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Design, Analysis and Optimization of front suspension wishbone of BAJA 2016 of All-terrain vehicle

[1] Reena Mishra, [2] Anand Baghe

[1] Research Scholar, Department of Automobile Engineering, RustamJi Institute of Technology, Tekanpur, Gwalior (M. P.)- 475005, India
[2] Assistant Professor, Department of Automobile Engineering, RustamJi Institute of Technology, Tekanpur, Gwalior (M. P.)- 475005, India

Abstract: The main objective of the paper only focused on the design and optimization front suspension wishbone of BAJA (ATV) vehicle. Analyze the entire suspension system for ATV vehicle for improving the stability and handling of wheel, the topic is focused to minimizing the sprung mass. To provide less weight, durable, efficient and less expensive. With the help of, solid work (2016) modeling software's and ANSYS (2015) workbench, the model was designed and optimized to serve the desired purpose. Material Was selected after conducting an extensive market survey and on the basis of Pugh`s concept method. This sequential approach was adopted for the front wishbone design of Baja vehicle and proved to be effective.

Keywords – Front Wishbone, Design Solid Work(2016), Anysis (2015),Lotus (Software)& Analyze.

ABSTRACT

The main objective of the paper only focused on the design and optimization front suspension wishbone of BAJA (ATV) vehicle. Analyze the entire suspension system for ATV vehicle for improving the stability and handling of wheel, the topic is focused to minimizing the sprung mass. To provide less weight, durable, efficient and less expensive. With the help of, solid work (2016) modeling software's and ANSYS (2015) workbench, the model was designed and optimized to serve the desired purpose. Material Was selected after conducting an extensive market survey and on the basis of Pugh`s concept method. This sequential approach was adopted for the front wishbone design of Baja vehicle and proved to be effective.

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1. INTRODUCTION

A vehicle suspension system is a linkage to allow the wheel to move relative to the body and some elastic element to support loads while allowing that motion. Most practical vehicles have some form of suspension, particularly when there are four or more wheels.

1.1 Types of Suspension System

1.2 The suspension system is classified into two main types [1]-

- Dependent Suspension System.
- Independent Suspension System

Dependent Suspension System

Following are the examples of dependent suspension system-
1. Leaf Spring Suspension.
Used in Heavy duty vehicles (trucks, bus, etc.)
2. Push and pull rod Suspension.
Used in F1 cars.
3. Anti-Roll Bar Suspension.
Used in passenger and luxury vehicles.

Independent Suspension System

Following are the examples of independent suspension system-
1. Macpherson Suspension.
Used in front suspension of most of the commercial cars.
2. Double Wishbone Suspension.

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Independent Suspension System

Following are the examples of independent suspension system-
1. Macpherson Suspension.
Used in front suspension of most of the commercial cars.
3. Double Wishbone Suspension.
Used in ATVs.

3. Trailing Arm Suspension.

Used in rear suspension of most of the commercial cars.

1.5 NEED OF DOUBLE WISBONE (A-arm) SUSPENTION SYSTEM

In automobiles, a double wishbone (or upper and lower A-rm) suspension is an independent suspension design using two (occasionally parallel) wishbone-shaped arms to 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 spring mount to the wishbones to control vertical movement. Double wishbone designs 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, scuff and more.

2. LETRATURE REVIEW

Y. Nadota, et al ,[1] Experimental device was developed to study fatigue phenomena for nodular cast iron automotive suspension arms. On the base of a detailed fracture analysis, it is shown that the major parameter influencing fatigue failure of casting components are casting defects: the High Cycle Fatigue behavior is controlled mainly by surface defects such as dross defects and oxides while the Low Cycle Fatigue is governed by multiple cracks initiated independently from casting defects.

Jong-Kyu Kim, et al [2], The shape of upper control arm was determined by applying the optimization technology.Work considers the static strength in the optimization process. The kriging-interpolation method is adopted to obtain the minimum weight satisfying the static strength constraint. The real experiments on 1/4 car is conducted to validate the FEM analysis. At last, the
correlation of each case about durability life is obtained.

Lihui Zhao, et al [3] Dynamic Structure Optimization Design of Lower Control Arm Based on ESL describes the new design obtained from the results of dynamic optimization based on ESL method can satisfy the actual requirements of manufacturing process. Structure dynamic optimization based on ESL method can avoid the dependency on personal experience, improve the accuracy of the optimal results, meanwhile decrease the time in product design.

