

# Design Review of Suspension Assembly of a BAJA ATV

Sumit Sharma<sup>1</sup>

<sup>1</sup>Department of Electrical Engineering, Rungta College of Engineering and Technology, Bhilai, Chhattisgarh, India

\*\*\*

**Abstract** - All Terrain Vehicle is designed to move through all terrain. Suspension system of this vehicle should be strong to provide better ride and handling with better comfort. Independent suspension systems are required for this purpose. It is designed using LSSA (Lotus Shark Suspension Analysis). After designing in Lotus, with the hard points the A-arms, front and rear uprights are then designed in CATIA and analyzed it by using ANSYS. It is the system that connects the wheels with chassis by means of an assembly, which provides firmness necessary to compensate shocks from roads. Suspension system defines roll/body angle, effortless steering, camber characteristics and many other factors. The suspension system must be quite rigid in order to bear sudden shocks due to drop, sudden dump etc. The suspension systems of vehicle support in maneuverability and driver's comfort. The suspension designing should withstand ruggedness, driver's fatigue-less driving.

**Key Words:** Suspension system, Lotus, Shocks, Double wishbone.

## 1. INTRODUCTION

The primary functions of a vehicle's suspension systems are to isolate the structure and the occupants from shocks and vibrations generated by the road surface. It is obvious that a suspension system must be able to withstand the loads acting on it. These forces may be in the longitudinal direction such as acceleration and braking forces, in the lateral direction such as cornering forces, and in the vertical direction.

The suspension system of BAJA vehicle is to absorb vibration due to irregularities of road conditions that require greater stiffness and tough/rigid body. Greater stiffness creates fatigue to the driver. So springing action will be the major factor behind designing. The choice of right type and its major benefit will play a major role in winning. So it is equally important to specify the correct type and validate it in software. Furthermore, it also designed to maximize the friction between tire contact patch and the road surfaces to provide vehicle stability under any circumstance associated with accelerating, braking, loaded or unloaded, uneven road, straight line or cornering. The suspension system significantly affects ride and handling of the vehicle that is 'vibrational' behavior including ride comfort, directional stability, steering characteristics and road holding. Generally, suspension system can be broadly classified as dependent and independent types.

The factor which primarily affect the choice of suspension type at the front or rear of the vehicle are the engine location and whether the front wheels are driven/undriven and steered/unsteered.

Independent suspension (i.e. double wishbone, McPherson and multilink) leads to better ride and handling capabilities. It is important to analyze the suspension systems that have been designed to predict the behavior of the system than followed with improvements.

Out of these available design, double wishbone type suspension is selected which gives adequate space for wheel assembly, better control over design characteristic and flexibility to designer in terms of performance variables.

### 1.1 Objective of Suspension System

It is important to analyze the suspension systems that have been designed to predict the behavior of the system than followed with improvements.

The suspension must be properly designed because it is a crucial subsystem in vehicle in order to:

- Carry the weight of vehicle and also its weight (unsprung weight).
- Keep the wheel perpendicular to the road for maximum grip resultant good ride and handling performance.
- Take the forces for accelerating or braking the vehicle.
- Ensure that steering control is maintained during maneuvering.
- Take the forces involved when cornering the vehicle.

### 1.2 Need of Double Wishbone Suspension System

In automobiles, a double wishbone (or upper lower A-arm) suspension is an independent suspension design using two (occasionally parallel) wishbone-shaped arms locate the wheel. each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. The shock absorber and coil springs mount to the wishbone to control vertical movement. Double wishbone design allow the engineer to carefully control the motion of the wheel throughout suspension travel, controlling such parameters as camber angle, caster angle, toe pattern, roll center height, scrub radius and more.

### 1.3 Concepts Related to Suspension System

**Camber:** Camber angle is the angle made by the wheels of a vehicle; specifically, it is the angle between the vertical axis of the wheels used for steering and the vertical axis of the vehicle when viewed from the front or rear. Camber angle can have both negative as well as positive orientation. Camber is considered positive if the top of the wheel leans outwards and negative if the top of the wheel leans inwards.

**Castor:** It is the angle between the pivot line (in a car an imaginary line that runs through the center of the upper ball joint to the center of the lower ball joint) and vertical.

