Zips Electric Racing Suspension Design

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FSAE Zips Electric Racing
Suspension Subsystem for ZER-20

Fall 2019 – Spring 2020

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Introduction

The Zips Electric Racing team started in 2013, and before this year the team had completed one rules-compliant car, ZER-19. This year, the team's most basic goals were to have a rules-compliant car and to compete in all events at the two competitions that were planned for the summer. The four students responsible for the steering and suspension subsystems and contents within this report were all new to the team as of this academic school year.

The major goals for the suspension and steering subsystems were to reduce weight and roll from last year's design. A secondary goal was to perform simulations and find other concrete justification for the decisions that were made in the design. This year the chassis was completely redesigned, which had a large effect on the steering column packaging and placement of suspension points. For most aspects of the subsystems, the components were designed one at a time, with a broad idea of how they would connect and interact with each other. First the suspension subsystem was designed, and after the geometry was finalized the team started designing the steering system. It was important to conduct finite element analysis and calculate the maximum forces that certain components would endure, as it was crucial to choose materials that would be lightweight and cost-effective, yet durable enough to endure the forces of competitive driving. Since this design was so customized, it was only realistic to purchase a few parts; the rest were manufactured by the team in The University of Akron's machine shop.

The design of the suspension and steering subsystems needed to be compatible with every other subsystem. As far as the scope of work, the suspension subsystem is responsible for everything from the uprights to the dampers, and the steering subsystem includes everything from the steering wheel to the tie-rod connection point on the steering knuckles. In other words, the suspension components include the uprights, wheel bearings, camber knuckles, upper and lower control arms, chassis tabs, tie-rods, pull-rods, bellcranks, and dampers, as well as their appropriate connecting components, and the steering components include the steering wheel, steering column, steering column support tubes, steering rack, tie-rods, steering knuckles, and all appropriate tabs and mounting components. Standard components including universal joints, needle bearings, a steering wheel quick-disconnect, and mounting hardware needed to be chosen and purchased to fit into the system. To cut back on cost, the purchased dampers and steering rack were reused from last year's design.

The senior design group worked together on the suspension and steering components in Solidworks, conducted simulations in MATLAB and Optimum Kinematics, and tested the components' structural integrity using finite element analysis. Next, the parts were manufactured in the university's machine shop. Upon the completion of the manufacturing phase, some of the components were assembled but few were installed in the car along with the rest of the subsystems. After months of hard work, the university was closed in March, then locked down soon after due to the contagious COVID-19. Much of the manufacturing was complete, but the car will not be tested or driven this semester, as the competitions that were scheduled for this summer were canceled. This design report includes an overview of the analysis, development, and manufacturing of the suspension and steering systems used for the 2020 Formula Electric car.

Design and Design Methodology

Control Arms

The first parts that were designed for the suspension system were the control arms. It was decided that double-wishbone control arms would be used, and that each wheel would have its own independent suspension system. Double-wishbone control arms were used over other options like MacPherson struts because of how much freedom they introduce into the design. Making small changes to the mounting points and length of the control arms helps fine-tune every aspect of the way the car drives. Four independent suspension systems were used over one dependent system because in a racing setting, it is not desirable for all four wheels to move every time one wheel goes over a bump. To determine where the control arms would mount to the frame, the anti-dive, anti-squat, pitch center, and roll centers had to be analyzed for each potential design. These are all crucial metrics, as they each have a uniquely powerful impact on how the car is driven.

Implementing anti-dive reduces the car's tendency to lurch forward during braking, which helps the rear tires maintain as much contact with the ground as possible. This is done by tilting the front wishbones, lessening the compression force that goes into the dampers during braking. If the altitude of each wishbone's fore mounting point were the same as that of the aft mounting point, then there would be no anti-dive. Since anti-dive is expressed as a percentage, its maximum value is 100%, for which there would be no compression of the dampers due to braking. If the anti-dive were negative, then it would be expressed as a positive percentage of pro-dive, and there would be excessive compression in the dampers due to braking. This would be done by tilting the front wishbones so that their side-view intersection point would be in front of the mounting points instead of behind them. This would cause the car to lurch forward during braking even more than it normally would, and so it is very rarely used.

Anti-squat is similar to anti-dive, but instead of dealing with the front suspension, it is only affected by the rear suspension. When a car accelerates, it has a tendency to lurch backward toward the rear wheels. Implementing anti-squat reduces this tendency, lessening the compression force that goes into the rear dampers during acceleration. If a car had rear control arms oriented such that the altitude of each wishbone's fore mounting point were the same as that of the aft mounting point, then it would have no anti-squat. A value of 100% anti-squat would correspond to a car that does not compress the rear dampers at all during acceleration. A negative value of anti-squat would be expressed as a positive percentage of pro-squat. A car with pro-squat would lurch backward excessively during acceleration, putting extra compression into the dampers, and increasing the likelihood that the bottom of the car would scrape across the ground. Because of this, pro-squat is very rarely used; instead, suspension systems usually implement a positive value of anti-squat that is relatively close to 0%.

Anti-dive and anti-squat are both determined by looking at how the wishbones are tilted from the side-view of the car. Figure 1 below shows the first step in finding these values. Lines are drawn through each of the upper and lower mounting points for the front and rear of the car. Intersection points are determined for the front and rear lines. Then a line (shown in red below) is drawn to connect each intersection point to the point where the bottom of the wheel makes contact with the ground. These red lines are called the front and rear side view swing arms (SVSA).

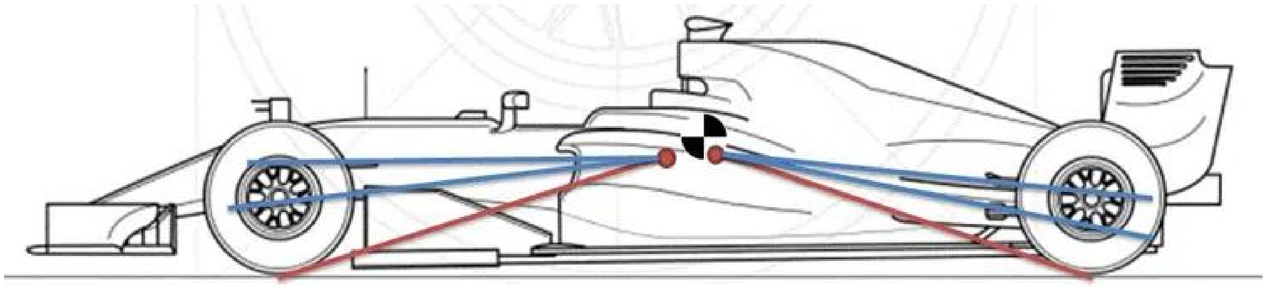


Figure 1: Side View Swing Arms (SVSA)

The next step is to draw vertical lines connecting the intersection points with the ground, forming front and rear triangles. First the front triangle can be analyzed as shown in Figure 2 below.

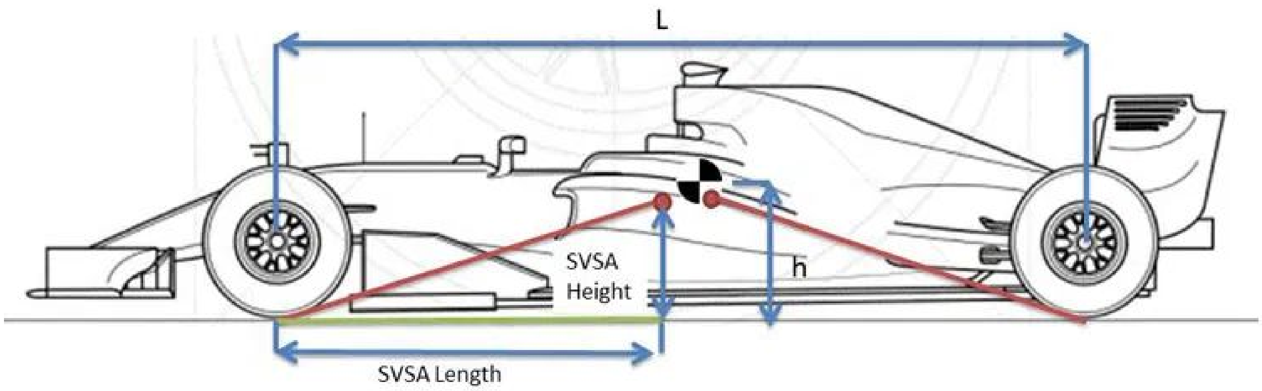


Figure 2: Front SVSA Triangle

Once all the variables are determined, they are substituted into the equation below to find the anti-dive:

$$\%Anti-dive = \%Front\ braking\ effort \cdot tan(\phi_F) \cdot \frac{L}{h}$$

where *% Front braking effort* is the percentage of braking effort that goes into the front tires as opposed to the rear ones. For this design, it was assumed that this was 60%.

