

DESIGN AND ANALYSIS OF LEAF SPRING FOR TANKER TRAILER SUSPENSION SYSTEM

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Abstract: A leaf spring is a simple form of spring, commonly used for the suspension in wheeled vehicles. Leaf Springs are long and narrow plates attached to the frame of a trailer that rest above or below the trailer's axle. For safe and comfortable riding, to prevent the road shocks from being transmitted to the vehicle components and to safeguard the occupants from road shocks it is necessary to determine the maximum safe stress and deflection. The objective is to find the stresses and deformation in the leaf spring by applying static load on it. These values are calculated analytically, by using finite element analysis and experimentation. Therefore in the present work, leaf spring is designed by considering static load on vehicle. The model of leaf spring is created in CATIA V5 and analysis is done using ANSYS 14.5 workbench. The experimental result for stress and deformation is in close agreement with FEA and analysis result.

Keywords: Suspension system, Mono suspension, di-suspension, leaf spring, design, analysis, FEM

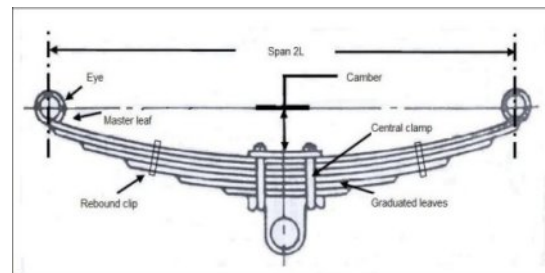
I. INTRODUCTION

Leaf springs are crucial suspension elements used on light and heavy passenger vehicle necessary to minimize the vertical vibrations impacts and bumps due to road irregularities and to create a comfortable ride. Leaf springs are widely used for automobile and rail road suspensions. It consists of series of flat plates, usually of semi- elliptical shapes. The leaves are held together by means of two U-bolts and a center clip. Rebound clips are provided to keep the leaves in alignment and prevent lateral shifting of the plate during the operation. The longest leaf called the master leaf is bent at both ends to form a spring eye. At the center, the spring is fixed at the axel of the vehicle. Multi- leaf springs are provided with one or two extra full length leaves in addition to the master leaf. This extra full-length leaves are stacked between the master leaf and the graduated length leaves. The extra full-length leaves are provided to support the transverse shear force. 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 so increasing the energy storage capabilities of a leaf spring and ensures a more compliant suspension system.

A spring is defined as an elastic body, whose function is to distort when loaded and to recover its original shape when the load is removed. Leaf springs absorb

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the vehicle vibrations, shocks and bump loads induced due to road irregularities by means of spring deflections, so that the potential energy is stored in the leaf spring and then relieved slowly. Ability to store and absorb more amount of strain energy ensures the comfortable suspension system. Semi-elliptic leaf springs are almost universally used for suspension in light and heavy commercial vehicles. For cars also, these are widely used in rear suspension. The spring consists of a number of leaves called blades. The blades are varying in length. The blades are usually given an initial curvature or cambered so that they will tend to straighten under the load. The leaf spring is based upon the theory of a beam of uniform strength. The lengthiest blade has eyes on its ends. This blade is called main or master leaf, the remaining blades are called graduated leaves. All the blades are bound together by means of steel straps. The spring is mounted on the axle of the vehicle. The entire vehicle load rests on the leaf spring. The front end of the spring is connected to the frame with a simple pin joint, while the rear end of the spring is connected with a shackle. Shackle is the flexible link which connects between leaf spring rear eye and frame. When the vehicle comes across a projection on the road surface, the wheel moves up, leading to deflection of the spring. This changes the length between the spring eyes. If both the ends are fixed, the spring will not be able to accommodate this change of length. So, to accommodate this change in length shackle is provided at one end, which gives a flexible connection. The front eye of the leaf spring is constrained in all the directions, where as rear eye is not constrained in X-direction. This rear eye is connected to the shackle.

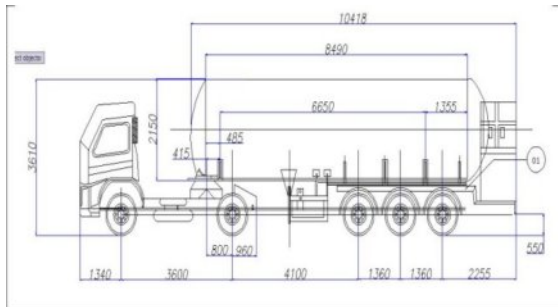


During loading the spring deflects and moves in the direction perpendicular to the load applied. When the leaf spring deflects, the upper side of each leaf tips slides or rubs against the lower side of the leaf above it. This produces some damping which reduces spring vibrations, but since this available damping may change with time, it is preferred not to avail of the same.

Moreover, it produces squeaking sound. Further if moisture is also present, such inter-leaf friction will cause fretting corrosion which decreases the fatigue strength of the spring, phosphate paint may reduce this problem fairly. The front end of the spring is connected to the frame with a simple pin joint, while the rear end of the spring is connected with a shackle as shown in figure.

A) Vehicle specification

The below figure is Tata LPS 4018 Tanker trailer. The trailer shown is heavy commercial vehicle with the capacity of 5675cc and with euro II emission norms. The trolley on the truck has a tri axle arrangement at the back.



B) Problem Statement

Leaf springs are crucial suspension element used on light passenger's vehicle necessary to minimize the vertical vibration impact and bump due to road irregularities and to create a comfortable ride. The leaf spring is analyzed for static strength and deflection using finite element analysis. The general purpose leaf spring design and finite element analysis software ANSYS is used for present study. The variations of stress and deflection values are predicted.

