Steering System and Anti Roll Bar Design

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2022 Formula FSAE

Capstone I: Technical Report

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Introduction: As a suspension team member on the 2022 York College Formula FSAE team, I was tasked with the research, design and analysis of our car's steering and anti-roll bar systems. The following report outlines all the research, design and analysis I have completed throughout the summer of 2021 semester.

Design Goals: The 2022 team set out to improve upon some of the major flaws found on the previous team's car. Unfortune to them and us, none of their hard work could be validated at the annual FSAE competition. This puts us in a unique position, and doesn't give us our team as much insight as we might have hoped in terms of what does and doesn't work for our car.

During one of our first team meetings, we outlined our team goals for FSAE competition. They were the following:

- Finish endurance withing 145% of the winner
- Improve upon acceleration time
- Emphasize focus on static events
- Pay special attention to comments judges make

As any proper engineer would do, setting goals both as a whole team, sub teams and individual goals allows you to validate what you are trying to achieve. With suspension being an important aspect to our vehicle's dynamics, we set out to outline some goals for our sub-team. The entire table can be found in Appendix II: Tables, Plots and figures (Table 2). Some notable design goals related with steering and ARB (anti-roll bar) are:

- Turning radius of 120 in
- Implement Ackermann steering geometry
- ARB to reduce body roll by 10%

Individually, my goals were to build a steering system that addressed the serious shortfalls of the 2021 car. The largest shortfall being the slop in the steering column. There was 20° of play before the wheels would being to move. Not only was this extremely dangerous, it is also against the FSAE rules. Knowing this, I set out to design and build a solid steering column, and essentially eliminate this slop. Since the steering wheel is one of the drivers only connections to the car, I understand that this is extremely important to me to get this right for the rest of the team.

Research: A significant number of factors have to considered when designing any component on a race-car. The following section outlines all of the research I have done to contribute the best possible components to our Formula Car. One of my personal goals on our car was to really left no stone unturned when researching and designing. I want all factors to be considered for both steering and anti-roll bars systems

Previous Teams Information: One of our first tasks was to come up with several design goals. To have some sort of reference frame I went back to all previous tech reports for those who designed steering and suspension components. I was able to compile some information that may be useful to future team members. First there is a spreadsheet with previous team's suspension goals, found in appendix II. A table with steering information can be seen below in table 1.

Table 1: Metrics from previous YCP FSAE teams, and who was responsible for design during that respective year

		Steering Metric				
Person(s) Responsible (Steering)	Car Year	Turning Radius (in)	Steering Wheel	U-Joint Type	Ackerman (%)	
Brett Tarlton	2017	147	210	Double	100 (Judges claimed it was higher)	
Jonathan Yee	2018	122	210	Double	100	
Lucas Rainville	2019	150	210	Double	100	
Gabriel Frangiadis	2020	125	Not Mentioned	Double	100	
Gerald MacDonald	2021	125	248	Double	100	

Suspension Terminology: My familiarity with terminology with suspension was originally not proficient. Overall, I knew I had a basic overview of how cars worked. However, I had never previously researched some of the important vehicle dynamics. A lot of these important dynamics are determined by the suspension team, so it was important to learn the following terms.

Camber Is the angle between the vertical axis of the wheel and the vehicle. It effects the handling of the car as whole. Negative camber is when the wheels are angled in towards the centerline of the car when viewing it from the front, and positive is when they are angled away.

Caster angle is the angle from the vertical axis of the car to the rest of the car. This will affect how the uprights sit in the wheels of the car.

Toe is the angle of the wheels looking at the car from a top down view. Toe in is when the front of wheels are angled towards the centerline of the car, and toe out is when they are angled away from the car.

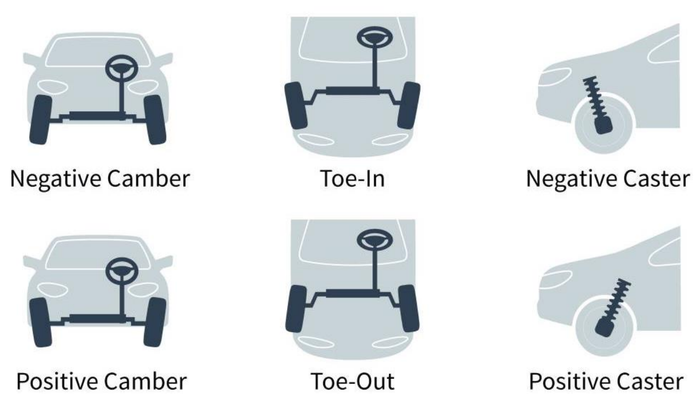


Figure 1: Camber, Caster, Toe all visualized clearly [https://www.cars.com/articles/when-should-you-get-a-wheel-alignment-1420681259841/]

King pin inclination is the angle of the entire wheel assembly relative to a vertical axis perpendicular to the ground. Typically, the car is not perfectly vertical and the king pin axis effects the position and location of the control arms.

Scrub Radius can be defined as the distance between the king pin axis and the center of the tire contact path.

Slip angle is the angle between where the car is actually travelling and the direction the tires are pointing. For future teams it would be worth exploring this a bit more and seeing how much you could optimize the slip angle. If done properly you could find the ideal tires and possibly take advantage of anti-Ackermann steering.

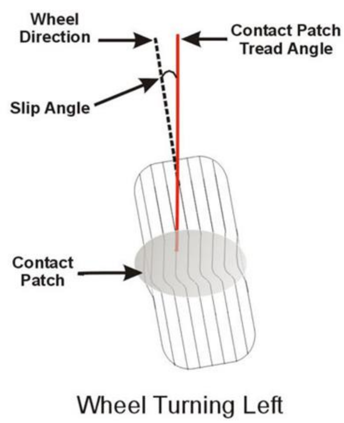


Figure 2: Identifying slip angle and where the angle lie relative to the rest of the wheel [http://www.clarks-garage.com/shop-manual/susp-15.htm]

Wheelbase and trackwidth is the overall length of the car from center of front axle to center of rear axle. Trackwidth is the width of the car in the front or the rear, from center of tire to center of tire. Typically, the rear trackwidth is slightly larger the front trackwidth. Our goal for trackwidth and wheelbase were (49.5in Front /47in Rear) and 61in respectively.

