Static Analysis of Helical Compression Spring Used in Two-Wheeler Horn

S. S. Gaikwad, P. S. Kachare

Abstract- Every two-wheeler has a provision of horn. It is used for maintaining a safe distance or to communicate for safe drive. The horn is critical element in vehicle. It is directly related to safety as well as goodwill of the company. Static analysis determines the safe stress and corresponding pay load of the helical compression spring. The present work attempts to analyze the safe load of the helical compression spring. A typical helical compression spring configuration of two wheeler horn is chosen for study. This work describes static analysis of the helical compression spring is performed using NASTRAN solver and compared with analytical results. The pre processing of the spring model is done by using HYPERMESH software.

Keywords : Geometric modeling, Helical compression spring, Static analysis, Two-wheeler horn.

I. INTRODUCTION

A spring is defined as an elastic body, 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. The torque can be used to cause a rotation, for example, to close a door on a cabinet or to provide a counter balance force for a machine element pivoting on a hinge[3].

Fig. 1 show the schematic representation of a helical spring acted upon by a compressive load F.

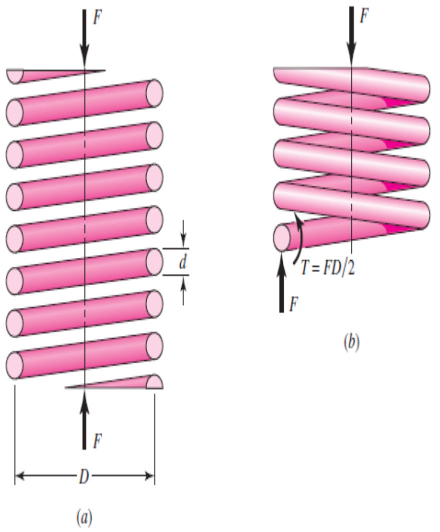

Fig. 1 Spring loaded by the axial force F (a & b)

Manuscript Received on February, 2013.

Gaikwad S. S., Mechanical Department, SVERI's C.O.Engg., Pandharpur, Solapur University, Solapur, Maharashtra, India.

Kachare P.S., Mechanical Department, SVERI's C.O.Engg., Pandharpur, Solapur University, Solapur, Maharashtra, India.

Fig. 1 (a) shows a round-wire helical compression spring loaded by the axial force F. We designate D as the mean coil diameter and d as the wire diameter. Now imagine that the spring is cut at some point Fig. 1 (b), a portion of it removed, and the effect of the removed portion replaced by the net internal reactions. Then, as shown in the fig. 1 (b), from equilibrium 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 $\tau_{max} = \frac{\tau r}{J}$

The result is, $\tau_{max} = \frac{\tau r}{J} + \frac{F}{A}$

At the inside fiber of the spring. Substitution of, $\tau_{max} = \tau$, $T = FD/2$, $r = d/2$,

$J = \pi d^4/32$ and $A = \pi d^2/4$ gives

$$\tau = \frac{8FD}{\pi d^3} + \frac{4F}{\pi d^2} \quad (1)$$

Now we define the spring index

$$C = \frac{D}{d} \quad (2)$$

This is a measure of coil curvature. With this relation, Equation (1) can be rearranged to give

$$\tau = K_s \frac{8FD}{\pi d^3} \quad (3)$$

Where K_s is a shear-stress correction factor and is defined by the equation

$$K_s = \frac{2C+1}{2C} \quad (4)$$

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

II. HELICAL COMPRESSION SPRING DESIGN FOR STATIC SERVICE

The preferred range of spring index is $4 \le C \le 12$, 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 $3 \le N_a \le 15$. To maintain linearity when a spring is about to close, it is necessary to avoid the gradual touching of coils (due to nonperfect 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,

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

$F = 0$, and closure, $F = F_s$ Thus, the maximum operating force should be limited to $F_{max} \leq \left(\frac{7}{8}\right) F_s$

Defining the fractional overrun to closure as ξ, where

$$F_s = (1 + \xi)F_{max} \quad (5)$$

it follows that

$$F_s = (1 + \xi)F_{max} = (1 + \xi)\left(\frac{7}{8}\right) F_s$$

From the outer equality, $\xi = \frac{1}{7} = 0.143 = 0.15$. Thus, it is recommended that, $\xi \geq 0.15$.

In addition to the relationships and material properties for springs, we now have some recommended design conditions to follow, namely:

$$4 \leq C \leq 12 \quad (6)$$

$$3 \leq N_a \leq 15 \quad (7)$$

$$\xi \geq 0.15 \quad (8)$$

$$N_s \geq 1.2 \quad (9)$$

Where, Ns is the factor of safety at closure (solid height) [3]. The theoritical variation of load verses shear stress as shown in table I.

