MILITARY TECHNICAL COLLEGE

CAIRO - EGYPT

OPTIMAL DESIGN OF HELICAL COMPRESSION SPRINGS SUBJECTED TO STATIC LOADS

G.A. Hassan*, A. Abdel-Moaty**

ABSTRACT

Computer aided optimal design of helical compression springs is urgently required to promote the establishment of a national Egyptian standard specifications for such springs. The study covers the optimal assignment of spring variables for maximum modulus of resilience and achieving some functional constraints required for the satisfactory operation of the designed springs. Appropriate models and a fast optimization technique are used in solving the constrained optimization problem in hand.

Springs having more resilience and less weight were obtained in comparison with the standard springs. The results are presented for loads from 182.5 N to 2080 N and covers: resilience, stress, volume, spring mass saving and spring stiffness.

1. INTRODUCTION

The rapid growth of computers and their software resulted in a great development in the field of design and utilization of available resources. The application of optimization techniques in the field of mechanical design leads to raw material saving and designed elements of high performance.

Helical compression springs have wide applications in the automotive, locomotive and manufacturing industries. The Egyptian production of such springs follows one of the well known international standards such as DIN, ISO and BS. The promotion of an Egyptian standard specifications for helical compression springs through the Egyptian Organization for Standardization (EOS) has a vital importance for the local springs industry. To establish such a standard using a high level scientific basis, an efficient optimization technique

* Assistant Professor, ** Graduate Student, Department of Mechanical Design, Faculty of Engineering, Cairo University, Giza, Egypt.

will be used.

The objective function, design variables and various functional constraints controlling the satisfactory operation of the spring are specified. The optimization problem is transformed from being constrained to an unconstrained one to facilitate using the Powell-optimization method.

The optimally designed springs are compared with the standard ones (according to DIN standards) and the difference in the design variables, modulus of resilience, shear stress and volume is illustrated and discussed.

2. MODELLING OF HELICAL COMPRESSION SPRINGS

The optimal design of the helical springs requires comperhensive modelling leading to thorough definition of all functions and criteria satisfying safe and economical operation of the springs. On the other hand, spring-modelling leads to the assignment of its critical design variables.

The main functions governing the operation of helical compression springs are as follows:

a) Spring Volume

The volume of the spring is given by:

$$V = \pi^2 d^2 D(n+Q)/4 \tag{1}$$

where d is the wire diameter, D is the mean coil diameter, n is the active number of coils and Q is the number of inactive coils which depends on the type of the spring ends (Q = 2 for squared and ground ends [1]). Fig.1 illustrates the spring geometry.

b) Static Deflection

The deflection of the spring δ corresponding to a static load P is given by [2] :

$$\delta = \frac{8 D^3 n P}{G d^4} \tag{2}$$

where G is the modulus of rigidity of the spring material.

c) Spring Stiffness

Spring stiffness plays an important role in the control of mechanical vibrations. It is function of the spring dimensions, number of coils and modulus of rigidity as follows [2] :

$$S = \frac{G d^4}{8 D^3 n} \tag{3}$$

d) Modulus of Resilience

The modulus of resilience of a spring is its strain energy (under the effect of certain load) per unit volume. The strain energy corresponding to a deflection δ equals $\int_0^\delta P\, d\delta$. Therefore, for a spring of linear characteristics, the strain energy μ is given by:

$$\mu = \frac{4\, P^2 D^3\, n}{G\, d^4} \qquad (4)$$

The modulus of resilience R is thus obtained by dividing μ (eq.(4)) by V(eq.(1)). That is:

$$R = 1.62\, P^2\, D^2\, n/G\, d^6\, (n + Q) \qquad (5)$$

e) Maximum Shear Stress

The maximum shear stress in the spring wire occurs at its inner side and it is given by [3] :

$$\tau = \frac{8\, P\, D\, K}{d^3} \left(1 + \frac{d}{2D} \right) \qquad (6)$$

where K is a factor known as "Wahl correction factor" [3] due to the wire curvature. It is related to the diameter ratio D/d through the equation [3]:

$$K = \frac{4(D/d) - 1}{4(D/d) - 4} + \frac{0.615}{(D/d)} \qquad (7)$$

f) Free Length

With an allowance of 12% of the spring deflection to prevent metal to metal contact, the free length of the spring H_f is given by [1]:

