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Optimal re-design of helical springs using fuzzy design and FEM

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ABSTRACT

The aim of this paper is to present results of using fundamental machine element design principles into re-designing optimally heavy duty springs used in terrain machinery and in industry. Use of standard procedures often results in recurring fatigue fracture failures. This reveals need for correcting also the present standards. Main causes of failures are the local bending due to eccentric highly impact force application at squared and ground ends and wearing away of the shot peening protection. Optimum design of the spring is obtained. Goals are minimisation of wire volume, space restriction, desired spring rate, avoidance of surging frequency and achieving reliably long fatigue life. Conclusions are verified by using full 3D solid FEM analysis with MSC Nastran by which the stresses and also strains, deformations and natural frequencies and modes are obtained.

1. Introduction

Background for this study is observation that conventionally designed helical springs did not have the expected long fatigue life time promised by standards. Analyses of many case studies have given clue that some overlooked effect may have contributed strongly.

The goals in this study are the following. First the main mechanisms causing failure are identified and re-design ideas are generated. Then optimal spring is found using models presented in texts of Hernandez et al. [1], Norton [2], Shigley and Mischke [3] and Spotts et al. [4]. The goals are formulated to maximise fuzzy satisfaction on performance of the spring, as in [11].

2. Materials and methods

2.1. Object of study

The object of study is a range of helical compression springs which are used in heavy duty application with very high life reliability requirements.

Ground ends cause local bending moment maximum as illustrated in Fig. 1. Definitions of spring variables and fuzzy satisfaction functions are shown in Fig. 2. The fuzzy function p_x has max height unity, but area is not unity.

2.2. Design goal formulation

2.2.1. The overall design goal

The overall goal is to maximise the satisfaction $P(G)$ of end user customer on the realised design event G. It is a union of partial design events. The "cost event" means now volume of the spring wire. Safety factor approach is used. The space for the spring is restricted in width and height.

$$\begin{aligned} G = & G_1(xx(1) = \text{cost}) \bullet G_2(xx(2) = N_{\text{goodm}}) \bullet G_3(xx(3) = N_{\text{misch}}) \bullet \\ & G_4(xx(4) = N_{\text{spott}}) \bullet G_5(xx(5) = N_{\text{taual}}) \bullet G_6(xx(6) = N_{\text{lifeA}}) \bullet \\ & G_7(xx(7) = f_{\text{surge}}) \bullet G_8(xx(8) = k_{\text{spring}}) \bullet G_9(xx(9) = \text{height}) \end{aligned} \quad (1)$$

Total satisfaction of this design event is product of partial functions

$$P(G) = P(G_1) \bullet P(G_2) \bullet P(G_3) \bullet \cdots \bullet P(G_7) \bullet P(G_8) \bullet P(G_9) \quad (2)$$

2.2.2. Design satisfaction functions

These are defined on four points to give a trapezoidal form, Fig. 2. Value of property, xx(IG) with index IG is on horizontal scale and satisfaction p_x on xx is on vertical scale, ranging from 0 no good to 1 unit or fully good.

2.2.3. Discrete variables and optimum search

Optimum strategy was done by exhaustive search loops. Preliminary choices are: (a) Choice of assuming (Isp = 1) no shot peening or assuming it (Isp = 2), (b) choice of impact factor V, (c) choice of estimating some volume fraction of inclusions.

1. Loop for material selection Im = 1 to N_{im},
2. Loop for helix diameter D (Idd) variation,

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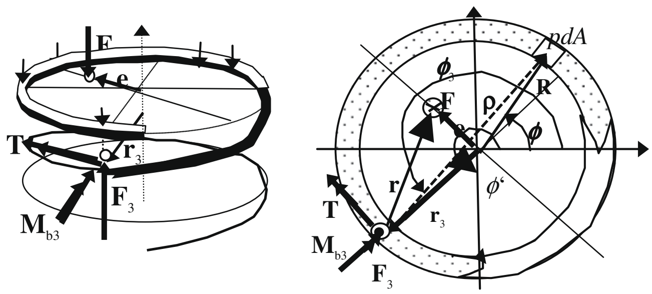

Fig. 1. Definitions of a helical spring with ground ends.

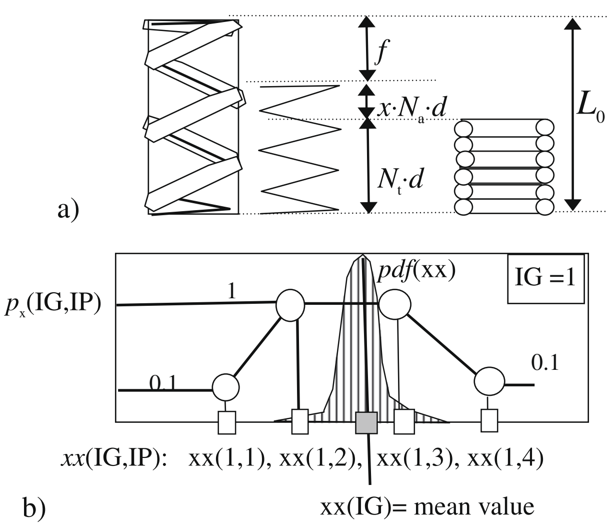

Fig. 2. Definitions. (a) Definition of spring variables. (b) Definitions of a satisfaction function. Probability density function of a selected variable.

