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Optimization of Formula SAE Electric Vehicle Frame with Finite Element Analysis

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DISCLAIMER

The material contained within this report has been prepared with the intent of being fully complaint and optimal with the 2016 Formula SAE Rules. However, the rules and the committee that enforces them are, simply put, human. If they don't like it, you don't run. In particular with Electric Vehicles, everything tends towards caution. Additionally, any grey areas in the rules can be in violation of Rule A3.6. It reads "The violation of intent of a rule will be considered a violation of the rule itself."

Additionally, the FSAE Rules undergo a major revision every two years. 2017 will be one of those years. While changes to the baseline frame design are unlikely, anything is possible. Any use of the final frame design and car layout presented within should consult the 2017 rules for any changes needed.

-Alex

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`INTRODUCTION, DESIGN REQUIRMENTS AND THEORY

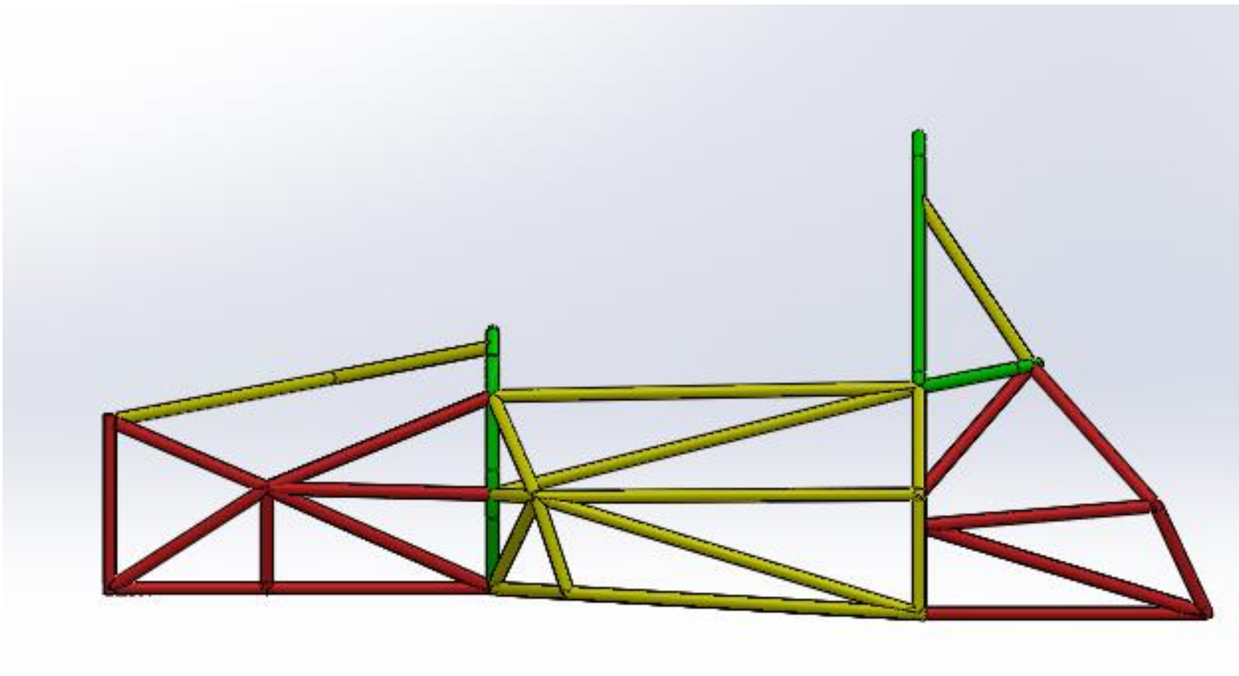
Introduction

Formula SAE is a student design competition, organized by the Society of Automotive Engineers (SAE) since 1979. In the last few years, spin-offs, using Electric and Hybrid powertrains have been started. The rules allow great freedom in the design and optimization of the cars. The University of Akron has a very successful Combustion team, and in the fall of 2013, the Electric team was started. The 2015 and 2016 vehicles use a 300 Volt Lithium-Polymer Battery pack (Accumulator), and a 3 Phase Electric Motor (Tractive System), limited to 80 KW (106 HP). The vehicle weighs around 550 lbs, without a driver, and can reach speeds of over 60mph.

Design Requirements

Simply put, the main design requirement is that the frame passes tech Inspection. The relevant parts of the 2016 rules can be seen in Appendix A. The baseline design is thin walled 1” mild steel tubing, welded into a space-frame. Below shows the 2016 Frame. Members are color coded by wall thickness.

Figure 1: 2016 Frame

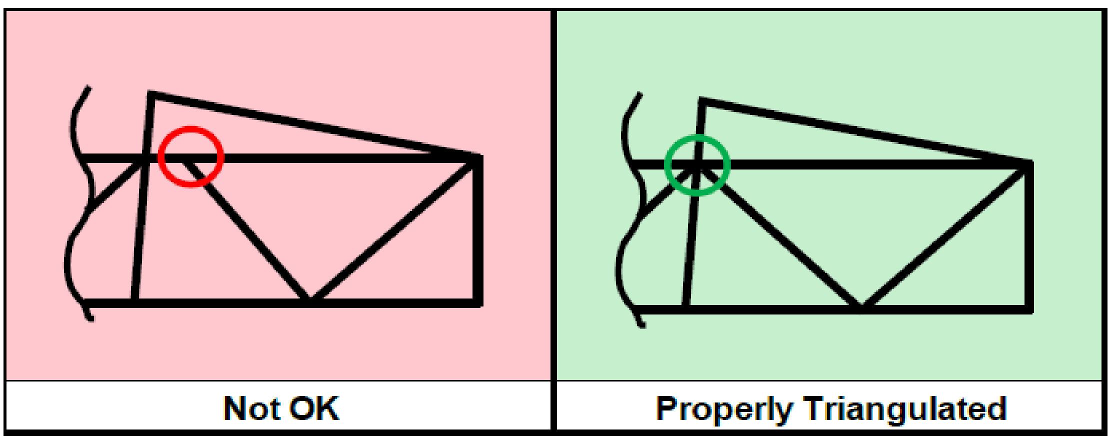


On the far left is the front Bulkhead (0.049” wall, square). Moving backwards, there is bracing for the bulkhead (0.049” wall, round) and for the front hoop (0.065” wall) Beyond those is the front roll hoop (0.095” wall). The driver’s cell, side impact structure and accumulator protection are all merged into one structure (0.065”). Behind the main hoop (0.095”) is the hoop’s bracing (0.065”) and the bent shoulder harness bar (0.095”). Moving downwards, there is the bracing supports, rear suspension box, and the tractive System protection (0.049”). The assigned colors, corresponding to wall thickness, are propagated through the report.

Additionally, two interior cross-section templates must be met. One specifies the area of the drivers cell, from opening to side-impact. The other regulates the area for his legs, from the front hoop, until 4” from the pedals. Finally, the car must be able to fit Percy, the 95th Percentile man template.

Per rule T3.3, all load paths must be properly triangulated, such that all members, given a planar load, are only in tension or compression, no shear. The front bulkhead supports, front hoop supports, main hoop supports, and side impact structure require a properly triangulated structure to connect them to each other and the roll hoops.

Figure 2: Proper Triangulation



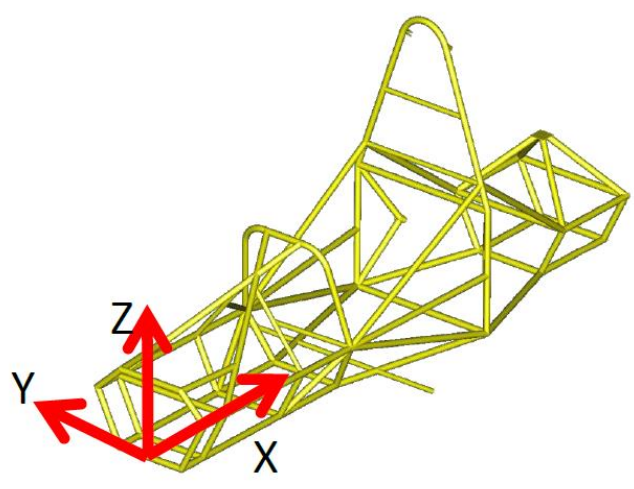
The rules are very flexible, allowing great design creativity. Alternate materials are permitted, provided that equivalency to AISI 1010 steel can be proved. These will be discussed later.

Augmenting the baseline rules, there is an additional set of rules, called Alternate Frame. These are optional, but state that if Finite Element Analysis is done; certain provisions in the baseline rules can be ignored. These provisions deal mostly with the mounting and angling of roll hoops and their braces. Analysis required for the alternate frame will be used as the criteria to optimize the frame. Provided it meets strength requirements, the optimization criterion for any section of the frame is number of members, then weight.

There are 8 loading conditions that the frame must be simulated in to meet Alternate Frame Requirements. These represent a wide range of worst case scenarios encountered in an FSAE Competition. Two are for rollover, with loads being applied to the top of the front hoop, and top of the main hoop. Two more are for a high speed Collison, Straight on and off-axis. For lower speed impacts, there is a test for side impact. For Electric Vehicles only, there are spin-off tests from the main side impact. These are the accumulator protection in a side impact, and Tractive System protection in side/Rear impact. Finally, there is loading on the shoulder harness mounting bar. Due to the limitations of Solidworks Simulations, the off axis impact will not be studied. Two other cases, regarding seat-belt tab locations, and loading of the accumulator itself will not be analyzed, as they are subject to other factors.

Table 1: Loading Conditions, Per Alt-Frame Rules

Top of Front Hoop		Top of Main Hoop	
X	6 KN	X	6 KN
Y	5 KN	Y	5 KN
Z	-9 KN	Z	-9 KN
Front Bulkhead		Rear Impact	
X	120 KN	X	-5.5 KN
Y	0 KN	Y	5.5 KN
Z	0 KN	Z	0 KN
Side Impact		Shoulder Points	
X	0 KN	Direction of Driver	
Y	7 KN		7 KN
Z	0 KN	Z	0 KN



All Simulations require the same boundary conditions, and same failure criteria. The Boundary Conditions are that the nodes at the base of each hoop are fixed against translation in three dimensions, but able to rotate in three dimensions. The maximum allowable deflection is 25 mm, and failure cannot occur anywhere in the structure. Failure is interpreted to mean exceeding the ultimate strength of the member. As a safety factor, my goal is to keep the stress under yield stress.

Finite Element Theory

Finite Element Analysis (FEA) is a powerful tool for analyzing any and all mechanical problems. To analyze a problem, first a geometry and material must be defined. Next, loads and boundary conditions are applied. Third, a mesh is applied, breaking the larger problem into a series of smaller problems. The mesh, in turn defines a large matrix of equations, which then must be solved simultaneously.

For a full 3D Simulation, the model will be broken into 3D elements, commonly either quadrilaterals or tetrahedrons. The 3D volume of the shape will then be filled with these elements. For our needs, a simpler element will suffice. A truss element takes into consideration the translation of an element in 2D or 3D space due to axial loading. For a truss element in 2D space, starting at point 1, ending at point 2:

$$\underline{F} = \underline{K}\,\underline{D}$$

$$\begin{Bmatrix} F1X \\ F1Y \\ F2X \\ F2Y \end{Bmatrix} = \frac{AE}{L} \begin{bmatrix} C^2 & CS & -C^2 & -CS \\ CS & S^2 & -CS & -S^2 \\ -C^2 & -CS & C^2 & CS \\ -CS & -S^2 & CS & S^2 \end{bmatrix} \begin{Bmatrix} X1 \\ Y1 \\ X2 \\ Y2 \end{Bmatrix}$$

S and C are sine and cosine, of theta. Theta is the angle between the x axis and the axis $\hat{x}$, witch runs from point 1 to point 2. A is the area, E is the Young's modulus, and L is the length of the member. The output is in global coordinates.

For a system of multiple members, a matrix as shown above is created for each member. The matrixes are then combined, and solved, using boundary conditions.

All finite element analysis is ultimately an approximation. Different element types have their advantages over others. In particular, truss elements don't consider the moment of inertia of the beam, I. As such, it is assumed that the beam neither twists, nor buckles. A beam element considers only the effects of bending stress, and functions as a continuation of a shear-moment diagram.

$$\underline{F} = \underline{K}\ \underline{D}$$

$$\begin{Bmatrix} F1Y \\ M1 \\ F2Y \\ M2 \end{Bmatrix} = \frac{EI}{L^3}\begin{bmatrix} 12 & 6L & -12 & 6L \\ 6L & 4L^2 & -6L & 2L^2 \\ -12 & -6L & 12 & -6L \\ 6L & 2L^2 & -6L & 4L^2 \end{bmatrix}\begin{Bmatrix} Y1 \\ \phi 1 \\ Y2 \\ \phi 2 \end{Bmatrix}$$

Here the output assumes that local and global coordinates are the same, and that there is no change in the x direction.

By combining the two sets of matricies, and accounting for local versus global coordinates, we obtain the following matrix for 2D Space.

$$\underline{F} = \underline{K}\ \underline{D}$$

$$\begin{Bmatrix} F1X \\ F1Y \\ M1 \\ F2X \\ F2Y \\ M2 \end{Bmatrix} = \underline{K} \begin{Bmatrix} X1 \\ Y1 \\ \phi 1 \\ X2 \\ Y2 \\ \phi 2 \end{Bmatrix}$$

$$\underline{k} = \frac{E}{L} \times$$

$$\begin{bmatrix} AC^2 + \frac{12I}{L^2}S^2 & \left(A - \frac{12I}{L^2}\right)CS & -\frac{6I}{L}S & -\left(AC^2 + \frac{12I}{L^2}S^2\right) & -\left(A - \frac{12I}{L^2}\right)CS & -\frac{6I}{L}S \\ & AS^2 + \frac{12I}{L^2}C^2 & \frac{6I}{L}C & -\left(A - \frac{12I}{L^2}\right)CS & -\left(AS^2 + \frac{12I}{L^2}C^2\right) & \frac{6I}{L}C \\ & & 4I & \frac{6I}{L}S & -\frac{6I}{L}C & 2I \\ & & & AC^2 + \frac{12I}{L^2}S^2 & \left(A - \frac{12I}{L^2}\right)CS & \frac{6I}{L}S \\ & & & & AS^2 + \frac{12I}{L^2}C^2 & -\frac{6I}{L}C \\ \text{Symmetry} & & & & & 4I \end{bmatrix} \tag{5.1.11}$$

This equation can then be extrapolated into a full 3D equation, with rotation and translation of each end in 3 dimensions, resulting in a 12x12 matrix.

$$
\underline{\hat{k}} = \begin{array}{c}
\begin{array}{cccccccccccc} \hat{d}_{1x} & \hat{d}_{1y} & \hat{d}_{1z} & \hat{\phi}_{1x} & \hat{\phi}_{1y} & \hat{\phi}_{1z} & \hat{d}_{2x} & \hat{d}_{2y} & \hat{d}_{2z} & \hat{\phi}_{2x} & \hat{\phi}_{2y} & \hat{\phi}_{2z} \end{array} \\
\left[\begin{array}{cccccc|cccccc}
\frac{AE}{L} & 0 & 0 & 0 & 0 & 0 & -\frac{AE}{L} & 0 & 0 & 0 & 0 & 0 \\
0 & \frac{12EI_z}{L^3} & 0 & 0 & 0 & \frac{6EI_z}{L^2} & 0 & -\frac{12EI_z}{L^3} & 0 & 0 & 0 & \frac{6EI_z}{L^2} \\
0 & 0 & \frac{12EI_y}{L^3} & 0 & -\frac{6EI_y}{L^2} & 0 & 0 & 0 & -\frac{12EI_y}{L^3} & 0 & -\frac{6EI_y}{L^2} & 0 \\
0 & 0 & 0 & \frac{GJ}{L} & 0 & 0 & 0 & 0 & 0 & -\frac{GJ}{L} & 0 & 0 \\
0 & 0 & -\frac{6EI_y}{L^2} & 0 & \frac{4EI_y}{L} & 0 & 0 & 0 & \frac{6EI_y}{L^2} & 0 & \frac{2EI_y}{L} & 0 \\
0 & \frac{6EI_z}{L^2} & 0 & 0 & 0 & \frac{4EI_z}{L} & 0 & -\frac{6EI_z}{L^2} & 0 & 0 & 0 & \frac{2EI_z}{L} \\
\hline
-\frac{AE}{L} & 0 & 0 & 0 & 0 & 0 & \frac{AE}{L} & 0 & 0 & 0 & 0 & 0 \\
0 & -\frac{12EI_z}{L^3} & 0 & 0 & 0 & -\frac{6EI_z}{L^2} & 0 & \frac{12EI_z}{L^3} & 0 & 0 & 0 & -\frac{6EI_z}{L^2} \\
0 & 0 & -\frac{12EI_y}{L^3} & 0 & \frac{6EI_y}{L^2} & 0 & 0 & 0 & \frac{12EI_y}{L^3} & 0 & \frac{6EI_y}{L^2} & 0 \\
0 & 0 & 0 & -\frac{GJ}{L} & 0 & 0 & 0 & 0 & 0 & \frac{GJ}{L} & 0 & 0 \\
0 & 0 & -\frac{6EI_y}{L^2} & 0 & \frac{2EI_y}{L} & 0 & 0 & 0 & \frac{6EI_y}{L^2} & 0 & \frac{4EI_y}{L} & 0 \\
0 & \frac{6EI_z}{L^2} & 0 & 0 & 0 & \frac{2EI_z}{L} & 0 & -\frac{6EI_z}{L^2} & 0 & 0 & 0 & \frac{4EI_z}{L}
\end{array}\right]
\end{array}
\tag{5.5.3}
$$

By generating a matrix for each of the beams, orienting them in the global coordinates system, and applying boundary conditions, the system of equations can be solved.

In finite element analysis, you always need to ask “is this result reasonable?” Knowing that a baseline rules frame should innately be safe, it will allow us know when to accept or reject the results FEA creates.

Vehicle Dynamics

In designing a FSAE Vehicle, an understanding of vehicle dynamics is important. Different measureable parameters include the vehicle's track width (t), wheelbase (L), center of gravity height (h), Tire stiffness (C) weight, and weight distribution,

First and foremost, there is Rule T6.7.2, the tilt table. The car must be able to be safely rolled to 60 degrees, (1.7 g's). Rollover stability, also called the Static Stability Factor is defined as follows. For maximum stability, you want a wide track width, and a low center of gravity.

$$SSF = g's = \frac{a_c}{g} = \frac{v^2}{r * g} = \frac{t}{2h}$$

Second, there are the changes between the vehicle at rest and in motion. At a standstill, there is only the force of gravity, through the center of gravity. Under any movement, there will be forces from the aerodynamic drag, through the center of pressure. Under accleleration, there will be a weight transfer backwards, causing "Power squat". In deceleration, "Break Dive" is observed. The magnitude of these changes depends on the stiffness of the springs. By rule, cars are not allowed to contact the ground in normal operation, and therefore decent ground clearance is needed.

Third, there is the effect of aerodynamics on the car. Aerodynamics is a balance between downforce and drag. Both downforce and drag increase as a square of the velocity. Drag is naturally inherent in the car, but can be reduced by body panels. Adding wings creates more drag, but also creates downforce. The added drag limits the top speed of the car, but FSAE Cars rarely reach their top speed in competition. Instead, adding wing has a minimal penalty, but then allows for better cornering, and better acceleration.

Finally, there is the steering nature of the vehicle. With any vehicle, the vehicle's behavior changes as speed increases. Road cars are said to understeer, where at higher speeds, more steering wheel turning is needed to make a turn. Race cars are set up to oversteer, where less effort is needed to make a turn at a given speed. If the understeer coefficient is positive, it is oversteering, and if negative, is understeering. The understeer coefficient is calculated by:

$$K_{US} = \frac{W_F}{C_F} - \frac{W_R}{C_R}$$

Where F and R denote the front and rear of the car

FSAE Cars have most of the weight in the rear, with the heaviest forward mass being the driver. FSAE Tires are unidirectional, meaning there are no designated front or back tires. However, lowering air pressure in the rear tires, or raising it in the front can shift the coefficient towards oversteer.

MATERIALS

In the rules, the baseline material is AISI 1010 steel. Alternate materials are permitted, provided that structural equivalency to the baseline can be proved. For equivalency, tensile yield Strength, tensile Ultimate Strength and Buckling Modulus must be equivalent.

For Alternate materials, there are no considerations made for using an alloy steel, such as 4130-N. For steel, the criteria simplifies down to an equivalent area and moment of inertia. By rule, there are minimum wall thicknesses that must be met, regardless of equivalency. To maintain equivalency with thinner walled tube, the outer diameter must be increased beyond the nominal 1". For steel, there are two "levels" of alternate materials. The thicker level only requires documentation; the thinner requires tensile testing, to prove weld quality.

Aluminum is another option, and can be considered superior, as it is lighter than steel. However, Aluminum also requires that the analysis be done considering the "as welded" strength of the material, unless otherwise show that it has been solution heat treated and artificially aged. The other issue with Aluminum is that the main hoop, and its supports must be made out of steel. This requires a mechanical joint, and adds undesired complexity.

Titanium and Magnesium are also permitted, per the rules. However, any Titanium or Magnesium that has been welded is strictly prohibited. This means that large quantities of mechanical joints must be made and used.

By using an iterative solver, different sizes of permitted alternate materials can be determined. The optimization criteria for the solver was minimum area, for a given size. For Aluminum, 2024-T351 is a common aircraft fuselage material, while 6061-T6 is a common, almost generic aluminum. Magnesium Alloy AM60 is a common cast alloy. Titanium Beta C, (Ti-3Al-8V-6Cr-4Mo-4Zr), is an alloy known for it's very high strength, light weight, and corrosion resistance.

