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Research Article

Static and Fatigue Failure Analysis of Rear Trailing Arm

(Case study at MIE)

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Abstract

This paper presents failure analysis and modeling of the rear trailing arm case study at Mesfin Industrial Engineering, Ethiopia. Considering loading type as 1G, 2G and 3G bump load condition, the static and fatigue analysis were carried out using SOLIDWORKS simulation finite element analysis. The maximum von Mises stress and static factor of safety for existing rear trailing arm in 3G bump load condition were 537.5MPa and 0.44 respectively. The endurance limit of the component 180MPa is lower than the alternative stress of 268.67MPa. Due to less factor of safety, the existing rear trailing arm failed statically and dynamically. From the analysis of the result, St-52 was chosen over St-37. The von Mises stress and factor of safety under 3G bump load condition for St-52 were 128.1MPa and 2.77MPa respectively. Hence, the design is statically safe. Fatigue analysis was also carried out for the 3G bump load of modified design of Rear Trailing Arm made of St-37, St-44 and St- 52 material for thickness increased 2mm to 5 mm materials. The endurance limit of the component for St-52 is 315MPa which is greater than the alternative stress of 64MPa. The modified rear trailing arm is safe statically and dynamically both.

Keywords: *Failure analysis, rear trailing arm, static analysis, fatigue analysis, von Mises stress*

1. Introduction

Suspension system is the most useful component of a vehicle in which its main function is to prevent the road shocks from being transmitted to the vehicle frame, to preserve the stability of the vehicle in pitching or rolling, to safe guard the occupants from road shocks, to provide steering stability with good handling and to provide good road holding while driving, cornering and braking. Elements of vehicle suspension are springs, dampers and connectors trailing arms of body with wheels. Springs are flexible members which act as the reservoir of energy during motion. When vehicles move on a bumpy road, energy is released with the action of damper and energy converted into heat energy consequently the vibration or bouncing of the body is avoided.

A trailing arm design can be used in an independent suspension arrangement. Each wheel hub is located only by a large, roughly U-shaped arm that pivots at one point, ahead of the wheel. Seen from the side, this arm is roughly parallel to the ground, with the angle changing based on road irregularities. A twist-beam rear suspension is very similar except that the arms are connected by a beam, used to locate the wheels and which twists and has an anti-roll effect. Trailing-arm and multilink suspension designs are much more commonly used for the rear wheels of a vehicle where they can allow for a flatter floor and more cargo room. The trailing arm of Geely Addis assembled by Mesfin Industrial Engineering PLC has been faced with repeatedly failure.

1.1 Background of Rear Trailing Arm

Rear Trailing Arm is one of the suspension system components that connect the dead axle and wheel. A trailing-arm suspension, sometimes referred as trailing-link is a vehicle suspension design in which one or more arms (or "links") are connected between (and perpendicular to and forward of) the axle and a pivot point (located on the chassis of a motor vehicle). It is typically used on the rear axle of a motor vehicle.

1.1.1 Working principle of Rear Trailing Arm

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

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the vehicle. The suspension arm stores the impact loads caused due to the uneven roads and provides better vehicle ride [A. Ossa *et al*, 2011; Vijaykumar *et al*, 2014; Abdelhamid Saoudi *et al*, 2011; Jitendram Shinde *et al*, 2016]. Trailing arm is the most visually striking component of a linked rear suspension. In addition to connecting the chassis, rear axle, and shocks, they provide fabricators with an opportunity for artistic expression. There are some intricate designs, along with some simpler stuff, and have been positively impressed with it all. A trailing arm which has a big job to do, the shocks are mounted to trailing arm so as to support the weight of the vehicle as well as the tension and compression forces generated between the rear wheels and the chassis. Finally, trailing arms need to be strong enough to withstand random hits from rocks, roots, and plain old dirt. Moreover; the trailing arm is used to keep the rear axle from moving left and right during turning. When the body of the automobile is going one way the axle wants to go the opposite side. During this time the trailing arm prevents this destructive action before catastrophic accident has happened. The existing damaged rear trailing arm picture is shown in Fig.1 given below.

Fig.1. The existing damaged rear trailing arm

1.2 Problem statement

Mesfin Industrial Engineering PLC has been set to get on assembling sedan automobiles the last seven years with the launch of its assembly car, called “Geely Addis,” in collaboration with Geely International Co, a China based manufacturer of Geely cars. Geely International, which was founded in Shanghai in July 2002, recently acquired Volvo, a Swedish brand known for the quality and endurance of its vehicles.

The cars are being initially assembled in semi knocked down (SKD) form, in which parts of the cars are wholly imported from China and assembled at the plant. The company has a long plan to eventually assemble the cars in completely knocked down (CKD) form, in which most parts are produced locally, with the exception of parts like the engines that will be imported.

However, its customers are not fully happy due to the failure of some parts. Among the parts of suspension system which fails repeatedly is the rear trailing arm. It fails due to excessive deformation that tends to be straightened. As a result, the tyre wears quickly due to camber effect. The remedial action which the maintenance personnel always take is only replacing the part by another new one.

The company has a long term plan to bring the full technology of Geely Addis. But it is strictly recommended that it has to study the potential failure causes and their consequences and to solve problem. Rear trailing arm with damaged part under maintenance is shown in Fig. 2 given below. 3D Modeling and 2D diagram of Trailing Arm is shown in Fig. 3.

Fig. 2. Damaged trailing arm under maintenance

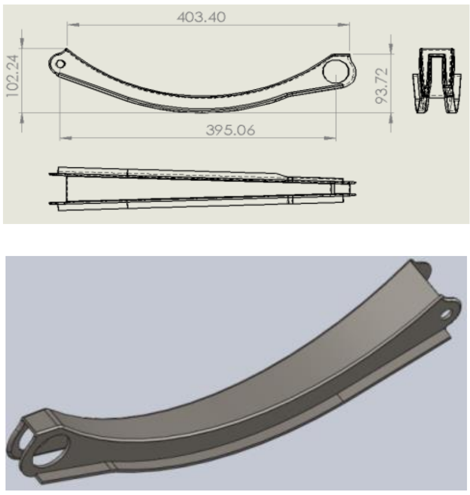


Fig. 3. 3D Modeling and 2D drawing of the original trailing arm

1.3. Objective

1.3.1 General objective

The main objective of this research work was to analyze the existing and modified damaged rear trailing arm analytically and propose alternative solution.

