Failure analysis of leaf spring suspension system for heavy load truck vehicle

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Abstract: The failure of leaf spring suspension system used in heavy load truck vehicle TATA LPT 1613TCIC model was investigated in the research reported in this paper. In order to analyse the variations in the chemical composition, micro-structural analysis along with material specification has been performed. The failed leaf spring fractured part was analysed by using a visual inspection technique and scanning electron microscope (SEM) analysis. Based on the fractography study, it was inferred that the failure of the fractured part was due to the cyclic load. This load lead to fatigue growth on leaf spring of the model truck vehicle. Then finite element analysis of leaf spring was carried out to find out the root cause of the leaf spring suspension system. The failure parameters were also optimised for the truck vehicle during safe operation on the road. The fatigue life of the proposed leaf spring has increased in comparison with the existing model lifecycles.

Keywords: leaf spring; finite element analysis; SEM; scanning electron microscope; fatigue life.

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1 Introduction

The suspension system of a vehicle delivers easy handling for a comfortable ride. In addition to this, the suspension system also provides better fatigue life for the automotive components (Zamanzadeh et al., 2015; Frey, 1996). The earliest forms of leaf spring suspension system are still being used in all military vehicles, commercial vehicles, and construction vehicles (Kong et al., 2016). The leaf springs are one of the major parts of the automobile suspension system and their contribution to effective functioning of the vehicle is crucial (Shamim and Anwer, 2014). The main components of an automobile suspension system are leaf spring, shock absorbers and linkages (Amrute et al., 2013). These automobile components enable a relative motion between the wheels (Rajesh and Sreekumar, 2016). A simple form of leaf spring suspension system is used in wheeled vehicles, which mainly has a laminated structure that absorbs energy when the load is applied (Prawoto et al., 2008). The laminated structure deforms with high deflections due to the impact developed by shear and compressive loads, and retained in the original structure when the load is released (Guan et al., 2017). When a vehicle is moved over a dip and bumps on the road the leaf spring absorbs the shock and the vibrations, thus providing good comfortable drive (D'Silva and Jain, 2014). The leaf spring helps to control a vehicle by keeping the wheels in contact with the road surface (Hou et al., 2007). When the vehicle moves over a dip and bump, these springs are helping to prevent it from bouncing uncontrollably. The main use for a leaf spring is carrying the whole weight of the vehicle (Kumar et al., 2014). The semi-elliptical leaf springs are normally multi-leaf springs in automobile vehicles (Gowd and Goud, 2012). The main disadvantage of leaf springs is the stress created during running condition, which restricts the use of more expensive high-strength materials. These components are subjected to millions of varying cyclic stresses leading to failure ultimately (Zadeh et al., 2000, Kong et al., 2014, 2016). The failure of the leaf springs could lead to hazardous accidents, thus it is also considered as a safety component. The leaf springs have also a low flexibility (Kumar and Vijayarangan, 2006).

The new trends in material sciences are to introduce composite materials replacing the conventional materials (Hou et al., 2007). The composite materials used for manufacturing the leaves are known as composite leaf springs (Patunkar and Dolas, 2011). Normally composite leaf springs are made from glass-fibre reinforced plastics (GFRPs), since this can meet the requirements of a vehicle suspension system (Yinhuan et al., 2011). These composite materials yield a significant weight reduction without any reducing the load carrying capacity (Ramakanth and Sowjanya, 2013). They have a good weight to strength ratio. Some notable demerits of composite materials are that the cost is high, when compared to steel and aluminium, and lower volume production methods. Also the database availability is lacking.

The leaf springs are used in different environmental conditions. Composite materials operating under wet condition may result in failure; they have a tendency to absorb moisture which changes their mechanical dimensions and properties (Mazumdar, 2002). Composites are very difficult to recycle (Yang et al., 2012). Thus the chances of failure are more for composite leaf springs in adverse environments (Shokrieh and Rezaei, 2003). These factors are considered as a necessity of modifying design and using conventional material in leaf spring design. Finally, decide on a new modified leaf spring design can be proposed. The CAD model is done in Solidworks software and the models are imported to ANSYS workbench. The analysis is used for static structural analysis and transient analysis is used for dynamic analysis. The modal analysis is used for finding mode shapes, which are used to determine the frequency response of designs using harmonic response module.

Several researchers have carried out investigations and reports such as performance and noise analysis of mono and multi leaf springs, development models of leaf spring to study the kinematics characteristics, analysis of leaf spring suspension characteristics, generation of different algorithms for analysing the suspension system, developing new materials and changing conventional material, etc. (Kumar and Aggarwal, 2017). A few notable studies include finite element analysis of multi leaf springs by Frey (1996). They have introduced air suspension as another system (Frey, 1996), performance and analysis, comparison of mono-, multi- and hybrid composite leaf springs with that of steel leaf springs (Guan et al., 2017), and a study of leaf spring suspension characteristics (Hoyle, 2004).

