A Thesis Presented to

The Faculty of Alfred University

Design, Analysis and Manufacture of a Tube Chassis and Related Components for SAE Baja

By

Karl Huber

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Under the Supervision of

Chair:

Assistant Professor Mechanical Engineering, Dr. Amit Maha

Committee Members:

Technician/Machinist, Mechanical Engineering, Jim Mighells

Associate Professor, Mechanical Engineering, Dr. Ehsan Ghotbi

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I. Introduction

The Society of Automotive Engineers (SAE) holds yearly International Collegiate Design Series (CDS) competitions. According to SAE, the CDS “takes students beyond textbook theory by enabling them to design, build, and test the performance of a real vehicle and then compete with other students from around the globe in exciting and intense competitions.” The SAE Baja competition is the most rugged of the CDS, requiring students to design and build an all-weather off-road vehicle from the ground up.

The SAE Baja competition consists of both static and dynamic events. The static events are an evaluation of the teams’ design reports and presentations, as well as the ability of the team to sell their vehicle as a potential production vehicle. Team design decisions, calculations and analysis must be well documented and well-reasoned. The dynamic events are designed to break competition vehicles. Each competition has three short dynamic events followed by a four hour endurance race. The short events test a vehicle’s acceleration, handling, suspension, and the ability to put power to the ground in hill climbs and sled pull events. The endurance race is a comprehensive test of vehicle performance and durability.

Unlike many other CDS competitions, SAE Baja is a holistic competition from design to execution. As opposed to modifying existing vehicles or production parts, the majority of a Baja vehicle is original custom design and manufacture. Vehicles consist of a few unique subsystems.

The frame of the vehicle is a custom designed welded steel, tube chassis, not dissimilar to custom roll cages used in professional off-roading. SAE tightly regulates the design and construction of the vehicle frame. The frame is a major source of vehicle weight and the excessive material in a poorly designed frame will lead to poor performance in acceleration, handling, and endurance events. Frame manufacture can be a considerable project for teams without access to CNC tube benders and notchers.

Vehicles typically employ independent suspension systems. Suspension arms, uprights, and knuckles one of the most common source of failure in competition. These components are directly targeted in suspension events featuring rock crawls, drops, jumps, whoops, and impact obstacles like railroad ties and telephone poles. The awkward center of gravity of most Baja vehicles requires creative suspension design to move vehicle roll centers and create desirable handling characteristics.

Drivetrain teams design custom gearboxes, yolks and axle shafts. Variable gear ratios are usually accomplished with continuously variable transmissions. While CVT’s are production devices, CVT tuning with belt tension, spring rate, spring preload, and design of flyweights pose a unique engineering challenge to drivetrain teams. Another step-down in gearing is required after the CVT and gearboxes are required.

The final subsystem of mist Baja vehicles is the Handling subsystem. This encompasses brakes, steering rack, and tie rods. The ability of a vehicle to lock its brakes and stop within 10 feet after a 100 foot acceleration is a key part of technical inspection. The handling teams must communicate closely with suspension in design of the upright in order to produce competitive turning radii and other steering and handling qualities.

In many aspects, the SAE Baja competition mirrors professional and amateur competitive off-roading sport, particularly in the practice of technical inspection. SAE Tech Inspection is the process where competition officials inspect vehicles with a fine tooth comb. The frame subsystem is one of the most heavily evaluated. Weld quality is examined, welders must submit weld samples for destructive testing. Safety features like firewalls, splash shields are tested for efficacy. Clearances and shielding all over the vehicle are picked apart. Vehicles that fail tech inspection can try to fix the violations and go through again, all though this is particularly time consuming in a three day competition. Technical inspection plays a crucial role in design series and directly affects the meta-competition of vehicle design. To be competitive, vehicles are designed on the bleeding edge of what is allowable by the competition.

The 2019-2020 Baja season saw the introduction of a new opportunity for vehicles, four wheel drive (4WD). Baja competition vehicles have not changed much in the last five years, well performing schools have highly tuned and perfected vehicle designs. These schools make minor changes to their vehicle each year for the sake of change, but functionally the top ten competition vehicles have been largely unchanged for the past few years. 4WD is an optional system in 2020 competitions with extra points awarded to vehicles that add this element, and 4WD systems will be required in the 2021 competition year.

The Alfred University Baja Team, Saxon Racing, decided to pursue the 4WD option. We made this decision even though we had many 2WD systems with finished or nearly finished designs at the time of SAE's announcement. My team had considered many different chassis designs and I had a completed 2WD chassis design. However, as frames can be used for two years in a row, my team need to make sure that we set up the 2020-2021 team up for success with a 4WD compatible vehicle. Furthermore, the sudden introduction of the 4WD requirement put all the teams on equal footing-with no well-tuned previous vehicles it would be very difficult for us to compete with schools that had historical success if Saxon Racing opted for 2WD in the 2019-2020 season. We also expected that competition events would be modified to reflect the presence of 4WD vehicles on the track, with easier handling sections and significantly more challenging terrain and physical obstacles. All of the work in the 2019-2020 season is distinguished by its relation to incorporating 4WD.

This thesis is a documentation of my contribution to Alfred University's Saxon Racing vehicle. I am the team lead for the Frame team. I also have made a number of contributions to the suspension team and performed organized testing for the drivetrain team. It is important to note that the Baja project is very distinctly a team project. My team has four other members, and all of them have contributed significantly to the project.

As the Frame team lead, my work on the chassis was comprehensive. My team handled design, analysis and manufacturing of the frame. Primary concerns in chassis design were guaranteeing a pass at tech inspection with as little weight as possible, and having a design that was compatible with the 4WD drivetrain components as well as well as being able to adapt to the multilink rear suspension system. Analysis on unique or novel members in my frame was performed with FEA software. Team Captain, Austin Gibson and I manufactured our welded tube chassis in house, and worked to resolve the numerous manufacturing challenges together to create a finished frame.

Additionally, throughout the season I offered input on the design of the front suspension components, most notably in creating an upright that was capable of being manufactured. My experience with ANSYS FEA led me to spearhead the analysis of suspension components using the software.

II. Chassis Design

A. Design Constraints

The chassis for Saxon Racing's 2020 vehicle was a new design from the ground up. The largest new problem raised in the 2020 competition season was the problem of incorporating a four wheel drive (4WD) system. There must be adequate space for all additional elements, and space for the driver while considering the 4WD elements. Furthermore, I have taken steps to mitigate historical problems associated with the chassis. Understeer and sluggish handling were considered. Also considered was the excessive weight caused by extraneous or overbuilt members. Finally, my team and I manufactured our chassis in house, which is particularly labor intensive process, and so the chassis was designed to increase ease of manufacture and assembly.

1. Structural Material Requirements

Rule B.3.2.16 specifies two types of members, Primary and Secondary. Each of these members has a specified location of use and a required material specifications.

The material used for the Primary Roll Cage Members and bracing must meet one of the following requirements:

- *Circular steel tubing with an outside diameter of 25 mm (0.984 in) and a wall thickness of 3 mm (0.118 in.) and a carbon content of at least 0.18%.*
- *A steel shape with bending stiffness and bending strength exceeding that of circular steel tubing with an outside diameter of 25 mm (0.984 in.) and a wall thickness of 3 mm (0.118 in.). The wall thickness must be at least 1.57 mm (0.062 in.) and the carbon content must be at least 0.18%,*

Further, the Rulebook specifies bending stiffness and bending strength equations:

Bending stiffness, kb, is given by:

$$k_b = EI$$

(Eq. 1)

Where:

E - Modulus of elasticity (205 GPa for all steels)

I - Second moment of area for the structural cross section about the neutral axis

Bending strength, Sb, is given by:

$$S_b = \frac{S_y I}{c}$$

(Eq. 2)

Where:

Sy - Yield strength (365 MPa for 1018 steel)

c - Distance from neutral axis to extreme fiber

The standard steel tubing specified by SAE is not ideal for building a strong or lightweight frame. Requisite bending strength and bending stiffness can be achieved with 1 ¼" x 0.065" 4130 "Chromoly" Steel Tubing, as shown below:

Bending Stiffness

Definitions:

E = Modulus of Elasticity (205 GPa for all steels)

I = Second Moment of Area for the

Requirement Definitions: 25.0mm x 3.00mm, 1018

D_o = 25.0mm

D_i = 19.0mm

I = $(\pi/64)*(D_o^4 - D_i^4)$

= $(\pi/64)*(25.0^4 - 19.0^4)$

= 1.28E+04 mm^4

= 1.28E-08 m^4

$K_{b,req}$ = E*I

= (205GPa * 1.28E-08 m^4)

= 2.62E+03 $N*m^2$

Bending Strength

Definitions:

S_y = Yield Strength (minimum specification value)

C = Distance from the neutral axis

Requirement Definitions: 25.0mm x 3.00mm, 1018

S_y = 365MPa

C = 12.5mm

= 0.0125m

$S_{b,req}$ = $(S_y * I)/C$

= (365MPa * 1.28E-08 m^4) / (0.0125m)

= 3.74E+02 N*m

Design Definitions: 31.75mm x 1.65mm, 4130

D_o = 31.75mm

D_i = 28.45mm

I = $(\pi/64)*(D_o^4 - D_i^4)$

= $(\pi/64)*(31.75^4 - 28.45^4)$

= 1.77E+04 mm^4

= 1.77E-08 m^4

$K_{b,req}$ = E*I

= (205GPa * 1.77E-08 m^4)

= 3.63E+03 $N*m^2$

Bending Strength

Definitions:

S_y = Yield Strength (minimum specification value)

C = Distance from the neutral axis

Design Definitions: 31.75mm x 1.65mm, 4130

S_y = 435MPa

C = 15.9mm

= 0.0159m

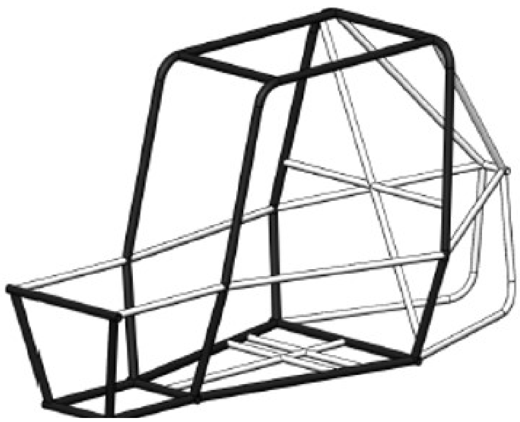
$S_{b,req}$ = $(S_y * I)/C$

= (435MPa * 1.77E-08 m^4) / (0.0159m)

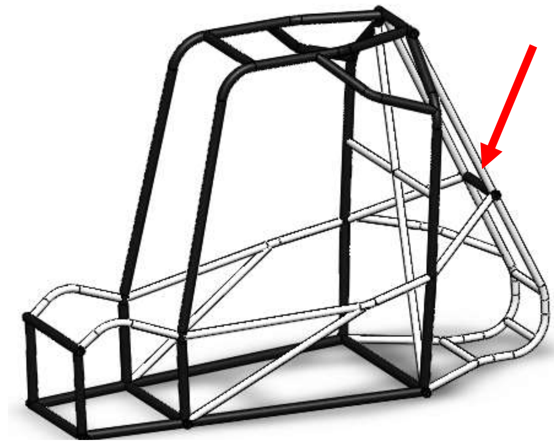
= 4.84E+02 N*m

2. Primary and Secondary Location

The rulebook specifies locations of where Primary and Secondary tubing should be used. Members meeting Primary material specifications may be used anywhere on the frame. Secondary material may only be used in the specified areas. Primary members are shown in black, Secondary members are shown in white.



(Figure 1- Rulebook Rear Braced Frame)



(Figure 2- Saxon Racing's Frame)

There are some key differences in the Saxon Racing's use of Primary and Secondary material. Notably our Rear Lateral Cross (indicated in Figure 2) is a primary member to satisfy lateral cross requirements, to serve as a mounting point for the fuel tank, and as the third point in the structural triangle constituting the Aft Bracing system. (See Appendix B)

3. Tubing Member Requirements

If any of the current design's tubing member requirements do not satisfy design rules, additional tubes must be added to support the offending member. Adhering to the requirements will reduce the need for extra members; these requirements must be kept in mind during design.

Rule B.3.2.1, includes the restrictions on the unsupported length of tubes and maximum bend angles.

Roll cage members must be made of steel tube and may be straight or bent. Straight members may not extend longer than 1016 mm (40 in.) between Named Points or comply with Rule B.3.2.4 - Additional Support Members. Bent members may not have a bend greater than 30 deg. that does not occur at a Named Point; and may not extend longer than 838 mm (33 in.) between Named Points or comply with Rule B.3.2.4 - Additional Support Members.

Rule B.3.2.4 - Additional Support Members outlines additional bracing requirements for members that do not satisfy these requirements.

4. Geometric Requirements

Geometric requirements on the frame primarily relate to the clearances between the driver and the envelope of the vehicle. While wearing a helmet, the driver's head must have 6" of clearance with envelope, and the driver's body must have 3" of clearance with the envelope of the Baja frame. These restrictions frequently result in the use of gussets on the frame to extend the envelope of the frame.

There are additional geometric requirements that specify dimensions that must be met regardless of the driver. These rules include:

B.3.2.6 - RRH – Roll Hoop
The RRH is a planar structure behind the driver's back and defines the boundary between the front-half (fore) and rear-half (aft) of the roll cage. The driver and seat must be entirely forward of this panel. The RRH is substantially vertical but may incline by up to 20 deg. from vertical. The minimum width of the RRH, measured at a point 686 mm (27 in.) above the inside seat bottom, is 736 mm (29 in.). The vertical members of the RRH may be straight or bent and are defined as beginning and ending where they intersect the top and bottom horizontal planes (points AR and AL, and BR and BL in Figure B-8). The vertical members must be continuous tubes (i.e. not multiple segments joined by welding). The vertical members must be joined by ALC and BLC members at the bottom and top. ALC and BLC members must be continuous tubes or adhere to B.3.2.14 - Butt Joints. ALC, BLC, RRH members, LDB and the shoulder belt member must all be coplanar.

Rule B.3.2.8.1 - Gussets for Lateral Clearance
If a gusset is used to brace the RHO and RRH to achieve the Lateral Clearance in Rule B.3.3.1 - Lateral Space the added members must be a primary material (B.3.2.16 - Roll Cage Materials); completely welded around the circumference of both ends of the gusset.
Gusset members connecting the SIM to RRH or FBM for the purposes of achieving the Lateral Clearance in Rule B.3.3.1 - Lateral Space may be primary or secondary material (B.3.2.3 - Secondary Members) and must be closed in with Body Panels

Rule B.3.2.12 - FBM – Front Bracing Members
Front Bracing Members must join the RHO, the SIM and the LFS at Points C, D and F. The upper Front Bracing Members (FBMUP) must join points C on the RHO to point D on the SIM. The lower Front Bracing Members (FBMLOW) must join point D to point F. The FBM must be continuous tubes. The angle between the FBMUP and the vertical must be less than or equal to 45 deg. If Front FAB, per Rule B.3.2.13.1 - Front Bracing, is used there is no angle requirement between FBM and vertical.

Rule B.3.2.13.2 - Rear Bracing
Rear systems of FAB must create a structural triangle, in the side view, on each side of the vehicle. Each triangle must be aft of the RRH, include the RRH vertical side as a member, and have one vertex at Point B and one vertex at either Point S or Point A. The members forming this structural triangle must be continuous members; but bends of less than 30 deg. are allowable. The third (aft) vertex of each rear bracing triangle, Point R (Figure B-19), must additionally be structurally connected to whichever Point, S or A, is not part of the structural triangle. This

additional connection is considered part of the FAB system, and is subject to B.3.2.1 - Member Requirements, but may be formed using multiple joined members, and this assembly, from endpoint to endpoint, may encompass a bend of greater than 30 deg.
Attachment of rear system FAB must be within 51 mm (2 in) of Point B, Point S and Point A, on each side of the vehicle. Distances are measured as a straight-line distance from centerline to centerline. The aft vertex of each rear bracing triangle defines Point R and must be joined by an LC of a minimum of 203.5 mm (8 in.) in length per B.3.2.5 - LC – Lateral Cross Member.

