SAE Mini BAJA: Suspension and Steering

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Team 11

Project Proposal

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Introduction

Every year, the Northern Arizona University's Society of Automotive Engineers supports individual teams to compete in SAE national events. This year, they are supporting a team to compete in the mini baja competition. Dr. John Tester is the current advisor to NAU’s SAE branch and will be our teams client for the project. Our teams goal this year will be to design and fabricate a lightweight and reliable suspension and steering system for the car.

Customer Needs and Goals

After speaking with our client, Dr. John Tester, he has developed a list of needs pertaining to the suspension and steering systems for the new SAE Baja vehicle. The previous Baja team did well last year, but there are many components and systems on the vehicle that could be improved.

The first need that Dr. Tester has stated, is that the approach angle of the current Baja vehicle is too small. A vehicles approach angle is the angle of the line drawn from the leading part of the vehicle to where the front tire meets the ground. At the competition, we know we will encounter large boulders and rocks driving through the course. Our job is to make sure we have a large enough approach angle to not high-center on any obstacles during competition. To address this need, our goal is to increase the approach angle of the vehicle.

Another need that was stated, is the lack of suspension mounts integrated into the current frame design. The existing Baja vehicle has extra bars welded to the frame specifically to mount the suspension system to the vehicle. Extra bars can add unwanted weight and provide no extra strength to the frame. To address this need, our goal is to work alongside the frame team to design suspension mounts into the new frame design. We hope, that by accomplishing this goal, we can minimize the overall weight of the vehicle and only have members that keep the frame and suspension structurally sound.

On the current Baja vehicle, the turning radius provided problems during the competition last year. Because of this, another need for the vehicle is that the overall turning radius is too large. The previous Baja team encountered multiple tight radius corners that required shifting the vehicle into reverse to get the right angle to make the corner. To resolve this need, our goal is to design a steering rack with an increased turning radius for improved maneuverability, mainly around sharp corners.

One of the biggest complaints about last years vehicle was the vehicle weight. Because of this, another need for the vehicle is that the suspension and steering components weight too much. The previous teams vehicle weighed 650 lbs. As an entire group, we would like to reduce weight drastically because reducing weight improves many key characteristics of the vehicle. To address this need, our goal is to design a suspension and steering system that is not only minimized in weight, but still provides high strength. We plan to reduce weight by using lighter materials for suspension components, and possibly designing a totally different suspension system requiring less suspension components.

Another need is that the track width of the existing vehicle is too wide. A specific trailer was rented in previous years in order to transport the vehicle to competition. This can become problematic, because money that could be spent on valuable vehicle components and testing was instead spent on transporting the vehicle to the competition. Due to this need, our goal is to design a suspension system that is minimized in track width. By achieving this goal, we will be able to fit the vehicle in the back of a pickup truck or trailer and not need to spend money renting a vehicle for transportation.

Finally, the last need from our client is that some of the suspension and steering components on the previous vehicle where not designed by our engineering team. In last years competition, the Baja team lost valuable points because some of the components were not designed but bought from a off-road vehicle manufacturer. To address this need, our goal is to design, build, and test suspension and steering components that were purchased by the previous Baja team. We hope that we can score higher in the competition provided we design all suspension and steering components.

Objectives

To quantify how we will measure the goals for our project, we have developed some objectives to accomplish before competition. All the objectives listed relate to a goal stated in the previous section. The table below outlines our objective, how we will measure that objective, and the type of measurement system we plan on using to quantify our objectives.

Objective	Measurement Basis	Units
Width	Track width of Mini Baja	in
Weight	Weight of all steering and suspension components	lb
Maneuverability	Turning radius	in
Approach Angle	Ride Height	in
Reliability	Repetition of suspension and steering components	reps

Figure 1- Objectives Table

QFD and HOQ

The purpose of this section is to ascertain what aspects of the design deserve the most attention. From the previous years car, we have learned that weight, maneuverability, and size are things that can be improved on. In the figures below, the clients needs are related to each other and the design requirements to give a more concrete understanding of what is required.

			Engineering Requirements					
		Weight	Yield Strength	Turning Radius	Ground Clearence	Suspension Travel	Weight	Cost
Customer Needs	Durability	9	9		3	3		
	Weight	10					9	3
	Cost Sensitive	7					3	3
	Safety	10	9		3	3		
	Manueverable	8		9	3	3		
		Score	18	9	9	9	12	6
		Weighted %	29	14	14	14	19	10

Figure 2- QFD

The QFD shows that strength and weight reduction should be the largest factors in producing a successful design. Cost is low in this calculation, because we are not working with a fixed budget and we have the ability to raise more money for parts.

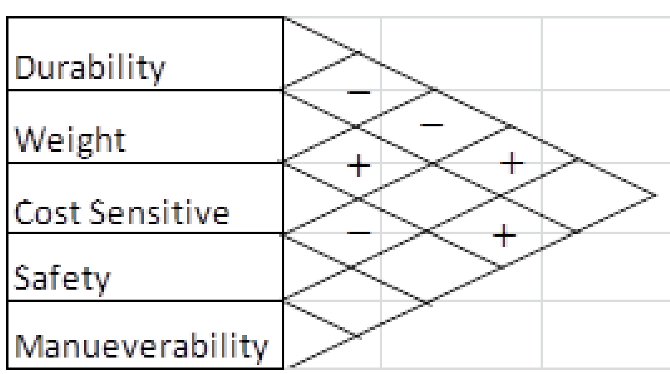


Figure 3- HOQ (+ is a positive correlation, - is a negative correlation)

Figure 2 gives a great example that the top two aspects picked in the QFD have a negative correlation. That is, that we cannot increase the strength of the design without sacrificing on weight. The design will then have to find a good medium of weight savings and strength.

Constraints

To fully satisfy the customer and participate legally in the SAE Mini Baja competition, there are a few constraints. Many constraints are from the customer, who would like to see an improvement from the previous vehicle. The one rule from SAE states that the track width can be no longer than 64”. However, the client has overridden this constraint saying the track width of the vehicle must be 59’’ or less. The previous car had a very long track width making it harder to turn. The customer would like the vehicle to be able to make a U-turn on a standard two lane road. All suspension mounts must be integrated into the frame with no extra bars for suspension only. The weight must be less than 450 lbs which is quite a significant decrease from the previous car weighing in at 650 lbs.

Testing Environment

Since the competition environment will be in Portland Oregon, the testing environment should be comparable to the region. This will be late spring so there should be an average amount of precipitation, and weather should be in the 50-70 degree fahrenheit range. Since the course is constructed only a few days before the competition, it is impossible to design a vehicle solely around the terrain. However, we can make educated guesses on what we expect to see on the course in Portland. We can assume that we will be spending the majority of the time in dirt, or mud if rain is in the forecast. We can also expect rugged terrain and many rocks. Taking this into account, we will test the vehicle out in the forest where there is ample rocks and rough terrain. Testing will also occur on fire roads as there should be many similar roads in the competition. Accelerometers will be used to determine if the suspension is adequate for the competition. Aside from testing outdoors, Finite element analysis on Solidworks and other testing software will be used.

Conclusion

This report details the project problem formulation and planning. The client needs were used to identify the problem with last years vehicle. The problem was divided into goals, objectives and constraints. A QFD and HOQ are used to identify and correlate constraints and objectives with others to allow for a better designed product. Only after these tasks are completed a Gantt chart is used to plan the project from beginning to end. The chart begins September, 14 to May, 27 which is the day of competition.

Concept Generation:

Throughout this section, the front suspension, rear suspension, and steering concepts will be analysed. In the front suspension, the concepts are Double A Arm, MacPherson, Torsion Bars, and Extended A-Arms. In the previous year, the team chose Double A-Arms. For the Rear suspension, the concepts are Double A-Arms, 2 link, and 3 link. In the previous year, the team chose 3-Link. For the steering, the concepts are back mounted rack and pinion, front mounted rack and pinion, and power assist. The previous team chose back mounted rack and pinion. The gear ratios will also be changed to 4-1 from 2-1 to make it easier to drive. All these concepts will work and will be put through a decision matrix to see what two concepts from each section will work the best

Front Suspension

Concept 1: Double A Arm

The double A arm suspension design is a proven concept across multiple platforms in all areas racing and conventional design. The reason for this is that the setup can be easily tuned and adjusted for camber, caster, and toe angles of the wheel. Also, by having multiple members and mounting points, the design ends up being very durable and resistant to impact on the wheels. An example of a traditional double A arm suspension design can be seen below.

