Michigan Electric Racing Lightweight Chassis Solution

ME450 Section 3 Team 1 - Fall 2020

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Executive Summary

Members of Michigan Electric Racing (MER) will be developing a lightweight chassis solution to help increase performance and design flexibility to earn more points in competition. MER is a student-led engineering project team at the University of Michigan in Ann Arbor that participates in Formula SAE Electric competition events each year. Our ME 450 team is composed of six senior members of MER with extensive experience in performance vehicle design, and our goal is to develop a new, innovative chassis solution that will increase overall performance in static and dynamic events at the competition.

Competition placement from previous years necessitated performance upgrades to keep team goals of winning attainable. MER's first car, MER19, was the heaviest car to compete in 2019. After gathering testing data to inform our vehicle simulator, mass was highlighted as a critical variable for improvement in all dynamic events. Relative to other components on the car such as motors, motor controllers, and gearboxes, the chassis was significantly heavier than what was deemed a competitive weight. MER20 saw a significant reduction in chassis weight, but was still more than 5kg heavier than the lightest steel tube frame chassis at competition.

After determining that decreasing the mass of the chassis was a feasible way of increasing performance at competition, a list of requirements was generated to facilitate the design process. These requirements and considerations were then quantified into verifiable specifications, some of which were mandated by the 2021 Official FSAE Rules. These specifications include reducing mass by at least 6kg, maintaining a torsional stiffness of at least 1100 Nm/deg, including all of the necessary structures and features to pass rules, and ensuring cost-effective manufacturability. Additionally, the lightweight chassis solution must not compromise the current performance levels of other subsystems.

After designing a carbon fiber monocoque solution to our problem definition, we completed several tasks necessary to ensure our solution was applicable and working. Specifically, a CAD model of the chassis was created and then used in several simulations. Additionally, research and simulations were performed to assist in material selection and thickness. Based on the simulation results, an estimated total vehicle mass reduction of 5% was expected when compared to MER19. In terms of material and thickness selection, it was determined that an aluminum honeycomb core thickness of 19.05mm and 6 layers of carbon fiber on each side met all FSAE rule requirements. Finally, the torsional stiffness of the carbon fiber monocoque, found through simulation, was found to be 12305 Nm/deg which exceeds the minimum set by our requirement and specifications.

Although numerous tests and simulations were performed over the course of the semester, we believe there is still more work to be done before we build and assemble this chassis design for competition. The next steps for the project include continuing research and development of our selected design, the full monocoque chassis. This requires further development in CAD and Ansys with further analysis in aerodynamics through Star-CCM+ and composites through Creo.

Problem Description

As members of MER, we participate in Formula SAE competitions hosted across North America. Over four days, we compete against other schools in dynamic and static events that test our car's acceleration, cornering ability, overall driving performance, and design acumen of the team. The dynamic events consist of acceleration, endurance, autocross, energy efficiency, and skidpad competitions. The static events consist of an engineering design evaluation, a presentation event, and cost analysis. The total possible score for the competition is 1000 points with the breakdown in each event as shown below in Table 1.

Table 1. Possible Points available in both Static and Dynamic Events

Static Events	**Points**
Design	150
Cost Analysis	100
Presentation	75
Dynamic Events	
Acceleration	100
Skidpad	75
Autocross	125
Energy Efficiency	100
Endurance	275
Total Points	**1000**

Each year, a new car is built using data from previous years to inform lighter, stronger, more capable mechanical designs, and more robust electrical systems. The competition vehicle from virtual FSAE 2020 can be seen in Figure 1 on page 5.

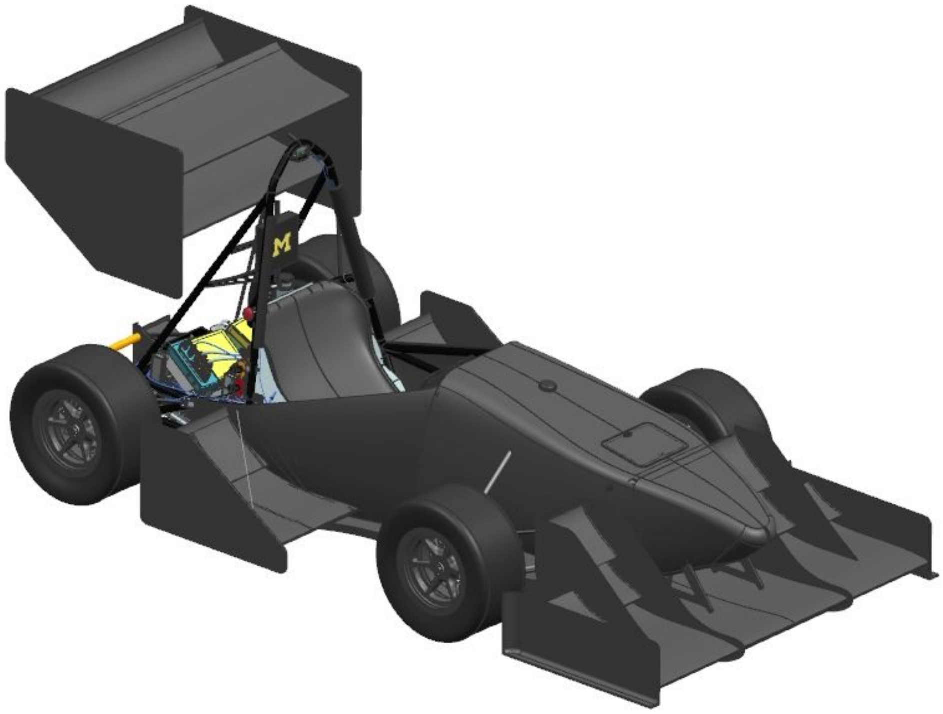

Figure 1. CAD Render of MER20

In order to score more points in both static and dynamic events, teams must design with competition-based goals in mind, for example a desired placement at the end of competition or achievable times for each event. After competitions are over, MER assesses each component of the car for areas of improvement and determines which subsystems should be given the most focus for the following year. From evaluating MER19, the chassis became the primary focus for improvement in MER20. Based on data from our simulation, reduced weight is correlated with lower lap times, which could, in turn, earn more points. Reducing weight in the chassis is not directly correlated to earning more design points, but alternatives to steel tube frame chassis typically have earned more points from judges due to the uniqueness of their solution. MER20's steel tube frame chassis was 14kg lighter than MER19's chassis, but due to the coronavirus, it was unable to be tested in competition. As a result, we have confidence in our ability to lightweight a chassis for MER21. However, we are quickly approaching the feasible rules-legal limit for steel frame chassis weight and need to explore other solutions.

MER has been developing an in-house vehicle simulation for the past two years, which has provided the background and reasoning for pursuing this project. Given the young age of the team, there is not an archive of testing data, nor are there experienced personnel to pilot new vehicle development. Consequently, a vehicle simulation was developed in MATLAB to guide each design cycle. The goal of this simulation was to generate a sensitivity plot that displays how a 5% increase or decrease in one aspect of the car affects our time in the acceleration event and a single autocross lap. Using GPS data from our autocross event in the 2019 Formula Lincoln competition, the track was imported into MATLAB using a cartesian coordinate system. Comparing the empirical vehicle speed from our 2019 Lincoln autocross lap with a simulated MER19 vehicle on the same track, there was a strong correlation in their performance characteristics as seen below in Figure 2. Furthermore, the confidence in the simulation results was strengthened by simulating the competitors' vehicles and comparing their acceleration times to the simulated times with similar results.

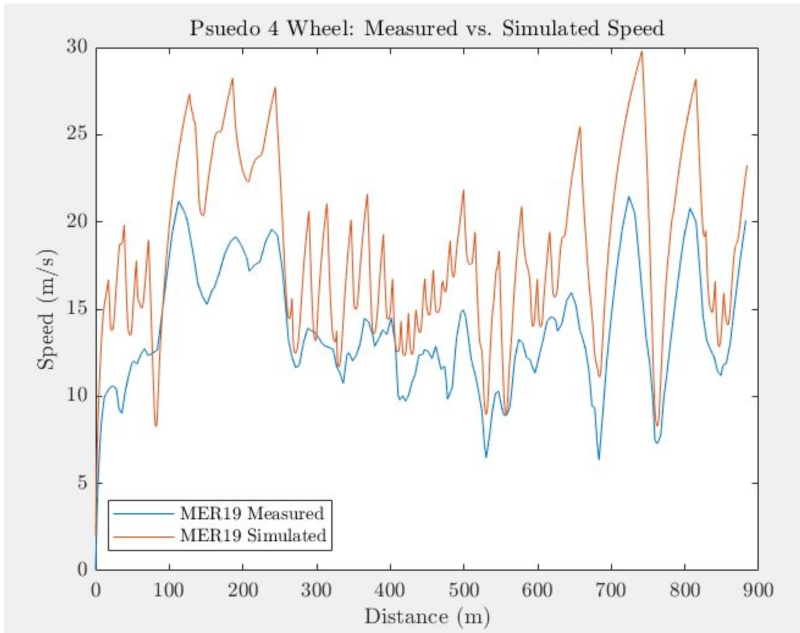

Figure 2. MER19 "Perfect Lap" and MER19 lap time from competition[12]

The simulation assumes that the tires are always at the limit of traction given the car's suspension setup. This means the tires have the best possible performance throughout the lap, making it the "perfect lap." The differences when comparing the simulated "perfect lap" to the real lap can be attributed to the fact that the simulation has perfect acceleration and braking, unlike the amateur driver of MER19. This means that the simulated speed rarely goes slower than the driver did across the lap.

Acknowledging the limitations of our autocross and lap simulation, the team felt comfortable progressing with further simulation of how attributes of the vehicle affect its dynamic performance. MER20 vehicle attributes were used for this instead of MER19 because the vehicle mass and drivetrain architecture changed drastically between the two.

Upon running the simulation, two sensitivity plots were generated. One shows the autocross lap's sensitivities, while the other shows the acceleration event. The plots in Figure 3 on page 7 show how changing certain aspects of the car affects its performance in each event.

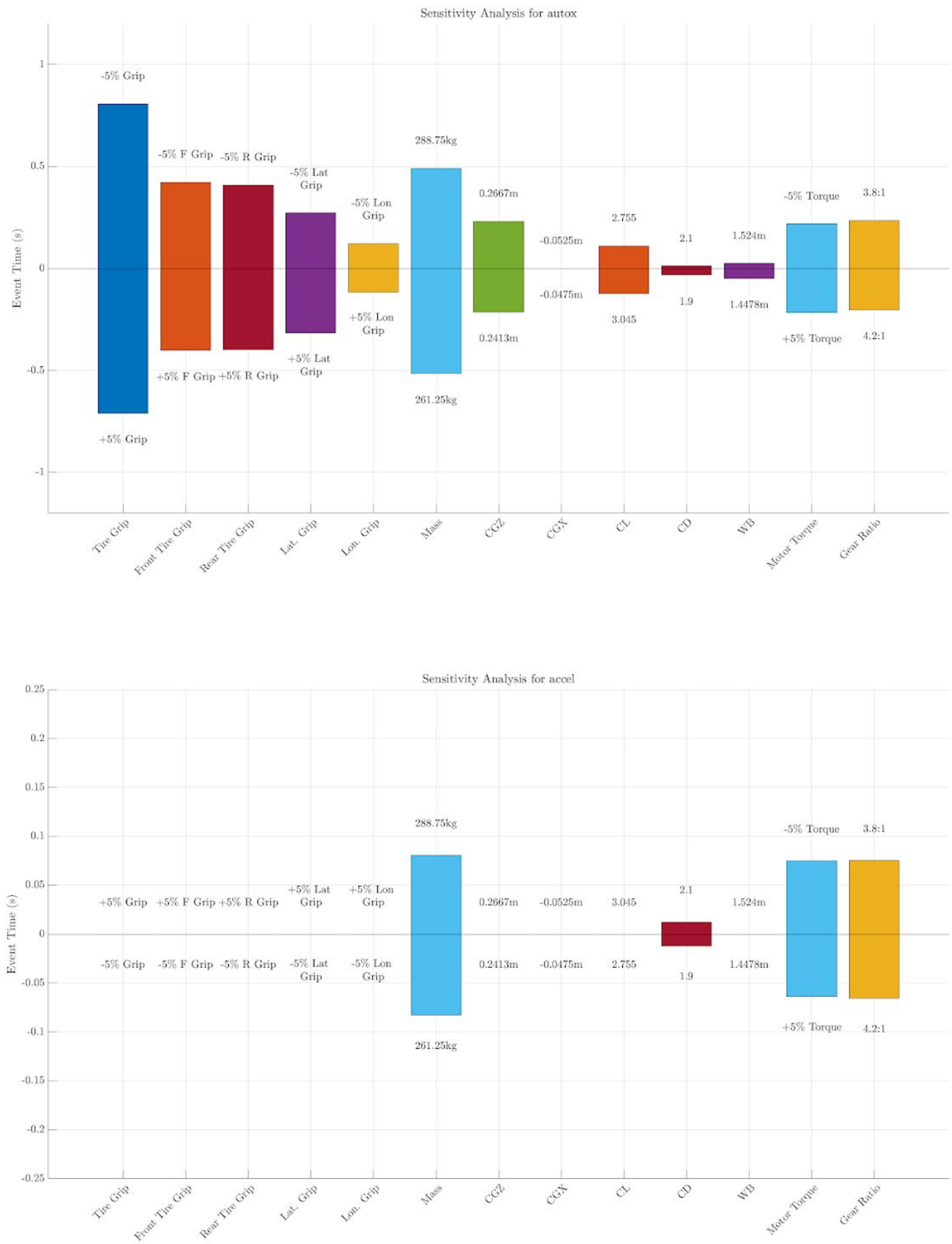

Figure 3. The top plot displays how a 5% change in a vehicle attribute affects autocross times. The second plot shows how a 5% change in a vehicle attribute affects the time in the acceleration event.[12]

Figure 3 on page 7 demonstrates the largest contributors to lap times: tire grip, vehicle mass, motor torque, and gearbox ratio. Concerning tire compounds, there are marginal differences in grip from off-the-shelf compounds available to FSAE teams as seen below in Figure 4. Tire performance is more affected by its camber, slip angle, normal load, contact patch area, and other vehicle dynamics measurables throughout the lap than its compound.

