ART OF FATIGUE ANALYSIS OF HELICAL COMPRESSION SPRING USED IN TWO-WHEELER HORN

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ABSTRACT

Two-wheeler has a provision for a sounding a horn to be used while commuting so as to warn the passerby of the approaching vehicle as well as a signal for maintaining a safe distance or to communicate for any other reasons for safety. The horn is crucial as it is directly related to safety and the malfunction may evoke secondary claims and may affect the good-will of the customer. In most malfunction problems of two wheeler horns is due to fatigue failure of spring in warranty period so it is important to calculate the fatigue life and reduce fatigue failure in its intended working period (i.e. before 3,00,000 cycles). A typical helical compression spring configuration of two wheeler horn is chosen for study. Life analysis determines fatigue life of spring by the safe stress and corresponding pay load of the helical compression spring. This work describes fatigue analysis of the helical compression spring is performed using NASRAN solver and compared with analytical results. The pre processing of the spring model is done by using HYPERMESH software. The present work attempts to analyze the life using MSC fatigue software and also verified by experimentation. Life of spring found to be infinity using software analysis as well as experimentally. To extend life in place of present spring co-axial (series) spring is analyzed. Also factors affecting fatigue life of this compression spring are stated.

Keywords: Fatigue analysis, Geometric modeling, Helical compression spring, Life analysis, Two-wheeler horn.
I. INTRODUCTION

A spring is an elastic object used to store mechanical energy, whose function is to compress when loaded and to recover its original shape when the load is removed. In other words it is also termed as a resilient member. Springs are elastic bodies (generally made up of metals) that can be twisted, pulled, or stretched by some force. A spring 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.

II. LITERATURE REVIEW

Robert Stone [1] this paper describes a working definition of fatigue and a brief discussion of fatigue characteristics. A short history will be presented as to how fatigue test data has been evaluated historically. (e.g., S-N curves, Weibull distribution, modified Goodman diagrams, etc.) Reviews the proper methods by which spring manufacturers should estimate the fatigue life of helical compression springs during the design phase. Modified Goodman diagrams have been sufficiently characterized to facilitate direct calculation of predicted life. By using Modified Goodman diagrams few calculations are presented along with a comparison to the results of traditional graphical methods.

William H. Skewis [2] discussed spring reliability factors, as springs tend to be highly stressed because they are designed to fit into small spaces with the least possible weight and lowest material cost and required to deliver the required force over a long period of time. The reliability of a spring is related to its material strength, design characteristics, and the operating environment. Corrosion protection of the spring steel has a significant impact on reliability and so material properties, the processes used in the manufacturing of the spring, operating temperature and corrosive media must all be known before any estimate of spring reliability can be made. Spring reliability is also directly related to the surface quality and the distribution, type and size of sub-surface impurities in the spring material.

G Harinath Goud and E Venugopal Goud [3] in this paper describe “static analysis of leaf spring”, used in automobile suspension systems. The advantage of leaf spring over helical spring is that the ends of the spring may be guided along a definite path as it deflects to act as a structural member in addition to energy absorbing device. The main function of leaf spring is not only to support vertical load but also to isolate road induced vibrations. It is subjected to millions of load cycles leading to fatigue failure. 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. Finite element analysis has been carried out to determine the safe stresses and pay loads.

Mr. V. K. Aher, and Mr. P. M. Sonawane [4] in this paper describe,“Static and Fatigue Analysis Of Multi Leaf Spring Used In The Suspension System of LCV”, has done the work regarding the leaf spring used in automobiles and one of the components of suspension system. The purpose of this paper is to predict the fatigue life of semi-elliptical steel leaf spring along with analytical stress and deflection calculations. This present work describes static and fatigue analysis of a modified steel leaf spring of a light commercial vehicle (LCV). The dimensions of a modified leaf spring of a LCV are taken and are verified by design calculations. The non-linear static analysis of 2D model of the leaf spring is
performed using NASTRAN solver and compared with analytical results. The pre processing of the modified model is done by using HYPERMESH software. The stiffness of the modified leaf spring is studied by plotting load versus deflection curve for working range loads. The simulation results are compared with analytical results. The fatigue life of the leaf spring is also predicted using MSC Fatigue software.

