

FORMULA SAE CAR CHASSIS DESIGN AND MANUFACTURING

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ENGINEERING PROJECT REPORT

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Abstract

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For this project our team wanted to design and build a chassis to be used in the FSAE competition. We felt the need to establish an SAE team at UOP and to market UOP's Mechanical Engineering program. In doing so we determined that a jig was needed to hold the chassis and we wanted the overall design of the chassis to be safe, weigh under 80 lbs, and be capable of fitting a six foot tall driver. Through design and analysis we successfully developed a chassis and jig in SolidWorks. Then through little over 100 hours of manufacturing, a complete chassis was built as well as a rotating jig set up. To confirm our engineering analysis we tested the torsional rigidity and verified our results. The chassis weight is 81 lbs, is capable of fitting a six foot tall driver, and provides the safety features necessary to protect the future driver.

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1. INTRODUCTION

Formula SAE is a competition held every year where university students compete to see who can design the best open wheel race car. Prior to this school year, UOP's ASME club never entered this competition. Building a complete car from scratch in less than a year is no small task. Because of this, it was necessary to break apart the construction of the car into different design groups: chassis, engine, drivetrain, suspension, cockpit, and brakes & steering. Our senior design group was motivated to design and build the chassis as it is one of the most critical parts of any successful race car.

Our goals for the chassis included designing and manufacturing:

- an inexpensive jig that will hold tubing in place within ± 1 mm and ± 0.5 degrees and allow for joints to be temporarily welded in order to construct the chassis.
- a chassis that is safe and protects the driver in the case of an accident.
- a chassis that is lightweight (under 80lbs), yet strong enough to withstand the forces of driving (up to 800lbs).
- a chassis that is capable of fitting a six foot tall driver.

2. CONCEPTUAL DESIGN

The FSAE rules and regulations allow for the design of the chassis to be either a monocoque or a tube frame design. One of the biggest design factors for our group was cost. Monocoque chassis by nature are very complex in terms of design, and they come at a hefty price. For comparison, a monocoque chassis can cost well over 10 times as much as a tube frame chassis. For this reason, going the tube frame route was the only option.

For the tubing material, hot rolled, low-carbon steel was chosen. This material was used throughout the frame and was picked due to its high strength, ductility, and low price. Another benefit of this type of steel is that it is very easy to machine, weld, and bend. All of the tubing used for the chassis is 1" in diameter and was purchased in bulk from McMaster.

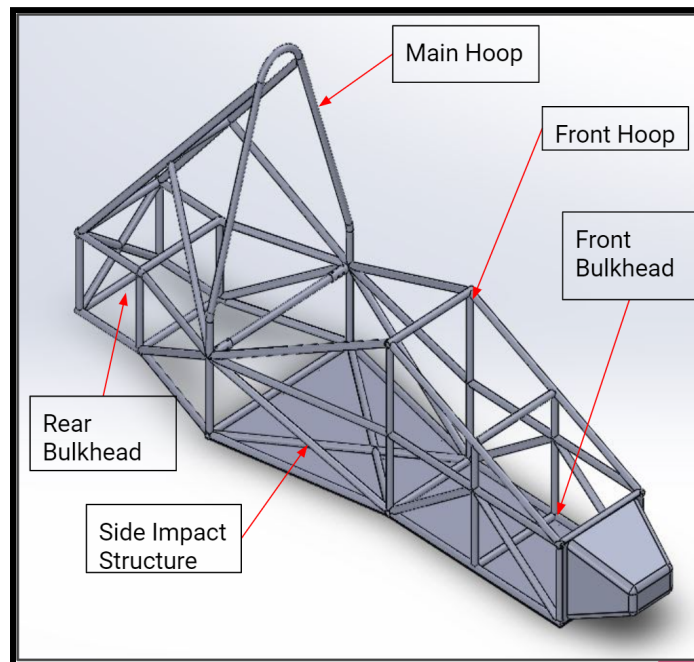


Figure 1: Isometric view depicting key parts of the chassis

The main sections of the chassis include the front bulkhead, front hoop, side impact structure, main hoop, and rear bulkhead as seen above in Figure 1. The tubing for the front hoop and main hoop feature a greater wall thickness (WT) than the rest of the chassis (0.095" vs 0.065"). Tubing thinner than the 0.065" WT could have been used in certain areas of the chassis, but since safety was a priority in our design, we opted to use the 0.065" WT tubing this throughout.

Per the FSAE rules, support tubes have to be triangulated ^[1]. Once again, we decided to err on the side of caution by including more support tubes, specifically in the front and rear bulkhead.

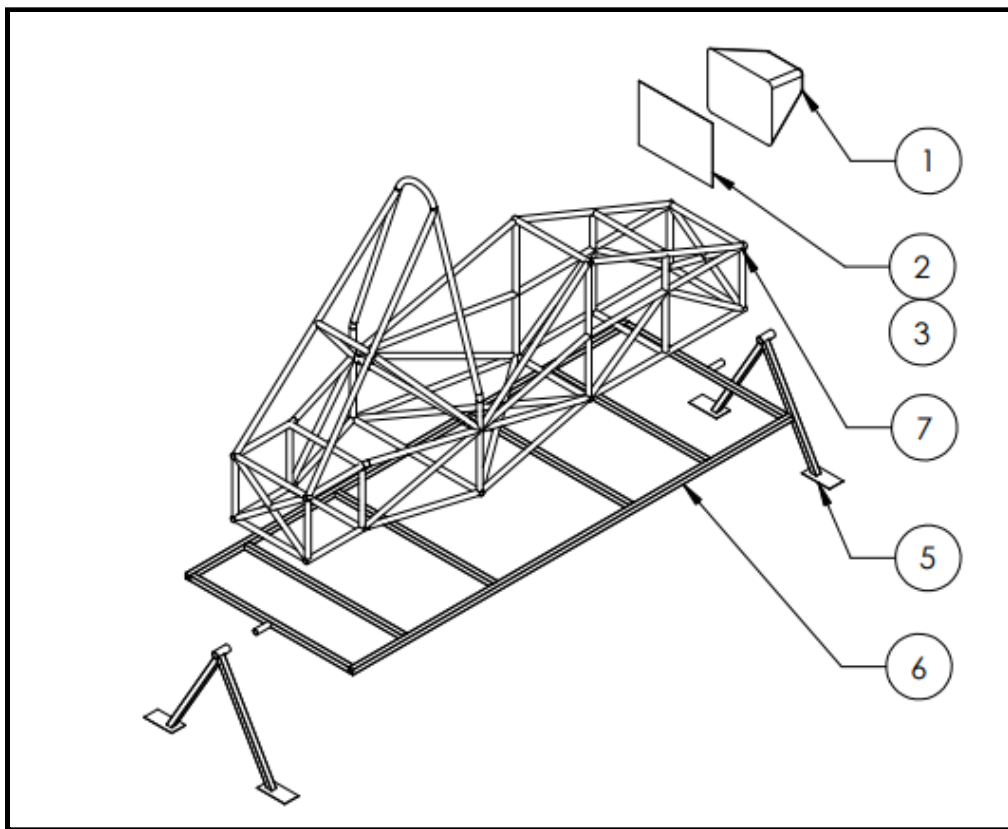


Figure 2: Chassis and jig major component list

The design of the chassis also includes a sleeve butt joint (Figure 1), impact attenuator, and an anti intrusion plate (Figure 2). The sleeve butt joint is directly

behind the driver's seat and allows for the engine to be easily inserted and removed from the car. The impact attenuator and anti intrusion plates are required by FSAE and are fixed to the front of the chassis to protect the driver in the case of a frontal collision. The chassis also features an aluminum floor that is mounted to the bottom of the frame using tabs leftover from the anti-intrusion plate. The floor plates were then riveted to the tabs.

We also had to design a mount stand for the chassis to sit on for when the car is assembled in the future. This stand was intentionally designed to be used for when we welded the chassis itself, but its use was changed once we acquired access to Delta's jigging table. The mount stand is made out of mild steel square stock and was welded at Delta. It has the same dimensions as the chassis and uses U-bolts to clamp to the chassis' lateral floor tubes.

3. DETAILED DESIGN and ANALYSIS

Head On Collision Analysis

We were curious to see what a head on collision with our current chassis would look like and defend the requirements from FSAE to have an Impact attenuator mounted to the front of the vehicle ^[1]. We used FEA to see how the chassis responds to a front end impact of roughly $3.28 \times 10^6 \text{ N}$ (80 km/hr collision). We assumed the collision to be elastic and that the vehicle would be impacting a stationary wall. We also assumed a safety factor of 4 on the mass (originally 250kg). We assumed the speed at impact to be 80 km/hr. This is an overcompensation as the average speed around the circuit will be roughly 40 km/hr. It is important to note that the distance traveled during the collision is assumed to be the same as the deflection. This required us to run multiple FEA and iterate because if we initially estimate the distance traveled (1 m) for the first point of impact until the vehicle is at rest (0 m/s) the impact energy would not be as great. This low impact energy would result in a small deflection, contradicting the distance traveled in the collision. Because of this, we had to decrease the distance traveled until it matched the deflection to get an accurate value for the maximum deflection.

Based on the FEA it is clear that there is a need for an impact attenuator. The maximum deflection is around 71 mm which is within 6 percent of the distance traveled during the impact (75 mm). The next locations of concern are the sides of the front hoop which sit right next to the drivers upper legs. There is a deflection of about 50 mm on both sides and with an initial distance of 375 mm between them, there is only 275 mm left for the driver to get his/her lower body out of the cockpit (slightly

less than 1 ft). Once again reiterating why we need the impact attenuator. With the impact attenuator we expect to see only 50 mm of deflection at the front of the vehicle and 35 mm of deflection at tubes on the side next to the driver's legs. The results are close to what we expected and reinforces the reasoning for the FSAE organization requiring an impact attenuator. For images of our FEA please reference appendix A.

Torsional Rigidity Analysis

Another important metric we wanted to analyze was the torsional rigidity. This is important to know because it can be used in kinematic analysis of handling and also compare our vehicle to others. We are assuming that the tires do not slip. The tube weldments feature in Solidworks only allows for the tube joints to be given conditions. This means that the joints of the 3 suspension mounts that are not being loaded are fixed. An arbitrary load of 1500 N is then applied at each of the 2 joints where the front right A-arms of the suspension meet the chassis. We placed the loads on the bottom 2 joints even though there are 4 hardpoints at each corner of the vehicle for the A-arms to mount to. This is because each set of 2 hard points share the same tube and are on the tubes themselves (not the joint) which can not be modeled due to the limitations of the weldments feature in Solidworks. To calculate the maximum deflection we used the calculated maximum force (4356 N) multiplied by the moment arm of 0.23 m^[2]. The moment arm is in the Y axis and represents the distance between the lower A-arm and the mounting point of the shock obtained from the FSAE car model. This is because it is assumed the load is transferred through the shock.

Based on the FEA the max deflection would be approximately 0.236 degrees on the side of the chassis experiencing loading. From this we can conclude that the chassis will be strong enough to withstand the forces experienced in the turns. The chassis itself experiences very little deflection, resulting in a positive effect on the overall handling of the car. With this small of a deflection we should be able to withstand higher loads. For images of our FEA please reference appendix A.

4. MANUFACTURING CONSIDERATIONS

Once we had all of our tubes, we were able to begin manufacturing the chassis. In total, we spent a little over 100 hours in total to produce the complete chassis and spent roughly \$1000 in materials (Appendix B). More detailed cost and time tables can be found in appendices B and C.

The first step in the manufacturing process was cutting and bending all the round and square tubes to size. The cutting was made easy by using a chop saw (Figure 3) and was completed fairly quickly. The main hoop and four other tubes featured bends, and this was carried out at Delta using their pneumatic tube bender (Figure 4).



Figure 3: Chop Saw



Figure 4: Pneumatic Bender

The most time consuming part of the project was notching and grinding the ends of the tubes so that they would mate with each other nicely. The tube notcher that we used can be seen in the figure 5. We initially used 1:1 notch patterns that we cut out and taped at the ends of the tubes (Appendix D). For the most part these worked, but lining up the patterns and taping them to the tubes accurately was a challenge.

About halfway through the notching process we moved our notching operations to Delta and ditched the paper patterns. The templates we had were useful for the more simple notches, but were too difficult to follow with the more complex notches. For the more challenging notches we used a combination of the tube notcher and grinders to complete the tube ends. We would reference the chassis, mark the tubes, grind, and repeat until we were satisfied with the fit. Roughly half of our manufacturing time was spent taping, grinding, and notching the tubes as seen in our Gantt chart in appendix C.



Figure 5: Square tube being cut with tube notcher

Welding was used to permanently assemble the chassis. We were fortunate to use Delta College's jigging table to carry out this process. Wedge clamps were used to hold the floor of the chassis on the table, and then tack welds were used to secure the joints so that they could be fully welded. TIG welding was used throughout the chassis as it is more precise than MIG. Only a quarter of a weld was completed at a time to reduce the effects of the heat distortion on the tubes. A completed weld can be seen in figure 6.

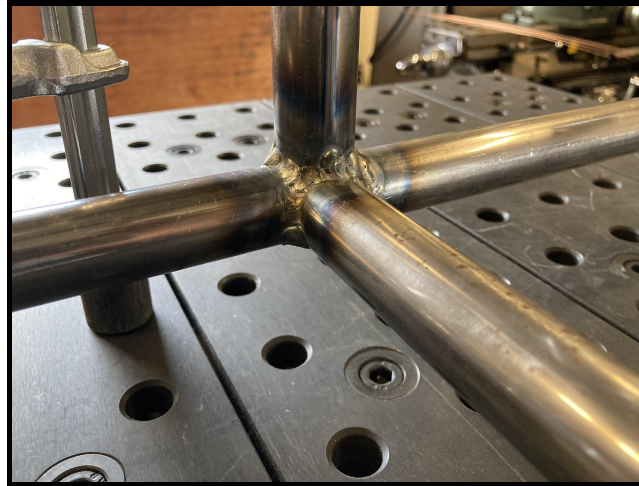


Figure 6: Completed weld up close

The other parts that we added to the chassis frame included the anti-intrusion plate, the impact attenuator, and the aluminum floor. The steel anti-intrusion plate was welded all around the perimeter of the front bulkhead. The foam impact attenuator was adhered to the anti-intrusion plate using 3M DP420 epoxy adhesive. Leftover steel from the anti-intrusion plate was cut into 1" by $\frac{3}{4}$ " tabs and were welded to the floor tubes of the chassis. The aluminum floor plates were then riveted to these tabs as seen in figure 7.

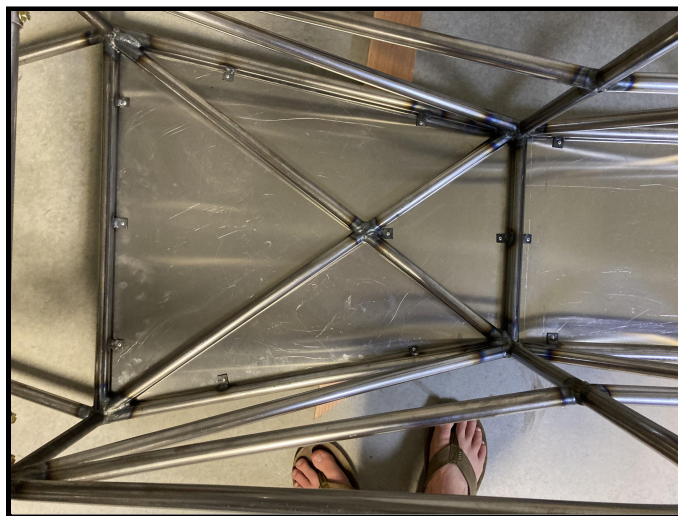


Figure 7: Completed aluminum floor panel

5. TESTING and EVALUATION of PERFORMANCE

Objectives

To test and evaluate the chassis we calculated the torsional rigidity and compared it to our value derived from FEA. To do this we set ourselves a list of objectives.