C) Objectives of the study

- To design leaf spring and axle of suspension system for TATA LPS 4018 tanker trailer.
- To calculate stress and deflection by applying various loads on the leaf spring.
- To model the leaf spring by using CATIA V5.
- To perform finite element analysis by using ANSYS 14.5 Workbench.
- To perform experimentation by using Universal testing machine (UTM).

D) Scope

The project begins with designing the leaf spring analytically. The use of software like CATIA V5 and ANSYS 14.5 has helped in modeling and static analysis of leaf spring by applying boundary conditions. Validation of the leaf spring model is done by using analytical, FEM and experimental methods.

E) Methodology

The methodology used in the project is as follows. The Analytical method is used to design the leaf spring then by considering the leaf spring dimensions modeling is done by using CATIA V5. This model is put in ANSYS 14.5 Workbench, by applying various boundary conditions and finite element analysis is done. Experimental work for evaluation of deflection and stress is done by using Universal testing machine

II. LITERATURE SURVEY

G.H. Goud and E. V. Goud [1] explained the Modeling and Design of leaf spring, used in automobile suspension systems. Static analysis determines the safe stress and corresponding pay load of the leaf spring and also to study the behavior of structures under practical conditions. The present work attempts to analyze the safe load of the leaf spring, which will indicate the speed at which a comfortable speed and safe drive is possible. A typical leaf spring configuration of TATA-407 light commercial vehicle is chosen for study. Finite element analysis has been carried out to determine the safe stresses and pay loads. In computer-aided design, geometric modeling is concerned with the computer compatible mathematical description of the geometry of an object. Static analysis is performed to find the Von-Mises stress by using ansys software and these results are compared with bending stresses calculated in mathematical analysis at various loads. It is concluded that for the given specifications of the leaf spring, the maximum safe load is 7700N. It is observed that the maximum stress is developed at the inner side of the eye sections, so care must be taken in eye design and fabrication and material selection. The selected material must have good ductility, resilience and toughness to avoid sudden fracture for providing safety and comfort to the occupants.

Davood Rezaei and Mahmood M. Shokrieh [2] a four-leaf steel spring used in the rear suspension system of light vehicles is analyzed using ANSYS V5.4 software. The finite element results showing stresses and deflections verified the existing analytical and experimental solutions. Using the results of the steel leaf spring, a composite one made from fibre glass with epoxy resin is designed and optimized using ANSYS. Main consideration is given to the optimization of the spring geometry. The objective was to obtain a spring with minimum weight that is capable of carrying given static external forces without failure. It is concluded that that the optimum spring width decreases hyperbolically and the thickness increases linearly from spring eye towards the axle seat. Compared to the steel leaf spring (9.2 kg) the optimized composite leaf spring without eye unit weights nearly 80% less than the steel spring.

III. ANALYSIS OF SUSPENSION SYSTEM

A) Materials for Leaf Spring

The basic requirements of a leaf spring steel is that the selected grade of steel must have sufficient harden ability for the size involved to ensure a full martensitic structure throughout the entire leaf section. In general

Material	Condition	Ultimate tensile strength (MPa)	Tensile yield strength (MP a)	% Minimum elongation $l=5.65 \sqrt{(area\ of\ Cross\ section)}$	Brinell hardness number
50Cr1	Hardened & Tempered	1680-2200	1540-1750	-	461-601
50 Cr 1v23	Hardened & Tempered	1900-2200	1680-1890	4	534-601
55 Si2 Mn90	Hardened & Tempered	1820-2060	1680-1920	6	534-601

terms higher alloy content is mandatory to ensure adequate harden ability when the thick leaf sections are used. The material used for the experimental work is 55Si2Mn90. The other designation of this material is shown in Table

The material used for leaf spring is usually a plain carbon steel having 0.90 to 1.0% carbon. For details refer Indian Standards [IS: 3431-1982 (Reaffirmed 1992)] on “Specification for steel for the manufacture of volute, helical and laminated springs for automotive suspension.”

International standard	Equivalent Grade			
	IS	DIN	BS	AISI
EN45	55Si2Mn90	55Si7	250A53	9255

Grade	C%	Si %	Mn %	Cr %	Mo %	P%	S%
55si2MN 90	0.55	1.7 4	0.87	0.1	0.0 2	0.05	0.05

Many industries manufacture steel leaf springs by 55si2Mn90 material. These materials are widely used for production of parabolic leaf springs and conventional multi leaf spring. Leaf spring absorbs the vertical vibration, shock and bump loads by means of spring deflection, so that the potential energy stored in the leaf spring and then relived slowly. Ability to store and absorb more amount of strain energy insures the comfortable suspension system. Following are the design parameter of the leaf spring. The Young modulus is considered as 200Gpa, poisons ratio is taken as 0.3with ultimate tensile strength is 1820-2060 Mpa and tensile yield strength is 1680-1920 Mpa. Brinell hardness number is 534-601. Density of the selected material is 7850kg/m^3 and thermal expansion of the material is $11 \times 10^{-6} / ^\circ\text{C}$.

