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Design, Manufacturing and Testing of Polymer Composite Multi-Leaf Spring for Light Passenger Automobiles - A Review

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1. Introduction

Weight reduction has been the main/primary focus of automobile manufactures. Suspension leaf spring, a potential item for weight reduction in automobiles, accounts for 10-25 percent of unsprung weight, (which is considered to be the mass not supported by leaf spring). Application of composite materials reduces the weight of leaf spring without any reduction on the load carrying capacity and stiffness in automobile suspension system(Daugherty,1981;Breadmore,1986;Morris,1986). A double tapered beam for automotive suspension leaf spring has been designed and optimized(Yu & Kim,1988). Composite mono leaf spring(Rajendran & Vijayarangan,2001) has also been analyzed and optimized.

Leaf spring should absorb vertical vibrations due to road irregularities by means of variations in the spring deflection so that potential energy is stored in the spring as strain energy and then released slowly. So, increasing energy storage capability of a leaf spring ensures a more compliant suspension system. A material with maximum strength and minimum modulus of elasticity in longitudinal direction is the most suitable material(Corvi,1990) for a leaf spring. Important characteristics of composites(Springer & Kollar,2003) that make them excellent for leaf spring instead of steel are higher strength-to-weight ratio, superior fatigue strength, excellent corrosion resistance, smoother ride, higher natural frequency, etc. Fatigue failure is the predominant mode of in-service failure of many automobile components, especially the springs used in automobile suspension systems. Fatigue behaviour of Glass Fiber Reinforced Plastic epoxy (GFRP) composite materials has been studied(Hawang & Han,1986). A composite mono-leaf spring has been designed and their end joints are anlysed and tested for a light weight vehicle (Shivasankar & Vijayarangan, 2006). Experimental and numerical analysis are carried out on a single leaf constant cross section composite leaf spring (Jadhao & Dalu, 2011). Theoretical equation for predicting fatigue life, formulated using fatigue modulus and its degrading rate, is simplified by strain failure criterion for practical application. A prediction method for fatigue strength of composite structures at an arbitrary combination of frequency, stress ratio and temperature has been presented (Yasushi,1997).

In the present work, a 7-leaf steel spring used in a passenger car is replaced with a composite multi leaf spring made of glass/epoxy composites. Dimensions and number of leaves of steel leaf spring (SLS) and composite leaf spring (CLS) are considered to be same. Primary objective is to compare their load carrying capacity, stiffness and weight savings of CLS. Ride comfort of both SLS and CLS is found and compared. Also, fatigue life of SLS and CLS is also predicted. This chapter of the book explores the work done on design optimisation, finite element analysis, analytical & experimental studies and life data analysis of steel and composite leaf springs (Senthilkumar & Vijayarangan,2007).

2. Steel Leaf Spring (SLS)

2.1 Design and finite element analysis

Design parameters of 7-leaf steel spring that exists in a light passenger car (available in India) are given in Table 1. The spring is assumed to be a double cantilever beam, even though the leaf spring is simply supported at the ends. Also, this spring is geometrically and materially symmetrical so that only one half is considered with cantilever beam boundary conditions for the analysis to save the calculation time. Axle seat is assumed to be fixed and loading is applied at free eye end.

Parameters	Values
Total length (eye-to-eye), mm	1150
Arc height at axle seat(Camber), mm	175
Spring rate, N/mm	20
Number of full length leaves	2
Number of graduated leaves	5
Width of the leaves, mm	34
Thickness of the leaves, mm	5.5
Full bump loading, N	3250
Spring weight, kg	13.5

Table 1. Parameters of steel leaf spring.

A stress analysis was performed using two-dimensional, plane strain finite element model (FEM). A plane strain solution is considered because of the high ratio of width to thickness of a leaf. Model is restrained to the right half part only because the spring is symmetric. The contact between leaves is emulated by interface elements and all the calculations are done using ANSYS (version 7.1) (Eliahu Zahavi,1992). Nodes are created based on the values of co-ordinates calculated and each pair of coincident nodes is joined by the interface elements that simulate action between neighboring leaves. Element selected for this analysis is SOLID42(Eliahu Zahavi,1992) which behaves as the spring having plasticity, creep, swelling, stress stiffening, large deflection, and large strain capabilities. Element is defined by four nodes having two degrees of freedom at each node: translations in the nodal x and y directions. Interface elements CONTA174 that is defined by eight nodes and TARGE170 are used to represent contact and sliding between adjacent surfaces of leaves. The contact elements themselves overlay the solid elements describing the boundary of a deformable body and are potentially in contact with the target surface, defined by TARGE170. This target surface is discretized by a set of target segment elements (TARGE170) and is paired with its associated contact surface via a shared real constant set. An average coefficient of friction 0.03 is taken between surfaces(SAE manual). Also, analytical solution is carried out using spring design SAE manual.