This design keeps the suspension members away from potential contact from obstacles because it is mounted on the sides of the vehicle and away from the underneath. The analysis of this design will be more complex due to the multiple mounting points. It also runs the risk of being heavier than other designs. However, since the current design is the same, as long as stress calculations are done correctly the design will end up being lighter.

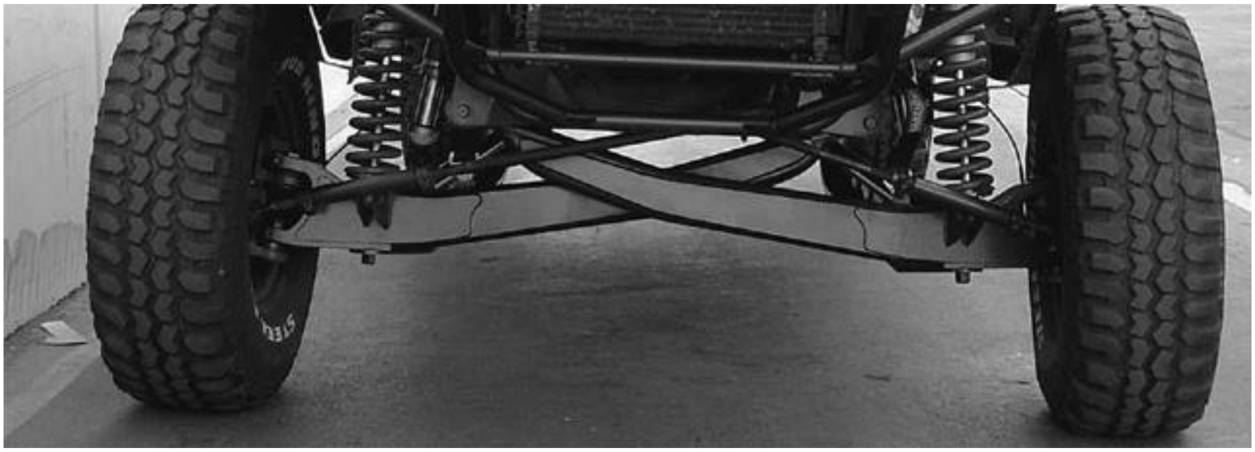
Concept 2: MacPherson Struts

This suspension setup was chosen in an attempt to reduce weight in the front of the car. While it is not very commonly used, it is favorable for lighter vehicles. This design only requires one lower A arm, because the strut is hard mounted to the top of the hub. A depiction of this suspension design can be seen below.

This design is less adjustable than the previous because of the way the strut needs to be mounted. It also puts significantly higher stresses on the strut and lower member which will require them to be either larger or very well designed. This design is also out of the way of potential impacts by obstacles. The stress analysis would be simplified due to only having two members.

Concept 3: I-Beam Suspension

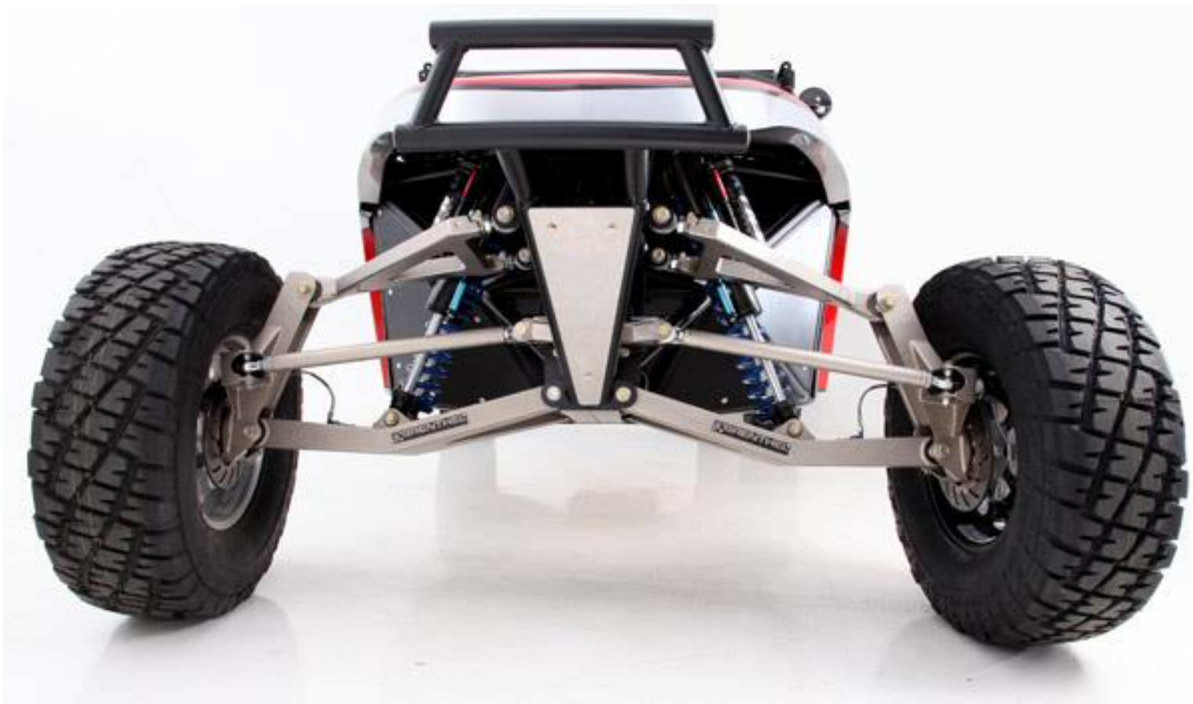
This design is more prominent with heavy vehicles that experience rough terrain and a high amount of suspension travel. The design is meant to be very durable to impacts and forces experienced during high amounts of travel. The setup can be repurposed for our vehicle by shrinking the members and engineering their geometry to match the shocks we specify. An example of this style of suspension can be seen below.



A major problem with this design is its lack of adjustment after it has been designed and installed. This will require a significant amount of forward thinking in the design process to remedy. Another issue is that even with proper analysis and design, the sheer size of the members will increase the weight of the vehicle. Also, because the members run under the vehicle, the ground clearance will be reduced.

Concept 4: Extended A Arms

This design is a modification of the original double A arms. It requires a reduction of the front section of the frame in order to lengthen the A arm members. The extended length will increase the amount of travel that can be seen in the front suspension. This increase in travel does come with a penalty in weight gain due to the extended length of the members. An example of this style of suspension can be seen below.



The only negatives to this design as opposed to the original double A arm setup is an increase in weight and a decrease in durability. The reduction in durability comes from the increase in lower member length. If the lower member was to impact an obstacle it would experience a significantly higher bending stress.

Front Suspension Design Analysis

In order to analyze the designs to more effectively choose which designs to carry forward in the design process, a decision matrix has been implemented. As seen below, the decision matrix has the designs listed on top with chosen engineering requirements to the left. Each requirement is weighted out of one hundred and the design is ranked on a scale from one to five.

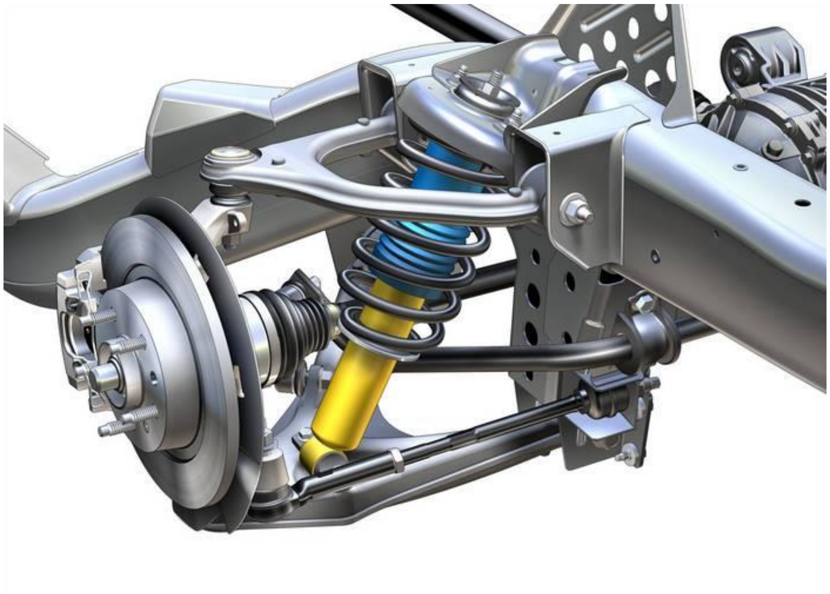
Front Suspension					
	Weight	McPherson	Double A-arms	Torsion Bars	Extended A arms
Cost	10	4	4	3	3
Weight	30	3	3	3	3
Strength	15	3	4	4	4
Ease of Machining	7.5	4	4	4	4
Ease of Design	7.5	3	5	3	3
Safety	2.5	4	4	4	4
Durability	10	3	4	4	4
Ground Clearance	10	3	3	3	4
Total Travel	7.5	3	3	3	3
Raw Total	100	30	34	31	32
Weighted		3.2	3.6	3.4	3.5

The raw total and weighted total can be seen at the bottom. The highest weighted engineering requirements are weight, strength, durability, and ground clearance. From this decision matrix, the highest scoring designs are the Double A Arms, and the Extended A Arms. Therefore, these are the designs that will be analyzed further in the design process.

