



Design of Upper A Arm of Double Wishbone Suspension System

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Authors' contributions

This work was carried out in collaboration among all authors. Author MA designed the study, performed the statistical analysis, wrote the protocol and wrote the first draft of the manuscript. Authors KSP and JB managed the analyses of the study. Author JB managed the literature searches. All authors read and approved the final manuscript.

Article Information

DOI: 10.9734/JERR/2020/v14i217118

Editor(s):

(1) Dr. Djordje Cica, University of Banja Luka, Bosnia and Herzegovina.

Reviewers:

(1) Bujoreanu Carmen, Technical University Gheorghe Asachi Iasi, Romania.

(2) Swati K. Dhamale, Savitribai Phule Pune University India.

Complete Peer review History: <http://www.sdiarticle4.com/review-history/58320>

Original Research Article

Received 06 June 2020

Accepted 30 June 2020

Published 04 July 2020

ABSTRACT

Every All-Terrain vehicle right now uses independent suspension system which consists of double wishbones connected to all the tires. As All-Terrain vehicles generally operated on different road conditions it is an absolute necessity to have a robust design of wish bones. A good deformation rate and good FOS determines how good a design. In this study we have designed three types of upper wishbones in Solid Works whose suspension geometry based on wheel base, track width, roll center and pitch center of the vehicle is validated in LOTUS software and the following graphs of camber, castor, toe, kingpin inclination are obtained. Linear static structural analysis is performed on all the three types designed in Ansys software and total deformation rate, equivalent stresses generated and FOS is calculated and the based on the results the best design is used for the vehicle. The design provided greater suspension travel, reducing the unsprung mass of the vehicle, maximizing the performance of the suspension system of the vehicle and better handling of vehicle while cornering. The design is used in SAE BAJA 2020 competition Conducted in Chitkara University Punjab.

Keywords: All-Terrain vehicle; wishbone; suspension; analysis; suspension geometry; design.

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1. INTRODUCTION

In the study by Hazem Ali Attia [1] the suspension system of a vehicle refers to the group of mechanical components that connect the wheels to the frame or body. Suspension systems improve vehicle ride and handling along with passenger safety and comfort. The wheels are supported by short upper and lower hinged arms holding them perpendicular to the road. In the study by V. V. Jagirdar, M. S. Dadar and V. P. Sulakhe [2] "Wishbone Structure for Front Independent Suspension of a Military Truck", a coil spring uses the support of either the upper or the lower arm to provide dampening. By shortening the upper arm wheel camber could be controlled to prevent edge loading tires while cornering.

1.1 What are the Advantages of Double Wishbone Suspension?

In the study by Hemim M. M., Rahman, M. M. and Omar R. M. [3] "Dynamic analysis of vehicle arm based on finite element approach" Double wishbone suspension offers drivers a smoother driving experience, especially on the roads which have irregularities as it doesn't affect wheel alignment. It gives technicians flexibility to adjust parameters like camber, caster and toe to meet the requirements of the track or road. Also, as there isn't a shock absorber sticking out of the top of the wheel hub, this type of suspension requires less vertical space. In the study by K. V. Reddy, M. Kodati, K. Chatra, and S. Bandyopadhyay [4] "A comprehensive kinematic analysis of the double wishbone and MacPherson strut suspension systems," this means you don't need to raise the ride height, negatively affects handling by increasing the center of gravity.

1.2 Double Wishbone Suspension System

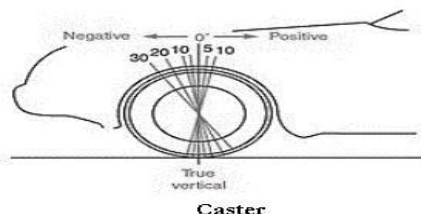


Fig. 1.1. Castor

In the study by A. Tandel, A. R. Deshpande, S. P. Deshmukh, and K. R. Jagtap, "Modeling,

analysis and PID controller implementation on double wishbone suspension using sim mechanics and Simulink" [5] Caster is the angle of the steering axis from the vertical as viewed from the side and is shown in Fig. Positive caster is defined as the steering axis inclined toward the rear of the vehicle.

Camber is the angle of the tire/wheel with respect to the vertical as viewed from the front of the vehicle, as shown in Fig. Camber angles usually are very small, on the order of 1°. Positive camber is defined as the top of the wheel being tilted away from the vehicle, whereas negative camber tilts the top of the wheel toward the vehicle.

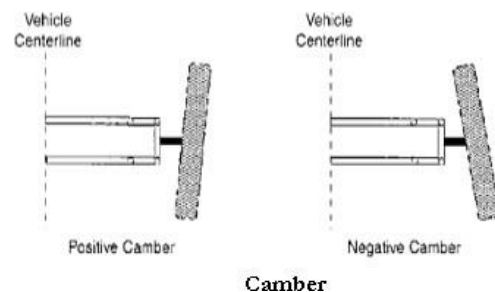


Fig. 1.2. Camber

King Pin axis inclination (a.k.a. Steering Axis Inclination) is the angle from the vertical defined by the centerline passing through the upper and lower ball joints. Usually, the upper ball joint is closer to the vehicle centerline than the lower, as shown in Fig.

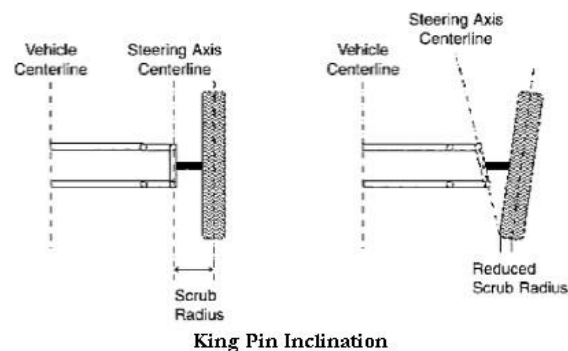


Fig. 1.3. King Pin Inclination

Toe is defined as the difference of the distance between the leading edge of the wheels and the distance between the trailing edge of the wheels when viewed from above. Toe-in means the front of the wheels are closer than the rear; toe-out

implies the opposite. Figure shows both cases.

