

Dynamic Analysis of the Front and Rear Suspension System of an All-Terrain Vehicle

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Abstract - Suspension system plays an important role in handling of the vehicle. This paper deals with the design and analysis of the parts used in the suspension system of an ATV in front and in rear. The purpose behind the design of this suspension system is to control camber and toe (based on iterations) with maximum damping and minimum wheelbase and track width change. This study summarizes in brief the design parameters, targets and considerations followed for the fabrication and assembly of the system. The suspension system plays an important role in stability of the vehicle in static and dynamic conditions.

Key Words: Isolation of masses, Double Wishbone, Multi-point, Force Interaction, Spring tuning, Material selection and analysis.

1. INTRODUCTION

Suspension system is an important part of vehicle which provides comfort, control and safety to the passengers. It allows vehicle to travel over rough surfaces with minimum up-and-down body movement. Suspension is defined as isolation of two masses hence while designing suspension, a balanced design should be provided in order to carry out functions like road holding, load carrying and passenger comfort. It consists of arms, damper, spring, joints, tire, knuckle and hub. It is necessary to design a suspension system that can handle the roughest of terrain and endure extreme force conditions without affecting the vehicle's stability and at the same time also provide a smooth ride to the driver. So it has been decided to go for the double wishbone in front and h-arm in rear with a dual rate spring system, so that it can sustain the design load and give excellent comfort and control over the off-road, providing safety to the driver.

1.1 Front suspension system- Double Wishbone

The designing of ATV's is considered under front steering conditions. As the front wheels need to steer in every condition, Double wishbone assembly is one of better option. It consists of two V- shape link. Suspension deals with restriction of degree of freedom. In double wishbone assembly each wishbone link restricts two Degrees of Freedom and one tie rod which controls one Degree of Freedom in all by double wishbone assembly five Degrees Of Freedom is restrained.

It provides a large amount of room for adjustment. Allows decent tire camber control resulting in enhanced handling characteristics, has high strength and rigidity. If an impact occurs and the suspension suffers damage it is unlikely that all the components will need replacing. Finally, it permits a low unsprung weight for the vehicle as only a small portion of the linkage weights are unsprung.

Dual unequal length A-arm were used as they provide optimum camber curves in wheel travel and the shorter upper arm helps to induce a camber curve which maintains maximum contact patch during wheel travel. During rolling motion the inner wheel undergoes droop while the outer wheel goes through bump motion. This increases the contact patches in the respective wheels which is very important in any off road vehicle. This was achieved by short long arm geometry.

1.2 Rear suspension system- H-Arm Suspension

In consideration of front steering condition, the rear wheels should have high toe and camber control. The Rear Wheel Drive in an ATV has more weight in rear part. So it is necessary to have the linkage which should provide Toe and Camber control with load bearing capacity. H arm and camber link is nothing but multi point suspension, where H arm provides restraining of four Degrees of Freedom with camber link which restricts one Degree of Freedom in all five Degrees Of Freedom restricted by H- arm with camber link.

H arm is nothing but link of shape 'H', where two mountings are on chassis and two are on knuckle. The H- arm has unique structure which provides zero steer condition in dynamic condition. The large space availability for damper mounting gives advantage in suspension. It provides low unsprung weight which is very essential for any suspension system.