The same process is followed to find anti-squat, using the rear SVSA triangle instead of the front one, as shown in Figure 3 below.

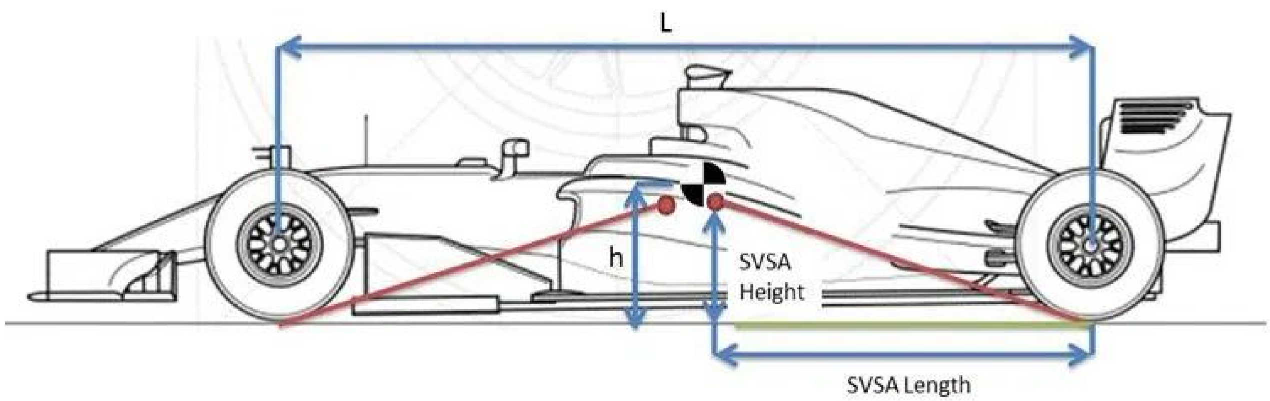


Figure 3: Rear SVSA Triangle

Once the variables are determined, they are substituted into the equation below to find the anti-squat:

$$\%Anti-squat = tan(\phi_R) \cdot \frac{L}{h}\ .$$

For the ZER-20 front suspension systems, the altitude of each of the lower fore and aft mounting points were set to be the same, and the upper wishbones were tilted so that the fore mounting points would be higher off the ground than the aft points. In the rear, both the upper and lower wishbones were tilted so that their fore mounting points were higher off the ground than the aft points. Solidworks was used to determine values of 18% anti-dive and 1% anti-squat. These metrics were verified with the hand-calculations listed in the Appendix. Having 18% anti-dive means that the car lurches forward less during braking, allowing for more grip of the rear tires, and lessening the amount of compression that occurs in the front dampers during braking. Having 1% anti-squat slightly lessens the tendency of the car to lurch backward during acceleration. This number was intentionally kept close to 0% to keep the maximum amount of grip possible in the rear tires while maintaining an ideal pitch center.

The pitch center depends on the values of anti-dive and anti-squat, which means that all three of these values need to be balanced in a design. When the car is driven, it lurches forward or backward around the pitch axis whenever it accelerates or decelerates. It is ideal for this axis to be as close to the center of the vehicle as possible. When the geometry of the control arms is changed to affect the percentages of anti-dive and anti-squat, this also changes the location of the pitch axis. In this design it was important to make sure that the pitch axis was kept very close to the center of the car.

The final metrics that needed to be balanced for the suspension design were the front and rear roll centers. It is important how high above or below the ground the roll centers are, because this determines how the car behaves while it turns. There is a roll center between the front wheels and one between the rear wheels, and they are connected to form the car's roll axis. If the front roll center is higher than the rear one, then the design has a negative roll axis inclination, and the car will experience oversteer while turning. This means that the car has a tendency to turn more than the driver intends, which can be unsafe for inexperienced drivers. Because of this, it was decided that this design would instead implement some understeer, giving the car a slight tendency to turn less than the driver intends it to, which is much easier to deal with than issues resulting from oversteer. To implement understeer, the front roll center was positioned slightly below the rear one, giving the car a positive rear axis inclination. Figure 4 below shows how each of the roll centers can be found, projecting lines from the control arms to find the instantaneous centers where they intersect, and then connecting lines from these instantaneous centers to the bottom of the opposite wheel. The point where these lines intersect is the roll center, indicated with a star in the picture.

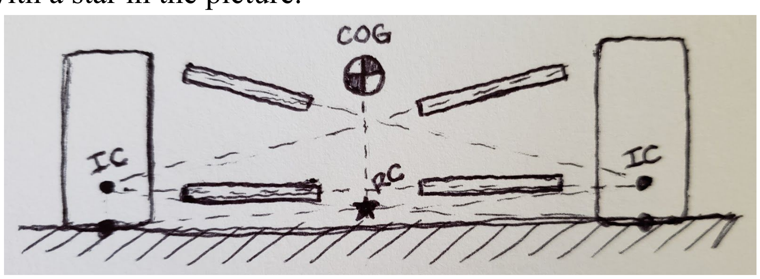


Figure 4: Finding Roll Centers

For this design, the front roll center was 1.18 inches off the ground, and the rear roll center was 1.89 inches off the ground. This means that the car has a positive roll center inclination, and the driver will experience slight understeering while turning, as desired.

Once all the desired parameters were analyzed, the orientation of the control arms was set in place, and the next steps of the design could be completed. The pictures below show this year's control arm design. The design uses a front track width of 47 inches, a rear track width of 46 inches, and a wheelbase of 60.8 inches.

