FORMULA SAE CAR CHASSIS DESIGN AND MANUFACTURING

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ENGINEERING PROJECT REPORT

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Spring 2021

ACKNOWLEDGMENTS

Thank you to Mr. Alex Taddei, San Joaquin Delta college welding instructor for his help in constructing the chassis and providing knowledge in manufacturing. Thank you to Delta college for letting our team use their facilities. Thank you to Dr. Watson and Dr. Steimel for their guidance and tools to aid in completing our project. Thank you to Dr. Andy Lutz for providing guidance on calculations for the chassis. Finally thank you to UOP ASME and the Department of Engineering for funding this project.

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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 drive, 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 ±1mm 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.

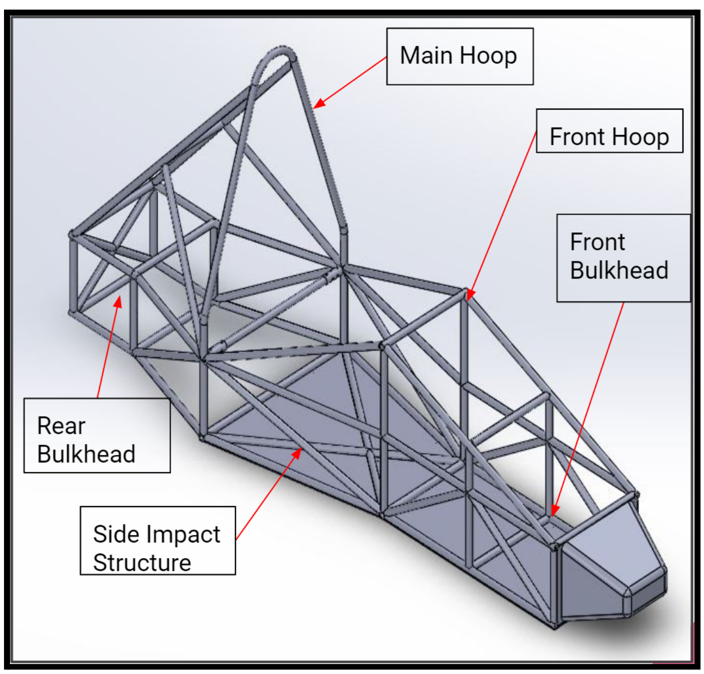


Figure 1: Isometric view depicting key parts of the chassis

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.

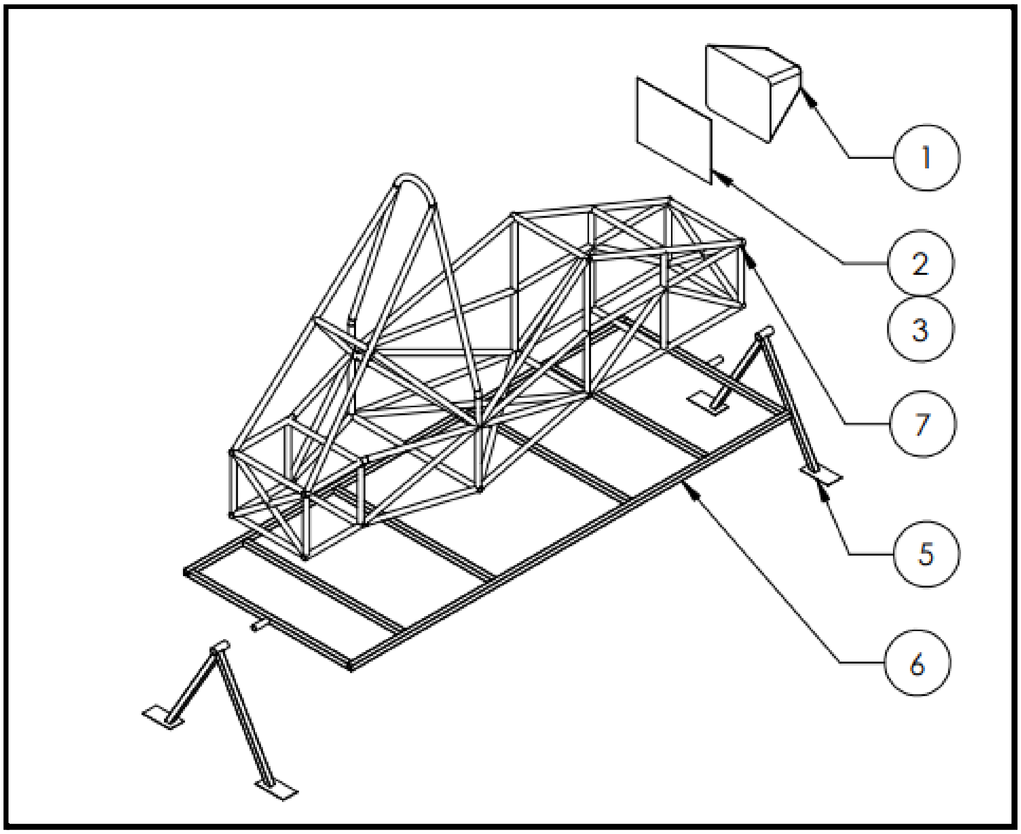


Figure 2: Chassis and jig major component list

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 $3.28\text{x}10^6$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

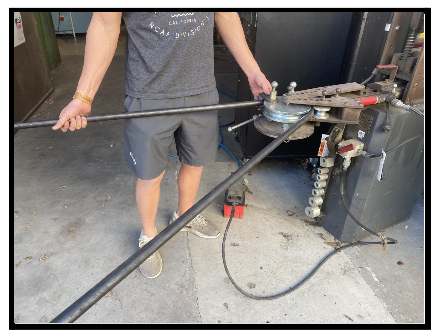


Figure 4: Pneumatic Bender

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.

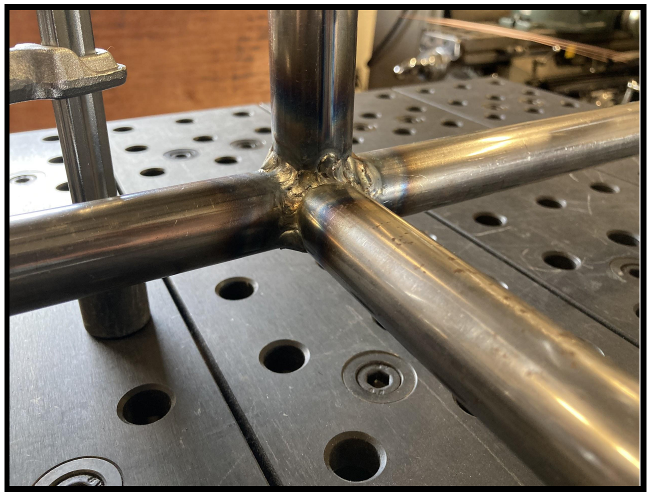


Figure 6: Completed weld up close

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 ¾” 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.

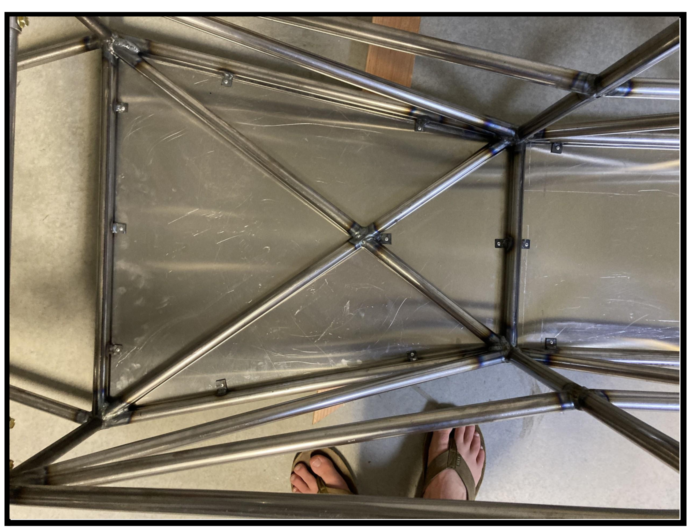


Figure 7: Completed aluminum floor panel

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.