2. Design Procedures

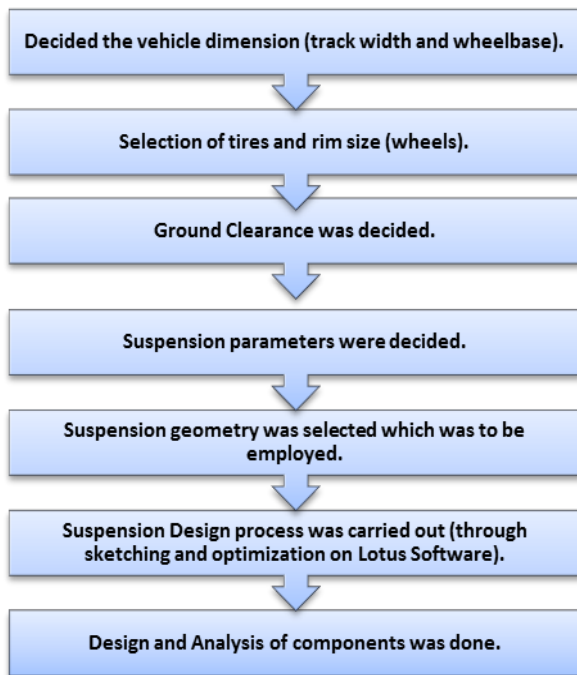


Chart1: Design methodology

2.1 Design Considerations

Table 1: Vehicle specifications

Parameter	Front	Rear
Ground Clearance(mm)	350	300
Track Width(inch)	49	46
Wheelbase(inch)	57.07	
Tire	23*7*10	

The tire dimensions decided was 23*7*10 for the better traction, ease of assembly of component, low unsprung mass and the rim is offset by 1.5 inches and uniform bolt dimensions for interchangeability.

The arms were decided to keep as long as possible for the stability of the vehicle and minimize the variations in geometry. The length of the upper arms was decided through geometry modeling, shorter upper arm meant negative camber in bump travel and optimum roll-camber coefficient.

The design of suspension begins with considering some important factors. The requirement for a good suspension system is to select good dampers, types of suspension system considering which is best suited for the current chassis design and steering system.

2.3 Force Analysis

There are three forces acting on the vehicle.

1. Momentum on the rear and frontal axle due to load transfer from the inner to the external wheel during the curve (cornering forces).
2. Longitudinal forces (braking and driving torque).
3. Vertical force due random obstacles.

3. Suspension Geometry

It refers to various types, arrangements of suspension arms. Different geometries produce different types of motion and can be used according to the need of purpose. Various geometries could be used for same purpose. The lotus software made it possible to select the effective geometry

3.1 Front Suspension Consideration

The front part of the suspension has to also deal with steering characteristics to maintain the control of the vehicle during the dynamic conditions.

Dual unequal length A-arms were used as they provide optimum camber curves in wheel travel and the shorter upper arm helps to induce a camber curve which maintains maximum contact patch during wheel travel. During rolling motion the inner wheel undergoes droop while the outer wheel goes through bump motion. This increases the contact patches in the respective wheels which is very important in any off road vehicle. This was achieved by short long arm geometry.

3.1.1 Double wishbone A-arms



Fig 1: Double Wishbone

1. It provides a large amount of room for adjustment like camber.
2. Allows decent tyre camber control resulting in enhanced handling characteristics, has high strength and rigidity.
3. If an impact occurs and the suspension suffers damage it is unlikely that all the components will need replacing.

- It permits a low unsprung weight for the vehicle as only a small portion of the linkage weights are unsprung.

3.1.2 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.

Table 2: Properties

Properties	AISI1018	AISI1040	AISI4130
Carbon Content (%)	0.18	0.40	0.30
Tensile Strength (MPa)	440	620	700
Yield Strength (MPa)	370	415	600
Hardness (BHN)	126	201	217
Cost (Rs/metre)	325	425	600

The material chosen for A-arms was *AISI 4130* due to its good tensile strength.

3.1.3 Joints selections

We select the following joints as we get the required degree of freedom and availability.

Table 3: Degree of freedom in joints

Joint type	Reasons (degree of freedom)
Ball joint	27°
Heim joint	13°

4. Rear suspension system:

In rear suspension design optimum Camber gain and minimalistic toe gain were main design considerations. Since most of the weight of the body is in the rear the suspension had to be designed to keep camber curve to provide max traction in different rolling situations. Lateral and cornering forces were taken into consideration while designing the components for rear.