1.3.2 Specific objective

i. Mechanical testing of rear trailing arm and reporting the results
ii. Analysis of rear trailing arm using Finite Element Analysis (FEA)

2. Literature Review

Different literature reviews which are familiar to the design, modeling and failure analysis of suspension system were inducted for project.

2.1 Maintenance and operational manual

Geely Addis operational and maintenance manual from Mesfin Industrial Engineering PLC were taken as a reference for this paper. From the maintenance manual, the Rear and Front Trailing Arms are inspected after every 2 years continues service.

Maintenance documents like job orders and inspection papers are the key references for determining the failure type, occurrence and effects of these failures on the other automobile parts [Geely Addis Manual *et al*]. Not only using these documents as an input for this project work but also gathering data from the technicians and maintenance personnel were helpful which can play a crucial role. The technicians conclude that every rear trailing arm they replace become out of use due to excessive deformation.

Therefore, in this paper work, it is proposed to carry out the structural analysis of trailing arm of Geely Addis sedan automobile. Diagram of Rear Trailing Arm from operational manual of Geely Addis is shown in Fig. 4 given below.

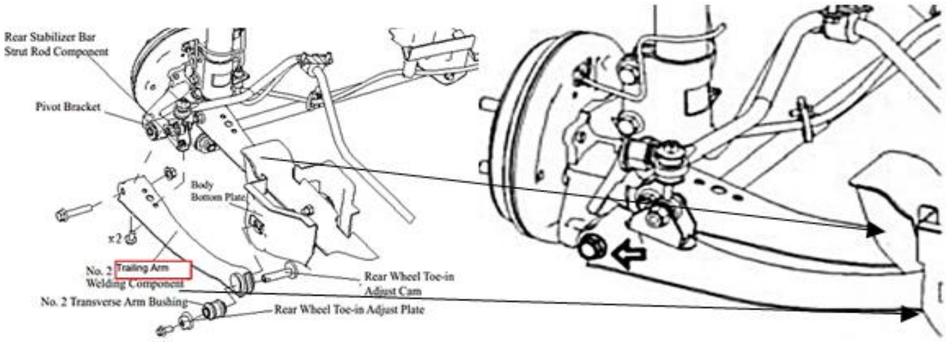


Fig. 4. Diagram of trailing arm taken from operational manual of Geely Addis

2.2 Suspension Lower Arm

The vehicle suspension system is responsible for driving comfort and safety as the suspension carries the vehicle-body and transmits all forces between body and road. Positively, in order influence these properties, semi -active or active components were introduced, which enable the suspension system to adapt to various driving conditions. From a design point of view, there are two main categories of disturbances on a vehicle namely the road and load disturbances. Road disturbances have the characteristics of large magnitude in low frequency (such as hills) and small magnitude in high frequency (such as road roughness). Load disturbances include the variation of loads induced by accelerating, braking and cornering. Therefore, a good suspension design is concerned with disturbance rejection from these disturbances to the outputs. A conventional suspension needs to be "soft" to insulate against road disturbances and "hard" to insulate against load disturbances. Consequently, the suspension design is an art of compromise between these two goals. The general function of control arms is to keep the wheels of a motor vehicle from uncontrollably diverging when the road conditions are not smooth.

In the automotive industry, the riding comfort and handling qualities of an automobile are greatly affected by the suspension system, in which the suspended portion of the vehicle is attached to the wheels by elastic members in order to cushion the impact of road irregularities. The specific nature of attaching linkages and spring elements varies widely among automobile models. The best rides are made possibly by independent suspension systems, which permit the wheels to move independently of each other. In these systems the unsprang weight of the vehicle is decreased, softer springs are permissible and front – wheel vibration problems are minimized. Spring elements are used for automobile suspension, increasing order of their ability to store elastic energy per unit of weight [A.V. Vanalkar *et al,* 2015; Aniket *et al,* 2014].

Load limits for parts made of materials that strain harden significantly when stressed in the plastic region can be estimated by limit analysis, as can those for parts made of other materials whose stress strain behavior differs from that of the idealized material. In these situations, the designer bases his design calculations on an assumed strength that may actually lie well within the plastic region for the material [D. Goldner*et al*, 1973].

The FEM approach is used for analysis of a suspension link for Static stress and Von-misses stress analysis of lower arm for deformation and stresses. Stress analysis of the lower wishbone arm is to be done considering Gross Vehicle Weight, Front axle Weight, Rear Axle Weight. Von-misses stress analysis is to be done after application of forces at lower control arm, bump stop, Spring, at wheel centre. And as stress is higher than safe limit new geometric change adopted in design to make it safer [Vijaykumar *et al*, 2014].

2.3 Static and Fatigue Failure Analysis of Trailing Arm

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. Prediction of fatigue life on lower suspension arm subjected to variable amplitude loading was studied using finite element analysis earlier [Z. Husin*et al*, 2010; SK As*et al*, 2005]. A bracket is welded at mid-point of the trailing arm where shocks were placed over it [M. E. M. EL-SAYED et *al*, 1996]. 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 research, Comparison of stresses produced in static analysis and fatigue analysis for the same applied load were carried out [Avinash, C. Suresh *et al*, 2016].Finally, this research has concluded solving the problem by improving the material property.

2.4 Role of Suspension Lower Arm

Most of the times it is called as a -type control arm. It joins the wheel hub to the vehicle frame allowing for a full range of motion while maintaining proper suspension alignment. Uneven tyre wear, suspension noise or misalignment, steering wheel shimmy or vibrations are the main causes of the failure of the lower suspension arm. Most of the cases the failures are catastrophic in nature. So the structural integrity of the suspension arm is crucial from design point of view both in static and dynamic conditions. As the Finite Element Method (FEM) gives better visualization of this kind of the failures so FEM analysis of the stress distributions around typical failure initiations sites is essential [A.V. Vanalkar*et al,* 2015; Aniket*et al, 2014*].

2.5 Working conditions

From the maintenance manual of the automobile, the vehicle is driven under one or more of the following circumstances,

A. Road Condition

i. Drive on rough, muddy or slippery roads [Geely Addis Automobile Manual *et al*].
ii. Drive on dusty roads [Geely Addis Automobile Manual *et al*].