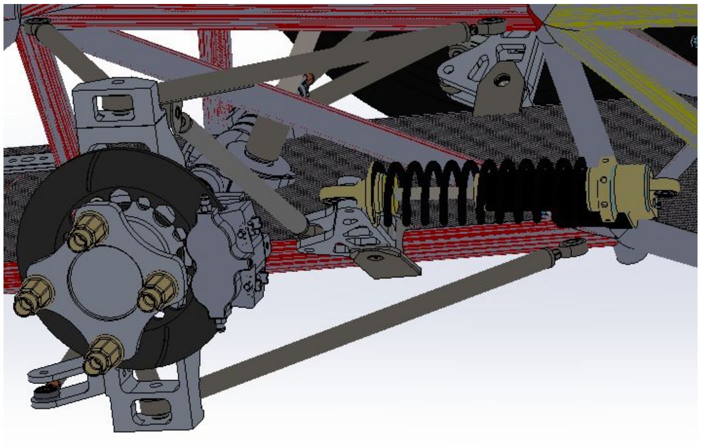


Figure 5: Front Suspension Design

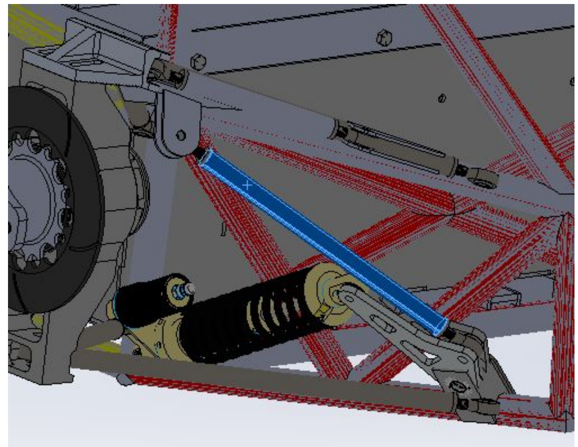


Figure 6: Rear Suspension Design

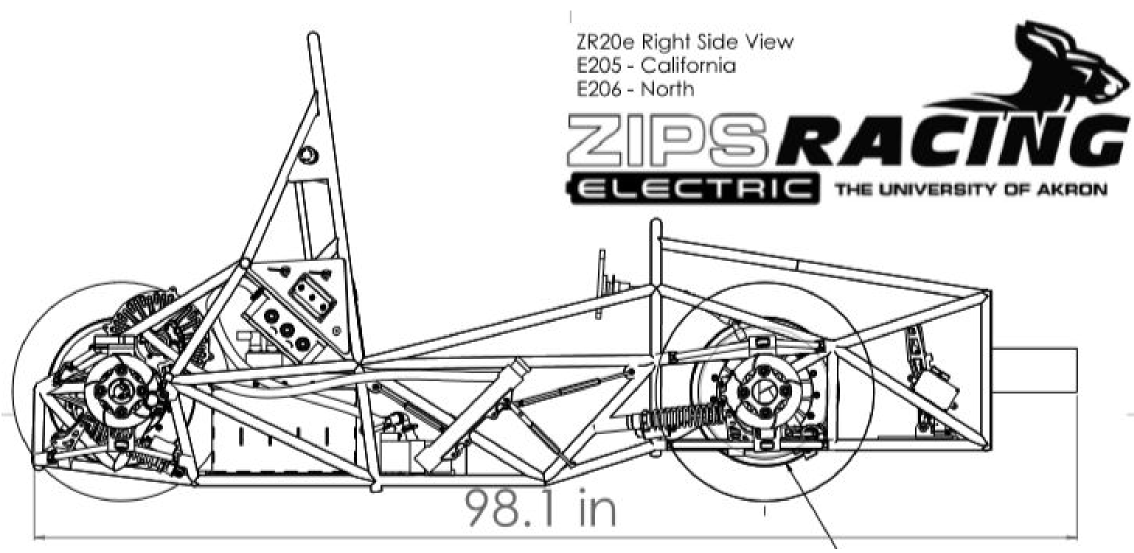


Figure 7: Side View of Suspension Design

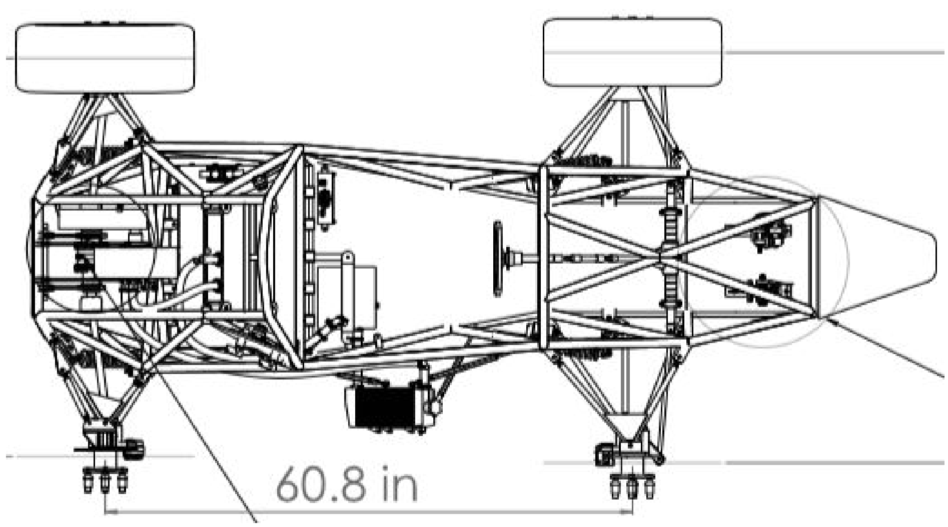


Figure 8: Top View of Suspension Design

Bellcranks and Pull-Rods

The ZER-20 suspension system utilizes pull-rod suspension with bellcranks in both the front and rear of the car. The pull-rod system is attached to the upper control arms, which pull the rod as the wheel is vertically displaced. The motion is then transferred into the dampers as the rod pulls on the bellcrank. The benefit of using pull-rods over push-rods is that this makes the center of gravity of the car lower as the packaging of the suspension components is also lower. The low center of gravity, combined with the wheelbase, reduces weight transfer and lateral roll during cornering by keeping the mass towards the center of the vehicle. The pull-rod system on ZER-20 aimed for a 1:1 motion ratio between the vertical displacement of the tires and the dampers. Motion studies were iterated until the desired ratio between the pivot of the bellcrank, the dampers, and the pull-rods mounting points was achieved. It is important to note that the pull-rods and dampers were kept as tangential as possible to the radius of motion in order to preserve linearity in the system, as the motion at the bellcrank attachment point is non-linear. The front pull rod is 10.79 inches long while the rear pull rod is 9.47 inches long.

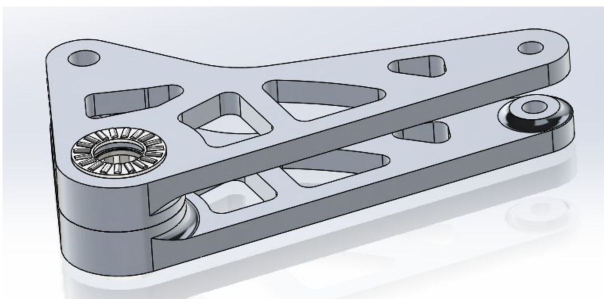


Figure 9: Front Bellcrank

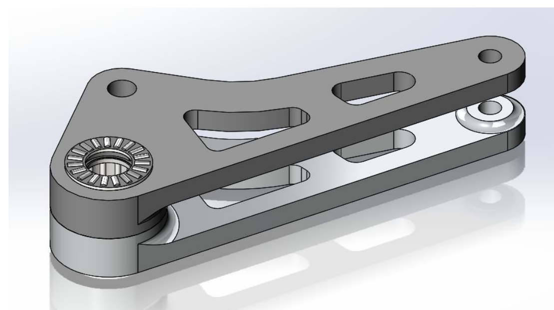


Figure 10: Rear Bellcrank

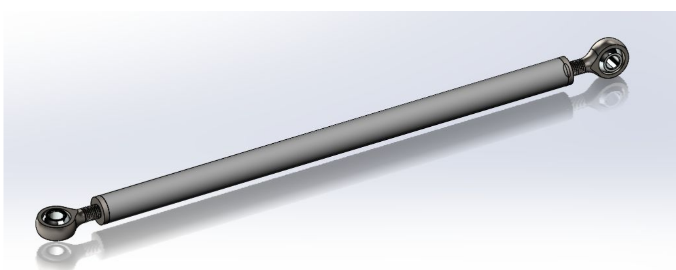


Figure 11: Front Pull-Rod

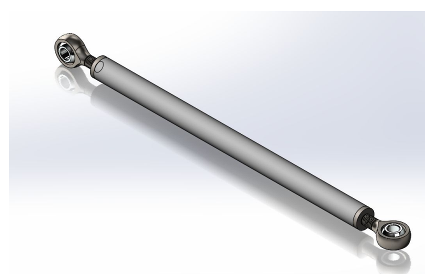


Figure 12: Rear Pull-Rod

Dampers

The final components of the pull-rods are the front and rear dampers. Since the dampers are the most expensive components of the suspension system, it was preferred that the dampers from last year's design would be reused if at all possible. The ZER-19 car used front dampers with a spring stiffness of 125 lbf/in and rear dampers with a spring stiffness of 175 lbf/in. To determine if these dampers would still be acceptable, the MATLAB/SIMULINK code in the Appendix was found and adapted for this design. The code uses a quarter-car model to evaluate the suspension system, given predicted values for mass and spring stiffness for each quarter of the car. Figure 13 on the left below shows the way that each quarter of the car was simplified so that it could be evaluated with the code. The code was run with predicted values of 480 lb for the weight of the car, a 40-60% front-back weight distribution, and a spring constant of 582 lb/in and damping constant of 500 N-s/m for the wheels and tires. Figure 14 on the right below shows how the suspension system would react to a bump.

