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ASHESI UNIVERSITY

DESIGN AND ANALYSIS OF SAE MINI BAJA SUSPENSION SYSTEM

CAPSTONE PROJECT

B.Sc. Mechanical Engineering

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CAPSTONE PROJECT

Capstone Project submitted to the Department of Engineering, Ashesi University in partial fulfilment of the requirements for the award of Bachelor of Science degree in Mechanical Engineering.

Romeo Nettey

2019

DECLARATION

I hereby declare that this capstone is the result of my own original work and that no part of it has been presented for another degree in this university or elsewhere.

Candidate's Signature:

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Candidate's Name:

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Date:

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I hereby declare that preparation and presentation of this capstone were supervised in accordance with the guidelines on supervision of capstone laid down by Ashesi University College.

Supervisor's Signature:

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Supervisor's Name:

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Date:

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Abstract

All-Terrain vehicle as the name implies refers to a car that has been designed to handle a more extensive variety of terrains than most other cars. ATVs are used for several purposes which include recreational purposes and others ranging from military to jungle safaris and desert rides. ATVs are rated highly based on their ability to sustain irregularities of the terrain with ease. The central system involved in supplying the damping characteristics of the vehicles is the suspension system. This paper discusses the development of the front and rear suspension systems for an off-road vehicle, mainly the mini Baja, which will be used to compete in the Baja SAE competitions.

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Chapter 1: Introduction

1.1 Introduction

An all-terrain Vehicle (ATV) is defined as a motorized off-highway vehicle designed to travel on four low-pressure or pneumatic tires, having a seat designed to be straddled by the operator and handlebars for steering control [1]. ATVs over the years have been designed to be a single seater but currently, one is able to find double seater all-terrain vehicles. All-terrain vehicles are made up of several systems that work together to cause the vehicle to function as required of it. These systems include the chassis system, steering system, suspension system, braking system, and drive train. These systems are all inter-dependent, which means that failure of one system affects the performance of the other systems that make up the vehicle. One of the critical systems that influences greatly the performance of the all-terrain vehicle is the suspension system. The suspension system mainly is made up of the spring and damper to perform the function of shock absorption. The main role of the suspension system is to support the weight of the vehicle and to provide comfort to the passenger [2].

The goal of this project is to design an off-road suspension system for Baja vehicles. This project is limited to**.** Physical models will not be built. Both rear and front suspension systems of the Baja vehicle will be considered for redesign and optimization.

1.2 Problem definition

Vehicles competing in the Baja SAE competitions have had suspension components failing during the course run. Some vehicles are not able to maneuver the course properly due to poor suspension geometry design. This renders the vehicles incapable of completing certain competition tasks.

1.3 Objectives

The objective of the design is to improve handling, maneuverability, traction, and bump absorption characteristics of the Baja. This will be done to meet the requirements of SAE. These objectives will be met by optimizing parameters such as camber, toe, caster, etc. that affect the dynamic performance of the suspension system.

1.4 Expected outcome

The target of the design is to have improved handling characteristics as well as improved maneuverability as compared to already existing Baja. The components of the suspension system should have a factor of safety not less than 2. Also, the suspension of the vehicle should allow better stability and shock absorption of all the vibrations it experiences from the terrain the vehicle drives on. Aside all that has been mentioned, it is also expected that the technical requirements and all other design related requirements given in the SAE Baja Rule book 2018 are met.

1.5 Motivation of project

The Mini Baja is an all-terrain vehicle designed to be used for recreational purposes. Baja competitors also need a solid design to guarantee their win. Major break downs that occur happen in the suspension system. This project aims to come up with an entirely reliable suspension design.

1.6 Research methodology

The approach to research concerning this project will be to search and source from pre-existing data about the competition and every area concerning the success of this project from the internet and other resources. Seeking knowledge from experts in the industry will also be a research method incorporated into ensuring objectives are met.

1.7 Facilities/materials to be used

The nature of the work is mainly CAD design and simulation. Some materials or facilities that will be used include:

- Ashesi online library resources
- SolidWorks
- MATLAB
- Simulink
- Lotus Engineering Suspension Analysis software
- VSusp online suspension software
- Baja SAE rules pdf
- Baja SAE evaluation criteria pdf

1.8 Scope of work

The project is limited to the design and simulation of the front and rear suspension system for a mini Baja.

Chapter 2: Literature Review

2.1 Review 1

Paper 1 - *Design and Optimization of Double wishbone suspension system for ATVs* [3]:

The purpose of this paper was to design and optimize a double wishbone system for an ATV as stated in the title of the paper. The objective of this paper is to optimize the front upper control A-arm by converting it into a single member to improve the suspension system performance to a certain extent. The results generated in this paper include camber angle, caster angle, roll steer, kingpin inclination, scrub radius and percent Ackerman. Software used include CATIA, which was used for modeling the components, ADAMS ANSYS, which was used for structural analysis to aid in finding out stress and deformation results of the parts.

Suspension system:

The paper focuses on a double wishbone suspension system. The paper explains double wishbones to be the most ideal suspension system, hence their reason for selecting it. Some other reasons that were mentioned to be reasons for selecting the double wishbones include the fact that it is an independent suspension type, and that it has near perfect camber control. The material chosen to be used for the member elements of the suspension system was AISI 4130 chromyl steel.

Suspension system analysis:

To perform their analysis on the designed wishbone, primary iterations were done to get three initial positions of the upper wishbone which was about the optimized. The three positions they came up with were:

- In center
- Front of spring
- Behind the spring

The center position was not selected because of the spring position. Their reason was that if they had to select that position, there would have to be a change in the spring position which will in turn cause a change in stiffness depending upon the angle of the spring mounting.

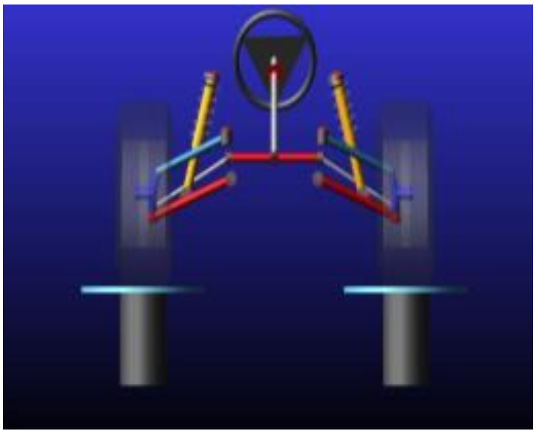

Figure 4. 1: Front view of suspension geometry

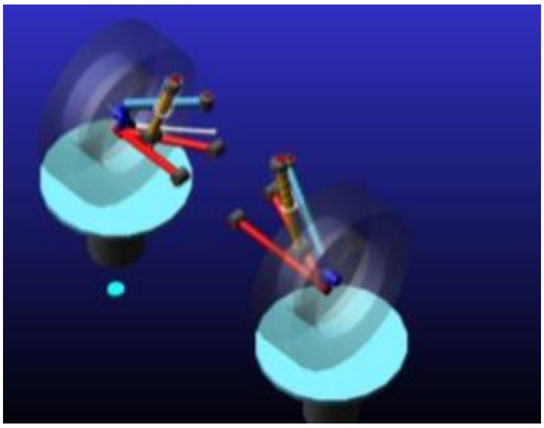

Figure 4. 2: Behind the spring design

Figure 1 shows the front view of the suspension geometry that was developed. Figure 2 above shows the design where the upper wishbone is placed behind the spring of the suspension system.

Finite Element Analysis:

The next step taken after developing the geometry of the suspension system was to model and conduct Finite Element Analysis on the suspension parts.

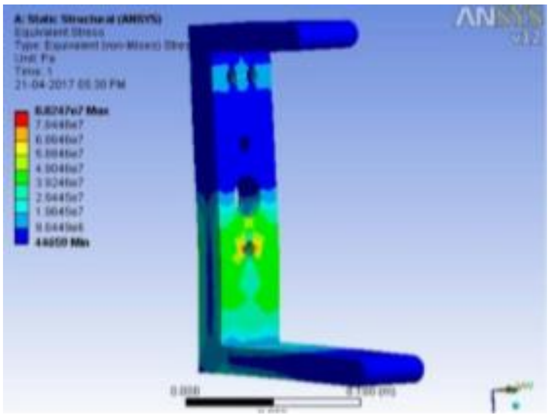

Figure 4. 3: ANSYS result for knuckle

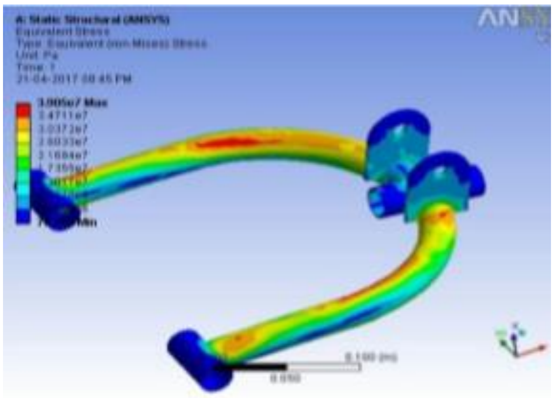

Figure 4. 4: ANSYS result for lower wishbone

The above images show the results of the Finite Element Analysis that was conducted on the hub and the wishbone designed by the team.

Results and Discussion:

After modeling and conducting FEA on the member elements, the next thing done was to generate results on the designed suspension geometry to analyze camber changes, caster changes, kingpin inclination, scrub radius, percent Ackerman and roll steer.

The results for camber change for the new geometry was from -6 to -9.5 degrees. Both values appear to be negative and the justification given to this was that negative camber provides cornering stability and traction while cornering [3].

The results of the caster graph showed caster change for new geometry to be from 3 to 10 degrees. In this case, both values are positive. The justification for this was that radial tires usually have caster angles over 7 degrees, and since they are running radial tires, they went with the range of 3 to 10 degrees to help the steering system.

The results for kingpin inclination showed a change in kingpin inclination for the new geometry from 7 to 8 degrees. Both values are also positive.

The results for scrub radius for the new geometry was also found to be 21 to 23mm. Both values of scrub radius are also positive. They defined positive scrub radius as the steering axis intersecting the ground plane between the vehicle centerline and the contact patch.

Conclusion:

They concluded based on the results that the new optimized geometry was ready to be installed on the vehicle since they were satisfied with the results generated.

2.2 Review 2

Paper 2 - *Design and Analysis of an ATV Suspension System* [4]:

The focus of this paper as stated in the abstract is to design, analyze and simulate an ATV suspension system mainly designed for a national level event namely Baja SAE INDIA.

Design:

In the design phase, they first decided to determine the desired system characteristics. The software that was used to design and analyze the suspension geometry was Lotus Engineering Suspension Analysis software. This software was used to design the hard points of the suspension to achieve the suspension characteristics that were being looked out for. The software that was used to create the CAD (Computer Aided Design) model was CATIA V5R21. The member elements of the suspension system were not just designed but designed for manufacturability. They were also modeled with assembly considerations in mind. After designing the member elements, ANSYS 15.0 was used to perform structural analysis on the suspension system to verify the flawless performance of the design.

The design targets set were to isolate amplitude obstacles by increasing travel, maintain undamped natural frequency from 1.2Hz to 1.5Hz, implement anti-dive geometry, and to minimize chassis roll by maintaining the roll gradient in the range of 1.5 degrees to 2 degrees/g.

Front suspension:

Short long arm type wishbone was selected for front suspension. Both upper and lower wishbones were designed in A-arm shape with a ball joint for attaching onto the knuckle. The wishbones were designed to be connected to the chassis at two pivots with a helm joint. They managed to design the suspension system in such a way that the tires remain in a proper orientation in all modes of motion. FOX progressive air shocks were selected as the shock absorbers for the suspension system design.

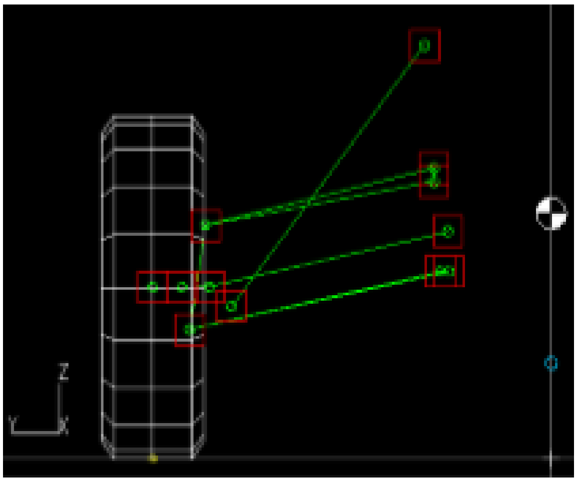

Figure 2. 1: Front suspension hard points created using Lotus software

Rear suspension:

Trailing arms were selected for the rear suspension system. This suspension type was also chosen due to its ability to allow a great deal of camber control and changes to be made. It was also chosen due to its ability to provide better wheel travel.

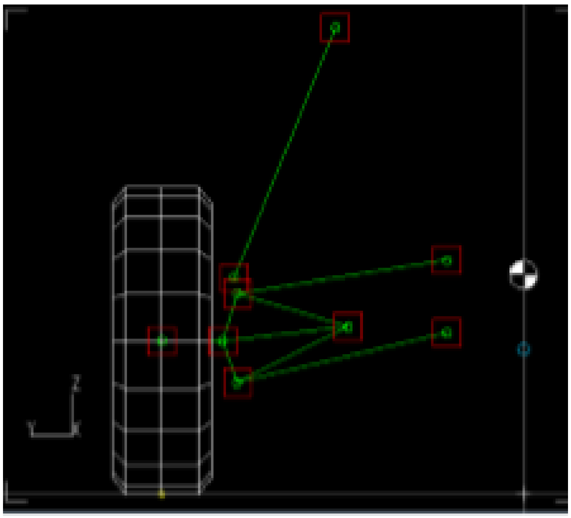

Figure 2. 2: Rear suspension hard points created using Lotus software

After developing the front and rear suspension systems using the Lotus software, results on camber change and toe change were developed from the software and analyzed.

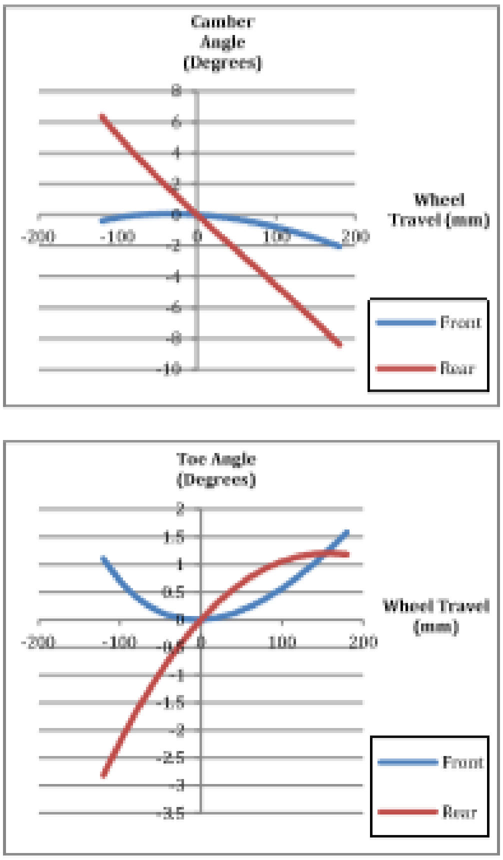

Figure 2. 3: Results on camber change and toe change

Finite Element Analysis:

After generating kinematic results, the next step taken was to conduct finite element analysis on the member elements of the suspension system. The software that was used to conduct FEA on the wishbone and the knuckle was ANSYS workbench 15.0. They simulated for a worst-case scenario meaning, a force greater than what is applied onto the part was used. Their results showed that the yield stress did not exceed the ultimate stress even under a worst-case scenario situation. 3g forces were used for the structural analysis of the wishbones and the trailing arms.

Conclusion:

In conclusion, the short long arm type wishbone suspension design provided efficient handling during cornering and enhanced wheel control. The designed trailing arm suspension system provided the desired wheel travel and static camber needed. The obtained wheel travel allowed for isolation from high amplitude obstacles. The quality of the desired was approved by performing FEA on the member elements of the suspension system.

Both papers followed about the same methodology in designing the suspension system. These processes are:

- Selecting suspension parameter values based on target of design
- Development of suspension geometry
- Using software to create developed suspension geometry
- Generating kinematic simulation results from software
- Using software to develop 3D model of suspension parts
- Conducting FEA on 3D modeled parts

Chapter 3: Design

This chapter purposely deals with the discussion of existing suspension designs that are mainly used on All-Terrain Vehicles (ATVs) and their technical specifications. The suspension systems that will be discussed in this chapter are limited based on research findings.

