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Abstract - The design of leaf spring has been a constant challenge for automotive and manufacturing engineers and it has undergone multiple revisions [2, 3 and 4]. The aim of this paper is to investigate and analyze how failure occurs on the semi-elliptical master leaf spring of a commercial car by analytical approach and using FEM simulation to ascertain the failure condition and to provide a cost-effective design modification for the same. The currently used 10 mm thick master leaf fails repeatedly at a particular zone close to the spring hanger end. After multiple trials for different thickness values and materials, recommendations were given for a better and modified design of the master leaf spring.

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I. Introduction

Leaf springs are crucial suspension elements used on light passenger vehicles to give a riding comfort. The leaf spring should absorb the vertical vibrations and impacts due to road irregularities by means of variations in the spring deflection so that the potential energy is stored in spring as strain energy and then released slowly, ensuring a more compliant suspension system. Leaf springs can serve both damping as well as springing functions. The leaf spring can either be attached directly to the frame at both ends or attached at one end, usually the front, with the other end attached through a shackle, a short swinging arm. The shackle takes up the tendency of the leaf spring to elongate when compressed and thus makes for softer springiness.

Failure prediction in large-scaled structures that are subjected to extreme loading conditions has been of utmost interest in the scientific and engineering community over the past century [4]. Failure of mechanical assembly component is a common phenomenon due to fracture that occurs almost everywhere in mechanical structures. The main cause of failure of leaf spring is due to large bending behavior [5-6].

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Figure 1 : Leaf Spring with Suspension Mechanism [5]

II. Literature Review

The shape of leaf springs has undergone multiple changes and revisions over time from 'flat' to 'elliptical' to the present-day shape of being parabolic. The parabolic spring is light-weighted, has superior capacity to store strain energy and offers better riding comfort and is widely used now-a-days in automotive applications. But it has manufacturing complications.

Different sub-assembly of vehicles, including leaf springs are made of steels with low strength and high ductility. Their failure modes are usually characterized by ductile tearing. Fatigue life prediction is based on knowledge of both the number of cycles the part will experience at any given stress level during that life cycle and environmental factors. The local strain-life method can be used pro-actively for a component during early design stage [7, 8]. For strain-based fatigue life prediction, Coffin–Manson relationship is normally applied [8], which is,

$$\varepsilon_a = \frac{\sigma'_f}{E}\left(2N_f\right)^b + \varepsilon'_f\left(2N_f\right)^c \qquad (1)$$

Where, E is the material modulus of elasticity, ε_a is the true strain amplitude, $2N_f$ is the number of reversals to failure, σ'_f is the fatigue strength coefficient, b is the fatigue strength exponent, ε'_f *is* the fatigue ductility coefficient and c is the fatigue ductility exponent.

Meanwhile, two mean stress effect models commonly used are the Morrow [8] and Smith-Watson-Topper (SWT) [5] strain-life models.

Mathematically, the Morrow model is defined by,

$$\varepsilon_a = \frac{\sigma'_f}{E}\left(1-\frac{\sigma^m}{\sigma'_f}\right)\left(2N_f\right)^b + \varepsilon'_f\left(2N_f\right)^c \quad (2)$$

The SWT model is defined by,

$$\sigma_{\max}\varepsilon_a = \frac{\left(\sigma'_f\right)^2}{E}\left(2N_f\right)^{2b} + \sigma'_f\,\varepsilon'_f\left(2N_f\right)^{b+c} \quad (3)$$

In 2008, Fuentes et al., studied leaf spring failure and concluded that the premature failure in the studied leaf springs which showed the fracture failure on a leaf was the result of mechanical fatigue and it was caused by a combination of design, metallurgical and manufacturing deficiencies [9].

III. Failure Analysis

The existing design parameters are listed in Table 1.

Table 1 : Design Parameters

Parameter	Value
Material selected	20MoCr4 (ISO grade)
Total span length (eye to eye)	1200 mm
Camber height	137 mm
Width of master leaf leaves	60 mm
Normal static load	1500 N

The leaf spring considered is of simply supported beam type, where the central location of the spring is fixed to the wheel axle. Therefore, the wheel exerts the force F on the spring and support reactions at the two ends of the spring come from the carriage. Maximum deflection, bending stress and Von-Mises stress distribution were estimated by considering the master leaf as a simply supported beam.

For uniform width of master leaf, the maximum stress and displacement were analytically calculated using,

$$\sigma_{\max} = \frac{3FL}{bh^2} \quad (4)$$

and

$$\delta_{\max} = \frac{3FL^3}{Ebh^3} \quad (5)$$

Where, *E, F, L, b* and *h* represent the Young's Modulus, normal load, span length, width and thickness of the master leaf.

