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Research Article

Design for Life Enhancement of a Compression Spring used for 2 W Horn using Fatigue Analysis

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Abstract

The horn switch has a helical compression spring at the base within the assembly to restore the position of the horn switch once used. The spring is subjected to cyclic loading with a maximum expected frequency of usage at about 150 cycles a day. Considering the design life of a two-wheeler at about 15 years, the spring should withstand a cyclic compression loading for about 0.8 million times. With a room for safety, the life of the spring is expected to be about a 0.3million cycles. This dissertation report gives the brief look on the life enhancement of compression spring used in two wheeler horn using fatigue analysis. The objective of the present work is to calculate the fatigue life and reduce the incidence of fatigue failure for the working life of compression spring in two wheeler horn. Hence the present spring (AISI 1065 : Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC) is analysed using MSC fatigue software, which shows the maximum stress is developed at the inner side of the spring. Its experimentation shows hair cracks with permanent set, which proves fatigue failure occurs before 3,00,000 cycles. Considering factors affecting fatigue life, material properties are changed (AISI 302/304 : Cr 17-19%, Ni 8-10%, Hardness- 45 HRC) and same is analysed using MSC fatigue software, which shows infinite life. And experimental test proved no any failure for 3,00,000 cycles.

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

1. 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.

2. Literature review

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 (Robert Stone, 2010).

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 (William H. Skewis, *Support Systems Technology Corporation*).

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 (G Harinath *et al* , 2012).

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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 (V Aher *et al*,2012).

The fatigue life of the leaf spring is also predicted using MSC Fatigue software. 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 (Shigley, 2006).

3. 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.

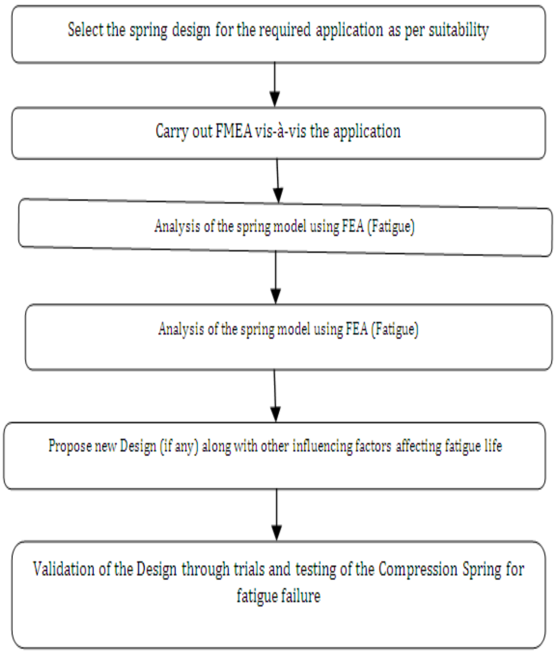

Fig. 1: Methodology for life analysis of two wheeler horn spring.

The standard spring drawing and specifications available from the past design/ validation are proved to be very useful for study.