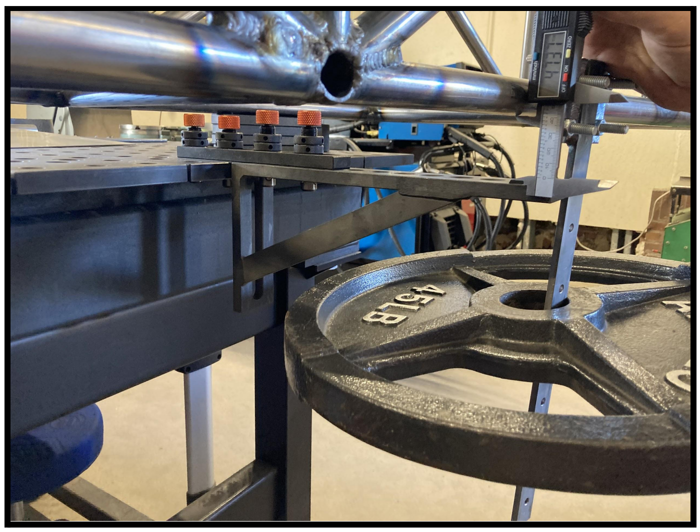


Figure 8: Deflection measuring setup

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 weakend - performance of the vehicle will decrease - overall safety decreases	5	- Improper welding pro - In a collision - Due to fatigue	2	-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	2	- Visual deformation	5	40
Risk analysis	Probability of caus	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.

REFERENCES

[1] 2020 SAE International. (2020, July 30). Formula SAE. https://www.fsaeonline.com/cdsweb/gen/DocumentResources.aspx.

[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.

[3] Krzikalla, D., Mesicek, J., Petru, J., Silva, A., & Smiraus, 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

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 $3.28\text{x}10^6$N (80 km/hr collision).

Theory:

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

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

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

$$Energy\ Absorbed\ (EA)\ 11,742\ Nm\ (J)$$

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

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

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

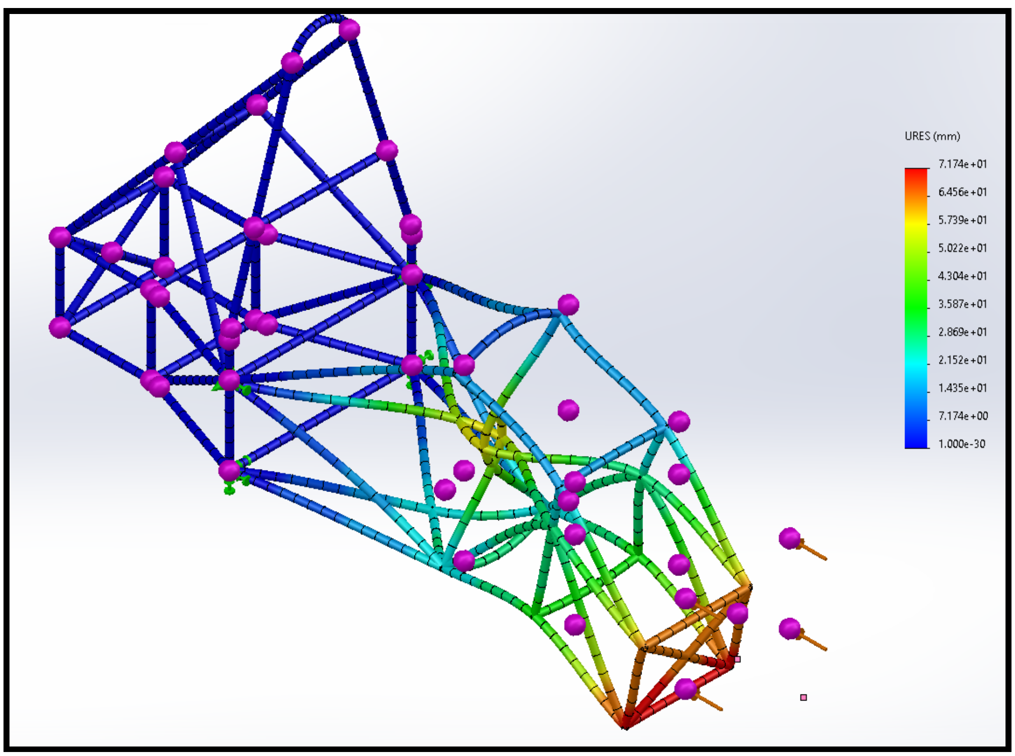
$$IA\ deflection\ =\ FEA\ deflection\ \cdot\ 0.719$$

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 .

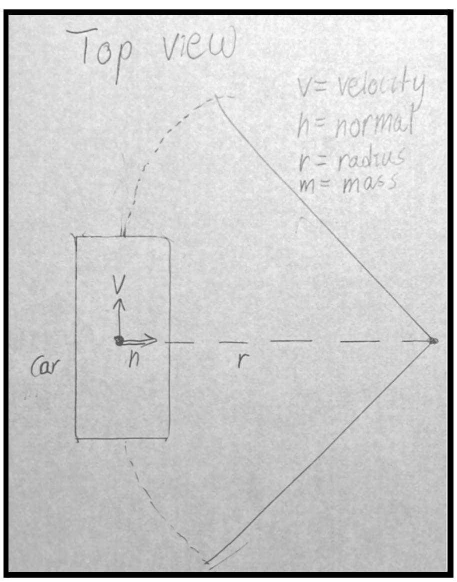
Comments: The results are close to what we expected and reinforces the reasoning for the FSAE organization requiring an impact attenuator. We tried to fully model the impact attenuator, but defining its material properties with the specific type from the manufacturer proved too difficult. We settled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



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

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.

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

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[1] Krzikalla, D., Mesicek, J., Petru, J., Silva, A., & Smiraus, 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.

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

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.