Mr. Balasaheb Gadade et al Oct-2015 [4] Work mainly focused on the finite element based stress analysis of A – Type lower suspension arm. The main objective is to calculate working life of the component under static loading. The A – Type lower suspension arm was developed by using CAD software. Actual model was manufacture as per Design by using AISI 1040 material. The finite element modeling and analysis was performed by using HYPER MESH software. Mesh was created with 10 node tetrahedral element.

Himanshu Deshwal et al [5] Suspension system is one of the most imperative sub-systems of an automobile. Its elementary function is to quarantine the driver from the road shocks and bumps. The Subordinate function of suspension system includes load transfer lateral stability and provides adequate wheel travel fortifying ergonomics and driving comforts. The geometric modeling of the suspension system is done in CATIA V5 R22. The structural analysis is done in ANSYS Workbench 14.0 and the shape optimization is carried out in Lotus Engineering Analysis V5.01 Software.

3. RESEARCH METHODOLOGY

Design of wishbones is the preliminary step to design the suspension system. Initially, the material is selected using Pugh's Concept of Optimization. Based on the properties of the selected material, the allowable stress is calculated using shear stress theory of failure [14, 19, 20, 24]. The roll-center is determined in order to find the tie-rod length. The designed wishbones are modeled using software and then analyzed using Ansys analysis software to find the maximum stress and maximum deflection in the wishbone. (Akshay Mukesh Mutake3 et.al)

3.1 MATERIAL SELECTION OF FRONT WISHBONE

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 [16, 19, 22]. The material selection also depends on number of factors such as carbon content, material properties, availability and the most important parameter is the cost.

Initially, three materials are considered based on their availability in the market- AISI 1018, AISI 1040 and AISI 4130.[1, 4, 7] By using Pugh's concept of optimization, we have chosen AISI 1040 for the wishbones. The main criteria were to have better material strength and lower weight along with optimum cost of the material.(Akshay Mukesh Mutake3 et.al)[8]

PUGH `S CONCEPT

This is a method for concept selection using a scoring matrix called the Pugh Matrix. It is implemented by establishing an evaluation team, and setting up a matrix of evaluation criteria versus alternative embodiments. This is the scoring matrix which is a form of prioritization matrix[6, 8, 10]. Usually, the options are scored relative to criteria using a symbolic approach (one symbol for better than, another for neutral, and another for worse than baseline). These get converted into scores and combined in the matrix to yield scores for each option.

The material used for the required front wishbone was circular steel tubing with an outside diameter of 25 mm (1 inch), wall thickness of 3.mm (0.120 inch) and a carbon content of at least 0.18 (Baja SAE et al, 2014). The research was conducted to choose the best possible material. The choice of material was limited to steel as per SAE rules. The material was selected on the basis of cost, availability, performance and weight of material. After thorough research, two best materials were found for the designing of the front wishbone i.e.: Steel AISI 4130 Chromoly alloy and Steel AISI 1018.[11, 14, 16] The reasons for using round tubing (seamless) were it is lighter than square tube as smaller gauge sizes can be used to handle the same stress as a wider square tube and a round tube always out performs the square tube. Table 1.4.1 shows Mechanical properties of Steel AISI 1018 tube & Steel AISI 4130 chromoly amalgam. (Akshay Mukesh Mutake3 et.al)

Table.1: Mechanical properties of Steel AISI 4130 chromoly alloy(Akshay Mukesh Mutake3 et.al)[8].

Physical properties	Steel AISI 1018 Properties	Steel AISI 4130 Chromoly alloy
Density	$7861 kg/m^3$	$7861 kg/m^3$
Ultimate Tensile Strength	$4398.85*10^5 N/m^2$	$6701.70*10^5 N/m^2$
Yield Tensile Strength	$3702.48*10^5 N/m^2$	$4350.59*10^5 N/m^2$
Modulus of Elasticity	$199.94*10^5 N/m^2$	$2047.74*10^5 N/m^2$
Bulk Modulus	$1399.63*10^5 N/m^2$	$1399.63*10^5 N/m^2$
Shear Modulus	$799.79*10^5 N/m^2$	$1061.79*10^5 N/m^2$
Poisson's ratio	0.290	0.290
Elongation Break	15%	25.5%
Hardness brinell	126	197

Front wishbone of vehicle of cross section of pipe which is used for providing primary support for all systems and subsystems of vehicle. As per the market parameters three materials were selected for roll cage materials which are AISI 1018, AISI 1020 and AISI 4130 chromoly. Every material having different parameters and properties as follows, from them AISI 4130 is selected due to its advantages over other materials mentioned in following table.

