cables can be tensioned by unscrewing the respective set screw from the aluminum blocks. In order to completely open the throttle body, the accelerator cable was required to displace 1.17 inches. Therefore, for ergonomics and throttle sensitivity, the accelerator was designed to travel through 30 degrees to rotation before completely opening the throttle. The clutch was setup for similar kinematic motion, except was placed at a greater degree of rest from vertical. The clutch and throttle cables each mount through a 0.5 inch OD 4130 tube, 1 inch in length that rests in the slots of the aluminum blocks. Through pedal movement, the aluminum sheet metal tabs force the 4130 tube down the slotted aluminum blocks for cable depression, and torsion springs return the pedals to the resting positions.

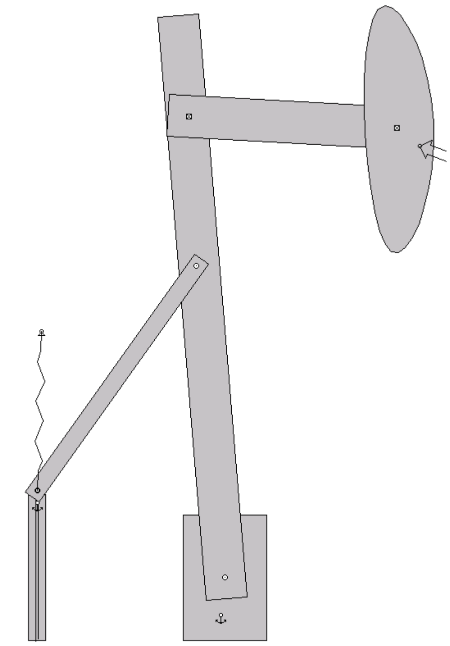

Figure 83: Working Model Kinematic Accelerator Pedal

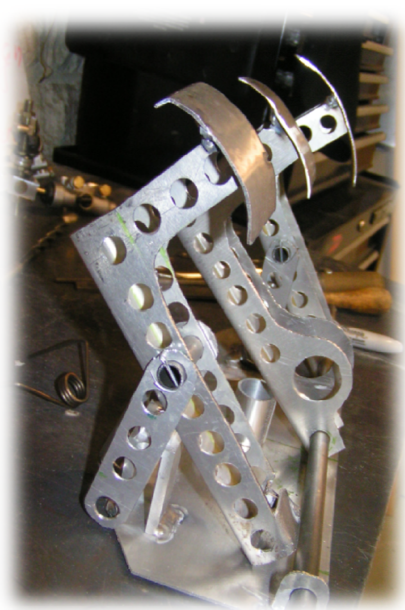

Figure 82: Fabricated Pedal System

The fabrication process, as mentioned, was focused on weight reduction of the pedal box. The pedals were all made from 1/16th inch thick aluminum sheet metal, as well as the base plate and accelerator and clutch linkage tabs. The master cylinder, accelerator cable, and clutch cable mounts were all made from 6061 aluminum stock that were manual mill machined for the necessary sizes and features. Additional features of the pedal box include the 4130 steel pedal pivot rod and joints for the accelerator and clutch linkages. The master cylinders were modified for size reduction as well. The bias bar aluminum clevis mounts were machined approximately one inch shorter and the master cylinder rods were cut down to better package the pedal box. The pedal box was slotted for adjustability of the entire system, which is secured using quick release clamps. For rigidity in the event of pedal torsion, addition Delrin bushings were inserted throughout the pedals. The entire pedal box was prepped and painted before installation and ergonomics tests were performed through the fabrication process to ensure all drivers could comfortably interact with the system.

ECU, Instruments and Wiring

The main goal of Worcester Polytechnic Institute's 2008 Formula SAE team is to complete all dynamic events and provide sufficient testing and data to properly tune, set-up and justify engineering decisions for the 2008 Formula SAE event in Detroit, Michigan. This requires the design of a completely new electrical and Engine Management System. Reliability, simplicity and cost are all important aspects to the electrical system. The system must also be integrated with all other vehicle systems while maintaining maximum performance.

Simplicity

The 2008 WPI Formula SAE design ideal is to design a simple, yet effective high performance vehicle. A common misconception is that simple is basic, or under-designed. This, however, is not the case for the 2008 WPI Formula SAE vehicle. In a racing vehicle where many possible failures car occur, overly complicated or unnecessary components and systems not only increase the cost of the vehicle, but also increase the likelihood of failure. The comparison of useful VS flashy summarizes the analysis and design decisions of all aspects of the 2008 WPI Formula SAE vehicle, and is especially important to the design and implementation of the electrical and ECU. Rather than complicate the system through the addition of sub-systems and gadgets, the idea is to provide the necessary controls and maximize the performance and reliability of the necessary vehicle systems, while providing a intuitive and predictable driver control system.

Weight, Packaging and Cooling

The weight of a vehicle is always important, but even more so in a racing environment. Intuitively, a vehicle with a lower mass will accelerate more rapidly and be more responsive than a vehicle of higher mass. Every system of the vehicle should therefore be monitored to ensure no unnecessary mass is added to the vehicle. While the electrical system may seem like a relatively small system when compared to the chassis, suspension, or other large systems, if care is not take, a significant amount of mass can be added. In an environment where victory is often contributed to ounces, not pounds, weight is considered in every component. Location and effect of the systems mass on the vehicles center of gravity (COG) should also be considered.

Packaging is another concern, for many reasons. Sensitive electronics must be protected from serious shock, heat, electrical interference, and other damage and interference. It also must allow the wiring harnesses, dash and necessary subsystems to be integrated without effecting driver ergonomics, controls or vehicle dynamics. Electronic chips and processers produce significant amounts of heat which, if not dissipated, could cause premature failure of the system. Proper airflow to these systems must be considered to prevent overheating. Additionally, multiple units should not be placed in close proximity and/or in enclosed spaces where the overall heat generation would be greater than the ability to cool the systems.

Cost

Cost is a significant factor in many forms of racing, and is one particular aspect in which the 2008 WPI Formula SAE vehicle will be judged. The limitations of a finite budget and small team are factors, requiring a cost/benefit analysis for each component of the vehicle. Electronics are classically a large source of the cost of a vehicle, especially a modern race vehicle. Engine Management Systems can often range from $1000, to over $3000 (USD.) The capabilities, requirements and reliability of all these units will greatly vary and will often guide many other design decisions and have a significant impact on the performance of the vehicle. Often, these units may be more “flashy” than useful. Proper justification of all components in the electrical system is critical to both the cost and design aspects of the vehicle and could greatly impact the team's success in the static events.

Safety: Fail Safe

Driver safety is of the utmost importance in Formula SAE vehicles. Safety is a major concern of the WPI Team, WPI Administration, the Formula SAE staff, and organizers, such as the Sports Car Club of America (SCCA). The majority of the 2008 Formula SAE rules are aimed to ensure a safe racing vehicle. This is of even greater importance when the intended vehicle use is for amateur, often untrained, drivers. Ensuring driver safety, especially during the event of a major system failure, as well as spectator and track marshal safety is critical.

Figure 84: Example of a Safety Shut-Off Switch

The 2008 Formula SAE rules mandate several safety requirements for the electrical system. These include three different kill switches which will shut off the vehicle systems if any are activated as well as a safety switch to eliminate accidental starting of the vehicle or movement of the vehicle during starting. These kill switches include 1 main kill switch accessible from outside the vehicle, a kill switch for the driver, and a switch which will activate if the brake pedal has traveled too far (i.e. during failure of the braking system.) The goal is to provide a fail-safe situation and a safe way to stop and shut down the vehicle. An example of a common type of main kill switch used can be seen in Figure 84.

Design

This section details the design decisions, justification and selection of the components of the 2008 Worcester Polytechnic Institute Formula SAE Vehicle electrical system.

Design Objectives

The requirements of the 2008 WPI Formula SAE vehicle, the 2008 Formula SAE Rules, and research and analysis of past Formula SAE electronics and road vehicle electronics system were all consideration in the design of the 2008 Electrical System and ECU design and selection. To ensure all goals and necessary requirements are met, a list of design objectives were determined. These design objectives are as follows:

- *Comply with the 2008 Formula SAE Rules*
    - *3 Required Safety Shut-Off switches*
    - *Clutch Safety Switch*
    - *Necessary Components (Brake Light, etc)*
- *Provide full control of the fuel system*
- *Provide full control of the ignition system*
- *Provide means for control of other vehicle systems*
    - *Cooling (fans, etc)*

    - *Additional Controls (Launch Control, Rev-limiter, etc)*
- *Mount on the vehicle without compromising other vehicle systems*
- *Support all necessary driver controls and instrumentation*
- *Minimize all unnecessary weight*
- *Eliminate and components not directly improving performance*
- *Minimize Cost*
    - *Cost VS Benefit analysis*
- *Offer a "fast" learning curve*
    - *Ability to teach others to use/install the system*

Terms and Abbreviations

The following is a list of commonly used abbreviations, and a definition of terms commonly used in electrical and fuel/spark management systems, as well as data acquisitions systems.

EMS - Engine Management System, The system comprised of the ECU, sensors and components used to control the fuel, spark and other engine and vehicle parameters

***ECU* –** Engine Management Unit, This is the main computer of the Engine Management System, Also called PCM (power-train control module)

***Electronic Fuel Injection (EFI)* –** A method of injecting fuel system using solenoids activated electronically by the EMS/ECU

***TPS* –** Throttle Position Sensor

IAT – Intake Air Temperature

MAT – Manifold Air Temperature, analogous to IAT

ECT – Engine Coolant Temperature

MAP – Manifold Absolute Pressure

MAF – Mass Air Flow

EGT – Exhaust Gas Temperature

WOT – Wide Open Throttle (maximum throttle position)

TCS – Traction Control System

ABS – Anti-lock Braking System

TDC – Top Dead Center, When the piston is at its top most point

BDC – Bottom Dead Center, When the piston is at its lowest most point

BTDC/BBDC – Before TDC/BDC, measured in degrees of crank rotation

ATDC/ABDC – After TCD/BDV, measured in degrees of crank rotation

VE – Volumetric Efficiency, The measure of an engine efficiency in filling it's cylinders

Boost/Vacuum- The relative positive/negative pressure in the manifold with reference to atmospheric

AFR – Air/Fuel Ratio, The measure of fuel in relation to the amount of Air present

02 Sensor – Oxygen Sensor, used to measure the AFR after combustion

Timing – Usually ignition, determines the location of the spark event relative to crank angle (usually BTDC)

Rev Limiter – A setting within the ECU which limits the RPMs an engine will turn at before fuel and/or ignition cut

FueIgnition Cut – Momentary suspension of fuel injection and/or spark

Knock – A definition used to define a vibration due to non-ideal ignition parameters

Knock Sensor – A sensor used to measure the amount of knock

Pre-Ignition – The premature ignition of the fuel/air mixture

Detonation – Ignition of end gasses following the normal spark event

Rich – A state defining a high presence of fuel in the air fuel ratio

Lean – A state defining a low presence of fuel in the air fuel ratio

Acceleration Enrichment – The momentary enriched mixture when the TPS signal changes at various rates

Load – Calculate value used to determine the amount of fuel to inject, can be either MAP, or TPS based

Sequential Fuel Injection – Fuel injection method in which the injector is fired every other crank rotation; Provides the best power and fuel economy

Semi-Sequential Fuel Injection- Fuel injection method in which an injector is fired every crank rotation; Provides the better power and fuel economy than batch fire

Batch Fire Fuel Injection - Fuel injection method in which all injectors are fired simultaneously, worst power and fuel economy; commonly used during cold startup

DAQ System - Data Acquisition System, a system used to measure various inputs and sensors

Analog – An input/output system which is infinitely variable, typically a 0 to 5v system on modern vehicle

Digital – An input/output system with an only an on and off state.