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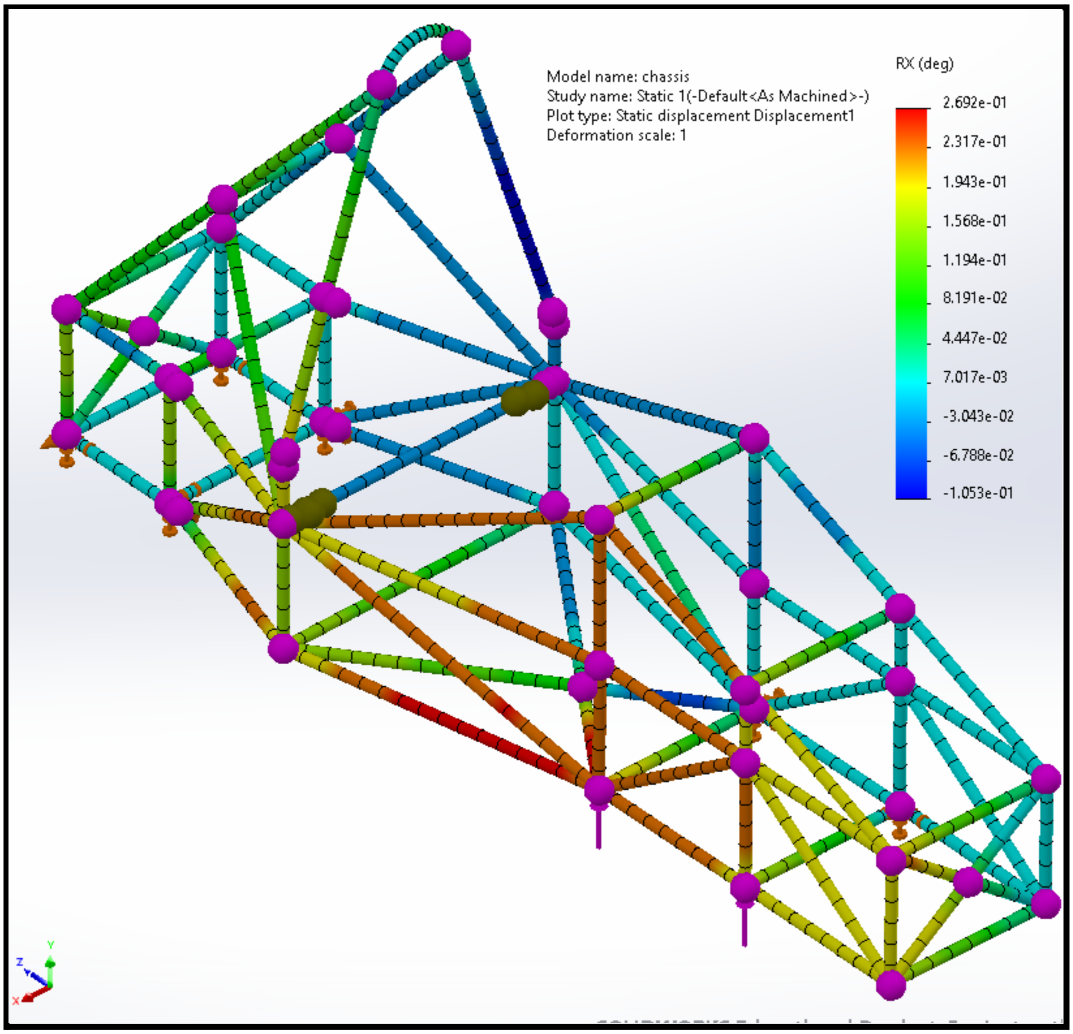
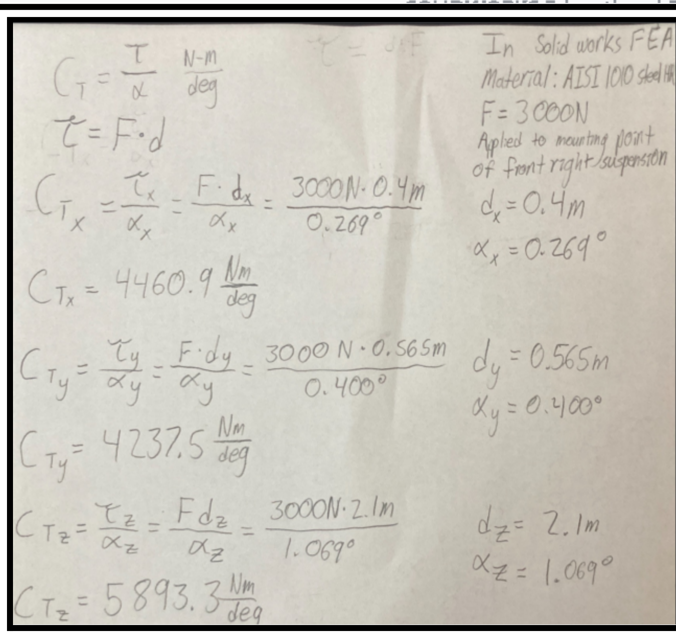
[2] Bartolomeo, M. V., Lombardo, A., Colella, M., & Delagrammatikas, G. J. (2019). (tech.). *Measuring the Traction Limits and Suspension Forces of a Formula SAE Racecar* (p. 25). American Society for Engineering Education.

Schematic, Plot, or Graphs:





In Solid works FEA
Material: AISI 1010 steel HR
F = 3000N
Aplied to mounting point of front right suspension

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

$\tau = F \cdot d$

$C_{T_x} = \frac{\tau_x}{\alpha_x} = \frac{F \cdot d_x}{\alpha_x} = \frac{3000N \cdot 0.4m}{0.269°}$

$d_x = 0.4m$

$\alpha_x = 0.269°$

$C_{T_x} = 4460.9 \frac{Nm}{deg}$

$C_{T_y} = \frac{\tau_y}{\alpha_y} = \frac{F \cdot d_y}{\alpha_y} = \frac{3000\,N \cdot 0.565m}{0.400°}$

$d_y = 0.565m$

$\alpha_y = 0.400°$

$C_{T_y} = 4237.5 \frac{Nm}{deg}$

$C_{T_z} = \frac{\tau_z}{\alpha_z} = \frac{F d_z}{\alpha_z} = \frac{3000N \cdot 2.1m}{1.069°}$

$d_z = 2.1m$

$\alpha_z = 1.069°$

$C_{T_z} = 5893.3 \frac{Nm}{deg}$

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.

Appendix B: Budget

Item	(Part #)	Cost Per Unit	Quantity	Shipping	Tax	total	Seller	Link	Priority	Explanation
Square Tube 6ft	6527K174	$17.98	8	$0.00	$0.00	$143.84	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the jig
U-Bolt 1"	3043T649	$0.99	12	$0.00	$0.00	$11.88	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the jig
Steel tube 1.25" 1ft (6.5" Chas	7767T48	$7.36	1	$0.00	$0.00	$7.36	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the jig
Tube size A 6ft	7767T39	$16.25	3	$0.00	$0.00	$48.75	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
Tube size B 6ft	7767T231	$11.93	23	$0.00	$0.00	$274.39	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
Tube size A 10ft	7767T999	$32.50	1	$76.86	$0.00	$109.36	McMASTER-CARR	https://www.mcmaster.com/	Primary	Part of the chassis
Small steel plate	204225705	$7.98	4	$0.00	$0.00	$31.92	Home Depot	https://www.homedepot.com/p/Everbilt-6-in-x-18	Primary	Part of the jig
Glue	N/A	$156.20	1	$0.00	$0.00	$156.20	Amazon	https://www.amazon.com/3M-Scotch-Weld-0212	Secondary	Part of the chassis
AIP	6544K14	$26.38	1	$0.00	$0.00	$26.38	McMASTER-CARR	https://www.mcmaster.com/6544K14/	Secondary	Part of the chassis
8mm Bolts	94036A529	$8.86	1	$0.00	$0.00	$8.86	McMASTER-CARR	https://www.mcmaster.com/94036a529	Secondary	Part of the chassis
Aluminium plate	89015K53	$157.35	1	$0.00	$0.00	$157.35	McMASTER-CARR	https://www.mcmaster.com/89015k53	Secondary	Part of the chassis
1/8" rivots	97447A015	$7.88	1	$0.00	$0.00	$7.88	McMASTER-CARR	https://www.mcmaster.com/97447A015/	Secondary	Part of the chassis
Impact Attenuator	N/A	$170.00	1	$0.00	$0.00	$170.00	BSCI	https://www.rollbarpadding.com/product/id-48	Secondary	Part of the chassis
8mm locknuts	97131A140	$7.86	1	$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 discription box should appear. Enter the quote #										
The dimentions 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%