Table 5: suspension specification in Rear

Parameter	bump	droop
Wheel travel (mm)	125	60
Ground clearance(mm)	300	

By using the torsion bar in the rear reduces the chances of vehicle to rollover. This also helps in inducing oversteer and which again helps in taking tighter turns by reducing the turning radius. The suspension geometry was setup in way such that the migration of the roll Centre was minimum at the rear thus inducing stability in the heavy rear part of the vehicle.

The geometry selected for the rear suspension system was semi trailing arm suspension but due to undesirable camber gain and variation in suspension parameter we go for H-arms with camber link.

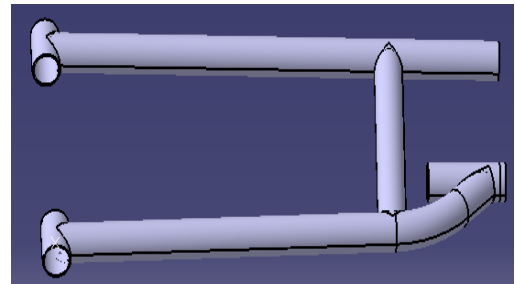
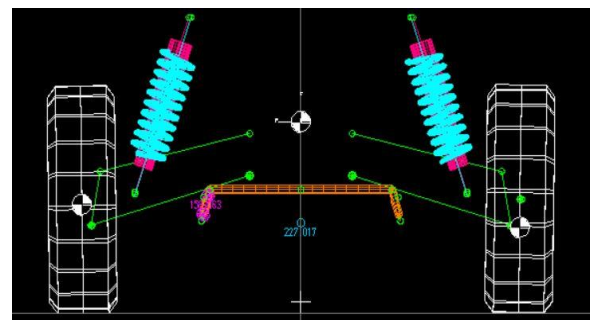


Fig 2: H arm

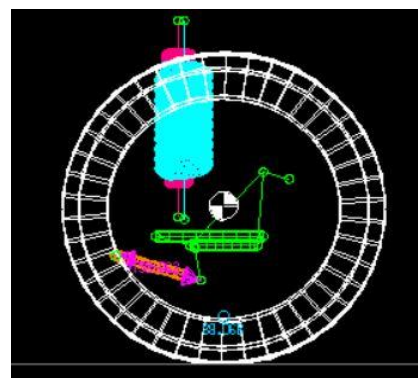
4.1 H-arms with camber link

They do not have any toe angle, kingpin axis due to the design of the knuckle. They have less degree of rotation. They are connected to the knuckle via nut and bolt.

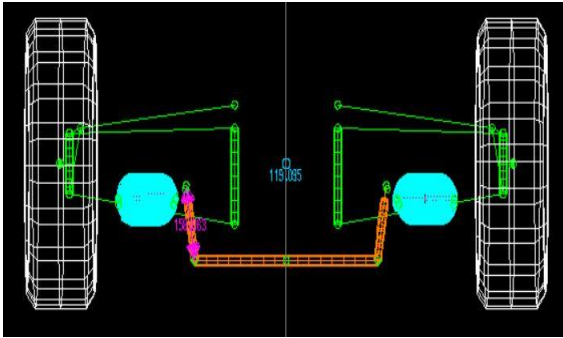
4.1.1 Final rear geometry



Front view



Side view



Top view

Fig 3: H arm geometries projection views

Suspension design parameters:

Table 6: Rear suspension specification

Total mass(kg)	201.17
Sprung mass(kg)	146.76
Unsprung mass(kg)	54.406
Castor(degree)	0
KPI(degree)	0
Ground clearance(mm)	300