2.2 Static testing

The static testing on existing steel leaf spring was carried out using an electro-hydraulic test rig which is depicted in Fig.1. The rig has the ability to apply a maximum static load of 10 kN. It has a display unit to show both load and corresponding deflection. The loading was gradually from no load to full bump load of 3250 N. The strain gauges were employed to measure strain and to calculate stress. The experimental data corresponding to steel leaf spring is given in Table 2. Maximum normal stress, $\sigma 11$ from FEM is compared to the experimental solution under full bump loading (error, 8.63%). There is a good correlation for stiffness in experimental, analytical and FEM methods (Table 2).

Parameters	Experiment	Analytical	FEM
Load, N	3250	3250	3250
Maximum stress, MPa	680.05	982.05	744.32
Maximum deflection, mm	155	133.03	134.67
Maximum stiffness, N/mm	20.96	24.43	24.13

Table 2. Stress analysis of steel leaf spring using experimental, analytical and FEM.

3. Composite Leaf Spring (CLS) (Senthilkumar & Vijayarangan,2007)

Applicability of CLS in automobiles is evaluated by considering the types of vehicles and different loading on them. Theoretical details of composite mono-leaf spring are reported (Ryan,1985; Richrad et al., 1990). In some designs, width is fixed and in each section the thickness is varied hyperbolically so that thickness is minimum at two edges and is maximum in the middle (Nickel,1986). Another design, in which width and thickness are fixed from eyes to middle of spring and towards the axle seat width decreases hyperbolically and thickness increases linearly, has been presented (Yu & Kim, 1988). In this design, curvature of spring and fiber misalignment in the width and thickness direction are neglected. A double tapered CLS has been designed and tested with optimizing its size for minimum weight(Rajendran & Vijayarangan,2002). A composite mono-leaf spring has also been designed and optimized with joint design(Mahmood & Davood,2003). The mono-leaf spring is not easily replaceable on its catastrophic failure. Hence, in this work, a composite multi leaf spring is designed and tested for its load carrying capacity, stiffness and fatigue life prediction using a more realistic situation.

3.1 Material selection

Material selected should be capable of storing more strain energy in leaf spring. Specific elastic strain energy can be written as:

$$S = \frac{1}{2}\frac{\sigma_t^2}{\rho E} \tag{1}$$

where σ_t is allowable stress, E is modulus of elasticity and ρ is density. Based on specific strain energy of steel spring and some composite materials(Yu&Kim,1988), E-glass/epoxy is selected as spring material having the mechanical properties given in Table 3. This material is assumed to be linearly elastic and orthotropic.

Parameters	Values
Modulus of elasticity, GPa	E_{11}, 38.6
Modulus of elasticity, GPa	E_{22}, 8.27
Modulus of shear, GPa	G_{12}, 4.14
Poisson ratio,	v_{xy}, 0.26
Tensile strength, MPa	σ_{t11}, 1062
Tensile strength, MPa	σ_{t22}, 31
Compressive strength, MPa	σ_{c11}, 610
Compressive strength, MPa	σ_{c22}, 118
Shear strength, MPa	τ_{12}, 71

Table 3. Mechanical properties of E-glass/epoxy(Springer & Kollar,2003).

3.2 Layup selection

Stored elastic strain energy in a leaf spring varies directly with the square of maximum allowable stress and inversely with the modulus of elasticity both in the longitudinal and transverse directions according to Eq. (1). Composite materials like E-glass/epoxy in the direction of fibers have good characteristics for storing strain energy. So, lay up is selected to be unidirectional along the longitudinal direction of spring. This also helped in fabricating process using filament-winding machine.

3.3 Design and finite element analysis of composite leaf spring

With the extensive use of laminated composite materials in almost all engineering fields, the optimal design of laminated composites has been an extensive subject of research in recent years. The dimensions of the composite leaf spring are taken as that of the conventional steel leaf spring. Each leaf of the composite leaf spring consists of 20 plies of thickness 0.275 mm each. The number of leaves is also the same for composite leaf spring. The design parameters selected are listed in Table 4.

A 3-D model of the leaf spring is used for the analysis in ANSYS 7.1, since the properties of the composite leaf spring vary with the directions of the fiber. The loading conditions are assumed to be static.The element chosen is SOLID46, which is a layered version of the 8-node structural solid element to model layered thick shells or solids. The element allows up to 250 different material layers. To establish contact between the leaves, the interface elements CONTACT174 and TARGET170 are chosen.