- Determine the torsional rigidity
- To measure the deflection of the chassis to within 0.01mm resolution
- To compare our values to the FEA values and determine the percent difference

Background

For some context on what the torsional rigidity actually means and what we can expect our value to be we will look at another FSAE team and the torsional rigidity of a go-kart. Go Karts are cited to have about 2000 Nm/deg while some FSAE teams calculate their torsional rigidity to be 600-700 Nm/degree^[3]. This number is important because it lets us know how much the chassis can potentially twist when a load is applied when in a turn or hitting curbs.

Equipment

To perform this test we need a way to measure deflection and to apply a load. To apply the load to the chassis we developed a metal bracket which can hold a variable amount of weights. To measure the deflection we used a set of calipers with a resolution of 0.01mm. To hold the chassis in place we used a jigging table and wedge clamps to ensure it would not move.

Experimental Method

We start off by mounting the weight holding device to a specific location to match the FAE as best as possible. Then we clamp the 3 points of the chassis that will not be loaded to the table using standard clamps. Once it is clamped we also need to set up a reference to measure to, from the chassis as shown in figure 8.

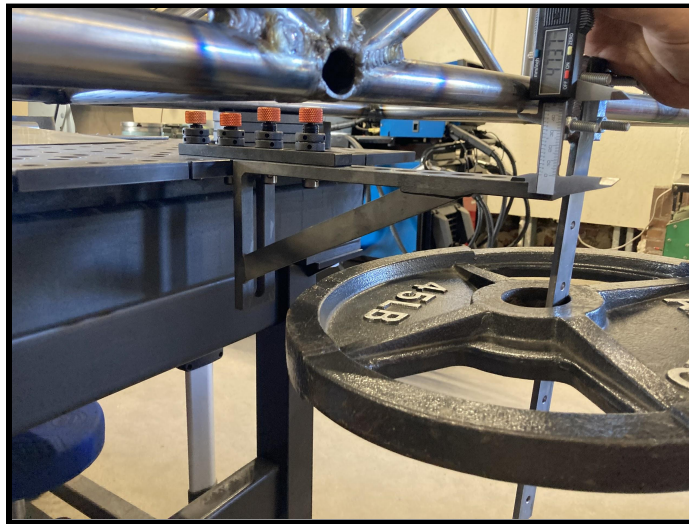


Figure 8: Deflection measuring setup

Then we can start loading the chassis and for each 45lb plate we take a measurement of the distance between the chassis and the reference point using the set of calipers. In this instance we only added up to 180lb because this gave us enough data and we did not want to permanently deform the chassis.

Test Conditions

We were able to test in an indoor facility on a flat and even reference table. The table and the environment allowed us to perform the test without any interferences or disturbances.

Results and Discussion

For our results we tabulated the data and determined the deflection by taking the difference between the measurements. With this deflection we were able to convert it to a deflection in degrees which gave us the torsional rigidity for each different load. We then took the average of this and compared it to our FEA. The results can be seen in table 1 where the torsional rigidities are averaged out based on each data point.

Table 1: Collected Data & Calculated Torsional Rigidity Values

Weight (lbs)	Measurement (mm)	Deflection (mm)	Torsional Rigidity Coefficient Nm/deg
0	47.78	0	n/a
50	47.37	0.41	3023
100	46.87	0.91	2724
145	46.61	1.17	3072
190	46.15	1.63	2890
		Average:	2928
		FEA Result:	4238
		Percent Difference:	31

This percent difference (31%) is acceptable to us when looking at the torsional rigidity in the context of other vehicles. We believe that the difference is due to our location of the weight mount since it was offset from the point at which the FEA was performed.

Conclusions

Our method involved having a controlled variable (the weight) and then measuring the deflection at a specific point repeatedly at different weights. This would then allow us to find the torsional rigidity which can then be compared to the calculation from the FEA. The chassis meets our requirements in terms of weight and strength, although not a perfect match to

our analysis we are still happy with the result. In the future we could apply the load in a similar location to the FEA for better results.

6. FAILURE MODE and EFFECTS ANALYSIS

Table 2: Failure Modes & Safety Concerns Scores

Failure Modes	Potential Impact	SEV (1-10)	Potential Causes	OCC (1-10)	Detection Modes	DET (1-10)	RPN (Risk Priority Number)
Joint Weld	If a weld cracks or a joint is weakened - performance of the vehicle will decrease - overall safety decreases	5	- Improper welding procedure - In a collision - Due to fatigue		- Visual crack - Welding X-ray's	3	30
Sleeve Butt Joint Failure	- performance of the vehicle will decrease - overall safety decreases	4	- Fatigue - Shear Stress - Bolt Failure		- Visual deformation	5	40
Risk analysis	Probability of cause	High, medium, low	Magnitude of the harm	High, medium, low	Possible solutions		RISK
Roll over	In a roll over the driver will be protected by the main hoop but any debris could hit the driver	low	broken bones, scrapes	medium	fully covered cockpit		medium
Collision	since the comp is at low speeds we expect minimal harm	low	whiplash	low	be prepared		low

Two potential failure modes include failures of a weld and of the sleeve butt joint. In terms of severity, we scored a weld failure as a 5 and the sleeve butt failure as a 4. We scored a weld failure as being more severe because it would reduce the performance and more importantly reduce the safety of the vehicle. Sleeve butt joint failure would still affect performance, but it poses a lower safety concern as the heat shield separates the sleeved tube from the cockpit.

For occurrence, both failure modes scored 2s, meaning they were unlikely to occur. We did not give them 1s as we did not test the structural integrity of either. However, we are confident that the welds will hold considering we had the guidance of a welding professor. We also have confidence in the sleeve butt joint because we used high grade bolts. If failure were to occur, it would likely come from a crash or due to excessive fatigue.

For detection, weld failure earned a score of 3 while sleeve butt failure earned a 5. Failure of a weld can be seen in advance in the form of cracking. X-rays may also be used to examine the internal structure of a weld. A failure of the sleeve butt joint

would be difficult to detect as shearing of the bolts would occur with little to no warning.

Although both had small risk priority numbers (weld failure 30, sleeve butt failure, 40), failure of the sleeve butt joint had a higher RPN due to the fact that it could harm performance and be difficult to detect.

To reduce the risk of weld failure, the gaps between the tubes will need to be as small as possible. Large gaps require more filler material which is not as strong as the base material. The occurrence of sleeve butt failure could also be reduced by using the highest grade possible bolts. Both of these changes would give occurrence scores of 1, and bring down the RPN of the weld failure and sleeve butt failure to 15 and 20 respectively. Fortunately these RPN values are very low, meaning that the risk of failure is also low.

7. SAFETY and LIABILITY

Two areas of concern we noted were the potential of a rollover or a collision. Fortunately, both of these would be unlikely to happen and there are design features that would mitigate the harm caused by either.

For a rollover to occur, the driver of the vehicle would have to be traveling at relatively high speeds and turn sharply. The competition course does not feature many high speed sections, let alone a section with a high speed corner. In the event that it did happen, the harness would keep the driver in the driver's seat, and the main hoop and front hoop would prevent the driver's head from hitting the ground.

A collision would also have a low probability of occurrence for multiple reasons. First, the course used in the competition is a cone circuit with very few walls. Second, as mentioned before the circuit does not feature many high speed sections, meaning the car will not be pushed to its limits. Third, there is only one car competing at a time, so a car to car collision will not happen. If the car were to be involved in a collision, the impact attenuator will absorb some of the impact energy and protect the driver.

In both cases, the driver would experience little to no harm due to the nature of the competition and the safety features in place. In the worst case scenario, the driver may experience some bruises and soreness.

8. SUMMARY and CONCLUSIONS

We can confirm that the production of the chassis can be considered a success. It looks professional and it will perform well when the rest of the car components are mated to it. Factoring in our modest budget, we are more than satisfied with how the final product turned out.

From our rigidity testing, we can conclude that the chassis is safe, can withstand the forces of driving, and that it will protect the driver in the case of an accident. Although unlikely to occur, the main hoop, front hoop, and impact attenuator offer great protection if the car were to experience a rollover or a collision.

We did not meet our goal of making the chassis less than 80 lbs, but we did get extremely close. Our final weight of the chassis was 81 lbs, just one pound over our goal. If we did not make the design so redundant for safety purposes, we could have easily met this weight goal. Nevertheless, we still consider our final weight a success.

Our final goal of designing our chassis to accommodate a six-foot tall driver was met. Jan is 6'1 and he had no issues fitting into the chassis. It may even be possible to fit drivers taller than 6'1, but this has yet to be confirmed. Overall, our design group views the production of the FSAE chassis as a great accomplishment.

9. RECOMMENDATIONS for FUTURE WORK

For future work our group would like to possibly reduce the weight of the chassis by using a different material such as chromoly steel although this is quite expensive and will be budget dependent. Another option in reducing the weight would be to use some Type C tubing as described by FSAE which provides a thinner tube thus reducing the weight while still providing structural stability. Also in the future we would like to reduce overall manufacturing time as this required approximately 100 man hours. To do this there is the option of outsourcing some of the labor or finding more efficient ways to notch the tubes. Finally our group would like to perform more rigorous testing provided more time.

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<https://doi.org/https://www.longdom.org/open-access/analysis-of-torsional-stiffness-of-the-frame-of-a-formula-student-vehicle.pdf>

APPENDICES

Appendix A: Final Engineering Analysis

Front Impact

Known: The chassis must be able to protect the driver in the case of an accident/impact. The chassis is made using 1010 hot-rolled steel tubing.

Find: We will be using FEA to see how the chassis responds to a front end impact of roughly $3.28 \times 10^6 \text{ N}$ (80 km/hr collision).

Theory:

$$\text{Total Force (TF)} = \frac{\text{Kinetic Energy}}{\text{Distance Traveled in Collision}} = \frac{\frac{1}{2}mv^2}{x} = \frac{\frac{1}{2} \cdot 1000 \text{ kg} \cdot \left(22.2 \frac{\text{m}}{\text{s}}\right)^2}{0.075 \text{ m}} = 3.285 \times 10^6 \text{ N}$$

Impact Attenuator (IA) @ 23°C and 50% compression can withstand 853kPa.

$$\text{Energy Absorbed (EA)} = P \cdot V = \left(853,000 \frac{\text{N}}{\text{m}^2}\right)(0.3556 \text{ m})(0.3048 \text{ m})(0.127 \text{ m})$$

$$\text{Energy Absorbed (EA)} = 11,742 \text{ Nm (J)}$$

$$\text{Force on Chassis (FC)} = \text{TF} - \frac{\text{EA}}{\text{IA compression}} = 3.285 \times 10^6 \text{ N} - 9.245 \times 10^5 \text{ N}$$

$$\text{Force on Chassis (FC)} = 2.361 \times 10^6 \text{ N}$$

$$\% \text{ Force on Chassis} = \frac{\text{FC}}{\text{TF}} (100) = 71.9\% ;$$

$$\text{IA deflection} = \text{FEA deflection} \cdot 0.719$$

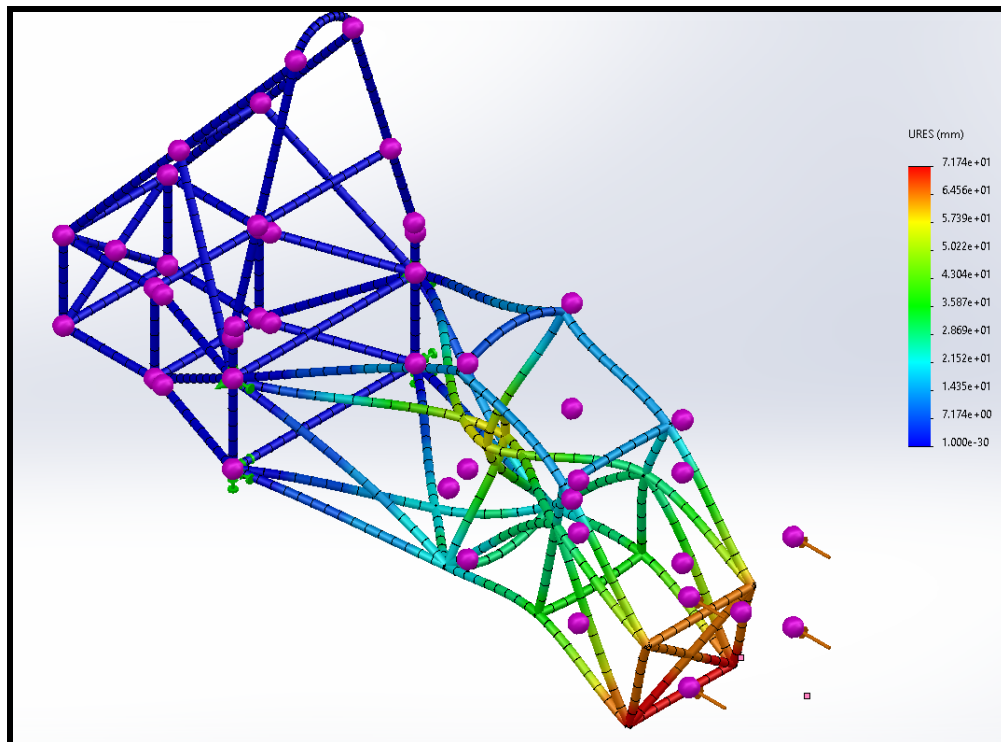
The impact attenuator reduces the force on the chassis by almost 30%. Because of this, we can estimate the deflection to be roughly 30% less all around when the impact attenuator is included.

Major Assumptions: We assumed the collision to be elastic and that the vehicle would be impacting a stationary wall. We also assumed a safety factor of 4 on the mass (originally 250kg). We assumed the speed at impact to be 80 km/hr. This is an overcompensation as the average speed around the circuit will be roughly 40 km/hr. In the FEA below, we did not include the impact attenuator (IA). We account for the IA in our calculations and assuming the impact force scales linearly with the deflection, we can estimate the deflection in a head on collision with an impact attenuator. For the FEA the force is distributed among four points on the front of the

chassis and every tube in the chassis is modeled as one continuous metal piece due to Solidworks limited abilities with tubes. In addition to this, when using the tube weldments feature the geometrically fixed points and the loaded points are only able to be at joints where the tubes meet. In reality the tubes will be welded together with ER70S-6 which has a tensile strength of 70 kpsi, more than double the tensile strength of the base metal. It is important to note that the distance traveled during the collision is assumed to be the same as the deflection. This required us to run multiple FEA and iterate because if we initially estimate the distance traveled (1 m) for the first point of impact until the vehicle is at rest (0 m/s) the impact energy would not be as great. This low impact energy would result in a small deflection, contradicting the distance traveled in the collision. Because of this, we had to decrease the distance traveled until it matched the deflection to get an accurate value for the maximum deflection.

Data: A mass of 1000 kilograms was used with a velocity of 80 km/hr and a distance of 0.075 m traveled during the collision due to the material (AISI 1010 Steel) failing and deforming.

Schematic, Plot, or Graphs:



Analysis: Based on the FEA it is clear that there is a need for an impact attenuator. The maximum deflection is around 71 mm which is within 6 percent of the distance traveled during the impact (75 mm). This is good. The next location of concern are

the sides of the front hoop which sit right next to the drivers upper legs. There is a deflection of about 50 mm on both sides and with an initial distance of 375 mm between them, there is only 275 mm left for the driver to get his/her lower body out of the cockpit (slightly less than 1 ft). Once again reiterating why we need the impact attenuator. With the impact attenuator we expect to see only 50 mm of deflection at the front of the vehicle and 35 mm of deflection at tubes on the side next to the driver's legs .

Comments: The results are close to what we expected and reinforces the reasoning for the FSAE organization requiring an impact attenuator. We tried to fully model the impact attenuator, but defining its material properties with the specific type from the manufacturer proved too difficult. We settled on the theoretical approach and derived the deflection through our equations and specs from the manufacturer of the impact attenuator.

