Review Article

GENERAL REVIEW OF MECHANICAL SPRINGS USED IN AUTOMOBILES SUSPENSION SYSTEM

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ABSTRACT

The main functions of automobile suspension systems are to isolate the structure and the occupants from shocks and vibrations generated by the road surface. The suspension systems basically consist of all the elements that provide the connection between the tyres and the vehicle body. A spring is an elastic object used to store mechanical energy. It is an elastic body that can be twisted, pulled, or stretched by some force. It can return to their original shape when the force is released. It is a flexible element used to exert a force or a torque and, at the same time, to store energy. The force can be a linear push or pull, or it can be radial, acting similarly to a rubber band around a roll of drawings. The torque can be used to cause a rotation. The main objective of this research paper is to through some light on the fatigue stress analysis of springs used in automobiles. Theoretical, Numerical and Experimental methods are used for the analysis of springs but Finite Element Method is the best for its analysis and calculating the fatigue stress, life cycle and shear stress springs.

KEY WORDS: Suspension system**,** helical compression spring, leaf spring, Maximum shear stress, Deflection, fatigue stress, Finite element method.

1. INTRODUCTION:

Suspension systems have been widely applied to vehicles, from the horse-drawn carriage with flexible leaf springs fixed in the four corners, to the modern automobile with complex control algorithms. The suspension of a road vehicle is usually designed with two objectives; to isolate the vehicle body from road irregularities and to maintain contact of the wheels with the roadway. Isolation is achieved by the use of springs and dampers and by rubber mountings at the connections of the individual suspension components. From a system design point of view, there are two main categories of disturbances on a vehicle, namely road and load disturbances. Road disturbances have the characteristics of large magnitude in low frequency such as hills and small magnitude in high frequency such as road roughness. Load disturbances include the variation of loads induced by accelerating, braking and cornering. Therefore, a good suspension design is concerned with disturbance rejection from these disturbances to the outputs. Roughly speaking, a conventional suspension needs to be "soft" to insulate against road disturbances and "hard" to insulate against load disturbances. Therefore, suspension design is an art of compromise between these two goals (Wang 2001).Today, nearly all passenger cars and light trucks use independent front suspensions, because of the better resistance to vibrations [1]. The main functions of a vehicle's suspension systems are to isolate the structure and the occupants from shocks and vibrations generated by the road surface. The suspension systems basically consist of all the elements that provide the connection between the tires and the vehicle body. The suspension system requires an elastic resistance to absorb the road shocks and this job is fulfilled by the suspension springs. According to Gillepsie (1992), the primary functions for suspension systems are;

- Provide vertical compliance so the wheels can follow the uneven road, isolating the chassis from roughness in the road.
- Maintain the wheels in the proper steer and camber attitudes to the road surface.
- React to the control forces produced by the tires- longitudinal (acceleration and braking) forces, lateral (cornering) forces, and braking and driving torques.
- Resist roll of the chassis.
- Keep the tires in contact with the road with minimal load variations

To accomplish all functions, the suspension system requires an elastic resistance to absorb the road shocks and this job is fulfilled by the suspension springs [2]. A spring is defined as an elastic machine element, which deflects under the action of the load & returns to its original shape when the load is removed [3]. Mechanical springs are used in machine designs to exert force, provide flexibility, and to store or absorb energy. Springs are manufactured for many different applications such as compression, extension, torsion, power, and constant force. Depending on the application, a spring may be in a static, cyclic or dynamic operating mode. A spring is usually considered to be static if a change in deflection or load occurs only a few times, such as less than 10,000 cycles during the expected life of the spring. A static spring may remain loaded for very long periods of time. The failure modes of interest for static springs include spring relaxation, set and creep. The main objectives of spring are as follows:

- **To apply force:** A majority industrial, e.g. To provide the operating force in brakes and clutches, to provide a clamping force, to provide a return load, to keep rotational mechanisms in contact, make electrical contacts, counterbalance loading, etc.
- **To control motion:** Typically storing energy, e.g. wind-up springs for motor, constant torque applications, torsion control, position control, etc.
- **To control vibration**: used in essence for noise and vibration control, e.g. flexible couplings, isolation mounts, spring and dampers, etc.
- **To reduce impact:** Used to reduce the magnitude of the transmitted force due to impact or shock loading, e.g. buffers, end stops, bump stops etc.

In practical situations, springs are used to provide more than one of the above functions at the same time. Because of superior strength and endurance characteristics under load, most springs are metallic.