$$H_f = (n+Q - 0.3)\, d + 1.12\delta \qquad (8)a$$

Combining eqs.(2) and (8)a gives H_f as:

$$H_f = (n+Q - 0.3)d + \frac{8.96\, D^3\, n\, P}{G\, d^4} \qquad (8)b$$

g) Buckling Load

The buckling load represents the upper limit of the load acting on a helical compression spring to prevent its instability. It is function of both the spring stiffness S and the free length H_f. It is given by [2] :

$$P_b = S\, H_f\, K_b \qquad (9)$$

where K_b is a buckling factor function of the ratio H_f/D. In a form suitable to the computer aided design, K_b is presented as a third order polynomial using curve fitting techniques. That is:

$$K_b = 0.737 + 0.01(H_f/D) - 0.0054(H_f/D)^2 - 0.001(H_f/D)^3 \quad (10)$$

h) Spring Material

Helical springs are usually manufactured using any of the materials: music wire, high carbon steel wire, oil tempered high carbon steel wire and/or alloy steel. The study presented in this paper covers only helical springs manufactured from oil tempered high carbon steel. It has a modulus of rigidity of 8.3×10^{10} N/m^2.

i) Allowable Shear Stress

The ultimate tensile strength of the spring material decreases as the wire diameter increases [1] . It varies from 2698.0 to 1520.5 MN/m^2 for a wire diameter changing from 0.5 to 10 mm respectively [1].

The allowable (design) shear stress τ_d is half the ultimate tensile strength of the spring material [1] . Therefore, the maximum design stress from the above analysis is 1050 MN/m^2.

3. DESIGN VARIABLES

From the previous analysis we see that the main variables governing the operation of the spring are:

a) **Wire diameter, d**

b) **Mean coil diameter, D**

and c) Number of active coils, n

The variables are subjected to an inequality constraints in the form:

$$g_i \leq x_i \leq h_i \quad (11)$$

where x_i are the design variables and g_i and h_i are the lower and upper limits respectively.

4. OBJECTIVE FUNCTION

The objective function defines the main goal of the optimization process. For helical springs, the objective function may be the weight(to be minimum) [4,5] and/or the modulus of resilience(to be maximum) [6] .

Because springs can be considered as energy accumulators, the strain energy stored in a spring under certain load is important. On the other hand, from economical point of view it is desirable to reduce the volume of the spring. Therefore, we adopt the modulus of resilience given by eq.(5) as an objective function for the optimal design problem in hand(to be maximum).

5. FUNCTIONAL CONSTRAINTS

The design problem of the helical spring is subjected to not only the variables constraints given by eq.(11), but also another functional constraints. The functional constraints represent extra conditions to be satisfied beside the objective function described in the previous section.

The functional constraints used in the computer aided design of the helical compression springs are as follows:

a) Volume Constraint:

To control the volume of the spring in order to reduce its cost, it is essential to define an upper limit for its volume.

b) Shear Stress Constraint:

Reducing the volume of the spring as in (a) above under the effect of the same load results in increasing the stress in the spring. Therefore, it is essential to constrain the stress so that for any applied load and optimally selected dimensions, it is less than the allowable shear stress given in section 1.

c) Stability Constraint:

The designed compression spring will buckle if its dimensions and number of coils result in a buckling load less than the applied load. Therefore, to achieve spring stability during operation, a stability constraint should exist. That is:

$$\text{The applied load} < \text{the buckling load}$$

d) Length Constraint:

The free length of the spring is function of both the load and the dimensions of the spring as indicated by eq.(8). This length is indirectly constrained by the stability constraint. However, the springs manufactured according to the international standards such as DIN and BS have definite lengths [1]. In order to compare the optimally designed springs and the standard ones, the free length of the spring is constrained to have the values assigned by the German standard: DIN 2098 [1].

6. STEADY STATE OPTIMIZATION OF CONSTRAINED PROBLEMS

The optimal design of helical springs is a steady state problem where several optimization techniques may be used[7]. However, the available techniques are mainly for unconstrained optimization problems such as direct search methods(e.g. Rosenbrock [8] and Powell [9])and gradient methods (e.g.