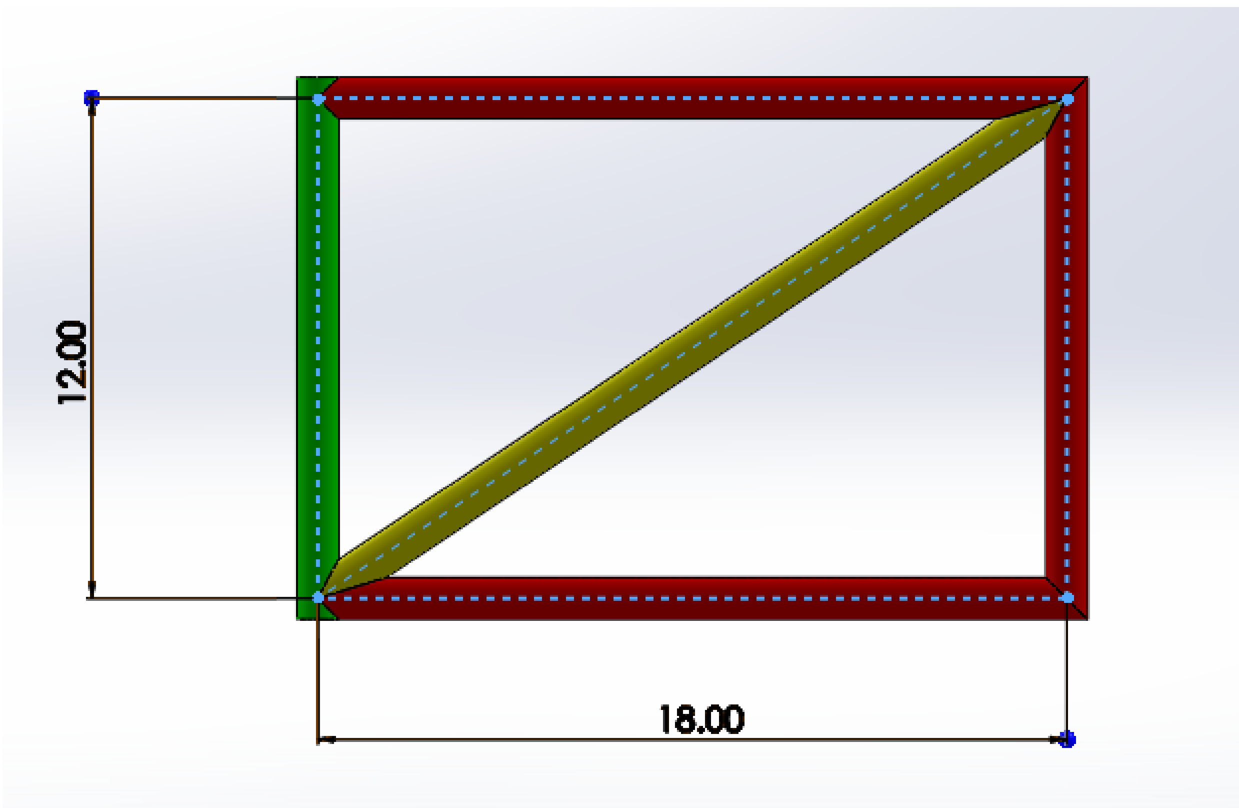
Table 2: Acceptable Alternate Materials

AISI 1010			OD (IN)	mm	Thick (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	2.90E+07	Psi	1	25.4	0.094	2.4	Round	2.69E-01	2.78E-02	8.08E+05	1.19E+04	1.42E+04
Syield	4.42E+04	psi	1	25.4	0.063	1.6	Round	1.85E-01	2.04E-02	5.93E+05	8.20E+03	9.81E+03
Sult	5.29E+04	psi	1	25.4	0.047	1.2	Round	1.41E-01	1.61E-02	4.66E+05	6.25E+03	7.48E+03
Non-Testing			OD (IN)	mm	Thick (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	2.90E+07	Psi	1.165354	29.6	0.079	2	Round	0.268795	3.99E-02	1.16E+06	1.19E+04	1.42E+04
Syield	4.42E+04	psi	1.296588	32.9	0.047	1.2	Round	0.18543	3.62E-02	1.05E+06	9.81E+03	0.00E+00
Sult	5.29E+04	psi	1	25.4	0.047	1.2	Round	0.14141	1.61E-02	4.66E+05	0.00E+00	4.10E+06
Testing			OD (IN)	mm	Thick (IN)	mm	mm	A in^2	I in^4	E*I	Yld Force	Ult Force
E	2.90E+07	Psi	1.42126	36.1	0.063	1.6	Round	0.268795	6.21E-02	1.80E+06	1.19E+04	1.42E+04
Syield	4.42E+04	psi	1.701225	43.2	0.035	0.9	Round	0.18543	6.43E-02	1.87E+06	9.81E+03	0.00E+00
Sult	5.29E+04	psi	1.305774	33.2	0.035	0.9	Round	0.14141	2.85E-02	8.28E+05	0.00E+00	4.10E+06
2024-T351			OD (IN)	mm	ID (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	1.06E+07	Psi	1.294364	32.9	0.118	3	Round	4.36E-01	7.62E-02	8.08E+05	2.05E+04	2.97E+04
Syield	4.70E+04	psi	1.178343	29.9	0.118	3	Round	3.93E-01	5.59E-02	5.93E+05	1.85E+04	2.67E+04
Sult	6.80E+04	psi	1.096209	27.8	0.118	3	Round	3.63E-01	4.40E-02	4.66E+05	1.70E+04	2.47E+04
6061-T6			OD (IN)	mm	ID (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	1.00E+07	Psi	1.317589	33.5	0.118	3	Round	4.45E-01	8.08E-02	8.08E+05	1.78E+04	2.00E+04
Syield	4.00E+04	psi	1.199309	30.5	0.118	3	Round	4.01E-01	5.93E-02	5.93E+05	1.60E+04	1.80E+04
Sult	4.50E+04	psi	1.115578	28.3	0.118	3	Round	3.70E-01	4.66E-02	4.66E+05	1.48E+04	1.66E+04
Ti-3Al-8V-6Cr-4Mo-4Zr			OD (IN)	mm	ID (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	1.51E+07	Psi	1.45333	36.9	0.049	1.2	Round	2.16E-01	5.34E-02	8.08E+05	3.43E+04	3.84E+04
Syield	1.59E+05	psi	1.315677	33.4	0.049	1.2	Round	1.95E-01	3.92E-02	5.93E+05	3.09E+04	3.46E+04
Sult	1.78E+05	psi	1.218293	30.9	0.049	1.2	Round	1.80E-01	3.08E-02	4.66E+05	2.86E+04	3.20E+04
Magnesium Alloy AM60			OD (IN)	mm	ID (IN)	mm		A	I	E*I	Yld Force	Ult Force
E	6.50E+06	Psi	1.43883	36.5	0.155041	3.9	Round	6.25E-01	1.31E-01	8.50E+05	1.19E+04	2.00E+04
Syield	1.90E+04	psi	1.402181	35.6	0.105927	2.7	Round	4.31E-01	9.12E-02	5.93E+05	8.20E+03	1.38E+04
Sult	3.20E+04	psi	1.39808	35.5	0.079407	2	Round	3.29E-01	7.18E-02	4.66E+05	6.25E+03	1.05E+04

It can be seen, that for the Steel, the area increases to the necessary size slower than the moment of Inertia. In both aluminum alloys, and the titanium, the buckling modulus is the slower growing factor. For the magnesium, the force at yield is the limiting factor. Not all of these are necessarily readily available in the materials and sizes, but are all structurally equivalent to AISI 1010 Steel. Of these materials, one of the easily available sizes is 1.375" x 0.035", replacing the 1" x 0.049"

All of these tubes can be said to be equivalent to the baseline. However, other factors influence how it behaves as well. As such, I have created an Arbitrarily sized, but representative structural equivalency test. The loads are 15KN Down, from the top left node, and 30 KN to the left, from the upper right node. The Boundary conditions are fixed translation and fixed rotation in X and Y for the bottom left node, and fixed Y translation at the bottom right node. The starting geometry can be seen in figure 3 below

Figure 3: Materials Test



All of the tests had approximately the same deformation pattern, as shown by the baseline below. The main change was the magnitude of the stress and deformation. The results are shown in table 2 below.

Figure 4: Materials Test - Deformed

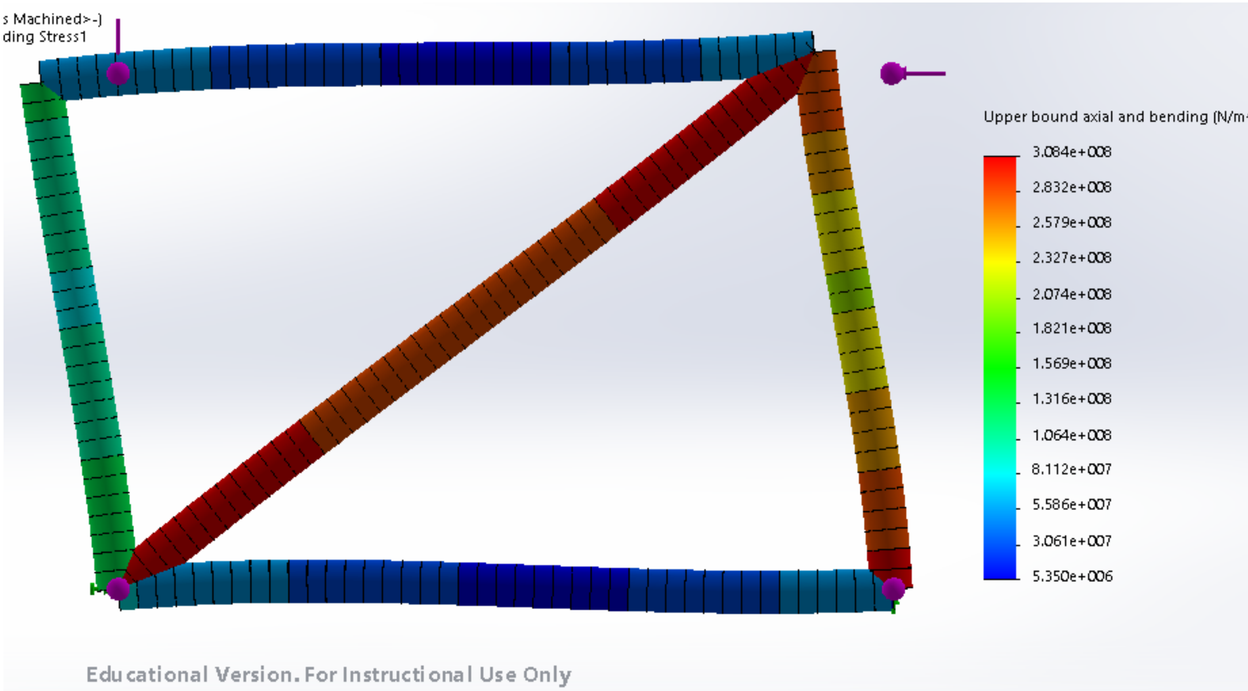


Table 3: Testing results.

	Stress (PA)	% Yield	Dspl (mm)	Wt
Baseline - 4130	3.08E+08	71%	1.173	4.04
Baseline - 1010	3.09E+08	102%	1.203	4.05
No Test - 1010	2.97E+08	98%	0.889	4.61
Testing - 1010	3.31E+08	109%	1.224	3.96
Aluminum - 2024	1.50E+08	47%	1.544	3.06
Aluminum - 6061	1.47E+08	53%	1.584	3.03
Titanium	3.12E+08	29%	2.227	2.54
Magnesium	1.45E+08	111%	2.338	1.96

Mechanical Fasteners

This suggests that Titanium would be an ideal material to make the frame out of. However, certain parts must be steel. As such, connecting two non-homogenous materials requires a mechanical fastener. There are two types, permitted by rule. The first is a double-lug, for attaching tubes at an angle to one-another. The second is a sleeve, for two tubes that are in-line.

Figure 5: Double-Lug Joint

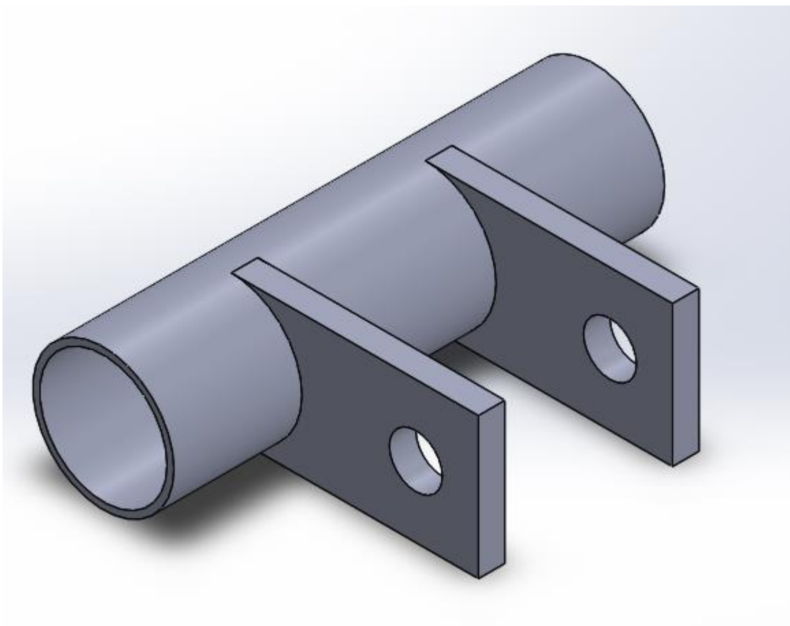
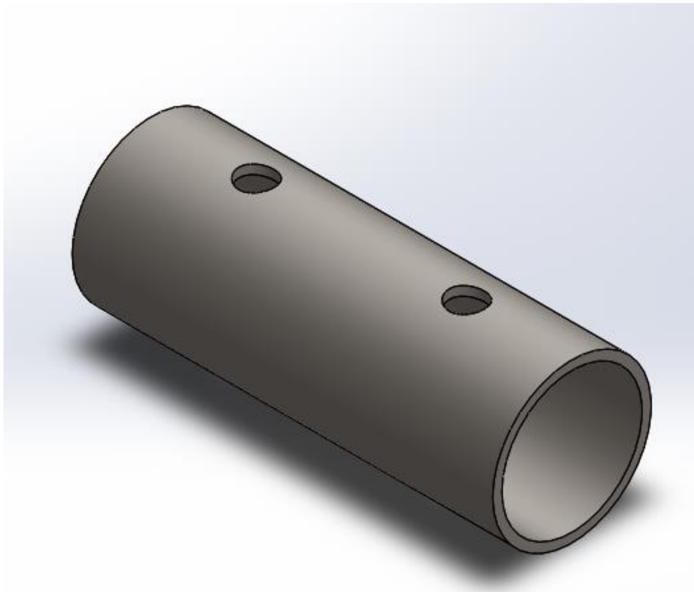


Figure 6: Sleeved joint



For manufacturing, the double lug section requires welding. No welding is permitted on Titanium and Magnesium alloys. Thus, roughly half of the frame has to be steel or aluminum. Additionally, the 3/16" tabs required are far thicker than other tabs on the car, incurring cost in buying stock, or a weight penalty. For the sleeve, 1.125" x 0.065" wall can be used for 1" OD tubes. Our team uses this material, in limited quantities, for the steering column supports.

The maximum tensile load a given tube can take is roughly 30 KN. Due to non-homogeneous geometry, stress concentrations occur. In the double-lug, this occurs at the edge of the weld, already a potentially weak point. In the sleeve, there is a stress build-up at the leading edge of the hole, where it could possibly tear-out

Figure 7: Double-Lug at 30KN

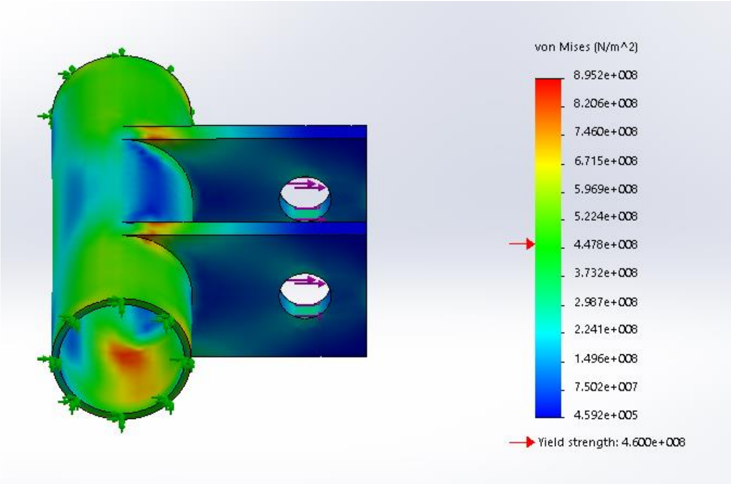
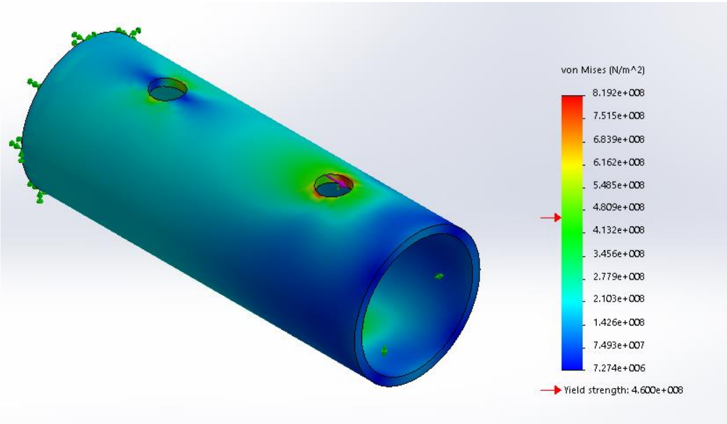


Figure 8: Sleeve Joint at 30KN



Ultimately, this creates an interesting predicament. There is increased complexity, weight, and manufacturing cost being added to the structure. The titanium, for its weight and high strength looks the most desirable, at least for this test. For manufacturing, having materials of different outer diameters can cause mitering issues. There is also an increase in cost for materials (magnesium and titanium), or post-processing (aluminum). Additionally, the university design center is not equipped to handle aluminum welding. Ultimately, the manufacturing inconvenience and increased cost outweigh any benefit gained by finite element analysis.

As previously mentioned, no considerations are made for using an alloy steel versus a mild steel, when computing equivalence. Alternate Frame Rules allow the superior properties to be considered. 4130-N has a yield strength of 63.1 ksi (434 MPa), and ultimate strength of 97.2 ksi (668 MPa). 4130-N Will be used as the material for all further analysis.

Composites

Similarly, Composite Monocoque's are also allowed. A popular trend is to create a partial monocoque, with bonded panels taking the place of the side impact structure. Composites may also be used to replace the Front Bulkhead, Bulkhead Supports, and Anti-Intrusion plate. In any case, equivalency to the baseline mild steel must be proved. Much like with the other alternate materials, any benefit gained from using composites is overshadowed by the increased cost and manufacturing work.

Standard Frame Members

As a whole, a frame designed by the rules is designed to be very safe with no analysis required. A standard 1" x 0.049" frame member of AISI 1010 Steel can take 6,400 pounds (28 KN) before yielding. The reason for triangulation along key load paths is to reduce the amount of force per member, to evenly distribute the force among the members, and keep it as tensile/compressive.

DESIGN OVERVIEW AND PLANAR SECTIONS

Design Process Overview

As can be seen, the frame is a complex three dimensional structure. To sample many geometries quickly, the frame has been divided into three isolated sections. Each of these sections will start by being modeled as a planar section. They will then be subjected to appropriate planar loads. The isolated front will be subjected to front impact and front hoop simulations. The isolated side will be subjected to side impact simulation. The isolated rear will be subjected to main hoop, shoulder bar, and tractive system impact simulation. Secondly, the parts will be brought back up to 3D, and finer details will be ironed out. After finding optimal isolated geometries, they will be knit together, and refinements made to create the final frame design. This design will be analyzed for final rules and FEA compliance.

As a guidance to teams, SAE has a document on their website showing structural configurations that are approved and rules compliant. This document can be seen in Appendix 2. While most of the designs presented within are less than ideal, some are simpler, and therefore possibly more desirable.

Manufacturing considerations are required in any engineering application. In ordering the 2016 frame, $500 was spent on material, and $2400 on set-up and mitering of 106 pieces. This total included 73 for the main envelope of the frame, and the rest for suspension, steering and powertrain. This is why the goal the optimization goal is to reduce the number of frame members.

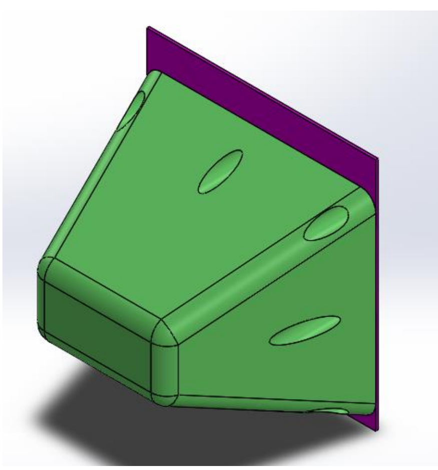
Considerations also need made for assembly. For suspension, there is a trend to have the front higher than the rear. This shifts the weight balance rearwards. For aerodynamics, having a flat bottom is preferable. The 2016 frame has the front and rear suspension boxes parallel to each other and the ground, separated by 2”. For ease of manufacturing, a flat bottom is desirable, and will be designed in. In running an alternate frame, it is permissible to have significantly tilted roll hoops, but we will not exercise this right. This adds assembly complexity, and will also incur additional shipping expenses. The full details of assembly will be discussed later.

Design – Isolated Front

The front of a Formula SAE car serves three main functions. The first is to connect the front roll hoop and front bulkhead. The second is to contain the front suspension points. The third is to hold the drivers legs and pedal box within.

The baseline material for the Front Bulkhead is 0.065” wall, round. A popular alternative is to make it out of 1” x 0.049” Square. This creates a flat surface for mounting the front crash structure to (T3.18). The crash structure and Anti-Intrusion plate are governed separately, and are outside the scope of this report.

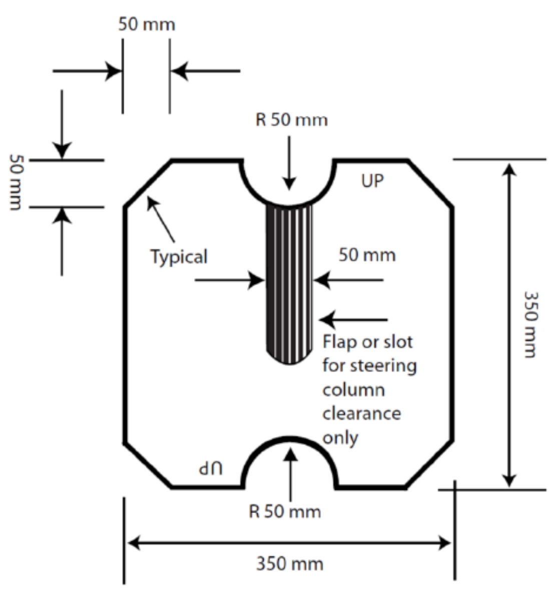
Figure 9: Standard front crash structure, and Anti-Intrusion plate.