In the research reported in this paper, failure of a leaf spring was analysed by means of visual inspection and SEM fractography mechanical characterisation. In the SEM analysis, the images of the sample were scanned using a focused beam of electrons (Pantazopoulos and Vazdirvanidis, 2008). This beam of electrons will interact with the atoms in the sample which is tested. This interaction produces different signals which contain the information about composition and surface topography.

The suspension system of a vehicle model was developed by the CAD modelling software package. Then static analysis and life cycle calculation were performed by using the FEA simulation software with two different material compositions and also parameters were optimised. These studies were carried out in the modified and the existing model to improve the displacement stresses involved, and a factor of safety under varying safe loading conditions was analysed. The main objective was to reduce the bending stress and improve the fatigue life of the proposed model.

2 Problem investigation

The proposed work is to investigate the possible causes of failure and suggest an improvement for the leaf spring of the TATA LPT 1613TCIC model truck. An existing leaf spring model was found to fail due to repeated cyclic stresses under varying load conditions. A leaf spring suspension system model truck TATA LPT 1613TCIC is shown in Figure 1. Thus a root cause analysis of the failure of leaf spring was carried out during this research. Then, based on this analysis report, a method to minimise the fracture growth by modifying the design parameters and material composition was conceived (Guimaraes et al., 2016). This research attempted to increase the safe stress and stiffness, and improve life cycle of the proposed model. The FEM method is an ideal tool, to find the payload and safe stresses for this kind of problem. The fatigue fracture of a leaf spring is shown in Figure 2.

Figure 1 Leaf spring model of TATA LPT 1613TCIC (see online version for colours)

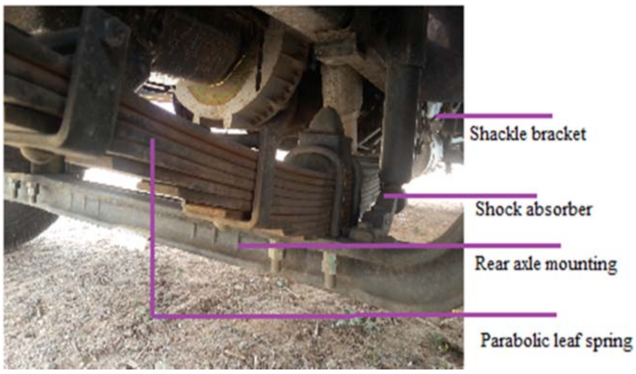
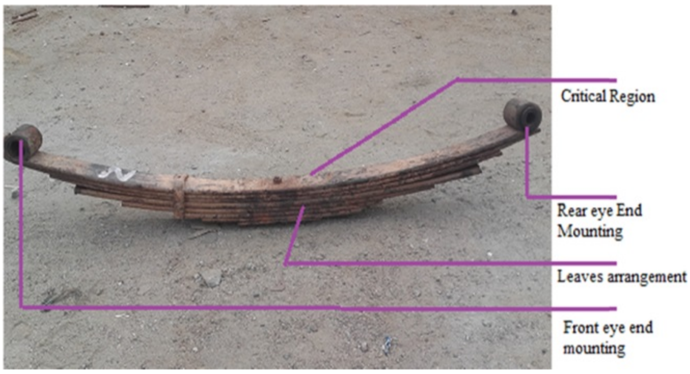


Figure 2 TATA LPT 1613TCIC model leaf spring (see online version for colours)



2.1 Failure analysis

Leaf springs are crucial part of heavy load vehicle. It is necessary to reduce the vertical bumps and vibration impacts due to an uneven road surface by means of consideration in the spring deflection. Thus potential energy will be saved as strain energy and then released slowly in order to increase the potential energy storage capabilities of a leaf spring and ensure more compliance in the spring. A TATA LPT 1613TCIC model truck has been designed for a maximum power of 130 BHP and the maximum torque is 2400 rpm. The investigation found frequent failure of leaf spring suspension system and half of the fractures had deep secondary cracking along the mid-plane, which could lead to accidents. In the end of this failure analysis, it was observed that the damaged leaf spring has failures due to fluctuating loads, with the static load of the vehicle and road imperfections affecting the payload during its lifetime cycle. The fracture failure of leaf spring suspension system is shown in Figure 3.

Figure 3 Fracture leaf spring deep secondary cracking (see online version for colours)

3 Material and geometry

The first step in a failure analysis was to select the desirable material. In general, a leaf spring material used is plain carbon steel-grade C25, which has 0.90–1.0% of carbon. The grade content of C25 steel has good mechanical properties and machinability. The hardness of the leaf spring was measured in Rockwell Hardness Tester; this test was conducted using ‘C’ scale under 55–65 HRC. The material properties observed during this test are given in Table 1 (SAE, USA, 1995).