B. Driving Condition

i. Repeat driving within 8km a few times, or when the outdoor temperature is below 0 degree centigrade [Geely Addis Automobile Manual *et al*].
ii. Idle the vehicle drive at a low speed for a long time, such as police vehicle, taxi, or door to door delivery vehicles [Geely Addis Automobile Manual *et al*].
iii. Continuously drive the vehicle at high speed for more than 2 hours (80% of the max speed) [Geely Addis Automobile Manual *et al*].

2.6 Failure due to overload

Every structure has a load limit beyond which it is considered unsafe or unreliable. Applied loads that exceed this limit are known as overloads and sometimes result (depending on the factor of safety used in design) in distortion or fracture of one or more structural members. Estimation of load limits is one of the most important aspects of design and is commonly computed by one of two methods-classical design or limit analysis.

2.7 Classical design

This classical approach inherently assumes that the stress to cause fracture is greater than the stress to cause yield. As fracture mechanics analysis clearly shows, this may not be the case. Fracture may occur at loads less than that required to cause yield if a sufficiently large imperfection is present in the material. Classical design keeps allowable stresses entirely within the elastic region and is used routinely in the design of parts [Shipley *et al*, 1973]. Allowable stresses for static service are generally set at one-half the yield strength for ductile materials and one-sixth for brittle materials, although other fractions may be more suitable for specific applications. For very brittle materials, there may be little difference between the "yield" and ultimate strength, and the latter is used in design computations.

3. Material and Research Methodology

3.1 Material

Materials, the existing rear trailing arm, were procured for conducting tests which was received from Mesfin Industrial Engineering PLC. The samples were prepared from this material to conduct various tests and analysis required in this project work.

3.2 Methodology

In this section, various methods were chosen in order to meet the specific objective and answers to the main problem of the project. This research work has been begun by collecting data and necessary information, which demonstrates comprehensively, and in detail. Definitions and theories from related literatures written by different authors based on this issue were provided.

3.2.1 Chemical composition testing

Five samples of existing rear trailing arm from Mesfin Industrial Engineering store were polished by polishing disc to remove the coating and debris. The material composition and grade of the samples were measured by using 6GSO-System Belec compact port spectrometry with high accuracies (C< 0.01percentage) the composition of the material was carried out for identifying the existing elements present in the material using argon gas flushing.

The following steps were used for determination of the composition of the plate using spectrometer according to ASTM E 350 – 95 (Reapproved 2000)" Standard test methods for chemical analysis of carbon steel". Chemical composition analysis **test** instrument is depicted in Fig. 5 given below.

3.2.2 Mechanical testing

To investigate the root cause of rear trailing arm failure, different types of mechanical testing had been conducted at Mekelle University and Mesfin Industrial Engineering PLC. The specimen preparation to conduct mechanical testing procedure is shown below.

3.2.2.1 Tensile testing

The standard dimension of specimens subjected for tensile strength testing was prepared according to ASTM E8/E8M – 09.The standard specimen size was marked and sketched on the flat stock plate using color marker. Tensile testing specimen drawing is shown in Fig. 6 given below.

Fig. 5. Belec compact port spectrometry machine

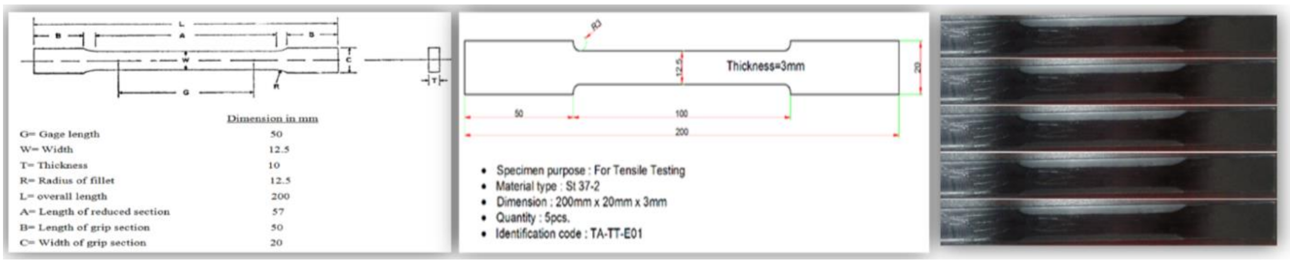


Fig. 6. Tensile testing specimen

3.3 Modeling and analysis

In this methodology, it was used the SOLIDWORKS Simulation finite element analysis (FEA) software to analyze the response of a component to an applied load. Finite element analysis is a powerful tool that allows engineers to quickly analyze and refine a design. It can be applied to problems involving vibrations, heat transfer, fluid flow, and many other areas. The existing trailing arm and the new proposed material were modeled using solid work. The modeling was assigned the proper material type from the library of the software. After giving the constraint (boundary conditions), it was meshed into finite elements. Loading applied and the stress strain analysis were carried and the final outputs generated data were stored for further analysis. Finally, the result was interpreted and the best solution for the failure was proposed as a solution for this problem.

4. Results and Discussion

4.1 Results

4.1.1 Chemical composition analysis

The composition analysis of the Rear Trailing Arm samples was carried out using Belec compact port spectrometric machine in MIE. The average composition weight percent results obtained from five specimens of five trials per each specimen and their detail trial results were tabulated. After testing, the obtained composition results for the materials are given in the Table 1 below.

Table 1: Summarized value of the material compositions for Rear Trailing Arm

No.	C	Si	Mn	Cu	Al	Cr	Mo	Ni	V	Ti	Nb	Co	W
Average	0.170	0.105	0.250	0.037	0.096	0.021	0.009	0.005	0.028	0.010	0.080	0.036	0.106
Grade 1: 1.0037, St37-2													

Table 2: Tensile testing results of existing rear trailing arm sample

Tensile Strength(in MPa)	Yield Strength(in MPa)	Percentage of elongation	Percentage of reduction of area	Modulus of elasticity, GPa
436.8	235.22	24.88	29.50	175

4.1.2 Mechanical properties analysis

Five specimens from different five existing Rear Trailing Arm were prepared. The tensile test specimens were subjected to uniaxial load until these were tested by using the Zhejiang SI-1000KN microcomputer controlled electrohydraulic servo universal testing machine. The test was performed at standard room temperature and humidity in Mekelle university main-campus civil (structural) engineering material testing laboratory. The data generated during tensile testing at room temperature is used to determine a mechanical property of trailing arm is in Table 2 given above.