5. Calculations:

Wheelbase (WB) = 1450 mm

Track width

Front = 1244.6mm Rear = 1168.4mm

Total mass=201.17 kg

Front mass (%) =41.78 % =84.05kg

Rear mass (%) =58.219 % = 117.12kg

Unsprung mass = 54.4059kg

Sprung mass =146.76 kg

Front sprung (M_{fsm}) = 53.861kg

Rear sprung (M_{rsm}) = 92.899 kg

$$\text{Motion Ratio} = \frac{\text{spring Travel}}{\text{wheel Travel}}$$

Front (M_F) =0.4894

Rear (M_R) = 0.64568

$$\text{Spring Stiffness} = 4 \times \pi^2 \times (f_{rt})^2 \times M_{fsm} \times Rf^2$$

K_{sf} (Front) = 23.48N/mm

K_{sr} (Rear) = 34.49N/mm

$$\text{Wheel Rate} = \frac{\text{Spring Rate}}{(\text{Motion Ratio})^2}$$

K_f (Front)= 98.03N/mm

K_r (Rear) = 82.73N/mm

C.G Height = 476.78 mm

$$\text{Roll Rate} = \frac{\pi \times (tf)^2 \times Kl_f \times Kl_r}{180 \times (K_r + K_f)}$$

Front (K_{θF}) = 110825.44 N/deg

Rear (K_{θR}) = 242571.75 N/deg

$$\text{Ride Rate} = (2 \times \pi \times frf)^2 \times Mf$$

K_{rF} (Front) = 12.11 N/mm

K_{rR} (Rear) = 26.67 N/mm

$$\text{Critical Damping} = 2 \times \sqrt{Kf \times Mfsm}$$

CrF(Front)=1162.37 kg/s

CrR(Rear) =1969 kg/s

$$\text{Damping Coefficient} = \text{Critical Damping} \times \text{Damping Ratio}$$

C_F(Front)= C_{rF} × Damping Ratio=1842.36 kg/s

C_R(Rear)=C_{rR} × Damping Ratio=2812.96 kg/s

$$\text{Damping Force} = \text{Damping Coefficient} \times \text{Velocity}$$

D_F (Front) = C_F × v =552.70 N

D_R (Rear)

=C_R × v=843.88 N

$$\text{Damper Stroke Length} = \frac{\text{Damping Ratio}}{\text{Spring Rate}}$$

Front = $\frac{Df}{K_{sf}}$ = 19.73 mm

Rear =

$\frac{Dr}{K_{sr}}$ = 21.09 mm

$$\text{Natural Frequency} = \left(\frac{1}{2\pi}\right) \times \sqrt{\frac{\text{Spring Rate}}{\text{Motion Ratio}}}$$

Front (F_f) = $\frac{1}{2\pi} \times \sqrt{\left(\frac{K_{sf}}{M_f}\right)}$ =2.3 Hz

Rear (R_r) = $\frac{1}{2\pi} \times \sqrt{\frac{K_{rf}}{M_r}}$ =2.8 Hz.

6. Damper selection:

By doing market survey and our design requirement following are consider for the comparisons

Table 7: availability of Damper and specifications

Parameter	AFCO 16 series	AFCO16 series	King shocks
Travel(inches)	6	6	5.5
Adjustment	single	double	single
Weight(kg)	1.34	2.30	1.5
Cost (per piece)	30000	45000	30000

Out of the above comparison we select the AFCO16 series single adjustable strut for the low weight required travel.

7. Springs

A mechanical spring is an elastic body which has primary function is to deflect under load and return to its original shape when load is removed. Spring undergoes different types of forces like twist, pull, stretch, etc.

In vehicle suspension system, helical compression spring is one of the primary elastic members. They attach vehicle body with wheel elastically and smooth out shocks which are received from wheel. Spring rate indicates energy absorption capacity. The fatigue failure of the suspension spring in a variety of ways Raw materials defect, surface imperfections, improper heat treatment, corrosion and decarburization are generally recognized causes of fatigue failure of suspension spring.

7.1 Parameters:

7.1.1 Selection of spring end

For the compression, spring has tendency to take load along the axis and flat surface. So, we selected the squared and ground end.