5. Capability Restrictions

Saxon Racing's fabrication shop has limited equipment for tube bending and notching. The shop has a pneumatic tube bender, a manual tube bender and a manual notching stand.

- Both tube benders have only one die for bending each thickness of tubing (1" and 1¼"). The die is for an inner radius of 4½", therefor the centerline radius of all tubes on the vehicle has been set to 5" for Secondary (1" OD) and 5 ⅛" for Primary (1¼" OD) tubing members.
- The process of bending with either of these methods is significantly less accurate and repeatable than a CNC tube bender. The geometric design must allow for large tolerances.
- Manual notching is limited at 1 notch direction per tube end. Complicated nodes will need to be ground to fit properly adding time and reducing precision.
- Attempting high angle bends with thin walled tubing may result in kinking. Kink prevention methods are limited to greasing, heating and sand-packing in the absence of a mandrel.

B. Design Goals

The tube chassis has a significant impact on the performance of an SAE Baja vehicle. It accounts for a major proportion of the vehicle's weight and constitutes a major manufacturing project in terms of time and resources. Furthermore, if the frame is improperly designed the tube chassis can drastically limit the design, fit and performance of the suspension and handling systems.

1. Weight Reduction

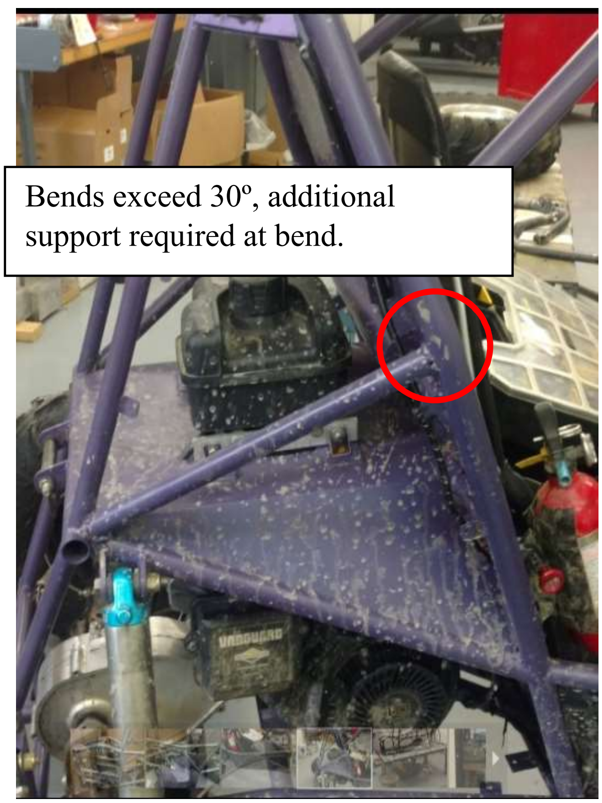
Our vehicle frame must be as lightweight as possible. The structural integrity and use of material are regulated to prevent any and all roll cage failure. If the frame meets the technical requirements of the rules, it will be a structurally sound frame. The tube chassis should avoid adding any members that are not required by the rules. That said, novel or unique critical areas should be analyzed with exact equations or Finite Element Analysis (FEA) and verified with testing.

There is room for improvement in 2020 when compared to previous years.

Previous vehicles have used AISI 4130 1 ¼” 0.065” wall thickness tubing (Primary tubing) in areas where only 1” 0.035” wall thickness tubing (Secondary tubing) is required. For example, Figure 3 shows a portion of the 2018-2029 vehicle. Every member pictured is a Primary member. Not every member pictured is required to be Primary. Some of these members are not required structural members, and do not have to conform to any external material requirements; use of primary material in these areas is unnecessary.

(Figure 3- 2019-2020 season vehicle foot box)

Design of previous frames has not accounted for bracing requirements. Successfully meeting critical member length and bend requirements will eliminate the need for additional bracing. The 2016-2017 vehicle in Figure 4 required additional aft bracing because the RRH had a bend exceeding 30º. The 2018-2019 vehicle has an additional brace on the Side Impact Member (See Appendix A).



(Figure 4-2016-2017 Aft Bracing)

(Figure 5-2018-2019, SIM Bracing)

Double Wishbone suspension using H-arms, requires a longer Aft Bracing section. The 2020 vehicle will utilize a multi-link suspension system with semi-trailing arms. This enables more flexible mounting points and allows for a reduced Aft Bracing Section.

2. Ease of Manufacture

The frame manufacturing process is historically time-consuming and labor intensive. Suspension tuning and design reevaluations require a complete frame. Our team has limited members capable of GTAW (TIG) welding. Delays in frame manufacture will overflow into delays in suspension and drivetrain mounting. It is imperative that frame manufacture takes as little time as possible. The design should carefully take into account the capability restrictions laid out in Design Constraints.

3. Additional Goals

There are additional desirable frame properties beyond weight and ease of manufacture that were considered in the design phase. Torsional rigidity, Roll Center, Center Of Mass will all have an impact on vehicle handling. Furthermore factor of safety in impacts could be of concern.

It is well documented that achieving desirable handling characteristics in the frame is extremely limited due to the geometric requirements of the suspension. With this knowledge, the team decided to address handling issues through flexible suspension design using a multi-link system. The role of the frame with respect to these parameters was to provide appropriate mounting points that allow for the development of this suspension system, without adding unnecessary members.

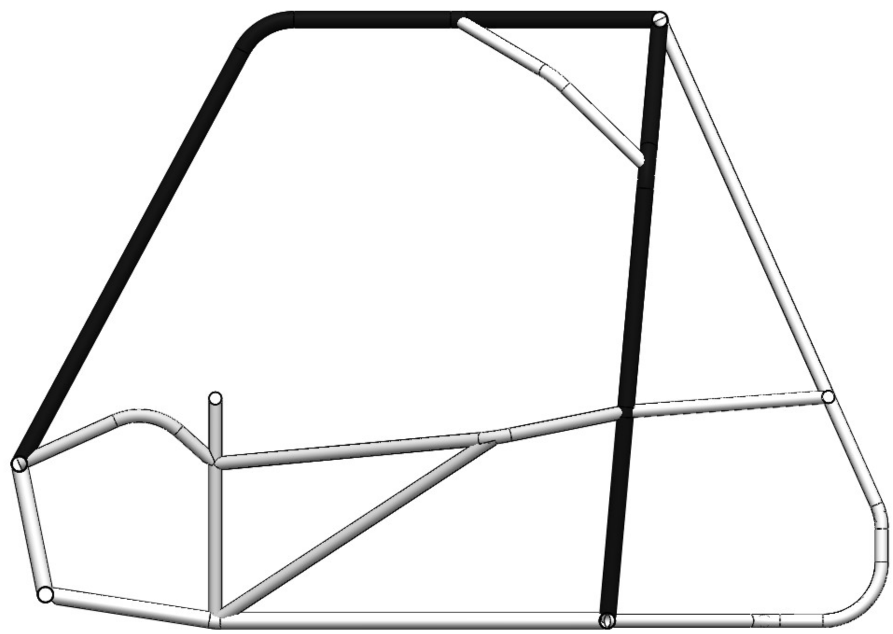
Unique or novel members that differ significantly from standard Baja vehicles or that have low clearances with other systems will need to have an adequate factor of safety of 1.3.

C. Design Decisions

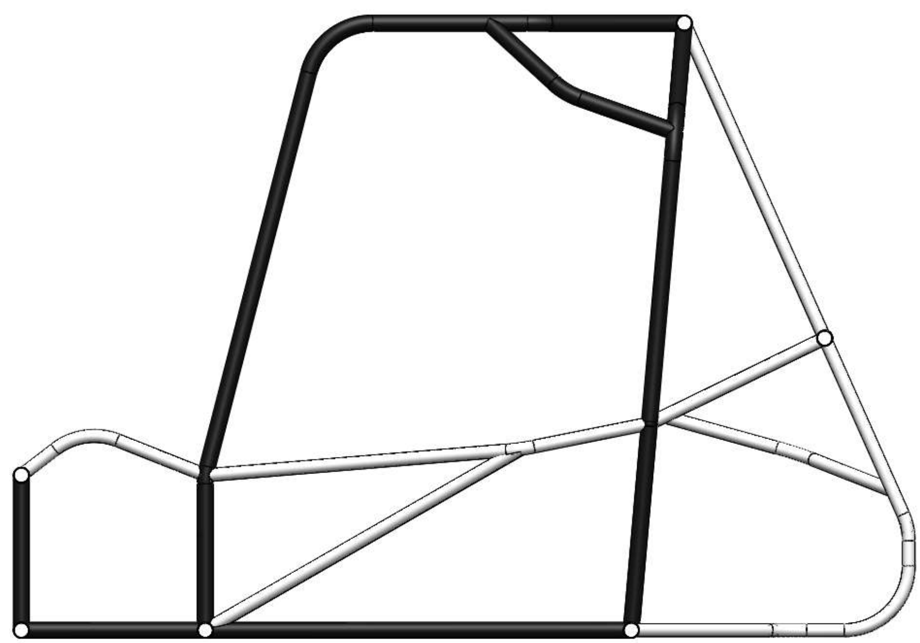
1. Nose vs. Non Nose

SAE allows for two configurations of FBMup geometry. One configuration is the connection of the FBMup to the front of the frame (a Non-Nose frame) or a connection behind the forward position and the construction of a separate extended foot box, colloquially referred to as the “Nose”. See Appendix for details. I considered the benefits and drawbacks of both.

Many models and design iterations were developed throughout the design process. Pictured below are two design iterations of a Non-Nose type and a Nose Type Tube Chassis



(Figure 6-Saxon Racing's 2020 Prototype Non-Nose Chassis)



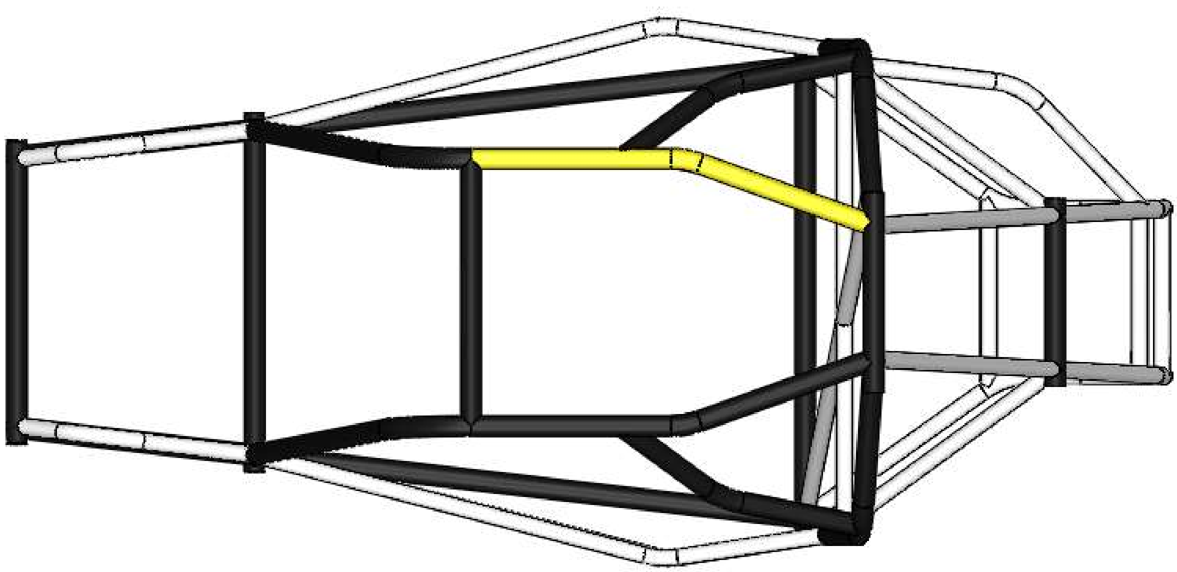
(Figure 7-Saxon Racing's 2020 Prototype Nose Chassis)

FACTOR	NOSE	NON-NOSE
Manufacturability	Decreased RHO/FBMup length and complexity.	
Design Flexibility	Ease of brake cylinder mounting.	Ease of placing front shock mounts. Location of point P is more flexible.
Driver Egress	Increased ease of egress over a fore braced vehicle.	Significantly increased if fore bracing not used. Significantly decreased if fore bracing required.
Weight	Reduced weight. Increased reduction over fore-braced designs.	
Impact on Other Members	Requires more use of primary material in foot box.	SIM will need altered geometry and additional bracing due to added length.
Chassis Characteristics	Aesthetically pleasing.	Fore bracing increases structural integrity and torsional stiffness. Center of mass is moved slightly forward.

(Table 1- Nose vs. Non-Nose)

A defining aspect of the design choice is whether fore bracing members are to be used. The rules governing fore bracing members are described in the appendix. If added, they connect the bend in the FBMup to a point on the SIM.The 4WD system requires approximately 6” of additional length in the cockpit to make room for the front differential and forward driveshaft. The additional length of the cockpit increases the FBMup’s angle from vertical to exceed 45º. Rule B.3.2.12 - FBM – Front Bracing Members mandates the use of fore bracing in this scenario. The costs of fore bracing were decided to outweigh the benefits. The geometric requirements ruled out a Non-Nose vehicle without fore-bracing (Figure 6) and therefore and a Nose style design was selected (Figure 7).

2. Bent RHO



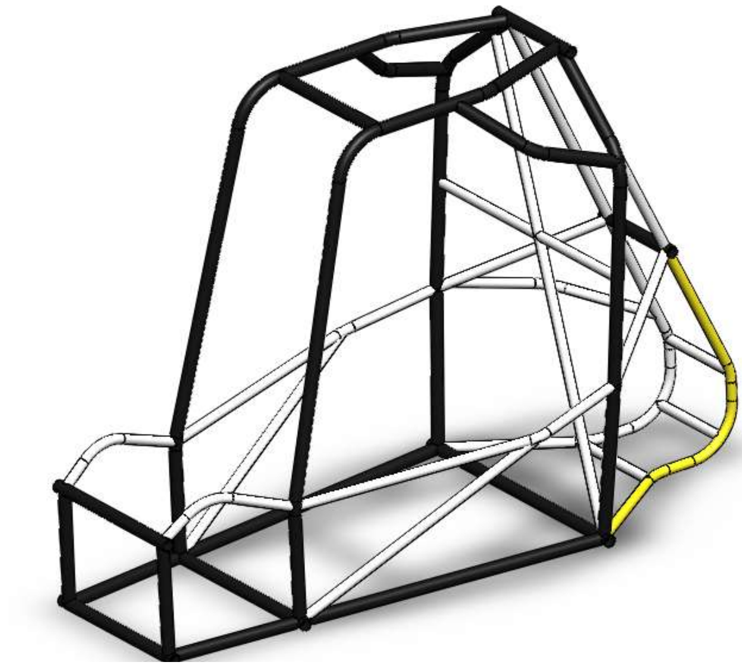
(Figure 8-Bent RHO)

Reducing the RHO to 8” at the connection with the RRH allows for a lighter RRH. 8” is the minimum distance for Lateral Crosses. Without this bend the RRH would be 12”-14” width at this connection with no benefit.