Torsional rigidity

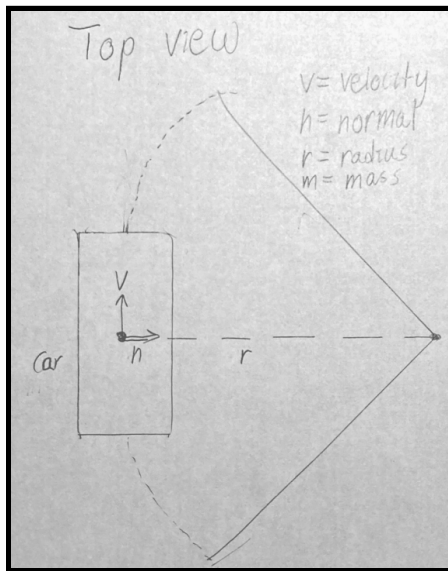
Known: The chassis has to be able to withstand the forces of driving (about 1.5G through turning). The chassis is made using 1010 hot-rolled steel tubing and we know the dimensions of the chassis. The goal for the torsional rigidity is 600 Nm/deg.¹

Find: We will be using FEA to determine the torsional rigidity and to see how the chassis responds to the forces experienced when turning. This analysis will give the torsional rigidity of the frame in degrees in the X, Y, and Z axis. With this information we can calculate the expected deflection when the chassis is experiencing the maximum load during cornering.

Theory: First, we must calculate the torsional rigidity using our FEA. The load is an arbitrary value and it is applied in the vertical direction on one of the 4 mounting points for the suspension (in our case the front right mount). The load is applied in the vertical direction because when testing a chassis in real life a vertical load is applied.¹ Another constraint is that the 3 remaining mounting points for the suspension are

fixed just as they are in a real world torsional rigidity test. Running the FEA gives us the deflection (deg) in each axis and the load (N) and distance (m) allows us to calculate the torsional rigidity in each axis.

$$C_T = \frac{\tau}{\alpha} = \frac{F \cdot d \text{ (Nm)}}{\alpha \text{ (deg)}}$$



With the torsional rigidity in each axis we can determine the deflection during max loading. To determine the max load from driving we will only be considering forces in the horizontal plane (XZ) since the maximum loads during turning occur in the horizontal direction due to centripetal acceleration.

$$\text{acceleration in the normal direction} = a_n = \frac{v^2}{r} = \frac{(6.6 \frac{m}{s})^2}{3m}$$

¹ Krzikalla, D., Mesicek, J., Petru, J., Silva, A., & Smirau, J. (2019). Analysis of Torsional Stiffness of the Frame of a Formula Student Vehicle. *Journal of Applied Mechanical Engineering*, 8(1), 1–5. <https://doi.org/https://www.longdom.org/open-access/analysis-of-torsional-stiffness-of-the-frame-of-a-formula-student-vehicle.pdf>

$$Force = m \cdot a_n = 300kg \cdot \frac{(6.6 \frac{m}{s})^2}{3m} = 4356 N$$

To determine the maximum deflection we will find the maximum torque by taking the maximum load and multiply it by its moment arm. Now we can divide our max torque by the torsional rigidity in that particular axis.

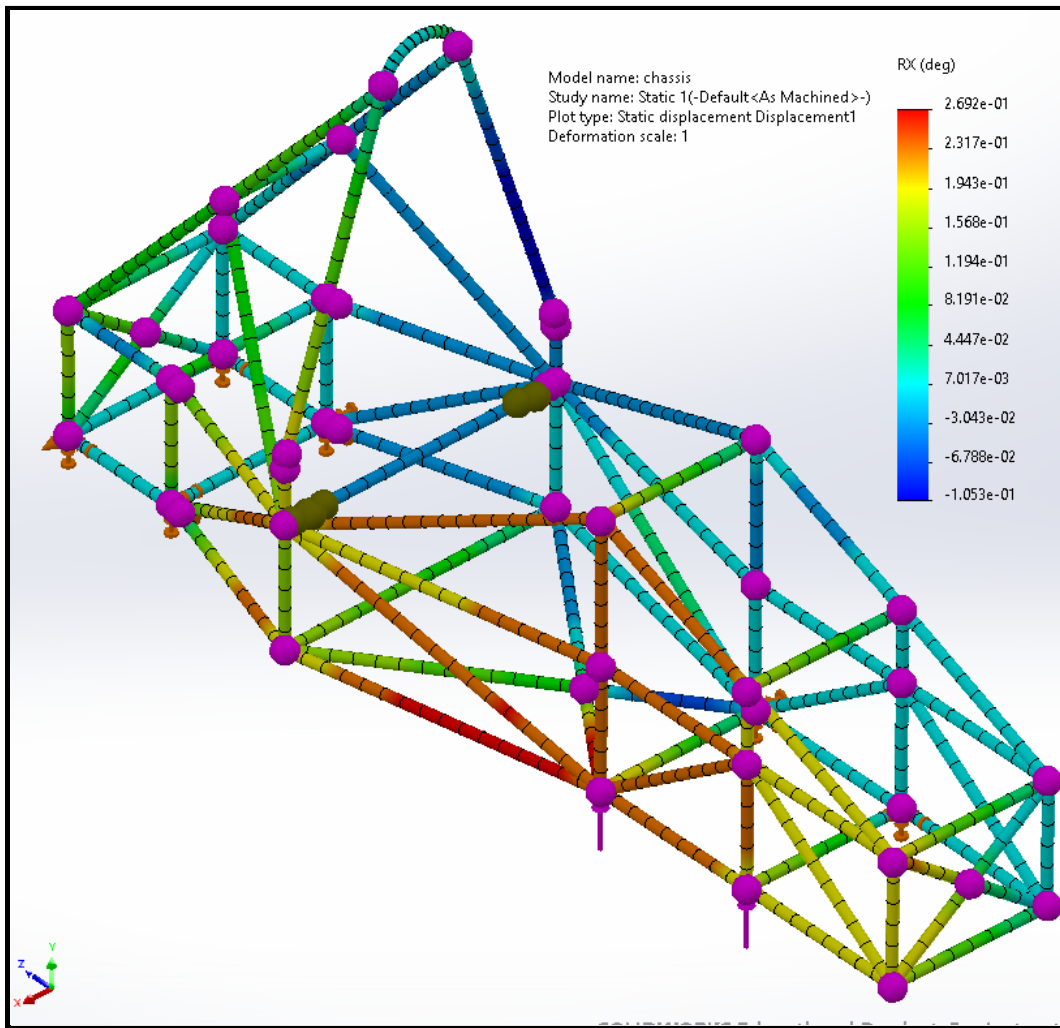
$$\alpha = \frac{\tau}{C_r} = \frac{F \cdot d (Nm)}{C_r (Nm/deg)} = \frac{4356 N \cdot 0.23 m}{4237.5 (Nm/deg)} = 0.236 deg$$

Major Assumptions: We are assuming that the tires do not slip. The tube weldments feature in Solidworks only allows for the tube joints to be given conditions. This means that the joints of the 3 suspension mounts that are not being loaded are fixed. An arbitrary load of 1500 N is then applied at each of the 2 joints where the front right A-arms of the suspension meet the chassis. We placed the loads on the bottom 2 joints even though there are 4 hardpoints at each corner of the vehicle for the A-arms to mount to. This is because each set of 2 hard points share the same tube and are on the tubes themselves (not the joint) which can not be modeled due to the limitations of the weldments feature in Solidworks. To calculate the maximum deflection we used the calculated maximum force (4356 N) multiplied by the moment arm of 0.23 m. The moment arm is in the Y axis and represents the distance between the lower A-arm and the mounting point of the shock obtained from the FSAE car model. This is because it is assumed the load is transferred through the shock.

Data: To solve this problem we will use a velocity of 6.6 m/s and a turn radius of 3 m for the tightest turn on the course. We use 6.6 m/s because this is the approximate speed needed to travel in a 3m radius turn to experience 1.5G.² Most sports cars will not pull more than 1.2G, but our vehicle is optimized for this and many teams cite 1.5G as a max. The mass is determined from the FSAE vehicle model which accounts for all of the components including the driversmass and gives a mass close to 300kg. To calculate the torsional rigidity we used the data of our chassis dimensions and the deflection in each axis given by the FEA.

² Bartolomeo, M. V., Lombardo, A., Colella, M., & Delagrammatikas, G. J. (2019). (tech.). *Measuring the Traction Limits and Suspension Forces of a Formula SAE Racecar* (p. 25). American Society for Engineering Education.

Schematic, Plot, or Graphs:



In Solid works FEA
Material: AISI 1010 steel
 $F = 3000\text{ N}$
Applied to mounting point
of front right suspension

$$C_T = \frac{T}{\alpha} \quad \frac{\text{N}\cdot\text{m}}{\text{deg}}$$

$$T = F \cdot d$$

$$C_{Tx} = \frac{T_x}{\alpha_x} = \frac{F \cdot d_x}{\alpha_x} = \frac{3000\text{ N} \cdot 0.4\text{ m}}{0.269^\circ}$$

$$C_{Tx} = 4460.9 \frac{\text{Nm}}{\text{deg}}$$

$$C_{Ty} = \frac{T_y}{\alpha_y} = \frac{F \cdot d_y}{\alpha_y} = \frac{3000\text{ N} \cdot 0.565\text{ m}}{0.400^\circ}$$

$$C_{Ty} = 4237.5 \frac{\text{Nm}}{\text{deg}}$$

$$C_{Tz} = \frac{T_z}{\alpha_z} = \frac{F \cdot d_z}{\alpha_z} = \frac{3000\text{ N} \cdot 2.1\text{ m}}{1.069^\circ}$$

$$C_{Tz} = 5893.3 \frac{\text{Nm}}{\text{deg}}$$

$d_x = 0.4\text{ m}$
 $\alpha_x = 0.269^\circ$

$d_y = 0.565\text{ m}$
 $\alpha_y = 0.400^\circ$

$d_z = 2.1\text{ m}$
 $\alpha_z = 1.069^\circ$

Analysis: Based on the theory the max deflection would be approximately 0.236 degrees on the side of the chassis experiencing loading. From this we can conclude that the chassis will be strong enough to withstand the forces experienced in the turns. The chassis itself experiences very little deflection, resulting in a positive effect on the overall handling of the car.

Comments: With this small of a deflection we should be able to withstand higher loads. We plan on validating the torsional rigidity with a strain gauge test in the real world. It is important to note that we far exceeded our goals when compared to other FSAE teams' chassis. However most teams included their suspension in the analysis which significantly lowers the torsional rigidity and explains the difference between our numbers. We are still satisfied with our results from an analytical standpoint because we do not expect to have large deflections. This is the most important metric when analysing the handling characteristics of a vehicle.

Item	(Part #)	Cost Per Unit	Quantity	Shipping	Tax	total	Seller	Link	Priority	Explanation
Square Tube 6ft U-Bolt 1" Steel tube 1.25" 1ft (6.5" Chas) Tube size A 6ft Tube size B 6ft Tube size A 10ft Small steel plate	6527K174	\$17.95	8	\$0.00	\$0.00	\$0.00	\$143.84 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the jlg
	3043T649	\$0.99	12	\$0.00	\$0.00	\$0.00	\$11.88 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the jlg
	7767T748	\$7.36	1	\$0.00	\$0.00	\$0.00	\$7.36 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the jlg
	7767T739	\$16.25	3	\$0.00	\$0.00	\$0.00	\$48.75 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
	7767T7231	\$11.93	23	\$0.00	\$0.00	\$0.00	\$274.39 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
	7767T7999	\$32.50	1	\$76.86	\$0.00	\$0.00	\$109.36 McMaster-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
	204225705	\$7.95	4	\$0.00	\$0.00	\$0.00	\$31.92 Home Depot	https://www.homedepot.com/p/Everbilt-6-in-x-18-in-Steel-Plate-4-Pack/304225705	Primary	Part of the jlg
	N/A	\$156.20	1	\$0.00	\$0.00	\$0.00	\$156.20 Amazon	https://www.amazon.com/3M-Scotch-Weld-9213	Secondary	Part of the chassis
	6544K14	\$26.38	1	\$0.00	\$0.00	\$0.00	\$26.38 McMaster-CARR	https://www.mcmaster.com/6544K14/	Secondary	Part of the chassis
	94036A529	\$8.86	1	\$0.00	\$0.00	\$0.00	\$8.86 McMaster-CARR	https://www.mcmaster.com/94036A529	Secondary	Part of the chassis
Aluminum plate 1.18" rivets Impact Attenuator 8mm locknuts	89015K53	\$157.35	1	\$0.00	\$0.00	\$0.00	\$157.35 McMaster-CARR	https://www.mcmaster.com/89015K53	Secondary	Part of the chassis
	97447A015	\$7.88	1	\$0.00	\$0.00	\$0.00	\$7.88 McMaster-CARR	https://www.mcmaster.com/97447A015/	Secondary	Part of the chassis
	N/A	\$170.00	1	\$0.00	\$0.00	\$0.00	\$170.00 BSCI	https://www.rollbarpadding.com/product/ind-48	Secondary	Part of the chassis
	97131A140	\$7.86	1	\$0.00	\$0.00	\$0.00	\$7.86 McMaster-CARR	https://www.mcmaster.com/97131A140/	Secondary	Part of the chassis
							\$1,162.03			
NOTE: Tube size A 10ft is not typically offered. Reference quote 68646. To reference enter the part number 7767T999 and a description box should appear. Enter the quote #										
The dimensions of size A tube are 1" OD x 0.095" wall and the value per unit and shipping was quoted to Jan. You should have an email of it.										
							Grand Total \$595.58			

Appendix C: Gantt Charts

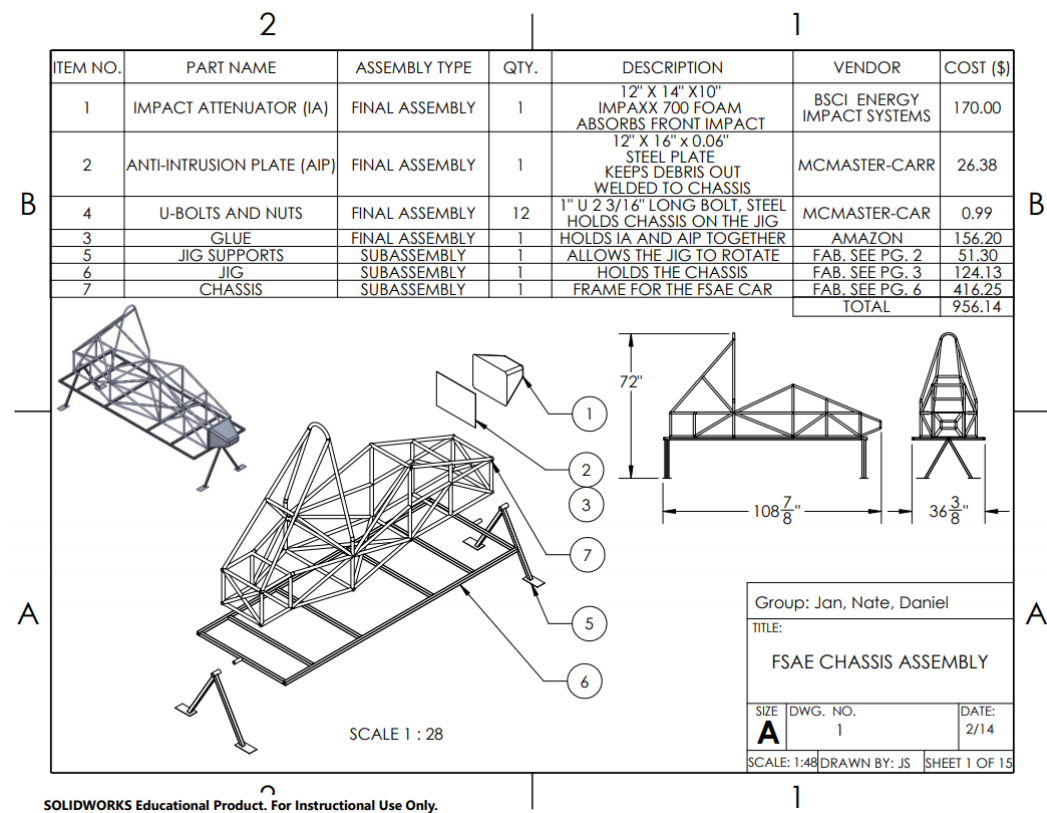
TASK NAME	START DATE	END DATE	DURATION (WORK DAYS)	TEAM MEMBER	PERCENT COMPLETE
Chassis Final Design					
Chassis 2nd drawing	1/18	1/22	5	Jan	100%
Perform Final Finite Element Analysis	2/10	2/20	10	Nate	100%
Chassis and Jig Construction Prep					
Cut tube	1/21	2/5	14	D/J/N	100%
Grind tube	1/28	4/18	80	D/J/N	80%
Check tube fitment	2/1	4/18	77	D/J/N	60%
Practice welds	2/5	2/11	6	Jan	100%
Order AIP, IA, glue, aluminium	1/30	2/5	5	Daniel	100%
SES					
Update graphics	1/18	1/20	2	Daniel	100%
Input sleeved butt joint	1/20	1/21	1	Nate	100%
Chassis and Jig Construction					
Weld jig together	2/22	3/2	10	Jan	100%
Assemble jig	3/2	3/2	0	D/J/N	100%
Weld chassis using jig	3/2	4/18	46	Jan	60%
Assemble chassis, AIP, and IA	3/15	4/18	6	D/J/N	60%
Test Chassis					
Test chassis rigidity using strain gauges	3/21	4/24	33	D/J/N	10%
Measure weight	3/30	4/24	1	D/J/N	0%
Final Submittals					
Final drawing submittal	1/26	3/3	6	D/J/N	100%
Final analysis submittal	4/1	4/7	6	D/J/N	10%
Design Presentation					
Senior Project Presentation	4/7	5/1	24	D/J/N	0%
FSAE Design Presentation	2/22	4/26	64	D/J/N	100%

