SAE Mini BAJA: Suspension and Steering

By
Zane Cross, Kyle Egan, Nick Garry, Trevor Hochhaus
Team 11

Design Selection and Analysis

Submitted towards partial fulfillment of the requirements for
Mechanical Engineering Design I – Fall 2014

Department of Mechanical Engineering
Northern Arizona University
Flagstaff, AZ 86011

Table of Contents

Introduction

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.

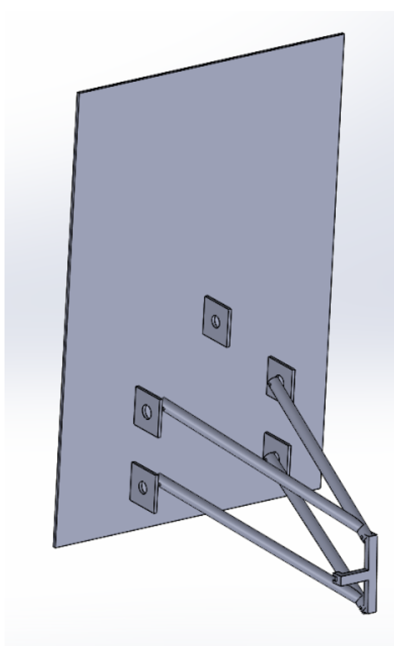

Figure 1: Dynamic motion assembly of A arms

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.

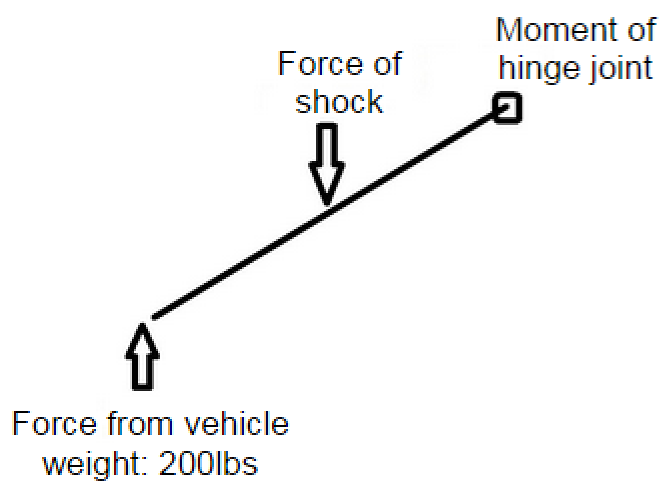

Figure 2: FBD of suspension system

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.