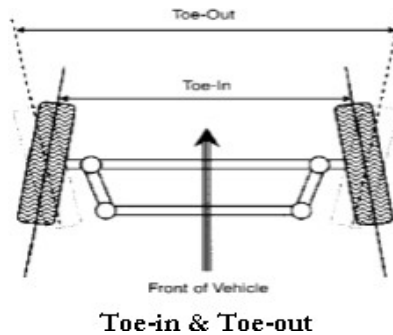


Fig. 1.4. Toe-in & toe-out

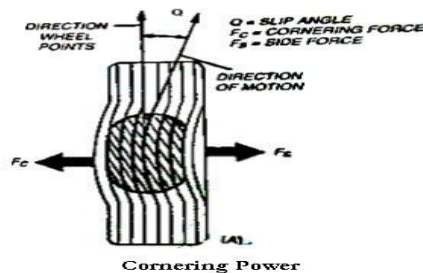


Fig. 1.5. Cornering Power

In the study by S. Chepkasov, G. Markin, and A. Akulova, "Suspension Kinematics Study of the 'Formula SAE' Sports Car," [6] Cornering power is the angle through which wheel has to turn to sustain side force is called slip angle and force produced due to this (at right angle to wheel plane) which counters side thrust is called 'cornering force'. The ratio of 'cornering force' to 'slip angle' is called cornering power.

In the study by Kamesh Jagtap and Yogesh Rathod, "Suspension system for an all-terrain vehicle: A review" [7] Roll center can be defined in two different ways, one based on geometric roll center (kinematic roll center) and another based on force roll center. While designing of suspension system we consider geometric based definition, because it plays a very important role in deciding the wishbone arm lengths and the geometry of wishbones. Geometric roll center also helps in determining length of tie rods, it is expected that both upper and lower wishbones and tie rods in a suspension system follow same

arc of rotation while cornering whose center is known as instantaneous center.

1.3 Problem Definition

The aim of this study is to design and analyze the upper A arm of a double wishbone suspension system. The process includes study of the suspension parameter in LOTUS. These parameters are optimized for the desired performance through iterative procedures. The best design is then modeled in Solid Works and analyzed for strength in Ansys.

2. METHODOLOGY

Once the changes are made such that all the parameters are confined in the optimal region, motion study is done on the system to analyze the behavior of the setup with actual road conditions. We analyze the graphs to verify and attain the values of the above parameters so as to keep them in range.

2.1 Formulation

We first determine the chassis dimensions, from this step, we decide the track width and finally the turning radius. From these steps, we get the total available space to accommodate the suspension arms. Hence, we consider the fixed points (hard points) and set up the theoretical values into LOTUS software. LOTUS suspension software is mainly used for designing the hard points such that the required kinematic behavior is achieved. Any number of results can be displayed graphically against bump motion, roll motion or steering motion. These results are updated in real time as the suspension hard points are moved. This software uses different templates to identify specific 3D suspension types. The lengths of both upper and lower control arms are fixed according to iterations done in the LOTUS software. At first sample lengths are taken and designed, different graphs of camber, caster, over steer, under steer and kingpin inclinations are obtained from LOTUS software and according to the results from the graphs the shapes and lengths of the control arms are fixed. The lengths are then modeled in the CAD software and then analysis is performed for forces and torques for verifying if they will sustain the loads.

Step 1 - The design process starts by first taking approximate dimensions of the arms and other components on paper.

These dimensions are derived from

- Wheelbase
- Track width
- Approximate roll center
- Approximate pitch center

Step 2 - Based on these approximate values, the design is formulated in the LOTUS software and the draft design is analyzed using the available graphs of

- Camber
- Castor
- King pin inclination
- Toe

Step 3 - These parameters are tested with respect to

- Bump
- Steer
- Roll

Step 4 - We want the values to be in the following these conditions

- Negative camber when wheel is on the ground
- Toe should have a max variation of 2 degrees (toe in)
- Castor should be positive for self-straightening
- Scrub radius must be a maximum of 20mm

Step 5 - Perform the steps 2, 3 and 4 until the desired suspension parameters are obtained.

Table 1. Spatial coordinates of the front suspension arms

Points	X(mm)	Y(mm)	Z(mm)
Point 1: Lower wishbone front pivot	3982	241.30	355.60
Point 2: Lower wishbone rear pivot	4202	241.30	355.60
Point 3: Lower wishbone outer ball joint	4089.50	583.30	247.10
Point 5: Upper wishbone front pivot	3942	241.30	442
Point 6: Upper wishbone rear pivot	4242	241.30	442
Point 7: Upper wishbone outer ball joint	4095.50	566.70	347.10
Point 8: Damper wishbone end	4146.50	446.50	327.50
Point 9: Damper body end	4180	289.65	708.45
Point 11: Outer track rod ball joint	4052.74	559.36	278.56
Point 12: Inner track rod ball joint	4041.50	203.20	384.25
Point 16: Upper spring pivot point	4180	328.09	801.54
Point 17: Lower spring pivot point	4145	446.50	287.50
Point 18: Wheel spindle point	4095.50	575	292.10
Point 19: Wheel centre point	4092.50	647.70	292.10
Point 20: Part 1 C of G	4030	440	195
Point 21: Part 2 C of G	4170	520	450
Point 22: Part 3 C of G	4230	525	220
Point 23: Part 4 C of G	4130	720	275

This is the table which shows the positions of all the hard points in the suspension geometry. These points are adjustable and these changes vary the suspension geometry. This variation gives different results on analysis.

Steer is a phenomenon in which the vehicle takes a turn. While taking the turn, the hard points also change it's position thus causing a change in the suspension parameters such as toe, camber, caster and king pin inclination. These changes can be viewed and visualized with the help of these graphs. The goal is to minimize these variations in the suspension parameters and through an iterative procedure, the hard points are reset to get the desired toe (0 degree change), camber (camber should reduce when the wheel rises from the ground), caster (camber should reduce when the wheel rises from the ground), caster

(always positive, max 10 degrees) and king pin inclination.

Roll is a condition in which the suspension system is analyzed for the vehicle's rolling nature when it passes over an obstacle. The aim is to observe the variation in the suspension parameters during the rolling condition of the vehicle. These graphs depict the variation of suspension parameters like toe, camber, caster and king pin inclination. The goal is to minimize these variations in the suspension parameters and through an iterative procedure, the hard points are reset to get the desired toe (0 degree change), camber (camber should reduce when the wheel rises from the ground), caster (always positive, max 10 degrees) and king pin inclination.

