

# DESIGN, SIMULATION AND ANALYSIS OF DOUBLE WISHBONE SUSPENSION SYSTEM FOR AN ALL-TERRAIN VEHICLE (ATV)

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**Abstract** - In an All Terrain Vehicle (ATV), the suspension system plays an essential role and contributes towards the isolation of heavy vibrations in a system. The primary function of the suspension system is to absorb the shocks coming constantly from the bumps and maintains comfortable contact between an ATV and road. This paper deals with the design, optimization, analysis and simulation of the front double wishbone suspension system for an All Terrain Vehicle. The double wishbone suspension system provides balanced stability under dynamic conditions. Considering various forces, cornering effect, camber and castor angle, king pin inclination, roll steer, scrub radius, etc acting on the tire, optimized suspension system is designed, analyzed and simulated for effective performance of an ATV. Use of A-type hollow metallic arms for front suspension is provided for weight reduction. The geometry is sketched with the use of CATIA V5. Various parts like hub, knuckle, control arms, etc are designed on CATIA V5 and modeling geometry is imported in ANSYS for structural analysis. Material optimization is focused for stress and deformation results within FOS limit. The simulation of the system is tested on MSC ADAMS software. Finally, after producing different parts for front suspension, testing and validation is implemented for effective performance and overall weight reduction of an ATV.

**Key Words:** Roll Centre, load transfer, camber, toe, knuckle, wheel hub

## 1. INTRODUCTION

All Terrain Vehicles are basically known as quad bikes or quadric-cycles. The important quality of an ATV is to travel on low pressure tires along any type of terrain with single driver operating handlebars or circular wheel for steering control. In an ATV system, the double wishbone suspension system is one of the types of an independent suspension system usually using two parallel wishbones locating one end on knuckle and another end to vehicle chassis frame. Shock absorber is mounted on upper or lower wishbone depending on simulation result which isolates the bump vibrations inducing in a vehicle. The maximum travel, camber, roll centre height, anti-dive and anti-squat are major aspects related to suspension system which are important for an ATV. The main perspective of suspension system is to provide better steering handling, vehicle stability and maintain comfort for passengers by absorbing the bumps. Forces like lateral, longitudinal, cornering and vertical forces are acted through tires. The motive of an upright or knuckle assembly is to provide a

physical mounting and links from the suspension arms to the hub and wheel assembly which carries braking components. It is a load-bearing member of the suspension system and is constantly rotating with the motion of the wheel. Various materials are opted for design considerations and sustainability throughout the racing track. Optimization is done at several steps for weight reduction. The primary objective of the double wishbone suspension system is to achieve effective performance to the total weight ratio of the vehicle.

### 1.1 Objective

The basic goal of the paper is to design double wishbone system which can endure the hard terrains with suspension travel of 3 inch for bump and rebound each considering significant cornering speed.

### 1.2 Methodology

- Determination of optimised Roll Centre Geometry using Adams Car Simulation Software.
- Design of components Software such as Wishbone, Knuckle, Wheel hub on the basis of optimised Roll Centre Geometry.
- Analysis of Designed components using Ansys Workbench Software.

### 1.3 Basic Parameters

- Track width = 1220 mm (48 inch)
- Wheelbase = 1168 mm (16 inch)
- CG Height = 330 mm (13 inch)
- Static Camber Angle = 0°
- Static Toe Angle = 0°
- King Pin Inclination = 8.5°
- Centre Offset = 62 mm
- Design Weight of Vehicle = 250 Kg
- Weight Distribution = 40:60 (Front : Rear)

## 2. ROLL CENTRE GEOMETRY

Roll centre is the imaginary point about which the vehicle rolls in dynamic condition. There are certain methodologies to determine the height of roll centre in suspension system. Locating the static position of roll centre in suspension geometry plays the pivotal roll as it further determines the lengths of wishbones, total height of knuckle, position of steering arm, length of tie rod, etc. After determining the position of roll centre, the respective geometry is simulated in Adams Car Simulation Software. Geometry providing the precise simulation results is further considered as optimized roll centre geometry. Procedure for locating roll centre for double wishbone suspension system is explained below.