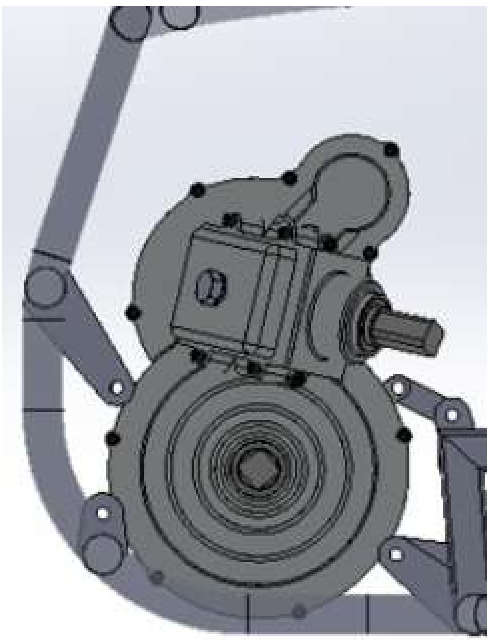
A narrow RRH requires larger head clearance gussets than a wider RRH. The savings in gusset size are counteracted by the larger RRH. The 8” RRH accomplished with bent RHO is the best option for weight reduction.

3. Curved FABlow

In accordance with DFMA principles, this choice reduces BOM costs and number of members. With the FABlow as a single curved piece, manufacturing time for notching and welding is reduced. Opportunities for notching errors to distort the frame from nominal are also reduced.



(Figure 9-Curved FABlow)



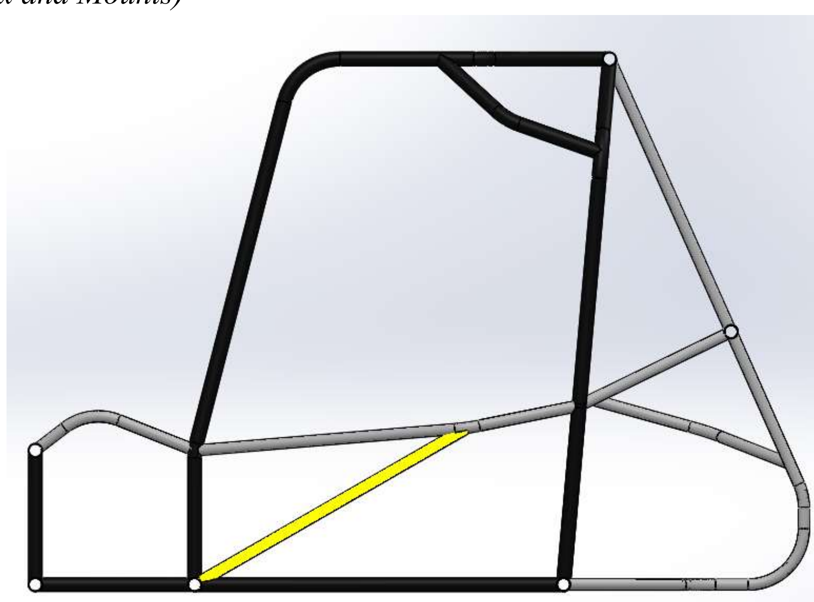
(Figure 10-FABlow with Gearbox and Mounts)

The curve of our FABlow mirrors the outer radius curve of our gearbox; there is no wasted space in our Aft Brace Compartment (FAB)

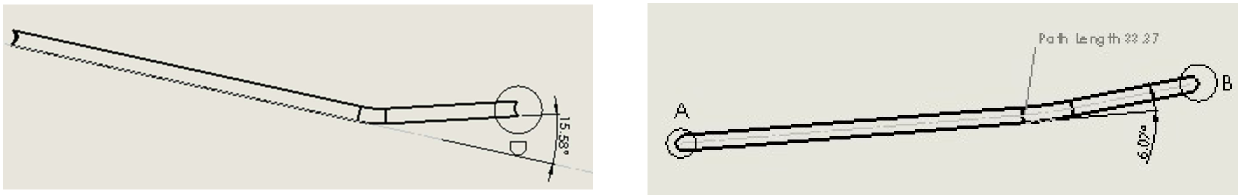
4. Single Braced SIM

The path length of the SIM surpasses 33 inches and the bend is under 30°.The member only requires one additional bracing member under Rule B.3.2.4.

Were the path length of the SIM under 33 inches, the member would require no additional bracing whatsoever. However, this was impossible to achieve while adhering to other rules.



(Figure 11-Single brace on SIM)

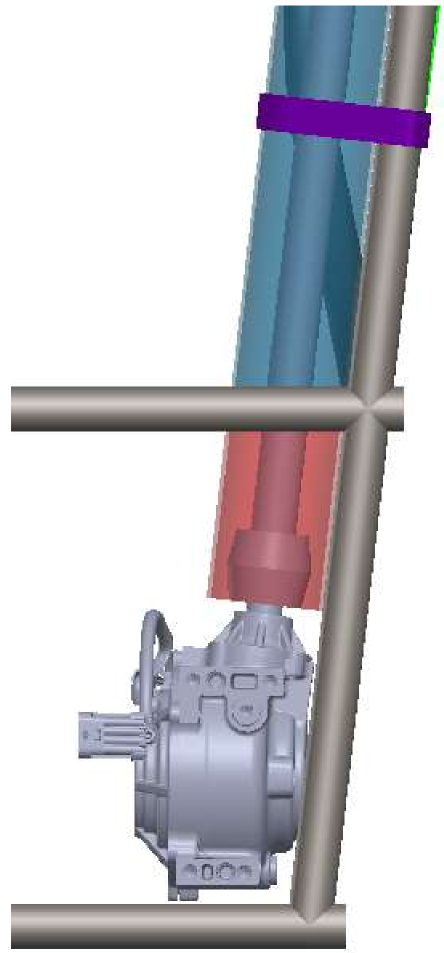


(Figure 12-SIM angle and Path Length)

5. Accommodation of Drivetrain Components

The front differential has a specified input angle of 83.5º from the output. The shape of the driver cockpit, particularly the angle of the LFS were changed to meet this angle. With the chassis and input angle in alignment, the driveshaft runs evenly down the side of the vehicle (Figure 13).

The foot box compartment was made wider and longer to house the front differential and axle-shafts. Variable engine mounts were designed so that belt tension on the CVT could be adjusted.



(Figure 13-SIMlow and Driveshaft)

III. FEA Analysis

I was responsible for FEA analysis of critical components not only for the Chassis but for front suspension members as well. FEA was conducted using ANSYS software. The team's desired factor of safety was 1.3. Analysis also required the identification of critical failure points in the vehicle.

A. Load Analysis

To appropriately use FEA software, it was necessary to have accurate numbers for the forces applied to the members being analyzed. Additionally, a method needed to be developed for representing the dynamic loading scenario as a static scenario for analysis with our team's limited ANSYS license and computing power.

1. Impulse Equation

In order to develop our equations, my team consulted with Alfred University's Dr. Ben Carlson. Our two options for estimating force in members were deformation equations and impulse theory. Relating deformation to force would require knowledge of both the amount of deformation on impact and the stiffness of the materials involved. It would also require knowledge of the behavior of the sprung versus the unsprung mass of the vehicle for suspension impacts. Use of impulse equations however, requires only the effective mass of the vehicle, the velocity change and the duration of the impact. Our team and Dr. Carlson agreed that impulse equations were more likely to generate accurate estimates.

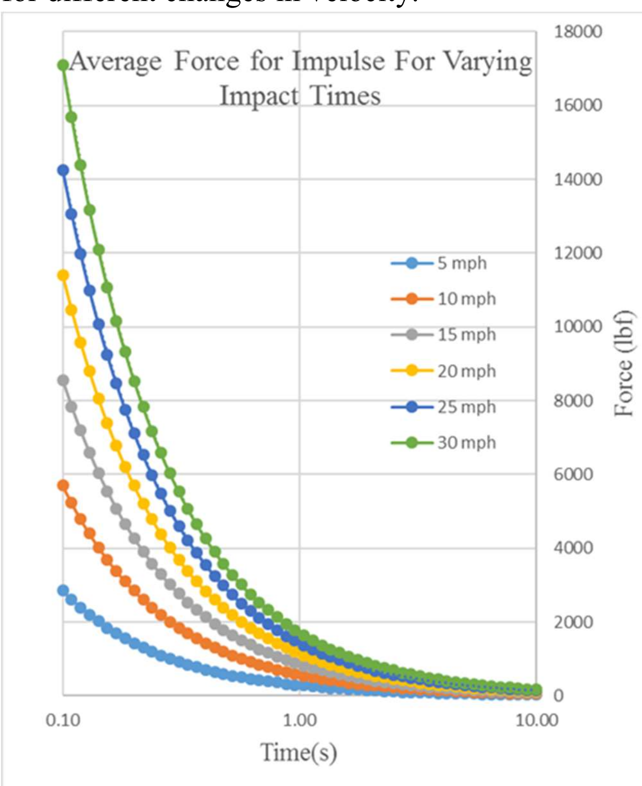
$$F_{avg} = \frac{2.5m(V_0 - V_1)}{(t_0 - t_1)}$$

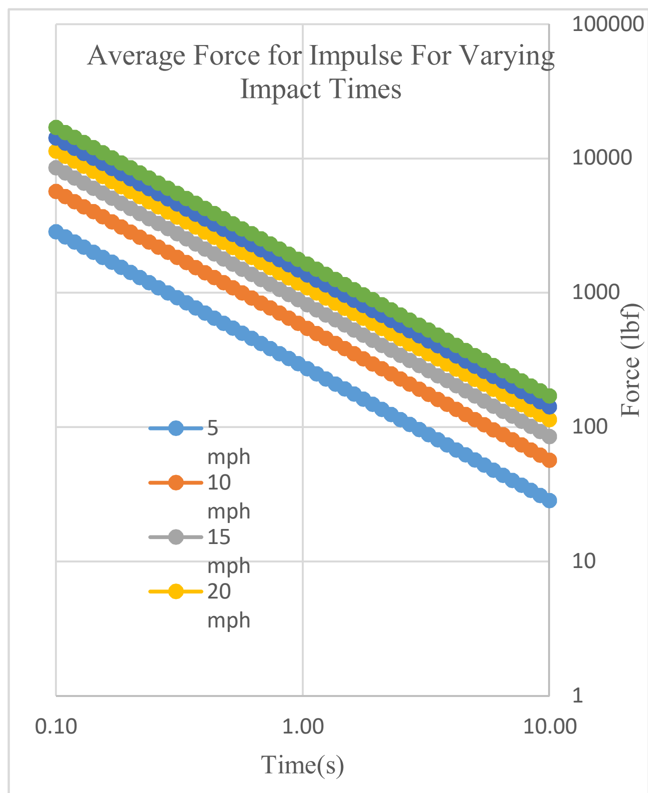
(Eq.3-Impulse Equation)

Impulse theory is limited in that it calculates average force, not peak force. It also requires as coefficient to account for the inertial mass of the vehicle to translate the dynamic problem into a static problem. A coefficient of 2.5 applied to the static mass is recommended by existing literature.

2. Parametric Plots

The impulse equation requires many estimations and correction factors. Furthermore, the vehicle will experience a range of loads, and a range of impact times. To best use this equation, I created parametric plots showing of the change in average force experience as impact time varies for different changes in velocity.





(Plot 1-Parametric Impact, Log x)

(Plot 2-Parametric Impact, Log-Log)

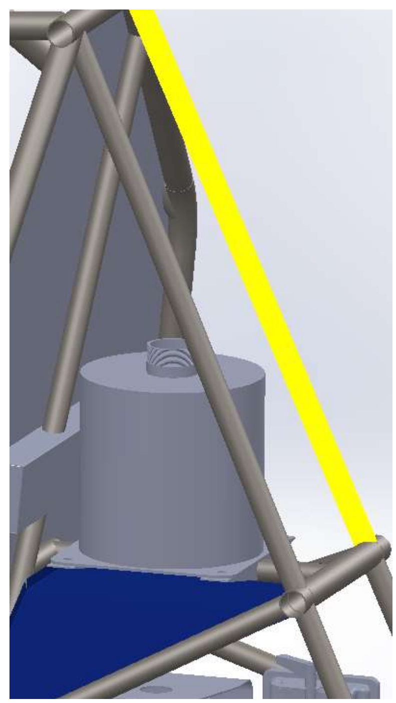
By consulting these plots, and the data behind them. My team and I were able to determine the range of average forces that we could expect for different impacts. I determined impact times by consulting video footage of Alfred's previous vehicles and confirmed with footage from other schools.

B. Chassis FEA

FEA can be a time consuming process, it can be difficult to preform FEA accurately and errors in final results are sometimes difficult to catch. A 3D element FEA needs to be justified before preformed. This is particularly relevant to the vehicle chassis which is composed of members with known structural properties. Exact solution methods and 1D FEA should be considered first.

Our primary material, AISI 4130 1¼", 0.065" wall thickness tubing is a common material. There is considerable existing analysis and empirical testing data of this material. Much of this data is specific to the SAE Baja completion due to the frequency of AISI 4130 use in the competition. For this reason is was not valuable to preform FEA on primary members that are tightly regulated or similar to other vehicles. The foot box (Nose) in a front collision scenario, the Front Roll Hoops in a roll-over, LFS in a side impact, are all very similar from one vehicle to another. We followed an industry requirement that results in a very high factor of safety, and does not require our team to independently verify factor of safety through FEA.

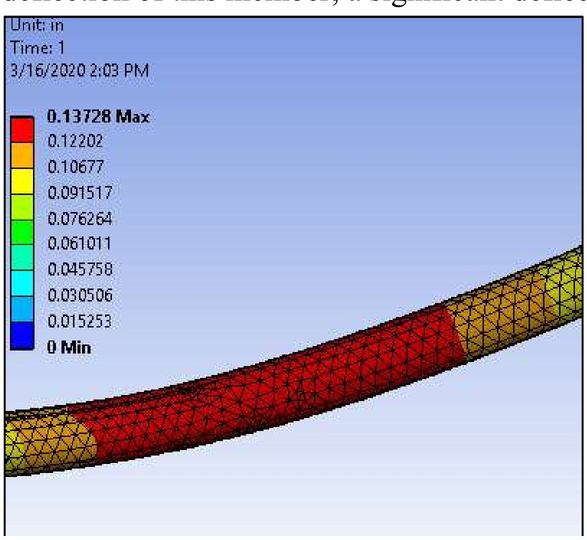
1. FABup Rollover



(Figure 14 – FABup with Fuel Tank)

The FABup is an area of concern due to its extreme proximity to the fuel tank. It is a distinct possibility that the FABup will be the first point of impact in a backward rollover on a hill climb or suspension event.

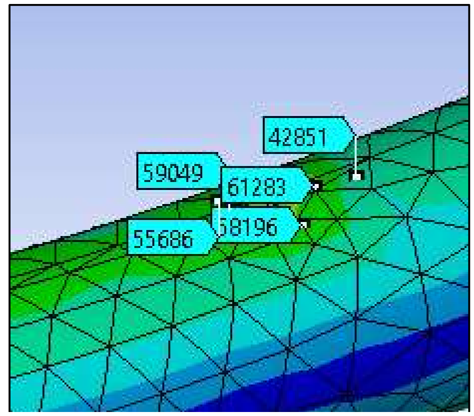
The loading of this scenario is not easily represented in a Mechanics problem. The primary concern of my team was the deflection of this member; a significant deflection would result in the fuel tank experiencing an impact. 1D bar elements could simulate the centerline deflection but would not be useful in simulating the deflection on the outer surface of the tube. FEA with 3D element was deemed necessary for this impact scenario.



(Figure 15 – FABup Displacement)

A 1500lbf load was determined to be the worst case scenario by consulting the parametric plots. This corresponds to an impact resulting from a 4 foot drop, with a 0.5 second impact time. This number is also consistent with similar studies conducted by other teams. Applying this force at different expected angles resulted in a worse case deformation of just over 1/8". Our tolerance for deformation on this member is 3/16" of an inch. Specifically, our factor of safety relating to deformation is 1.36.

Mesh size was refined to ensure convergence. Peak stress rose by 5% from a 1/4 " mesh to a 1/64" mesh, however deformation varied by less than 1%. The team is not particularly concerned with the plastic deformation of this member- deformation is expected in worst case scenarios. SAE rules disqualify a frame after three rollovers. We do not expect repeated rollover impacts. However, in the worst case scenario the peak stress experienced is less than the yield stress of the material.