However, other resilient materials, e.g. polymers, where special properties such as a low modulus and high internal damping capacity are required

2. CLASSIFICATION OF SPRING:

2.1. Leaf spring: Fig.1 shows the plate spring or leaf spring in which the major stresses are tensile and compressive. Leaf springs may be of cantilever type or semi elliptical or elliptical. A leaf spring consists of flat leaves or plates of varying lengths clamped together so as to obtain greater efficiency and resilience.

Fig.1 Leaf spring

2.2. Cylindrical helical spring: Fig.2 shows cylindrical helical spring which may be in compression or tension. The major stresses produced in this are shear due to twisting. The load applied is parallel to the axis of spring. The cross-section of the wire may be round, square or rectangular. These springs are wound in the form of a helix of a wire.

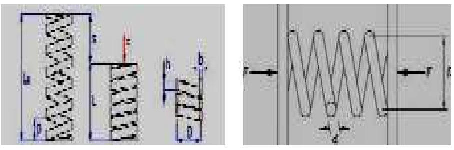
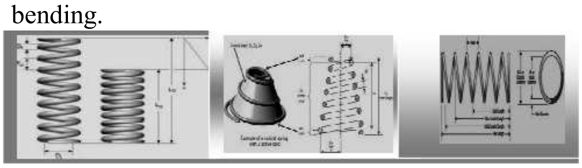


Fig.2. (a) **Fig.2. (b)**

Fig.2. (a) & (b) Cylindrical conical spring

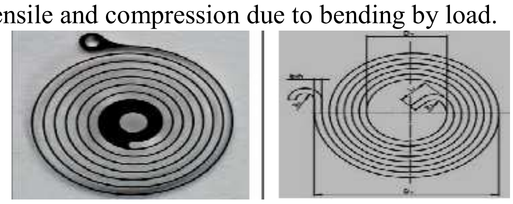
2.3. Helical conical spring: The fig.3 is shown a helical conical spring. The major stresses produced in this are also shear due to twisting. If the radius of the coils of a helical spring is constant, then it becomes a cylindrical helical. If a helical spring works in torsion, i.e. the torsional moment is applied about the axis of the helix, then the spring obtained is helical torsion spring. Major stresses produced in this spring are tensile and compression due to bending.



(a) **(b)** **(c)**

Fig.3 (a), (b) & (c). Helical conical spring

2.4. Spiral spring: If the angle of helix is zero as shown in fig.4, then it is a spiral spring, consists of a flat strip wound in the form of a spiral and loaded in torsion. The major stresses produced in this are tensile and compression due to bending by load.



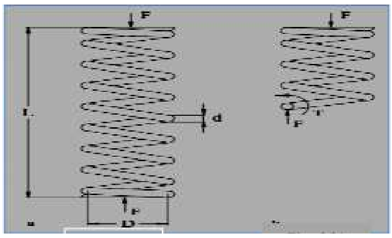
(a) **(b)**

Fig.4 (a) & (b) Spiral spring [4]

3. STRESSES IN HELICAL SPRINGS OF CIRCULAR WIRE:

To determine the stress generated in the spring; consider a helical spring subjected to an axial load *F*. Let

D= Mean Diameter of the spring coil,
d = Diameter of the spring wire,
n = No. of active coils,
G = Modulus of rigidity for the spring material,
F = Axial load on the spring,
τ = Max. shear stress induced in the wire,
C = spring index = D/d
P= Pitch of the coils, and
δ = Deflection of the spring, as a result of an axial load F.



(a) **(b)**

Fig. 5(a) Helical spring with axial load and (b) free body diagram.

If we remove a portion of the spring, the internal reactions will be a direct shear and a torque T = F× D/2 where each will cause a shear stress, and the maximum shear will occur at the inner surface of the wire which is equal to:

$\tau_{max} = Tr/J + F/A$

Substituting T= F× D/2, r= d/2, $J = \pi/32\ d^4$, $A= \pi/4\ d^2$; gives

$\tau = \frac{8FD}{\pi\, d^3} + \frac{4F}{\pi\, d^2}$

Defining the spring index which is a measure of coil curvature as:

C = spring index = D/d, for most springs C ranges from 6 to 12

We get:

$\tau = \frac{2C+1}{2C}\left(\frac{8FD}{\pi\, d^3}\right) = K_s \frac{8FD}{\pi\, d^3}$

$K_s = \frac{2C+1}{2C}$;

where K_s is called the “Shear stress correction factor”

This equation assumes the spring wire to be straight and subjected to torsion and direct shear. However, the wire is curved and the curvature increases the shear stress and this is accounted for by another correction factor K_C and thus the equation becomes:

$\tau = K_c K_s\ 8FD/\pi\, d^3$

Where K_c is the “curvature correction factor”.