Rear Suspension

Concept 1: Double A Arm

The double A Arm suspension, as described previously in the Front Suspension section, is a proven design across multiple platforms, from off-road to on-road use. The basic design of a double A arm suspension system consists of two A arms that provide a connection between the chassis and the hub of the vehicle. The two points at the base of the “A” connect the arms to the chassis and the tip of the “A” connects the arms to the hub. An example of a double A Arm rear suspension can be seen below.

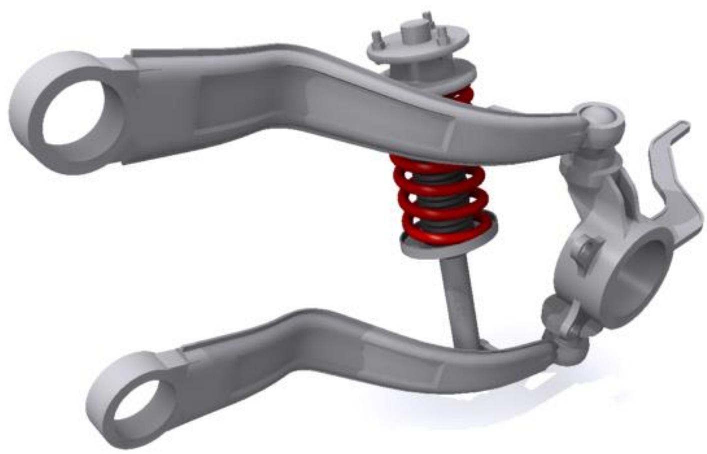


Some advantages of using this type of design include: versatility, amount of ground clearance, increased handling characteristics, and light weight. A double A arm suspension system can have a high versatility because of how well the suspension system can be adopted to all four corners of the vehicle. Once one corner of the suspension system is analyzed and engineered it can easily be replicated to the other three corners, because it can be assumed that the other corners of the vehicle will see the same impact and load. This type of design also provides a large amount of ground clearance. With this type of design, the A arms are mounted to the side of the frame, meaning there are no suspension members running underneath the frame. This means that suspension members will be higher off the ground compared to other suspension systems that would need members to run under the frame. Depending on how we design the A arms, we can provide increased handling characteristics utilizing this design. Increased handling can be accomplished by using a shorter upper A arm compared to the lower A arm. This increases handling because when entering a corner the suspension compresses and the wheel to the outside of the corner will produce negative camber, providing an increased contact patch between the tire and the ground. This type of suspension system can also provide a lightweight. Depending on which A arm we mount the shock strut to, we can lighten the other A arm by utilizing a lightweight material such as Aluminum to provide a slightly lighter weight compared to making all members out of a heavier material.

Some disadvantages to this type of design include: difficulty to produce, high cost, and space constraints. This design could potentially be difficult to produce because of the complexity of the members. The A arms could potentially be difficult to machine with the tools we have access to at the machine shop. If this design is chosen, it will be important make sure we have all tools needed to produce the A arms. This type of design also comes at a higher cost. The high cost can mainly be attributed to the fact that this type of system utilizes more material than other suspension systems. With more material being used, we will have an increase in weight. By utilizing this type of design in the rear, we could run into space constraints between the placement of the driveshaft and shock. The driveshaft will need to be mounted in the centerline of the rear suspension, so we will have to design the rear suspension taking this factor into account.

Concept 2: 2 Link

This type of suspension design utilizes two links to connect the suspension system to the frame of the vehicle, like the name implies. This type of design is very similar to the double A arm except that the suspension members connecting the frame to the hub are not in the shape of an A. One suspension link is connected to the frame and the top of the hub, while the other link is connected to the frame and the bottom of the hub. An example of a two link suspension is depicted below.

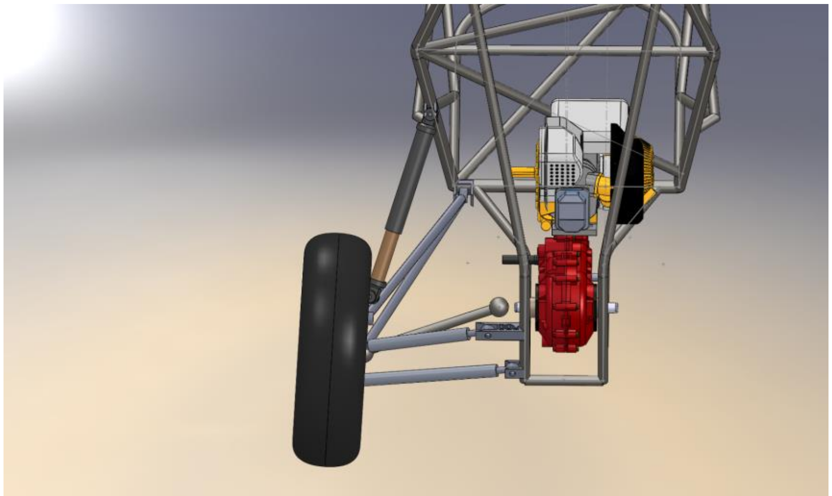


Some advantages to this type of design include: decrease in weight, and lower cost. This design can provide a decrease in weight of the entire suspension system because usually less material is used to make the members compared to other types of suspension. This type of design also provides a lower cost because the design is very simple and makes use of a small amount of material. Since less material is needed for the design, we won’t need to spend as much money on material for the members.

Some disadvantages with this type of design include: lower strength, decreased ground clearance, and decreased handling characteristics. This design makes use of only two members, usually in the form of bars, connecting the frame to the hub, and because of this, the strength of the system could be an issue. By utilizing this design, we also will have compromised ground clearance. One of the members would need to be mounted under the frame. By having a member under the frame, the chance of that member hitting a large boulder or rock is more likely, lowering the reliability of the suspension system. This design could also potentially decrease the handling characteristics of the vehicle. Because of the way the suspension is designed, the adjustment of camber, caster, and toe will be difficult to adjust once the suspension is mounted.

Concept 3: 3 Link

This type of suspension utilizes three links to connect the suspension system to the frame of the vehicle, like the name implies. Usually, a large suspension link runs from the middle of the frame to the hub, and the 2 other suspension links run from the rear of the frame to the hub. Of the two suspension links in the rear, one link mounts to the top of the hub, while the other link mounts to the bottom of the hub. This type of design is depicted in the figure below.



Some advantages to this design include: High strength, and reliability. A three link suspension can provide our team with a high strength system because this type of suspension has the most amount of members connecting the frame to the hub. With the use of three links, we can distribute the forces encountered by the wheel to three separate links, meaning that each link won’t see as high of forces as other designs. Yet another advantage to utilizing this type of design is the high reliability. A high reliability can be achieved because in this design there are more suspension links that distribute forces encountered at the wheel. By having more members we hope that the suspension will be more reliable to impacts from various objects during the competition.

Some disadvantages to this design include: difficulty of engineering analysis, increased weight, and increased cost. This design could be difficult to analyze because of the various points of placement of the members to the hub and frame. Have all these variables could increase the amount of analysis that will need to be done to utilize this design. This design would also increase weight of the suspension system. This design would increase weight because more members are needed to complete the design, compared to other suspension systems. Weight would also increase because one member needs to be fairly large to account for impacts from large boulders and rocks. Utilizing this design would also see an increase in cost. Since this design uses the most amount of suspension members when compared to others designs, this design will cost more because of the amount of material needed to complete the design.

Rear Suspension Design Analysis

To compare how the various rear suspension designs discussed will help or hurt our design goals for the vehicle, a decision matrix was created. Multiple design goals were compiled, and weighted with respect to how important they are to our design. Each goal is weighted out of one hundred and the design is ranked on a scale from one to five. The decision matrix for the rear suspension design can be seen below.