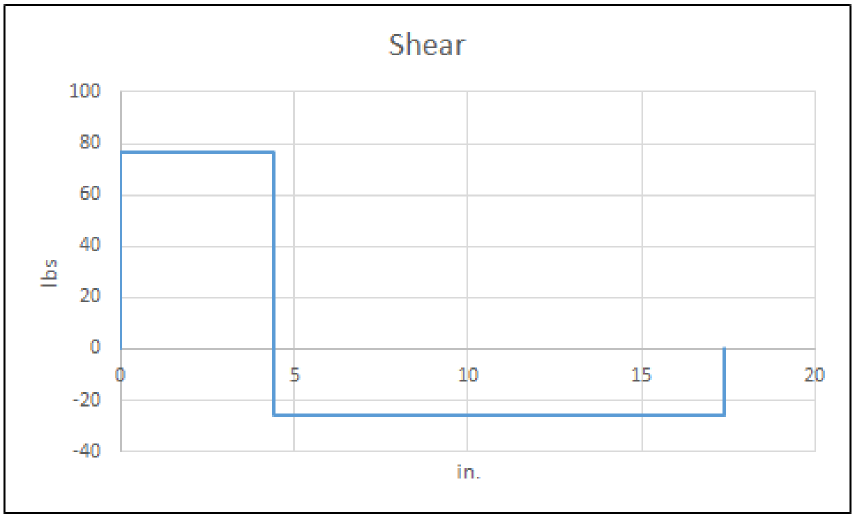

Figure 3: Shear Diagram

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

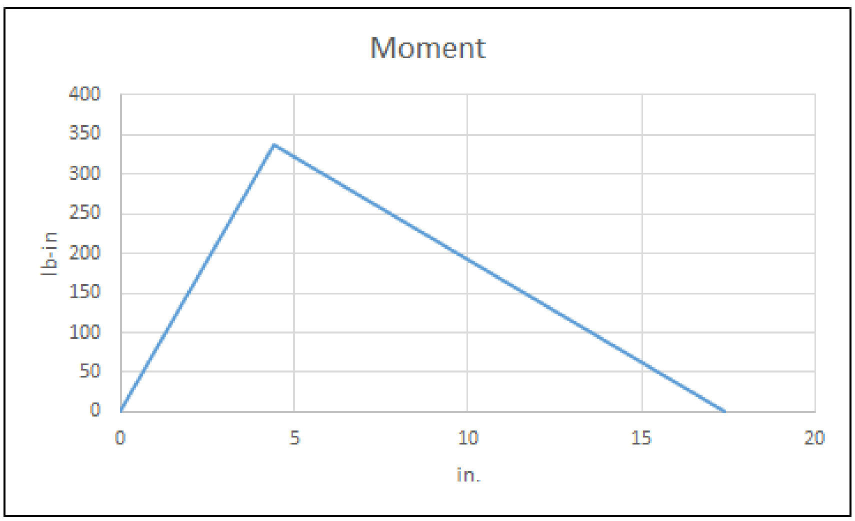

Figure 4: Moment Diagram

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.

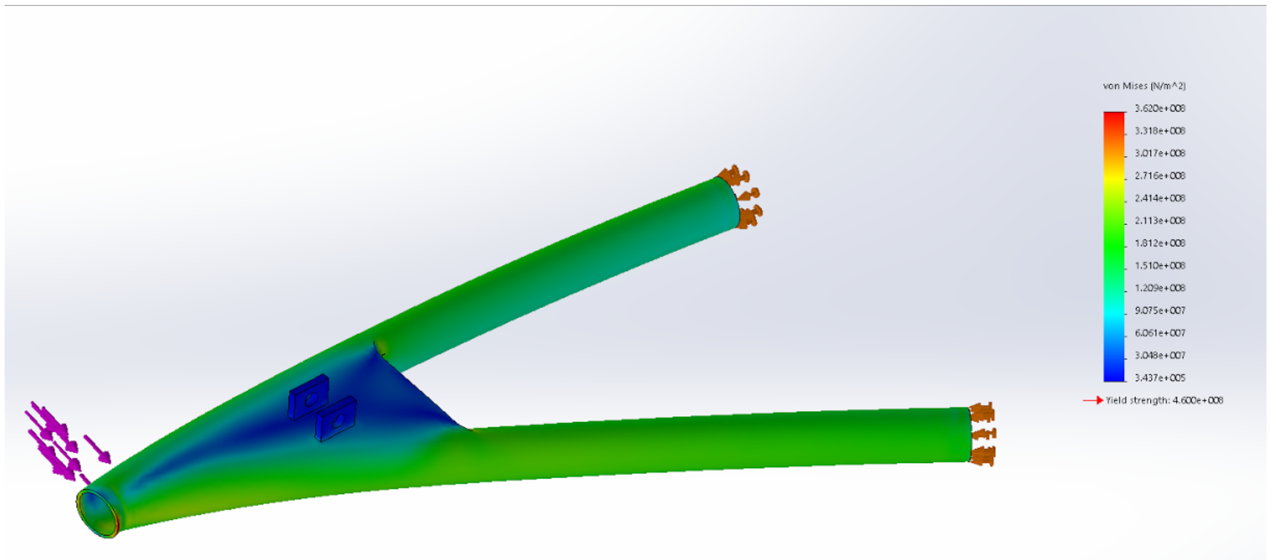

Figure 5: Front impact final FEA stress printout

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.

