

# Design Modification and Analysis of Suspension Trailing Arm Using Finite Element Analysis

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**ABSTRACT:** The Suspension system is the most important system in vehicle for balancing of vehicle and driver comfort. The number of studies had done on suspension system from past few years. This study describes the analysis of cause of the failure of a suspension system trailing arm geometry installed in the rear of an All-Terrain Vehicle. Thus, the topic is focused on designing and analyzing of suspension system in vehicle for driver comfort with providing optimum performance. The trailing arm suspension system is used in auto rickshaw. It is play important role in suspension system. This trailing arm fails suddenly without any sign of failure and this case is a dangerous as well as disturbing factor for driver. Some times because of failure of trailing arm parameters like camber, wheel travel, roll center were not performing well as per desired performance. We have studies to change design of trailing arm for resolve the problem. Design of the present trailing arm has been improved and from obtained result we can say that design modified and within safety limit.

**KEYWORDS:** Trailing Arm, Modelling, Stress Analysis, Design modification.

## I. INTRODUCTION

In many countries bus, cars as well as three wheeled vehicles are a public transport vehicles, but three wheeler is common transport vehicle in India with a maximum speed of about 14 m/s[1]. This vehicle is commonly known as an Auto-rickshaw. Especially these vehicles are used in Asian countries as public transport. The total weight of the vehicle is around 1025 kg including the driver and four passengers [4]. The auto rickshaw is used in Urban as well as ruler area. It front wheel of auto rickshaw attached to linkage suspension attached to the steering column and two rear wheels attached to corresponding trailing arms that are pivoted to the frame.

The trailing arm connected between the wheel spindle/hub and chassis. Trailing arm placed longitudinally along the driving direction and mounted on suspension sub frame or body on both the sides. The trailing arm has to subjected forces in all direction because of that the possibility of bending and torsional stress increased. If only vertical and lateral forces acting on trailing arm then no camber and toe change. Trailing arm suspension is mostly used for the rear suspension of the three wheelers. It is also called as crank axle suspension system. Trailing arm is subjected to high torsional and bending stresses. It allows the wheel to move up and down only. No change will be in camber and toe due to lateral and vertical forces. But when increasing unsprung mass of trailing arm then reduce the quality of ride.

The overall study of suspension trailing arm for minimize the failure of trailing arm during operation. As these all references says that the failure occurs in rear suspension trailing arm will be find out by study the exact failure parameter of trailing arm by different test . The failure of trailing arm we can reduce by improving the failure parameter. The failure parameter is improved by changing design of suspension system arm, so it will help to you improve life of suspension trailing arm and reducing the accidents and reducing weight of three wheeler.

## II. RELATED WORK

The suspension trailing arm is big subject for researchers. Many of researchers were work on the failure occurs in trailing arm in three wheeler with different methods. The many paper published on trailing arm and many possibilities are drawn from existing data of geometry condition by literature review. Boundary condition and nonlinearly of

material properties and some other parameter are required to study in detail. The software models and it's analysis done in few researches with experimentation.

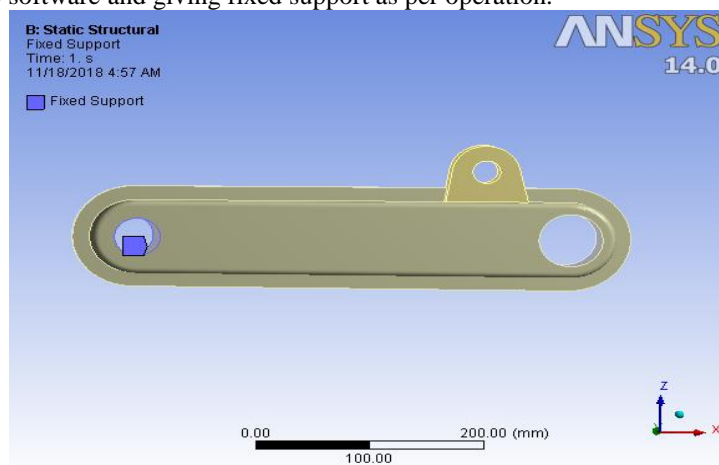
**Kiran Chaudhary et al. [2]** In this study design of the quarter car trailing arm suspension system suspension geometry is carried out in CAD software. The optimized values of spring stiffness, damping coefficient of sprung and un-sprung masses are coupled with the geometry design and is analysed in Lotus Suspension Analyser to get various values of the suspension geometry. All the obtained suspension parameters are checked by the Lotus suspension analyser for kinematic analysis. For the calculated values and it is observed that there is no change in the camber and toe angle values, and even change in wheel base is also within the desired limit.