7.1.2 Material selection for spring

Any of the below material can be used for spring but it is good to prefer the alloy steel of chromo vanadium ASTM A231 because of high impact and shock absorbing capacity along with high temperature sustainability but due to unavailability of the chromo vanadium material in *India* and high cost we go for the IS4454 grade 2 **cold drawn steel wire** spring material.

7.2 Concept of dual rate spring

In this arrangement two or more springs of different stiffness are arranged in series. This method gives better performance rather than a single spring. Let assume we take the two springs of different stiffness

Damper Characteristics	Rear
AFCO 16 series (travel)(inch)	6
Damper Stroke(mm)	21.09
Damping (N)	843.88

One is less stiff say k_1 and another is stiffer say k_2 . And the relation between the total stiffness and this stiffness is

$$1/\text{total stiffness (k)} = 1/k_1 + 1/k_2$$

$$\text{Total force (f)} = f_1 = f_2$$

$$\text{Total deflection of spring } (\delta) = \delta_1 + \delta_2$$

7.3 Calculations:

Properties of IS4454 grade 2 cold drawn steel:

Minimum tensile strength=1290 Mpa

Modulus of rigidity (G) = 73000 Mpa

Factor of safety=1.5

Allowable tensile strength=1290 N/mm

Allowable shear stress=0.6*1290=774/FOS=516 N/mm

Required travel of spring (front & rear) =6 inches

Spring stiffness

$$\text{Front } (K_{sf}) = 23.4818 \text{ N/mm}$$

$$\text{Rear } (K_{sr}) = 34.485 \text{ N/mm}$$

As it is decided to use 2 springs in series because we get the linear graph between loads vs. deflection.

So that equivalent stiffness (K) is given by,

$$\frac{1}{K} = \frac{1}{k_1} + \frac{1}{k_2}$$

$$\frac{1}{23.56} = \frac{1}{35} + \frac{1}{k_2}$$

$$\frac{1}{k_2} = \frac{1}{35} - \frac{1}{23.56}$$

$$k_2 = 68.56 \text{ N/mm}$$

Front equivalent springs constants

$$K_1 = 35 \text{ N/mm}$$

$$K_2 = 68.56 \text{ N/mm}$$

Rear equivalent spring constant

$$K_1 = 55 \text{ N/mm}$$

$$K_2 = 90.01 \text{ N/mm}$$

Spring dimensions

Front spring design

Spring 1

Wire diameter (mm) =9
 Mean coil diameter (mm) =59
 Number of turns =8
 Total no of turns = 10
 Spring index =6.55
 Free length (mm) =210
 Solid length (mm) =90
 Pitch (mm) =24
 Travel 1 (mm) =120

$$k1 = \frac{G * d}{8 * C^3 * n}$$

$$K1 = 36.5N/mm$$

Spring 2

Wire diameter (mm) =8
 Mean coil diameter (mm) =57
 Spring index =7.12
 Number of turns =3
 Total no of turns = 5
 Free length (mm) =70
 Solid length (mm) =40
 Pitch (mm) =18
 Travel 2 (mm) =30

K2=68.9N/mm

Total travel (T) =travel 1+travel2
 =120+30=150mm
 =5.9 inch

Rear spring design

Wire diameter (mm) =9
 Mean coil diameter (mm) =56
 Spring index =6.22
 Number of turns =6
 Total no of turns = 8
 Free length (mm) =180
 Solid length (mm) =72
 Pitch (mm) =27
 Travel (mm) =108

K1=55N/mm

Wire diameter (mm) =9

Mean coil diameter (mm) =59
 Spring index =6.55
 Number of turns =3
 Total no of turns = 5
 Free length (mm) =87
 Solid length (mm) =45
 Pitch (mm) =23
 Travel (mm) =42
K2=97.2 N/m
Total travel=travel 1+travel2
=108+42=150mm
=5.9 inch