Table 1 Material properties

Material	*50 Cr1V23* (*existing*)	*50Si2Mn90* (*proposed*)
Young’s modulus (GPa)	201	206
Ultimate tensile strength (MPa)	158.40	160.00

3.1 Compositional analysis

The chemical compositional analysis of the leaf spring was carried out by spectrophotometry and the results are presented in Table 2 conforming to specifications of 50Cr1V23 material. From the investigation, it was found a manganese silicon steel (50Si2Mn90) is a suitable material for increasing a fatigue life of a leaf spring compared to the existing material.

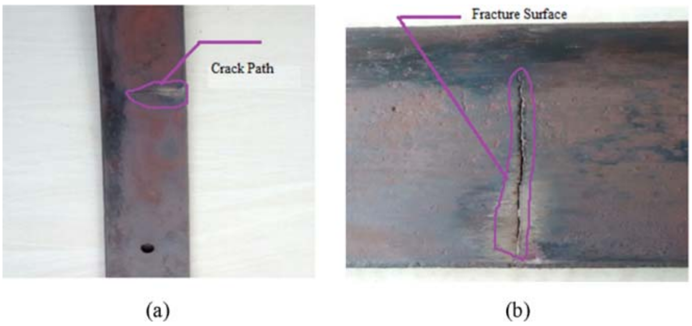
Table 2 Material composition

Specification		*Composition %*							
Steel	*Grade*	*C*	*Si*	*Mn*	*Pmax*	*Smax*	*Cr*	*Mo*	*V*
50Cr1V2 (Existing)	1	0.45–0.55	0.10–0.35	0.080–1.10	0.04	0.035	0.09–1.2	–	0.15–0.30
50 Si 2 Mn 90(Proposed)	3	0.50–0.60	1.50–2.00	0.80–1.00	0.05	0.05	–	–-	

3.2 Visual analysis

In the next stage, an examination of a failed leaf spring was carried out. The visual inspection revealed that a fracture occurred near the mid region of the master leaf as shown in Figure 4(a). Due to fatigue, a fracture was observed on failure part. The fractured surface is shown Figure 4(b). When a failed leaf spring is cut into parts, the halves of fractured surface rub against each other leading to surface damage. Finally, a typical cyclic loading fracture was observed. This fatigue crack initiated at the end of the corner region, where the crack path marks are present, and propagated along the mid plane surface area along a critical path length due to cyclic loads resulting in failure of the master leaf spring.

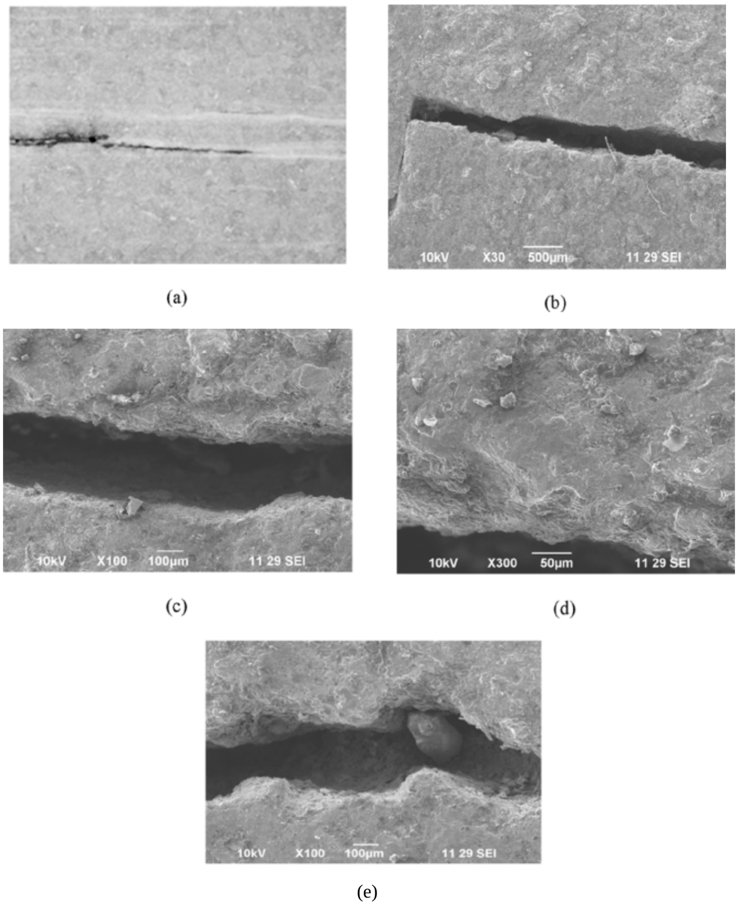
Figure 4 (a) Crack regions and (b) fracture crack path length (see online version for colours)