Design Considerations

To design the electrical system and ECU there are several aspects to consider, including end goals, vehicle and engine requirements, user and team skill and knowledge, reliability, etc. These all will impact the vehicle performance and success in the 2008 Formula SAE completion. Judges will be looking for justification and engineering knowledge of the system and determining whether the decisions correspond with the engineering goals, application, and marketing plan.

Engine Management System (EMS)

The engine management system (EMS) is essentially the heart of the entire electrical system. It controls all dynamic aspects of the engine and related systems, including spark and injection events. The engine management system defines the maps, or curves/charts, used to calculate the amount of fuel to inject and when the start the ignition event. There are many options available for engine management systems which include O.E.M., Piggyback , or Standalone. Each option offers its own strengths and weaknesses and were each analyzed to determine the best system for the 2008 WPI Formula SAE Vehicle

OEM (Original Equipment Manufacturer)

This unit would be considered the “stock” unit. This is the engine management system provided by the manufacturer, if one exits. This uses the manufacturer determined fuel and timing maps and includes all the sensors to properly control the engine. In the case of the Honda CBR 600 F4i engine this is the unit sold with the motorcycle.

The main advantage of this option is simplicity. The engineering work, mapping, sensors and all other requirements have already been done. This unit was designed specifically for the application, typically ensuring a high level of reliability. It is a “plug and play” system. This method would allow the team to have the engine running with little time and effort, thus maximizing testing time. Use of the OEM system requires minimal expertise to install and use and could be performed by virtually any member of the team.

There are, however, several disadvantages to an OEM system, the most critical being the lack of control. The majority of OEM engine management systems do not allow the user to control the maps and aspects of engine tuning. This is due to many reasons, including liability, safety, cost, and warranty considerations. Additionally, the stock maps may not be optimally tuned for a specific engine, as the parameters must safely and reliably operate all engine produced in a variety of operating conditions and environments. Further, most engines are not designed with the Formula SAE mandated restrictor, thus limiting the usefulness of the factory maps. The factory tune may in fact be harmful to the engine in its modified operating state, and will certainly not provide maximum performance from the engine. The OEM system will most likely offer little to no data acquisition capabilities, as the common user has minimal use for such data.

Piggyback

This system is one step above the OEM/stock. A piggyback engine management system, such as the power commander used in the 2004 and 2007 Formula SAE vehicles, utilizes the OEM unit, but adds an additional unit which interrupts and modifies the signal to achieve a desired operating parameter. As with the stock system, this unit offers a relatively simple and quick method of engine control. The stock ECU, sensors, and wiring are maintained with usually little modification. This unit will allow control of some engine parameters, within the limits of the stock ECU. Many piggyback units also allow a certain measure of data acquisition using the OEM sensors. This can be useful for basic tuning of the engine. The learning curve for a piggyback system is also very fast. Since the system starts with a base tune, one can simply modify the parameters until the desired conditions are met. A simple air fuel ratio and monitoring of knock sensors are used to perform tuning. Current team mates are familiar with piggyback systems including the Power Commander, Gizzmo CAM FC, Greddy eManage/eManage Ultimate and AEM AFC. Since most piggybacks rely on the same basic signals and functionality, the installation and use would be similar to these units.

Figure 85 Power Commander Piggyback ECU

As some past WPI Formula SAE vehicle have shown, there are limitations to piggyback systems. They will not always offer control of all the engine systems. The power commander, one of few systems available for OEM motorcycles, could not control spark events until recent models. Even the new models require the addition of an extra module. This is very common among piggyback systems, as the ignition system require additional calculations and hardware to provide a safe and reliable system.

Further, since the OEM system is retained, the tune may be inconsistent, as the piggyback and OEM ECU “fight” for control of the system. In the cases of the 2004 and 2007 vehicles, this was the case. The amount of fuel necessary to remove to tune for the modified air flow was greater than the piggyback could maintain, and the unit would revert back to a near stock tune, resulting an extremely rich air fuel ratio. This severely impacted performance and cause maintenance issues, such as fouled spark plugs.

A piggyback system also adds additional mass to the vehicle, as additional hardware and wiring is required to control the engine. This can be multiple additional units at times when the demand for additional controls is added. This also creates packaging and cooling considerations.

Standalone

A standalone engine management system is the most complex and also the most versatile system available. As its name suggests, a standalone system is one capable of control engine and vehicle parameters absent of any other system. Common standalone systems include MoTec, AEM (Advanced Engine Management), Haltec, and performance electronics as well as “build you own” options such as custom boards, or “megasquirt n spark”. The standalone units vary greatly in cost, functionality, reliability and ease of use. The cost alone can range from under $1000 (USD) to $3000(USD) or more.

Figure 86 Performance Electronics Standalone ECU

The major advantage of a standalone system is the ability to control virtually all parameters of the engine systems. Various size maps are available for both injection and ignition control, and it is often possible to access a different map, depending on operating conditions (i.e. cold start, high acceleration, or even component failure.) It is also possible to add additional controls, not present in stock form, such as launch or traction control, shift control, etc. Since there are many systems to choose from, the unit can be selected, or even designed to suit an individual application.

A standalone unit typically does not require additional hardware or control boxes, thus typically minimizing weight, and can eliminate many of the unnecessary components of the stock system, not required for race use and even simplify the electrical system. It also will typically offer very consistent results (assuming proper use) as there are no other systems to interfere with the standalone unit. Many standalone systems also provide data acquisition capabilities, ranging from basic, to virtually a full DAQ system.

The largest drawback, however, is the typical complexity of the system. In even the simplest unit, a basic understanding of a vehicle electrical system and fuel injection and ignition controls in mandatory. Additionally, unless the manufacturer or another user has provided a base tune, there are no ready to run maps, as these must be defined by the user. Unless the installer and tuner is familiar with engine and electrical basics, the system could become overwhelming, and difficult to implement. This, along with possible inherent flaws, could lead to a high level of unreliability in the engine management system. This was the case of one system attempted on the 2007 WPI Formula SAE vehicle, and ultimately resulted in many of the performance drawback from which the vehicle suffered.

Depending on the unit, some or all of the OEM sensors may not be useable. This may require the addition of extra sensors and the modification of the engine and supporting systems to accept the sensors. In some cases, the ECU may require additional programming to ensure proper calibration of the sensor and functioning of critical aspects of the Engine Management System.

Standalone units are also among the most expensive options, though greatly varying in their cost. Many units may simply fall outside of the teams financial capabilities, this is often the case of the MoTec unit, as this is amongst the most expensive options. A careful cost VS benefit analysis and research, as well as definition of goals and engineering justification in necessary when dealing with standalone engine management systems.

ECU Selection

It was decided that a standalone engine management system would be used on the 2008 WPI Formula SAE vehicle chassis, despite the potential cost and complexity. Piggyback and OEM units do not offer the control necessary to adequately tune the CBR600f4i engine with the restriction required. Use of either a piggyback or OEM system would result in a poorly tuned vehicle as well as cause many reliability concerns, thus offsetting the gain in simplicity and time gained.

The performance gains of a standalone EMS are necessary to create a competitive vehicle for the 2008 Formula SAE competition. Carful research and analysis, along with a cost/benefit analysis allow a standalone system to be selected, while still maintaining the desired reliability and simplicity of the system. The elimination of many of the unnecessary stock items will simplify the overall electrical design and once designed the system can be maintained and modified for future vehicle with proper knowledge transfer from team to team.

Standalone Engine Control Unit

As mentioned in the previous section, there are many different choices for standalone engine management systems. This section will evaluate the primary of these options and provide a justification for the final ECU selection. The major factors to consider in the selection of a standalone ECU include cost, ease of use/installation, and reliability. Additional factors such as data acquisition capabilities, additional inputs/outputs, weight and footprint will also be considered.

MoTec

Figure 87: MoTec M400 Engine Control Unit

MoTec is arguably one of the most powerful engine management systems available today. The MoTec unit offers full control of spark and injection events, offering very fine and precise control of the curves defining these events. The unit utilizes a 40 x 21 table for the main fuel calibration (840 sites) and also offers additional tables for modifications and alternate maps. The ignition offers the same level of control.

The MoTec unit requires calibration of the sensors used, and accepts a broad range of commonly used sensors (commonly GM or Delco sensors.) Most sensors can be found on stock applications or easily adapted.

The MoTec unit also offers many additional features, including data acquisition, traction and launch control, and cam control. These, however, are not standard features, and must be purchased separately, which is a major contributor to the high cost of the unit. The base unit is already amongst the most expensive run by a formula SAE team, with the additional options, the cost becomes even higher. This is a major detriment to the cost score of the vehicle, and possibly even the design judging. The part can be considered over complicated for the required situation. Many of the additional features can be achieved on a less expensive unit using little additional hardware. Additionally, the MoTec unit is amongst the most complex, requiring creation of a custom wiring harness, integration of various non-stock sensors, and most likely the necessity to complete build the base map from scratch. While the large tables offer a fine range of adjustability, in the case of the base tune, it requires even more work just to get the unit up and running. This limits the amount of testing, tuning, and in the end, lack of confidence in the vehicles engine management system. If poorly implemented, the MoTec unit could create a serious weak point for the 2008 WPI Formula SAE vehicle.

Advanced Engine Management (AEM)

Figure 88: AEM's Engine Management System

AEM is a company well known in the aftermarket automotive crowd for providing a decently affordable standalone engine management system, often preferred because of the "plug and play" adaptors offered for many street vehicles. The unit most situated for Formula SAE use, however, is actually provided through a third party, Superior Engine Technology.

The FSAE version of the AEM ECU uses the stock wiring harness and all stock sensors, with the wiring harness re-terminated to connect to the AEM unit. This would most likely still require some modification to the wiring harness as not all the sensor may lie in their stock location. However, with minimal modification to the stock harness, the AEM unit may be adapted to control the engine and related system of a Formula SAE vehicle. It should be note that the unit will not drive stock ignition coils. This will require the addition of a Capacitive Discharge Ignition system (CDI) or ignition driver. These can range from $100 to $500 (USD) or more.

As with the MoTec unit, the AEM unit offers the data logging abilities, as well as the possibility to integrate traction control and launch control, and control shifting. The unit uses a 17 x 21 table (357 sites) for fuel and ignition control, providing adequate control for most applications. Unlike the MoTec unit, these abilities are all standard, and not "options."

Superior Engine Technology also offers technical support specifically for Formula SAE applications. It should be noted, however, that response time is very slow, and during the 2008 electrical design the value and level of technical support was questioned. This, however, may be the exception, and not the norm. Base maps for the Honda CBR 600 f4i are also provided for various basic parameters (runner length, etc.)

The AEM unit is in the middle of the cost range, costing just over $1500 (USD.) This, however, does not include a stock wiring harness, nor the injector driver required to run the F4i ignition coils. While the AEM unit is not as simple as some methods, with carful planning, basic understanding of engine and electrical systems and careful fabrication the unit provides a reasonably reliable and cost effective engine management system.