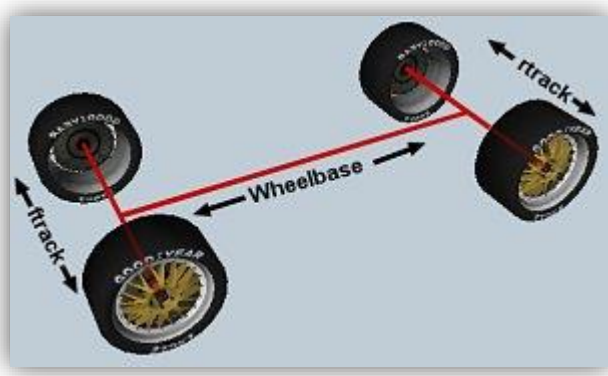


Figure 3 : Wheelbase and trackwidth diagram [http://speedsims.org/tutorials/NH_physics_basics.html]

Understeer/oversteer are important concepts to the driver and designers of the car. Understeer is when the car isn't turned enough and goes to the outside edge of the turn. Whereas oversteer is the opposite when the car is turned too much into the turn. There are serious conversations about which is better for the car, however it largely comes down to driver preference. It is easier to manage understeer than oversteer in a turn. If the car takes the turn too sharp, and oversteer occurs, it would be increasingly difficult to recover. If you understeer the car stays stable and you just take the turn a bit wide.

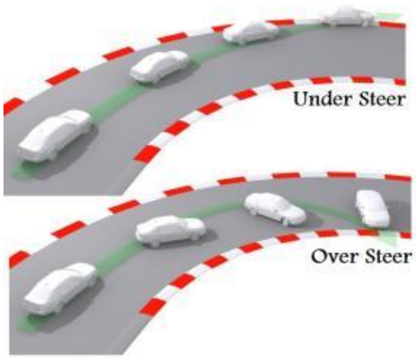


Figure 4: An informative diagram on how under and oversteer work [https://topalwaysdown.wordpress.com/2011/04/15/snap-oversteer-making-the-bear-dance/]

Steering Characteristics:

Turning radius is the smallest radius the car can turn itself upon. This is important for both the endurance and skidpad events. During the endurance event, there is at least one hair pin turn per lap. We don't want our drivers not being able to follow the driving line they would want. Based on the 2021 FSAE rules the tightest turn on the endurance event would have a 9m (29.5ft) outside diameter, with a minimum track width of 3.5m (11.5ft). For the skidpad event the minimum inside circle radius is 15.25m (50ft). When analyzing what our goal turning radius should be I created some SolidWorks sketches that were able to give me a visual. We determined our goal for minimum turning radius should be 120 inches. This should give us plenty of clearance when turning during both endurance and skipad events. I used our trackwidth goal of 49.5in when creating my sketches.

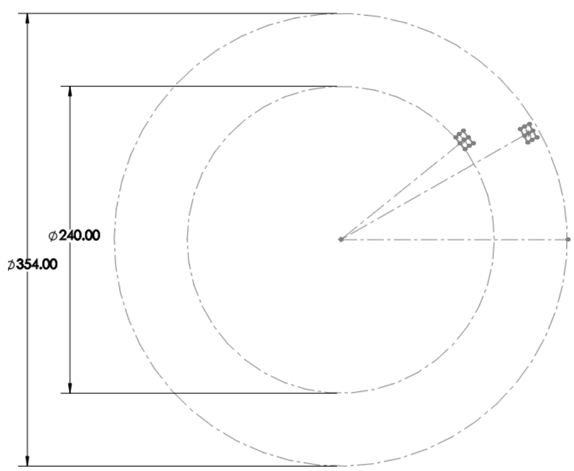


Figure 5: SolidWorks sketch detailing the varying turn radii of the track, allowed me to get a visual to determine our turning radius

Bump steer is an important concept that should be accounted for when determining where to put your tie rods and steering rack. Bump steer happens when you go over a bump and it unintentionally takes control of the steering wheel for you. This can be extremely dangerous, especially when travelling at high rates of speed. To mitigate the effects of bump steer, you will need to best align the tie rod, lower and upper control arm planes to form an imaginary instant center.

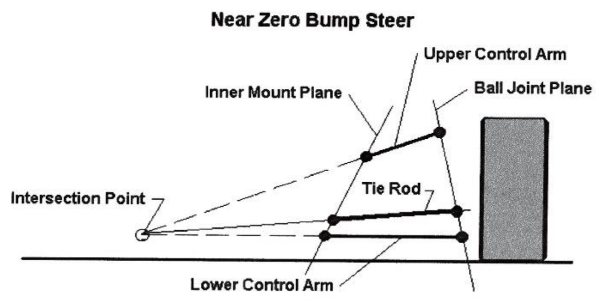


Figure 6: Diagram of how to align several planes to form an ICR to achieve no bump steer [https://honda-tech.com/forums/road-racing-autocross-time-attack-19/truth-behind-bump-steer-3285683/]

Ackermann geometry was developed to eliminate the effects of your inside wheel having to travel at a different angle than the outside turning wheel. When turning if both your inside and outside wheels turned at the same angle you would have significant slip in the tires. Ackermann pro-rates the angles of the tires based on how tight of a turn you are making. In pro-Ackermann geometry the inside wheel turns more than the outside wheel, in anti-Ackermann the inside wheels turns less than the outside wheel. Ackermann steering can be implemented by drawing an imaginary line from the center of the rear axle, through the kingpin axis line. If the tie rods lay on any points on that line, Ackermann would be implemented. It is possible to adjust the amount Ackermann implemented, by moving the imaginary line a bit inside. Anti-ackermann principles can be taken advantage of at high rates of speed (+100mph) and could be complimented nicely with optimizing the slip angle of the tires.

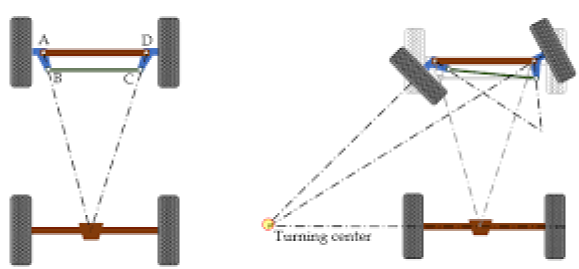


Figure 7: Ackermann steering geometry principles visualized [https://www.ripublication.com/ijm&s17/ijm&sv12n1_05.pdf]

Front vs Rear Steer is something I would recommend future teams dive into a bit more. Front steer is when the steering rack is placed in front of the front axle, and rear steer is when its placed behind the front axle. I could not find significant research supporting why either front or rear steer is more beneficial. Although I did gather the following information. Front steer reportedly improves handling, stability and will feel more stable in corners. Whilst rear steer makes steering a bit easier, its adds a small oversteer effect and doesn't feel as stable.