### 2.1 Determination of Roll Centre

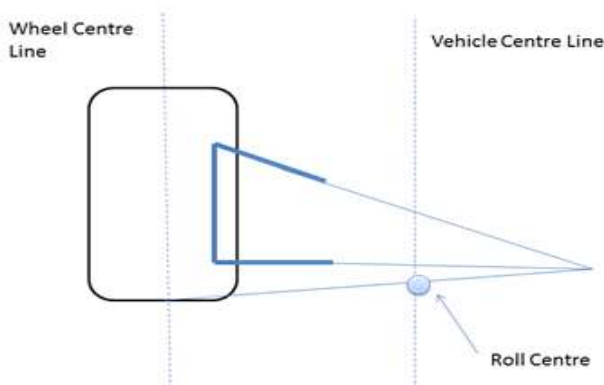


Fig -1: Determination of Roll Centre

Above figure explains the basic course of action to determine the roll centre. As it can be seen from the figure, it is solely dependent on suspension geometry as well as basic principle of Instantaneous centre. Initially, the ICR (instantaneous centre of rotation) of suspension arms i.e. wishbones is located through which imaginary line from wheel contact patch is passed. The point of intersection of this line and vehicle centre axis defines the static roll centre of the given suspension geometry. Furthermore it should be noted that, relative distance between the Roll Centre and Centre of Gravity (C.G) directly affects the magnitude of jacking force induced.

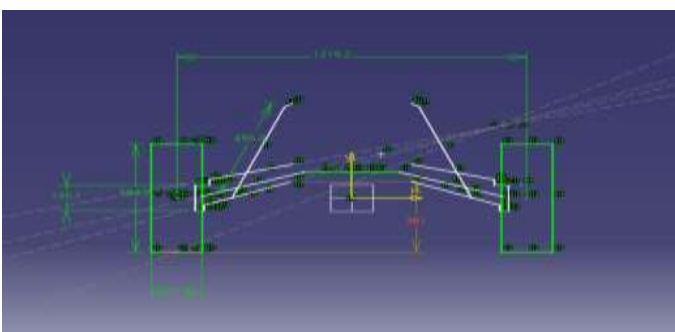


Fig -2: Roll Centre Geometry

### 2.2 Simulation of Roll Centre Geometry

After plotting the RC geometry, kinematic simulation is performed using MSC Adams Car Software to find out whether the designed system provides the optimized results or not. MSC Adams software provides good simulation platform for the user giving specific graphical outputs for the various dynamic input conditions.

Hard-points of several suspension locations are entered in Adams software which further generates the 3D model of designed suspension geometry. Basic parameters such as vehicle weight, weight distribution ratio, CG height, static wheel parameters (camber and toe angle) must be provided in the iteration before performing the final simulation as it affects the performance of a vehicle in dynamic condition.

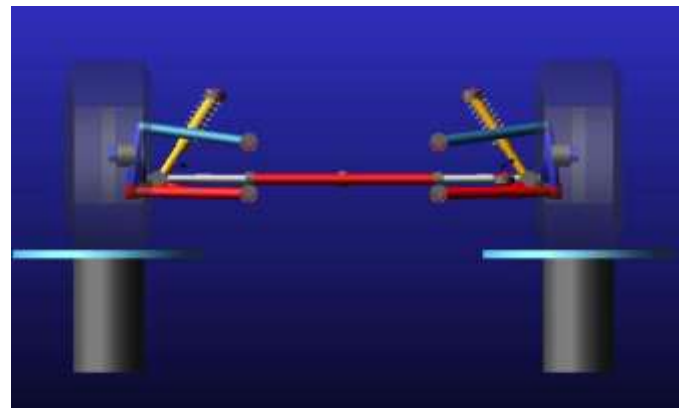


Fig -3: Suspension model in ADAMS

Simulation results can be obtained in the graphical form for different wheel parameters such as camber, caster, toe, ride rate against wheel travel. It is important to note that, dynamic change in the wheel parameters should be minimum for better handling performance and less tire wear.

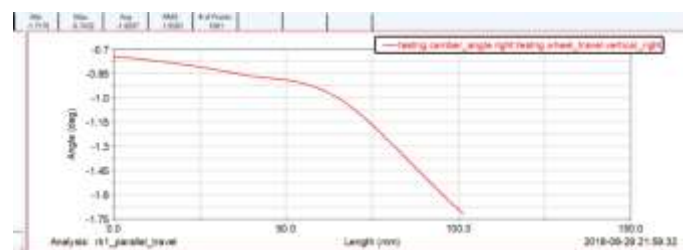


Chart -1: Camber Angle VS Wheel Travel

It can be seen from the results that, dynamic change of the camber angle and toe angle against wheel travel is within limit. Hence, the respective RC Geometry should be considered as optimized one for the further design.