B) Specification of TATA LPS 4018

1	Manufacturer	Tata Motors
2	Class of vehicle	Tractor Tanker Trailer
3	Type	Tanker Trailer, Tri-axle
4	Prime Mover Used	TATA LPS4018 Tractor
5	Unladen weight of the trailer	59939.1 N
6	Payload	235440 N
7	Gross vehicle weight of the trailer	295379.1 N
8	Under carriage & 3 axle	38259 N
9	Overall length of trailer	10.418 m
10	Overall width of trailer	2.495 m
11	Overall Height of trailer	1.270 m
12	Overall length of tanker trailer combination	14.015
13	Cross bearers	C Section 125x65 mm
14	Wheel base	6.260 m
15	Axle	3 no. 150mmx150mm Round beam of axle, 16 mm wall thickness
16	Leaf spring	75x13mm flat:940mm eye center-8
17	Tyres	10.00x20, 16R, 12 nos.

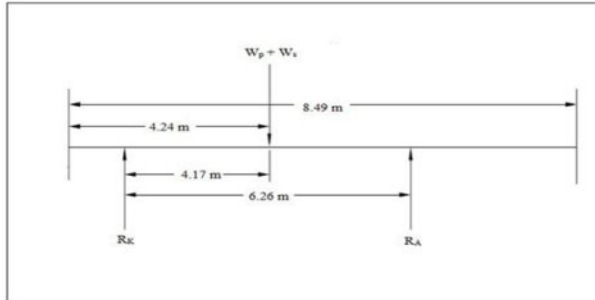
C) Analytical analysis of chassis frame

1) Load distribution on kingpin & axle:

The specification data was obtained from TATA Motors.

- Payload (W_p) = 235440 N
- Gross vehicle weight of the trailer (under load) (W_s) = 295379.1 N
- Self weight of tanker (W_s) = 59939.1N
= (Design input)

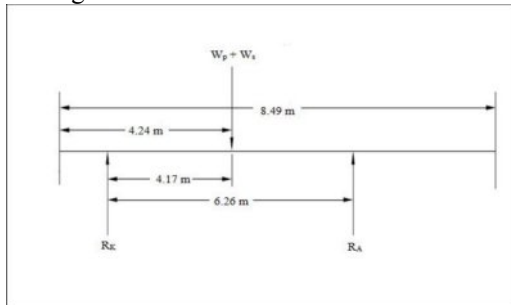
The payload and the self weight are assumed to be acting at the centre of the tanker loading area. The loading condition of the tanker is as shown below



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Taking moment around Reaction on Axle (R_A) Reaction on King Pin

$$(R_k) R_A \times 6.260 = (W_p \times 4.175) + (W_s \times 4.175)$$

$$= (235440 \times 4.175) + 59939.1 \times 4.175$$

$$R_A = 196998.04 = 196998 \text{ N}$$

$$R_k = W_p + W_s - R_A$$

$$= 235440 + 59939.1 - 196998$$

$$= 98381.1 \text{ N}$$

$$= \text{Load per axle} = (R_A + W_a) / 3$$

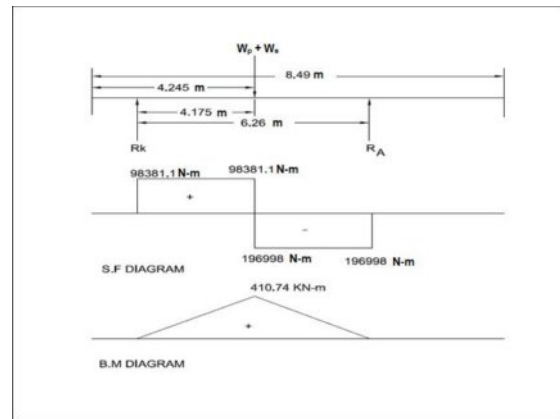
$$= (196998 + 29430) / 3 = 75476 \text{ N}$$

Load per axle = 75476 N

Kingpin imposed load = 98381.1 N

Axle capacity is 117720 N The imposed load is 75476 N. Therefore the axle weight is within the permissible capacity limit. Selected axle is 12T

2) Load distribution on tanker trailer axle:

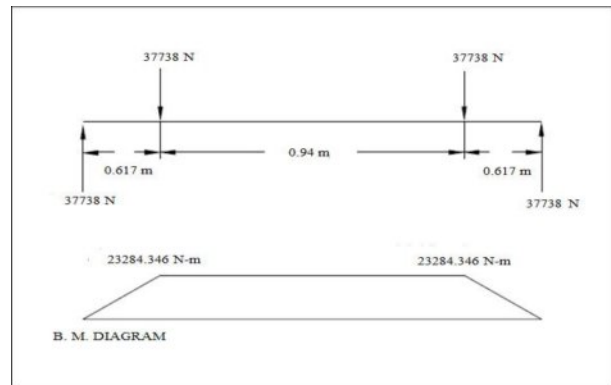


Calculation of Bending Movement

- B.M at left point = 0
- B.M at king pin = 0
- B.M where the lode is act = $98381.1 \times 4.175 = 410.74 \text{ KN-m}$
- B.M at Axle = 0
- B.M at right point = 0

3) Axle calculation:

- Total Load on axle (P_a) = 196998 N
- Axle weight (W_a) = 29430
- Load on tri-axle ($3P_a$) = $P_a + W_a$
= $196998 + 29430 = 226428$ Load on each axle = $226428/3 = 75476 \text{ N}$



- Max. Bending Moment (M_b) = $37738 \times 0.617 = 23284.346 \text{ Nm}$
- Yield stress of axle material (σ_{ye}) = 470MPa
F.O.S = 2.5
- Safe working stress (S) = $470 / 2.5 = 188 \text{ N/mm}^2 = 188 \times 10^6 \text{ N/m}^2$