Anti Roll Bar: Anti roll bars are also known as torsional, or anti sway bars. All of these names can be interchangeably used. ARB are added to cars to increase their stiffness and keep the cars load transfer to a minimum whilst cornering. Typically connected to the rockers the ARB transfers the body roll into a torque transmitting through the bar. As we all known Newtons 2nd law, for every action there is an equal and opposite reaction. So as the body of the car rolls on side, it provides an equal reaction to the other. This keeps the cars center of gravity closer to the ground. Reducing the overall body roll of the car helps with its dynamics, Implementing an ARB provides some adjustability if we determine that there is too much under or over steer. Our design goal is to reduce body roll by 10%. The 2021 team implemented an ARB but since they did not go to competition we are unaware if it provided significant benefit.

Design & Analysis: Using what I have learned it was time to design the steering and anti-roll bar systems for the 2022 car. I understood the importance of these took my time to verify the designs I have come up with.

Determining final suspension points: Noah Dekker and I worked hard implementing all aspects of steering (bump steer, Ackermann) into our final suspension points. This turned out to be more challenging that originally thought of. However, the best approach we developed when determining points in both Optimun G was using a SolidWorks sketch. This allowed us to get a good reference on where our tie rod points could lie to provide us with near zero bump steer and 100% Ackermann steering. If you want to read more about determining the rest of the suspension points, Noah Dekker's tech report would be a great reference.

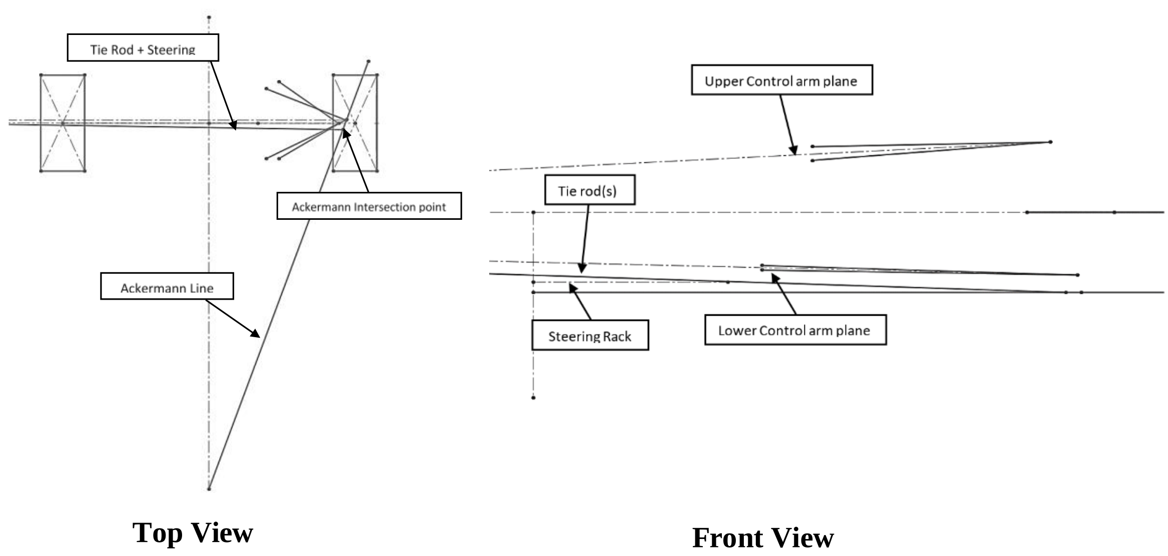


Figure 8: SolidWorks sketches Noah Dekker and I created to assist our determination of the final suspension points. You can clearly see the bump steer and Ackermann lines labelled

Steering: Upon my research, almost all of the teams in the past have gone with a double u-joint design for their steering column. Typically, the difference in the steering system would be where the u-joints were located and the angle at which they were positioned, as well how well the system as a whole was built. My goals for steering was to build a very solid, rigidly connected system to reduce the amount of play in the steering wheel. I had a general idea of where our steering wheel would be positioned, however I waited for the results of the ergonomic study Jake McGrath (Frame, Cockpit) did on prospective drivers. He let me know the ideal angle of the steering wheel relative to the ground should be between 20-30°.

Ackermann Analysis: When exploring research for FSAE cars, it was apparent not every team has gone with 100% and it seems as if this is an ongoing discussion. Every YCP FSAE team in the past has planned on implementing 100% Ackermann. I wanted to explore the difference in wheel angles at differing Ackermann percentages and much the wheels would be angled. I wanted to see how much the wheel would be angled with respect to the radius of the turn. Figure 9: Showing differing Ackermann percentages and the inner wheel angle vs the radius of the turn, it shows that implementing Ackermann is a nonlinear relationship between the amount you steer vs the amount the wheels are turned Figure 9 shows that as you turn tighter, the inside wheel angle increases exponentially and not linearly. As seen, adjusting the Ackermann % makes it more difficult to turn, as the wheel angle doesn't turn as sharp, so you would need to turn the steering wheel more. This can found using the expression seen below.

$$\delta_{inner\ wheel} = \tan^{-1}\left(\frac{L}{R - \frac{T}{2}}\right)$$

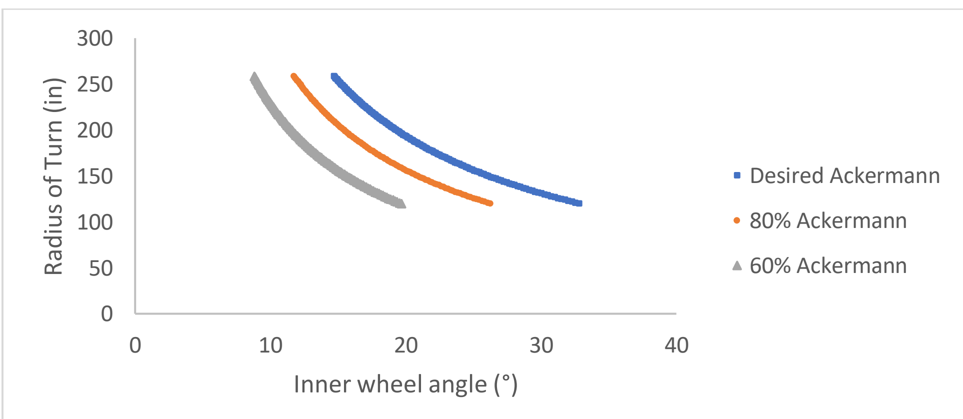


Figure 9: Showing differing Ackermann percentages and the inner wheel angle vs the radius of the turn, it shows that implementing Ackermann is a nonlinear relationship between the amount you steer vs the amount the wheels are turned

I also plotted the inside vs outside tire angle to get a reference of the max angle our wheel should be turning for 100% Ackermann steering. I would encourage future teams to research a bit more on whether the Ackermann percentage makes a significant enough difference. From my understanding if its not complimented with slip angle correctly, it won't have a significant impact. From my Ackermann geometry analysis, I determined the largest angle the wheel should turn is 33° on the inner wheel, and 23° on the outside wheel. This gives the wheels 56° of travel from lock to lock.