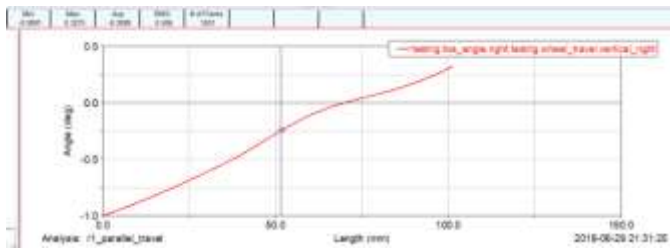


Chart -2: Toe Angle VS Wheel Travel

Following are the important parameters which are obtained from Optimized Roll Centre Geometry.

Table -1: Obtained Parameters

Parameter	Value
Length of Upper Wishbone	246 mm
Length of Lower Wishbone	282 mm
Centre to Centre Height of Knuckle	140mm
Position of Steering Arm on Knuckle (from centre of upper ball joint mount)	60 mm
Length of Steering Tie Rod	395 mm

### 3. MATERIAL SELECTION

Material Selection plays an important role in given design and analysis as selected material should satisfy the design criteria. Additionally, maximum and minimum stress induced in the dynamic loading condition should also be within factor of safety limit. Material mainly used for manufacturing of wheel assembly i.e. Knuckle and hub are Mild steel and Aluminum. Factor of safety while designing these suspension components should be considered between 1 to 2.5 for given material. There are other factors such as cost, availability, weight which should also be considered while selecting the appropriate material.

#### 3.1 Material Selection for Wishbone

Wishbones are the connecting link between wheel assembly and chassis of a vehicle. Thus, dynamic lateral load transfer induced while cornering acts directly through these suspension links. Hence, wishbone must sustain these dynamic loading conditions before structural failure. This factor should be considered while selecting the material for wishbones.

According to market availability of different materials, grades AISI 1018, AISI 1020, AISI 4130 are shortlisted for material selection amongst which grade AISI 1020 with outer pipe diameter of 22 mm having 1.6 mm thickness is selected for wishbones considering yield strength.

Table 2 -: Material Comparison for Wishbone

Material Properties	AISI 1018	AISI 1020	AISI 4130
Yield Strength (MPa)	440	394	560
Ultimate Tensile Strength (Mpa)	370	294	460
Hardness (BHN)	126	111	217
Modulus of Rigidity (GPa)	205	200	210

#### 3.2 Material Selection for Wheel Assembly

Wheel assembly mainly consists of knuckle, stub axle, brake disk, caliper mounting and wheel hub. As the scope of this paper is limited to knuckle and hub, we will select material for the same by performing similar material comparison as earlier.

Hub is one of critical part of the wheel assembly connecting tires and brake disk on either end. It sustains static forces such as wheel reaction, distributed vehicle weight as well as dynamic loading such as lateral load transfer, longitudinal load transfer, braking torque and bump forces. These forces are further directed towards knuckle through the stub axle which is accountable for holding the wheel assembly together. Because of this reason, vehicle hub and knuckle should be made of the material having high ultimate tensile strength with factor of safety more than 1.5.

Table 3 -: Material Comparison for Wheel Assembly

Material Properties	Al 6061	Al 7071	EN 24
Yield Strength (MPa)	310	572	1000
Ultimate Tensile Strength (MPa)	276	503	680
Hardness (BHN)	95	150	302
Modulus of Rigidity (GPa)	68.9	71.7	207

Considering all above factors along with criticality of components, Al 7071 is selected for wheel assembly providing highest tensile stress with high weight to performance ratio.

#### 4. DESIGN AND ANALYSIS

Design mainly includes force calculation as follows:

- Cornering Force –

It is calculated using Newton's Second Law of Motion and acts in radially outward direction on wheel assembly.

Consider, vehicle is cornering at 45 KMPH

(12.5 m/s) with track turning radius of 2.6 m, then cornering force induced on inner tire is given by,

$$F_c = \frac{MV^2}{R}$$

Substituting above values we get,

$$F_c = \frac{250 \times 0.4 \times 12.5 \times 12.5}{2.6}$$

$$F_c = 6010 \text{ N}$$

Above equation gives the magnitude of cornering force acting on inner wheel of the vehicle. Definite percentage of this force is transferred dynamically to outer tire in lateral direction. This is called as dynamic lateral load transfer. It is calculated by

$$\text{Lateral Load Transfer} =$$

$$\frac{\text{Lateral Acceleration (g)} \times \text{mass (kg)} \times \text{CG Height (m)}}{\text{Track Width (m)}}$$

It can be seen in above equation, product of lateral acceleration and mass gives the value of Cornering force calculated earlier. Thus, substituting values in above equation we get,

$$\text{Load Transfer} = \frac{6010 \times 0.33}{1.22}$$

$$\text{Load Transfer} = 1626 \text{ N}$$

This simply implies that, under steady state condition, 1626 N of the load on the inside tire is transferred laterally to outside tire achieving dynamic lateral load transfer of 27 %. It should be noted that, load transfer is mainly calculated to check the feasibility of the suspension system under dynamic condition. However, cornering force calculated earlier must be used while performing the static structural analysis.