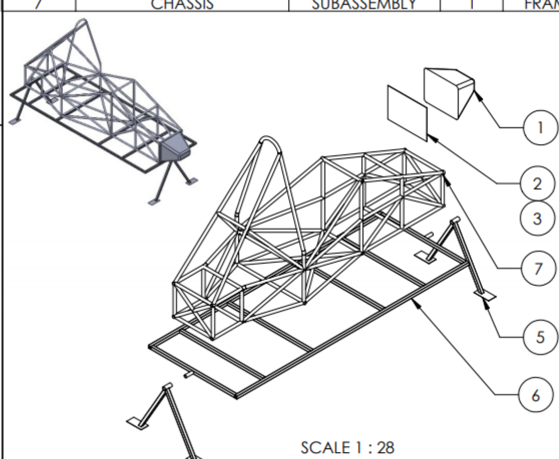
Appendix D: Final Engineering Drawings

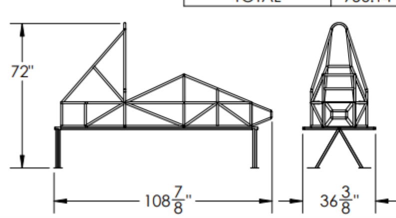
TABLE OF CONTENTS

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

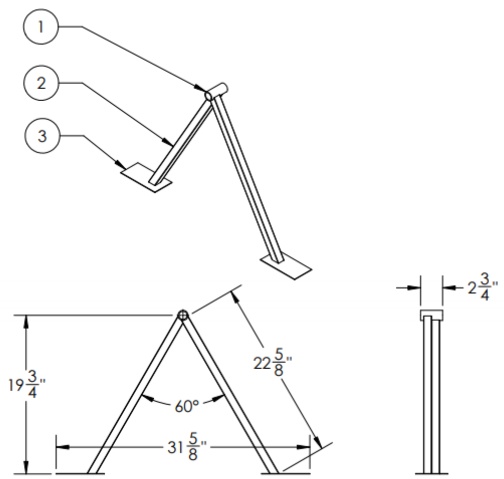
ITEM NO.	PART NAME	ASSEMBLY TYPE	QTY.	DESCRIPTION	VENDOR	COST ($)
1	IMPACT ATTENUATOR (IA)	FINAL ASSEMBLY	1	12" X 14" X10" IMPAXX 700 FOAM ABSORBS FRONT IMPACT	BSCI ENERGY IMPACT SYSTEMS	170.00
2	ANTI-INTRUSION PLATE (AIP)	FINAL ASSEMBLY	1	12" X 16" x 0.06" STEEL PLATE KEEPS DEBRIS OUT WELDED TO CHASSIS	MCMASTER-CARR	26.38
4	U-BOLTS AND NUTS	FINAL ASSEMBLY	12	1" U 2 3/16" LONG BOLT, STEEL HOLDS CHASSIS ON THE JIG	MCMASTER-CAR	0.99
3	GLUE	FINAL ASSEMBLY	1	HOLDS IA AND AIP TOGETHER	AMAZON	156.20
5	JIG SUPPORTS	SUBASSEMBLY	1	ALLOWS THE JIG TO ROTATE	FAB. SEE PG. 2	51.30
6	JIG	SUBASSEMBLY	1	HOLDS THE CHASSIS	FAB. SEE PG. 3	124.13
7	CHASSIS	SUBASSEMBLY	1	FRAME FOR THE FSAE CAR	FAB. SEE PG. 6	416.25
					TOTAL	956.14





Group: Jan, Nate, Daniel
TITLE: FSAE CHASSIS ASSEMBLY
SIZE A | DWG. NO. 1 | DATE: 2/14
SCALE: 1:48 | DRAWN BY: JS | SHEET 1 OF 15

ITEM NO.	PART NAME	QTY.	DESCRIPTION	VENDOR	COST ($)
1	TUBE	1	1.25" OD 1" ID X 1FT CUT TO TWO 2 3/4" TUBES TOLERANCE +/- 1/16" LOW CARBON 1010 HOT ROLLED STEEL ALLOWS THE JIG TO ROTATE	MCMASTER-CARR	7.36
2	SQUARE TUBE	2	1" X 1" X 6FT WITH 0.065 WALL THICKNESS CUT TO FOUR 22 5/8" TUBES, TOLERANCE +/- 1/16" LOW CARBON 1005 HOT ROLLED STEEL ELEVATES THE JIG OFF THE GROUND SO IT CAN ROTATE	MCMASTER-CARR	17.98
3	PLATE	1	6" X 18" 16GA CUT TO FOUR 6" X 3" PLATES TOLERANCE +/- 1/16", PLAIN STEEL ALLOWS THE SUPPORTS TO BE HELD IN PLACE BY WEIGHT	HOME DEPOT	7.98
				TOTAL	51.30



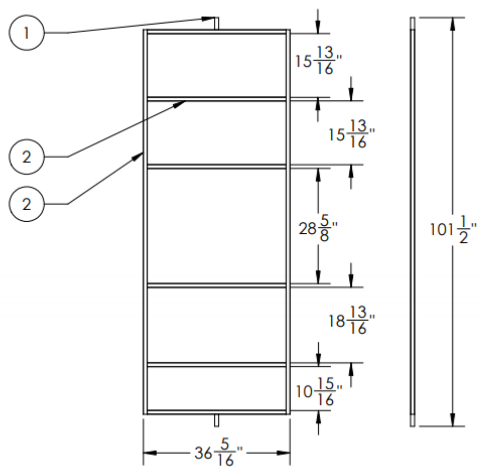
Group: Jan, Nate, Daniel

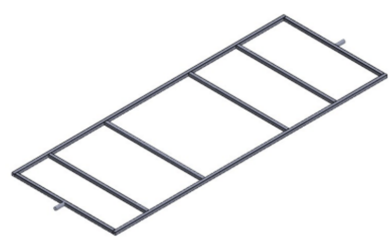
TITLE: JIG SUPPORTS

SIZE A | DWG. NO. 2 | DATE: 2/14

SCALE: 1:12 | DRAWN BY: JS | SHEET 2 OF 15

ITEM NO.	PART NAME	QTY.	DESCRIPTION	VENDOR	COST ($)
1	STOCK ROUND TUBE A	1	1" OD 0.81" ID X 6FT LOW CARBON 1010 HOT ROLLED STEEL ALLOWS THE JIG TO ROTATE	MCMASTER-CARR	16.25
2	STOCK SQUARE TUBE	6	1" X 1" X 6FT WITH 0.065 WALL THICKNESS LOW CARBON 1005 HOT ROLLED STEEL USED TO CREATE THE JIG WHERE THE CHASSIS WILL MOUNT TO	MCMASTER-CARR	17.98
				TOTAL	124.13