Rear Suspension				
	Weight	Double A arm	2 link	3 link
Cost	10	3	4	4
Weight	30	3	3	3
Strength	15	5	4	3
Ease of Machining	7.5	3	4	3
Ease of Design	7.5	5	3	3
Safety	2.5	5	5	5
Durability	10	5	4	4
Ground Clearance	10	5	3	3
Total Travel	7.5	4	4	4
Raw Total	100	38	34	32
Weighted		4	3.6	3.3

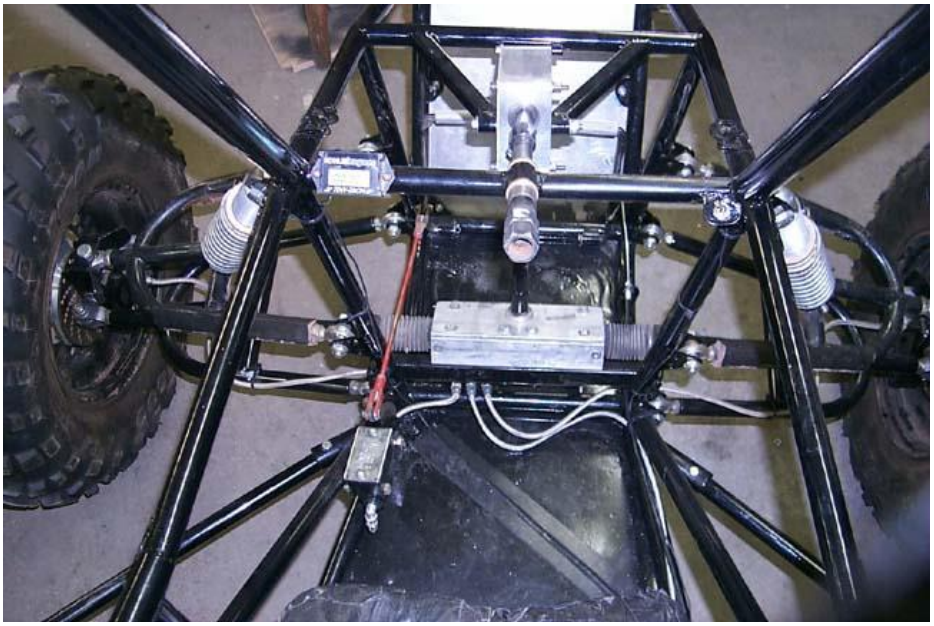
From this decision matrix, the highest scoring designs are the Double A Arms, and the 2 link design. The lowest scoring design was the 3 link, and will not be analyzed any further. Therefore, the designs that will be analyzed further in the design process will be the Double A Arms and 2 link design.

Steering

The designs that follow denote where, on the wheel hub, the tie rod end will be attached. The rack must be mounted on the same side of the wheel center as the tire rod end for the best possible performance. Because of this, the rack can either be mounted forward of the centerline or behind the center line.

Back Mounted Rack and Pinion:

The back mounted design uses attachment points on the back of the hub to mount the tie rod end. This design is often much more durable because the tie rod is shielded from debris, that may hit the front of the front of the vehicle, by the suspension components. A downside to this design is that there is less room for the drivers legs which could make it difficult for the driver to get in and out of the vehicle. There is also a possibility for the u-joints in the system to bind if not designed properly, which would lead to a vehicle that cannot turn.



Front Mounted Rack and Pinion:

The front mounted system is much more popular with other teams at competition because of the room that it gives the driver. Much needed space is cleared up when the rack is pushed as far out as possible. The driver could more easily get in and out of the vehicle. However a side effect of pushing the rack farther away from the driver is weight. More material is needed to attach the steering wheel to the rack thereby increasing the weight. Another disadvantage of this design is that the tie rod is exposed in front of the suspension components making it less durable.

Power Assist Steering:

A power assist system uses either electric power to run a pump or a pump mounted to the engine to run the fluid through the system to turn the wheels. This system can be tuned to driver comfort as well as response giving a much better handling vehicle. The major disadvantage of the system is the weight and power needed to run the pump. The pump would sap about half of our engines power which is only just enough to move the vehicle. Any loss in power to the wheels would decrease the competitiveness of our vehicle dramatically. The system would also increase weight by at least 100%.

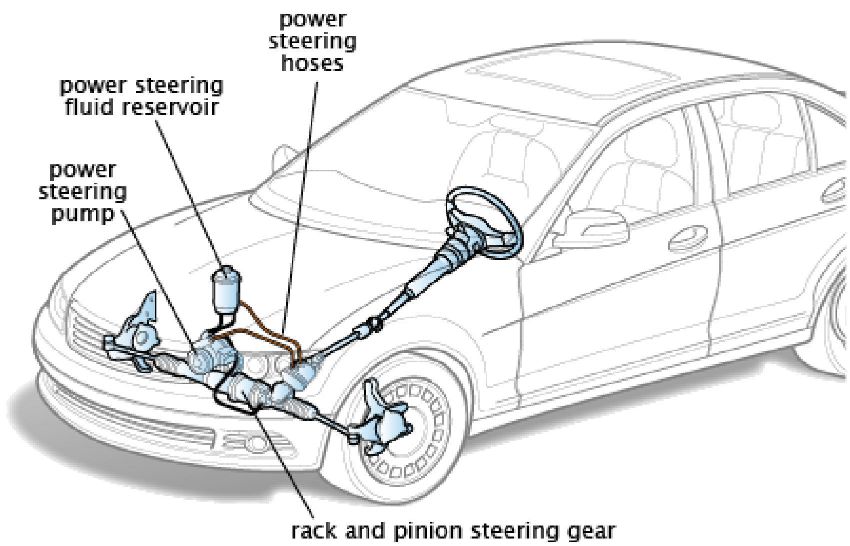


Image courtesy of ClearMechanic.com

Steering Design Analysis:

Steering				
	Weight	Back Mounted 4-1	Front Mounted 4-1	Power Steering
Cost	10	4	4	1
Weight	15	4	4	2
Strength	10	2	2	4
Ease of Machining	5	4	4	1
Ease of Design	5	5	5	1
Safety	5	4	4	4
Durability	10	5	3	4
Turning Radius	20	5	5	3
Ease of turning	15	4	4	3
Foot room	5	2	4	3
Raw Total	100	39	39	26
Weighted		4.1	4	2.7

The designs were put into a weighted matrix. The design with the highest score would be the best design for our goals. The Back Mounted design received the highest score. The Front Mounted design received a score just slightly smaller than Back Mounted. The Power Assist design received the lowest score and therefore will not be evaluated any further. The Back Mounted and Front Mounted will be re evaluated with the entire vehicles ergonomics in mind before a final design is chosen.

Concept generation conclusion

The final designs have been selected for the front suspension, rear suspension, and the steering. For the front, the Double A-Arms and Extended A-arms were selected. The team selected these because A-Arms worked great for the previous team and our current team can improve on the previous design. For the rear suspension, the double A-Arms and the 2-Link were chosen. The double A-Arms were chosen because they are a common rear suspension in the previous mini baja races. In addition, if Double A-Arms were selected in the front, less analyzation time will be needed to improve on the design; this is because the data from one wheel can be transferred to all wheels. For the steering, the back and front mounted rack and pinion were selected, both with a 4-1 gear ratio. All these designs have been carefully selected to improve on the suspension and steering systems of this Mini Baja. Further design and analysis will be conducted to see what component will be implemented on the final baja car.

Design Selection and Analysis:

The competition in which the car we are designing will participate, is known for being difficult to survive. Our designs must therefore be analyzed and tested greatly before being implemented in this rigorous competition. The chosen design for the front suspension is a simple true A style a-arm. The steering design chosen was a back mounted rack and pinion. This report will discuss in depth analysis of the chosen front suspension and steering system for the Baja vehicle.

Suspension Analysis

The first step towards analyzing the suspension was to create the geometry of the length of the members and the mounting position of the shock. The overall track width of the car is limited to 59 inches. Knowing this, the width of the front of the frame, and the distance from the outside of the tire to the mounting point of the hub, the length the A arms could span could be calculated. Next, the car needed to have a ride of at least one foot when the driver was inside. Using these X and Y values plus the specifications of last years shocks, final dimensions were established for the A arm members. From previous dynamic and force analysis, it was found that the closer the shock could be mounted to the hub, the less force the members would be subjected to. The arms were set at 17.40 inches with the shock mounted 15 inches from the chassis mounting point. This gave the car a 14-15 inch ride height with the driver with only an inch of compression from the shock. The members were then modeled in SolidWorks to confirm that there was no binding or other issues along the entirety of the suspensions travel. This model can be seen below.