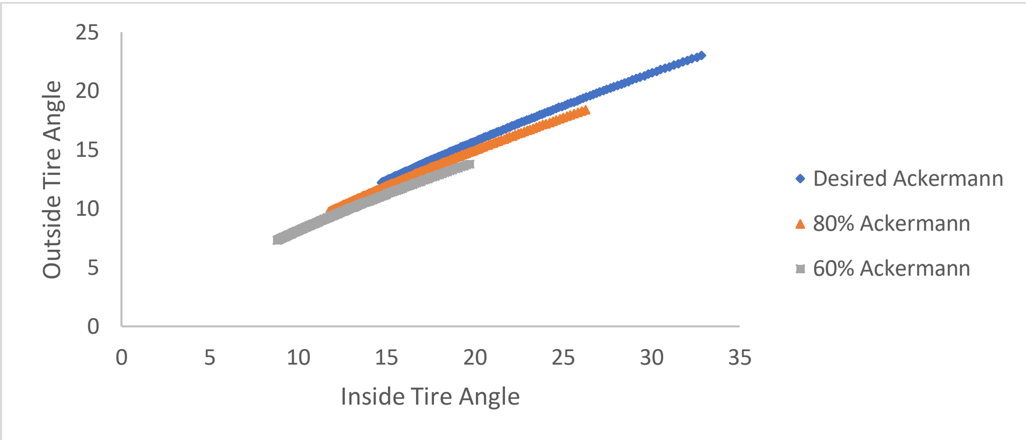


Figure 10: Outside tire angle vs inside tire angle when differing percentages of Ackermann steering are implemented

Design Concepts: I wanted to explore alternative options for our steering system rather than just saying "this is what teams have used in the past". I spent some time modelling steering gearboxes use bevel gears. I thought this was exploit the disadvantages of u-joints, in that we could obtain a constant velocity into our steering rack. These were my initial concepts so I didn't put an insane amount of time into them.

Figure 11: SolidWorks model of my two-steering gearbox, left side angle was 60°. Right picture is 120° gear angle

However, spending the time to make these concepts allowed was a great lesson. I got significantly more familiar with how the steering column would fit, and seeing the CAD models allowed me to realize the major exploits the designs had. Per the FSAE 2021 rules, a cockpit jig has to fit through the entirety of the cockpit. However, all moving suspension components are exempt from the jig. As you might be able to see in the photos above, the lower column would have to be supported somehow. As soon as I realized this I abandoned this concept and moved on to analyzing the u-joints.

U- Joint Analysis: Universal Joints (u-joints) allow us to transmit power through a misaligned shaft. There comes a slight disadvantage when using u-joints, this is that the power is not transmitted at a constant velocity through the u-joint. Another small disadvantage is that you are fairly limited on the severity of the angle you can transfer power between. However, teams have used a double u-joint in the past with varying degrees of success. To visualize this concept, I plotted the angular velocity of the output shaft vs steering input angle for a single u-joint and its misalignment angle. I assumed one could turn the steering wheel at a constant velocity of 2 rad/s. The equation of motion for a shaft with a constant velocity is below. I did the same for a double u-joint, where you can see that the second shafts angular velocity is more than the intermediate and input shaft. Gabriella Frangiadis's tech report from summer 2019 was also a great resource on this. The equation can be seen below

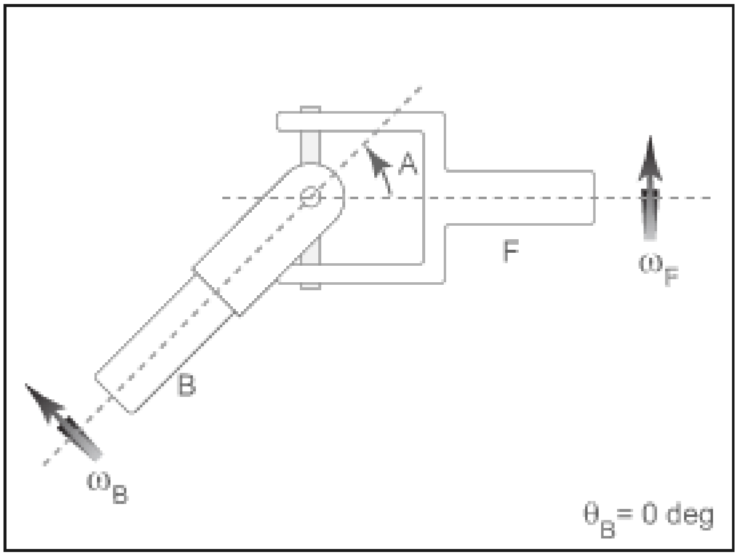


Figure 12: Visual aid for variables that determine output speed of a universal joint [https://www.mathworks.com/help/physmod/sdl/ref/universaljoint.html#]