The baseline material for the front hoop is 0.095" wall, and the hoop must extend from the bottom of the frame, up, over and back down. (T3.12). The Front Bulkhead must be connected back to the main hoop by three 0.049" wall members per side, an upper, lower, and a diagonal (T3.19). The Front Hoop must have bracing, one tube per side, 0.065" wall (T3.14).

Regarding holding the drivers legs, the following template must be met (T4.2). It must extend from the front hoop, until 4" from the pedals. The 2016 car has a 5" deep pedal box.

Figure 10: Internal Cross Section Template



In considering these first two requirements, a multitude of design options become available. The first specifies the shape, and the second the size. It is fully permissible to have a single member be the upper bulkhead support and the front hoop brace. The lower member that defines the floor will serve as the lower bulkhead support. The question is then, how to efficiently add the minimum number of tubes to accommodate suspension points and create good triangulation.

FSAE is very much a sport of trends. One team succeeds with it, and ten teams will copy it. This can be seen with composite side-impact structures, suspension design, and aero design. One very prevalent trend has what I consider to be a very inefficient design. It can be seen as the baseline front below, and on the 2016 frame in Figure 1, above. The fabrication of it is difficult and expensive, and the benefits appear to be minimal. At the absolute, the entire weight of the vehicle is channeled through the member. 800 lbs across a single 1” x 0.049” Tube of 1010 Steel gives a strain of 0.00018. Over a 10 inch tube, that gives a deflection of 0.0018”. Regarding stress, this has a safety factor of 8 against yielding.

The boundary vonditions were the base of the main hoop, restricted from translating, but not rotating, while the lower edge of the bulkhead was restricted in the y-direction. The first load case is half of the front impact, 30KN at the top and bottom of the Bulkhead. The second load case is half of the front hoop in rollover, with 4.5KN down, and 3KN backwards.

Figure 11: Baseline Front Design

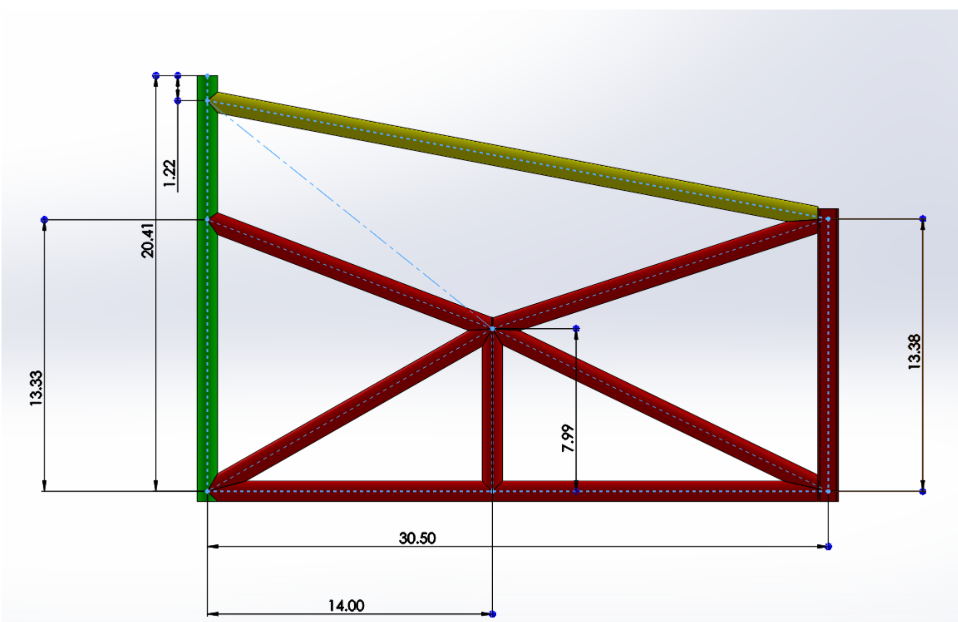


Figure 12: Front Impact

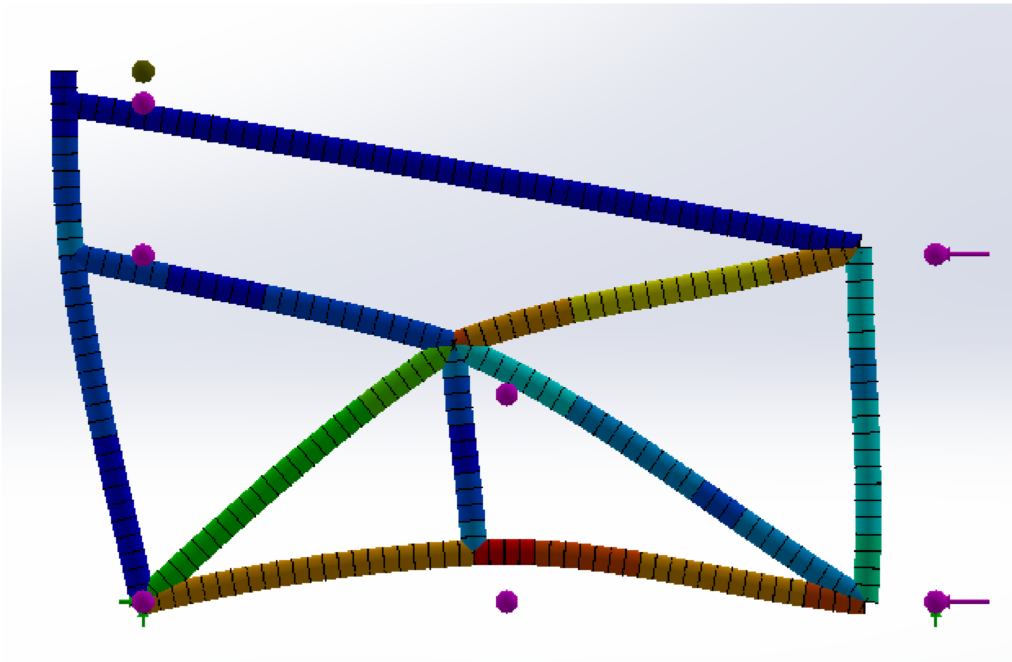
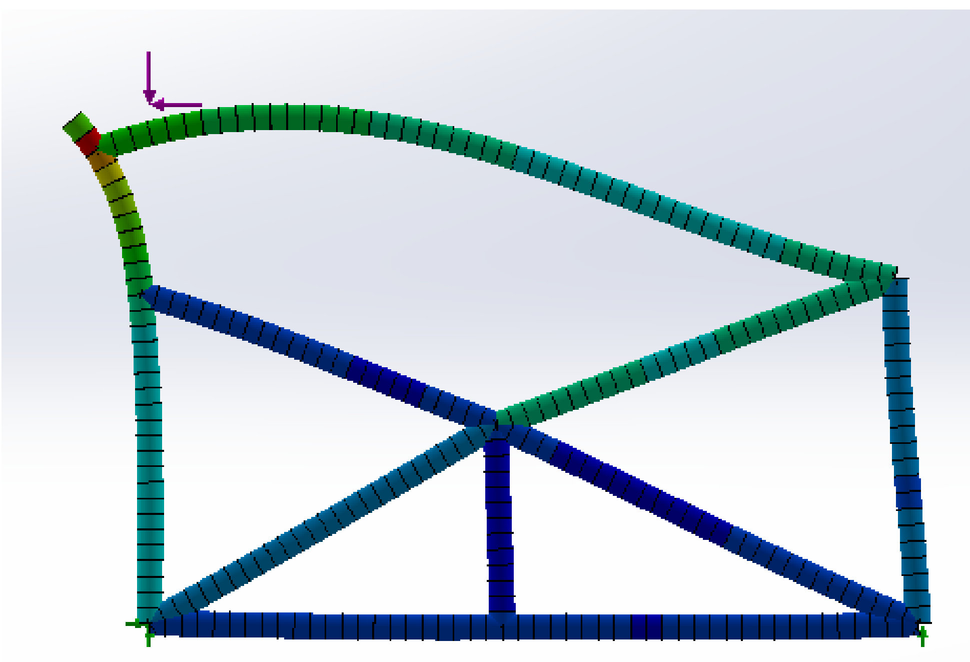


Figure 13: Rollover- Front Hoop

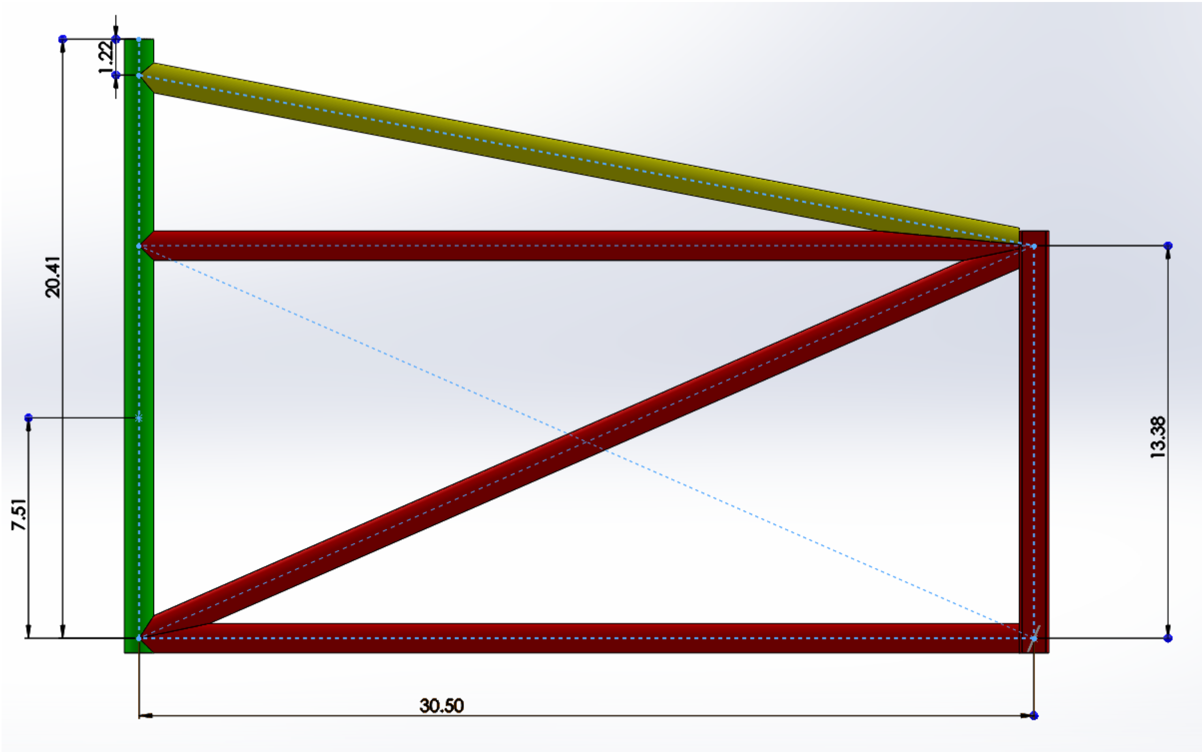


Nine Alternatives were then tested, and they can be seen in the Appendix. The results can be seen below, in Table 4. Yield strength of the material is 434 MPa, and the ultimate strength is 668 MPa.

Table 4: Front Iterations and Alternatives

		Baseline		No Bracing		Alternate 1		Alternate 2		Alternate 2B
Front Hoop	mm	0.359		14.66		0.4625		0.652		0.3759
	N/m2	1.28E+08		5.96E+08		1.28E+08		1.27E+08		1.28E+08
Front Bulkhead	mm	1.61		141		2.881		4.932		1.316
	N/m2	4.7E+08		6.09E+09		6.13E+08		7.77E+08		3.60E+08
Physical Properties	Weight	8.5		5.2		7.43		8.06		8.05
	Tubes	9		4		8		6		6
		Alternate 3		Alternate 3B		Alternate 3C		Alternate 3D		Alternate 4
Front Hoop	mm	0.4717		0.4717		0.4425		0.4425		0.3958
	N/m2	1.34E+08		1.35E+08		1.35E+08		1.35E+08		1.28E+08
Front Bulkhead	mm	4.897		4.897		4.342		4.432		1.619
	N/m2	7.54E+08		7.54E+08		5.84E+08		5.84E+08		4.33E+08
Physical Properties	Weight	7.08		7.04		7.41		7.41		8.21
	Tubes	5		5		5		5		8

Figure 14: Alternate Front 2B, the preferred front design.



Design: - Isolated Side

The sides of the car serve at least two functions. The first is to contain the driver, and the second is to contain the side impact. Depending on the packaging, there may also be vital components in the side-pods.

By rule, there are two templates that must be met for the drivers cell. The first template is the cockpit opening template. It must be met until the bottom of the side impact members. The second is Percy, the 95th percentile man. He must be able to be fitted into the seat, with his head 2” below a line connecting the two roll hoops.

Figure 15: The 95th Percentile Man template, Percy

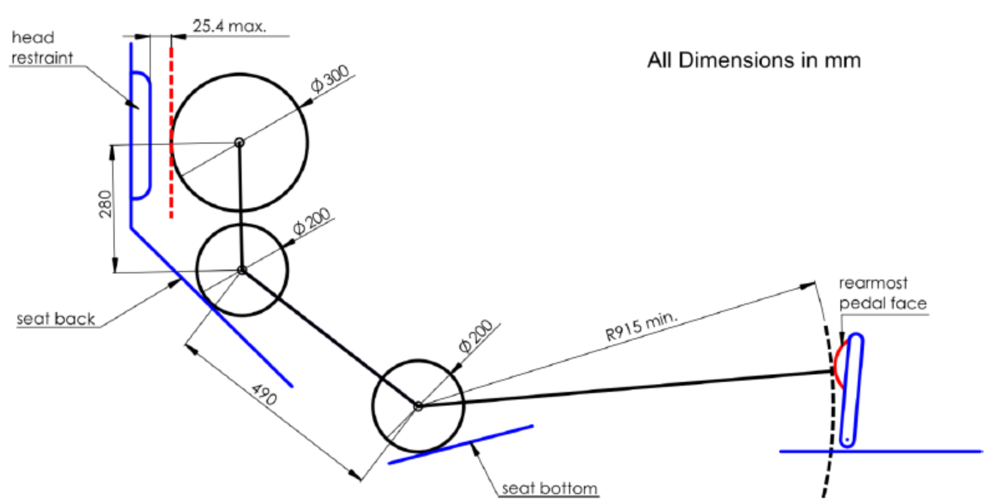


Figure 16: Cockpit Opening Template.

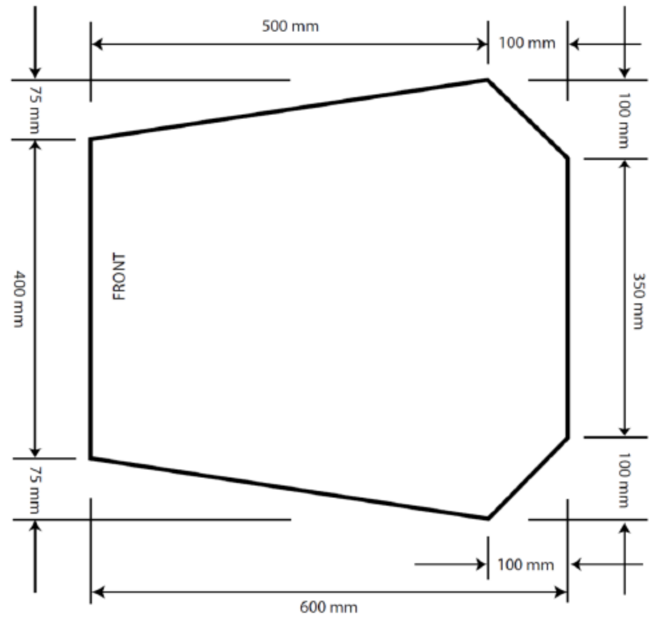


FIGURE 8

The side Impact structure consists of three members. There has to be an upper and lower member, and a properly triangulated connector. The entirety of the upper member needs to be in the "side impact zone", 11.8" to 13.8" (300mm to 350mm) above the ground. As such, the lower member of the cockpit can be considered the lower side impact member. However, a dedicated upper and diagonal will be needed.

The baseline design is shown in figure 17 below, and shown deformed in figures 18 and 19 below. The load is 7KN, and it is applied at the top, and then bottom outboard nodes. The boundary conditions are the base of the hoop segment is fixed, while the bottom node is restricted in the Y-Direction.

Figure 17: Baseline Side Design

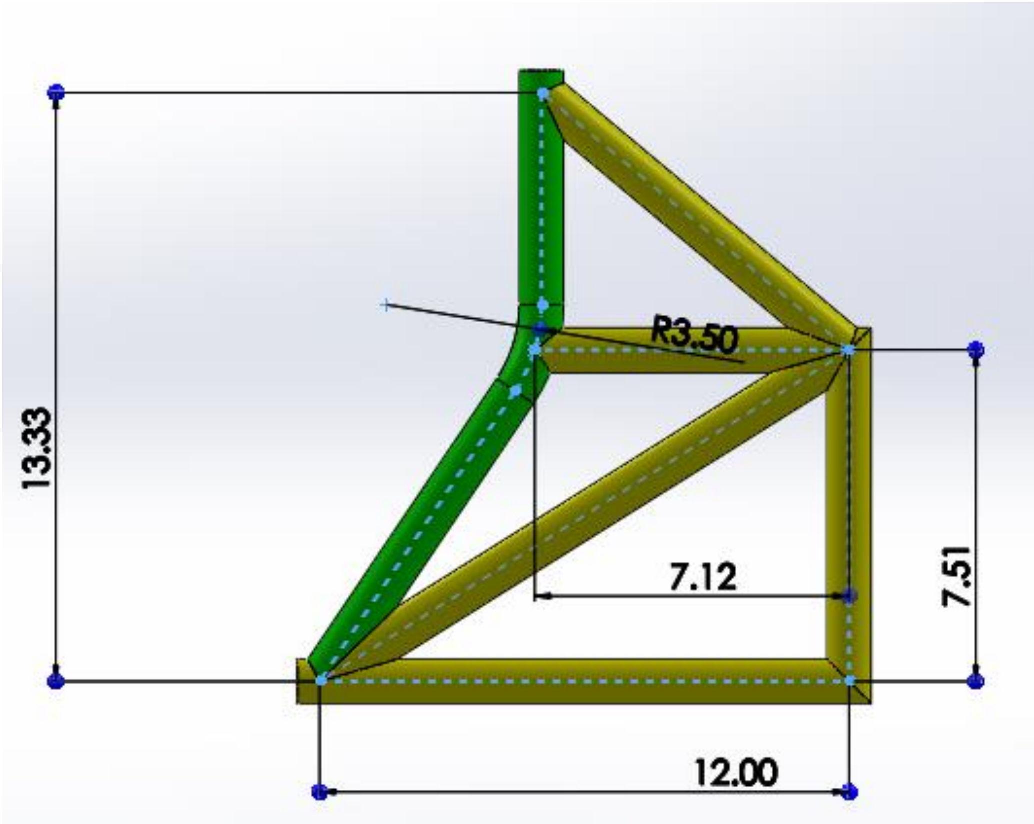


Figure 18: Side Impact, Upper member

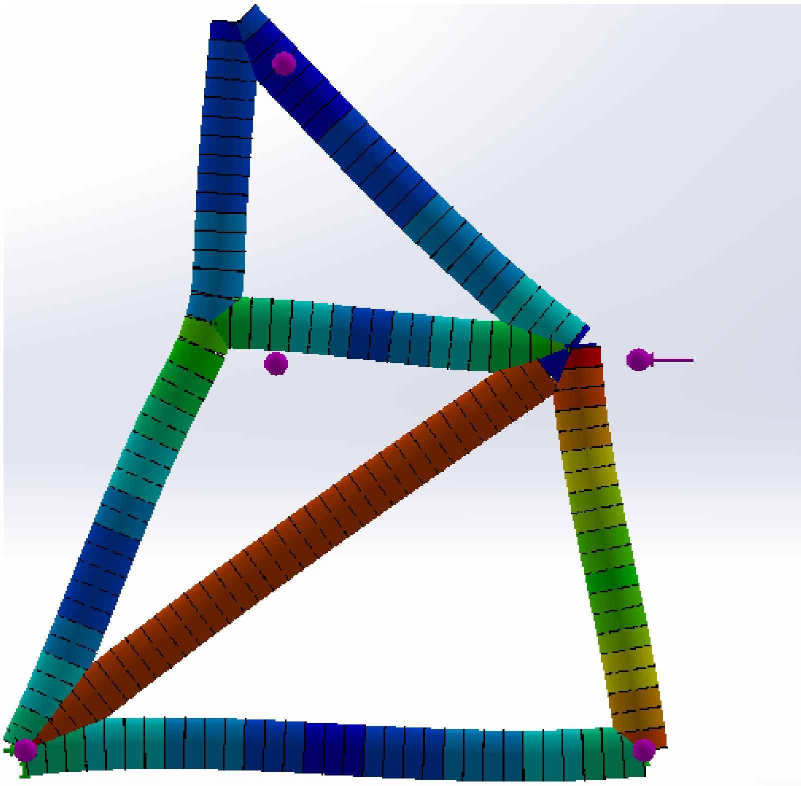
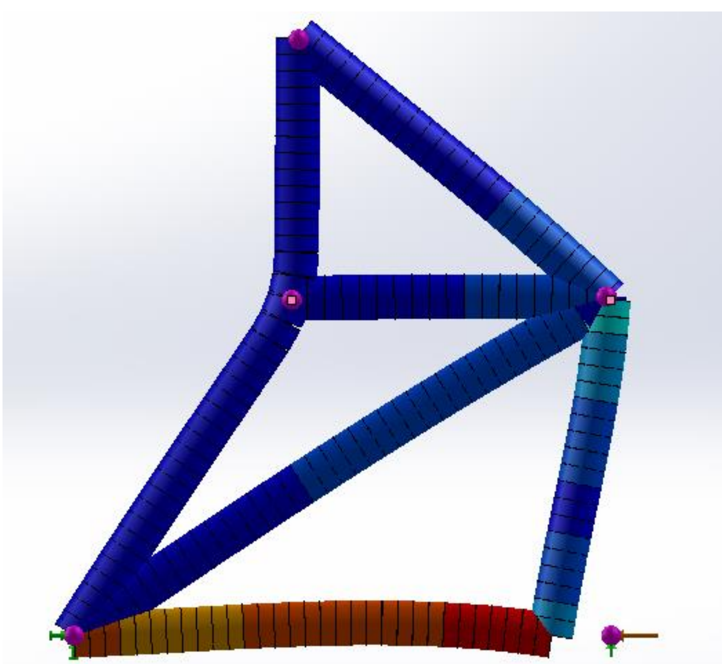


Figure 19: Side Impact, Lower Member



Having obtained a baseline, two alternatives were tested. They can be seen in the Appendix. The results can be seen in Table 5, below. Yield strength of the material is 434 MPa, and the ultimate strength is 668 MPa.