3.3 Scanning electron fractography

In the research reported in this paper, scanning electron microscope (SEM) fractography was used to examine the fractured part at high magnification with a focused electron beam to generate a different level of signals at the surface of solid specimens. Also it was capable of performing SEM at selected points on a specimen. During this investigation, no surface coating was applied in the fractured area of failure part. Additionally, through this SEM analysis, an accelerating voltage of 10 kV was used. The fractured part was scoured ultrasonically using acetone. The two halves of the fractured surface area were observed and results were found in failure regions. Images were taken at specific regions: at middle of the fracture surface and close to the edge. Fractured part of the surface was damaged due to rubbing. The crack along most of the inner surface was clearly rusted. The half of the fractured area of a cross from inner to outer surface area was rusted. A micro-structure of the failure part was examined by using SEM.

Figure 5 (a) Mid-plane crack in leaf spring follows a white band; (b) crack shows rusted surface compared with fracture next to it; (c) Marks were observed propagating from the edges (inner edge) and (d) SEM fractography shows the fatigue fracture



The SEM fractography of the fractured surface clearly shows the crack growth starts from the end region surface up to inner mid-plane area following the white band shown in Figure 5(a). This crack was indicative of a corroded surface at outer region shown in Figure 5(b). The crack at the middle plane was rusted but comparatively smaller, than the inner region, and marks were observed propagating from the edges (inner edge) of the examination as illustrated in Figure 5(c). Due to cyclic loading, micro-cracks and oxide particles were visible on the surface of the fractured part. The failure of the leaf spring part will occur owing to fatigue fracture as shown in Figure 5(d) and (e) which are due to high cycle fatigue event.

4 Leaf spring design

When designing a leaf spring, the maximum loads are fixed for operational requirements. Next, design variables such as camber and span length are fixed based on accessible area and a factor safety is designed as per standard. Other parameters, such as the number of full length leaves, diameter of the eye and the number of graduated leaves are considered arbitrarily. In order to carry heavy loads, additional full length leaves are added to the master leaf for a suspension system. Then, a standard procedure is applied with the leaf spring design to achieve the length and width and of leaves in the first iteration (SAE, 1990). Next, static structural analysis results are compared within the safe design. If a design is not within the safe limit, a second iteration design is done and tested until the stresses and deflections are found within a safe limit. Finally, the parameters are calculated and designed in the safest condition. This formula was used for the TATA LPT 1613 TCIC model truck to design the chosen leaf spring.

Design methodology for maximum allowable stress is calculated by using the following relation,

$$\sigma_{max} = \frac{S_{yt}}{FoS} \tag{1}$$

The correction factor for maximum allowable deflection was,

$$m = \frac{N_f}{N} \tag{2}$$

The standard laminated leaf spring modification does not change the stress value, but deflection equation needs changes as follows:

$$\delta_c = \frac{1-4m+2m^2[1.5-\ln(m)]}{(1-m^3)} \tag{3}$$

The maximum allowable stress and deflection are given by equations (4) and (5) respectively,

Bending stress in graduated leaves $\sigma_{bg} = \dfrac{18pL}{3(n_f+2n_g)bt^2}$ (4)

Bending stress in extra full length leaves $\sigma_{be} = \dfrac{12pL}{3(n_f+2n_g)bt^2}$ (5)

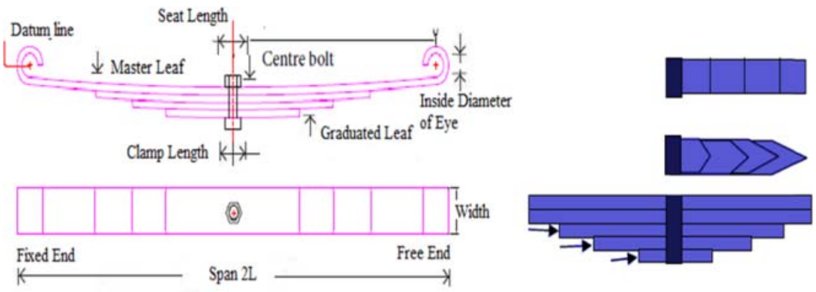
The leaf spring is considered as a simply supported beam and the value of the maximum allowable deflection after modification is given by the equations (6) and (7).

$$\delta_{\max} = \frac{12pL^3}{Ebt^3(3n_f+2n_g)} \tag{6}$$

$$K = \frac{8Enbt^3}{3L^3} \tag{7}$$

The maximum bending stress, deflection and stiffness of leaf spring can be calculated. Figure 6 shows the construction of leaf spring model.