Calculations:

$$k = \frac{G * d}{8 * C^3 * n}$$

$$35.56 = \frac{73000 * 9}{8 * (6.5)^3 * n}$$

Number of turns (n) =8.4=8
 Free length (Lf) =Pn+2d
 210=P*8+2*9
 P=24mm

Checking in Torsional shear stress
 Considering static failure
 Let k_s static shear stress factor

$$k_s = 1 + \frac{0.5d}{D}$$

$k_s=1.07$
 Static shear stress=

$$\tau_s = k_s \left(\frac{8 * w * D}{\pi d^3} \right)$$

$$\tau_s = 1.07 \left(\frac{8 * 4000 * 59}{\pi 9^3} \right)$$

$$=882.02N/mm$$

Now, by considering curvature effect

$$K_w = \left(\frac{4C-1}{4C-4} \right) + \left(\frac{0.615}{C} \right)$$

$$= \left(\frac{4 * 6.5 - 1}{4 * 6.5 - 4} \right) + \left(\frac{0.615}{6.5} \right)$$

$$=1.231$$

Shear stress induced considering curvature effect is given by

$$\tau_c = kw \left(\frac{8 * w * D}{\pi d^3} \right)$$

$$\tau_c = 1.231 \left(\frac{8 * 4000 * 59}{\pi 9^3} \right)$$

$$= 1014 \text{ n/mm}$$

Checking for buckling for both ends fixed spring

$$\frac{Ls}{D} \leq 5.2$$

$$\frac{210}{59} \leq 5.2$$

$$3.6 \leq 5.2$$

So the buckling is not occurs in spring.

8. Rear wheel hub:

Which would be the force acting on the hub when vehicle travels from a bump? Considering the above parameters analysis has been carried out and maximum von mises stress as well as maximum deformation has been calculated.

8.1 Mesh and refinement:

Defeaturing	
Statistics	
<input type="checkbox"/> Nodes	1225441
<input type="checkbox"/> Elements	766834
Mesh Metric	
<input type="checkbox"/> Min	0.107757710635553
<input type="checkbox"/> Max	0.999977098794229
<input type="checkbox"/> Average	0.756742295168794
<input type="checkbox"/> Standard Deviation	0.131707301406643