Individual leaves are fabricated using a filament-winding machine. A fiber volume fraction of 0.6 is used. All individual leaves are assembled together using a center bolt and four side clamps. Also metal spring eyes are fixed at both the ends. CLS is tested with an electro-hydraulic leaf spring test rig (Fig. 1). Four CLSs were manufactured and tested. The spring, which provided the lowest stiffness and highest stress values, has been considered for comparative purpose because it satisfies the fail-safe condition. The reason for the stiffness and stress variations may be due to variation in volume fraction obtained in the fabrication process or due to lack of complete curing. A reasonably good weight reduction (68.15%) is achieved by using CLS (4.3 kg) in place of SLS (13.5 kg).

Parameters	Values
Thickness of each leaf, mm	5.5
Width of the each leaf ,mm	34
Thickness of the fiber, mm	0.2
Width of the fiber, mm	34
Thickness of the resin, mm	0.075
Width of the resin, mm	34
Thickness of single layer, mm	0.275
Number of layers	20

Table 4. Design parameters of composite leaf spring.

Load indicator

Load cell

Leaf spring

Fig. 1. Electro-hydraulic leaf spring test rig.

For a light passenger vehicle with a camber height of 175 mm, static load to flatten the leaf spring is theoretically estimated to be 3250 N. Therefore, a static vertical force of 3250 N is applied to determine the load-deflection curves (Fig. 2).The load is gradually increased to obtain the deflection of steel spring first untill it becomes comelpetely flat. Then, for similar deflection in composite leaf spring, the loads are measured for composite leaf spring. From Fig.2, it is understood that the deflection increases linearly as load increases in both steel and composite leaf springs. For a full bump load of 3250 N, composite leaf spring deflects to 94 mm only while steel leaf spring deflects 175 mm. The FEM results of longitudinal stress and deflection of CLS are shown in Figs.3&4. During full bump load test, experimental stress measurement (Fig. 5) is carried out to verify the results of FEM analysis (Figs 3 & 4). Fig.3 shows the variation of stress in CLS along the length of the spring. Fig.4 shows the deflection of CLS at various points along the length. It is found that CLS developes the maximum stress of about 215 MPa and it deflectes about 60 mm. E-glass/epoxy composite leaf spring has spring constants 34.57-53.59 N/mm. Thus, all the data of spring constants for CLSs are greater than the design value, 20 N/mm. The reason for increased stiffness is lower density of E-glass/epoxy composite combination.

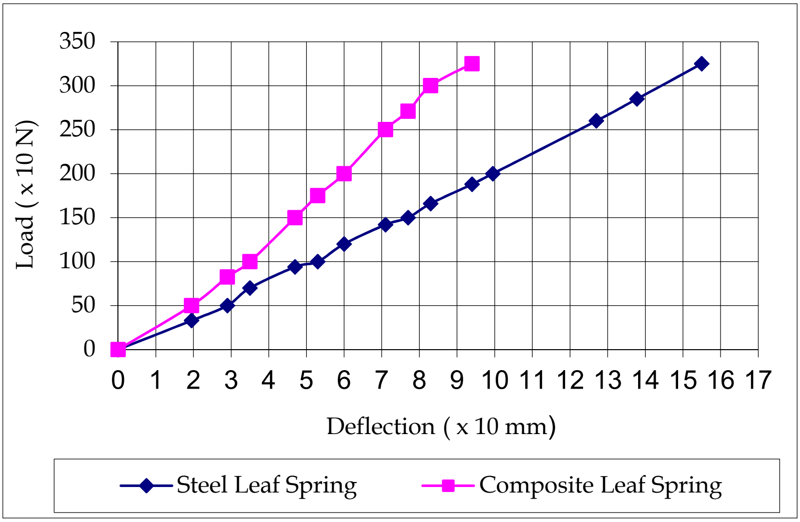


Fig. 2. Load – deflection curves for steel and composite leaf springs.

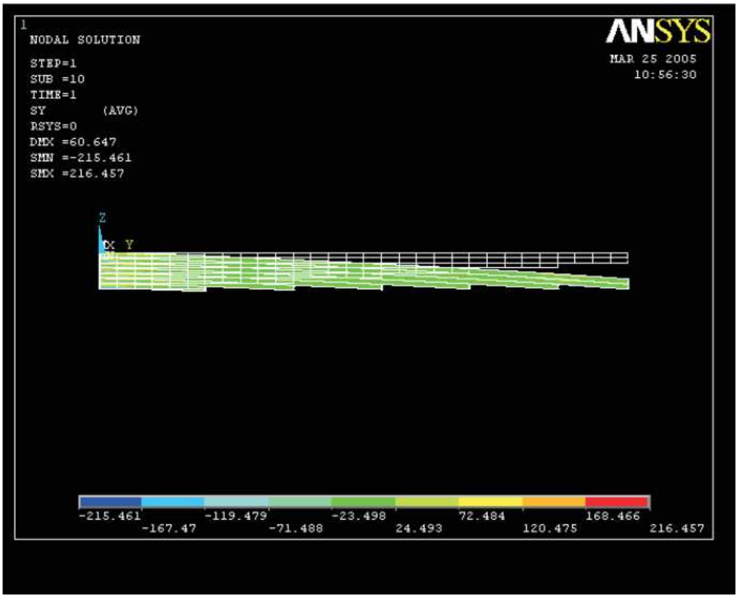


Fig. 3. FEM results of longitudinal stress of composite spring.