$$\omega_F = \frac{\cos(A)}{1 - \sin(A)^2 * \cos(\theta_B)^2}$$

$$\omega_F = Angular\ Velocity\ of\ Output\ shaft$$

$$A = Angle\ between\ shafts$$

$$\theta_B = Angle\ of\ input\ shaft$$

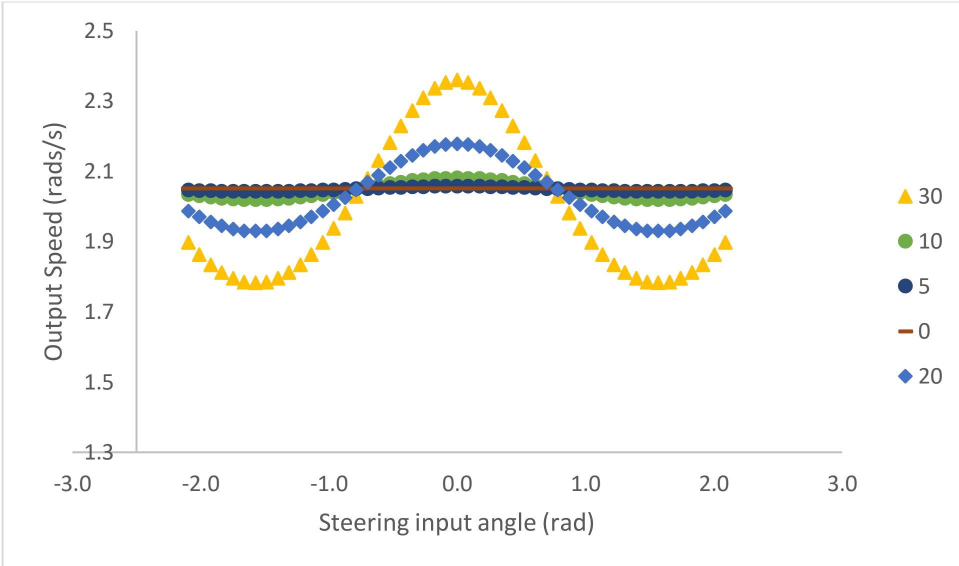


Figure 13: Angular velocity of a u-joint at varying different degrees

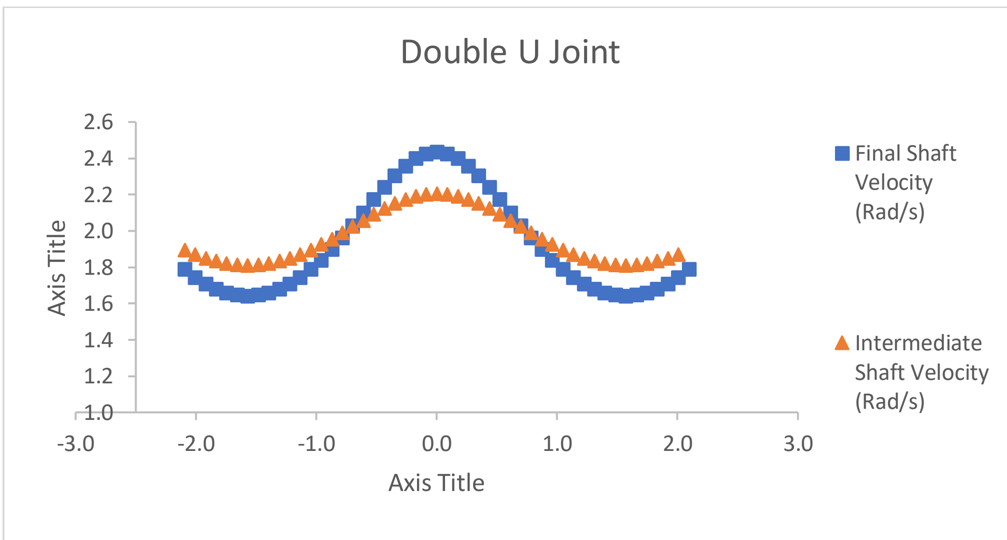


Figure 14: Angular Velocity of both the intermediate and final shafts for a double u-joint configuration

Steering Wheel & Quick Connect: The 2021 team purchased a new 11.75in wide steering wheel from jegs.com. It's very nice, has not been used in competition, however a critical rule was overlooked and we will therefore be using the older one. Which is a Alpha Flat Suede steering wheel. The smaller suede one was preferred by the prospective drivers anyway. A new quick connect and spline will need to be purchased. Upon a quick survey from prospective drivers, many prefer the quick release sleeve type over the long button. This is safer anyway, since removing the steering wheel should be easier than having to click a button, like on the 2021 FSAE car. This attached to the column via ¾-32 spline that will be welded in.

Figure 15: Steering components chosen to be used, left picture is steering wheel we own, right picture is a new quick disconnect. See Bom for more details

Steering Rack: The 2021 team had run into issues at some point. Gerald MacDonald had noted to me that he took the rack that was used on the 2021 off of the FSAE electric car in the Kinsley lobby. Upon discussion with Dr. Ericson, he had noted this rack was only used in the electric car, which only competed once. The rack was put on the 2021 car, its in great shape. Its very wide at 16 ¾ in which actually plays to our favor since we had some small issues with frame clearance. It also allows the data acquisition team to mount a steering angle sensor on the bottom, which was requested by systems integration. Chris C's tech report is a good resource on how the steering angle sensor works. Luckily, we already possess the steering angle sensor. It's a KAZ Technologies FSAE steering rack, with a rack travel ratio of 4.71" in per rev, and has 248° of travel, and weighs 3lbs.

Figure 16: KAZ Technologies Steering rack chosen to be used for 2022 FSAE car

Tie Rods: According to the simulations that were run in Optimum G, the maximum force that the tie rod will experience is 315 lbf of compression. I think we all know most materials could handle this kind of loading. However, 4130 chromoly steel (0.500 OD x 0.035") was chosen for a couple of key reasons. Noah Dekker is already planning on purchasing a bunch, not much more is needed for 2 tie rods (~15in a piece). Large reason I chose to use an alloyed steel is its ductility. If for whatever reason we get into an accident, or hit a cone, and the tie rod experiences some unexpected loading, steel will bend quite a bit and not snap like carbon fiber. This ductility could save us from further damaging more components that would prevent us from further racing. Welded inserts will allow us to minutely adjust the length as necessary.

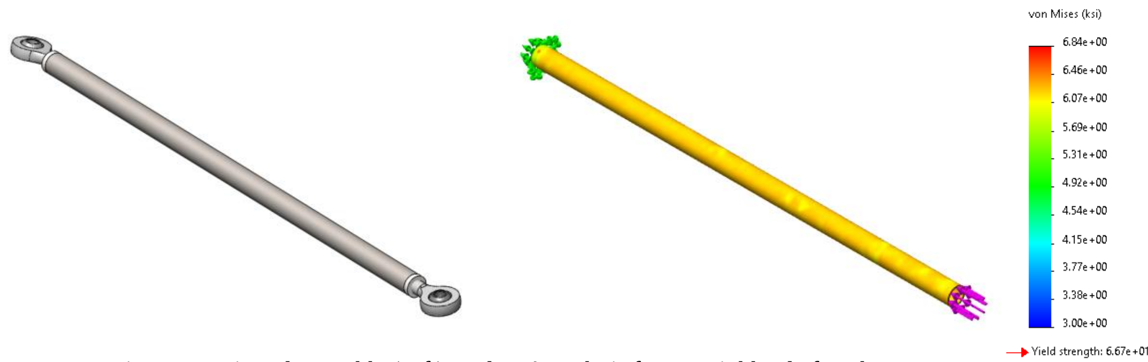


Figure 17: Tie rod assembly (Left), and FEA analysis for an axial load of 315bs

Final Design: Once I had all the desired information from Jake McGrath, I made a SolidWorks sketch to get some numbers on where the column mount would be located. As stated, before he had noted the steering wheel should be between 20° and 30°. I planned on using a 30° angle since this gives me the most amount of room for the u joints. Since the rack was already determined, I needed to find a couple of key items. First was a coupler between the rack and steering column. Then the welded inserts that would connect the u-joints together.

