

# Static and Fatigue Failure Analysis of Trailing Arm Bracket of Rear Suspension System

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**ABSTRACT:** Trailing arms for the vehicle suspension systems, such as trailer suspension system, are well-known to those skilled in the art. Different designs, which maintain various advantages over one another are customary in the industry. Suspension system comprises of trailing arm having a leading portion which is coupled to the vehicle chassis and rear end is connected perpendicular to the axle of the system. A bracket is welded at mid-point of the trailing arm where shocks are placed over it. The main function of the trailing arm is to allow the axle to move up and down using the coil spring and limit the movement, when driving on the uneven roads. Because of this the bracket area is highly affected by torsional loading resulting in break torque, hence there will be chances of failure of bracket. In this paper trailing arm is designed by utilizing modelling software Solid works, meshing was carried by using ANSA V15.2 and analysis was carried out in the ANSYS APDL.

**KEYWORDS:** Trailing arm, static analysis, fatigue analysis.

## I. INTRODUCTION

A vehicle system is considered as a system of complex structure made with numerous mechanical components. That are subjected to complex cyclic loads as a result of ordinary use. Dynamic suspension system pulled in countless researchers in recent couple of years. Suspension is a term given to the system of linkages, shock absorbers springs that connects the vehicle to its wheel and allows the relative motion between them. Generally the main advantage of suspension system is passenger comfort and vehicle control that is by keeping the driver and passenger comfortable by well confined from road, noise vibration, which caused due to uneven road, bumps or potholes.

In this paper, study is carried out trailing arm bracket of a rear suspension system to know the failure modes as well as life cycle. The model is created in solid works, meshing is done by using ANSA and analysis is carried out in the ANSYS.

A trailing arm is a sub part of suspension system, there are different types of suspension systems are used for both front and rear suspension system. In automobiles trailing arm suspension system is independent suspension design, uses a long hollow tubular solid to locate the wheel. Foremost end is connected to the chassis while rear end is connected perpendicularly to the axle of the suspension system. The coil spring and shock absorbers mounted on the trailing arm to manipulate the vertical motion.

Trailing arm / lower control arm works with the upper arm to keep the wheel alignment in all positions. The wheel is connected to the spindle, the spindle is connected to the ball joints of both upper and lower control arms in suspension system. While working condition this enables the up/down swivel motion of the vehicle. The suspension arm stores the impact loads caused due to the uneven roads and provides better vehicle ride.

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## II. TYPES OF SUSPENSION SYSTEM

Suspension system is classified into two main types

- a) Dependent type: -In dependent type suspension system movement of wheel on one side of the vehicle is subjected to the movement of its accomplice on the other side. These rarely used in modern passenger car.Example: - Solid axle Suspension system.
- b) Independent suspension: -in this kind of suspension motion of wheel pair is independent. For the purpose that whenever there will be any disturbance at wheel not transmit to the other wheel of the system. Now a day's most of the modern vehicles uses independent suspension system for both rear and front suspension system. One of the main advantage of this suspension system is better ride and better handling they requires less space, easiest steer-ability and low cost. Examples of independent suspension system as follows  
Examples: - Leaf Spring Suspension, McPherson Suspension. Wishbone suspension, Trailing arm suspension

## III. LOAD CALCULATIONS

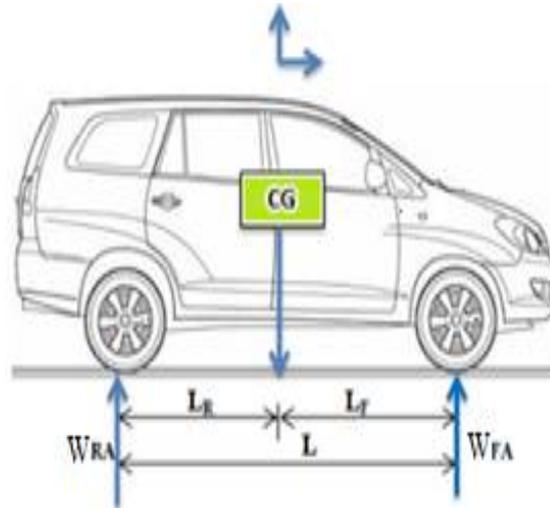


Figure 1: Typical vehicle line diagram.