- Bump Force –

It is calculated using principle of change in kinetic energy. It is given by,

$$\text{Change in KE} = \text{Work Done}$$

$$\text{Final KE} - \text{Initial KE} = \text{Bump Force} \times \text{Displacement}$$

Consider, the vehicle accelerating from rest position achieving the final velocity of 45 KMPH (12.5 m/s) at the instance of 6ft (1.82 m) bump i.e. displacement,

$$F_{\text{Bump}} = 10684 \text{ N}$$

- Force on Steering Arm –

It is the product of static wheel reaction and that of coefficient of friction.

$$\text{Force} = \mu \times \text{Static Wheel Reaction}$$

Consider, wheel reaction distributed according to weight distribution ratio and coefficient of friction of 0.6 we get,

$$\text{Force} = \frac{0.6 \times 0.4 \times 250 \times 9.81}{2}$$

$$2$$

$$\text{Force} = 294.3 \text{ N}$$

This represents the magnitude of force acting through steering arm for single knuckle.

- Brake Torque –

It is simply the product of braking force (N) and effective wheel radius (mm).

$$\text{Brake Torque} = \text{Force}_{\text{brake}} \times \text{Effective Wheel Radius}$$

Considering the brake force of 1758 N and effective wheel radius of 10.25 inch, we get,

$$\text{Brake Torque} = 458 \text{ Nmm}$$

#### 4.1 Design of Wishbones

We have obtained the length of upper and lower wishbones from optimized RC Geometry (refer table 1). Additionally, we have also selected the appropriate material for wishbones. Shock absorber is mounted on lower wishbones with motion ratio of 0.8 providing precise simulation results. Thus, following figure represents the 3D model of designed lower wishbone.

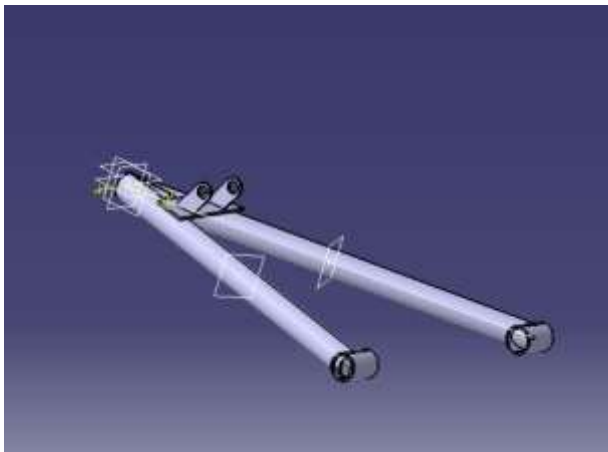


Fig -4: CAD Model of Lower Wishbone

After the applying the discussed loading conditions, analysis is performed which provided following results.

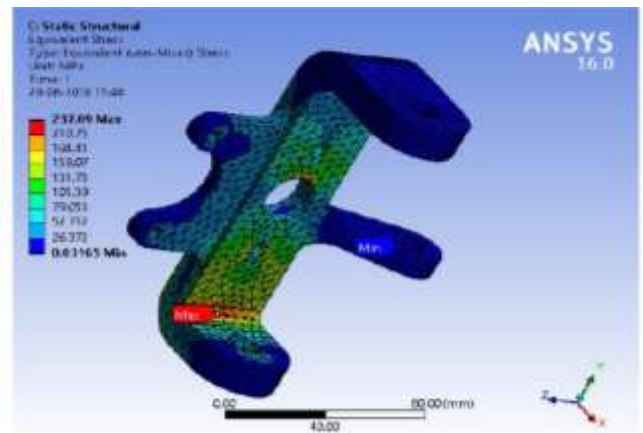


Fig -6: Stress Analysis of Knuckle

#### 4.2 Design and Analysis of Knuckle

It is the stationary part of the wheel assembly which has mountings for suspension links as well as brake caliper. Suspension links are mounted on knuckle mainly using ball joint which provides total angle of action about 30°. It is important to note that, ball joint should not be mounted at angle greater than 40° which can cause repeating failure of ball joint due to application instantaneous peak load. This can lead to severe damage of wheel assembly.