Figure 6 Construction of leaf spring (see online version for colours)



The specification of the vehicle model TATA LPT 1613TCIC truck is given in Table 3. The thickness and number of graduated leaves are modified in the proposed design and also the master leaf and the full length of first leaf thickness are changed but width dimension is maintained while designing. In this research, the Solidworks software package was used for CAD modelling of the leaf spring. The model was prepared as per the specification given in Table 3.

Table 3 Design parameters of leaf spring

Material	*50 Cr 1 V 23(existing)*	*50 Si 2 Mn 90 (Proposed)*
Total length of the span (eye to eye) (mm)	1340	1340
Distance between U- bolts (mm)	80	80
Free camber(At no load condition)mm	96	96
No. of full length leaves	2	2
Total number of leaves (N)	8	6
Thickness of each leaves (mm)	10	12
Width of each leaf spring (mm)	70	70
Maximum load (Metal to Metal Position) 'N.'	10,500	10,500
Factor of safety (FOS)	2.0	2.0
Weight (kg)	34.9	33.8
Leaf spring span	Front – 1150 mm Rear – 1220 mm	
Leaf spring width	Front – 70 mm. Rear – 70 mm	

4.1 Loads on leaf spring

The leaf spring is considered to be a simply supported beam, where one end is fixed and the other end is loaded (IS 1135, 1995). The leaves are arranged in concentric arc model type. In this arrangement, each leaf has a contact at all loading conditions. A spring is fixed to the central point of the wheel axle; hence a wheel exerts force in the spring and the end support reactions of the spring come from the carriage. The normal operation of a leaf spring is to compress and absorb road shocks, so that it slides and bends to allow for suspension movement. In addition to shocks, springs may carry lateral loads, brake torque and driving torque. The leaf spring and two ends form an eye shape. A front eye is directly coupled with a pin to the frame side, so that it rotates freely about the pin, but with no translation movement. The rear eye is attached to shackle; this is a flexible link that slides along the X-axis direction.

In the finite element model, the leaf spring model was formed with elements or nodes. The model was constrained to two rotational degrees of freedom and all translational degrees of freedom. The load transfers to the chassis through its contact, and this causes stresses in each leaf member under all loading conditions. When load is applied to the rear eye side, it can also rotate about the pin direction. When a load is applied or removed on the link, it oscillates. The maximum level of load is applied 10,500 N at the leaf spring model and the number of associated nodes is 12, so the load at each node is 750 N. The load is equally split among all the nodes and elements related to the centre location of the leaf spring.

4.2 Results and discussions

The varying loading conditions of the existing model were analysed and compared with the modified design. Stress analysis of the existing model reveals that the stress level is analysed to show that the maximum load acts at the rear end plate which is fixed to the axle. Thus, there is a need to modify the design parameters and change the material composition. In order to achieve safe stress value, several modifications were carried out in the existing model. The modified model was analysed by using ANSYS 14.0 software package. Figure 7(a) and (b) shows the deformation under the maximum loading condition of leaf spring models.

From Figure 7(c) and (d), the maximum equivalent von Mises stress developed in existing model under static load condition was calculated. The von Mises equivalent stress is maximum in the existing model compared with the proposed model. The stress reduction of 23.42% was achieved in the proposed design when compared to the existing model. It was also observed that the vertical deflection of the model throughout the loading was minimised.