Or easier the two correction factors are combined together as a single correction factor K_B where:

$K_B = K_c \times K_s = \frac{4C+2}{4C-3}$

Thus, $\tau = K_B \frac{8FD}{\pi d^3}$

3.1. Deflection of Helical Springs:

The deflection-force relation can be obtained using Castigliano's theorem. The total strain energy in the spring wire has two components torsional and shear.

$U = \frac{T^2 L}{2GJ} + \frac{F^2 L}{2AG}$

Substituting for T, A & J and knowing that L= πDN

Where N = N_a is the *Number of active coils*, we get:

$U = \frac{4F^2 D^3 N}{d^4 G} + \frac{2F^2 DN}{d^2 G}$

Applying *Castigliano's* theorem to get the deflection “δ”;

$\delta = \frac{\partial y}{\partial x} = \frac{8FD^3 N}{d^4 G} + \frac{4FDN}{d^2 G}$

Since c= D/d, we can write

$\delta = \frac{8FD^3 N}{d^4 G}\left(1+\frac{1}{2c^2}\right) \approx \frac{8FD^3 N}{d^4 G}$

[$1+\frac{1}{2c^2}$ = the effect of transverse shear is neglected]

Knowing that the Spring Rate, k= F/δ

Thus, $k = \frac{d^4\, G}{8D^3\, N_a}$

where N_a= Number of active coils

4. CONCEPT OF SPRING DESIGN:

The design of a new spring involves the following considerations:

- Space into which the spring must fit and operate.
- Values of working forces and deflections.
- Accuracy and reliability needed.
- Tolerances and permissible variations in specifications.
- Environmental conditions such as temperature, presence of a corrosive atmosphere.
- Cost and qualities needed.

The designers use these factors to select a material and specify suitable values for the wire size, the number of turns, the coil diameter and the free length, type of ends and the spring rate needed to satisfy working force deflection requirements. The primary design constraints are that the wire size should be commercially available and that the stress at the solid length be no longer greater than the torsional yield strength [4, 5].

4.1. Spring materials:

One of the important considerations in spring design is the choice of the spring material. Springs are usually made from alloys of steel. The most common spring steels are music wire, oil tempered wire, chrome silicon, chrome vanadium, and 302 and 17-7 stainless. Other materials can also be formed into springs, depending on the characteristics needed. Some of the more common of these exotic metals include beryllium copper, phosphor bronze, Inconel, Monel, and titanium. Titanium is the strongest material, but it is very expensive. Next come chrome vanadium and chrome silicon, then music wire, and then oil tempered wire. The stainless and exotic materials are all weaker than the rest.

Some of the common spring materials are given below.

- **Hard-drawn wire:** This is cold drawn, cheapest spring steel. Normally used for low stress and static load. The material is not suitable at subzero temperatures or at temperatures above 1200C.
- **Oil-tempered wire:** It is a cold drawn, quenched, tempered, and general purpose spring steel. It is not suitable for fatigue or sudden loads, at subzero temperatures and at temperatures above 1800C.
- **Chrome Vanadium:** This alloy spring steel is used for high stress conditions and at high temperature up to 2200C. It is good for fatigue resistance and long endurance for shock and impact loads.
- **Chrome Silicon:** This material can be used for highly stressed springs. It offers excellent service for long life, shock loading and for temperature up to 2500C.
- Music wire: - This spring material is most widely used for small springs. It is the toughest and has highest tensile strength and can withstand repeated loading at high stresses. It cannot be used at subzero temperatures or at temperatures above 1200C.
- **Stainless steel:** Widely used alloy spring materials.
- **Phosphor Bronze / Spring Brass:** It has good corrosion resistance and electrical conductivity. It is commonly used for contacts in electrical switches. Spring brass can be used at subzero temperatures [6].

The following table shows the physical properties of spring materials:

Table: - 1 Physical properties of spring materials [7]