Fig 4: mesh refinement

8.2 Parameters applied:

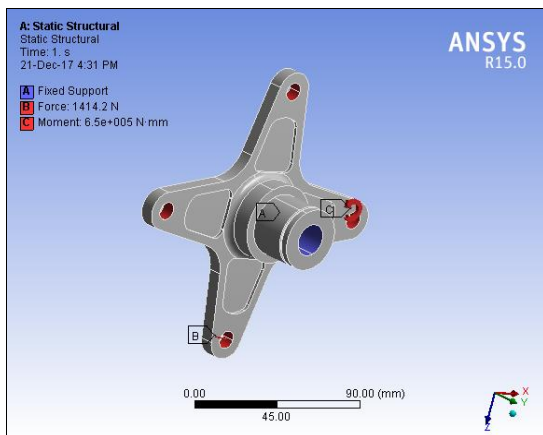


Fig 5: Force/Moment on Hub

8.3 Stress:

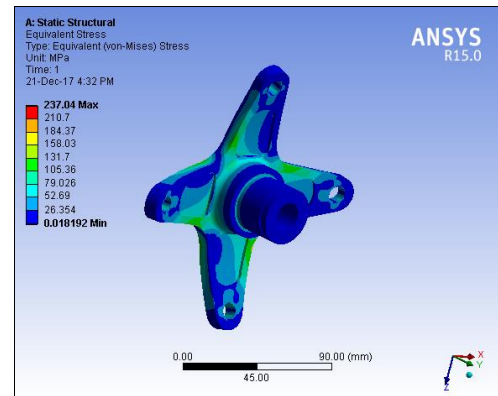


Fig 6: Stress Analysis

8.4 Deformation:

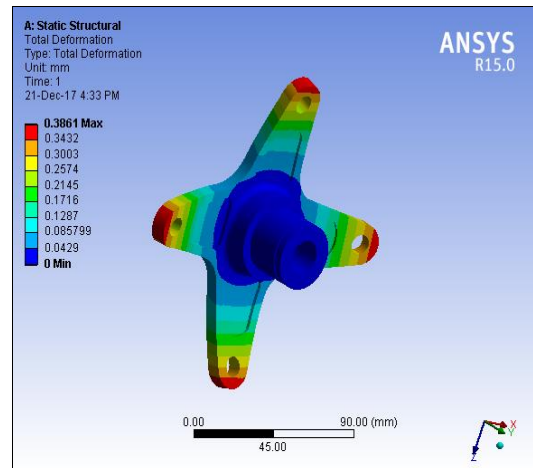


Fig 7: Deformation chart

After carrying out analysis the maximum stress occurring is about **237.04 MPa** and the deformation was about **0.3861mm** with FOS **2.12**

We also performed fatigue analysis and the life obtained for the component was about 1×10^8 cycles which was beyond the safety limits and hence the design was safe against the fatigue loading.

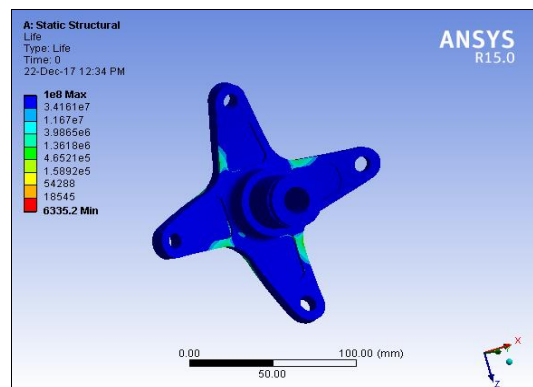


Fig 8: fatigue analysis

9. Rear knuckle:

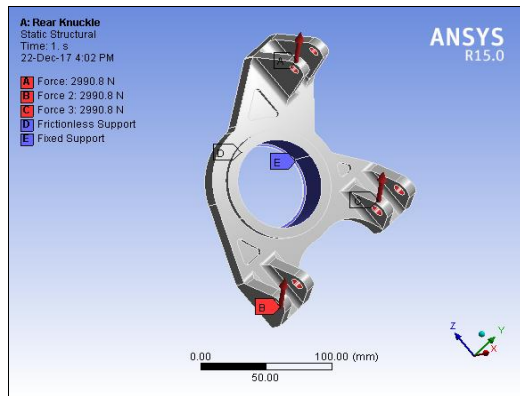


Fig 9: Force/Moment application

9.1 Stress:

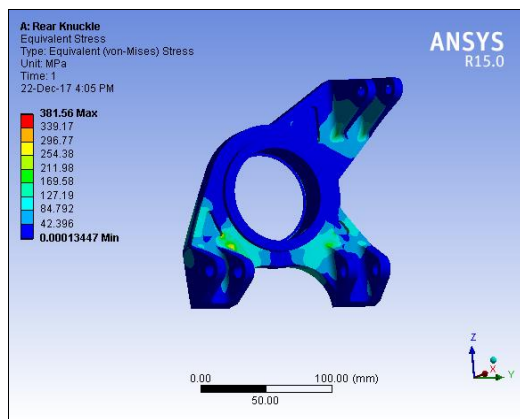


Fig 10: Stress Analysis

9.2 Deformation:

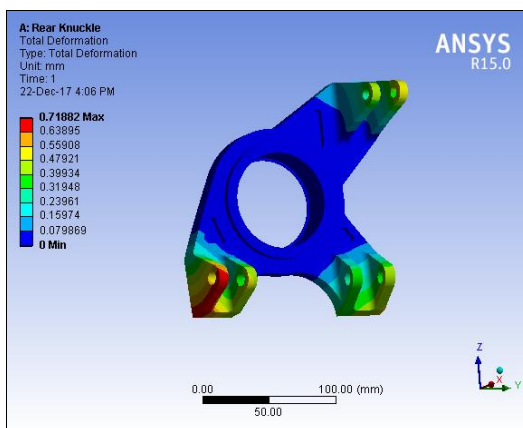


Fig 11: Deformation Analysis

9.3 Result:

Maximum stress occurring in knuckle is **381.56Mpa** and deformation is **0.71** having FOS **1.59**.