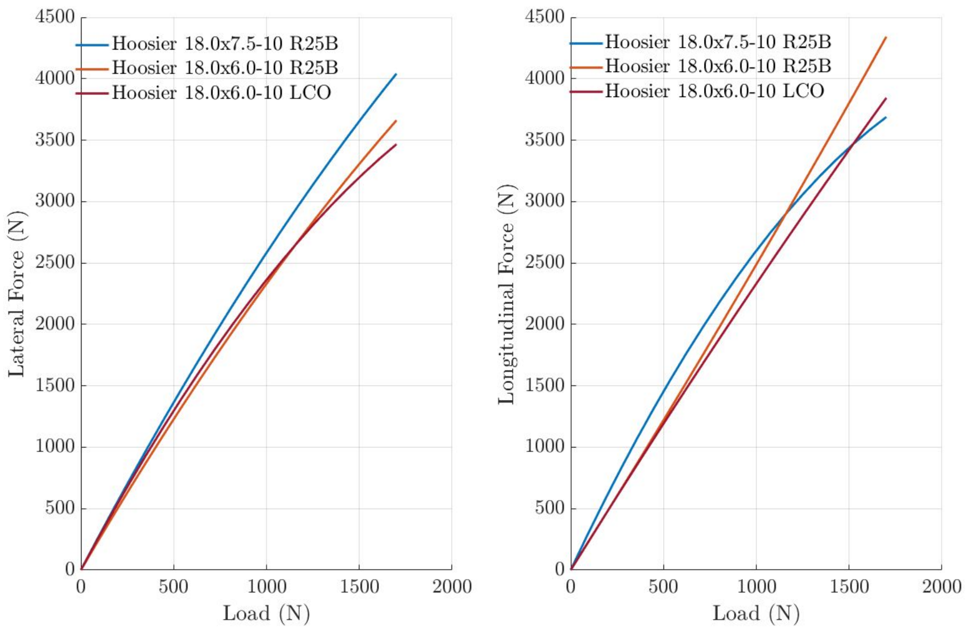

Figure 4. This figure shows the lateral and longitudinal capability of the three tires available from Hoosier Tire for use on our vehicle. The 18.0x7.5(blue) and 18.0x6.0(orange) are both R25B compounds while the 18.0x6.0 (red) is the LCO compound. The biggest differences in lines can be attributed to the larger contact patch of the 7.5 inch wide tire compared to the 6.0 inch wide tire.[12]

Motor torque is also a hard attribute to improve because it either requires larger, heavier motors, which add more mass, or a custom solution to improve upon the off-the-shelf EMRAX 188 that the team employs currently. Adding mass is detrimental, and a custom solution is not feasible for the team at this point. Gearbox ratio was selected for MER20 using a simpler simulation, but the gearbox is a single stage planetary, so changing that ratio will be a relatively trivial task and will be implemented in the MER21 vehicle. This leaves only mass as the major variable we can tune to improve the dynamic performance of the vehicle. The chassis is the single heaviest line item of our vehicle, making it the obvious choice for weight reduction.

Design Considerations

There are a few things we need to consider when designing a chassis to ensure that all of the vehicle systems are integrated harmoniously. The chassis is designed to hold all critical components of the vehicle, including the drivetrain, powertrain, and control systems, while being as stiff as possible to mitigate efficiency losses. It is also crucial that the chassis keeps the driver safe in cases of collisions or rolls. Other considerations include cost, documentation, and manufacturability of each chassis design [2].

Some of the primary design considerations come from packaging concerns. The drivetrain includes the motors, gearbox, tripods, and corners, while the powertrain includes the battery pack, motor controllers, and wiring, all of which need to fit in a rigid, weight-balanced manner within the car as shown below in Figure 5. The heaviest component in the car is the accumulator, our custom-designed and manufactured battery pack. There are special rules that dictate the structure surrounding the accumulator to make sure it is properly protected from impact. It is also crucial to keep the accumulator as low to the ground as possible to ensure a low center of gravity. A low center of gravity allows for higher lateral acceleration capability by reducing lateral load transfer. Another component of the powertrain that requires special allocation is the motor controllers. Because we are running a 2-motor architecture, we have two custom-built motor controllers that must be packaged safely.

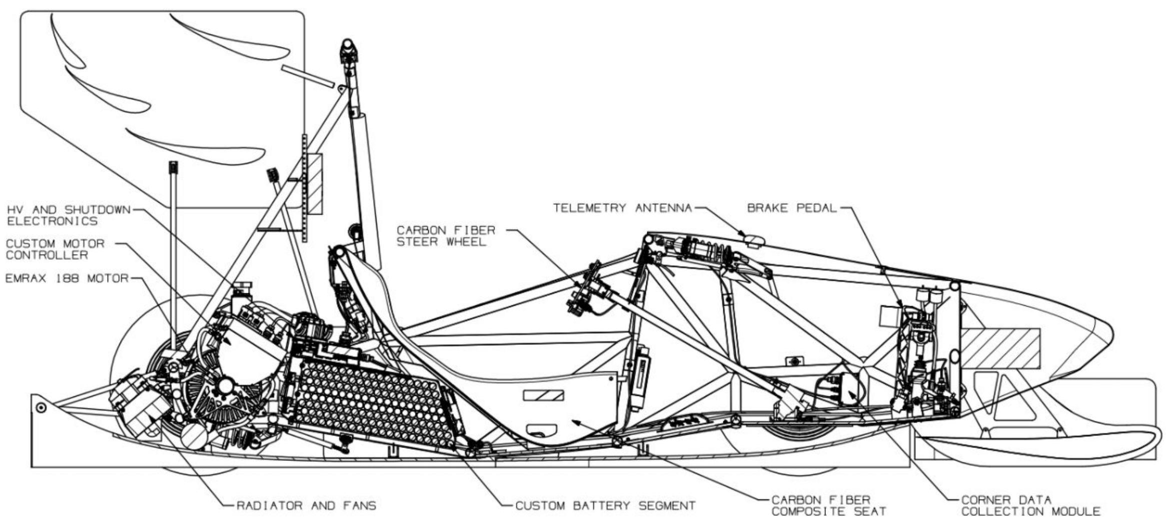

Figure 5. Right view drawing of MER20 Packaging [13]

The drivetrain includes the motors and gearboxes inside of the car. We are running an independent rear wheel drive configuration, so the motors and gearboxes must fit within the chassis in the rear of the car, behind the driver. There must also be a way for the gearbox to transmit rotational motion to the wheel hubs, requiring some kind of port or cavity on the body of the chassis such that a shaft can pass through to deliver rotational motion.

Another important consideration is the comfort of the driver. While designing an innovative, lightweight chassis, we must not sacrifice the safety or comfort of the driver in any way. Driver comfort is very important in that it allows the driver to quickly and effectively respond to road input without succumbing to fatigue from an uncomfortable position.

Literature Search and Benchmarking

We chose to benchmark from existing solutions in FSAE Electric, Formula Student (the European equivalent of FSAE), professional motorsport, and the automotive industry. Three teams were chosen for benchmarking: TU Fast, Carnegie Mellon Electric Racing, and Wisconsin Electric Racing as shown in Figure 6 on page 10. The TU Fast 2019 car was chosen because it was considered one of the best cars in the history of the competition, winning 6 out of 6 competitions at Formula Student Germany, aided by a carbon fiber monocoque (CFMC) chassis [9]. Carnegie Mellon 2018 and Wisconsin 2018 were chosen because Carnegie was a steel tube space frame and Wisconsin used a hybrid carbon fiber/steel tube space frame chassis, the two most common non-CFMC routes [10][11].

Figure 6. Left to right: TU Fast, Carnegie Mellon Racing, Wisconsin Racing

Relevant metrics for each FSAE vehicle used for benchmarking are shown below in Table 2. Weight of both chassis and full vehicle are good performance benchmarks, while design finish shows how focus on innovative chassis solutions can yield benefits in static events.

Table 2. A table of the attributes of the previous MER vehicles benchmarked to the competitors.

Car	Chassis	Chassis Weight (Kg)	Curb Weight (Kg)	Design Points	Design Finish	Overall Points	Overall Finish
MER19	Steel Space Frame	48	272	80	13th	490.6	9th
MER20	Steel Space Frame	34	204	N/A	N/A	N/A	N/A
TU Fast	CF Monocoque	25	158	129.5	1st	947	1st
Carnegie Mellon	Steel Space Frame	33	213	90	9th	799.6	1st
Wisconsin	Hybrid Chassis	29	192	150	1st	548.3	4th

From each team we referenced, we have multiple takeaways, all of which will influence our design process, and the future designs of MER. From images of each car, it can be clearly determined how mounting of components varied between chassis structures. Issues faced by MER in the past regarding packaging flexibility are a shared experience within all FSAE teams who choose to design a steel frame chassis. The hybrid chassis appeared to provide more flexibility, while the CFMC offers the greatest range of mounting freedoms.

Industry and professional racing offer insight into other creative solutions we can reference, as shown below in Figure 7. The Ford GT MkII chassis is an FIA-rules-compliant roll frame made from 15cdv6 low carbon steel [8]. While this is an example of a lightweight steel chassis solution, it is not in the tube frame form encouraged by FSAE rules. The Dallara F2 CFMC represents the most common chassis solution in open wheel racing, providing both stiffness and variable mounting opportunities [7]. The Tesla Model 3 aluminium monocoque is a metal unibody chassis option used in industry [6]. For production cars, it meets safety and stiffness needs very well, however it requires more material and weight than would be desired for competition settings.

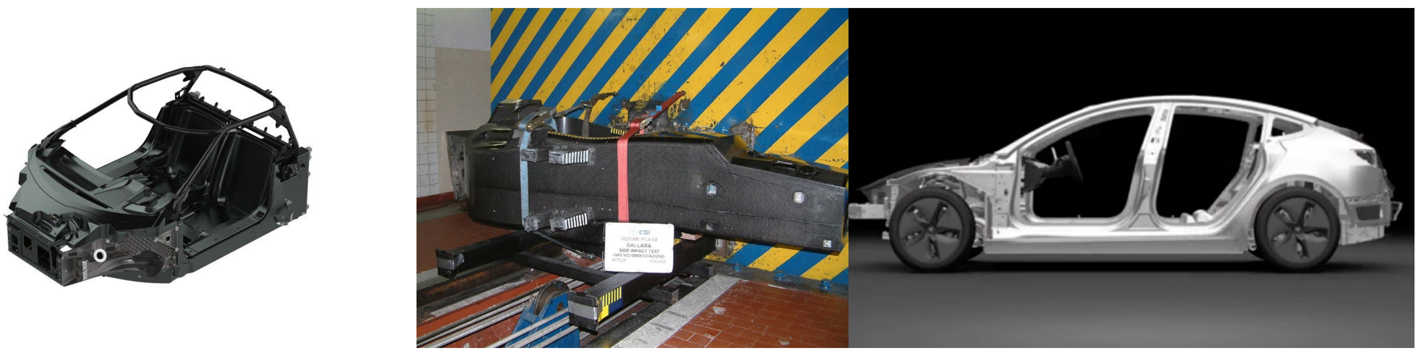

Figure 7. Left to right: Multimatic Ford GT Mk II, Dallara F2 Monocoque, Tesla Model 3

The benchmarks above provide baseline information for us to compare our design and ideas to. It can be seen from Table 2 that there is a correlation between lighter vehicle mass and overall finish, with lighter cars performing better in the dynamic events. It also highlights the effect that chassis design may have on the design score. The reason Wisconsin did not place higher in the overall competition was due to critical failures that did not allow them to run in some dynamic events. The information in Table 2 further justifies the idea that an innovative, lightweight solution can help MER score more points at competition.

The industry benchmarks give us insight as to how solutions are implemented with fewer performance and cost constraints. These solutions allow us to generate more concepts and synthesize more innovative solutions by taking advantage of the engineering principles used in commercial products and other performance applications. For instance, a possible solution is using bent sheet metal (stamped) or cast metal parts as individual members of a space frame chassis, methods used by consumer automotive manufacturers. Due to the prohibitive cost and intensive labor of creating a die for each part, this is not feasible, but using bent sheet metal to reinforce parts of a composite/hybrid chassis is now another option.

Requirements and Specifications

After comprehensive research, analysis, and benchmarking, we developed a list of requirements for a feasible lightweight chassis solution that can be implemented by a future team. MER was originally planning to develop this project over summer 2020 such that it could be manufactured and integrated by the 2021 competition, but resource and time limitations imposed by the coronavirus pandemic pushed the design cycle back by nearly 6 months.

The requirements are based on the performance goals set through simulation and competition data. Many of the requirements are inherent from the FSAE rulebook and can be easily converted into specifications. We must incorporate each of these rules in our design such that we pass the technical inspection at competition. Teams who do not pass this inspection do not get to compete, which would negate any improvements we strove to make in dynamic events. The rules are written to ensure that the drivers competing in the dynamic events will not get injured. These safety rules also extend to design and manufacturing, making sure there are not any components that can hurt the engineers, technicians, and inspectors over the course of the design cycle and competition. Because the conformity to these rules is binary, there is no quantifiable specification that can be verified; the vehicle must simply be rules-compliant at competition.

The performance requirements were translated into specifications by filtering them through various benchmarks and patterns of improvement from previous iterations of MER. A necessarily high torsional stiffness is not required for a high-performance race car chassis [3]. The stiffness must only surpass a certain threshold to mitigate efficiency losses from strain during load transfer between the suspension mounting locations. The previously-held threshold has been 1000 Nm/deg for MER, and we have not documented any negative repercussions or malfunctions leading us to believe that we have built an overly-compliant chassis. While the steel space frame chassis for MER20 has not yet been validated for stiffness, it was simulated to have a stiffness of 1100 Nm/deg. To ensure that the team does not regress in any fashion if they choose to implement our solution, we have kept this as our minimum requirement.

The primary goal in developing this solution is to reduce the overall mass of the largest component on the car. Previously, emphasis was not placed on minimizing the mass of the chassis in an effort to pass all rules. The iteration from 2019 to 2020 saw a ~25% reduction in mass, leading to an overall chassis mass of around 34kg. The lightest documented chassis we found in our research was 25kg with the top 20 teams close to that range. In order for our team to place itself among these ranks, we decided we must reduce mass by at least 6kg, placing us in an attainable range near the top. Mass reduction results in two distinct performance metric improvements, acceleration, and efficiency. The vehicle will gain acceleration from the motors having less total mass to propel. The vehicle will also see better energy efficiency from lower usage due to the reduced kinetic energy necessary to keep the vehicle at a certain speed. This comes from the relation that energy is proportional to mass.

