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Fabrication and analysis of spring testing machine

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Abstract

The helical spring is the most common element that has been used in automobile industries. In this work the spring constant and deflection of the spring at different loads have been calculated. A closed coil helical spring is used for calculating tension and open coil helical spring is used for compression. The component used for the fabrication of the spring testing machine are frame, hydraulic, spring weighing machine, open coiled and closed coiled helical spring etc. In this machine the combined testing of both tension and compression can be done. The data obtained from the machine the compared with the manually calculated data and all the result obtained are in good agreement.

Key words: Helical Tension spring, Deflection, Spring Constant, Loads

1. Introduction

The appropriate equations determining the effectiveness of dynamic stress reduction in resonant conditions as a function of coating parameters were derived. It was proved that rubber coating will not perform in satisfactory manner due to its low modulus of elasticity in shear. It was also demonstrated that about resonance areas of increased stresses are wider and wider along with the successive resonances and achieve significant values even at large distances from the resonance frequencies [1]. Long-term fatigue tests on shot peened helical compression springs were conducted by means of a special spring fatigue testing machine at 40 Hz. Test springs were made of three different spring materials oil hardened and tempered SiCr- and SiCrV-alloyed valve spring steel and stainless-steel. With a special test strategy in a test run, up to 500 springs with a wire diameter of d = 3.0 mm or 900 springs with d = 1.6 mm were tested simultaneously at different stress levels. Based on fatigue investigations of springs with d = 3.0 mm up to a number of cycles $N = 10^9$ an analysis was done after the test was continued to $N = 1.5 \times 10^9$ and their results were compared. The influence of different shot peening conditions were investigated in springs with d = 1.6 mm. Fractured test springs were examined under optical microscope, scanning electron microscope (SEM) and by means of metallographic micro sections in order to analyze the fracture behavior and the failure mechanisms [2]. Elastomeric components have wide usage in many industries. The typical service loading for most of these components is variable amplitude and multi axial. In this study a general methodology for life prediction of elastomeric components under these typical loading conditions was developed and illustrated for a passenger vehicle cradle mount. Crack initiation life prediction was performed using different damage criteria. The methodology was validated with component testing under different loading conditions including constant and variable amplitude in-phase and out-of-phase axial–torsion experiments. The optimum method for crack initiation life prediction for complex multiaxial variable amplitude loading was found to be a critical plane approach based on maximum normal strain plane and damage quantification by cracking energy density on that plane. Rain flow cycle counting method and Miner's linear damage rule were used for predicting fatigue life under variable amplitude loadings [3]. Very high cycle fatigue (VHCF) properties of newly developed clean spring steel were experimentally examined under rotating bending and axial loading. As a result, this steel represents the duplex S–N property only for surface-induced failure under rotating bending, whereas it represents the single S–N property for surface-induced failure and interior inhomogeneous micro structure induced failure under axial loading. The surface morphology of the interior inhomogeneous microstructure with distinct plastic deformation is much rougher than that of the ambient matrix, which means the stress concentration resulted from the strain inconsistency between the micro structural in