Consider above vehicle for load calculation we use Duygu Guler method. Numerical data with parameter is given below.

L = Wheel base length = 2705 mm.

Distance from front axle to CG, LF =0.55 x L = 1487.75mm

Distance from rear axle to CG, LR =0.45\*L = 1217.25mm.

Total weight of the car is 1995 Kg

$$W = mg \quad \dots \dots \dots (1)$$

The load acting on front and rear wheels can be calculated by using following equilibrium equations.

$$W_{FA} = W \frac{L_R}{L} \quad \dots \dots \dots (2)$$

$$W_{RA} = W \frac{L_F}{L} \quad \dots \dots \dots (3)$$

The static load on one wheel calculated as follows.

➤ Load acting on one front wheel is

$$W_{FAW} = \frac{W_{FA}}{2} \quad \dots \dots \dots (4)$$

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Load acting on one rear wheel

$$W_{RAW} = \frac{W_{RA}}{2} \dots \dots \dots (5)$$

By using above formulae we can calculate force acting on the trailing arm bracket.

Total weight of the vehicle from equation (1).

$$W = 1995 \times 9.81 = 19570.95 \text{ N}$$

The load acting on the front wheel can be calculated by using equation number (2)

$$W_{FA} = 19570.95 \times \frac{1217.25}{2705} = 8806.9275 \text{ N}$$

Similarly, Load acting on rear wheel is calculated by using equation number (3)

$$W_{RA} = 19570.95 \times \frac{1487.75}{2705} = 10764.0225 \text{ N}$$

Load acting on one wheel of front and rear wheel can be calculated by using equation numbers (4) and (5).

$$W_{FAW} = \frac{8806.9275}{2} = 4403.46375 \text{ N}$$

$$W_{RAW} = \frac{10764.0225}{2} = 5382.01125 \text{ N}$$

Reaction on each wheel of front side = 4403.46375 N

Reaction on each wheel of rear side = 5382.011 N

Our area of interest is on rear suspension which is almost equal to 5400N. This load is used for static analysis of rear suspension trailing arm for 1G load condition.

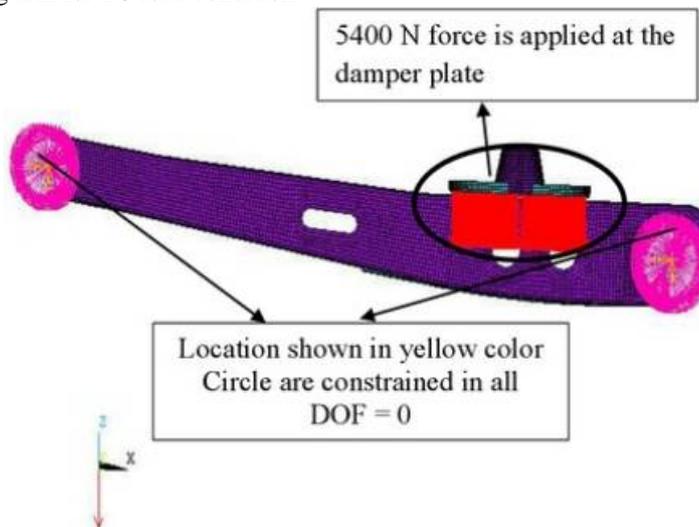


Figure 2: Physical boundary condition.

## IV. ANALYSIS OF EXISTING TRAILING ARM WITH BRACKET

It is very important to analyse the present design of the trailing arm in order to assure the strength features in the freshly designed bracket. Static analysis was carried out for the current arm. The material of the existing bracket is steel AISI1018 with Young's modulus of 210GPa and Poisson's ratio of 0.3. The trailing arm was meshed using ANSA.