Table. I variation of Shear Stress with load

Load (N)	Maximum Shear Stress (N/mm2)
1	139.79
2	279.58
3	419.38
4	559.17
5	698.97
6	838.77
7	978.56

A. Geometric Properties of Helical Compression Spring

Wire Diameter - 0.45mm, Outer Diameter of Coil - 4.8mm, Mean Diameter of Coil - 4.35mm, No. of Active Coils - 4, Total No of Coils - 6, Free Length - 10.2mm.

III. LITERATURE REVIEW

G Harinath Gowd and E Venugopal Goud [1] 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 [2] 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 [3] 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.

IV. MODELING & 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 [1].

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 [1]. Geometric modeling CATIA is used for computer-aided design. Finite element method HYPERMESH is used for meshing.

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

International Journal of Engineering and Advanced Technology (IJEAT)
ISSN: 2249 – 8958, Volume-2 Issue-3 February 2013

Fig. 2 FEM model of helical compression spring with meshing

V. 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 discritization called meshing, boundary conditions and loading.

A. Meshing

Meshing involves division of the entire of 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. 2 shows the meshed model of the helical compression spring.

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 vehicle. The ends of the helical compression spring are closed and ground. The helical spring is fixed in between 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. Figure 3 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. 3. 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.

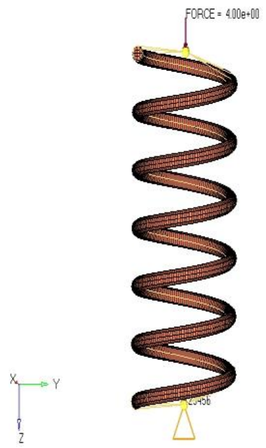

Fig. 3 Meshing, boundary conditions and loading of compression spring

VI. RESULTS AND DISCUSSIONS

The deformed and undeformed shape of the leaf spring is shown in Fig. 4 and the table II gives the Von-Mises stress and shear stress at various loads.

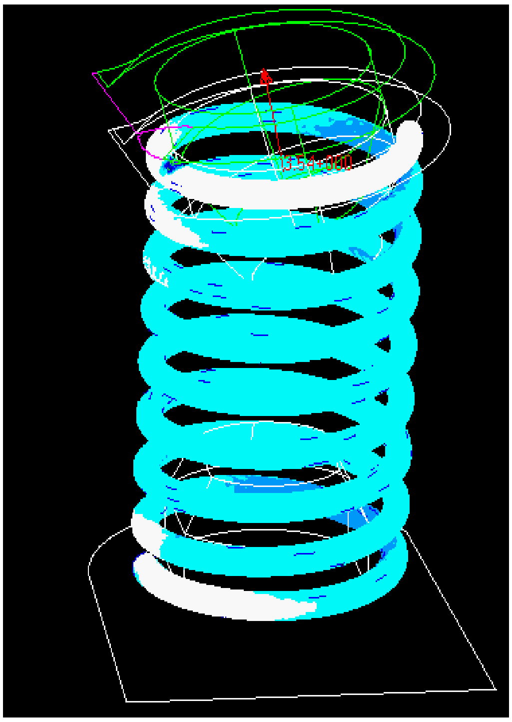

Fig. 4 The deformed and undeformed shape of the helical compression spring

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

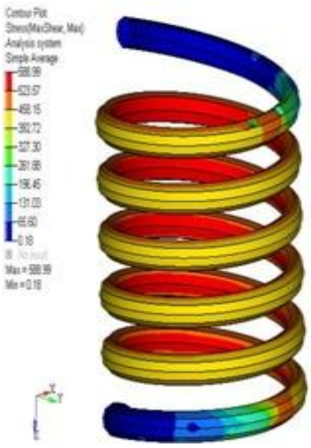

Fig. 5 Maximum shear stress at design load 4N

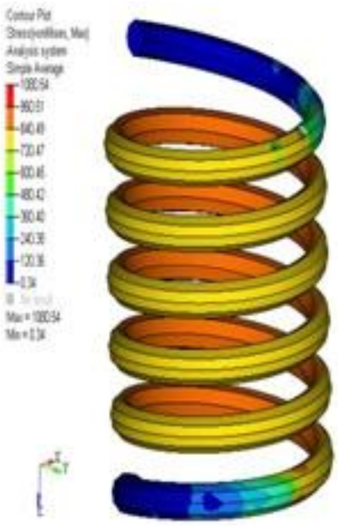

Fig. 6 Von-Mises stress at design load 4N

Table. II Variation of Von-Mises stress and maximum shear stress with load

Load (N)	Maximum Shear Stress (N/mm2)	Von-Mises Stress (N/mm2)
1	102.72	180.84
2	205.44	361.67
3	391.20	633.40
4	410.88	723.35
5	513.60	904.18
6	616.32	1085.02

Static analysis is performed to find the Von-Mises stress and maximum shear stress by using NASTRAN software and these results are compared with calculated in mathematical analysis at various loads. It is shown in table III and table IV.