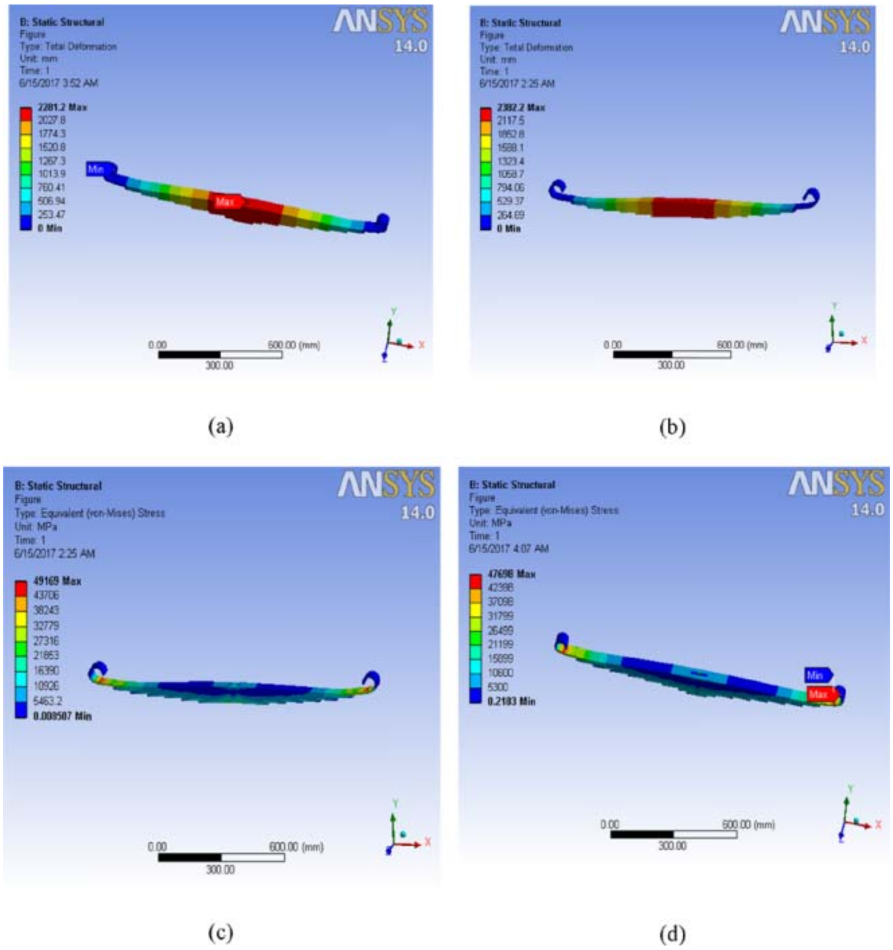
4.3 Modal analysis

Modal analysis is an essential of the harmonic response analysis system to analyse the natural frequency of the model. Modal analysis was carried out for finding frequencies which are tabulated in Table 4. The system is considered to be about three-degrees of freedom, since the system of modal frequencies has the corresponding mode shapes shown in Figure 8(a) and (b).

Table 4 Mode shapes

Mode shape	*1*	*2*	*3*	*4*	*5*	*6*	*7*	*8*	*9*	*10*
Frequency (Hz) Existing	27.99	53.97	70.71	111.5	125.09	149.08	239.65	271.3	327.49	364.48
Frequency (Hz) proposed	29.54	54.172	72.369	117.31	125.93	143.36	239.62	271.9	338.4	353.23

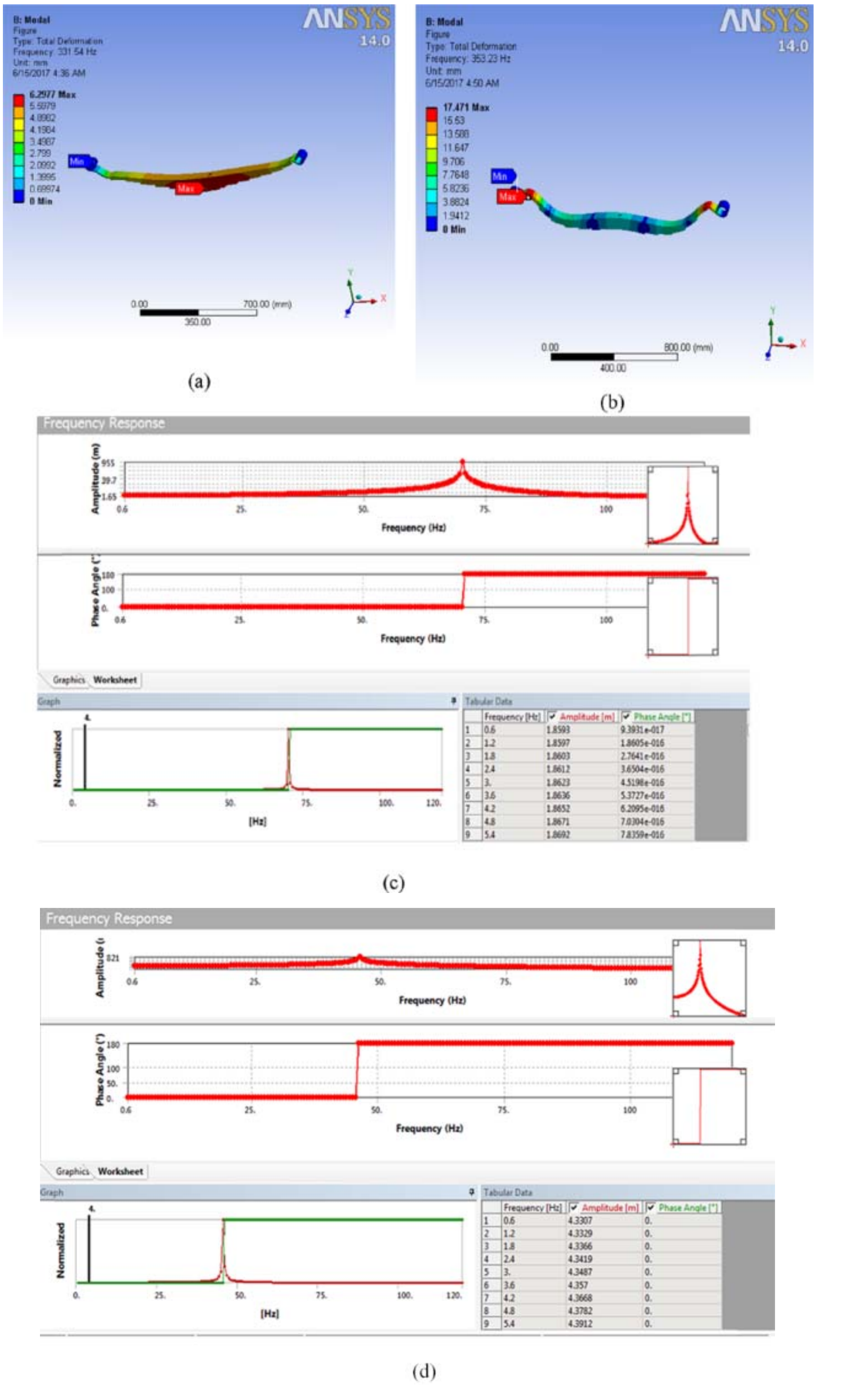
Figure 7 (a) Deflection load at 10,500 N existing leaf spring; (b) deflection load at 10,500 N proposed leaf spring; (c) von Mises stress at load 10,500 N existing leaf spring and (d) von Mises stress at load of 10,500 N proposed leaf spring (see online version for colours)