In this structural analysis is carried out steady state or static condition and solution is independent of time. Assumptions are made as follows,

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- First FE model static structural analysis is carried out for calculated load that is 1G load.
- Than further calculations are made for 2G and 3G bump loads.
- In this result is compared with respect to yield strength of the material.
- Deflection observed is neglected.

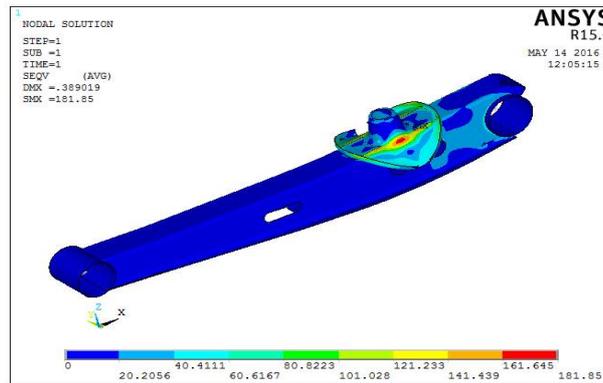


Figure 3: Von Mises stress for 1G bump loads

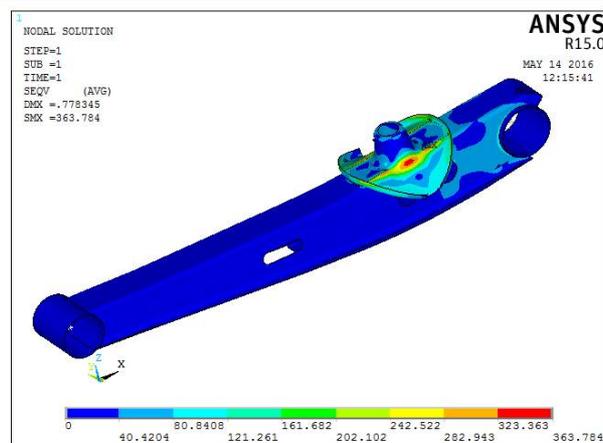


Figure 4: Von Mises stress for 2G bump loads.

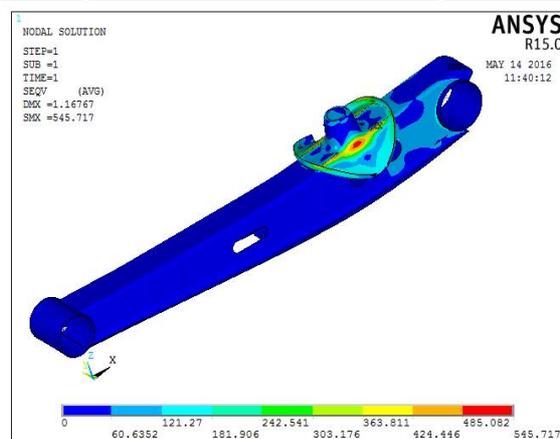


Figure 5: Von mises stress for 3G bump loads

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Boundary conditions were applied and solved for stresses for a load of 5400 N and it was observed that the maximum stress produced was 181.7 Mpa for 1G Bump load, 363.7 MPa for 2G bump load and 545.717 MPa for 3G bump load where the failure of bracket occurs. Hence design modification is required.

### V. DESIGN MODIFICATION AND ANALYSIS

In static analysis for 3G load case more stress observed at the shock mounting bracket the region is shown below figure. In the modified design model, bracket design changes has been mad, the thickness of the plate is increased from 2.5 mm to 4 mm to avoid the failure for 3G bump load condition Therefore the modified new design model of shock mounting bracket has been created using CAD Tool solid works.