Performance Electronics (pe-ecu-1)

Figure 89: Performance Electronics ECU

The PE-ECU-1 produced by Performance Electronics LTD is among the cheaper options for standalone engine management systems, costing just under $1000 (USD). Performance Electronics also offers a FSAE student discount, bringing the price to a little over $700. Similar to the AEM unit provided by Superior Engine Technology, the PE-ECU-1 has been marketed towards Formula SAE teams. Unlike the AEM unit, however, this is not a stock unit modified to meet Formula SAE requirements. The PE-ECU-1 is a standalone system which has the functionality to be easily adapted to use of Formula SAE vehicles. Some stock sensors may be used, while some require replacement with similar GM versions, as not all sensors can be calibrated (MAP, IAT, and ECT.) The unit also has the capability to run the stock F4i coils, eliminating the need for a separate ignition system.

For fuel and ignition control the PE-ECU-1 offers 16 x 16 tabled (256 sites). This is the most limited of the standalone system covered, however, offers sufficient control to tune the f4i engine, without overly complicating the tuning process. The unit also offers controls of the rev limit, and fuel and ignition cut, allowing integration of traction control and/or launch control systems (these however would require additional hardware.) Like the other units, it also allows compensation maps for various events such as acceleration, deceleration, etc.

Performance Electronics also provide technical support and base maps of the Honda CBR 600 f4i. These maps are specifically built for a Formula SAE engine using a 20mm restrictor and can aid in the initial start up and tuning of the vehicle. The response from Performance Electronics technical support staff has also been timely, helpful and informative.

The unit offers basic data logging capabilities, but lacks a significant amount of programmable inputs. Since a standalone data acquisition system need not be included in the cost of the vehicle, however, this option still is favorable for the cost section of the competition.

Another possible disadvantage is the inability to provide fully sequential fuel injection. This can lead to slightly lower horsepower and decreased fuel economy, as the 2 injectors must fire at the same time. Additionally, the unit requires the replacement of the stock crank trigger wheel. This allows the unit to detect the location of the crank shaft and determine fuel injection timing. This modification if not highly involved, but requires some accurate tools to determine the TDC of cylinder 1 and patience to ensure proper installation of the new device.

The PE-ECU-1 offers not only a cost effective option, but a simple, and usually reliable option for a standalone engine management system. The system can easily be adapted for a

Honda CBR600 F4i using provided documentation and with basic introduction and knowledge, the use, tuning, and installation of the unit be carried over year to year.

Megasquirt n' Spark/Custom Build

The Megasquirt n' Spark system is an example of a simplified custom built engine management system. These systems require the user to assemble the circuitry, sensors, and other necessary hardware for the ECU. This, in most cases, results in a very low cost system. Pre-designed systems, such as the megasquirt n' spark have components you can add, depending on your application. This allows the unit to be customized to fit the exact need of the user and the vehicle. These units may or may not allow Traction/Launch control, data acquisition of other features, depending on the specific build. If an entire custom build is used, custom software must also be written to control the unit.

This method requires extensive knowledge of vehicle electrical and engine systems, as well as fabrication skills and knowledge of circuits and controls. While the unit could offer a very customized range of controls suited for a specific application, if manufactured incorrectly, could fail to function as all. Additionally, the hand fabrication of the unit will typically mean less reliability, as the process control and precision are much lower than that of most commercially available units. While this unit offers an extremely low cost option, while maintaining a high level of control, only those very skilled and knowledgeable should attempt such a method due to the difficulties and possible unreliability of the system.

Results

The engine management system selected for use on the 2008 WPI Formula SAE vehicle is the PE-ECU-1 by Performance Electronics. This unit was selected because of its low cost, and simplicity of use, while maintaining the control deemed necessary to properly tune the engine. The simplicity of both the installation and use of the system is critical, as knowledge carry over has been sub-standard on the WPI formula SAE team in past years. In order to maintain a successful program and provide future teams with a unit that is easy to learn and quick to implement will aid in teaching future teams about engine management systems and allow further improvements to be made to the system.

The main goal of this year's vehicle is to build a simple, but reliable system within budget and properly engineered. It is exactly this goal that the PE-ECU-1 achieves. It does not add any necessary of overly complicated systems or provide a weak link in the engine management system. It is useful, rather than flashy and ensures that the vehicle will be properly tuned, with adequate time for testing. If further additions, such as traction control, are desired, they can be integrated using the digital and analog inputs of the ECU.

The unit also improves the cost and marketability of the vehicle. The unit is much cheaper than its competition, partly due to the lack of data acquisition capabilities. This, however, in the long run, is more cost beneficial to the team, both financially, and competitively. The use of a standalone DAQ system does not have to be factored into the cost report, and can provide a customized system to collect and record the desired information. This system can subsequently be removed from the car, without affecting the running condition, and installed and a future car, with no new cost incurred. This leaves previous cars intact for testing while still providing the data acquisition capabilities necessary for proper set-up of the new vehicle.

Wiring Diagrams

Using the provided documentation, shown in Appendix A, from performance electronics, wiring diagrams and pin-outs of each connector were created. This diagram includes all necessary sensors, relays, power circuits, and vital engine components (injectors, coils, etc). The intention of the wiring diagram is to provide documentation from which a wiring harness and electrical system can be fabricated and diagnosed in case of failure. The application, wiring and functions of the engine management system component will be discussed the next section.

Following the theme from the rest of the vehicle, the wiring harness was designed to be modular. By disconnecting the appropriate connectors, the main wiring harness, power harness, dash harness, and pedal harness can be removed, without the need to remove additional wires. This allows parts and sections of the vehicle to be removed for service, without the risk of damaging another section of wiring or the difficulty of removing and reinstalling the entire harness. The engine control unit can also be removed without removal of any of the harness, in the case of failure, or to preserve the unit during vehicle maintenance, as certain operations, such as welding, pose significant risk to the electronics.

Components

The engine management system is comprised of many components, including sensors, coils, injectors and relays. Each component plays a vital role in the behavior and control of the engine system. This section provides a brief overview of the major components.

Intake Air Temperature Sensor

This sensor is used to measure the temperature of the incoming air. The PE-ECU-1 uses only A Delphi GM IAT Sensor. The intake air temperature is used to make certain corrections to the engine's performance parameters, such as detecting a cold start. In vehicle equipped with a Mass Airflow Sensor (MAF sensor) the intake air temperature is used along with the MAF sensor to determine the density of the incoming air. The GM sensor used in the 2008 vehicle works using a very simple principle. The sensor is a thermister, with a +5 volt signal being fed into the sensor, and a return signal going back to the ECU. The change in voltage is then correlated to a specific temperature.

Figure 90: GM Delphi Intake Air Temperature

Engine Coolant Temperature (ECT) Sensor

The temperature of the engine's coolant is measure for a variety of reasons. One reason is to activate appropriate sections of the cooling system, such as the fans, as is done on the 2008 Formula SAE vehicle. The ECT is also important to many of the ECUs critical functions, including ignition timing and fuel injection control. The ECU monitors the ECT to ensure the engine is within proper operating range before running a more aggressive tune. The Engine coolant temperature sensor operates using the same principles as the IAT sensor, just using a sensor with a different range.

Figure 91: GM Delphi ECT Sensor

Manifold Absolute Pressure (MAP)

The MAP sensor is used to measure the pressure of the incoming air in the manifold, this, along with throttle position, can be used to calculate Load and the proper amount of fuel to inject. The MAP sensor used in the 2008 Formula SAE Vehicle has a silicon inside, which can flex. The movement of this chip causes a change in its resistance. A vacuum tube is connected from in plenum of the intake manifold to the MAP sensor, and it is the boost or vacuum in the manifold that causes the flex of the chip. Note: MAP sensors often have limited ranges, in applications were forced induction is used care should be taken to ensure the MAP sensor will support the expected manifold pressures created under boost.

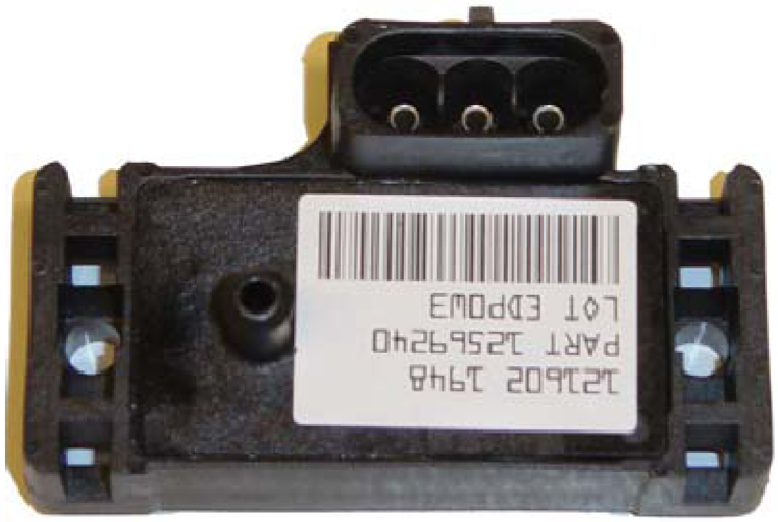

Figure 92: GM MAP Sensor

The Map sensor has a Signal, +5 volt, and ground wire connected. The resistance of the chip will cause a change in the returned voltage, from which the ECU will calculate the Manifold Absolute Pressure.

Throttle Position Sensor (TPS)

The throttle position sensor is a potentiometer which measures the relative position of the throttle. This measurement, like the manifold absolute pressure, is used to calculate load and determine engine performance parameters. The PE-ECU-1 can utilize virtually and Throttle Position Sensor or form of three wire potentiometer, such as the linear potentiometer used on the 2007 Formula SAE Vehicle. The TPS has a +5 volt, Ground, and Signal connection. As the resistance changes as the TPS moves, the returned voltage is altered and the throttle position calculated. To calibrate the throttle position sensor the maximum and minimum values are set by opening the accelerator all the way to set the maximum, and leaving it in its resting state to set the minimum. The ECU bases the calibration of this data given the linear behavior of potentiometers.

Figure 93: Example of a Throttle Position Sensor

Crank Position

Figure 94: Hall Effect Sensor

The crank position sensor is used to determine the relative position of the crankshaft and time injection and ignition events. In many engines, including the Honda CBR600 F4i, a Hall Effect sensor is used. A hall effect sensor uses a rotating wheel with teeth passing through a magnetic field. The position of the wheel and teeth allow the ECU to determine crankshaft position.

Fuel Injectors

Fuel injectors are essentially an electronic valve used to meter and control fuel flow to the engine. A fuel injector uses a solenoid coil to activate the opening of the injector. When the signal is removed, a return spring causes the fuel injector to close, thus stopping the flow of fuel.

A diagram of a fuel injector can be seen in Figure 95. Fuel injectors are activated by triggering of the ground signal, not the power. Injectors are typically fed constant power once the ignition is turned on, and the ECU will trigger a ground signal every time the injector is to fire.

