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Research Article

Study of Dynamic Characteristics of Helical Spring of Variable Wire Diameter

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Abstract

Helical springs are among the most commonly used components in mechanical systems. In order to provide precise spring applications, mechanical design research has been conducted leading to significant improvements in knowledge of springs. The purpose of this paper is to study a dynamic characteristic of springs for which test can be carried out at different speeds and loads for the sprung and unsprung mass with Variable wire diameter and Uniform wire diameter springs on test rig of quarter model of car.

Keywords: *Design, modification of Cam, Heat treatment*

Introduction

Helical springs are among the most commonly used components in mechanical systems. In order to provide precise spring applications, mechanical design research has been conducted leading to significant improvements in knowledge of springs. Recent research concentrates on nonlinear effects, such as small variable pitch angle, end effects, variable pitch etc. The research efforts, directed towards improvement of helical spring by varying radius and pitch angle in context of spring design is facing difficulties such as controlling pitch angle precisely during coiling.

Mechanical systems often use springs to store energy though their axial length must sometimes be significantly reduced. Conical spring provides a commonly used as solution for applications with space constraints. Indeed they are often chosen for their ability to telescope, meaning they take up very little space at maximum compression while storing as much energy as cylindrical spring.

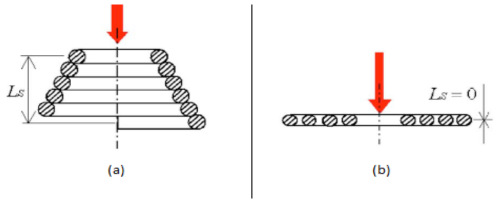

Fig (a) Non-Telescoping Spring (b) Telescoping Spring Conical spring also have their specific features such as having nonlinear load-length characteristic

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The test of spring under dynamic condition is necessary to analyze the performance of spring under dynamic loading condition; this test can be carried out on model test rig representing a car. The model may be full, half or quarter. Looking at the complexity of manufacturing full and half car quarter car model seems more practical and simple. Since it is difficult to conduct tests and take measurements on actual car, by making quarter car model, we can get the performance of spring in terms of force transmissibility, displacement of the unsprung and sprung mass relative to each other etc.

Nonlinear springs enhance the performance of many applications, including prosthetics and artificial implants, micro-electromechanical systems (MEMS) devices, statically balanced mechanisms, designs for crashworthiness, robotic joints and vibration absorption systems. Each nonlinear spring application requires a unique load-displacement function and space constraints. Emmanuel Rodriguez has given a relationship between load and length in nonlinear region of conical spring.

Literature Review

Substantial work has been reported over the past years in the field of the nonlinear vibrations of spring. Some work is listed below:

Sanket Modi studied helical Compression Spring of Varying Wire diameter. Designed and developed test set up for dynamic testing also compared results with spring of uniform wire diameter. Manufactured a spring of tapered wire diameter and equivalent spring of uniform wire diameter. The dynamic testing is carried out on quarter car model. The disturbance is produced by cam. The values of acceleration, velocity and displacement at both the ends of spring are measured by sensors and data acquisition system. Following conclusions are drawn from this study,

1. In case of uniform spring the displacement for the sprung mass is less than that of unsprung mass, and at higher exciting frequency the displacement of sprung mass is larger than the sprung mass.
2. In case of Variable wire diameter spring in its linear zone the sprung mass displacement is larger than the unsprung mass displacement but in nonlinear zone the vibration level is reduced to a significant level. (Sanket Modi *et al*, 2012).

Emmanuel Rodriguez described the behavior of constant pitch conical compression spring. The non-linear phase is determined by discretizing algorithm. Presented analytical continuous expressions of length as a function of load and load as a function of length, for a constant pitch conical compression spring in the nonlinear phase.

Emanuel has presented two models for determining the behavior of a constant pitch conical spring. They were developed to improve currently available conical spring design software. To complete the well-known constant spring rate of behavior linear phase, the proposed models involve two analytical equations for the nonlinear phase, with the first giving spring length as a function of load and the second being the exact inverse by giving load as a function of length. Moreover, the length expression is proved to be written as a polynomial. The new models were successfully confronted with experimental data. The results of this study provide a very fast and precise calculation process. Using it, designers will be able to simply obtain any conical spring characteristic, avoiding painstaking algorithm programming. In answer to our requirement to use load-length relations, the new models will pave the way for the development of an interactive conical spring design synthesis tool based on optimization methods. (Emmanuel Rodriguez *et al*, 2006).