3.1 Review of existing designs

First, to briefly define what a suspension is, we can say that it is a system of tires, tire air, springs, shock absorbers and linkages that allow a vehicle to connect to its wheels and allows relative motion between the two. All these components including several others are put together to form the suspension system of a vehicle. The most basic functionality of the suspension system of a vehicle is to support the handling or road holding capabilities and improve the ride quality experienced by the passengers.

The suspension systems of vehicles can be broadly classified into two subgroups. These subgroups are dependent and independent suspension systems [5]. The difference between the two subgroups is the ability of opposite wheels to move independently of each other. With the dependent subgroup, a beam or live axle usually holds the wheels parallel to each other and perpendicular to the axle. Most Mini Baja vehicles had independent suspension systems incorporated in the design of the vehicle. Suspension systems designs to be discussed include MacPherson strut, Double wishbone, Multi-link suspension, and Semi-trailing arm suspension.

3.1.1 MacPherson Strut suspension

This suspension system was developed by Earle S. MacPherson of General Motors in 1947 [5]. It has since been the most widely used front suspension system, especially in cars of European origin [5]. It is widely used in the front suspension of modern vehicles. This type of suspension system combines a shock absorber and a coil spring into a single unit. Some advantages of this suspension system include:

- More compact and lighter suspension system to be used for front-wheel drive vehicles.
- Due to the simple nature of its design, there are fewer joints in the suspension to wear over time.
- Monotube struts that are inverted can provide extra rigidity in the front suspension.

Some disadvantages of using this suspension system include:

- Geometric analysis has shown that the system cannot allow vertical movement of the wheel without some degree of either camber angle change, sideways movement or both.
- Compared to double wishbone or multi-link suspension, it does not give as much good handling. It limits the freedom of engineers to choose camber change and roll center.
- McPherson struts are not suitable to be used on vehicles with a cockpit adjustable ride height due to camber changes that cannot be avoided.
- Vehicle is subjected to suffer from almost same vertical motion from shock absorbers as the wheel causing relatively little leverage to break the stiction in the seals.

3.1.2 Double Wishbone suspension

This type of suspension has been used on several vehicles starting from performance vehicles all the way down to the most common of cars. The setup of a double wishbone is such that the absorbers and coil springs connect upper and lower control arms, whereby the steering knuckle and hub carrier are found on the lower control arm and the upper control arm attached to the frame. It is designed mainly for body-type vehicles [6].

Some advantages of using this suspension system include:

- When cornering hard and the car starts to roll, this system maintains a better tire contact patch with the road.
- Allows for greater control over camber, caster, and roll center.
- It gives more freedom with the placement of dampers.
- Since the damper does not stick out with this system design, it is much more economical in terms of vertical space.
- More rigid as compared to McPherson Strut.

Some disadvantages of incorporating this suspension system into vehicle design includes:

- It is relatively more expensive and complex as compared to McPherson Strut suspension system.
- Since there are more joints in this system, the problem of higher service costs rises.
- In terms of packaging, those it does not take up much vertical space, it takes up more horizontal room due to the location of the upper arm.

3.1.3 Multi-link Suspension

The Multi-link suspension system design deals with using several short links (or arms) to attach the hub carrier to the car's body. Each link is configured to make sure that the camber angle of the wheel remains unchanged during suspension movement. The number of links used in the design may vary based on the user's target. The links may vary from as low as three (3) to as high as five (5). The system uses three or more lateral arms and one or more longitudinal arms that do not have to be of equal length and can be angled away from their natural direction [7]. It is considered the best and most functional independent suspension system to be used on a production car.

Some advantages of a Multi-link suspension system include:

- Due to its ability to allow a vehicle to flex more, it makes it a good solution for off-road driving.
- Designers can alter one parameter in the suspension without affecting the entire assembly.

Some disadvantages of a Multi-link suspension system include:

- It is more complex and incorporates more components.
- It is more expensive to design and produce.

3.1.4 Semi-trailing arm suspension

This suspension type is a flexible independent rear suspension system for automobiles where each wheel hub is located only by a large, roughly triangular arm that pivots at two points. This type of suspension system is commonly used for the rear wheels of vehicles to allow for a flatter floor and more cargo room.

Some few advantages of a semi-trailing arm suspension are as follows:

- It has better rolling characteristics

- It does well in handling lateral forces in a better way
- Allows better control of vehicle when cornering.
- Offers camber gain to the wheels due to the pivots mounted at inclinations to the chassis

Some few disadvantages of a semi-trailing arm suspension are as follows:

- When the wheel moves up and down, camber angle changes.
- Since they are firmly attached to the wheels, more shock and noise could be transferred to the car body.
- Unsprang weight of the trailing arm leads to poor ride quality.

All the above stated suspension types are the suspension systems that are found being used in several of the Mini Baja vehicles being built to be tested in the Baja SAE intercollegiate competition. Several teams move to the competition grounds carrying spare components especially for their suspension systems because the has been a recurring problem of suspension failures especially during the 4-hour course challenge.

Problem Statement: Several competing Mini Baja vehicles fail to complete the 4-hour course challenge of hardcore testing of the vehicles due to failure of the suspension system of the vehicle. This comes about because of poor design of suspension system.

3.2 Thesis design objective

The goal of this project is to design and analysis a suspension system for a mini Baja, and how to integrate them into whole vehicle system for best results.

3.3 Design decisions

Designing the suspension system requires choosing a suspension type for both the front and rear of the vehicle. Review of already existing suspension types as seen in subsection 3.1 was conducted, which informed the choice of suspension type to be used. Also, review of team reports on the suspension system from teams that have taken part in the SAE Baja challenge were studied to understand the reason behind the selections that were made for the suspension type for both front and rear of the Baja. After reviewing suspension system types that are mostly used for the front and rear of the vehicle, and after carefully weighing their strengths and weaknesses in terms of advantages and disadvantages as well as their technicalities, a conclusion was drawn on the types of suspension system to be used for the front and rear of the Baja. Unequal length double wishbone with damper mounting on the lower A-arm was chosen for the front suspension system whereas unequal length double wishbone with damper mounting on the upper wishbone was chosen as the rear suspension system. The meaning of the unequal length is that the upper A-arms are shorter in length than the lower A-arms. This is to induce negative camber on the wheels. This decision was made based on how well the chosen suspension systems for both the front and rear of the vehicle satisfy the objectives of the project.

3.3.1 Pugh Matrix

Table 3. 1: Front suspension type decision matrix

5 = best, 1 = worst			
Criteria	Rating	Unequal-length double A-arms	McPherson Strut
Manufacturing and Serviceability	20%	4	5
Weight	15%	4	5
Performance (handling)	25%	5	3
Clearance	20%	5	2
Cost	20%	3	5
	100%	21	20

Table 3. 2: Rear Suspension Type Decision Matrix

5 = best, 1 = worst				
Criteria	Rating	Unequal-length double A-arms	3-link Trailing Arm	Pure Trailing Arm
Manufacturing and Serviceability	20%	4	3	3
Weight	15%	5	4	4
Performance	25%			
Rear impact protection	20%	3	2	4
Cost	20%	4	4	4
	100%	16	13	15

Chapter 4: Methodology

From the Pugh matrix in chapter 3, unequal-length double A-arms were selected for both front and rear suspension systems of the Baja. After making this decision, several other procedures were carried out to fully design the selected suspension systems to meet the stated objectives of the project. Each procedure or method will be explained in subsections in this chapter.

4.1 Computational setup

4.1.1 Software

The main software used to carry out the design phase of the project include SolidWorks, MATLAB, VSusp online software, and Lotus Suspension Analysis software.

- SolidWorks:

  SolidWorks is basically a computer-aided design and computer-aided engineering software used for solid modeling. SolidWorks offers a lot of options when it comes to 3D modelling. It has several sections of itself which are used for various purposes. For this project, SolidWorks is used to model the suspension components and the entire Baja vehicle to help visualize the suspension systems on the vehicle. The FEA (Finite Element Analysis) simulation package of SolidWorks is also used to conduct static tests on the suspension components.

- MATLAB:

  MATLAB is a software that combines a desktop environment tuned for iterative analysis and design processes with a programming language that expresses matrix and array mathematics directly [8]. This software is used in studying the amount vibration the vehicle experiences as it goes over bumps. MATLAB helps simulate the oscillation of both unsprung and sprung mass of the vehicle to assess how well the designed suspension system works.

- Lotus Suspension Analysis:

  This is a user-friendly suspension geometric and kinematic modelling tool, which allows and makes it easy to apply changes to developed geometry as well as instantaneously assess impacts via graphical results [9].

- VSusp online software:

  This is an online running software that allows users to develop suspension geometry by putting in values for various parameters. This was used to generate the suspension geometry which was worked with.

4.2 Suspension Design

4.2.1 Overview

This section entails detailed information on the various steps that were followed to execute the design of the suspension system for both front and rear of the Baja. For the front wheels, unequal length A-arms was chosen

4.2.2 Tire Selection

Much time can be spent on the design of the suspension system. However, its capabilities will be observed based on the set of tires that would be used as the front wheels and rear wheels of the vehicle. Tire selection was done based on the objectives or design goals for the Baja suspension system, also keeping in mind the type of terrain the vehicle will be used on. Tire types include radial and bias tires. Radial tires from research are said to be much stronger and last longer as compared to bias tires. Also, tires come either tubeless or with a tube in it. Tires with tubes mean more weight and vice versa for tubeless tires. With the design goal of lesser unsprung weight, tubeless tires were selected for both front and rear of the vehicle. Since the tires are the parts that connect the vehicle to the ground, allow it to accelerate, brake, and maneuver, the tires were carefully selected considering the tread pattern and width. The tread pattern influences the grip the tire has on the ground when moving. For the front tires, more grip and lateral stability or lateral g-force capabilities of the car is required therefore, the "knobby" style tread pattern was selected. They also provide more grip in corners. The rear tire tread pattern must allow the vehicle to slide while cornering, and since a design goal is to allow oversteer characteristics, the bar style tread pattern was chosen. The tires selected also have good width to aid with gripping the rough track. With ground clearance also in mind, the tires that have been selected aid substantially in obtaining the desired ground clearance for the Baja. The front tires were chosen to be **Maxxis M943 iRazr 22x7R-10** and the rear tires were chosen to be **GBC Dirt Devil 22x8R-10**. Specification tires can be found below.

Table 4. 1: Maxxis iRazr Front tire specification

MAXXIS IRAZR AT22X7R-10	
Deepest Tread Depth	1 4/32 in.
Ply Rating	6 Ply
Position - Tire	Front
Tire Classification	Race
Tire Construction	Radial
Tire Size	22x7-10
Type	Tubeless
Units	Each
Weight	12.75 lbs.

Table 4. 2: GBC Motorsports Dirt Devil Rear tire specification

GBC Motorsports Dirt Devil AT22X8R-10	
Brand	GBC Motorsports
Model	Dirt Devil
Item Weight	14.55 pounds
Product Dimensions	22 x 22 x 8 inches
Section Width	22 inches
Aspect Ratio	8
Construction	Bias
Rim Diameter	10 inches
Speed Rating	B
Tread Depth	0.63 inches
Position Tire	Rear

4.2.3 Front Suspension design

The overall dimension of the car was selected or decided on based by considering the body-to-length ratio of the vehicle that would allow better performance of vehicle during cornering and as it moves along the tracks. The track width and body length were selected based on the maximum values set for these parameters by Baja SAE. To allow easy movement of the vehicle through the course especially during cornering, a track width of 52” and a wheelbase of 60” were selected. To maximize obstacle avoidance on the tracks also, a ground clearance of 11” was selected. The double wishbone suspension system was selected due to its flexibility and provision of better ride comfort on bumpy terrain. The double wishbone allows more control on parameters of suspension geometry. It allows change to parameters such as camber angle of the wheel to be done easily. Parameters that were necessary to draw the front suspension geometry, which plays a huge role in determining the length of the wishbones can be found in the next section.

4.2.3.1 Determination of length of wishbones

Table 4. 3: Input values for front suspension geometry

PARAMETERS	VALUES
Track width	52”
Wheel base	60”
Ground clearance	11”
Camber angle	0.5 deg
Kingpin Inclination	11.4 deg
Roll center height	3.50”
Tire	22x7R-10

The data above was used to draw the optimum suspension geometry to fulfill the given requirements. The generation of the front suspension geometry was done with the aid of an online platform that deals with suspension geometry design, VSusp online. Using this online platform, the above data was inputted into the software, which then generated the front suspension geometry. It always allows for changes to be made. The results of the front suspension geometry can be seen below:

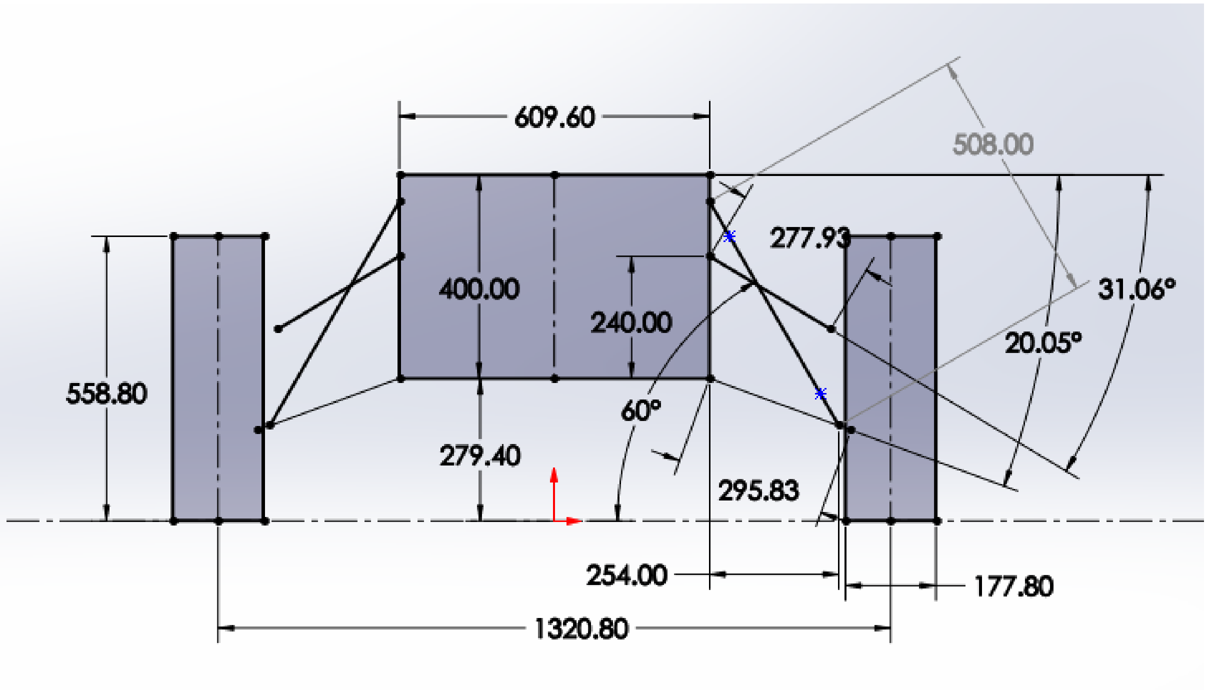

Figure 4. 5: Front suspension geometry

By drawing the suspension geometry, the length of the upper and lower wishbones for the front suspension system were determines.

Table 4. 4: Final values obtained for front wishbone design

Parameter	Value
Length of upper wish bones	10.942”
Length of lower wishbones	11.647”
Inclination of wishbone with upper horizontal (α)	31.06˚
Inclination of wishbone with upper horizontal (β)	20.04˚

4.2.4 Front spring design

After arriving at the front suspension geometry, the next step was to design the spring for the front suspension system based on the weight of the vehicle. To begin the spring design, the various masses that make up the vehicle were measured and group to aid in the development of the spring. The various design considerations made for the vehicle suspension are as follows:

Table 4. 5: Coil spring design parameters used

Parameters	Values
Sprung mass	270 kg
Unsprung mass	80 kg
Estimated weight	260 kg
Driver with accessories	90 kg
Mass distribution (Front:Rear)	40:60
Mass per front wheel	54 kg
Mass per rear wheel	81 kg
Static to Dynamic amplification factor	2.5

The front suspension unequal-length double A-arm system has the spring-damper system mounted on the lower arm therefore calculations were done based on the lower arm. The length of the wishbone was measured horizontally from the chassis. The mounting point of the spring on the lower wishbone was also measured horizontally from the chassis.

Material of coil selected: 17-7 PH ASTM A313 (631)

This is a stainless-steel material. This material was chosen out of several options due to the nature of the terrain and sorts of challenges that the event would entail, especially the muddy parts of the terrain. The stainless-steel material is most preferred for moist conditions, thus the justification of material choice.