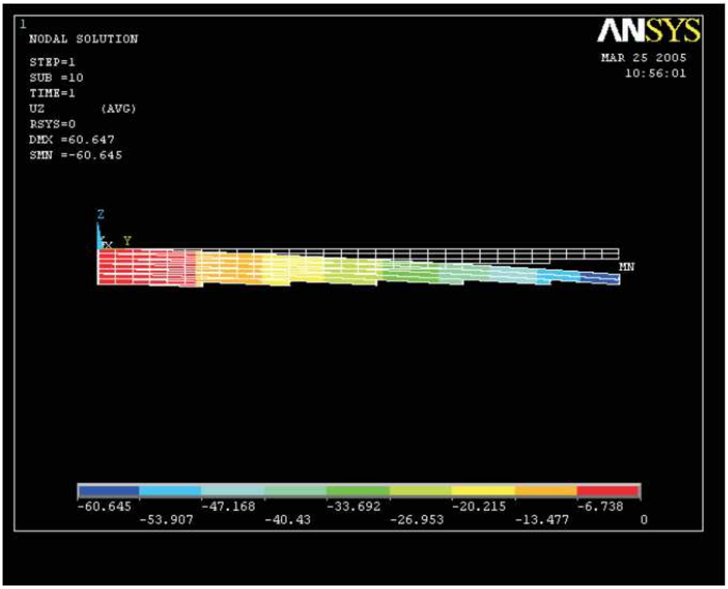


Fig. 4. FEM results of deflection of composite spring.

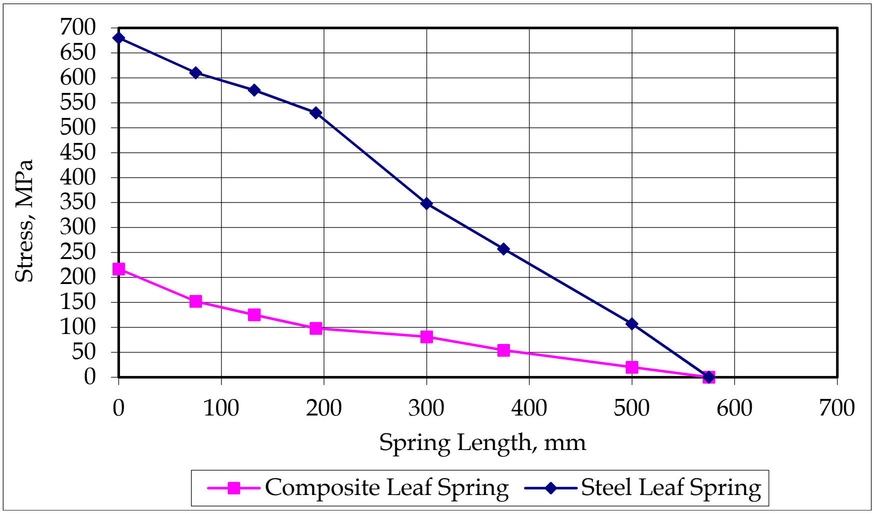


Fig. 5. Variation of experimental stress of steel and composite springs.

Leaf spring is analyzed under transverse loading condition. The longitudinal compressive strength of composite used in this analysis is less than its longitudinal tensile strength. So failure criterion stress is longitudinal compressive stress. Maximum longitudinal compressive stress (Fig. 5) is 222 MPa for CLS. At a same loading, maximum stress developed in SLS is 680 MPa (Table 2, Fig. 5). When compared with stress developed in SLS, less stress (67.35%) is developed in CLS. Compressive strength of fiber glass/epoxy is 610

MPa and yielding stress of steel is 1175 MPa. So, factor of safety obtained in SLS is 1.73, while in CLS it is 2.75. Experimental deflection of CLS under full bump loading is 94 mm (Fig. 2), which is less than the maximum value (175 mm). It shows that CLS is stiffer (64.95 %) than SLS. Table 5 gives the results of analysis of CLS using experimental, analytical and finite element methods. The variation of longitudinal stress of SLS and CLS is also presented in Figs.6&7. Fig. 6 shows the variation of longitudinal stress of steel leaf spring in FEM and experimental analysis. There is about 8% higher stress value obtained in the FEM than experiments. This is due to the fact that the constraints given at the ends of leaf spring are stiffer than the actual stiffness due the cantilever configuration in FEM. There could also be deviation from the material properties used in FEM analysis.