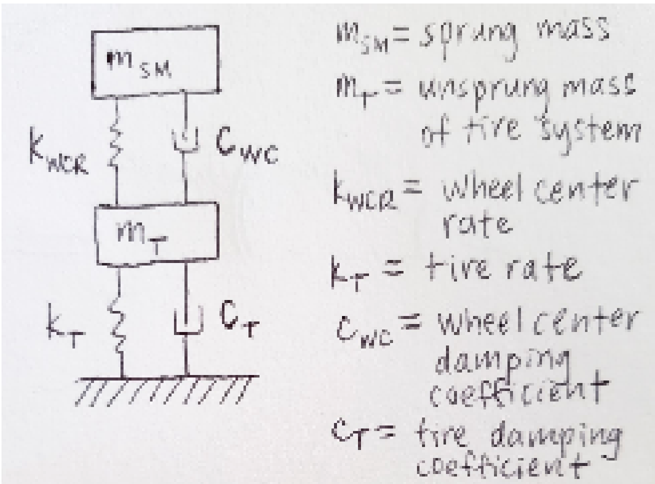


Figure 13: Quarter-Car Model

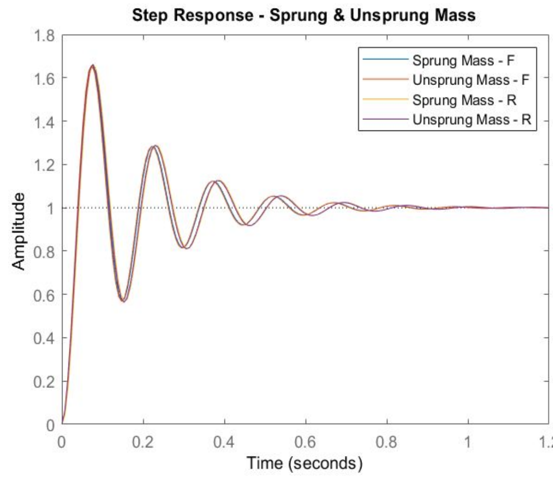


Figure 14: Suspension System Step Response

Figure 14 shows that the suspension system would be underdamped, with approximately 66% overshoot and a 2% settling time of 0.72 seconds. This translates to a stiff suspension system, with little delay between the front and rear of the car, which is desirable for a race car. Since this shows that the old dampers would be acceptable, they were reused in this year's design.

Steering Column

The first step in designing the steering column was to decide how many sections there would be, and how these sections would be connected. The column could either use u-joints or gears to connect the sections, each having unique advantages and disadvantages. U-joints are cheaper and simpler than gears, and do not require the design of a gearbox. Gears allow for much more freedom in the design, but they are relatively heavy and expensive, and for safety reasons, the design would need to include a bulky gearbox to surround them. It was decided that u-joints would be used, and since each u-joint has a maximum operating angle of 30°, the steering column would need to include at least two u-joints with three sections.

Since the potential steering column designs are confined by the geometry of the frame, it was very difficult to design a steering column for which all three sections exceeded the minimum length needed to attach the u-joints. The tight space and the limiting factor of the u-joint angles forced the decisions to tilt the steering wheel upward and to move the steering rack as far forward as possible. Tilting the steering wheel upward to be 6° from horizontal would help with driver ergonomics, so this was not a problem. However, moving the steering rack forward would have consequences for the rest of the steering design later on, so it was only done when all other options were exhausted. Figure 15 below shows the final geometric design and how it fits into the frame of the car.

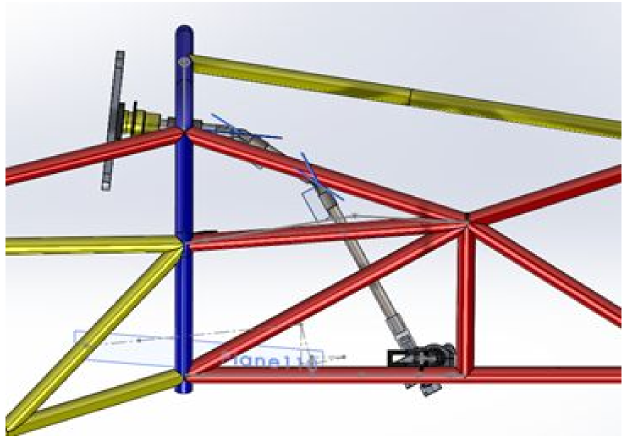


Figure 15: Final Steering Column Design

Steering Rack and Tie-Rods

To cut back on cost, the steering rack from the ZER-19 car was reused in this design. This steering rack has a maximum travel distance of 3.25 inches for a steering wheel with a range of at least 248°. However, since the rules dictate that the steering wheel can have a maximum range of only 180°, this means that the steering rack has a range of motion of only 2.36 inches, or 1.18 inches in either direction. This value is important for the Ackermann design in the next section.

A component needed to be designed to attach the steering rack to the frame. Originally the idea was to use something similar to the L-bracket that was used in last year's design, shown in Figure 16 on the left below. However, it was determined that these were unnecessarily heavy, so instead a thin plate was designed and used, shown in Figure 17 on the right below. The plates were each welded to the frame on two sides, and then the steering rack was fastened to the plates.

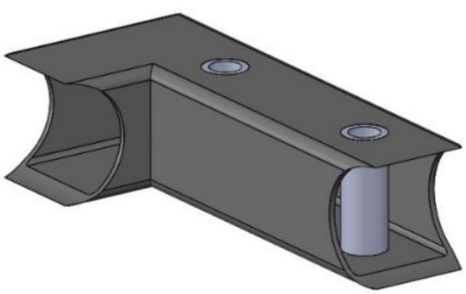


Figure 16: L-Bracket

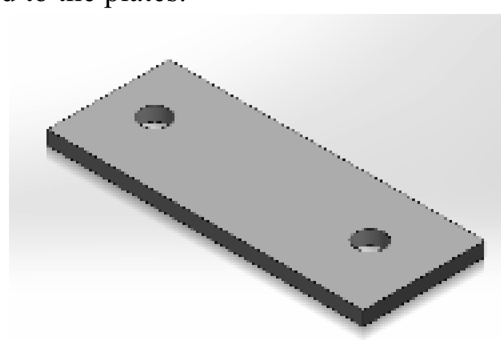


Figure 17: Thin Plate

Ackermann

One of the most important aspects of the steering system design is the Ackermann percentage. When a car that is not designed with any Ackermann turns, the two front wheels are parallel to each other, so they travel along different circular paths that do not share the same center, as is shown in Figure 18 on the left below. Since the front tires travel along circles with different centers, the tires fight each other to compensate for it. This causes tire slip, especially during low-speed turns. When the tires slip, the traction and tire contact patch areas decrease, resulting in wasted energy. A car implements an Ackermann design to prevent this from happening. If a car has 100% Ackermann, then the front tires travel along circular paths that share the same center, as is shown in Figure 19 on the right below.

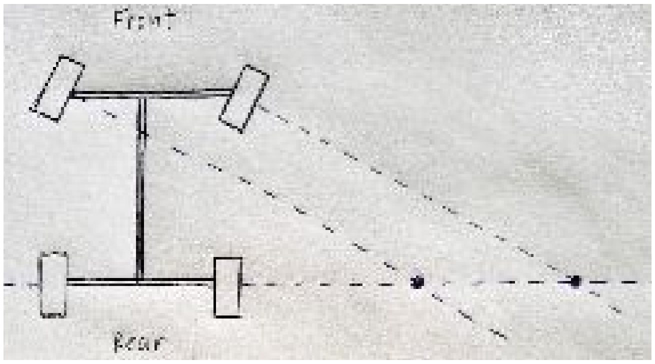


Figure 18: No Ackermann

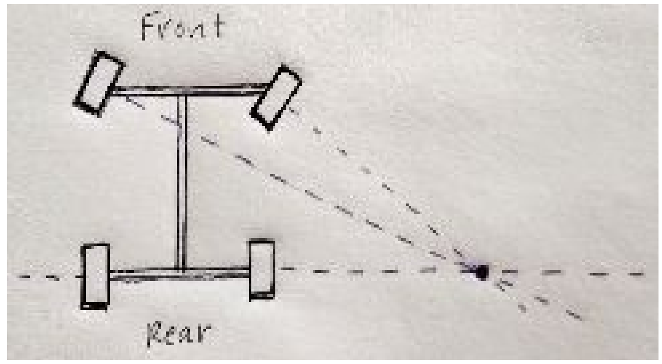


Figure 19: 100% Ackermann

To do this, the front tires can no longer be parallel to each other during a turn; instead they need to be offset approximately 5°-15° from each other, depending on the geometry of the car. To adjust the Ackermann percentage, the angle of the steering knuckles needs to be changed. Figure 20 below shows how angling the steering knuckles affects the amount of Ackermann in the design.