Therefore Section modulus required

$$(Z) = (23284.346) \div (188 \times 10^6) = 1.238529 \times 10^{-04} \text{ m}^3$$

The axle beam is 150mm round beam with wall thickness of 16 mm. Section modulus of round axle beam

$$(Z) = 282447.36 \text{ mm}^3 = 2.8244736 \times 10^{-04} \text{ m}^3$$

$$\begin{aligned} \text{Stress induced (S)} &= (23284.346) \div (2.8244736 \times 10^{-04}) \\ &= 82.43 \times 10^06 \text{ N/m}^2 \end{aligned}$$

Therefore 150 mm x 150 mm round axle beam induced stress $82.43 \times 10^06 \text{ N/m}^2$ is less than allowable stress $186.39 \times 10^06 \text{ N/m}^2$, so it is safe for operation.

d) Leaf Spring calculation:

Load on tri-axle $(3P_a) = 226428 \text{ N}$

Axle self weight $(W_a) = 29430 \text{ N}$

Load on leaf spring $(P) = 226428 - 29430$
 $= 196998 \text{ N}$

Load per spring $= 196998 / 6 = 32833 \text{ N}$

The yield stress of this spring material is $(\sigma_{ye}) = 1750 \text{ N/mm}^2$
 $= 1750 \times 10^06 \text{ N/m}^2$

Factor of safety = 2

Safe working stress $(S) = 1750 \times 10^06 / 2$
 $= 875 \times 10^06 \text{ N/m}^2$

Leaf spring dimensions:-

Eye to eye centre $(L) = 0.940 \text{ m}$

Width of spring $(b) = 0.075 \text{ m}$

Thickness $(t) = 0.013 \text{ m}$

Working stress $(S_{ws}) = 600 \times 10^06 \text{ N/m}^2$

Load $(P) = (2 \times n \times S \times b \times t^2) / (3 \times L)$

Therefore, no. of spring $(n) = (32833 \times 3 \times .940) / (2 \times 600 \times 10^06 \times .075 \times 0.013 \times 0.013)$
 $= 6.08$

Therefore, minimum no. of leaves required = 7

No. of spring available $(n) = 8$

Stress induced $(S) = (3 \times 32833 \times .940) / (2 \times 8 \times 0.075 \times 0.013 \times 0.013)$
 $= 456.5 \times 10^06 \text{ N/m}^2$

Therefore, the leaf spring bundle of eight springs is safe for operation.

e) Formulae:

1) Total no. of leaves (n)

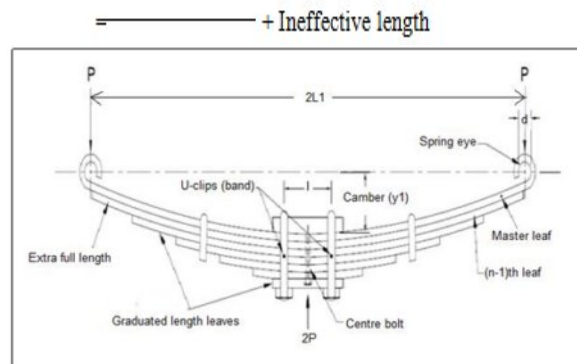
$$n = n_e + n_g$$

2) Effective length:

$$2L = 2L_1 - 1 \quad \dots \text{ (when band is used)}$$

$$2L = 2L_1 - 1 \quad \dots \text{ (when U-bolts are used)}$$

3) Length of smallest leaf :



4) Length of (n-1)th leaf

$$= \dots \times (n-1) + \text{Ineffective length}$$

5) Length of master leaf

$$= 2L_1 + \pi (d + t) * 2$$

6) Stress in leaf spring

$$\sigma_b = \frac{\sigma}{\dots}$$

7) Deflection in leaf spring

$$y = \frac{\sigma}{\dots}$$

8) Bending Stress in extra full length leaves

$$\sigma_{be} = \dots$$

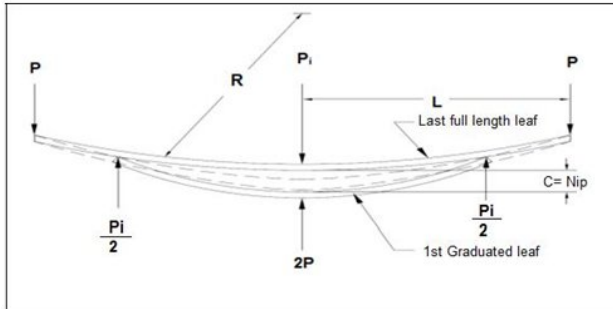
9) Bending Stress in graduated leaves:

$$\sigma_{bg} = \dots$$

$$10) \sigma_{be} = 3/2 \sigma_{bg}$$

11) Deflection in full length & graduated leaves

$$y = \dots$$



**Design of leaf Spring:
Design Input:**

- Axle load carrying capacity = 12 T = 12000 kg
- Load on spring (2P) = 6000 kg
- Load on spring at one side (P) = 3000 kg
- Length of Span (2L₁) = 876 mm
- Length of half Span (L₁) = 438 mm
- Ineffective length (l) = 104 mm
- Length of hanger (l₁) = 150 mm
- Width of the leaves (b) = 75
- Thickness of leaves (t) = 13
- Number of leaves in the spring (n_e + n_g = n) = 8
- Number of graduated leaves including master leaf (n_g) = 6
- Number of extra full length leaves (n_e) = 2
- Modulus of elasticity (E) = 2 x 10⁴ kg/mm²