**Akshay G Bharadwaj et al. [3]** during this study, components were fabricated according to the design and assembled. The vehicle performance was then validated by performing various tests. Measurement of the static parameters such as track width, height, length was carried out and was found to be within the limits specified by the SAE for the E-Baja vehicles. The dynamic testing of the vehicle was carried out multiple times and was found to be satisfactory in the aspects of driver safety, driver comfort, ease of steering and manoeuvrability. In conclusion, the implementation of the design in the vehicle achieved the goals of optimum performance by flexibility of camber adjustment, minimum toeing, driver comfort and maximum travel; these were achieved at a reasonable cost.

### III. MODELLING AND ANALYSIS OF SUSPENSION TRAILING ARM

The CATIA software is used for modelling of geometry. The measurements of main part of trailing arm have been made in order to create a three dimensional geometry.

Initially the 3D drawing of Trailing Arm is done by CATIA according to dimension specified. Then, the model has been imported to ANSYS software and giving fixed support as per operation.



**Fig 1: 3D Modelling of Trailing Arm**

### IV. MATERIAL

According to chemical composition analysis by OES [7], the trailing arm was manufactured using an EN 18D steel alloy, EN 18D steel alloy properties:

- Young's modulus (E)= 190Gpa
- Yield stress=565Mpa
- Ultimate tensile strength=887MPa
- Density=7.85g/cm<sup>3</sup>
- Poisson's ratio=0.29

### V. MESHING GENERATION

Meshing is important to convert from infinite element to finite element because of that we can divide all geometry in small element. There are two convergence studies, h-convergence study and p-convergence study h-Convergence study is done by increasing number of elements which can be done by making mesh size finer, and it is

important to maintain continuity in meshing and element check should be done for aspect ratio, warping angle, skew ratio and others. p- Convergence study is done by increasing number of nodes.

Meshing of a given model will be done depending on geometry of the model, it is better to have more degrees of freedom hence more number of elements so that results obtained will be closure to analytical results.

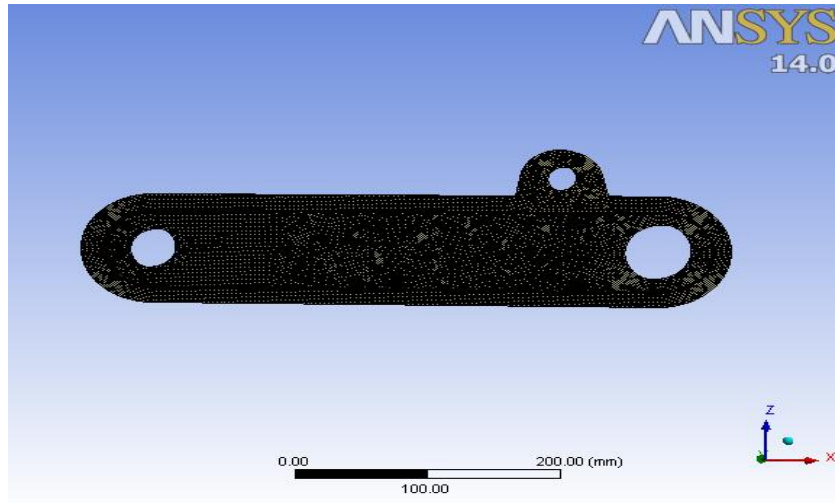


Fig 2: Meshing of Trailing Arm

#### VI. LOADING AND BOUNDRY CONDITION

The gross weight of three wheeler distributes on each wheel, gross weight and weights of persons riding the Auto rickshaw. The full static load located at the centroid of three wheeler., each tire has a share of triple of that full static load, The gross weight of rickshaw is around 1025kg, assuming the weight of passenger is 100kg of each person and Luggage weight add separately 100kg, the total static load will be 1025 kg .The assuming 425 kg kerb weight by data available so, each tire will have a share of 388kg. when the a car has an impact within the motion, control arm subjects to longitudinal load equals nearly to double of its tire share static load as well as horizontal load equals to that longitudinal one but it directs a horizontally along the length of Auto. The fatigue stress and its safety factor have been created in these two forms of numerical analysis, the load on trailing arm is fluctuating and the static load which only includes the

weights of AVT, freight and persons.