**King Pin Inclination:** Kingpin inclination is defined as the angle at which kingpin axis is inclined to the vertical axis passing through the wheel center. Kingpin angle affects the performance of the car when the wheels are steered. More the kingpin angle more the car will lift when steered. When the kingpin axis is extended up to the ground, it intersects the ground at a particular point. The distance of that point from the center of wheel contact patch is called scrub radius. Scrub radius increases the wear of the tire but some amount of negative scrub radius is required so that the wheel purely rolls when steered.

**Roll Center:** The Roll Centre of a suspension system is that point in the transverse plane of the axles, about which the sprung mass of that end of the vehicle will roll under the influence of that end of the vehicle will roll under the influence of centrifugal force.

**Toe:** Toe is final parameter used to describe a vehicle's alignment. Toe is the symmetric angle that each wheel makes with the longitudinal axis of the vehicle, as a function of static geometry, kinematic and compliant effects. Tire wear is heavily dependent on the toe distances. Dynamic factors induce change in toe, which lead to two conditions- toe in and toe out. In rear wheel drive cars, increased front toe in provides greater straight-line stability at the cost of some sluggishness in steering response.

**Table -1.1:** Essential parameters related to suspension system

S.N O.	PARAMETERS	VALUES	
1.	Wheelbase	56"	
2.	Track-width	Front	52"
		Rear	52"
3.	Ground Clearance	13"	
4.	Kingpin Inclination	18.16 degree	
5.	Kingpin Offset(ground)	8.408 mm	

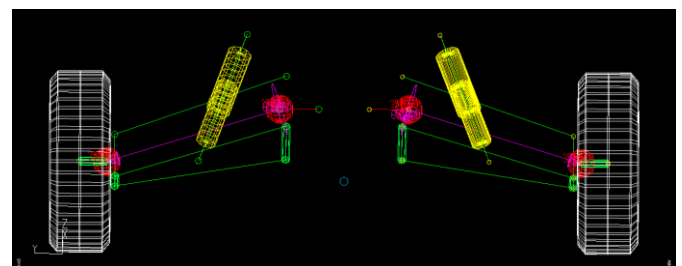
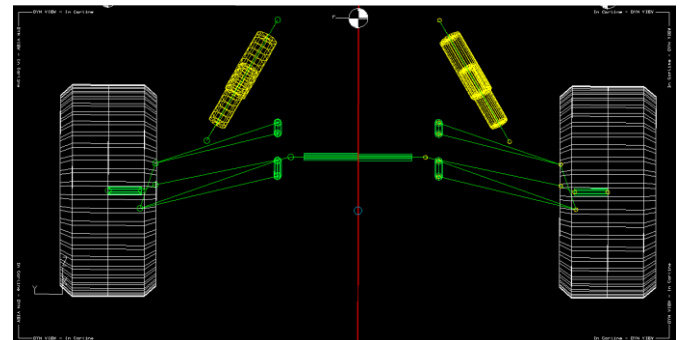
6.	Castor Angle	0 degree	
7.	Camber	0 degree	
8.	Tire Dimension	Front	21*7R10
		Rear	22*8R10

**Table -1.2:** Final geometry parameter

PARAMETER		VALUES
Upper wishbone	Length	354.9 mm
	Angle	25.2 degree
Lower wishbone	Length	385.04 mm
	Angle	24.9 degree
Roll center height		226.14 mm

### 2. DESIGN METHODOLOGY

The design is completed in LOTUS SHARK suspension designing software system. The calculations are made concerning the analysis papers. The design is formed keeping the anti-dive properties to the minimum to stay the ATV stable altogether conditions. Before designing the suspension of vehicle we have concentrate upon a number of the essential parameters of the vehicle needed for it.



**Fig -2.1:** Lotus Shark Suspension Analysis

LOTUS Shark suspension analysis tool is employed for initial calculation of the vehicle. 3D moving models are often creating and change in LOTUS Shark Suspension Analysis (LSSA). Using LSSA hard points are drawn and graphical and numerical values are found out. This modelling approach permits user to form their own suspension models and work on them simply. The changes in any parameters can be find out camber angle, toe angle are often displayed diagrammatically against motions like roll motion, bump motion, steering motion. many parameters are thought of to

induce the hard points of the suspension system like damping magnitude relation, sprung and un-sprung weight, spring rate, camber angle, caster angle, roll center, wheelbase, track dimension, toe angle, ground clearance. So before coming up with in LSA we've got design considerations:

- 1) Kingpin and caster angle are kept in such a way that they'll compensate every others camber gain well, by providing their individual function.
- 2) A positive king pin angle is kept to assist in steering the vehicle. Roll center below CG to avoid unwanted force.
- 3) Front ride frequency is more than rear portion.
- 4) Roll axis inclined towards front portion to present understeer characteristic to the vehicle.
- 5) Front double wishbone unequal para Camber angle.