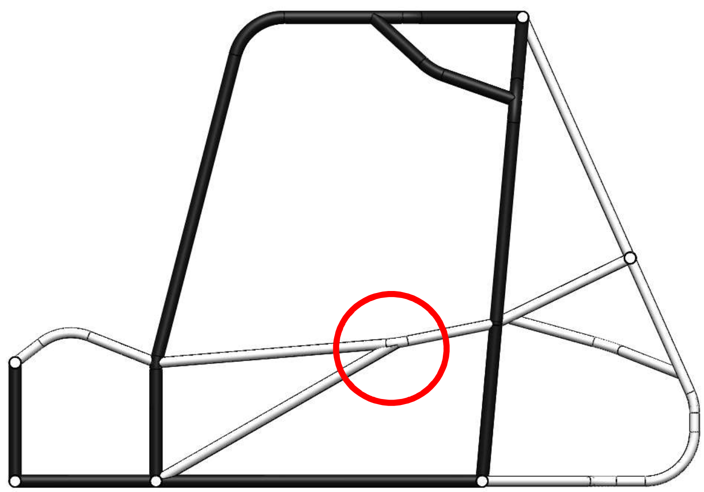


(Figure 16 – FABup Stress at Critical Section)

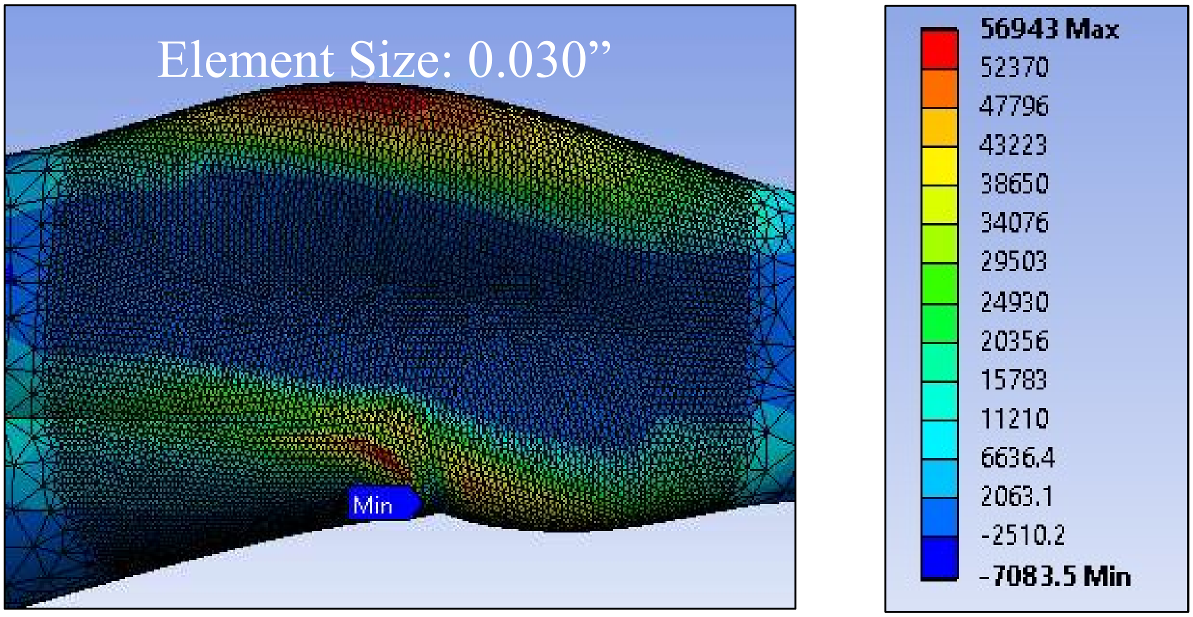
2. SIM Side Impact

Unlike the LFS, which is a very standard primary member, the SIM is a secondary member with limited bracing. Bracing designs and the shape of the SIM are highly variable; my team could not rely on historical data. Additionally the 2019-2020 frame incorporates only one SIM brace, an infrequently used design and distinct from previous Alfred University vehicles.

Impacts between vehicles are extremely rare in SAE Baja. A side impact refers to the impact between the side of the frame and the ground in a horizontal rollover. Side impacts typically occur at the external bend of the SIM, directly on the node of the brace. My team justified the use of 3D FEA, from the complexity that the weld and notched tube added to the system.



(Figure 17 – Side Impact Location)



(Figure 18 – Side Impact Von Mises Stress)

Side impacts involve considerably less velocity change and they are generally slightly longer impacts. Consulting videos and the parametric plots yielded an expected impact of slightly less than 500lbf, 500lbf was used as a conservative estimate.

Mesh size was refined from 1/4 “to 1/64” to verify convergence of results. Peak Von Mises stress increased less than 4% over refinement. Factor of safety concerning yield for this scenario is 1.1. The SIM has significant clearance, even plastic deformations exceeding 1” are not an issue. Failure is the most relevant concern, the peak stress (56,943psi) is well below ultimate failure of AISI 4130 (97,200psi), FOS vs. Ultimate Tensile Stress is 1.7.

C. Front Uprights

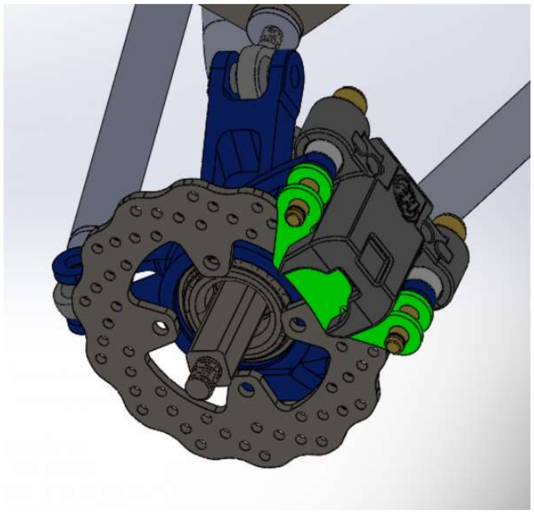
Our front uprights are a new design with complex geometry; they are best analyzed in FEA. I took over the analysis of front suspension components because of my experience with ANSYS. The uprights will experience much more frequent and regular loading than the frame, so fatigue life must also be estimated. An S-N curve describes the relation between cyclic stress amplitude and number of cycles to failure. The loading of the front upright is not cyclic, but conservative fatigue life estimates can be obtained from S-N curves for this scenario. Uprights are manufactured out of 6061 T6 Al, which has a yield strength of 40ksi.

1. Braking Scenario

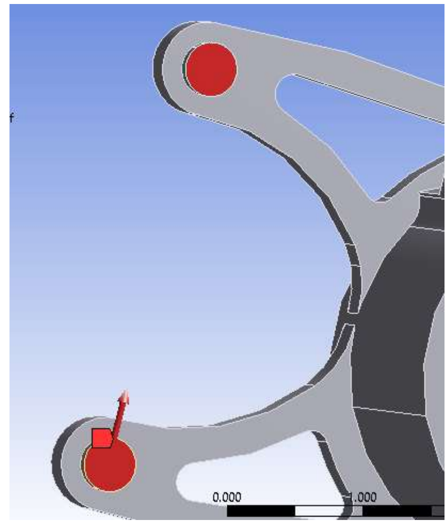
The wheel lock force exerted by the caliper on the rotor will be experienced by the upright, through the caliper bolts. Calculations preformed by the front suspension team identified the wheel lock force as 300lbf. The desired FOS for this component is 1.3 and the desired fatigue life for this scenario is to exceed 3,000 cycles.

Figure 19 shows how the caliper and rotor are attached to the upright.

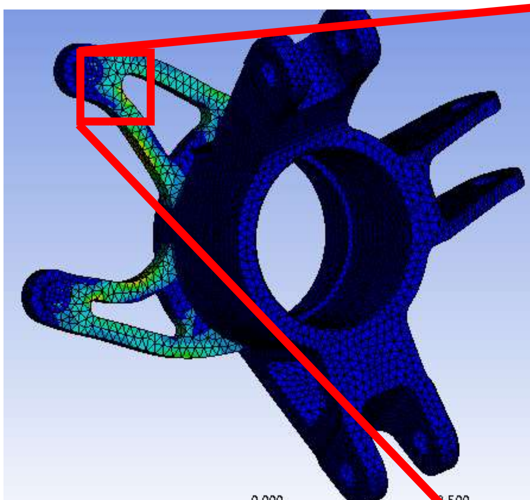
The angle of the force in Figure 20, is determined from this model.

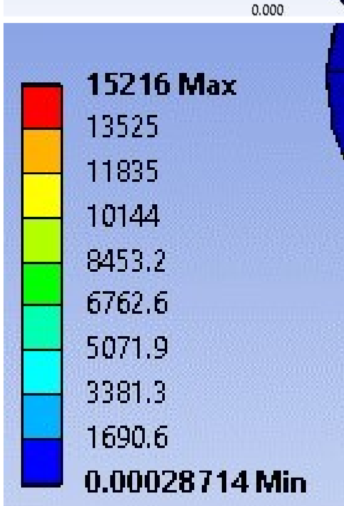


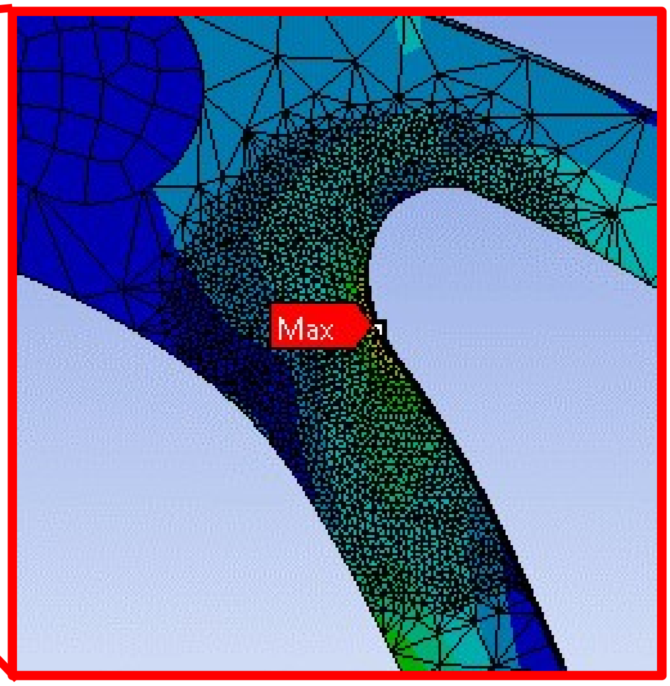
(Figure 19 Brake Caliper Assembly)



(Figure 20- Brake Force Applied)







(Figure 22 Upright Braking Stress, Critical Area)

Figure 21 shows, and Figure 22 clarifies the region of highest stress in this scenario. The critical area is along the inside of a weight reduction cut-out. If the stress proves too high in this area, it could be alleviated by a small design change.

(Figure 21 Upright Braking Stress, General)

A coarse mesh was used to identify critical area of the upright. Once the critical areas were identified, mesh was refined. The max stress in the upright was 15.5ksi in the upright, corresponding to a factor of safety of 2.52. A conservative fatigue life was estimated from a 6061 T6 Al S-N curve to be 5000 cycles.

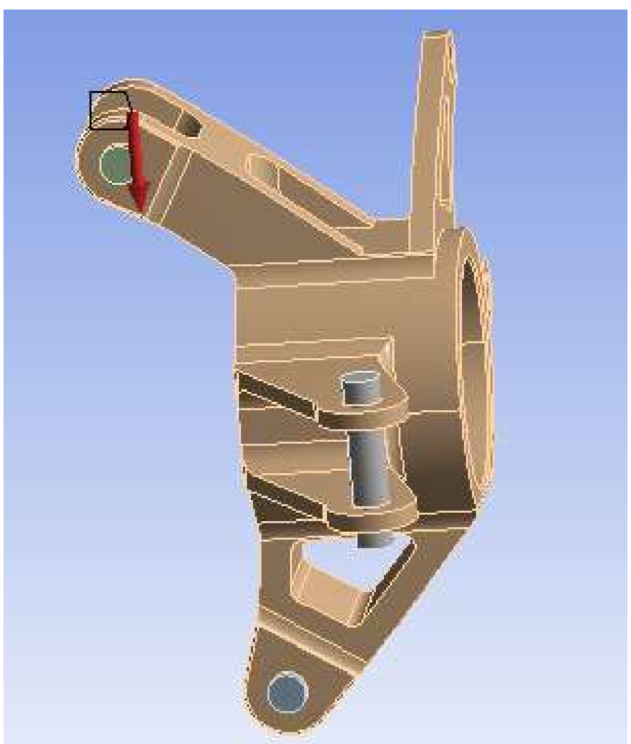
BRAKING SCENARIO			
Loading Conditions		300lbf	Total
		25°	From x-axis
Mesh Density (in)	Max Stress (Von Mises)		FOS
	Stress (ksi)	% change	
0.125	12.57	-	3.10
0.0625	14.18	12.81	2.75
0.03125	14.95	5.43	2.61
0.015	15.5	3.68	2.52
Deformation (.015" mesh)		.0035"	
Fatigue (Cycles)		5000	

(Table 2 – Upright Braking)

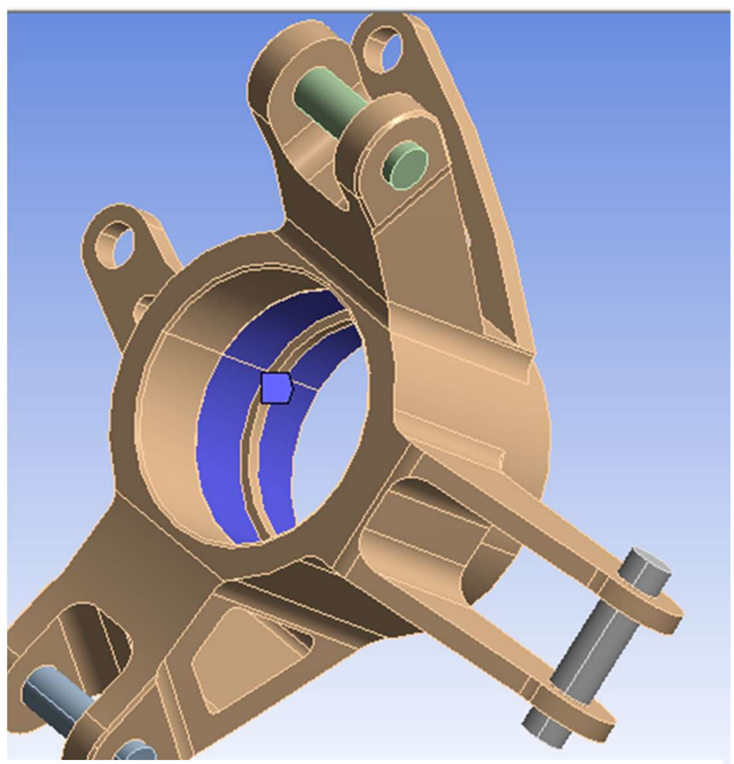
2. Vertical Drop

Suspension tests in competition can involve jumps, see-saws and whoops, all designed to apply vertical loads to vehicle suspension. The team required that our uprights withstand 500 maximum impacts.