Analytical results by applying various loads

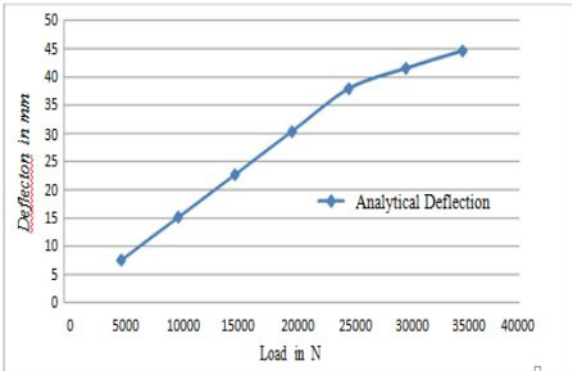
The below shown results are analytically calculated by considering 12 T load on axle and the load of 3 T is distributed towards the spring. The 3 T load has given the deflection 44.68mm by hand calculation and stress of 710.70x10⁶N/m². The value of stress obtained in analytical result is less than the safe working stress. The graphs formed below are plotted in the excel sheet.

Sr.No.	Load(N)	Deflection(mm)	Stress(N/m ²)
1	5000	7.59	108.22 x10 ⁰⁶
2	10000	15.18	216.45 x10 ⁰⁶
3	15000	22.77	324.68 x10 ⁰⁶
4	20000	30.36	432.91 x10 ⁰⁶
5	25000	37.95	541.14 x10 ⁰⁶
6	30000	41.55	649.37 x10 ⁰⁶
7	32833	44.68	710.70 x10 ⁰⁶

f) Leaf spring Result:
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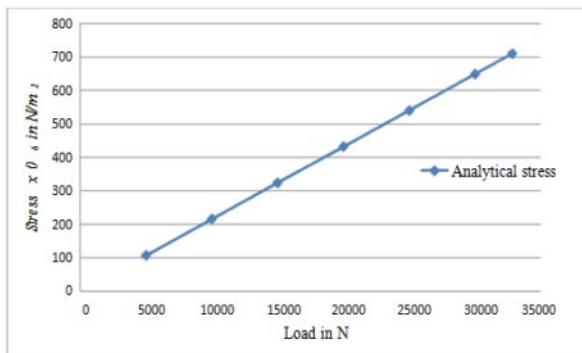
Sr. No.	Parameters	(8x75x13), 876mm 12 ton axle capacity
1	Axle load carrying capacity, kg	12000 kg
2	Load on spring(2P),kg	6000 kg
3	Load on spring at one side (P),kg	3000 kg
4	Length of span(2L ₁),mm	876 mm
5	Ineffective length(l), mm	104 mm
6	Total no. of leaves(n)	8
7	No. of extra full length leaves(n)	2
8	No. of graduated leaves (n) including master leaf	6
9	Width of leaves(b), mm	75 mm
10	Thickness of leaves (t),mm	13 mm
11	Effective length (2L),mm	806.67mm
12	Bending stress in leaf spring(σ _b), kg/mm ²	71.53 kg/mm ²
13	Deflection in leaf spring (y), mm	44.68 mm
14	Initial gap (C), mm	14.89 mm
15	Initial load (P _i), kg	500 kg
16	Radius of curvature (R),mm	2434.8 mm
17	Material (55Si2Mn90):	
18	a. Ultimate tensile strength, kg/mm ²	135.52-209.98 kg/mm ²
	b. Tensile yield strength, kg/mm ²	171.25-195.71kg/mm ²
19	Factor of safety on yield	2.73
20	Deflection limit (2L ₁ /y)	19.6

Sr.No.	Load(N)	Deflection(mm)	Stress(N/m ²)
1	5000	7.59	108.22 x10 ⁰⁶
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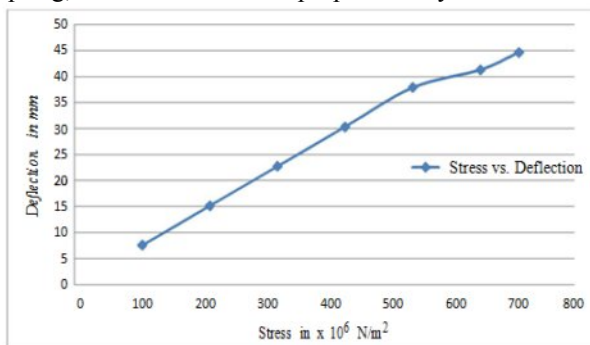


Graph 1: load vs. deflection graph. By applying various loads the deflection

Varies at different points. At 32833N the deflection is 44.68mm



Graph 2: when the value of load increases on the spring, stress also increases proportionally with the load



Graph 3: The stress increases the deflection increases till the point the deflection reaches 30-40 mm and then it increases gradually.

IV. NUMERICAL ANALYSIS OF LEAF SPRING

A) Finite Element Analysis

The Basic concept in FEA is that the body or structure may be divided into smaller elements of finite dimensions called “Finite Elements”. The original body or the structure is then considered as an assemblage of these elements connected at a finite number of joints called “Nodes” or “Nodal Points”. Simple functions are chosen to approximate the displacements over each finite element. Such assumed functions are called “shape functions”. This will represent the displacement within the element in terms of the displacement at the nodes of the element. Mathematically, the structure to be analyzed is subdivided into a mesh of finite sized elements of simple shape. Within each element, the variation of displacement is assumed to be determined by simple polynomial shape functions and nodal displacements. Equations for the strains and stresses are developed in terms of the unknown nodal displacements. From this, the equations of equilibrium are assembled in a matrix form which can be easily be programmed and solved in software. After applying the appropriate boundary conditions, the nodal displacements are found by solving the matrix stiffness equation. Once the nodal displacements are known, element stresses and strains can be calculated.

