Failure Analysis of Suspension System: Case Study-Tata Indica

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Abstract— This paper presents the failure analysis of a suspension system. As a case study a suspension system has been taken as a case study. An actual suspension system of TATA INDICA car has been selected. A premature failed system of passenger car is analyzed mathematically and metallurgical. Force and deflection analysis has been performed. At the same time metallurgical analysis has been done on the basis of microstructure of suspension system. Using High resolution SCM pictures for spring has been captured and analyzed. It has been observed that the system failed because of Poor surface finish, poor operating condition and excessive loading.

Key words: Suspension System, Microstructure, Fatigue Failure

I. Introduction

The coil spring being a part of suspension system; it is subjected to numerous influences in service life. The suspension springs are also subjected to road conditions and driving maneuvers. Hence it is needed to check the fatigue strength of spring to have better and problem free service life. The fatigue strength of suspension springs is vitally affected by material properties, coating and environmental influences. There will be chances of accident if spring fails. The accident may lead to injury to the occupants and/ or damage to the vehicle. In this project work we will be analyzing the suspension spring for passenger car in order to predict its fatigue life.

A typical vehicle suspension is made up of two components: a spring and a damper. The spring is chosen based solely on the weight of the vehicle, while the damper is the component that defines the suspensions placement on the compromise curve. Depending on the type of vehicle, a damper is chosen to make the vehicle perform best in its application. Ideally, the damper should isolate passengers from low-frequency road disturbances and absorb high frequency road disturbances. Passengers are best isolated from low-frequency disturbances when the damping is high. However, high damping provides poor high frequency absorption. Conversely, when the damping is low, the damper offers sufficient high-frequency absorption, at the expense of low-frequency isolation.

This work includes failure analysis of a Suspension system which gone under premature fatigue failure. TATA INDIACA passenger car was choosen for case study.

1) Significance of a Suspension System

- Low initial cost
- Minimum tyre wear
- Minimum deflection consists with required stability
- Minimum weight

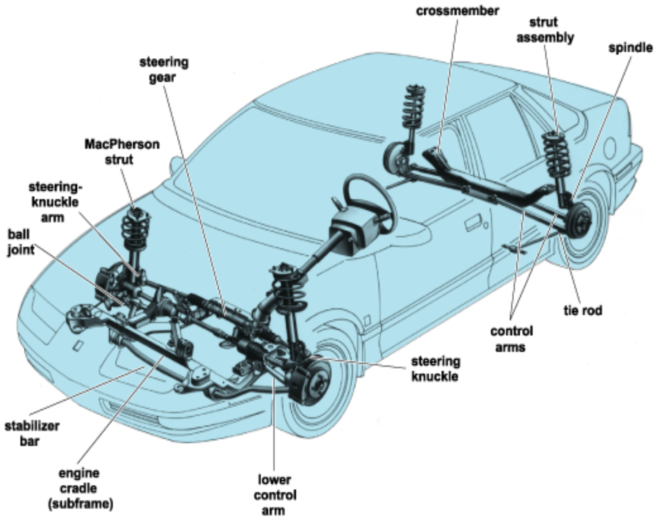

Fig. 1: Suspension System

2) Types of Suspension System

a) Active Suspension System

An active system, on the other hand, has the capability to adjust itself continuously to changing road conditions. Character to respond to varying road conditions, active suspension offers superior handling, road feel, responsiveness and safety.

Active suspension systems (also known as Computerized Ride Control) consist of the following components. a computer or two (called an electronic control unit, or ECU, for short), adjustable shocks and springs, a series of sensors at each wheel and throughout the car, and an actuator or servo atop each shock and spring. The components may vary slightly from manufacturer to manufacturer, but these are the basic parts that make up an active suspension system. As mentioned above, active suspension works by constantly sensing changes in the road surface and feeding that information, via the ECU, to the outlying components. These components then act upon the system to modify its character, adjusting shock stiffness, spring rate and the like, to improve ride performance, drivability, responsiveness, etc.