4.1.3 Force Analysis of the Rear Trailing Arm

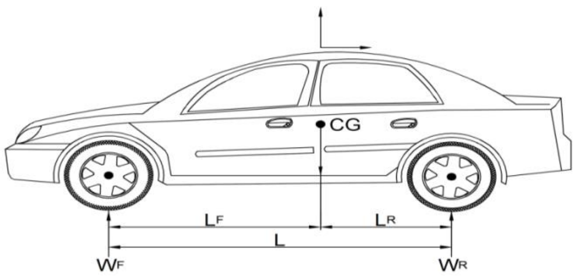


Fig.7. Free body diagram of Geely Addis Automobile

Consider the above vehicle for load calculation we use DuyguGuler method [Avinash C. Suresh *et al*, 2016; Geely Addis Automobile Manual]. Free body diagram of Geely Addis Automobile used for force analysis of the rear trailing arm is shown in Fig. 7 above.

Numerical data with parameter is given below.

Distance between the front and rear wheel /axle (Wheel base) is **L**
Distance from front axle to CG, **LF** =0.55* L
Distance from rear axle to CG, **LR** =0.45*L
Total weight of the car (Gross vehicle weight) is 7383.65 Kg
The reaction force created at the front wheel is $\mathbf{W_F}$
The reaction force created at the rear wheel is $\mathbf{W_R}$
The load the vehicle can be calculated as
W=mg; W=7383.65kg*9.85m/s^2=72728.95N
The distance from center of gravity to both front and rear axle can be calculated as

$\mathbf{L_F}$ =0.55* L =1347.5mm and $\mathbf{L_R}$=0.45*L=1102.5mm
The load acting on front and rear wheels can be calculated by using the following equilibrium equations.

$$W_F = \frac{W*L_R}{L};\ W_F = \frac{72728.95N*1102.5}{2450} = 32728.03N$$

$$W_R = \frac{W*L_F}{L};\ W_R = \frac{72728.95*1347.5}{2450} = 40{,}000.92N$$

The reactions or Loads acting on each wheel of front and rear wheel can be calculated by using equation the following equations

Load acting on each front wheel

$$W_{FW} = \frac{W_F}{2} = \frac{32728.03}{2} = 16364.02$$

Load acting on each rear wheel

$$W_{RW} = \frac{W_R}{2} = \frac{40{,}000.92}{2} = 20{,}000.46N$$

At the rear axle there are four trailing arms (two at the left rear and two at the right rear wheels) and the weight should be shared in to these trailing arms. Hence;

$$W_{R,singleTA} = \frac{W_{RW}}{2} = \frac{20{,}000.46}{2} = 10{,}000.23N = 10.00023kN$$

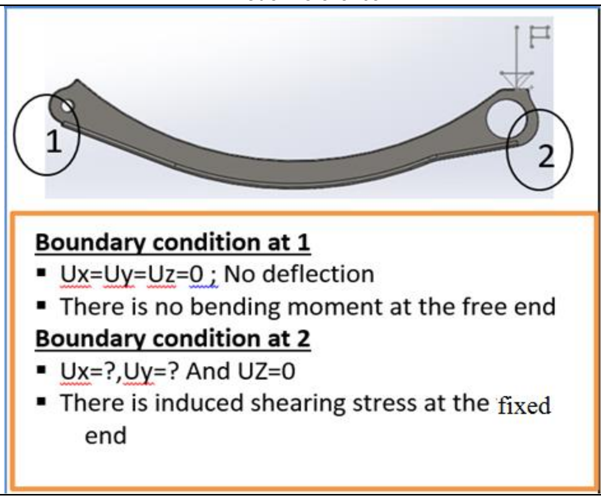
Our area of interest is on rear suspension which is almost equal to 10.00023kN. This load is used for static analysis of rear suspension trailing arm.

4.1.4 Finite Element Analysis of existing Rear Trailing Arm

FE model static structural analysis is carried out for 1G, 2G and 3 G loading condition. Analysis of the existing design of the Rear Trailing Arm is very important to design and assure the strength in the newly designed product. Static analysis was carried out for the current rear trailing arm using FE model. The mechanical properties of the existing trailing arm St 37-2 considered during the solid model preparation with Young's modulus of 175GPa,Poisson's ratio of 0.28, Yield strength of 235MPa and Tensile strength 360MPa.The Rear Trailing Arm was meshed using solid work software.

Table 3: Material property of the existing Rear Trailing Arm

Model Reference



Properties

Description	Material type
	St-37
Model type:	Linear Elastic Isotropic
Yield strength:	2.35e+008 N/m^2
Tensile strength:	3.6e+008 N/m^2
Elastic modulus:	1.75e+011 N/m^2
Poisson's ratio:	0.28
Mass density:	7850 kg/m^3