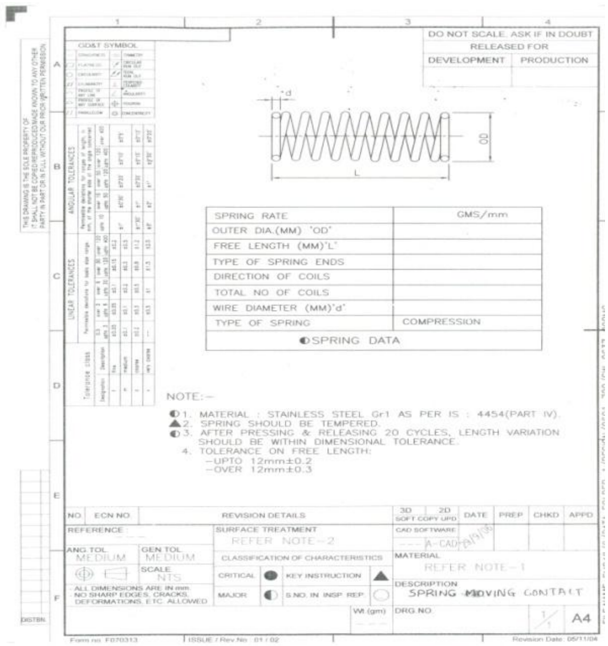

Fig.2: Two wheeler horn compression spring drawings

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

$\tau = F/A$ and the torsional shear stress given by

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

At the inside fiber of the spring. Substitution of,

$\tau_{max} = \tau$, $T = FD/2$, $r = D/2$, $J = \dfrac{\pi d^4}{32}$ and $A = \dfrac{\pi d^2}{4}$ gives,

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

Now we define the spring index, $C = D/d$ (2)

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

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

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

$$K_s = \left(1 + \frac{1}{2C}\right) \qquad (4)$$

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

4. 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,

$\frac{Free\ length}{Mean\ coil\ diameter} \leq$ 2.6 ….(Guide is not required)

$\frac{Free\ length}{Mean\ coil\ diameter} \geq$ 2.6 ….(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, $C = \frac{Mean\ dia.of\ coil}{Dia.of\ spring\ wire} = \frac{D}{d} = \frac{4.35}{0.45}$ = 9.67
Wahl's stress factor,
$Ks = K_W = \frac{4(9.67)-1}{4(9.67)-4} + \frac{0.615}{9.67}$ =1.15
Shear stress , $\tau = \frac{8FD}{\pi d^3} Ks$
$= \frac{8\times0.4\times9.8\times4.35}{3.142(0.45)^3} \times 1.15 = 547.92$ N/mm^2

Axial Deflection, $y = \frac{8FD^3 \times i}{Gd^4}$
$= \frac{8\times0.4\times9.8\times(4.35)^3\times6}{7300\times9.8\times(0.45)^4}$ =5.27 mm
Stiffness or Rate of spring:
$F_o = \frac{F}{y} = \frac{0.4}{5.27}$ = 0.076 kg/mm
Free length, $I\mathbf{o} \geq (I + n)d + y + a$
$I\mathbf{o} \geq (6 + 1)0.45 + 5.16 + 0.25(5.16)$
$I\mathbf{o} \geq 9.6$ 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 mm

5. 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 $F_{max} \leq (7/8)F_s$.
Defining the fractional overrun to closure as ξ ,
where $F_s = (1+\xi) F_{max}$
it follows that, $F_s = (1+\xi)(7/8)F_s$. From the outer equality, ξ = (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) Ns ≥ 1.2. The theoretical variation of load verses shear stress as shown in table 1.

Table 1: Variation of Shear Stress with load

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

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.

6. 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 (G Harinath *et al* , 2012). 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 (G Harinath *et al* , 2012).

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

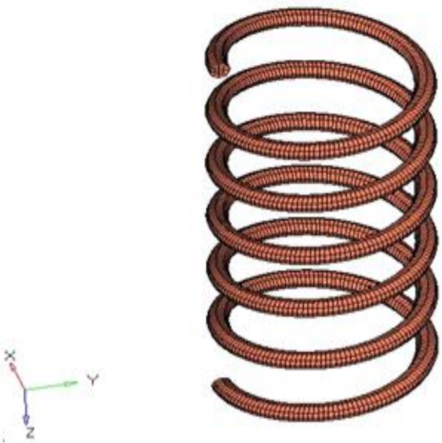

Fig. 3 FEM model of helical compression spring with meshing

7. 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. 4 shows the meshed model of the helical compression spring.

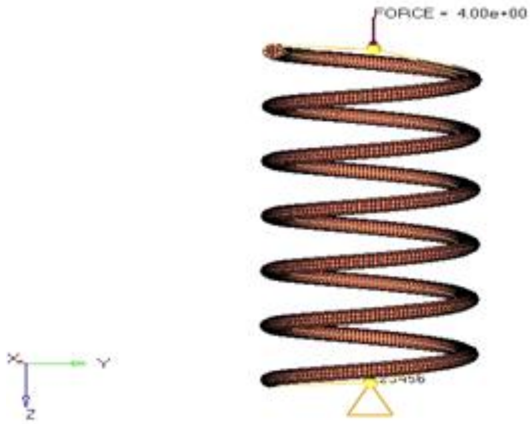

Fig. 4 Meshing, boundary conditions and loading of compression spring

B. Material Properties of the Given Helical Compression Spring

Material = Stainless Steel, Young's Modulus = 193000 N/mm^2, Density = 8E-009 tones/mm^3, Poisson's ratio = 0.3 and shear stress = 588.99 N/mm^2.