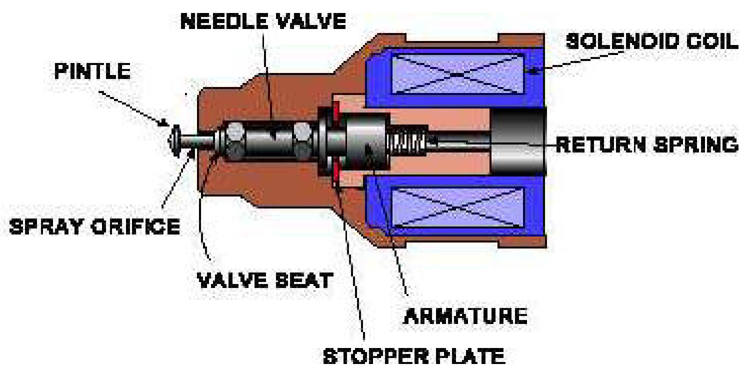

Figure 95: Fuel Injector Diagram

The fuel injector sees constant fuel, regulated at a specific pressure. This fuel pressure will affect the duration, or pulse-width, of the injector. The pulse width defines the amount of time which the injector is opened, which in turns, along with injector size, determines the amount of fuel injected. The PE-ECU-1 calculates the pulse width using the following formula (similar for most ECUs):

Open Time = (Base Open Time x AT x CT x CR x AC x CC x U1** x U2** x U3**) + BA*

Equation 1 – Total Open Time Calculation

Where:

Base Open Time = Basic open time based on engine load and RPM (2D Table)

AT = Air temperature compensation (1D table)

CT = Coolant temperature compensation (1D table)

CR = Cranking compensation (1D table)

AC = Acceleration compensation (User defined parameters)

CC = Individual cylinder bank compensation (User defined parameter)*

*U1** = User selectable analog input number 1 (1D table)*

*U2** = User selectable analog input number 2 (1D table)*

*U3** = User selectable analog input number 3 (1D table)*

BA = Battery voltage compensation (1D table)

** Cylinder bank compensation is adjustable for injector banks 2,3 and 4. These are set as a percent of bank 1.*

***Only included in calculation if set to modify the fuel flow*

The various compensation factors are additional parameters the user can define to control the injection system under various conditions, such as high acceleration, during cranking, etc. In a more basic and generic form the equation is as follows:

Pulse Width = [(MAP voltage ÷ 5) x UAP] ÷ POT

Where:

UAP = User Adjustable Pulse-Width

POT = Pulse Width Offset Time

Or in a more accurate and realistic form, which accounts for Volumetric Efficiency (VE):

Pulse Width = [(MAP voltage ÷ 5) x UAP x (VE Absolute% ÷ 100)] ÷ POT

Where:

UAP = User Adjustable Pulse-Width

VE Absolute % = Volumetric Efficiency Percentage

POT = Pulse Width Offset Time

With the addition of all correction factors, the equation becomes as follows:

Pulse Width = [(MAP voltage ÷ 5) x UAP x (VE Absolute% ÷ 100) x TPS% x CTS% x IAT% x EGO% x SE5] ÷ POT

Where:

UAP = User Adjustable Pulse-Width

VE Absolute % = Volumetric Efficiency Percentage

TPS = TPS sensor signal

CTS = Coolant Temperature Sensor

IAT = Intake Air Tempeature

EGO = Exhaust Gas Oxygen

SE = Starting Enrichment Requirements

POT = Pulse Width Offset Time

Ignition System

Ignition coils rely on inductance to produce the charge necessary to ignite the air/fuel mixture. Voltage is fed to the coil, charging it, and when a trigger is fired, the coil releases the charge, firing the spark plug and causing combustion. Similar to the injection system, there are many parameters that are used to determine exact ignition timing.

Improper ignition timing can result in knock, or even pre-ignition due to premature ignition of the fuel/air mixture. A late ignition will result in lost power. The spark event must carefully be timed to allow full and proper combustion. Ignition timing can be retarded (moved closer to TDC) or advanced (moved further from TDC.) Typically advanced timing will result in higher horsepower, however, too aggressive of a timing advance will cause knock and eventually engine failure as the combustion attempts to move the piston backwards.

The ignition system on the 2008 Formula SAE vehicle is configured in a wasted spark mode. This means 2 coils are fired at the same time. One of the sparks causes no combustion event, thus the name “wasted spark.” The PE-ECU-1 uses the following basic equation to calculate ignition timing.

Total Ignition Timing (degrees)= Base Timing + AT + U1 + U2* + U3**

Equation 2 – Total Ignition Timing Calculation

Where:

Base Ignition Timing = Basic ignition timing based on engine load and RPM (2D Table)

AT = Air temperature compensation (1D table)

U1 = User selectable analog input number 1 (1D table)*

U2 = User selectable analog input number 2 (1D table)*

U3 = User selectable analog input number 3 (1D table)*

** Only included in calculation if set to modify ignition*

Relays

Automotive relays are used to activate items such as fuel pumps and cooling fans. The relay uses a trigger signal the close the relay and passes a 12 volt signal through to the end source. A fuse should always be used when wiring the +12 volt source voltage to the relay. The fuel pump and fans are both activate when the ECU triggers the relay, and +12 volts powers the device. The starter system of the care is actuated in this way as well.

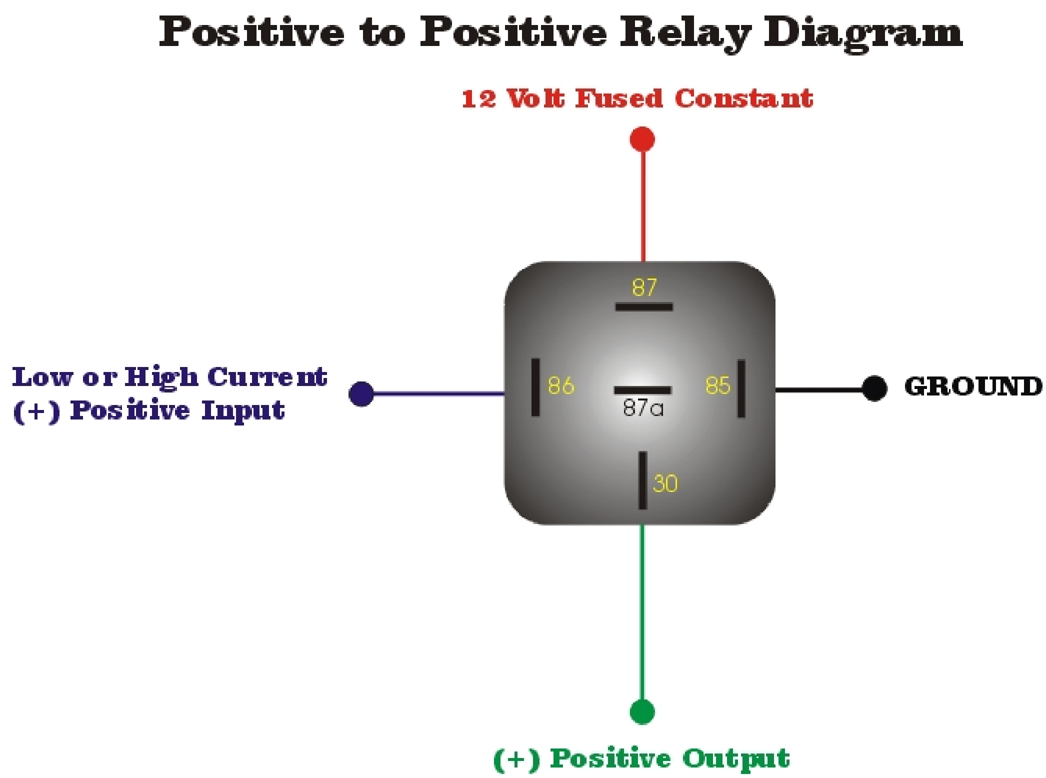

Figure 96: Typical Automotive Style Relay

For the radiator fans on the 2008 WPI Formula SAE vehicle, 2 separate fan circuits will be use. They will be an AUTO setting, where the ECU has full control of the radiator fans and their on/off state, and an on setting, which will have the radiator fans always on, regardless of the ECU signal. This is important because it allows the drive to turn the radiator fans on when the engine isn't running, such as sitting in the paddock, or to keep the ECT temperatures low before a run. Additionally, it offers a fail-safe option. If the ECU is failing to properly control the engine temperature and is not turning the fans on, the driver may choose to manually run the fans to control the engine coolant temperature.

Launch Control

The PE-ECU-1 offers the ability to cut ignition, fuel, or both at the same time, or even to set a second stage rev-limiter when certain criteria are met. This is achieved using user defined digital inputs. When a “high state” is observed, the ECU can take one of the four previously mentioned actions. A high state is defined in the ECU as a signal between 3.1 and 5.0 volts

In order to activate the launch control, a +5 volt signal will be sent to the ECU. This will be achieved through use of a dash mounted switch to arm the system, and use of the clutch safety switch to activate it. The clutch switch already uses a +5volt signal to which is passed to the starter switch and subsequently the starter relay. When the clutch is depressed, the second stage rev-limiter will be activated. This rev-limiter will be set to allow the vehicle to launched in a quick, yet controlled state.

Traction Control System (TCS)

Currently, the traction control system is not being implemented on the vehicle, due to the lack of the control and logic portion of the device. Rather than spend significant time researching and building a possible faulty system, the requirements and methodology have been designed. The current electrical system leaves the ability for the addition of traction control at a later time.

Similar to the launch control system, the traction control system will be activated using one of the user definable inputs, and have an on/off setting which the driver can control. This can either be achieved using the digital input, and logic to ensure the signal does not bounce in and out of the threshold values, as this would cause a pulsing in the TCS and instability in the vehicle, or with the analog input. The digital signal would cut ignition during a “high state.”

While the analog system can be set to remove fuel and/or spark in a level adequate for the amount of slip the wheels are seeing.

The monitoring of the rear wheels would be achieved through a wheel speed sensors mounted at the rear wheels. Additionally, an accelerometer would be mounted on the vehicle to measure the acceleration of vehicle, and compare it with the acceleration of the wheels. The speed of both wheels would be calculated, and the acceleration of the vehicle calculated, and then compared with the measure acceleration of the vehicle. If the measured vehicle differs greatly from the measured value, the TCS would be triggered.

Care must be taken when determining the frequency of measurements for the traction control system, as a high amount of lag in the system can result in further vehicle instability, while and overly sensitive system will damage vehicle performance. Various limits of slip should also be programmed, and allow the driver to change the settings via a cockpit mounted control. This would allow the driver to adjust the sensitivity of the traction control to suit various driving conditions.

Safety Switches and Power System

Sections 3.2.5.2 and 3.9 define the necessary kill switches that must be mounted on the car. These include the master kill switch, brake over travel switch, and cockpit mounted kill switch. These switches are intended to shut off the car in the case of a problem or emergency, and ensure drive, spectator and track worker safety. The brake over travel switch and driver cockpit switch must shut off the engine and kill all power to both the ignition and fuel pump. The brake travel switch may not also be reset by repeated actuation (i.e. pressing the brakes again may not restore power to the car.) The master kill switch must be located near the main roll bar and must disable all electrical systems. All battery current must flow through this switch.

These three switches will be wired in series, so the actuation of any switch will kill all power to the vehicle. The positive lead of the battery will be connected to the input of the master kill switch. The alternator and starter relay will also be connected to the main kill switch. The main kill a power lead into the cockpit mounted switch, which will then be connected to the brake over travel switch and finally to the power system.

Engine/Drivetrain

Differential Carrier/Chain Tensioner

The completely redesigned chain tensioning system of the 2007-2008 vehicle attempts to correct the dependability and simplicity of previous years' attempts. This year, a double slider-crank design was utilized for rigidity and ease of adjustment. The major components included in the chain tensioning system design are the two lightweight 7075-T6 water-jet machined differential carriers and the dual slider cranks for tensioning. Previous year's typically consisted of two oversized differential carrier plates that were heavy and awkward to work around. Redesign of the differential carriers created two significantly smaller aluminum plates that allow for other components to be located around the differential and sprocket; namely the dual slider cranks.