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

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

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

For the elastic collision, the impact force is as calculated from eqn.(1)

$W_{net}=1/2mv^2_{final}=1/2(m)(v^2)_{initial}$ ………(1)

Where, W_{net} is net workdone on account of an inelastic collision.

$W_{net}= -1/2(m)(v^2)_{initial}$ …...(2)

But, W_{net}=Impact force*d……(3)

Where d is the distance travelled during impact.

(It is concidered that for static analysis,the vehicle comes to rest 0.1 seconds after impact. Therefor,for a vehicle which moves at 16.66m/s (or60km/h),the travel of the vehicle after impact is 1.66m from equ.(1),(2)and(3), we get)

II.Impact force determination by Accleration limit

Force =m × a

Where m=275kg and a=7.9 × 9.81m/s^2

(The moter insurance repair center` has analysied that baja car will see max. of 7.9G`s of force during impact. And also mass of the vehicle)

III. Applied 2G and 3G force :-

Force applied: 8829 N(2G) Stress produced:- 22.76N/mm^2 Deformation: 0.019 mm F.O.S.: 16.25	Force applied: 13243 N (3G) Stress produced: 186.29N/mm^2 Deformation: 0.050 mm F.O.S.: 1.28

IV. The most common example is in a vehicle's suspension, where it is used to describe the displacement and forces in the springs and shock absorbers. The force in the spring is (roughly) the vertical force at the contact patch divided by the motion ratio, and the wheel rate is the spring rate multiplied by the motion ratio squared.

$$\mathbf{MR} = \frac{\textbf{Spring displacement}}{\textbf{Wheel displacement}}$$

Wheel Rate= SpringRateMR^2

It basically depends on 2 factors in case of front suspension system

a) Bending Moment: To reduce the bending moment the strut point should be near to the wheel.

b) Suspension Stiffness: The suspension tends to get stiff when its inclination of the shock absorber to horizontal tends to 90 deg.

STEP 1: MOTION RATIO

In developing a basic spring setup, first step is determining motion ratio.

MR = Motion ratio = 0.63

MR = $\left(\frac{d2}{d1}\right)$

Where,

d_1 = Distance from spring centerline to control arm inner pivot center (in) or (mm) = ?

d_2 = Distance from outer ball joint to control arm inner pivot center (in) or (mm) = 470mm

D1= $\left(\frac{470}{0.63}\right)$ =746.03mm

ACF = 30^0

Angle correction factor = ACF = (Cosine<(30^0))
Spring angle from vertical (see diagram)

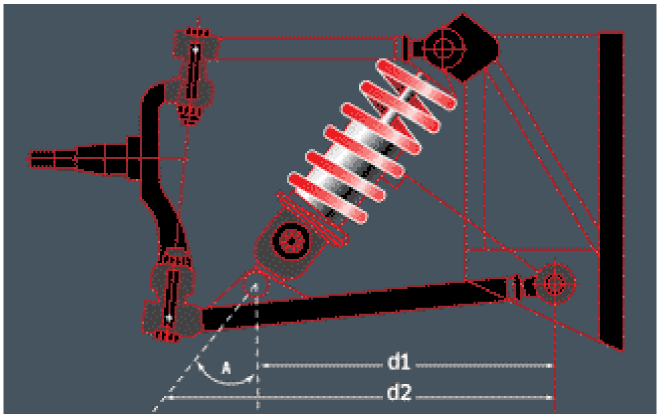


Fig.1: (A-ARM SUSPENSION)

The motion ration is a lever arm effect of the control arm acting on the spring. If the spring is mounted at an angle, the reduced motion of the spring must also be taken in account.

STEP 2: WHEEL RATE

Wheel Rate is the actual rate of a spring acting at the tire contact patch. This value is measured in lbs/inch or N/mm, just as spring rate is. The wheel rate can be determined by using the formula below.