10. Rear H arm

Upper suspension arm is connected to the upper mounting of the knuckle it is also analyzed under the force of **2804N** laterally and **2600 N** on upward direction.

10.1 Parameters applied

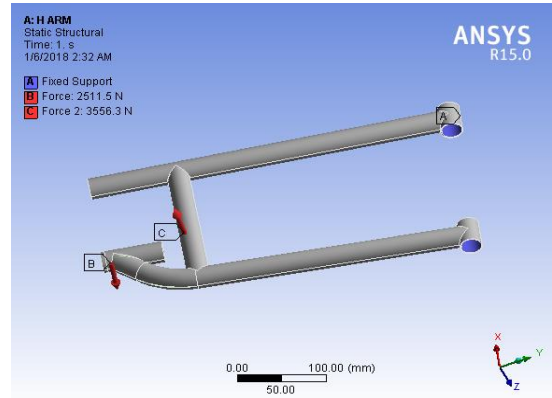


Fig 12: Force/Moment application

10.2 Stress

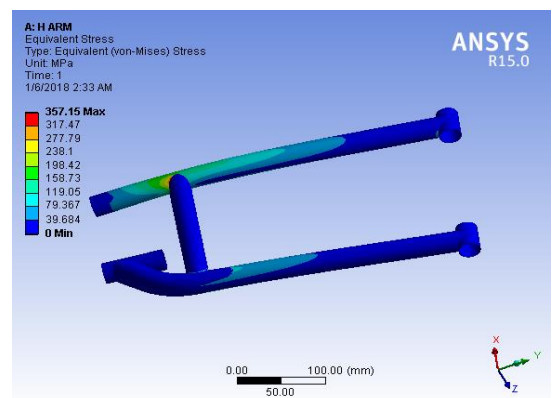


Fig 13: Stress Analysis

10.3 Deformation

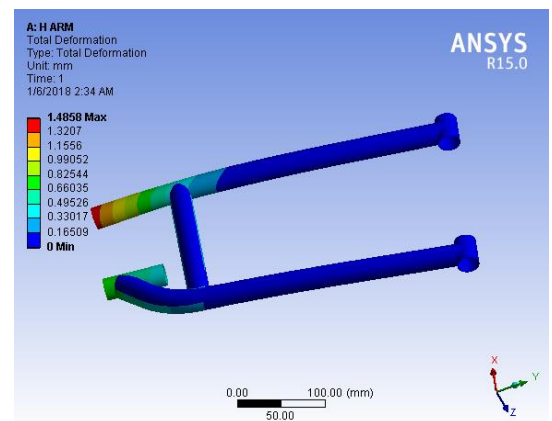


Fig 14: Deformation Analysis

10.4 Result

The max stress that acts on the arms is about **357.15MPa** and max deformation is about **1.4858mm**.

11. CONCLUSION

Suspension Deals with three C's comfort, contact, control. The agenda for designing the suspension was to enhance the dynamic stability of an ATV. The Front and Rear geometries were designed by using Lotus software with the help of CatiaV5 and analysis was carried out by the software 'Ansys'. The suspension parameters like Ground clearances, Wheelbase, Track width and other dynamic parameter such as Camber gain, toe gain etc. kept optimum so as to get minimum force interaction with the tire. Thus with Double Wishbone and H-arm linkage system the ATV was designed and analyzed successfully.

12. REFERENCES

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