B. Review of Literature

In literature several works has been reported. Some significant work is presented here.

K. Michalczyk [1] the analysis of elastomeric coating influence on dynamic resonant stresses values in spring is presented in this paper. The appropriate equations determining the effectiveness of dynamic stress reduction in resonant conditions as a function of coating parameters were derived. It was proved that rubber coating will not perform in satisfactory manner due to its low modulus of elasticity in shear. It was also demonstrated that about resonance areas of increased stresses are wider and wider along with the successive resonances and achieve significant values even at large distances from the resonance frequencies.

B. Pyttel , I. et al. [2] Long-term fatigue tests on shot peened helical compression springs were conducted by means of a special spring fatigue testing machine at 40 Hz. Test springs were made of three different spring materials – oil hardened and tempered SiCr and SiCrV-alloyed valve spring steel and stainless steel. With a special test strategy in a test run, up to 500 springs with a wire diameter of d = 3.0 mm or 900 springs with d = 1.6 mm were tested simultaneously at different stress levels. Comparison the results for the springs with d = 1.6 mm and d = 3.0 mm and Ps = 98% are summarized in Fig. 1. Except for springs made of the stainless steel wire, the fatigue strength of springs with d = 3.0 mm is higher than for springs with d = 1.6 mm. The size effect would imply higher fatigue strength for smaller wire diameters.

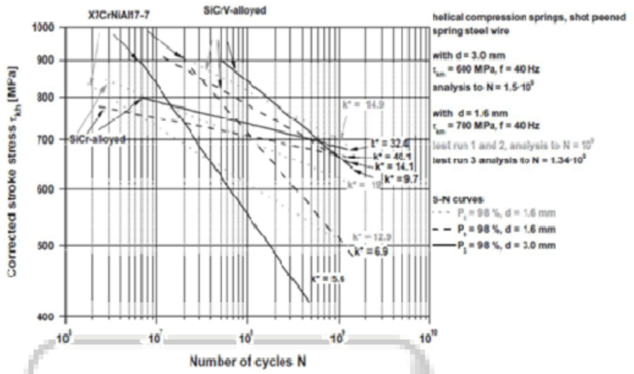

Fig. 2: Graph

Touhid Zarrin-Ghalami, et al.[3] Elastomeric components have wide usage in many industries. The typical service loading for most of these components is variable amplitude and multi axial. In this study a general methodology for life prediction of elastomeric components under these typical loading conditions was developed and illustrated for a passenger vehicle cradle mount. Crack initiation life prediction was performed using different damage criteria. The methodology was validated with component testing under different loading conditions including constant and variable amplitude in-phase and out-of-phase axial–torsion experiments. The optimum method for crack initiation life prediction for complex multi axial variable amplitude loading was found to be a critical plane approach based on maximum normal strain plane and damage quantification by cracking energy density on that plane.

Wei Li, et al. [4] Very high cycle fatigue (VHCF) properties of a newly developed clean spring steel were experimentally examined under rotating bending and axial loading. As a result, this steel represents the duplex S–N property only for surface-induced failure under rotating bending, whereas it represents the single S–N property for surface-induced failure and interior inhomogeneous microstructure induced failure under axial loading.

Sid Ali Kaoua a, et al. [5] This paper presents a 3D geometric modeling of a twin helical spring and its finite element analysis to study the spring mechanical behavior under tensile axial loading. The spiraled shape graphic design is achieved through the use of Computer Aided Design (CAD) tools, of which a finite element model is generated. Thus, a 3D 18-dof pentaedric elements are employed to discrete the complex ‘‘wired-shape” of the spring, allowing the analysis of the mechanical response of the twin spiraled helical spring under an axial load. The study provides a clear match between the evolution of the theoretical and the numerical

B. Pyttel , et al. [6] The paper gives an overview of the present state of research on fatigue strength and failure mechanisms at very high number of cycles (Nf > 107). Testing facilities are listed. A classification of materials with typical S–N curves and influencing factors like notches, residual stresses and environment are given. Different failure mechanisms which occur especially in the VHCF-region like subsurface failure are explained. There micro structural in homogeneities and statistical conditions play an important role.