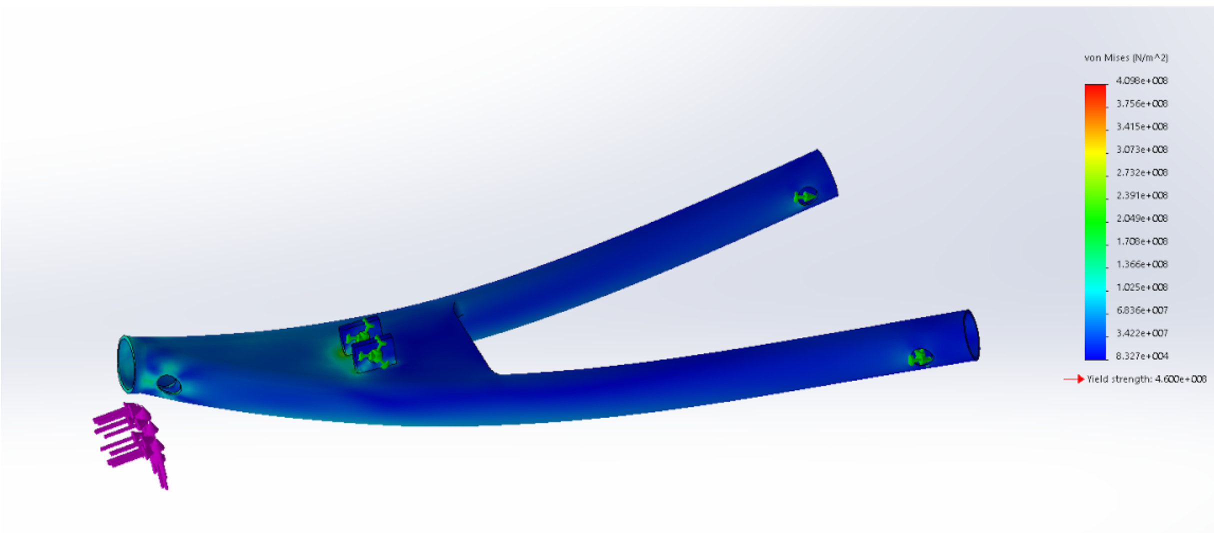

Figure 6: Vertical impact final FEA stress printout

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.

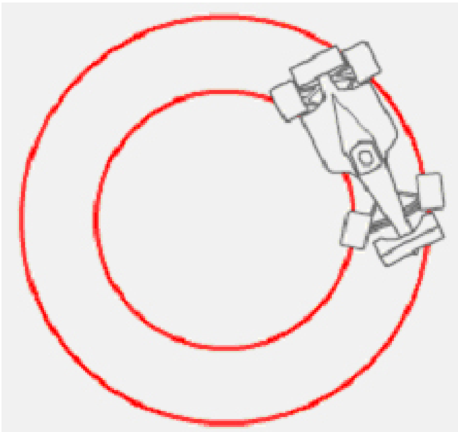

Figure 7: Depiction of Ackerman Angle Theory

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.

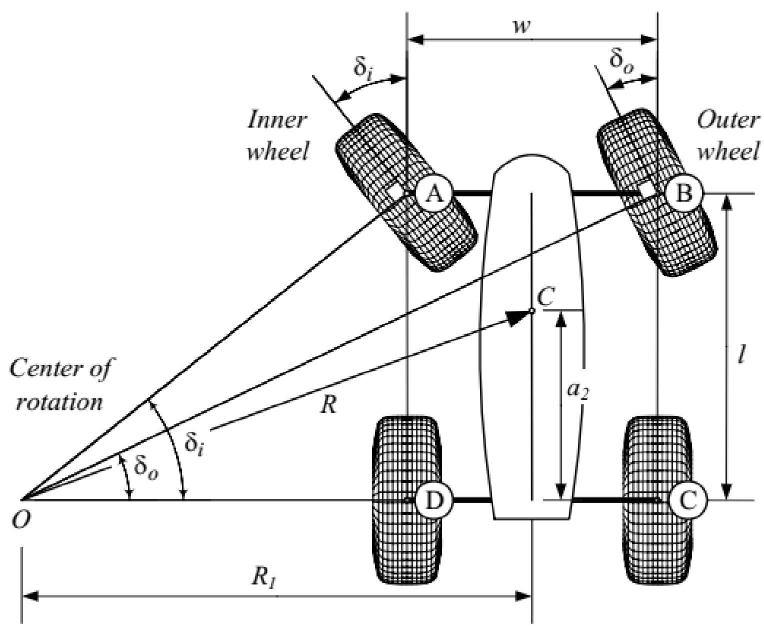

Figure 8: Front wheel steer vehicle, and the steer angles of the outer and inner wheel

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.