IV. Modified Design

The spring steels commonly used for making leaf springs are low alloy steels like Carbon steel, Si steel, Mn steel, Si–Mn steel, Si–Cr steel, Mn–Cr steel, Cr–V steel, Si–Cr–V steel, Si–Ni–Cr steel, Ni–Cr–Mo steel and Cr–Mo steel. In this paper the material property selected for analysis is a Carbon steel of 56SiCr7, tempered in the temperature range of 400°C∼550°C [10,11].

Table 1 : Spring steel standards - ISO683-14(1992-08-15) [6]

No	Specification		Composition %						
	Steel grade	grade	C	Si	Mn	P_{max}	S_{max}	Cr	Mo
1	59Si7	5	0.55- 0.63	1.60-2.0	0.60 - 1.00	0.030	0.03	---	---
2	56SiCr7	3	0.52-0.59	1.6-2.0	0.70-1.00	0.030	0.03	0.2-0.4	----
3	61SiCr7	7	0.57-0.65	1.6-2.0	0.70-1.00	0.030	0.03	0.2-0.4	----
4	55SiCr63	2	0.51-0.59	1.6-2.0	0.50-0.80	0.030	0.03	0.55-0.85	----
5	55Cr3	8	0.52-0.59	0.15-0.4	0.70-1.00	0.030	0.03	0.70-1.0	----

V. FEM - Based Failure Analysis

The semi-elliptical master leaf was modeled using Solidworks 2012 software. Shackle and bushing were considered for boundary conditions only. Shot-peening and Nip stresses and the frictional effect were also omitted.

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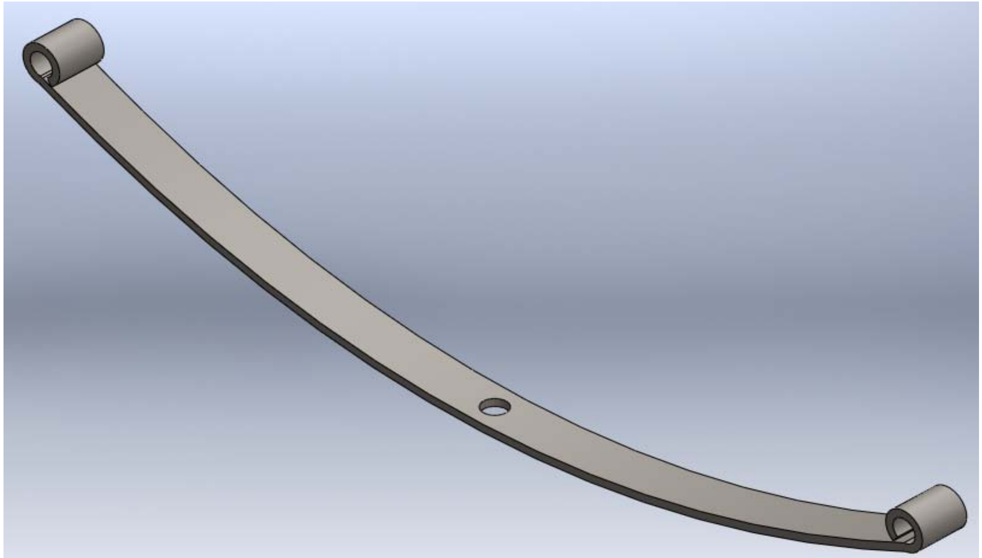

Figure 2 : Master Leaf (CAD Model by Solid Work 2012)

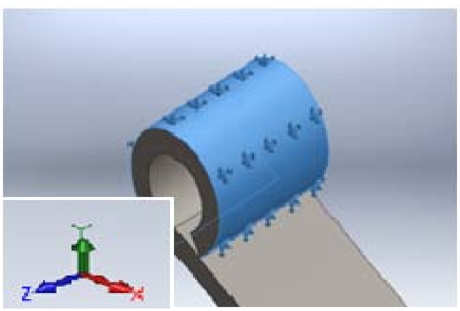

Figure 3 : Hanger end

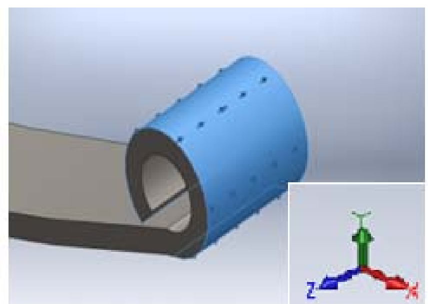

Figure 4 : Shackle end

Table 2 : Mesh Details

Jacobian points	4 Points
Element Size	9.49789 mm
Tolerance	0.474894 mm
Total Nodes	16022
Total Elements	8513

VI. Results and Discussion

The post processing of the modeled master leaf (existing), gave the stress, strain and displacement plots as shown in Fig. 6.