Length of lower wishbone = 11.72" (297.71mm)

Spring-damper mounting angle (inclined to the horizontal) = 60˚

Mounting point of the spring on the lower wishbone = 10" (254mm)

Reaction force from the ground when wheel goes over a bump:

$F_{wheel} = (mass\ per\ front\ wheel)(9.81)N$

$F_{wheel} = (54kg)(9.81)$

$F_{wheel} = 529.74N$

A force slightly greater than the calculated force above will cause the wheel to move upwards. Despite that fact, for calculation purposes, the value of the force obtained above will be used.

Calculation the spring force:

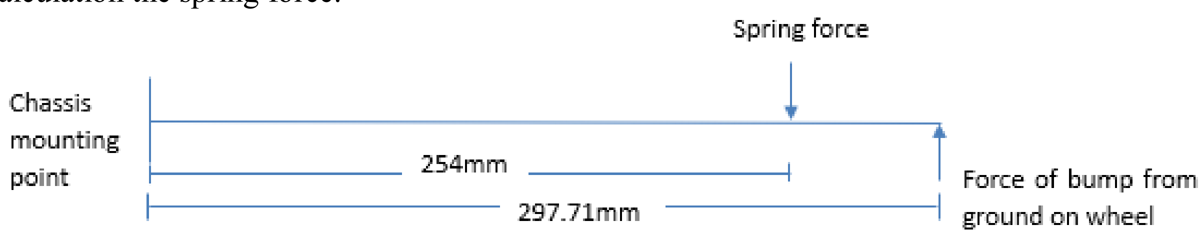

Figure 4. 6: Forces on front upper wishbone

Taking moments about the wishbone hinge point on the chassis:

$$\Rightarrow (F_{spring})(254mm) = (F_{wheel})(297.71mm)$$

$$\Rightarrow F_{spring} = \frac{(529.74)(297.71)}{(254)} = 620.90N$$

Applying dynamic amplification factor:

$$\Rightarrow F_{dynamic,spring} = (2.5)(620.90)$$

$$\Rightarrow F_{dynamic,spring} = 1552.25N$$

Calculating stiffness constant (ks):

To calculate spring stiffness, the spring deflection for the vehicle must be identified. As an ideal condition for ATV (All-Terrain vehicles) front wheel travel, a spring deflection of 4" was selected.

$$\text{Stiffness (ks)} = \frac{Dynamic\ spring\ force}{Spring\ travel}$$

$$\text{Stiffness (Ks)} = \frac{1552.25}{101.6} = 15.28N/mm$$

Calculating number of coils in the spring from the force:

The formula used to calculate the number of coils in the spring is given below.

$$ks = \frac{Gd^4}{8nD^3}$$

d = 8.45mm

D = 64.96mm

D' = 73.41mm

G = $75.8 \times 10^3 MPa$

The variables in the formula are defined below:

n = number of coils in the spring

k = spring stiffness value

G = Modulus of rigidity of the spring material

d = diameter of the spring wire

D = Mean diameter of the coil spring

D' = outer coil spring diameter

$$n = \frac{Gd^4}{8D^3ks} = \frac{(75.8 \times 10^3)(8.45^4)}{(8)(64.96^3)(15.28)}$$

$$n = 11.53$$

$$n \approx 12\ coils$$

4.2.5 Rear Suspension design

Developing the rear suspension geometry, a track width of 50" was chosen as compared to the front suspension design. The smaller track width at the rear is help increase the oversteer characteristics of the vehicle to take corners with less effort. It was also made smaller to improve the stability of the vehicle. According to the Baja SAE technical specification standards, the vehicle must have four or more wheels not in a straight line [10]. This improves the stability of the vehicle as it moves along the track.

4.2.5.1 Determination of length of wishbones

Table 4. 6: Input values for rear suspension geometry

PARAMETERS	VALUES
Track width	50”
Wheel base	60”
Ground clearance	10”
Camber angle	0.5 deg
Kingpin Inclination	9.04 deg
Roll center height	-0.157”
Tire	22x8R-10

The above data was used in developing the rear suspension geometry. This was also done with the aid of VSusp online platform. The detailed drawing of the rear suspension geometry can be found below.

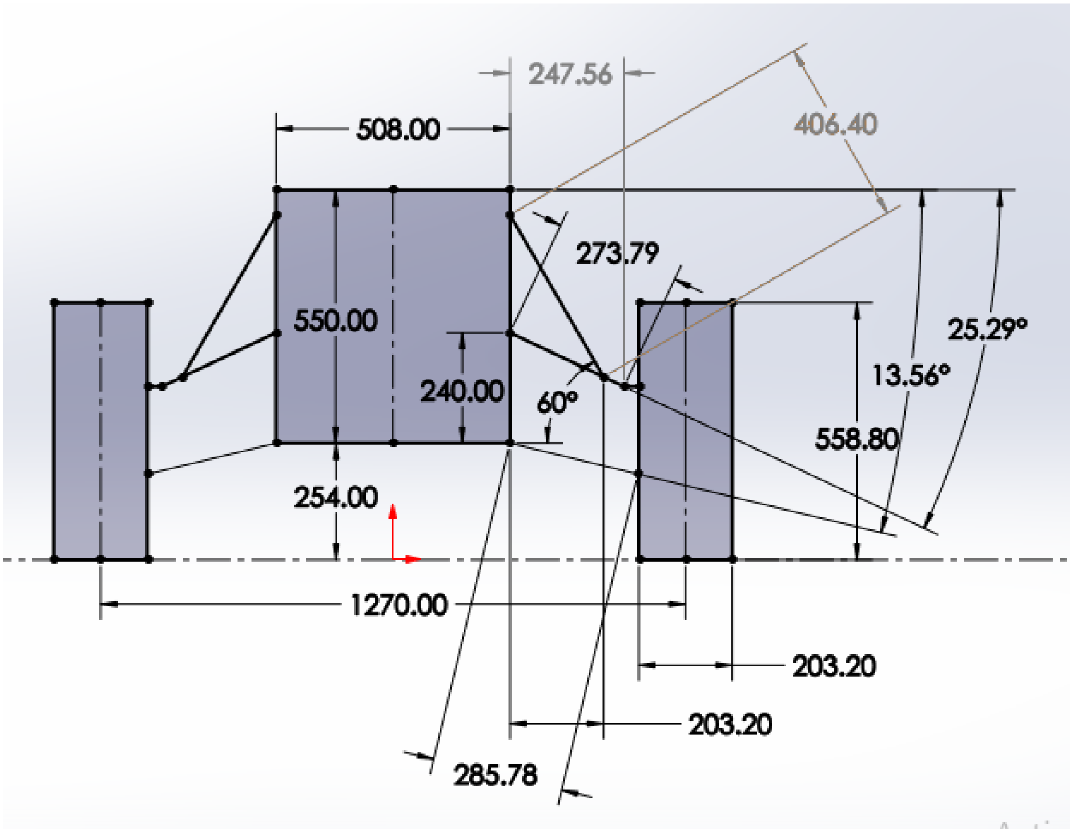

Figure 4. 7: Rear suspension geometry

The generated geometry above assisted in measuring the lengths of the upper and lower wishbones.

Table 4. 7: Final value for rear wishbone design

Parameter	**Value**
Length of upper wish bones	10.779”
Length of lower wishbones	11.251”
Inclination of wishbone with upper horizontal (α)	25.285˚
Inclination of wishbone with upper horizontal (β)	13.56˚

4.2.6 Rear spring design

Length of lower wishbone = 9.75” (247.56mm)

Spring-damper mounting angle (inclined to the horizontal) = 60˚

Mounting point of the spring on the lower wishbone = 8” (203.20mm)

Reaction force from the ground when wheel goes over a bump:

$$F_{wheel} = (mass\ per\ rear\ wheel)(9.81)N$$

$$F_{wheel} = (81kg)(9.81)$$

$$F_{wheel} = 794.61N$$

A force slightly greater than the calculated force above just like the front wheel will cause the wheel to move upwards. Despite that fact, for calculation purposes, the value of the force obtained above will be used.

Calculation the spring force:

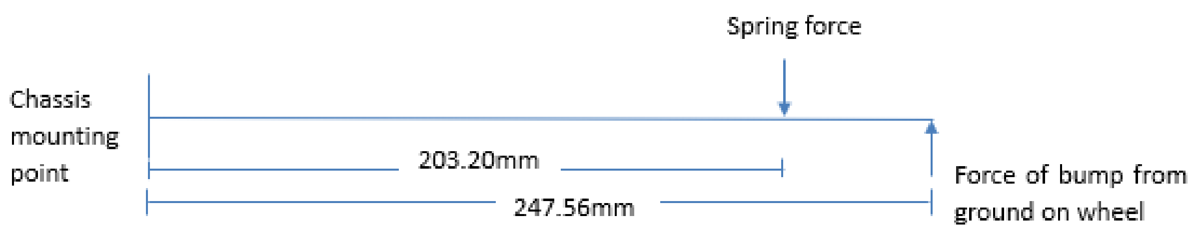

Figure 4. 8: Forces on rear upper wishbone

Taking moments about the wishbone hinge point on the chassis:

$$\Rightarrow (F_{spring})(203.20mm) = (F_{wheel})(247.56mm)$$

$$\Rightarrow F_{spring} = \frac{(794.61)(247.56)}{(203.20)} = 968.08N$$

Applying dynamic amplification factor:

$$\Rightarrow F_{dynamic,spring} = (2.5)(968.08)$$

$$\Rightarrow F_{dynamic,spring} = 2420.197N$$

Calculating stiffness constant (ks):

To calculate spring stiffness, the spring deflection for the vehicle must also be identified for the rear wheel. As an ideal condition for ATV (All-Terrain vehicles) rear wheel travel, a spring deflection of 4" was selected.

$$\text{Stiffness (ks)} = \frac{Dynamic\ spring\ force}{Spring\ travel}$$

$$\text{Stiffness (Ks)} = \frac{2420.197}{101.6} = 23.82N/mm$$

Calculating number of coils in the spring from the force:

The formula used to calculate the number of coils in the spring is given below.

$$ks = \frac{Gd^4}{8nD^3}$$

d = 8.45mm

D = 64.96mm

D' = 73.41mm

G = 75.8 $\times 10^3 MPa$

$$n = \frac{Gd^4}{8D^3ks} = \frac{(75.8 \times 10^3)(8.45^4)}{(8)(64.96^3)(23.82)}$$

$$n = 7.398$$

$$n \approx 8\ coils$$

Table 4. 8: Spring Parameters

SI. NO.	Parameter	Front Spring	Rear Spring
1	Diameter of wire	8.45mm	8.45mm
2	Outer diameter	73.41mm	73.41mm
3	No. of turns	12	8
4	Free length of spring	411.00mm	316.40mm
5	Pitch of spring	32.14mm	36.38mm
6	Eye-to-eye length of spring-damper(unloaded)	15.28N/mm	23.82N/mm
7	Stiffness of spring	144.03mm	96.02mm
8	Maximum travel	508mm	406.40mm

4.2.7 Design of wishbone- Front and Rear

After generating the suspension geometries for both front and rear suspension systems, the next step was to model the wishbone to allow FEA analysis to be conducted on them. This is to test how well they would perform under stress. SolidWorks was used to model both front and rear wishbones which can be seen in the figure below. The 3D model was created using dimensional length obtained from the suspension geometry. The designs were also developed based on the chassis dimensions as well as the track width that were chosen with regards to the restrictions given by Baja SAE. The upper and lower arms were designed to be of unequal length, whereby the lower arm was made longer than the upper arm.

For the front suspension setup, the spring-damper component is mounted on the lower wishbone, both the upper and lower wishbones are attached to the upright via a ball joint. For the rear suspension setup, the spring-damper component is mounted on the upper wishbone. Its upper and lower wishbones were also designed to be attached onto the upright by a ball joint. Material chosen for the wishbones was AISI 4130 chromoly steel. 3D models of the lower and upper arms of the front and rear wishbones are shown in the figures below.

Figure 4. 9: Front lower/rear upper wishbone

Figure 4. 10: Front upper/rear lower wishbone

4.2.8 Design of hub and upright

The wheel upright refers to the mounting part of the suspension system that is responsible for connecting all suspension, steering and braking parts to stabilize the vehicle

[11]. The upright basically acts as a connector of the chassis to the wheel assembly of the vehicle with the help of the wishbones. The upright also serves as a connector to the steering arm, hence, allowing the driver of the vehicle to control/steer the vehicle. The hub is usually found at the rear of the upright. It is connected to the wheels of the vehicle. The upright is a stationary part whereas the hub moves/rotates with the wheel. Since the baja being designed is a rear-wheel drive, the front upright was designed to be suitable for non-drive suspension. The wheel hub and upright were designed using SolidWorks. Not only were these part designed for FEA (Finite Element Analysis) to be conducted on them individually but they were also designed capable of being put together as an assembly with the remaining suspension parts. The material chosen for these components is AISI 4130 chromoly steel. The next step is to conduct FEA analysis to determine the components' behaviour to receiving stress. The figures below show the 3D model of the hub and upright of the wheel asembly.

Figure 4. 11: Hub

Figure 4. 12: Front upright

Figure 4. 13: Rear Upright

4.2.9 Conducting Finite Element Analysis on suspension components

After designing all the above suspension components using Solidworks, the next step was to simulate each model. Finite Element Analysis is the simulation of any given phenomenon using the numerical technique called Finite Element Method (FEM) [12]. The simulations were done to check the response of each component to stresses they are likely to experience during the competition. The end goal of conducting FEA on the components is to generate results on the life cycle of the components, their strength, and to obtain the factor of safety of the components. To achieve these results, static analysis and fatigue analysis were conducted to obtain the desired results. Finite Element Analysis was conducted using the simulation package in Solidworks. The material selected for all the components was AISI 4130 chromoly steel.

4.2.9.1 Conducting FEA - Wishbones

The wishbones of the suspension system were simulated as a frontal impact situation. Frontal impact test involves simulating a component as though it were stationary, and another object or force runs into it head-on. It is also known as a crash test. To carry out the frontal impact test on the wishbones, a fixed geometry constraint was applied to the chassis mounting ends of the wishbones. A 1200kg force pointing to the rear of the vehicle was applied to the ball joint housing of the wishbones. The ball joint housing refers to the front of the wishbone that fixes onto the upright. With both the fixed geometry constraint and a force of 12000N applied to the wishbone, the simulation was carried out. Results were generated for both static (frontal impact) and fatigue analysis of the components. This was done for both upper and lower wishbones of the front suspension system.

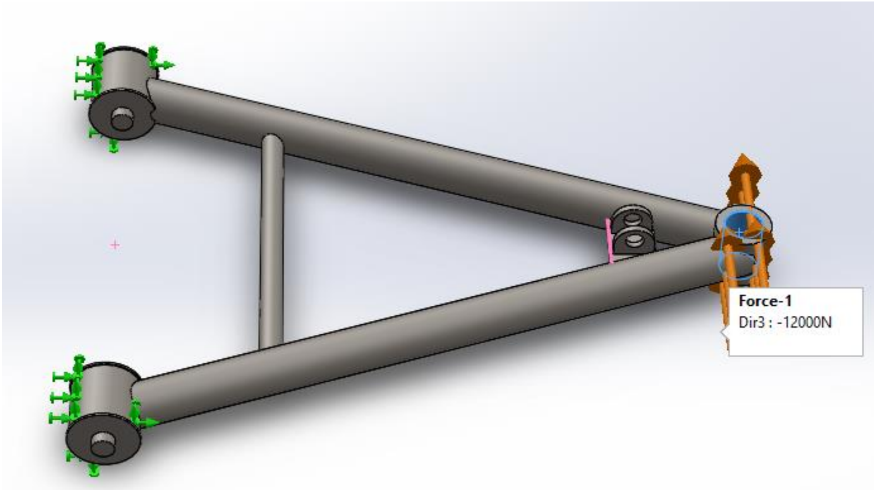

Figure 4. 14: Wishbone with fixture geometry constraint and force applied

The green arrows indicate the fixed geometry constraint on the wishbone, and the orange highlighted arrows pointing to the rear-end of the vehicle symbolizes the force applied on the ball joint housing of the wishbone.

4.2.9.2 Conducting FEA – Hub

To conduct FEA on the wheel hub, a fixed geometry constraint was applied to the portion of the hub that is mounted onto the suspension upright. The wheel hub is simulated in a case where it has experienced a bump force. A bump force is the force that a vehicle or its components experience when it hits a road bump. The force applied onto the hub is a remote force. It was applied as a remote force because the hub does not experience the bump force directly but rather through the tires. The force is applied from a distance that implicates the point where the tire has contact with the ground. In this case, the remote load was applied at a distance of 279.40mm, which is the point the tire has contact with the ground. The force is applied onto the wheel hub by selecting the stud holes that were designed. The stud holes contain rods that fix the tires onto the vehicle. To effectively test the design of the components, the forces per tire used were:

- Normal force: 3G’s
- Lateral force: 2G’s
- Longitudinal force: 2G’s

The Baja suspension system was designed with a front to rear weight ratio which is 40:60. The total weight of the vehicle is 350kg, therefore 140kg of the vehicles weight is skewed to the front. The simulation was done based on a worst-case scenario where the hub experiences the whole 140kg of vehicle weight on itself, hence the forces that were applied onto the hub are as follows:

Table 4. 9: Forces acting on tire

Force type	**Value (N)**
Longitudinal	2800
Lateral	2800
Normal	4200

Using this data, the simulation was carried out and results were generated.