Parameters	Experiment	Analytical	FEM
Load, N	3250	3250	3250
Maximum stress, MPa	222	310.82	215.46
Maximum deflection, mm	94	59.20	60.65
Maximum stiffness, N/mm	34.57	54.89	53.59

Table 5. Stress analysis of composite leaf spring using experimental, analytical and FEM.

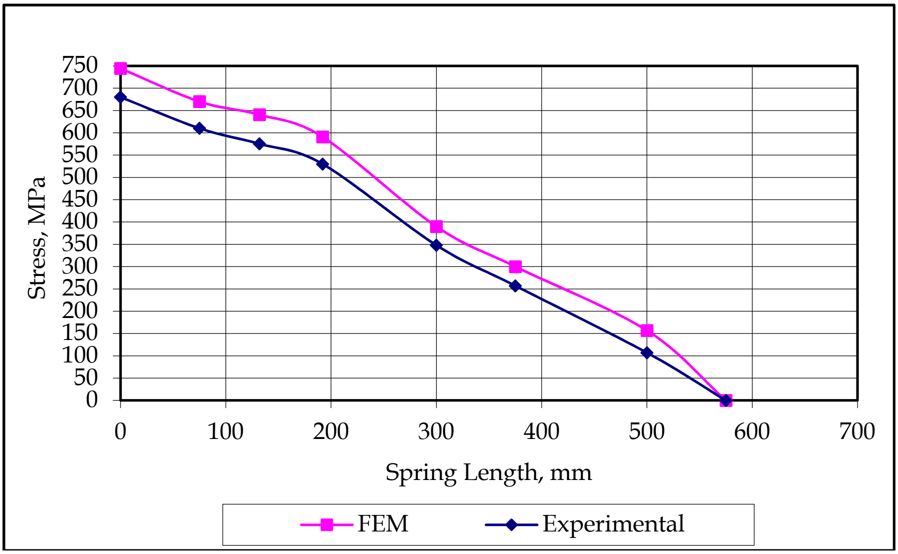


Fig. 6. Variation of longitudinal stress of steel leaf spring.

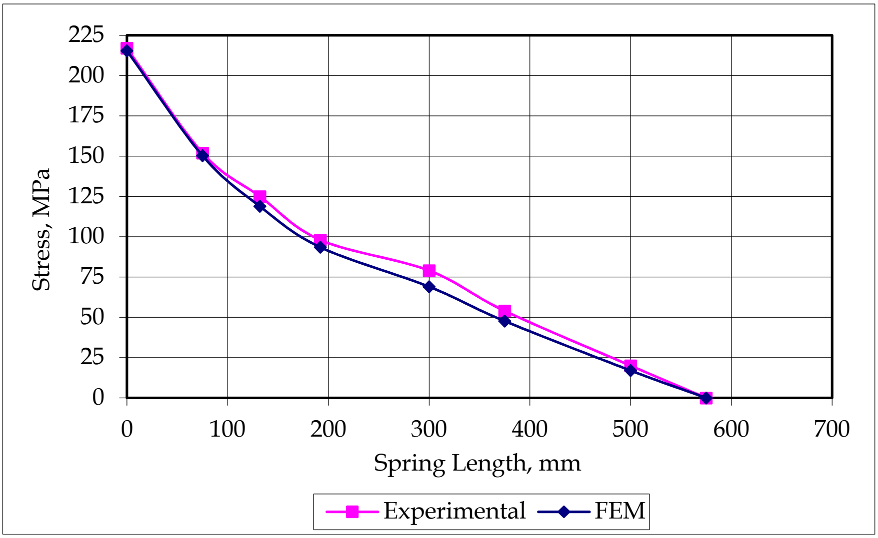


Fig. 7. Variation of longitudinal stress of composite leaf spring.