An analytical method has been presented for investigating the free-vibrations of the bending–torsional–shearing coupled cylindrical helical springs. This is the first free-vibration analysis of helical springs that takes into account the warping effect upon the natural frequencies without employing any approximations concerning the warping. The method has been illustrated by its application to the springs for three different cross-sections, and the expressions developed can be further utilized in cylindrical or non-cylindrical helical springs with other cross-sectional shapes. The natural frequencies are presented for a range of geometric parameters. In the case of elliptical wires, results are presented for the aspect ratio l=a/b ranging from 3/5 to 5/3, the helix pitch angle a from 5 to 12.5, the number of active turns n from 6 to 12, and the ratio of cylinder radius to minor axis R/2a ranging from 20/3 to 50/3. Validation of the proposed model has been achieved through comparison with a finite element model. (A.M. Yu *et al*, 2011)

R. Champion developed nonlinear expressions that model the mechanical behavior of a vertically suspended loaded helical spring, operating within the elastic limits of the material. In the absence of applied forces the spring is assumed to form a uniform helix, but both the static extension model and the dynamic model incorporate the natural variation in pitch angle of the spring when it is suspended.

The models are successful in explaining the departure from linear behavior observed in the static extension and the periods of oscillation data for a helical, wire laboratory spring. Independent analyses of the static and dynamic data yield consistent estimates of the spring rate (R. Champion *et al*, 2011).

Sridhar Kota presented comprehensive approach to develop various compliant elements of prescribed nonlinear stiffness. This includes a generalized synthesis methodology for designing a nonlinear spring for a prescribed load-displacement function, while also meeting stress, material, stability, and space constraints. To relax the restrictions on the solution space, scaling guidelines were included within the design process. This enabled the optimizer to focus on the nonlinearities of the problem, i.e., the spring's shape function. Design options for nonlinear springs were illustrated by:

(1) Considering them as building blocks of larger nonlinear structures.

(2) Investigating their scaling potential.

In both cases, it was imperative to preserve the nonlinear behavior of the spring. Using linear beam theory, various scaling guidelines were provided for nonlinear springs enabling existing spring designs to be modified for new design specifications. Two nonlinear spring designs, the J-curve and S-curve, were generated to match the prescribed design specifications. The objective formulation within this paper enabled spring designs to meet exact displacement-ranges and not cross over themselves during deformation. Both designs resulted in one primary spline with a fixed end point. With the addition of the deformed crossover penalty, spring designs are likely to have fewer splines. For future work, we plan to determine the optimal number of splines for the topology parameterization (Sridhar Kota *et al*, 2010).

Manuel Paredes presented comprehensive process for optimal conical spring design. The assistance tool i.e. software is based on the work of related to

cylindrical compression springs. It has been adapted to fit the requirements for conical compression springs. It consists of selecting a design or an operating parameter that must be maximized or minimized. An application example is also mentioned in paper. This example is of electrical contactor. Different values which are entered in software are spring material, wire diameter, large outside diameter, large inside diameter, free coils, fixed coils, percentage travel of spring etc.
The process had main advantage of linking industrial and mathematical knowledge to propose an optimal design that functions directly from global specifications (Manuel Paredes *et al*, 2009).

Faruk Firat Calim investigated dynamic response of helical rods having arbitrary shape under time dependent loads, using an efficient method of analysis in the Laplace domain. The material of the rod is assumed to be homogeneous, linear elastic and isotropic. The curvature of the rod axis, effect of the rotary inertia, axial and shear deformation are considered in the formulations. Ordinary differential equations in scalar form obtained in the Laplace domain are solved using complementary functions method. The forced vibrations of non-cylindrical helical rods are analyzed through various examples. This brings great convenience in the solution of the physical problems having general boundary conditions. Another advantage of using the complementary functions method-based solution is that the helical rods with variable cross-section and geometry, which yield ordinary differential equations having variable coefficients, can also easily be handled. The differential equations can be solved by using the complementary functions method as accurate as required with an appropriate integration step-size. In the present method, fifty elements were used to achieve the desired accuracy as opposed to 1000 elements needed in ANSYS. The dynamic stiffness matrix was calculated in the Laplace domain by applying the complementary functions method to the differential equations in canonical form. This brought great convenience in the solution of the physical problems having general boundary conditions (Faruk Firat Calim *et al*, 2009).