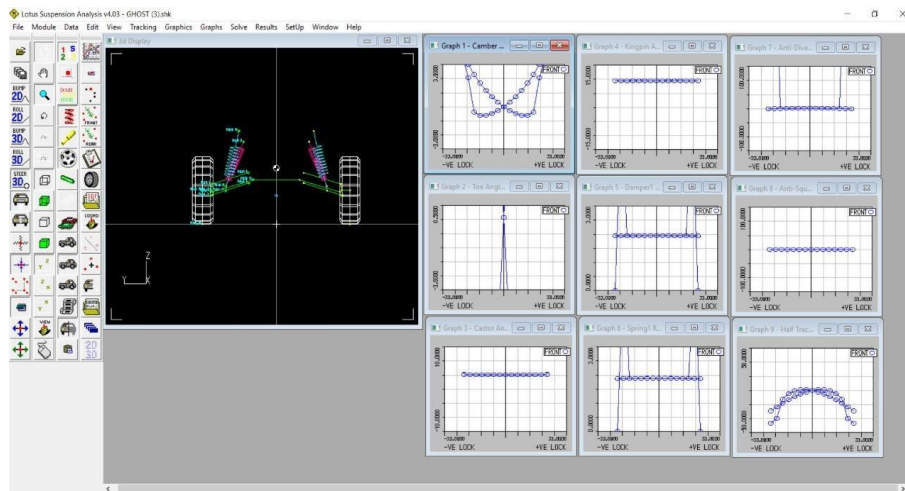


Fig. 2. Analysis of the graphs in steer

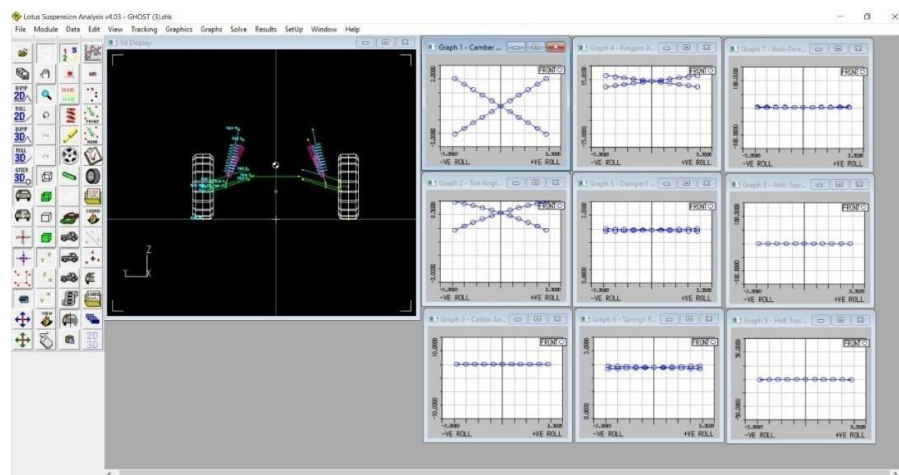


Fig. 3. Analysis of the graph in roll

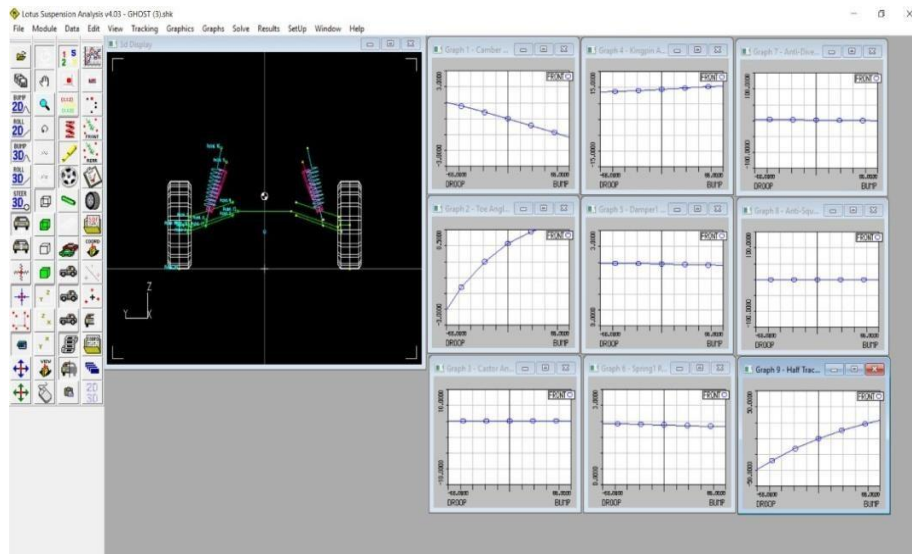


Fig. 4. Analysis of the graph in bump

Bump is the final test condition where the suspension parameters are observed and corrected for the required values. These graphs show the variation of the parameters when the vehicle passes over a bump (an obstacle). Here too, the aim like in the previous two cases, is to set the hard points in such a way so as to ensure that all the parameters are as desired.

The ultimate aim is to ensure that all the suspension parameters are within the desired range in all the three conditions (bump, steer and roll).

1 inch (outer dia.) AISI 4130 rods using Weldments tool of Solid works. We considered

three designs for the arms namely, straight arms, curved arms and W arms, we analyzed the forces developed on it and chose the best one. The bushes are then developed with another tube of thickness such that the inner diameter of the bush is able accommodate a M10 bolt and nut system to hold it securely in place with the knuckle.

The lower arm has an additional component in form of suspension clamps to connect the lower arm with the suspension damper. This is the only difference between the upper and lower arm. These clamps have a pivotal role to transmit forces from the wheel to the suspension damper.