Figure 18: KAZ Technologies weld spline inserts 3/4-20 for 0.750 tubes with a 0.065 wall

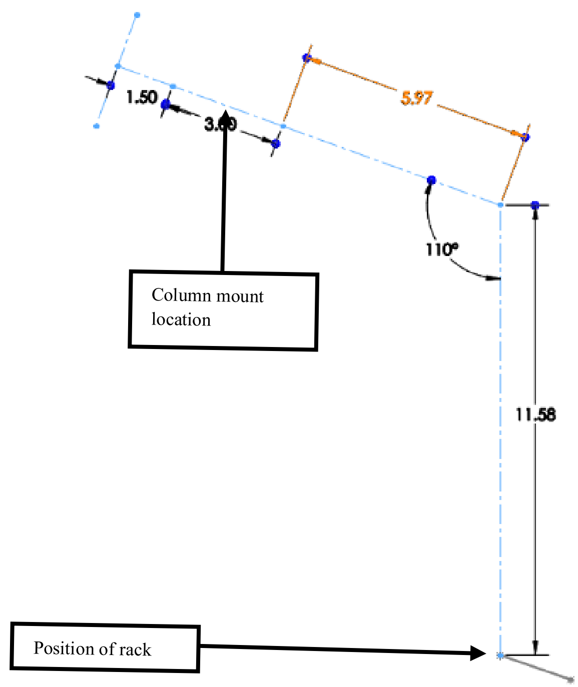


Figure 19: SolidWorks sketch I made to give me the dimensions of where the steering column mount is relative to the steering rack position

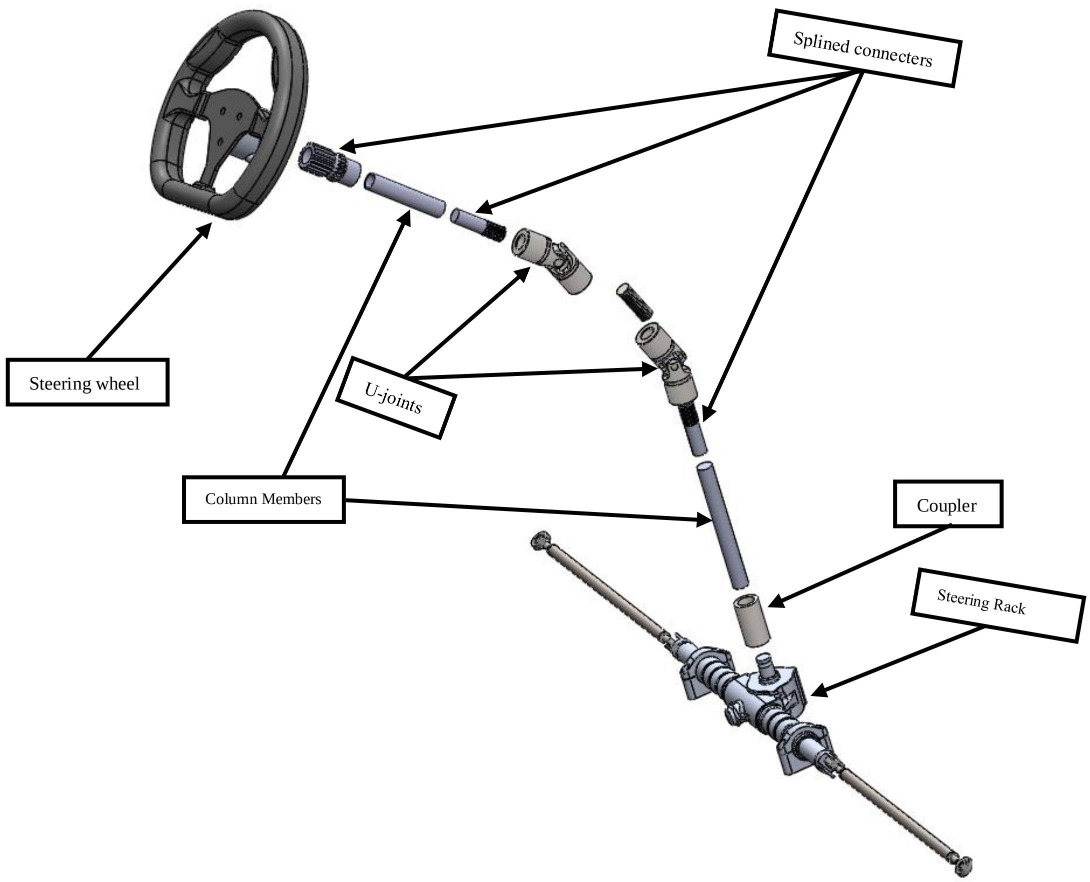


Figure 20: Full exploded view of the 2022 FSAE Steering system I designed

With a steering wheel angled at 30°, both u-joints can be angled at 20°. The u-joints are from KAZ Technologies, and I plan on buying splined inserts which can then be welded to the column. The upper and lower column members are going to made out of low carbon steel, with a .750 OD.

Analysis: According to Steven Fox's tech report, the largest force you can exert on the steering column is 60ft-lb. Which seems a bit high, however my analysis was done under this assumption. Using what we have learned from EGR264, its fairly easy to determine the ID of our column members. My hand calculation can be found in Appendix I: Calculations and Equations. We know the equation that defines the stress produced by a torque is seen below:

$$\sigma = \frac{\tau r}{J}$$

I calculated the ID to be 0.607in, which gives us a wall thickness of 0.072in. I performed FEA on two components of the steering column. First the short-splined coupler between the u-joints. Then the coupler connecting the steering column to the rack. These are the smallest members, they should experience the largest amount of stress. The FS on the splined insert is 1.25. The smallest FS on the splined coupler is 5.03.