While a majority of the points in FSAE competition events come from the dynamic events involving drivers operating the vehicles, a significant portion of the points come from static events, including an engineering design presentation, followed by judging panel scoring and subsequent team ranking. A steel space frame chassis is the default option encouraged by FSAE, which they do by restricting the use of materials to mostly steel tubes. This often results in a design score consistent with the average teams at competition. With an overall goal to win, we must perform well in every category of competition, including the design presentation and scoring.

By looking at MER's previous design scores and cross-referencing them with design scores from other teams and their chassis designs, we determined that an innovative chassis solution consistently leads to more points in design scoring, and therefore made it a requirement to increase our design score (more than 80 points), verifiable only at competition. The design points come from a culmination of everything in design. Even if we can't prove that an advanced chassis solution leads to more design points, it does lead to more design freedom for other subsystems, allowing us to be more creative when designing solutions, leading to more potential design points in more than just chassis.

Outside of designing a chassis that can be manufactured within the limitations of the team resources, the only other requirement is making sure there is enough space to efficiently mount the critical systems of the car. Ensuring that there is high mounting flexibility allows for other systems to optimize their design's for vehicle performance. For instance, if the side panels of the chassis were made of a material with which mounting points can be put anywhere, then the suspension packaging could welcome more complex designs to control more aspects of the vehicle state, such as heave. The requirements and specifications for this project are summarized in Table 3 on page 14, along with justifications for each requirement in the furthest right column.

Table 3. Requirements and Specifications for lightweight chassis solution

Reqs	Specs	Justification
Pass Strength Rules	Passes Impact Attenuator test and 3-point bend test	FSAE Rules 2021 F.4.3, F.3.2, F.8.7 Requirements [1]
Required structures	• front, main roll hoop • side impact structures • Proper roll hoop attachments, bracing • Front chassis protection • Impact attenuator	FSAE Rules 2021 F.5, F.7, F.8 Requirement [1]
Safety	Must pass FSAE competition template tests and meet cockpit safety regulations	FSAE Rules 2021 T.1 Requirement [1]
Electrically Grounded	Grounded resistance less than 5 Ohms	FSAE Rules 2021 EV.5.4 & EV.7.7 Requirement [1]
Manufacturable	Must be able to be fabricated using resources in the Wilson Student Team Project Center, and from Sponsors	We need the chassis to be fabricated to run in competition
Lightweight	Reduce chassis weight by 6 kg when compared to MER20	Aligns our chassis weight closely to average of world top 20 (~25kg)
Stiffness	A chassis torsional stiffness of at least 1100 Nm/deg	Maintain or improve stiffness of MER20
Advanced Chassis Solution	Score more than 80 points in design (total 150 points)	Improve Design score from MER19
Improve Mounting Flexibility	Increase usable area for mounting critical components(Suspension, Gear boxes, Battery pack, motors, etc)	Allows for other systems to be mounted and well-packaged without additional mass

Concept Generation

With the refined technical requirements and specifications, our group began generating concepts to solve the design problem. Our goal with concept generation was to develop a variety of ideas and compare them against each other to determine which would best meet our requirements. In order to explore the full solution space, we used multiple concept generation methods including functional decomposition, a morphological matrix, benchmark reviewing, and brainstorming.

Functional decomposition allowed us to generate a functional tree for our project. Although our requirements and specifications were already confirmed, a functional tree is another method of verifying the goals of the project from the scope of how the final project should function. We began by listing the overall functions of the final product, then describing sub-functions or design elements that were desirable for each. This helps to remember what elements of the project affect the overall functions so that every element is considered in our design. The functional tree can be seen below in Figure 8 on page 15.

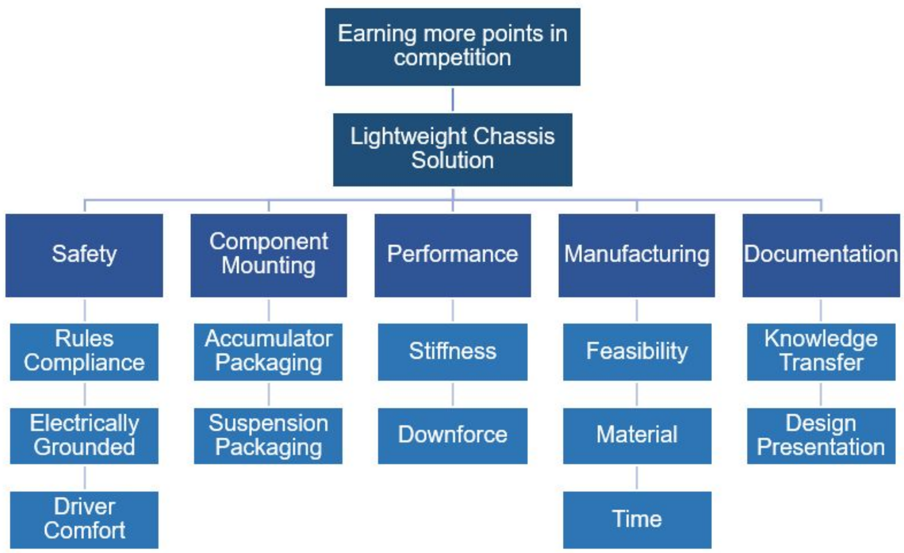

Figure 8. Shown above is a functional tree for a lightweight chassis solution design for MER. Project and design functions are shown in darker blue and subfunctions or elements of the design are shown in lighter blue. This tree helps ensure that when design decisions are made, any effects that decision may have on other components are considered.

The functions listed in the functional tree were then used to generate a morphological matrix. A morphological matrix was used to develop solutions for each function. The functions were listed vertically along the first column and the concepts horizontally along each row, resulting in multiple partial concepts for each function. The morphological matrix developed for this project can be seen in Table 4 on page 16.

Table 4. Morphological Matrix

Functions	1	2	3	4	5
Mounting Freedom	Aluminum Inserts	Steel Tabs	JB Weld	More tubes/nodes	Additional Plates
Lightweight	Thin walled steel tubing	3D printed aluminum nodes	Carbon Fiber paneling	Carbon Fiber tubing	
Downforce	Carbon Fiber panels	Molded body (monocoque)	No aero package	Add ballast mass	Additional wings
Safety	Steel roll cage	Airbags	Crumple Zones	Rigid seat belt mounting	
Rules Compliance	Following the rules	Cheat	Bribe Judges	Modify	
Stiffness	Carbon Fiber	Lots of nodes (steel tube)	Titanium	Aluminum	
Electrical Grounded	Steel tube frame	Inserts	Aluminum honeycomb		
Design Points	Material usage	Heavily involved design	Complex & Justified		

We used a morphological matrix to develop our concepts. We chose this method to generate a wide variety of ideas for each aspect of our solution. Compared to other concept generation techniques, such as TRIZ, SCAMPER and utilizing design heuristic cards, the morphological matrix allowed us to easily and quickly organize potential solutions without restricting creativity. The act of rapid-firing ideas actually led to a more comprehensive and diverse set of potential solutions than if we had gotten too detailed with individual ideas. From this matrix, we are now able to identify partial solutions and dive into greater detail.

We then reviewed our benchmarking to remind ourselves of the industry methods in chassis design and our competitors. The benchmarking summary table can be found below in Table 2 on page 10.

Significantly, what we took away from the benchmarking review was that hyper and formula vehicles typically use a carbon fiber monocoque.

We brainstormed ideas directly using industry solutions and competitors as inspiration, we generated a list of potential chassis solutions which can be found below in Table 5.

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Table 5. List potential chassis solutions

Type of Chassis	Potential Material
Space Frame	Steel
Space Frame	Aluminum
Space Frame	Titanium
Hybrid	Front composite, rear metal space frame
Hybrid	Front metal space frame, rear composite
Hybrid	Bottom composite, top space frame
Hybrid	Composite tubes, machined metal interface nodes
Monocoque	Machined billet of metal
Monocoque	Kevlar
Monocoque	Carbon Fiber

From this list, we have preliminarily identified three potential solutions: a steel space frame, a front composite/rear space frame hybrid, and a carbon fiber monocoque. Aluminum was not selected for the space frame because aluminum has roughly ⅓ of the elastic modulus of steel, so the weight savings from the lighter density is replaced with additional material to meet stiffness requirements. Titanium was not selected because even though its strength to weight ratio is better than steel, the increase in cost of 5-7 times is not worth the increase in performance. The front composite/rear space frame makes the most sense of the hybrid solutions for our needs because if the front is a composite monocoque then the curb weight will decrease. This is because the monocoque replaces body panels & some aerodynamic components, and front mounting flexibility increases. The rear space frame in the hybrid chassis is heavier than a full monocoque, but it makes passing rules easier because of required roll hoops rules. Lastly, out of the potential monocoque materials carbon fiber is the best solution because it has the highest stiffness to weight ratio, and it could feasibly be manufactured by our team, unlike a machined chassis from a large billet of metal [4].

Concept Evaluation / Selection

To evaluate our design ideas, we focused on the three main options that could be made to meet every requirement in some fashion: space frame, hybrid structure, and a monocoque. A space frame chassis is a skeleton-like structure of connecting tubes/members that form a frame to which components can be mounted. A hybrid solution combines two or more structural subsystems (such as tubes and composite) to make the chassis. Finally, a monocoque chassis is made of one cohesive piece and also forms the exterior of the vehicle.

A steel space frame chassis could be made to pass rules easily, as we have experience making space frame chassis. Electrical resistance for grounding would not be an issue since it is made of metal, and additional testing for panel stiffness would not be necessary, reducing complexity overall. However, a space frame chassis is limited to how light it can safely be designed for the EV class because of side impact structure rules. A space frame chassis is relatively easy in its construction with it being the easiest to fabricate out of the three solutions. In terms of innovation, the steel space frame has been used for over 30 years in FSAE competition and has been refined continuously as competition rules have evolved, making it hard to gain design points if our team elected to use it. Also, the skeletal nature of a space frame chassis makes mounting and packaging difficult because structural parts are limited to mount to where the nodes are.

A carbon fiber composite front and rear steel tube space frame hybrid chassis would satisfy our requirements in Table 3 on page 14, but there are concerns in terms of manufacturing. This hybrid chassis would be harder to manufacture than a full space frame chassis, as we would need to lay up the composite portion, which includes fabricating and preparing the mold, while also welding the rear space frame steel tubing. Grounding would not be an issue, as half the chassis is steel, and a conductive layer can be added to the composite part of the chassis. This solution would be made lighter than a full space frame chassis, since a significant portion of it would be made of carbon fiber composite material instead of steel. A hybrid chassis would certainly get more points in design when compared to the conventional space frame, as it is a more advanced solution and more innovations with other systems can be made. However, while front mounting would be made easier due to the nature of the composite chassis, rear mounting would not be improved. The space frame portion would need to accommodate both suspension mounting and internal mounting for the motors & accumulator, which would make that design difficult to package. The biggest downside to this design is the region of the chassis connecting the space frame and the monocoque would be a liability for the chassis torsional stiffness. When FSAE teams have elected to go for a hybrid chassis instead of a full monocoque it has usually been because of thermal concerns from their large internal combustion engines. Since we use an electric powertrain, we do not share those same concerns.

A full composite monocoque chassis could be made to meet all of the requirements in Table 3 on page 14. The monocoque would be easier to manufacture than the hybrid solution, as we do not need to manufacture two types of structures. Electrical grounding could be done with a conductive layer in the composite just like the hybrid chassis. The full carbon fiber monocoque would be lighter than the hybrid chassis due to the lack of space frame tubing in the back and lack of hardware to join the two halves. This would also make the full monocoque stiffer than the hybrid. Additionally, a full composite monocoque would score more points than MER19 in design as it is a more advanced chassis solution. Finally, it allows for full flexibility in mounting. Components can be placed anywhere along the body through insert reinforcements. There are no nodes that dictate favorable locations to apply force and mount components. Mounting flexibility can also have implications in center of gravity placement, allowing us to refine our vehicle dynamics even more. Lastly, a monocoque would decrease the curb weight more than the other two proposed solutions by reducing mounting weight and eliminating a considerable amount of aerodynamic components such as side panels, bulkhead, and part of the undertray [5].

We decided to rank our concepts relative to one another with scores of “good”, “neutral”, or “bad.” Good represents an option which is significantly better than the others, and bad means the option is notably worse to the point that it is unacceptable. This category considers how novel and interesting the concept is. These rankings are summarized in this color-coded Table 6 on page 19. The colors allow us to quickly see which options were the less or more favorable.

Table 6. Chassis Design Evaluation Matrix

	Weight	**Manufacturing**	**Stiffness**	**Cost**	**Mounting Flexibility**	**Innovation**
Space Frame	Bad	Good	Neutral	Neutral	Bad	Bad
Hybrid Chassis	Neutral	Bad	Bad	Bad	Neutral	Neutral
Full Monocoque	Good	Neutral	Good	Bad	Good	Good

This matrix is similar to a Pugh chart, but it doesn't have scores for rankings or category weights. We chose to avoid weighting each category because they are all critical to our vehicle’s performance at competition. Should two concepts score similarly, we would weigh the manufacturability and cost more, as those are the limiting factors for our team’s ability to implement those solutions. Based on the scoring distribution of each chassis solution, we have decided to use the Full Monocoque for our lightweight chassis solution.

Solution Development and Verification

With a design concept selected, the next step is to begin system design and start making design decisions (that will likely be iterated) that will carry through to the final product. All chassis design begins with the driver because driver safety is the number one priority in performance racing. FSAE’s basic 2D driver template, nicknamed Percy, represents a 95th percentile male driver that the chassis must be able to accommodate. This is shown in Figure 9 on page 20 [1]. The position of the driver can be manipulated, but the listed dimensions cannot. Along with this driver, there are two templates that must be able to pass through portions of the vehicle without being obstructed by any permanently-mounted components as shown below in Figure 9. The cockpit template must be able to pass through the entire cockpit vertically between the required front and main roll hoops and the bulkhead template must be able to pass through the bulkhead (where the driver's legs go) from the front roll hoop to the pedals without any obstructions.