From Table 4, the amplitude of vibration is maximum at modal frequency 353.2 Hz for proposed design and hence natural frequency is considered. Similarly, an existing model of maximum modal frequency 368.9 Hz is equal to the natural frequency and for the model. It is also noticed that when compared, the natural frequency is maximum and it is minimum for the existing model.

A harmonic analysis is used for determining a steady-state sinusoidal response of the structure under varying loading conditions at a given frequency. A frequency response loading considers only the frequency. The peak response considers a natural frequency of the structure in a harmonic analysis. Since the natural frequencies are known, simulation can cluster the results near the natural frequencies instead of using evenly spaced results from the graphs shown in Figure 8(c) and (d). The graph result shows the amplitude response of existing model magnification interval is less than that of the proposed model.

Figure 8 (a) Mode shape existing model; (b) mode shape proposed model; (c) frequency response existing model and (d) frequency response proposed model (see online version for colours)



From Table 5, it is clearly observed that the stiffness is calculated by equation (7) is high for proposed designs, and hence it is a safer design which has 26.8% higher stiffness comparing to existing design. Stiffness analysis is relation between stiffness (K), natural frequency (ωn) and mass (m) is given by the equation is

$$K = m\omega n^2 \tag{7}$$

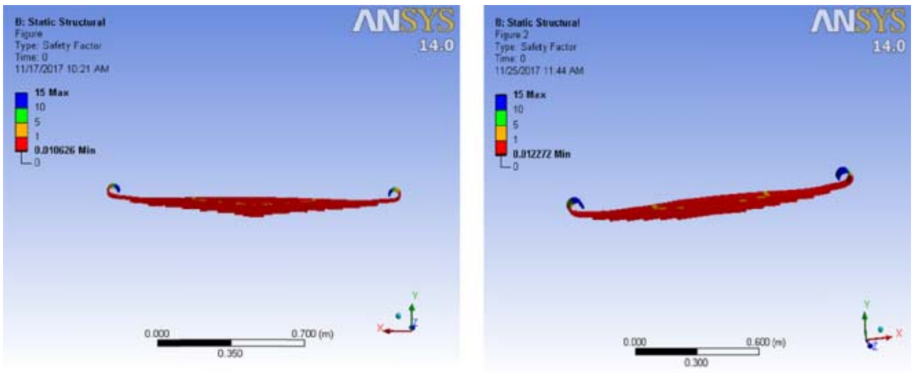
Table 5 Stiffness and mass

Design	*Mass* (kg)	*Natural frequency* ω_n (rad/s)	*Stiffness K* (N/mm)
Existing	34.9	149.08	12.47
Proposed	33.8	143.36	16.467

4.4 Fatigue analysis

Fatigue analysis was carried out during this research, and predicts component fatigue failures during the design phase. The fatigue strength calculated by applying varying levels of cyclic stress to individual analysis of materials used and high cycle fatigue life was measured. The graphical representation of fatigue data plots the load stress amplitude on the horizontal axis against the number of cycles to failure on the vertical axis. The two materials were tested in a series of dropping stress levels until failure occurs within a selected maximum number of cycles. The simulation and experimental results showed that safety and fatigue life were improved in proposed design. Fatigue life and factor of safety tests were conducted through FEM software for existing and proposed models, as shown in Figures 9(a), (b) and 10(a) and (b). The results are show that fatigue life is improved in the proposed design.