3. Loop for wire diameter d(Id) variation,
4. Loop for total number of coils N_t(INtot) variation.

Inside the loops the best optimum choice is made.

2.2.4. Material property data for optimisation

Now a reasonable selection for materials is ASTM A232 chrome Vanadium steels AISI 6150. According to Norton [2] it is suitable for fatigue loading. It is good for shock and impact loads for temperatures to 220 °C. Maximum wire diameter recommendation is 12 mm. Shear modulus is G = 79,000 MPa. Ultimate tensile strength S_{ut} depends on the diameter d of the wire

$$R_m = S_{ut} = \frac{A}{d^m}, \quad R_m \text{ [MPa]}, \quad d \text{ [mm]}, \quad A = 1880 \text{ [MPa]}, \quad m = 0.192 \tag{3}$$

Many other strength values are derived from S_{us}. Here R_e is yield strength in tension, S_{ys} is yield strength in torsion and S_{us} is ultimate strength in shear

$$R_e = 0.75S_{ut}, \quad S_{ys} = (0.577\ldots0.6)S_{ut}, \quad S_{us} = 0.67S_{ut} \tag{4}$$

2.2.5. Load stresses

Load to the spring comes from a cam mechanism. Nominal shear stress depends on load force

$$\tau_n = T_{nF} \cdot F = \frac{8}{\pi}\frac{D}{d^3} \cdot F, \quad T_{nF} = \frac{8}{\pi}\frac{D}{d^3} \tag{5}$$

Shear stress is maximal in the inner coil since it has the smallest curvature

$$\tau_{xy,F} = K_w\tau_n = K_wT_{nF} \cdot F = T_F \cdot F, \quad T_F = K_wT_{nF} \tag{6}$$

Here the correction factor K_w of nominal shear stress is

$$K_w = \frac{4C-1}{4C-4} + \frac{0.615}{C}, \quad C = \frac{D}{d} \tag{7}$$

Spring force F and spring rate k are

$$F = \frac{Gd^4}{8N_aD^3}f, \quad F = kf \tag{8}$$

Shear stress dependence on deflection is

$$\tau_{xy,F} = K_k \cdot f = K_w\frac{1}{\pi}\frac{d \cdot G}{D^2N_a} \cdot f \tag{9}$$

$$K_k = K_w\frac{1}{\pi}\frac{d \cdot G}{D^2N_a} \tag{10}$$

The springs are generally pre-stressed with deflection $f = f_{pre}$. Adding this to f_{cam} gives maximum deflection

$$f_{pre} = f_{min}, \quad f_{cam} = \Delta h, \quad f_{max} = f_{min} + f_{cam} \tag{11}$$

Shear stresses are by Eq. (6)

$$\tau = K_kf \Rightarrow \tau_{min} = K_kf_{min} = \tau_i, \quad \tau_{max} = K_kf_{max} \tag{12}$$

The mean and amplitude shear stresses are

$$\tau_{mean} = \frac{1}{2}(\tau_{max} + \tau_{min}), \quad \tau_{ampl} = \frac{1}{2}(\tau_{max} - \tau_{min}) \tag{13}$$

2.2.6. Properties evaluated by satisfaction functions

The relevant properties of springs for user are cost and endurance. Endurance is evaluated as fatigue life in crack initiation. Some common fatigue diagrams are shown in Fig. 3.

The following nine properties are selected as relevant.

2.2.6.1. Material cost of wire spring. Cost property is now wire material volume

$$xx(1) = \text{Volume} = \text{Length} \bullet \text{Area} = \pi DN_a\frac{\pi}{4}d^2, \quad N_a = N_{tot} - 2 \tag{14}$$

Here N_{tot} is the total number of coils and N_a is the active number. It is by 2 turns less than total number when the spring ends are bent and ground as shown in Fig. 2b.

2.2.6.2. Torsional safety factor estimation using Goodman diagram. The torsional endurance strength is reversed strength S_{ew} which is independent of size and alloy composition [2].

S_{ew} = 310 MPa applies for not shot peened springs or who have lost their protective layer.

S_{ew} = 465 MPa applies for shot peened springs.

In this model the basic stress level is the initial pre-stressing load defined as the minimum shear stress.

This stress τ_i is defined as τ_{min} at initial pre-stressing

$$\tau_i = \tau_{min} \tag{15}$$

Endurance shear stress is

$$S_{es} = \frac{0.5S_{ew} \cdot S_{us}}{S_{us} - 0.5S_{ew}} \tag{16}$$

The torsional safety factor by Goodman diagram [2] is

$$N_{Goodm} = \frac{\left(1 - \frac{\tau_i}{S_{us}}\right)}{\frac{\tau_m - \tau_i}{S_{us}} + \frac{\tau_a}{S_{es}}}, \quad xx(2) = N_{Goodm} \tag{17}$$