Spring Design and Test Rig Modification

Spring Design

In this total experimentation, procedure that followed for spring design is, first deciding the parameters of helical spring and then for conical spring. For deciding this parameters, constraints that are taken into consideration are as follows:

1. Height between sprung and unsprung mass.
2. Lowest diameter of spring that can accommodate in damper.

Conical springs

A helical spring of specifications given in fig. is manufactured.

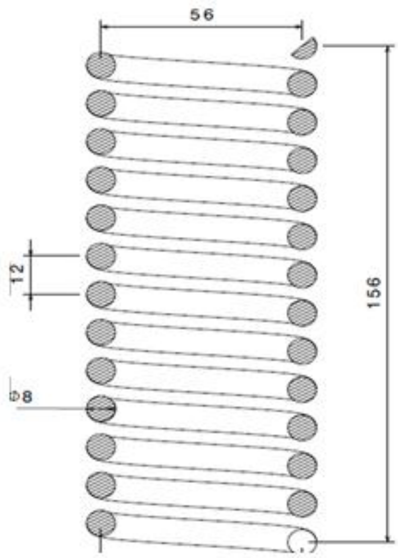

Fig Equivalent spring

Depending on the mean diameter (56 mm) of above spring 3 conical springs are designed. In all the springs wire diameter is same i.e. 8 mm.

Table Conical Spring Dimensions

Spring No.	D1 (mm)	D2(mm)	Cone Angle
0	56	56	0
1	50	62	5.47
2	44	68	9.28
3	41	71	12

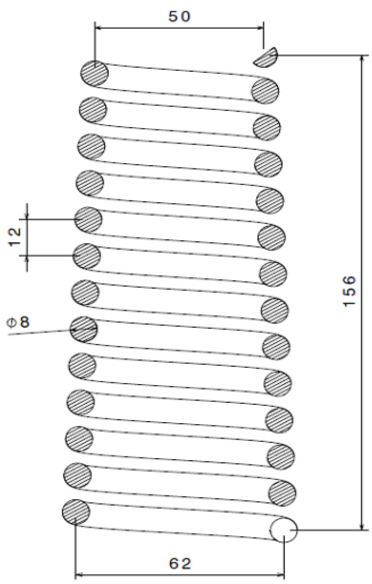

Fig Spring with cone angle 5.47^0

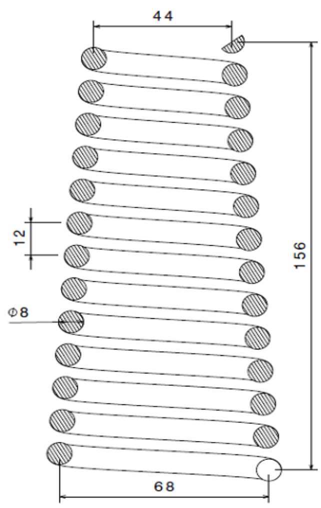

Fig Spring with cone angle 9.28^0

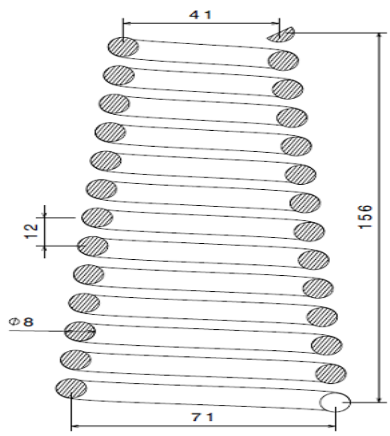

Fig Spring with cone angle 12^0

Taper Wire Spring

Following parameters are taken into consideration for the manufacturing of spring.
Mean spring diameter = 60 mm
End diameters of spring wire = d_1 =12.5 mm
d_2 = 5.5 mm
Total number of coils = 8
Number of active coils = 6
Pitch = 25 mm
Free length = 165 mm
End effect = end coils are ground