The design of the carrier plates was focused on weight and size reduction, therefore FEA was critical for the optimization process. The strength analysis of the differential carriers was performed by applying torsional loads to the center of the carriers. The mounting points of the carriers were defined as fixed and 600 ft. lbs. of torque was applied (analysis was performed on one carrier and the load was split from 1200 ft. lbs. torque of differential). The result, was approximately a safety factor of 2, which was more than enough considering the carriers would not be subject to all of the torsional force; there would be displacement of forces through the joints and other components. The aluminum differential carriers mount to the lower, steel carrier plate of the sub-frame using 4130 steel tabs. The top end of the carriers mount to the dual slider cranks and through adjustment of the cranks, the carriers pivot about the lower carrier mounts. Originally, the dual slider cranks were designed to mount to the lower sub-frame cross members with double sided clevises that extended up to the differential carriers. However, during the

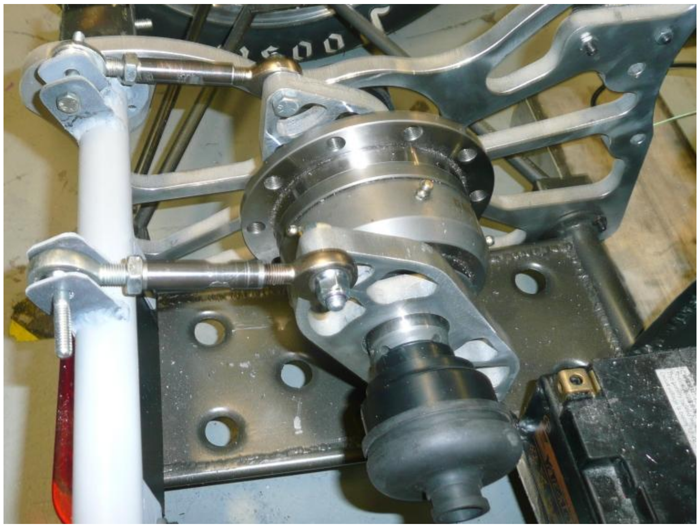

Figure 97: Fabricated Chain Tensioning System

fabrication process, the clevises proved to be oversized for the space constraints and custom double heim slider cranks (similar concept of the push rods) were used. These slider cranks mount to the upper rear sub-frame cross member using 4130 steel tabs. The reverse thread of the dual slider cranks allows for easy adjustment with a wrench to provided infinite adjustability of the drive shafts within the boundary conditions of the sub-frame. Therefore, with this system, the possibility of slip from tensioning nuts vibrating loose has been eliminated, as well as the awkwardness of large differential carrier plates and awkward tensioning system location. The redesigned chain tensioning system provides weight savings, reliability, and ease of adjustment; critical characteristics for the competition environment that the vehicle was designed for.

Sprocket

The sprocket is the way which the power is transferred from the engine to rear wheel on a motorcycle. We use the same method for our FSAE car except with different ratios. A bike has a fairly large engine sprocket and a slightly larger wheel sprocket in order to have a wide range of speeds. In order to optimize the engine speeds for our application we use a different final drive ratio between this sprockets in our design.

Sprocket design and ratio is geared toward making the transmission more effective and reducing the weight. These design goals are achieved through use of aluminum 6061 for lighter weight and the final drive ratio is determined by matching the top speed of the car with the assumed top speed that will be reached on the competition to utilize all six gears. Desired top speed is selected as 80 mph from the top speeds reached during previous competitions. Therefore 11 tooth front and 45 tooth rear sprockets is chosen to obtain 4.09 final drive ratio to achieve 88 mph top speed.

Gas Tank

Previous years' attempts to design a fuel tank resulted in large and awkward containers that were difficult to install, remove, and access. The 2007-2008 fuel tank was designed for easy access and proper volume for the competition endurance race. The design called for corrosion and fire resistant material due to flammable nature of gasoline; therefore, 1/16th inch sheet metal aluminum was selected. The target volume of the fuel tank was 1.75 gallons, just over the estimated fuel usage in the endurance race assuming worst case scenario of less than 20 miles per gallon (even this is an overestimation). Tolerances and Formula SAE rules determined the final geometry of the fuel tank, including the placement of the filler neck. One major consideration was that the fuel tank should be designed to be removable from the vehicle without having to remove the exhaust system or engine. Considering the fuel tank was to be mounted and placed behind the seat to the left side of the vehicle, easy removal was a challenge. Therefore, the fuel tank was designed to be narrow enough to contain the fuel pump and short enough to not come in contact with the exhaust headers. After several tolerance checks with the engine, headers, seat, and ground, the overall geometry was finalized. The fabrication of the fuel tank was performed using a 1/16th inch piece of sheet metal from which the expanded geometry of the entire tank was drawn on. The sheet metal was then cut using a vertical saw, bent to the appropriate angles, and welded. As one last check for clearances, the fuel tank was placed in the car in the appropriate position and the filler neck position was marked appropriately. An adaptor for the fuel pump was cut from ¼ inch aluminum and welded to the base of the fuel tank once six 0.25-20 threaded holes were tapped. Once completely welded, the fuel tank was checked for leaks using water and the final volume of 1.74 gallons was verified.

Intake System

The 2007-2008 intake system was a complete redesign from previous years. This year, a carbon fiber 1800cc plenum was utilized for light weight and ease of manufacturability. In depth CFD analysis, using COSMOS FlowWorks, was performed to optimize runner length, plenum size, trumpet placement, and angle of entry of the restrictor. The complete design was performed by the graduate students mentioned in the Authorship section of this paper and a separate detailed report of the intake was written. The fabrication of the intake system was performed by the graduate students as well as members of the 2007-2008 Formula SAE team.

Aerodynamics

This year aerodynamics was a very important part of our finished vehicle. A separate MQP was dedicated to the research, testing and implementation of the body and front and rear wings as well as a diffuser to be used on the car this year. More details can be found on this section of the vehicle by referring to the 2008 FSAE Aerodynamic Design MQP.

Conclusions/Recommendations

This year's FSAE team managed to improve upon many of the shortcomings that arose in the vehicles of previous years. The team has constructed a list of recommendations to help future teams improve upon the undesirable aspects of the 2008 car. There are a number of design changes that team members would make if given the chance to repeat the process. These are:

- Design a larger cockpit for the driver. The frame was built to accommodate the 95th percentile male, but taller drivers found the seating arrangement to be cramped.
- Perform FEA on the pedal system. This would aid in material removal from the pedals.
- Look at other designs for the control arms. Consider airfoil profiles, aerodynamic covers, and alternative materials.
- Perform weight reduction of the engine.
- Place and weld the harness bar higher relative to where the driver's shoulders will be. SAE maintains strict regulations regarding the driver restraints.
- Using a shorter wheelbase would bring the car's wheelbase-to-track-width ratio closer to the Golden Ratio, improving handling characteristics and turning.
- Implementing steering rack placement and attachment of the tie rods near the top of the vehicle would alleviate the tight packaging situation if using pull rods in the front of the vehicle.
- The rear rocker mounting system for the monoshock setup could have been better optimized in terms of weight.
- Carefully considering material benefits versus cost would have helped the team to more efficiently use their budget by using less expensive materials wherever possible.
- Rear hubs should be machined by the team, rather than borrowed from an existing vehicle.
- Research lightweight and less expensive wheels.
- Use the best of the work conducted by the previous year's team instead of completely starting from scratch.

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Appendices

Cost Report

On electronic copies double click below to open the full cost report.

Worcester Polytechnic Institute

Tech Racing #53

2008 Formula SAE Cost Report

Weight Transfer Sheet

Designation	Front		Total		Rear	
Measured						
Weight (lbs)	WF=	270	W=	600	WR=	330
Un-sprung Weight (lbs)	UWF=	42			UWR=	55
Un-sprung CoG Height (in.)	UGF=	10			UGR=	10
Track Width (in.)	TF=	42			TR=	38
Roll Center Height (in.)	CF=	1.23			CR=	1.11
Sprung CoG Height (in.)	SGF=	10	SG=	12	SGR=	13
Proportional Roll Stiffness	DrF=	0.5			DrR=	0.5
Wheel Base (in.)			WB=	68		
Calculated						
Sprung Weight (lbs)	SwF=	228	Sw=	503	SwR=	275
Un-sprung Weight Transfer (lbs)	UtF=	10			UtR=	14.47368
Weight Transfer via Roll Center (lbs)	CtF=	6.677143			CtR=	8.032895
Proportion of Sw on Rear (%)			WDR=	0.5467		
Mean Track Sw (in.)			TM=	39.813		
Mean Roll Center Sw (in.)			CM=	1.1644		
Mean CoG Sw (in)			GM=	11.64		
Mean Roll Moment (in.)			LM=	10.476		
Weight Transferred Sprung Mass (lbs)			St=	132.35		
Total Weight Transfer Sideways (lbs)	WtF=	82.85269	Wt=	171.53	WtR=	88.68212
Weight Transfer Front to Back	WtoF=	5.662023			WtoR=	-5.66202

1	g Constant Cornering	
	Inside	Outside
Front Force (lbs)	54.98	220.68
Front Percent	9.2%	36.8%
Rear Force (lbs)	73.49	250.85
Rear Percent	12.2%	41.8%

1	g Accel	
	Inside	Outside
Front Force (lbs)	82.06	82.06
Front Percent	13.7%	13.7%
Rear Force (lbs)	217.94	217.94
Rear Percent	36.3%	36.3%

1.5	g Decel	
	Inside	Outside
Front Force (lbs)	214.41	214.41
Front Percent	35.7%	35.7%
Rear Force (lbs)	85.59	85.59
Rear Percent	14.3%	14.3%

Design Report

2008 Worcester Polytechnic Institute Formula SAE Design Report

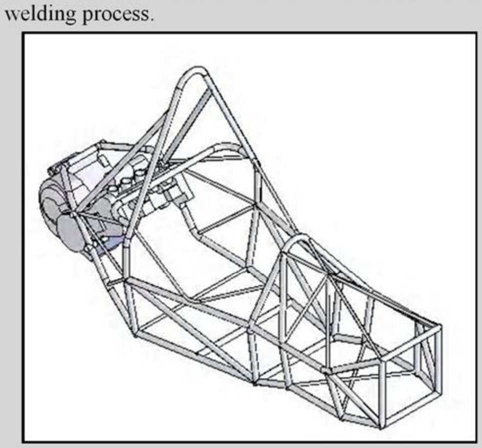
Introduction

WPI has seen a variety of success since the teams inception in 1987. It was decided early on that the 2008 vehicle will be one of a simple, yet competitive and reliable design, focusing on proper design and optimization of the necessary components without over complication of the vehicle design. This approach required a complete restructuring of the team and the program. The result is a car radically different than any vehicle WPI has produced in the past.

Chassis

The 2008 vehicle is built upon an AISI 4130 Steel space frame with a billet aluminum rear sub-frame, utilizing the engine as a partially stressed member. The space-frame and sub-frame were then optimized using Finite Element Analysis in order to ensure adequate strength and rigidity, while reducing overall weight.