Total Nodes	10933
Total Elements	5259

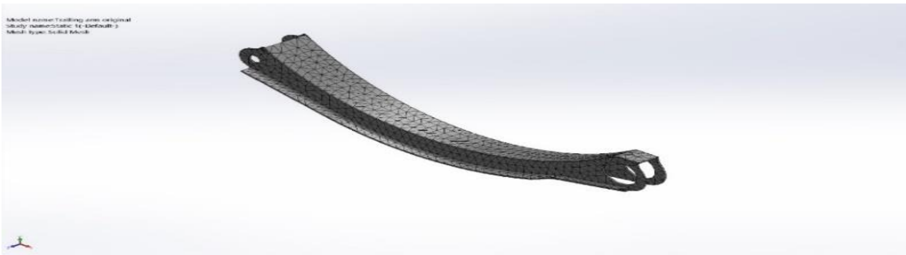


Fig. 8. Meshed existing Rear Trailing Arm

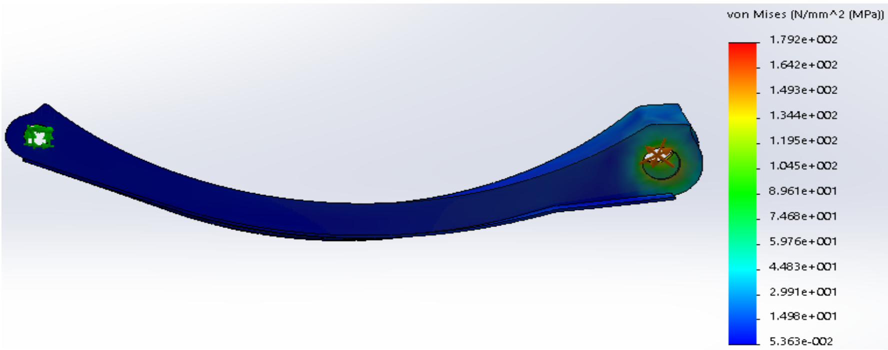


Fig. 9. Von mises stress for the existing Rear Trailing Arm of 1G bump load

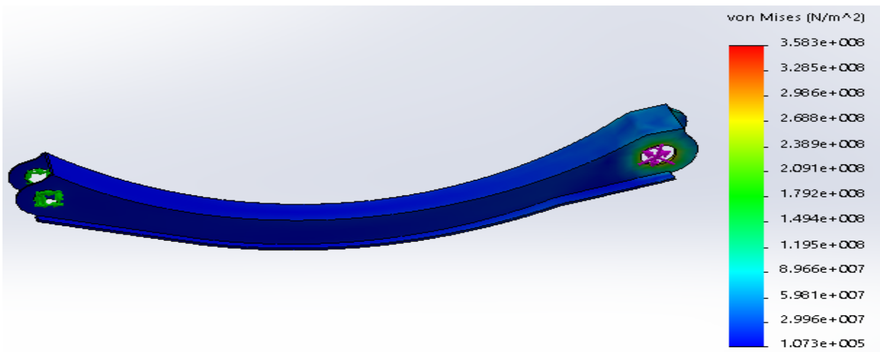


Fig. 10. Von mises stress for the existing Rear Trailing Arm of 2G bump load

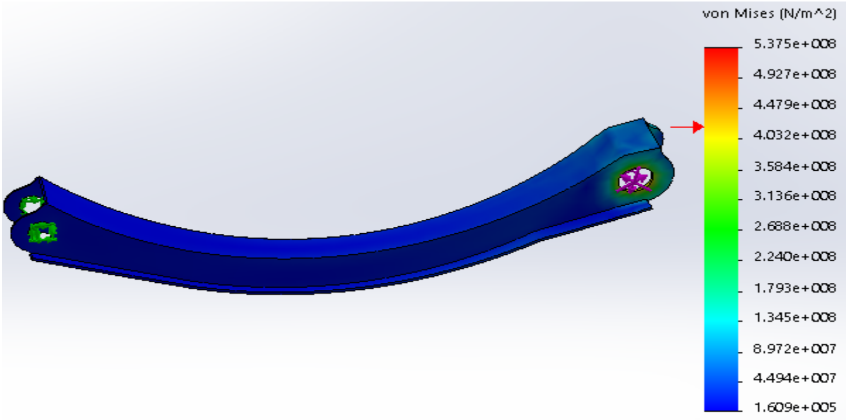


Fig. 11. Von mises stress for the existing Rear Trailing Arm of 3G bump load

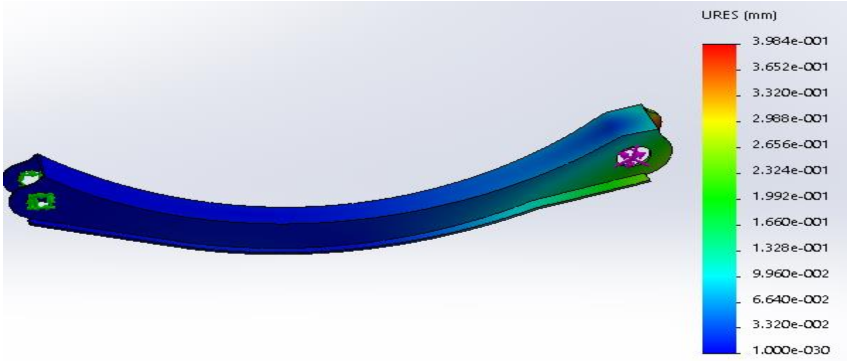


Fig. 12. Deformation for the existing rear trailing arm of 1G bump load

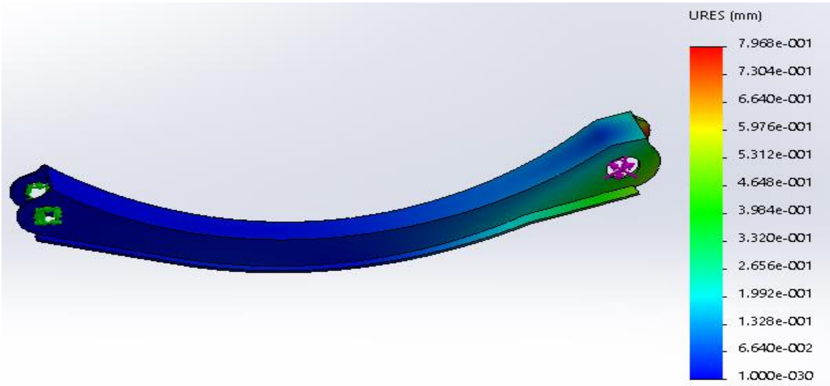


Fig. 13. Deformation for the existing rear trailing arm of 2G bump load

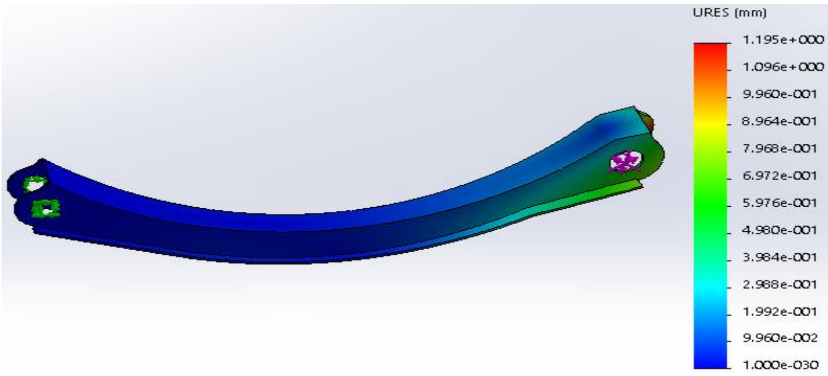


Fig. 14. Deformation for the existing rear trailing arm of 3G bump load

4.1.5 Static analysis of the existing Rear Trailing Arm

Meshed Rear Trailing Arm modeled in 3D is shown in Figure 8 given above. Material property of the existing Rear Trailing Arm is shown in Table 3 given above.