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. 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.

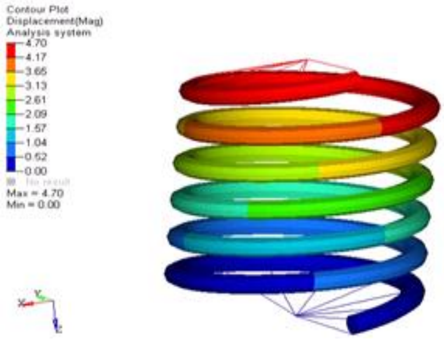

Fig. 5: Deformation

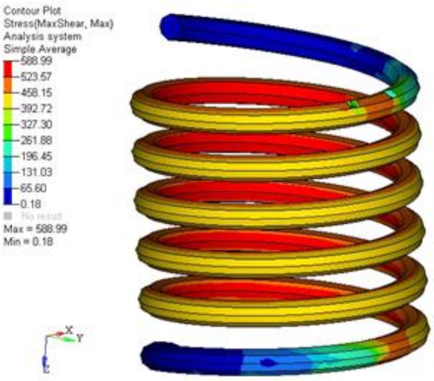

Fig. 6: Maximum shear stress at design load 4N

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.

8. Fatigue Analysis using MSC Fatigue

Present Spring: Using MSC Fatigue life is analyzed by importing present helical compression spring(AISI 1065 : Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC) which is modeled and static analyzed. The result obtained is as shown in fig. 7. Which shows the maximum stress is developed at the inner side of the spring.

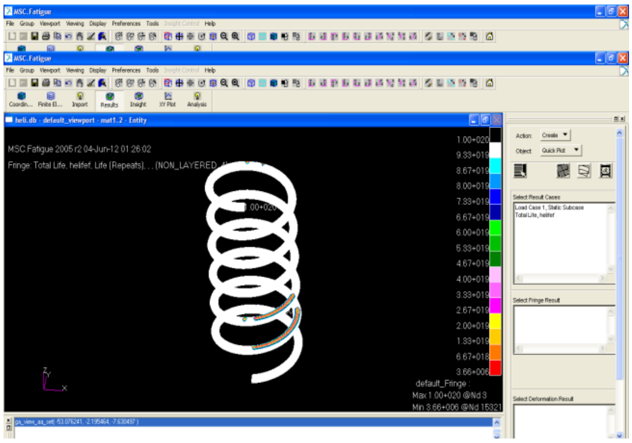

Fig.7: Life (Present spring)

Proposed Spring: Life of proposed helical compression spring (AISI 302/304 : Cr 17-19%, Ni 8-10%, Hardness- 45 HRC) which is analyzed. The result obtained is as shown in fig. 8. shows infinite life.

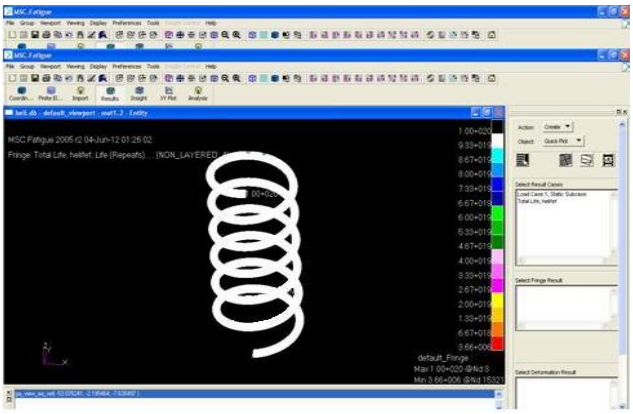

Fig.8: Life (Proposed spring)

9. Experimentation

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 3,00,000 cycles trial is conducted to ensure consistency/ repeatability of the spring behavior which is shown in table 2.