We have taken Damper to upper wishbone for the front suspension and damper to lower H-arm in rear suspension system. We have design by using these parameters and additionally by checking various properties like roll motion roll center, bump motion, steering motion. These properties are controlled by controlling the camber, caster, toe, kingpin angle etc.

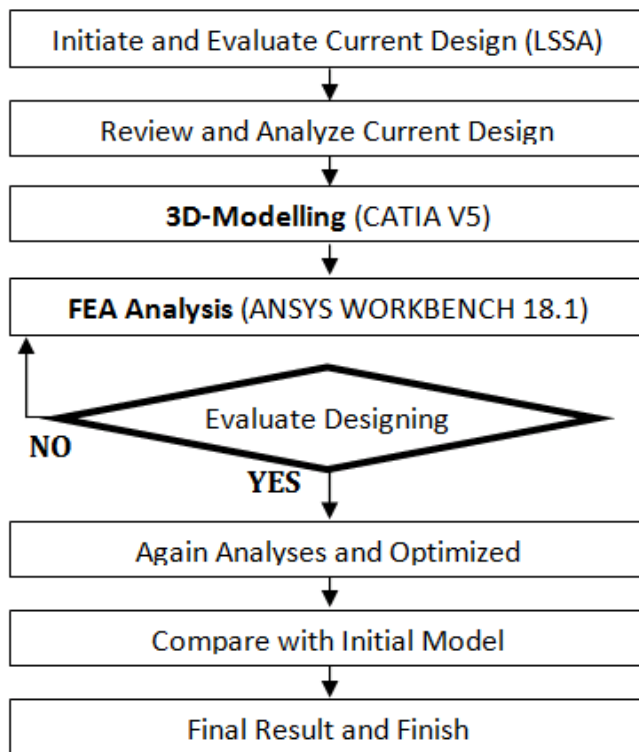


Fig. 2.2: Design Methodology Process Flowchart

### 2.1 Design of Wishbone

Design of the suspension was administered in computer aided designing (CAD) using CATIA for designing purpose. Design of wishbones is that the necessary step to construct a

suspension system. Initially, for designing we've located coordinates from Lotus.

### Front Suspension

Double wishbone suspension system of unequal length and parallel was implemented for Front suspension additionally referred to as short long arms (SLA). We tend to finalize the form of wishbone to be an arms because it will distribute stresses over the members effectively and in a very proper method. Upper wishbone is shorter than lower wishbone. The advantage of getting completely different lengths is that once the vehicle takes a flip a negative camber is induced that will increase the stability.

