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FINITE ELEMENT ANALYSIS OF HELICAL COIL COMPRESSION SPRING FOR THREE WHEELER AUTOMOTIVE FRONT SUSPENSION

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Abstract-The current paper deals with the stress analysis of a helical coil compression spring, which is employed in three wheeler's auto-rickshaw belonging to the medium segment of the Indian automotive market. In the design of this kind of spring both the elastic characteristics and the fatigue strength have to be considered as significant aspects. In addition to this particular elastic property, as a consequence of the research effort in reducing the mass of components typical of the automotive industry, these springs have to face very high working stresses. The structural reliability of the spring must therefore be ensured. So for this purpose the static stress analysis using finite element method has been done in order to find out the detailed stress distribution of the spring.

Keywords- Helical Coil Compression Spring, Finite Element Analysis, Fatigue Strength

1. INTRODUCTION

Three-Wheeled Vehicles (TWVs) form an essential part of public transport for the urban middle class population of India. Apart from India, TWVs are also being used world over for public transport and for carrying freight. The suspension system for such three wheeler's vehicles is very poor as concerned with the ride comfort of the passengers. Relatively higher centre of gravity, the lack of differential for the driving rear axle have been cited as contributors to rollovers and pitching. This also adds the discomfort to passengers. Very few publications evaluate the performance of TWVs in everyday use for mass transportation. Nowadays the trend in the industries is moving towards the weight reduction in every component and springs are also not exempted. Over the country side roads where these TWVs are widely used for carrying freight, the stress levels in the springs go high. As helical coil compression springs are one of the main parts of the suspension system, it becomes quite necessary to do the complete stress analysis of the spring. As these springs undergo the fluctuating loading over the service life, it becomes essential to find out the fatigue limit of the same.

Mechanical spring is defined as an elastic body that has the primary function to deflect or distort under load, and to return to its original shape when the load is removed. First step in the design of spring in general, is to determine the loads and the deflections required for a given spring application depending upon the type of the loading. In addition to this tentative selection of the material must be made. In case of the most general approach for the spring design, the maximum stress in the spring wire may be computed by superposition of direct shear and the torsional shear stress. To design the helical coil compression spring for small pitch angle, a very common approach called as approximate theory considering direct shear and effect of curvature is referenced here.[1] The assumption is that an element of an axially loaded helical spring behaves as a straight bar in pure torsion. If P be the load acting axially on the spring, d is the diameter of the spring wire, D is the mean diameter of the coil, and then forces acting on the element are resolved into a twisting moment $PD/2$, acting in a radial plane and a direct axial shearing force P. The stresses set up by the twisting moment are considered first and then superimposed on the stresses due to the direct shear. But the maximum shear stress occurs at the inner side of the coil and may be considerably higher than the stress given by elementary theory by an amount depending on the spring index. The reason for this is the short fiber length a^1b^1 at the inner side of the coil as shown in the figure 1. Thus if the radial sections bb^1 and aa^1 rotate through an angle with respect to each other about spring axis, the inner fiber a^1b^1 will be subjected to much higher stress and shearing strain than outside fiber $a\,b$. Also at the inner side of the coil, the shear stress due to the direct axial load P added such that it produce the torque moment $PD/2$, at this point. Thus the stress range at inner side of the coil is normally much higher than elsewhere and for this reason fatigue failure generally starts at this region. Therefore maximum shear stress at the inside of the coil given by

$$\tau_{max} = \frac{8PD}{\pi d^3} K \qquad \text{Eq.1}$$

where

$$K = \left(\frac{4C-1}{4C-4} + \frac{0.615}{C}\right) \qquad \text{Eq.2}$$

$$C = \frac{D}{d}$$

K is called as Wahl's factor used for considering stress correction because of curvature effect and C is called as spring index. Form this it is clear that spring index has straight impact on the stress distribution. Stress obtained by these formulae can be cross checked by finite element method for better understanding of the stress distribution. The deflection is given by formula

$$\delta = \frac{Gd^4}{8D^3N} \qquad \text{Eq.3}$$

where G is the shear modulus of material and N is the number of active coils in the spring.[1]

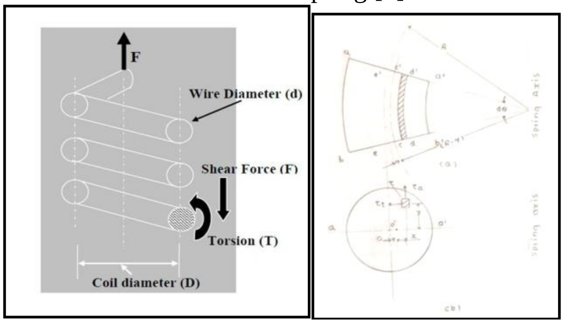


Figure 1- Elemental Section of Helical Coil Compression Spring [1]

2. FINITE ELEMENT ANALYSIS

A primary reason to use FEA in coil spring design is the ability to reduce error caused by the simplification of equations, mainly concerning the pitch angle. An FEA-based design begins with the selection of the element type, how the model should be constructed, how accurate the results should be, and how fast the model should run. The most accurate FEA results can be obtained by creating 3-D parts of a coil spring and its seats, followed by meshing the parts with a 3-D solid element. Finer meshing with higher-order elements will produce more accurate results. However, with a higher number of elements and nonlinearity due to the contact between a coil spring and its seats, or between the coil and itself, each analysis could take hours to run. While the accuracy of the result is important, the computational time must be reasonable to incorporate FEA into coil spring design. To resolve lengthy computational time in a solid model, a 3-D beam element is often selected to model a coil spring and seats. Since the deformation of a seat under compression is very minimal and can be ignored, the material properties of seats are set very high to act as a rigid body. Contact between a coil and seat, or between the coil and itself, is detected by gap elements or rigid elements.