In finite element analysis 3D model of leaf spring is developed in CATIA v5. After modeling of leaf spring then give the actual supporting boundary condition in ANSYS 14.5 i.e. fixed support in front side, displacement in the rear side and in center 32833N force is applied. In fixed support there is no any degree of freedom i.e. zero displacement at any direction. In rear side only z-axis displacement is allowed. This condition is real in TATA LPS-4018 Leaf spring. After giving the boundary condition meshing of leaf spring is done by obtaining the nodes 46588 and element 7864. The results of the solution are available at the node of the elements. Finite element analysis can display them in graphical form to analyse them, to make design decisions and recommendations. Conventional analytical method for solving stress and strain becomes very complex. In such cases finite element modeling becomes very convenient.

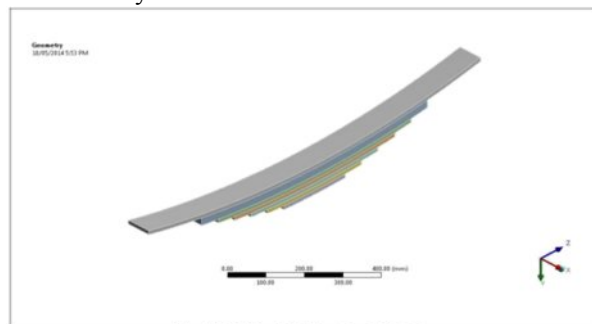


Figure 1: Model of the leaf spring in CATIA

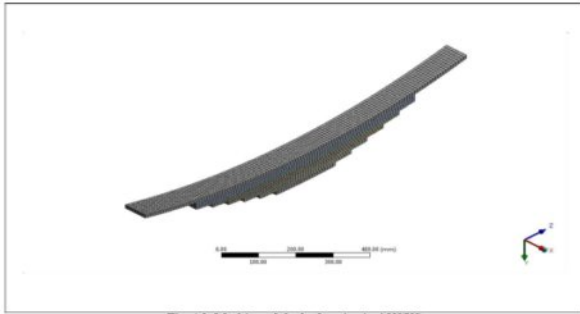


Figure 2: Meshing of the leaf spring in ANSYS
After giving all boundary condition and load i.e. 32833N which is applied to the leaf spring and calculated by considering 12 Ton capacity load vehicle. Also applying all material properties of 55Si2Mn90 i.e. Young's Modulus (E), Poisson's Ratio, Tensile strength ultimate, tensile strength yield, Density, thermal expansion. In still leaf spring total deformation is found 46.104mm and equivalent (von-mises) stress 1361.3mpa. The von- mises stress and total deformation is shown