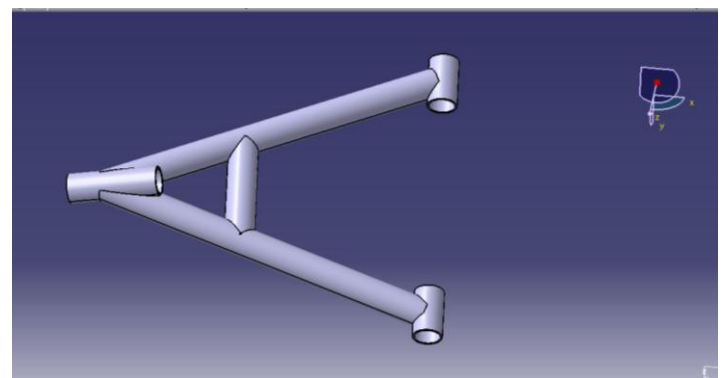


Fig. 2.3: Front Wishbone

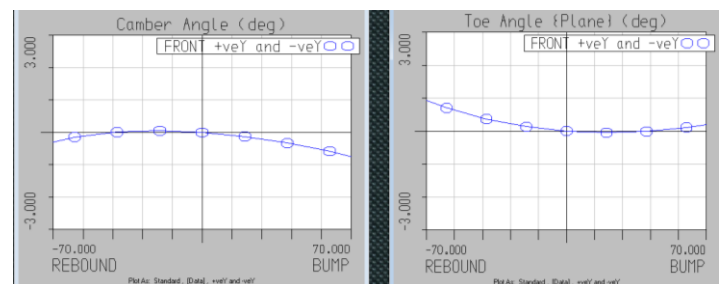


Fig. 2.4: Camber angle change

Fig. 2.5: Toe angle change

### Rear Suspension

It consists of equal H-arms and I-arm design. Independent H-arm are widely accepted as camber changes is simply eliminated and controlled. One end is connected to the chassis and alternative end is connected to the knuckle. A H-arm provides great deal of travel and principally equal the front suspension. At rear side the camber angle and the toe angle can be zero. As we are having damper to lower arm in rear side therefore the upper arms are of I -shape and therefore the rear lower arms can be of h-shape. As we've camber angle 0° that makes our front suspension equal and parallel. we'll mount the spring within the higher stiffness which is able to be tilted towards the axle side.

### Uprights

Upright is design by CATIA software by using the right tire dimensions. It's designed in such a way that it may hold the upper and lower pivot points of the arms and therefore the

output shaft long with the hub and brake callipers without any difficulties. After this it's tested by collecting all the remaining elements of the suspension system.

Properties	
Volume	2.8589e+005 mm <sup>3</sup>
Mass	2.2442 kg
Scale Factor Value	1.

Table 2.1: Upright Material Properties

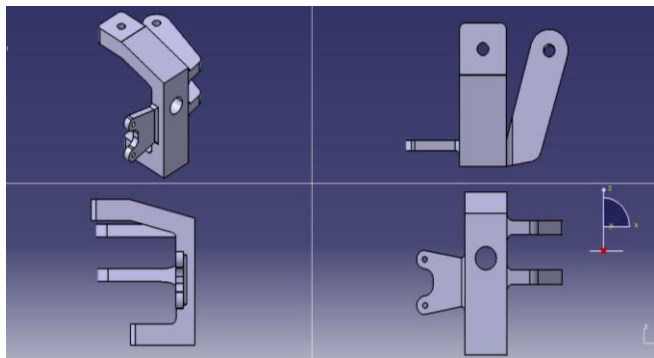


Fig. 2.6: Upright different directional view

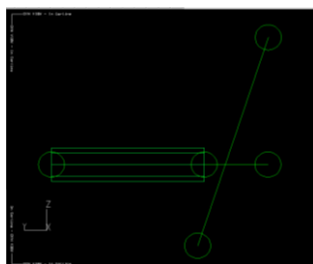


Fig. 2.7: Upright (Pivot Point)

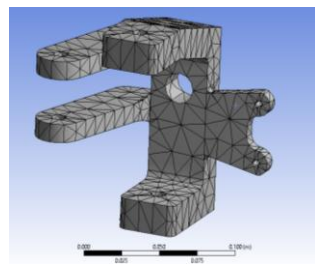


Fig. 2.8: Upright (Mess Modal)

### 3. MATERIAL SELECTION

Material consideration for the wishbone becomes the most primary need for design and fabrication. The strength of the material should be well enough to withstand all the loads acting on it in dynamic conditions. The material selection also depends on number of factors such as carbon content, material properties, availability and the most important parameter is the cost.