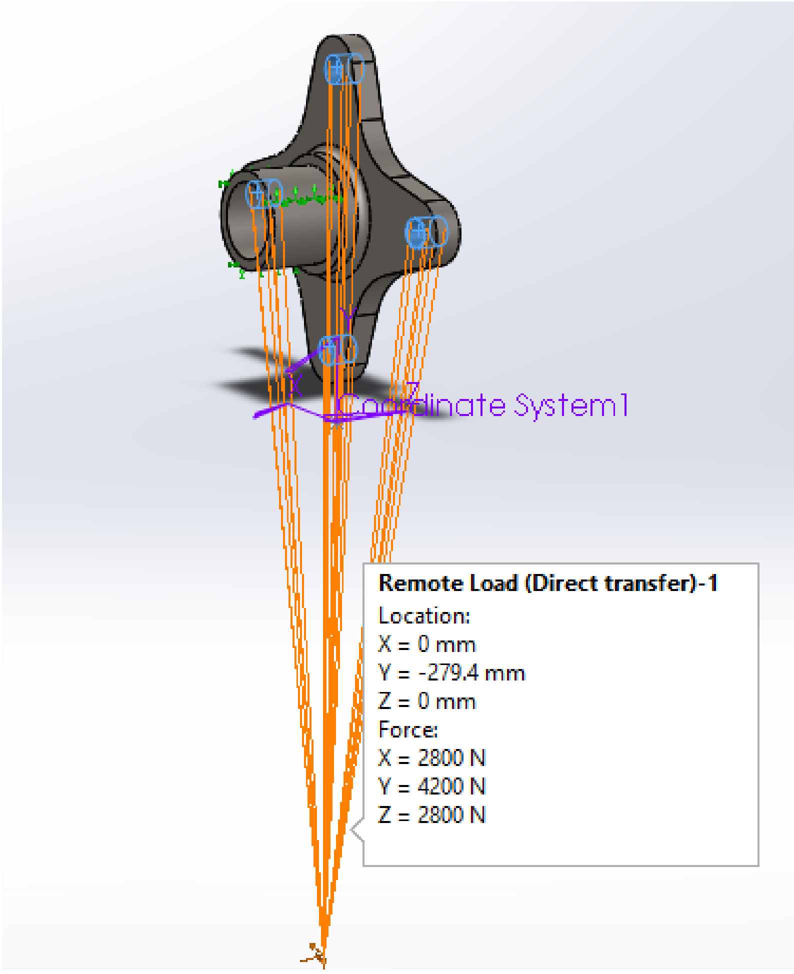

Figure 4. 15: Fixture constraints and force application points on hub

The image above shows the points at which the fixture geometry (green arrows) was applied onto the hub, the stud holes where the forces were applied (blue highlights), and the distance at which the remote load was set. The remote load is indicated by the orange lines.

4.2.9.3 Conducting FEA – Upright

In conducting FEA on the suspension upright, real-world movements of the wishbones and the upright were considered. The mounting points on the uprights for the wishbones were constrained to prevent it from moving in the lateral and longitudinal directions of the tire. The lower arm mounting point on the upright was also constrained from moving downwards beyond its static point. The steering knuckle was constrained from moving in the outward direction. A remote load is applied from a distance of $279.4mm$ i.e. the distance where the tire is in contact with the ground. Since the uprights are also linked to the tires, normal, lateral and longitudinal forces from the tire experiencing a bump was applied onto the uprights where the hub is mounted. The same forces (table 1) that were used on the hub apply to the simulation of the front upright. Since the vehicle's rear weight is 210kg, the rear upright experiences more force on it as compared to the front. Using the same method by which the forces for the hub were calculated, the forces expected to be acting on the rear hub are as follows:

Table 4. 10: Forces acting on rear upright

Force type	**Value (N)**
Longitudinal	4200
Lateral	4200
Normal	6300

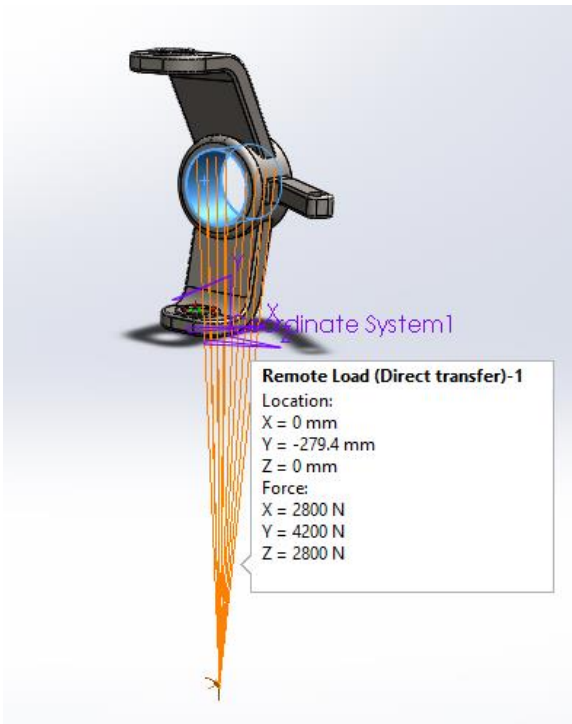

Figure 4. 16: Force application on front upright

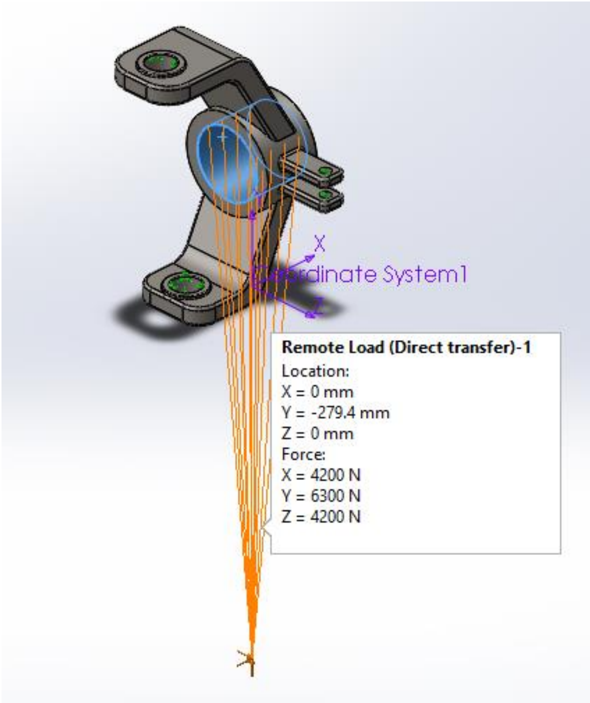

Figure 4. 17: Force application on rear upright

4.2.10 Suspension simulation – Lotus Shark

After finalizing simulations on the designed suspension components, the next step was to move the suspension geometry into Lotus Shark suspension simulation software to generate kinematic results. The designed front and rear suspension geometries were used to develop the front and rear suspensions using the software. Several parameters ranging from the tires selected to the track width, wheelbase, spring stiffness, and several calculated and derived values for the suspension system were all inputted to arrive at the designed suspension system based on the geometries developed. Limitations given by SAE were also considered. With help from *Getting Started With Lotus Suspension Analysis* [13], suspension system was created using Lotus Shark software. Results were then generated on camber variation, caster, toe, roll center, halftrack, and kingpin angle variations. Parameters and values used in setting up both front and rear suspension systems in Lotus Shark are:

Table 4. 11: Lotus Shark suspension settings

Settings	**Front**	**Rear**
Toe	$0°$	$0°$
Camber	$-0.5°$	$-0.5°$
Caster	$6°$	$0°$
KPI	$11.38°$	$9.04°$
Anti-dive	-	-
Anti-squat	-	-
Ackerman	110%	-
% braking	60%	40%
Suspension travel		
Bump	3”	3”
Rebound	1”	1”
Tire		
Rolling radius	11”	11”
Width	7”	8”

The mentioned results generated are the kinematics of the suspension system. Kinematics refer to how the system behaves as the vehicle travels over an obstacle or maneuvers around a turn. Lotus Shark software is used to analyze the kinematics of the suspension.

4.2.10.1 Front suspension simulation

Based on the front suspension geometry, wishbone design, and other several generated values, the hard points on the chassis were calculated. Static camber of the wheels was taken to be $-0.5\ degrees$ to maximize contact of tire surface with ground. This is to maximize traction. The travel was limited to $3''$ bump and $1''$ rebound. All these values were fed into the software to generate the front suspension system as show in the figure below.

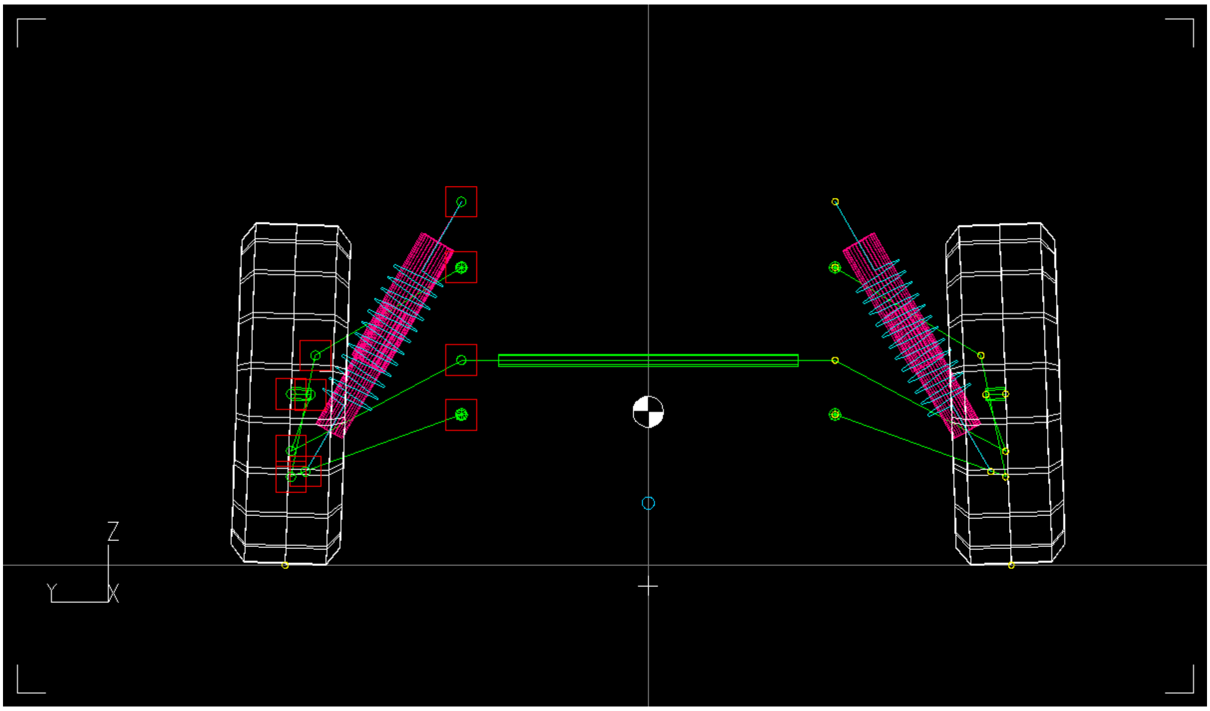

Figure 4. 18: Lotus generated front suspension

Figure 4.14 above shows the image of the front suspension setup that was created. It shows the wishbones and their mounting positions as well as the damper and coil mounting positions. The software helps visualize what the suspension system would be like and allows generation of results to see how best it performs.

4.2.10.2 Rear suspension simulation

The rear suspension hard points were also determined from the rear suspension geometry that was developed. The static camber value was also set to $-0.5\ degrees$. The travel was kept the same as the front suspension. The rear suspension data settings were entered to generate the rear suspension separately. This was developed in another file. Below is the generated rear suspension system.

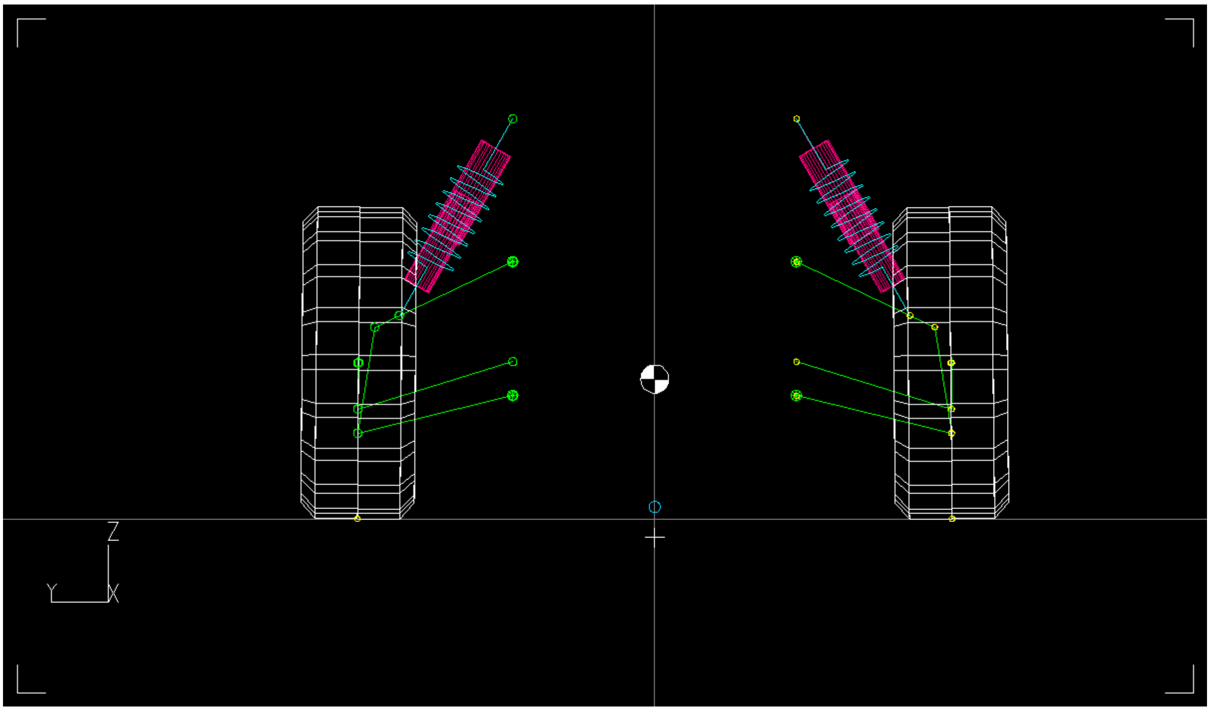

Figure 4. 19: Lotus generated rear suspension

After generating the rear suspension in a different file, the two files were merged to obtain a full vehicle with front and rear suspension setups which can be seen below.

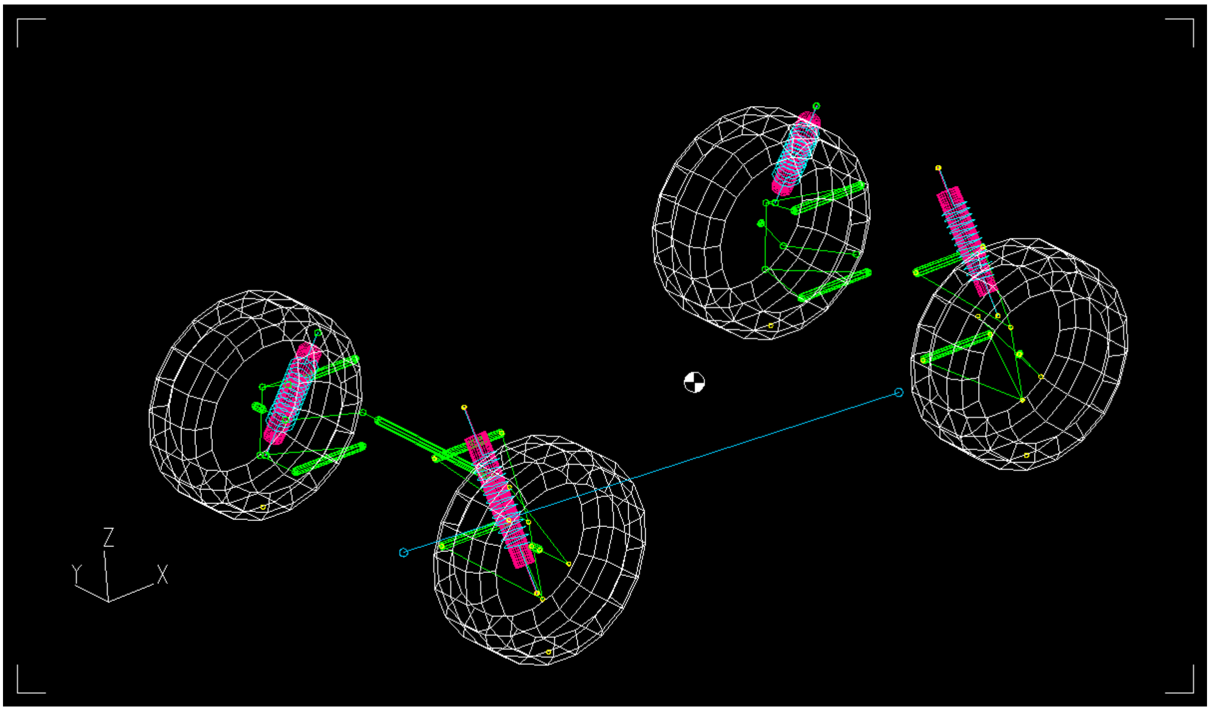

Figure 4. 20: Lotus generated front and rear suspension setups together

After creating both front and rear suspension setups in Lotus Shark, bump simulation results were generated and analyzed.