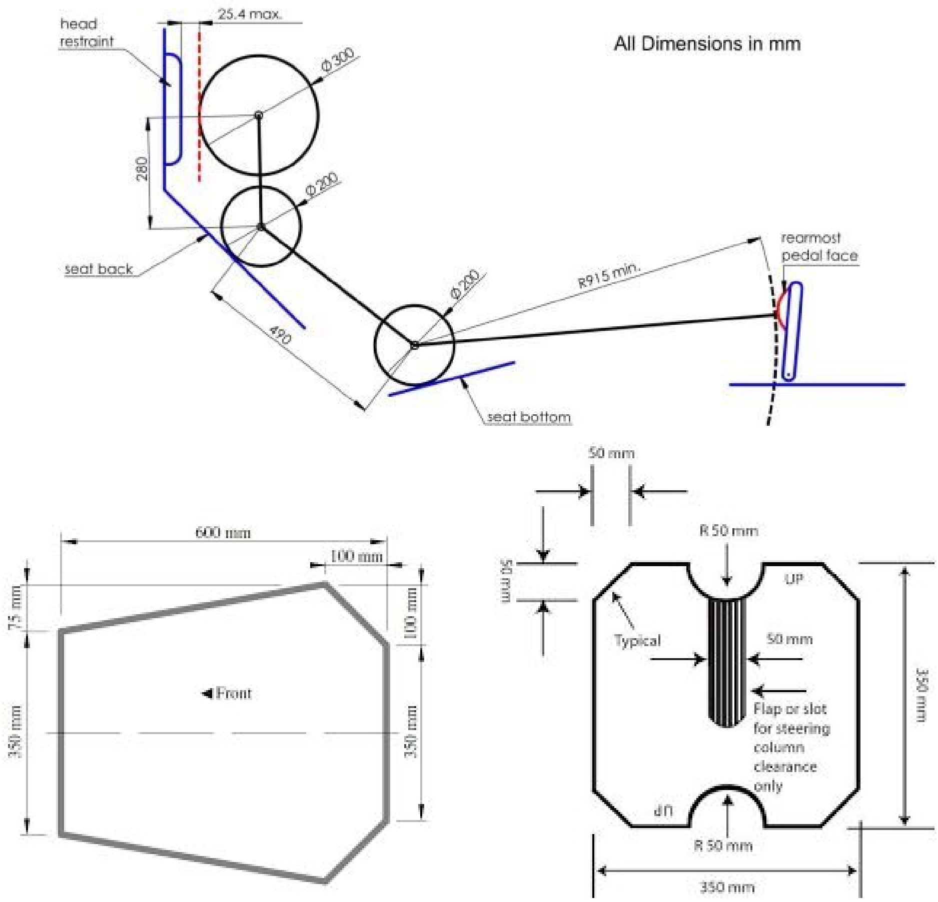

Figure 9. The top image shows the driver template (Percy). The position can be manipulated but the size cannot be changed. The bottom left image shows the cockpit template and the bottom right shows the bulkhead template, both with dimensions.

Along with these templates, there are a number of material usage rules and keep out zones that must be considered. These constraints provide us with a "design zone," or a bounded 3D space that we can have chassis parts in. This is exactly how CAD design will begin. We'll draw a basic 2D shape around Percy and the templates, and then begin applying other constraints and required structures until we have a reasonable preliminary 2D design. From there, we'll extend/extrude parts of the drawing to make it 3D and keep applying constraints as they become relevant, while using metrics such as component mass, component volume, and suspension points to dictate where there needs to be a place to mount. Chassis design elements related to vehicle component interaction are shown in Figure 10 on page 21.

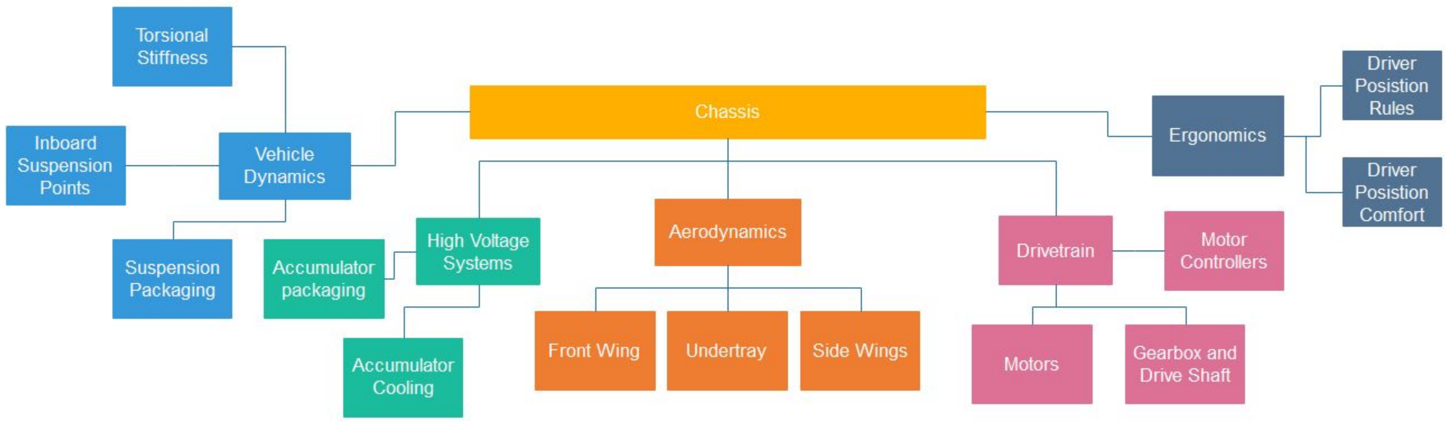

Figure 10. The design tree above shows vehicle interactions that directly affect chassis design decisions. This tree will help us make sure that we do not neglect any vehicle systems in chassis design that compromise performance or other non-chassis design goals.

Chassis is not only influenced by our competition rules but also from the other divisions on Michigan Electric Racing: Vehicle Dynamics, High Voltage Systems, Aerodynamics, Drivetrain, and Ergonomics. Ergonomics mostly deals with items mentioned above about driver position but also driver comfort. The chassis must also package the drivetrain suite: motor, motor controllers, and gearbox while allowing a half shaft to pass through for power to reach the wheels. Aerodynamics play a role because the chassis will interact with the performance of the front wing, side wings, and undertray. Additionally the chassis must package the battery pack and control systems (Accumulator) and must meet the cooling needs of the chassis. Lastly vehicle dynamics will greatly affect the design of the monocoque through suspension packaging and inboard suspension points. The chassis can also affect the performance of the suspension system if it does not meet the torsional stiffness requirements so it is also listed.

Risk Assessment

At the onset of this project, certain risks were understood as paramount to our design considerations. Firstly, the safety of the driver was critical not only for passing rules, but for dynamic events as well. Certain structural strength and stiffness requirements were to be met to enable participation, and those same requirements would ensure the safety of our driver in the event of a crash. Component failure could also result from fatigue stresses. Fatigue failure during a dynamic event would result in a loss of points and potential injury to the driver. This risk must be mitigated by engineering components with a sufficient safety factor, in our case 1.5. This was chosen so as to not add unnecessary weight and hinder performance while still ensuring driver safety.

Detailed Design Solution

This section will explain the process of how the monocoque in Figure 11 on page 22 was designed and what the optimization process will entail regarding the outlined specifications.

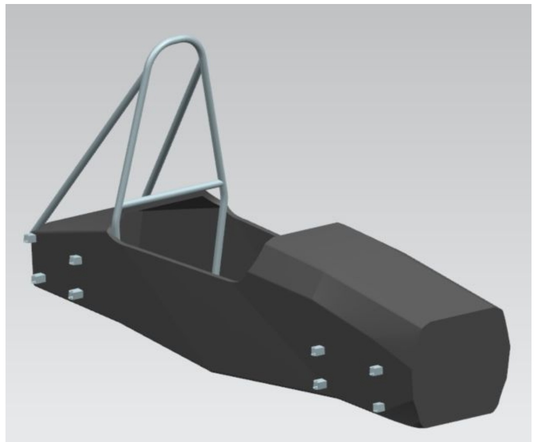

Figure 11. The figure above shows the first revision of the proposed monocoque design in CAD.

CAD of the monocoque started by looking at the requirements that directly influence its physical design as shown below in Table 7. These requirements were prioritized based on their relative importance to meeting our competition goals. The two requirements closest to the top are specified by competition rules and are not flexible. For example, if the chassis does not pass the safety or template rules, we will not get a chance to compete at competition. These requirements will be implemented with additional safety factors as they are the most critical. The requirements at the bottom are adjustable based on other design considerations. When looking at something like mounting flexibility, an inherent benefit of a monocoque over a space frame, it has been ranked fourth because we understand that the departure from a space frame chassis will satisfy most components of this requirement.

Table 7. Requirements and specifications for the lightweight chassis solution

Priority	Reqs	Specs
1	Required structures	● Front & main roll hoop ● Side impact structures ● Proper roll hoop attachments & bracing ● Front chassis protection ● Impact attenuator
2	Safety	Must pass FSAE competition template tests and meet cockpit safety regulations
3	Manufacturable	Must be able to be fabricated using resources in the Wilson Student Team Project Center, and from sponsors
4	Improve Mounting Flexibility	Increase usable area for mounting critical components (suspension, gear boxes, battery pack, motors, etc)
5	Lightweight	Reduce chassis weight by 6 kg when compared to MER20
6	Stiffness	Must have a chassis torsional stiffness of at least 1100 Nm/deg

Using these requirements listed in Table 7 on page 22, the design began around the required templates and safety rules. The safety rules that dictate the shape of the chassis and set guidelines for minimum cockpit size and height of side impact structures can be seen in Figure 9 on page 20. Figures 12 and 13 show that the current carbon fiber monocoque design passes the templates and rules highlighted previously.

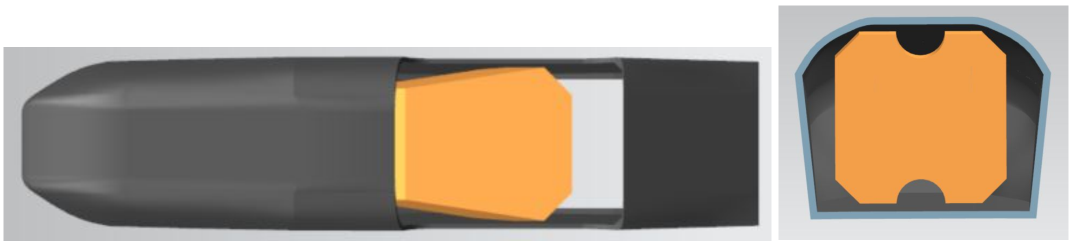

Figure 12. Shown on the left is a top down view of the bulkhead template passing through the opening of our cockpit. Shown on the right is a cross section view of the bulkhead template in the bulkhead of the chassis.

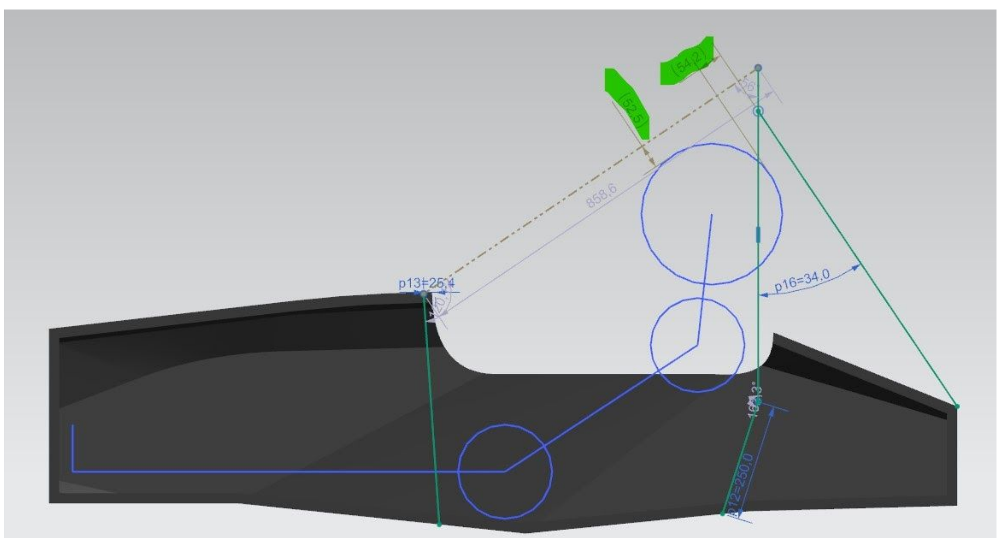

Figure 13. The figure above shows "Percy," the 95th percentile male mentioned in Figure 9 on page 20. This cross section also shows the Percy is protected in a roll over accident by being more than 50 millimeters from both the line connecting the roll hoops and roll hoop supports as highlighted.

After satisfying rules that dictate the minimum dimensions of the chassis, packaging optimization can begin. This involves placing spatial representations of major components of the car in the proposed monocoque. Figure 14 on page 24 shows some of these major components arranged in the current design: driver (Percy), accumulator, drivetrain, and suspension. Since the team did not design a new accumulator or drivetrain for 2021, the MER 20 systems were used as representative volumes.

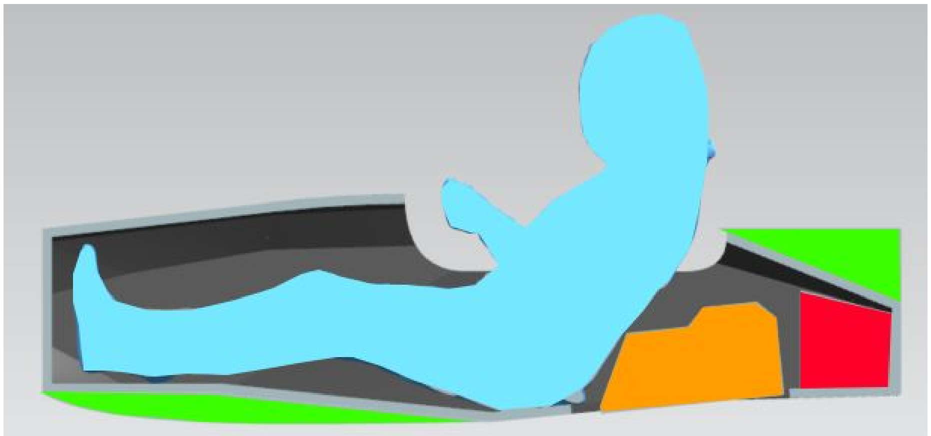

Figure 14. A cross sectional view of the volumes of major packaging concerns for the monocoque. Green represents suspension packaging, orange is the accumulator and blue is the drivetrain. Percy is included in this cross section to show that the driver will fit even with these components.

The requirement to improve mounting flexibility applies to all major systems, but it is targeted at the suspension system. While the chassis currently meets the mounting requirement for the other major systems, further analysis is required to verify that suspension mounting flexibility is improved when compared to MER 20, Figure 15 shows how this was done.