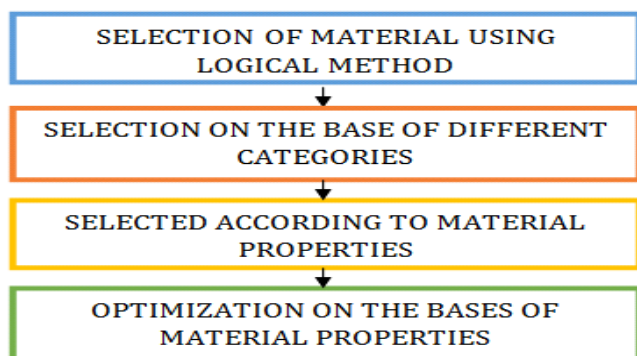


Fig. 3.1: Material Selection Process Flowchart

Physical properties	Steel AISI 1018 Properties	Steel AISI 4130 Chromoly alloy
Density	7861kg/m <sup>3</sup>	7861kg/m <sup>3</sup>
Ultimate Tensile Strength	4398.85*10 <sup>5</sup> N/m <sup>2</sup>	6701.70*10 <sup>5</sup> N/m <sup>2</sup>
Yield Tensile Strength	3702.48*10 <sup>5</sup> N/m <sup>2</sup>	4350.59*10 <sup>5</sup> N/m <sup>2</sup>
Modulus of Elasticity	199.94*10 <sup>5</sup> N/m <sup>2</sup>	2047.74*10 <sup>5</sup> N/m <sup>2</sup>
Bulk Modulus	1399.63*10 <sup>5</sup> N/m <sup>2</sup>	1399.63*10 <sup>5</sup> N/m <sup>2</sup>
Elongation Break	15%	25.5%

Table.3.2: Mechanical properties of Steel AISI 4130 chromoly alloy

## 4. IMPACT FORCES CALCULATION OF FRONT SUSPENSION SYSTEM OF ATV BAJA VEHICLE

### 4.1. Impact force determination by speed limit (for static analysis)

(According to the rule book. The maximum speed of the car is assumed to be 60km/h around 16.66m/s)

For the elastic collision,

$$W = 1/2mv^2_{final} = 1/2mv^2_{initial} \dots\dots (1)$$

Where,

$W_N$  is net work done on account of an inelastic collision.

$$W_N = -1/2(m)(v^2)_{initial} \dots\dots (2)$$

But,

$$W_N = \text{Impact force} \cdot d \dots\dots (3)$$

Where d is the distance travelled during impact.

(It is considered that for static analysis, the vehicle comes to rest 0.1 seconds after impact. Therefore, for a vehicle which moves at 16.66m/s (or 60km/h), the travel of the vehicle after impact is 1.66m from equation (1), (2) and (3), we get).

### 4.2. Impact force determination by Acceleration limit

$$\text{Force} = m \times a$$

Where m=261.5 kg and a=7.9 × 9.81m/s<sup>2</sup>

## 5. CALCULATIONS

- Total Sprung mass ( $S_m$ ): 116 kg
- Total Unsprung mass ( $US_m$ ): 65 kg
- Total Weight: 181 kg
- Front track width ( $F_{tw}$ ): 52 inches
- Rear track width ( $R_{tw}$ ): 52 inches
- Static ride height ( $S_{rh}$ ): 13 inches
- Tire diameter ( $T_d$ ): 22 inches
- Ride frequency front ( $R_f$ ): 2 Hz
- Ride frequency rear ( $R_r$ ): 2.5 Hz

### 5.1 Spring Constant:

$w = \sqrt{(k/m)}$   
 where  $w$  = amplitude  
 $k$  = spring constant  
 $m$  = sprung mass  
 We also know that  
 $w = 2\pi f$   
 where  $f$  = ride frequency  
 on equating both the equations we get,  
 $k = 4\pi^2 m f^2$  N/m  
 using the above formula, the spring constants for the front and rear suspension system are calculated. Quarter body analysis of the sprung mass is done so that sprung mass acting on each wheel is determined.  
 The weight distribution according to the calculation is front: rear = 42:58.

### Front spring constant:

Entire sprung mass ( $S_m$ ) = 116 kg  
 Front sprung mass = 48.72 kg  
 Mutual mass ( $M_f$ ):  $48.72 \times 1.7 = 82.824$  kg  
 The load acting on each wheel at front ( $m_1$ ) = 41.412 kg  
 $k = 4\pi^2 m_1 f^2$   
 $k = 4 \times \pi^2 \times 41.412 \times 2^2$   
 $k = 6532.89$  N/m or 6.532 N/mm

### Rear spring constant:

Entire sprung mass ( $S_m$ ) = 116 kg  
 Rear sprung mass =  $116 \times 0.58 = 67.28$  kg  
 Mutual mass ( $M_r$ ) = 114.37 kg  
 The load acting on each wheel at front ( $m_2$ ) = 57.18 kg  
 $k = 4\pi^2 m_2 f^2$   
 $k = 4 \times \pi^2 \times 57.18 \times 2.5^2$   
 $k = 14094.29$  N/m or 14.094 N/mm

### 5.2 Wheel rate

Wheel rate = spring rate  $\times$  (motion ratio)<sup>2</sup>  
 Front wheel rate =  $6.532 \times (0.5)^2 = 1.63$  N/mm  
 Rear wheel rate =  $14.094 \times (0.7)^2 = 6.90$  N/mm