Stefanie Stanzl-Tschegg [7] Ever since high-strength steels were found to fail below the traditional fatigue limit when loaded with more than 108 cycles, the investigation of metals’ and alloys’ very high cycle fatigue properties has received increased attention. A lot of research was invested in developing methods and machinery to reduce testing times.

Yuxin Penga, et al. [8] A stranded wire helical spring (SWHS) is a unique cylindrically helical spring, which is reeled by a strand that is formed of 216 wires. In this paper, parametric modeling method and the corresponding 3D model of a closed-end SWHS are presented based on the forming principle of the spring. By utilizing a PC + PLC based model as the motion control system, a prototype machine tool is designed and constructed, which improves the manufacturing of the SWHS. Via the commercial CAD package Pro/Engineering, numerical simulation is carried out to test the validity of the parametric modeling method and the performance of the machine tool. The scheme of the tension control system is analyzed and the control mechanism is set up, which have achieved the constant tension of each wire. A human machine interface is also proposed to achieve the motion control and the tension control. Experimental results show that the tension control system is well qualified with high control precision.

A.González Rodríguez, et al. [9] An adjustable-stiffness actuator composed of two antagonistic non-linear springs is proposed in this paper. The elastic device consists of two pairs of leaf springs working in bending conditions under large displacements. Owing to this geometric non-linearity, the global stiffness of the actuator can be adjusted by modifying the shape of the leaf springs. A mathematical model has been developed in order to predict the mechanical behavior of our proposal.

Matjazˇ Mršnik, Janko Slavicˇ, et al. [10] the characterization of vibration-fatigue strength is one of the key parts of mechanical design. It is closely related to structural dynamics, which is generally studied in the frequency domain, particularly when working with vibratory loads. Fatigue-life estimation in the frequency domain can therefore prove advantageous with respect to time-domain estimation, especially when taking into consideration the significant performance gains it offers, regarding numerical computations.

Nenad Gubeljaka, et al. [11] High strength steel grade 51CrV4 in thermo-mechanical treated condition is used as bending parabolic spring of heavy vehicles. Several investigations show that fatigue threshold for very high cycle fatigue depends on inclusion's size and material hardness. In order to determine allowed size of inclusions in spring's steel the Murakami's and Chapetti's model have been used. The stress loading limit regarding to inclusion size and applied stress has been determine for loading ratio R=-1 in form of S-N curves. Experimental results and prediction of S-N curve by model for given size of inclusion and R ratio show very good agreement. Pre-stressing and shot-penning causes higher compress stress magnitude and consequently change of loading ratio to more negative value and additionally extended life time of spring.

James M. Meagher et al. (1996)[12] the author presents 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 is calculated by principal stresses and von misses stress and it is useful for fatigue studies. A three dimensional finite element model is used for two coil of different wire model, one is MP35N tube with a 25% silver core and other a solid MP35N wire material helical conductor and the result is compared with the proposed strength of material model for flexural loading.

M. T. Todinov (1999) [13] author gives for helical compression spring with a large coil radius to wire radius ratio, the most highly stressed region is at the outer surface of the helix rather than inside. The fatigue crack origin is 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.

Kotaro watanabe et al. (2001)[14] a new type rectangular wire helical spring was contrived by the authors is 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 is clarified. Manufacturing equipment for this spring is proposed.

B. Ravi kumar et al. (2003)[15] author was analyzed 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.

Dammak Fakhreddine et al. (2005)[16] In this paper the author presents 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 is subjected to different cases of static and dynamic loads and different type of method (finite element method, dynamic stiffness matrix method) is governing equations by the motion of helical spring.

The literature review discussed above Dynamic modeling and failure analysis of passenger car 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. Failure Analysis of different car's suspension system is reported in literature but TATA INDICA is not considered for this analysis.

II. Failure Analysis of Suspension System

In this section mathematical analysis and metallurgical analysis is presented to investigate the premature failure of suspension system. Numerical analysis done is covered in next section.