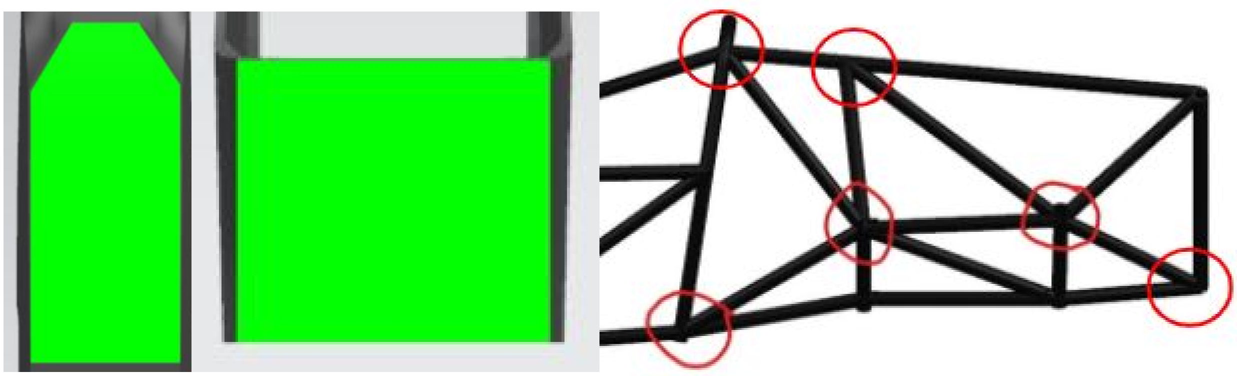

Figure 15. Shown on the left is the monocoque usable surface area for suspension mounting. Shown in the center is the area for rear suspension mounting. Shown on the right is the usable MER 20 suspension mounting points. The surface area was found by going out 4” from the center of the node.

Referencing Figure 15, 4” from each node was determined through previous years’ simulation data to be an acceptable distance based on the amount of deflection a tube experienced under suspension loading. Deflection increased as the loads were applied farther from the node. It was found that mountable surface area increased by 205% in the front and 211% in the rear. With the rules and packaging requirements met, we shifted focus to meeting the weight goal. Because the internal structure of the composite has not been finalized, variable thickness does not currently exist in the CAD model, and therefore an average ply thickness and core thickness was chosen to provide a representative weight.

Table 8. Densities and masses of various materials used to construct the monocoque. Total mass is 23.4kg.

Material	Density (g/cm^3)	Mass (kg)
Prepreg (30% Epoxy, 70% Carbon Fiber)	1.55	14.3
Aluminium Honeycomb	0.070	2.1
Chromoly Steel Roll Hoop Structure	7.85	7

The total mass shown in Table 3 on page 14 is well below the requirement of 27 kg for total chassis mass. However, that does not include the aluminium mounting inserts for major components. A high estimate of this hardware would be around 5 kg. This puts the chassis at 28 kg and exceeds the target value. Since the mass requirement is not met, the chassis will be analyzed to see where safety factors can be reduced for template, or optimize mass in different areas that don't need as high rigidity.

The chassis shown in Figure 12 on page 23 was then reviewed with the Michigan Electric Racing Aerodynamics lead to refine the aerodynamic design of the chassis. It was decided that the chassis should narrow at the nose to provide more air to flow to the side of the chassis where MER has side wings. Additionally there was concern about running a "high nose" aerodynamic concept with the nosecone of the car. Since the nose cone does not fall in the scope of this project it was not optimized for the purpose of ME 450 but the change in concept was reflected in the new front of the chassis.

The lines of the chassis were also changed to make manufacturing the monocoque easier with the curves of the bulkhead being changed to straight lines to accommodate the aluminum honeycomb not being able to bend tight radii. Additionally it was discovered that the rear of the monocoque would need to be closed for rules so the rear is now a solid panel.

Engineering Analysis

FSAE Structural Equivalency Rules

In order for our full carbon fiber monocoque chassis to pass FSAE rules, it needs to be structurally equivalent to a steel space-frame chassis. The summary of all the performance requirements for certain chassis applications is shown in Table 9 on page 26. In order to prove that these requirements are met, composite sandwich panels must be manufactured and tested. The required tests are 3-point bend test and perimeter shear. For this project, we want to focus on meeting the requirements of major components of the chassis, such as the side impact structure and front bulkhead. The relevant equivalency rules, which pertain to the major chassis components, are highlighted in Table 9 on page 26. Thus, the two primary tests to be conducted for this project is the 3 point bend test and perimeter shear test with the lap joint adhesion test remaining unnecessary for the time being.

Table 9. FSAE Structural Equivalency Testing Rules

<table>
<tr><th colspan="2" rowspan="2">Rule/requirement/car location</th><th rowspan="2">Type</th><th colspan="2">Performance requirement</th></tr>
<tr><th>Property</th><th>Value</th></tr>
<tr><td rowspan="3">Side impact structure</td><td rowspan="3">Side impact zone</td><td>Bending</td><td>Buckling modulus</td><td>3 baseline steel tubes</td></tr>
<tr><td>Bending</td><td>Energy absorption</td><td>2 baseline steel tubes</td></tr>
<tr><td>Puncture</td><td>Minimum shear force</td><td>7.5 kN</td></tr>
<tr><td>Floor</td><td>Floor</td><td>Bending</td><td>Buckling modulus</td><td>1 baseline steel tube</td></tr>
<tr><td rowspan="3">Front bulkhead</td><td rowspan="2">Support</td><td>Bending</td><td>Buckling modulus</td><td>1 baseline steel tubes</td></tr>
<tr><td>Puncture</td><td>Perimeter shear strength</td><td>> 4 kN</td></tr>
<tr><td>Bulkhead</td><td>Bending</td><td>Buckling modulus</td><td>6 baseline steel tubes</td></tr>
<tr><td rowspan="2">Monocoque attachments</td><td>Primary structure</td><td>Attachments</td><td>Maximum force</td><td>> 30 kN</td></tr>
<tr><td>Impact attenuator</td><td>Attachments</td><td>Maximum force</td><td>8mm bolts (X4)</td></tr>
<tr><td rowspan="4">Driver harness attachments</td><td>Shoulder belts</td><td>Attachment</td><td>Maximum force</td><td>13 kN</td></tr>
<tr><td>Lap belts</td><td>Attachment</td><td>Maximum force</td><td>13 kN</td></tr>
<tr><td>Anti-submarine belts</td><td>Attachment</td><td>Maximum force</td><td>6.5 kN</td></tr>
<tr><td>Lap + anti-submarine</td><td>Attachment</td><td>Maximum force</td><td>19.5 kN</td></tr>
</table>

The chassis needs to have the same buckling modulus as the steel tubes that make up the different parts of the chassis. With this in mind, the baseline values of the buckling modulus required for each area of the chassis can be calculated and are shown below in Table 10.

Table 10. Calculation of Buckling Modulus Requirement for each Chassis Area

Chassis section	Requirement
Side Impact Zone lower side	Buckling Modulus = One Size B baseline steel tubes 1*(200*10^9 Pa)*(8.509*10^-9 m^4) = **1.7018 [Kpa*m^4]**
320mm Above the floor of the side impact zone	Buckling Modulus = Two Size B baseline steel tubes 2*(200*10^9 Pa)*(8.509*10^-9 m^4) = **3.4036 [Kpa*m^4]**
Front Bulkhead	Buckling Modulus = Two Size B baseline steel tubes 2*(200*10^9 Pa)*(8.509*10^-9 m^4) = **3.4036 [Kpa*m^4]**
Front Bulkhead Support vertical side	Buckling Modulus = One Size C baseline steel tube 1*(200*10^9 Pa)*(6.695*10^-9 m^4) = **1.3390 [Kpa*m^4]**
Front Bulkhead Support one side	Buckling Modulus = Three Size B baseline steel tubes 3*(200*10^9 Pa)*(8.509*10^-9 m^4) = **5.1054 [Kpa*m^4]**

As can be seen in Table 10 on page 26 on the previous page, there are different tube sizes mentioned for each area of the chassis. This is due to the FSAE rules requiring specific chassis steel tube sizes for each application as shown below in Figure 16.

	Application	Steel Tube Must Meet Size per F.3.4:	Alternative Tubing Material Permitted per F.3.5 ?
a.	Front Bulkhead	Size B	Yes
b.	Front Bulkhead Support	Size C	Yes
c.	Front Hoop	Size A	Yes
d.	Front Hoop Bracing	Size B	Yes
e.	Side Impact Structure	Size B	Yes
f.	Bent Upper Side Impact Member	Size D	Yes
g.	Main Hoop	Size A	NO
h.	Main Hoop Bracing	Size B	NO
i.	Main Hoop Bracing Supports	Size C	Yes
j.	Driver Restraint Harness Attachment	Size B	Yes
k.	Shoulder Harness Mounting Bar	Size A	NO
l.	Shoulder Harness Mounting Bar Bracing	Size C	Yes
m.	(EV) Accumulator Protection Structure	Size B	Yes
n.	Component Protection	Size C	Yes
o.	Other Structural Tubing	Size C	Yes

Figure 16. Steel Tube Requirements for each Chassis Application [1]

The values used in Table 10 on page 26 to calculate the Buckling Modulus are also from the FSAE rulebook. The dimensions and specifications for each tube size are shown in Figure 17.

F.3.4.1 Minimum Requirements for Steel Tubing

A tube must meet <u>all four</u> minimum requirements for each Size specified:

	Tube	Minimum Area Moment of Inertia	Minimum Cross Sectional Area	Minimum Outside Diameter or Square Width	Minimum Wall Thickness	*Example Sizes of Round Tube*
a.	Size A	11320 mm^4	173 mm^2	25.0 mm	2.0 mm	*1.0" x 0.095"* *25 x 2.5 mm*
b.	Size B	8509 mm^4	114 mm^2	25.0 mm	1.2 mm	*1.0" x 0.065"* *25.4 x 1.6 mm*
c.	Size C	6695 mm^4	91 mm^2	25.0 mm	1.2 mm	*1.0" x 0.049"* *25.4 x 1.2 mm*
d.	Size D	18015 mm^4	126 mm^2	35.0 mm	1.2 mm	*1.375" x 0.049"* *35 x 1.2 mm*

Figure 17. Specifications of Different Steel Tube Sizes [1]

Simulation Validation

After identifying the baseline values needed to satisfy structural equivalency rules, we proceeded to pinpoint the ideal sandwich panel stackup that would satisfy all the requirements through simulations. The software we used for panel FEA simulation was Creo, a 3D CAD program with analysis features. Creo was chosen based on the prior knowledge available to us from a Michigan Electric Racing alumnus, Grace Stridick. Grace completed her Engineering Honors project on how varying weave orientations affect composite panel strength, which lended itself quite well to our project. Figure 18 below shows one of the simulation results from her final report in Creo.

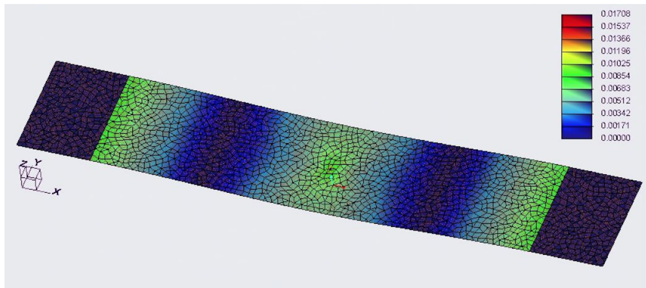

Figure 18. Example Image of Creo Composite Simulation Result [14]

Grace used a strain value of 0.017 [14] as the failure reference for carbon fiber-epoxy structures. This value was provided by a University of Michigan Properties of Advanced Materials professor. Due to the lack of availability of composite strength data (typically obtained from a lab test), this strain value is necessary to validate our simulation data. A tradeoff associated with using software instead of lab equipment to complete the strength equivalency tests is increased error. Our simulation does not account for layup tolerances, epoxy/hardener mix ratio error, and dimensional tolerances on stock. This error would exist in a physical test, as the effects of each source would be reflected in the data output. The material properties used in the simulation were from data sheets, and they may not reflect the exact properties of the materials we would use. The source of the fabric and resin depends on availability and costs prior to manufacturing. Due to the sole method of validation currently being simulation, strain failure will be assessed on a variety of loads and carbon fiber panel stackups, in turn, requiring a significant time contribution to complete the myriad of necessary simulations.

After determining that Creo was the best software to complete the composite analysis, we needed to verify our process and assumptions with an actual test and results. The verification process was based on an article titled “Structural Equivalency Analysis” by Muhammad Yaqoob [15]. The article provided data and results from a physical perimeter shear test on four different plain weave carbon fiber layups. The results showed the 4-ply layup (our comparison value) failed at a load of roughly 4800N. Using the same experimental setup, we simulated the perimeter shear test in Creo across varying loads until we achieved a strain close to 0.017. A snapshot of the simulation results can be seen in Figure 19 on page 29.

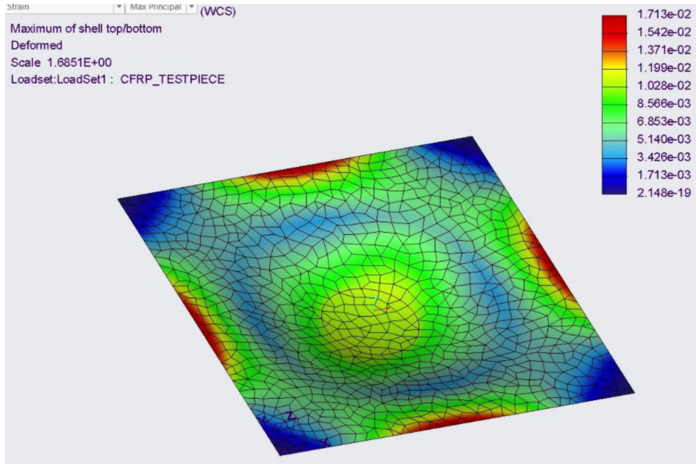

Figure 19. Simulation Result of Perimeter Shear Test

Based on the figure, we observed that an applied load of 4850N had a strain value of 0.0171. The similarity of these results validates our simulation process and parameters and allows us to continue simulating and confidently report simulation results for the test requirements.

Three-Point Bend Test Simulation

As mentioned previously, the chassis needs to have the same buckling modulus as the steel tubes that make up the different parts of the chassis. The data needed to show this required equivalency can be obtained by performing a 3-point bend test. According to FSAE rules, teams that are using a composite structure for the side impact structure and front bulkhead must build a flat panel of size 500 X 275 mm and perform a three-point bend test on said panel. The panel must also be supported by a span distance of 400mm and have a metallic load applicator with a radius of 50mm. An example of a three-point bend test assembly is shown below in Figure 20.