Wheel Rate (non beam)

WR Wheel Rate (lbs/in) or (N/mm)
C = Spring Rate (lbs/in) or (N/mm)
MR =Motion Ratio =0.63
ACF is on Factor=(Cosine<(30^0)) Angle

WR = (MR)^2(C)(ACF)

STEP 3 SUSPENSION FREQUENCIES

Suspension Frequency refers to the number of oscillations or "cycles" of the suspension over a fixed time period.
When a load is applied to the vehicle

Suspension Frequency

SF = Suspension Frequency (cpm)*=SF = $(187.8)\left(\sqrt{\frac{WR}{sprung\ weight}}\right)$

CPM=cycles per minute

Suspension frequency in hertz divide by 60 =$\frac{SF}{60}$

WR =Wheel Rate (lbs/in) or (N/mm) = $(\frac{SF}{187.8})$^2(sprung weight)

Sprung weight= Vehicle corner weight less un-sprung weight

Calculation of Wheel Rate for a given frequency

WR = Wheel Rate (lbs/in) or (N/mm) (see step 2) = $(\frac{48}{187.8})$^2(sprung weight) =0.25559N/mm
Sprung weight= Vehicle corner weight less un-sprung weight

Calculation of Spring Rate needed for a given Wheel Rate

C = Spring Rate (lbs/in) or (N/mm)

WR = Wheel Rate (lbs/in) or (N/mm) (see step 2)

C= $(\frac{WR}{(MR)^2(ACF)}) = (\frac{0.25559}{(0.63)^2(Cosine\angle(30))})$ = 0.3218 N/mm

MR Motion Ratio

ACF =Angle Correction Factor

V. Applied Boundary conditions front lower wishbone arm :

The total load (2000N) is applied over the front wishbone arm, which contains 1825 nodes. So load on each node is given by Force on each node present at top surface of side rail= (total load)/ (number of nodes) Force on each node = 2000/1825 Force per node =1.095

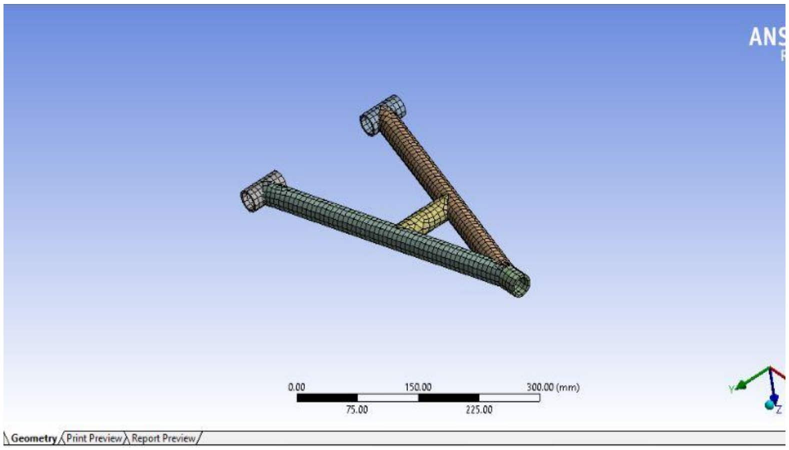


Fig.2 : Front lower wishbone meshing part

Finite element analysis has also been conducted on the front arms. The stresses created in the part can be seen in Figure. The biggest reason for choosing this design is that it only requires one piece, using a simple jig, to be fabricated. It has been determined that the tubing used for the suspension arms will be ASTM 106a steel. It will be 1” diameter with 3” wall thickness. This was determined after comparing the weight and material properties for several sizes of tubing’s.

4.1 Front Suspension

For front suspension there is many choices to select but from them double wishbone type of suspension had been selected because of its high load handling capacity and rigid support to the wheel geometry. It is ease to control Camber angle, Castor angle and King Pin inclination angle with double wishbone system.[1]For this the parameters which are considered are as follows;

Table.2: Specification of ATV Baja suspension system

i.	Length of spring	250mm
ii.	Total length (spring + damper)	550mm
iii.	Weir diameter	8mm
iv.	Mean coil diameter	75mm
v.	Allowed travel of spring	75mm
vi.	No. of active turns	20
vii.	Total no. of turns	22
viii.	Spring rate	11.42N/m
ix.	Wheel rate	4.532N/m
x.	Motion ratio	0.63
xi.	Camber angle	2 degree
xii.	Caster angle	+5degree
xiii.	Toe in	5 degree

Table. 3 : Front suspension system(Force analysis)

S.No	Condition	G force	
1.	Bump	3 g vertical	2g horizontal
2.	Drop	4g	1.5 g
3	Cornering	0.75 g	1.25g

5. Design introduction

Analysis of wishbone is performed in Ansys 2015. Analysis Software is necessary in order to determine the induced maximum stress and maximum deflection in wishbones. For analysis, wishbones are first needed to be molded in software. The modeling of wishbones is done in solid work modeling 2016 software.