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homogeneity as soft phase and the ambient matrix as hard phase plays a key role in causing interior crack initiation [4]. A 3D geometric modelling of a twin helical spring and its finite element analysis to study the spring mechanical behavior under tensile axial loading. The spiraled shape graphic design is achieved through the use of Computer Aided Design (CAD) tools, of which a finite element model is generated. Thus, a 3D 18-dof pentaedric elements are employed to discrete the complex ‘wired-shape” of the spring, allowing the analysis of the mechanical response of the twin spiraled helical spring under an axial load. The study provides a clear match between the evolution of the theoretical and the numerical tensile and compression normal stresses, being of sinusoidal behavior. On the other hand, the minimum stress level is located in the center of the filament cross section [5]. The paper gives an overview of the present state of research on fatigue strength and failure mechanisms at very high number of cycles ($N_f > 10^7$). Testing facilities are listed. A classification of materials with typical S–N curves and influencing factors like notches, residual stresses and environment are given. Different failure mechanisms which occur especially in the VHCF-region like subsurface failure are explained. There micro structural in homogeneities and statistical conditions play an important role. A double S–N curve is suggested to describe fatigue behavior considering different failure mechanisms. Investigated materials are different metals with body-centered cubic lattice like low- or high strength steels and quenched and tempered steels but also materials with a face-centered cubic lattice like aluminum alloys and copper [6]. Ever since high-strength steels were found to fail below the traditional fatigue limit when loaded with more than 10^8 cycles, the investigation of metals’ and alloys’ very high cycle fatigue properties has received increased attention. A lot of research was invested in developing methods and machinery to reduce testing times. This overview outlines the principles and testing procedures of very high cycle fatigue tests and reports findings in the areas of crack formation, non-propagating small cracks, long crack propagation and thresholds. Furthermore, superimposed and variable amplitude loading as well as frequency effects are reported [7]. A stranded wire helical spring (SWHS) is a unique cylindrically helical spring, which is reeled by a strand that is formed of 2∼16 wires. In this paper, a parametric modelling method and the corresponding 3D model of a closed-end SWHS are presented based on the forming principle of the spring. By utilizing a PC + PLC based model as the motion control system, a prototype machine tool is designed and constructed, which improves the manufacturing of the SWHS. Via the commercial CAD package Pro/Engineering, numerical simulation is carried out to test the validity of the parametric modeling method and the performance of the machine tool. The scheme of the tension control system is analyzed and the control mechanism is set up, which have achieved the constant tension of each wire. A human machine interface is also proposed to achieve the motion control and the tension control. Experimental results show that the tension control system is well-qualified with high control precision [8].An adjustable-stiffness actuator composed of two antagonistic non-linear springs is proposed in this paper. The elastic device consists of two pairs of leaf springs working in bending conditions under large displacements. Owing to this geometric non-linearity, the global stiffness of the actuator can be adjusted by modifying the shape of the leaf springs. A mathematical model has been developed in order to predict the mechanical behavior of our proposal. The non-linear differential equation derived from the model is solved, obtaining large stiffness variations [9]. The characterization of vibration-fatigue strength is one of the key parts of mechanical design. It is closely related to structural dynamics, which is generally studied in the frequency domain, particularly when working with vibratory loads. Fatigue-life estimation in the frequency domain can therefore prove advantageous with respect to a time-domain estimation, especially when taking into consideration the significant performance gains it offers, regarding numerical computations. Several frequency-domain methods for a vibration-fatigue-life estimation have been developed based on numerically simulated signals. This research focuses on a comparison of different frequency-domain methods with respect to real experiments that are typical in structural dynamics and the automotive industry. The methods researched are: Wirsching–Light, the a0.75 method, the experimental comparison researches the resistance to close-modes, to increased background noise, to the influence of spectral width, and multi-vibration-mode influences. Additionally, typical vibration profiles in the automotive industry are also researched. For the experiment an electro-dynamic shaker with a vibration controller was used. The reference-life estimation is the rain flow- counting method with the Palmgren–Miner summation rule [10]. High strength steel grade 51CrV4 in thermo-mechanical treated condition is used as bending parabolic spring of heavy vehicles. Several investigations show that fatigue threshold for very high cycle fatigue depends on inclusion’s size and material hardness. In order to determine allowed size of inclusions in spring’s steel the Murakami's and Chapetti's model have been used. The stress loading limit regarding to inclusion size and applied stress has been determine for loading ratio R=-1 in form of S-N curves. Experimental results and prediction of S-N curve by model for given size of inclusion and R ratio show very good agreement. Pre-stressing and shot-penning cause higher compress stress magnitude and consequently change of loading ratio to more negative value and additionally extended life time of spring [11]. A spring is basically an elastic body whose purpose is to detect or distort under loading conditions and consequently store energy and release it slowly or rapidly depending on the particular application. In 1932, Lucien Lacoste invented the zero-length spring. A zero length spring has a physical length equal to its stretched length. Its force is proportional to its entire length, not just the stretched length and is therefore constant over the range of flexures in which the spring is elastic [12]. Springs are usually made from alloys of steel. Extension and compression springs are literally on opposite sides of the spring spectrum. Extension springs are used primarily to hold two components together, while compression springs are best for keeping components from meeting one another. Both employ a coil design for elasticity and strength, but they work under two different principles of elastic potential