2.1 Geometry and Material Properties

Detailed specifications of the spring under consideration are given in the table 1. This spring is used in the TWV's front suspension. Spring is made up of ASTM A227 material with squares and grounded ends. The material properties are given in the table 2. This spring had undergone hardening process with oil quenching and followed by the stress relieving treatment at 400 Degree Celsius for 25 minutes. Then shot peening has been done after some time with ball size of 2 mm for good surface finish.

Wire Diameter, d	8.0	mm
Outer Diameter, D_o	54.0	mm
Inner Diameter, D_i	38.0	mm
Mean Diameter, D_m *or* D	46.0	mm
No of Active Turns, N_a	10.0	
No of Total Turns, N_t	12.0	
Free Length, l_0	195.0	mm
Solid Length, l_s	96.0	mm
Spring Index, C	5.75	

Table 1-Spring Specifications

The effect of residual stresses has been neglected in this analysis. A basic solid model of the spring is made in the Pro-E Wildfire 4 software as shown in figure 2 (A). The spring seats are not modeled in this analysis and contact of seats with spring surface is considered as rigid body.

Modulus of Elasticity, E	196.50	GPa
Poisson's Ratio, μ	0.250	
Shear Modulus, G	78.60	GPa

Table 2-Material Properties of ASTM A227 Spring Steel [2]

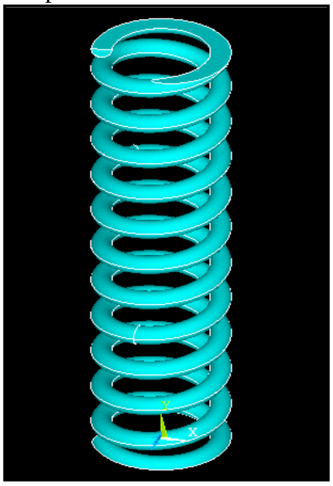
2.2 Element Type and Mesh Convergence Criteria

ANSYS 13.0 is the software used for the pre and post processing. This spring is meshed with different elements and different meshing types and then comparative study has been done in order to find out the convergence criteria. At first the spring was meshed with element SOLID187. This element is a higher order 3-dimentional, 10-node element. SOLID187 has a quadratic displacement behavior and is well suited to modeling irregular meshes (such as those produced from various CAD/CAM systems). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions. This element is used for the Tet-Meshing.

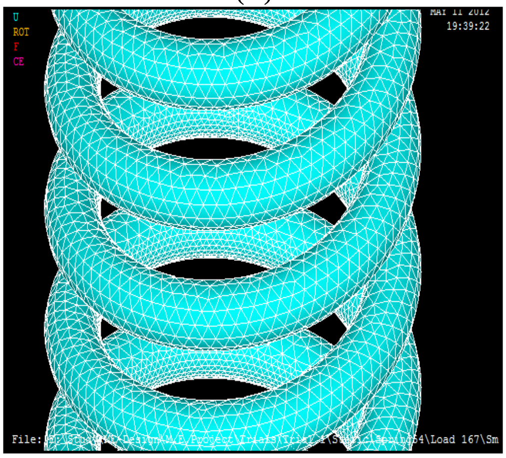
In the second case the spring was modeled with element SOLID95/SOLID186. This is used for 3-D modeling of solid structures having 20 nodes. It can tolerate irregular shapes without as much loss of accuracy. SOLID186 elements have compatible displacement shapes and are well suited to model curved boundaries. It is defined by eight nodes having three degrees of freedom at each node:

translations in the nodal x, y, and z directions. This element is used for Hex-meshing.

Mesh study is performed on the FE model to ensure sufficiently fine sizes are employed for accuracy of calculated results but at competitive cost (CPU time). In the process, the shear stress is the specified field variable is selected and its convergence is monitored and evaluated. Selecting the right techniques of meshing are based on the geometry, model topology, analysis objectives. Tetrahedral meshing produces high quality meshing for boundary representation solids model imported from the most CAD system.