### 5.3 Roll gradient:

$$f_g = \frac{-W_v \times H_{cg}}{G_g \text{ RR}_{ff} + \text{RR}_{fr}}$$

Where,  
 $H_{cg}$  = cg to Roll axis distance (m)  
 $W_v$  = Vehicle weight (N)  
 $F_g / G_g$  = Roll gradient from ride springs (degree/g)

### Front Roll Rate:

$$\text{RR}_{ff} = \frac{3.14 \times (t_f)^2 K_{lf} K_{rf}}{180 \times (\text{WR}_{lf} + \text{WR}_{rf})}$$

$\text{RR}_{ff}$  = Front roll rate (Nm/degree roll)  
 $t_f$  = Front track width (m)  
 $\text{WR}_{lf}$  = LF Wheel rate (N/m)  
 $\text{WR}_{rf}$  = RF Wheel rate (N/m)

$$\text{RR}_{ff} = \frac{3.14 \times (1.3208)^2 \times 2420 \times 2420}{180 \times (2420 + 2420)}$$

$\text{RR}_{ff} = 36.841$  Nm/degree

### Rear Roll Rate:

$$\text{RR}_{rr} = \frac{3.14 \times (t_r)^2 K_{lr} K_{rr}}{180 \times (K_{lr} + K_{rr})}$$

$\text{RR}_{rr}$  = Rear roll rate (Nm/degree roll)  
 $t_r$  = Rear track width (m)  
 $\text{WR}_{lr}$  = LR Wheel rate (N/m)  
 $\text{WR}_{rr}$  = RR Wheel rate (N/m)

$$\text{RR}_{rr} = \frac{3.14 \times (1.3208)^2 \times 1114 \times 1114}{180 \times (1114 + 1114)}$$

$\text{RR}_{rr} = 16.959$  Nm/degree

### Roll gradient:

$$f_g = \frac{181 \times 0.2818}{0.0368 + 0.1695}$$

Roll gradient = 53.85 degree/g

## 6. ANALYSIS AND RESULTS

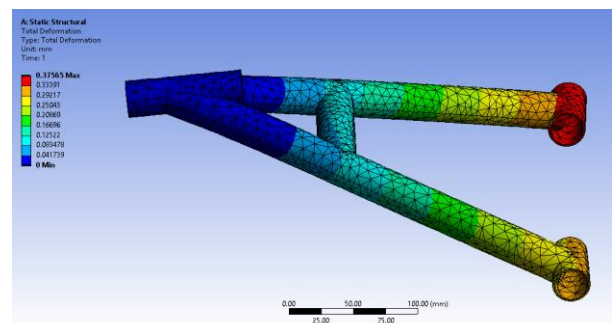


Fig. 6.1: Suspension A-arm (Total Deformation)

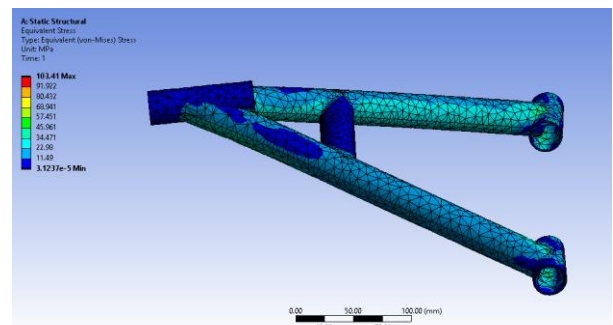
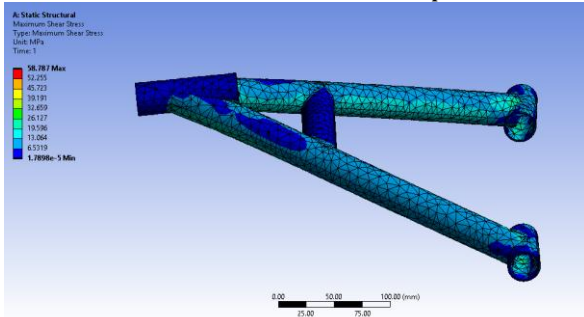


Fig. 6.2: Suspension A-arm (Equivalent Stress)

Properties	
Volume	1.4255e+005 mm <sup>3</sup>
Mass	1.119 kg
Scale Factor Value	1.