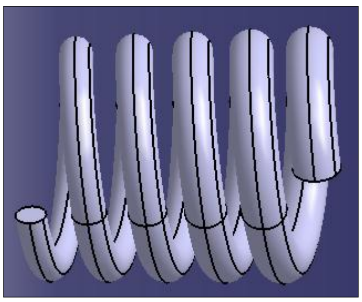

Fig Tapered wire spring

Outer diameter is maintained constant. For this wire is wounded on taper mandrel. Equivalent helical spring is also manufactured of following dimensions.
Mean spring diameter = 60 mm.
Spring wire diameter = d =9 mm.
Total number of coils = 8.
End effect = end coils are ground.
Free length = 165 mm.
Number of active coils = 6.
Pitch = 25 mm

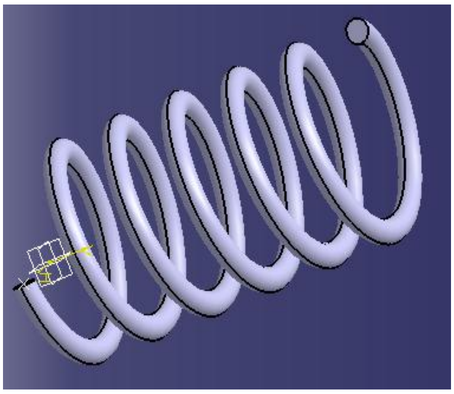

Fig Equivalent Spring

Commonly used spring materials

One of the important considerations in spring design is the choice of the spring material. Some of the common spring materials are given below.

Hard-drawn wire: This is cold drawn, cheapest spring steel. Normally used for low stress and static load. The material is not suitable at subzero temperatures or at temperatures above 1200⁰C.

Oil-tempered wire: It is a cold drawn, quenched, tempered, and general purpose spring steel. However, it is not suitable for fatigue or sudden loads, at subzero temperatures and at temperatures above 1800⁰C. When we go for highly stressed conditions then alloy steels are useful.

Chrome Vanadium: This alloy spring steel is used for high stress conditions and at high temperature up to 2200⁰C. It is good for fatigue resistance and long endurance for shock and impact loads.

Chrome Silicon: This material can be used for highly stressed springs. It offers excellent service for long life, shock loading and for temperature up to 2500⁰C.
Music wire: This spring material is most widely used for small springs. It is the toughest and has highest tensile strength and can withstand repeated loading at high stresses. However, it cannot be used at subzero temperatures or at temperatures above 1200⁰C. Normally when we talk about springs we will find that the music wire is a common choice for springs.

Stainless steel: Widely used alloy spring materials.
Phosphor Bronze / Spring Brass: It has good corrosion resistance and electrical conductivity. That's the reason it is commonly used for contacts in electrical switches. Spring brass can be used at subzero temperatures

From the above materials and as per availability the spring steel of grade EN 47 is used.

Spring manufacturing processes

If springs are of very small diameter and the wire diameter is also small then the springs are normally manufactured by a cold drawn process through a mangle. However, for very large springs having also large coil diameter and wire diameter one has to go for manufacture by hot processes. First one has to heat the wire and then use a proper mandrel to wound the coils. There are two process of spring manufacturing

1. Cold Rolling: Helical compression spring having diameter up to 12 mm is cold- wound.
2. Hot rolling: Helical compression spring having diameter larger than 12 mm to 16 mm are either cold-wound or hot-wound.

As average spring wire diameter is 9 mm the spring is cold rolled.

Heat Treatment of spring steel

After the formation of spring it is necessary to be heat treated. Generally annealing process is done on the spring to reduce the residual stresses induced in the spring which form during the coiling of spring. Pre-stressing is necessary to avoid permanent setting in spring.

Test Rig

The set-up is developed by considering sprung and unsprung masses and a unit to provide sinusoidal input to unsprung mass. This unit consists of a cam and follower mechanism. The arrangement is shown below:

Fig Test Rig

In this test rig linear bearings are used which guides on the vertical rods, in turn they are transmitting the rotary motion of cam into to and fro motion. This to and fro motion will excite the lower C-channel and amplitude of the disturbance is same as that of rise of the cam. Also there is an arrangement for load adjustment at the top which will apply uniform load on spring.