- Calculations for weight distribution on each wheel: According to OES[4]

Gross Weight = Kerb Weight + Passenger weight + Luggage weight

$$WG = 425 + 500 + 100 \text{ Kg}$$

$$WG = 1025 \text{ Kg}$$

From standard

The ratio of weight distribution

Weight acting on each wheel

$$W = (0.66 \times WG) / 2$$

$$= (0.66 \times 1025) / 2$$

$$W = 338.25 \text{ Kg}$$

So, the reaction force,

$$P = W \times g$$

$$= 338.25 \times 9.81$$

$P = 3318.23\text{N}$

$P = 3500\text{N Appx.}$

Assumed condition is of bump, in which the extremity of load is present.

The load considerations are Double Gravity ( 2g acceleration )

$P = 7000\text{ N}$  (taken for Analysis)

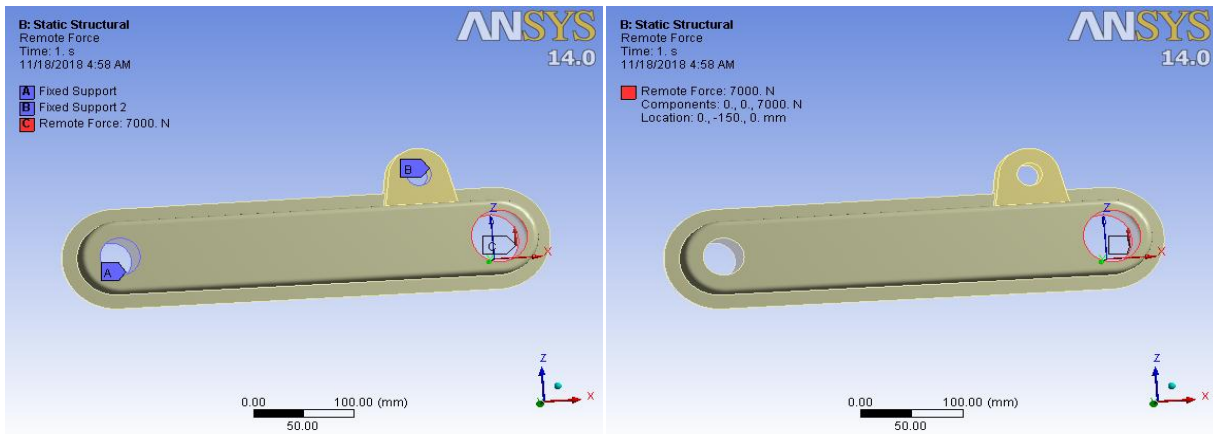


Fig 3: 3D Loading and boundary condition

### VII. RESULT FOR ORIGINAL GEOMETRY

From all above loading and boundary condition following results are obtained. According to von mises's theory, a ductile solid will yield when the distortion energy density reaches a critical value for that material.