5.1 Solid works Modeling

Pro-E is modeling software which allows 3D- modeling and 2-D drafting of elements.

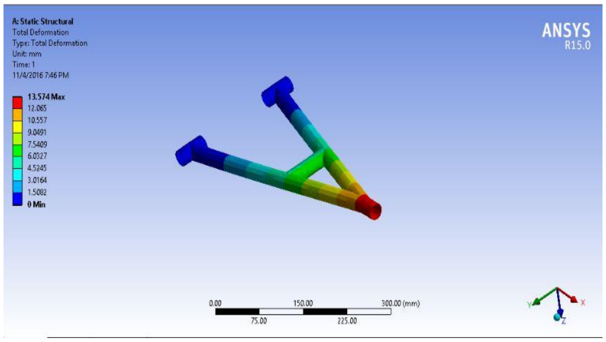


Fig.7 : Applied force Front lower arm wishbone analysistotal deformation (in anysis)

(a) Total deformation of front wishbone lower arm (1mm)

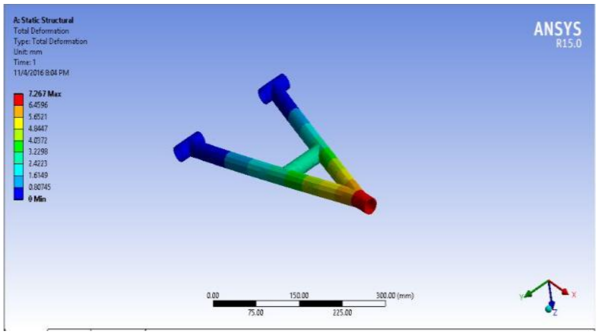
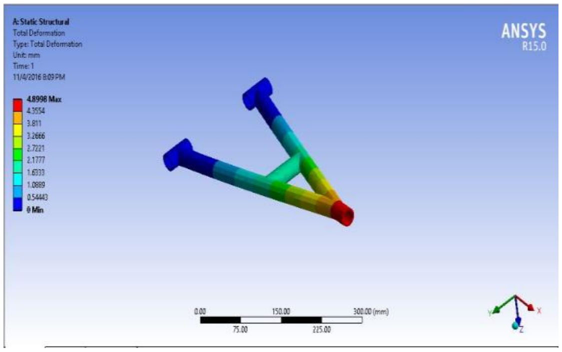


Fig (b) Total deformation of front wishbone lower arm (2mm)



(c) Total deformation of front wishbone lower arm (3mm)