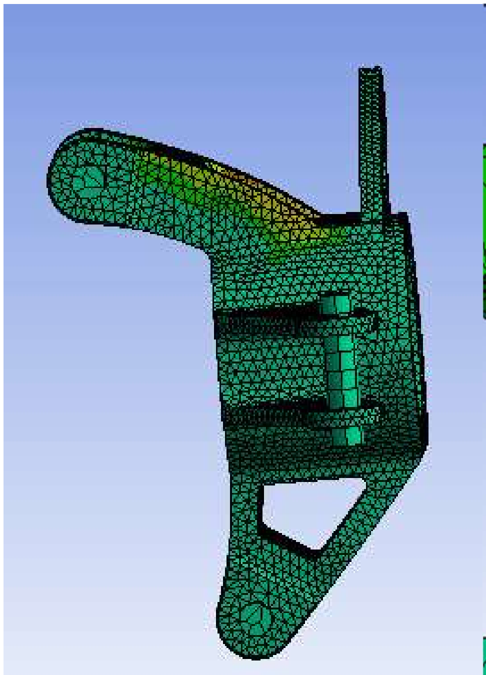
The scenario considered is a 4ft drop with the vehicle landing on a single front tire only. In the vertical drop scenario, the forces will be applied to the shocks. Our shocks mount on the upper A-Arm which is attached to the upright with a heim bolt. The lower A-Arm is unspung; it will not experience forces of the magnitude experienced by the upper. The model is fixed on its bearing surfaces.

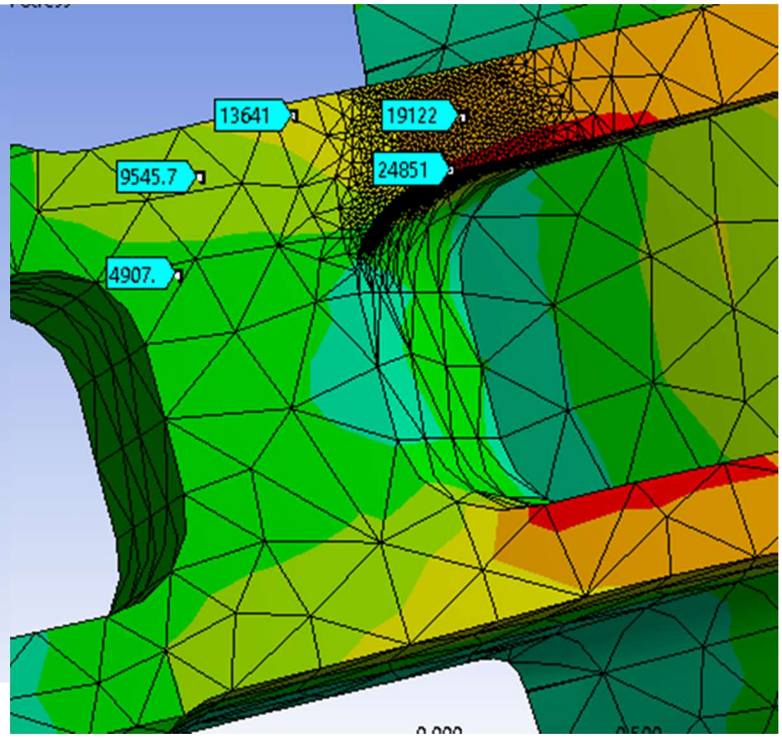


(Figure 22- Upright Vertical Loading)



(Figure 23- Upright Vertical Fixed Points)





(Figure 24- Upright Vertical Loading Mesh) *(Figure 25- Upright Vertical Critical Area)*

The primary concern in this scenario is focused on ensuring that the heim bolt will fail before the upright. The front uprights are complicated and expensive components that the team does not have back-ups for. A damaged upright will take a considerable amount of time to repair or replace in competition. 3/8" heim bolts are inexpensive (by comparison) and they are easily changed out if damaged. The heim bolts are acting as shear pins in our front suspension design. Calculations by other members of my team have determined that the heims will fail at 500lbf in this scenario. Out of an abundance of precaution, I analyzed the uprights in ANSYS with the forces that they would experience if the heim did not fail- 1500lbf.

DROP SCENARIO			
Drop		4ft	
Velocity (Fps)		16.050	
Velocity (m/s)		4.892	
Velocity (mph)		10.943	
Estimated Impact Time		0.5s	
Loading Conditions		1500lbf	Total
		At the upper Mount	
Mesh Density (in)	Max Principal Stress (ksi)		FOS
	Stress	% change	
0.125	23.12	-	1.69
0.0625	23.03	-0.39	1.69
0.03125	24.35	5.73	1.60
0.015	24.85	2.05	1.57
Fatigue (Cycles)		1000	

(Table 3 - Upright Vertical Loading)

Using the conservative loading model, the upright has a factor of safety of 1.57 and converges on the solution. Evaluating this loading with an S-N curve yields a cycle life of 1000, above the required 500 cycle durability.

3. Forward Impact

The rough terrain and presence of obstacles in the endurance race, make high speed forward impacts inevitable in competition. The most destructive impact for suspension components is a high speed collision between a single wheel and a large rock or obstacle. If the wheel is trapped and cannot clear the obstacle, the impact can be particularly devastating. Experience and analysis of competition footage indicate that these impacts bring a slow to medium speed vehicle to a full a stop, and significantly decelerate vehicles moving at high speed. If a vehicle moving over 20mph is brought to a full stop by a wheel-trapping obstacle, the suspension is almost invariably damaged.

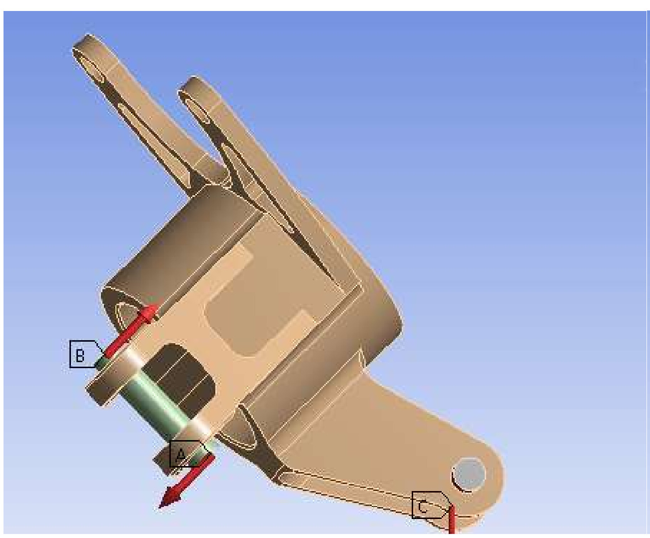
In this collision scenario, the wheel will be pulled outward and the upright will rotate until it is restrained by the heim bolts connected to the A-Arms. The tie rod will also exert a load on the upright in this scenario.

The goal of the suspension design team was not to create indestructible components. The criteria for durability is to withstand a 15mph change in velocity in less than half a second. Consulting the parametric plots, this corresponds to a total load of approximately 3000lbf. I considered failures along the upper heim connection, the lower heim connection and the tie rod connection.

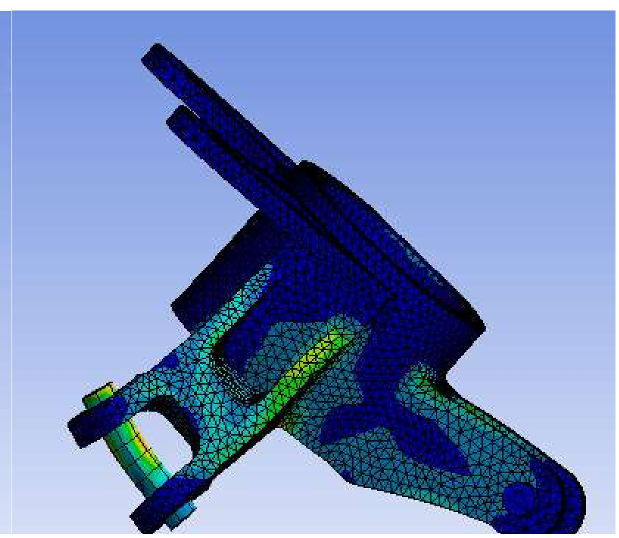
FORWARD IMPACT		
Initial Velocity (mph)	25	15
Final Velocity (mph)	10	0
Estimated Impact Time	0.25s	
Impact Conditions	Force is split between Hiems	
Impact	3000lb	
Modeled	1500lb	Hiem
Tie Rod Conditions		
	500lb	Hiem

(Table 4- Upright Forward Forces)

a.) Upper Heim Support

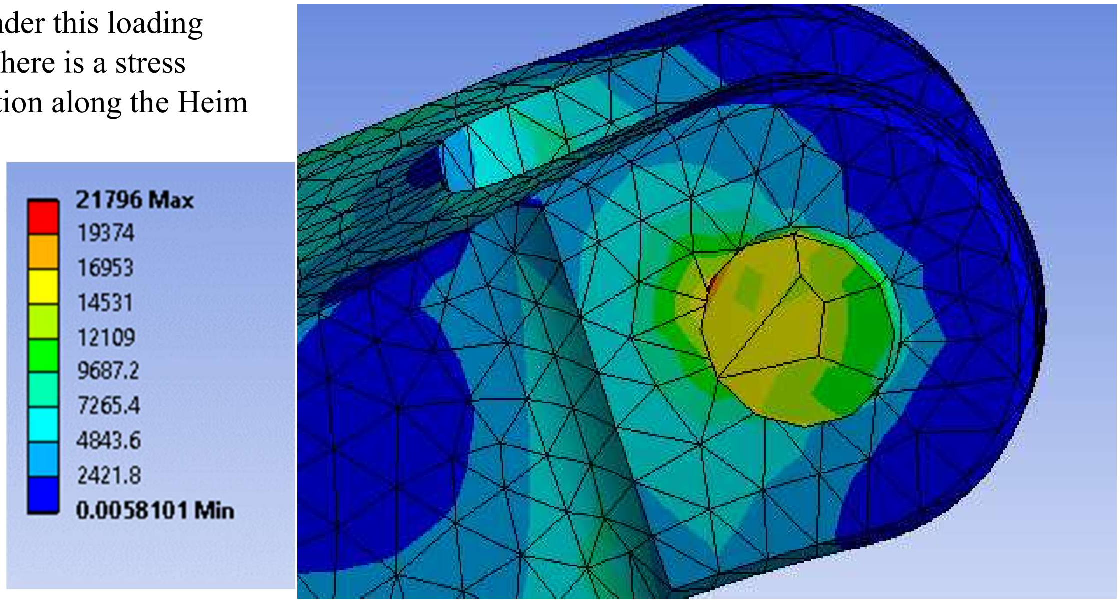


(Figure 26- Upright Forward Loading Forces)



(Figure 27-Upright Forward Loading Mesh)

Under this loading scenario, there is a stress concentration along the Heim Bolt Hole



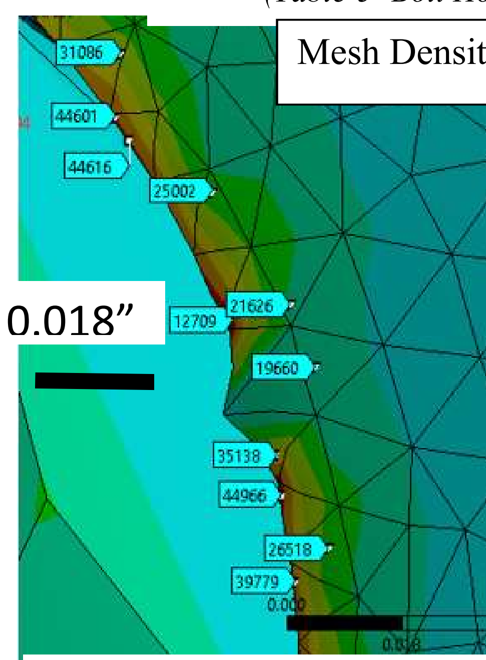
(Figure 28-Upright Heim Support Coarse)

As I refined the mesh it became clear that the solution was not converging at this point.

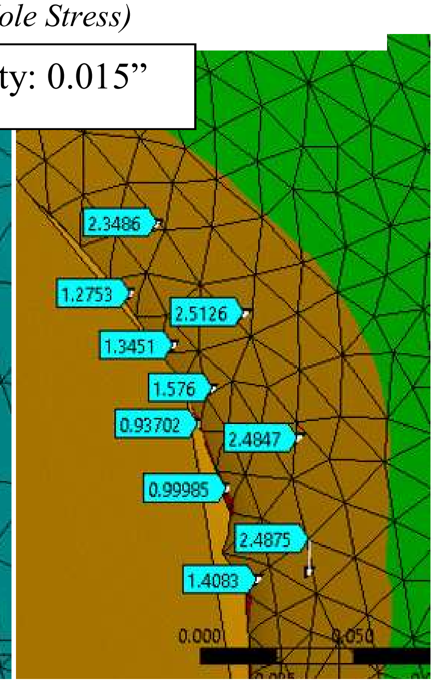
UPPER HEIM			
Mesh Density (in)	Max Stress (Von Mises)		FOS
	Stress	% change	
0.125	25.23		1.55
0.0625	29.8	15.34%	1.31
0.03125	36.93	19.31%	1.06
0.015	45.00	17.93%	0.87

(Table-5- Bolt Hole Stress)

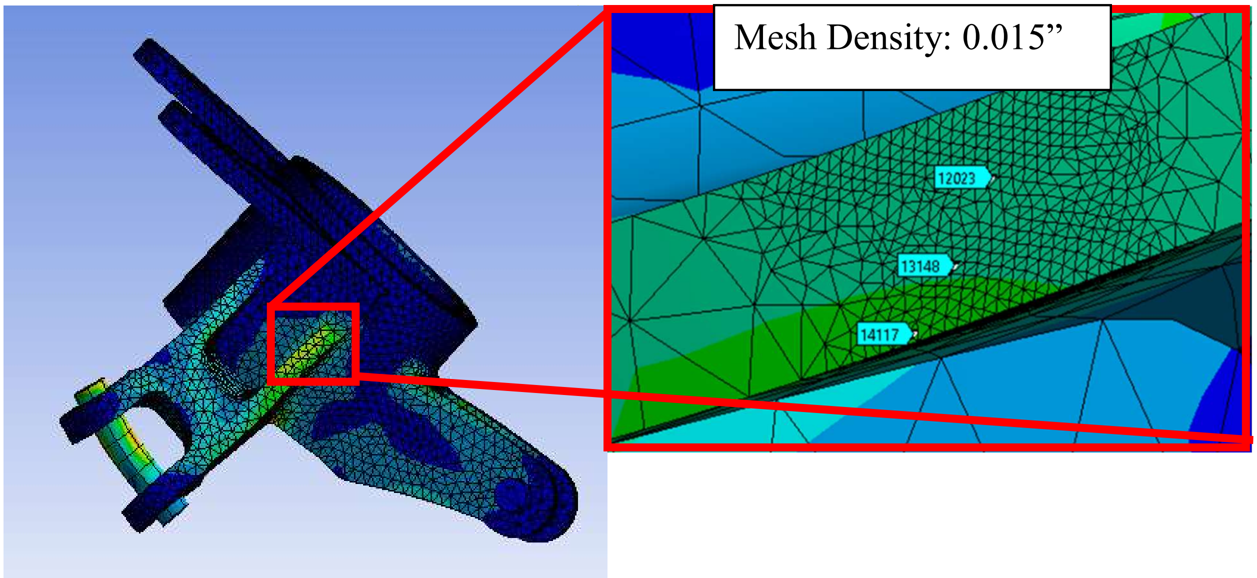
However, the stress in this area falls off drastically over a very short distance. It is my interpretation that the sharp edge on this feature is resulting in a stress riser that ANSYS cannot accurately model. More importantly, this is the side of the bolt hole that is under compression. If this region deforms, it will be deformation that blunts the edge of the bolt hole, and reduces the local stress. Deformation along this edge, especially less than 0.005” is not a concern. It is worth noting that under this loading condition, calculations show that the heim bolt would fail well before the upright.