4. Fatigue analysis (Senthilkumar & Vijayarangan, 2007)

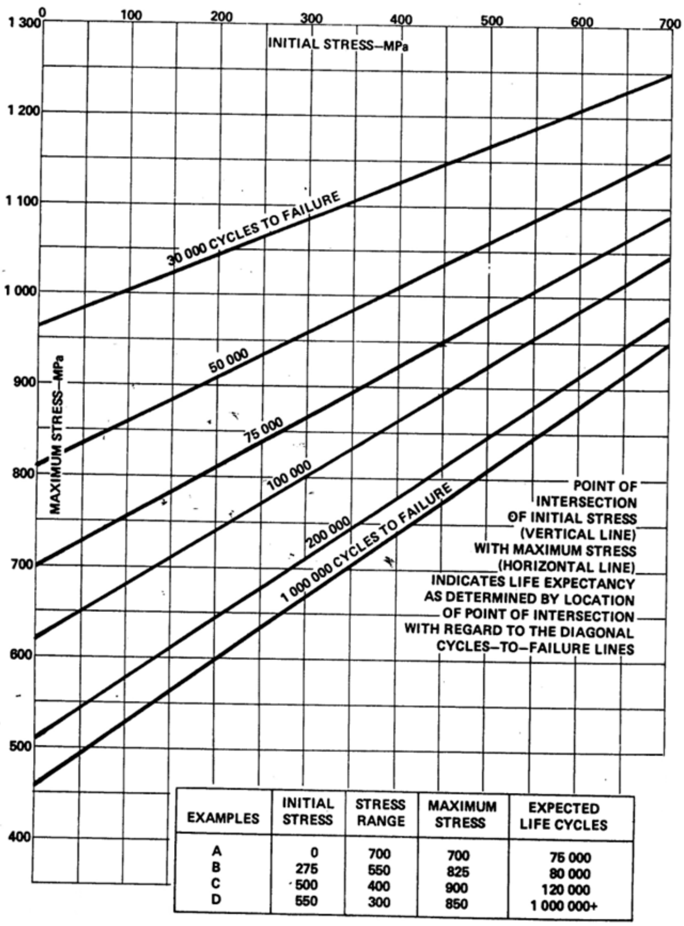
Main factors that contribute to fatigue failures include number of load cycles experienced, range of stress and mean stress experienced in each load cycle and presence of local stress concentrations. Testing of leaf springs using the regular procedure consumes a lot of time. Hence (SAE manual,1990) suggests a procedure for accelerated tests, which give quick results, particularly for SLSs. As per the outlined procedure(SAE manual,1990 & Yasushi,1997), fatigue tests are conducted on SLSs and CLSs. Fatigue life(Yasushi,1997) is expressed as the number of deflection cycles a spring will withstand without failure (Fig. 8).

4.1 Fatigue life of Steel Leaf Spring (SLS)

Fatigue life calculation of SLS is given as follows: stroke available in fatigue testing machine, 0-200 mm; initial deflection of SLS, 100 mm; initial stress (measured by experiment), 420 MPa; final deflection of SLS (camber), 175 mm; maximum stress in the final position (measured by experiment), 805 MPa. Fatigue life cycles predicted for SLS is less than 10,00,000 cycles (Fig. 8) by the procedure outlined in (SAE manual,1990).

4.2 Fatigue life of Composite Leaf Spring (CLS)

A load is applied further from the static load to maximum load with the help of the electro-hydraulic test rig, up to 3250 N, which is already obtained in static analysis. Test rig is set to operate for a deflection of 75 mm. This is the amplitude of loading cycle, which is very high. Frequency of load cycle is fixed at 33 mHz, as only 20 strokes/min is available in the test rig. This leads to high amplitude low frequency fatigue test.



EXAMPLES	INITIAL STRESS	STRESS RANGE	MAXIMUM STRESS	EXPECTED LIFE CYCLES
A	0	700	700	75 000
B	275	550	825	80 000
C	500	400	900	120 000
D	550	300	850	1 000 000+

Fig. 8. Estimation of fatigue life cycles of steel leaf springs (SAE manual,1990).

Maximum and minimum stress values obtained at the first cycle of the CLS are 222 MPa and 133 MPa respectively. As the cycles go on increasing, stress convergence is happening only after 25000 cycles. These maximum and minimum operating stress values are 240 MPa and 140 MPa respectively. Because of very low stress level, fatigue life of CLS is very high under simulated conditions.

4.2.1 Life data analysis

Life data analysis (Weibull,1961) which is a statistical approach is used to find the reliability of predictions about the life of composite leaf springs by fitting a statistical distribution to life data from representative sample units. For the GFRP leaf spring, the life data is measured in terms of the number of cycles to fail for the four leaf springs and are presented in Table 6.

Leaf spring No.	Cycles to fail	Stress level
1	10,800	0.65
2	6,950	0.65
3	19,240	0.65
4	14,350	0.65

Table 6. Number of cycles to failure for composite leaf spring.

4.2.2 Life time distribution

Weibull life distribution model is selected which has previously been used successfully for the same or a similar failure mechanism. The Weibull distribution is used to find the reliability of the life data and it helps in selecting the particular data that is to be used in life prediction model. The Weibull distribution uses two parameters, namely, 'b' and 'Θ' to estimate the reliability of the life data. 'b' is referred to as shape parameter and 'Θ' is referred to as scale parameter.

The reliability of Weibull distribution is given by,

$$R(t) = 1\text{-}\exp[-x / \theta]^b \tag{2}$$

where, X is the life; b is the Weibull slope or 'shape' parameter and Θ is the characteristic life or 'scale' parameter.

The parameters of the Weibull distribution are calculated using probability plotting (Weibull, 1961). The life cycles of leaf spring are arranged in increasing order and the median rank is calculated using the Equation (3) and are shown in Table 7.

$$Medianrank = 100 * (j\text{-}0.3)/(\mathrm{N} + 0.4) \tag{3}$$

where, j is the order number and N is the total quantity of sample.