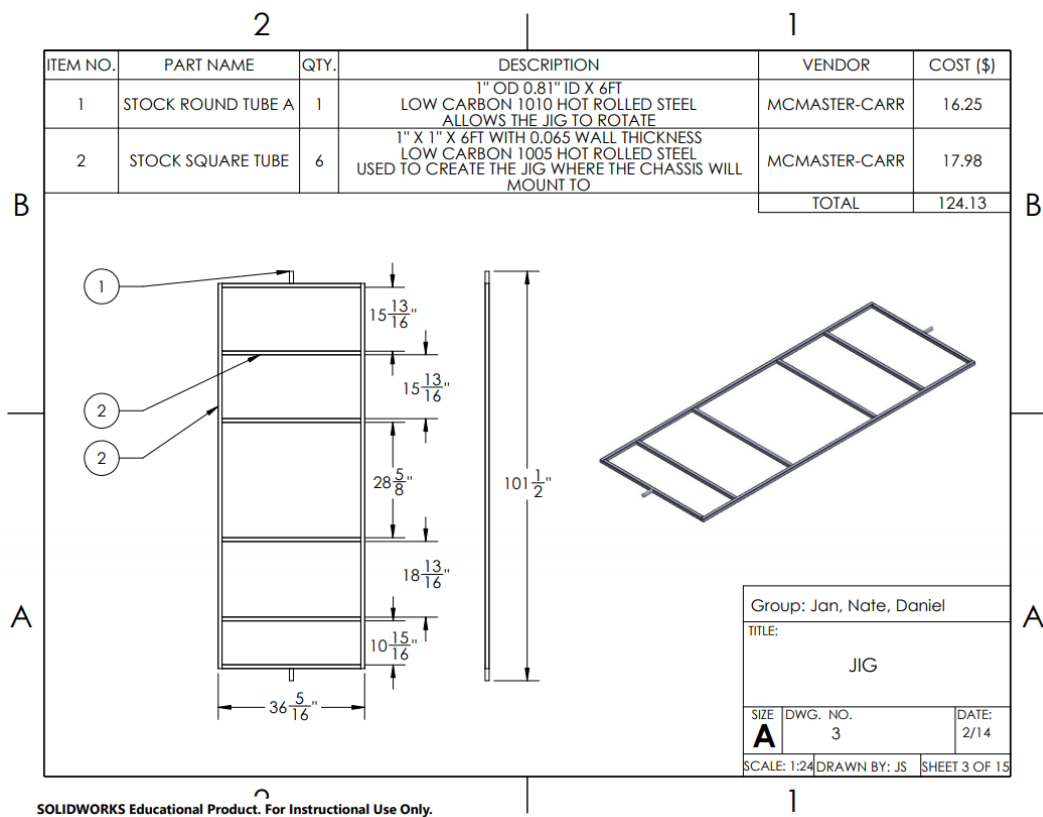
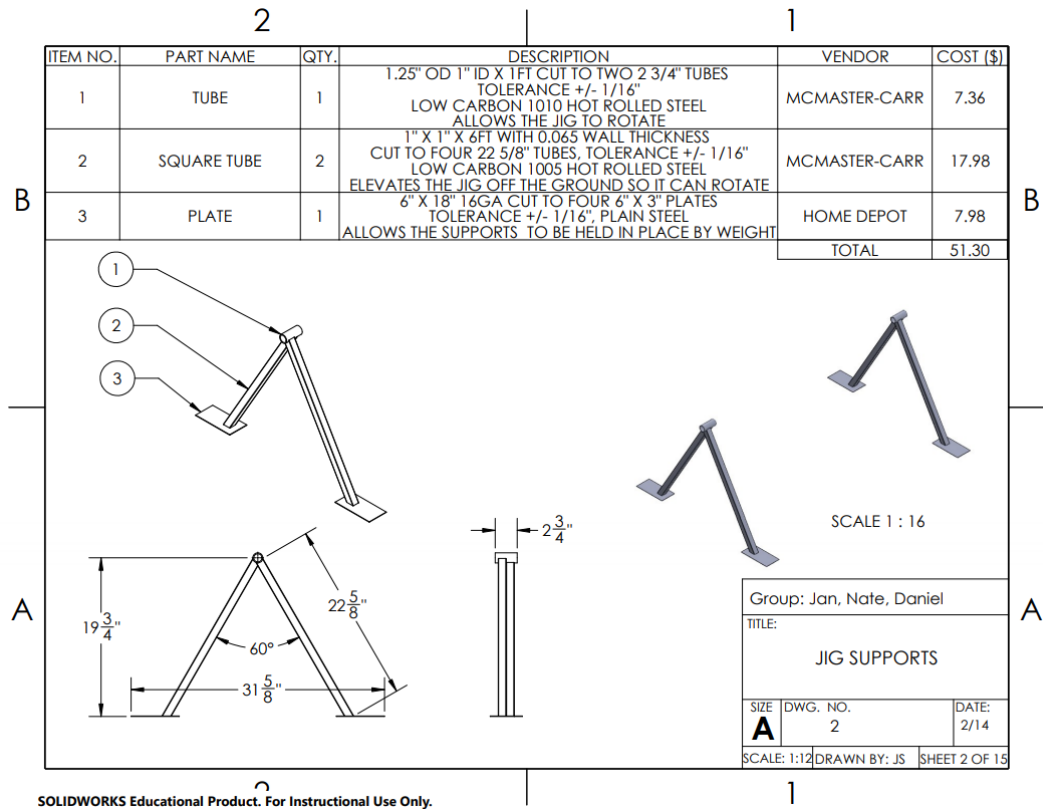
Appendix D: Final Engineering Drawings

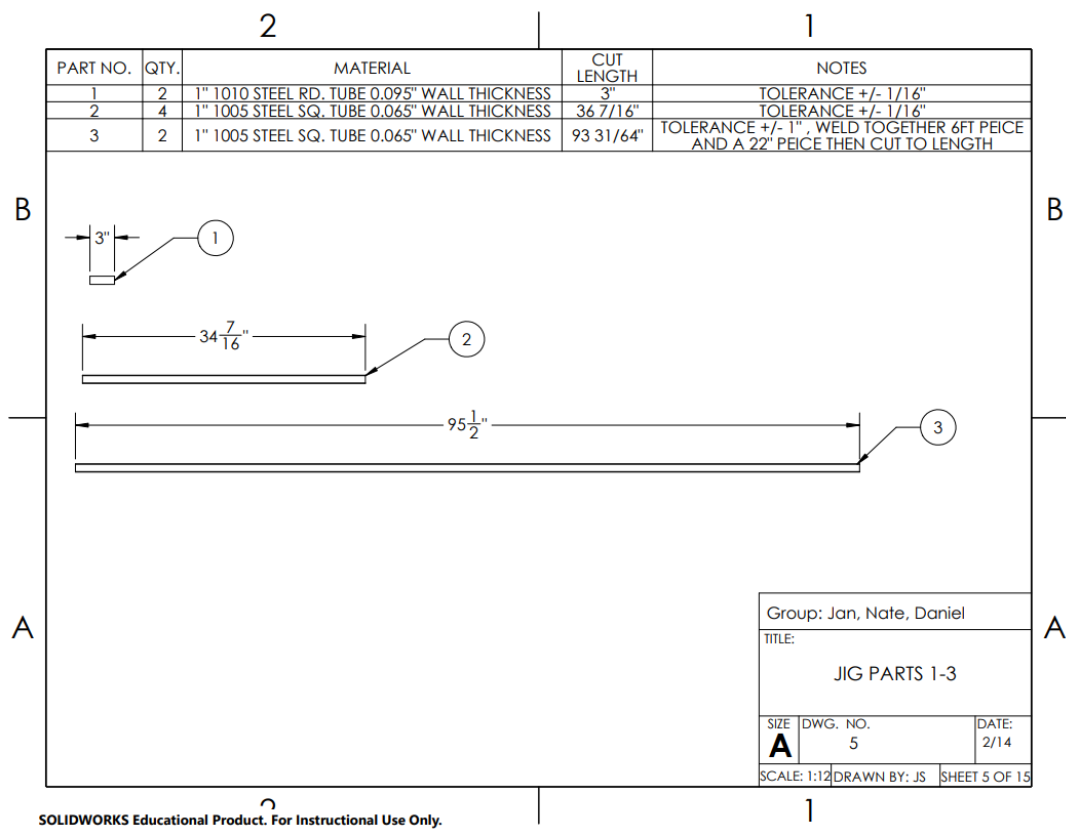
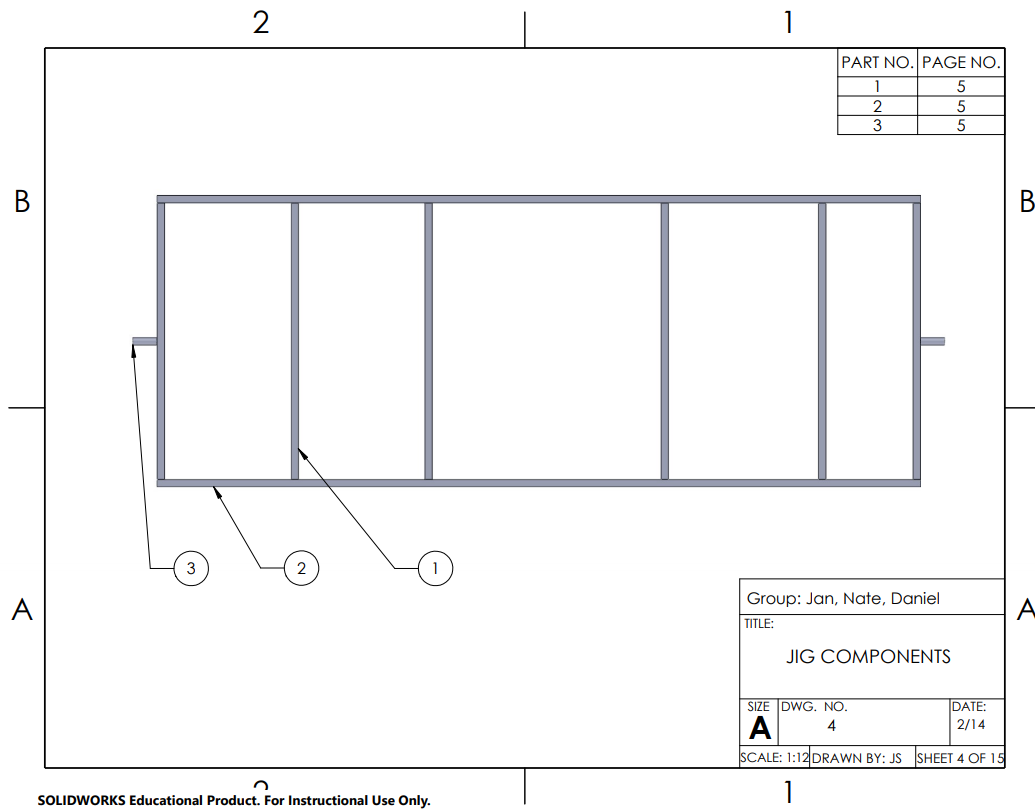
TABLE OF CONTENTS	
PAGE/SHEET NO.	DISCRIPTION
1	COMPLETE ASSEMBLY OF ALL OF THE COMPONENTS FOR THE FSEA CHASSIS
2	DETAILS FOR THE FABRICATION OF THE JIG SUPPORTS
3	DETAILS FOR THE FABRICATION OF THE JIG
4	DIAGRAM DETAILING THE CUT LIST FOR THE JIG COMPONENTS
5	CUT LIST FOR ALL OF THE JIG PARTS
6	DETAILS FOR THE FABRICATION OF THE CHASSIS
7	DIAGRAM DETAILING THE CUT LIST FOR THE CHASSIS COMPONENTS
8	CUT LIST FOR PARTS 1-6 OF THE CHASSIS
9	CUT LIST FOR PARTS 7-9 OF THE CHASSIS
10	CUT LIST FOR PARTS 10-13 OF THE CHASSIS
11	CUT LIST FOR PARTS 14-16 OF THE CHASSIS
12	CUT LIST FOR PARTS 17-19 OF THE CHASSIS
13	CUT LIST FOR PARTS 20-22 OF THE CHASSIS
14	CUT LIST FOR PARTS 23-26 OF THE CHASSIS
15	BILL OF ALL RAW MATERIALS FROM A VENDOR

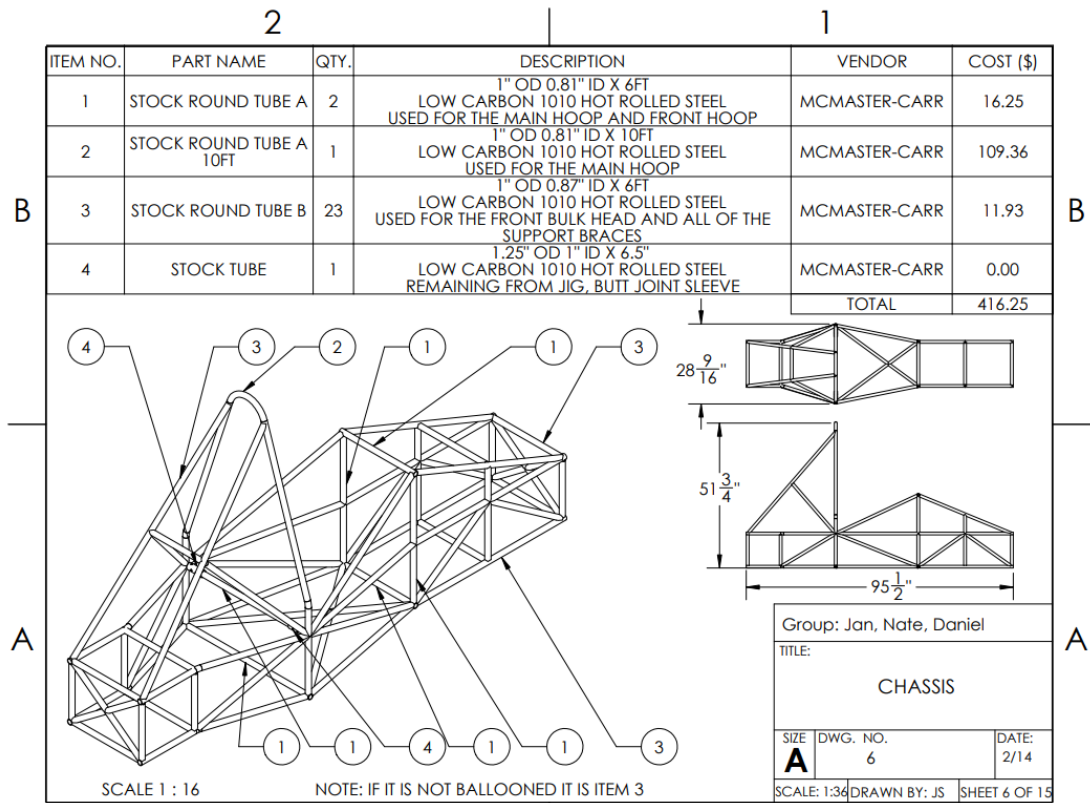
Group: Jan, Nate, Daniel
 TITLE:
 TABLE OF CONTENTS
 SIZE **A** DWG. NO. 0 DATE: 2/14
 SCALE: 1:1 DRAWN BY: JS SHEET 0 OF 15