Table 5: Side Itterations and Alternatives

		Baseline	Alternate 1	Alternate 2
Upper Impact	mm	0.1589	0.1549	0.1611
	N/m2	6.75E+07	7.43E+07	6.02E+07
Lower Impact	mm	0.08349	N/A	0.08297
	N/m2	6.63E+07	N/A	6.89E+07
Physical Properties	Weight	3.67	2.67	4.47
	Tubes	6	4	6

Unlike with the front and rear sections, a superior design cannot be directly picked, due to packaging and other considerations. One design, impossible to properly model in side view, is to have the side impact entirely between the roll hoops. This design can then be further iterated to a bonded composite panel.

Design: - Isolated Rear

The Rear of the frame must comply to the rules, and the needs of the team. By rule, the Main Hoop (0.095”) must be supported by 2 Main Hoop Braces (0.065”). In turn, these braces must be supported (0.049”) back to the upper and lower side impact member. For electrical vehicles only, the tractive system must be protected (0.049”) from side and rear impacts. Additionally, the shoulder harness bar must be mounted, such that it creates a +10/-20 Degree angle with the driver’s shoulders.

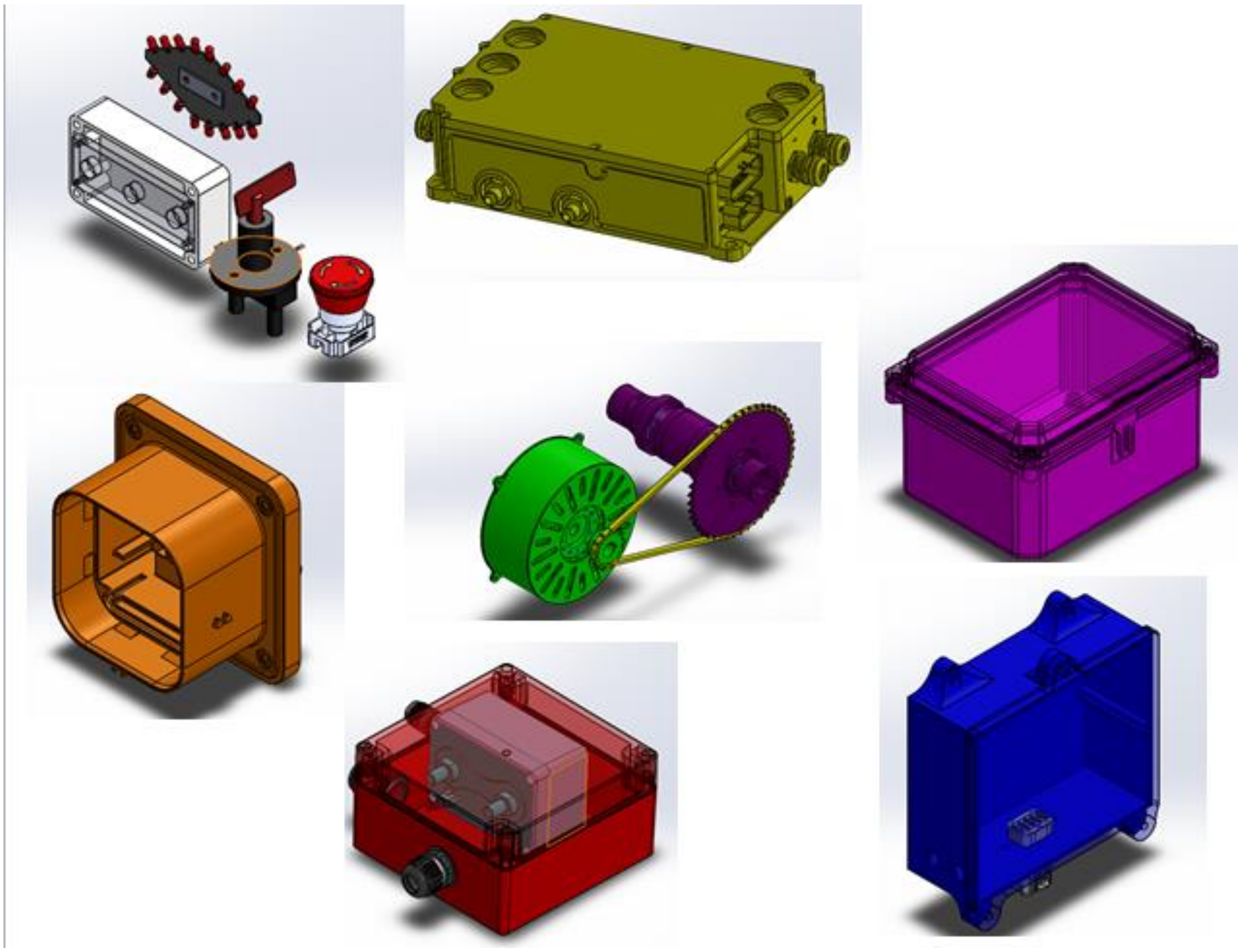
Packaging

Unlike the front, the rear of the car is dictated by the major components in the frame. Major components may also be stored in the sides, either alongside or beneath the driver. The low voltage electrical items are capable of being located anywhere in the frame. The high voltage components, connected with 3/8" wire, should be located near each other. The drivetrain components are similarly restricted. The differential must be centered, and the sprockets on the motor and differential must be aligned.

Table 6: Internal Components

Component	**Class.**	**Color**	**2016 Location**	**Size**	**Weight**
Backplane Box (ECU)	LV	Pink	Left Sidepod	6" x 8" x 5"	2 lbs
Low Voltage Batteries	LV	Navy	Right Sidepod	7" x 5.75" x 2.75"	5 lbs
Accumulator (High Voltage Batteries)	HV	Sky	Behind Driver, Left Sidepod	<See Note>	100 lbs
Motor Controller	HV	Yellow	Behind Driver	8" x 12" x 3.5"	30 lbs
Motor	HV Drive	Green	Centered, Rear	10" OD x 4" Long	30 lbs
Differential	Drive	Purple	Centered, Rear	4" OD, 13" Long (10" OD Sprocket)	13lbs
Exterior panel mounted components.					
High Voltage Disconnect	HV	Orange	Panel, Rear	4.25 x 4.25 x 2.5"	1 lb
E-Meter	HV	Red	Panel, Rear	6.25" x 6.25" x 3.5"	2 lbs
TSMP	LV	White	Panel, Rear	6" x 1" x 2"	Negligible
Power Switches (x2)	LV	Black and Red	Panel, Rear	3" OD, 3" Long	Negligible
E-Stop (x2)	LV	White	Main Hoop (2 Sides)	1" OD x 2" Long	Negligible
TSAL	LV	Grey and Red	Main Hoop	6" x 1.5 x .25"	Negligible

Figure 20: Internal Components

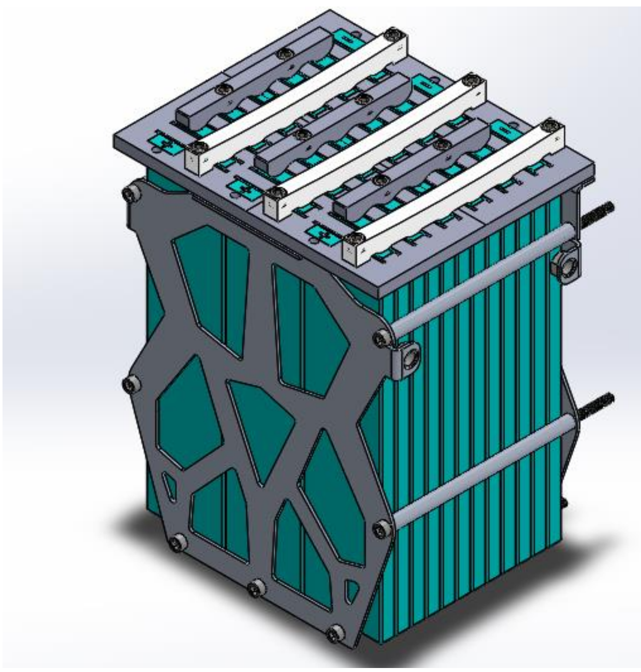


The largest and heaviest of these components is the accumulator. For safety and to alleviate congestion at competition, the accumulator must be removed from the car before charging. As such, accumulator location dictates the rest of the packaging.

The 2016 car uses 216 cells in a 72 series/ 3 parallel arrangement. The accumulator is a large, “J” shape that wraps around the drivers left hip. The accumulator is located with the majority of the box beneath the drivers back. This configuration requires disassembly of the cockpit for charging.

Within the 2016 accumulator, there are 6 segments, each containing 36 cells (12S/3P) at 50 Volts. For design of packaging, these 6 segments will be considered, with a 7th "segment" being allocated for electrical components. Each segment equivalence is 6.125" x 4.25" x 8.25"

Figure 21: Actual Accumulator Segment

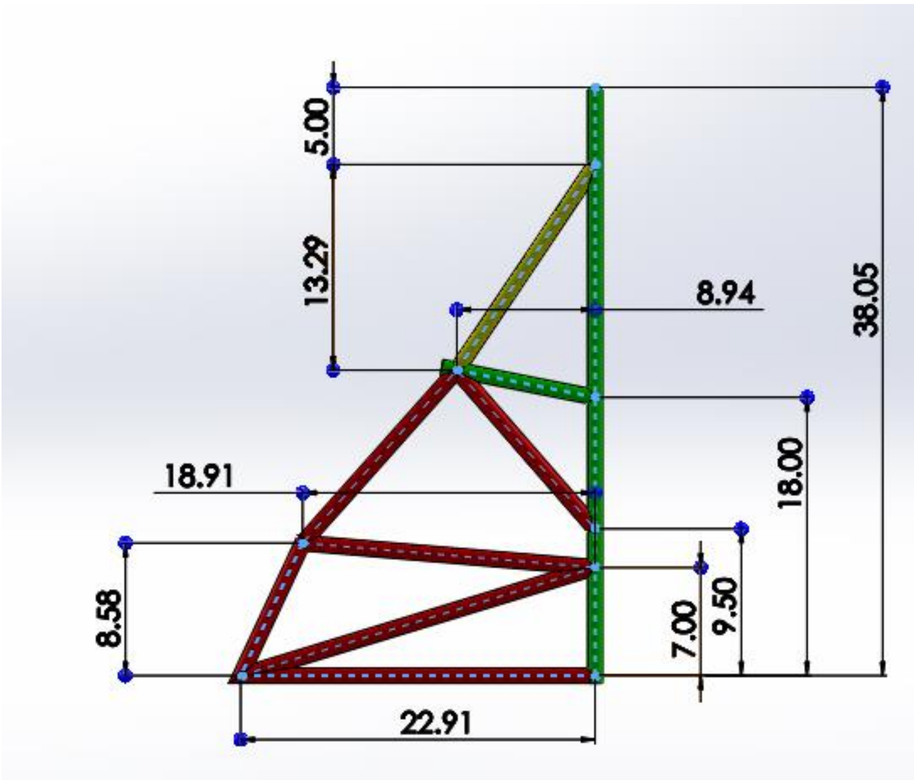


Rear Iterations

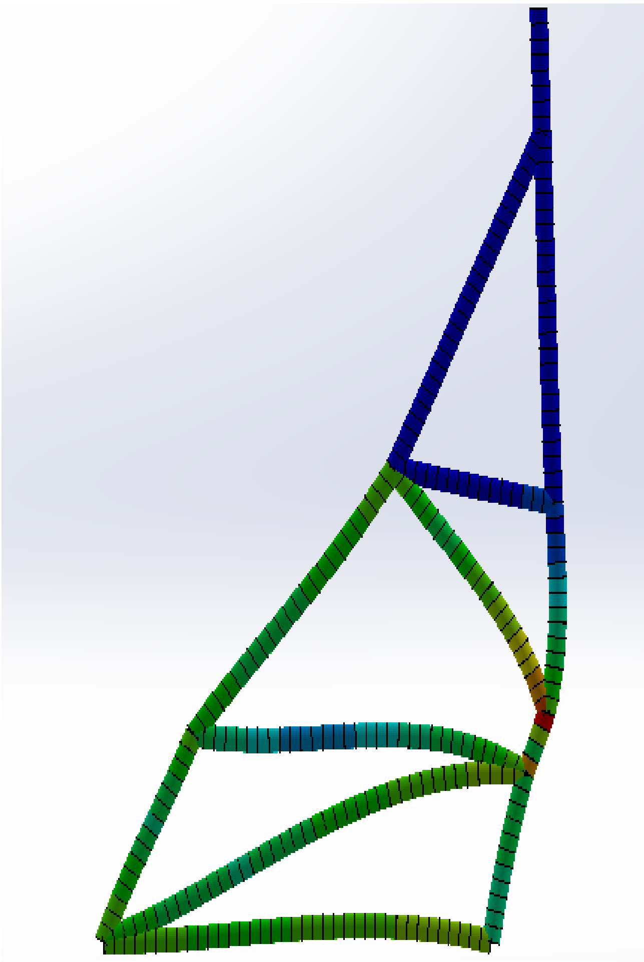
For initial designing, we will not directly consider the need for packaging. We will assume that all components will fit, and concern ourselves only with the relative geometry. This assumption will be validated later

For the finite element analysis, the base of the hoop will be constrained against translating, and the trailing edge will be constrained against translation in the Y-direction. The first load case is 7KN, horizontal from the shoulder harness bar. The second load case is half of the front hoop in rollover, with 4.5KN down, and 3KN backwards. The final case is 2.5 KN, for a rear impact.

Figure 22: Baseline Design



Figures 23 and 24: Shoulder Bar loading, and Rollover Loading



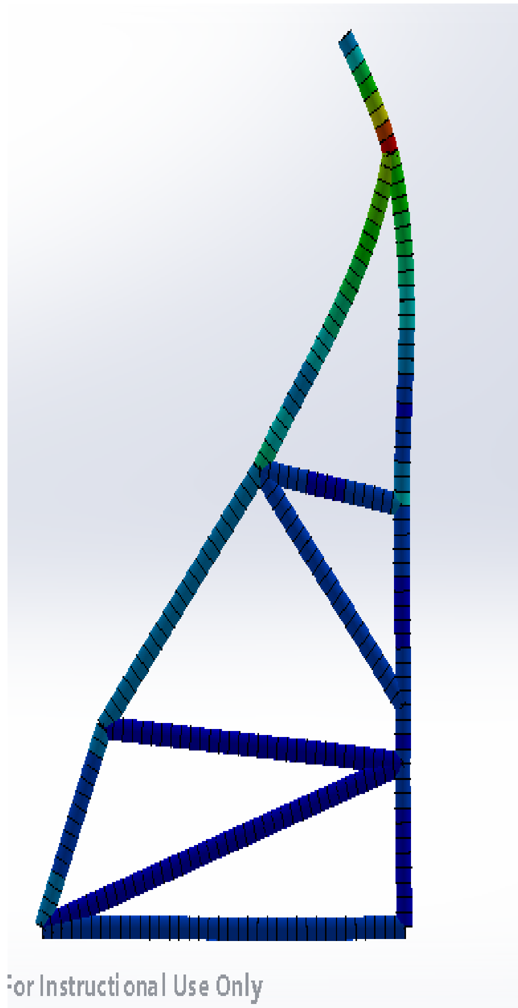
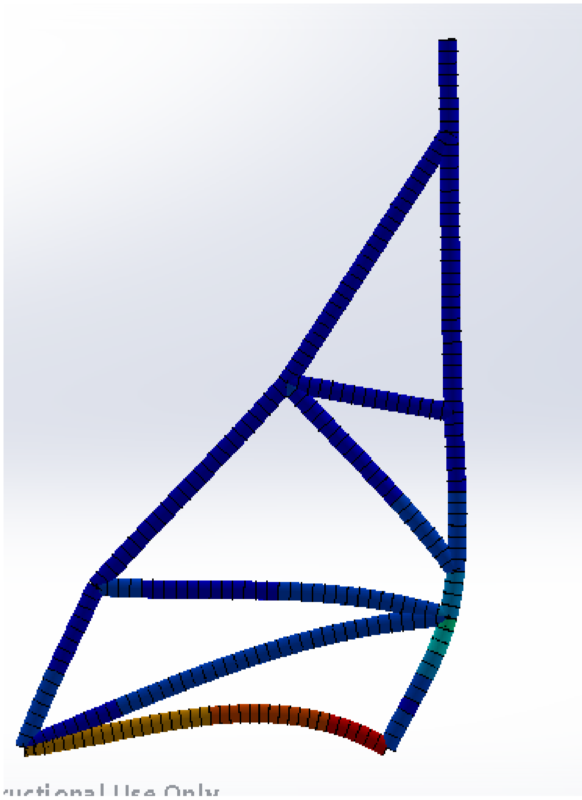


Figure 25: Rear Impact

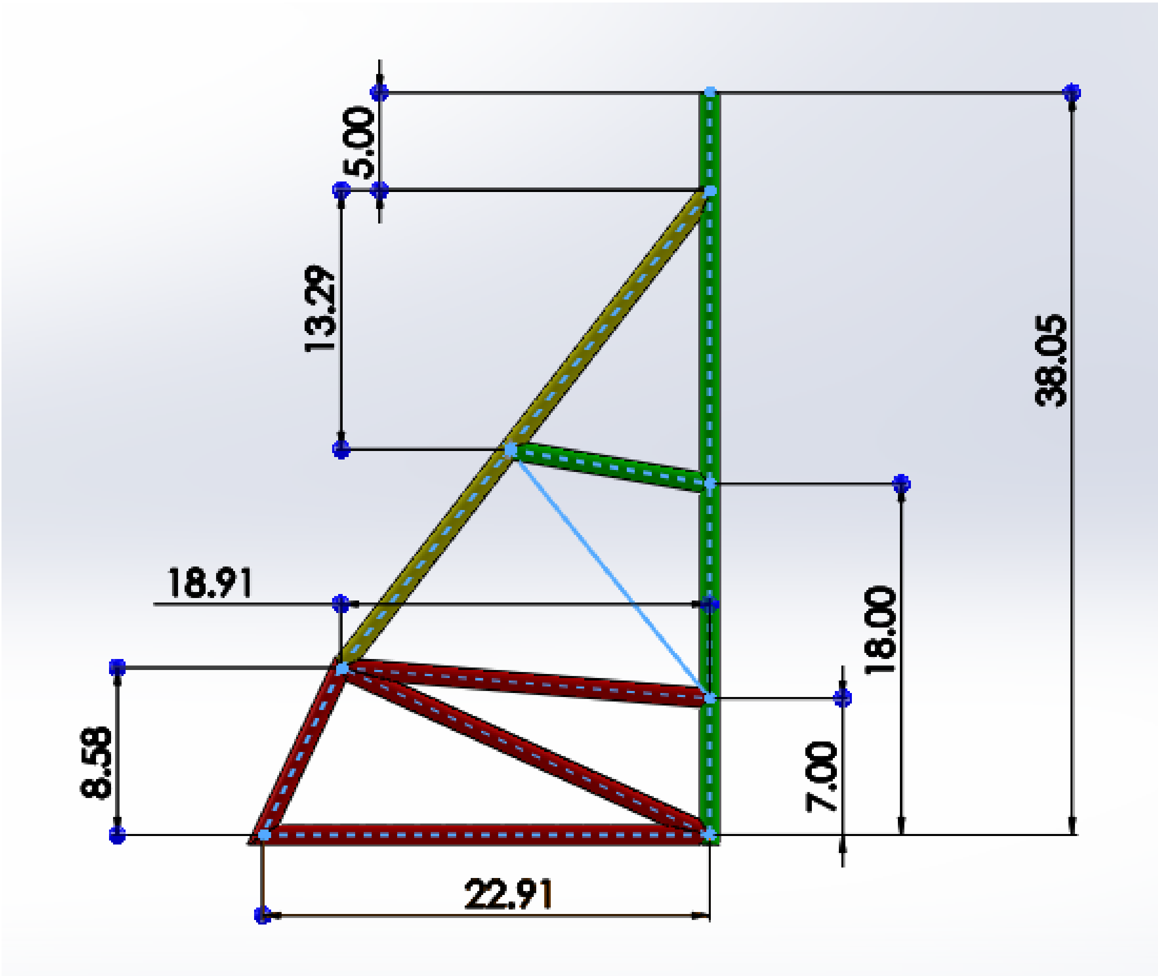


Having obtained a baseline for comparison, 12 alternatives were tested. The alternate designs can be seen in the Appendix. The results can be seen in Table 7. Yield strength of the material is 434 MPa, and the ultimate strength is 668 MPa.

Table 7: Rear Alternates and Iterations

	Baseline		Alternate 1		Alternate 1B		Alternate 1C	
Shoulder Bar	mm	0.7509		0.5562		0.5672		0.6005
	N/m2	1.49E+08		1.08E+08		1.06E+08		1.04E+08
Main Hoop	mm	3.053		3.272		3.228		3.048
	N/m2	4.42E+08		4.42E+08		4.42E+08		4.42E+08
Front Bulkhead	mm	0.081		0.07847		0.0769		0.0778
	N/m2	3.23E+07		3.24E+07		3.16E+07		3.16E+07
Physical Properties	Weight	8.60		8.62		8.60		8.64
	Tubes	9		9		9		9
	Alternate 2		Alternate 2B		Alternate 2C		Alternate 2D	
Shoulder Bar	mm	11.88		0.451		0.4284		9.652
	N/m2	9.46E+08		8.08E+07		8.08E+07		7.30E+08
Main Hoop	mm	3.674		3.264		3.028		3.319
	N/m2	4.42E+08		4.42E+08		4.42E+08		4.42E+08
Front Bulkhead	mm	0.0743		0.07432		0.0743		0.07435
	N/m2	3.02E+07		3.03E+07		3.03E+07		3.02E+07
Physical Properties	Weight	7.73		8.22		8.35		8.40
	Tubes	6		8		7		6
	Alternate 2E		Alternate 3		Alternate 3B		Alternate 3C	
Shoulder Bar	mm	3.616		3.65		10.44		7.927
	N/m2	4.94E+08		4.87E+08		8.50E+08		7.47E+08
Main Hoop	mm	3.09		12.75		13.45		3.673
	N/m2	4.42E+08		4.42E+08		4.42E+08		4.42E+08
Front Bulkhead	mm	0.07432		0.3449		0.3848		0.07437
	N/m2	3.03E+07		3.03E+07		3.18E+07		3.02E+07
Physical Properties	Weight	8.19		9.41		8.35		7.73
	Tubes	7		10		8		0

Figure 26: Alt Rear 2E, the best Alternative.