Order No.	Cycles to fail	Median rank
1	6,950	15.9
2	10,800	38.5
3	14,350	61.4
4	19,240	84.1

Table 7. Median rank of composite leaf springs.

The value of Θ is found to be 14,600 cycles. The reliability of the life data is calculated and shown in Table 8. It is found that the reliability of 3rd GFRP spring is higher than that of other leaf springs and the fatigue life data of 3rd GFRP spring has been considered for fatigue life prediction.

The fatigue test is conducted up to 20000 cycles and it is examined that no crack initiation is visible. The details of test results at 0 and 20000 cycles are as follows: maximum load cycle range, 1850-3250 N; amplitude, 75 mm; frequency, 33 mHz; spring rate, 27.66 N/mm; maximum operating stress, 240 MPa; minimum operating stress, 140MPa and time taken 17 h. The experimental results are available only up to 20000 cycles. With no crack initiation,

Leaf spring no.	Life (cycles)	Median rank	Reliability (%)
1.	10,800	38.5	43
2.	6,950	15.9	39
3.	19,240	84.1	73
4.	14,350	61.4	62.7

Table 8. Reliability of fatigue life data.

there is a necessity to go for analytical model for finding number of cycles to failure from analytical results. An analytical fatigue model to predict the number of fatigue cycles to failure for the components made up of composite material has been developed (Hwang and Han,1986). They have proposed two constants in their relation on the basis of experimental results. It is proved that the analytical formula predicts the fatigue life of component with E-Glass/epoxy composite material.

$$\text{Hwang and Han relation: } N=\{B(1-r)\}^{1/C} \tag{4}$$

where, N is the number of cycles to failure; B= 10.33; C= 0.14012; r = σ_{max}/σ_U; σ_{max} is the maximum stress; σ_U is the ultimate tensile strength and r is the applied stress level. Equation (4) is applied for different stress levels and fatigue life is calculated for the composite leaf spring. The results are obtained based on the analytical results (Table 9) and the resulting S-N graph is shown in Fig.9. From Fig.9, it is observed that the composite leaf spring, which is made up of E-glass/epoxy is withstanding more than 10 Lakh cycles under the stress level of 0.24.

The test was conducted for nearly 17 hours to complete 20000 cycles. The variations in stress level were reduced to very low level after 20000 cycles. There was no crack initiation up to 20000 cycles. The stress level of 0.24 is obtained from experimental analysis. This is very much helpful for the determination of remaining number of cycles to failure using fatigue model (Hwang and Han,1986). According to this fatigue model, the failure of the composite leaf spring takes place only after 10 Lakh cycles. Since the composite leaf spring is expected to crack only after 10 Lakh cycles, it is required to conduct the leaf spring fatigue test up to 10 Lakh cycles for finding type and place of crack initiation and propagation. For completing full fatigue test up to crack initiation with the same frequency, nearly 830 hours of fatigue test is required.

From the design and experimental fatigue analysis of composite multi leaf spring using glass fibre reinforced polymer are carried out using life data analysis, it is found that the composite leaf spring is found to have 67.35% lesser stress, 64.95% higher stiffness and 126.98% higher natural frequency than that of existing steel leaf spring. The conventional multi leaf spring weighs about 13.5 kg whereas the E-glass/Epoxy multi leaf spring weighs only 4.3 kg. Thus the weight reduction of 68.15% is achieved. Besides the reduction of weight, the fatigue life of composite leaf spring is predicted to be higher than that of steel leaf spring. Life data analysis is found to be a tool to predict the fatigue life of composite multi leaf spring. It is found that the life of composite leaf spring is much higher than that of steel leaf spring.

Maximum stress MPa	Applied stress level	Number of cycles to failure
100	0.1	8143500
200	0.2	3515500
300	0.3	1354800
400	0.4	450900
500	0.5	122700
600	0.6	25000
700	0.7	3200
800	0.8	200
900	0.9	-
1000	1.0	-

Table 9. Fatigue life at different stress levels of composite leaf spring.