Figure 9 (a) Factor of safety for existing and (b) factor of safety for proposed (see online version for colours)



(a) (b)

Table 6 shows the deflection and bending stress of the existing and the proposed model. The allowable safe stress determined between the corresponding loads is 2500 to 10,500 N; hence the near corresponding safe load is given in Table 6. From the table it was found that the maximum value of deflection and bending stress of the existing and modified models were compared with the results of the leaf spring. At the end of the research, the cost estimation for the proposed model was calculated. The cost estimation inferred that the cost of the product decreases by 10.25% for the proposed model.

Figure 10 (a) Fatigue life – existing and (b) fatigue life – proposed (see online version for colours)

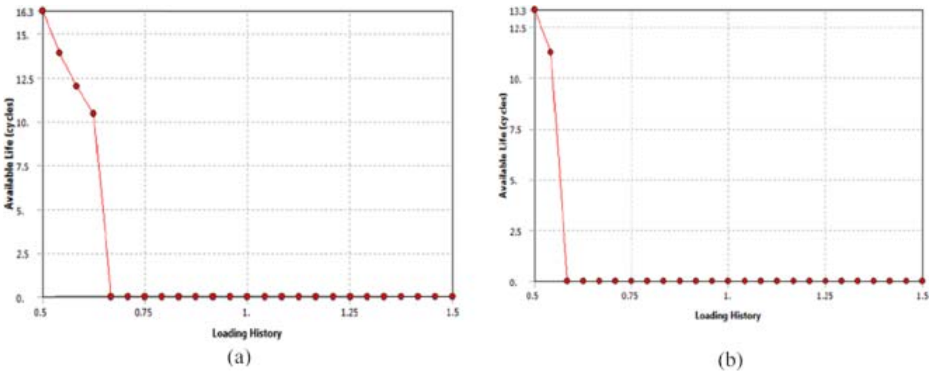
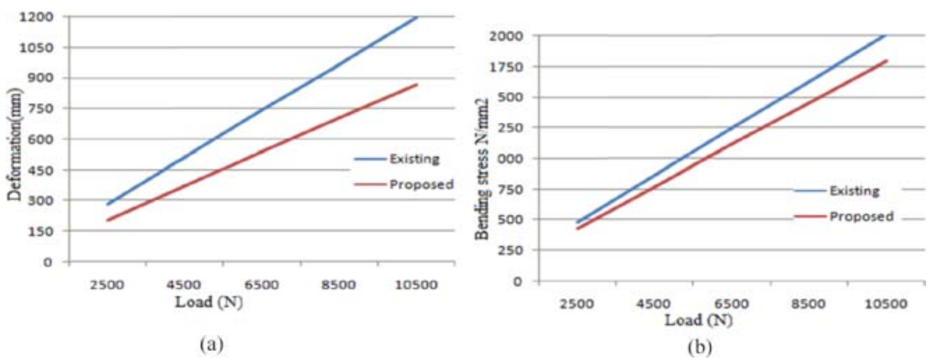


Table 6 Results comparison

	Existing model		*Modified model*	
Load (N)	*Deflection* (mm)	*Bending stress* (N/mm^2)	*Deflection* (mm)	*Bending stress* (N/mm^2)
2500	285.01	478.57	206.91	427.29
4500	513.02	861.42	372.45	769.13
6500	741.04	1244.28	537.98	1110.96
8500	969.05	1627.17	703.52	1452.8
10500	1197.06	2010.02	869.05	1794.6

The graphs shown in Figure 11 indicate the comparison results of deflection and bending stress of the existing and modified models. The results show an improvement in the life cycle of a model with varying load.

Figure 11 Load vs. deflection load vs. bending stress (see online version for colours)



5 Conclusion

At the end of this research work, the new leaf spring suspension system has been designed. The visual inspection and fractography examination using SEM analysis of the fracture proved that the fracture was initiated due to the cyclic loading. FEM simulation results indicate that there is a notable reduction of stress in the proposed design, when compared with the existing design. In leaf spring design the main parameters noticed from stiffness analysis were fatigue life and factor of safety. The mass of the proposed design as per assembly was reduced by 3.25% when compared with the existing design. Also the stiffness was improved by 32.02% when compared to the existing model. The proposed model also has a higher natural frequency which is a notable finding. The damping ability also can be improved by the fact that the new design provides stiffness to the leaf spring that is higher than the existing design when spaced conveniently. The proposed design of the leaf spring model is also a cost-effective solution. Thus the objective of providing an economical and feasible design revision for the existing leaf spring to prevent frequent failure was achieved.

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Nomenclature

A	Area, mm^2
L	Length, mm^2
σ_{max}	Maximum allowable stress, N/mm^2
δ	Deflection, mm^2
Nf	Number of full length leaves
Ng	Number of graduated leaves

N	Total number of leaves
t	Thickness, mm
k	Stiffness of the system (N/m)
$2L$	Overall length of span, mm
M	Mass of the system, kg
ω_n	Natural frequency (rad/s)
m	Ratio of the number of full length leaves to the total number of leaves