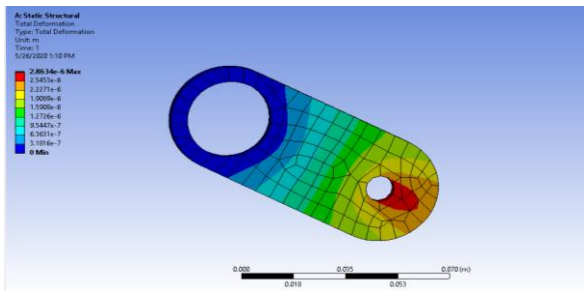
**Table 6.1:** A-Arm Material Properties



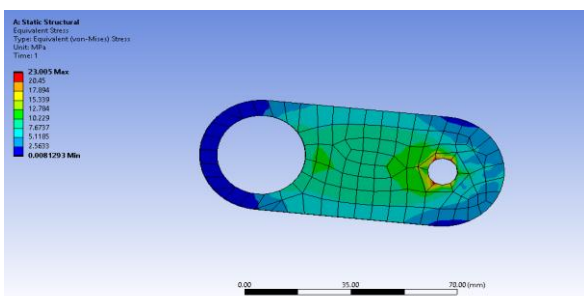
**Fig. 6.3:** Suspension A-arm (Maximum Shear Stress)

**Analysis Details:**

Material: AISI 4130 Chromoly  
 OD\*Thickness: 29.2\*1.65 mm  
 Bending Stiffness: 2764 Nm<sup>2</sup>  
 Bending Strength: 438 Nm  
 Load Applied: 3G  
 Maximum Stress: 103.41 MPa  
 Maximum Deformation: 0.37 mm



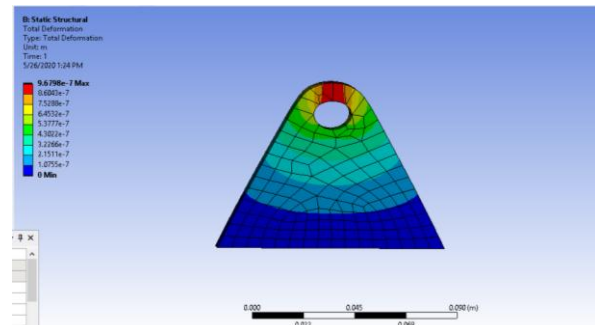
**Fig. 6.4:** Suspension mounting (Total Deformation)



**Fig. 6.5:** Suspension mounting (Equivalent Stress)

**Analysis Details:**

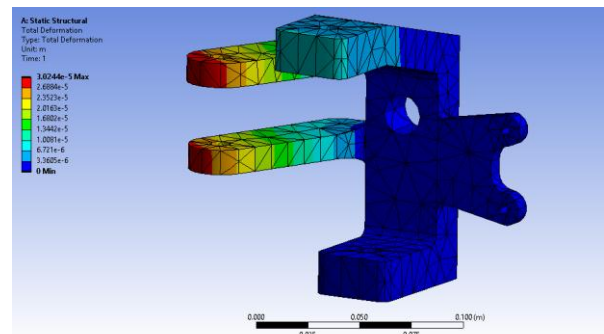
Material used: AISI 4130 Chromoly metals  
 Maximum Stress: 23 MPa



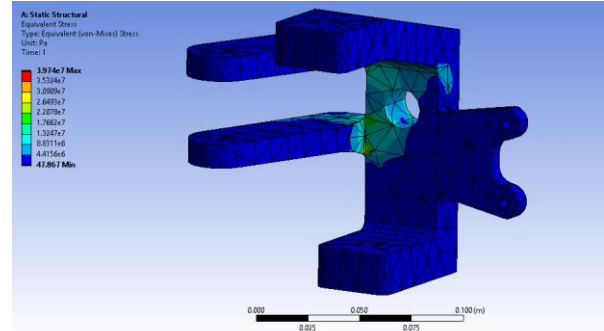
**Fig. 6.6:** Shock absorber mounting (Total Deformation)

**Analysis Details:**

Poisson's Ratio: 0.29  
 Thickness: 5mm  
 Maximum Principal Stress: 2.46 MPa  
 Equivalent Stress: 5.89 MPa