Material	Analysis		Tensile properties			Torsional properties of wire			Process of manufacture, chief uses, special properties
	Element	Percent	Ultimate strength, MN/m^2	Elastic limit, MN/m^2	Modulus of elasticity, MN/m^2	Ultimate strength, MN/m^2	Elastic limit, MN/m^2	Modulus of torsion, MN/m^2	
Watch spring steel	C Mn	1.10-1.19 0.15-0.25	2260 2350	2060 2275	0.221×10^6 0.221×10^6	Not used -Do-	Not used -Do-	Not used -Do-	Cold-rolled & heat treated before forming main springs for watches & similar uses.
Clock-spring steel	C Mn	0.90-1.05 0.30-.50	1240 2340	1035 2060	0.206×10^6	-Do-	Do	-Do-	Cold-rolled & heat treated before forming clock& motor springs, misc. flat springs for high stress.
Flat spring steel	C Mn	0.65-0.80 0.50-0.90	1100 2210	860 1930	0.206×10^6	-Do-	-Do-	-Do-	Cold-rolled or annealed or tempered misc. flat springs.
High carbon wire	C Mn	0.85-0.85 0.25-0.60	1380 1725	1100 1450	0.206×10^6	1100 1380	760 1035	0.07845×10^6	Cold rolled or drawn high grade helical springs or wire forms.
Oil-tempered wire	C Mn	0.60-0.70 0.60-0.90	1070 2070	820 1725	0.200×10^6	795 1380	5515 900	0.07845×10^6	Cold-drawn & heat treated before coiling general spring use.
Music wire	C Mn	0.70-1.00 0.30-0.60	1725 3790	1035 2410	0.206×10^6	1035 2060	620 1240	0.07845×10^6	Patented & cold- drawn misc. small springs of various types high quality.
Hard-drawn spring wire	C Mn	0.60-0.70 0.90-1.20	1035 2060	685 1380	0.200×10^6	830 1520	520 90	0.07845×10^6	Patented & cold drawn same uses as music wire but lower quality wire.
Hot rolled bars special steel	C Mn	0.90-1.05 0.25-0.50	1210 1380	725 965	0.196×10^6	758 965	520 758	0.07257×10^6	Hot rolled heavy coil or flat springs.
Chrome-vanadium	C Mn	0.45-0.55 0.50-0.80	1380	1240	0.206×10^6	965	690	0.07845×10^6	Cold rolled or drawn special applications.

alloy steel 0.50%C	Cr V	0.80-1.10 0.15-0.18	1725	1590		1205	900		
Silicon Manganese alloy steel	C Mn Si	0.55-0.65 0.60-0.90 1.80-2.20	About the same as Chrome Vanadium			About the same as Chrome Vanadium.			Hot or cold rolled or drawn, in some applications may be used as a lower cost material in place of Chrome Vanadium.
Type 18-8 Stainless	Cr Ni C Mn	17-20 7-10 0.08-0.15 2 max.	1100 2275	415 1795	0.193×10^6	830 1650	300 965	0.06865×10^6	Cold rolled or drawn, best corrosion resistance, fair temp. resistance.
Cutlery-type stainless	Si Cr C	0.30-075 12-14 0.25-0.40	1180 1725	900 1380	0.193×10^6	830 1240	550 830	0.07551×10^6	Cold rolled or drawn, heat treated after forming, resists corrosion when polished, good temp. Resistance.
Spring brass	Cu Zn	64-72 balance	690 900	275 414	0.103×10^6	310 620	206 414	0.03825×10^6	Cold-rolled or drawn for electrical conductivity at low stresses for corrosion resistance.
Nickel Silver	Cu Zn Ni	56 25 18	900 1035	550 760	0.110×10^6	590 685	414 480	0.03825×10^6	Cold-rolled or drawn; better quality than brass corrosion resistance.
Phosphor bronze	Cu Sn Or Cu Sn	91-93 7-9 94-96 4-6	690 1035	414 760	0.103×10^6	550 725	345 590	0.04315×10^6	Cold-rolled or drawn; used for corrosion resistance and electrical conductivity.
Silicon bronze	Si Sn or Mn Cu	2-3 Small amounts balance	Properties similar to phosphor bronze.			Properties similar to phosphor bronze			Cold-rolled or drawn; used as substitute for phosphor bronze where lower cost is necessary.
Monel	Ni Cu Mn Fe	64 26 2.5 2.25	690 965	550 830	0.180×10^6	520 760	310 480	0.06571×10^6	Cold-rolled or drawn; resists corrosion; moderate stresses to 200ºC.
Inconel	Ni Ce Fe	80 14 balance	965 1210	760 930	0.210×10^6	660 830	380 550	0.07551×10^6	Cold-rolled or drawn; resists corrosion; high stresses to 340ºC.
K-monel	Ni Cu Al Fe	66 29 2.75 0.90	1100 1240	795 1000	0.180×10^6	725 860	450 590	0.06571×10^6	Cold-rolled or drawn; precipitation hardened by heat-treatment resists corrosion; high stresses to 230ºC.
Z-nickel	Ni Cu Mn Fe Si	98 Small amounts	1240 1590	900 1180	0.206×10^6	830 1035	414 620	0.07551×10^6	Cold-rolled or drawn; precipitation hardened by heat-treatment; resists corrosion; high stresses to 290ºC.
Beryllium-copper	Cu Be	98 2	1100 1380	685 1035	0.100×10^6 0.128×10^6	690 900	450 660	0.04120×10^6 0.04805×10^6 Subject to heat-treatment.	Cold-rolled or drawn; corrosion resistance like copper, high physicals for electrical work; low hysteresis.