Table. III Comparison between theoretical and NASTRAN for Shear Stress

Load (N)	Theoritical Maxi. Shear Stress (N/mm2)	Nastran Maxi. Shear Stress (N/mm2)`
1	139.79	102.72
2	279.58	205.44
3	419.38	391.20
4	559.17	410.88
5	698.97	513.60
6	838.76	616.32

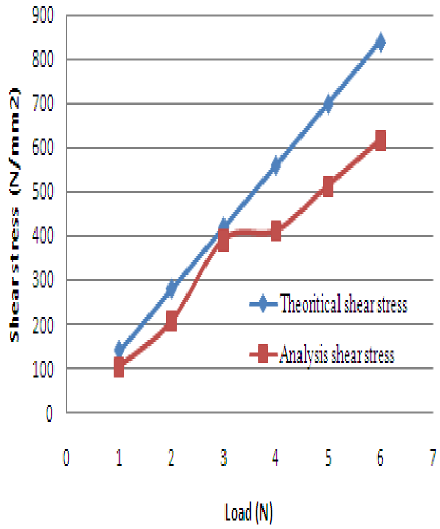

Fig. 7 Variation of shear stress with load

A graph is plotted as shown in Fig. 7 between load verses theoretical shear stress and nastran shear stress. Load on the X - axis and shear stress on the Y - axis. Another graph is plotted as shown in Fig. 8 between load versus nastran Von-mises stress and theoretical shear stress. Load on the X - axis and stress on the Y - Axis.

Table. IV Comparison between Theoretical and NASTRAN for Shear Stress and von-mises stress

Load (N)	Theoritical Shear Stress (N/mm2)	Nastran Von-Mises Stress (N/mm2)
1	139.79	180.84
2	279.58	361.67
3	419.38	633.40
4	559.17	723.35
5	698.97	904.18
6	838.76	1085.02

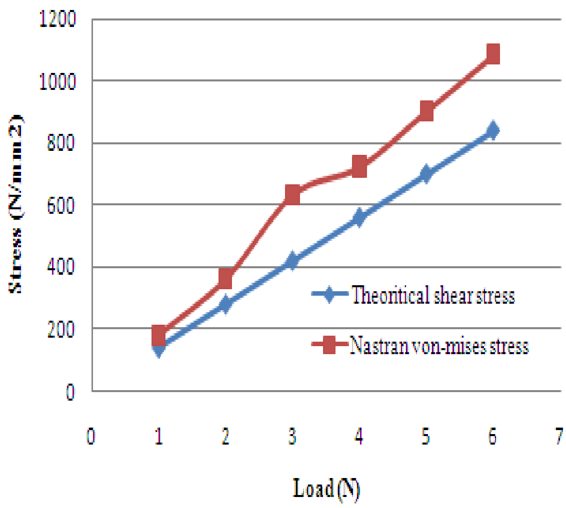

Fig. 8 Variation of shear stress and von-mises stress with load

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication

It is seen that in fig. 7, graph that when load increases the shear stress increases linearly. So load - stress graph gives the straight line relationship. At lower loads both theoretical and NASTRAN results are very close, but when load increases the NASTRAN results are uniformly reduced compared to theoretical results and it is reversed in fig. 8. Variation of shear stress and von-mises stress with load.

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.

VII. CONCLUSION

To prevent the accident and to safeguard the occupants from accident, horn system is necessary to be analysed 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. 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/mm2). By considering the factor of safety 1.5 to 2. It is obvious that the allowable design stress is 419 to 838 N/mm2. 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. At lower loads both theoretical and NASTRAN results are very close, but when load increases the NASTRAN results are uniformly reduced compared to theoretical results.

VIII. ACKNOWLEDGMENTS

This study was carried out through a Able Technologies (I) Pvt. Ltd., has sponsored me this project work. The authors wish to thank Prof. P. S. Kachare and Mr. S. S. Kulkarni for his helpful advices.

REFERENCES

[1] G. Harinath Gowd and E. Venugopal Goud, “Static analysis of leaf spring”, International Journal of Engineering Science and Technology, Vol. 4, pp. 3794-3803, 2012.
[2] Mr. V. K. Aher and Mr. P. M. Sonawane, “Static and fatigue analysis of multi leaf spring used in the suspension system of LCV”, International Journal of Engineering Research and Applications, Vol. 2, pp. 786-1791, 2012.
[3] Shigley, “ Mechanical Engineering Design”, Eighth Edition, Budynas−Nisbett, McGraw−Hill Primis , 2006, Ch. 10.
[4] Mehdi Bakhshesh and Majid Bakhshesh, “Optimization of steel helical spring by composite spring”, International Journal of Multidisciplinary Sciences and Engineering, Vol. 3, pp. 47-51, 2012.
[5] M. Senthil Kumar and S. Vijayarangan,“Static analysis and fatigue life prediction of steel and composite leaf spring for light passenger vehicles”, Journal of Scientific & Industrial Research, Vol. 66, pp. 128-134, 2007.