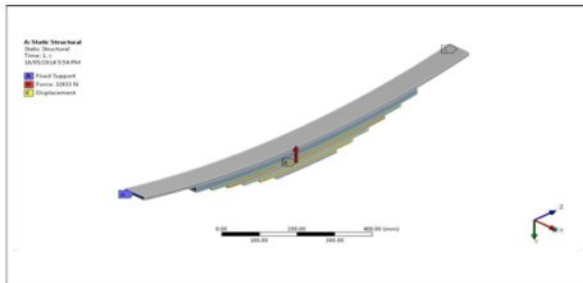


Figure 3: Boundary conditions applied on leaf spring model

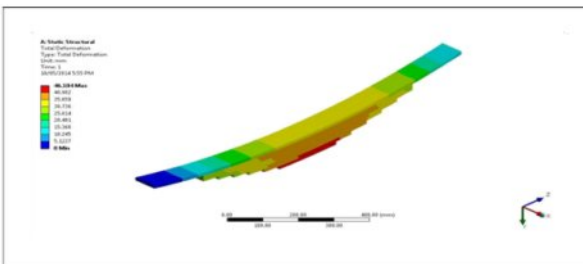


Figure 4: Total deformation of the leaf spring

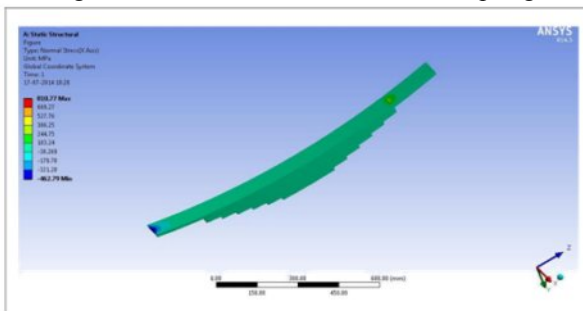


Figure 5: Normal stress of the leaf spring

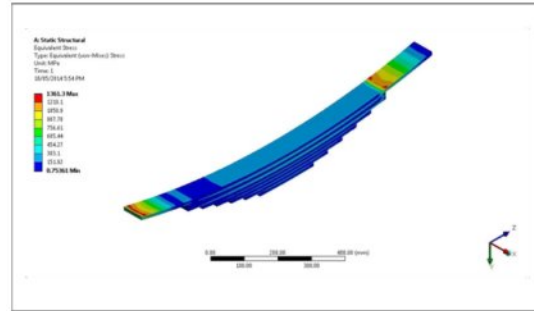
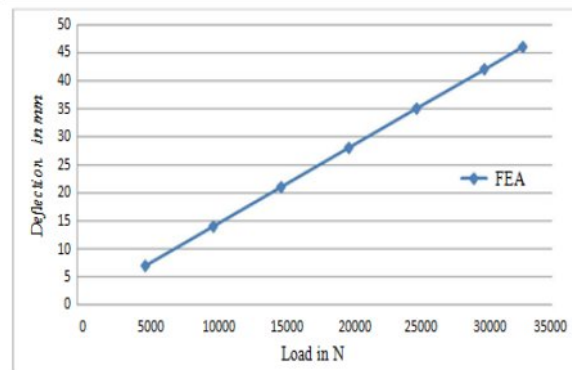


Figure 6: Equivalent stress of the leaf spring

B) FEA results by applying various loads:

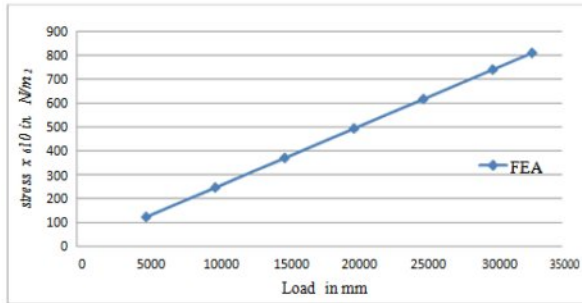
The show results are calculated using finite element method by Ansys14.5. By considering 12 T load in axle and the load of 3tons is distributed towards the spring.

Sr. No.	Load(N)	Deflection(mm)	Stress(N/m ²)
1	5000	7.02	123.46 x10 ⁰⁶
2	10000	14.04	246.93x10 ⁰⁶
3	15000	21.06	370.40 x10 ⁰⁶
4	20000	28.083	493.87 x10 ⁰⁶
5	25000	35.104	617.34x10 ⁰⁶
6	30000	42.125	740.81x10 ⁰⁶
7	32833	46.104	810.77 x10 ⁰⁶



Graph 4: Variation in output deflection with load for leaf spring

The shows that the result obtained by hand calculation and FEA shows the same result. According to FEA the deflection is 46.10mm.



Graph 5: Variation in output stress with load for leaf spring

The above graph shows that the result obtained by hand calculation and FEA shows the same result. The graph is made considering allowable value of safe working stress.

V. EXPERIMENTAL VALIDATION OF DESIGN PARAMETERS FOR LEAF SPRING

In the experimental analysis of the steel leaf spring are taken. The deflection or the bending stress of the spring is taken on the universal testing machine. A universal testing machine (UTM), also known as a universal tester, materials testing machine or materials test frame, is used to calculate various stresses, strength and deflection of the material. It is named after the fact that it can perform many standard tensile and compression tests on materials, components, and structures.

The set-up and usage are detailed in a test method, often published by a standards organization. This specifies the sample preparation, fixturing gauge length. The specimen is placed in the machine between the grips and an extensometer if required can automatically record the change in gauge length during the test. If an extensometer is not fitted, the machine itself can record the displacement between its cross heads on which the specimen is held. However, this method not only records the change in length of the specimen but also all other extending elastic components of the testing machine and its drive systems including any slipping of the specimen in the grips. Once the machine is started it begins to apply an increasing load on specimen. Throughout the tests the control system and its associated software record the load and extension or compression of the specimen.



Figure 7: Experimental Setup of Leaf spring on the UTM

Experimental Procedure

The first step is to purchase the leaf spring from the manufacturer as per dimensions. Then get it coupled as per the actual situation used in vehicle while assembling. After purchasing the leaf spring, it is put on the Universal testing machine where the rear end of the spring is fixed, the front end is kept free and the load of 32.83kN is applied at the centre of the leaf spring. The way leaf spring is used or fitted in the actual vehicle, the same boundary conditions are applied while experimentation. Then the results are being noted. The deflection is calculated using digital vernier caliper and the stress is also shown by the UTM machine. The figure shown below is the leaf spring on the universal testing machine used in the experimentation. The deflection value using vernier caliper



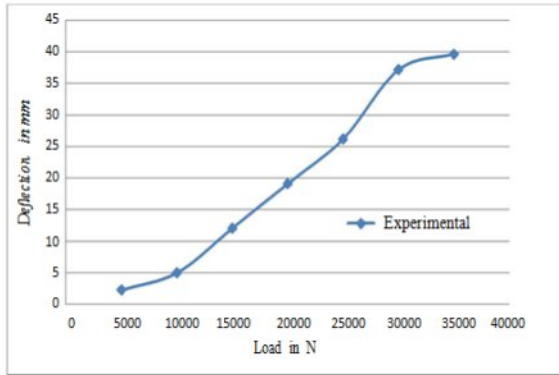
Figure 8: Deflection in the leaf spring

Experimental Result

The table contains the deflection and stress calculated by applying various load on the spring by using universal testing machine.