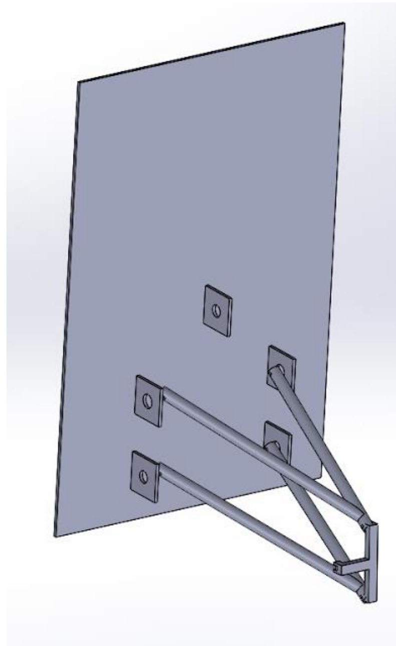


Figure 4: Dynamic motion assembly of A arms

After the geometry was decided for the A arms, hand calculations are needed to find the outer and inner diameters of the suspension members. To do this, analysis of a simple beam was done. The suspension system is comprised of two A arms but the hand calculations were done on one straight beam for a few reasons. The first reason is that most of the force seen on the suspension system will be on the member with the shock due to a bending force. Because of this, we are left to analyze one member. Second, due to the symmetric geometry of the A arm, we can treat the A shaped member as a simple beam and then half the resulting forces seen on the beam. A free body diagram is shown below to show the forces in the system.

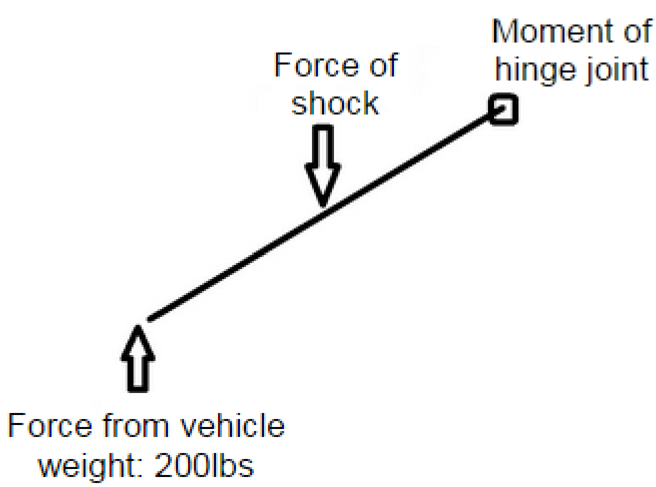


Figure 5: FBD of suspension system

Here a moment around the hinge joint is taken to get the forces in the x and y direction at the hinge joint and at the force of the shock. The results of this calculation is as follows:

Moment around hinge: Force of shock= **325.83lbf**
Sum of forces in Y direction: Force of hinge Y dir=**51.85lbf**
Sum of forces in X direction: Force of hinge X dir=**124.66lbf**

Because of the geometry of the beam, the forces can now be divided by two to represent the full A arm.

Force of shock= **162.92lbf**
Force of hinge Y dir=**25.93lbf**
Force of hinge X dir=**62.33lbf**

From these forces, shear and moment diagrams were composed. Below is the shear diagram.

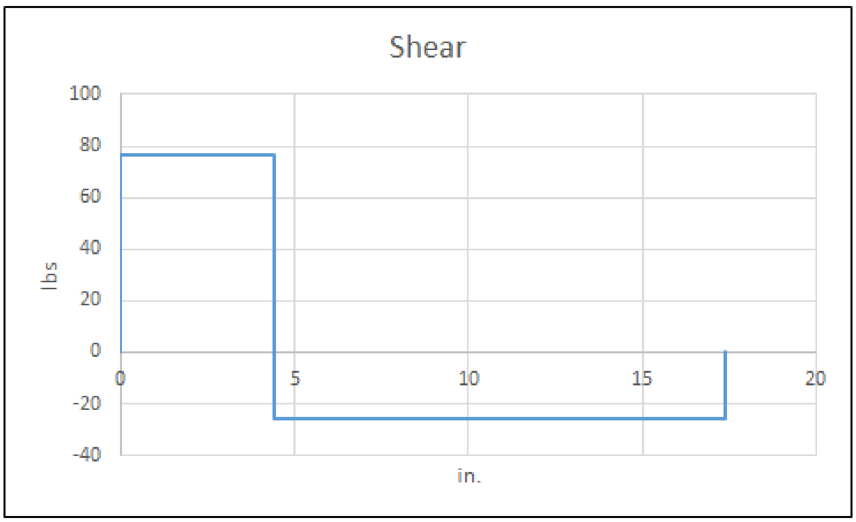


Figure 6: Shear Diagram

From this shear diagram, we can calculate the moment diagram which will give us our max moment.

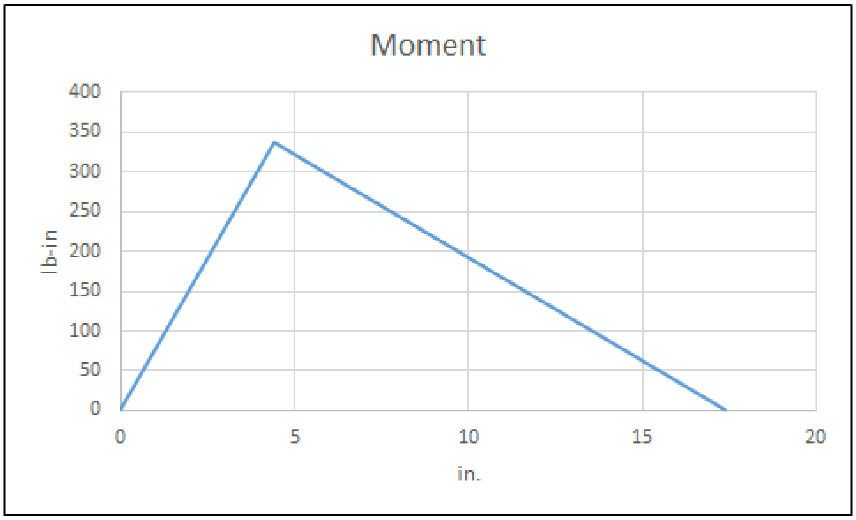


Figure 7: Moment Diagram

This moment diagram will give us our max moment. The max moment for this equation is 337.04lb-in. With this value, a max stress equation can be used for bending.

$$\sigma = \frac{Mc}{I}$$

Equation: (1)

This equation will output the outer and inner diameters for the suspension members. The diameters are imbedded in the variable “I” in the inertia equation above. This inertia equation is as follows:

$$I = \frac{\pi}{64}(D^4 - d^4)$$

Equation: (2)

This equation is for a hollow pipe with an inner and an outer diameter. An assumption was made to have the inner diameter be 80% of the outer diameter. Lastly, the “c” in equation (1) is half the outer diameter. For the stress, a value of 36 Ksi was chosen for A36 structural steel from a Mechanics of Material book. The calculated diameter values are D = .78in, and d = .624in for a factor of safety of 1. With a factor of safety fo 2, the values went to D = .98in and d = .78in. Stress analysis can now be performed to determine the final pipe material and dimensions.

Two scenarios were chosen to ascertain the correct properties of the A arms. Both situations are supposed to be extreme cases in order to guarantee the survivability of the suspension. The first scenario assumes the car weighs 650lbs with a driver and is moving at 25mph. The car then hits an obstacle, like a rock or tree, that brings it to a complete stop. The impact would take place solely on one front tire. Through energy and momentum calculations, the force on that side of the suspension came out to be 6200lbf. For ease of analysis, that force was divided across the two suspension members so that only one member would be analyzed. The desired factor of safety for both scenarios was 2. The material for the member was chosen initially to be 4130 steel. Through an iterative process of stress analysis and manipulating pipe diameters, a final size for the member was chosen. Below, the final stress analysis shown for this impact scenario.