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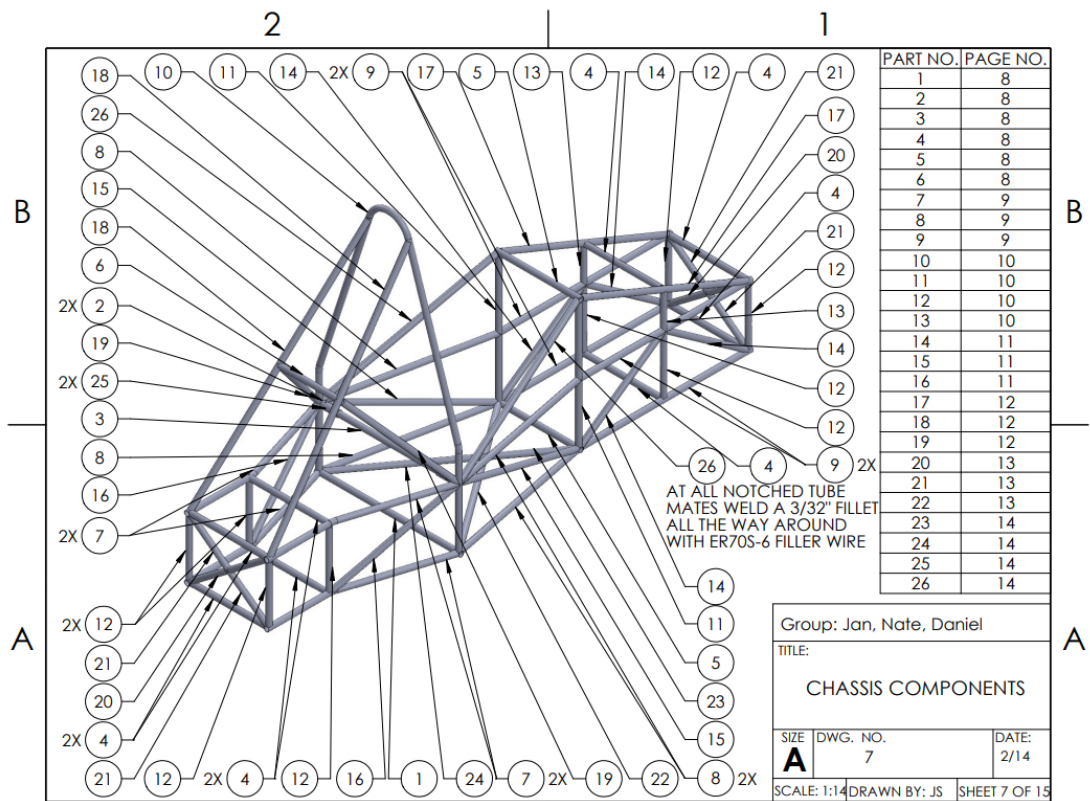




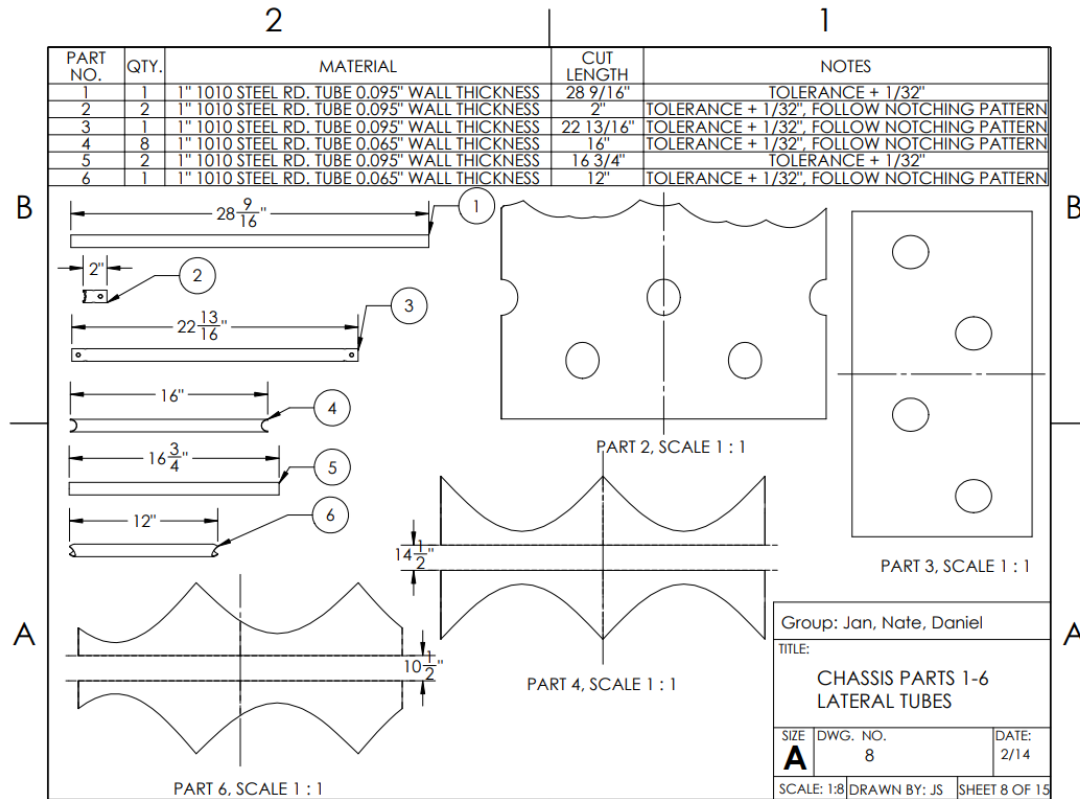




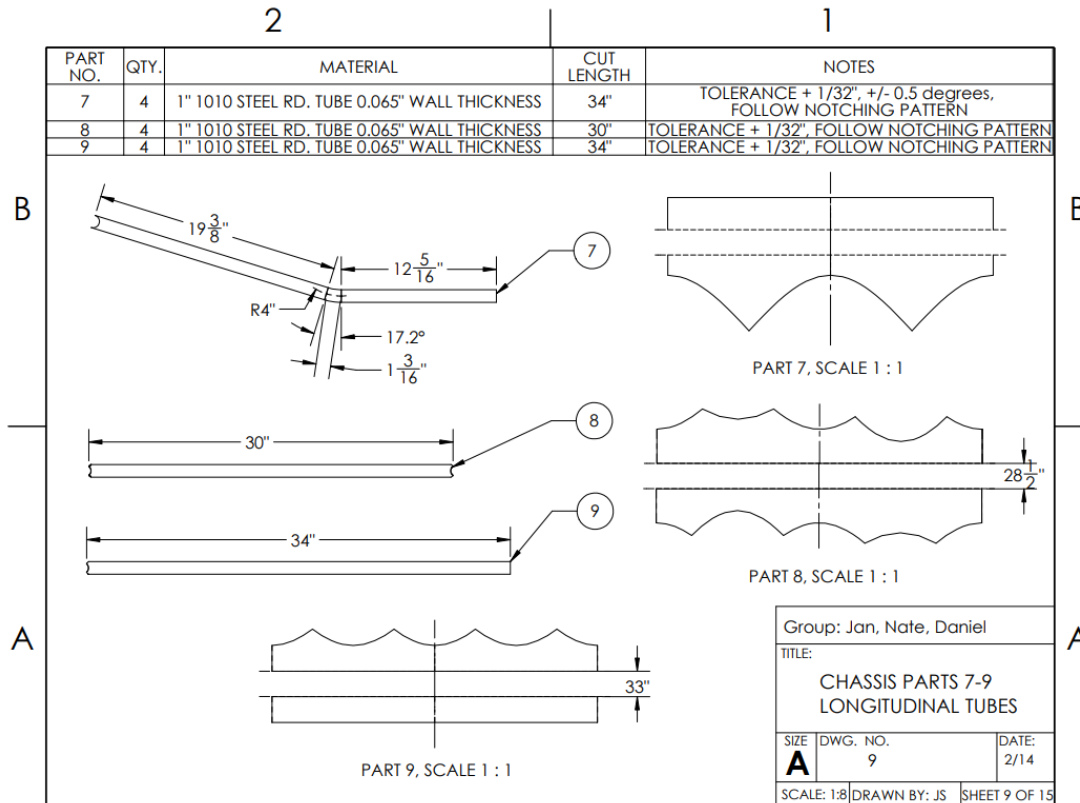
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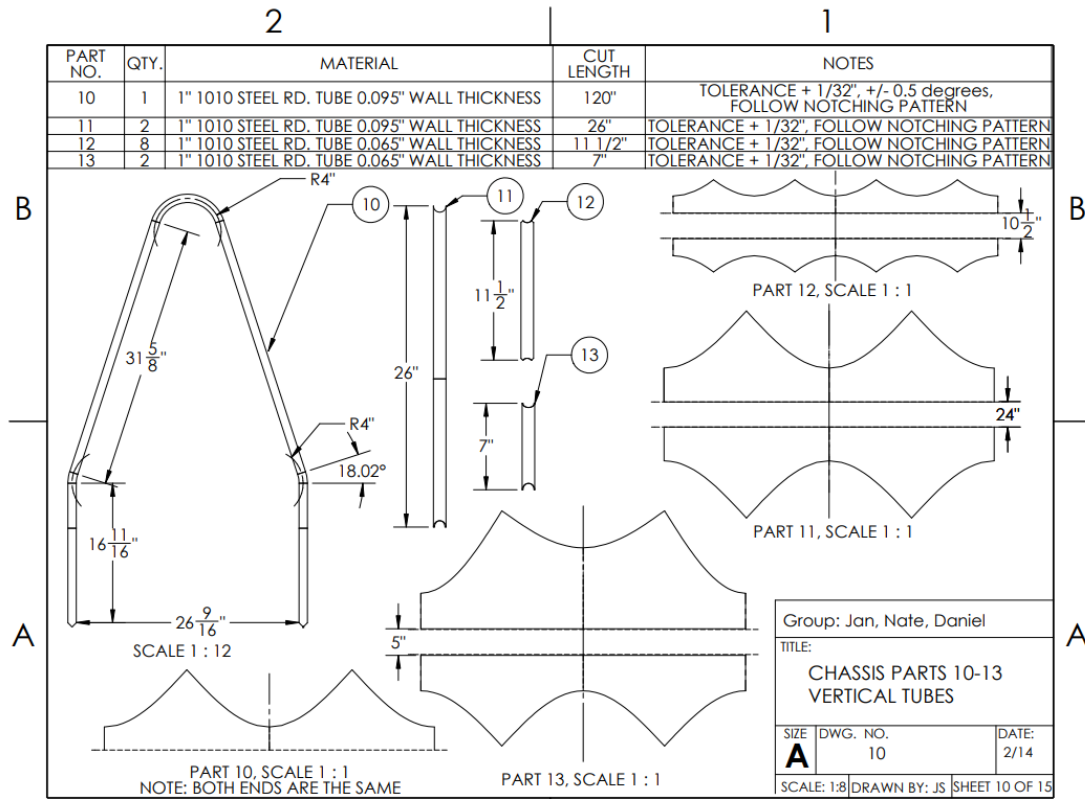
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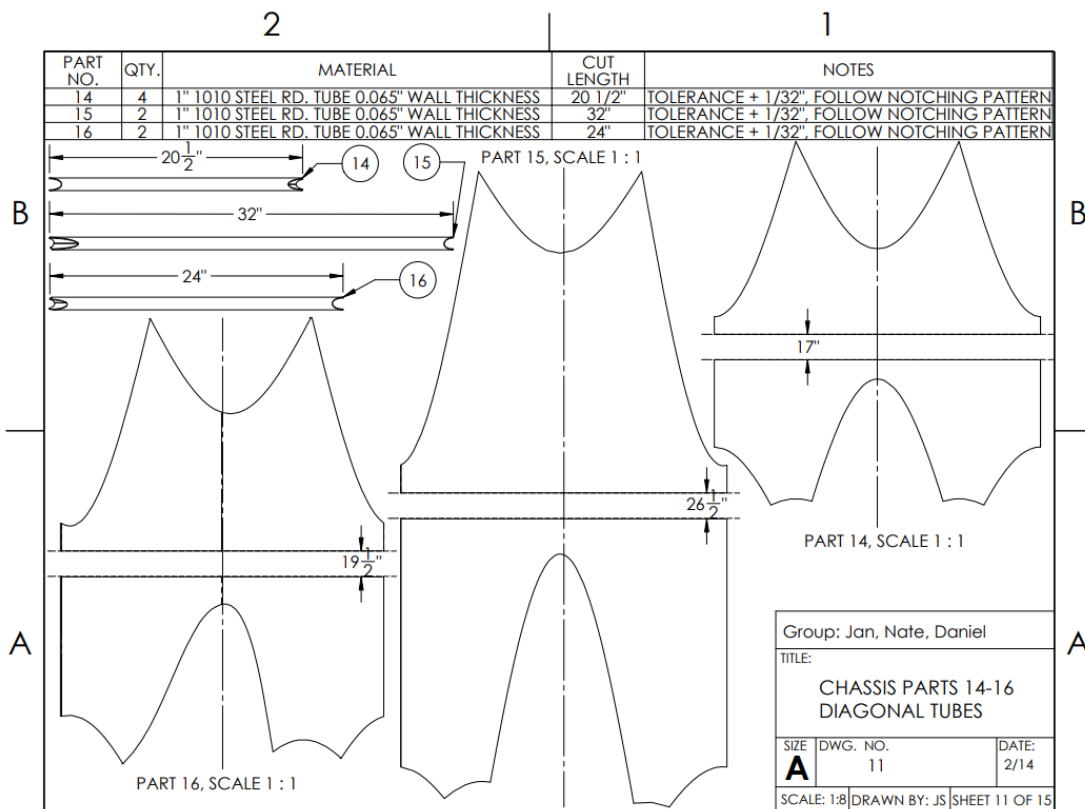
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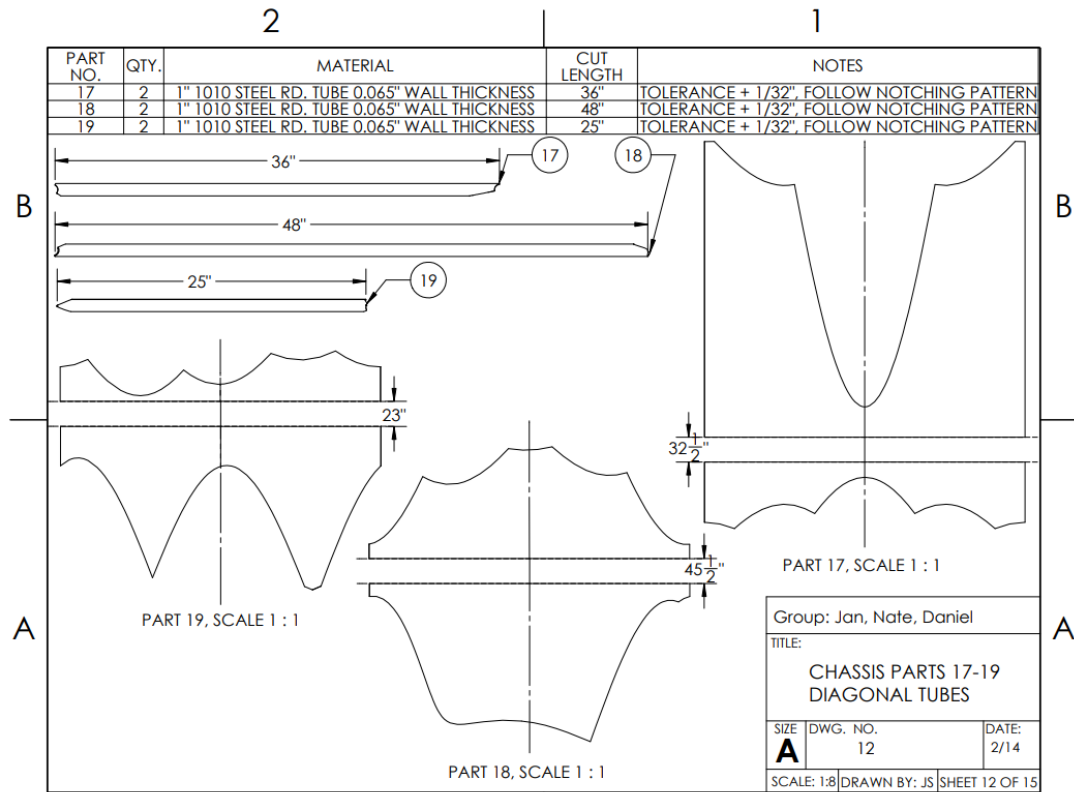
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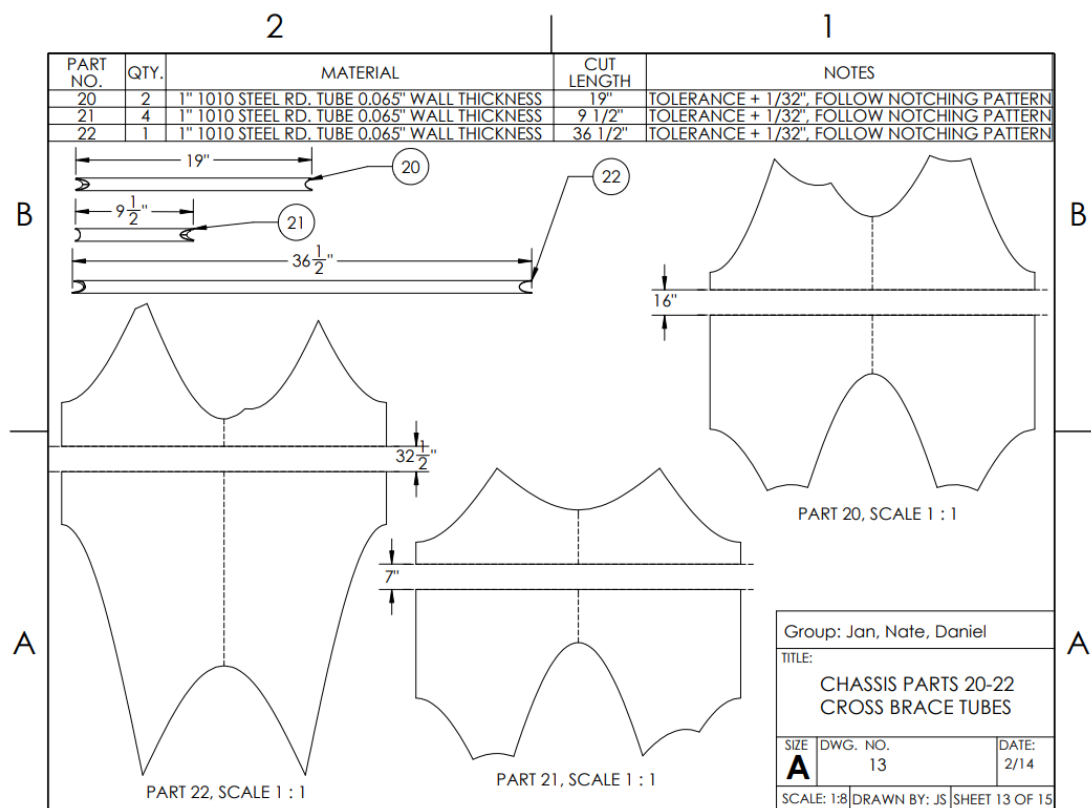
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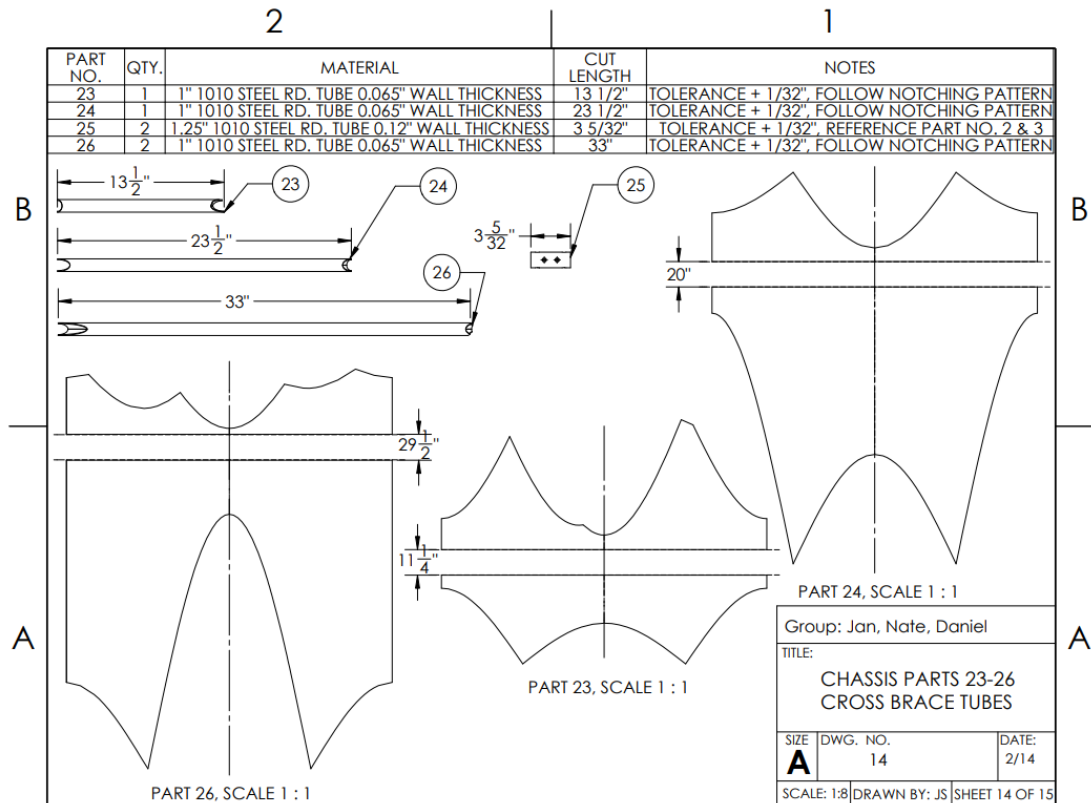
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2 1