Present helical compression spring's (AISI 1065 : Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC) five sample springs tested. It is observed that out of five, four failed, one shown hair crack with permanent set. (i.e. 80% failure)

Proposed helical compression spring's (AISI 302/304 : Cr 17-19%, Ni 8-10%, Hardness- 45 HRC)also five sample springs tested. Each spring recorded maximum sag within permissible limits. During and after testing no cracks or any breakage found for 3,00,000 cycles, as shown in table 3.

Spring Fatigue Test : To determine number of cycles during cyclic loading.
Test conditions : (Required and actual)

- Working temperature: Room temperature.
- Stroke: 5 mm
- Frequency: 1 cycles/sec
- Cycles to be completed: 3 lac cycles
- Test sample: 5 nos.

Acceptance criteria: During and after testing the spring should not sag or permanent set not to be more than 0.5mm over the original free length or to be within specification. No cracks permitted.

Present spring

Material: AISI 1065 (ASTM A 227): Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC

Table 2: Test report of present spring

Parameters	Free length in mm		Load at height 8.58 mm
Values as per drawing	10.2 mm		0.4kg
Readings before test	1	10.10	0.305
	2	10.30	0.28
	3	9.30	0.3
	4	9.60	0.29
	5	9.80	0.3
Readings after test	1	9.60	0.28
	2	9.90	0.265
	3	8.90	0.275
	4	9.50	0.27
	5	9.30	0.28

Results and Discussion: 5 nos. springs tested for fatigue loading. 4 nos. spring failed with crack. 1 no. spring showed hair cracks with the sag (permanent set) reaching the maximum limit permitted.

Proposed spring

Material: AISI 302/304 (ASTM A 313): Cr 17-19%, Ni 8-10%, Hardness- 45 HRC

Table 3: Test report of proposed spring

Parameters	Free length in mm		Load at height 8.58 mm
Values as per drawing	10.2 mm		0.4kg
Readings before test	1	10.10	0.290
	2	10.25	0.285
	3	10.00	0.31
	4	9.85	0.285
	5	10.15	0.3
Readings after test	1	9.80	0.28
	2	10.00	0.28
	3	9.70	0.295
	4	9.60	0.275
	5	9.90	0.285

Results and Discussion: Five springs tested for fatigue loading. Each spring recorded maximum sag (permanent set) within permissible limits. During and after testing no cracks or any breakage found.

10. 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.
For improving fatigue life following methods can be used.

- Upgrade material to a higher tensile range or a higher quality grade.
- lowering stress.
- Shotpeening
- The surface of the wire is shaved before the final draw to eliminate surface defects.

Hence to enhance fatigue life material is changed
In fatigue analysis using M SC fatigue: Present helical compression spring (AISI 1065 : Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC) shows maximum stress is developed at the inner side of the spring coil and fatigue failure occurs.
To avoid fatigue failure material is changed (AISI 302/304 :Cr 17-19%, Ni 8-10%, Hardness- 45 HRC). It's life is found to be infinite.
In experimentation: Present helical compression spring's (AISI 1065 : Co 45%, Mn 0.6-1.3%, Hardness- 42 HRC) five sample springs tested. It is observed that out of five, four failed, (i.e. 80% failure) one shown hair crack with permanent set.
Proposed helical compression spring's (AISI 302/304 : Cr 17-19%, Ni 8-10%, Hardness- 45 HRC)also five sample springs tested. Each spring recorded maximum sag within permissible limits. During and after testing no cracks or any breakage found. (i.e. no failure)

11. Future scope

Future scope for this study is to study micro-structure, stress corrosion, buckling, coil clash, wear, non-axial forces, and dynamic loading. Shock loading and resonance can seriously reduce cycle life. Also care must be taken in spring surface, fabrication. The material must have good ductility, resilience and toughness to avoid sudden fracture.

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