energy. Torsion spring provide torque around the axis of the helix, rather than a force in line with the axis of the helix, as in compression and extension springs [13]. Hooke's law of elasticity is an approximation which state that the amount by which a material body is deformed (strain) is linearly related to the force causing the deformation (stress). The materials for which Hooke's law is a useful approximation are known as linear- elastic or "Hookean" materials. For systems that obey Hooke's law, the extension, x produced is proportional to the load [14]. There are three basic principles in spring design: The heavier the wire, the stronger the spring, the smaller the coil, the stronger the spring and the more active the coils, the less load you will have to apply in order to get it to move a certain distance [15]. Hydraulic refers to pressurized fluid, which can be oil, water or other liquids. It employs these to transmit energy from an energy generating source to areas where it is needed. In general, any application that requires a large force to be applied smoothly by small linear or rotary displacement unit (actuator) needs hydraulic and powered fluid technology. Pump converts mechanical energy into hydraulic energy [16]. The pseudo elastic response of shape memory alloy (SMA) helical springs under axial force is studied both analytically and numerically. In the analytical solution two different approximations are considered. In the first approximation, both the curvature and pitch effects are assumed to be negligible. This is the case for helical springs with large ratios of mean coil radius to the cross sectional radius (spring index) and small pitch angles. Using this assumption, analysis of the helical spring is reduced to that of the pure torsion of a straight bar with circular cross section. A three dimensional phenomenological macroscopic constitutive model for polycrystalline SMAs is reduced to the one-dimensional pure shear case and a closed-form solution for torsional response of SMA bars in loading and unloading is obtained. In the next step, the curvature effect is included and the SMA helical spring is analyzed using the exact solution presented for torsion of curved SMA bars [17]. This paper is a discussion about automotive suspension coil springs, their fundamental stress distribution, materials characteristic, manufacturing and common failures. An in depth discussion on the parameters influencing the quality of coil springs is also presented. Following the trend of the auto industry to continuously achieve weight reduction, coil springs are not exempt. A consequence of the weight reduction effort is the need to employ spring materials with significantly larger stresses compared to similar designs decades ago. Utilizing a higher strength of steel possesses both advantages and motivated by a specific failure mode known as lateral wire buckling occurring in the tensile armor layers of flexible pipes. The tensile armor is usually constituted by two layers of initially helically wound steel wires with opposite lay directions. During pipe laying in ultra-deep waters, a flexible pipe experiences repeated bending cycles and longitudinal compression [18]. An automotive suspension system is designed to provide both safety and comfort for the vehicle occupants. In this study, finite element models were developed to optimize the material and geometry of the composite elliptical spring based on the spring rate, log life and shear stress parameters. The influence of the ellipticity ratio on the performance of woven roving–wrapped composite elliptical springs was investigated both experimentally and numerically. The study demonstrated that composite elliptical springs can be used for light and heavy trucks with substantial weight reduction. The results showed that the ellipticity ratio significantly influenced the design parameters. Composite elliptic springs with ellipticity ratios of a/b = 2 had the optimum spring parameters [19]. Long-term fatigue tests on shot penned helical compression springs were conducted by means of a special spring fatigue testing machine at 40 Hz. Test springs were made of three different spring materials – oil hardened and tempered SiCr and SiCrV-alloyed valve spring steel and stainless steel. With a special test strategy in a test run, up to 500 springs with a wire diameter of d = 3.0 mm or 900 springs with d = 1.6 mm were tested simultaneously at different stress levels. Based on fatigue investigations of springs with d = 3.0 mm up to a number of cycles $N = 10^9$ an analysis was done after the test was continued to $N = 1.5x10^9$ and their results were compared. The influence of different shot peening conditions were investigated in springs with d = 1.6 mm. Fractured test springs were examined under optical microscope, scanning electron microscope (SEM) and by means of metallographic micro sections in order to analyze the fracture behavior and the failure mechanisms. The paper includes a comparison of the results of the different spring sizes, materials, number of cycles and shot peening conditions and outlines further investigations in the VHCF-region [20, 21]. A pioneering theory for the description of the maximum bending and shear stresses on spring coils. He was the first to realize that the standard formulas for deflection and force exerted by springs were inaccurate because the stress distribution in the spring, due to pure torsion, did not have a linear relationship with the distance from the wire axis to the spring fiber. The so called Ro-ever effect indicates that additional stresses appear due to the curvature, coil pitch angle and wire cross section, and a stress correction factor must be applied [22]. A stress correction factor to take into account those additional stresses using torsion theory and carried out tests that were in good agreement with the values theoretically predicted[23]. A new correction factor that included the displacement of the actual center of rotation of the fibers from the geometrical center of the cross section, carrying out tests to confirm his predictions [24]. Control to the different parts but variation in damping coefficient is having much effect on vibration amplitudes. From the results it has seen that the transmission of vibration to the upper part is of maximum amplitudes. The amplitude of vibration is minimum to the tire. It has been observed from the results that the bump amplitude to the road should be of optimum value. Random vibration analysis can be further done of a road vehicle is investigated using different car models which are quarter car model, bicycle car model, and half car model. Computer programs in Mathematical are developed for all car models. To understand the base excitation response behaviors of the sprung mass in all car models, firstly deterministic vibration analysis are carried out and the results are presented by graphs. This graphs show the vibration amplitudes vs time under the excitation frequencies which are near to and far from the natural