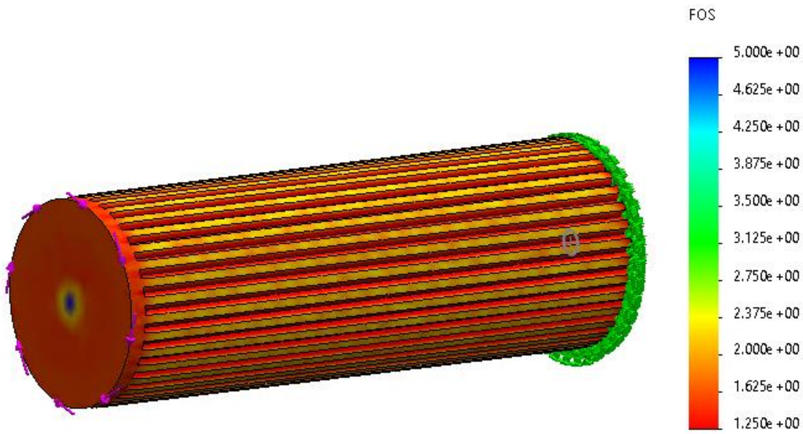


Figure 21: Factor of safety plot produced from running FEA on splined insert. Loading condition of 60ft-lbs applied, Minimum factor of 1.25 with respect to maximum loading

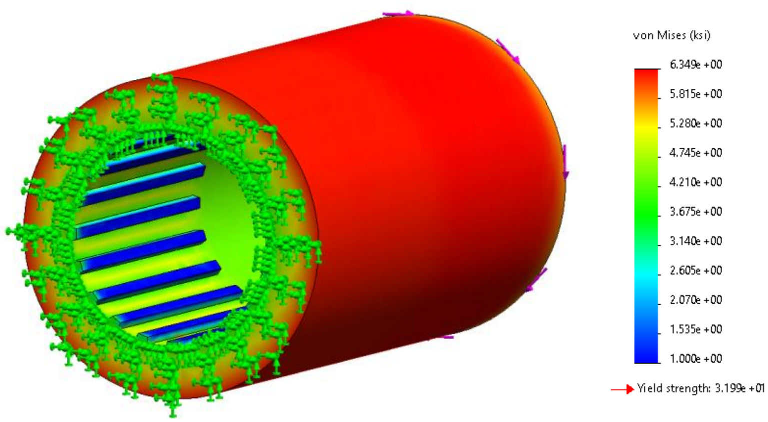
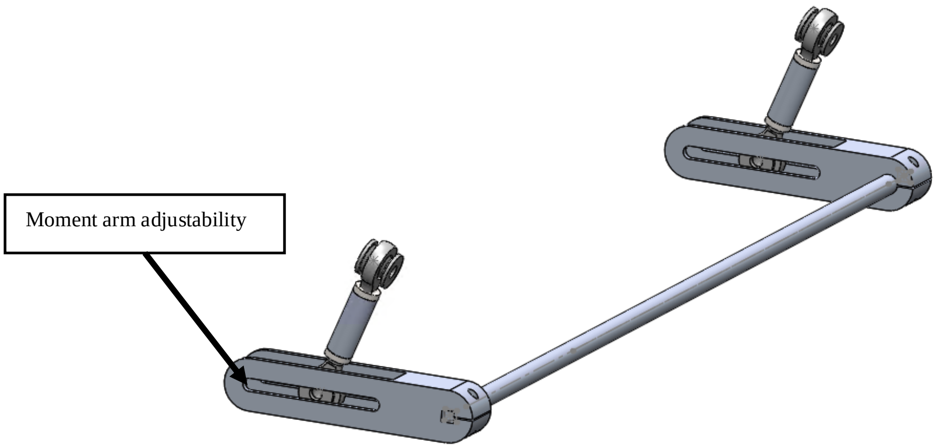


Figure 22: FEA stress plot of steering coupler loaded with a torque of 60ft-lbs, maximum stress on the outside of the coupler is 6.4 ksi

Anti-Roll Bar: (ARB) I knew that the anti-roll was important to the car, in terms of the suspension and the cars dynamics. Simply put I wasn't able to put as much time into the ARB as I would have liked, and I would encourage future teams to research a bit more. My goal for the ARB was to make it adjustable, and I didn't want to have to move around various other suspension components to make them fit. My original concept conceived on getting some round tube, heating it up, bending two 90° curves on, and flattening the ends and drilling holes. This issue with this design is that everything in suspension has to be mounted in double shear. My second design was based off of some stock ARB's I had saw online. I decided to go with the design seen below based on a couple key factors. Its lightweight and relatively small, significantly smaller than the one on last year's car. It also allows for some sort of adjustability within the car.



Analysis: The connecters I plan to make out of 6061-T6, and they allow for full adjustability if we determine we need the car stiffer or softer. Determining the thickness was actually pretty easy. Once I had a rough range of moment arm values (1.5in to 4in) I could solve for torsional stiffness and determine how thick the bar should be. It uses 0.375in round stock bar, with milled down square ends. As for material of the actual ARB, any steel should work almost identically since the modulus of rigidity (G) is around the same magnitude for all steels. See the Appendix I: Calculations and Equations for these calculations. The stiffness of our ARB ranges from (43-308) lb-ft/deg.

Conclusion: Capstone is most definitely a challenge for anyone willing to take it on. I can say with confidence the summer is the hardest semester yet. Balancing everything is incredibly difficult, and that even goes for capstone. It may be apparent in my report, but my time spent on steering vs anti-roll bars was about 80-20%. This was for a couple of reasons, mainly being that you *need* a steering system to drive the car. In terms of grading the ARB wasn't weighted as heavily. If you happen to be in charge of the anti-roll bar I would suggest doing a bit more analysis. For steering I hope you found this report extremely helpful in terms of analysis and design. Although not too different than what previous teams have used in the past. I hope that this report is a foundational step in terms of the entire steering system. Otherwise as we as a team enter the build semester my goal is build a system will almost no slop or free play in the steering whatsoever. This is important to me as I took on this responsibility. Simply put if the steering fails in any way, its comes down on me. I look forward to beginning the build phase of this project and hope we can succeed on achieving our goals (hopefully) at competition.

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Appendices

Appendix I: Calculations and Equations

(A) Stress in Tie Rods

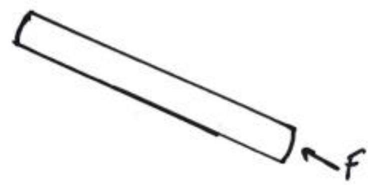
GIVEN: F = 315 lbs

OD = 0.50 IN

ID = 0.43 IN

L: 14.83 IN.

DRAW:



FIND: STRESS

SOLVE:

$$\sigma = \frac{F}{A} = \frac{315\ \text{lbs}}{\pi\left(\frac{0.50}{2}^2 - \frac{0.43}{2}^2\right)} = \boxed{6160\ \text{PSI} = 6.16\ \text{KSI}}$$

$$F.S = \frac{\text{ALLOWABLE}}{\text{ACTUAL}} = \frac{63.1\ \text{KSI}}{6.16\ \text{KSI}} = F.S \text{ OF } \underline{10.2} \text{ W/ RESPECT TO YIELD}$$

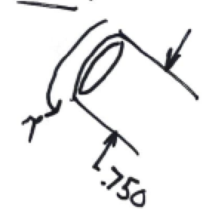
(B) ID of steering column members

GIVEN: τ = 60 ft·lbs OD = 0.750 IN

MATL: LOW CARBON STEEL.