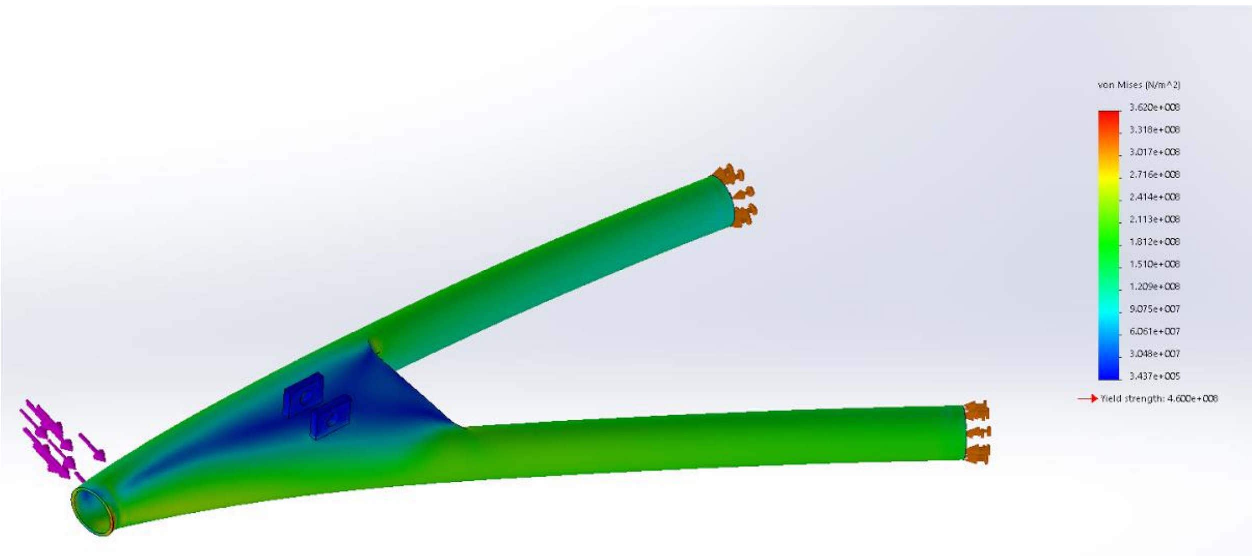


Figure 8: Front impact final FEA stress printout

In this analysis the load was applied to the end of the A arm. The member was fixed at the mounting points to the frame to prevent rotation. The small, high stress zones seen in the stress analysis can be omitted, because the force the member would experience would be more evenly distributed by a collar insert. The factor of safety for this analysis came out to be 2.1. This was with an outside diameter of 1.25in and an inside diameter of 1.15in. The second scenario assumes that the car and driver, weighing 650lbs, jumps of a ledge or obstacle and reaches a height of three feet. The car then lands on only one wheel. Through dynamic analysis of this impact using the shocks known spring rate, the magnitude of the force was calculated to be 3100lbf. This force is only applied to the member the shock mounts to, so that member is the only one that needs to be analyzed. The same material and dimensions that succeeded in the last analysis were used in this scenario to determine a factor of safety. The results from the stress analysis of the member can be seen below.

Figure 9: Vertical impact final FEA stress printout

In this analysis the load was also applied to the end of the member. The A arm was pinned at the chassis mounting locations as well as the shock mounting location to the simulate the suspension bottoming out. As with the previous stress analysis there was a high stress point in this analysis that can be omitted because of its location on the mounting tab. With the material and dimensions of the previous analysis, the factor of safety for this loading came out to be 3.2. With these factors of safety, the next step in the design process would be to chose a tubing size that is commonly manufactured. In order to maintain the the durability of the design and to increase the factor of safety, the next highest wall thickness for an outside diameter of 1.25in would be chosen. With the current dimensions, the expected weight loss from last years design would be 10-12lbs overall. This would be a significant reduction in unsprung weight and would allow for the suspension to respond better to inputs from the course.

Steering Analysis

To begin with the steering analysis, we needed to make a design choice on the type of steering design that would best suit our situation. After compiling different steering design ideas, and comparing them in a decision matrix, the top two steering designs were: front mounted, and back mounted steering. Front mounted steering means that the steering system is in front of the front axle centerline, where as back mounted steering is behind the front axle centerline. After looking at different factors, a back mounted steering design was chosen. One reason for going with a back mounted design was because of the minor modification to the front hub assembly. The hub on the last years baja vehicle has the steering mounted in the rear, and we plan on using the same hub in this years competition. The only modification that will need to be made is the mounting location of where the tie rod mounts to the front hub. Another reason for choosing a back mounted steering design is the space constraint of the front of the frame. The frame team is trying to reduce weight of the frame, therefore minimizing the front of the frame. Since the frame will be much smaller at the front, a back mounted steering system was chosen because frame member length would need to increase in order to fit a front mounted steering system. The final reason for choosing a back mounted steering was because mounting the tie rods on the front of the hub assembly would cause space issues. Since the brake caliper mounts on the front of the hub assembly, we would run into issues figuring out a place to mount the tie rod to the hub.

Once a back mounted steering system was chosen, analyzation of the steering components was started. The first type of steering analysis that took place was researching Ackerman steering angles. Ackerman steering design says that when cornering, the inside tire must turn at a greater angle than the outside tire. This is true, because the inside tire follows a smaller radius than the outside tire. A depiction of this scenario is shown in Figure 7.

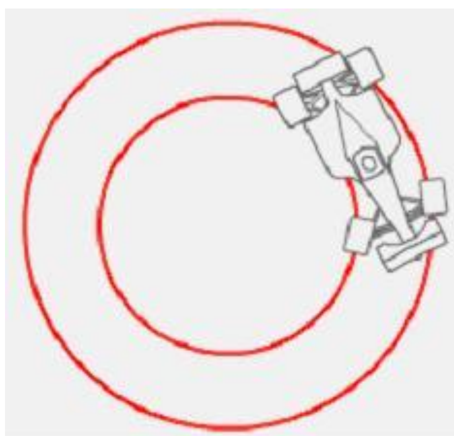


Figure 10: Depiction of Ackerman Angle Theory

One of the objectives of this years Baja vehicle was to achieve a 180 degree U-turn within the width of two lanes. In order to achieve this objective, the max steering angles of both the inside and outside tires needed to be found. The max steering angles of the inside and outside tire can be determined from Equation (3) and Equation (4), respectively.

$$\tan(\delta_i) = \frac{L}{R_1 - \frac{W}{2}}$$

Equation: (3)

$$\tan(\delta_o) = \frac{L}{R_1 - \frac{W}{2}}$$

Equation: (4)

All the variables in Equation (3) and Equation (4) are depicted in Figure 8, where L is the wheelbase of the vehicle, W is the track width of the vehicle, and R1 is the mid-radius between the inner radius and outer radius of the inside and outside tire respectively.

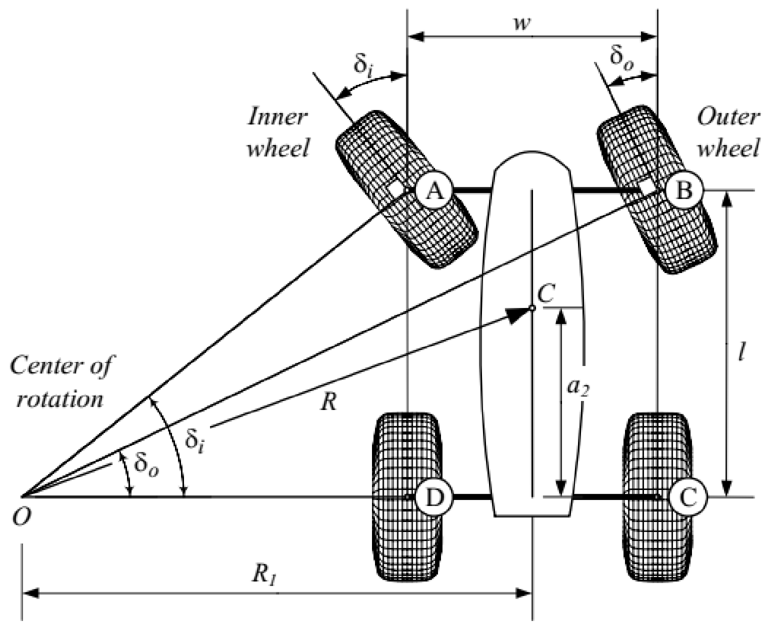


Figure 11: Front wheel steer vehicle, and the steer angles of the outer and inner wheel

Next, the variables needed for calculating the max inner and outer steering angles was determined. The track width,W, was determined to be 49 inches from the known width of the front tires and the constraint for vehicle width given to us by our client. The wheelbase, L, was determined to be 65 inches based on the Solidworks drawing of our suspension system and length of the frame. The last variable, R1, was determined to be 115.5 inches based on the known outside tire radius and inside tire radius. With all the variables in Equations (3) & (4) determined, the max inside and outside angles were calculated.