5. LITERATURE REVIEW:

A few papers where discussed about the different types of mechanical springs used in the suspension system of automobiles. Also the different modes of spring failure and modifications were developed and validating the procedures for predicting the fatigue stress analysis.

James M. Meagher et al. (1996) the author presented the theoretical model for predicting stress from bending agreed with the stiffness and finite element model within the precision of convergence for the finite element analysis. The equation was calculated by principal stresses and von misses stress and it was useful for fatigue studies. A three dimensional finite element model was used for two coil of different wire model, one was MP35N tube with a 25% silver core and other a solid MP35N wire material helical conductor and the result was compared with the proposed strength of material model for flexural loading[8]. M. T. Todinov (1999) author had given for helical compression spring with a large coil radius to wire radius ratio, the most highly stressed region was at the outer surface of the helix rather than inside. The fatigue crack origin was located on the outer surface of the helix where the maximum amplitude of the principal tensile stress was calculated during cyclic loading; according to the author fatigue design should be based on the range of the maximum principal tensile stress [9]. Kotaro watanabe et al. (2001) a new type rectangular wire helical spring was contrived by the authors was used as suspension springs for rally cars, the stress was checked by FEM analysis theory on the twisting part. The spring characteristic of the suspension helper spring in a body was clarified. Manufacturing equipment for this spring had been proposed [10]. B. Ravi kumar et al. (2003) author was analysed the failure of a helical compression spring employed in coke oven batteries surface corrosion product was analyzed by X-ray diffraction (XRD) and scanning electron microscope - energy dispersive spectroscopy (SEM–EDS). Here used various testing procedure as chemical, surface corrosion product, fracture surface analysis. The conclusion of this work that the most probable cause of failure of the helical compression springs was corrosion fatigue accentuated by loss of surface residual compressive stress [11]. Dammak Fakhreddine et al. (2005), In this paper the author presented an efficient two nodes finite element with six degrees of freedom per node, capable to model the total behavior of a helical spring. The working on this spring was subjected to different cases of static and dynamic loads and different type of method (finite element method, dynamic stiffness matrix method) were governing equations by the motion of helical spring. This element permitted to get the distribution of different stresses along the spring and through the wire surface without meshing the structure or its surface [12]. Gulur Siddaramanna Shiva Shankar et al. (2006), this paper presented a low cost fabrication of complete mono composite leaf spring and mono composite leaf spring with bonded end joints. The 3D modeling of both steel and composite leaf spring was done and analysed using ANSYS. A comparative study was made between composite and steel leaf spring with respect to weight, cost and strength. The analytical results were compared with FEA and the results show good agreement with test results. From the results, it was observed that the composite leaf spring was lighter and more economical than the conventional steel spring with similar design specifications. Adhesively bonded end joints enhanced the performance of composite leaf spring for delamination and stress concentration at the end in compare with bolted joints. Composite mono leaf spring reduced the weight by 85 % for E-Glass/Epoxy, 91 % for Graphite/Epoxy, and 90 % for Carbon/Epoxy over conventional leaf spring [13]. L. Del Llano-Vizcaya et al. (2006) in this paper author used a critical plane approach, Fatemi–Socie and Wang–Brown, and the Coffin–Manson method based on shear deformation. The stress analysis was carried out in the finite element code ANSYS and the multiaxial fatigue study were performed using the fatigue software nCode and compared with experimental results in order to assess the different criteria. A failure analysis was conducted in order to determine the fatigue crack initiation point and a comparison of that location with the most damaged zone predicted by the numerical analysis is made. The M (Manson) method to estimate strain-life properties from the monotonic uniaxial tension test, gives better predictions of the