A. Calculations for Suspension Spring (Coil)

Specifications of tata indica eV2
Kerb Weight: 1080 Kg
Seating Capacity: 5
Gross Weight = Kerb Weight + Passenger weight + Luggage weight
WG = 1080 + (5*70) + (5*10)
WG = 1480 Kg
From standard
The ratio of weight distribution is F/R: 49/51
Weight acting on each front wheel
W = (0.49* WG)/2
= (0.49* 1480)/2
W = 362.6 Kg
So, the reaction force acting on wheel,
P = W*g
= 362.6*9.81
P = 3557 N (Approx.)

1) Parameters of Helical Spring

Inner diameter of the spring=12.5mm
Outer diameter=137mm
Total no of coil=7
Active no of coil=5, due to square ends spring
Rigidity G=80N/mm^2

B. Mathematical Calculations for Spring Design

1) Solid Length

When the compression spring is compressed until the coils come in contact with each other, then the spring is said to be solid. The solid length of a spring is the product of total number of coils and the diameter of the wire. Mathematically,
Solid length of the spring,
$L_S = n'.d$
Where
n' = Total number of coils, and
d = Diameter of the wire.

2) Free Length

The free length of a compression spring is the length of the spring in the free or unloaded condition. It is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed). Mathematically, Free length of the spring,
L_F = Solid length + Maximum compression + *Clearance between adjacent coils (or clash allowance)

$$= n'.d + \delta_{max} + 0.15\ \delta_{max}$$

3) Spring Index

The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire. Mathematically,

C=D/d
D=Mean diameter of the coil
d=diameter of the coil

4) *Spring Rate*

The spring rate (or stiffness or spring constant) is defined as the load required per unit deflection of the spring. Mathematically,

k=W/ δ
W=Load
δ = Deflection of the spring.

5) *Pitch*

The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state. Mathematically,

P=free length/ *n'-1*

Symbol of the spring,
D = Mean diameter of the spring coil,
d = Diameter of the spring wire,
n = Number of active coils,
G = Modulus of rigidity for the spring material,
W = Axial load on the spring,
τ = Maximum shear stress induced in the wire,
C=Spring index,
P= Pitch of the coils,
δ = Deflection of the spring,
n' =Total no of coil

a) $Solid\ length\ L_s = n'.d$
$=77 \times 12.5 = 87.5\text{mm}$

b) $Spring\ index, C = D/d$

$$\mathrm{D = D_0 - d = 137 - 12.5 = 124.5mm}$$

$$C = \frac{124.5}{12.5} = 9.96$$

c) $Shear\ stress\ factor\ K_s = 1 + \frac{1}{2c} = 1.050$

d) $The\ torsional\ shear\ stress, \tau_1 = {8WD}/{\pi d^3} = 577.67N/mm^2$

Direct shear stress, $\tau_2 = {4W}/{\pi d^2} = 28.99\mathrm{N/mm^2}$

Resultant shear stress,

$\tau = \tau_1 + \tau_2 = 606.66$ (Positive sign to be used for inner edge of the wire

$\tau = \tau_1 - \tau_2 = 548.68$(Negative sign to be used for outer edge of the wire)

e) $Stress\ factor, K = \frac{4c-1}{4c-4} + \frac{0.615}{c} = 1.145$

f) Maximum shear stress induced in wire $\tau, = \mathrm{K} \times \frac{8\mathrm{wD}}{\pi \mathrm{d}^3} = 661.43\mathrm{N/mm^2}$

g) Deflection of the spring, $\delta = \frac{8\mathrm{WD}^3\mathrm{n}}{\mathrm{G\,d}^4} = 140.57\mathrm{mm}$

h) Free length of the wire, $\mathrm{L_F = n'.d} + \partial_{\max} + 0.15\,\partial_{\max} = 249.15$

i) $Spring\ rate\ or\ stifness, K = {W}/{\delta} = 25.30$

j) $Pitch = {free\ length}/{n' - 1} = 41.525$

Solid length L_s	87.5mm
Spring index, C	9.96
Shear stress factor K_s	1.050
The torsional shear stress, τ_1	577.67
Direct shear stress, τ_2	28.99
Resultant shear stress, τ	606.66 and 548.66
Stress factor, K	1.145
Maximum shear stress induced in wire τ,	$661.43\mathrm{N/mm^2}$
Deflection of the spring, δ	140.57mm
Free length of the wire, L_F	249.15
Spring rate or stifness, K	25.30
Pitch	41.525