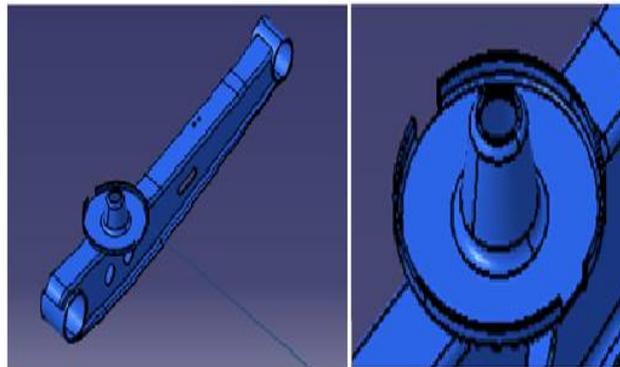


Figure 6: Modified design.

Static structural analysis is carried out for the modified design for 3G bump load condition. Fatigue analysis is also carried out for the 3G bump load of modified design for finding the fatigue life of the component. The maximum stress observed for 3G load condition for trailing arm of shock mounting bracket is 205.7411 MPa, which lies within the yield limit of the material. Hence the design is safe. Further fatigue analysis is also carried out for the new modified design.

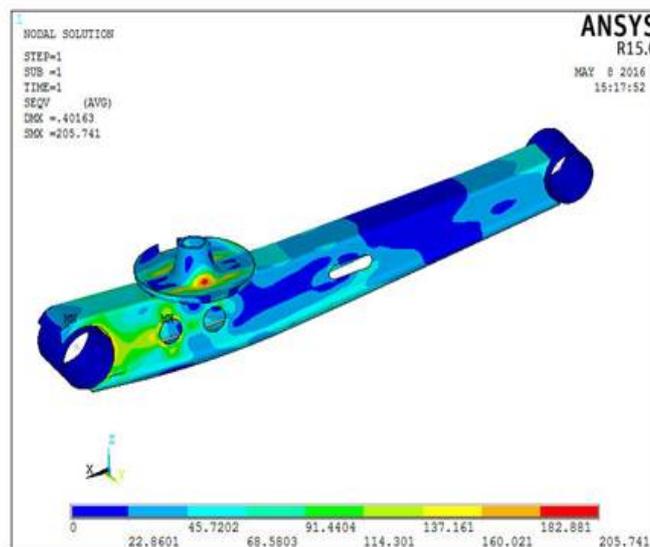


Figure 7: Von mises stress for modified design.

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## VI. FATIGUE ANALYSIS

The Fatigue life analysis of the trailing arm is predicted using stress life approach. In this analysis is concentrated on critical location nodes 29854 and 11343. It is observed that shock mounting place of the bracket is the critical location of the trailing arm bracket.

Fatigue life requirement is  $1 \times 10^6$  cycles, fatigue analysis is carried out only on 3G bump loads. Only damaging signals resulting alternating stresses have taken for cumulative fatigue damage.

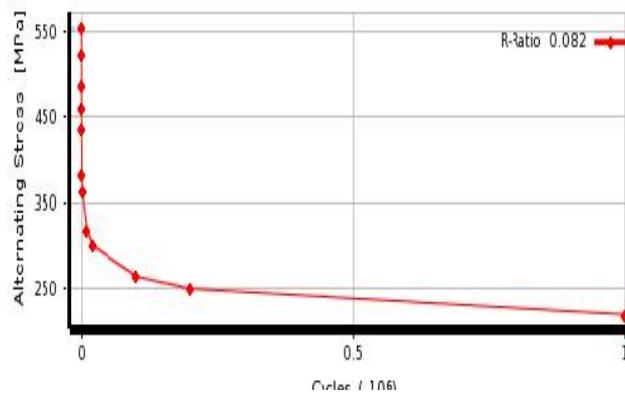


Figure 8: Graph for alternating cycles vs. Number of cycles

The Wohler S-N diagram is plotted as shown in graph 8. The S-N diagram represents nominal stress amplitude on Y-axis and number of cycles on X-axis.

Table 1 Different node number location and stresses.