3D TESTING

Having determined the ideal 2D geometries, the next step is to bring it back up to 3D. To test local geometries, without the hassle of mitering and working on the full model, three simplified “sleds” were made. Front and rear sleds each contains half of the designed car, and half of a stand in. The side sled contains both roll hoops, in addition to the sides. This allows the final boundary conditions to be used, and to see to what extent loads are being transferred from one roll hoop to the other.

This transition will require assigning widths to different areas. In the front, this will be at the Bulkhead’s top and bottom, and Front Hoop’s base, first bend, and second bend. In the rear, this will be at the Bulkhead’s top and bottom, and the Main Hoop’s base, first bend and second bend. The selected values will be discussed in appropriate sections.

Additionally, there is the length of the car. By rule T2.3, the wheelbase must be at least 60”, and the 2016 car has a wheelbase of 61.5”. To achieve this, a hoop-to-hoop distance is set at 33”.

3D Front

To transition the front to 3D, widths had to be selected for the components. 13” was selected for the front bulkhead bottom. This is the same as on the current car, to allow for reuse of the front crash structure. The same dimension will be chosen for the bulkhead top and base of front hoop. 15” Inches was chosen for the width at each of the bends. To keep consistent with the 2016 car, the lower bend height was set at 7.51”. This can be seen by the marked point in in Figures 11 and 14.

Combining all of these factors, we obtain the isolated 3D front. This can be seen in Figure 27. The boundary conditions and load conditions were set as those for the full analysis. (See Table 1). The deformations can be seen in Figures 28 and 29, and the results in table 8.

Figure 27: Isolated 3D Front.

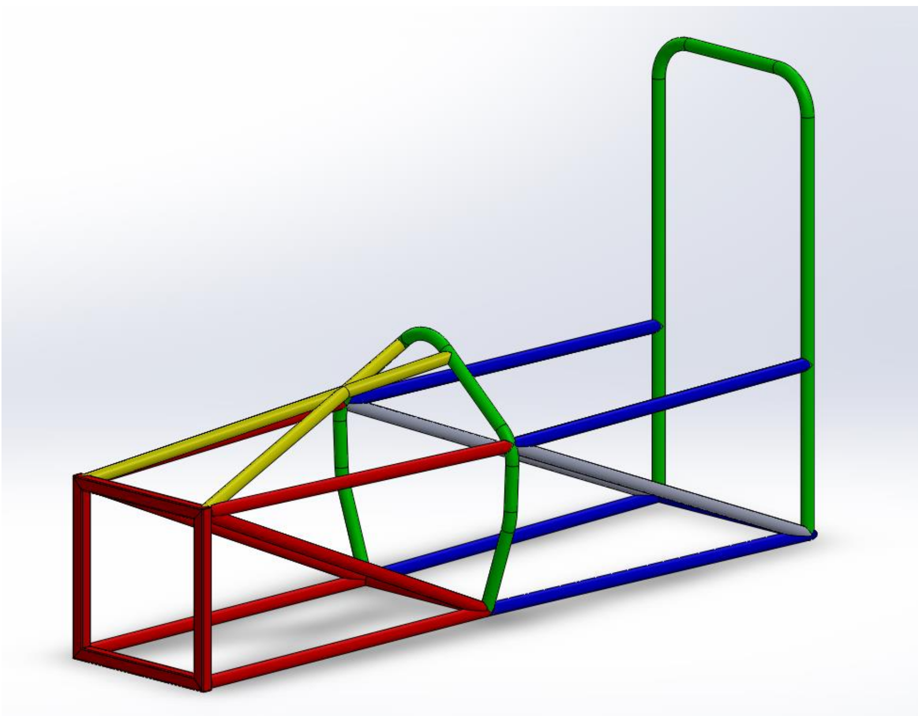


Figure 28: Front Impact

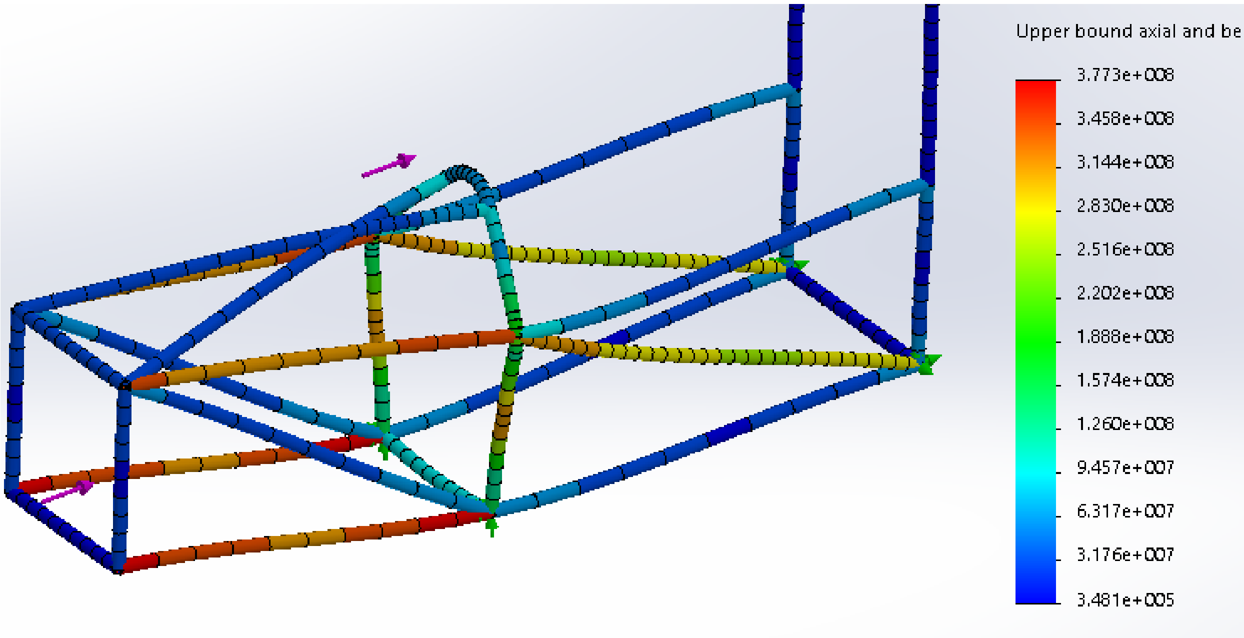


Figure 29: Front Rollover

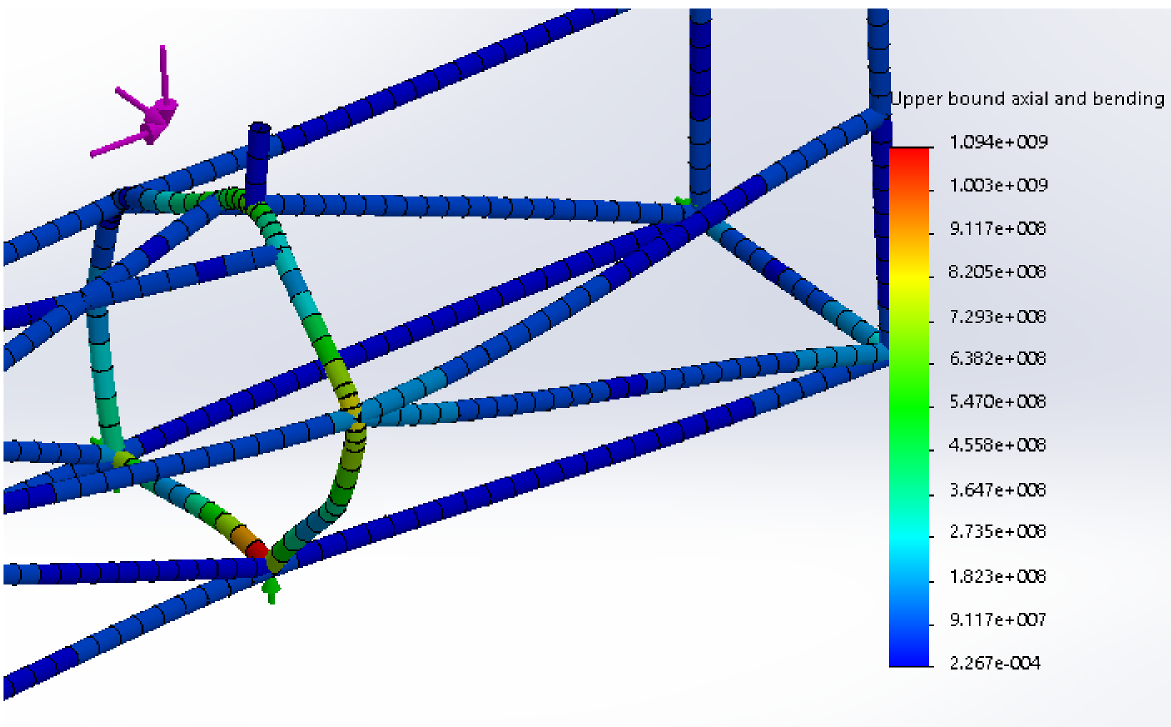


Table 8:

BASELINE GEOMETRY				
Baseline	Angle 1	Angle 2	Stress	Deflection
Front	10.79	23.68	3.773E+08	6.257
Hoop	10.79	23.68	1.09E+09	14.5

The stress increase in the front hoop load case is worrying, but will be temporarily accepted. When the full side-impact structure is added in, the torsional stiffness should increase. I also suspect an error in the Solidworks simulation.

Iterations were run on the structure, altering each of the 4 dimensions controlling the underlying sketch by a small amount each way, to see how it reacted. Seeing as how the front hoop geometry is mostly left as-is, only front impact was considered.

Figure 14 (Repeated): Front 2B

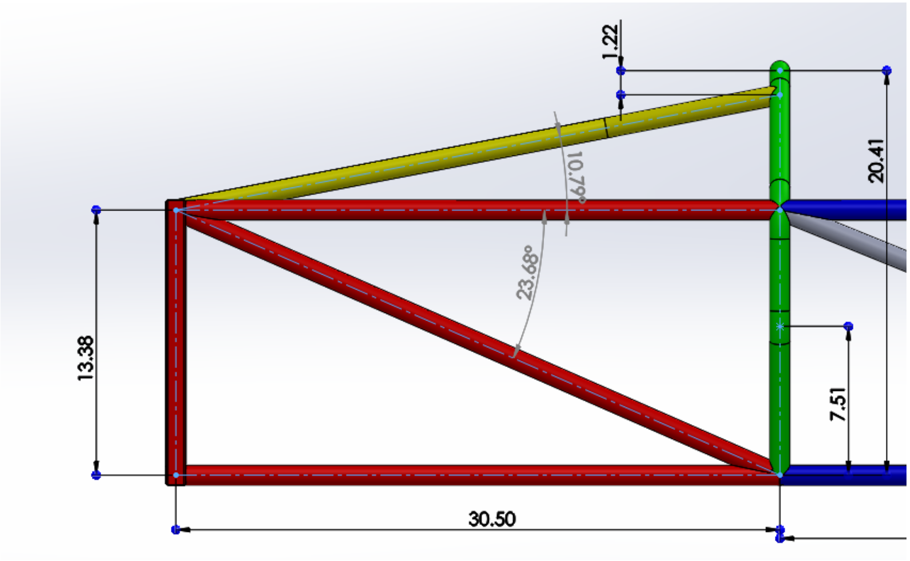


Table 9: Results of Micro-Iterations.

	Angle 1	Angle 2	Stress	Deflection
Baseline	10.79	23.68	3.773E+08	6.257
DIMENSION 1: HEIGHT OF FRONT HOOP				
+ 0.5"	11.7	23.68	3.774E+08	6.233
- 0.5"	9.89	23.68	3.782E+08	6.196
DIMENSION 2: FRONT HOOP TO BRACE				
+ 0.25"	10.34	23.68	3.788E+08	6.225
- 0.25"	11.25	23.68	3.785E+08	6.244
DIMENSION 3: HOOP TO BULKHEAD				
+ 0.5"	10.62	23.34	3.773E+08	6.341
- 0.5"	10.97	24.03	3.766E+08	6.230
DIMENSION 4: HEIGHT OF BULKHEAD				
+ 0.5"	9.89	24.46	4.321E+08	6.001
- 0.5"	11.7	22.89	4.111E+08	6.608

It is interesting to note that while most of the iterations had a minimal effect, the adjustment of the height of the Front Bulkhead had a substantial negative effect. I theorize that there must be a parabolic or quadratic relationship between bulkhead height and maximum stress.

<u>Roll Hoop Bracing</u>

As previously stated, FSAE cars are subject to trends. One of them is with the front bracing. The old design is have two straight rails, from bulkhead to hoop. The current trend is to have the braces cross, near the steering column. This allows the semi-circle on the template to be utilized when clearing the steering column. This adds build complexity and weight. It also makes any overhead pushrod suspension impossible. Due to a desire to have a simple suspension, and to have a simple frame, the old design will be evaluated.

Figure 30: Front with Alternate Bracing

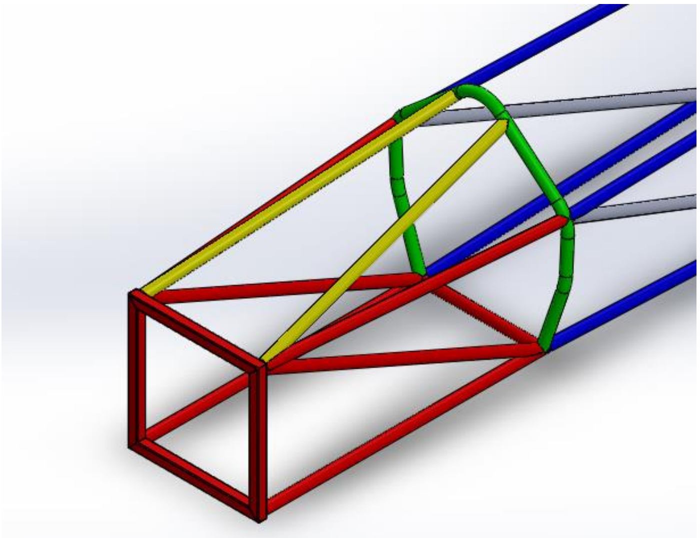


Table 10: Baseline vs. Straight Bracing, for Front Impact.

	Stress	Deflection
Baseline	3.773E+08	6.257
Straight	3.770E+08	6.236

As we can see, the two designs are nearly identical. However, the simplicity, weight, and having one less tube make it ideal.

Front Suspension and Template

In selecting the baseline widths, the decision was made to keep the same sized bulkhead as our current car has. That way, an old crash structure can be used, if desired. That width, used on the old frame works fine, passing template easily. However, when doing the interior template test, the new car fails the test miserably.

In looking at Figure 1, the original design, we can see one of the benefits of the 5-point star. It allows the front suspension node to be pulled out, allowing the template to be satisfied. To do this, we will consider two ways. Both will involve having a suspension point, offset ½" above the lower bend in the front hoop, and 1" out, while still maintaining the same side profile.

The first method is to use two tubes, with a mitered joint, and then a third tube dropping down to the frame to locate the lower point. The second option is to use a single tube, and bend it to create a dogleg, with the bend centered at the suspension point. The same drop-down support will then be used.

Figure 31: Dogleg front Suspension

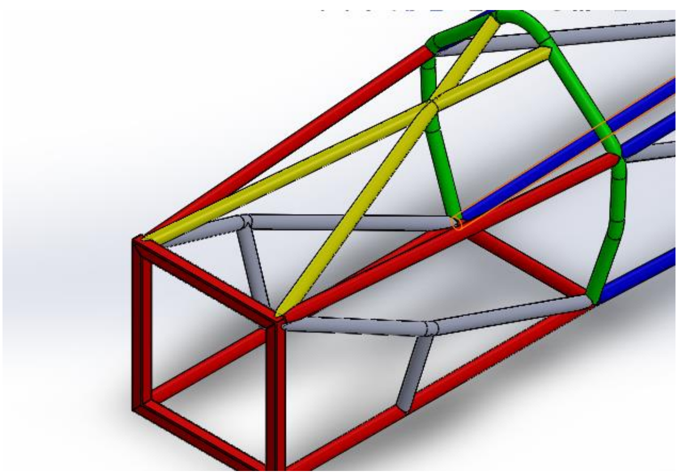


Figure 32: Mitered Front Suspension

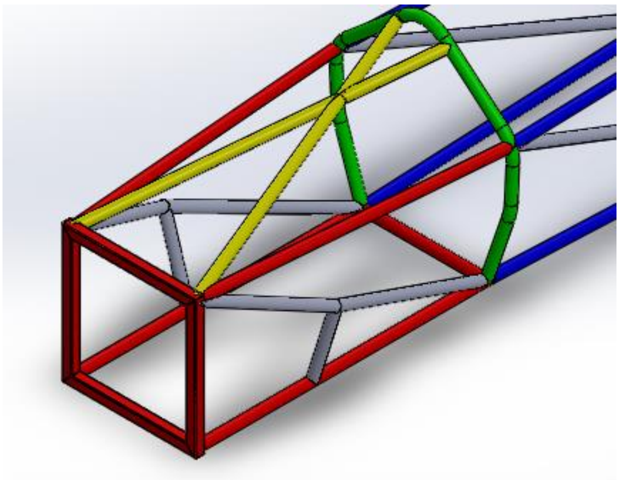


Table 11: Wide Front Suspension Analysis

	Stress	Deflection
Baseline	3.773E+08	6.257
Dogleg	4.017E+08	6.018
Mitered	4.011E+08	6.048

From this analysis, we can see that the two options are equivalent, even though not as strong as the baseline. Because the dogleg is simpler to manufacture, and has less members, it is the preferred design.

Roll Hoops

In transitioning to 3D, one of the things that will change is the roll hoops becoming bent. The radius of the bends, however, is limited by the dies of the tube bender. For our preferred vendor, VR3 Engineering in Stratford, ON the two applicable sizes for our use are 3.233" and 5.625"

The other issue is where to locate the bends. In all designing the planar sections, there was just a point for the location of the bend. In the 3D models, the locations of the bends are specified by a point in space. In creating the bend and its radius, there are three options, those being locating the point at either edge, or the middle of the bend. The top of the roll hoop will naturally be at the middle of the bend. For the rest, the most neutral option is for it to be the midpoint of the bend. Another option is for that to be either the upper or lower edge of the bend. Most, but not all combinations of points are valid. Provided the combination is valid, the sketch will be fully defined, with only one dimension, that being the radius of the bends.

In creating further connected 3D Sketches, Solidworks will automatically make attaching to the midpoint of the bend an option. This can create an undesirable situation, where the node you've selected isn't directly controllable. Placing the nodes at mid-bend helps to pre-empt any problems.

For clarity, a naming scheme has been devised. The words leading, center, and trailing refer to the position of point on the bend. Consider the roll hoop as a curving vector, departing from its base point towards the bend point. If the leading edge of the bend is on the point, it's leading. If the trailing edge of the bend is on the point, it's trailing. The first word refers to the upper bend, and the second to the lower bend. The third bend, at the top of the hoop, is understood to be centered. As an example, a Trailing-Leading configuration can be seen in Figure 33.

Figure 33: Trailing-Leading Alternative Front Roll Hoop

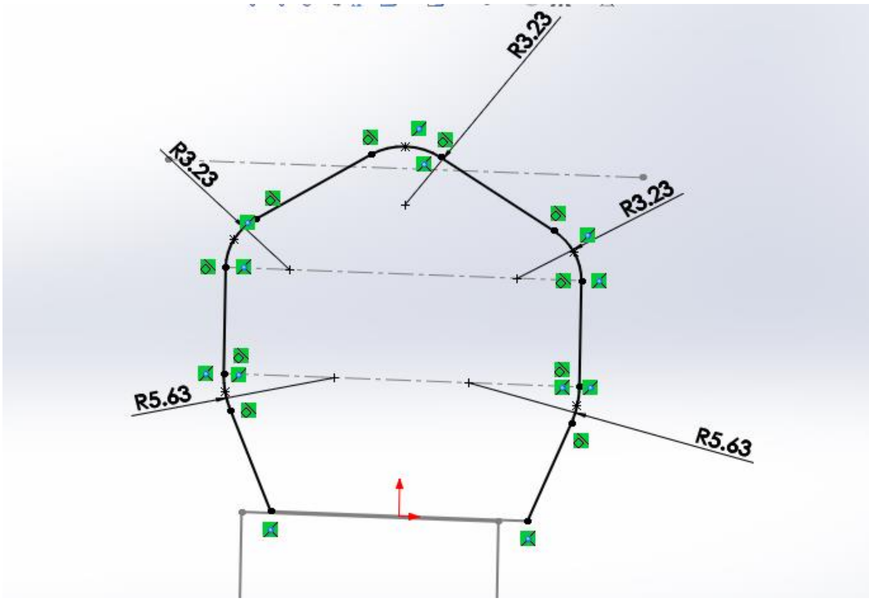


Table 12: Alternative Front Roll Hoop Geometries.

	Stress	Deflection
Baseline	1.094E+09	14.5
Trail - Lead	1.105E+09	14.81
Center - Lead	1.080E+09	14.19
Trail - Center	1.095E+09	15.13

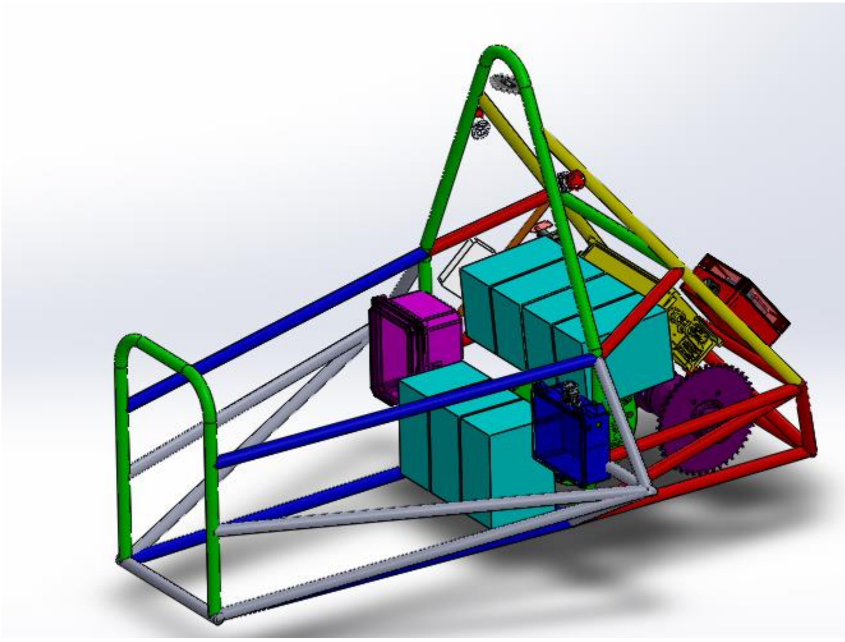
As we can see, there is no substantial difference. To prevent issues with Solidworks, as mentioned above, a Center-Center configuration will be used on all bends. Additionally, in a rollover, forces in the beam may cause the beam to buckle, and possibly un-bend. Using center-center attachments means that any bend will only have half as long of a lever arm, and can only half un-bend, without having to break the weld.