2. Problem Description

In this work the spring constant and deflection of the spring at different loads have been calculated. A closed coil helical spring is used for calculating tension and open coil helical spring is used for compression. The component used for the fabrication of the spring testing machine are frame, hydraulic, spring weighing machine, open coiled and closed coiled helical spring etc. The combined testing of both tension and compression can be done as shown in figure 1

(a)

(b)

Figure 1. (a) Machine before completion, (b) Machine after completion.

3. Result and Discussion

The deflection and spring constant due to compression is shown in table 1.Similarly, the deflection and spring constant due to tension is shown in table 2. Further, combined effect due to compression and tension is shown in table 3.

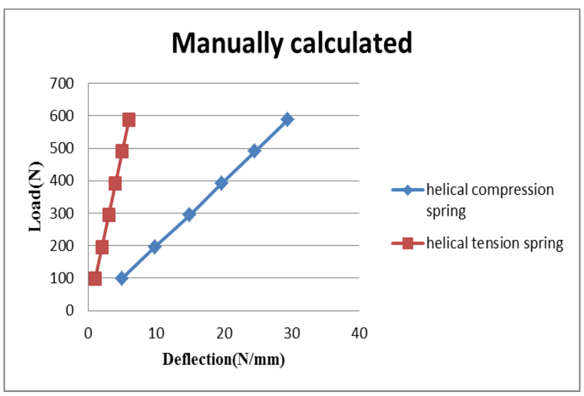
Table 1:- Deflection and spring constant of helical compression spring

Load (N)	Manually Calculated Data			Experimental Data		
	Deflection (mm) Compression	Spring Constant (N/mm)	Average Value	Deflection (mm) Compression	Spring Constant (N/mm)	Average Value
98.1	4.92	19.93	19.818	4.8	20.43	20.29
196.2	9.82	19.97		9.5	20.65	
294.3	14.98	19.64		14.8	19.88	
392.4	19.72	19.91		19.3	20.33	
490.5	24.63	19.94		24.5	20.02	
588.6	29.51	19.95		28.9	20.36	