Inside Tire Max Turning Angle: 35.54 Degrees

Outside Tire Max Turning Angle: 24.90 Degrees

Turning Radius: 9.63 ft

Now that the max Ackerman Angles for the steering system where determined, the location of where the tie rod mounts to the hub needed to be determined. In order to achieve unequal angles on the inside and outside tire, where the tie rod mounts to the hub needs to be non-vertically aligned with the kingpin. The kingpin is where the suspension members mount to the hub, and is also the pivot point of the steering. With the tie rod mounts non-vertically aligned with the kingpin, the angles of the tires will differ from inner to outer wheel. A depiction of this setup is shown in Figure 9.

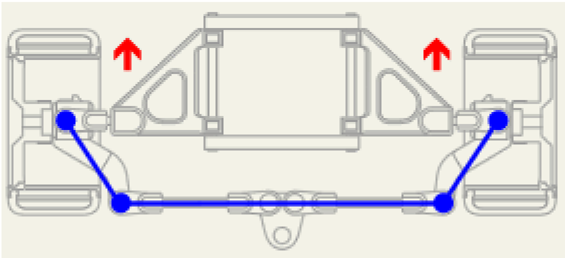


Figure 12: Tie rod mount Illustration

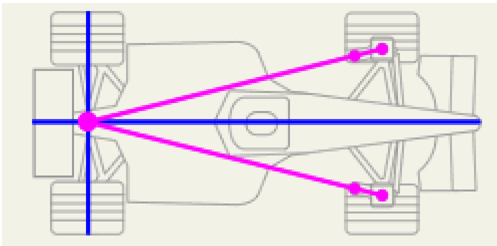


Figure 13: Tie rod mount illustration with neutral toe on turn in

In order to determine how far from the kingpin the tie rods need to be mounted, the depiction shown in Figure 10 is used to achieve neutral toe when the car enters a corner. Through the use of similar triangles, the exact distance from the kingpin to the desired tie rod mount location can be calculated. As shown in Figure 11, one triangle is known, with lengths of both sides measuring 24.5 inches and 65 inches. The other similar triangle has a known length of one side to be 3.45 inches. Once all these values are known, the distance from the kingpin to the tie rod location can be calculated.

Distance from Kingpin to Tie Rod Mount Location: 1.43 in

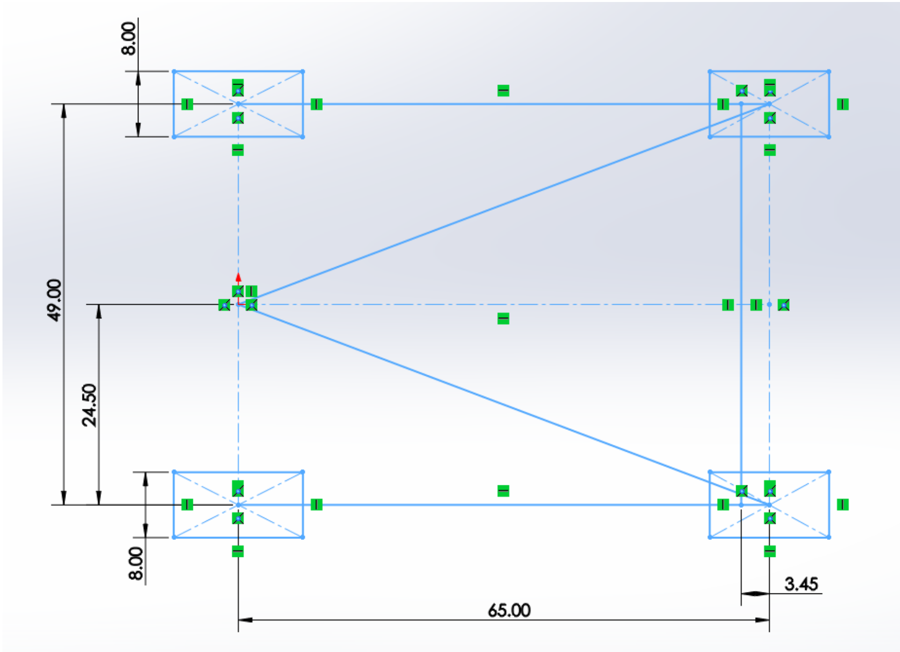


Figure 14: Variables used for tie rod mount location

Last years Baja vehicle lost points in the competition from the steering wheel being too difficult to turn. Because of this, analysis of the steering ratio needed to be accomplished to make sure we have a responsive steering system and also a steering system that takes less effort to turn the wheel. After researching why the steering system was so difficult to turn, we discovered that the steering system had a steering quickener, reducing the steering ratio. The original steering ratio of the existing rack and pinion is 12:1. With the steering quickener installed, the steering ratio was reduced to 6:1. After removing the steering quickener, and testing the response and difficulty of the steering, our team realized that a steering ratio of 12:1 would give the responsive steering, as well as a steering system that wasn't difficult to turn.

Rack and Pinion Ratio: 12:1

The final analysis that was done on the steering system was determining the outer diameter of the tie rods so that they would survive during SAE Baja competition. In order to start determining the tie rod diameter, the axial force the tie rod encounters during competition needs to be determined. Therefore, a scenario was assumed that would be a depiction of the worst situation the vehicle would encounter. This scenario included the vehicle traveling at max velocity, hitting an obstacle such as a boulder or tree, and coming to a sudden stop. In our situation, the vehicle is assumed to be traveling at 20 mph and, after hitting the obstacle, the vehicle goes from 20 mph to 0 mph in half a second. Figure 12 shows a free body diagram of the forces and moments encountered from the above scenario.

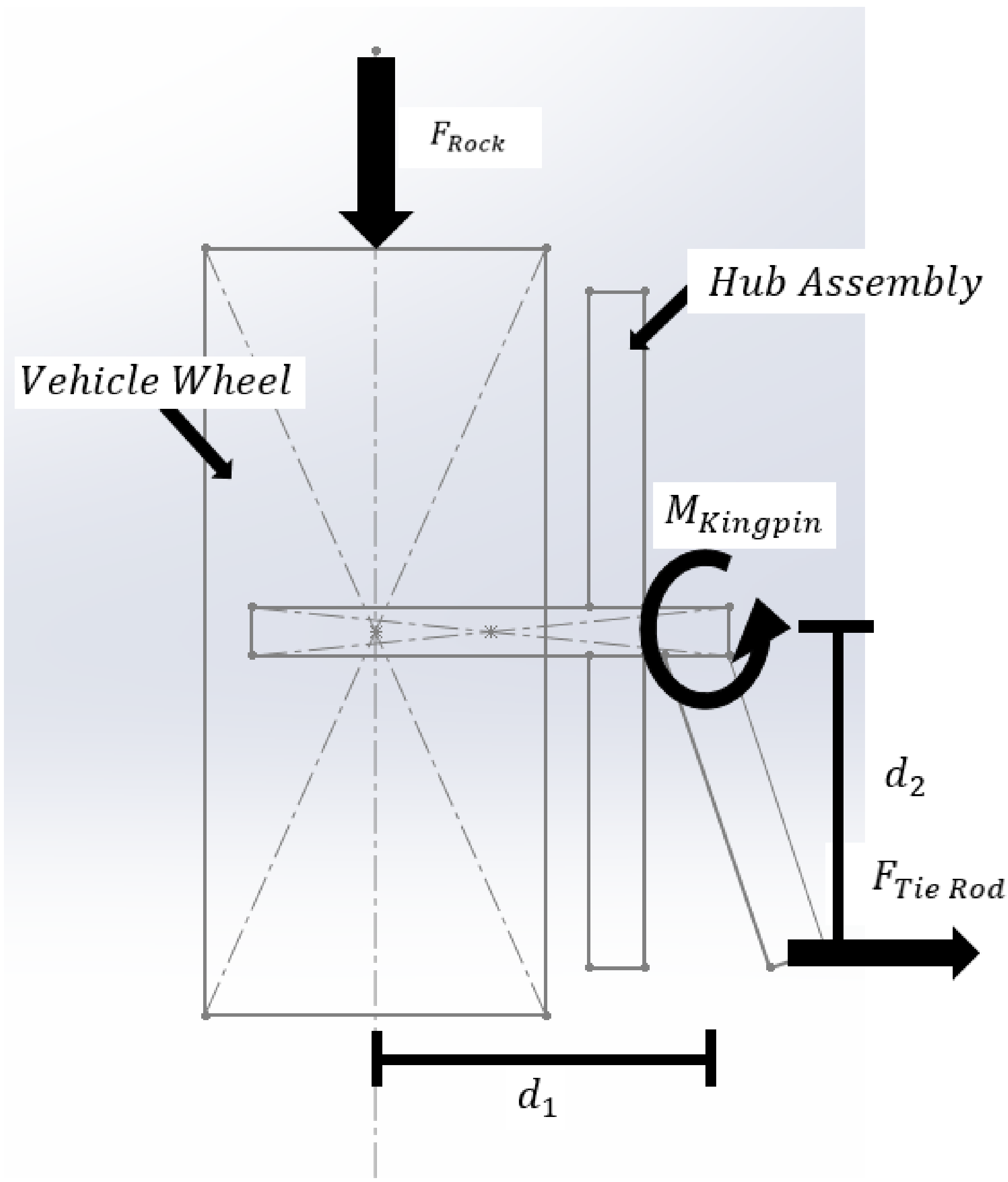


Figure 15: Free Body Diagram for analysis of tie rod

The first force that needs to be calculated is the force that the rock exerts on the wheel. This is accomplished by using Equation (5) assuming the mass of the vehicle is 272.4 kg and the declaration experienced is 17.88 m/s^2.