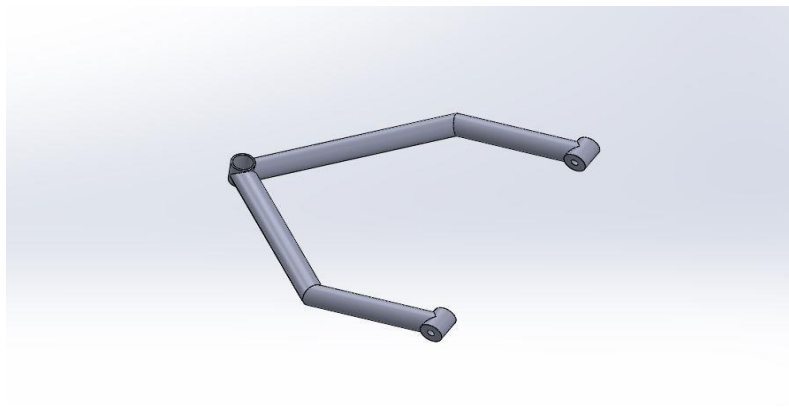


Fig. 5. Upper A-arm (W shape)

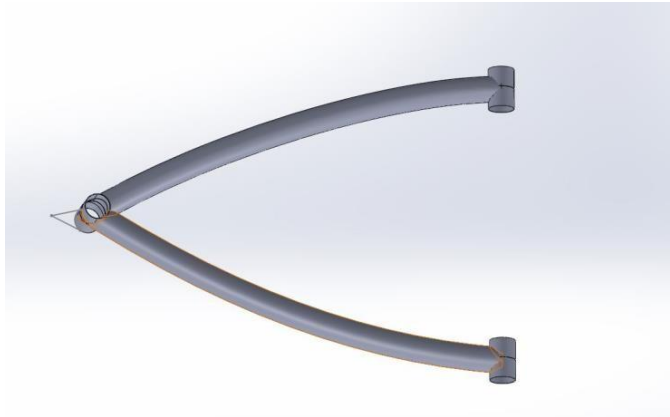


Fig. 6. Upper A arm with curved arms

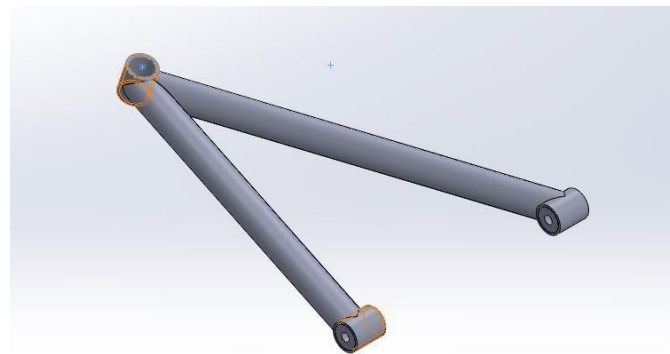


Fig. 7. Upper A arm with straight arms

2.2 Analysis

Analysis work was carried out in Ansys R2020. This software uses Finite element method to find out the results. The finite element method is the most widely used method for solving problems of engineering and mathematical models. It includes the use of mesh generation techniques for dividing a complex problem into small elements, as well as the use of software program coded with Finite element method algorithm. Static structural analysis was implemented on the above 3 designs. A static structural analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. The assumptions which were made before conducting analysis were the loads applied are time independent, the direction of loads does not change.

AISI 4130 steel is selected as the material to be used. It is a type of stainless steel. Mesh is taken

as 1 mm and at the complicated areas of the Arm size of 0.5 mm is taken. Two cases for performing analysis are taken. Case 1 is when the vehicle is falling from a height so that height impact forces will be generated on the wishbone so the two ends of the wishbone which are connected to chassis are taken as fixed support and the impact load is taken on the end which is connected to knuckle. Case 2 is when the vehicle is suddenly stopped, the braking forces will be acted so that the end which is connected to the knuckle is taken as the fixed support and a force which is parallel to the direction of chassis is applied on the two ends which are connected to chassis. Total deformation, equivalent Von Mises stresses and factor of safety are evaluated.

Selection of better type of A-arms is done if the total deformation rate and equivalent stress are less in both the above cases. There are various types of stress life curves to consider for calculating FOS. Goodman curve is used to find

out FOS. Finally, the design is validated by good FOS number. FOS tells how good the design can withstand the loads acting upon it.

The analysis is carried out for the following cases

Case 1: Sudden impact force when car lands of a single tire:

For this case force = $3 \times g \times \text{mass of the car}$
 $= 3 \times 9.81 \times 200 = 5800 \text{ N}$

Case 2: Torque of arms when sudden brake is applied:

For this case torque = braking force \times distance between A arms

$$= 4500 \times 0.4$$

$$= 1800 \text{ Nm}$$

Static structural analysis is done on all the types to find out total deformation, equivalent stress and factor of safety

Three types of wishbone that are designed are:

- A. Straight A-arm
- B. Curved A-arm
- C. W shaped A-arm

Straight A-Arm Analysis

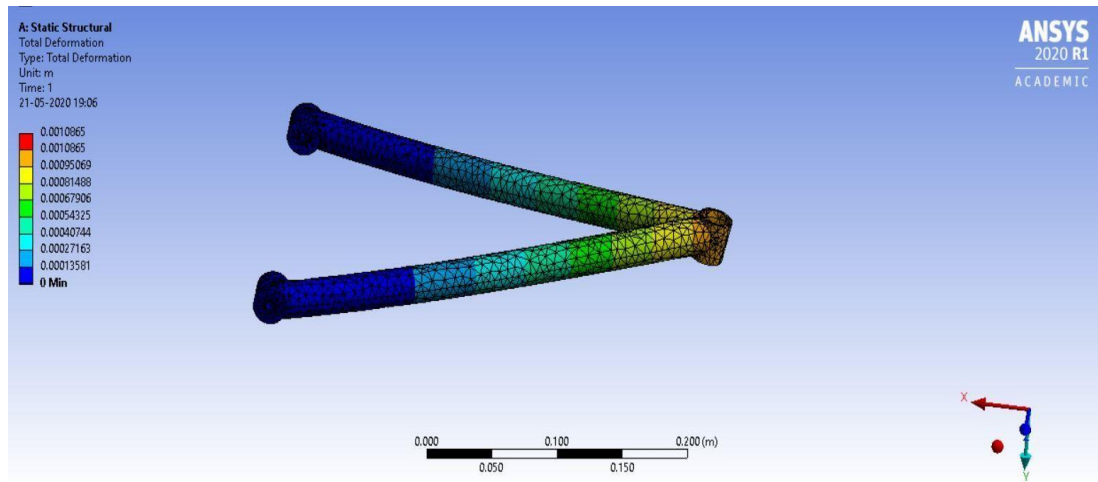


Fig. A.1. Total Deformation when case 1 load is applied

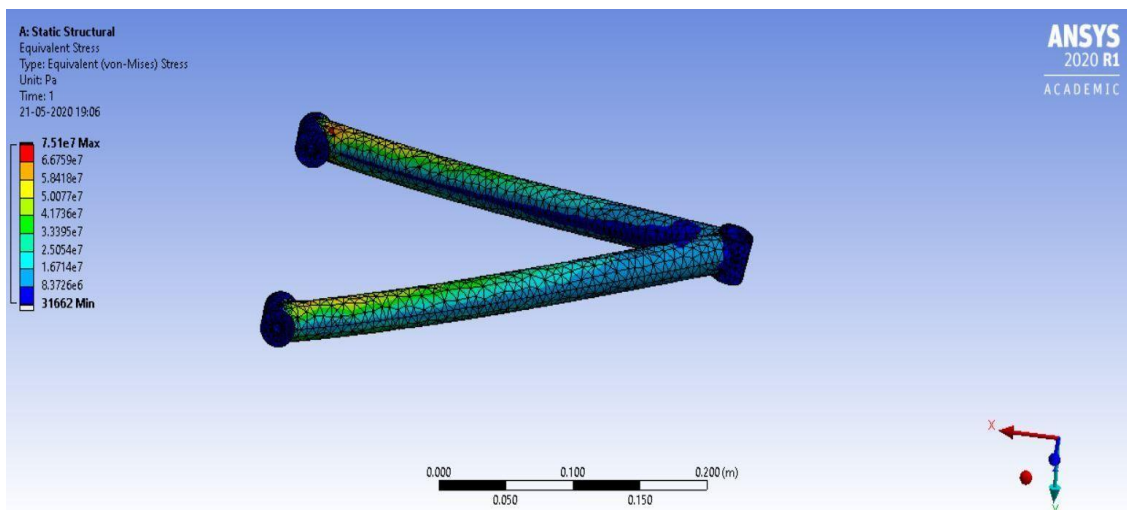


Fig. A.2. Equivalent stress when case 1 load is applied

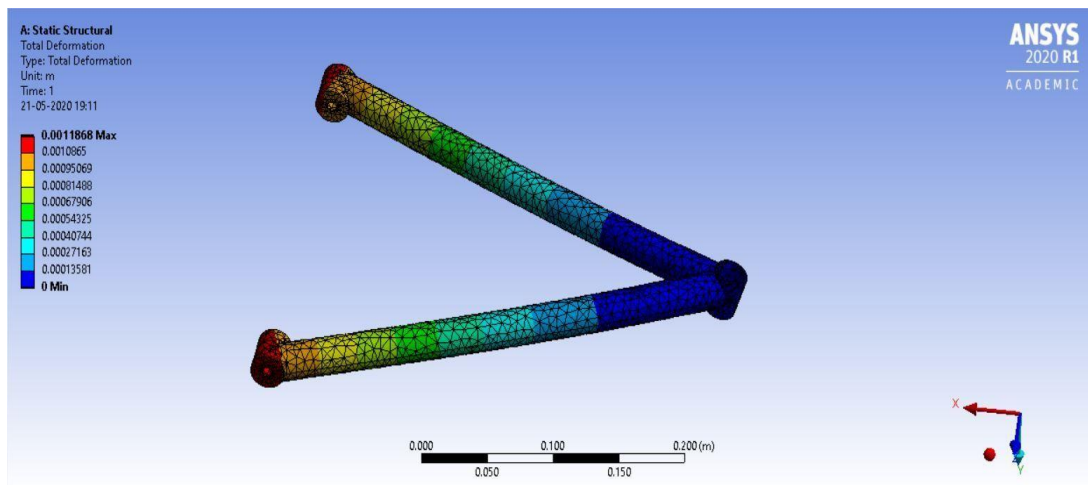


Fig. A.3. Total Deformation when case 2 load is applied

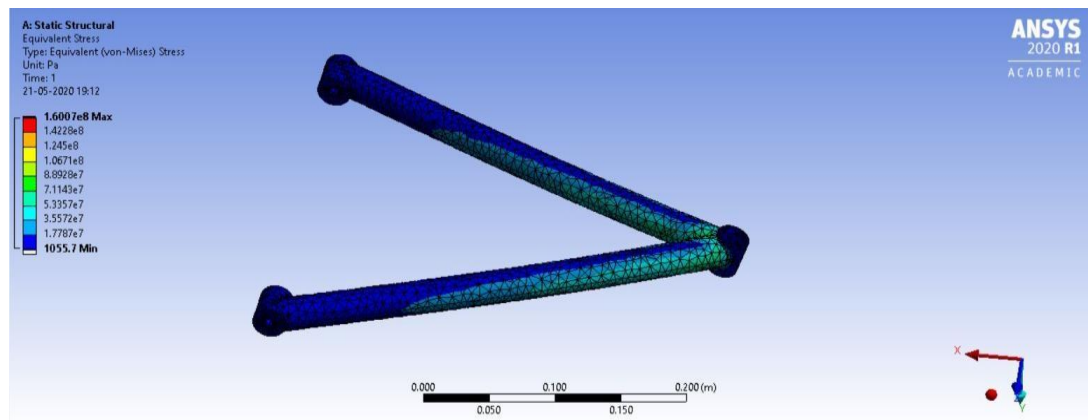


Fig. A.4. Equivalent stress when case 2 load is applied

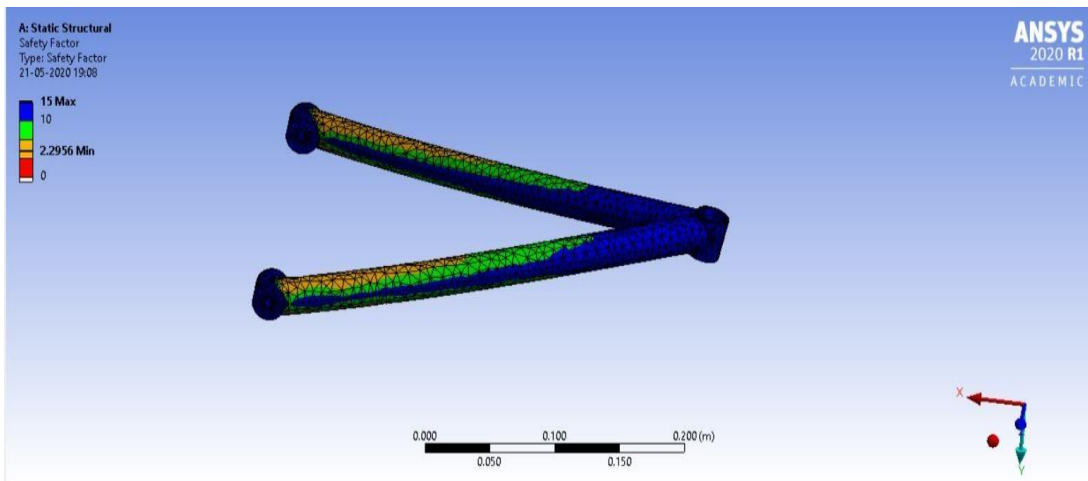


Fig A.5. Factor of safety

Curved A-arm Analysis

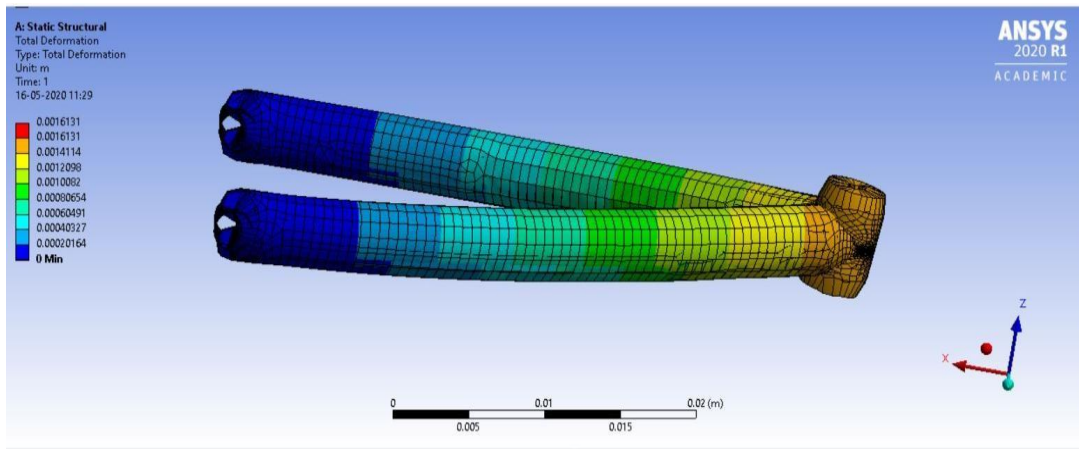


Fig. B.1. Total Deformation when case 1 load is applied

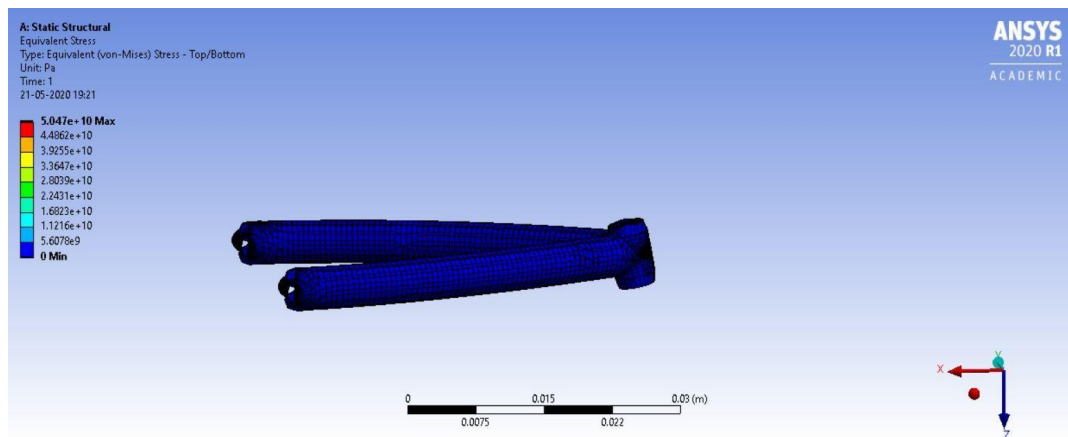


Fig. B.2. Equivalent stress when case 1 load is applied

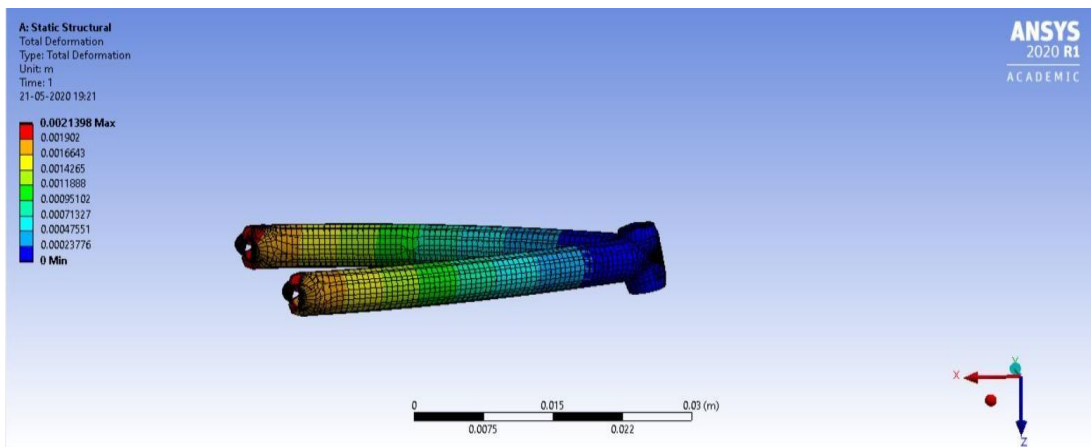


Fig. B.3. Total Deformation when case 2 load is applied

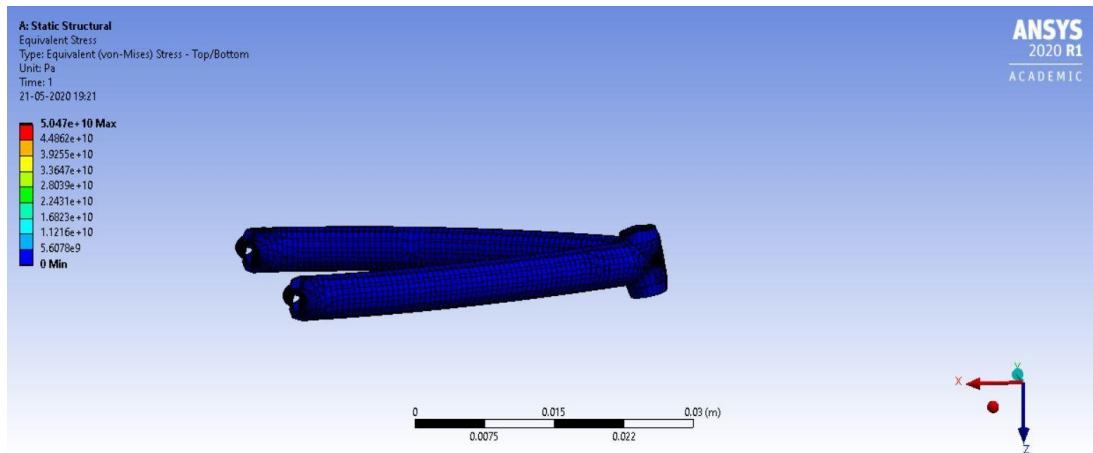


Fig. B.4. Equivalent stress when case 2 load is applied

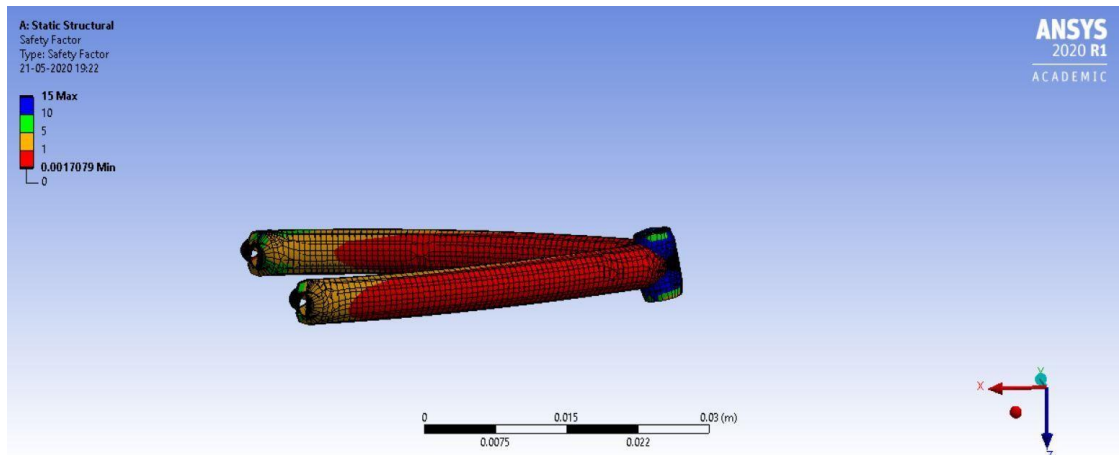