ITEM NO.	PART NAME	QTY.	DESCRIPTION	VENDOR	COST (\$)
1	IMPACT ATTENUATOR (IA)	1	12" X 14" X 10", IMPAXX 700 FOAM ABSORBS FRONT IMPACT	BSCI ENERGY IMPACT SYSTEMS	170.00
2	ANTI-INTRUSION PLATE (AIP)	1	12" X 16" X 0.06", STEEL PLATE KEEPS DEBRIS OUT, WELDED TO CHASSIS	MCMaster-CARR	26.38
3	U-BOLTS AND NUTS	12	1" U 2 3/16" LONG BOLT, STEEL HOLDS CHASSIS ON THE JIG	MCMaster-CARR	0.99
4	GLUE	1	HOLDS IA AND AIP TOGETHER	AMAZON	156.20
5	TUBE	1	1.25" OD 1" ID X 1FT CUT TO 2 3" TUBES TOLERANCE +/- 1/16" LOW CARBON 1010 HOT ROLLED STEEL ALLOWS THE JIG TO ROTATE	MCMaster-CARR	7.36
6	SQUARE TUBE	8	1" X 1" X 6FT WITH 0.065 WALL THICKNESS CUT TO 4 22 9/16" TUBES, TOLERANCE +/- 1/16" LOW CARBON 1005 HOT ROLLED STEEL ELEVATES THE JIG OFF THE GROUND SO IT CAN ROTATE	MCMaster-CARR	17.98
7	PLATE	1	6" X 18" 16GA CUT TO 4 6" X 3" PLATES TOLERANCE +/- 1/16", PLAIN STEEL ALLOWS THE SUPPORTS TO BE HELD IN PLACE BY WEIGHT	HOME DEPOT	7.98
8	ROUND TUBE A	3	1" OD 0.81" ID X 6FT LOW CARBON 1010 HOT ROLLED STEEL ALLOWS THE JIG TO ROTATE	MCMaster-CARR	16.25
9	ROUND TUBE A 10FT	1	1" OD 0.81" ID X 10FT LOW CARBON 1010 HOT ROLLED STEEL USED FOR THE MAIN HOOP	MCMaster-CARR	109.36
10	ROUND TUBE B	23	1" OD 0.87" ID X 6FT LOW CARBON 1010 HOT ROLLED STEEL USED FOR THE FRONT BULK HEAD AND ALL OF THE SUPPORT BRACES	MCMaster-CARR	11.93
TOTAL					956.14

Group: Jan, Nate, Daniel

TITLE:

BILL OF MATERIALS

SIZE DWG. NO. DATE
A 15 2/14

SCALE: 1:1 DRAWN BY: JS SHEET 15 OF 15

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I am creating this update 4 years later because during one of my recent interviews I was pushed for a more thorough explanation of the torsional rigidity and what its characteristics are. I did not have a great explanation and this prompted me to review this project in depth and I noticed the units for torsional rigidity did not match. I reviewed further and rethought through the analysis of a head on collision. It was on the right track, but the original analysis was incomplete and misleading. I will correct these errors in this paper so I can understand the solution.

Looking at torsional rigidity, what are its units and where do they come from?

$$\text{Torsional Rigidity} = G \cdot J \quad (N \cdot m^2)$$

Where G is the shear modulus (N/m^2) and J is the torsional constant (m^4). G is an inherent property of the material, the chassis is 1010 steel, so $8 \times 10^{10} (N/m^2)$. J is a property of the geometric structure and it is different depending on the cross section (*1). So for a chassis an equivalent torsional constant can be defined (*2). In this analysis I will determine the torsional constant J_{eq} . This is also where torsional stiffness comes into play k_t . Note k_t has to be in radians to continue.

$$k_t = T/\theta \quad (N \cdot m/deg) \quad \text{multiply by } 57.2958 \text{ to get to radian}$$

$$(G \cdot J)_{eq} = k_{t_{rad}} \cdot L = T \cdot L / \theta \quad (N \cdot m^2)$$

$$J_{eq} = T \cdot L / (G \cdot \theta) \quad (m^4)$$

T being a torque (force times distance) and L being the length, or span of the twist. Now that I have both the torsional rigidity and torsional stiffness defined, I will reference the original paper. I did not mention torsional stiffness ever, but I did call it torsional rigidity and denoted it as C_t . I also did not specify what torsional rigidity was at any point. I will go through the analysis from the beginning and clarify any of these issues, and re-calculate the results.

*1 Not the same as polar moment of inertia using the Saint-Venant definition (property of a cross-section that quantifies its resistance to twisting). The polar moment of inertia only is for circular cross sections.

*2 The only other ways to calculate J are through FEA, which usually uses this method also or through large assumptions, like thin-walled cell theory.

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In the paper the point of the analysis was to determine what the max expected forces were from the suspension in the most extreme cases (cornering). Then through Finite Element Analysis (FEA) I determine the max deflection of the chassis. This gave a theoretical value for torsional stiffness. The assumption was that max deformation occurred during cornering due to the load from the suspension and chassis interfaces. To continue this analysis I will rederive the numbers.

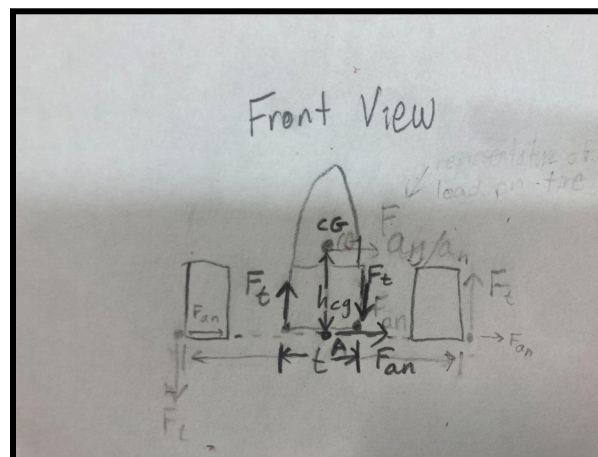
To determine the max force expected in cornering I will use the max acceleration times the mass (385kg or 849lbs) of the car. The max amount of acceleration comes from the tires coefficient of friction, estimating 1.1 for the tires used. The acceleration would therefore be $a_n = 1.1G$.

This is the equation for the maximum lateral force a tire can generate

$$F_n = \mu \cdot W \text{ (N) and because } F = m \cdot a$$

$$\text{The limit of cornering occurs when } a_n = \mu \cdot g \text{ (m/s}^2\text{)}$$

The lateral acceleration produces a tire force toward the center of the corner. This force acts in the plane of the road, not vertically on the chassis. What matters for chassis torsion is the roll moment, generated because the center of gravity is above the tire contact patch. That roll moment is resisted by equal-and-opposite vertical forces at the left and right suspension mounts. These vertical forces create the twisting load on the chassis. To estimate them, I will calculate the roll moment and distribute it to the wheel corners using a sum of moments.



$$\Sigma M_A = h_{cg} \cdot F_{an} + 2F_t \cdot 0.5t = 0$$

$$h_{cg} = \text{center of gravity height} = 0.39 \text{ m}$$

$$F_{an} = \text{normal force on tries} = a_n \cdot m = 1.1 \cdot 9.81 \cdot 385 = 4154 \text{ N}$$

$$t = \text{chassis width} = 0.40 \text{ m}$$

As a result $F_t = 4050 \text{ N}$ rotating in the opposite direction of F_{an}

Unfortunately since I used the weldments feature and defined all tube interfaces in Solidworks, no other CAD program is able to run a mesh on the body. Other CAD programs consider the tubing as individual pieces. I do not have Solidworks and I have tried converting the file format but it does not work. For now I will just explain my next steps.

Continuing with the theoretical analysis, I would use F_t as the load on the suspension. I would redo the FEA on the chassis, using the front right and left suspension hardpoints (connection between chassis and suspension) as the loading points. It is not the same as the real world test I did (3 fixed 1 loaded) but it is still comparable. I would then have the max deflection and the deflection angle from the front left to the front right suspension hardpoint.

Using small angle approximation this would be the deflection $\theta \approx \frac{\delta_R - \delta_L}{t} \cdot \frac{180}{\pi}$

This gives me the following theoretical values.

Torsional Stiffness; $k_t = T/\theta$ ($\text{N}\cdot\text{m}/\text{deg}$) $\rightarrow$ then convert to ($\text{N}\cdot\text{m}/\text{radian}$)

Torsional Rigidity; $(G \cdot J)_{eq} = k_t \cdot L = T \cdot L / \theta$ ($\text{N}\cdot\text{m}^2$)

Torsional Constant; $J_{eq} = T \cdot L / (G \cdot \theta)$ (m^4)

How would it compare to the real world test? I will have to recalculate the real world data so it could be compared. I will use the same linear to angle calculation for deflection as above but $\delta_L = 0$ since it is fixed. This then gives me these torsional stiffnesses. For the torsional stiffness equation I am using the width of the

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chassis as the length in the torque calculation (t), since it is the distance between δ_R and δ_L in this cross section.