$$F_{Rock} = M_V * A$$

Equation: (5)

Once the force on the rock is calculated, the moment at the kingpin can be calculated using the force from the rock and knowing distance d1. After using Equation (5) the Force exerted by the rock on the tire is 4,870.51 N. We also know that the distance, d1, is 0.1016 m from hand measurements of the hub. With the variables known, we can calculate the moment experienced at the kingpin using Equation (6).

$$M_{Kingpin} = F_{Rock} * d_1$$

Equation: (6)

After plugging in the variables known into Equation (6), the moment experienced at the kingpin comes to 494.84 N*m. Now that this moment is known, and we know the distance, d2, is equal to 0.0876 m, we can determine the axial force that the tie rod experiences. With the variables known, we can calculate the axial force that the tie rod experiences using Equation (7).

$$F_{Tie\ Rod} = \frac{M_{Kingpin}}{d_2}$$

Equation: (7)

From the use of Equation (7), the axial force that the tie rod experiences was calculated to be 5,648.86 N. When this value is converted to English units, the axial force changes to 1,269.86 lbf.

Axial Force in Tie Rod: 1269.86 lbf

Now that we know the axial force that the tie rod experiences, we can find the outer diameter of the tie rod. One of the main modes of failure that tie rods can experience is from buckling. Because of this, the diameter of the rod can be determined using the buckling formula. Equation (8) shows the buckling formula used to determine the value of “I”, and from the value of “I” we can get diameter of the tie rod.

$$F_{Tie\ Rod} = \frac{\pi^2 EI}{(KL)^2}$$

Equation: (9)

In Equation (9) the variables that need to be determined include, modulus of elasticity (E), support factor (K), and length of the member (L). The modulus of elasticity was assumed to be (29*10^3) ksi because, the material chosen was A36 structural steel. The support factor was determined to be 1, because we know that the tie rod will have pinned supports at both ends. Finally, the length of the member was determined to be

15 inches from our solidworks assembly of the suspension and steering. With all variables known, the “I” value from the buckling calculation can be determined.

Second Moment of Inertia (I) = .001997 in^4

Finally, the diameter of the tie rod can be determined from the second moment of inertia equation. There are different formulas for the second moment of inertia depending on the cross-section of the part being analyzed. In our case, the formula for a circular cross-section was found. The second moment of inertia for a circular cross-section is shown in Equation (10).

$$I_x = \frac{\pi}{64} D^4$$

Equation: (10)

To get the final diameter of the tie rod, the second moment of inertia value calculated above was plugged into Equation (10). After solving with a safety factor of 2, our final diameter came out to .449 inches. Due to the fact that tie rods are known to be a weak point on these vehicles the final diameter chosen was .50 inches.

Tie Rod Diameter = .50 inches

Bolt Analysis

After the analyzation of the suspension components, we knew what kind of forces we would see in the A-arms. Based on a full speed collision with the impact of a solid object (rock, or tree) going into one A-arm, we calculated the max shearing force on the bolt to be about 5,000 lbf. Knowing the max shearing force the bolt would encounter, we could play with bolt diameter values to achieve a good safety factor and lightweight. We also assumed that the bolt would be experiencing double shear. Bolt diameter was chosen using the Equation (11).

$$\tau = \frac{F}{2A}$$

Equation (11)

We also assumed we would be using Grade 8 bolts as they are readily available, a popular grade, and is sold from various suppliers. Last years Baja vehicle used ⅜ inch diameter bolts, but we realized we could use a 5/16 inch bolt and still achieve a factor of safety of 2.76. Therefore, 5/16 in diameter Grade 8 bolts will be used to attach the suspension and steering components to the frame.

Bill of Materials

The designs there were chosen have a cost associated with them.Listed in our bill of materials is everything that our team would need to created a steering and suspension system. The last column account for all the materials that we already have or that we will get donated.

Table 3: Bill of Materials

Description	Qty.	Price Each	Total Cost	Free/Donated
Tubing 1.25"	168"	n/a	$100.00	x
Shocks	4	$270.00	$1,080.00	x
5/16-16 Bolt	50/pack	n/a	$61.00	x
5/16-16 Nut	50/pack	n/a	$20.00	x
Female Hyme Joint	2	$27.40	$54.80	
Male Hyme Joint	16	$27.40	$438.40	
Uniball	8	$15.00	$120.00	
Plate Stock 1x2'	1	$30.72	$30.72	x
Tie Rod Material 3'	1	$10.41	$10.41	x
Steering Rack	1	$98.00	$98.00	x
Steering Wheel	1	$49.99	$49.99	x
Front Wheel Hub	2	$128.00	$256.00	x
Rear Wheel Hub	2	$32.50	$65.00	x
Front Brakes	2	$25.50	$51.00	x
Rear Brakes	2	$25.50	$51.00	x
Wheels/Tires	5	$100.00	$500.00	
		Total	**$2,986.32**	**$1,113.20**

Conclusion

Static suspension analysis using simple beams showed that our members needed to be .98inch outer diameter and inner diameter of .78inch at minimum. From that point we used solidworks to calculate dynamic loading of the members. To achieve a factor of safety greater than two, the members diameters had to be increased to 1.25inch outer diameter and an inside diameter of 1.15inch. Even with these increased dimensions the weight savings over last year's vehicle will be 10-12lbs. The turning radius will be reduced using proper Ackerman Angles to 9.63 ft. The team decided to remove the steering quickener giving us a ratio of 12:1 which would make it easier for the driver to turn the wheels. A buckle analysis of the tie rod gave a solid diameter of .5 inches.The tie rod would then be able to withstand a front impact at full speed. We calculated a bolt of 5/16in diameter would be sufficient for our application and needs. These results are only preliminary and are likely to change as we further refine our designs of both suspension and steering. We plan on ordering material and parts over the break to be able to start building early next semester.

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Appendix A

Task Name	Duration	Start	Finish	Predeces
Start	0 days	Mon 9/15/14	Mon 9/15/14	
Research	**21 days**	**Mon 9/15/14**	**Mon 10/13/14**	
Pros/Cons of Previous Design	**10 days**	**Mon 9/15/14**	**Fri 9/26/14**	
Suspension	10 days	Mon 9/15/14	Fri 9/26/14	1
Steering	10 days	Mon 9/15/14	Fri 9/26/14	1
New Research	**11 days**	**Mon 9/29/14**	**Mon 10/13/14**	
Suspension	11 days	Mon 9/29/14	Mon 10/13/14	4
Steering	11 days	Mon 9/29/14	Mon 10/13/14	5
Design	**24 days**	**Tue 10/14/14**	**Fri 11/14/14**	
Suspension	10 days	Tue 10/14/14	Mon 10/27/14	7
Suspension Solid Model	6 days	Tue 10/28/14	Tue 11/4/14	10
Suspension FEA	8 days	Wed 11/5/14	Fri 11/14/14	11
Steering	10 days	Tue 10/14/14	Mon 10/27/14	8
Steering Solid Model	6 days	Tue 10/28/14	Tue 11/4/14	13
Steering FEA	8 days	Wed 11/5/14	Fri 11/14/14	14
Build	**15 days**	**Mon 11/17/1**	**Fri 12/5/14**	
Suspension	15 days	Mon 11/17/1	Fri 12/5/14	12
Steering	15 days	Mon 11/17/1	Fri 12/5/14	15
Sign up for competition	0 days	Tue 10/7/14	Tue 10/7/14