spring fatigue lives than the MM (Muralidharan) method [14]. C. Berger, B. Kaiser (2006), in this paper the author presented the first results of very high cycle fatigue tests on helical compression springs. The springs tested were manufactured of Si–Cr-alloyed valve spring wire with a wire diameter between 2mm and 5 mm, shot-peened and the fatigue tests were continued up to 108cycles or even more. The aim should be to elaborate results about and insights concerning the level of the fatigue range in the stress cycle regime up to 109cycles, about the mechanisms causing failures and about possible remedies or measures of improvement [15]. Chang-Hsuan Chiu et al. (2007) In this paper the author presented four different types of helical composite springs were made of structures including unidirectional laminates (AU), rubber core unidirectional laminates (UR), unidirectional laminates with a braided outer layer (BU), and rubber core unidirectional laminates with a braided outer layer (BUR), respectively. It aims to investigate the effects of rubber core and braided outer layer on the mechanical properties of the aforementioned four helical springs. According to the experimental results, the helical composite spring with a rubber core can increase its failure load in compression. Therefore, author wanted to say that the shock absorbers with high performance might be expected to come soon [16]**.** L. Del Llano-Vizcaya et al. (2007) according to the author had given an experimental investigation been conducted to assess the stress relief influence on helical spring fatigue properties. First S–N curves were determined for springs treated under different conditions (times and temperatures) on a testing machine. Next the stress relief effect on spring relaxation induced by cyclic loading was evaluated. This methodology used in the experimental work and procedures used in the relaxation tests, fatigue tests and residual stress measurements. Finally, residual stresses were measured on the inner and outer coil surfaces to analyze the effect of heat treatment [17]. Y. Prawoto et al. (2008) the author had given an automotive suspension coil springs, their fundamental stress distribution, materials characteristic, manufacturing and common failures. A coil's failure to perform its function properly can be more catastrophic than if the coil springs were used in lower stress. As the stress level was increased, material and manufacturing quality became more critical. This paper discussed several case studies of suspension spring failures. The finite element analyses of representative cases were finite element modeling in metallurgical failure analysis synergizes the power of failure analysis into convincing quantitative analysis [18]. Yuuji Shimatani et al. (2010) In this paper author presented two types of specimen were processed by grinder and cutting tool, both specimens showed clear duplex *S-N* curve, composed of three or two types of failure mode depending on the stress amplitude, such as, a surface inclusion induced failure mode (S-mode), a subsurface inclusion-induced failure mode without (I-mode) and with granular bright facet (GBF) area in the vicinity of inclusion (IG-mode). Fatigue life in very high-cycle regime was almost same between the both specimens because of the existence of almost same size inclusion at crack origin [19]. Hsu Hsin-Tsun et al. (2010), In this paper the dynamic analysis of an electric vehicle (EV) has been investigated. The vehicle suspension system was built using multi-body dynamics (MBD) software, Altair MotionView/MotionSolve. Using the model, the dynamic properties of a target vehicle with gasoline engine and an electric vehicle with motor and batteries were simulated. When the engine was replaced with an electric motor and batteries, the lateral acceleration and the yaw rate of the vehicle was decreased slightly for a fixed steering wheel angle. Roll angle was increased due to the increase in vehicle weight. By using optimization software, the solution to adjust the lateral acceleration, roll angle and yaw rate of the EV car to meet similar performance of the original engine car was to increase the spring rate of front and rear suspension and the diameter of stabilizer bar, especially the diameter of stabilizer bar. Also, the adjustment of the design variables makes the roll rate of rear suspension increase. The kinematic simulation of the suspension system was also carried out to analyze their kinematic performance. Then the optimization of dynamic properties of the electric vehicle suspension system was carried out [20]**.** C. Berger, B.