(Figure 29- Bolt Hole Stress)



(Figure 30-Bolt Hole FOS)



(Figure 31- Upper Heim Support)

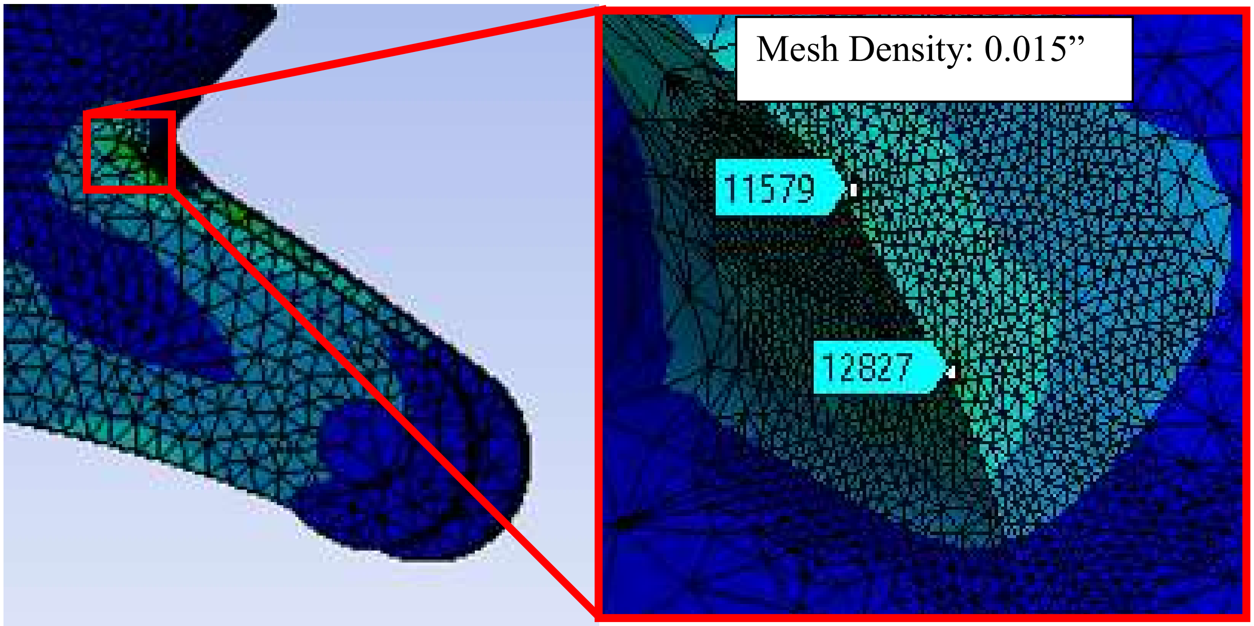
Another possible critical area is the upper arm of the upright. Analysis on this region yielded a FOS of 2.72 and a series of solutions that show convergence.

UPPER SUPPORT			
Mesh Density (in)	Max Stress (Von Mises)		FOS
	Stress (Ksi)	% change	
0.125	13.75		2.84
0.0625	14.17	2.96%	2.75
0.03125	14.25	0.56%	2.74
0.015	14.36	0.77%	2.72
Fatigue (Cycles)		4000	

(Table 6- Upper Support Stress)

b.) Tie Rod Connection

The suspension team calculated that the maximum force the tie rod will exert on the upright is 500lbf. The upright was analyzed under these conditions, and the mesh refined at the critical region shown



(Figure 32-Upright Tie Rod Stress)

The mesh refinement shows that the results are converging. The high factor of safety in this region led to a topology study on this section of the upright and ultimately material was removed,

TIE ROD CONNECTION			
Mesh Density (in)	Max Stress (Von Mises)		FOS
	Stress	% change	
0.125	11.25		3.47
0.0625	12.16	7.48%	3.21
0.03125	12.34	1.46%	3.16
0.015	12.83	3.80%	3.04
Fatigue (Cycles)		5000	

(Table 7- Tie Rod Support Stress)

4. Summary

A summary of the FEA analysis of the front suspension upright is presented below.

Scenario	Max Stress (ksi)	FOS	Fatigue Life	Result
Braking	15.5	2.53	4000	Sufficient
Drop	24.85	1.57	1000	Sufficient, 3 competitions may be issue
Forward-Heim	45.00	0.87	-	Bolt will fail first, Some deformation along edge of hole
Forward-Support	14.36	2.72	4000	Sufficient
Forward-Tie Rod	12.83	3.04	5000	Sufficient

(Table 8- Upright FEA Summary)

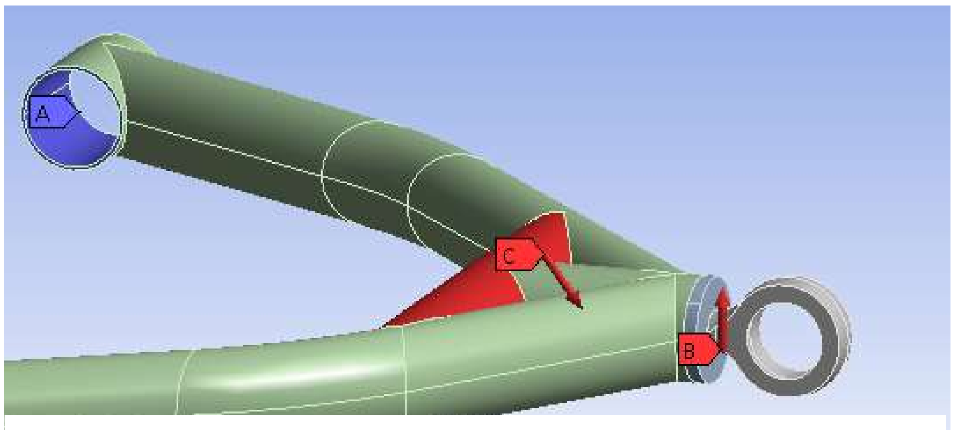
D. A-Arm FEA

The upright is directly linked to the A-Arms; the A-Arms should be evaluated under the same impact conditions. The Upper A-Arm is made from the team's Secondary material: AISI 4130 Chromoly Steel 1" Diameter 0.035" wall thickness tubing. The yield strength of this material is 63ksi. The lower A-Arm is manufactured from 6061 T6 Aluminum with a yield stress of 40ksi.

1. Vertical Impact

In a vertical impact, the Upper A-Arm will see a force from the shock and a force from the upright connection.

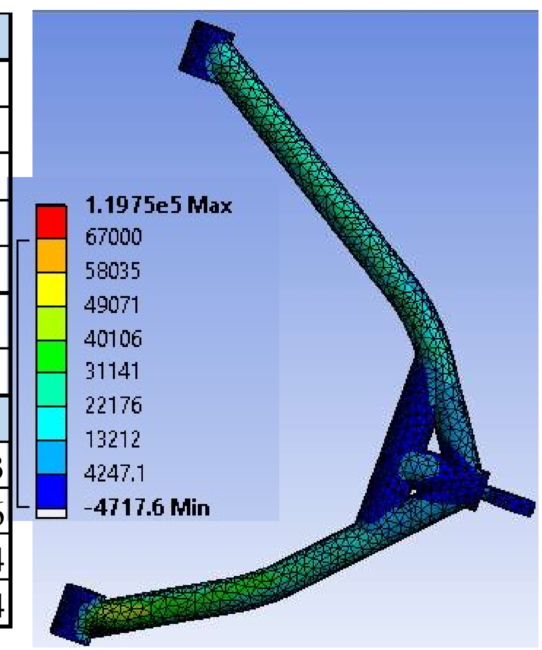
Analysis of the critical area shows that peak Von Mises stress, is well below yield strength, FOS is 1.54.



(Figure 33-A-Arm Vertical Impact)

VERTICAL IMPACT		
Drop	3ft	
Velocity (Fps)	13.900	
Velocity (m/s)	4.237	
Velocity (mph)	9.477	
Estimated Impact Time	0.5s	
Loading Conditions	500lb	Shock Mount
	500lb	Upright
Mesh Density (in)	Max Stress	FOS
0.125	44.00	1.43
0.0625	43.17	1.46
0.03125	40.91	1.54
0.015	40.85	1.54

(Table 9 – A-Arm Vertical Impact)

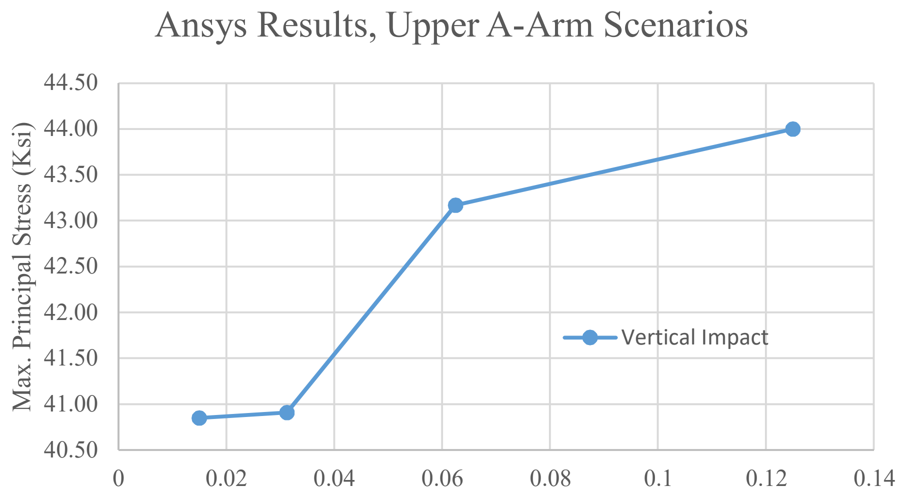


(Figure 34- A-Arm Vertical Mesh)

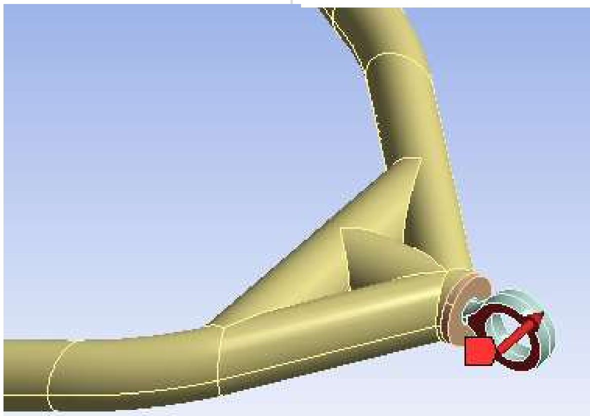
The left plot demonstrates the convergence of results as the mesh is refined.

2. Forward Impact.

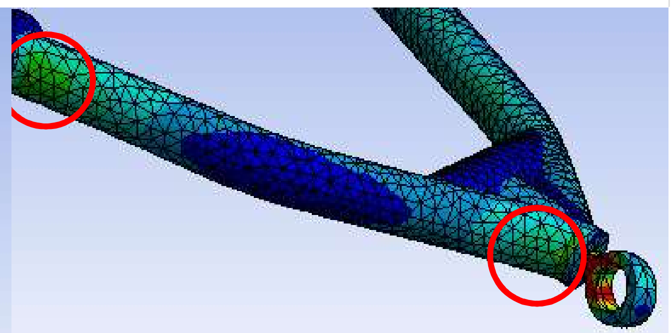
I preformed FEA analysis on the A-Arm in the same forward impact scenario as the upright.



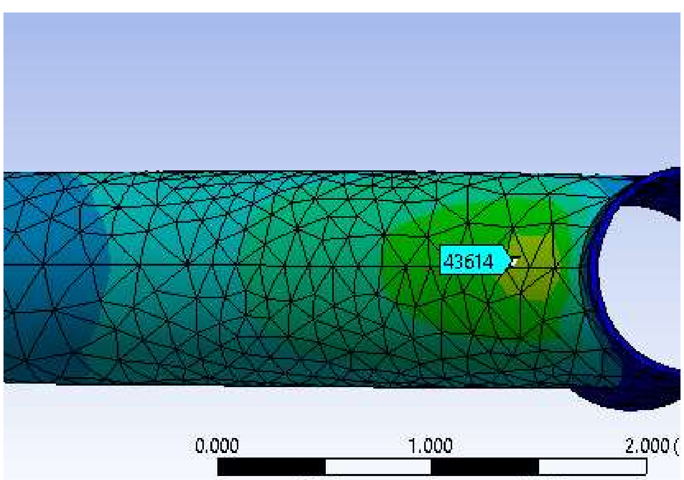
(Plot 3 – A-Arm Vertical Stress Convergence)



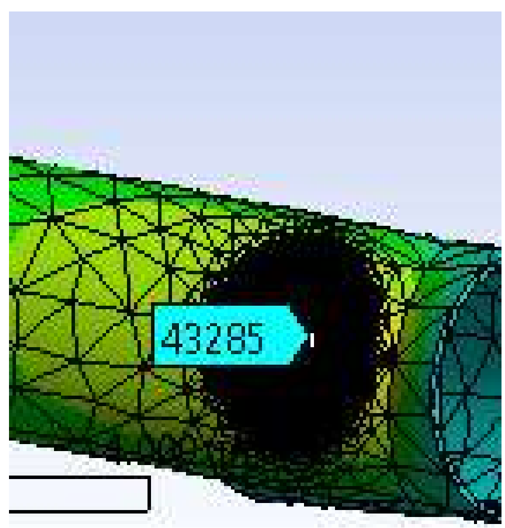
(Figure 35 – A-Arm Forward Loading)



(Figure 36 – A-Arm Forward General Mesh)

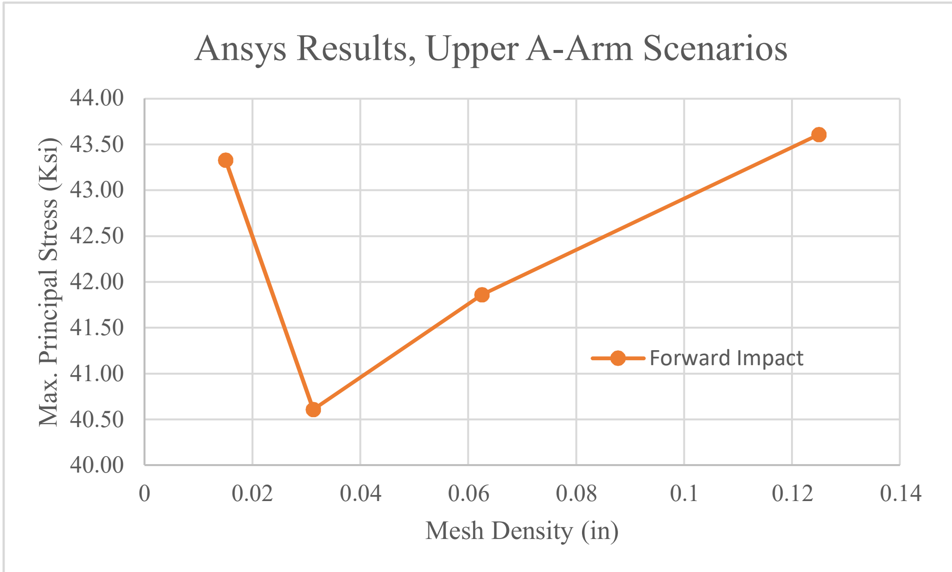


(Figure 37 – A-Arm Forward Area 1)



(Figure 38 – A-Arm Forward Area 2)

The A-Arm also withstands this loading scenario with a FOS of 1.45. Convergence is not as clear in this case, however the general trend is lower stress values as the mesh is refined.



(Plot 4 – A-Arm Forward Stress Convergence)

FORWARD IMPACT		
Initial Velocity (mph)	25	
Final Velocity (mph)	10	
Estimated Impact Time	0.25s	
Loading Conditions	Force is split between arms	
Total Impact	3000lb	
Modeled	1500lb	Hiem
Mesh Density (in)	Max Stress	FOS
0.125	43.61	1.44
0.0625	41.86	1.51
0.03125	40.61	1.55
0.015	43.33	1.45

(Table 10 – A-Arm Forward Impact)

IV. Manufacture

Due to the limited financial resources of Saxon Racing, our chassis was not manufactured professionally; our team was required to manufacture the chassis in our STEP Lab.

A. Equipment Selection

Our lab is equipped with both a pneumatic tube bender and a manual tube bender.