3D Rear

In bringing the rear up to full 3D, values had to be chosen for different widths. The width of the main hoop at its base and the bulkhead at the base were chosen to be the same. At its lowest point, the main hoop requires 15” inside to inside width. Therefore, a 16” centerline width was nominated. The width of the hoop at each of the bends and width of the bulkhead were chosen to all be 25”

The other thing that must be confirmed is the assumption that all components will fit. A re-designed accumulator has been proposed, splitting the seven segments into two pieces. One will fit under the driver’s seat, and the other behind the driver. A few other pieces can be rotated around, to get everything to fit. It’s not a perfect solution, but good enough to allow designing to proceed.

Figure 34: Provisional Packaging



The load conditions were then applied. (See Table 1). The deformations can be seen in Figures 35, 36 and 37, and the results in table 13.

Figure 35: Shoulder Bar

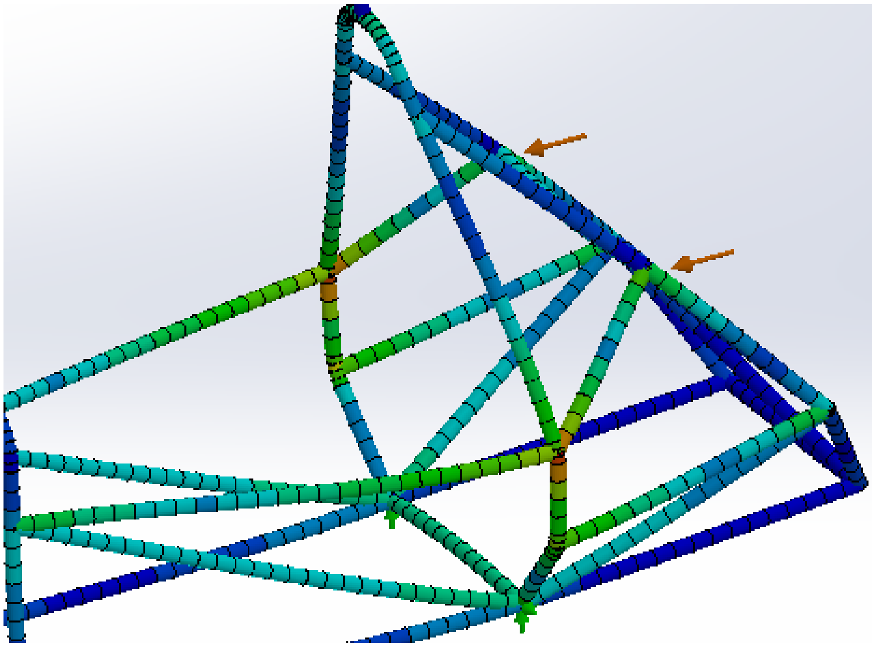


Figure 36: Main Hoop

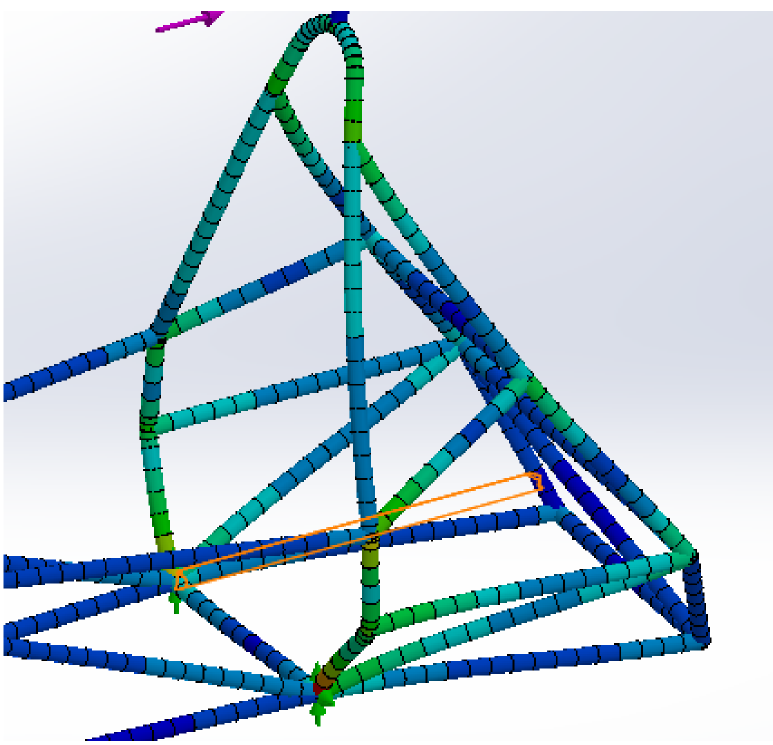


Figure 37: Rear Impact

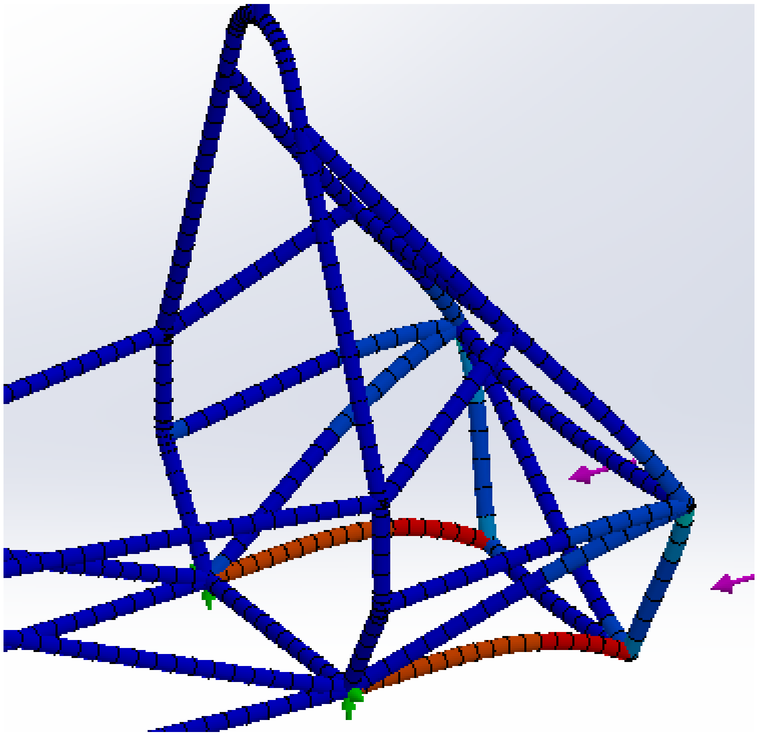


Table 13: 3D Rear Baseline Deformations

	Angle	Stress	Deflection
Shoulder	37.70	4.84E+08	18.22
Hoop	37.70	9.30E+08	23.21
Rear	37.70	2.96E+07	0.08792

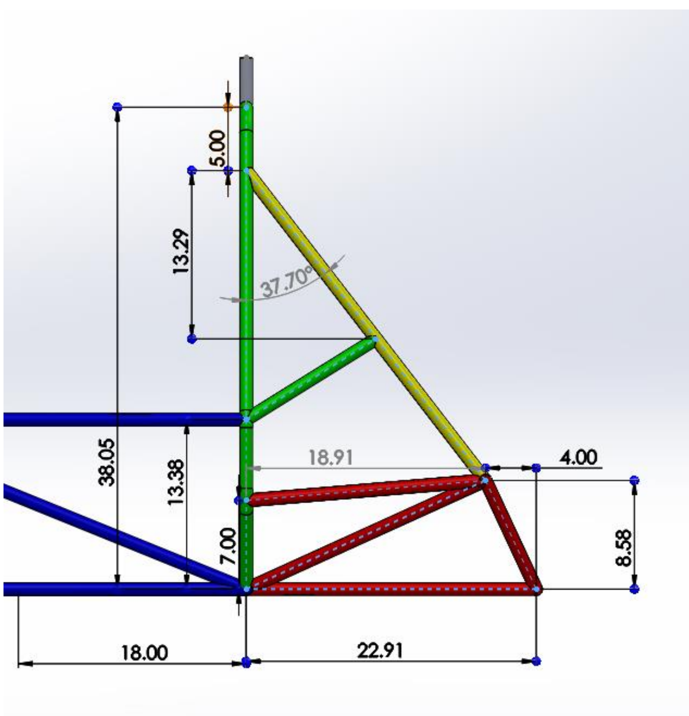
Same as with the front, the stress in the hoop rollover is too high. Once again, increasing the torsional rigidity in the full model should help to resolve this problem. I also suspect an issue with the Solidworks simulation, regarding the roll hoop base nodes.

The next goal was to understand how the frame behaves when dimensions are changed. Therefore, iterations were performed on the dimensions controlling the underlying sketch, to see how it responds to small changes.

Table 14: Rear Micro-Iterations

	Angle	Shldr Str	Shldr Disp	Hoop Str	Hoop Disp
Baseline	37.70	4.84E+08	18.22	9.30E+08	23.21
DIMENSION 1: HEIGHT OF MAIN HOOP					
	Angle	Shldr Str	Shldr Disp	Hoop Str	Hoop Disp
+ 1"	36.59	4.19E+08	18.36	8.67E+08	20.05
- 1"	38.86	3.41E+08	15.39	8.52E+08	18.98
DIMENSION 2: HOOP-BRACING OFFSET					
+ 0.5"	38.27	4.51E+08	17.21	9.06E+08	23.55
- 0.5"	37.14	5.18E+08	19.23	9.16E+08	21.84
DIMENSION 3: REAR BOX LENGTH					
+ 0.5"	38.42	4.84E+08	18.2	9.21E+08	22.67
- 0.5"	36.96	4.85E+08	18.22	8.97E+08	22.57

Figure 26 (Repeated)



Similar to the effect of changing the height of the front Bulkhead, there appear to be non-linear effects in play with how changing dimensions effects stress in the structure. In particular, it’s interesting that both raising and lowering the main hoop will lower the stress. Lowering the hoop seems intuitive, with a shorter lever arm meaning less moment, and therefore less stress. Raising the hoop seems counterintuitive, until you consider the movement of the braces upwards with it, providing more strength.

Shoulder Bar

While Alt Rear 3C failed, it is still an interesting option. I theorize that having a properly set-up side structure might better distribute the load. It also must be considered that with such a wide shoulder bar, that approximating the load by applying it to the edges of the bar may not be ideal.

Figure 38: Alternate Shoulder Bar

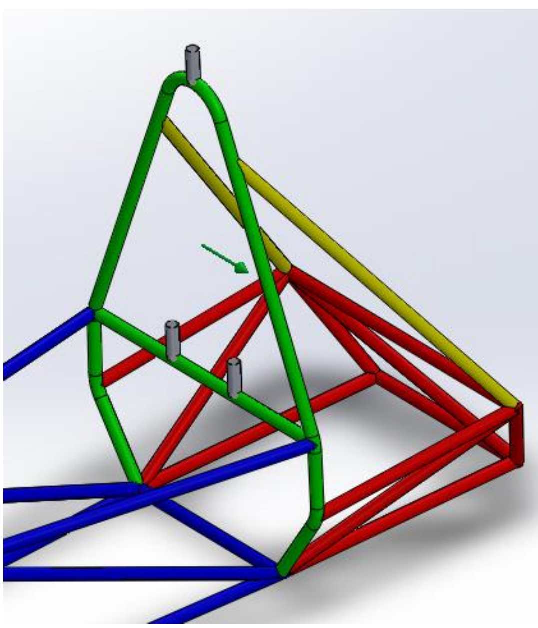


Table 15: Shoulder Bar Deflection

	Shoulder	
	Stress	Deflectior
Baseline	4.84E+08	18.22
Shoulder Bar 3C	9.34E+08	12.86

Due to the extreme width of the bar, versus the narrowness of the mounting points, the bending stress in the bar is too large to make this a viable option. It also presents logistical challenges with the seatbelts.

Main Hoop Spar

In the wonderful world of trends in FSAE, one trend is a main hoop spar. It is a member that runs between the lower bends of the main hoop, to increase the torsional rigidity. It may also increase the strength in the rollover test, particularly the 5KN side-to-side load.

Figure 39: Rear with Spar

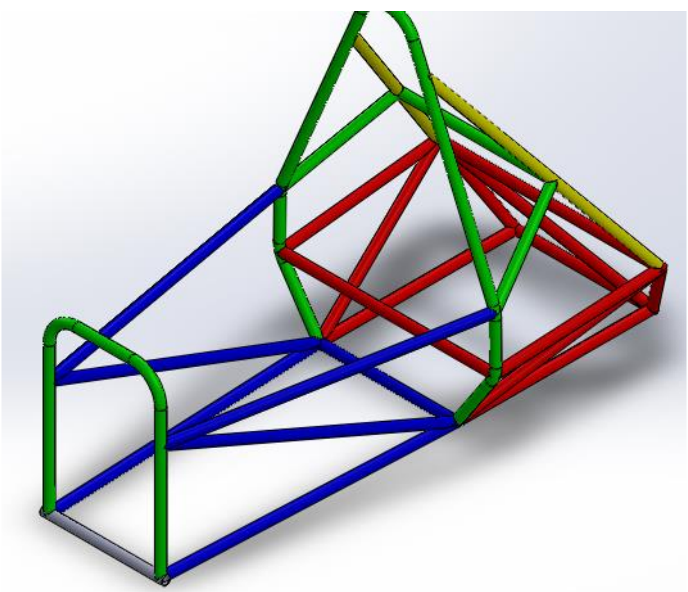


Table 16: Deflections with Spar

	Shoulder		Main Hoop	
	Stress	Deflection	Stress	Deflection
Baseline	4.84E+08	18.22	9.30E+08	23.21
Main Hoop Spar	4.30E+08	11.94	8.06E+08	22.1

Adding this spar across the main hoop has helped lower the stress in rollover situations dramatically, by increasing the torsional rigidity. It will be used in further designing.

3D Sides

As was previously mentioned, the side geometry is dependent on the packaging. In all cases, one issue is the location of the side impact member. In the front and rear, the first bends are offset the same height distance from the base of the respective hoop as they were on the 2016 frame. However, other changes to the geometry means this is no longer valid. Instead, the 7.51" at the front will be used for the sides and rear. This frees up a bit of space for drivetrain in the rear, where it's needed.

Depending on the final packaging, there are a variety of options for the sides available. While all of them are rules equivalent, the preferred one is the one with the least members. It is still of interest to see the strength of each. Unlike with the front and rear, there are not multitudes of constraining dimensions to iterate with.

For the analysis, the boundary conditions are those specified by SAE, with the base of the hoops fixed in translation, but with permitted rotation. The 7 KN load was applied at each end of the upper side-impact member.

Figure 40 below shows the four different options, overlaid. Flat (blue), Hybrid (grey), Triangular (pink) and Square (yellow) Side-impact designs. Individual designs can be seen in the Appendix. All materials are 1 x 0.065"

Figure 40: Options for Side Design

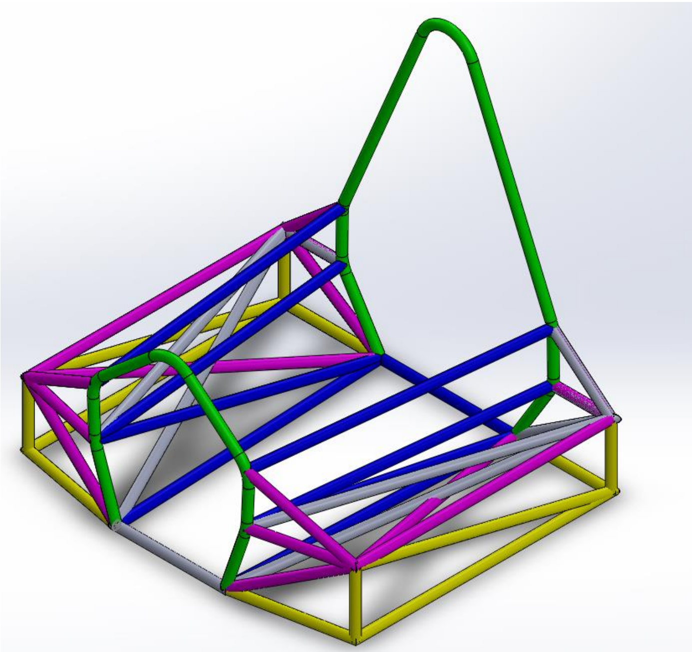
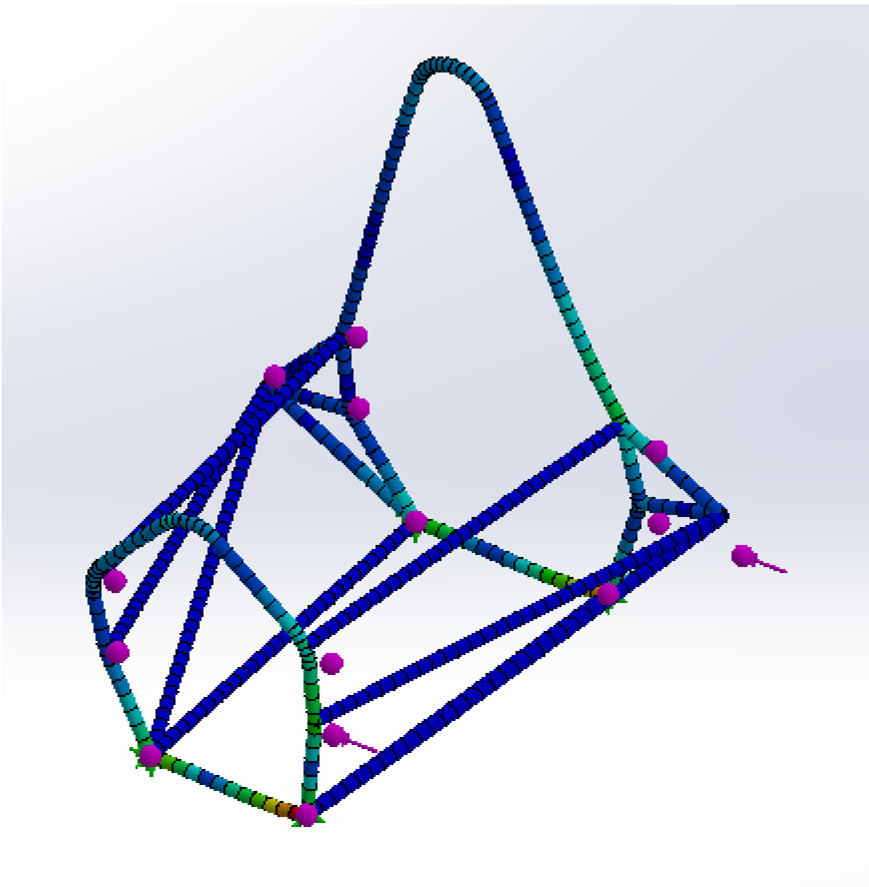


Table 17: Side Impact Comparison

	Flat	Hybrid	Triangular	Square
Members	8	14	20	30
Weight	25.47	29.14	32.34	39.43
Stress	5.713E+08	5.729E+08	4.953E+08	3.159E+08
Displ.	10.960	9.945	9.710	3.649

While the stress is above yield in most cases, I believe it to be a by-product of not having the full front and rear sections in this model. Once again, the highest stress is in the base of the roll hoops. By the packaging shown in the 3D rear section, the triangular hybrid sides are the smallest sides that meet all of the packaging requirements. The results for the Hybrid (grey) can be seen in Figure 41

Figure 41: Side Impact FEA



FINE TUNNING AND RULES COMPLIANCE

By combining the isolated front, side and rear models, the full frame comes into focus. However, there are a multitude of details that need to be sorted out.

Center of Gravity

For lack of a better alternative, this frame is being designed with the same axle locations for 2016. 61.5” wheelbase, with the rear axle offset 15.5” from the back of the main roll hoop. The tires, when loaded and inflated have a diameter of 20”. Therefore, the axles will be offset 10” from the ground.

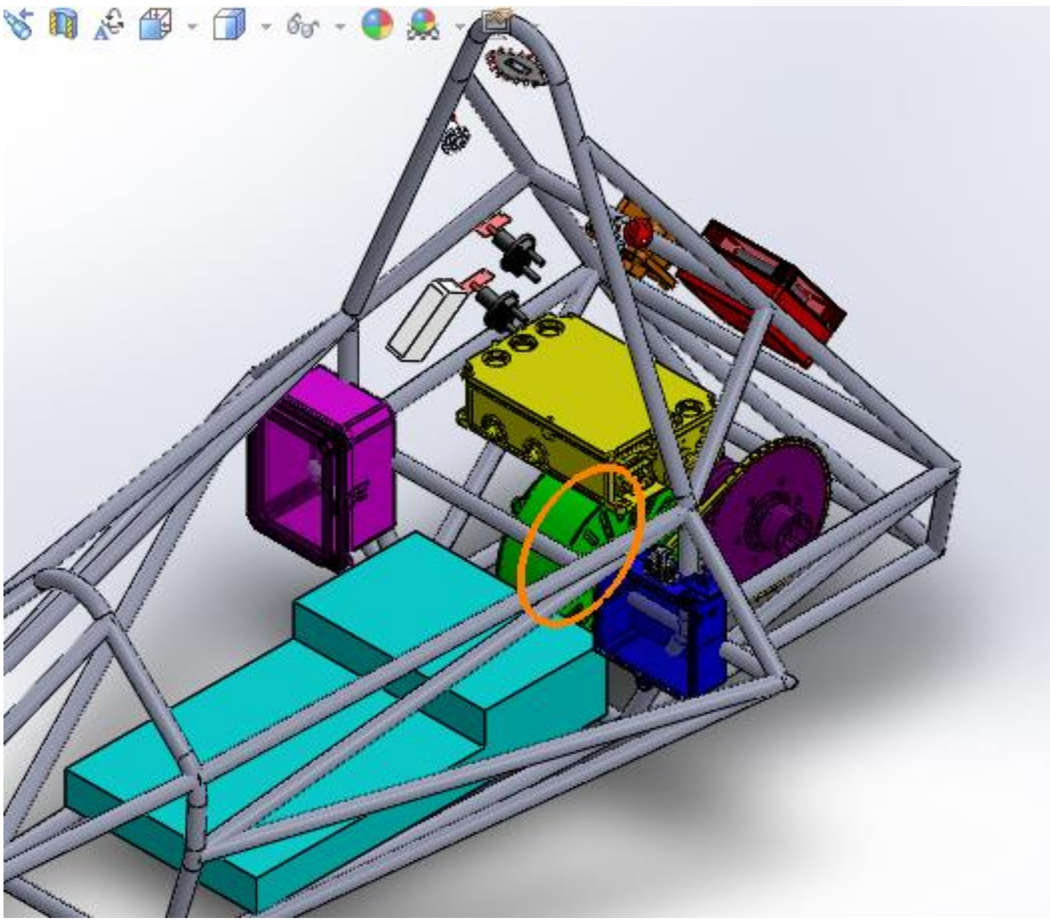
The height of the side impact structure is one of the more stringent rules. The entire height of the member (1” OD) must be between 11.8 and 13.8 inches (300 to 350mm) above the ground. This effect can be obtained by a combination of raising the ride height, and adjusting the dimensions of the suspension box. To avoid altering the finite element analysis, the ride height should be set at 4.5”. For comparison, the 2016 car had a front ride height of 5”, and a rear height of 3”.