Shigley’s [5] book of “Design of Mechanical Elements”, include, spring chapter. In this chapter we will discuss the more frequently used types of springs, their necessary parametric relationships, and their design.

III. METHODOLOGY

Spring design in fatigue applications generally based on following consideration:

- Available space and required loads and deflections
- Method of stressing
- Rate of load application
- Operation environment
- Minimum fatigue life at required reliability

For life analysis and life enhancement of helical compression spring used in two-wheeler horn following steps are used as a methodology.

![Methodology for Life Analysis of Two Wheeler Horn Spring](image)

**Fig. 1:** Methodology for Life Analysis of Two Wheeler Horn Spring.
The standard spring drawing and specifications available from the past design/validation are proved to be very useful for study.

Fig. 2: Two wheeler horn compression spring drawings

Fig. 2 shows two wheeler horn compression spring drawings. We designate ‘d’ as the mean coil diameter and d as the wire diameter. Now imagine that the spring is cut at some point a portion of it removed, and the effect of the removed portion replaced by the net internal reactions. The cut portion would contain a direct shear force \( F \) and a torsion \( T = Fd/2 \). To visualize the torsion, picture a coiled garden hose. Now pull one end of the hose in a straight line perpendicular to the plane of the coil. As each turn of hose is pulled off the coil, the hose twists or turns about its own axis. The flexing of a helical spring creates torsion in the wire in a similar manner. The maximum stress in the wire may be computed by superposition of the direct shear stress given by

\[
\tau = \frac{F}{A}
\]

and the torsional shear stress given by \( \tau_{\text{max}} = \frac{\tau}{r} \). The result is,

\[
\tau_{\text{max}} = \frac{\tau r}{J} + \frac{F}{A}.
\]

At the inside fiber of the spring. Substitution of, \( \tau_{\text{max}} = \frac{\tau r}{J} = T/D, r = D/2, J = \frac{\pi d^4}{32} \),

\[
A = \frac{\pi d^2}{4}
\]

gives,

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

\[\ldots\ldots(1)\]

Now we define the spring index, \( C = D/d \)  \[\ldots\ldots(2)\]
This is a measure of coil curvature. With this relation, equation (1) can be rearranged to give
\[
\tau = \frac{8FD}{\pi d^3} K_s \quad \text{.........(3)}
\]
Where \(K_s\) is a shear-stress correction factor and is defined by the equation,
\[
K_s = \left(1 + \frac{1}{2C}\right) \quad \text{.........(4)}
\]

For most springs, \(C\) ranges from about 6 to 12. Equation (3) is quite general and applies for both static and dynamic loads [5].

IV. GEOMETRIC PROPERTIES OF HELICAL COMPRESSION SPRING

Mode of loading: Cyclic loading
Outer diameter of coil, \(D_o = 4.8\) mm
Inner diameter of coil, \(D_i = 3.9\) mm

Mean coil diameter, \(d = \frac{D_o - D_i}{2} = \frac{4.8 - 3.9}{2} = 0.45\) mm

Mean diameter of coil, \(D = \frac{D_o + D_i}{2} = \frac{4.8 + 3.9}{2} = 4.35\) mm

Necessity of guide: Compression spring may buckle at low axial force for this reason spring needs guide its necessity is checked by,
\[
\frac{\text{Free length}}{\text{Mean coil diameter}} \leq 2.6 \quad \text{(Guide is not required)}
\]
\[
\frac{\text{Free length}}{\text{Mean coil diameter}} \geq 2.6 \quad \text{(Guide is required)}
\]