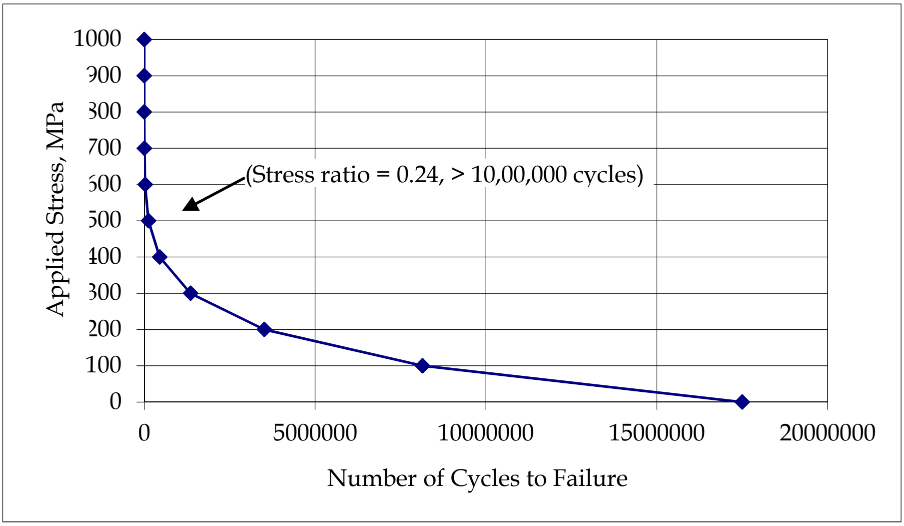


Fig. 9. S-N curve for composite leaf spring.

5. Optimisation

Optimisation of weight reduction in CLS has been carried out(Senthilkumar & Vijayarangan,2007). The finite element analysis of composite multi leaf spring with 7 leaves shows that maximum deflection of the leaf spring and the bending stress induced are well within the allowable limit (factor of safety 2.8). So, it is decided to optimize the number of leaves with minimum of 2 leaves and maximum of 7 leaves using ANSYS 7.1 itself, for the minimum weight. The number of leaves is reduced one by one, without changing any of the dimensions, and found out results for same full bump load. The results are shown in Table 10. From the FEM results in Table 10, it is understood that the leaf spring with 3 leaves violates the constraints of deflection and stress. Therefore, further optimization is terminated at this point. It is evident that only 4 leaves (2 full length leaves and 2 graduated leaves) are sufficient to withstand the applied load. The composite leaf spring with 4 leaves weighs about 3.18 kg only. It gives a weight reduction of 76.4%.

No.of Leaves	**Maximum Deflection (mm)**		**Maximum direct stress along the length of leaf spring (MPa)**	
	ANSYS	Allowable	ANSYS	Allowable
7	61	175	217	610
6	97		324	
5	104		409	
4	150		607	
3	308		1079	

Table 10. Results of optimization of number of leaves.

6. Ride comfort

To provide ride comfort to passenger, leaf spring has to be designed in such a way that its natural frequency is maintained to avoid resonant condition with respect to road frequency. The road irregularities usually have the maximum frequency of 12 Hz(Yu&Kim,1988). Therefore, leaf spring should be designed to have a natural frequency, which is away from 12 Hz to avoid the resonance. Stiffness is more and weight is lower of CLS than that of SLS. Therefore, first natural frequency of CLS (14.3 Hz) will be higher (126.98%) than that of SLS (6.3 Hz). First natural frequency of CLS is nearly 1.2 times the maximum road frequency and therefore resonance will not occur, and it provides improved ride comfort. After optimization, CLS has a fundamental natural frequency of 41.5 Hz, which is 3.46 times the maximum road frequency, which ensures that resonance will not occur.

7. Conclusion

Design and experimental analysis of composite multi leaf spring using glass fibre reinforced polymer are carried out. Compared to steel spring, the composite leaf spring is found to have 67.35% lesser stress, 64.95% higher stiffness and 126.98% higher natural frequency than that of existing steel leaf spring. The conventional multi leaf spring weighs about 13.5 kg whereas the E-glass/Epoxy multi leaf spring weighs only 4.3 kg. Thus the weight reduction of 68.15% is achieved. Besides the reduction of weight, the performance of the leaf spring is also increased. Compared to the steel leaf spring (13.5 kg), the optimised composite leaf spring weighs nearly 76.4% less than the steel spring. Ride comfort and life of CLS are also more when compared to SLS. Therefore, it is concluded that composite multi leaf spring is an effective replacement for the existing steel leaf spring in light passenger vehicles.

8. Acknowledgement

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Materials are important to mankind because of the benefits that can be derived from the manipulation of their properties, for example electrical conductivity, dielectric constant, magnetization, optical transmittance, strength and toughness. Materials science is a broad field and can be considered to be an interdisciplinary area. Included within it are the studies of the structure and properties of any material, the creation of new types of materials, and the manipulation of a material's properties to suit the needs of a specific application. The contributors of the chapters in this book have various areas of expertise. therefore this book is interdisciplinary and is written for readers with backgrounds in physical science. The book consists of fourteen chapters that have been divided into four sections. Section one includes five chapters on advanced materials and processing. Section two includes two chapters on bio-materials which deal with the preparation and modification of new types of bio-materials. Section three consists of three chapters on nanomaterials, specifically the study of carbon nanotubes, nano-machining, and nanoparticles. Section four includes four chapters on optical materials.

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