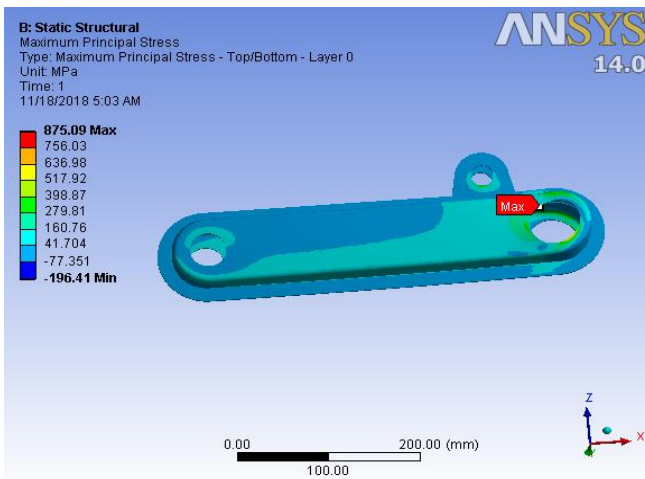


Fig 4 : Max Principal Stress

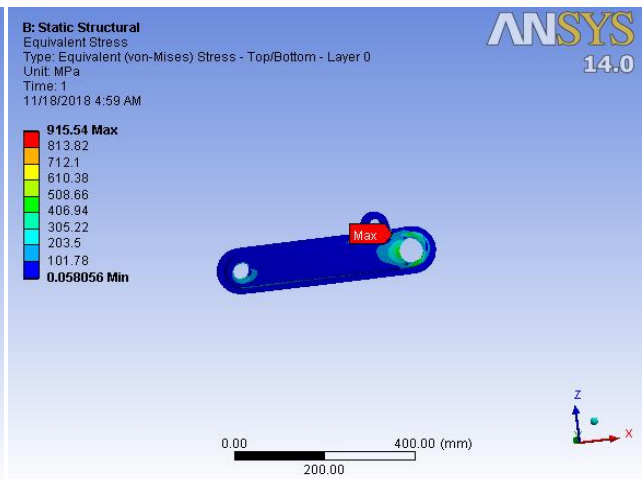
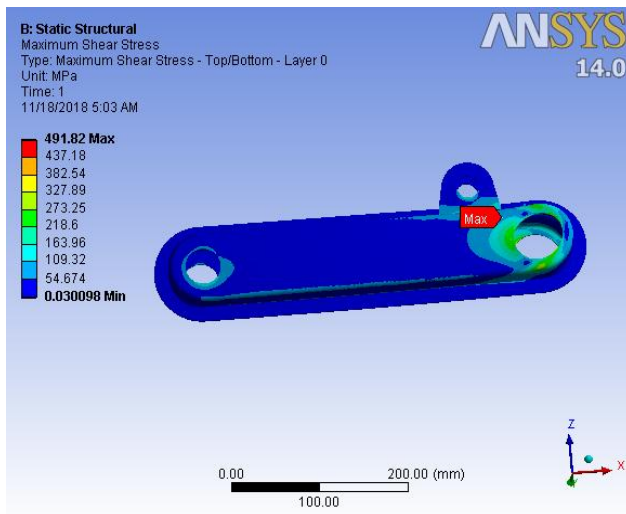
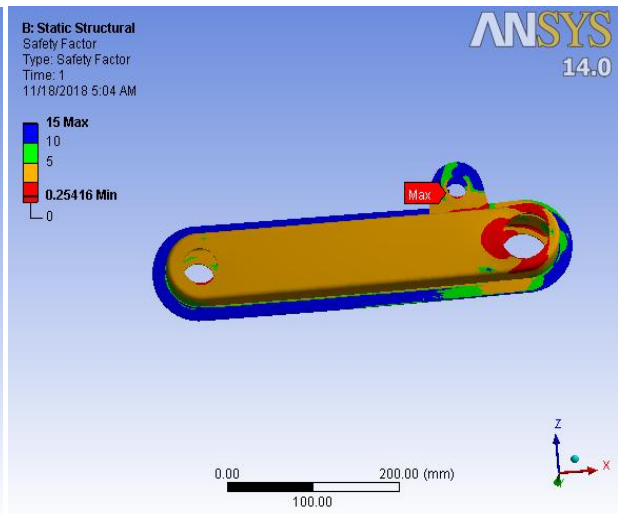