Figure 20. Three-point Bending Test Assembly [15]

Using the requirements of the three-point bend test, the same test was modeled in Creo. The plan for the simulation was to first apply sequential loads to obtain a strain failure close to a value of 0.017. The force and displacement needed to calculate Young's modulus (E) and the buckling modulus were then recorded.

The buckling modulus is determined by the product of the Young's modulus (E) and the second moment of area (I). Using the traditional beam theory equation, Young's modulus, E, was calculated as shown in Eq.1,

$$E_{bend} = \frac{L^3F}{48I\delta} \qquad \text{(Eq.}\mathbf{1}\text{)}$$

where E_{bend} is the Young's modulus, L is the length of the panel, F is the load applied on the panel, I is the second moment of area, and δ is the deflection of the panel. The calculation of the second moment of area is shown in Eq.2,

$$I = \frac{1}{12}wh^3 \qquad \text{(Eq.}\mathbf{2}\text{)}$$

where w is the width of the panel and h is the distance between the facing skin centres. An example of performing a hand calculation with these prior mentioned equations in order to determine the buckling modulus for a 2 ply sandwich panel is shown below in Figure 21.

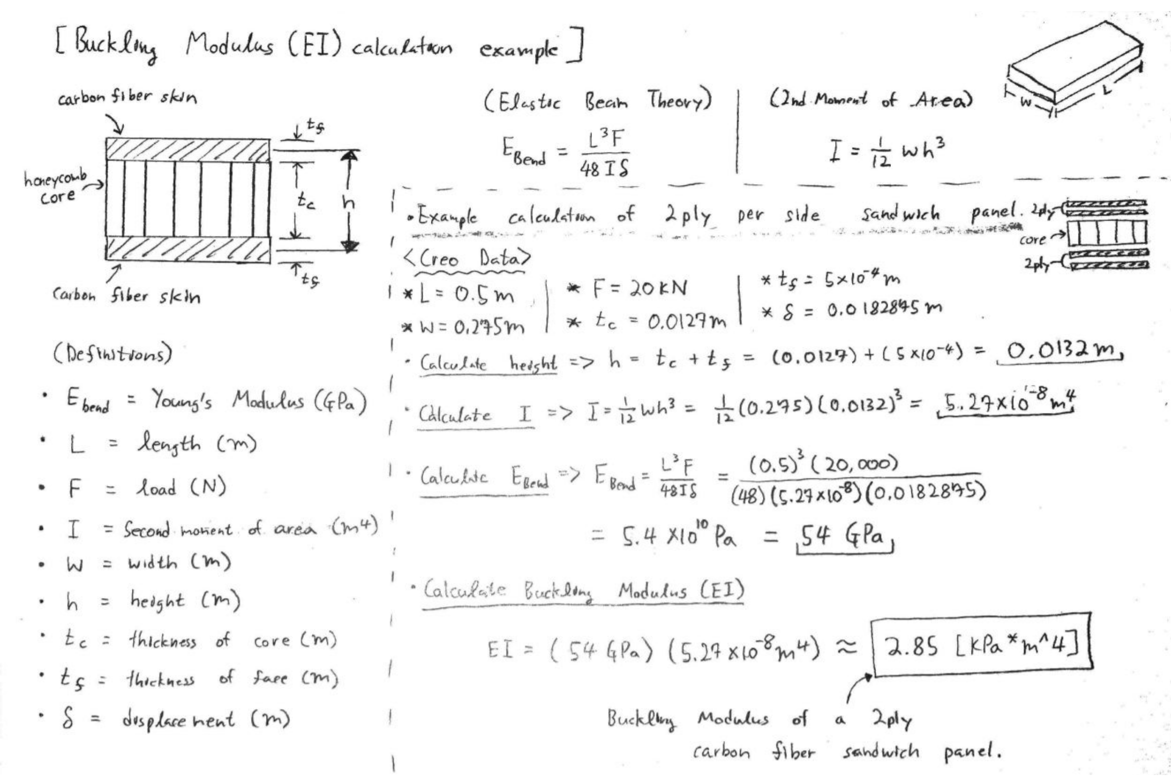

Figure 21. Hand Calculation Example of Buckling Modulus

After verifying the simulation and performing hand calculations, we then determined the different ply stack ups and core thicknesses to experiment with. According to the FSAE rules, the composite layup is required to be quasi-isotropic, which means the layup must have equal fiber properties and mass properties in the 0/90/45/-45 directions. Our initial assumption of this statement was that the layup required an equal number of layups in both the 0/90 and 45/-45 directions. As a result, the different ply stack ups used as part of our experiment and simulations can be seen in Table 11; the table alternates through different core thicknesses of 12.7, 19.05,and 25.4 mm. The topmost row of the table represents the outermost skin of the panel and can see our assumption of equal number of layups in both 0/90 and 45/-45 directions.

Table 11. Ply Stack Up Plan with Varying Core Thicknesses

0/90	0/90	0/90	0/90	0/90	0/90	0/90
45/-45	45/-45	45/-45	45/-45	45/-45	45/-45	45/-45
Core	0/90	0/90	0/90	0/90	0/90	0/90
	45/-45	45/-45	45/-45	45/-45	45/-45	45/-45
	Core	0/90	0/90	0/90	0/90	0/90
		45/-45	45/-45	45/-45	45/-45	45/-45
		Core	0/90	0/90	0/90	0/90
			45/-45	45/-45	45/-45	45/-45
			Core	0/90	0/90	0/90
				45/-45	45/-45	45/-45
				Core	0/90	0/90
					45/-45	45/-45
					Core	0/90
						45/-45
						Core

Once the experiment process was organized, each configuration was applied to our model in Creo and the simulations were completed. The three-point bend test simulation can be seen below in Figure 22.

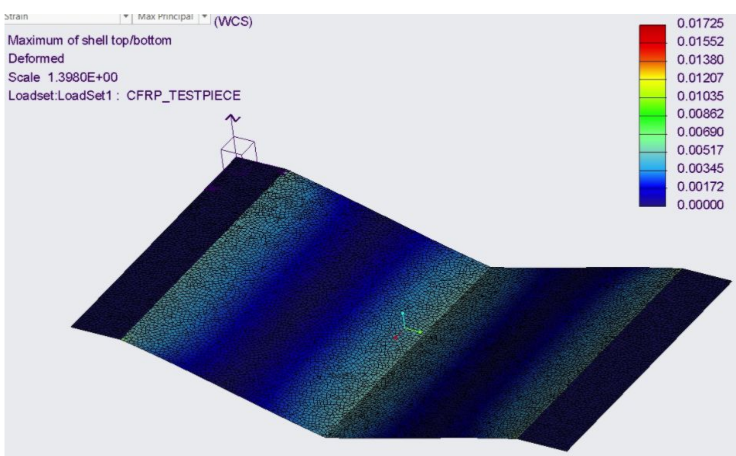

Figure 22. Simulation Result of Three-Point Bend Test

We iterated through varying loads until we observed a strain of 0.017. The simulation provided the dimensions, load, and displacement of the sandwich panel stackup and was used to calculate the buckling modulus with Eq 1 and 2. The relationship between the number of carbon plies per side and the buckling modulus is shown in Figure 23. The relationship between the core thickness and buckling modulus can be seen in Figure 24.

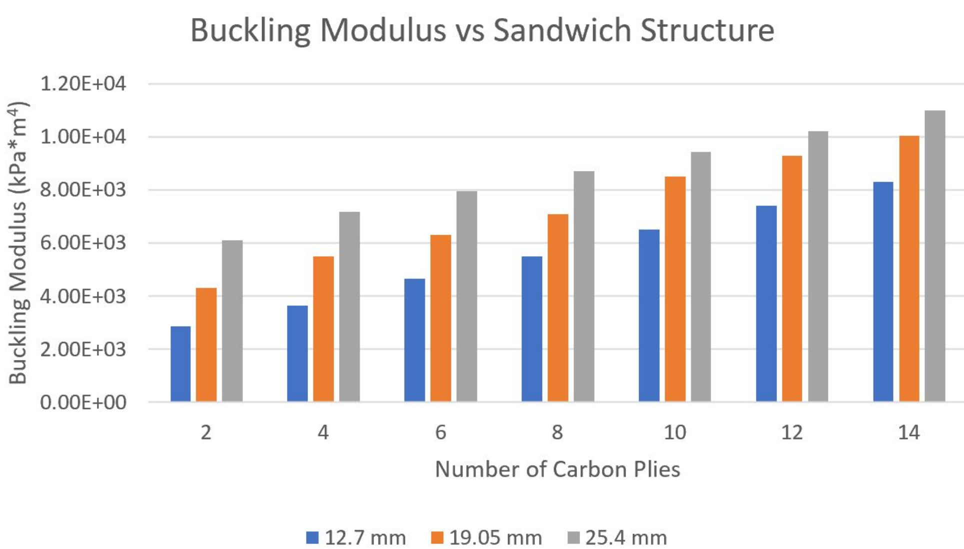

Figure 23. Shown above is the buckling modulus as a function of the number of plies on each side of the panel. As you can see, there is an increasing trend in buckling modulus with an increasing number of plies. This makes intuitive sense in that, as a general rule, more material leads to a stronger product.

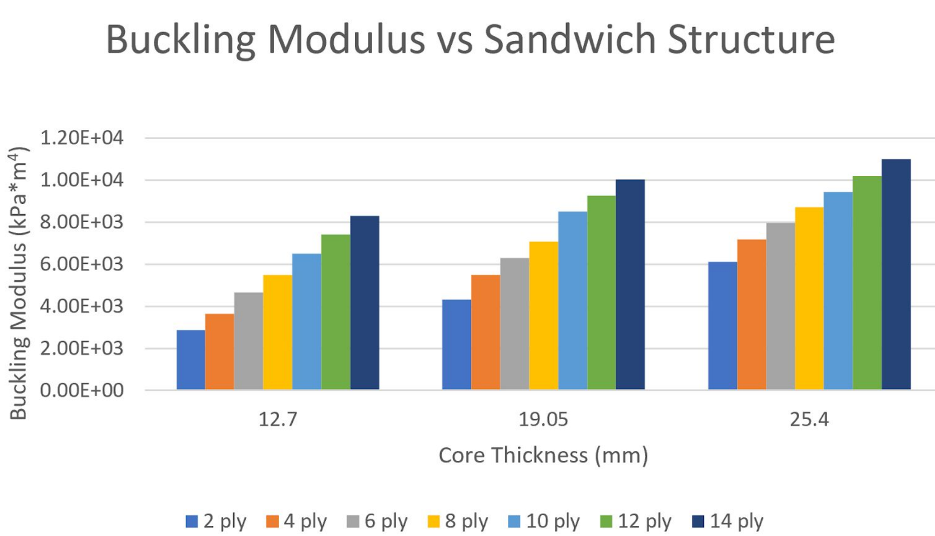

Figure 24. Shown above is the buckling modulus as a function of core thickness.

Similar to the trend in increasing the number of plies, increasing the core thickness leads to a stiffer panel. This is less intuitive, but makes sense in that the bending stiffness of something is inversely proportional to the distance between the center of area of the cross section and the bending axis. In simple terms, the more material that is further from the bending axis (middle of the core), the stiffer the panel.

After completing the simulations and gathering the results, we needed to determine the ideal panel stackup for each strength requirement at different locations in the chassis (within the scope of this project), allowing us to design the monocoque with a non-uniform mass and thickness distribution. With an array of results and strength requirements, we decided to organize them in a table with each requirement set as "fail" (red) and used boolean logic to set a minimum value for each cell. We applied a safety factor of 1.5 (used in MER19) to each buckling modulus calculated from simulation and entered our results until the cell showed the requirement as "passing" (green). Table 12 shows the results of this panel placement analysis. The minimum requirement for each panel is represented by the first green box going down each column.

Table 12. Panel Placement Decision Matrix

Core Thickness (mm) [t_c]	Number of layers (per side)	Front Bulkhead (2 B tubes) 5.11 [Kpa*m^4]	Front Bulkhead Support (Vertical Side - 1 C tube) 2.01 [Kpa*m^4]	Front Bulkhead Support (One side - 3 C tubes) 6.03 [Kpa*m^4]	Side Impact Zone (lower - 1 B tube) 2.55 [Kpa*m^4]	320mm Above Side Impact Zone (2 B tubes) 5.11 [Kpa*m^4]
12.7	2					
12.7	4					
12.7	6					
12.7	8					
12.7	10					
12.7	12					
12.7	14					
19.05	2					
19.05	4					
19.05	6					
19.05	8					
19.05	10					
19.05	12					
19.05	14					
25.4	2					
25.4	4					
25.4	6					
25.4	8					
25.4	10					
25.4	12					
25.4	14					

Table 12 shows that most of the panels meet chassis requirements. Based on our requirements and specifications from Table 3 on page 14, we wanted to choose a sandwich panel that meets FSAE rules requirements , torsional stiffness standards, and weight reduction goals. According to our subject matter expert, a core thickness of 19.05mm and 5 layers per side satisfied all FSAE requirements [19]. Based on Table 12, we found that a core thickness of 19.05mm and 6 layers per side would be our stackup plan for our monocoque design. The stackup plan meets the chassis requirement and the torsional stiffness validation will be discussed in the next section.

Torsional Stiffness Simulation

As part of our requirements and specifications, we wanted to maintain a torsional stiffness of at least 1100 Nm/deg. The previous MER vehicle had a torsional stiffness of about 1100 Nm/deg according to simulations run. While torsional stiffness is not required for a high performance race car chassis [3], we wanted to maintain a minimum stiffness to ensure no losses occur in strain during load transfer between the suspension and mounting locations. As expected, the carbon fiber monocoque solution not only met the minimum torsional stiffness requirement of 1100 Nm/deg, but well surpassed it with a simulated torsional stiffness of 12305 Nm/deg. Shown below in Figure 25 is the deformation simulation performed using Ansys 2020 R2.