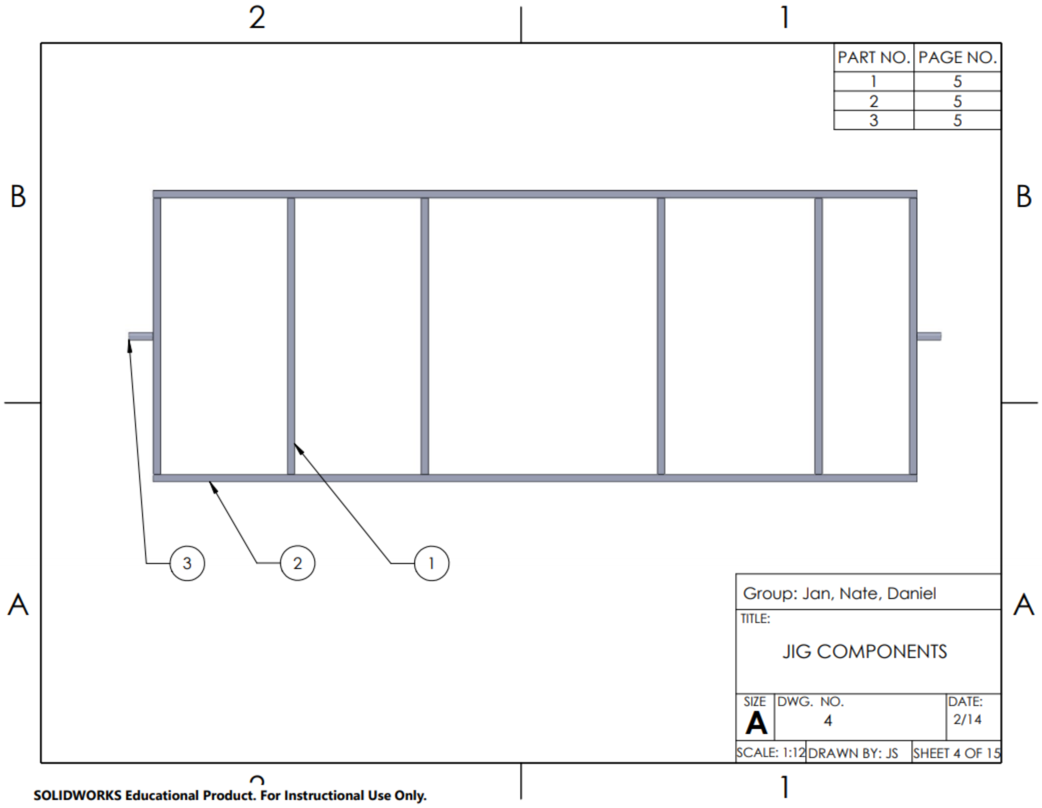


Group: Jan, Nate, Daniel

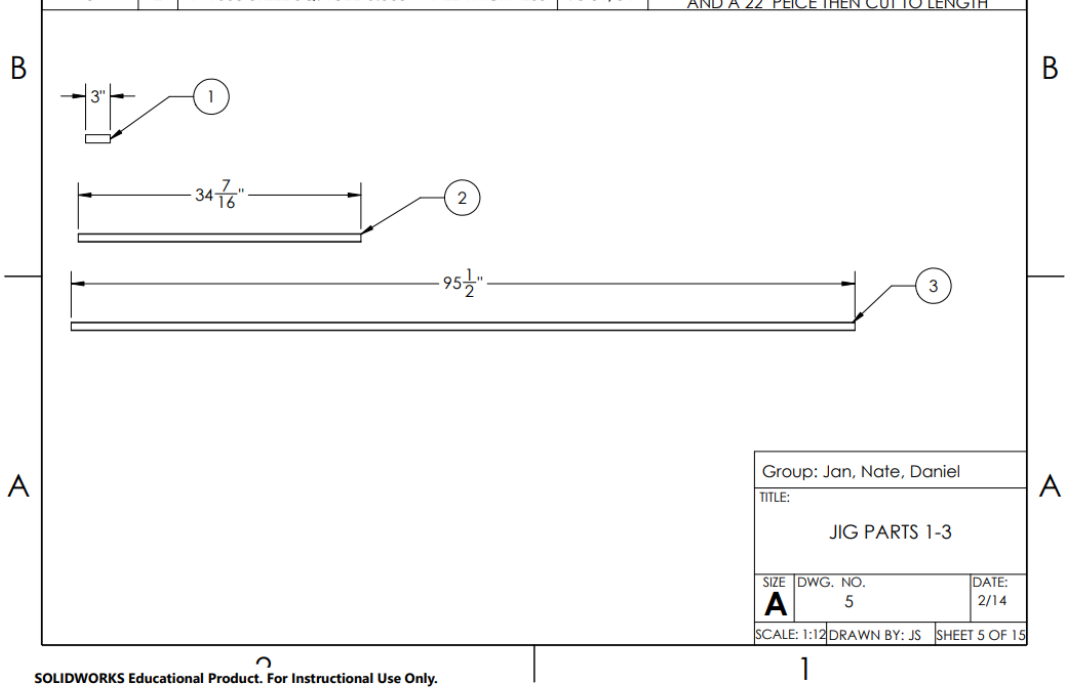
TITLE: JIG

SIZE A | DWG. NO. 3 | DATE: 2/14

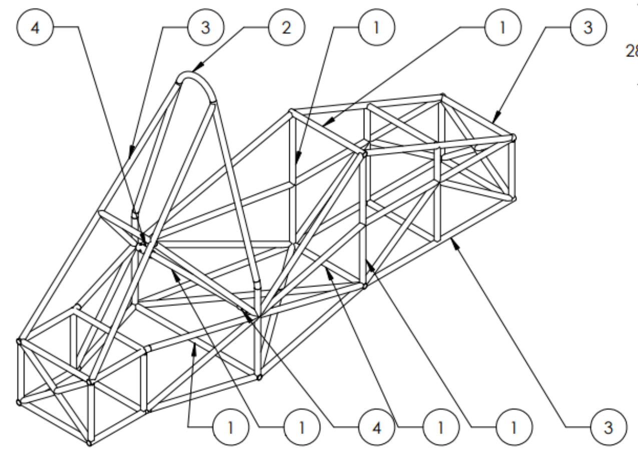
SCALE: 1:24 | DRAWN BY: JS | SHEET 3 OF 15

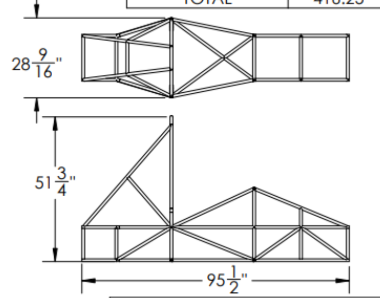


PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
1	2	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	3"	TOLERANCE +/- 1/16"
2	4	1" 1005 STEEL SQ. TUBE 0.065" WALL THICKNESS	36 7/16"	TOLERANCE +/- 1/16"
3	2	1" 1005 STEEL SQ. TUBE 0.065" WALL THICKNESS	93 31/64"	TOLERANCE +/- 1" , WELD TOGETHER 6FT PEICE AND A 22" PEICE THEN CUT TO LENGTH



ITEM NO.	PART NAME	QTY.	DESCRIPTION	VENDOR	COST ($)
1	STOCK ROUND TUBE A	2	1" OD 0.81" ID X 6FT LOW CARBON 1010 HOT ROLLED STEEL USED FOR THE MAIN HOOP AND FRONT HOOP	MCMASTER-CARR	16.25
2	STOCK 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
3	STOCK 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
4	STOCK TUBE	1	1.25" OD 1" ID X 6.5" LOW CARBON 1010 HOT ROLLED STEEL REMAINING FROM JIG, BUTT JOINT SLEEVE	MCMASTER-CARR	0.00
				TOTAL	416.25





SCALE 1 : 16

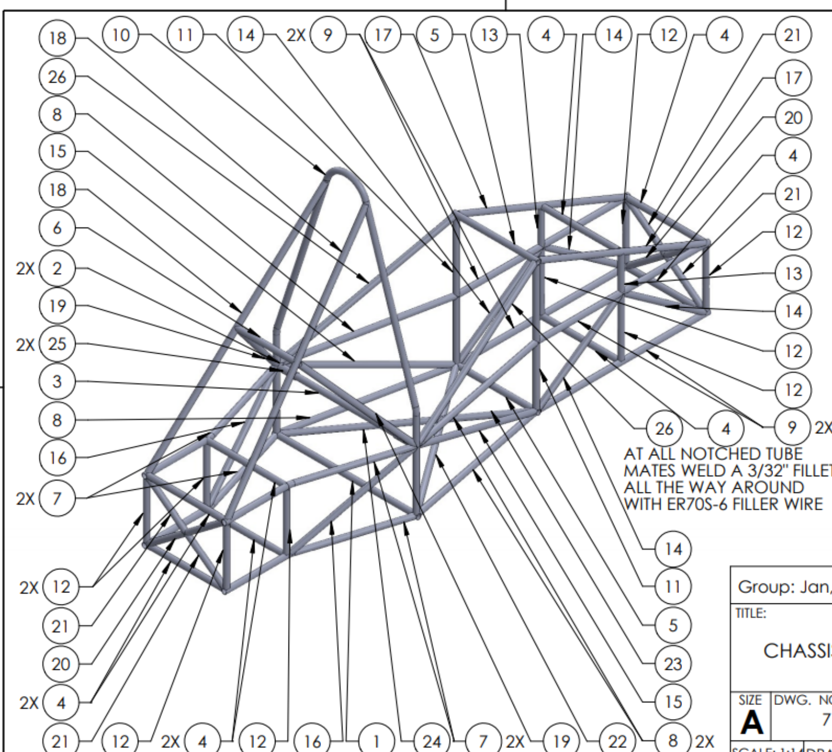
NOTE: IF IT IS NOT BALLOONED IT IS ITEM 3