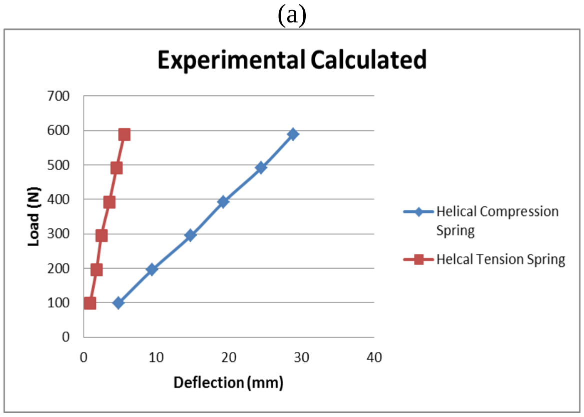
Table 2:- Deflection and spring constant for helical tension spring

Load (N)	Manually Calculated Data			Experimental Data		
	Deflection (mm) Tension	Spring Constant (N/mm)	Average Value	Deflection (mm) Tension	Spring Constant (N/mm)	Average Value
98.1	0.95	103.26		0.89	121.11	
196.2	1.90	103.26		1.78	110.22	
294.3	2.68	109.81	104.17	2.45	120.12	112.53
392.4	3.81	102.99		3.55	110.53	
490.5	4.77	102.83		4.53	108.27	
588.6	5.72	102.90		5.61	104.91	

Similar loads were applied in all three categories at different levels. The minimum and the maximum loads applied for testing is 98.1N and 588.6N respectively. The minimum and the maximum deflection given by the machine due to compression are 4.8mm and 28.91mm respectively and for tension these are 0.89mm and 5.61mm respectively.



(a)



(b)

Figure 2:- Variation between Load and Deflection for helical compression spring (a) Manually, (b) Experimentally

The figure 2 (a) and (b) shows he graph drawn between the load and the deflection of the spring. The figure (a) shows the graph plotted for the manually calculated values of the helical tension and compression spring, whereas (b) shows the experimental values.

3.1 Combined Testing of spring

In the combined testing, both tension and compression springs were used simultaneously in the machine for testing as shown in figure 1. The tension spring was fixed in the upper part and the compression spring was fixed in the lower part and different results were obtained by applying different loads at different levels as shown in table 3.

Table 3:- Combined testing

Experimental Data (Combined testing)						
Load	Deflection (mm)		Spring Constant (N/mm)			
	Tension	Compression	Tension	Average Value	Compression	Average Value
98.1	0.87	4.71	112.75		20.82	
196.2	1.75	9.50	112.11		20.65	
294.3	2.44	14.75	120.61	111.81	19.95	
392.4	3.52	19.12	111.47		20.52	
490.5	4.52	24.41	108.51		20.09	
588.5	5.58	28.86	105.46		20.53	20.42

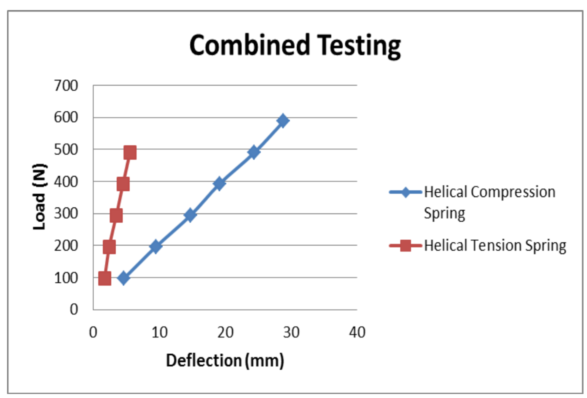


Figure 3- :-Variation between load and Deflection for helical compression spring

Figure 3 shows the combined testing of the spring under the same load applied for the tension and compression individually. The load is on the Y-axis and the deflection obtained is on the X-axis. In the combined testing of the spring, the deflection obtained is in between the experimental value of the compression and the tension as the load applied is same.

4. Conclusion

The spring constant and deflection of the spring at different loads have been calculated. A closed coil helical spring is used for calculating tension and open coil helical spring is used for compression. In this machine the combined testing of both tension and compression can be done. The data obtained from the machine the compared with the manually calculated data and all the result obtained are in good agreement.

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