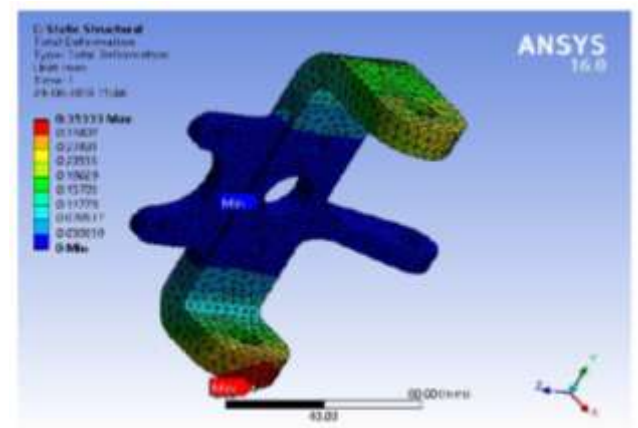


Fig -7: Total Deformation of Knuckle

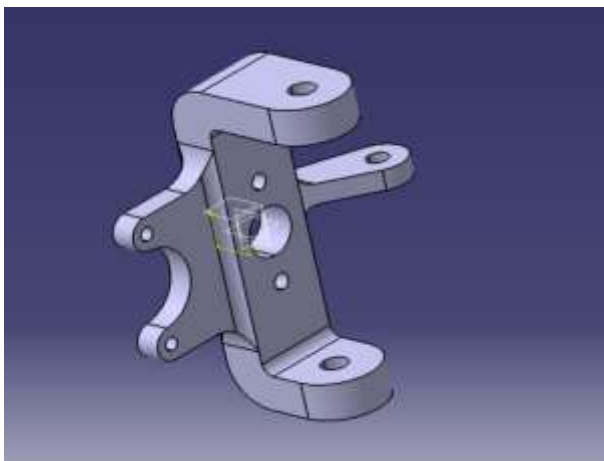


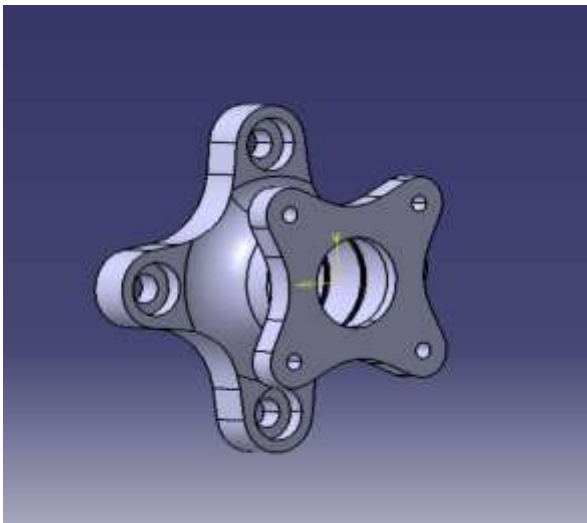
Fig -5: CAD Model of Knuckle

Based on force calculation performed earlier, we will now discuss the exact location at which forces must be applied while performing the Static Structural analysis in ANSYS Workbench. Importantly, fixed support must be applied to stub axle hole and then other loading conditions are applicable. Cornering force is applied at ball joint mounting hole in lateral direction while braking torque is applied at caliper mounting hole in either anticlockwise or clockwise direction. Steering force is applicable to steering arm as discussed earlier and bump force must applied at lower ball joint mounting hole where initial impact takes place.

As it can be seen that, maximum induced stress as well as total deformation are well within Factor of Safety limit, we can conclude that design and analysis of the knuckle is successfully completed.