It is evident that the Von-Mises stress at the hanger end is critical (604 MPa) and is close to the yield stress value (650 MPa), even in static loading conditions. Reversed fatigue loading affects the life of master leaf causing pre-mature failure in the same zone (near hanger end) as reported in the passenger car service station.

To overcome this failure, multiple trials have been made in terms of change of material and thickness of the semi-elliptical master leaf. Si steel substantially increases the elastic limit of the steel and improves the resistance to permanent set of springs. Hence Si steel of ISO specification 56SiCr7 is chosen from the ISO spring steel standard shown in Table 1. Similarly, after repeated trials for varying thicknesses, 14 mm thickness is chosen for the uniform thickness of master leaf. The FEA results for the modified design were depicted in Fig. 7.

The fatigue test result (S-N curve) for dynamic loading of master leaf and the comparison of the results obtained were shown in Fig. 8 and Table 3 respectively.

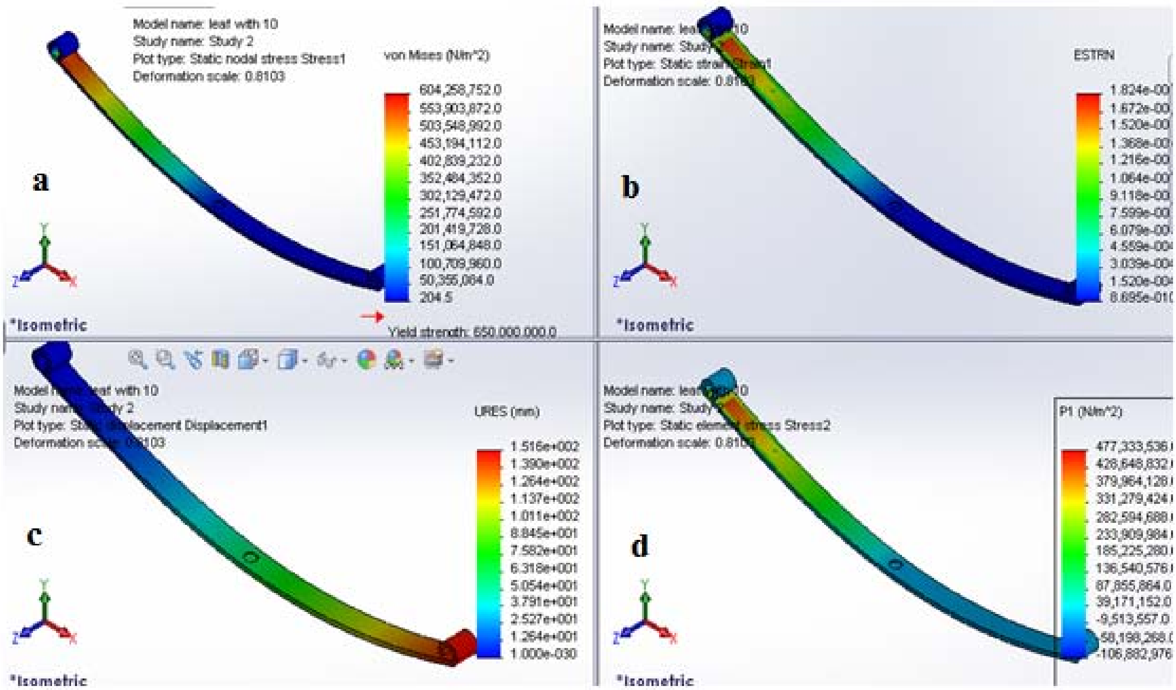

Figure 6 : For Existing Design

[(a) Stress plot (b) Strain plot (c) Displacement plot (d) First principal stress]