4.2.11 MATLAB Simulation

Aside using Lotus Shark Suspension software, MATLAB was also used to generate the bump effects on the sprung and unsprung mass of the Baja vehicle. A quarter car model was developed and used to simulate the bump effect on the vehicle. A quarter car model is a model of a suspension system of a vehicle. It has the title ‘quarter’ because out of the 4 tires that exist on the vehicle, only one of the tires is modeled for simulation. By modeling a quarter version of the vehicle, the effects that a bump or bumps would have on the vehicle can be seen. Based on the results obtained, it can be applied to the remaining three tires of the vehicle. The MATLAB simulation shows the impact of the bump and the time it takes for both the sprung and unsprung mass to settle after oscillation of the masses due to bump force.

To simulate a quarter car model of the, a set of equations were derived from the quarter car model below:

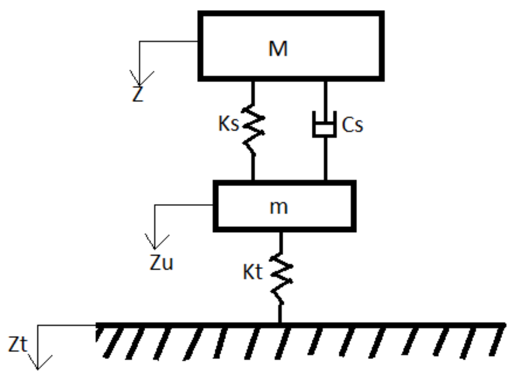

Figure 4. 21: Quarter car model

M = sprung mass

m = unsprung mass

Ks = spring rate

Cs = damping coefficient

Kt = wheel rate

Z = Sprung mass displacement

Zu = Unsprung mass displacement

Zr = Road displacement

Fb = Force on sprung mass

Fw = Force on the sprung mass

Derived sprung mass equation:

$$m\ddot{Z} = C_s(\dot{Z}_u - \dot{Z}) + K_s(Z_u - Z) + F_b$$

Derived unsprung mass equation:

$$m\ddot{Z}_u = C_s(\dot{Z} - \dot{Z}_u) + K_s(Z - Z_u) + K_t(Z_r - Z_u) + F_w$$

After deriving the above equations, a MATLAB Simulink model of the quarter car was developed to be able to generate results from the equations. Since most of the parameters involved have already been identified, these values were put into the Simulink model and simulated to generate results. The developed quarter car Simulink model can be seen below in figure 4.18.

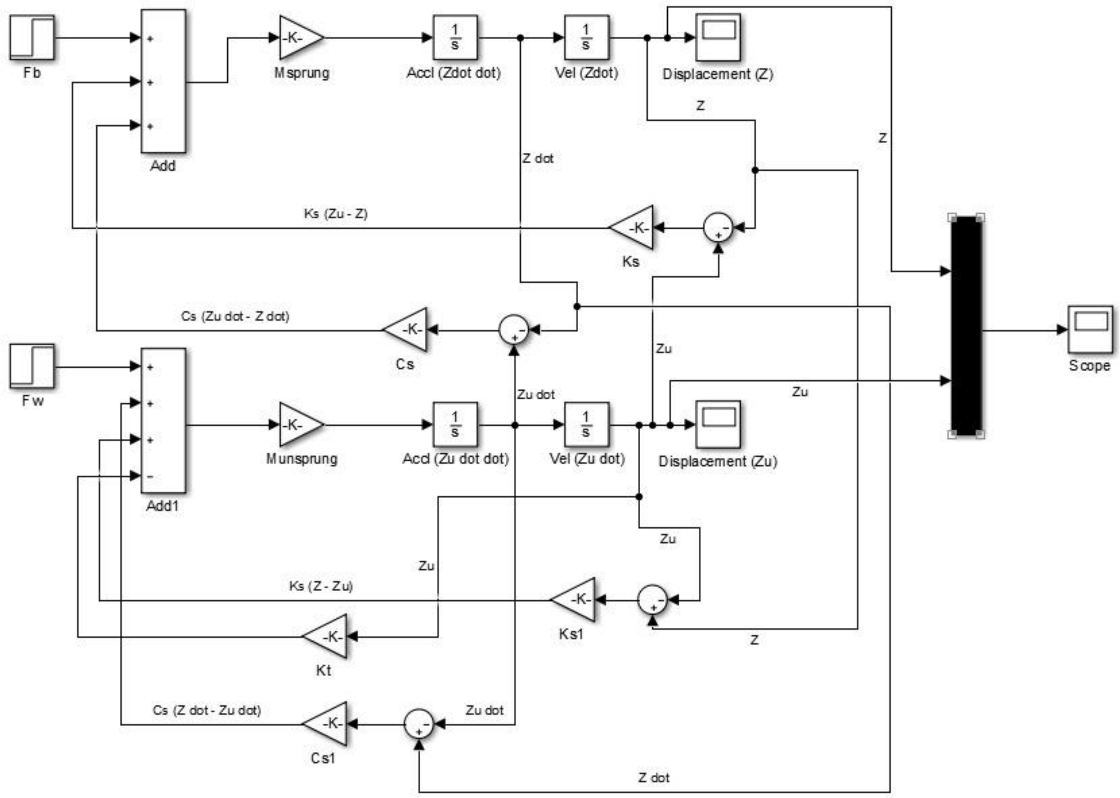

Figure 4. 22: Quarter car Simulink model

Chapter 5: Results

The results section of this paper shows all the generated results with a brief discussion on what the data shows. Simulation was used to perform the various studies that are shown in this section using Solidworks, Lotus Shark, and MATLAB Simulink software.

5.1 Results from Wishbone FEA simulation

5.1.1 Lower wishbone analysis

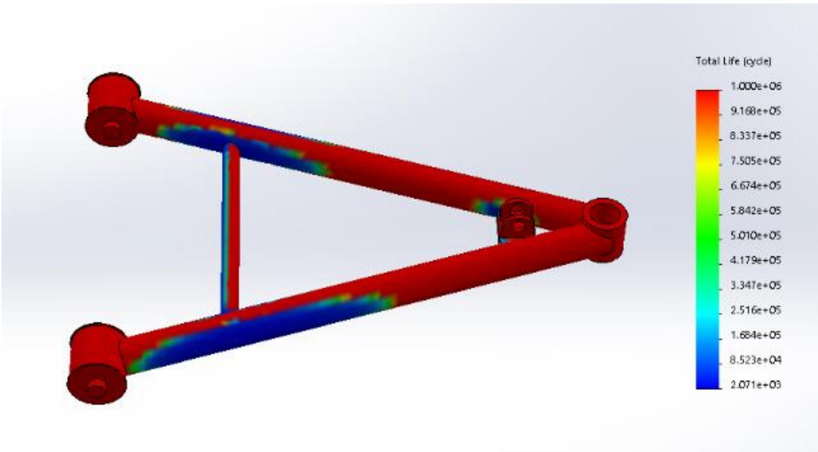

Figure 5. 1: Fatigue analysis results on lower A-arm

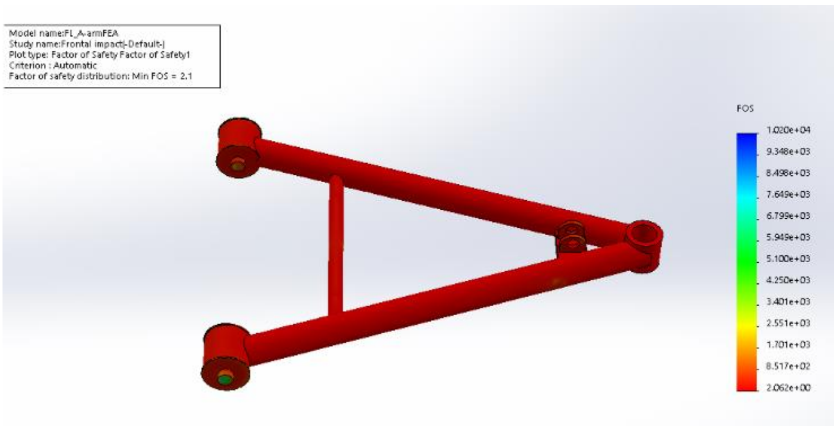

Figure 5. 2: Factor of safety analysis results on lower A-arm

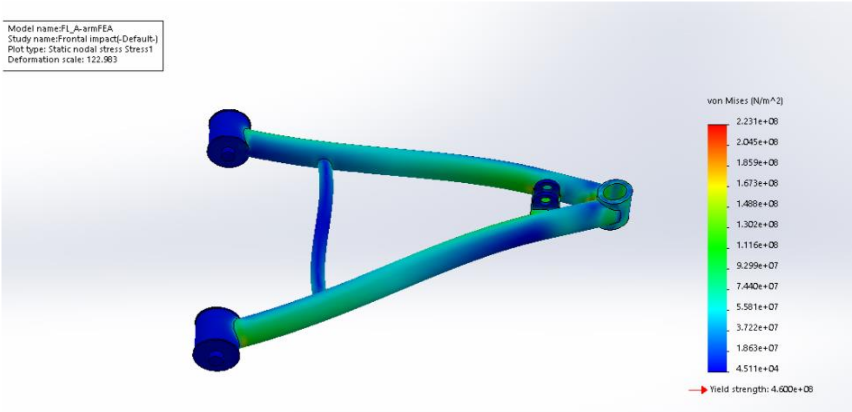

Figure 5. 3: Stress analysis (frontal impact) results on lower A-arm

The lower wishbone was tested for its strength during a frontal impact. The total weight of the vehicle was estimated to be 350kg. It was also determined that the maximum acceleration a Baja vehicle is likely to endure during the competition is 3G [14]. A greater mass of 400kg was used to simulate a worst-case scenario, therefore with this knowledge, a 1200kg force was used to simulate the frontal impact.

Being an off-road vehicle, the components must be able to withstand minor collisions and bumps without failure. Figure 5.1 shows the stress results from the stress analysis of the lower A-arm after experiencing a frontal impact load of 1200kg force. The results show a maximum stress of $2.231 \times 10^8\ N/m^2$ which is less than the yield strength of AISI 4130 chromoly steel which $4.600 \times 10^8\ N/m^2$ as shown in the Von misses stress scale. Figure 5.2 also shows the factor of safety results from the stress analysis of the A-arm. The minimum factor of safety is 2.1, which is good enough for the design. Lastly, figure 5.1 shows the results from the fatigue analysis of the A-arm. The result of the fatigue analysis helps to predict the number of life cycles the component has before it gets totally damaged. The lower A-arm has a maximum life cycle of 2071 cycles before it reaches its point of no use.

5.1.1 Upper wishbone analysis

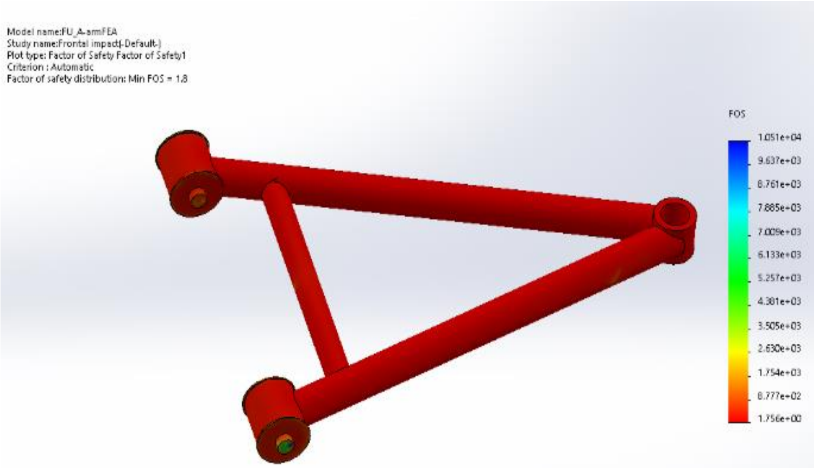

Figure 5. 4: Fatigue analysis result on upper A-arm

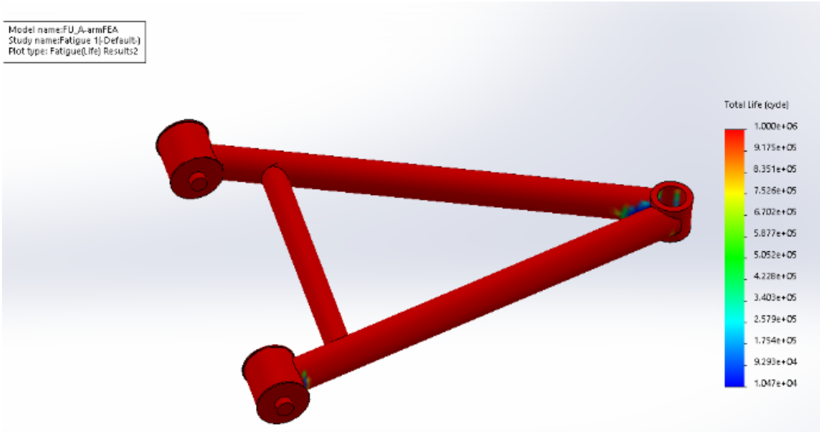

Figure 5. 5: Factor of safety analysis result on upper A-arm

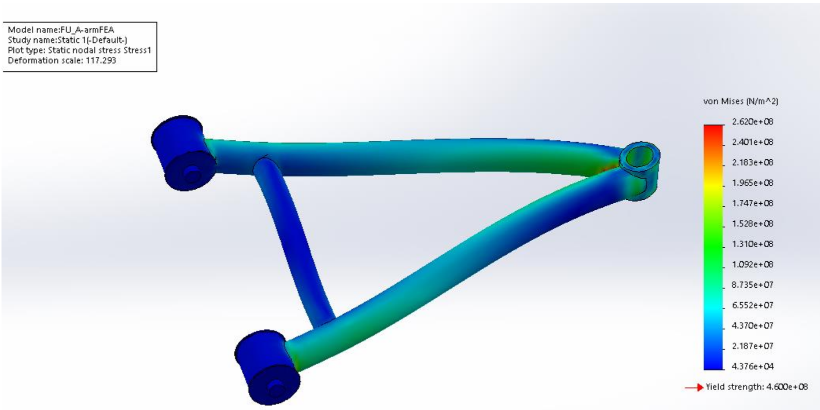

Figure 5. 6: Stress analysis result on upper A-arm (frontal impact)

Using the same force as used on the lower A-arm, the frontal impact test conducted on the upper A-arm resulted in the figures above. Figure 5.6 shows the results of the stress analysis on the upper A-arm. The maximum stress that occurs on the upper A-arm because of the applied frontal impact load results to $2.620 \times 10^{8}\ N/m^{2}$. This value is very well below the yield strength of AISI 4130 chromoly steel used as the material for the component. The factor of safety results shown in figure 5.5 identified the factor of safety of the upper A-arm to be 1.8. The results of the fatigue analysis also shown in figure 5.4 shows the total number of life cycles the upper A-arm has before getting totally damaged. The component has a life cycle of 10470, which is far greater than that of the lower A-arm when compared with each other.