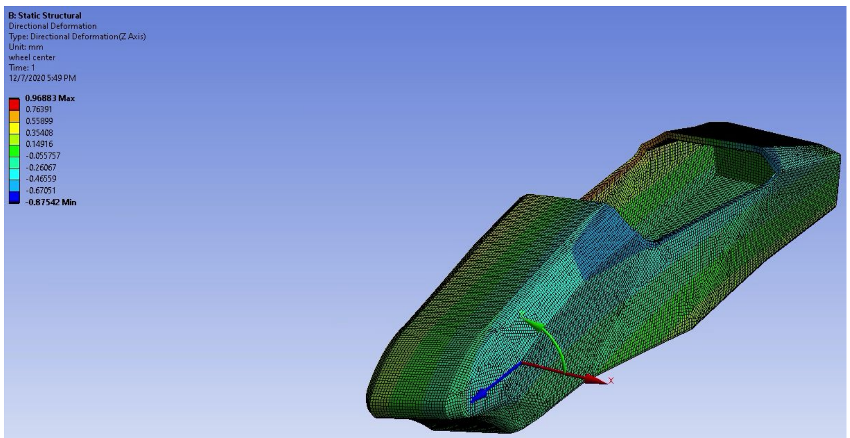

Figure 25. Shown above is the deformation simulation run in Ansys 2020 R2 with the deformation results in the top left in mm.

The max deformation is obtained by placing two forces on either side of the chassis acting at the hub of the wheel and where the suspension is attached to the chassis. Meanwhile, a fixed support is maintained at the rear of the chassis. In order to calculate torsional stiffness from deformation, the torsional moment was calculated using the equation for torsional moment,

$$M = 2 \times F \times L \qquad \text{(Eq. 3)[18]}$$

where M is the torsional moment in Nm, F is the force applied in Newtons and L is the distance from the center of the chassis to the point where the force is applied in meters. After calculating the torsional moment, the angle of deformation was found using the equation for deformation,

$$\alpha = tan^{-1}\left(\frac{\mu_1+\mu_2}{L_{12}}\right) \qquad \text{(Eq. 4)[18]}$$

where α is alpha, or the angle of deformation, in radians, μ_1 and μ_2 is the deformation of the chassis as seen below in Figure 26 in meters, and L_{12} is the width of the chassis in meters. After determining the angle of deformation and converting to degrees, dividing the torsional moment by the angle of deformation gives the torsional stiffness in Nm/deg as shown in the equation for torsional stiffness. The equation for torsional stiffness, or k_t, is

$$k_t = \frac{M}{\alpha} \quad \text{(Eq. 5)[18]}$$

where k_t is torsional stiffness in Nm/deg, M is the torsional moment in Nm as discussed above and α is the angle of deformation in degrees.

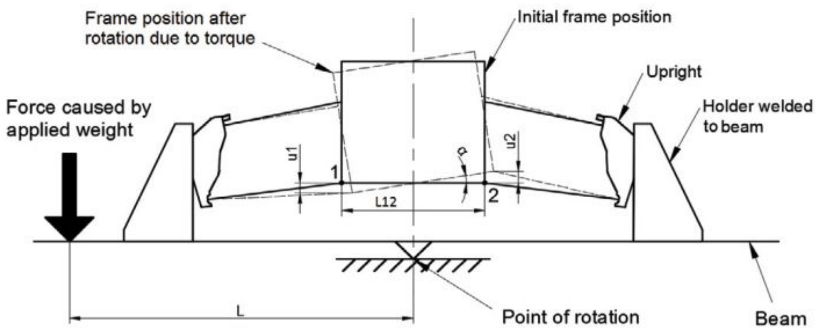

Figure 26. Visual representation of the parameters mentioned above. Clearly shown is α, μ_1, μ_2, L and L_{12} . [18]

After using Equations 3 - 5, values for all prior mentioned parameters were found. These results can be found below in Table 13.

Table 13. Values used to calculate torsional stiffness as well as the results obtained from using the above mentioned equations.

Parameter	Result	Unit
F	6870	N
L	.237	m
L_{12}	.419	m
$\mu_1 + \mu_2$	.0019	m
M	3256.38	Nm
α	.265	degrees
k_t	12305	Nm/degree

With a simulated torsional stiffness of 12305 Nm/deg, we are confident that even if we perform a physical torsional stiffness test, the monocoque will still meet the requirement set out of meeting the minimum 1100 Nm/deg torsional stiffness value. If we find that the simulation overestimated stiffness by even a magnitude of 10, the stiffness will still meet the requirement set out and thus, there is a lot of room for the simulation to have due to the drastic increase in torsional stiffness. However, we do not see any factor that can cause such a massive error in our simulation.

Computational Fluid Dynamics (CFD) Simulation

A carbon fiber monocoque chassis replaces several aerodynamics components that would traditionally need to be added separately on a tube chassis. The body of the monocoque forms the nose cone, bulkhead, side impact structure, floor, and rear of the car, and it is therefore essential to ensure the solution does not negatively impact aerodynamic performance. The most important aerodynamic metrics in performance racing are downforce (negative lift generated by the wings) and drag. In our analysis, the body of the monocoque is isolated from the wings and the rest of the aero package. When the wings are added to the car, further analysis must be done to ensure the necessary downforce is provided for optimal vehicle dynamics.

The monocoque does not function to produce any downforce. The only necessities are that it does not produce significant lift (this would take load off of the wheels, which negatively affects traction limits) and that it minimizes drag. Input flow should ideally be routed to the wings and cooling systems of the car, but this can only be accounted for when the aerodynamics package is designed. We can verify that the monocoque minimizes drag and lift with a 3D computational fluid dynamics (CFD) analysis. To do this we used Siemens Star CCM+, and the assistance of the MER21 Aerodynamics division leader, Mitchell Houghtaling, to set up the simulation and parameters [20]. Traditionally, a race car would be tested in a wind tunnel to verify aerodynamics effects, so we simulated a wind tunnel environment in Star CCM+, and applied an inlet air velocity of 35 mph (the average speed of our dynamics events in the FSAE competition). The streamlines from the results of the simulation are shown below in Figure 27.

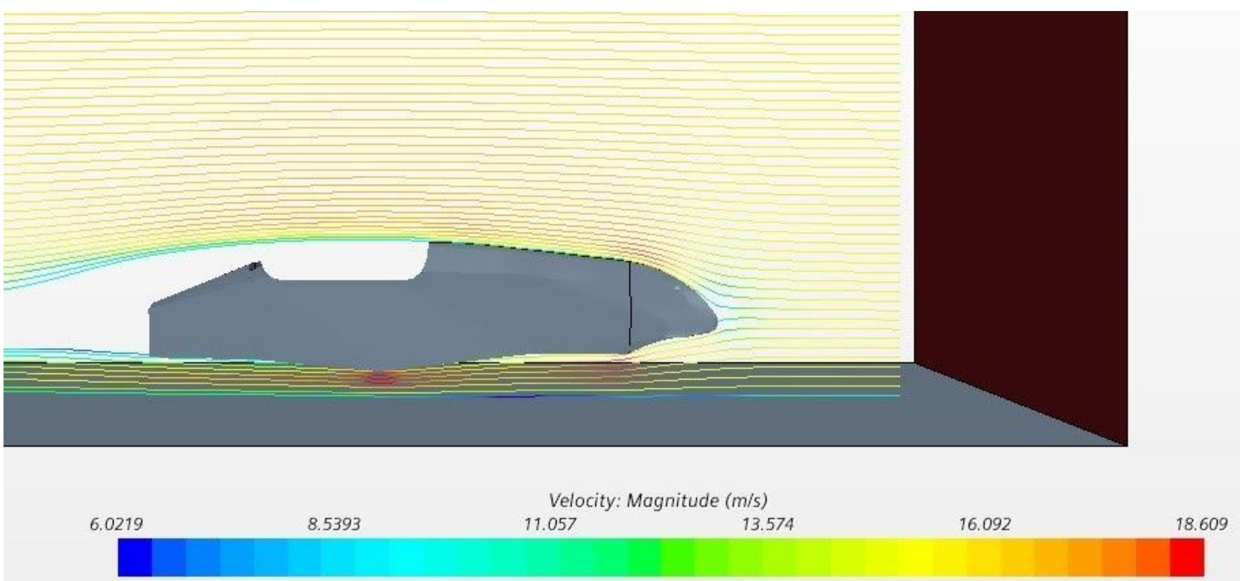

Figure 27. The image above shows a side view of the monocoque in Star CCM+. The environment is meant to mimic a real wind tunnel, and the applied inlet air velocity is 35 mph. The colorbar in the bottom of the image shows the approximate magnitude of air velocity in the simulation.

The nose cone in Figure 27 on page 36 is an updated version of the previous nose cone. A short meeting with Mitchell revealed that the original nose cone was producing unwanted lift. It can clearly be seen that the velocity of air at the tip of the nose cone is lower than the velocity of the air around the top and bottom of the bulkhead. The boundary layer from the bulkhead ensures that there is no additional turbulence towards the rear of the vehicle and that the only high pressure zone that exists outside of the wings is the tip of the nose cone. The lack of streamlines entering the cockpit ensures there are no high pressure zones within the car, as well.

One of the most important aerodynamics metrics in performance racing is downforce, or negative lift with respect to the road. This is important because most components on a race car are made light weight for better acceleration, but this sacrifices traction (a function of normal force). A good aerodynamics package adds normal force (as a function of vehicle velocity and orientation) such that the maximum tractive capabilities are increased at higher speeds, making up for the low mass of the vehicle. The monocoque itself is not designed to produce any downforce, but it is important that it does not act as a wing and produce lift, for the same reasons as detailed above. Figure 28 below shows a graph of the downforce produced by the monocoque as a function of simulation iteration.

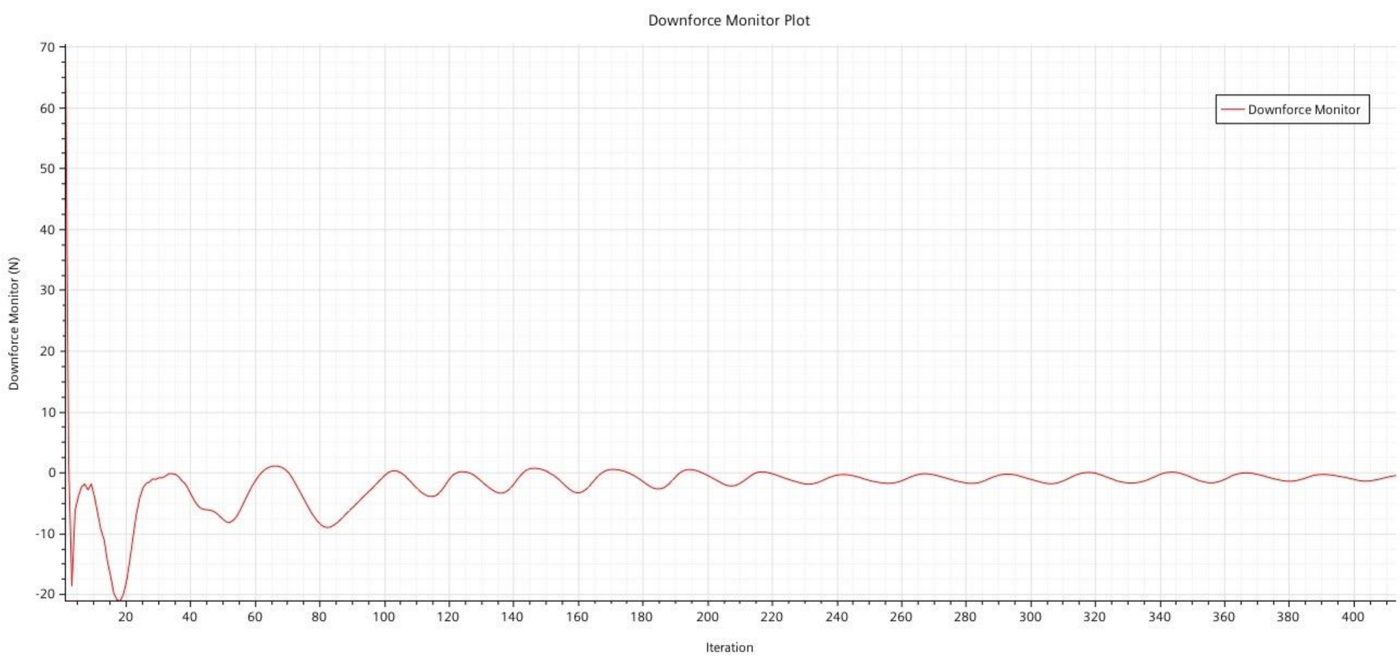

Figure 28. The graph above shows the downforce in Newtons with respect to the simulation iteration. As you can see, the downforce starts negative (unwanted lift) but approaches zero as the simulation iterates, which meets our needs.

The other important aerodynamics metric when it comes to performance racing is drag. Drag causes negative acceleration, which is only useful when braking in performance racing. Otherwise, to be fast, a vehicle must accelerate as fast as possible. Therefore, for dynamic events, it is necessary to reduce drag as much as possible unless you're using an active drag reduction system (DRS). Figure 29 below shows a graph of the drag produced by the monocoque as a function of simulation iteration.

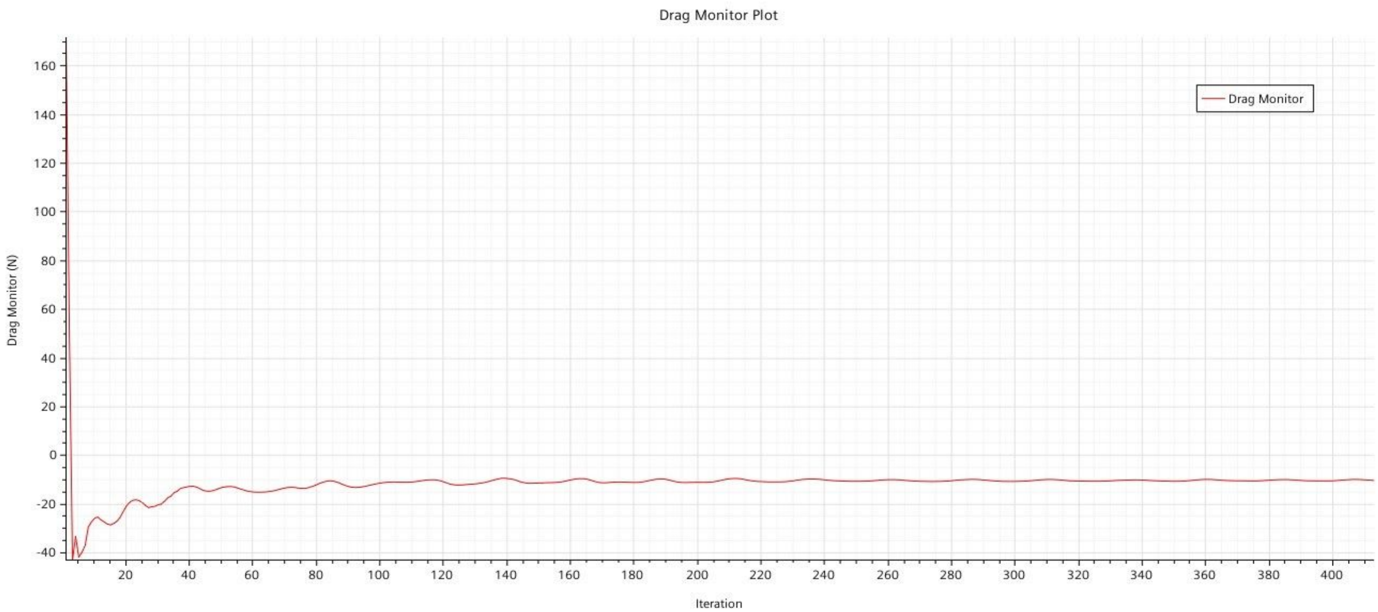

Figure 29. The graph above shows the drag in Newtons with respect to the iteration in the simulation. The average drag produced by the monocoque is about 10 N, which is 2.2 pounds. There is no reference value for us to achieve when it comes to drag, but 2.2 pounds is sufficiently low when compared to the torque produced by our powertrain.