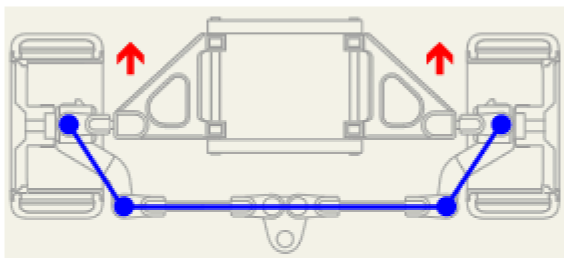

Figure 9: Tie rod mount Illustration

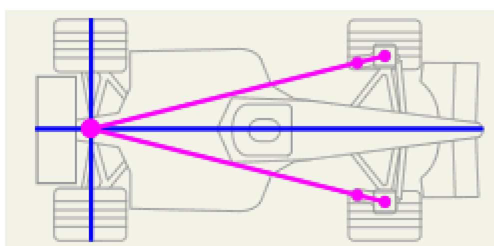

Figure 10: Tie rod mount illustration with neutral toe on turn in

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

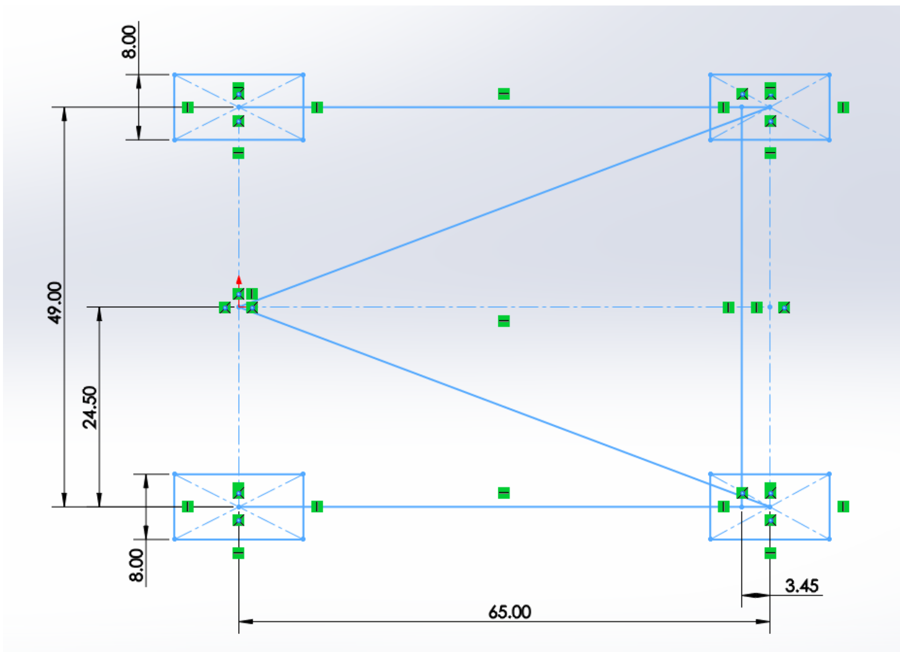

Figure 11: Variables used for tie rod mount location

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.