The chassis was joined using Gas Tungsten Arc Welding (GTAW) using an inert 4130 filler rod by a skilled welder. Upon completion of the construction, the frame was stress relieved in order to ensure the strength of the material was not diminish due to the heat of the welding process.



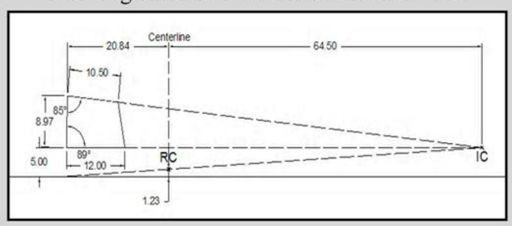
Front Chassis Design

The rear sub-frame is constructed using 2 billet 7075 T6 plates connected with AISI 4130 steel cross members. By incorporating the Aluminum sub-frame into the chassis design a 20% decrease in overall weight was possible. The overall chassis weight including the rear sub-frame is 58 lbs, over a 35% reduction in weight from the 2007 and 2006 chassis designs.

Suspension

Geometry and Control Arms

The suspension design this year was one of the significant changes from past years and was one of the major focuses design the design process. Coordinating with the chassis design, a goal wheel base was set of 68 inches with a target front and rear track width of 42inches and 38 inches, respectively. This provides a wheelbase to track width ratio of 1.5, staying slightly under the "golden ratio" of 1.618 to provide a high level of vehicle maneuverability while still offering high speed stability. The lower control arms were calculated to be 12 inches long based on the chassis width and track.



Roll Center Diagram

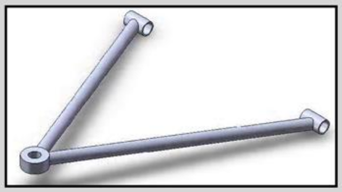
The desired roll center is one that is as close to the ground as possible but that would not cross the plane of the ground during the expected suspension travel. While considering the rules mandate the suspension must have a travel of one inch in compression and rebound it was decided that the suspension geometry would be tuned to allow three quarters of an inch in compression and rebound within the desired operation, with the capability to overextend in either direction to a little over an inch, thus satisfying the minimum requirements.

The result is a static roll center 1.22 inches above the ground which moves to just over a tenth of an inch off the ground under compression. This necessitated that the front upper control arms would be 10.5 inches long.

To simplify the design and manufacturing and lower production costs the rear control arms and sub frame placement were design to use the same effective lengths as the front. Due to the slightly narrower track in the rear the roll center is moved lower slightly but still remains above ground plane during driving conditions.

For the rear control arms a 60 degree angle chosen to aid balancing the forces on each member when under load. The front control arms utilize a 45 degree angle between members, allowing the required steering angle while also controlling the forces in the control arm members. All of the control arms are made of 0.5 inch AISI 4130 Steel with varying sidewall thickness based on Finite Element Analysis performed with the Cosmos Works package in Solidworks. In order to ensure a reliability and durability various tests were performed using higher loading situations and scenarios similar to contacting a curb or cone as well as improper handling or securing of the vehicle by the control arms. Full weight transfer calculations based on theoretical values were performed to ensure accurate numbers for all testing.

To eliminate rod end in bending all heim joints were eliminated from the control arms. A delrin busing with a brass sleeve is utilized for the inboard mount while a spherical bearing is pressed into the control arm at the outboard location. A fixture for the front and rear control arms was designed and fabricated to ensure proper tolerances and dimensions were maintained and provide a repeatable process from which the control arms may be produced.



Control Arm With Bushings

Uprights

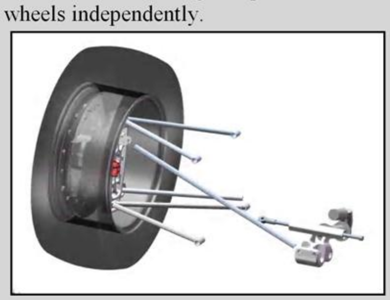
The prototype vehicle carries billet 7075-t6 aluminum uprights weighing in at 1.3 lbs each with and incorporate many unique features. The design allows a single casting to be used for a production version of the front and rear upright, with slightly varying finish machining operations. The most notable feature is the adjustable camber method utilized through manipulation of the upper control arm mounting point. The mounting point of the upper control arms uses a slot with a separate plate to ensure proper clamping forces are still maintained allowing infinite adjustment between 0 and –2 degrees of camber. The uprights also contain an integral caliper mount and steering point. This minimizes the weight and complexity of the part and would aid in the mass production of the vehicle.

Front Upright

The design for the vehicle currently incorporate castor adjustment as a set castor value was desired for the prototype vehicle. However the uprights could be altered slightly on the bottom to incorporate a slot from front to back to allow adjustment of the vehicles castor to suit individual driver's desires and operation requirements.

Dampers

The 2008 vehicle utilizes a unique monoshock arrangement in the front and rear. This arrangement allows the roll and compression/rebound forces to be separated and independently tuned. The addition of a independent roll damper also provides adjustable damping for in roll as well. Additionally, this layout reduces weight and cost. The pull/push rods from the uprights or control arms are connected to a single rocker, from which all loads are transferred either by translation of the rocker, into the roll springs, or by rotation about a shaft, into the shock controlling compression and rebound. Such an arrangement allows for an infinite adjustment in roll, without affecting the compression/rebound of the vehicle. This allows the vehicle to remain flat through a corner while still allowing compliance and movement of the wheels independently.



Pull Rod Mono-Shock Assembly

The 2008 vehicle utilizes Cane Creek Double Barrel dampers which allow both high and low speed adjustment in compression and rebound. This provides a nearly infinite range of adjustment for all conditions the car may see.

Steering

The steering design of the 2008 vehicle focused on reducing bump steer while also providing a more ergonomic mounting location of the steering rack than has previously been used. Ackerman angle was secondary to these design criteria.

By placing the rack lower and further aft in the vehicle the drivers feet easily clear the rack. This set up also allows the bump steer to be minimized, while providing adequate Ackerman angle for vehicle maneuverability and driver feedback.

Brakes

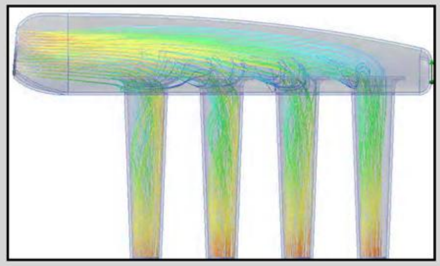
The vehicle uses 4 wheel independent outboard disk brakes with Polaris single piston floating calipers. The rotors are attached to the hubs and all loads were analyzed using finite element software.

Powertrain

WPI Formula SAE vehicle has historically used Honda CBR 600cc engines, and the same is true for the 2008 Vehicle. The Vehicle utilizes a 599cc engine from a Honda CBR 600 F4i. This engine was selected because of the great reliability, compatibility and access of OEM and aftermarket parts, and the relative ease of tuning and modification of the engine. The internals of the engine were left stock to retain reliability of the powertrain package.

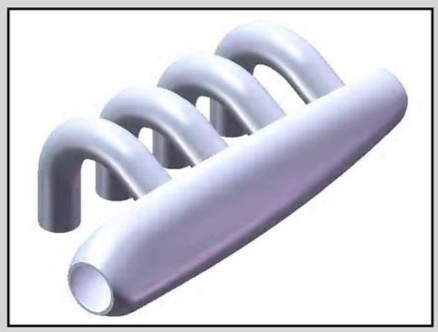
Intake and Exhaust

The intake system utilizes a 40mm Aluminum Throttle body with a delrin barrel type valve. The use of the barrel in the throttle body design minimizes downstream turbulence found with the use of typical butterfly type valves and offers and much better linearity. This allowed the team to achieve maximum flow through the restrictor at full throttle. The throttle body is followed by the restrictor tube, with an 8 degree from the mandated 20mm restrictor to an opening of just over 1.5 inches inlet to the plenum. The length of the restrictor tube was designed and analyzed to maximize airflow into the plenum while reducing turbulence.



Plenum CFD Analysis

Hand calculations were first used to determine geometry and then the entire system was optimized using Computational Fluid Dynamic (CFD) analysis. Careful analysis of the plenum was performed to ensure even flow to all four cylinders. After all iterations were complete a plenum size of just over 2 times the displacement of the engine was decided upon which provide even flow to all cylinders.



Plenum and Runner Design

Air will be fed into the runners via trumpeted velocity stacks with their inlet 0.5 inches from the bottom of the plenum. This will ensure reduce turbulent flow into the runners, while ensure no airflow is lost and aid in the even airflow of all runners. The length and diameter of the runners were calculated for peak torque at 11,000 RRMs and the lengths to utilize the second pressure wave set. The length of the runners, including the effective runner length of the heads, was calculate to be 13.086 inches. This resulted in a manufactured runner length of just over 9 inches.

The restrictor and plenum will be constructed of carbon fiber, reinforced with an aluminum honeycomb core. By constructing these pieces out of Carbon Fiber instead of Aluminum it was calculate that weight of the system will be reduced by an estimated 15% .

The exhaust is a 4 to 2 to 1 design utilizing slip on merge collectors constructed from stainless steel with equal length primaries and secondaries. The exhaust system is terminated with a carbon fiber cased, 16 inch Yoshimura TRC muffler. Flow through the exhaust system was optimized using Computational Fluid Dynamics.

Cooling

The cooling system of the 2008 vehicle uses dual radiators mounted inside the vehicle's side pods with electronically controlled fans. The stock water pump has been retained for reliability purposes. The driver also has the option of overriding the ECU forcing the fans on.

Engine Management System (EMS)

The engine management system is based around Performance Electronics LTD. PE-ECU. This ECU was selected because of its small size and weight, ease of use, adjustability and cost. The intake air temperature sensors, coolant temperature sensor, and manifold absolute pressure sensors were changed to standard GM sensors while all other stock sensors were retained.

Key features of the engine management system include launch control, which, when the clutch is depressed, activates a secondary rev limiter. A wideband lambda sensor has also been integrated, using one of the ECUs analog inputs and aids in controlling the fuel system and ensuring maximum performance from the engine and vehicle. The ECU relays all critical data and warnings to the driver via LED warning lamps.

By integration with the electric over air pneumatic shifting system, automatic up-shifting is also achieved. The driver may activate the system through a cock-pit mounted toggle switch. When the specified RPM and driving criteria are met, the ECU will output a signal to shift controller to shift up.

Drivetrain

The differential used on the 2008 vehicle is a Quaife Automatic Torque Biasing differential originally intended for a Honda 1.8 liter front wheel drive vehicle and will be chain driven via a sprocket mounted to the differential housing. The final drive ration is 4.25:1 which results in a top speed of just over 80 mph.

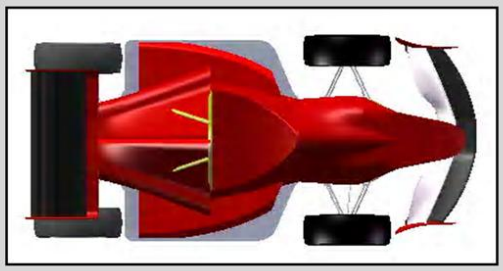
A pre-fabricated differential was selected to reduce manufacturing time and cost of the prototype and increase reliability. The Quaife differential offers an excellent torque biasing ratio while the sealed grease lubricated design offers a robust and reliable leak-free unit.