Group: Jan, Nate, Daniel

TITLE: CHASSIS

SIZE A | DWG. NO. 6 | DATE: 2/14

SCALE: 1:36 | DRAWN BY: JS | SHEET 6 OF 15



AT ALL NOTCHED TUBE MATES WELD A 3/32" FILLET ALL THE WAY AROUND WITH ER70S-6 FILLER WIRE

PART NO.	PAGE NO.
1	8
2	8
3	8
4	8
5	8
6	8
7	9
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9	9
10	10
11	10
12	10
13	10
14	11
15	11
16	11
17	12
18	12
19	12
20	13
21	13
22	13
23	14
24	14
25	14
26	14

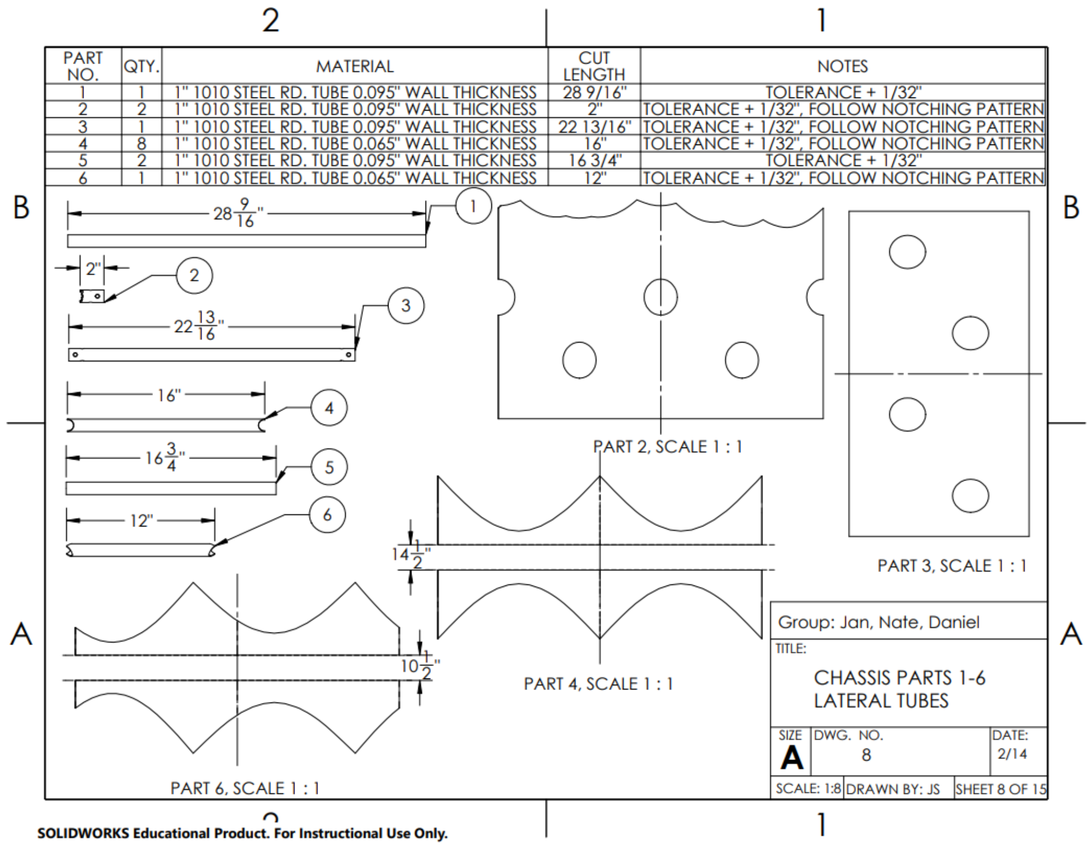
Group: Jan, Nate, Daniel

TITLE: CHASSIS COMPONENTS

SIZE A | DWG. NO. 7 | DATE: 2/14

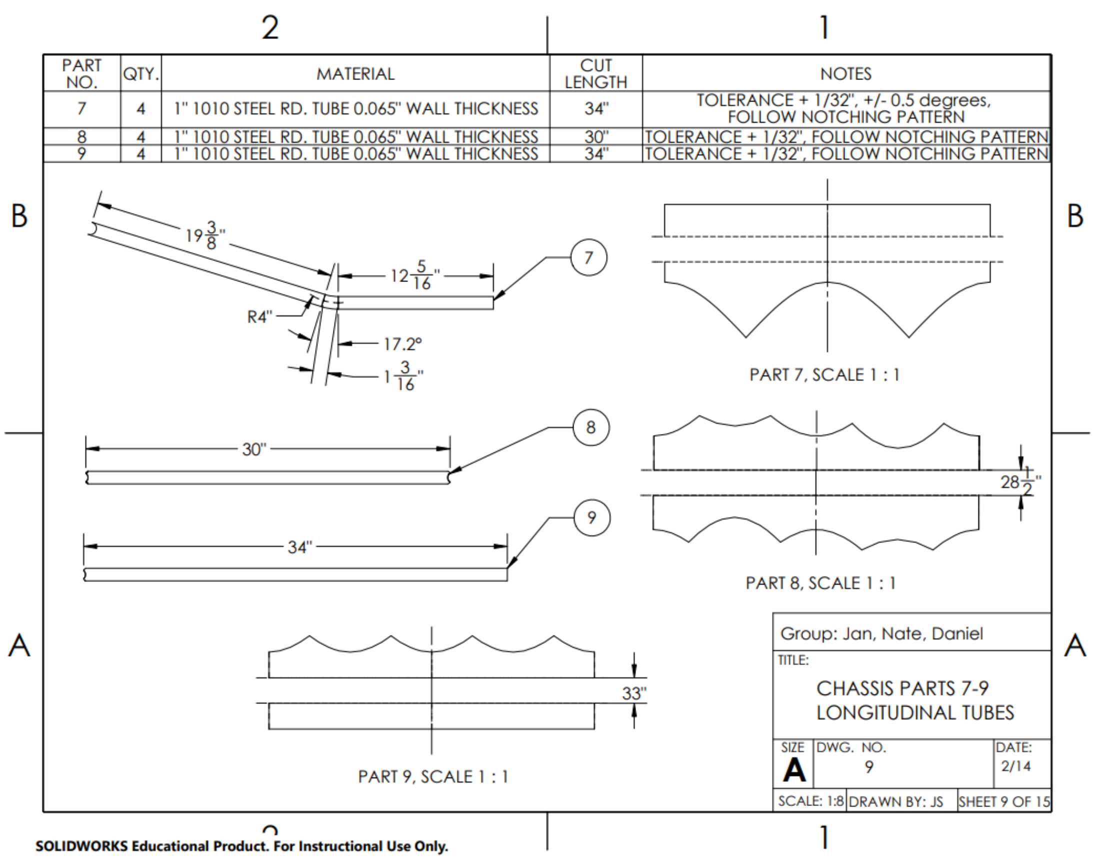
SCALE: 1:14 | DRAWN BY: JS | SHEET 7 OF 15