Kaiser et al. (2011), In this paper the author presents a long-term fatigue tests up to a number of 109 cycles on shot peened helical compression springs with two basic dimensions, made of three different spring materials. The test springs were manufactured of oil hardened and tempered of SiCr and SiCrV-alloyed valve spring steel wires and of a stainless steel wire with diameters of 1.6 mm and 3.0 mm with shot peened. Method to be used experimental procedure the VHCF-test on spring. It becomes obvious that the various spring types in test exhibit different fatigue properties and different failure mechanisms in the VHCF regime [21]. M.Venkatesan et al. (2012),this paper described the design and experimental analysis of composite leaf spring made of glass fiber reinforced polymer and the objective was to compare the load carrying capacity, stiffness and weight savings of composite leaf spring with that of steel leaf spring. The design constraints were stresses and deflections. The dimensions of an existing conventional steel leaf spring of a light commercial vehicle were used to fabricate a composite multi leaf spring using E- Glass/Epoxy unidirectional laminates. Finite element analysis with full load on 3D model of composite multi leaf spring was done using ANSYS 10 and the analytical results were compared with experimental results. Compared to steel spring, the composite leaf spring was found to have 67.35% lesser stress, 64.95% higher stiffness and 126.98% higher natural frequency than that of existing steel leaf spring. A weight reduction of 76.4% was achieved by using optimized composite leaf spring [22].Mehdi Bakhshesh et al. (2012) In this paper author used helical spring is the most common used in car suspension system, steel helical spring related to light vehicle suspension system under the effect of a uniform loading has been studied and finite element analysis has been compared with analytical solution and steel spring has been replaced by three different composite helical springs including E-glass/Epoxy, Carbon/Epoxy and Kevlar/Epoxy. Numerical results have been compared with theoretical results and found to be in good agreement [23]. Y. N. V. Santhosh Kumar et al. (2012), In this paper author has shown the. This work deals with the replacement of conventional steel leaf spring with a Mono Composite leaf spring using E-Glass/Epoxy due to high strength to weight ratio. The design parameters were selected and analyzed with the objective of minimizing weight of the composite leaf spring as compared to the steel leaf spring. The leaf spring was modeled in Pro/E and the analysis was done using ANSYS Metaphysics software. It was observed that the composite leaf spring weighed only 39.4% of the steel leaf spring for the analyzed stresses [24]. Brita pyttel et al. (2012) in this paper author present is helical compression springs are used generally in fuel injection system of diesel engines, where it undergoes cyclic loading for more than 108 numbers of cycles and along the length of the spring at inner side. Finite element analyses were carried out, using ABAQUS 6.10. The simulation results show an oscillatory behavior of stresses along the length at inner side. Shear stresses along the length of the spring were found to be asymmetrical and with local maxims at starting of each middle coil. The FEA need to be modified further for cyclic analysis and failure analyses to be used with ABAQUS [25]. Priyanka Ghate et al. (2012; In the present investigation, it was found that the existing primary suspensions with composite spring assembly could sustain loads in normal operating conditions and maintain the required ride index, however, during cornering and hunting speeds failure of outer spring of primary suspension was observed. In the present work, an attempt was made to analyze in detail the reason for failure and a single non linear spring had been suggested to improve durability of the primary suspension and in the meantime the required ride index. Failure of the composite spring assembly was analyzed by applying the forces obtained from dynamic analysis. The dynamic analysis was performed using ADAMS/Rail at four different velocities and three different track conditions. The critical loading condition was achieved at a hunting speed of 132km/h on a curved track. A single spring set was considered in ADAMS/View to perform stress analysis to know durability. Numerical simulation showed that the stress level in the composite set which has both outer and inner spring was above allowable limit of 412 MPa where as a newly designed single non linear spring replacing the composite spring was found to have stress level of 160 MPa. The stress value obtained from numerical simulations in ADAMS was verified with analytical design calculations for the spring and the ride index was found to 1.78 which was 8% better than the earlier spring. It was concluded that the new spring design can enhance durability and ride index [26]. Kushal A Jolapara, et al. (2012), this paper showcase that there was a clear difference between the two cases of spring loading under static conditions. The load, load rates and stress values were higher for restricted uncoiling springs compared to unrestricted uncoiling. Both the experimental and FEM analysis results support this claim. Failure of the bearing in the suspension strut results in increased load on the spring, which increases the chances of failure. It was concluded that there was a definite change in spring performance under the two conditions [27]. V.K. Aher et al. (2012. In the present work static and fatigue analysis of a steel leaf spring of a light commercial vehicle (LCV) was carried out using NASTRAN solver and compared with analytical results. The preprocessing of the model was done by using HYPERMESH software. The stiffness of the leaf spring was studied by plotting load versus deflection curve for various load applications which shows the linear relationship. The simulation results were compared with analytical results. From the damage contour, the highest damage value was in acceptable range [28]. Tausif M. Mulla et al. (2012), the elastic behavior and the stress analysis of springs used in the Three Wheeler Vehicle’s front automotive suspension was discussed in this paper. The results obtained by a fully 3D FE analysis also highlighted the poor accuracy that can be provided by the classical spring model when dealing with these spring geometries. Relative errors on maximum shear stress ranging from 1.5 to 4 per cent, with reference to the applied loads, obtained when compared with the values calculated by