5.2 Results from Hub FEA simulation

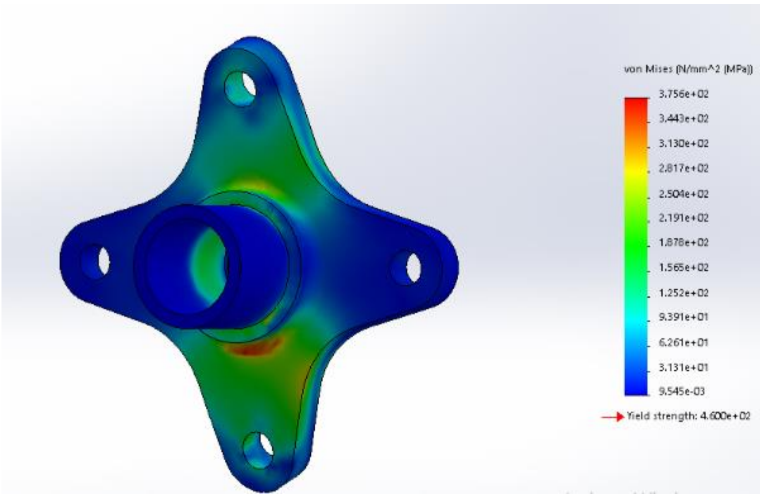

Figure 5. 7: Stress analysis result of hub

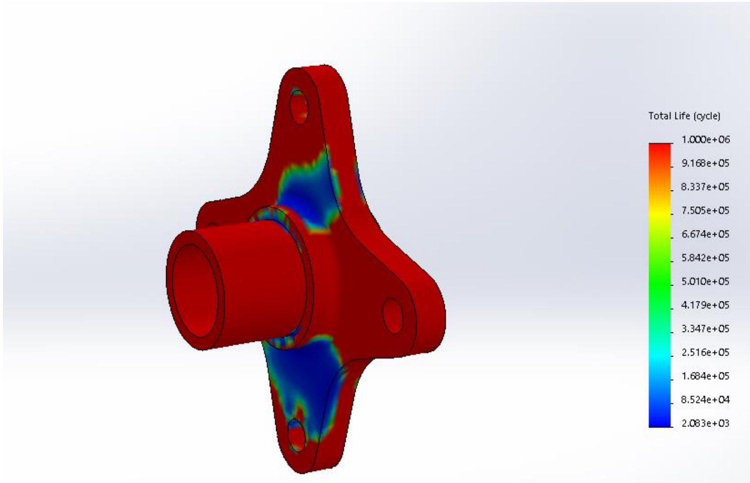

Figure 5. 8: Fatigue analysis result on hub

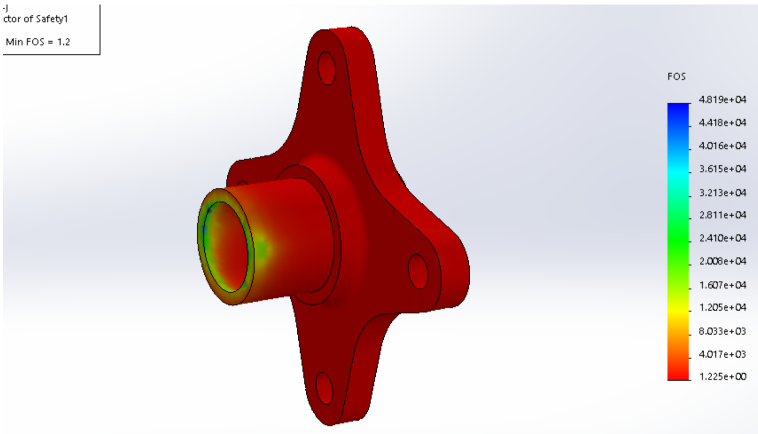

Figure 5. 9: Factor of safety result on hub

The hub was tested for strength during a bump impact experience. The maximum stress the hub experiences from the impact is $375.6MPa$ as shown on the Von Misses stress scale in figure 5.7. This maximum stress experienced by the hub does not exceed the yield strength of the material used to make it which is AISI 4130 chromoly steel. The maximum stress indicating in red on the hub body can be seen to be occurring below the part of the hub that mounts onto the upright. The minimum factor of safety the design of the hub has is 1.2 as shown in figure 5.9. The fatigue analysis results as shown in figure 5.8 shows or helps predict the total life of the hub if it continues to experience forces as that which was applied. The maximum life of the hub has been predicted to be 2083 life cycles.

5.3 Results from Upright FEA simulation

5.3.1 Front Upright Analysis

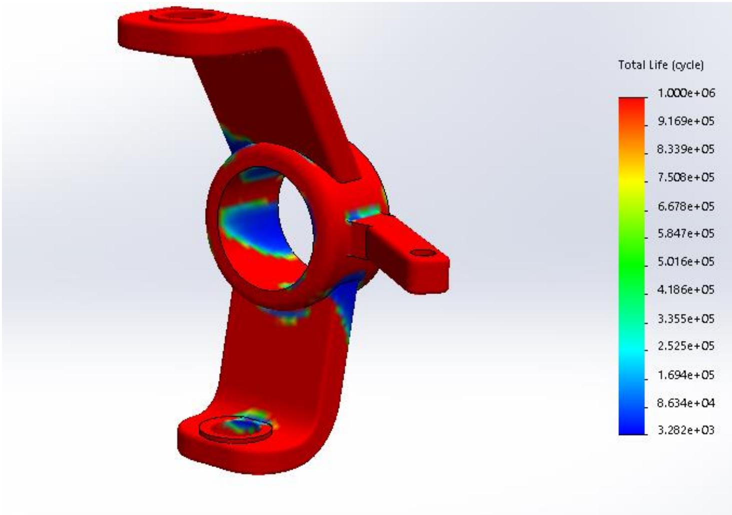

Figure 5. 10: Fatigue analysis result on front upright

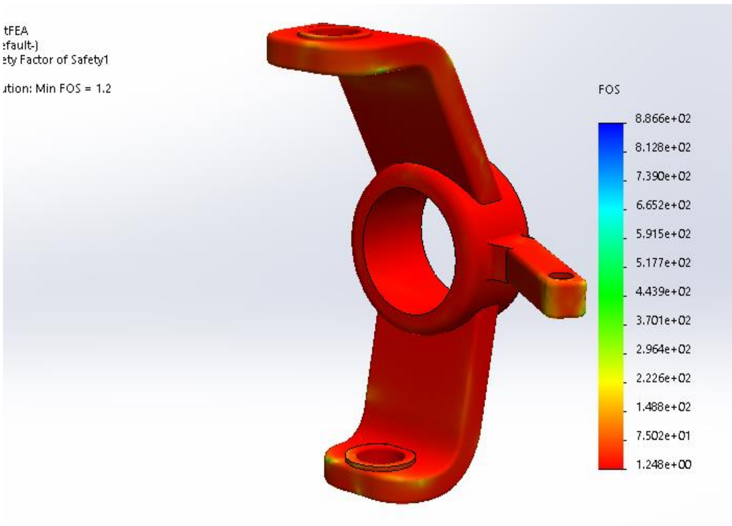

Figure 5. 11: Factor of safety result on front upright

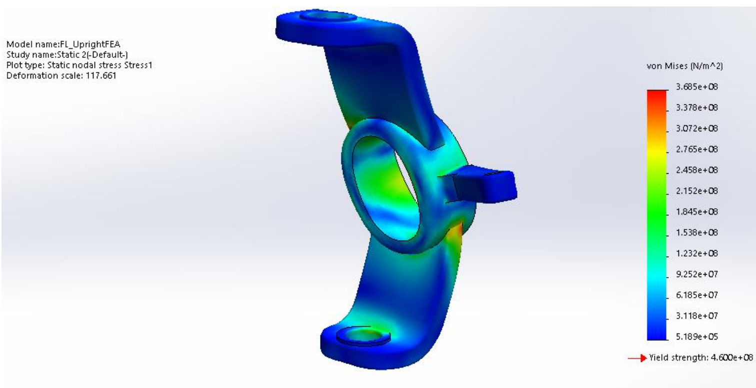

Figure 5. 12: Stress analysis result on front upright

The above figures show the results of the stress, fatigue and factor of safety analysis conducted on the front upright. Figure 5.12 shows the stress analysis results on the front upright. The maximum stress the upright experiences as shown on the Von Misses stress scale is $3.685 \times 10^{8}\ N/m^{2}$. Also seen from the scale, it shows that the maximum stress experienced by some portions of the upright does not exceed the yield strength of the material used, which is $4.600 \times 10^{8}\ N/m^{2}$. The factor of safety results generated as shown in figure 5.11 indicates that the minimum factor of safety the design of the front upright has is 1.2. The fatigue analysis results shown in figure 5.10 indicates that the maximum life the front upright has is 3282. Exceeding this number of cycles will begin excessive damage of the component.

5.3.2 Rear Upright Analysis

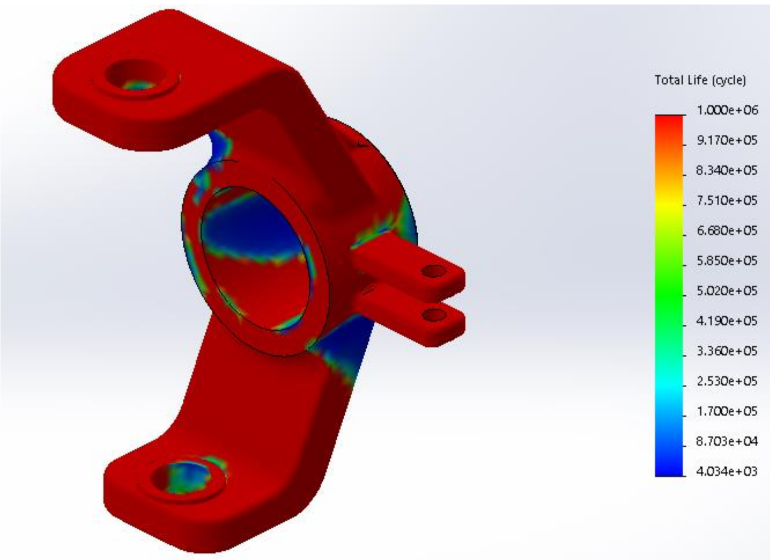

Figure 5. 13: Fatigue analysis result on rear upright

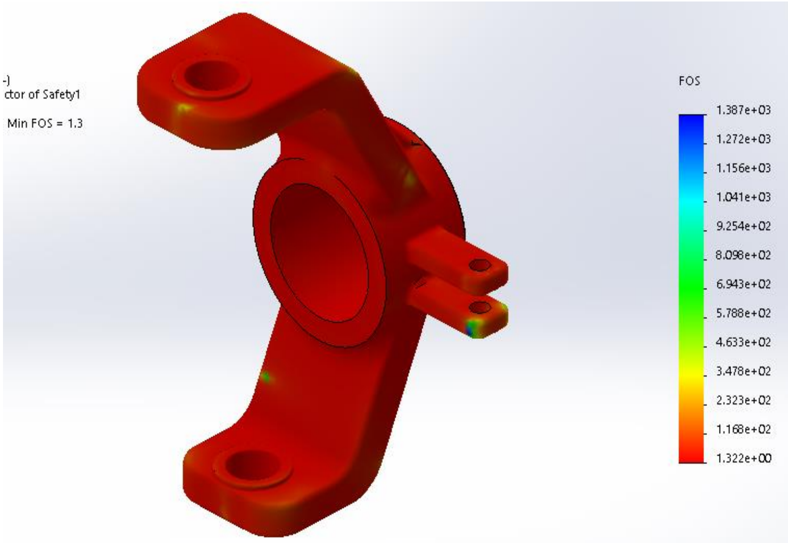

Figure 5. 14: Factor of safety result on rear upright

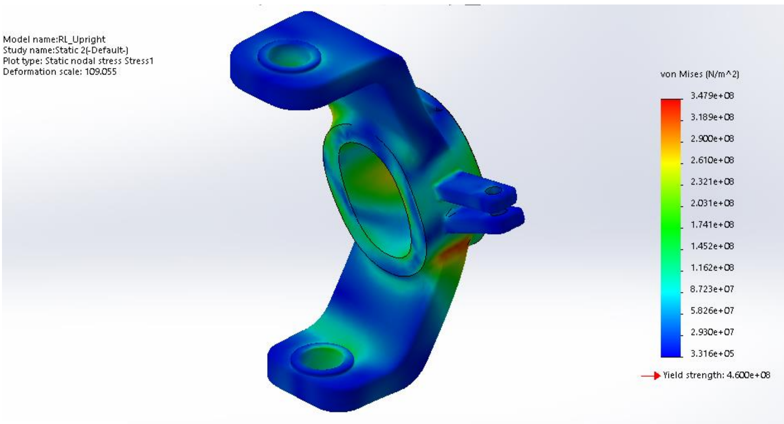

Figure 5. 15: Stress analysis result on rear upright

With regards to the rear upright which has a very similar design to the front upright, tests were carried out on it due to the difference in weight of the vehicle it experiences. The results of the stress analysis as shown on the Von Mises stress scale in figure 5.15 indicates that the maximum stress the body of the upright experiences is $3.479 \times 10^{8}\ N/m^{2}$. The minimum factor of safety for the design of the rear upright is 1.3 as shown in figure 44. The fatigue analysis results also indicate that the component has a maximum life of 4034 as shown in figure 5.13 before it begins to fail dangerously.

5.4 Results from Lotus Shark suspension simulation

This section discusses the generated kinematics results of the suspension geometries developed using Lotus Shark suspension software. From the results, it will be determined how best the objectives of the suspension systems' designs were met.

5.4.1 Camber variation result

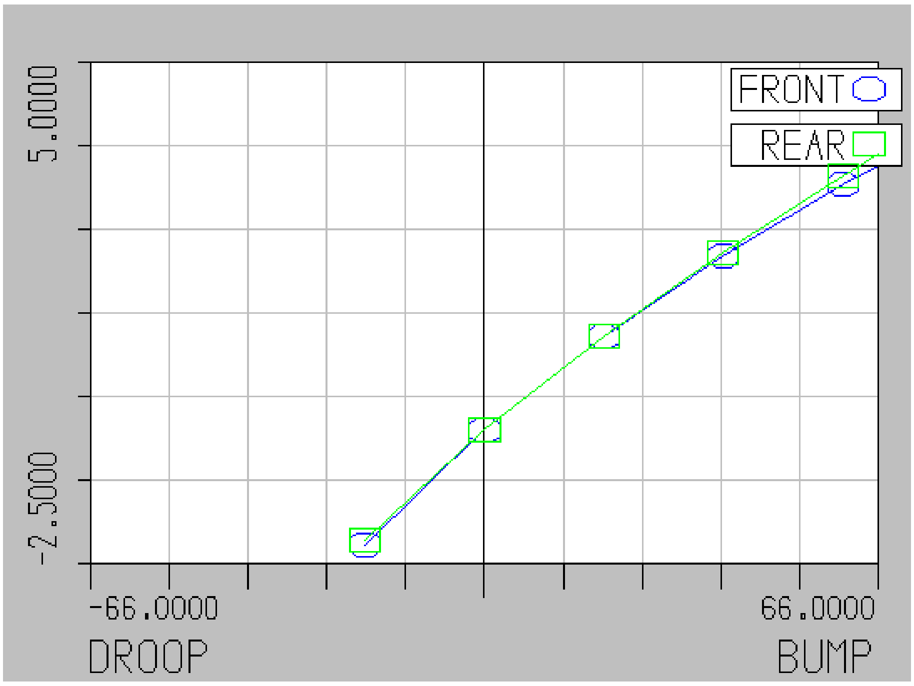

Figure 5. 16: Camber change graph

From the graph results shown above, the camber is seen to be within plus and minus 5 degrees. For the front suspension geometry in full bump, camber changes from its static orientation which is at $-0.50\,°$ to $4.11\,°$. The results also show that the front suspension in full droop/rebound gains a camber angle of $-2.24\,°$. For the rear suspension geometry in full bump, the camber gains a value of $4.36\,°$. In full droop, the camber gains a value of $-2.15\,°$.

5.4.2 Toe change result

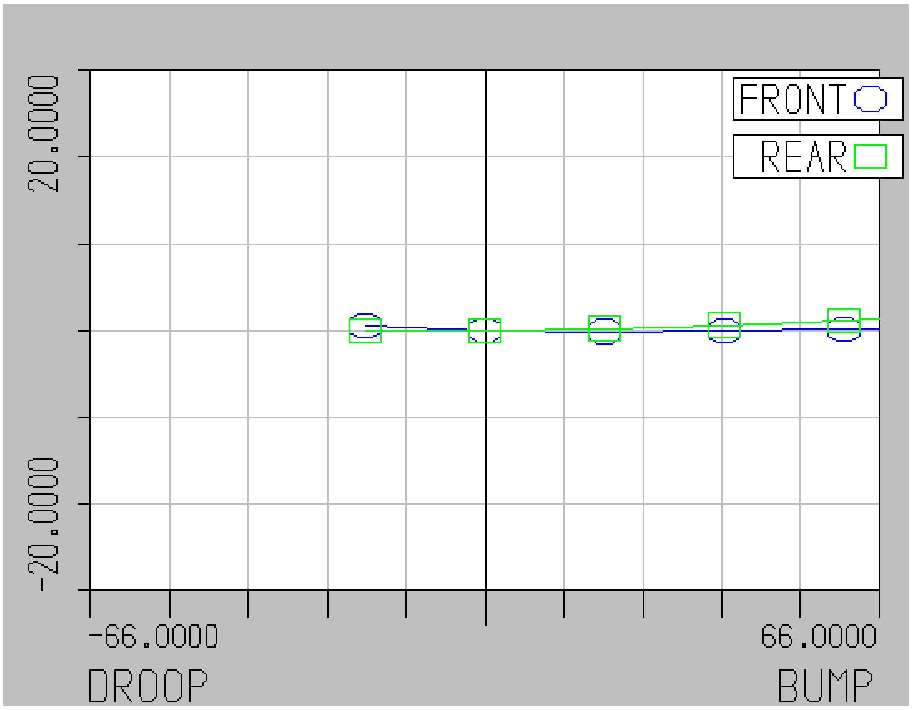

Figure 5. 17: Toe change results

Graph results for toe change show that the toe angle for the front suspension in full bump results to 0.33 °. In full droop, toe angle for the front suspension geometry becomes 0.37 °. Toe change for the front suspension also represents bump steer for the front of the car. This is minimal, making it difficult to be noticed by the naked eye. The toe change for the rear suspension geometry in full bump is 1.07 ° and in full droop is -0.05 °. The static value set for both front and rear suspension geometries was 0 °.