Disturbed Input

The disturbed input is given by eccentric cam as shown in fig. an eccentric cam is actually a shaft and a disc arrangement. Circular disk with eccentricity of 2.5mm is used for Sinusoidal excitation to the spring-mass system. The cam used is having specifications as follow,
Circular disk diameter = 130 mm
Eccentricity = 2.5 mm
Bore diameter = 36 mm

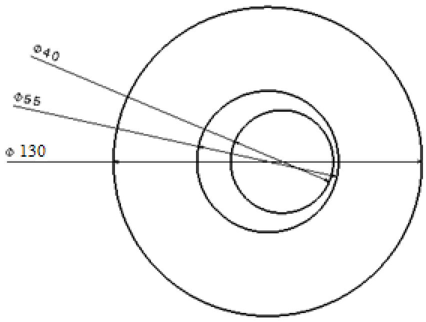

Fig Eccentric Cam

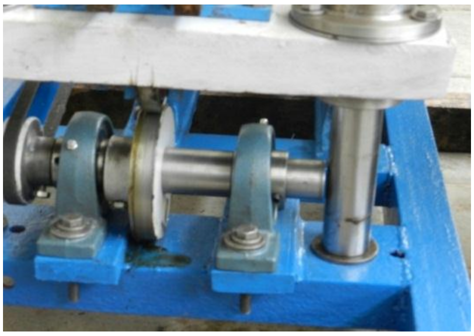

Fig Photograph of Previously used Cam and follower mechanism Sinusoidal motion given by Cam,

$$y = \mathbf{a}\ \mathbf{sin}(\omega t)$$

Where,
a=2.5
Displacement v/s time plot is given below:

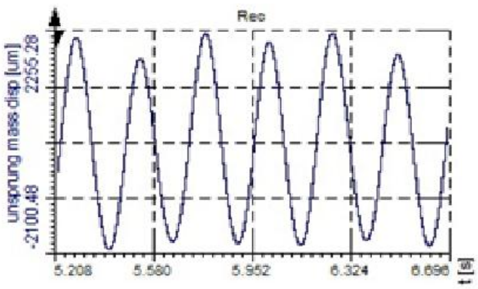

Fig Graph for input disturbance

The peaks are not steady for higher rotating speeds. This cam design becomes obsolete for further use, so there arises a need of modification in cam which will give exact sinusoidal input.

Dynamic Analysis

The dynamic testing of spring-mass system is carried out on manufactured test set up. The spring-damper system with top mass is attached between two c-channel as shown in fig. The acceleration plots of spring mass system are carried out at different speeds of cam. With the help of mathematical operations in DEWESoft software velocity and displacement graphs are also plotted. The speeds are so chosen that they are either above or below natural frequency of spring-mass system.

The dynamic testing of spring is done in both linear and nonlinear zone. The top mass is adjusted so as to get deflection of spring in linear and nonlinear zone. As mentioned above the readings are taken at speeds which are away from the natural frequency of the spring mass system being tested.

In both the cases dynamic testing of linear spring is done on same test set up. Since the results of tests are to be compared with nonlinear spring the operating condition were required to be same. The attachment of spring mass system is as shown in figure below,

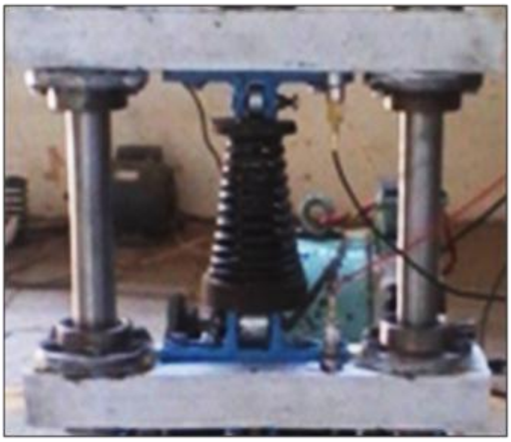

Fig Photograph of attachment of spring

Data acquisition system is used for dynamic measurement uses the software of DEWESoft company.