Fig. B.5. Factor of safetyW shaped A-arm analysis

W shaped A-arm Analysis

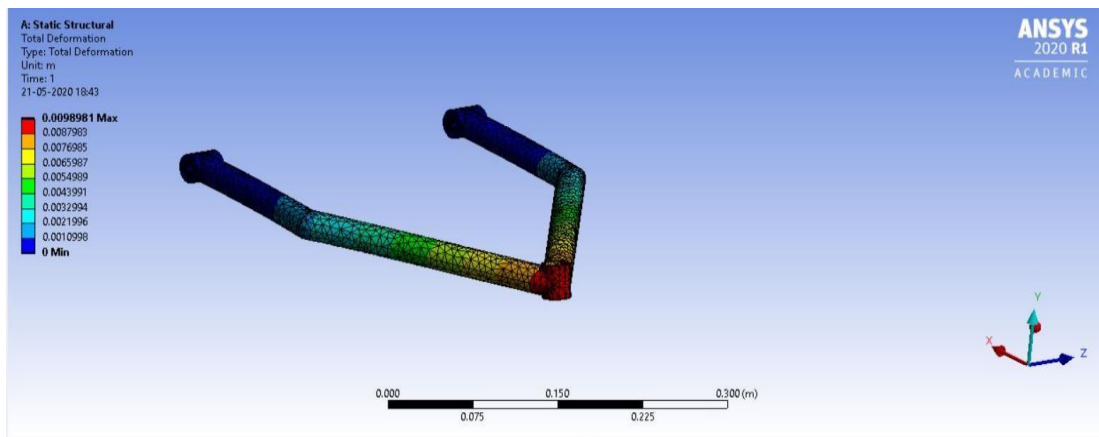


Fig. C.1. Total Deformation when case 1 load is applied

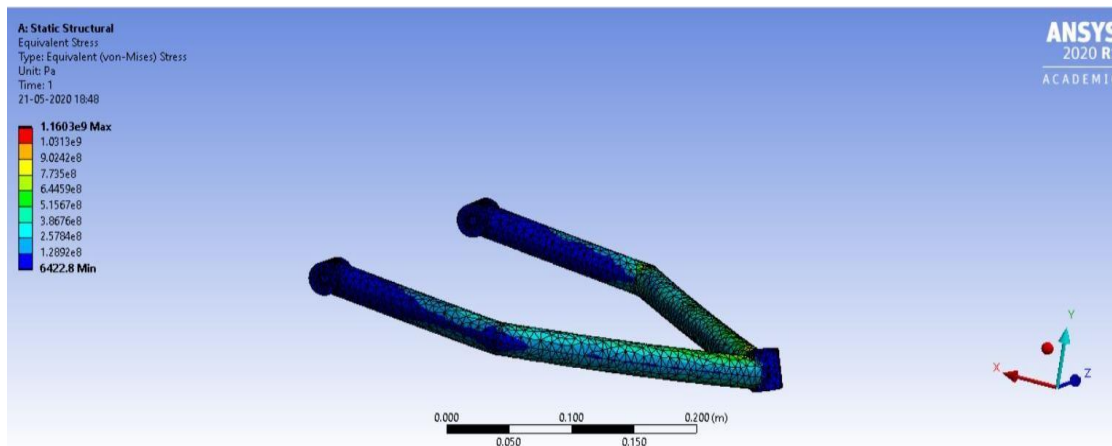


Fig. C.2. Equivalent stress when case 1 load is applied

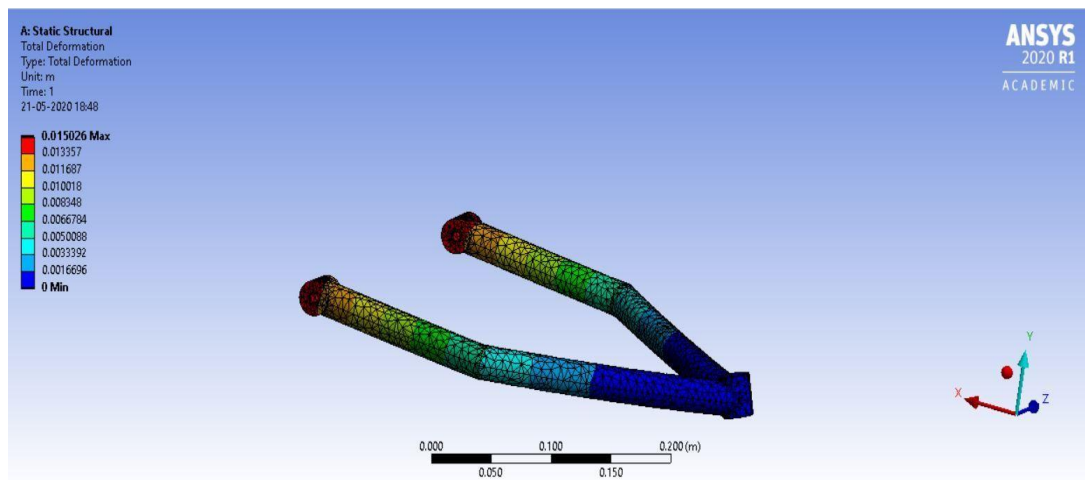


Fig. C.3. Total Deformation when case 2 load is applied

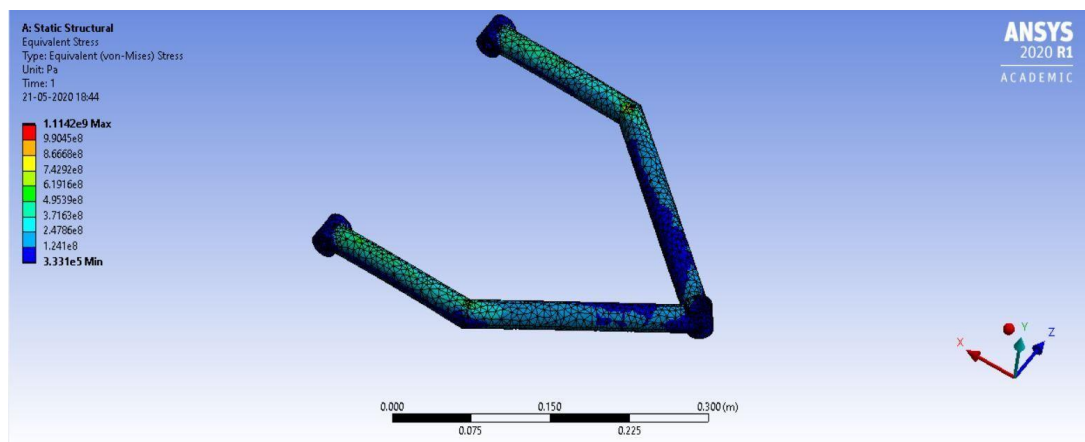


Fig. C.4. Equivalent stress when case 2 load is applied

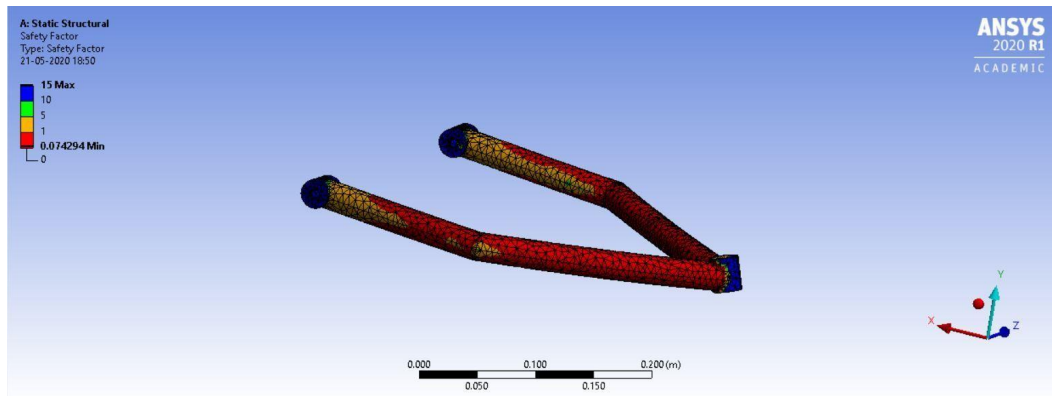


Fig. C.5. Factor of safety

Table 2. Deformation and factor of safety for different arms attained on analysis

Type	Deformation	Factor of safety
Straight A-arm	1.1 mm	2.2
Curved A-arm	2.4 mm	1.81
W shaped A-arm	5 mm	1.57

Table 3. Suspension kinematics

Positive camber	1 degree
Caster	4 degrees
Toe	3 mm
Kingpin inclination	6 degrees
Scrub radius	2 cm

3. RESULTS