5.4.3 Half-Track change result

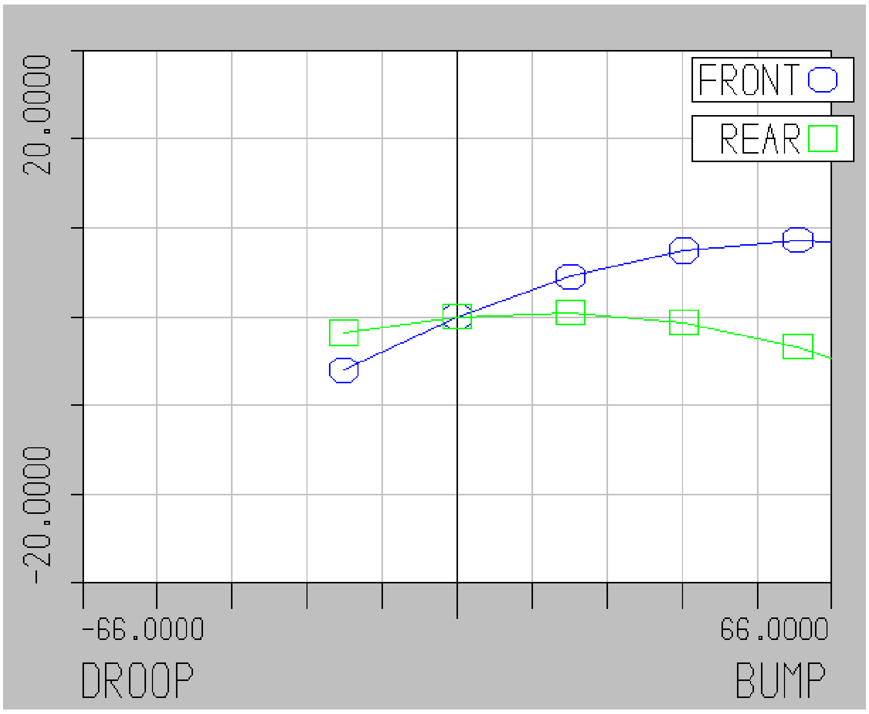

Figure 5. 18: Half Track change results

The Half Track change for the front suspension geometry in full bump as shown by the graph result in figure 48 is 5.43 inches and in full droop, the Half Track change is -3.91 inches. The Half Track change of the rear suspension geometry in full bump is -5.29 inches whereas in full droop it is -1.22 inches. The static value for half Track as identified from the graph result is 0 inches. This Half Track change is also known as the scrubbing radius of the tires. The higher the scrubbing radius, the more wear is caused to the tires of the vehicle. It is best to minimize the scrubbing radius so to slow down wearing away of the tires.

5.4.4 Roll Center Height result

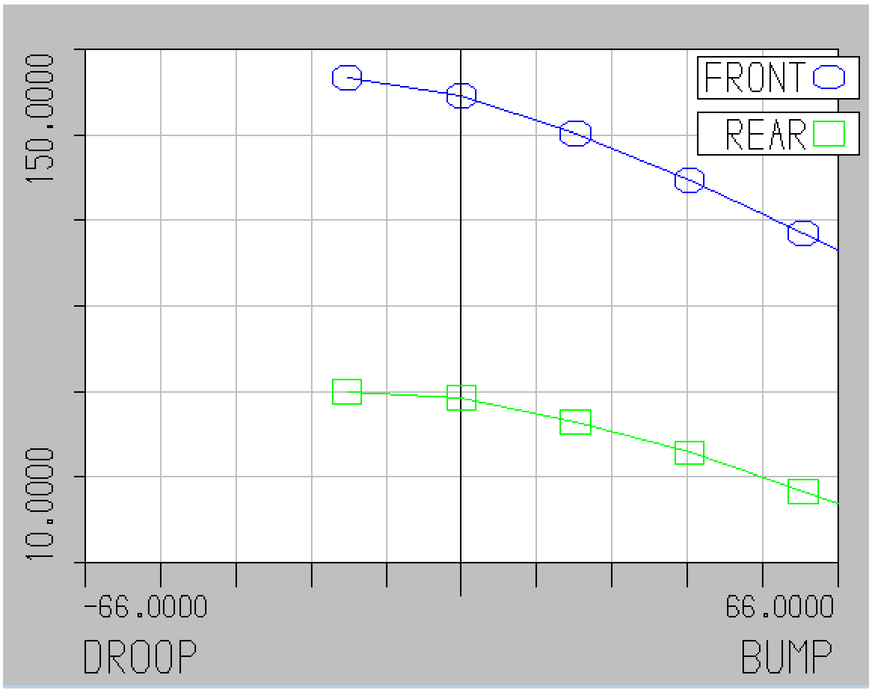

Figure 5. 19: Roll center height change results

The graph result produced for the roll center height kinematics shows that for the front suspension geometry in full bump, the roll center height becomes 86.54 inches from the ground whereas in full droop, the roll center height becomes 138.34 inches from the ground. For the rear suspension geometry, at full bump, the roll center height becomes 17.85 inches from the ground whereas that for full droop becomes 56.61 inches from the ground. The static vehicle roll center heights for front and rear are 135.74 inches and 54.87 inches from the ground respectively. In the design goals of the suspension geometries, it was desired to have the front suspension roll center height from ground higher than that of the rear, which clearly has been met.

5.4.5 Caster change result

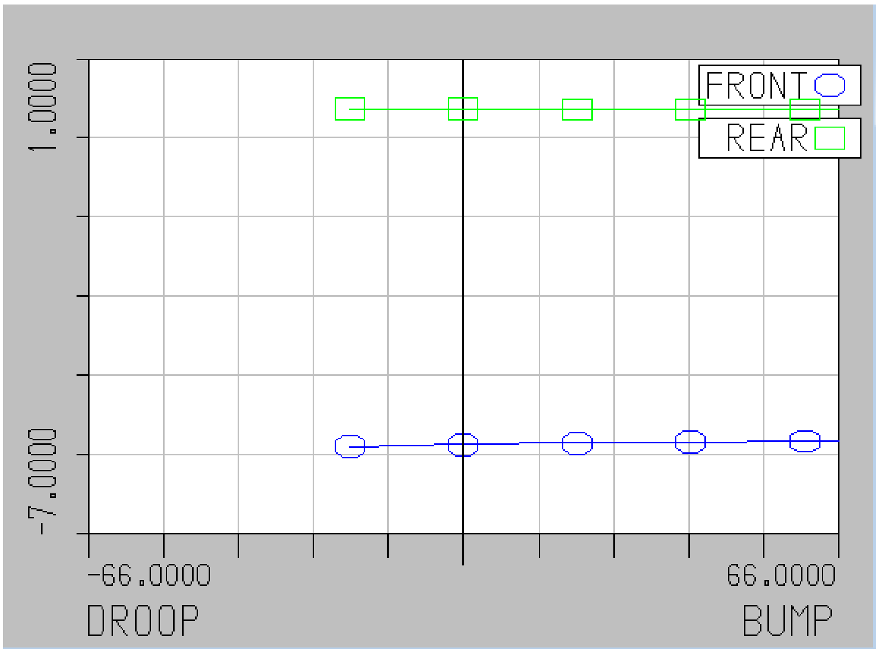

Figure 5. 20: Caster change results

Analyzing the graph results carefully, for the front suspension geometry in full bump, the caster value obtained is $-5.43\ degrees$ whereas in full droop, the caster value becomes $-5.53\ degrees$. For the rear suspension geometry in full bump, the caster value becomes $0.149\ degrees$ whereas in full droop, the caster value obtained is $0.152\ degrees$. The static caster values for front and rear suspension geometries are $-5.49\ degrees$ and $0.15\ degrees$ respectively.

5.4.6 Kingpin angle variation result

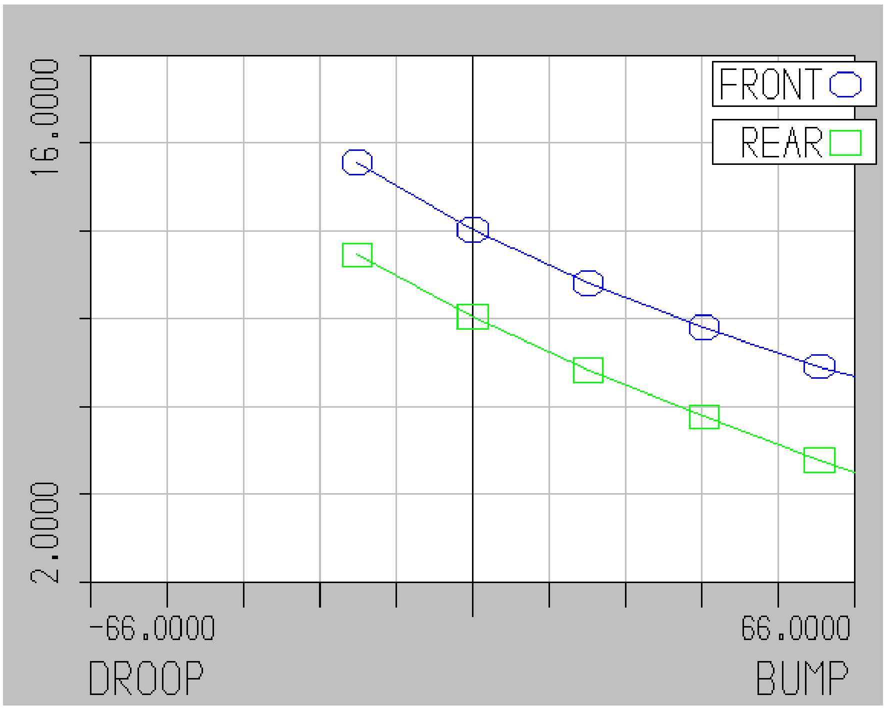

Figure 5. 21: Kingpin inclination change results

Analyzing the graph results for the kingpin angle kinematics shows that for the front suspension in full bump, resultant kingpin angle becomes $6.78\ degrees$ whereas in full droop, the resultant kingpin angle is $13.15\ degrees$. For the rear suspension geometry in full bump, the resultant kingpin angle is $4.18\ degrees$ whereas in full droop, the resultant kingpin angle is $10.69\ degrees$. The static kingpin angles of front and rear suspension geometries are $11.37\ degrees$ and $9.04\ degrees$ respectively.

5.5 Results from MATLAB Quarter car simulation

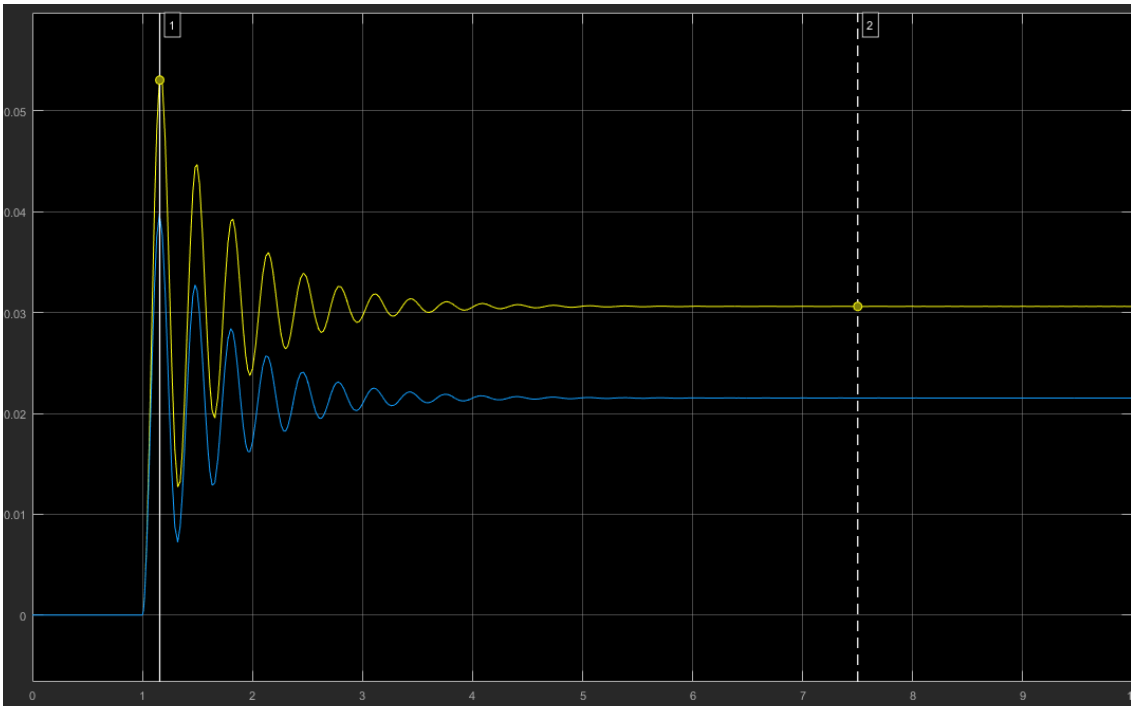

Figure 5. 22: Oscillation results of sprung and unsprung mass

The results above generated from MATLAB Simulink shows the oscillation the sprung and unsprung mass undergo when they encounter a bump. The yellow oscillatory line indicates the sprung mass whereas the blue oscillatory line indicates the unsprung mass. Analyzing the graph carefully, it can be seen how quickly both masses settle after vibrating for a while when the vehicle encounters a bump.

Chapter 6: Conclusion

6.1 Discussion

Based on the results derived from all the tests that were conducted, a decision had to be made on how well designed the front and rear suspension systems are typically for off-road use since it is being designed for the mini Baja vehicle. Results from the FEA of the suspension member elements which are the upper and lower wishbones, wheel hub, and front and rear uprights show that the design of those components pass for manufacturing. This conclusion is being made because all these components or parts were simulated based on worst-case scenarios that might occur on the suspension system and the results from the stress analysis of each component individually shows that the maximum stresses that occurred on the members did not exceed the yield strength of the material used. This shows that the design of each component is safe and will withstand heavy forces that act on it all the time. It is also safe say that AISI 4130 chromyl steel is a good choice for manufacturing the member elements of an off-road suspension system since its yield strength remained above the maximum stresses acting on all simulated bodies. Fatigue analysis also showed that the total life for the hub, wishbones, and uprights all exist within the range of 1000 to 10000 cycles. The mini Baja vehicle is not driven daily as commercial vehicles are driven hence, having a total life within this range is good enough for the vehicle. The suspension system is being designed for a vehicle that is used during a competition, therefore having a total life that falls within that obtained range is good for the suspension system.

The goal of the suspension system design was to improve handling, traction, maneuverability through the courses, and its ability to isolate the sprung mass from the terrain which the vehicle navigates. Talking on traction first, having a high ratio of sprung to unsprung weight passes as best for vehicle traction [4]. This is because a higher proportion of the sprung weight can push down on the wheels and tires with more force. This will in return force the tires to have more contact with whatever surface it finds itself travelling on. The suspension system in this paper was designed with a sprung mass of 270kg and an unsprung mass of 80kg. Since the weight of the sprung mass is far greater than that of the unsprung mass, it improves the vehicle's traction. This provides good traction for the vehicle. Since traction plays a role in the handling characteristics of a vehicle, designing the suspension system to have an unsprung weight lower than the sprung weight results in better handling characteristics of the vehicle.

Discussing the kinematics results obtained, minimal toe changes have occurred for both front and rear wheels. Toe affects three major areas of performance which are tire wear, straight-line stability and corner entry handling [15]. The design goal was always to keep toe at 0 degrees but having the changes of toe occur will require that some adjustment must be made to maximize straight-line stability of vehicle. The toe changes that occur is very minimal and hence handling remains intact. Camber changes that are shown by the results to occur are generally accepted. Ideally, it is best to design the suspension such that the wheel camber relative to the chassis becomes increasingly negative as the suspension deflects upward [15]. The results generated for camber changes fits well into the situation in the previous statement. This plays a good role in the maneuverability of the vehicle through the course, especially when cornering. The roll center result is as desired since the design goal was to get the front to have a higher roll center than the rear to promote oversteer of the vehicle when cornering. This will allow less effort to be applied when steering in a corner. This aids in the maneuverability of the vehicle throughout the course.