Static structural analysis was carried out for the existing Rear Trailing Arm under 1G, 2G and 3G bumps load conditions. The von misses stress for the existing rear trailing arm observed for 1G, 2G and 3G loadings are shown in the Fig. 9, 10 and 11 given above.

The maximum stress produced in the existing Rear Trailing Arm was 179.2MPa for 1G bump load, 358.3MPa for 2G bump load and 537.5MPa for 3Gbump load conditions. In this case, the existing Rear Trailing Arm has failed statically. Static stress analysis result for existing rear trailing arm is shown in Table 4 given below. Deformation graph for the existing rear trailing arm of 1G, 2G and 3G loading condition are shown in Fig. 12, 13 and 14 given above.

Table 4: Static stress analysis result for existing Rear Trailing Arm

Bump load	Von Mises Stress in MPa.	Static factor of safety
1G	179.2	1.31
2G	358.32	0.66
3G	537.5	0.44

4.1.6 Proposed solution selection

As it is indicated from the very beginning, the existing rear trailing arm material was ductile and soft. It was observed that the mechanical property can be improved by heat treatment. Though, exiting materials was failed statically, it might be due to lower safety factor as well as mechanical property could not sustained enough and it deformed at lower load. Solution could be optional, either it can work by changing the composition or replaced by other materials which have better mechanical properties rather than exiting materials.

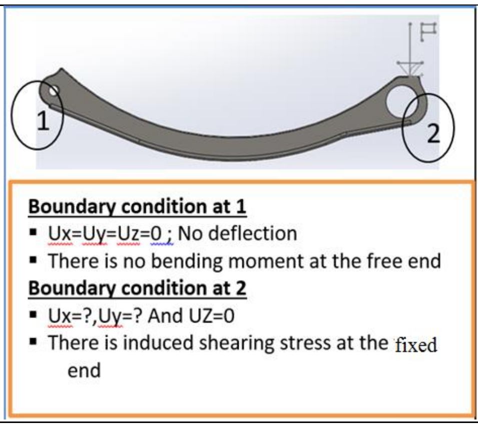
Analytical proposed solution could be done due to lack of time and further cost of the heat treatment work it was thought to go software analysis work only with changed materials.

(i) Static analysis of the modified design of Rear Trailing Arm

In the existing Rear Trailing Arm, more stress was observed. In the modified design model, the thickness and material property was improved by changing 2mm to 5mm of St-37, St-44 and St-52material quality. Static structural analysis is carried out for the modified design under 3G bump load condition for each material types. Material property of the modified rear trailing arm is shown in Table 5.

Table 5: Material property of the modified Rear Trailing Arm

Model Reference



Properties

Description	Material types		
	St-37	St-44	S-44
Model type:	Linear Elastic Isotropic	Linear Elastic Isotropic	Linear Elastic Isotropic
Yield strength:	2.35e+008 N/m^2	2.75e+008 N/m^2	3.55e+008 N/m^2
Tensile strength:	5.3e+008 N/m^2	5.3e+008 N/m^2	6.3e+008 N/m^2
Elastic modulus:	2.1e+011 N/m^2	2.1e+011 N/m^2	2.1e+011 N/m^2
Poisson's ratio:	0.28	0.28	0.28
Mass density:	7850 kg/m^3	7850 kg/m^3	7850 kg/m^3

The von Misses stress observed for 3G load condition of the Rear Trailing Arm is 128.1MPa for St-52, which lies within the yield limit of the material. Von Mises Stress for modified design of Rear Trailing Arm is shown in Fig.15 given below. Hence the design is safe. Fatigue analysis was also carried out for the new modified design of Rear Trailing Arm. Deformation for the modified design Rear Trailing Arm graph is shown in Fig. 16. Static stress analysis result for different materials of Rear Trailing Arm is shown in Table 6.

Table 6: Static stress analysis result for different materials of Rear Trailing Arm

Type of load	Material type	Von Mises Stress (MPa.)	Static factor of safety
3G	St-37	128.1	1.83
	St-44	128.1	2.15
	St-52	128.1	2.77

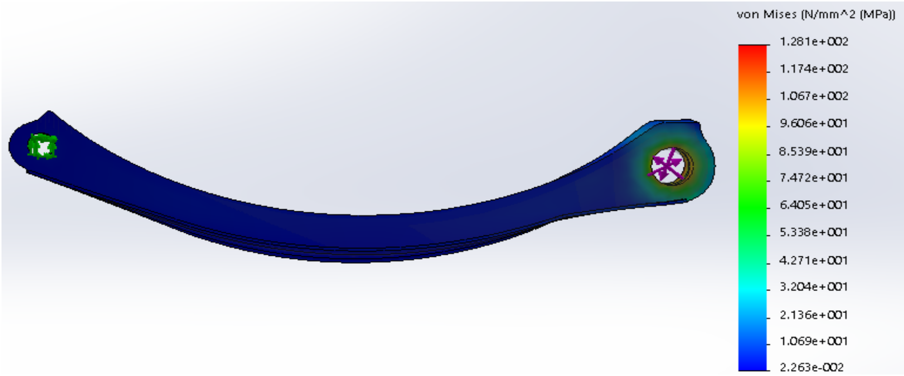


Fig. 15. Von Mises Stress for modified design of Rear Trailing Arm

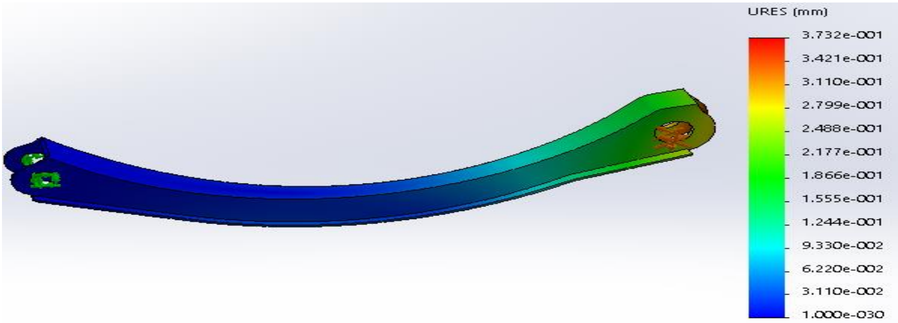
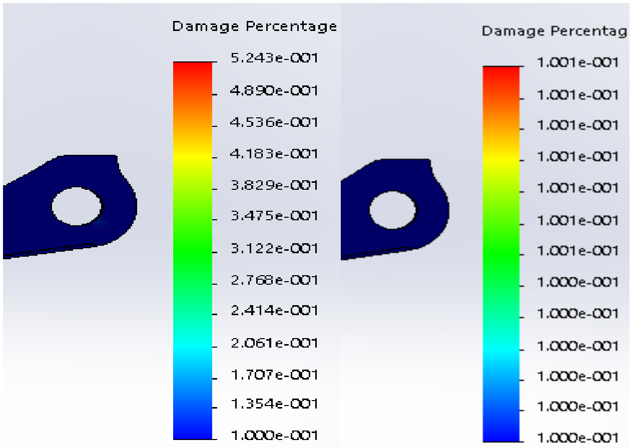