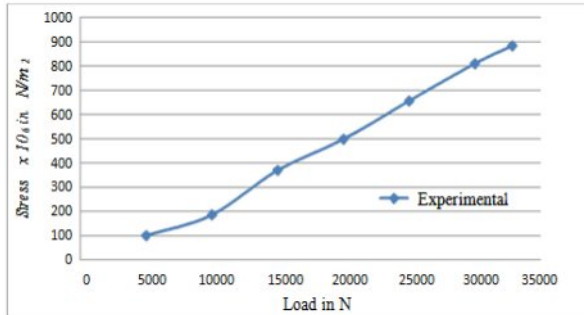
Sr. No	Load(N)	Deflection(mm)	Stress(N/m ²)
1	5000	2.3	100.64x10 ⁰⁶
2	10000	5	186.29x10 ⁰⁶
3	15000	12.07	369.94x10 ⁰⁶
4	20000	19.15	498.59x10 ⁰⁶
5	25000	26.190	657.24x10 ⁰⁶
6	30000	37.22	810.88x10 ⁰⁶
7	32833	39.65	884.18x10 ⁰⁶

Experimental output Results for Deflection and Stress



Graph 6: Load vs. Deflection output

The fig. showed the experimental results. After applying the load on the leaf spring the leaf spring gives the deflection of 39.65mm. The reading of the digital vernier calliper is shown in the fig. The deflection shown at the intermediate stage of applying load is less and increases at the certain load of 10-15kN.



Graph 7: Load vs. Stress output

The stress found in the graph is calculated by using the experimental result. The maximum stress is obtained $884.18 \times 10^6 \text{ N/m}^2$ which are comparatively similar to the safe working stress of the material which is 875mpa.

Validation of Result

Sr.No.	Result	Analytical	FEA	Experimental
1	Deflection in mm	44.68	46.104	39.65
2	Stress in N/m^2	710.70×10^6	810.77×10^6	884.18×10^6

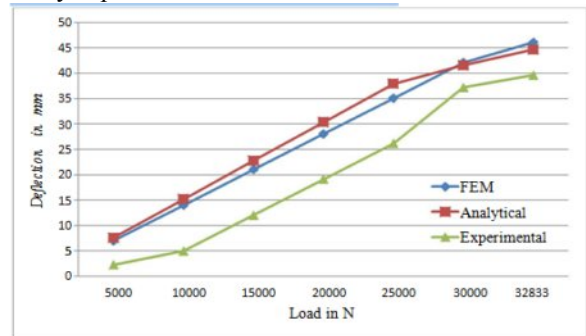
Deflection and stress result

VI. RESULT AND DISCUSSION

Though the detailed results are presented in earlier, here an attempt is made to show compare the result obtained analytically, FEA and experimentation. For comparison of stress safe working stress and ultimate stress is considered.

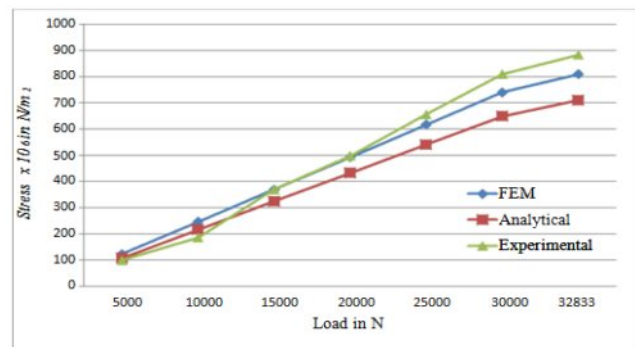
The deflections are evaluated at various loads varying from 5000N to 32833N. The results are given in the tables and it is observed that the experimental deflection as compared to analytical and FEA deflection is less and maximum deflection is shown by finite element analysis. The results shown by experimentation are near to the values of the result

shown by analytical and FEA. The deflection by simulation is 46.10 mm, by analytical it is 44.68 mm and by experimental 39.65mm.



Graph 8: Comparison of FEM, Analytical and Experimental Deflection

The fig shows the stress value which is calculated by formula and analytical methods are within permissible limit. The value obtained of von mises stress by using FEA is 1361.3mpa which is less than the ultimate tensile stress of the material-1750mpa and the stress obtained by experimental method is less than the value of safe working stress as required analytical permissible limit 875 Mpa.



Graph 9: Comparison of FEM, Analytical and Experimental Stress

The graph schemes comparing the results by analytical, FEA and experimental. By analytical calculation the safe value obtain was $710.70 \times 10^6 \text{ N/m}^2$. The safe stress value for material is $875 \times 10^6 \text{ N/m}^2$. The FEA result for deflection is 46.104 mm is in close agreement with analytical value 44.68 mm. The comparison in the graphs is shown in fig

VII. CONCLUSION

Leaf spring is a device which is used in suspension system to safeguard the vehicle and the occupants. To prevent the road shock from being transmitting to vehicle component and to safeguard the occupants for comfortable riding, it is necessary to determine the maximum safe stress and deflection. Therefore in the present work, leaf spring is designed by using 12T load for TATA LPS 4018. Model of leaf spring is prepared using CATIA V5 and analysis is done using ANSYS 14.5 workbench and it is concluded that for the given

specifications of the leaf spring, the stress is found below its safe working stress of material. Also, von mises stress of 1361.3 Mpa is below its ultimate tensile stress. The deflection calculated by using analytical, FEA and experimental methods was found nearly same.

VIII. FUTURE SCOPE

The leaf spring made of the 55Mn2Si90 material composition gives the value of stress and deflection. An extended study of this nature by replacing the steel leaf spring by composite leaf spring reduces the stress value and comfortable ride can be achieved. This variation can be considered from strength, stiffness and vehicle stability point of view. An efficient design and manufacturing process of composite material leaf spring can reduce the cost and weight of the vehicle.

Verify the analysis results with different analysis software's like solid works etc

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