Fletcher and Reeves [10]). The helical spring problem is a constrained one, where both the design variables and some otther functions are constrained. To use the available optimization techniques, the constrained problem has to be transformed to an apparently unconstrained one through using variables transformation [11] and modifying the objective function [12] as follows:

a) Variables Transformation:

The inequality constraints of the design variables x_i take the form given by eq.(11). Box [11] suggested using what is called variables transformation where the constrained design variables x_i are replaced with unconstrained ones y_i using the equation:

$$x_i = g_i + (h_i - g_i) \sin^2 y_i \qquad (12)$$

where g_i and h_i are the limits of x_i.

b) Modifying The Objective Function:

The functional constraints presented in section 5 are subjected to lower and upper limits so that:

$$G_j \leq C_j(\bar{x}) \leq H_j$$

where $C_j(\bar{x})$, G_j are the functional constraints, lower limits and upper limits respectively.

The functional constraints can be eliminated by using a new modified objective function $P(\bar{x})$ related to the original objective function $F(\bar{x})$ through the equation [12] :

$$P(\bar{x}) = F(\bar{x}) + \sum_{j=1}^{M} \frac{1}{K_j} S_1 \left[\Delta C_j(\bar{x})\right]^2 \qquad (13)$$

where M is the number of functional constraints, K_j are penalty constants of small values and $\Delta C_j(\bar{x})$ are the difference between the functional constraints $C_j(\bar{x})$ and the limits G_j and H_j as follows:

$$\Delta C_j(\bar{x}) = G_j - C_j(\bar{x}) \qquad \text{if} \quad C_j(\bar{x}) \leq G_j$$

and /or $$\Delta C_j(\bar{x}) = C_j(\bar{x}) - H_j \qquad \text{if} \quad C_j(\bar{x}) \geq H_j$$

The parameter S_1 is a unit step function of a value depends on the value of $C_j(\bar{x})$. That is:

$$S_1 = 1 \qquad \text{if} \; \Delta C_j(\bar{x}) > 0$$

and/or $$S_1 = 0 \qquad \text{if} \; \Delta C_j(\bar{x}) \leq 0$$

The optimization technique used in the present work is Powell's method [9] which is a direct search method that does not require any derivatives of the objective function.

It is suitable for linear or nonlinear, quadratic or nonquadratic optimization problems.

The optimization method, the variables transformation and the modification of the objective function have been programed in FORTRAN language. One master program and five subroutines have been used to optimize any constrained problem.

The optimization technique and computer programs were tested using well known problems such as the post office parcel [11] . On the other hand, the technique was used successfully in the evaluation of the steady state optimal control variables of the plastics extrusion process [13] .

7. OPTIMAL DESIGN VARIABLES OF HELICAL COMPRESSION SPRINGS

The present study covers an operating static load range from 182.5 to 2080 N. The objective function (section 4), functional constraints (section 5) and optimization technique (section 6) have been used to define the optimal spring variables for each static load.

The free length H_f of the spring is constrained to have specific values depending on the load P (according to DIN 2098, 1968). For example:

- At P = 623 N, H_f = 120, 180, 275, 395, 585 mm.
- At P = 2080 N, H_f = 202, 315, 475, 690, 1015 mm.

The parameters of the optimally designed springs are presented in Tables 1 and 2 for the loads 623 and 2080 N respectively. The parameters include the wire diameter, the mean coil diameter, the number of active coils, the spring deflection, the modulus of resilience (objective function), the shear stress, buckling load, mass saving and the spring stiffness (useful in vibration isolation problems).

8. COMPARISON BETWEEN OPTIMAL AND STANDARD SPRINGS

To examine the effectiveness and usefulness of the optimally designed springs , the results are compared with the DIN standard 2098 issued in 1968 and titled: "Helical Spring Made of Round Wire".

The comparison covers static loads from 182.5 to 2080 N and free length from 71.5 to 1015 mm (parameters of the optimal and standard springs are compared in Tables 1 and 2).

9. DISCUSSIONS

The results presented in this paper is only a sample of the results of a complete program of an M.Sc. research work carried out in the Mechanical Design Department, Cairo University. It extends to cover springs having up to 17.5 active coils as indicated in Tables 1 and 2 and free length up to 1015 mm at 2080 N load.

The difference between the optimal and standard wire diameter and mean coil diameter increases with increased load (Tables 1 and 2). It starts from 0.2 and 2.2 mm at 182.5 N load to 0.67 and 6.1 mm at 2080 N load for the wire diameter and the mean coil diameter respectively. The number of active coils has almost a constant difference of 0.5 independent of the load.