For 2 W horn compression spring, \(\frac{10.2}{4.35} = 2.34 < 2.6\)

Hence, guide is not required.(i.e. no need to consider effect of buckling)

Spring index, \(C = \frac{\text{Mean dia of coil}}{\text{Dia of spring wire}} = \frac{D}{d} = \frac{4.35}{0.45} = 9.67\)

Wahl’s stress factor, \(K_s = K_W = \frac{4(9.67) - 1}{4(9.67) - 4} + 0.615 = 1.15\)

Shear stress , \(\tau = \frac{8 F D}{\pi d^3} K_s = \frac{8 \times 0.4 \times 9.8 \times 4.35}{3.142(0.45)^3} \times 1.15 = 547.92 \ \text{N/mm}^2\)

Axial Deflection, \(y = \frac{8FD^3 \times i}{Gd^4} = \frac{8 \times 0.4 \times 9.8 \times (4.35)^3 \times 6}{7300 \times 9.8 \times (0.45)^4} = 5.27 \ \text{mm}\)

Stiffness or Rate of spring: \(F_o = \frac{F}{y} = \frac{0.4}{5.27} = 0.076 \ \text{kg/mm}\)

Free length, \(I_o \geq (1 + n)d + y + a\)
\(I_o \geq (6 + 1)0.45 + 5.16 + 0.25(5.16)\)
\(I_o \geq 9.6 \ \text{mm}\)

Actual free length is taken as 10.2 mm

Pitch, \(p = \frac{I_o - 2d}{i} = \frac{10.2 - 2(0.45)}{6} = 1.55 \ \text{mm}\)
V. HELICAL COMPRESSION SPRING DESIGN FOR STATIC SERVICE

The preferred range of spring index is, $4 \leq C \leq 12$, with the lower indexes being more difficult to form (because of the danger of surface cracking) and springs with higher indexes tending to tangle often enough to require individual packing. This can be the first item of the design assessment. The recommended range of active turns is $3 \leq N_a \leq 15$.

To maintain linearity when a spring is about to close, it is necessary to avoid the gradual touching of coils (due to non perfect pitch).

A helical coil spring force-deflection characteristic is ideally linear. Practically, it is nearly so, but not at each end of the force-deflection curve.

The spring force is not reproducible for very small deflections, and near closure, nonlinear behavior begins as the number of active turns diminishes as coils begin to touch. The designer confines the spring’s operating point to the central 75 percent of the curve between no load, $F = 0$, and closure, $F = F_s$.

Thus, the maximum operating force should be limited to $F_{\text{max}} \leq (7/8)F_s$.

Defining the fractional overrun to closure as $\xi$, where $F_s = (1+\xi)F_{\text{max}}$ it follows that, $F_s = (1+\xi)(7/8)F_s$. From the outer equality, $\xi = (1/7) = 0.143 = 0.15$.

Thus, it is recommended that, $\xi \geq 0.15$.

In addition to the relationships and material properties for springs, we now have taken these recommended design conditions with the factor of safety at closure (solid height) $N_s \geq 1.2$.

The theoretical variation of load verses shear stress as shown in table I.

<table>
<thead>
<tr>
<th>Load (N)</th>
<th>Maximum Shear Stress (N/mm2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>139.79</td>
</tr>
<tr>
<td>2</td>
<td>279.58</td>
</tr>
<tr>
<td>3</td>
<td>419.38</td>
</tr>
<tr>
<td>4</td>
<td>559.17</td>
</tr>
<tr>
<td>5</td>
<td>698.97</td>
</tr>
<tr>
<td>6</td>
<td>838.77</td>
</tr>
<tr>
<td>7</td>
<td>978.56</td>
</tr>
</tbody>
</table>

Table I: Variation of Shear Stress with load

To prevent the accident and to safeguard the occupants from accident, horn system is necessary to be analyzed in context of the maximum safe load of a helical compression spring.