In considering the center of gravity, the weight of the components must also be considered. The weight of the motor, motor controller and diff is close to 80 lbs. The driver is 170 pounds, and another 100 in batteries. Based on the knowledge gained from the micro-iterations, we can safely increase the size of the suspension box, and lower the ride height to 3.5”

Packaging and 95% Man Template

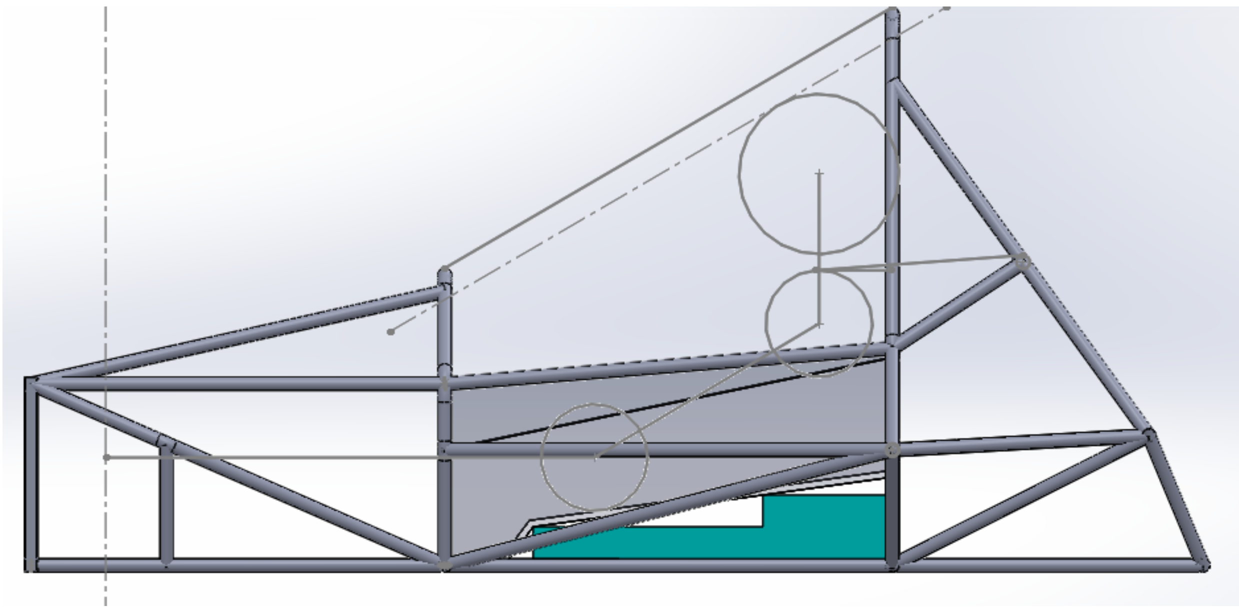
The accumulator and packaging design previously created is still not perfect. It raises the center of gravity, and that is not desirable. It also requires odd, not orthogonal mounting of the motor controller. In considering the number of cells, a flat-pack accumulator was suggested. It would have the 70S/4P configuration of the 2015 car. The basic geometry could also be re-configured for a different, stronger cell, in a 90S/1P configuration.

Figure 42: Revised Packaging



With this, the 95% man can be placed into the car, and the height of the main hoop adjusted to make the frame rules compliant. The positioning of the template also allows for additional Accumulator parts to be placed under his lower back.

Figure 43: Positioning of 95% Man



Hoop-to-Hoop Bracing

The final thing that needs to be considered is hoop-to-hoop bracing. In the sled testing, there were braces going from the top bend in the front hoop to the base of the main hoop. In order to make the accumulator packaging work, different braces were considered. The analysis method was applying the two hoop load conditions to a provisional full frame model. The members in question are 1 x 0.065”, and are shown in grey. Same as with the sides, the different options are superimposed.

Figure 44: Hoop-to-Hoop Bracing Test Model

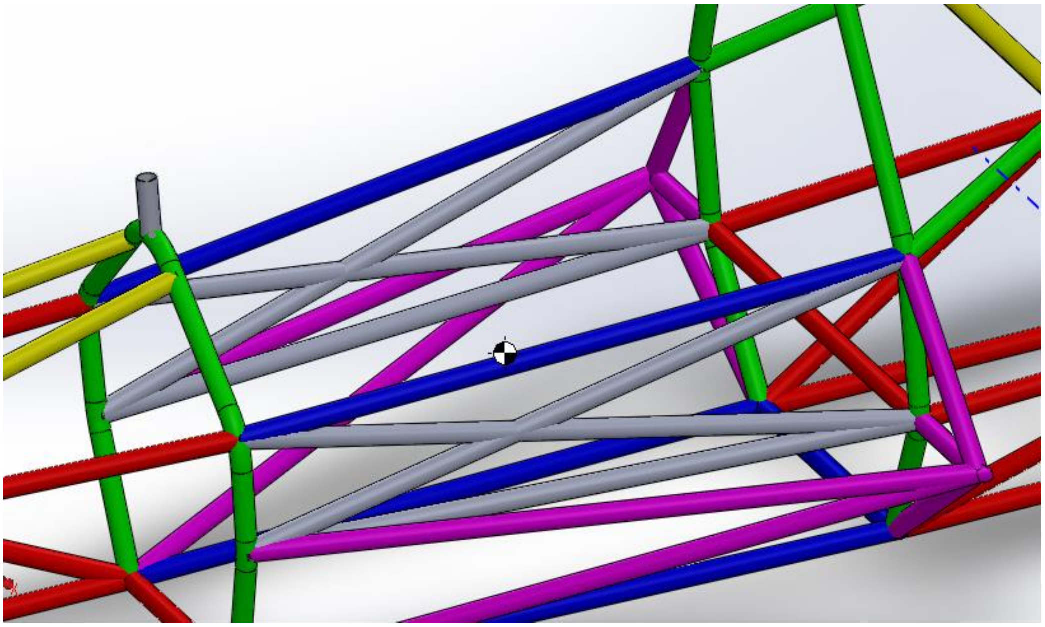


Table 18: Hoop-to-Hoop Bracing Results

	Mid Front High Back		Mid Front Mid Back		High Front Mid Back	
	Front Hp	Main Hp	Front Hp	Main Hp	Front Hp	Main Hp
Stress (PA)	8.29E+08	9.12E+08	8.21E+08	9.16E+08	8.57E+08	9.18E+08
Displ (mm)	11.91	29.17	12.26	28.68	12.31	32.58

While the straight braces were the most effective, attaching to rear hoop lower bend node is not desirable for packaging. Hence, the Mid Front-Upper Rear configuration was chosen. It is also interesting to see that in combining the models, the stress in the front hoop rollover has decreased, while the stress in the main hoop rollover has increased.

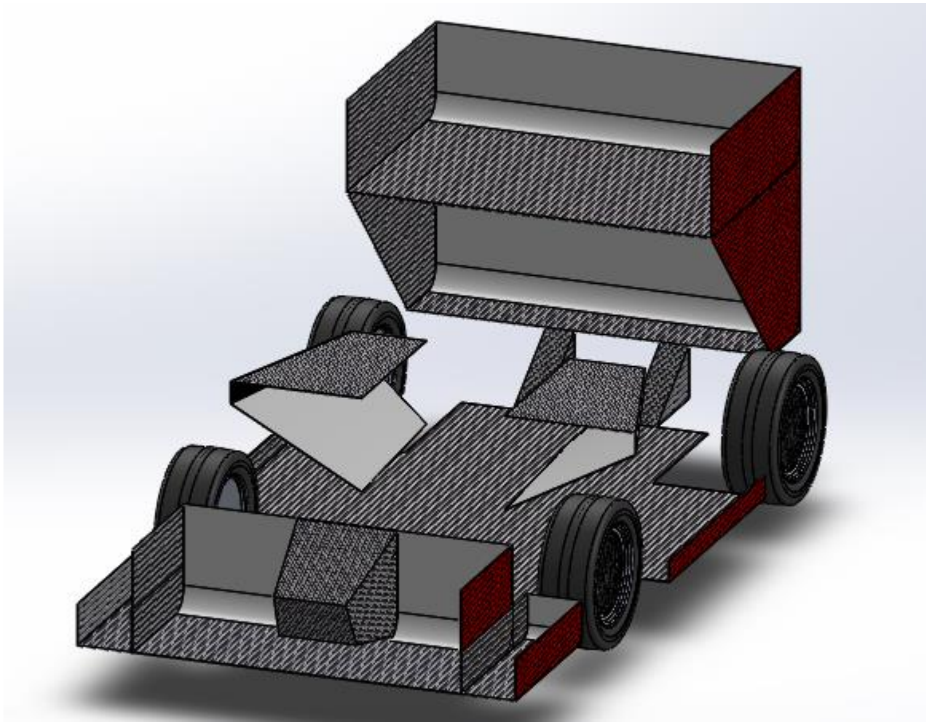
Aerodynamic Considerations

The aerodynamics, generally speaking, have to work around the frame and other subsystems, not the other way around. Regarding the frame: tire size and axle height effects ride height, and this effects the side impact structure. The ride height then effects any possible under-tray or front wing, with a lower ride height being preferable.

One way that aero effects the frame is through the packaging. While difficult to make, there is great benefit from non-structural side pods. On gas cars, these can be used to house radiators, exhaust, and other miscellaneous components. For an electric vehicle, it would be possible to have circuit boards, or low voltage batteries inside of them. This would allow eliminating the triangular sides all together. With both set-ups, a side impact would be a “fatal” injury. However, a radiator is much cheaper and quicker to repair than a printed circuit board.

In a FSAE car, there are certain compulsory aerodynamic pieces. The driver’s cell must be enclosed by a series of panels, including a floor close-out. For most non-monocoque cars, this takes the form of a series of removable carbon fiber, fiberglass, or plastic panels. These are then shaped around the frame to reduce drag from the airflow. While not required, a nose cone is almost always present. In this sense, the frame can only hinder aero, and never help.

Figure 45: Full Aero Package



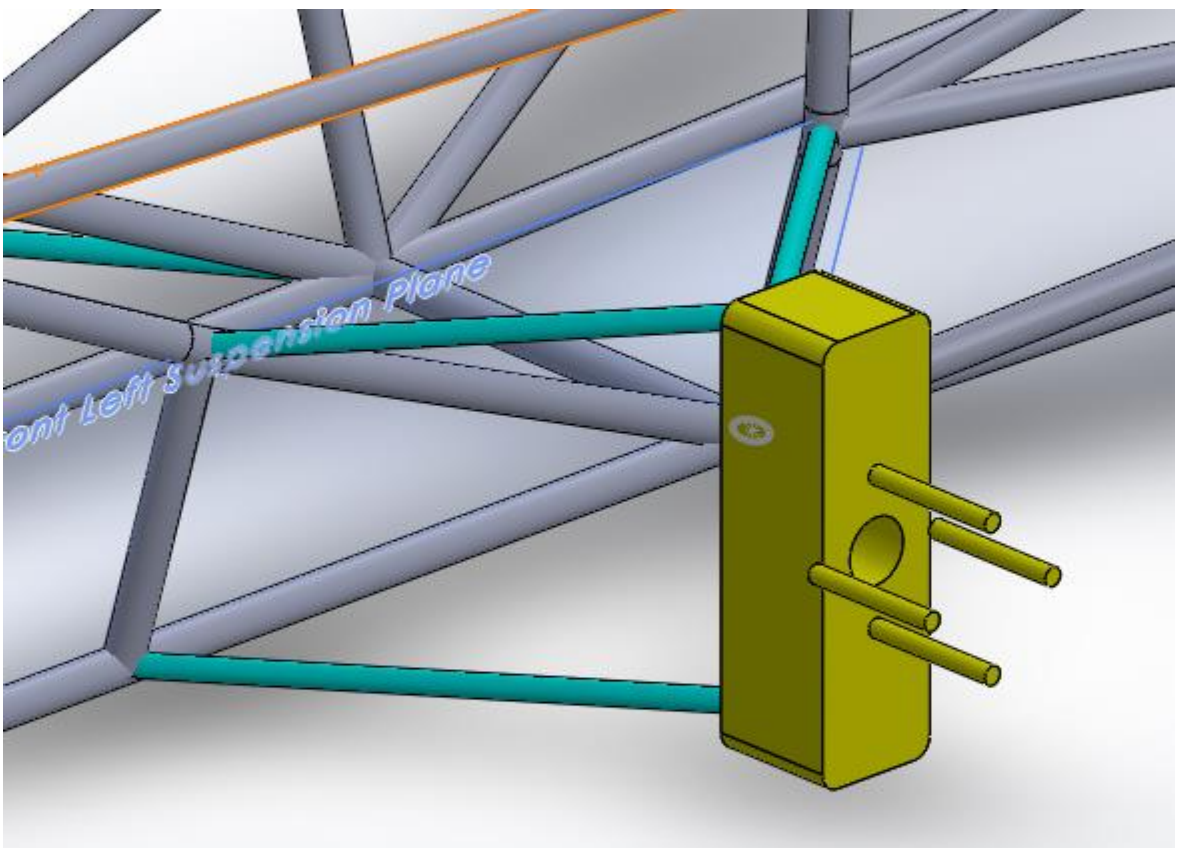
Suspension Considerations

Suspension is a key consideration in designing a FSAE car. In order to induce the desirable camber (wheel tilting, from a front/back view) in cornering, the upper suspension arm needs to be shorter than the lower ones. This means having upper suspension points that are farther out from the center than the lower ones. It is also desirable that certain points be higher than others, to combat diving and squatting. As previously seen, in order to pass the legs template test, the front suspension points were pulled up and out. In order to make other suspension factors more predictable, it is ideal to have all the points lie on a plane.

As a quick reminder, planes can be defined in a number of ways. These include 2 parallel lines, two intersecting lines, a point and line, or three points. For these applications, three points will be used to define the planes. All of these points are all related back to the planar and width sketches.

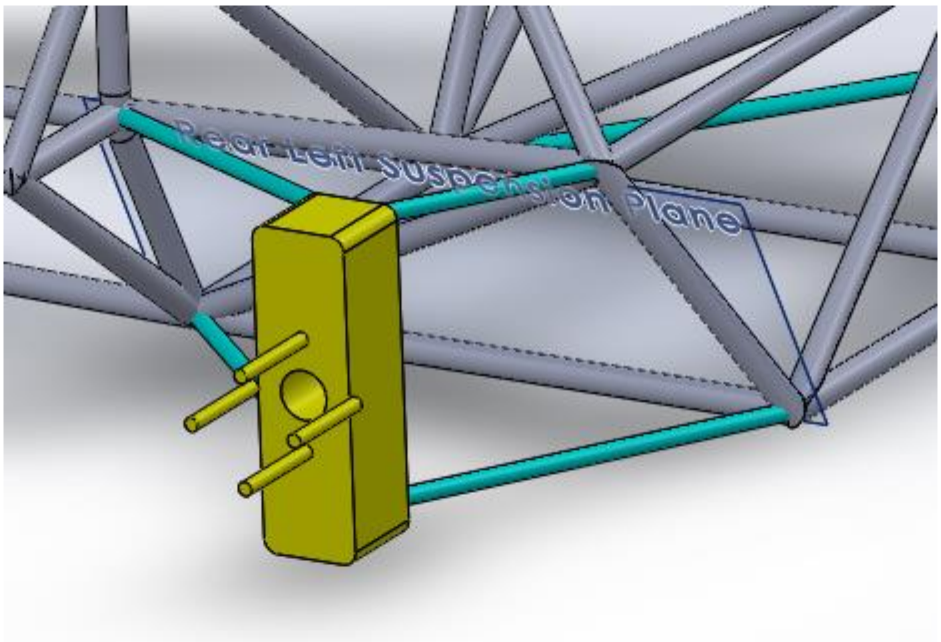
For the front suspension, a plane is defined, with the three points being the base of the hoop, the lower hoop bend, and the base of the bulkhead. The first two of these will be the actual suspension points. The third point is said to lie on the plane, as well as a projection of the 2D suspension diagonal, and 1/2” above the lower hoop bend node. The fourth point is directly below the third point, on the lower bulkhead support member.

Figure 46: Front Suspension Plane



For the rear suspension, the plane is defined at the base of the main hoop, lower bend in the main hoop, and the bottom of the rear bulkhead. The top of the rear bulkhead then is made to be on this plane, projected from the 2D Sketch. The raise, for anti-squat is already built into the underlying 2D Sketch. While this adjusts the rear end geometry, there is no effect on the tube count, and minimum effect on weight. It also removes the width of the top of the bulkhead as a variable.

Figure 47: Rear Suspension Plane



Final Finite Element

<u>Procedure</u>

Having considered all of these sub-systems, the final finite element analysis needs to be run. This analysis will be run in a dedicated FEA Program, Abaqus. To import the Geometry into Abaqus, the Model was converted into a wireframe, by deleting all Weldments structural members. The remaining sketches had all references to other sketches deleted, and then all construction lines were deleted. The geometry was exported from solidworksas a STEP file.

The geometry was imported into Abaqus, combining all of the wireframe “parts” into a single part. To correct import errors, the geometry was changed to precise. The material was created, followed by creating the three bream profiles. The beam profiles were then assigned to members. For the Front Bulkhead, a 1” x 0.049” Square member, it is equivalent to the yellow 1” x 0.065” tube, and was included in the yellow set. The part was seeded and meshed, with a global seed size of around 1.5. Additional seeds were placed on the bends, and the total was 750-850 elements. The boundary conditions and loads were created per the rules, and previous analysis.

In fine-tuning the analysis, there are a variety of options, depending on the expected and observed behavior of the model. The Nelgerom, used when large displacements are present, was turned on. Analysis was ran on one load condition, using the baseline mesh, hybrid (quadric) elements, and a 3000 element mesh. The results can be seen in Table 17.

Table 19: Comparison of Different Abaqus Meshing Options

	Stress (PA)	Displ (mm)
Shoulder Bar - Baseline	1.93E+08	5.68452
Shoulder Bar - Quadradic	1.96E+08	5.68452
Shoulde Bar - Fine Mesh	1.90E+08	5.68198

We can see that for this model, there is no major effect in the behavior by changing these factors on the model. If there was any doubt, a model with more nodes and more degrees of freedom provides a better correlation to reality. The Analysis was ran using the baseline mesh conditions, and the results can be seen in Table 18 below

Table 20: Finite Element Analysis Results

	Stress (PA)	Displ (mm)
Front Impact	6.37E+08	21.07438
Front Hoop	6.83E+08	16.60906
Side Impact	3.88E+08	7.49808
Rear Impact	2.66E+07	0.0862584
Main Hoop	8.46E+08	32.3088
Shoulder Bar	1.86E+08	9.8425

These results are almost acceptable. Before, there were large stress concentrations at the bases of both roll hoops, in both rollover scenarios. While the stress concentration at the base of the front hoop has diminished, the main hoop still substantially exceeds the allowable stress. It also comes close to material failure at the front hoop and front impact tests. By increasing the wall thickness of the material beneath each roll hoop, the energy dissipated there should causes less stress. For safety, the material under both hoops is now the same as the hoops themselves, 0.095" wall tube. The results of this can be seen in Table 21:

Table 21: Reinforcement of Roll Hoop Bases

	Stress	Displ
Main Hoop	8.46E+08	32.3088
Reinforced	5.92E+08	30.3022
Front Hoop	6.83E+08	16.60906
Reinforced	5.35E+08	16.39824
Front Impact	6.37E+08	21.07438
Reinforced	6.15E+08	20.701

This solves the issue of stress in the Main Hoop rollover, but there is still too much deflection. The magnitudes of the deflections in the two relevant directions can be seen in Figures 48 and 49 below.