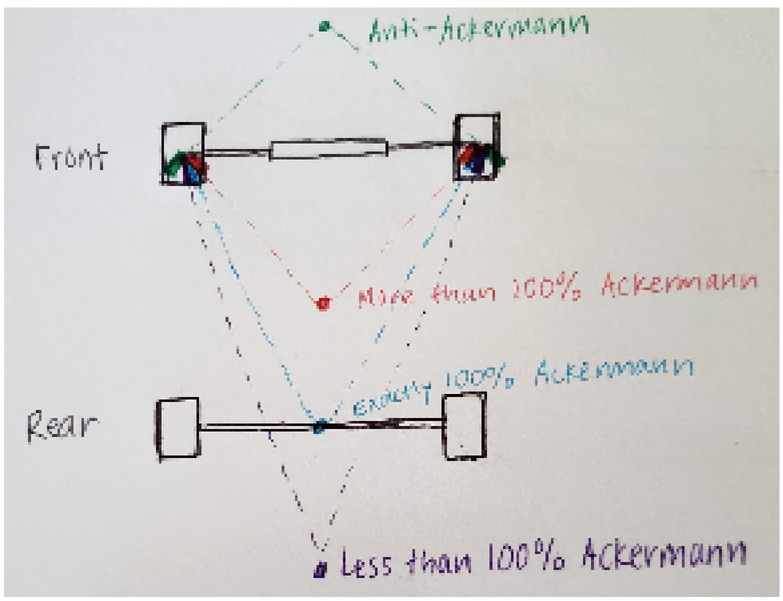
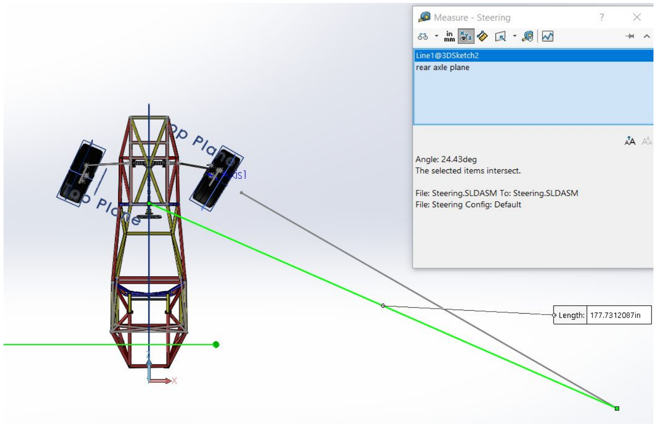
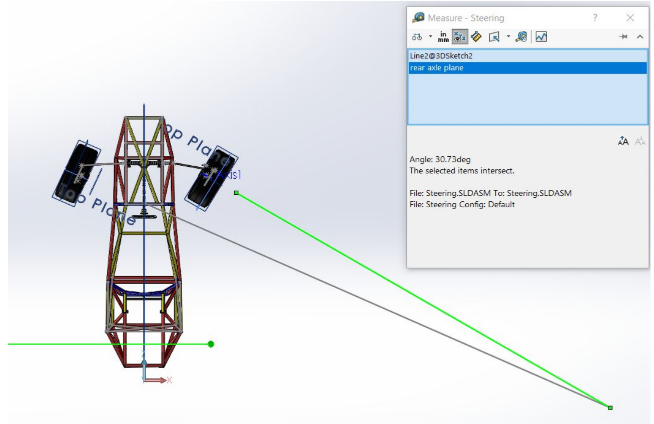


Figure 20: How Steering Knuckle Angle Affects Ackermann

It was decided that this year's design should implement somewhere between 50-80% of Ackermann. This would be an increase from the 40% that was used last year, since driver feedback indicated that the car was difficult to steer during low-speed turns. A simplified Solidworks model was created, and different tie-rod lengths and steering knuckle angles were tested until a desirable geometry was achieved. Figures 21 and 22 below show how the inner and outer front wheel angles were found for a full right turn for each potential design. As was discussed in the previous section, the steering rack can move 1.18 inches left or right for a full turn, so this number was used in the Solidworks model.





Figures 21 and 22: Ackermann Design in Solidworks

In the end, the inner angle was 30.73° and the outer angle was 24.43° for a full turn. The difference in these angles is 6.30°, so it could be said that this design uses 6.30° of pro-Ackermann; however, this does not mean much without the context of knowing which angle corresponds to 100% Ackermann, so the angle was converted to a percentage using the equation below.

$$\%\ Ackermann = \frac{cot(\delta_o) - cot(\delta_i)}{Track\ width} \cdot Wheelbase$$

Using the front track width of 47 inches and the wheelbase of 60.8 inches, it was determined that the final design uses 67.2% of Ackermann. This was accomplished by angling the short surface of the steering knuckle and by making the center-to-center distance 15 inches for each tie-rod. The steering knuckle and tie-rod are shown in Figures 23 and 24 below, respectively.

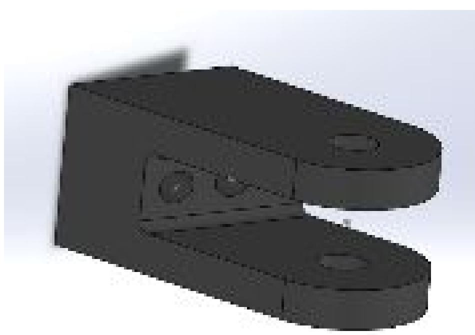


Figure 23: Steering Knuckle

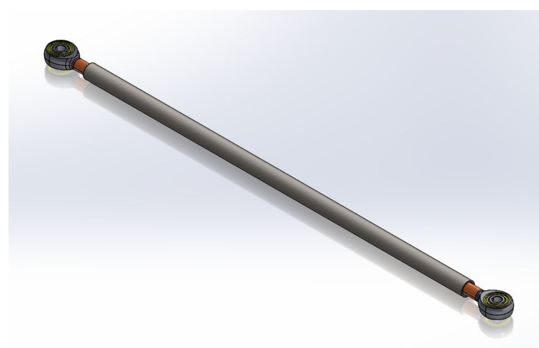


Figure 24: Tie-Rod

Uprights

The uprights are important components in this design, as they connect the suspension and steering systems to each other. Although they should be as lightweight as possible, they also need to be able to withstand the forces applied on them during driving. Figure 25 below shows how these forces were calculated and used to create an appropriate design. The maximum vertical and lateral forces were calculated to be 380 lb and 570 lb, respectively. This information was used to develop a lightweight, efficient design that was individualized for each upright. These front and rear upright designs are shown in Figures 26 and 27 below.