Sensor and Weight Attachment on Test Set Up

The sensors (Accelerometers) are attached to the C-channels (Sprung-Unsprung) masses. The accelerometers are used to measure the displacements of sprung unsprung mass in time domain and frequency domain which are then plotted using Dewesoft Software.

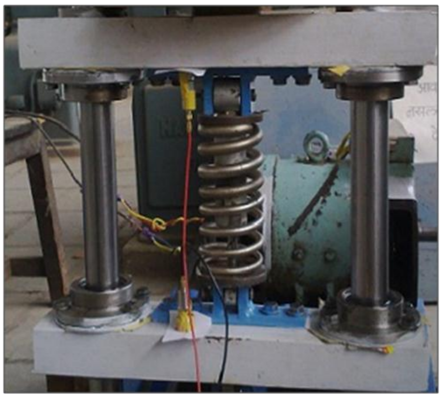

Fig Photograph of attachments of sensors

DATA Acquisition System

For the time-displacement plots of sprung unsprung mass the data acquisition system used is DEWE-soft 43, which has DEWE43 7.0 data acquisition system software.

Fig Data Acquisition System

Two accelerometers are used for the testing, which mounted on both channels. 43V data acquisition system has 8 analog channels and 8 digital channels as shown in above figure.

Fig DEWE 43 V Data acquisition system

Fig Accelerometer

The accelerometers are connected to the data acquisition system with help of connecter BR-ACC. Details regarding dynamic testing of spring are described in this chapter. The test is carried out at different exciting frequencies at different mass applied on the spring. The response for input is plotted. The readings were taken for both spring i.e. variable wire diameter and uniform wire diameter spring.

Selection of Various Experimental Parameters

1. Selection of Amplitude of Excitation

Velocity of Vehicle = 70 Km/hr.
= 19.44 m/s.
Excitation Frequency = 200 rpm.
Power Spectral Density (PSD) = $\frac{C*V^{(N-1)}}{f^N}$
Where,

C = Roughness coefficient (m).
N = Wave number.
Rough Runway is chosen, corresponding values for C and N are as follows,
C = 8.1 * $10^{(-6)}$
N = 2.1
So,
PSD = 1.69 * $10^{(-5)}$
Amplitude =$\sqrt{2} * 1000 * \sqrt{(FRV * PSD * \sqrt{2})}$
Where,
FRV = 0.305
∴ Amplitude = 3.82 mm
∴ So rise of the cam is chosen as 4 mm.

2. Excitation Frequencies

Excitation frequencies are chosen as follow,
Velocity of Vehicle (V) = 70 Km/hr.
= 19.44 m/s.
Frequency of the base excitation is a function of the vehicle speed and road roughness.

$$N = \frac{V}{\lambda}$$

Where,
N = Excitation Frequency (rps).
λ = Wavelength of road roughness.
From Table type of irregularity considered is Macro texture, the range of wavelength is 0.5-5 mm.
Since the lower and higher excitation frequencies are chosen on the basis of stability of system while operating. When excitation frequency goes beyond 500 rpm the system starts vibrating very violently

Sr. No.	Wavelength (mm)	Excitation Frequency (rpm)
1	1.08	350
2	1.23	400
3	1.38	450
4	1.54	500

3. *Load Steps*

Four load steps are chosen for the experiment. Out of these first two load steps are in linear zone springs and last two are in non-linear zone of springs.
For Conical Springs following load steps are chosen,
Load step 1 = 22 kg.
Load step 2 = 34 kg.
Load step 3 = 45 kg.
Load step 4 = 56 kg.
For Taper wire Springs following load steps are chosen,
Load step 1 = 22 kg.
Load step 2 = 34 kg.
Load step 3 = 45 kg.
Load step 4 = 56 kg.

Results and Discussion

The test can be carried out at different speeds and loads. For the sprung and unsprung mass displacement can be plotted in time domain and frequency domain, for both the springs i.e. Variable wire diameter and Uniform wire diameter springs. Readings will be taken below and above the natural frequency of system.

Results for Conical Springs will be

1) As the load increases displacement of upper C channel decreases.
2) For low loading condition difference between taper wire spring and equivalent spring is minimum but at high loading condition that difference increases that means taper wire spring transmit very less displacement as the load go on increasing. 3) At low loading conditions acceleration for both the spring is same.

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