The unit is mounted in sealed roller bearings in a 6061 T6 aluminum carrier. Tension of the chain is maintained using an inverter slider-crank design using the carrier as one of the links.

Power is transmitted to the wheels through unequal length and diameter AISI 4130 steel half-shafts. The shafts are fully heat treated after welding and designed to ensure equal angles of twist in both shafts.

Aerodynamics

The aerodynamic package of the vehicle includes a front wing, rear wing, under car diffuser, and full body, including a rear cover. The front wing is comprised of 2 elements and is mounted below the nose. The lift to drag ration of the front wing in 5.9 with an 8 degree angle of attach. The rear wing consists of 3 straight elements. The wing profile for both the front and rear wings are custom, base on the Selig 1223 profile.



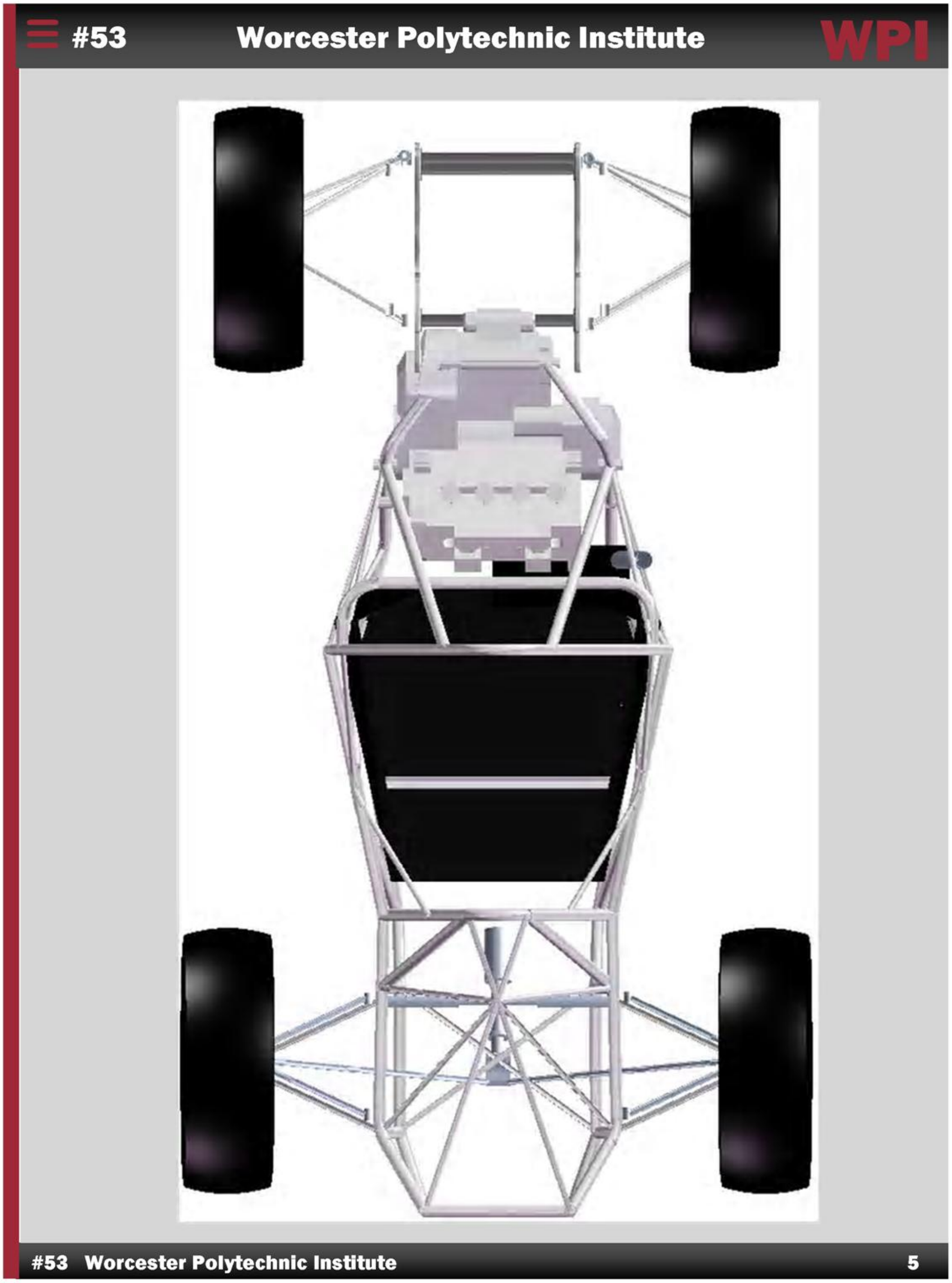
Full Body, Front and Rear Wings and Diffuser

All body work and wings are constructed from carbon fiber, with the wings utilizing an integrated I-beam shaped spar to minimize weight while providing strength and stiffness.

Miscellaneous Design Features

Shift Actuation

The shifting mechanism for the 2008 vehicle is an electric over air pneumatic system. Shifting is actuated using the paddles mounted behind the steering wheel. These paddles will close a circuit, sending a signal to the electronic shift controller. The drive may also choose to activate that automatic up-shifting feature and allow the ECU to control the up-shifting of the vehicle.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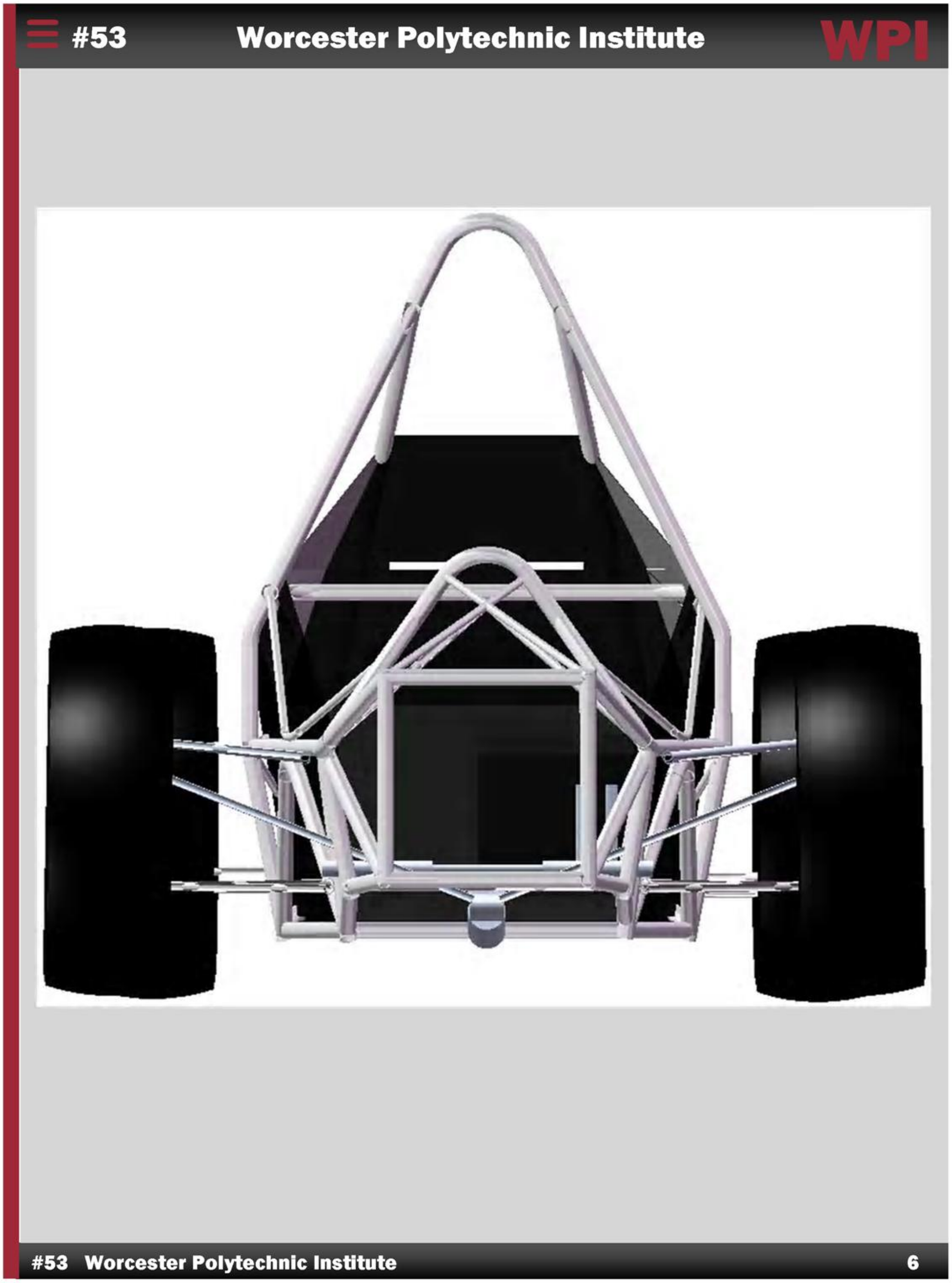
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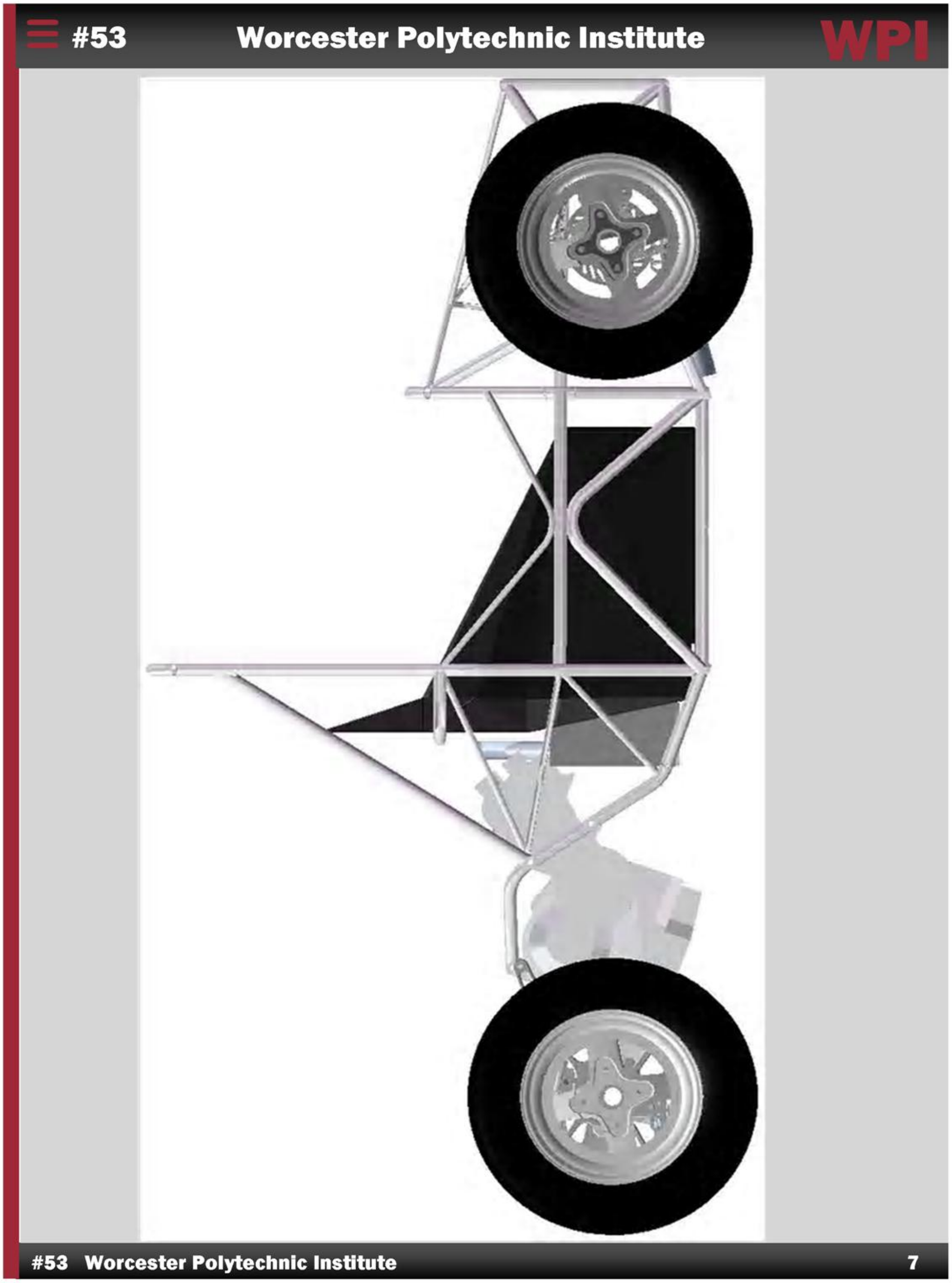
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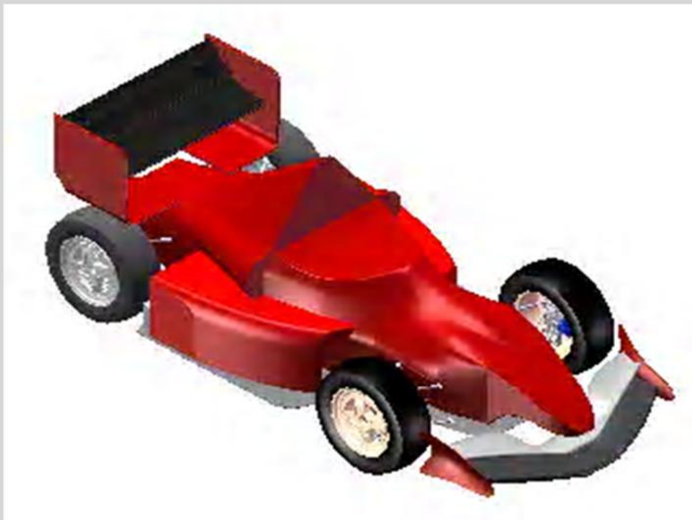
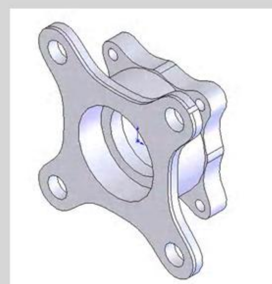