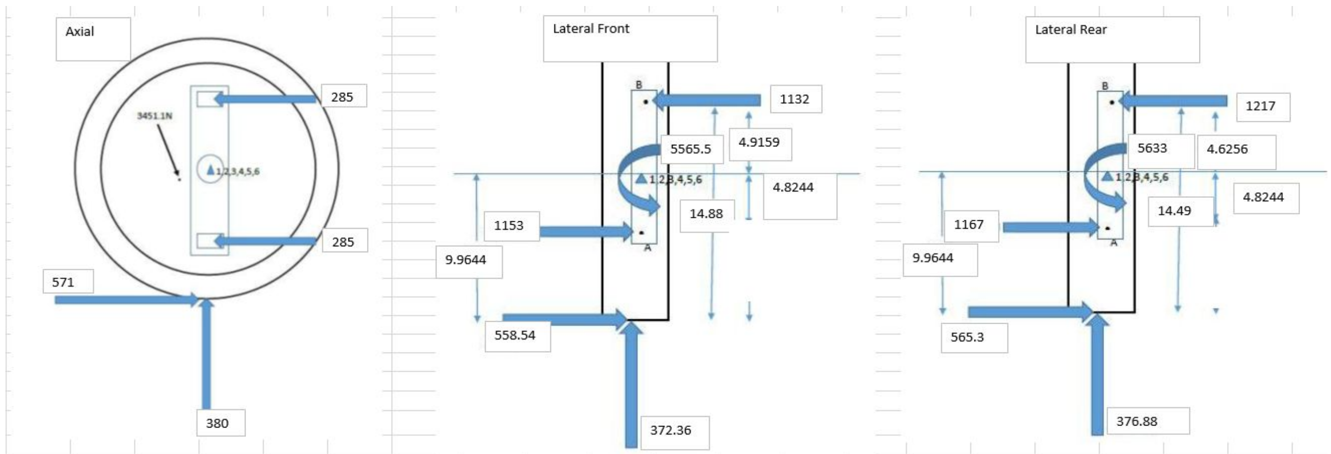
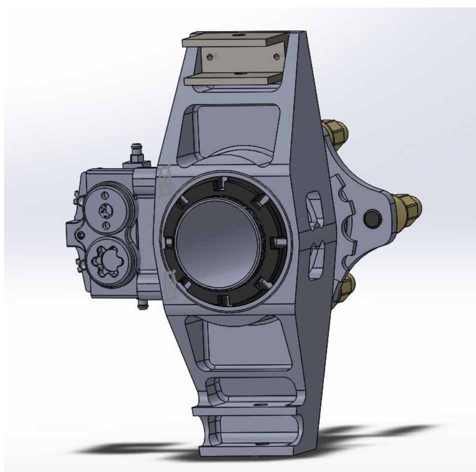
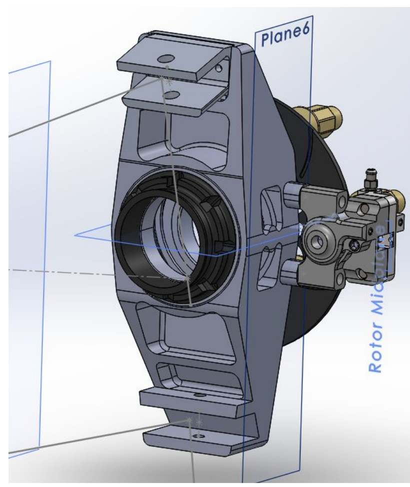


Figure 25: Upright Forces





Figures 26 and 27: Initial Front and Rear Upright Designs

Although the new uprights were designed to be efficient and lightweight, it was determined that they would be too difficult to machine, so instead the uprights from last year's car were reused in this design. This was done in order to maintain simplicity and to cut back on time. If there had been more time, the new uprights would have been modified until they were appropriate to use. The reused uprights are pictured below.

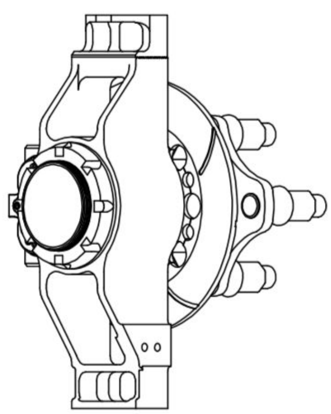


Figure 28: Final Upright Design

Manufacturing

When designing the different components of the suspension and steering systems, strength and weight were the two main deciding factors for material choice. Most parts were made from 6061 aluminum, as this is a relatively cheap, lightweight material that is still strong enough to function for most of the components. However, the uprights were made from 7075 aluminum because of the intense force they need to withstand. The steering column was made from 4130 steel, chosen for its weldability and fatigue strength. Each section had different thicknesses: the upper column thickness was 0.13” and the lower thicknesses were 0.065”. The supports that hold these were made to be 0.095” thick, and the supports that connect to the frame were 0.035” thick. Each of the thicknesses was selected to fit the bearings and adapters as well as to ensure that the tubes could withstand the necessary amount of force. Once the material designations were chosen, manufacturing could begin.

Since most of the suspension and steering components were custom-designed and could not be purchased, they were manufactured in The University of Akron’s machine shop. Each of the team members had to complete eighteen modules within the Tooling U-SME manufacturing education course. The course covered various machines and safety information that one would need before entering a manufacturing setting.

Most of the components required the use of the cold saw or bandsaw to cut the parts to length. However, some components like the stanchions needed to have more precise dimensions, so the mill was used to gradually cut the parts down. The lathe was used to hollow out or add threads to some parts, like the placeholder tubes pictured in Figure 29 below. The drill press was used to drill holes into the steering column sections and the thin plates. For the most complex parts, like the uprights and steering knuckles, the CNC machine had to be used to achieve the necessary level of precision. After machining, several parts like the steering column tubes were put through the sandblaster to properly finish the surface. The deburring machine was used for any part that had been machined. Figure 30 below shows most of the manufactured steering system components.

Figures 29 and 30: Placeholder Tubes and Manufactured Steering System Components

Testing

Since the University of Akron had to shut down and the competitions were canceled due to COVID-19, this year's design has not yet been fully assembled and tested. Figures 31 and 32 below show the latest progress, and it can be seen that the suspension system has not yet been installed. There is no way to know for sure how successful the design would have been until the car is able to be finished, but for now the models and simulations can be analyzed to get an idea of the way the car would perform. The previous sections outlined the MATLAB/SIMULINK simulation that used the quarter-car model to show how the car would have raced. The results showed that the car would have a stiff, reactive suspension system, which is desirable in a race car. The motion studies that were iterated for the pull-rod design contribute to ideal mounting points with an optimal 1:1 ratio of the pivot of the bellcrank, dampers, and pull-rods. The various simulations and studies that were conducted lead to the conclusion that the car would perform as desired under typical conditions at the testing facility and in a competition.

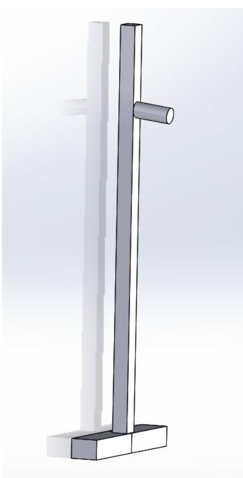
Figures 31 and 32: Latest Installation Progress

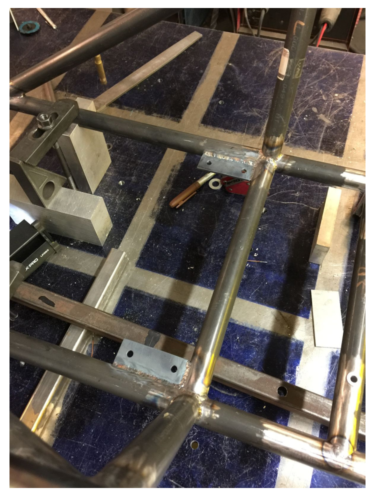
Although the ZER-20 car will not be fully assembled this semester, its construction should be completed as soon as possible so that future designers can learn from the way that it drives. The simulations and studies are extremely valuable, but ultimately they are only theoretical, and the best way to measure the true success of the design is to actually build it and test the physical model. Particular attention should be paid to the way it feels to drive the car during braking, acceleration, and cornering, as these are the times when the suspension system is pushed to its limits and becomes the most useful.

Assembly/Installation

After all components were machined, the team began to weld and fit the pieces together. All welding was tungsten inert gas (TIG) welding and was done by Jimmy Volcansek, lead of the accumulator system for the Formula Electric team. Before welding, all pieces were measured relative to each other and loosely fitted as an assembly to ensure proper fitment. Once fitment was ensured, the parts were sanded using a wire wheel and then cleaned using acetone. When trying to fit the tubes for the steering column supports, the team grinded away small portions of the tube ends, trying to mate them with the curved frame tubes.

While many of the suspension components were not able to be assembled due to the COVID-19 shutdown, the steering column was mostly assembled. The three column tubes were welded together and attached to the steering rack to ensure that the assembly fit. A T-bar was designed and manufactured to ensure that the upper steering column support could be welded to the frame in the right location. Figure 33 shows the T-bar design and Figure 34 shows one of our group members checking its measurements prior to installation use. Figure 35 below shows the steering rack mounting plates after they had been welded to the frame. Once this was done, the steering rack was fastened to the plates, and the team ensured that each side of the steering rack could move outward without interfering with the frame.