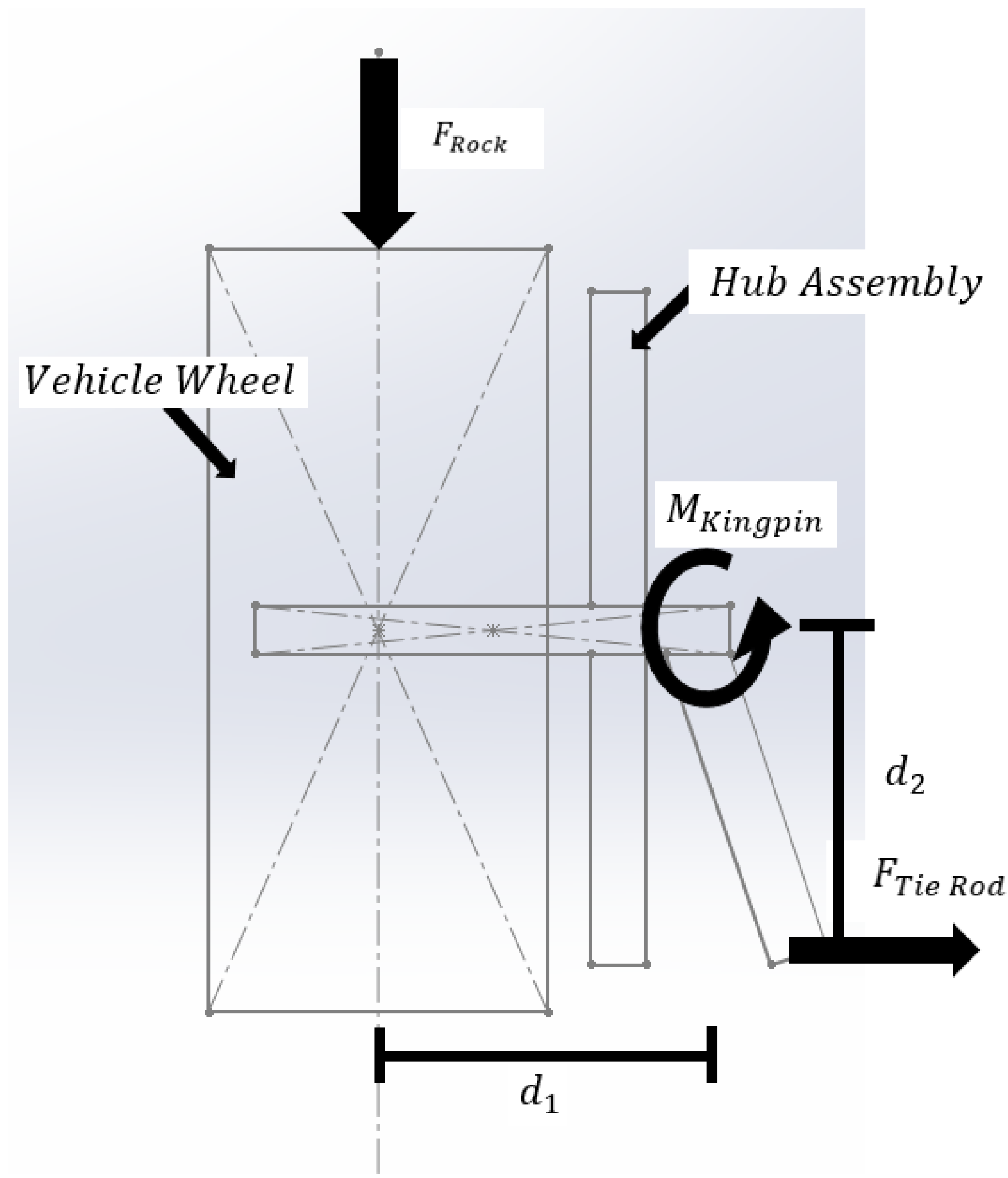

Figure 12: Free Body Diagram for analysis of tie rod

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

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. These results are only preliminary and are likely to change as we further refine our designs of both suspension and steering.

References:

- www.lostjeeps.com
- brenthelindustries.com
- www.ultimatecarpage.com
- tortoracer.blogspot.com
- www.mech.utah.edu
- www.cougar-racing.com
- http://forum.kerbalspaceprogram.com/threads/42596-Rover-Steering-to-use-Ackerman-Principle
- http://www.idsc.ethz.ch/Courses/vehicle_dynamics_and_design/11_0_0_Steering_Theroy.pdf
- http://www.rctek.com/technical/handling/ackerman_steering_principle.html
- www.desertkarts.com
- R. Hibbeler, Mechanics of Materials, Upper Saddle River : Pearson Prentice Hall, 2011.

- http://forum.kerbalspaceprogram.com/threads/42596-Rover-Steering-to-use-Ackerman-Principle
- http://www.idsc.ethz.ch/Courses/vehicle_dynamics_and_design/11_0_0_Steering_Theroy.pdf
- http://www.rctek.com/technical/handling/ackerman_steering_principle.html
- www.desertkarts.com

- http://www.engineeringtoolbox.com/euler-column-formula-d_1813.html