Table 1: Mathematical Calculation Results

C. *The Coil Spring Design Method*

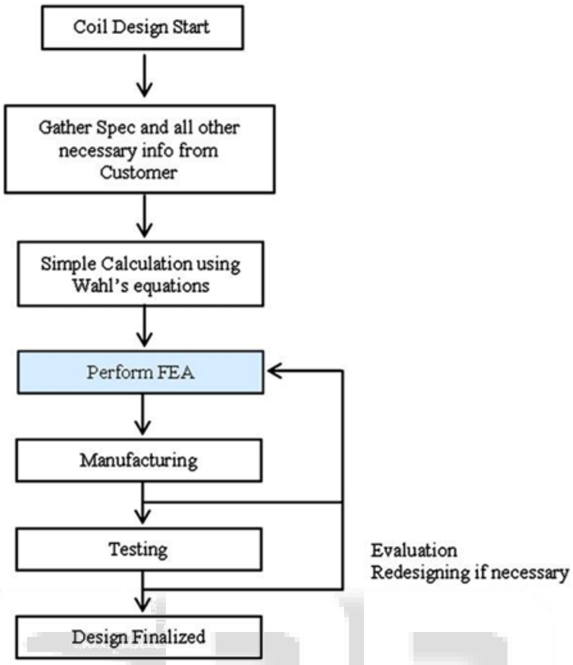

Fig. 3: Simple flow chart of coil spring design

III. FAILURE ANALYSIS: CASE STUDIES

A. *Raw Materials Defect*

A typical raw material defect is the existence of a foreign material inside the steel, such as non-metallic inclusions. Figs. 11a and 11b show the fracture surface and SEM fractograph, as well as the EDS spectrum of an inclusion located _1 mm below the surface. This particular coil failed early despite all other parameters being normal.

Fig. 12 shows a raw material defect that is usually very diffcult to find after a coil is formed. This type of defect is easy to detect during the cold drawing process of coil manufacturing preparation. An ideal raw material has the form of ferrite pearlite. However, a raw material can also have local binate inside the ferrite pearlite matrix. Due to a hardness difference, such raw materials may exhibit internal cracking.

B. *Surface Imperfections*

Surface imperfections can occur as small hardening cracks, tool marks, scale embedded to the base material during cold drawing, or surface flaws inherited by the raw material. Fig. 13 shows two different surface flaws deep enough to cause a coil spring to fail early. On the left side, the surface imperfection is inherited from the raw material. This type of defect can occur when the surface flaw detector does not function normally.

Poorly shot penned surfaces can also be classified as surface imperfections. Fig. 4 shows a comparison between two different coils that failed at similar locations, but possessed completely different fatigue lives. On the left side, the surface was shot penned poorly and therefore exhibited a shorter life. On the right side, the surface was shot penned sufficiently and therefore a longer life.

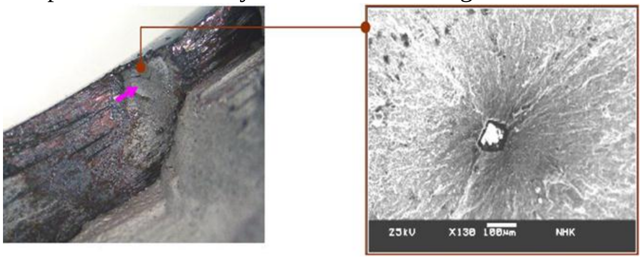

Fig. 4: Fracture surface of a coil that failed early due to an inclusion and Its SEM appearance.