Weight(N)	Measurement (mm)	Deflection (mm)	Deflection (deg)	k_t (Nm/deg)
0	47.78	0	0	0
222.4	47.37	0.41	0.06	1482.7
444.8	46.87	0.91	0.13	1368.6
645.0	46.61	1.17	0.17	1517.6
845.2	46.15	1.63	0.23	1469.9

The average k_t is 1459.7 (Nm/deg) giving the following for the other properties.

$$\text{Torsional Stiffness } k_t = 83700 \quad (N \cdot m / rad)$$

$$\text{Torsional Rigidity } (G \cdot J)_{eq} = 33500 \quad (N \cdot m^2)$$

$$\text{Torsional Constant } J_{eq} = 0.000000423 \quad (m^4) \quad \text{or} \quad 4.23 \times 10^{-7}$$

What are good values for torsional stiffness, and torsional constant? As a sanity check a 1" tube with similar wall thickness (0.095"-0.062") that the car uses has a polar moment of inertia on the order of $J = \times 10^{-8} m^4$ while a 3" tube with a similar wall thickness is on the order of $J = \times 10^{-6} m^4$. So the determined torsional constant makes sense, although not on the high performing end. From research I see that production cars are around 10-30 kNm/deg and FSAE chassis are between 2-10 kNm/deg. The chassis was at 1.5 kNm/deg, but this might be thrown off by the fact that I tested by fixing 3 points and loading 1. This causes both a torsion and bending element to be present. Meaning if the bending component were removed, the deflection would be less, giving a higher torsional stiffness.

Higher is better, but why do you want the chassis to be stiff? You want the chassis to be stiff so that the suspension, not the frame, controls wheel motion. A stiff frame preserves suspension geometry, makes handling predictable, distributes

loads properly, and builds driver confidence. The practical design target should balance stiffness against weight.

Does the torsional stiffness change depending on where the load is suspended from when analysing a chassis? The same chassis will show different torsional stiffness values depending on where you suspend and load it. That doesn't mean that any of the values are wrong, it just means in order to compare values the testing method, and loading point have to be the same. And what about torsional rigidity? Well G does not change unless the material does, but J will change with different cross sections, across the chassis.

Moving on to the head on collision. The questions to answer are; how much energy must the Impact Attenuator (IA) absorb, what is the expected deceleration, are the forces transmitted into the chassis survivable / within design, after running FEA what experienced the most deformation, how did you reinforce it or why not? I am going to use the energy method and compare it to my original approach.

On the note of energy, I will first find the energy of an impact into a wall, given that a collision into another vehicle or moving object is highly unlikely. I will use some of the requirements and input for the FSAE rules, using the mass of my car (385kg) and the standard Impact Attenuator for FSAE. All values come from its documents found here <https://www.rollbarpadding.com/product/id-48>.

F.8.8.2 Impact Attenuator Functional Requirements

These are not test requirements

- a. Decelerates the vehicle at a rate not exceeding 20 g average and 40 g peak
- b. Energy absorbed must be more than 7350 J

When:

- Total mass of Vehicle is 300 kg
- Impact velocity is 7.0 m/s

$$E_{\text{impact}} = \frac{1}{2} m \cdot v_0^2 = \frac{1}{2} 385 \text{ kg} \cdot 7 \text{ m/s}^2 = 9432.5 \text{ J}$$

For the Impact attenuator I will use this equation below, assuming 50% max crush so $\epsilon_{\text{max}} = 0.5$, and A = IA frontal area (using the average of the largest and smallest cross section because it is tapered, front 0.203m×0.101m, back 0.355m×0.304m,

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and $0.254m$ long), $\sigma_p = 700kPa$ (this is conservative since as it crushes these values go up), t = crush thickness.

$$E_{IA} \approx \sigma_p \cdot \varepsilon_{max} \cdot A \cdot t = 700000 \frac{N}{m^2} \cdot 0.5 \cdot 0.0645 m^2 \cdot 0.254 m = 5730 J$$

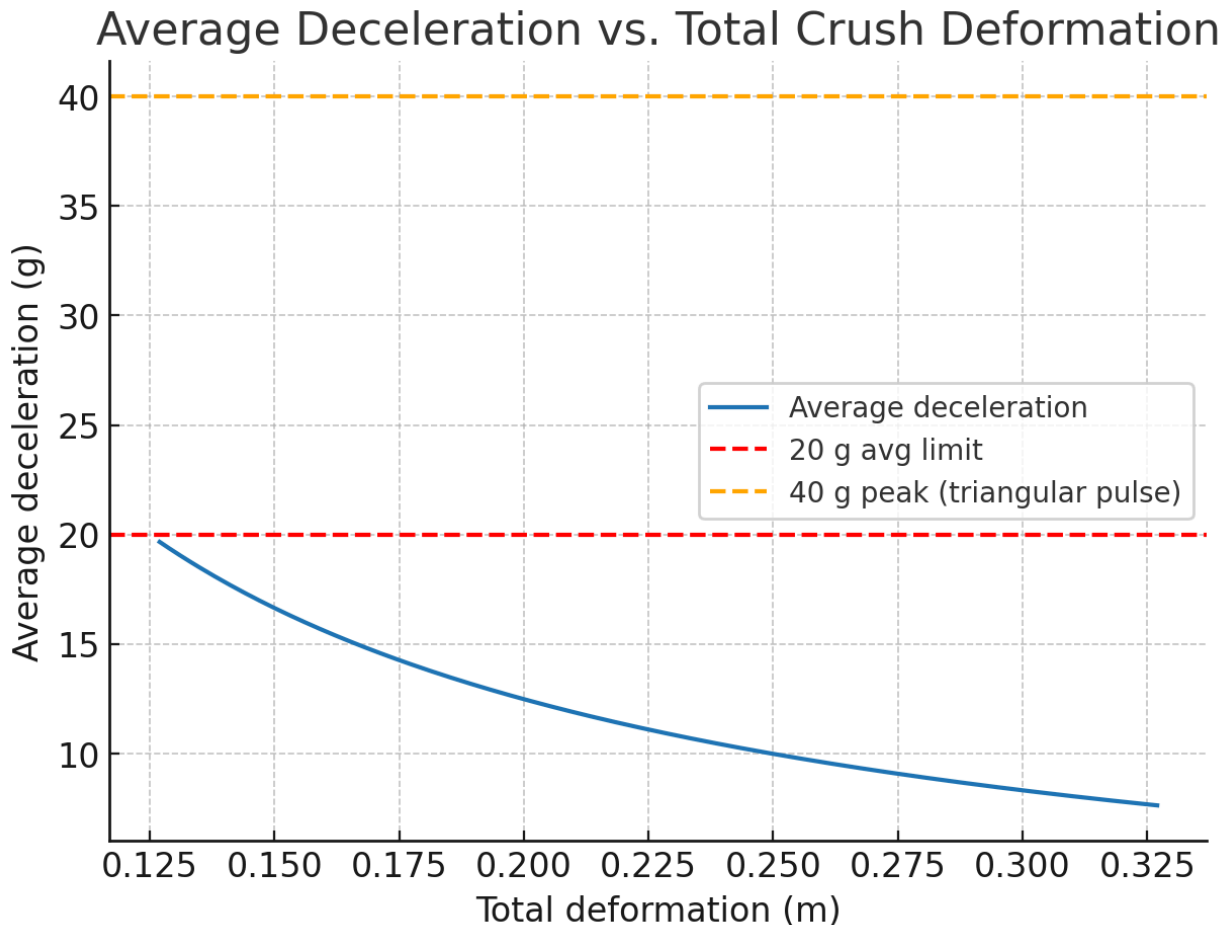
Now we determine how much energy went into the chassis

$$E_{impact} - E_{IA} = E_{chassis} = 3702 J$$

Using $F = m \cdot a$ and $E = F \cdot d$ I will find the average deceleration

$$a_{avg} = \frac{E}{m \cdot d} = \frac{9432.5}{385 \cdot (d_{chassis} + d_{IA})}$$

$$d_{IA} = \varepsilon_{max} \cdot t = 0.5 \cdot 0.254m = 0.127m$$



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What this graph shows is that even with just the impact attenuator absorbing all of the impact of the car to 50% crush, it should not see above an average of about 20Gs (the car is heavy at 385kg and at a speed of 7m/s, ~16mph). In reality I expect the IA to crush further and absorb more energy over a larger distance. This meets the requirements for custom FSAE impact attenuators.

I am still curious how much the chassis would actually deflect if 3702 J made it to the chassis. Given my issue with running FEA on the chassis CAD, I am using the previous FEA as a data point and assuming a linear relationship to the stiffness, the math is below.

$$F_{FEA} = 2.361 \times 10^6 \text{ N}$$

$$\delta_{FEA} = 0.075 \text{ m}$$

$$\text{Stiffness is } k = \frac{F}{\delta} = 3.148 \times 10^7 \text{ N/m}$$

Energy hitting the Chassis is 3702J

$$\text{Using } E = \frac{1}{2} k \delta^2, \delta_{chassis} = 0.0153 \text{ mm}$$

$$\text{To find the average force } F_{avg} = \frac{E}{\delta}, F_{avg} = 2.41 \times 10^5 \text{ N}$$

Now $F=ma$ can be used to find the Average acceleration which is 64 g, meaning peak deceleration could reach 128 g. The graph above does not capture this well since it is the average deceleration of the entire system and not giving a specific history on the IA and the chassis deformation and deceleration. Since I am not able to determine an exact force history and therefore an acceleration vs time graph, it is safest to assume the peak g's are 2x the average g's. Again in real life I expect the IA to deform significantly more, reducing the Gs to acceptable levels (exact total is 208.9mm out of 254mm to absorb all the energy, from below).

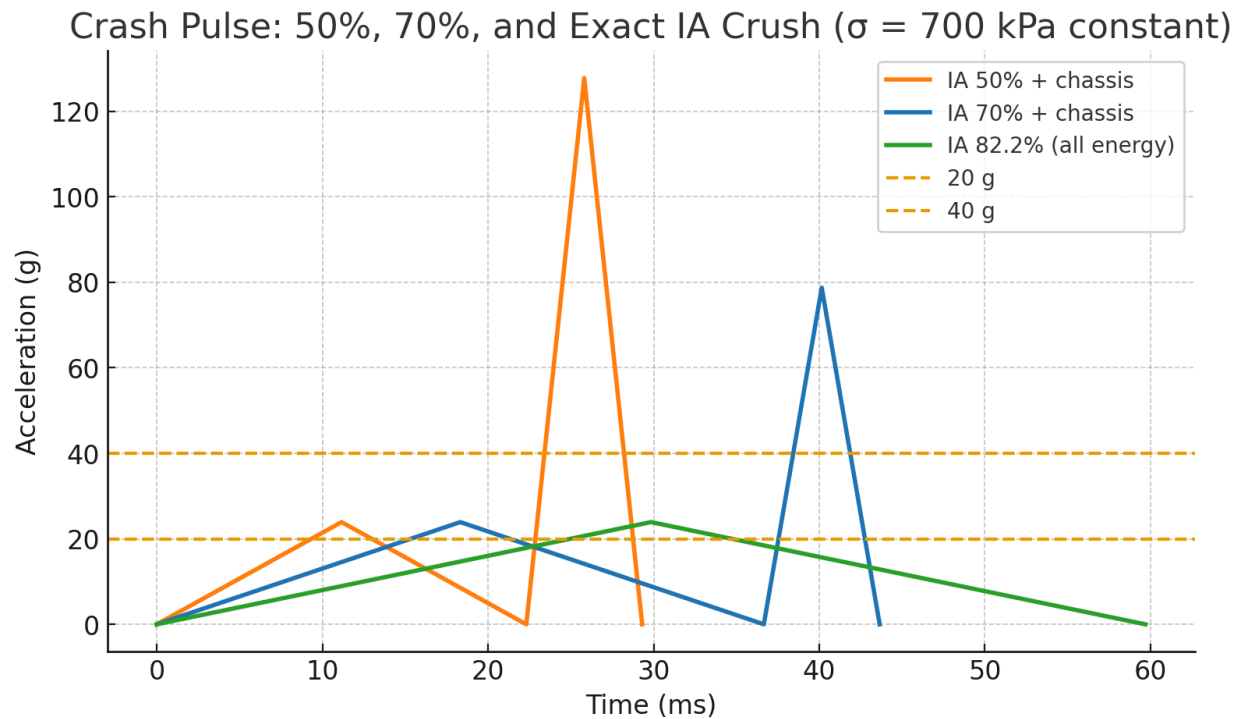
The reason for using 2x the average g's for peak g's is that when comparing impact impulses the triangular pulse is one of the most conservative (over estimating).

This paper explains the types of pulses more and has great visuals

(<https://www.nhtsa.gov/sites/nhtsa.gov/files/18esv-000501.pdf>). There are more types of pulses like trapezoidal pulses. These pulse shapes are determined for

specific impact types. For example the Haversine pulse is most representative of a frontal barrier impact (like in this paper), but the triangular pulse is more conservative, yielding a higher peak deceleration or g. Ideally you would use actual force data from a test crash and use the mass of the car and occupants to create the acceleration vs time graph.

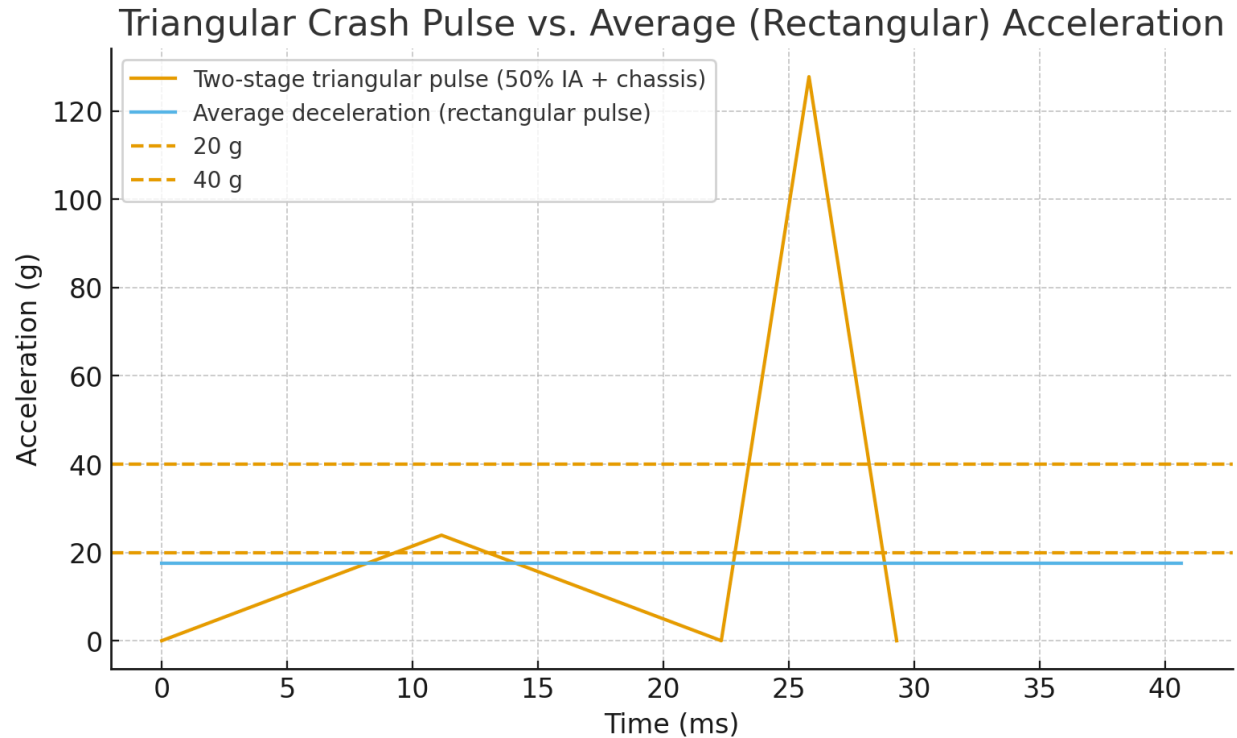
Using the numbers above, here is a graph of what an impact pulse would look like. I also input some different crush percentages of the Impact attenuator to see the impact on peak loading. The small pulse is the IA, the large pulse is the steel chassis. The IA would absorb the impact at just 82.2% crush at well below peak G limits of 40G.