In the present work, helical compression spring is modeled and static analysis is carried out by using NASTRAN software. Also its life is analyzed.

It is observed that the maximum stress is developed at the inner side of the spring coil. From the theoretical and the NASTRAN, the allowable design stress is found between the corresponding loads 3 to 6 N.

It is seen that at 7N load, it crosses the yield stress (yield stress is 903 N/mm$^2$).

By considering the factor of safety 1.5 to 2. It is obvious that the allowable design stress is 419 to 838 N/mm$^2$. 

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VI. MODELING AND ANALYSIS OF HELICAL COMPRESSION SPRING

In computer-aided design, geometric modeling is concerned with the computer compatible mathematical description of the geometry of an object. The mathematical description allows the model of the object to be displayed and manipulated on a graphics terminal through signals from the CPU of the CAD system.

The software that provides geometric modeling capabilities must be designed for efficient use both by the computer and the human designer [3].

To use geometric modeling, the designer constructs the graphical model of the object on the CRT screen of the ICG system by inputting three types of commands to the computer. The first type of command generates basic geometric elements such as points, lines, and circles. The second type command is used to accomplish scaling, rotation, or other transformations of these elements. The third type of command causes the various elements to be joined into the desired shape of the object being created on the ICG system. During this geometric process, the computer converts the commands into a mathematical model, stores it in the computer data files, and displays it as an image on the CRT screen.

The model can subsequently be called from the data files for review, analysis, or alteration. The most advanced method of geometric modeling is solid modeling in three dimensions [3]. Geometric modeling CATIA is used for computer-aided design. Finite element method HYPERMESH is used for meshing.

![FEM model of helical compression spring with meshing](image)

**Fig 3:** FEM model of helical compression spring with meshing

VII. STATIC ANALYSIS

For the above given specification of the helical compression spring, the static analysis is performed using NASTRAN to find the maximum safe stress and the corresponding pay load. After geometric modeling of the helical compression spring with given specifications it is subjected to analysis. The analysis involves the following discretization called meshing, boundary conditions and loading.

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A. Meshing
Meshing involves division of the entire model into small pieces called elements. This is done by meshing. It is convenient to select the hex mesh because of high accuracy in result. To mesh the helical compression spring, the element type must be decided first. Here, the element type is solid 45. Fig. 4 shows the meshed model of the helical compression spring.

![Fig 4: Meshing, boundary conditions and loading of compression spring](image)

B. The Following are the Material Properties of the Given Helical Compression Spring
Material = Stainless Steel, Young’s Modulus = 193000 N/mm², Density = 8E-009 tones/mm³, Poisson’s ratio = 0.3 and shear stress = 588.99 N/mm².

C. Boundary Conditions
The helical compression spring is mounted in the horn of the two-wheeler automobile. The casing of the horn is connected to the handle of the vehicle. The ends of the helical compression spring are closed and ground. The helical spring is fixed in between the horn button and casing directly with a frame so that the spring can move longitudinally about the shaft translation is occurred. The bottom end of the spring is fixed and the other end of the spring is connected to the button of the vehicle. The horn button has the flexibility to slide along the X-direction when load applied on the spring and also it can move in longitudinal direction. The spring moves along Y-direction during load applied and removed. Therefore the nodes of bottom end of the compression spring are constrained in all translational degrees of freedom. So the top end is constrained as, UY, ROTY and the nodes of the bottom end are constrained as UY, UZ, UX. Fig. 4 shows the boundary conditions of the helical compression spring.