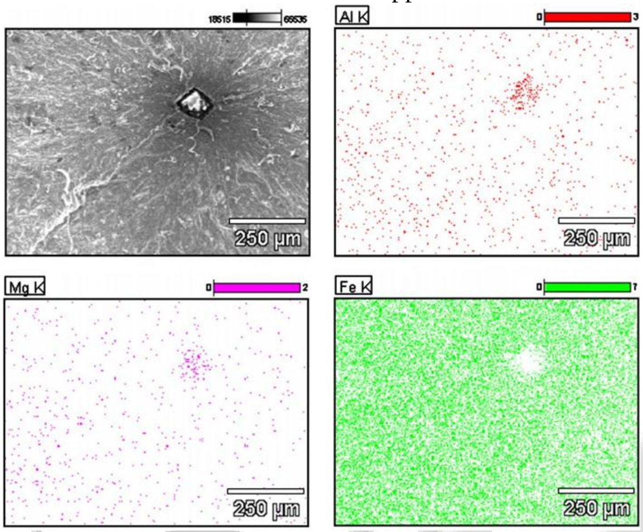

Fig. 5: Eds spectrum mapping of the inc

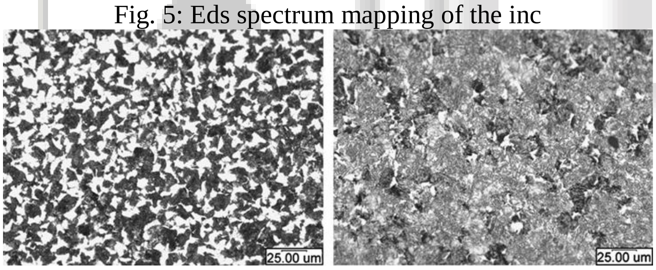

Fig. 6: Appearance of different microstructures extracted from the same bar.

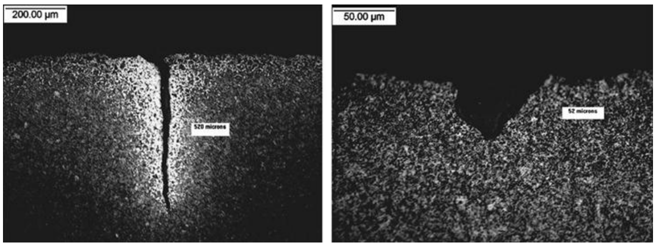

Fig. 7: Inherited from raw material (left) and surface imperfection due to manufacturing (right).

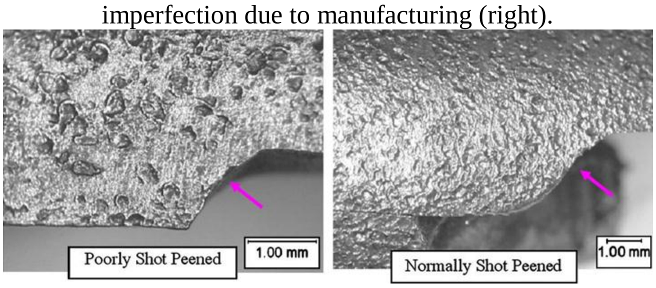

Fig. 8: Surface imperfections due to poor shot peening condition

On the left side, the microstructure is normal ferrite pearlite. on the right side, the same material has biteain structure inside the ferrite pearlite matrix.

IV. Conclusions

From micro structural analysis it appears that the spring was failed prematurely due to the inadequate shot penning process used to impart residual compressive stresses on the surface. The presence of excessive oxide inclusions in the steel might have also aggravated the case. Mathematical calculation for suspension system given the dimension of helical spring. The actual suspension system was as per our calculation. So the premature failure of the suspension system was further analyzed and it has been observed that the working condition which includes road and load on the vehicle was excessive. These parameters were confirmed by the owner of car as well as the service center of the vehicle manufacturer. Vehicle was used as a taxi in the remote location of north India where road condition was poor.

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