The objective function which is the modulus of resilience of the designed spring decreases as load increases (Tables 1 and 2). There is a remarkable difference between the optimal and standard modulus of resilience at small loads (typically, the increase is 36% at 182.5 N load).

As the operating load increases, the volume of the spring increases (Tables 1 and 2). The difference between optimal and standard spring volumes increases with increased load. It starts from 0.67 cm^3 at 182.5 N load to 69.0 cm^3 at 2080 N load. The last difference repesents about 10.9% saving in volume. Money wise, this saving in volume represents raw material saving which is illustrated in Tables 1 and 2 . The significance of Tables 1 and 2 appears from the following example: Suppose that a factory is intended to produce 10000 spring a year from the size corresponding to 2080 N load. The material saving is therefore about 5.4 tons of spring material which may lead to reducing the cost of the spring for certain profits.

The shear stress in the optimally designed springs is more than the corresponding value in the standard springs (Tables 1 and 2). The maximum difference is about 20%. However, the shear stress in the optimally designed springs corresponding to any load is less than the allowable shear stress (section 2).

All the optimally designed springs are safe from point of view of buckling (stability constraint). For example, the buckling load corresponding to an applied load of 2080 N is 2192 N (using eq.(9)).

The range of springs presented in this paper covers a stiffness range from 3.4 KN/m at 182.5 N load to 14.7 KN/m at 2080 N load as illustrated in Tables 1 and 2 .

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11. NOMENCLATURE

$C_j(x)$ = Functional constraints
d = Wire diameter
D = Mean coil diameter
F = Objective function
g_i = Lower limits of design variables
G = Modulus of Rigidity
G_i = Lower limits of functional constraints
h_i = Upper limits of design variables
H_f = Free length
H_i = Upper limits of functional constraints
K = Wahl correc tion factor
K_b = Buckling factor
K_j = Penalty constants
n = Number of active coils
P = Static load
$P(x)$ = Modified objective function
P_b = Buckling load
Q = Number of unactive number of coils
S = Spring stiffness
S_1 = Unit step function
V = Spring volume
x_i = Constrained design variables
y_i = Unconstrained design variables
δ = Spring deflection
$\Delta C(x)$ = Difference between a functional constraint and its limit
μ = Strain energy
τ = Shear stress

All units belong to the "SI" system of units.

Table 1 Comparison between DIN 2098 standard and optimal spring for a load 623 N

	Geometrical parameters d (mm)	D (mm)	n Coils	Free length (mm)	Def. (mm)	Modulus of resilience (MN/m^2) Value	Dev%	Shear stress (MN/m^2) Value	Dev%	Buckling load (N)	Total volume (mm^3) Value	Dev%	$M_{st} - M_{op.}$ (gram)	Stiffness (N/mm)
o	5.0	63	3.5	120	87.7	1.225	-22.0	890.4	-13.55	623	21373.9	19.7	32.8	7.2
*	4.75	61.6	3.0	120	87.1	1.485		1011.0		630	17174			
o	5.0	63	5.5	180	135	1.412	-10.4	890.4	-6.6	623	29146.2	14.6	33.3	4.7
*	4.85	61.2	5.0	180	131.7	1.559		949.0		625	24879			
o	5.0	63	8.5	275	210	1.558	-5.4	890.4	-3.5	623	40806.7	10.7	34.2	3.07
*	4.9	61.3	8.0	275	203	1.642		921.2		624.6	36425			
o	5.0	63	12.5	395	304	1.66	-3.9	890.4	-2.6	623	56349.3	11.2	39.1	2.13
*	4.89	60.4	12.0	395	292.9	1.724		913.1		624.2	50059			
o	5.0	63	18.5	585	455	1.74	-9.5	890.4	-5.3	623	79666.2	14.4	89.2	1.4
*	4.87	61.3	17.0	585	441.2	1.905		937.6		623.7	68228			

Table 2 Comparison between DIN 2098 standard and optimal spring for a load 2080 N