Fig 5: Equivalent Von-Mises stress



**Fig 6: Maximum Shear Stress**

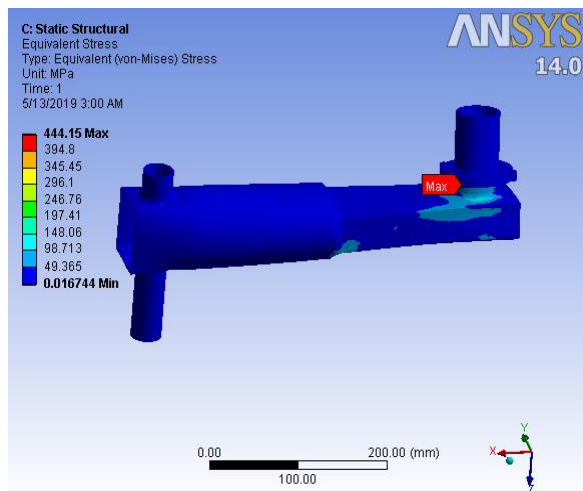


**Fig 7: Factor of Safety**

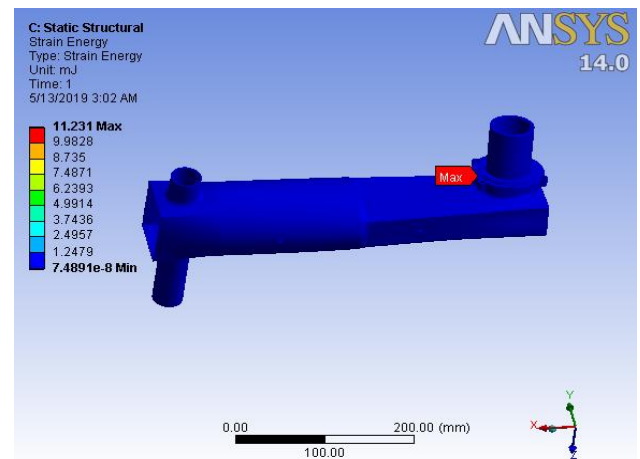
We can conclude that, equivalent von-mises stress goes beyond ultimate tensile strength of the material and hence there is crack initiation in the trailing arm at the specified loading conditions and that crack leads towards the fatigue failure of the trailing arm as the trailing arm is under cyclic loading as per the description given in the problem definition. After knowing the result, it is need to change design of trailing arm at failure section. Change the design to get desired result.

**VIII. RESULT FOR MODIFIED TRAILING ARM**

The design had been changed from previous and for same loading and boundary condition for trailing arm as below:



**Fig 8: Equivalent Von-Mises stress**



**Fig 9: Strain Energy**

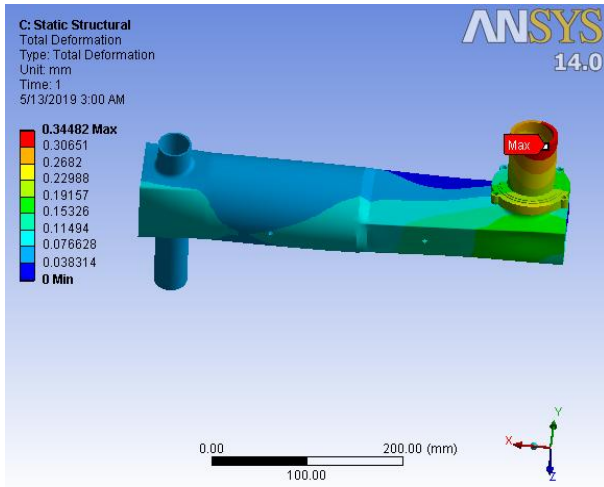


Fig 10: Total Deformation

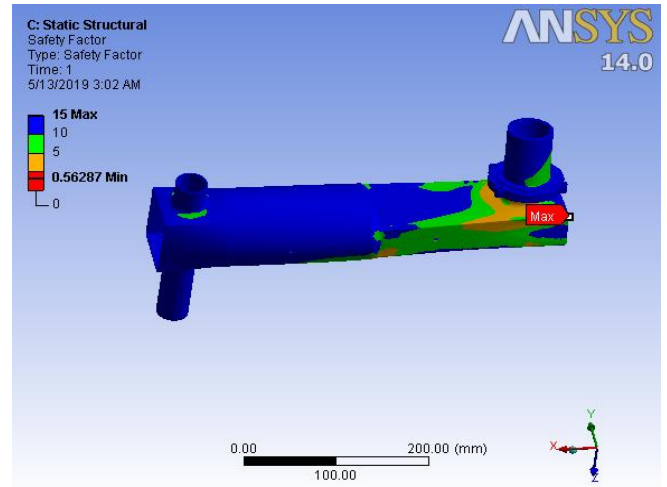


Fig 11: Safety Factor

The design of trailing arm and required design modification are done and results of stress fatigue analysis of modified trailing arm show that stress value has been brought to the safety limit.

IX. COMPARISON OF RESULTS

Sr. No.	Components	BEFORE MODIFICATION	AFTER MODIFICATION
1	Total Deformation	0.8	4.9
2	Equivalent Stress Plot	915.54	444.15
3	Major Principal Stress Plot	875.09	470.92
4	Equivalent Elastic Strain Plot	0.0047	0.0029
5	Max Shear Stress	491.82	79.67
6	Factor Of Safety	0.25	0.56

Table No.1 Stress analysis result

X. CONCLUSION

Modify the design of trailing arm to make safe reliable and durable. We worked on the stress raiser in the previous design of trailing arm and required design modification is done. We got some results which defines the stress distribution, deformation and safety factor. We compare all results before and after modification.

Design of the present trailing arm has been improved and from obtained results we can say that the design of trailing arm is optimized and within safety limit, the component is now have considerably good life.



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