By consummating analysis on the following types, we found out that Straight type A arms have less deformation rates and less stresses were developed among other types and the factor of safety is also good. According to the standards FOS of 2 and above is considered as an optimal design as it can bear twice the ultimate load which acts on it.

4. CONCLUSION

- The design provided greater suspension travel, reducing the un-sprung mass of the vehicle, maximizing the performance of the suspension system of the vehicle and better handling of vehicle while cornering.
- The suspension spring design was found to be safe and was of desired condition of ride.
- The quality of the design was assured by performing FEA which portrayed

deformation and equivalent stresses well within the desired range.

- Thus the conclusion is made that the modified suspension which is suitable to the Indian road standards by conducting various modes in analysis and it is found to be feasible.
- The above designed was manufactured and used in an All-Terrain vehicle which participated in SAE BAJA 2020 which took place in Chitkara University Punjab and did not succumb to any deformations.

By the above analysis we can say that selecting A-arm with straight profile is optimal for use.

DISCLAIMER

The products used for this research are commonly and predominantly use products in our area of research and country. There is absolutely no conflict of interest between the authors and producers of the products because we do not intend to use these products as an avenue for any litigation but for the advancement of knowledge. Also, the research was not funded by the producing company rather it was funded by personal efforts of the authors.

ACKNOWLEDGEMENTS

We would also like to thank our mentor Mr. K. Veladri (Associate Professor, Mechanical Engineering, Vasavi College of Engineering) for guiding us through this study.

COMPETING INTERESTS

Authors have declared that no competing interests exist.

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Peer-review history:
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