The pneumatic tube bender is superior many aspects. However, with limited material, kinking is unacceptable. It is possible that the kinking in the pneumatic bender could be remedied.

Kinking is common in hard materials and in tubes with low OD to wall thickness. Kinking can also be caused or exacerbated by new dies; dies need to be broken in before they can bend tubes consistently. The tube must also move smoothly through the die.

My team lubricated the tubes and the dies of the pneumatic bender to prevent kinking. There was some improvement, but not enough. To truly prevent kinking a tube bending mandrel must be used. The team lacks the funds for a mandrel. With no knowledge of how long the dies would take to break in, or if that would even solve the issue, and with limited material, we opted with the manual tube bender. This choice meant that all of the bending operations would need to be performed by myself and the team captain, Austin Gibson.

FACTOR	PNEUMATIC	MANUAL
Speed	Considerably Faster	
Operator Training	Requires little training	Requires considerable experience. Operator needs to use intuition, good judgement, and patience to be accurate
Accuracy	Small adjustments are difficult, can make accuracy difficult after initial bend.	Accuracy is highly dependent on operator, but tolerances within ±0.5º are achievable
Precision	Extremely precise	Extremely imprecise, each operation must be treated individually
Kinking	High rates of kinking in primary material, kinking prevention methods ineffective.	Primary tubing does not kink in this bender.

(Table 11 – Bending Equipment Selection)

The manual tube bender was selected for use in the bending operations, notching operations were performed with a manual tube notcher, and adjusted with angle grinders and dremel.

B. Issues and Resolutions.

While the manual tube bender did not present kinking issues in primary tubing, kinking was a major issue in the thinner walled secondary tubing. This OD to wall thickness ratio of this tubing is not recommended to be bent without the use of a mandrel. There were relatively few members of the chassis where bent thin walled secondary was specified other than the SIM and the FABlow. There are a few methods to prevent kinking that Austin and I tried:

METHOD	DESCRIPTION	CONCERN	RESULT
Multiple bends close together	A bend less than the critical angle for kinking is made, the tube is pulled slightly farther into the die, and then another bend is performed.	This increases the radius of the overall bend. Final radius is difficult to predict, changes from original design of one member requires changes to others.	This method was unsuccessful at stopping kinking unless bends were significantly far apart. The design change that this would mandate was deemed unacceptable.

Water Filled Tubes	Tubes are filled with water and capped at either end with custom plugs.	There is no drawback to this method if successful.	The water did not provide enough resistance and the tube kinked
Sand Packing	Sand is tamped into the tubes for bending. Sand needs to be very dry, Ideally dried in the tubes,	Drying the sand takes considerable time and is difficult to do correctly.	Large angle bends caused tubes to kink. Kinking occurred inconsistently. It is hard to tell if this method was ineffective or performed incorrectly
Heat	Tubes and dies are heated with an oxyacetylene torch to make them more pliable.	Distortion of mechanical properties of the material likely. Not best practice in tube bending. Testing required to verify integrity.	Tubes kinked with large angle bends.
Heat and Sand	Considerably more heat was applied in this method	See heat	Some bends were accomplished. It is very likely that tubes properties are highly distorted.
Thick walled secondary	AISI 4130 1" tubing with 0.065" wall thickness instead of 0.035"	Weight is added to the chassis.	Bends smoothly with no kinking at any angle.

(Table 12 – Kink Prevention Methods)

Only considering the bending capability of each method, using thick-walled secondary is by far the best way to achieve high angle bends, the only other method that can achieve bends of over 30º is the combination of heat and sand. Austin and I judged the likelihood that the heating process weakened or otherwise altered the tubing to be too high and thick walled secondary was used for the FABlow, a 90º bent member.

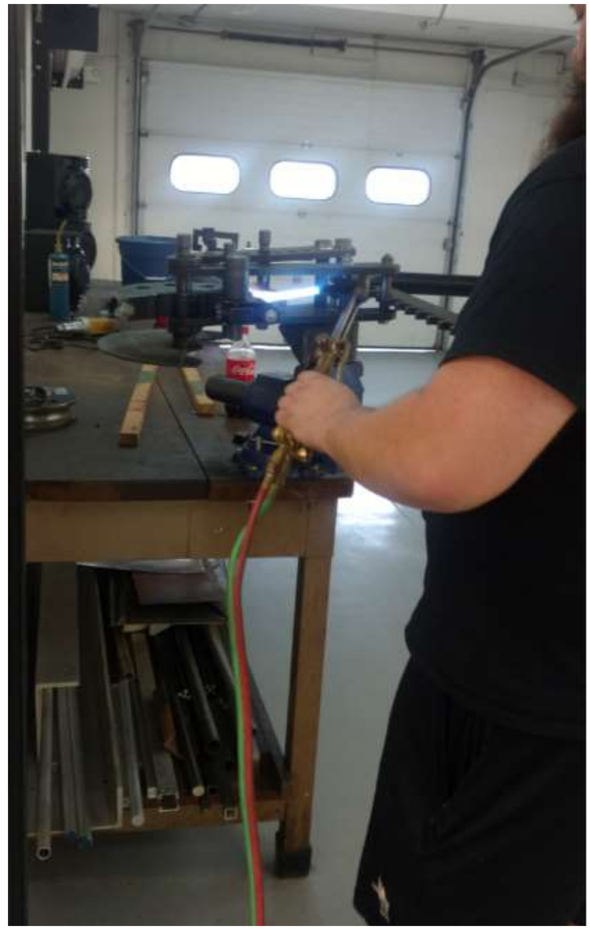
The SIM has a 22º angle bend. While sand-packing was not a consistently successful method, some bends were achieved. Luckily, the bend in the SIM occurs far to one side of the member, if a bend failed, only around a foot of material would be wasted as opposed to the whole 3' member. The low cost of error, and the considerable weight that would be added by using thick-walled tubing for the SIM led us to attempt to manufacture the SIM from thin-walled tubing using sand packing without heat. Ultimately this method was successful.

V. Lessons Learned

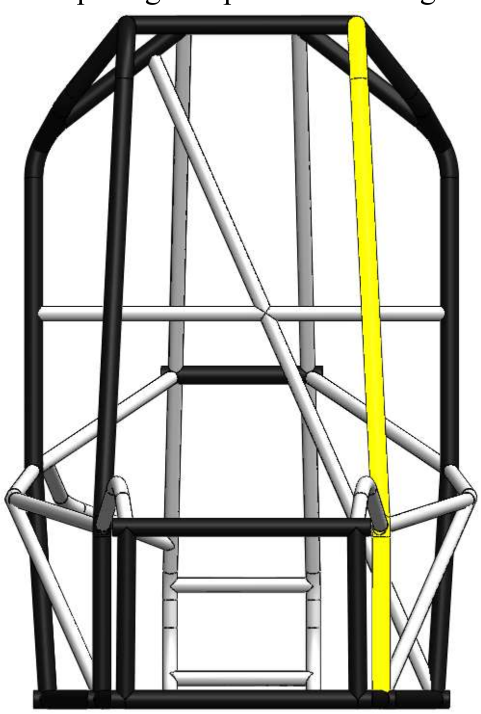
The manufacture of the frame could have been made an easier, and faster process with some changes at the design table. While manufacturable, some design elements led to difficulties in manufacturing.

A. Non-Planar Bends

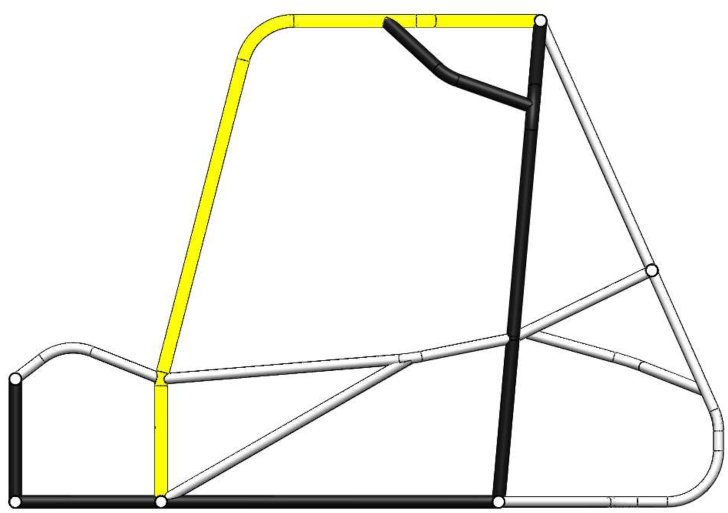
Most members in the frame are designed so that each section of the member lies in the XY,YZ or XZ plane, different sections of the member may be lie in different planes – but the sections will be perpendicular to the other. The most difficult members to manufacture were the Front Roll Hoops. These are the longest members on the frame and by design they had members that ran along lines with X,YX, Y and Z components, not lying in a global plane and requiring complicated bending.



(Figure 39 – Team Captain, Austin Gibson, Applies the Heat & Sand Packing Technique)



(Figure 40– FBMup Non-Planar Bend, Front)



(Figure 41 – FBMup Non-Planar Bend, Side)

The member is vertical in the cockpit then bends inward *and* backwards as it goes to the top of the frame. At the top of the frame the member must bend to lie perfectly horizontally and to run straight to the back of the frame (perpendicular to the front plane). Regarding set-up, it was very difficult and time consuming to figure out how the member needed to be oriented in the tube bender. Once the correct orientation was determined, it was still difficult to achieve that orientation within a tight tolerance. A small change in angle over a long member will cause significant displacement. If I were to redesign the frame this member would not bend inward, only backwards and stay in the XZ plane until the top of the frame. I would accept the greater width at the top of the frame that this would cause.

B. Nodes

The nodes of the frame are where three or more members meet at a single point. Nodes can be valuable in the frame, they tend to add torsional stiffness and overall structural stability, however they are particularly time consuming given our equipment. At a node, a tube must be notched to fit with the multiple other tubes at the node. Without CNC bending and notching there will be errors, even within tolerance, that results in the multiple tubes of the node not meeting at the exact same point. For this reason, it is nearly impossible to determine the angles of more than one notch from the drawing; it would be unwise to notch a tube at multiple angles to meet the print because it will not meet the frame.

The only method that ensures tubes will meet properly at a node, is to make the major notch and then gradually grind the tube to fit with the others at the node. This process is time consuming and requires skill, experience, and patience.

In many respects, my team and I did a good job reducing nodes, particularly in the aft bracing section, but the front section of the frame could have been designed so that there were fewer instances of multiple tubes converging at single points.

C. Tolerances

As discussed in the introduction, the chassis is one of the largest contributors to the weight of the frame and to remain competitive, teams design chassis on the very edge of what is allowed by the rules. Tight clearances increase the length of time for chassis manufacture, make the process more stressful, and require that only the most skilled team members work on the chassis. Shaving weight for competition is a trap in some respects, and my team and I fell into it in certain areas. All of our members meet the technical requirements- but just barely.

- The proximity of the fuel tank to the FABup (Figure 14) is an outright design error. There should be greater tolerance in this section, given the importance of protecting the fuel tank.
- The width of the tombstone (RRH) is acceptable, but tight. Our first tombstone had to be rejected because it did not meet driver shoulder width clearance requirements.
- The footbox is very tight with drivetrain components.

Reducing frame weight was one of our stated goals and it certainly was achieved with an over 20% weight reduction from previous frames even while being 4WD compatible. However this weight reduction was not achieved by designing thin tolerances into the components listed above. The reduction was achieved by having a design that eliminated extraneous members and required the use of fewer members. It was achieved by using Secondary material where available. The added weight from giving a 1 1/2" clearance on the fuel tank instead a 3/16" clearance would have been negligible. Future chassis designs should focus on intelligent macro design and not concern themselves with shaving minor weight at the expense of spatial tolerance.

VI. Conclusion

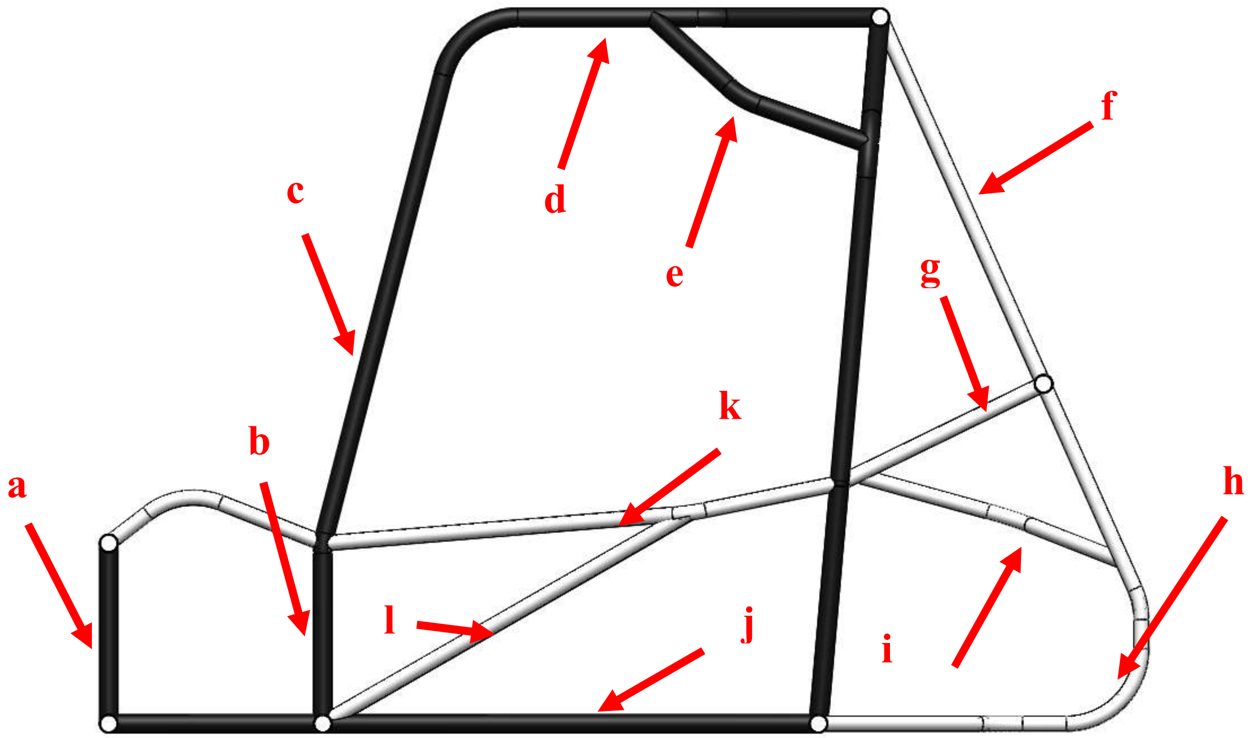
The frame was successfully designed and manufactured in 2020. A - Arm manufacture was also completed by myself and the frame team. Unfortunately, this vehicle will not see any physical competitions in the 2019-2020 season. All competitions were cancelled due the COVID-19 pandemic.