Fig. 16. Deformation for the modified design Rear Trailing Arm



(a) (b)

Fig. 17. Damage percentage of Rear Trailing Arm (a) existing (b) modified

The damage plot Fig. 17above (a) shows the percentage of the existing Rear Trailing Arm's and Fig. 17(b) after modified damaged percentage life consumed by the specific fatigue event. The fatigue event consumes only 0.52 % of the existing Rear Trailing Arm's life. The damage plot shows the percentage of the modified Arm's life consumed by the specific fatigue event. The fatigue event consumes only 0.1 % of the existing Rear Trailing Arm's life.

4.1.7 Fatigue analysis of the modified Rear Trailing Arm

The fatigue analysis of the existing Rear Trailing Arm is done as follows. Repeated stress cycle curve is shown in Fig. 18.

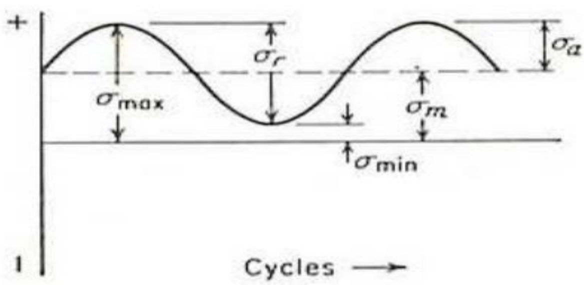


Fig. 18. Repeated stress cycle

a. Cyclic loading analysis for the existing Rear Trailing ArmSt-37 with 2mm material thickness

$$m_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$
$$m_m = \frac{537.5 + 0.161}{2} = 268.83MPa.$$

The stress ratio is given by;

$$R = \frac{\sigma_{min}}{\sigma_{max}} = \frac{0.161}{537.5} \approx 0$$

The amplitude ratio is given by

$$A = \frac{\sigma_a}{\sigma_m} = \frac{268.67}{268.83} \approx 1$$

The induced alternating stress developed in the Rear Trailing Arm is

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$
$$\sigma_a = \frac{537.5 - 0.161}{2} = 268.67MPa.$$

The endurance limit for steel material is given by

$$\sigma_e = 0.5 * \sigma_{UT} = 0.5 * 360 = 180MPa \ldots \text{for St} - 37 \text{ material quality with 2mm thickness}$$

b. Cyclic loading analysis for the existing Rear Trailing Arm St-44 and St-52 with 5mm thickness

$$m_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$
$$m_m = \frac{128.1 + 0.0023}{2} = 64.05MPa.$$

The stress ratio is given by;

$$R = \frac{\sigma_{min}}{\sigma_{max}} = \frac{0.0023}{128.1} \approx 0$$

The amplitude ratio is given by

$$A = \frac{\sigma_a}{\sigma_m} = \frac{64}{64.07} \approx 1$$

The induced alternating stress developed in the Rear Trailing Arm is

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$
$$\sigma_a = \frac{128.1 - 0.0023}{2} = 64MPa.$$

The endurance limit for steel material is given by

$$\sigma_e = 0.5 * \sigma_{UT} = 0.5 * 530 = 265MPa \text{ for St} - 44 \text{ material with 5mm thickness}$$

$$\sigma_e = 0.5 * \sigma_{UT} = 0.5 * 630 = 315MPa \text{ for St} - 52 \text{ material with 5mm thickness}$$

When the trailer is at motion, the suspension system has energy storing due to different springs. It can simply be modeled as a spring.

A bump in the road causes the wheel to move up and down perpendicular to the road surface.

$$FOS_{dynamic} = \frac{FOS_{static}}{IF}$$

$IF = 1 + \sqrt{1 + \frac{2h}{\partial_{rta}}}$[Prof. J.E. Akin *et al*].

Where:

- $h =$ average roughness of the ethiopian sphalt road

According to the Ethiopian road transport authority (ERTA), h=20mm

- ∂_{rta} = deflection of Rear Trailing Arm the minimum deflection, 32mm
- IF = Impact factor

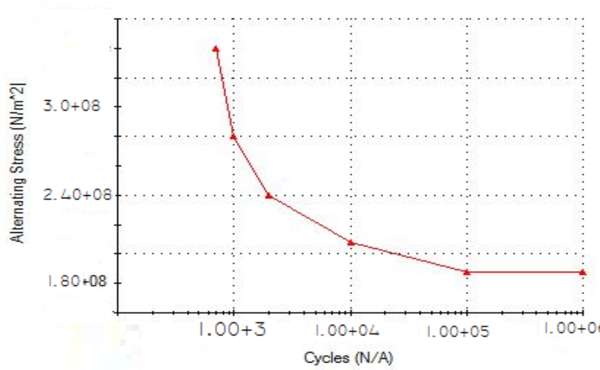


Fig. 19. Alternating cycles vs. number of cycles for the existing Rear Trailing Arm

The S-N diagram for the existing Rear Trailing Arm is plotted as shown in Fig. 19 given above. The S-N diagram represents nominal stress amplitude on Y-axis and number of cycles on X-axis. The endurance limit is 180 MPa which confirms from the Fig. 19 given above.