Figures 33, 34, and 35: T-Bar Design, Measuring the T-Bar, Mounting Plates Welded to Frame

Rules Compliance

Listed in the Appendix are the rules pertaining to the steering and suspension subsystems according to the FSAE Rulebook. Each rule was carefully analyzed to ensure that this year's design would pass all inspections at competition. Although this design complies with all the rules, a select few of them had a direct impact on the design.

Rule V.3.2.5 restricts the free play in the steering system to 7°, which led to the decision to use tapered pins instead of straight pins to secure the u-joints in the steering column. Although the car has not been tested yet, it is anticipated that the tapered pins will help limit the amount of free play to approximately 2°.

Rule V.3.3.4 specifies that the steering wheel needs to have a continuous perimeter with no concave sections, so this design uses a steering wheel that resembles an oval with several cutouts. Because of Rule V.3.3.1, the steering wheel is well under the front roll hoop to ensure that even if the driver's hands are at either end of the steering wheel, they will be lower than the front roll hoop, ensuring they would not be crushed in a crash that resulted in an overturned car. To comply with Rule V.3.3.2, the steering wheel must be quickly removable without the use of tools by a seated driver. This was accomplished by welding a quick-release spline to the top end of the steering column; this mates with a quick-release that is mounted on the back side of the steering wheel. The final steering wheel design is shown in Figure 36 below. Figure 37 shows the initial finite element analysis that was conducted to ensure the steering wheel could withstand the force a driver would place on it during steering.

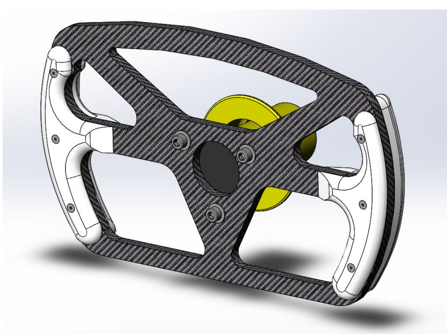


Figure 36: Steering Wheel Design

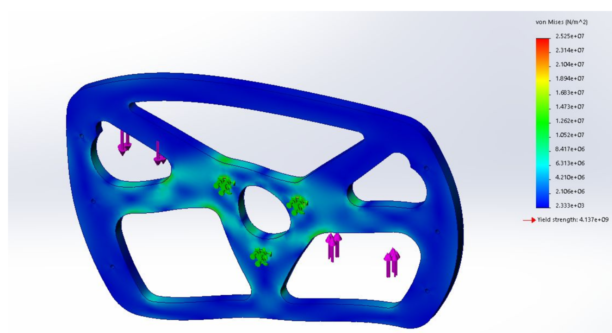


Figure 37: Steering Wheel FEA

In accordance with Rule V.3.2.4, nylon spacers were placed on either side of the steering rack to stop any steering components from coming in contact with the suspension, body or the frame of the car. This prevents steering at an unreasonably great angle, reducing the possibility of locking up the front tires when steering as much as is possible to one side. Since Rule V.3.2.6 requires that the steering rack be mechanically attached to the frame, the thin plates described in the Design and Design Methodology section of this report were manufactured and welded to the frame. To comply with Rules V.3.2.1-V.3.2.3, the steering of the front tires occurs manually with driver input to the steering wheel with no driver aids such as power steering or any other electrical means of steering. The steering of the front tires is mechanical in nature, accomplished with the use of a rack and pinion gear that is turned by a steering column connected to the pinion gear at one end and a steering wheel at the other. Once the wheel is turned, the tie rods that are connected on either side of the steering rack and also connected to the steering knuckles on the hub assemblies then move linearly with the pinion gear to effectively turn the car left and right.

Conclusions/Discussion

Although it is believed that this year's suspension and steering designs will be successful, there is no way to know for sure until the car can be fully assembled and driven. Next year's team should use this car as a way to test different aspects of these subsystems so that they can be improved upon in the future. While the theoretical analysis and simulations indicate that the suspension system will be relatively stiff and reactive, as desired, and perform well during tight cornering, there are still a few recommendations that would help improve future designs.

This year's design did not utilize anti-roll bars because of concerns regarding time, cost, weight, and packaging. However, if they are able to be included, using a front and rear anti-roll bar will give some added control over the stiffness of the suspension. They can be designed with multiple mounting holes, allowing them to be adjustable. This opens up the possibility of making significant adjustments after assembly and testing, which would be very useful.

There are two major choices for ways to connect the sections of the steering column: universal joints and gears. Universal joints are cheaper, assuming the design is sufficient only using two of them, and there is an advantage in the fact that the team has used them for several years now. However, it would be beneficial to consider using gears instead. Although they are slightly more expensive, they offer more freedom in the design of the steering column. They also introduce the possibility of moving away from a 1:1 rotation ratio; bevel gears with a 2:1 ratio can be used, allowing for the full range of motion of the steering rack. This year, this range of motion was reduced to 2.36 inches instead of the full 3.25 inches, taking away from the ability to make sharper turns. Although the team has never used gears before, they should be considered for future steering designs.

One of the team goals is always to reduce the weight of the car, and for the suspension system, this can most easily be accomplished by redesigning the uprights. This year, new front and rear uprights were designed to be lightweight and efficient. Unfortunately, these designs were not able to be used due to concerns about machinability. Future teams should rethink the way that the uprights are currently designed, but also keep in mind that they need to be machined in a practical way.

While this year's suspension and steering designs will likely perform well, the potential improvements described above should be considered. Clear team and subsystem goals should be decided upon early, and communication within the subsystem as well as with the other subsystems should be a priority throughout the entire design process. Completing the design and simulation early allows for time to make adjustments and complete manufacturing and assembly on time. Since the suspension and steering systems are so broad and include so many unique components, it is easy to get caught up with one aspect of the design and fall behind. This makes it important to set several goals and to understand how much of a commitment it is to design and manufacture these subsystems.

References

Bauer, N., Kuhar, A., Sheridan, S. (2019). FSAE Zips Racing Electric: Suspension Subsystem for ZER-19. The University of Akron.

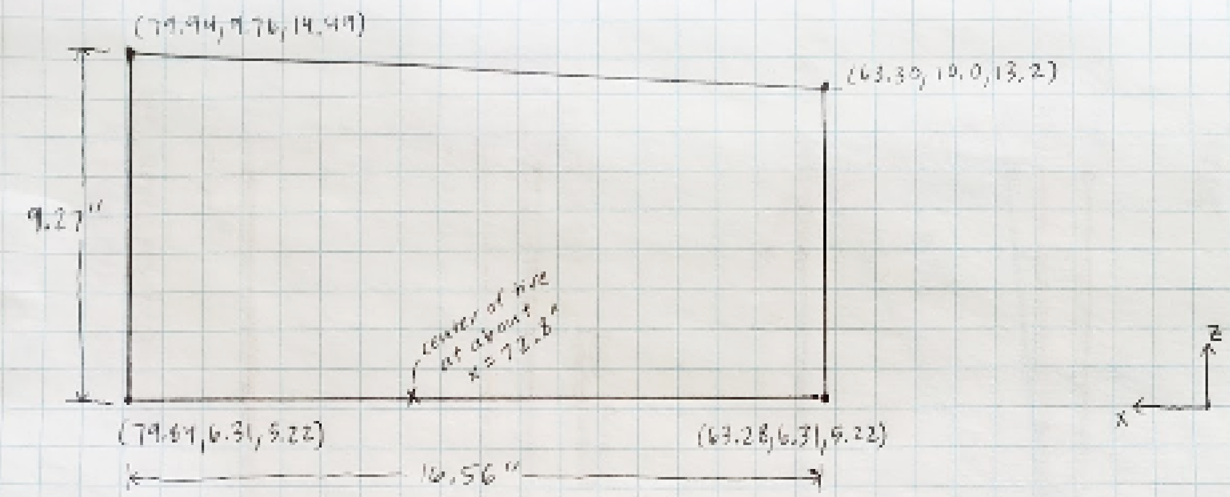
Milliken, William F., and Douglas L. Milliken. Race Car Vehicle Dynamics. Society of Automotive Engineers, 1995.