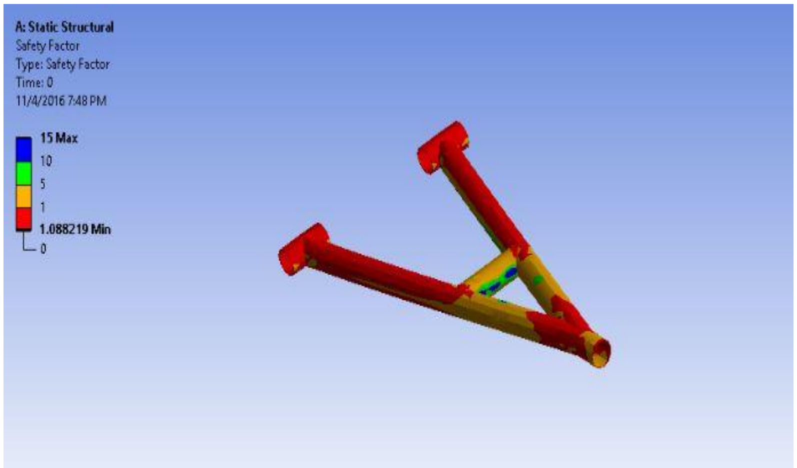
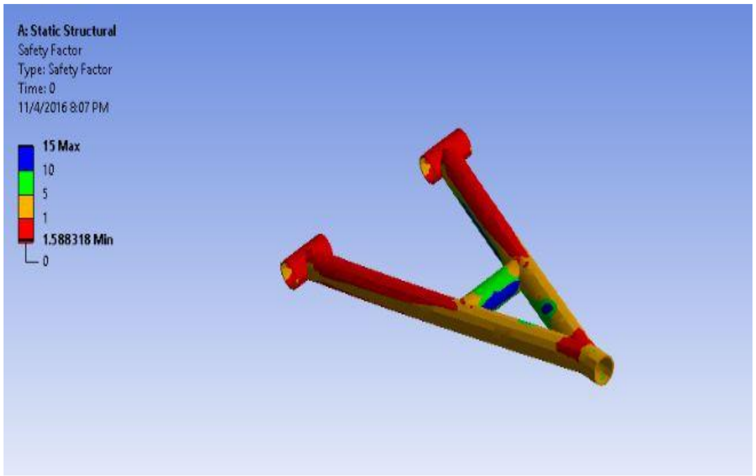
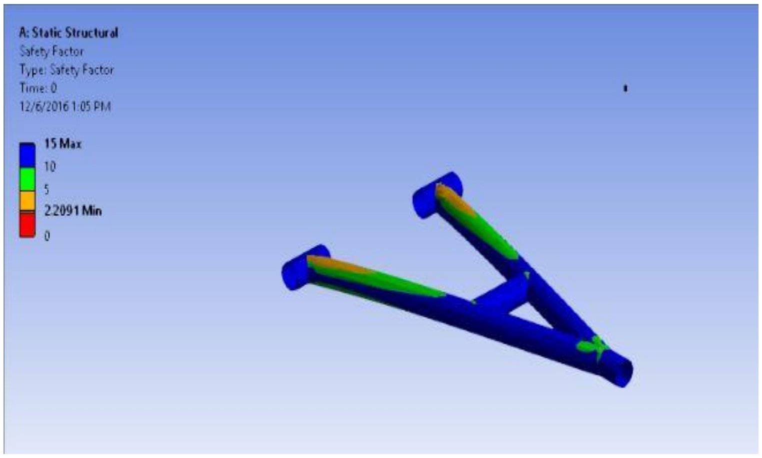


Fig.8 : Applied force front lower arm wishbone analysis Factor of safety (in anysis)

(a) Factor of safety of front wishbone lower arm (1mm)



(b) Factor of safety of front wishbone lower arm (2mm)



(c) Factor of safety of front wishbone lower arm (3mm)

RESULT ANALYSIS

Table.4:-Result

Thickness of front wishbone (mm)	Stress (N/mm) of front wishbone	Total deformation (mm) of front wishbone	Factor of safety
1mm	2255.4 N/mm	1.5082 mm	1.08
2mm	976.02 N/mm	0.8074 mm	1.58
3mm	535.69 N/mm	0.5443 mm	2.20

CONCLUSION

1. From the structural analysis of the front wishbone arm the 3 mm thickness of arm is found out to be suitable for the current application of BAJA SAE ATV.

2. In comparison to 1 mm thickness of front wishbone arm the stress concentration reduced by 76.24% in 3 mm thickness of front wishbone arm of the ATV Baja vehicle.

3. In case of 3 mm thickness the total deformation of the front wishbone arm is decreased by 63.9%.

4. The factor of safety of the suspension system is increased by 68.56% by using 3mm thickness in comparison to 1 mm.

5. Reduce weight cause less fuel consumption of the ATV Baja vehicle.

FUTURE SCOPE

1. The suspension system can be further modified for decreasing the weight and cost. Current research considers material for the front wishbone suspension system. Further research should explore the effect of material modeling to reduce the weight of the roll-cage.

2. Further research has to be carried out towards material modeling has to be done to reduce the generated accelerations and reaction forces on the roll cage structure and to find out the optimum material properties for the roll cage structure.

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[5]Deshwal Himanshu #1, Jangid Rahul #1, Mehra Kamlesh #1 , Singh Yashpal #1 , Singh Neha #1, Yadav Neetu #1 Finite Element Analysis and Geometric Optimization of Double Wishbone Pushrod Actuation Suspension System of "SUPRA SAEINDIA" Vehicle ISSN: 2348 – 8360 volume 3 Issue 5 – (May 2016)