#### 4.3 Design and Analysis of Hub

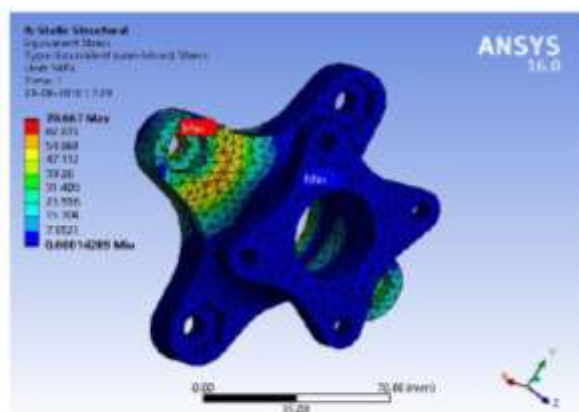
It is rotating part of the wheel assembly which has two petals for wheel mounting and brake disc mounting on the either side. Hub should be precisely designed as there are chances of fouling of brake disk petal with mounting bolts. Additionally, hub failure mainly occurs due torsion, hence petal thickness should be kept appropriately according to selected material. Further weight reduction can also be achieved by providing slots on the wheel mounting petal. However, stress analysis must be performed for the same.



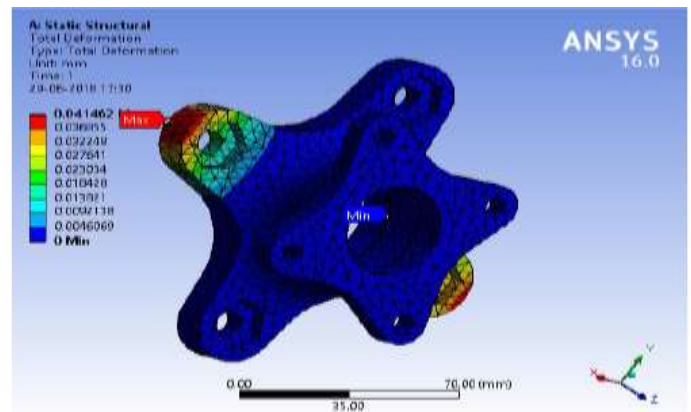
**Fig -8:** CAD Model of Hub

Similar to that of knuckle, we will now discuss the exact location at which forces must be applied while performing the Static Structural analysis in ANSYS Workbench. It should be noted that, while applying the cornering force in lateral direction and bump force in vertical direction on wheel mounting holes, fixed support must be applied to bearing mounting location. On the other hand, while applying brake torque on wheel mounting holes in either clockwise or anticlockwise direction, fixed support must be applied to brake disk mounting holes.

Following results are obtained after performing the analysis.



**Fig -9:** Stress Analysis of Hub



**Fig -10:** Total Deformation of Hub

From the static structural analysis of wheel hub, it is observed that total deformation and induced maximum stress is well within limit. Hence, we can conclude that, design and analysis of hub is successfully completed.

## 5. SELECTION OF SHOCK ABSORBERS

Radflo Air Shocks 2.0 are selected, as the stiffness can be adjusted easily by varying the air pressure. It has total eye to eye length of 19.5 inches which is used while drawing the roll centre geometry. As per design, it is mounted on lower wishbone with the motion ratio of 0.8 at an angle of 54° with respect to vertical axis.

## 6. CONCLUSION

After designing and simulating the optimized roll centre geometry using MSC Adams car software, lengths of important suspension components are obtained. This further led to material selection and designing of the components such as wishbones, knuckle, hub, etc using CATIA V5 software. Furthermore, static structural analysis of these components is performed using force analysis in ANSYS Workbench Software. This provided stress and deformation results within acceptable factor of safety limit which confirms the successful design and analysis of the suspension components.

## REFERENCES

1. Smith, Caroll. 1978, "Tune to win", Chapter 2, Aero publishers Inc.
2. Thomas D. Gillespie, "Fundamentals of Vehicle Dynamics", SAE Inc.
3. Trushar Bhoje, Prof. Anurag Nema, Prof. Anantharma, "Design, Analysis and Experimental Investigation of Double Wishbone Independent Suspension System for Agriculture Trolley", International Research Journal of Engineering and Technology (IRJET)' e-ISSN: 2395-0056, Volume 3, Issue 9, Sep-2016, pp.1067-1071.

4. Asst. Prof. N.Vivekanandan, Abhilash Gunaki, Chinmaya Acharya, Savio Gilbert and Rushikesh Bodake, "Design, Analysis And Simulation of Double Wishbone Suspension System", "International Journal of Mechanical Engineering (IJME)", Volume 2, Issue 6, June 2014,pp.1-7.
5. Reena Mishra, Anand Baghel, "Design, Analysis and Optimization of front suspension wishbone of BAJA 2016 of All-terrain vehicle- A Review", International Journal of Research in Mechanical, Mechatronics and Automobile Engineering (IJRMMAE), ISSN : 2454-1435, Volume 2, Issue 3, 28th Feb, 2017, pp.40-52.