Theme, R. (2019, September 10). Suspension Secrets - Suspension Geometry Explained & Applied. Retrieved May 02, 2020, from https://suspensionsecrets.co.uk/

Appendix

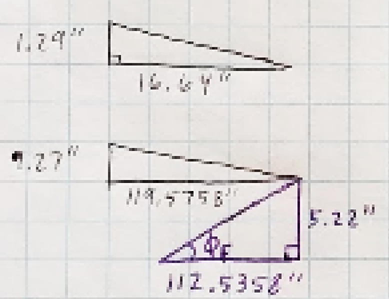
Anti-Dive Hand-Calculations

Anti-Dive Calculation



$14.49'' - 13.2'' = 1.29''$

$79.94'' - 63.30'' = 16.64''$



$14.49'' - 5.22'' = 9.27''$

$\frac{9.27''}{1.29''}(16.64'') = 119.5758''$

$119.5758'' - (79.84'' - 72.8'') = 112.5358''$

$$\%\ \text{Anti-dive} = (\%\ \text{front braking effort})(\tan \Phi_F)\left(\frac{L}{h}\right)$$

Assume % front braking effort = 60%

$L = 60.8''$ (wheelbase)

$h = 11.05''$ (center of gravity height)

$$\Rightarrow \%\ \text{Anti-dive} = (0.6)\left(\frac{5.22''}{112.5358''}\right)\left(\frac{60.8''}{11.05''}\right)$$

$$= \boxed{\%\ \text{Anti-dive} = 15.3\%}$$

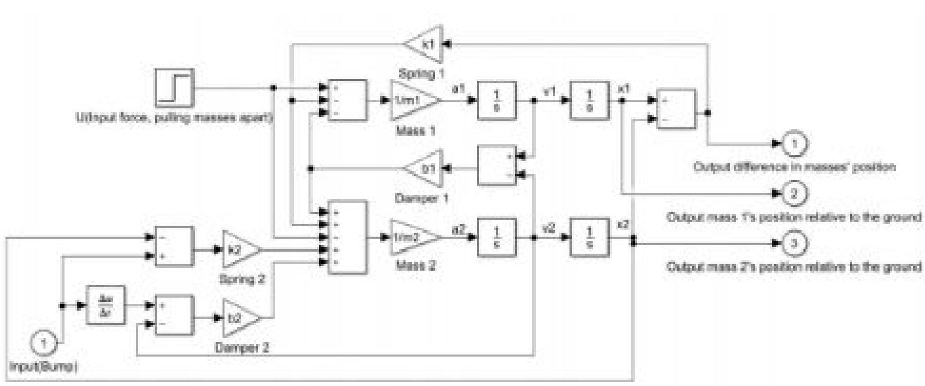
Computer software gave Anti-dive = 18% ≈ 15.3%

MATLAB Simulation Code

```
% Preliminary system model for a quarter car
% Same base code as was used for ZER-19, which was based on code from:
% "http://ctms.engin.umich.edu/CTMS/index.php?example=Suspension&section=SystemModeling"
% "http://ctms.engin.umich.edu/CTMS/index.php?example=Suspension&section=SimulinkControl"

% Nomenclature

% Front
% (m1) 1/2 front mass 58.87 kg
%
% (m2) individual suspension mass 27.94 kg
%
% (k1) spring constant of suspension system 125 lbf/in - 21890 N/m
%
% (k2) spring constant of wheel and tire 582 lbf/in - 101,923 N/m
%
% (b1) damping constant of suspension system 150 lbf.s/in - 26269 N.s/m
%
% (b2) damping constant of wheel and tire 500 N.s/m
% Back % - The same as the front unless listed
% (m1) 1/2 back mass 64.86 kg
%
% (bk1) spring constant of suspension system 175 lbf/in - 30647 N/m

% Establish Simulink Model 'suspension2' Parameters

clear all;
close all;
clc

% Note: confusion using the model provided and the definition/nomenclature \
% of system parameters: M1,M2,K1,K2 used in declaration but m1,
% m2,k1,k2 used in model
% % Fix: changed all M1,M2,K1,K2 to lower case, i.e. m1,m2,k1,k2

M1 = [58.87 64.86]; % (M1) 1/4 bus body mass
m2 = 12.8175; % (M2) suspension mass
K1 = [21890 30647]; % (K1) spring constant of suspension system
k2 = 101923; % (K2) spring constant of wheel and tire
b1 = 26269; % (b1) damping constant of suspension system
b2 = 500; % (b2) damping constant of wheel and tire

s = tf('s');

for i = 1:1:2
% Select mass and spring depending on case
m1 = M1(i);
k1 = K1(i);
if i == 1
        fprintf('\nFront:\n')
else
        fprintf('\nBack: \n')
end

% Extracting a linear model into MATLAB
[A,B,C,D]=linmod('suspension2');

% Extract for u = x2-x1
[uNum,uDen]=ss2tf(A,B,C(1,:),D(1));
uT = tf(uNum,uDen);
```

```
% Extract for position of the sprung mass relative to ground: x1 + x2
[smNum,smDen]=ss2tf(A,B,C(2,:),D(2));
smT = tf(smNum,smDen);

% Extract for position of the unsprung mass relative to ground: x2
[umNum,umDen]=ss2tf(A,B,C(3,:),D(3));
umT = tf(umNum,umDen);

figure(1)
step(uT)
hold on
fprintf('\nu(t):\n')
display(stepinfo(uT))
damp(uT)
title('Step Response - u(t) = x1-x2')
xlabel('Time')
ylabel('Amplitude')

figure(2)
step(smT)
hold on
fprintf('\nSprung Mass:\n')
display(stepinfo(smT))
damp(smT)
step(umT)
fprintf('\nUnsprung Mass:\n')
display(stepinfo(umT))
damp(umT)
title('Step Response - Sprung & Unsprung Mass')
xlabel('Time')
ylabel('Amplitude')
end

figure(1)
legend('F','R')

figure(2)
legend('Sprung Mass - F','Unsprung Mass - F','Sprung Mass - R','Unsprung Mass - R')
```

SIMULINK Model



FSAE Rules for Competition

V.3 SUSPENSION AND STEERING

V.3.1 Suspension

V.3.1.1 The vehicle must be equipped with a fully operational suspension system with shock absorbers, front and rear, with usable wheel travel of at least 50 mm, with a driver seated.

V.3.1.2 Officials may disqualify vehicles which do not represent a serious attempt at an operational suspension system, or which demonstrate handling inappropriate for an autocross circuit.

V.3.1.3 All suspension mounting points must be visible at Technical Inspection by direct view or by removing any covers.

V.3.1.4 Fasteners in the Suspension system are **Critical Fasteners**.

V.3.1.5 All spherical rod ends and spherical bearings on the suspension and steering must be one of:

- Mounted in double shear.
- Captured by having a screw/bolt head or washer with an outside diameter that is larger than spherical bearing housing inside diameter.

V.3.2 Steering

V.3.2.1 The Steering Wheel must be mechanically connected to the front wheels.

V.3.2.2 Electrically actuated steering of the front wheels is prohibited.

V.3.2.3 Steering systems must use a rigid mechanical linkage capable of tension and compression loads for actuation.

V.3.2.4 The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four bar linkage at one of the pivots). The stops may be placed on the uprights or on the rack and must prevent the wheels and tires from contacting suspension, body, or frame members during the track events.

V.3.2.5 Allowable steering system free play is limited to seven degrees (7°) total measured at the steering wheel.

V.3.2.6 The steering rack must be mechanically attached to the frame.

V.3.2.7 Joints between all components attaching the Steering Wheel to the steering rack must be mechanical and be visible at Technical Inspection. Bonded joints without a mechanical backup are not permitted. V.3.2.8 Fasteners in the steering system are **Critical Fasteners**, see **T.8.2** V.3.2.9 Spherical rod ends and spherical bearings in the steering must meet **V.3.1.5 above** V.3.2.10 Rear wheel steering may be used.

V.3.3 Steering Wheel

V.3.3.1 In any angular position, the top of the Steering Wheel must be no higher than the top-most surface of the Front Hoop. See figure following **F.5.8.6**

V.3.3.2 The Steering Wheel must be attached to the column with a quick disconnect.

V.3.3.3 The driver must be able to operate the quick disconnect while in the normal driving position with gloves on.

V.3.3.4 The Steering Wheel must have a continuous perimeter that is near circular or near oval. The outer perimeter profile may have some straight sections, but no concave sections.