	Geometrical parameters d (mm)	D (mm)	n Coils	Free length (mm)	Def. (mm)	Modulus of resilience (MN/m^2) Value	Dev%	Shear stress (MN/m^2) Value	Dev%	Buckling load (N)	Total volume (mm^3) Value	Dev%	$M_{st} - M_{op.}$ (gram)	Stiffness (N/mm)
o	10.0	125	3.5	202	138	0.842	-29.0	737.8	-17.0	2080	169633.8	24.6	325.5	14.7
*	9.33	118.9	3.0	202	141.3	1.086		863.4		2192	127910			
o	10.0	125	5.5	315	220	0.968	-24.0	737.8	-12.9	2080	231318.9	20.4	367.7	9.27
*	9.45	119.2	5.0	315	224.8	1.2		832.8		2138	184180			
o	10.0	125	8.5	475	338	1.069	-16.2	737.8	-8.6	2080	323846.4	16.5	417.6	6.09
*	9.58	119.2	8.0	475	341.4	1.242		801.59		2117.1	270310			
o	10.0	125	12.5	690	497	1.138	-12.7	737.8	-6.8	2080	447216.4	14.5	507.2	4.17
*	9.64	119.0	12.0	690	498.0	1.283		787.7		2102	382190			
o	10.0	125	18.5	1015	738	1.192	-8.33	737.8	-4.5	2080	632271.5	10.9	538.4	2.82
	9.8	122.2	17.5	1015	739	1.291		771.2		2083	563250			

o DIN value
* Optimal value

$$Dev\% = \frac{DIN - Optimal}{DIN} \times 100$$

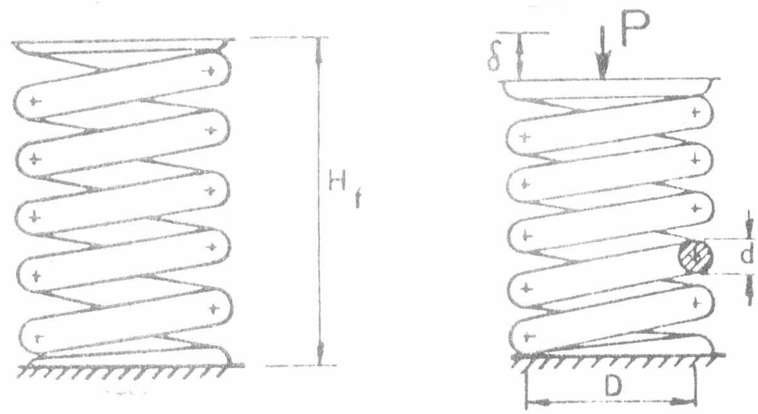

Fig. (1) Configuration and Main Dimensions of Helical Compression Springs

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All units belong to the "SI" system of units.

Table 1 Comparison between DIN 2098 standard and optimal spring for a load 623 N

	Geometrical parameters			Free length	Def.	Modulus of resilience (MN/m²)		Shear stress (MN/m²)		Buckling load	Total volume (mm³)		$M_{st} - M_{op.}$	Stiffness
	d (mm)	D (mm)	n Coils	(mm)	(mm)	Value	Dev%	Value	Dev%	(N)	Value	Dev%	(gram)	(N/mm)
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Table 2 Comparison between DIN 2098 standard and optimal spring for a load 2080 N

	Geometrical parameters			Free length	Def.	Modulus of resilience (MN/m²)		Shear stress (MN/m²)		Buckling load	Total volume (mm³)		$M_{st} - M_{op.}$	Stiffness
	d (mm)	D (mm)	n Coils	(mm)	(mm)	Value	Dev%	Value	Dev%	(N)	Value	Dev%	(gram)	(N/mm)
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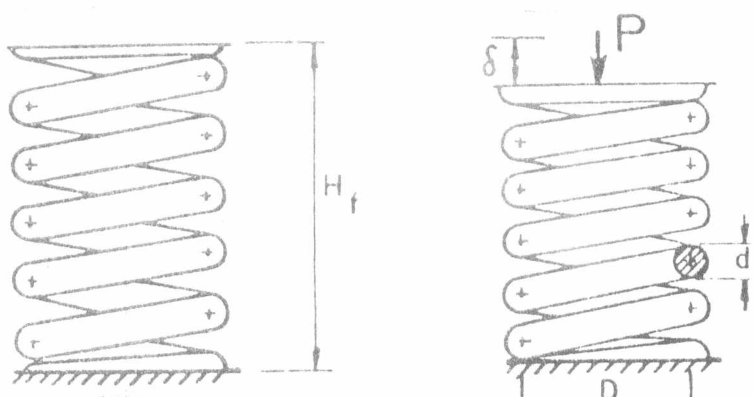

Fig. (1) Configuration and Main Dimensions of Helical Compression Springs