FIND: ID OF STEERING COLUMN

DRAW:



SOLVE

$$\sigma = \frac{Tr}{J} = \frac{\left(60\ \text{ft}\cdot\text{lbs} \times \frac{12\ \text{IN}}{1\ \text{FT}}\right)(0.375\ \text{IN})}{\frac{\pi}{32}(D^4 - d^4)}$$

$$J = \frac{\pi}{32}(D^4 - d^4)$$

$$r = 0.375\ \text{IN}$$

$$d = \sqrt[4]{\frac{Tr}{\sigma_y\left(\frac{\pi}{32}\right)D^4}} = \sqrt[4]{\frac{(720\ \text{lb}\cdot\text{IN})(0.375\ \text{IN})}{(64\ \text{ksi})\left(\frac{\pi}{32}\right)(0.750^4)}} = \boxed{0.607\ \text{IN or } t = 0.072}$$

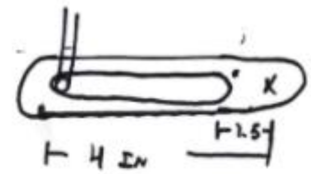
(C) Stiffness of Anti-roll bar

GIVEN: $\varnothing = 0.375$ IN

MOMENT ARM RANGES FROM 1.5 – 4 IN

$L = 21.75$ IN $G = 11.3$ KSI

DRAW



FIND: STIFFNESS RANGE

SOLVE:

$$K = \frac{JG}{r^2 L}$$

$$J = \frac{\pi}{32} r^4$$

$$K_s = \frac{\left(\frac{\pi}{32}\left(\frac{0.375}{2}\right)^4\right)(11{,}300 \text{ PSI})}{(1.5^2)(21.75 \text{ IN})} = 449. \ \frac{\text{lbs}}{\text{IN}}$$

$$K_n = 63.0 \ \frac{\text{lbs}}{\text{IN}}$$

$$\boxed{\text{Range from } 63 \frac{\text{lbs}}{\text{IN}} - 449 \frac{\text{lbs}}{\text{IN}}}$$

$$21.75^2 \ 63 \frac{\text{lbs}}{\text{IN}} \cdot \frac{\pi}{180} = \boxed{520 \frac{\text{lb} \cdot \text{IN}}{\text{Deg}} - 3700 \frac{\text{lb} \cdot \text{IN}}{\text{Deg}}}$$

$$\boxed{43 \frac{\text{lb} \cdot \text{FT}}{\text{Deg}} - 308 \frac{\text{lb} \cdot \text{FT}}{\text{Deg}}}$$

Appendix II: Tables, Plots and figures

Table 2: 2022 Suspension Team Design goals, as well as previous teams design goals

Suspension Characteristics	UOM (Unit of Measurement)	Who is responsible ?	2022 Initial Goals (Us)	2021	2020
Suspension Travel	in	Noah	2in wheel travel 1.25in suspension	1.25	1.25
Ride Height	in	Noah	1.75in	1.75	1.75
Turning Radius	in	Justin (Steering)	125	125	125
Weight Distribution	% of total wei	Noah	45/55	22/19F 29/30R	22/19F 29/30R
Track Width	in	Noah/Alex	49.5F 47R	49.5	47
Wheel Base	in	Noah/Alex	61	60.375/60.625 Unknown Why two	60.375/60.625 Unknown Why two
Spring Constant	lb/in	Noah/Yomaicol	158 front 307 rear		
Damping Coefficient	lb*s/in	Noah/Yomaicol	18 front and rear		
Camber	°	Noah/Alex	-5	-3.5/-4/-6.2/-6	-3.5/-4/-6.2/-6
Caster	°	Noah/Mike/Alex	about 3 deg	?	?
Toe	°	Noah/Alex	3 (front toe out) 2 (rear toe in)	1/0.8/1.4/-1.6	1/0.8/1.4/-1.6
Ackerman	%	Justin (Steering)	Implenment	100	100
Anti-dive/Anti-squat	%	Noah	4% anti dive	4 Anti-Dive	4 Anti-Dive

Appendix III: Bill of Materials (BOM)

(A) Steering BOM

Product	Vendor	Part #	Price	QTY	Description		Prices
Steering Wheel	Pegasus Auto	3406-Blac	$174.99	1	Steering Wheel		$174.99
Steering wheel Quick connect + spline	Speedway Mtrs		$89.99	1	New quick connect		$89.99
Steering Column Tubing 0.750 OD	McMaster Carr	7767T32	$6.43	1	Welded attachments to column		$6.43
Spline Attachments Steering Column	KAZ Tech	-	$11.00	1	Splined Inserts (3/4"-20)		$11.00
U Joints	KAZ Tech	-	$103.50	2	U joints		$207.00
Steering Rack	KAZ Tech	-	$670.00	1	Steering Rack Previoulsy owned		$670.00
Steering Coupler	KAZ Tech	-	$35.00	1	New coupler to connect rack to column		$35.00
Column Mount Bearins	McMaster Carr	5905K26	$6.94	2	Used in the frame to secure the column		$13.88
Spline Shaft for U joints	?	?	?	1			
0.5"-0.035" 4130 Chromoly Round Hollow Tube	Stock Car Steel	41	$7.20	2	Tie rod Raw Material	$0.00	$14.40
10-24 1.125" Shoulder Bolt	McMaster Carr	91259A509	$3.39	2	Misc Hardware for Tie Rods		$3.39
10-24 0.875" Shoulder Bolt	McMaster Carr	91259A541	$3.08	2	Misc Hardware for Tie Rods		$3.08
M6 25mm Shoulder bolt	McMaster Carr	92981A204	$1.62	8	Misc Hardware for Tie Rods		$1.62
Right Hand Ball Joint Rod End	McMaster Carr	60645K821	$4.81	2	For tie Rod ends		$4.81
Left Hand Ball Joint Rod End	McMaster Carr	60645K822	$4.81	2	For tie Rod ends		$4.81

(B) Anti Roll Bar BOM

Product	Vendor	Part #	Price	QTY	Description		Prices
Right Hand Ball Joint Rod End	McMaster Carr	60645K821	$4.81	4	For connecter to rockers		$4.81
Left Hand Ball Joint Rod End	McMaster Carr	60645K822	$4.81	4	For connecter to rockers		$4.81
Set Screw	McMaster Carr	94355A229	$6.40	1 (Pack of 25)	for shaft connection		$6.40
ARB Rods	McMaster Carr	8920K135	8.36	1	Arb Rods (0.375 OD Steel)		8.36
ARB Connecters	McMaster Carr	9008K14	17.04	1	Connecter 4x 6061		17.04