**Fig. 6.7:** Suspension Knuckle (Total Deformation)



**Fig. 6.8:** Suspension Knuckle (Equivalent Stress)

**Analysis Details:**

Material: Aluminium 6061  
 Yield Strength: 705MPa  
 Load Applied: 3G  
 Maximum Stress: 105.7 MPa  
 Maximum Deformation: 0.37 mm

**Table 6.1:** Result of Simulation and Calculations

PARAMETER		VALUE
Roll Rate	Front	36.841 Nm/degree
	Rear	16.959 Nm/degree
Wheel	Front	1.63 N/mm

Rate	Rear	6.90 N/mm
Motion Ratio	Front	0.5
	Rear	0.7
Spring Constant	Front	6.532 N/mm
	Rear	14.094 N/mm

### CONCLUSION

After initial testing of the sub-systems designed, it was observed that the systems fulfilled their individual performance expectations. The suspension system has near about optimum steer characteristics. The research work concludes the safe, comfortable and balanced wheel assembly arrangements for BAJA ATV. The calculations helped to achieve better under steer all-terrain vehicle which is able to handle tough conditions. The suspension composed of the hydraulic dampers and springs provides the sturdiness along with the flexibility to absorb shocks and bumps. The material 4130 Chromoly withstand enough for all-terrain conditions leading to conclusion for inexpensive, weld ability material. The calculations describe values of roll rate at front, wheel rate and coil rate along with wheel frequency. In this way, the design of suspension system is successfully completed and serves as ground for further research on said topic.

easy to manufacture. If a component can be easily manufactured, it means the productions costs can be taken to a minimum and production speed can be improved. For this project it was taken into consideration to design the components of the suspension to be ease to manufacture. In other words, most of the parts can be fabricated with basic tools. The upright is one of the most important parts of this system since it connects all parts of the suspension together. Thus it was important for our team to come up with an efficient, strong, light weight model. The model proposed is a very exceptional piece which its benefits are low mass, light weight, improved use of space and durability. The drawbacks of this part are the awkward shape of this part which increases the difficulty in manufacturability. The upright needs to possess several connection links that need to be strong enough to hold the critical loads which the cart is going to be put into. Such links need to be welded in place, which means change in geometry is going to be present leading to creating stress concentration factors.

### REFERENCES

- [1] William F Milliken and Doglus L Milliken 1995. Suspension geometry. Racecar vehicle dynamics. Society of Automotive engineers, Inc. 624-625.
- [2] William F Milliken and Doglus L Milliken 1995. Ride and roll rates. Racecar vehicle dynamics. Society of Automotive engineers, Inc. 582.
- [3] Thomas D. Gillespie, "Fundamentals of Vehicle Dynamics", SAE Inc.
- [4]OptimumgTechtipshttp://www.optimumg.com/docs/Springs & Dampers\_Tech\_Tip\_1.pdf
- [5] Lotus. software.http://www.lotuscars.com/engineering/engineering-software.
- [6] International Journal of Mechanical Engineering and Technology (IJMET) Vol.8, (June 2017) E-ISSN 0976-6340, Kancharana Sunil and J Kranthi Kiran: "SUSPENSION SYSTEM FOR AN ALL TERRAIN VEHICLE"
- [7] International Journal of Advance Engineering and Research (IJAER) Vol.4, (Dec 2017) E-ISSN 2348-4470, P-ISSN 2348-6406 Nayan Deshmukh\*, Vivek Singh Negi\* and Amit Deshpande: "Design of Double Wishbone Suspension System of BAJA Vehicle"
- [8] Suspension\_Baja\_Final\_Report\_2007
- [9] https://www.google.com
- [10] https://www.ansys.com/en-in



Fig: Front suspension

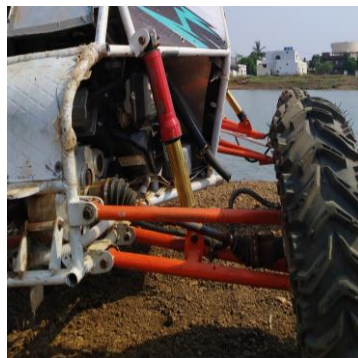


Fig: Rear suspension



Fig: Completed Fabricated Vehicle

### Manufacturability/Machining

It is very important for engineers to design for manufacturability. In other words, the components, parts or the entire assembly being designed need to be relatively