using simple analytical model which was found in textbooks. The stress distributions clearly show that the shear stress was maximum at the inner side of the every coil. So the probability of failure of spring in every coil was same except end turns [29]. K.A. SaiAnuraag et al. (2012), In this paper the authors compared the static, dynamic & shock analysis for two & five layered composite leaf spring. The leaf springs were modeled with Unigraphics software NX7.5 and the analysis was carried out using ANSYS 11.0 FEA software to predict the behavior. The composite material used was E-Glass Epoxy. In static analysis the maximum displacement was observed in two layered i.e. 101.5mm compared to 83.23mm in five layered. Also during the static analysis Von-mises stress for the five layered was more than two layered i.e. 948Mpa for five layered compared to 795.4Mpa for two layered. For modal analysis various nodes were obtained and a comparative table was drawn for various nodes. The range of frequencies for two layers was 19.2 Hz to 1433 Hz and for five layers was 21.2 Hz to 1612 Hz. In Hormonic analysis amplitude vs. frequency graph for two layered and five layered were considered. For two layered amplitude decrease to a minimum and then increases & remains constant. For five layered, amplitude was remain constant initially but increased rapidly at the end. In shock analysis, when the time was increased, initially the displacement was increased and reached the maximum value and the decrease for two layer mode and for five layered mode, the displacement initially increased, became minimum and the increased with the time [30]. U. S Ramakanth et al. (2013); In this paper the stress analysis of conventional steel spring and composite leaf spring was carried out under the same static load conditions. Stresses in composite leaf springs were found out to be less as compared to the conventional steel leaf springs. Also a new combination of steel and composite leaf springs (hybrid leaf springs) were given the same static loading and was found to have values of stresses in between that of steel and composite leaf springs. Conventional 65Si7 (SUP9) leaf springs were found to weight about 58.757kgs, while the composite leaf springs weighed only 19.461kgs, and the hybrid leaf springs weighed 41.14 kg for the same specifications. The cost of the GFRP composite was very high when compared to conventional steel leaf springs, while the cost of hybrid leaf springs might be lesser when compared to GFRP composite leaf springs. The fatigue analysis of the steel leaf springs were carried with four approaches, Soderberg's approach was found out to give better results for the analysis of life data for leaf springs[31]. D.V Dodiya et al. (2013); In this work attempt was made to analyze a leading arm in a horizontally oriented spring damper assembly and the geometric and space and force requirements were studied to improve road handling abilities. The leading arm had been incorporated which essentially transmits the road undulations to the suspension shock absorber with the help of an arm which was robustly built and designed and its analysis had been carried out with ANSYS. The suspension system of the front wheels has a steering system to accommodate. The system had to be so designed to run in harmony with the vehicle steering system [32]. Pankaj Saini et al. (2013), In this paper, design and analysis of composite leaf spring analysis was done using ANSYS 9.0 software. The design constraint is stiffness. The material selected was glass fiber reinforced polymer (E-glass/epoxy), carbon epoxy and graphite epoxy is used against conventional steel. From the static analysis results it is found that there is a maximum displacement of 10.16mm in the steel leaf spring and the corresponding displacements in E-glass / epoxy, graphite/epoxy, and carbon/epoxy are 15.mm, 15.75mm and 16.21mm and all the values are below the camber length for a given uniformly distributed load 67 N/mm over the ineffective length & the von-mises stress in the steel is 453.92 MPa. The von-mises stress in Eglass/ epoxy, Graphite/ epoxy and Carbon/epoxy is 163.22MPa, 653.68 MPa and 300.3 MPa respectively. Among the three composite leaf springs, only graphite/epoxy composite leaf spring has higher stresses than the steel leaf spring. E-glass/epoxy composite leaf spring can be suggested for replacing the steel leaf spring from stress and stiffness point of view. Composite mono leaf spring reduces the weight by 81.22% for E-Glass/Epoxy, 91.95% for Graphite/Epoxy, and 90.51 % for Carbon/Epoxy over conventional leaf spring [33].C. Berger, B. Kaiser et al. (2013) In this paper the author presents a long-term fatigue tests on shot peened helical compression springs were conducted by means of a special spring fatigue testing machine at 40Hz. Test springs were made of three different spring materials- oil hardened and tempered SiCr- and SiCrV-alloyed valve spring steel and stainless steel up to 500 spring with a wire diameter of d = 3.0 mm or 900 spring with d = 1.6 mm were tested at different stress levels. Method to be used experimental procedure the VHCF-test on different wire diameter of spring. The paper includes a comparison of the result of the different spring sizes, materials, number of cycles and shot peening conditions and outlines further investigations in the VHCF-region [34].

CONCLUSION:

The literature review discussed above depicts that the design of mechanical springs used in automobiles is quite necessary to do it's deign analysis which involves stress distribution analysis, maximum displacement and different mode of failure. The springs undergo the fluctuating loading over the whole span of service life. In addition, various Design softwares like ANSYS, SolidWorks, Pro-E, CATIA, Autodesk Inventor, etc., have been used for performing the stress analysis of mechanical springs. Almost in all of the above cases, fatigue stress, shear stress, maximum displacement calculation, play significant role in the design of mechanical springs. This study shows that shear stress and deflection equation is used for calculating the number of active turns and mean diameter in helical compression springs. Comparison of the theoretical results obtained by the shear stress equation and Finite Element Analysis (FEM) of springs provides the better solution of the problems arises in the existing design of the mechanical spring. In future, it will help the designers for predicting the safe design of mechanical springs used in the automobiles to get better and comfortable ride.

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