Figure 48: Displacement in X Direction (Max: 1.090”)

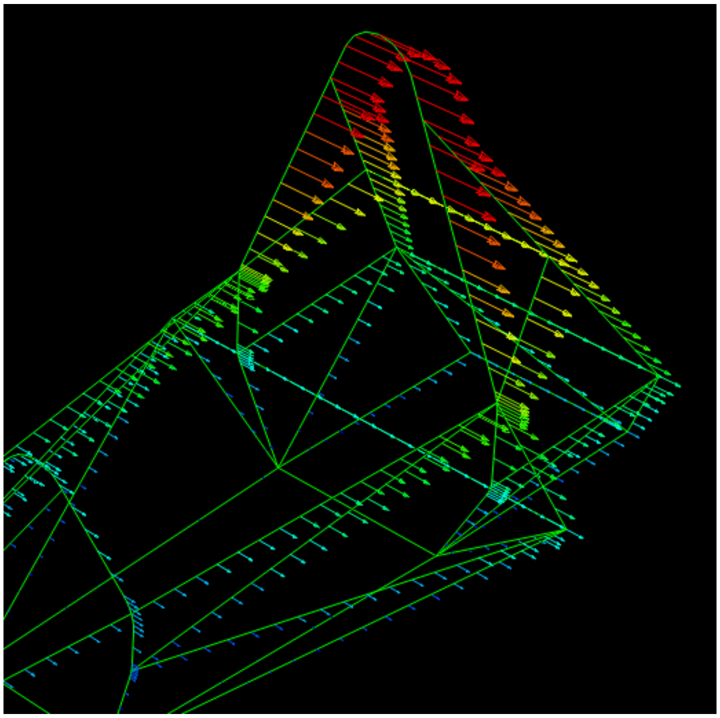
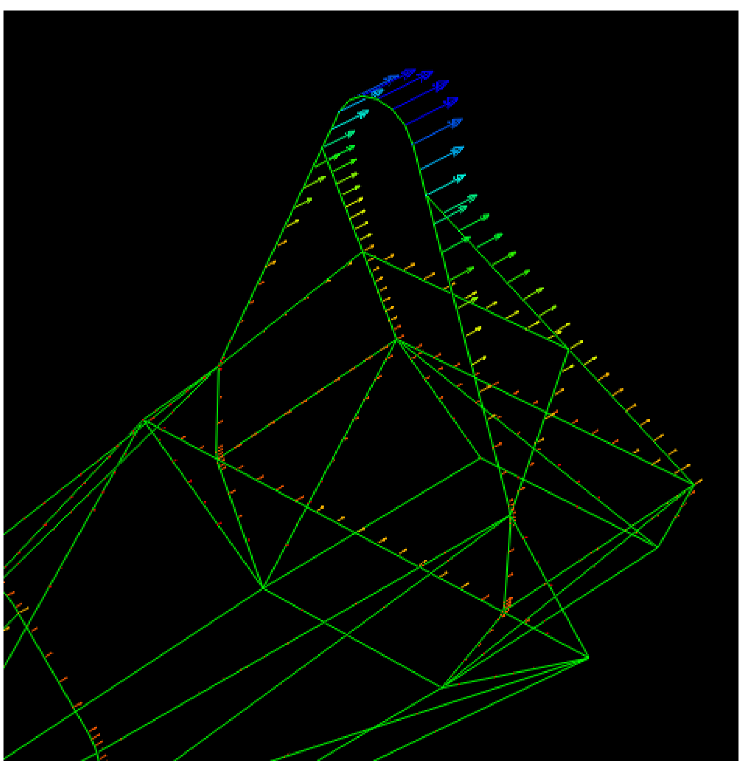


Figure 49: Displacement in Z Direction (Max 0.667”)



This presents an interesting conundrum. In consulting the micro-iterations done on the rear geometry, raising or lowering the main hoop braces should provide added stiffness. However, in consulting Figures 48, there needs to be stiffness added in the main hoop plane. Using an idea from SAE Baja, the main hoop spar was rotated. The results can be seen in Table 22 Below. Because of the asymmetry, it was tested with the lateral portion of the load applied in both directions

Table 22: Main Hoop Deflection Reduction

	Stress	Displ	Displ 1	Displ 3
Reinforced	5.92E+08	30.3022	27.686	16.95704
6" Bracing	6.23E+08	31.369	25.527	18.66392
4" Bracing	6.30E+08	29.718	25.4762	15.59052
Spar Diagonal	3.52E+08	20.54352	7.88162	18.43532
Spar Diagonal	4.34E+08	16.27886	8.90524	13.9954

As was expected, the change in the bracing dimension had a minimal effect, but the diagonal spar was much more effective. The final step is to make sure that modifying the spar has had no effect on the other impact scenarios, particularly because of the asymmetry.

Table 23: Final Analysis Reconfirmation

	Stress (PA)	Displ (mm)
Front Impact 120 KN	5.10E+08	21.40966
Front Hoop 6,5,9 KN	4.83E+08	15.54734
Front Hoop 6,5,9 KN	4.90E+08	16.36522
Side Impact 7KN	2.19E+08	4.445
Side Impact 7KN	2.46E+08	4.52882
Rear Impact 5KN	2.72E+07	0.087884
Main Hoop 6,5,9 KN	3.52E+08	20.54352
Main Hoop 6,5,9 KN	4.34E+08	16.27886
Shoulder Bar 14KN	2.02E+08	11.10742

With this, the Design is finished and approved.

DESIGN SUMMARY

The 2016 FSAE car has 75 members, weighing 76.4 lbs. The optimized design has 51 members, weighing 63.4 pounds. This represents a savings of 32% on Members, and 17% on weight.

Figure 50: Full Frame Isometric

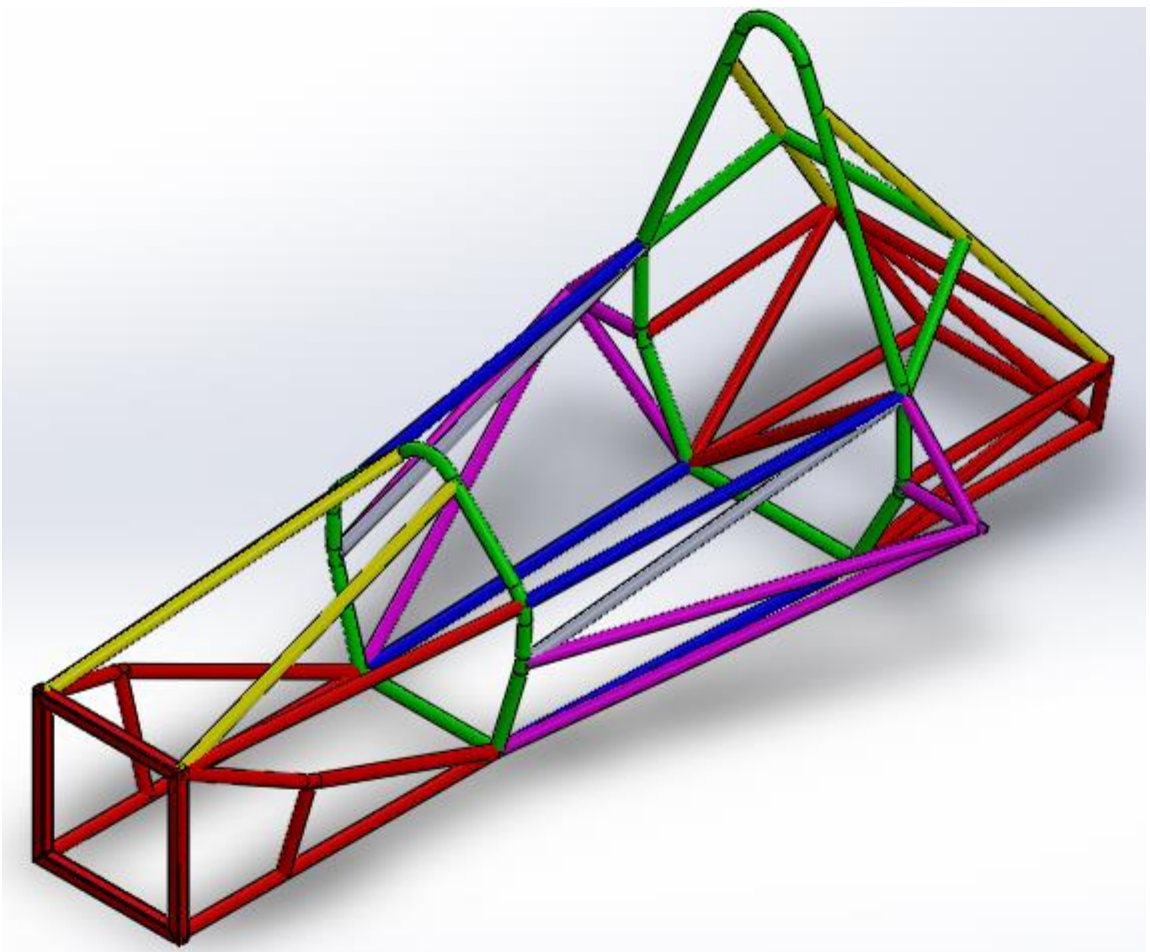


Figure 51: Full Frame, Side

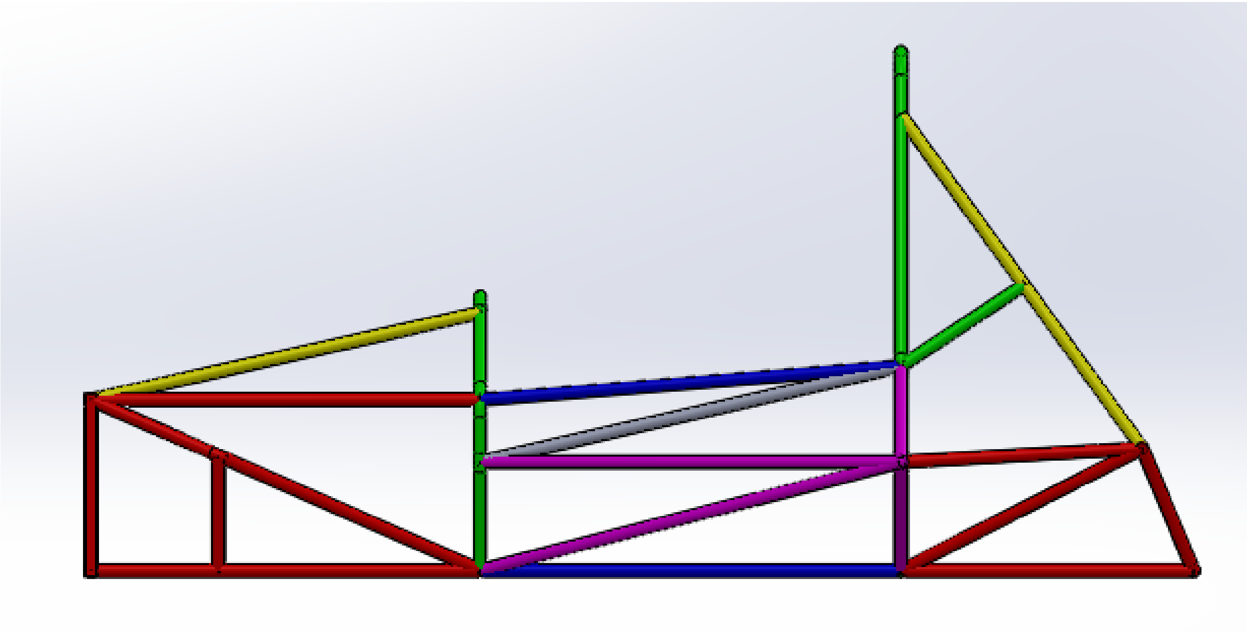


Figure 52: Full Frame, Top

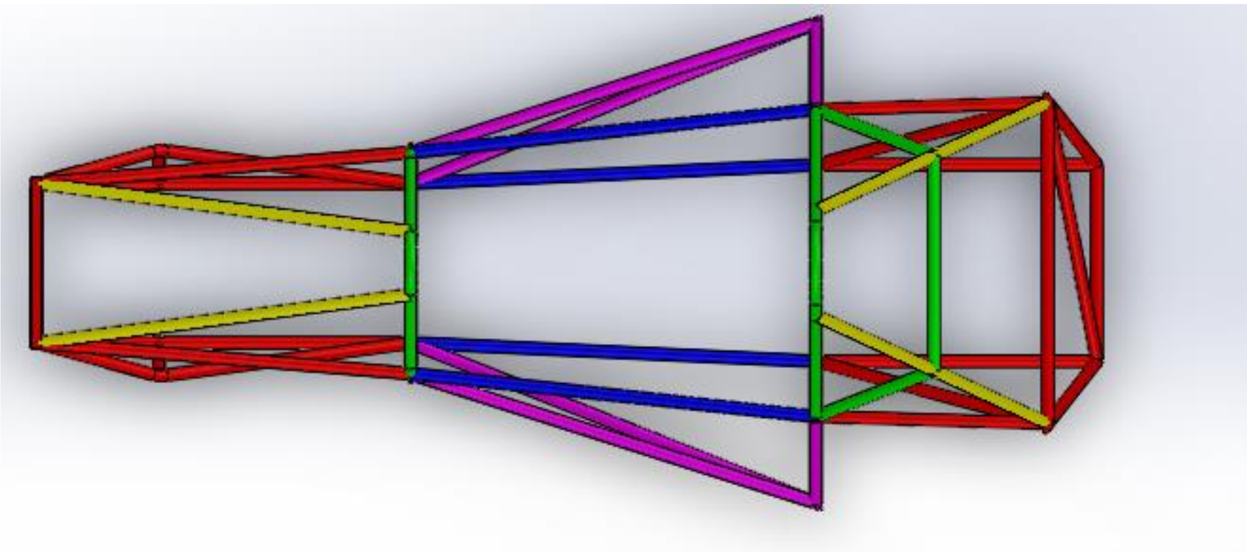
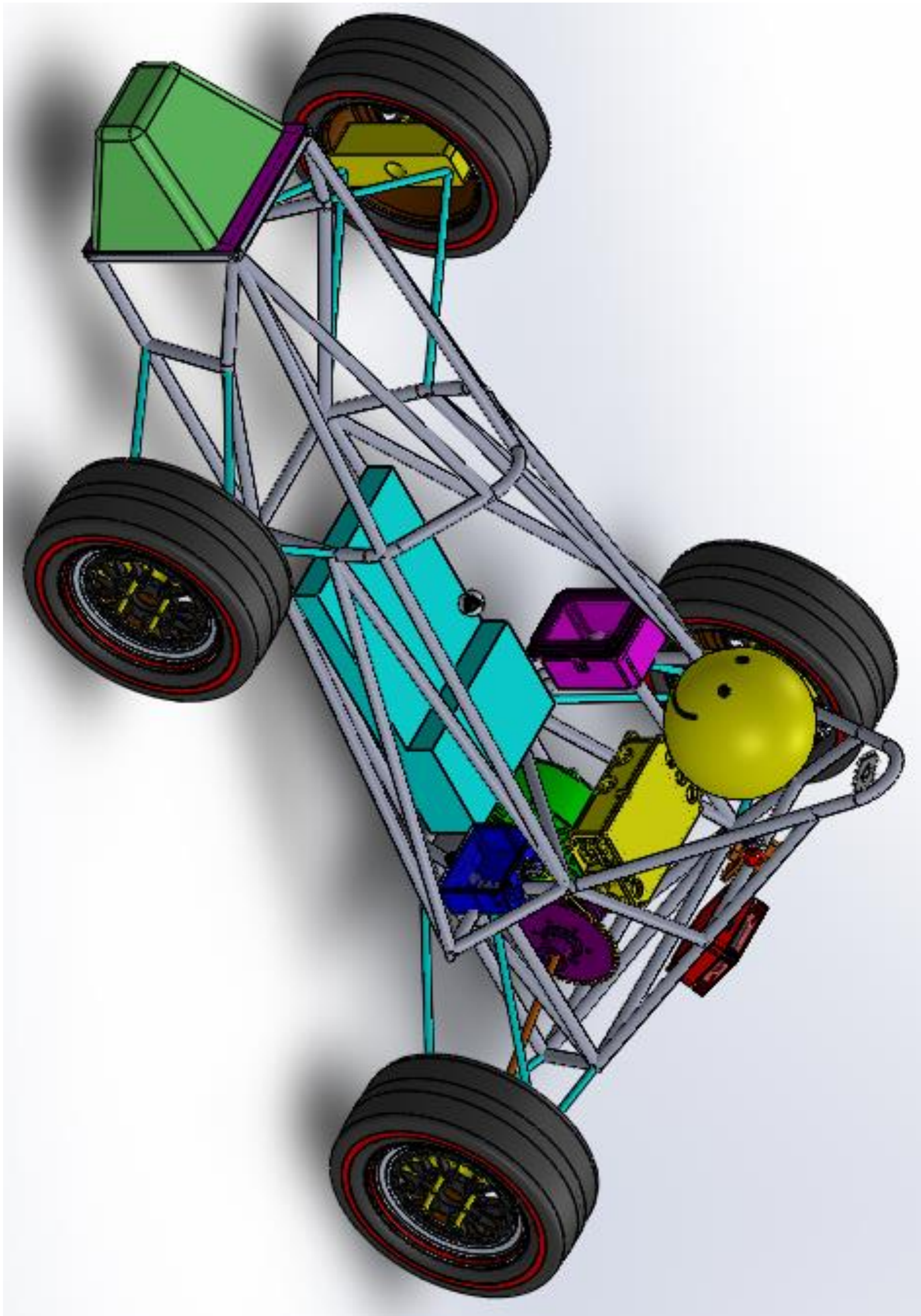


Figure 53: Full Car, Isometric



MANUFACTURING AND ASSEMBLY

As was previously mentioned, there were a few basic manufacturing considerations made in designing the car. Once the design is mostly finished, the next phase is to make detailed plans for welding and manufacturing. Our preferred vendor is VR3 Engineering, in Stratford, ON. If you provided a mitered 3D Model, they will quickly and precisely cut and bend the tubes.

One requirement for submitting an order is to provide a drawing and bill of materials, with each unique part having it's own part number. It is entirely possible to arbitrarily number all of the bodies in a single isometric view. However, for clarity, a numbering scheme is preferable. For the order of our 2016 frame, the frame was divided into 15 groups, from A to Q. (I and O were skipped). Bodies in those groups were then numbered, starting with 01 for each group. For identification of bodies, I believe that all 6 views are necessary to identify all components. The drawings and bill of materials (BOM) for the 2016 and optimized frames can be seen in the Appendix

Table 24: BOM Summary Comparison

Group	2016 – Ref	Optimized
A – Front Bulkhead	4	4
B – Front Floor	4	3
C – Front Hoop Bracing	3	2
D – Front Suspension	12	6 (2x 1 Bend)
E – Front Hoop	1 (5 Bends)	1 (5 Bends)
F – Cockpit Floor	7	3
G – Cockpit Vertical Bracing	10	4
H – Upper Side Impact	6	4
J – Cockpit Upper Bracing	8	6
K – Main Hoop	1 (5 Bends)	1 (5 Bends)
L – Shoulder Bar	1 (2 Bends)	1 (No Bends)
M – Main Hoop Bracing	2	2
N – Rear Suspension Box	6	6
P – Rear Bulkhead	5	5

Q – Rear Bracing	5	3
Total	**75**	**51**

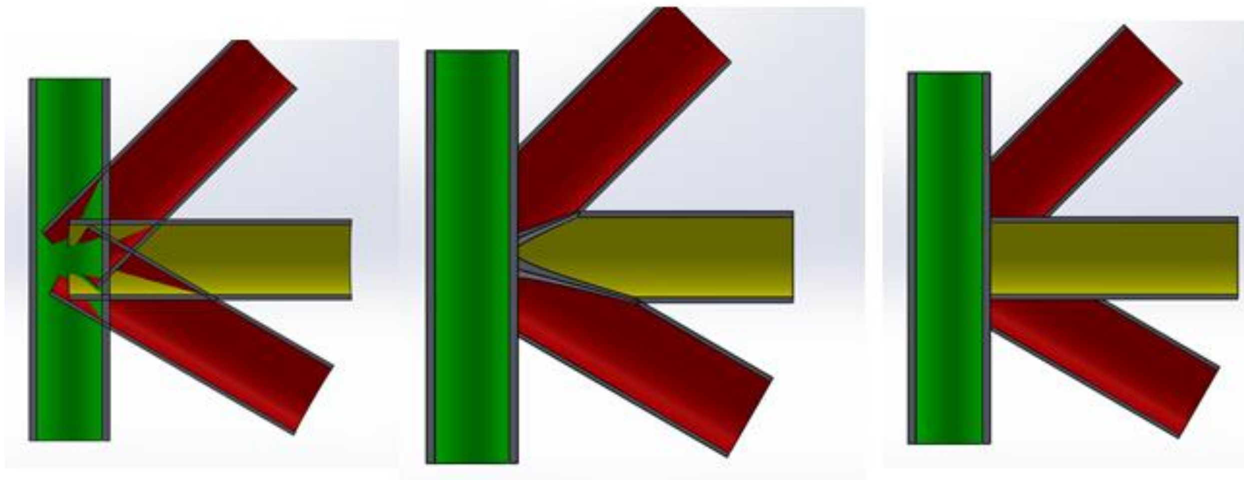
To build the frame on the welding table, planar sections are desired. The parts can be tacked, and possibly full-welded. This makes final assembly much easier. In manufacturing the 2016 frame, 1:1 print-offs of all of the planar sections were made. This allows for local identification of bodies and dimensions. 1:1 prints were also made of all of the bent components, to verify their geometry.

Table 25: List of Planar Sections for Assembly

Planar Section	2016 - Ref	Optimized
Front Bulkhead	4X A	4x A
Front Floor	4X B	3X B
Front Left Suspension	3X D	
Front Right Suspension	3X D	
Lower Left Cockpit Floor	2X F	3X F
Lower Right Cockpit Floor	2X F	
Side Impact Left	3X H	2X H
Side Impact Right	3X H	2X H
Main Hoop	1X K, 1X Q	1X K, 1X Q
Rear Bulkhead	5X P	5X P
Total	**31 (41.3%)**	**21 (41.1%)**

Once planar sections have been determined, mitering can begin. Care must be used in mitering, to prevent intersections and hollow node. An intersection is an impossible geometry, one that would need fixing with an angle grinder. Hollow node is when there is insufficient metal-to-metal contact, when multiple tubes converge onto a single point.

Figure 54: No Mitering, Incorrect Mitering, and Correct Mitering



With the part fully mitered, it can be sent out, quoted, and manufactured. For the frame alone on the 2016 car, the cost was approximately $2660. For the optimized frame, it is $2250, a savings of $410 dollars. These figures don't include FSAE student discounts

The Finished frame can be seen in Appendix 12

CONCLUSION

By using Finite Element Analysis, and re-organazing the packaging within the frame, a great weight and cost savings was created. This can be seen in the 17% decrease in weight, 32% decrease in frame members, and a savings of $400 in manufacturing. The frame still has sufficient strength in all tested load configurations. It accomplishes this through innovative geometry, cunning use of materials, and advanced packaging.

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Appendix

Rules and Regulations

Appendix 1: 2016 FSAE Rules, Sections T3, T4, T5, AF, EV3.4, EV4.2

Appendix 2: Approved Bulkhead Support Structures, Excel.

Appendix 3: TBD

2016 Frame: As manufactured

Appendix 4: W701-09.PDF and W701-09B.PDF 1-23-16

Appendix 5: W701-09 Bill of Materials 1-23-16

Frame Optimization

Appendix 6: FEA Summary – Planar (Solidworks)

Appendix 7: FEA Summary – Isolated 3D (Solidworks)

Appendix 8: FEA Summary – Full Frame (Solidworks and ABAQUS)

Appendix 9: Assembly Drawing

Appendix 10: Bill of Materials

Appendix 11: Abaqus CAE File

Appendix 12: Final Frame