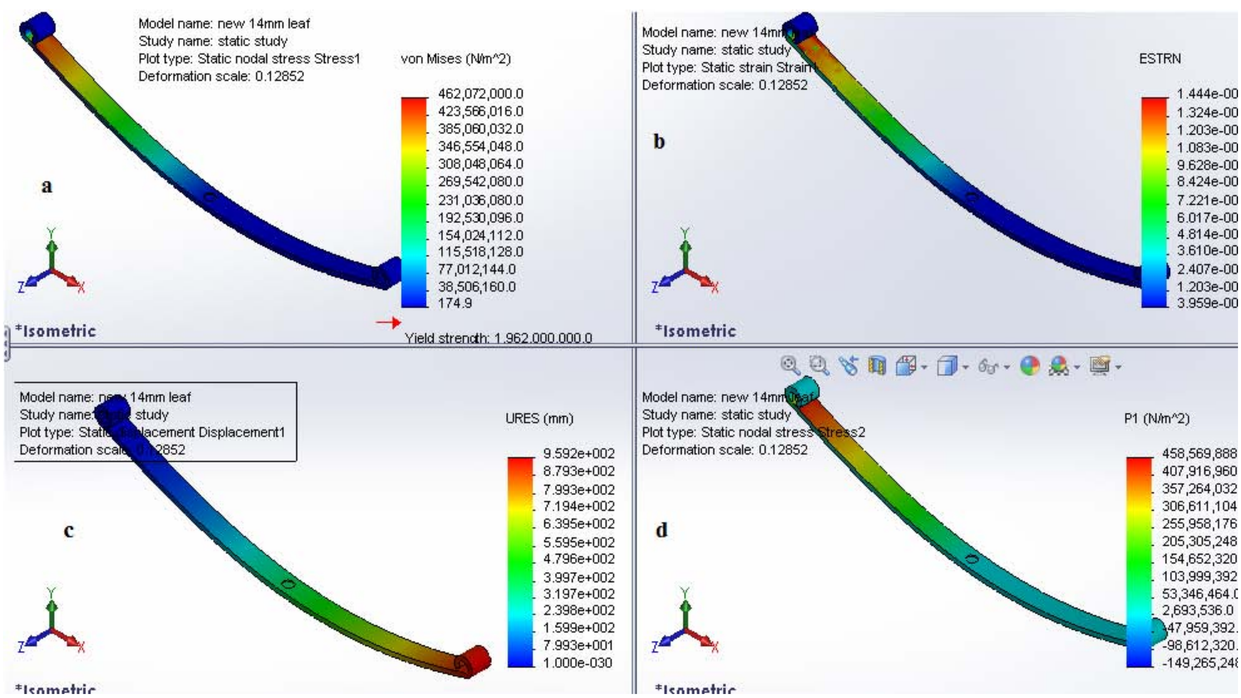

Figure 7 : For Modified Design

(a) Stress plot (b) Strain plot (c) Displacement plot (d) First principal stress

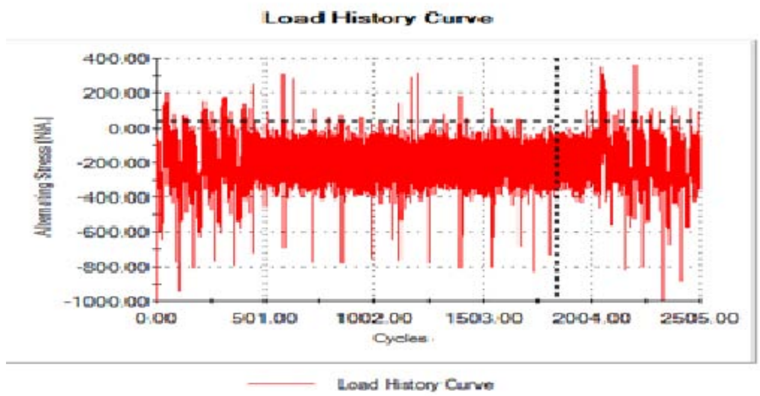

Figure 8 : S-N Curve for Master leaf (Modified Design)

Table 3 : Comparison of Results (Existing and Modified Designs)

S. No.	Parameter	Existing Design	Modified Design
	Analytical		
1	Bending stress (MPa)	450	230
2	Max. deflection (mm)	7.7	28
	FEM-based		
1	Von mises stress (MPa)	604	402
2	Resultant displacement (mm)	151.6	101
3	Elastic Strain	0.0304	0.0018

The following inferences can be taken from the above results:

- ✓ The revised design shows a marked reduction in Von Misses stress. The maximum Von Misses stress induced reduced by 33%. The yield strength of 56SiCr7 steel used in revised design is 1962 MPa, which is nearly 5 times that of maximum Von-Mises stress induced. This ensures high factor of safety and reliable operation even under dynamic conditions.
- ✓ The maximum bending stress induced (analytical) for static loading conditions reduced by 49%.
- ✓ FEM based resultant displacement registered 33% reduction.

Thus the modified design involving change of material with an increased thickness of 14 mm has substantial improvements in terms of reduction of V o n M i s e s stress, higher yield strength, lessened resultant displacement and higher factor of safety. Hence the authors recommend this as a cost-effective solution, as desired by the customer.

The other alternatives like use of parabolic master leaf with varying thickness and use of composite materials are not advocated, since the objective was to give an economic and feasible design revision for the existing semi-elliptical master leaf, which is prone to frequent failure.

VII. Acknowledgment

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