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
1	1	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	28 9/16"	TOLERANCE + 1/32"
2	2	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
3	1	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	22 13/16"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
4	8	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	16"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
5	2	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	16 3/4"	TOLERANCE + 1/32"
6	1	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	12"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



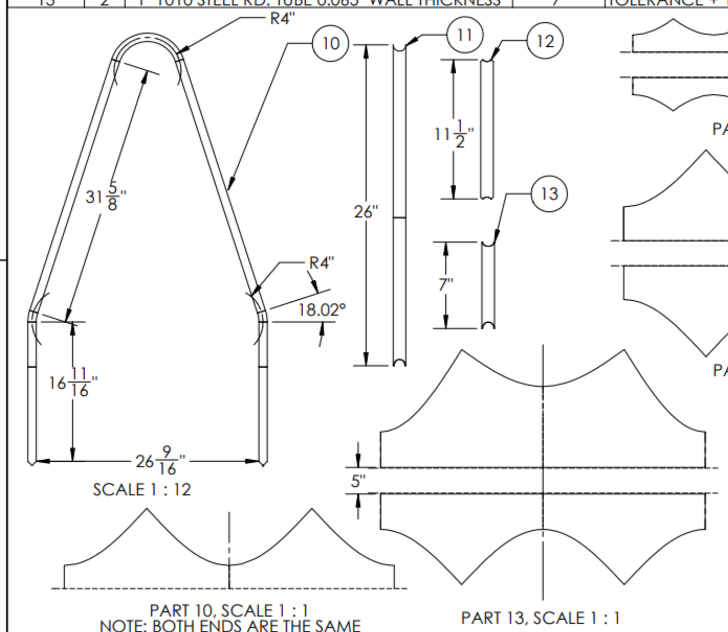
SOLIDWORKS Educational Product. For Instructional Use Only.

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
7	4	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	34"	TOLERANCE + 1/32", +/- 0.5 degrees, FOLLOW NOTCHING PATTERN
8	4	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	30"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
9	4	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	34"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



SOLIDWORKS Educational Product. For Instructional Use Only.

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
10	1	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	120"	TOLERANCE + 1/32", +/- 0.5 degrees, FOLLOW NOTCHING PATTERN
11	2	1" 1010 STEEL RD. TUBE 0.095" WALL THICKNESS	26"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
12	8	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	11 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
13	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	7"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



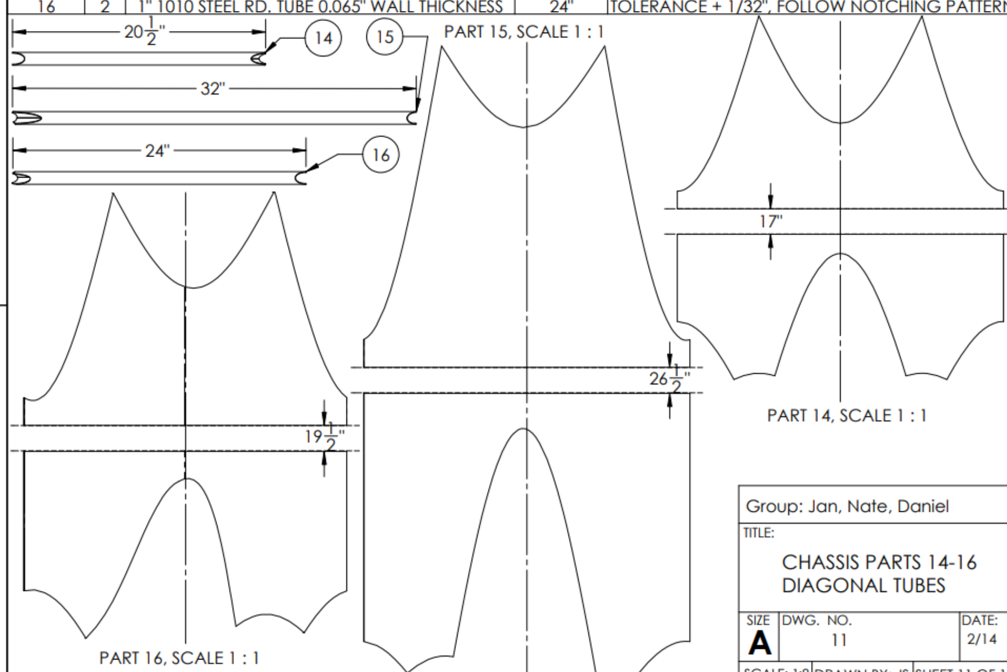
Group: Jan, Nate, Daniel

TITLE:

CHASSIS PARTS 10-13
VERTICAL TUBES

SIZE	DWG. NO.	DATE:
A	10	2/14
SCALE: 1:8	DRAWN BY: JS	SHEET 10 OF 15

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
14	4	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	20 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
15	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	32"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
16	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	24"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



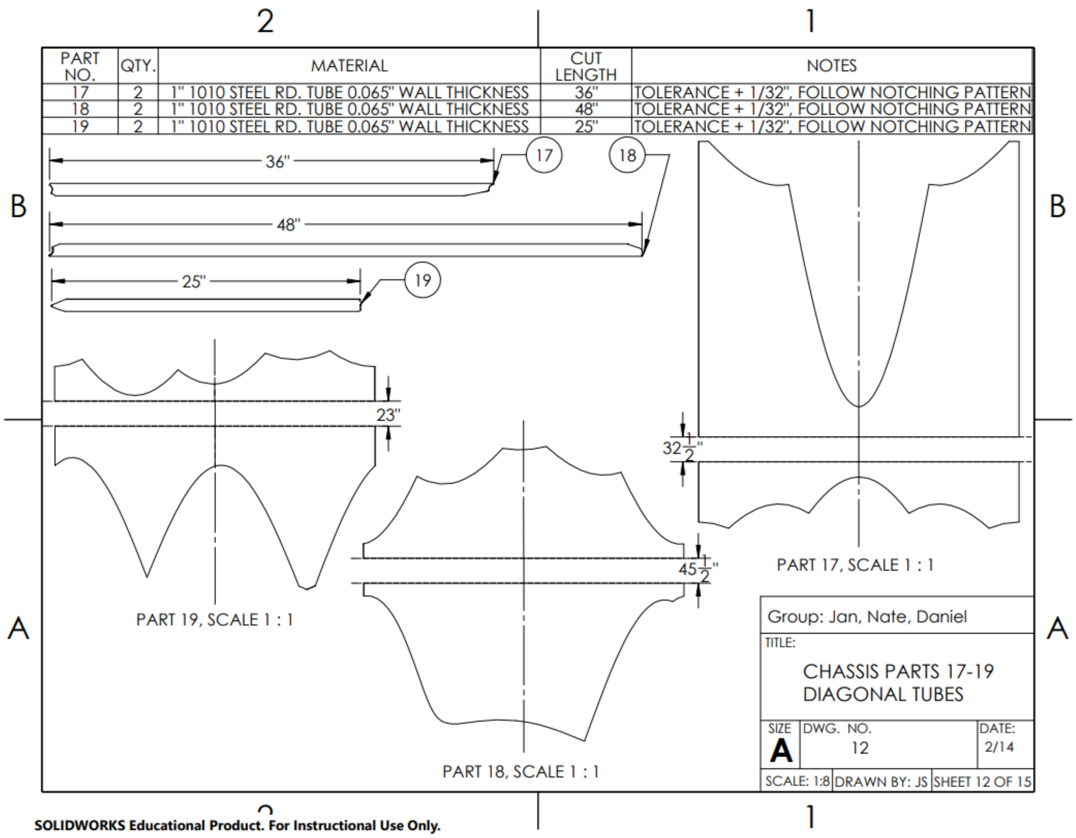
Group: Jan, Nate, Daniel

TITLE:

CHASSIS PARTS 14-16
DIAGONAL TUBES

SIZE	DWG. NO.	DATE:
A	11	2/14
SCALE: 1:8	DRAWN BY: JS	SHEET 11 OF 15

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
17	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	36"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
18	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	48"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
19	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	25"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



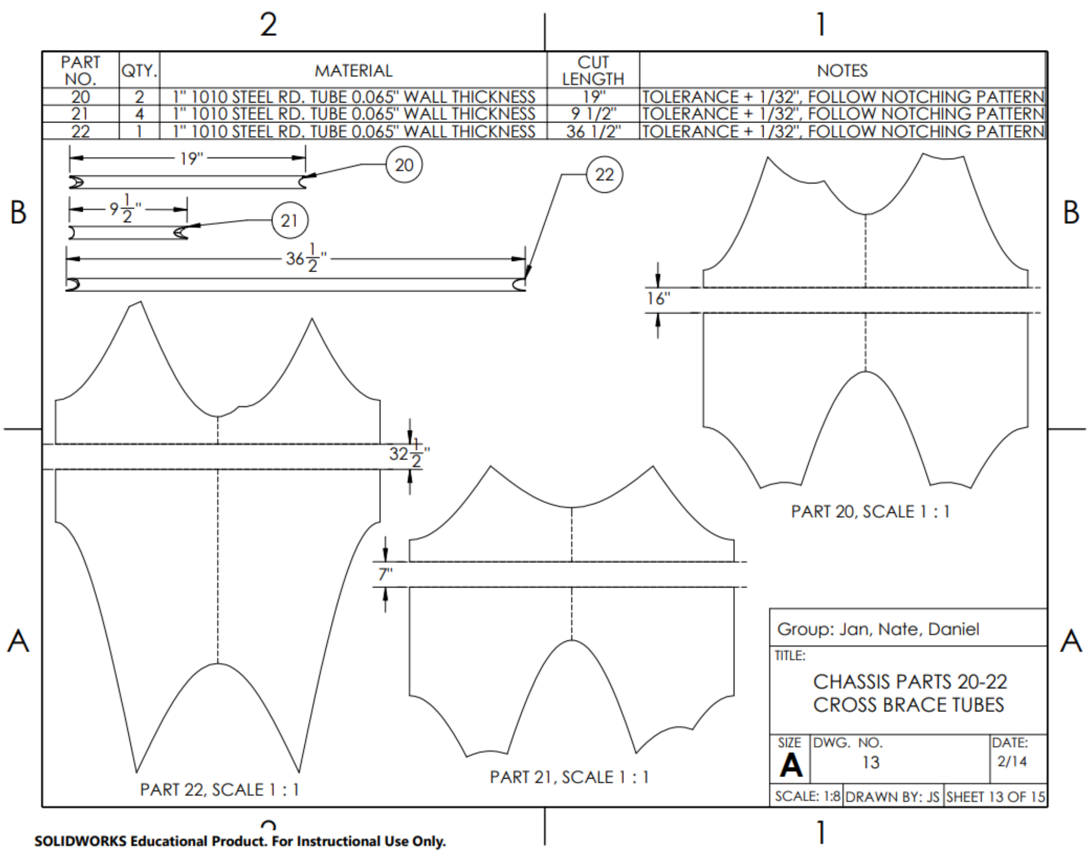
Group: Jan, Nate, Daniel

TITLE:

CHASSIS PARTS 17-19
DIAGONAL TUBES

SIZE	DWG. NO.	DATE:
A	12	2/14
SCALE: 1:8	DRAWN BY: JS	SHEET 12 OF 15

PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
20	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	19"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
21	4	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	9 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
22	1	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	36 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



Group: Jan, Nate, Daniel

TITLE:

CHASSIS PARTS 20-22
CROSS BRACE TUBES

SIZE	DWG. NO.	DATE:
A	13	2/14
SCALE: 1:8	DRAWN BY: JS	SHEET 13 OF 15

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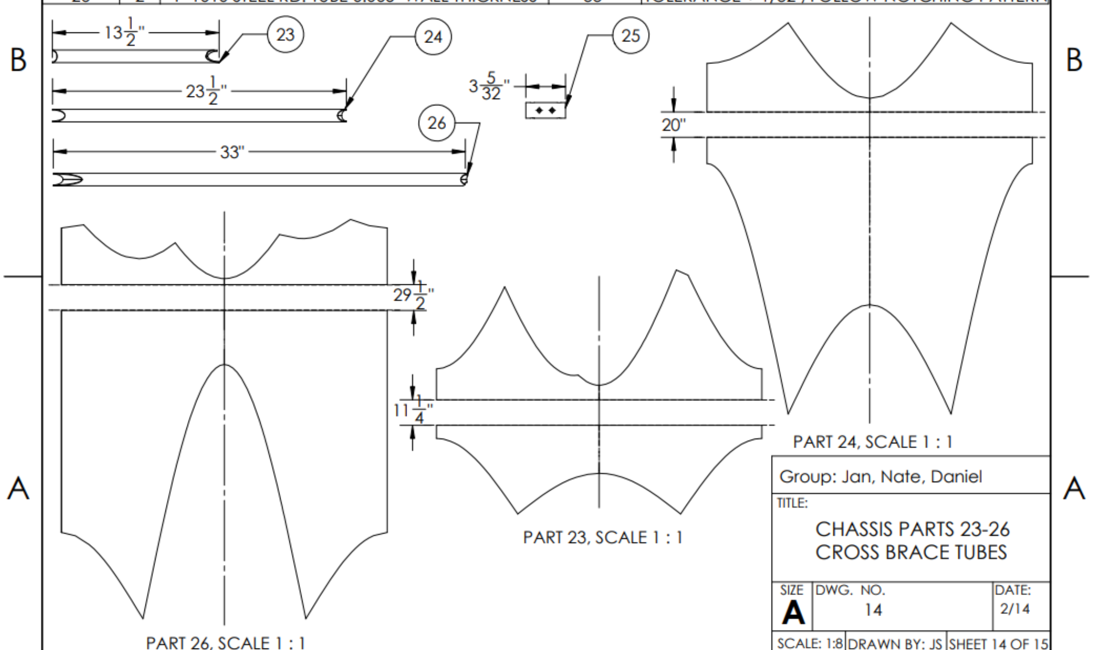
PART NO.	QTY.	MATERIAL	CUT LENGTH	NOTES
23	1	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	13 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
24	1	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	23 1/2"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN
25	2	1.25" 1010 STEEL RD. TUBE 0.12" WALL THICKNESS	3 5/32"	TOLERANCE + 1/32", REFERENCE PART NO. 2 & 3
26	2	1" 1010 STEEL RD. TUBE 0.065" WALL THICKNESS	33"	TOLERANCE + 1/32", FOLLOW NOTCHING PATTERN



Group: Jan, Nate, Daniel

TITLE:

CHASSIS PARTS 23-26
CROSS BRACE TUBES

SIZE A | DWG. NO. 14 | DATE: 2/14

SCALE: 1:8 | DRAWN BY: JS | SHEET 14 OF 15

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2 1

B A

ITEM NO.	PART NAME	QTY.	DESCRIPTION	VENDOR	COST ($)
1	IMPACT ATTENUATOR (IA)	1	12" X 14" X10" , 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-CAR	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 A | DWG. NO. 15 | DATE: 2/14

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

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