D. Loads Applied
The load is distributed equally by all the nodes associated with the center of the spring. The load is applied along FY direction as shown in Fig. 4. To apply load, it is necessary to select the circumference of the spring centre and consequently the nodes associated with it. It is necessary to observe the number of nodes associated with the circumference of the spring centre, because the applied load need to divide with the number of nodes associated with the circumference of the spring centre.
In the present work, helical compression spring is modeled and static analysis is carried out by using NASTRAN software. It is observed that the maximum stress is developed at the inner side of the spring coil. From the theoretical and the NASTRAN, the allowable design stress is found between the corresponding loads 3 to 6 N. It is seen that at 7N load, it crosses the yield stress (yield stress is 903 N/mm$^2$). By considering the factor of safety 1.5 to 2, it is obvious that the allowable design stress is 419 to 838 N/mm$^2$. So the corresponding loads are 3 to 6 N. Therefore it is concluded that the maximum safe pay load for the given specification of the helical compression spring is 4 N.

From Fig. 5 and Fig. 6, it is obvious that maximum stress developed is at inner side of the spring sections. The red color indicates maximum stress, because the constraints applied at the interior of the spring. Since the inner surfaces of the spring are subjected to maximum stress, care must be taken in spring surface, fabrication and material selection. The material must have good ductility, resilience and toughness to avoid sudden fracture.
VIII. FATIGUE ANALYSIS OF PRESENT SPRING

Using MSC Fatigue life is analyzed by importing present helical compression spring which is modeled and static analyzed. The result obtained is as shown in fig. 7. It’s life is found to be more than 3,00,000 cycles.

![Fig. 7: Life](image)

 IX. STATIC ANALYSIS OF MODIFIED CO-AXIAL TWO WHEELER HORN SPRING

Similarly, static analysis modified co-axial spring is carried out which is shown in fig. 8. Then importing the same in Msc. Fatigue, fatigue analysis is completed. For modified co-axial spring also life is found to be infinite.

![Fig. 8: Shear Stress Analysis (Max. Shear)](image)
X. EXPERIMENTATION

The standard or reference data available for the past design/validation as standards or reference proved to be very useful for experimentation.

The spring design once identified as suitable for the application is subjected to trials for checking the fatigue life by using MSC. Fatigue experimentation is carried out using a `Life Testing SPM’ of a suitable type and capacity at Kiran Machine Tools Ltd. I-1, M. I. D. C. area, Jalgaon (MH). The spring is held in position with other operating conditions identical to the application.

The trials are conducted in a very controlled environment with focus on the variables influencing the fatigue life. A 300000 cycles trial is conducted to ensure consistency/repeatability of the spring behavior which is shown in fig. 10. Present spring is found to be safe for 300000 cycles. Fig. 10 shows test report of the same spring.

Fig 9: Life (Max. Shear)

Fig 10: Test Report of Present Spring
XI. RESULTS AND DISCUSSIONS

From static analysis, the allowable design stress is 419 to 838 N/mm$^2$. So the corresponding loads are 3 to 6 N. Hence, analysis is carried at load 4 N.

Life of present spring found to be more than 3,00,000 cycles using software analysis as well as experimentally. To extend life in place of present spring co-axial (series) spring is analyzed and its life is also more than 3,00,000 cycles.

The inner surfaces of the spring are subjected to maximum stress, care must be taken in spring surface, fabrication and material selection.

For improving fatigue life following methods can be used.

- The most obvious way to improve fatigue life is to design a spring with a lower stress.
- Shotpeening can greatly improve the fatigue life of springs.
- The set taken during pressing must be allowed for in the coiling process so that the correct loads are obtained after press.
- Upgrade material to a higher tensile range or a higher quality grade.
- The surface of the wire is shaved before the final draw to eliminate surface defects.
- Shock loading and resonance can seriously reduce cycle life.
- Some fatigue problem can be improved by stress corrosion, buckling, coil clash, wear, non-axial forces, and dynamic loading.

XII. FUTURE SCOPE

Future scope for this study is to study micro-structure, change material of spring design the same spring.

The inner surfaces of the spring are subjected to maximum stress, care must be taken in spring surface, fabrication and material selection. The material must have good ductility, resilience and toughness to avoid sudden fracture.

XII. ACKNOWLEDGMENTS

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