This fluid dynamics analysis is by no means complete for a performance race car, as the addition of other components such as wings and wheels, and viscous effects from the road would need to be taken into consideration before the design is finalized. This analysis does, however, provide proof that as a standalone component the monocoque does not produce any unwanted aerodynamic effects.

Verification

Moving forward, there are several actions that must be performed in order to continue with our solution. For example, we must select a sandwich structure, perform perimeter shear test simulations, energy absorption calculations, comparison test simulations, lap joint strength calculations, CFD analysis with our aerodynamics packaging, torsional stiffness simulation and integrate variable sandwich thicknesses into our CAD model while further refining the current CAD model.

More specifically, we must determine the ideal sandwich construction for all areas of the racecar. As mentioned above, we are still determining the ideal thickness and materials to be used in the sandwich which will help ensure we develop a lightweight solution. Now with the knowledge that our layup does not need to alternate 0/90/45/-45, but rather only needs to have the same number of plies, we can further optimize our layer options. Once we re-run our simulations, we will choose 2-3 sandwiches to use throughout the monocoque using Table 12 on page 33.

Once we select our sandwich structures, we will move onto simulated testing. We will run our simulated 3-point, perimeter shear, and energy absorption test simulations. We plan to compare our test results to the FSAE rules requirements for each test to ensure our selections pass rules.

Another test we need to simulate is the comparison test. This test is required by rules to "establish an absorbed energy value of the baseline tubes," [1]. Although we don't have a physical test rig to experimentally test the compliance, we still need to establish a baseline energy absorption for the steel tubes we are replacing. This energy value will be compared to the energy the composite panels absorb. Figure 30 shows an example.

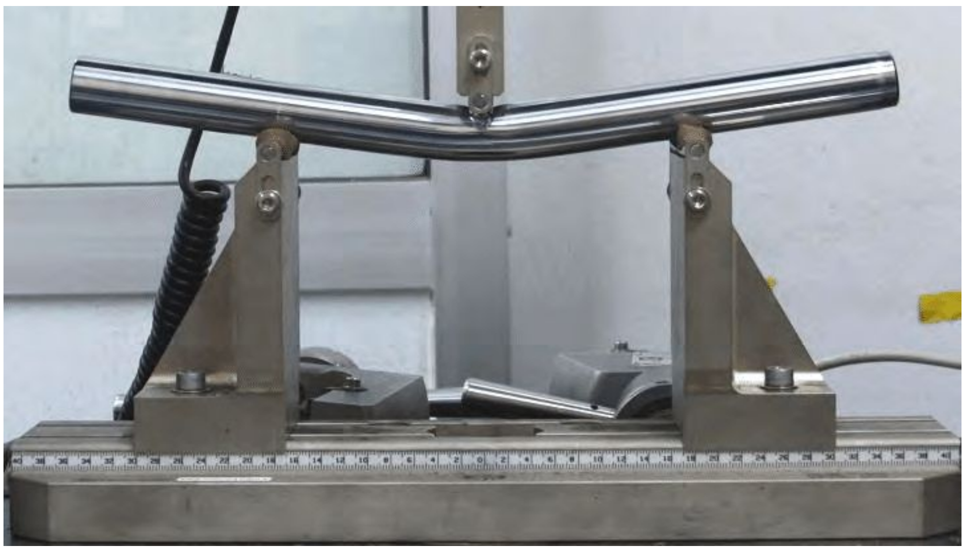

Figure 30. Shown above is an example of a tube undergoing a 3-point bend test [17].

We also need to test our monocoque's aerodynamics performance. To do this, we will be comparing the monocoque's aerodynamic coefficient of lift and drag to the current car. We have analyzed the current car using Star-CCM+, simulating airflow to determine the overall downforce and drag with the full aero package. To compare the monocoque to the current chassis, we will use MER20's aero package with the monocoque to analyze downforce and drag. We'll then compare the numbers from both simulations to determine if any performance has been lost by changing the chassis shapes. Documented CFD and other aerodynamic simulation data can be presented at the design event at competition to help reach our design event points benchmark set by MER19. Figure 31 on page 40 shows CFD from MER19.

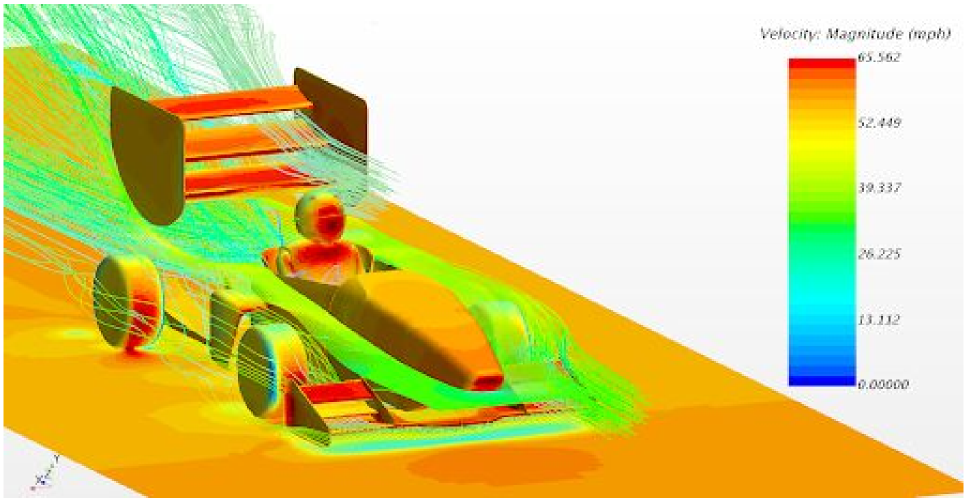

Figure 31. Shown above is the CFD of the MER19 iteration with the full aero package [17].

With more feedback from CFD, we will refine our monocoque's shape to reduce drag and increase flow to wing elements to increase downforce. We will also incorporate the thickness of each section into the CAD model, based on the sandwich structures we select. This thickness will allow us to further refine the shape with regards to templates, suspension points, and driver comfort.

With all these elements combined, we can perform a more realistic torsional stiffness simulation. Incorporating the thickness of the panels and their stiffness will make the torsional stiffness FEA much more accurate. With the data from this simulation we will be able to refine the shape and if we greatly surpass our torsional stiffness requirements or do not meet them, then we can work to change the mass of the chassis in areas or modify the geometry to better meet the requirements.

Discussion and Recommendations

As mentioned before, our chassis meets our requirements for mass, but we could probably be lighter. Based on Table 12 on page 33, there are sandwich structures that we can use in some parts of the car that have fewer layers of carbon. These sandwiches would be less dense than the current sandwich we chose to use for the entire vehicle. Using sandwiches that use the same core but fewer layers of carbon lowers the mass, but still allows us to be cost effective by using the least amount of different cores.

Due to our 450 team's limited knowledge of aerodynamics, we were not able to fully optimize the chassis for drag reduction. Our current iteration of the chassis has low drag and almost no lift (in either direction). With proper CFD analysis, we could further reduce the drag the chassis generates, and increase the amount of downforce. Some aerodynamic components are built into the chassis, such as the nose cone and the undertray. These would need to be designed by the aerodynamics division of the team in order for the chassis to fully be incorporated into the vehicle's aerodynamic package.

Conclusion

As an FSAE team, we design, build, and test a new car each year with the intention of winning a competition. We iterate on previous designs and look for all possible areas of improvement using test data from prior vehicles. After MER19, the chassis was highlighted as a major area for improvement. The next year, we produced a chassis 14kg lighter than MER19, but still with potential to be even more competitive. Lighter cars have proven - through simulation and competition data - to be better performing in dynamic events, and innovative lightweighting solutions have also proven to earn more points in static events. Referencing other successful FSAE cars as benchmarks, we formulated requirements and specifications for what we believe a highly competitive, rules-passing lightweight chassis solution would have. We have generated three broad concepts for possible solutions: a space frame chassis, a hybrid chassis, and a full monocoque chassis. These concepts will be evaluated in order to determine the best possible solution. We have estimated a budget of $380 from material costs for testing as required by FSAE rules. The lightweight chassis solution design for the Michigan Electric Racing Team should be completed on December 8th 2020.

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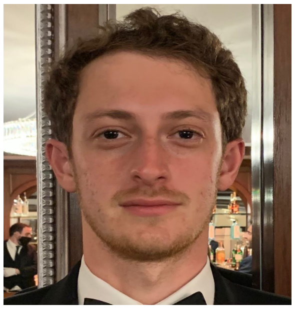
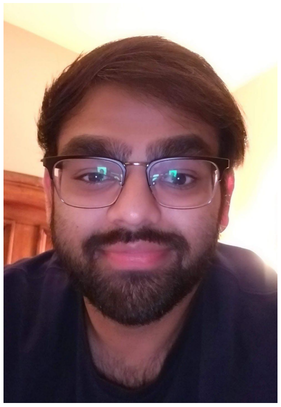
Team Member Bios



Jeremy Herskowitz is a senior mechanical engineering student from New York, NY. His passions include problem solving and critical thinking. Ever since he picked up a hammer and began making custom furniture he realized that engineering problems were everywhere. This inspired him to pursue a degree in mechanical engineering in order to incorporate his passions of problem solving and creating. As a sophomore, he joined the Michigan Electric Racing team at the University of Michigan and helped manufacture parts as well as assemble systems. As a member of the Vehicle Dynamics and Chassis division he assists in the design, manufacture and assembly of numerous systems, especially the chassis of the MER racecar.

Michael Hahm is a senior mechanical engineering student from Troy, Michigan. His passion is in design and manufacturing from a young age in Korea where he got to compete in different robotics competitions. Now he joined the Michigan Electric Racing design team at the University of Michigan and helped provide learning resources in CAD and machine work. He is currently part of the Vehicle Dynamics and Chassis team working on jig designs for assemblies.

Fred Ouding is a senior mechanical engineering student from Kalamazoo, Michigan. In Kalamazoo he grew up working in the family greenhouse where found a passion for design and fabricating. In college he participated in the Michigan Engineering Zone and also became involved in Michigan Electric Racing where he was the suspension and dynamics lead and now is the technical director.



Neel Patel is a senior mechanical engineering student from Kansas City, Kansas. There, he spends most of his time working for the family businesses by either providing labor, performing maintenance/upkeep, or helping in some way to increase revenue and review scores or reduce costs. He participated in project teams in high school, and joined the Michigan Electric racing design team after enrolling at the University of Michigan. He is currently the leader of the Vehicle Dynamics and Chassis subteam.

Sahil Saini is a senior mechanical engineering student from Ellicott City, Maryland. He has a passion for mechanical design and automotive racing. He enjoys modeling mechanical systems with code and associated software in order to better understand their design and optimize them. He was the Vehicle Dynamics and Chassis lead for the previous two years, but has now found his most enjoyable role as Public Relations Chair for MER.

Sean Phelan is a senior mechanical engineering student from Chicago, Illinois. He previously attended DePaul University, studying digital cinema. After a few years he made the gigantic mistake of deciding to study mechanical engineering instead. He transferred to the University of Michigan. There, he joined the Michigan Electric Racing team. He worked in the VDC and ergonomics. This year, he was promoted to Project Manager.

Appendix

Engineering Standards: We designed our chassis to the Formula SAE 2020 rules standards. Our team competes in the FSAE Electric competition, so our vehicle has to conform to those standards. Since this is the only application of the chassis, we deemed it only necessary to follow these standards.

Engineering Inclusivity: With our problem definition of "score more points in both static and dynamic events" at FSAE competitions, the scope of our solution is narrow. As a result, we did not consider many aspects of inclusivity. However, there is some inclusivity built into the FSAE rules. Driver accommodation rules state that our car "must be able to accommodate drivers of sizes ranging from 5th percentile female up to 95th percentile male." [1] We have designed our chassis to these rules, so anyone up to the 95th percentile male would be able to fit into this car. The vehicle would be able to accommodate smaller drivers with adjustment of controls, which is outside the scope of this project.

Environmental Context Assessment: Our system did not make significant progress towards the environmental or social challenge. The goal for our team is to design an innovative chassis solution for the Michigan Electric Racing Team (MER). Although our design solution for a carbon fiber monocoque reduces chassis weight, the use of carbon fiber may introduce undesirable consequences toward the environment. It was found that carbon fiber requires 14 times more energy than producing steel, which results in a significant amount of greenhouse gas emissions.

Social Context Assessment: Considering the niche application of composite monocoque chassis, our design is not likely to be adopted or self-sustaining in the market. Because of that, our system would not worsen planetary or social systems. Due to how expensive prepreg carbon fiber cab be, the feasibility of our product might not be resilient to disruptions. The team would ostensibly have gotten the materials from sponsors, so any disruption to the market would probably affect the team's ability to manufacture this chassis.

Ethical Decision Making: In racing, less weight is key to performance, hence our goal to lower chassis weight. However, achieving the absolute lowest weight would mean cutting a lot of corners and ignoring driver safety. In our design, we accounted for driver safety to ensure that our driver is not at risk while operating the car. To do this, we followed the safety guidelines in the rules. Our impact structures absorb the necessary amount of energy to keep the driver safe in the event of a crash. The roll hoops protect the driver if the vehicle rolls over. These structures add weight, but are essential in guaranteeing the driver's safety, so we've included them in the design.