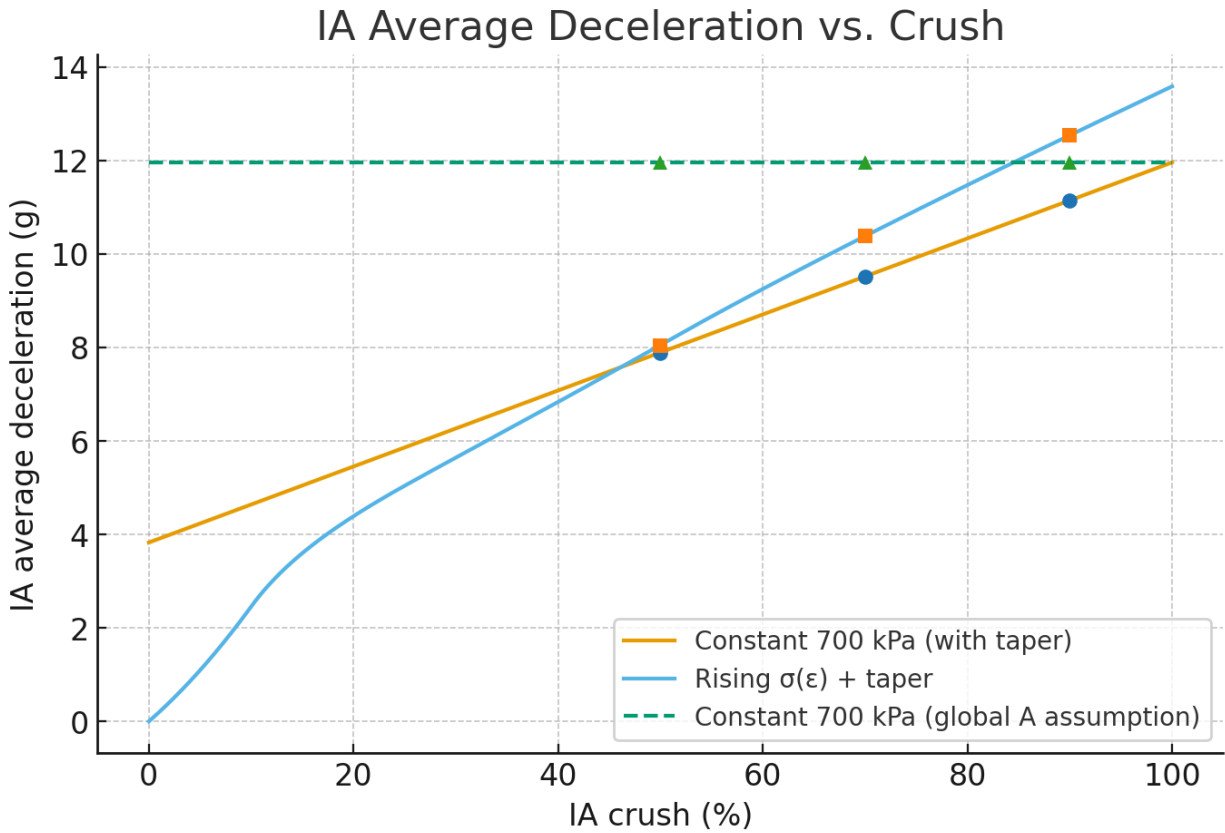
One additional thing to answer is, why are the accelerations here not the same as in the initial estimate? Because it was a general estimate looking at the system as a whole, it is the average of the pulses, or a constant-acceleration level equal to the event's overall average. The reason these are equal is due to the momentum which is

$$\Delta p = m \cdot \Delta v = \int_0^T F(t) dt$$

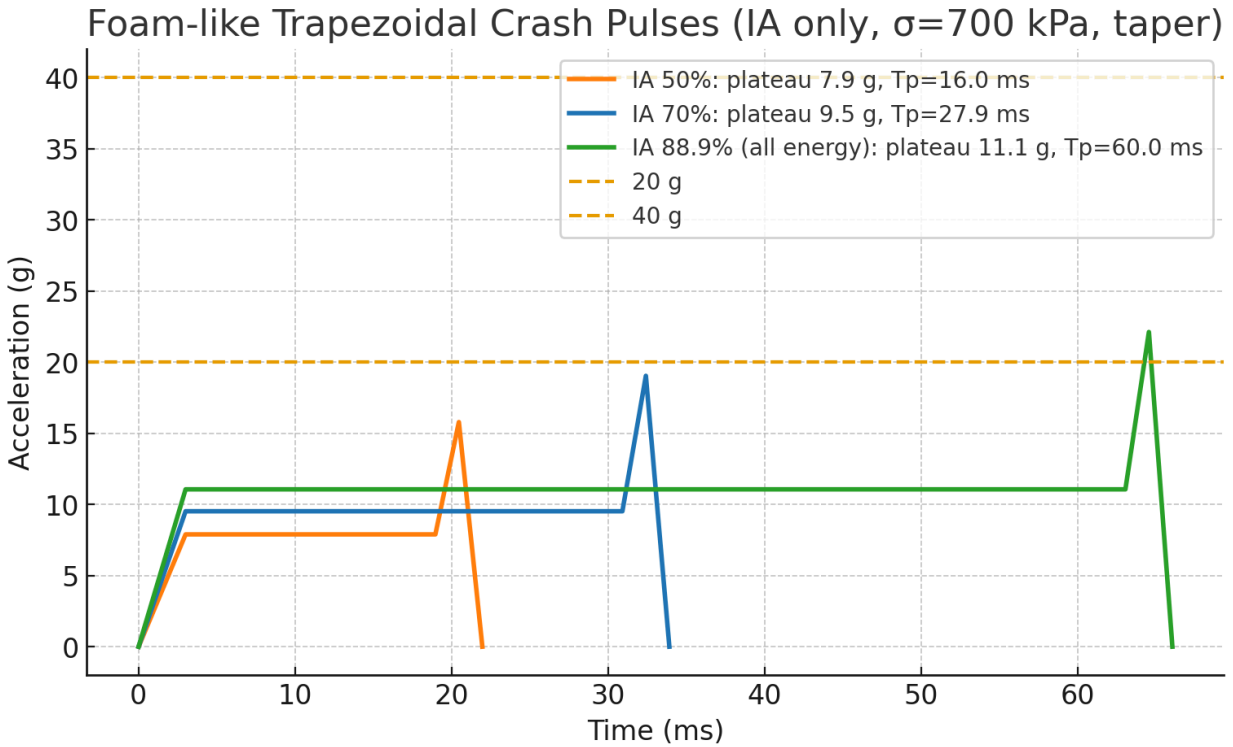
And since the acceleration vs time graph is derived from the force vs time graph via mass, the area under the curve is correlated to the change in momentum ($N \cdot s$)



Is the standard FSAE Impact Attenuator (IA) safe for this car? Based on the conditions required it is, but I would be concerned about it at higher speeds than 17 mph (100% crush of IA). Keeping in mind though that area is averaged and the foam's stiffness is assumed to be constant, I expect the impulse to more gradually increase, with a higher peak G. Here is a graph of the average deceleration for the IA with the different assumption

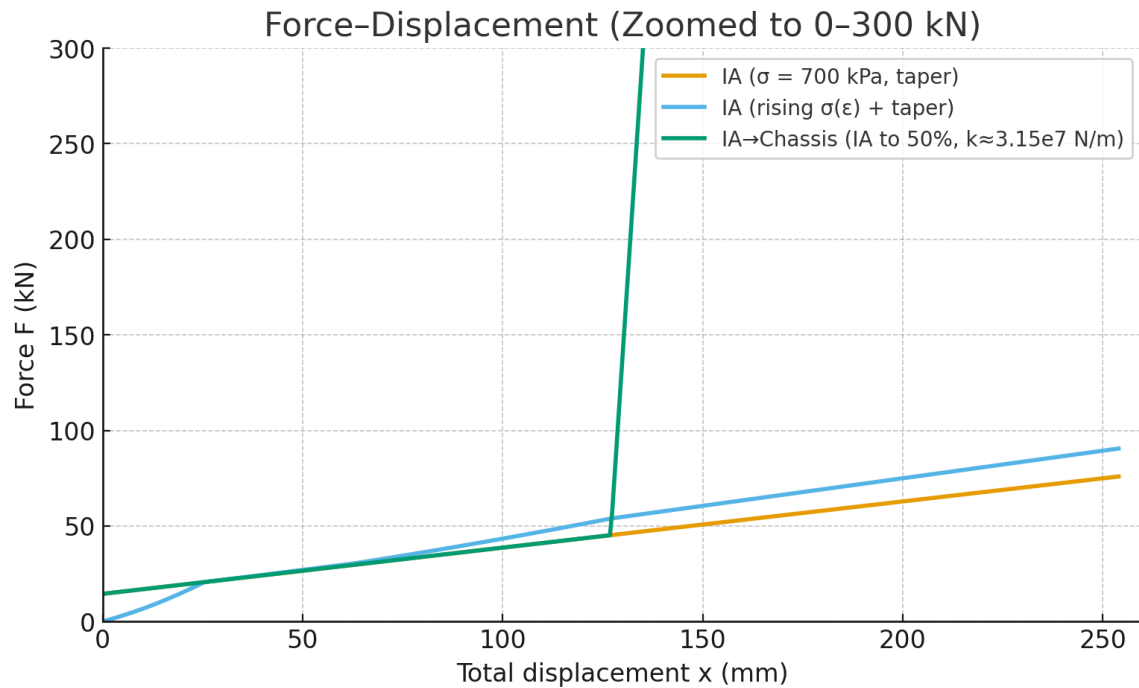
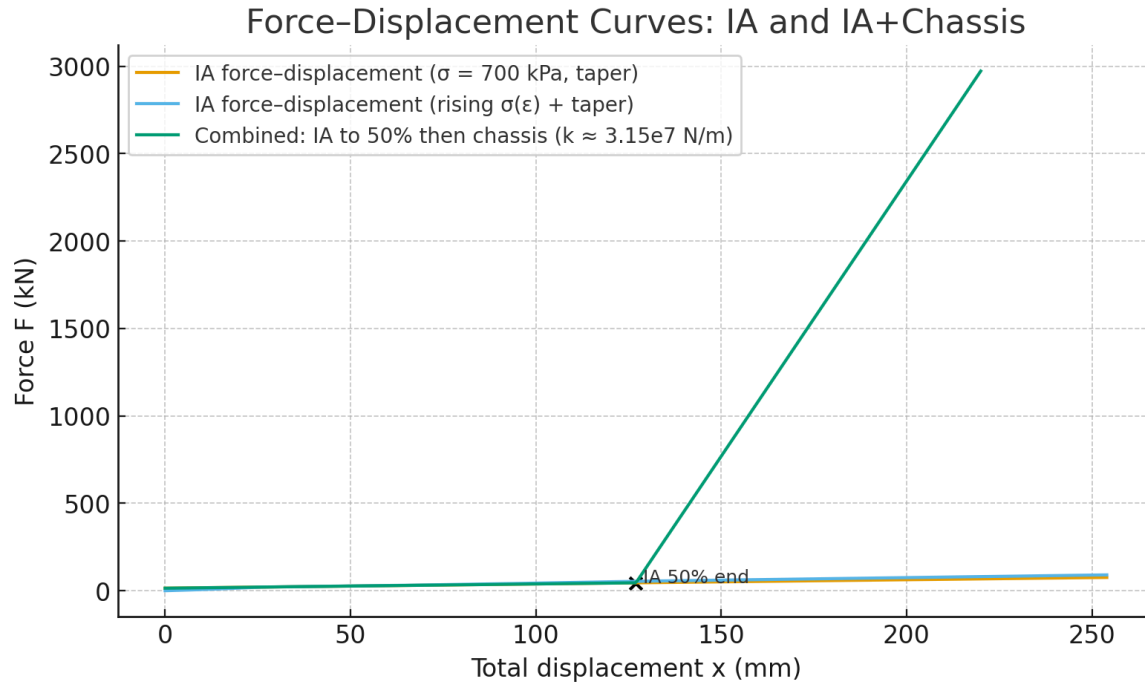


What would an impulse curve look like for a foam block with increasing area and stiffness as it is crushed? It would look more trapezoidal and have a smaller peak at the end near max densification.



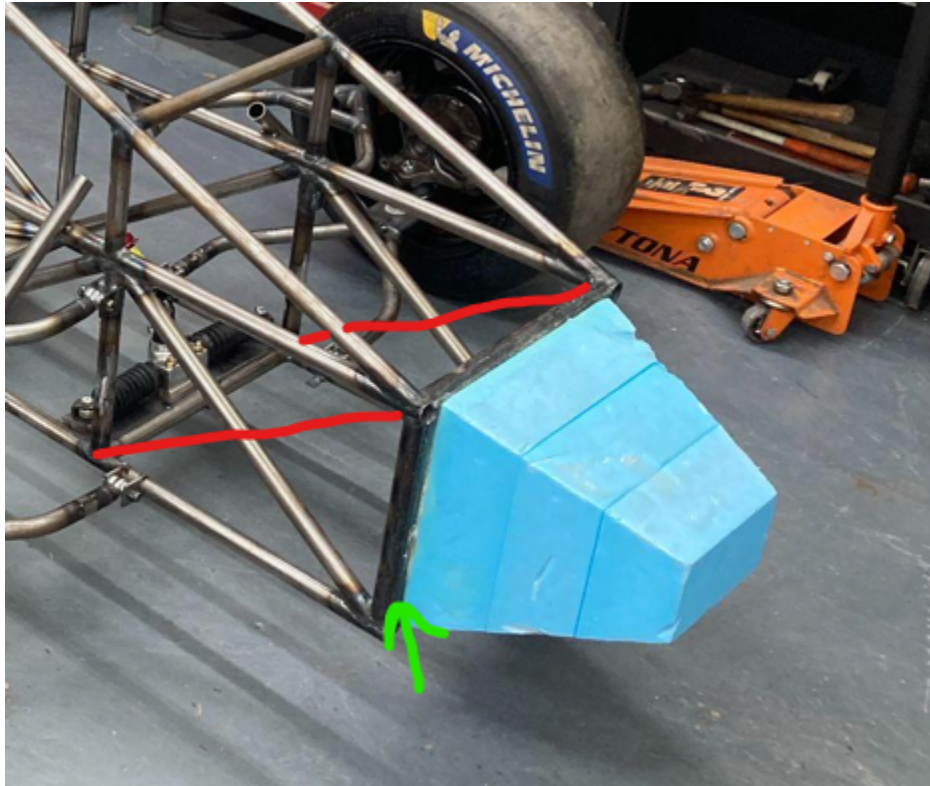
What is the correlation between force time and force distance curves? They are correlated through kinematics equations and the velocity, specifically.
The force displacement graph looks like this

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Now let's assume the chassis did get hit, what happens. Given the max deflection in the worst case (green) is around 16mm, I would expect about 500kN of force. Resulting in the 128G peak. The deformation starts at the lower members of the chassis. The structure is stiff enough to not risk intrusion or injury to the driver

since the pedals (tip of the drivers foot) sit a minimum 3in into the chassis. If I want to stiffen the front I would add a cross member to create a full X shape (in red) on the left and right side of the front of the chassis. Also this simulation does not account for the anti intrusion plate (in green) which would stiffen the front as well. If I wanted to increase the safety or ability to handle impact, I would add to the impact attenuator by either lengthening (more capacity and gentler decel) or add more material to it (more capacity but harder decel).



In summary the original paper had many misunderstandings for the torsional behavior and very surface level analysis of an impact. This paper should add clarity and detail to the original work. This paper is also much more conservative and realistic with the results, so with this I can finally answer the question, was this vehicle safe? Yes it was at speeds around 17mph, for head on collisions (the least likely scenario in FSAE).

GLOSSARY

A = cross sectional area

a = deceleration, units are typically in gravity (g) or (m/s²)

G = shear modulus

J = torsional constant

k = spring constant (axial or linear)

L = length

T = torque

θ = angle in radians or degrees

F = force

μ = coefficient of friction

W = weight of the vehicle (m*g)

g = gravity

M = Moment

h = height

t = crush depth

E = Energy

σ = stress

ϵ = strain

m = mass

d/ δ = displacement/ deformation

p = momentum

v = velocity

Tp = Impulse time