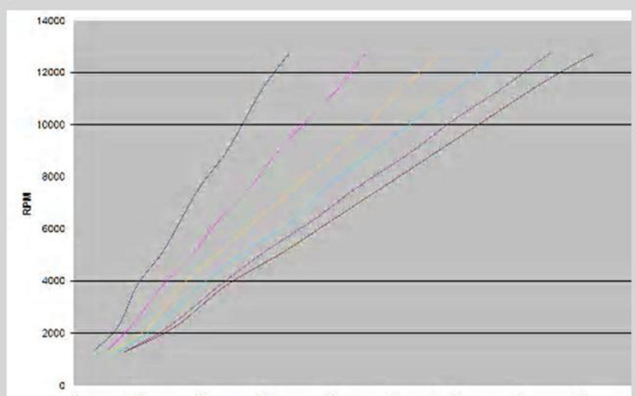
Intake Restrictor Tube



Front Wheel Hub

Plenum - Iteration 9			
Inlet Velocity	1200 in/s		
Plenum Volume	1741 cc/106.25 in^3		
	Volumetric Flow Rate (in^3/s)	Mass Flow Rate (lb/s)	% Dev
			4.7 %
Cylinder 1	841	0.036	
Cylinder 2	869	0.0373	
Cylinder 3	879	0.0377	
Cylinder 4	882	0.0379	

Plenum Analysis Report



Gearing and Speed Plot

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FSAE Design Spec Sheet 2008

Car No.	53
School	Worcester Polytechnic Institute (WPI)

Dimensions	Front	Rear
Overall Length, Width, Height	89 inches, 48 inches, 43 inches	
Wheelbase	68 inches	
Track	42 inches	38 inches
Weight with 150lb driver	270 lbs (est)	330 lbs (est)

Suspension Parameters	Front	Rear
Suspension Type	Non-Parallel unequal length A-Arm. Pull Rod monoshock with Cane Creek 4 way adjustable damper. Roll damping integrated on rocker	Non-Parallel unequal length A-Arm. Push Rod monoshock with Cane Creek 4 way adjustable damper. Roll damping integrated on rocker
Tire Size and Compound Type	20x7-13 R25A Hoosier	20x7-13 R25A Hoosier
Wheels	13 inch 3 pc Al Kaiser Wheels	13 inch 3 pc Al Kaiser Wheels
Design ride height (chassis to ground)	1.0 inches	1.0 inches
Center of Gravity Design Height	12 inches	
Suspension design travel	1.0 inches jounce/ 1.0 inch rebound	1.0 inches jounce/ 1.0 inch rebound
Wheel rate (chassis to wheel center)	141.75 lb/in (not final- multi spring package)	162 lb/in (not final- multi spring package)
Roll rate (chassis to wheel center)	Adjustable- pending testing	
Sprung mass natural frequency	2.79 Hz (not final- multi spring package)	2.71 Hz (not final- multi spring package)
Jounce Damping	High and Low speed Adjustable	High and Low speed Adjustable
Rebound Damping	High and Low speed Adjustable	High and Low speed Adjustable
Motion ratio / type	0.9 / Linear	0.9/ Linear
Camber coefficient in bump (deg / in)	0.7 deg / in	0.7 deg / in
Camber coefficient in roll (deg / deg)	0.82 deg / deg	0.82 deg /deg
Static Toe and adjustment method	-0.25 to 0.25 inch toe via adj steering links	-.025 to 0.25 inch toe via adj toe links
Static camber and adjustment method	1 deg static, adjustable from 0-2 deg slotted upright	1 deg static, adjustable from 0-2 deg slotted upright
Front Caster and adjustment method	5 degrees non-adjustable	
Front Kingpin Axis	1 deg static, equivalent to camber	
Kingpin offset and trail	.913 inches offset 0 inches trail	
Static Akermann and adjustment method	10% non-adjustable	
Anti dive / Anti Squat	15%	5%
Roll center position static	1.22 inches above ground	1.2 inches above ground
Roll center position at 1g lateral acc	Pending Testing and Tuning	Pending Testing and Tuning
Steer location, Gear ratio, Steer Arm Length	Rear Steer, 1 inch above lower control arm, 12:1 Ratio 12 inch steer arm	

Brake System / Hub & Axle	Front	Rear
Rotors	7.75 inch dia. Cross Drilled Polaris ATV Rotors	7.75 inch dia. Cross Drilled Polaris ATV Rotors
Master Cylinder	One 1/2 inch bore Polaris master cylinder for each front and rear. Adjustable bias bar	
Calipers	1.187 inch dia., Floating single Piston Caliper	1.187 inch dia., Floating single Piston Caliper
Hub Bearings	Taper Roller Bearings	Taper Roller Bearings
Upright Assembly	CNC 7075-Al, integral caliper mount	CNC 7075-Al, integral caliper mount
Axle type, size, and material	Fixed spindle, 1 inch dia, 4140 steel	Rotating axle, 1.075 inch diam, 4130 steel

Ergonomics	
Driver Size Adjustments	Fixed seat and steering wheel. Pedals adjust through a range of 3.5 inches
Seat (materials, padding)	Carbon Fiber, Foam insert and headrest
Driver Visibility (angle of side view, mirrors?)	190 degree side visibility, side mirrors mounted to nose
Shift Actuator (type, location)	Steering Column mounted paddle shifter, pnematic shifter activation
Clutch Actuator (type, location)	Foot pedal, cable actuated
Instrumentation	Dash mounted temp and oil warning light, shift light, neutral light

Frame	
Frame Construction	Main structure tube frame with an aluminum rear subframe
Material	1"/ 0.5" DIA AISI 4130 Steel Tubing / Billet 7075 Aluminum rear subframe with 1" AISI 4130 Supports
Joining method and material	Gas Tungsten Arc Welding, Inert 4130 Filler Rod
Targets (Torsional Stiffness or other)	2500 ft-lb / deg
Torsional stiffness and validation method	3500 ft-lb/deg FEA beam element model
Bare frame weight with brackets and paint	58 lbs
Crush zone material	Aluminum Honeycomb
Crush zone length	200mm / 7.87 inches
Crush zone energy capacity	Calculations --80719 in*lbs (91200 Joules)

Powertrain	
Manufacture / Model	2004 Honda CBR600-F4i
Bore / Stroke / Cylinders / Displacement	62mm/48.5mm/4 cylinder/ 599cc
Compression ratio	12.5:1
Induction	Naturally Aspirated
Throttle Body / Mechanism	40 mm Barrel Throttle Body
Fuel Type	93 Octane
Max Power design RPM	11,500
Max Torque design RPM	10,500
Min RPM for 80% max torque	8000
Fuel System (manf'r, and type)	Electronic Semi Sequential Fuel Injection
Fuel System Sensors (used in fuel mapping)	Air Temp, Coolant Temp, Throttle Pos, Crank Pos, MAP, Lambda/oxygen
Fuel Pressure	50 psi
Injector location	5 inches before and pointing toward intake valve
Intake Plenum volume and runner length(s)	1740 cc/9 inches
Exhaust header design	4-2-1 equal length (+/- 0.25"), 1.5"/2.25" collector
Effective Exhaust runner length	14 inches
Ignition System	Performance Electronics PE-ECU-1 engine management system
Ignition Timing	3-D map, RPM and Throttle position, 45 deg BTDC max advance (Tuning still in progress)
Oiling System (wet/dry sump, mods)	Wet Sump, Mechanically Driven Oil Pump
Coolant System and Radiator location	twin side pod mounted radiators with thermostatic controlled electric fans
Fuel Tank Location, Type	Mounted to firewall between seat and engine
Muffler	Yohismura Carbon Fiber, 2 liter volume
Other significant engine modifications	

Drivetrain	
Drive Type	Chain
Differential Type	Quaife ATB Limited Slip
Final Drive Ratio	4.25:1
Vehicle Speed @ max power (design) rpm	82 mph @ 11500 RPM
1st gear	34 mph
2nd gear	46 mph
3rd gear	58 mph
4th gear	67 mph
5th gear	75 mph
6th gear	82 mph
Half shaft size and material	1 inch OD, .035 Wall Thickness, AISI 4130 Steel
Joint type	Honda Civic tulip style on the inboard, 86mm Stub Axle

Aerodynamics (if applicable)	
Front Wing (lift/drag coef., material, weight)	5.9, Carbon Fiber, TBD
Rear Wing (lift/drag coef., material, weight)	5.9, Carbon Fiber, TBD
Undertray (downforce/speed)	Pending Analysis
Wing mounting	Front: Struts to Nose Cone Assembly, Rear: Struts to sub-frame assembly