(A)



(B)

Figure 2- Solid Model (A) and FEM Model (B) with Tet Mesh (SOLID 187)

After testing the spring model for different element sizes with above elements SOLID187 and SOLID 186 one by one it was seen that smaller mesh size captures the higher stress value. The element SOLID187 with Tet Mesh is shown in figure 2 (B). But smaller element size less than 2 mm for both the types of elements was not implemented because it creates the large number of nodes and it requires comparatively too much time and space for the storage and calculation for the computer. Then for different meshing options, comparing the accuracy of the solution and time required for the same; the most effective FEM model obtained when compared with the theoretical value of the stress counter of maximum shear stress obtained by using Wahl's Equation. The stress counter for the load of 167 kg or 1638.27 N having theoretical value about 474.08 N/Sq.mm is compared with different element sizes is shown in figure 3. Tet-Mesh (SOLID 187) with element size 2 mm and refined at inner side of the coil at level one is the most converging model with the theoretical values.

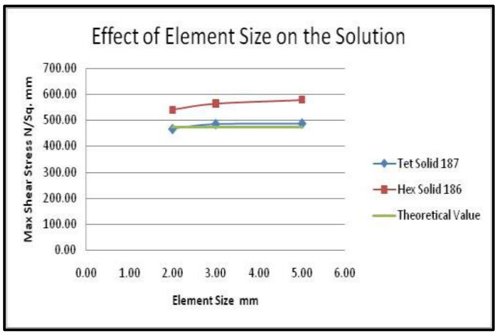


Figure 3- Comparison of Different Meshing Types

2.3 Loading Conditions

As this spring is used in the TWV's front suspension it is necessary to find out the load acting on the spring in actual practice in static condition as well as in dynamic condition. Normally total weight of the vehicle with driver and one passenger is about 405 Kg, but for safer side it is taken as 500 Kg concentrated at the center of gravity of the vehicle. It is assumed that this total weight is equally divided into two springs of rear suspension and one spring of front suspension. So the front suspension spring is experiencing approximately 167 Kg load. This load is modeled in the analysis with the help of mass element. Then rigid body constraint equations are applied for giving contact between this element and the surface elements of the spring on upper side.

2.4 Static Stress Analysis

The linear static analysis was performed to determine the stress and strain results from the finite element model. The material utilized in this work consists of a linear elastic, isotropic material. The choice of the linear elastic material model is essentially mandated. Model loading consist of the applied mechanical load, which is modeled as the load control and the displacement control. From the analysis, the inner side of the coil is found to experience the largest stresses. The maximum shear stress distribution of the coil spring is shown in figure 4. From the result, the maximum shear stress of value 467.664 N/sq. mm in *xy* plane occurred at nodes on inner side of the every coil. Figure 5 shows the effect of load on the shear stress obtained by FEM.

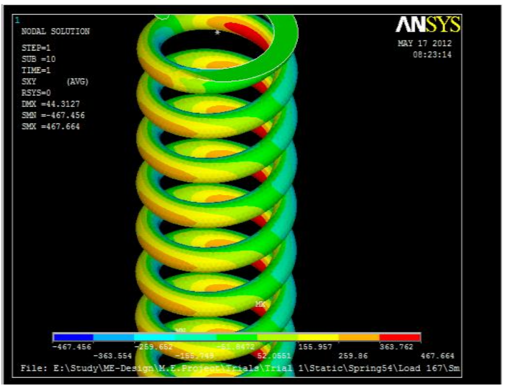


Figure 4-Shear Stress Counter

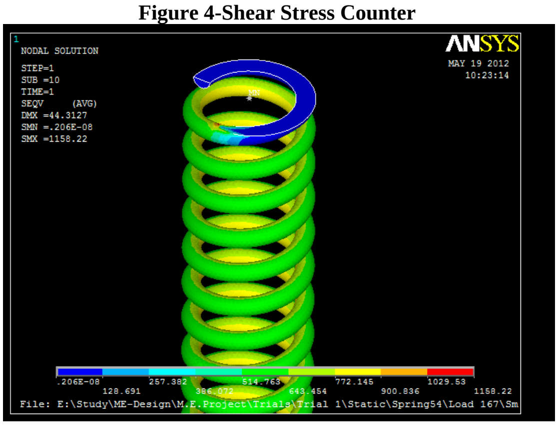


Figure 5-Eq. Von Mises Stress Counter

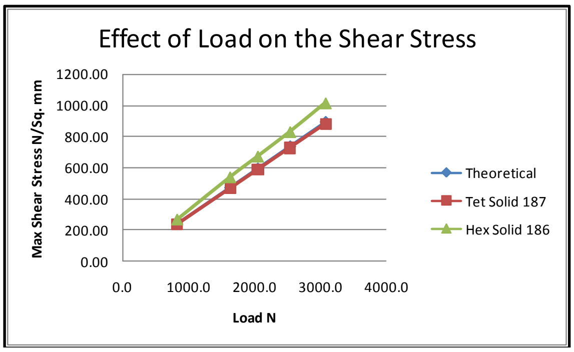


Figure 5-Load Vs Shear Stress Effect on spring

3. CONCLUSION

The elastic behavior and the stress analysis of springs employed in the TWV's front automotive suspension have been presented and 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 is found in textbooks. The stress distribution clearly shows that the shear stress is having maximum value at the inner side of the every coil. The distribution of the stress is similar in every coil. So the probability of failure of spring in every coil is same except end turns. In such case residual stress in every coil may be important factor which influence the failure.

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