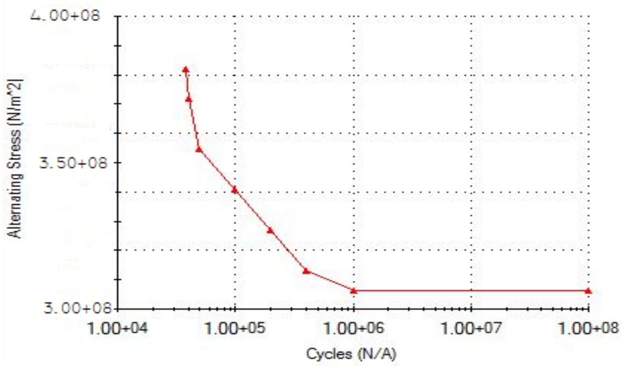


Fig. 20. Alternating cycles vs. number of cycles for the modified Rear Trailing Arm

The S-N diagram for the modified Rear Trailing Arm is plotted as shown in Fig. 20 given above. The S-N diagram represents nominal stress amplitude on Y-axis and number of cycles on X-axis. The endurance limit is 315MPa.

4.1.8 Summary of the Analysis

The summarized result analysis for the existing and new Rear Trailing Arm under1G, 2G and 3G loading conditions is tabulated in Table 7. Complete diagram analysis procedure is shown in Fig. 21 given below.

Table 7: Summarized static and dynamic factor of safety for existing and new Rear Trailing Arm

Material type	Thickness in mm	Loading type	FOS $_{static}$	FOS $_{dynamic}$
St-37 (Existing Rear Trailing Arm)	2	1G	1.31	0.53
		2G	0.66	0.26
		3G	0.44	0.17
St-37	2	1G	1.31	0.53
	3		1.69	0.68
	5		5.47	2.19
	2	2G	0.66	0.26
	3		1.03	0.41
	5		2.76	1.11
	2	3G	0.44	0.17
	3		0.69	0.28
	5		1.83	0.73
St-44	2	1G	1.54	0.61
	3		1.98	0.79
	5		6.40	2.56
	2	2G	0.77	0.31
	3		1.21	0.48
	5		3.24	1.29
	2	3G	0.51	0.20
	3		0.81	0.32
	5		2.15	0.86
St-52	2	1G	1.98	0.79
	3		2.55	1.02
	5		8.26	3.30
	2	2G	0.99	0.40
	3		1.56	0.62
	5		4.18	1.67
	2	3G	0.66	0.26
	3		1.04	0.42
	5		2.77	1.11

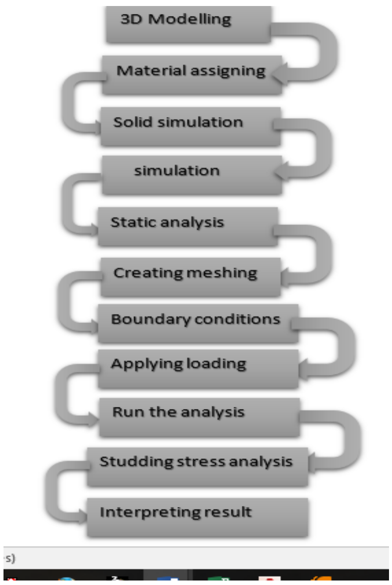


Fig. 21. Diagram of analysis procedure

4.2 Discussion

Experimental tests analysis like chemical compositions as well as mechanical tests were carried out only for existing St-37, not for St-44 and St-52. Material grade St-37 was confirmed form the average composition weight percent results obtained from the chemical testing experiment carried out using Belec Spectrometer. Fatigue analysis of St-44 and St-52 were chosen only after when found results of existing rear trailing arm was not satisfactory.

Form the literature point of view, mechanical property like yield strength and tensile strength result was supposed to be 235MPa and 360MPa for St-37. Mechanical testing was carried out using materials testing system for heat treatment and non-heat treated existing rear trailing arm. The average value of yield strength and tensile strength of the existing rear trailing arm were found 235.22MPa and 436.8MPa respectively.

The force that could be applied to the existing rear trailing arm was calculated. The static stress analysis and fatigue analysis were done using the SOLIDWORK simulation. The maximum stress produced in the existing rear trailing arm was 179.2MPa for 1G bump load, 358.3MPa for 2G bump load and 537.5MPa for 3G bump load conditions. The static factor of safety in 3G bump load condition was 0.44. In this case, the existing rear trailing arm has failed statically.

The fatigue analysis was also studied for existing rear trailing arm. The damage plot shows the percentage of the existing rear trailing arm's life consumed by the specific fatigue event. The fatigue event consumes only 0.52 %, while modified arm's life consumed by the specific fatigue event is only 0.1 % Arm's life.

As a result, the life cycle at which the product can fail was observed 190,700 cycles. From the static and fatigue analysis of the existing rear trailing arm, the product was directed to be fail by excessive deformation because of the softness, ductility and less rigidity.

New design modification to achieve the mechanical target of trailing arm was done by changing the material St-52. The existing material trial analysis was experimented by changing St -44 and St- 52 with an incremental material thickness from 2mm to 5mm and they were analyzed statically and fatigue under 3G bump load condition. The maximum stress produced in the new modified rear trailing arm was 128.1MPa for 3G bump load condition. The static factor of safety in 3G bump load condition was 2.15 for St-44 and 2.77 for St-52. The modified rear trailing St-52 arm has been observed safe with factor of safety 2.77 and 1.11 static and dynamic respectively.

Conclusion

The objective of this study is to make an analysis on how the rear trailing arm fails. Objective of this research study was achieved by choosing the St-52 materials instead of St-37. FEA to study the stress and fatigue analysis were carried out on the existing component as well as for St-44 and St-52. From the practical observation, it was found that the rear trailing arm gets out of use after a minimum deformation of 2.0 mm in length.

Compositional analysis done by chemical analysis confirms that material is St37-2. The mechanical properties data achieved by mechanical tests entails the confirmation of St37-2. Mechanical property data analysis elongation as well as reduction in area carried out for non-heat treated samples indicates that material is soft, ductile, less rigidity and easily exposed to deformation. From the static analysis of 3G bump load condition, von misses stress for existing trailing arm is 537.5MPa. As a result, it was indicated that the component has failed statically.

In the modified design model, the thickness and material quality was improved from 2mm in to 5mm of St-52 material quality. Static structural analysis was carried out for the modified design under 3G bump load condition. Von-Mises stress observed was 128.1MPa which is below the yield strength. The damage plot shows the percentage of the existing rear trailing arm's life consumed by the specific fatigue event.

Therefore, it is to be concluded that the failure life can be chosen by the St-52 instead of St-37 and St-44 and its changing thickness from 2mm to 5mm. The new modified design is statically and dynamically safe.

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