The track width and the wheelbase selected with respect to the limitations given by SAE Baja allow the vehicle to maneuver through the course easily.

Overall, the performance cannot be rated just yet because it has not been compared to an already existing, fully function Baja suspension system that has been used and survived the Baja competition course. This however does not mean it is not a good design for a Baja suspension system. Also, the design goals cannot fully be checked since the vehicle was not built and tested. Therefore, a conclusion on the performance of the cannot yet be made.

6.2 Limitations

A major limitation to this project was the fact that it was a one-man job and therefore the manufacturing part of the project had to be taken out to safe time. Also, even if the Baja was to be built, the parts are relatively expensive. Since there was a limited amount of funding given by the school, the building aspect of the project was not considered. The inability to build the vehicle did not allow for real-world testing of the suspension design to check if the design goals were met or not.

6.3 Future works

Plans include reviewing the entire design and making sure the designed suspension system when manufactured, will perform as expected.

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Glossary

Definitions of terms used in this paper:

Wheel base: The distance between the center of the front and rear tires of a vehicle as viewed from the side of the vehicle.

Track width: The measure of the center distance between the two front wheels or the two rear wheels.

Half Track: This is the distance that exist between the center of the tires and the center of the vehicle as viewed from the front or rear of the vehicle.

Spring rate: Refers to the force per unit of displacement of the spring or shock absorber.

Roll center: Refers to the point which the suspension system rotates around in an instance.

Roll axis: The axis that connects the front and rear roll centers which the vehicle rotates around in an instance.

Oversteer: when lateral acceleration on the center of gravity of the vehicle causes the rear wheels to slip more than the front.

Understeer: when lateral acceleration on the center of gravity of the vehicle causes the front wheels to slip more than the rear wheels.

Unsprung mass: the mass of the vehicle components between suspension and road surface (upright, hub, wheels, etc.)

Sprung mass: mass of the vehicle that rides on the suspension system i.e. chassis, driver, and all other components of the vehicle.

Travel: measure of the distance from the bottom of the suspension stroke to the top of the suspension stroke. The distance which the bottoming or lifting of the wheels can reach.

Camber angle: refers to the angle between the vertical axis of the wheel and the vertical axis of the vehicle when viewed from the front or rear.

Caster angle: refers to the angular displacement from the vertical axis of the suspension of a steered wheel in a vehicle, measured in a longitudinal direction.

Toe angle: the angle of the tires when viewed from the top view of the vehicle relative to the longitudinal axis of the vehicle.

Kingpin inclination: the angle which arises between the steering axis and a vertical axis to the road.

Bump: refers to the vertical movement of the wheel up to the chassis. The amount of upward travel the wheel can go before it cannot go anymore.

Droop/rebound: refers to the vertical movement of the wheel down from the chassis. The amount of downward travel that the coil-over can do before bottoming out.

Bump steer: change in steering angle when the wheel is in a bump or droop without the driver having to turn the steering wheel or any form if lateral movement occurring in the steering rack.

Ground clearance: Distance from the lowest point of the vehicle body to the ground.

FEA: refers to the simulation of any given physical phenomenon using the numerical technique called Finite Element Method (FEM).

CAD: Computer-Aided Design (CAD) refers to a computer technology that designs a product and documents the design's process [16].

Longitudinal force: refers to a force the tire receives from the road that acts along the X axis [17].

Lateral force: refers to a force the tire receives from the road that acts along the Y axis [17].

Normal force: refers to a force the tire receives from the road that acts along the Z axis. [17]

Appendix A

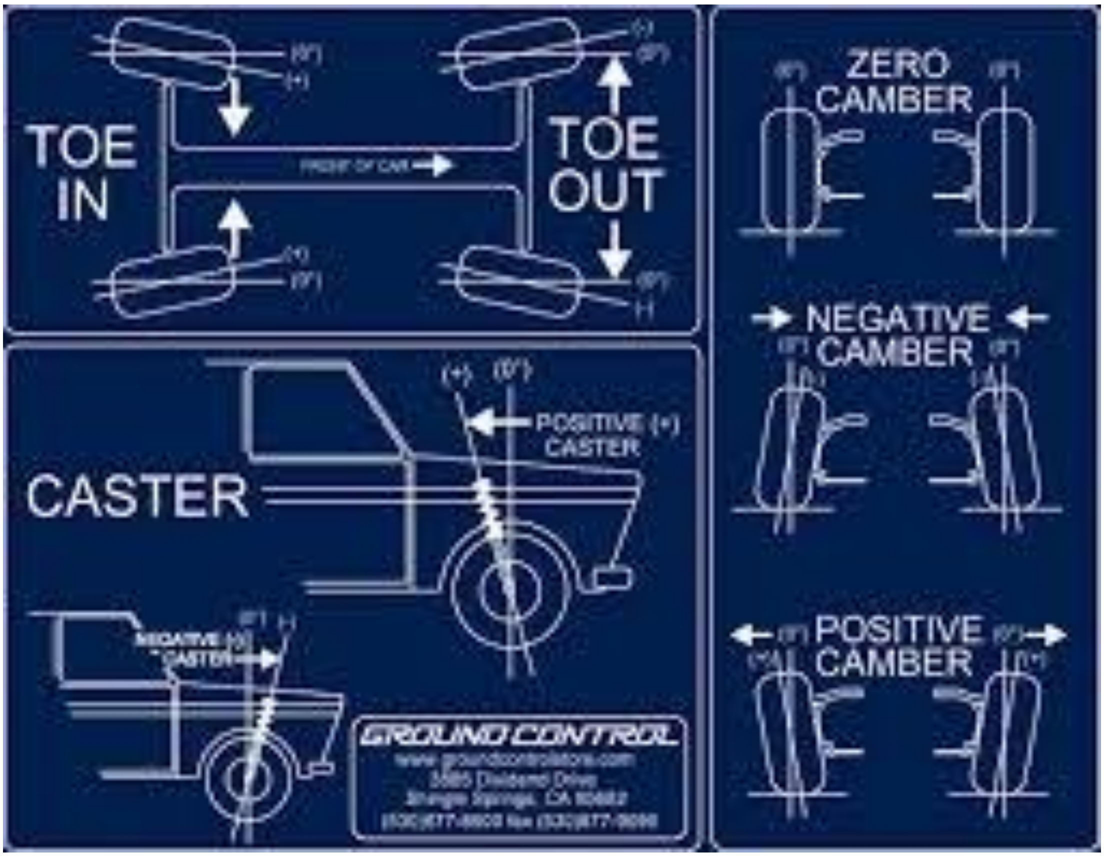

Figure A. 1: Camber, Toe and Caster visual representation

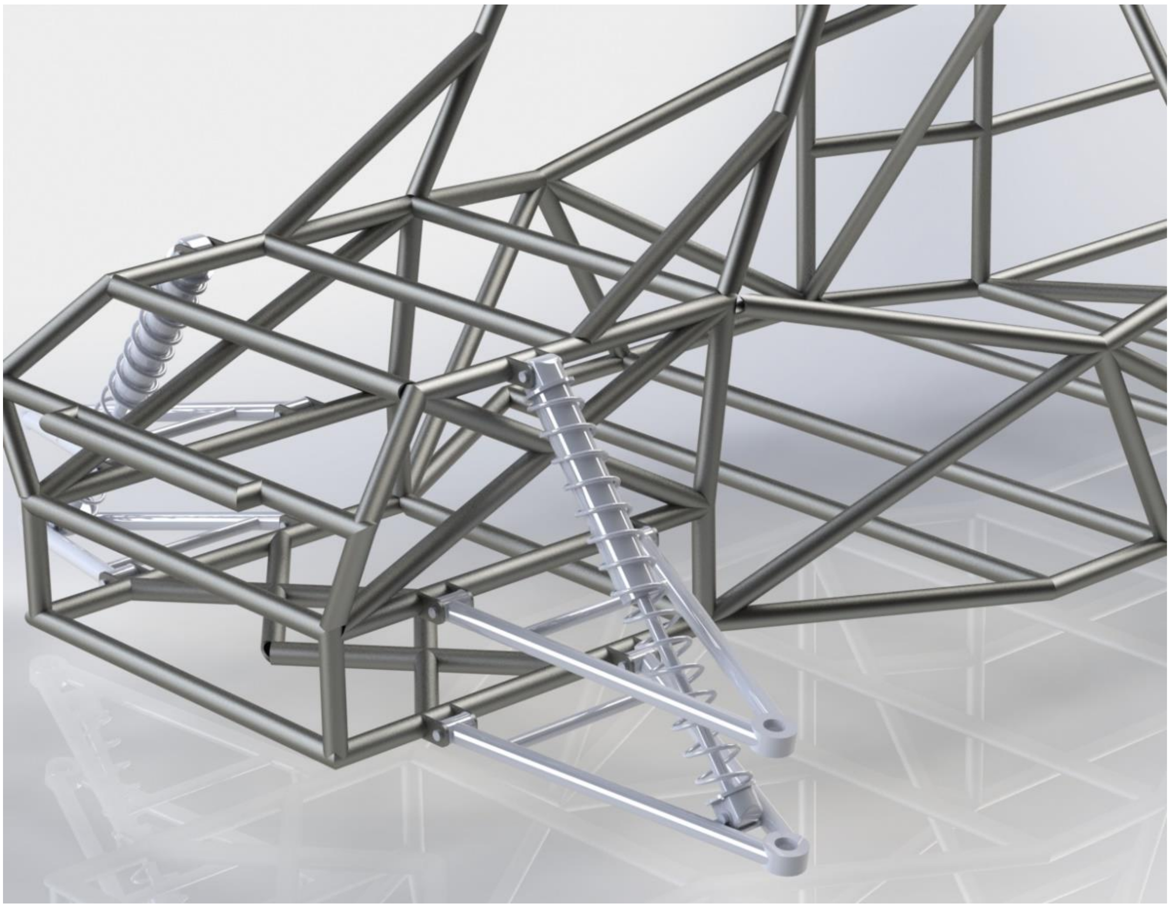

Figure A. 2: Designed front double wishbone suspension system with damper mounting on lower A-arm

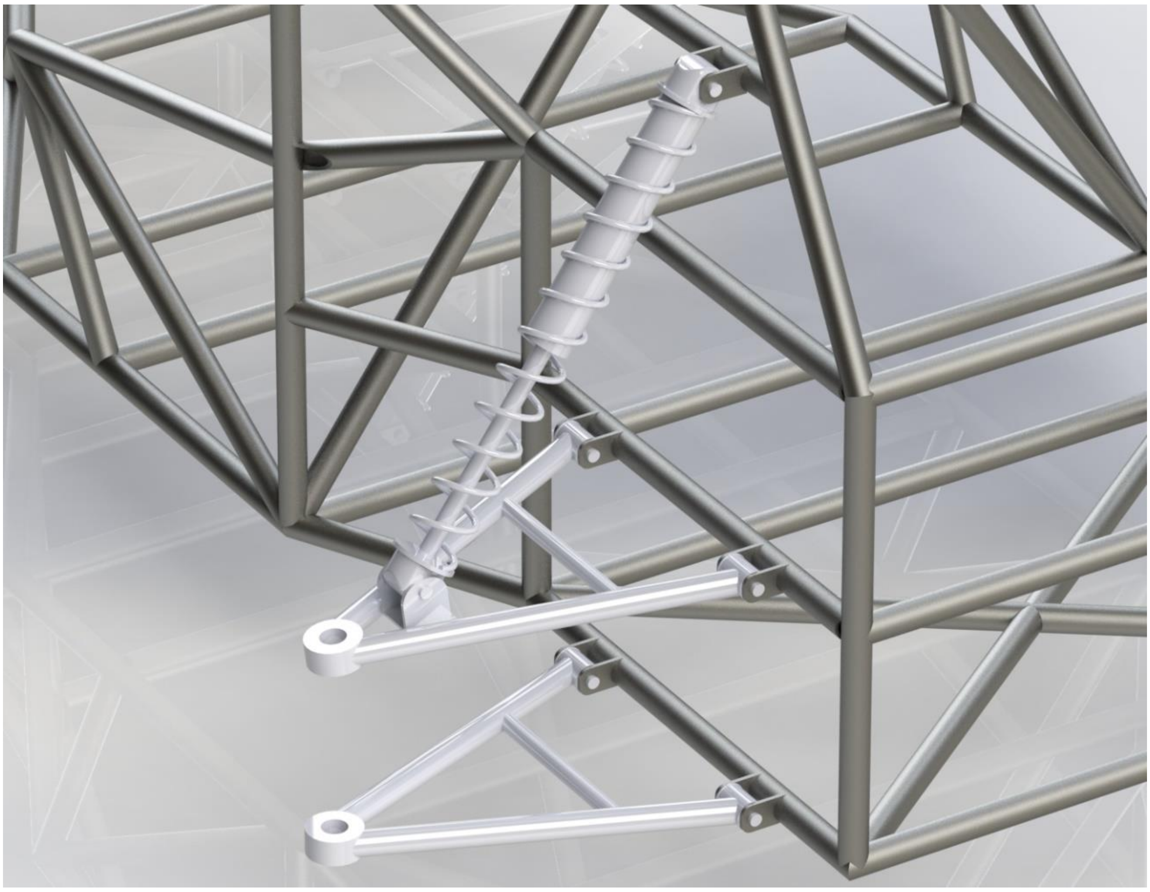

Figure A. 3: Designed rear double wishbone suspension system with damper mounting on upper A-arm

Figure A. 4: Top view of Baja with designed suspension system mounted on chassis

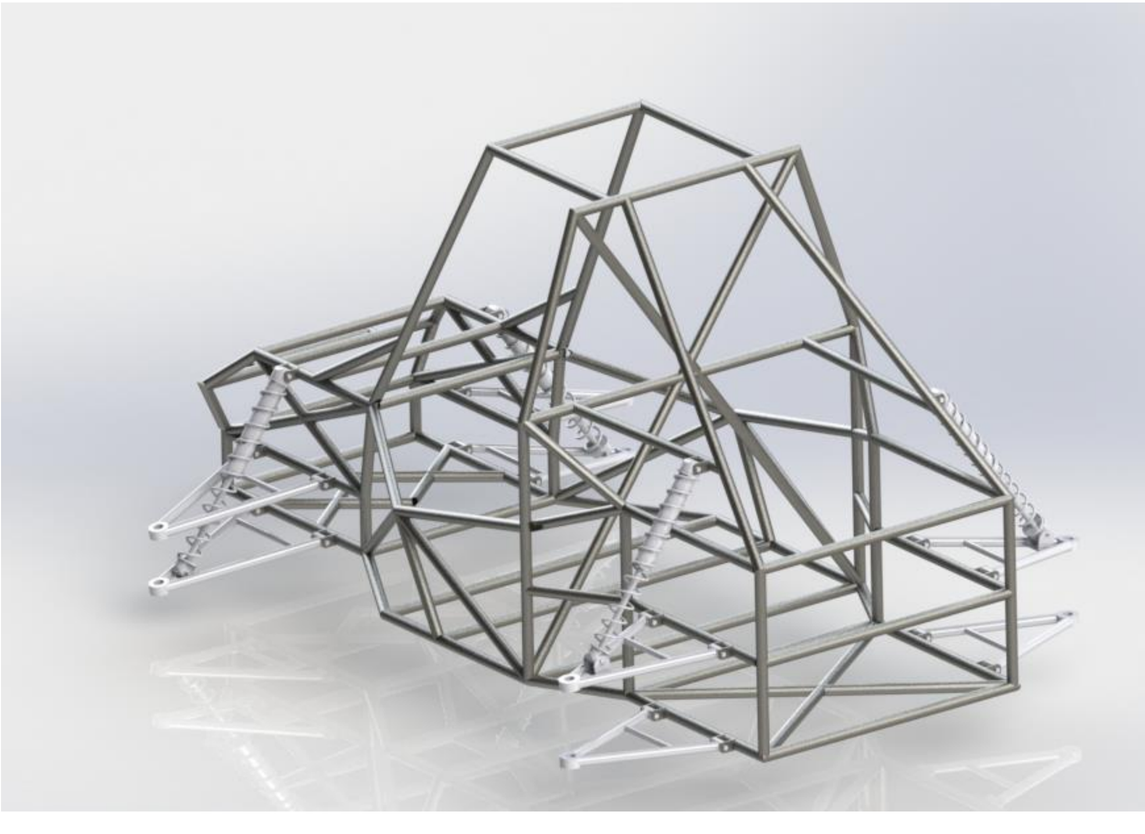

Figure A. 5: Isometric view of Baja showing suspension system mounted on chassis