(Figure 42 - Completed Frame)

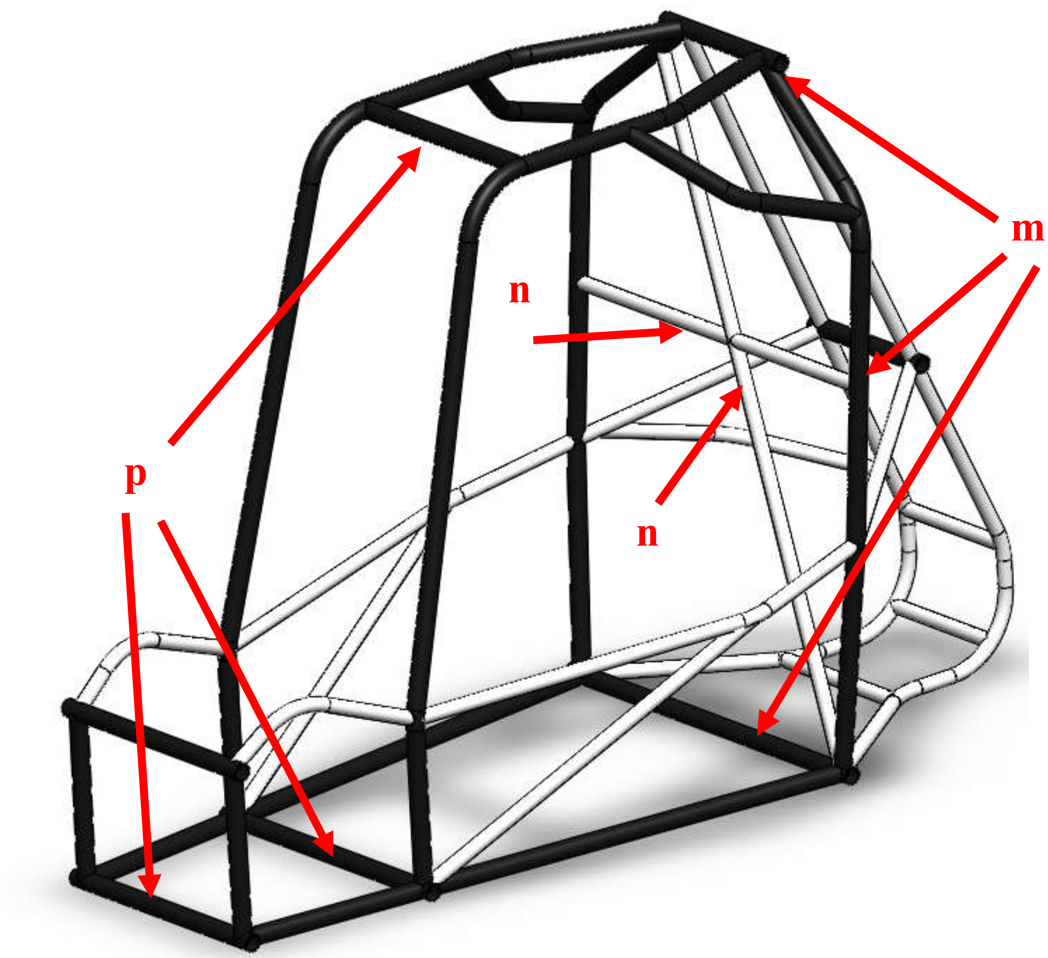
Lack of lab access prevented testing of the vehicle and the ability to benchmark performance against previous vehicles, in terms of frame qualities like torsional rigidity, and center of gravity.

A few results are known. The weight of the frame with all suspension tabs and brackets (not pictured) is 55lbs, a 10lb reduction from 2018-2019. Our tube chassis is extremely close to print with all bends within 0.5º. The frame is also very symmetrical and square, much more than previous years (special thanks to Austin Gibson). As the chassis was not used in competition this season, it has the ability to compete in the 2020-2021 and in the 2021-2022 competition season.

Appendix A-Labeling of Frame Members



a.) FBMlow’	Fore Bracing Member
b.) FBMlow	
c.) FBMup	
d.) RHO	Roll Hoop Overhead. The FBMlow, FMBup, and RHO are a single tube, referred to as the front/forward roll hoop
e.) Head Gusset	
f) FABup	Fore-Aft Brace
g.)FABmid	
h.) FABlow	
i.) Engine Containment Gusset	
j.) LFS	Lower Frame Side Member -Some literature refers to this as the SIMlow
k.) SIM	Side Impact Member
l.) SIM Brace	



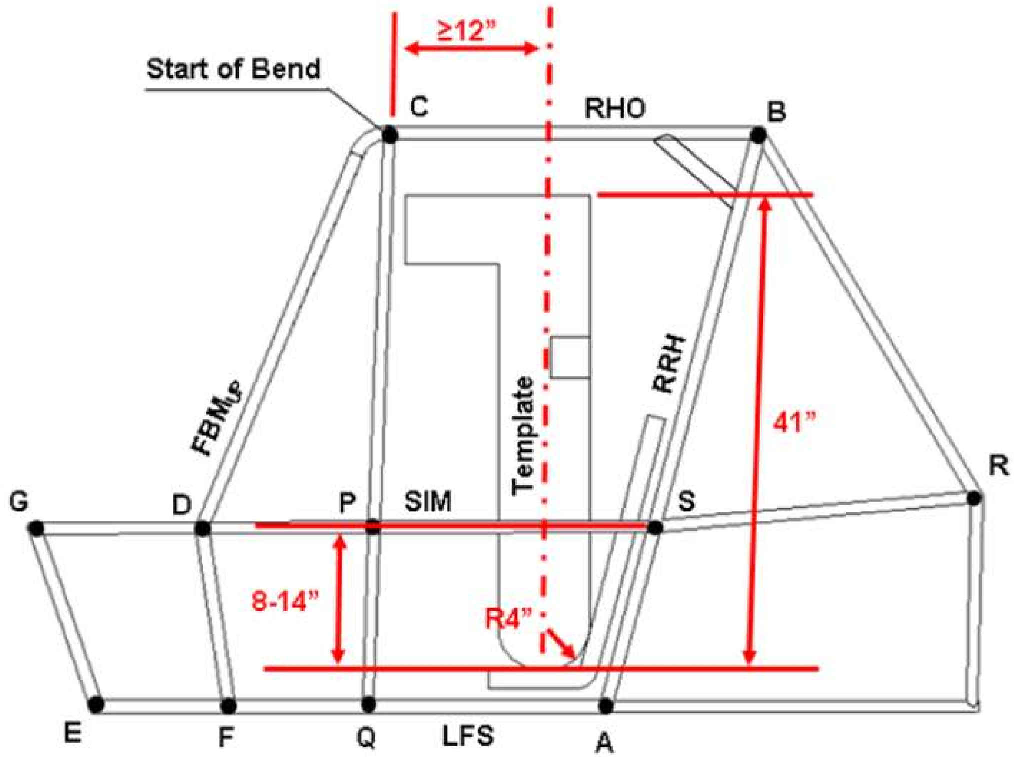
m.) RRH — Refers to the entire Rear Roll Hoop composed of two side members and two lateral crosses

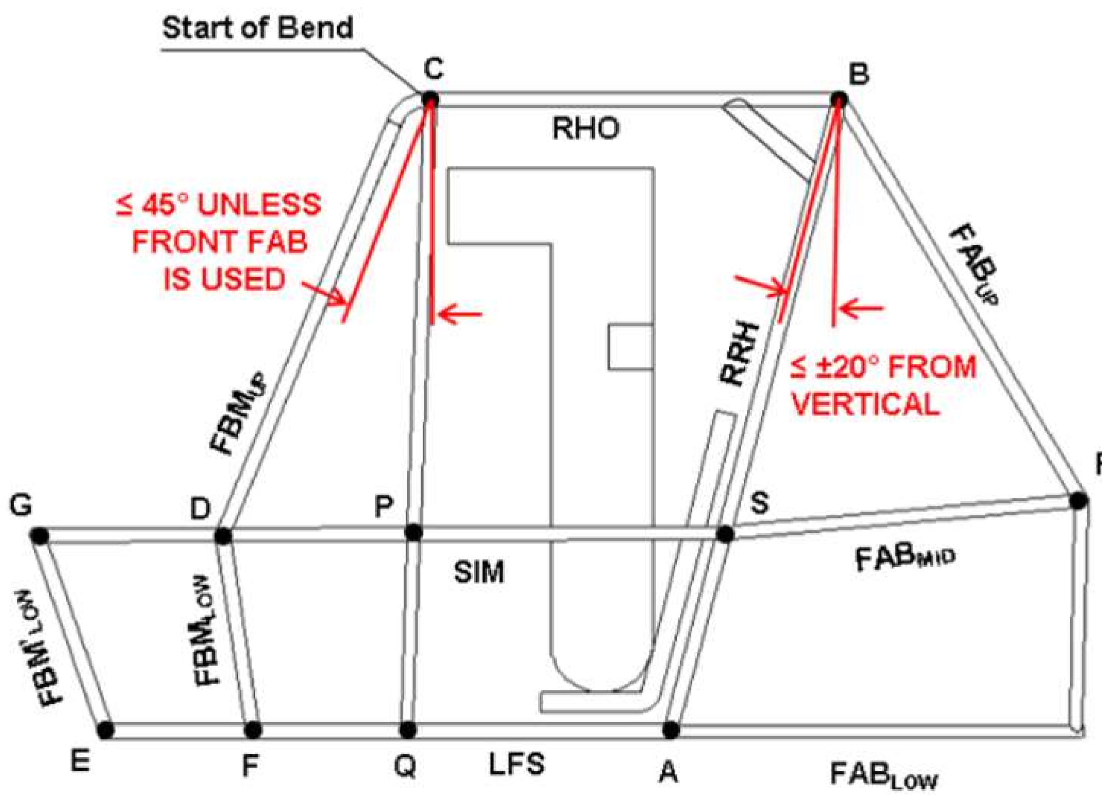
n.) LDB — Lateral Diagonal Brace

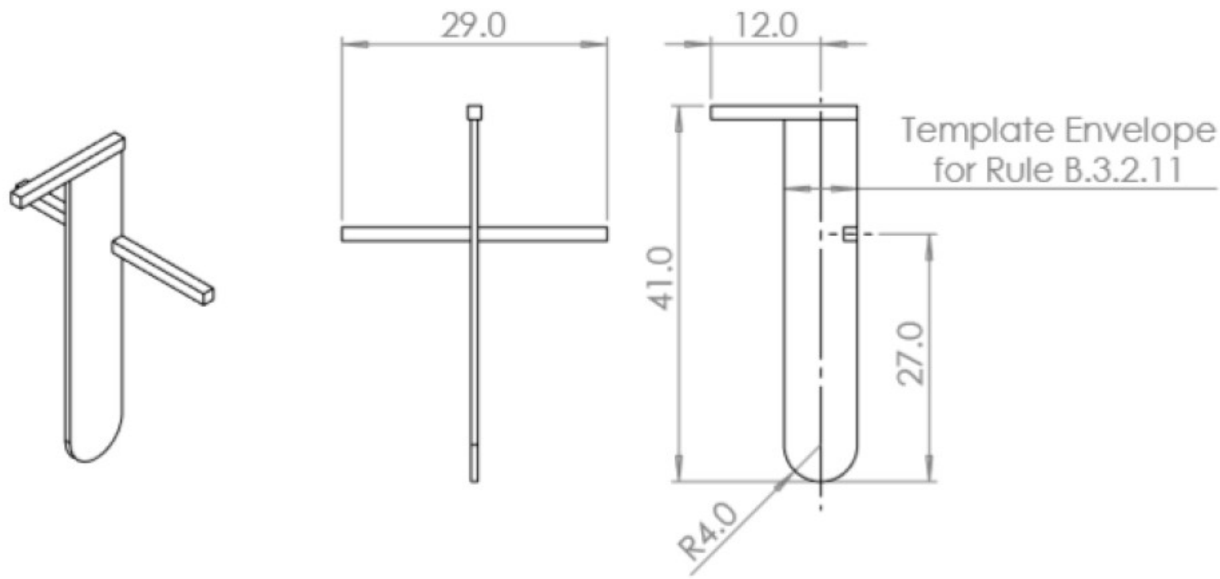
o.) LDB Cross Brace

p.) LC — Lateral Crosses connect the left and right halves of the chassis. LC's have specific three letter designations, but are usually referred to by location e.g. Rear LC, Overhead LC.

Appendix B – Selected Geometric Rules and Requirements







B.3.2.6 - RRH – Roll Hoop
The RRH is a planar structure behind the driver's back and defines the boundary between the front-half (fore) and rear-half (aft) of the roll cage. The driver and seat must be entirely forward of this panel. The RRH is substantially vertical but may incline by up to 20 deg. from vertical. The minimum width of the RRH, measured at a point 686 mm (27 in.) above the inside seat bottom, is 736 mm (29 in.). The vertical members of the RRH may be straight or bent and are defined as beginning and ending where they intersect the top and bottom horizontal planes (points AR and AL, and BR and BL in Figure B-8). The vertical members must be continuous tubes (i.e. not multiple segments joined by welding). The vertical members must be joined by ALC and BLC members at the bottom and top. ALC and BLC members must be continuous tubes or adhere to B.3.2.14 - Butt Joints. ALC, BLC, RRH members, LDB and the shoulder belt member must all be coplanar.

Rule B.3.2.8.1 - Gussets for Lateral Clearance
If a gusset is used to brace the RHO and RRH to achieve the Lateral Clearance in Rule B.3.3.1 - Lateral Space the added members must be a primary material (B.3.2.16 - Roll Cage Materials); completely welded around the circumference of both ends of the gusset.
Gusset members connecting the SIM to RRH or FBM for the purposes of achieving the Lateral Clearance in Rule B.3.3.1 - Lateral Space may be primary or secondary material (B.3.2.3 - Secondary Members) and must be closed in with Body Panels

Rule B.3.2.12 - FBM – Front Bracing Members
Front Bracing Members must join the RHO, the SIM and the LFS at Points C, D and F. The upper Front Bracing Members (FBMUP) must join points C on the RHO to point D on the SIM. The lower Front Bracing Members (FBMLOW) must join point D to point F. The FBM must be continuous tubes. The angle between the FBMUP and the vertical must be less than or equal to 45 deg. If Front FAB, per Rule B.3.2.13.1 - Front Bracing, is used there is no angle requirement between FBM and vertical.

Rule B.3.2.13.2 - Rear Bracing
Rear systems of FAB must create a structural triangle, in the side view, on each side of the vehicle. Each triangle must be aft of the RRH, include the RRH vertical side as a member, and have one vertex at Point B and one vertex at either Point S or Point A. The members forming this structural triangle must be continuous members; but bends of less than 30 deg. are allowable.
The third (aft) vertex of each rear bracing triangle, Point R (Figure B-19), must additionally be structurally connected to whichever Point, S or A, is not part of the structural triangle. This additional connection is considered part of the FAB system, and is subject to B.3.2.1 - Member Requirements, but may be formed using multiple joined members, and this assembly, from endpoint to endpoint, may encompass a bend of greater than 30 deg.
Attachment of rear system FAB must be within 51 mm (2 in) of Point B, Point S and Point A, on each side of the vehicle. Distances are measured as a straight-line distance from centerline to centerline. The aft vertex of each rear bracing triangle defines Point R and must be joined by an LC of a minimum of 203.5 mm (8 in.) in length per B.3.2.5 - LC – Lateral Cross Member.

ACKNOWLEDGEMENTS

SAE's Baja Design Series is a team competition, and the project and legacy is huge. My work outlined in this thesis exists as a small part of a larger project and as a contribution to the enduring Saxon Racing Program

I would like to thank my frame team: Lucas Chaplain-Koski, Ian King, Chris Becker, and Nick Wieme. They each have made significant contributions to the Frame subsystem, and supported in the design and manufacture of the tube chassis.

Much of the analysis I preformed was on front suspension components. Chris Becker designed the A-Arms featured in my analysis and Randy Saffir designed and re-designed, and re-re-designed the uprights.

A special thanks needs to be extended to the Saxon Racing Club E-board, and Club Advisor Dr. Ghotbi. President Nathanial Blakely picked all the right fights with all the right people so that the team could do what the team needed to do.

Dr. Maha vigilantly oversaw the Baja Senior Project and made sure that the team was able to present and discuss engineering decisions intelligently.

Jim Mighells let Austin and my team make a mess and a lot of noise in the STEP Lab almost 24/7, and was reservoir of much needed technical information and advice.

Finally, I need to thank Team Captain Austin Gibson. He was able to keep a fire under everyone while simultaneously doing all of the welding on the vehicle, and designing his own rear suspension. He taught me tube bending and notching. He called out bad design ideas. The two of us did most of the frame manufacturing together. The team absolutely could not have succeeded without him, and neither could I.