Location	Node number	Stress observed (MPa)
1	29854	212.97
1	11343	213.39

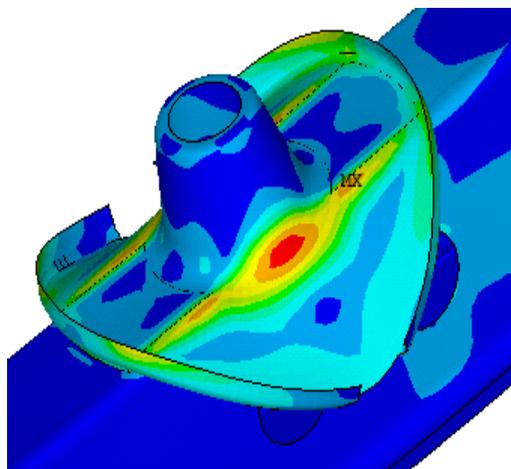


Figure 9: Maximum stressed area shown in red colour

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In the above figure stress distribution in ANSYS we found the red colour which is high stressed area in the bracket. The possibility of failure of bracket. For node locations 1 at 29854 and node location 2 at 11343 are critical node locations, where fatigue failure occurs and we compute the life at these node locations.

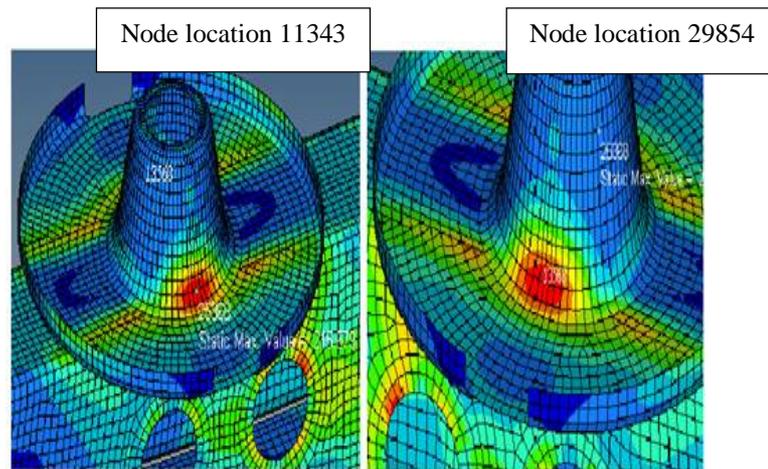


Figure 10: Maximum stress at critical locations

The combination of event 1, load 1 and event 2, load 2 yields an alternating stress intensity of 212.97 N/mm<sup>2</sup> and 213.39 N/mm<sup>2</sup>. Subjected to 0.5e6 cycles while from the S-N Table, the maximum number of cycles permitted at that stress intensity is 0.1e7. The partial usage value, 0.5, is the ratio of cycles used/cycles allowed. The Cumulative Fatigue Usage value is sum of the partial usage factors (Miner's rule).

## VII. RESULTS AND DISCUSSIONS

Comparison of stresses produced in static analysis for the same applied load of 5400 N, the comparison of stresses produced in the brackets is shown below.

Table 1: Stresses for static analysis for 3G bump loads

Bracket	Type of load	Observed vvon Mises stresses
Current bracket	3G	545.717 Mpa
Modified Bracket	3G	205.741 MPa

As we can see clearly from the above comparison that the stresses produced in the modified bracket is very less compared to current. Fatigue life observed for modified design is  $1 \times 10^6$  in analysis. Hence the design is safe.

## VIII. CONCLUSION

From analysis we conclude that,

- Prediction of fatigue life is mainly concentrated over the critical locations.
- For modified design the Von-Mises stress observed well below the yield strength of the material and fatigue life observed is  $1 \times 10^6$  cycles under 3G bump load condition.
- The stress and deflection observed is well below the permissible limit of the material.
- The Von-Mises stress obtained is 205.741 N which is less than the yield limit for 3G bump load, hence the design is safe.

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