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DESIGN METHOD FOR ELASTIC SYSTEMS USED IN THE CONSTRUCTION OF THE SMALL AND MEDIUM CALIBER ANTIAIRCRAFT GUNS

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Abstract: *The antiaircrafts cannons guns housing is endowed by a helical compression spring, which have the function to accumulate the recul energy, energy used to reposition a reculant part. In this case the helical compression spring loads are dynamic, a fatigue-stress situation exists in the spring. In this paper is presented an method to optimization designing for this helical compression springs.*

Keywords: *helical compression spring, antiaircrafts cannons, recul energy, reculant part.*

1. INTRODUCTION

Mechanical system pipes are composed of antiaircraft guns coil springs compression springs known as recovery. Springs can be round section where antiaircraft guns small caliber (Fig. 1) or rectangular for medium caliber (Fig. 2).

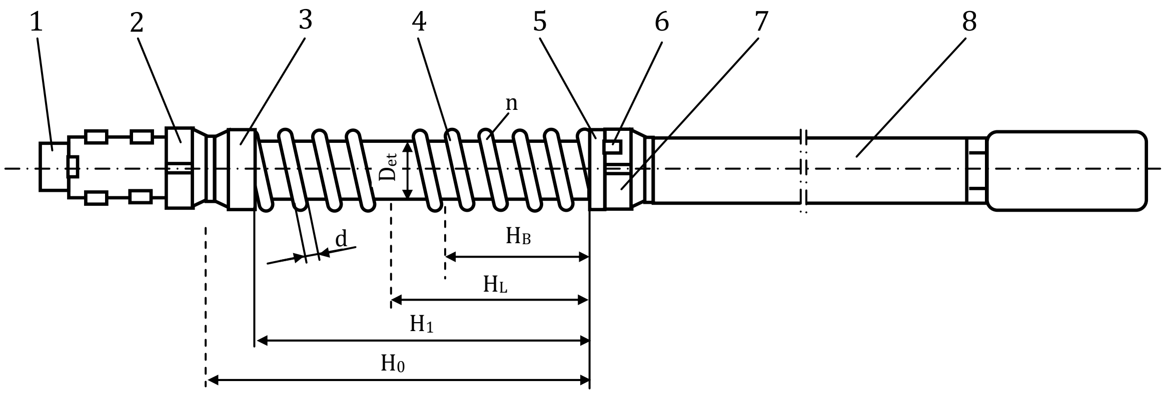

Fig. 1. Barrel overall automatic antiaircraft gun
1 - lainer, 2 - nut, 3 - socket, 4 - spring recovery, 5 - spacer; 6 - safety; 7 - nut, 8 - barrel itself

These functional building blocks are designed to store a part of the recoil energy, energy used for restoring the original position recoil mass. Spring design methodology is adopted based on constructive version adopted by the working of their application type (static or dynamic). In the general case the initial data of the problem of an spring design are overall dimensions, maximum deflection, maximum load taken by bow and rigidity.

After the choice of material that will make up the spring, depending on type adopted will

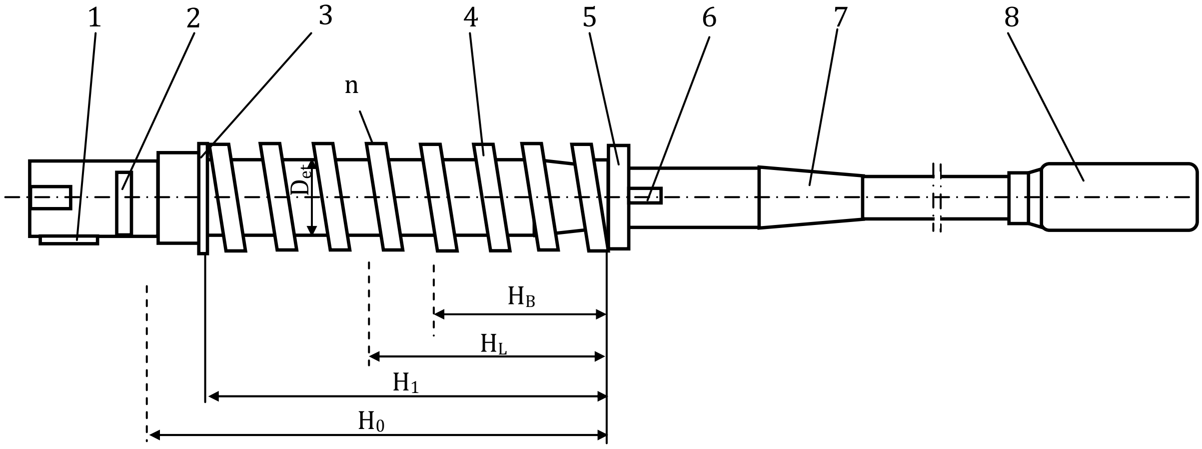

Fig. 2. Automatic antiaircraft gun barrel assembly
1 - safe plate, 2 - bearing plate, 3 - washer, 4 - spring recovery;
5 - threaded bush, 6 - removable tube, 7 - barrel itself; 8 - brake mouth

determine the geometric parameters of stress and strain conditions, determining the deformation and mechanical work. Must also set the degree of nonlinearity of this feature.

2. DESIGN METHOD OF RECOVERY ELASTIC SYSTEM

In that case, the spring recovering from antiaircraft guns barrels, requested by variable load, aiming at the natural frequency of oscillation of longitudinal vibration foj take maximum values in order preîntâmpinării occurrence of resonance phenomena. Own frequency values helical compression spring, when running drawdowns are calculated with [1], [2]:

$$f_{oj} = \frac{\omega_{oj}}{2\pi} = j\sqrt{\frac{c/m}{2}} \qquad (1)$$

where: $j \ \varepsilon \ N^*$,

c - representing stiffness spring:

$$c = \frac{Gd^4}{8nD_m^3} \qquad (2)$$

G - transversal modulus,

d - wire diameter,

Dm - average diameter of the spring,

$$m = \frac{\pi^2 d^2 D_m n\gamma}{4g} \qquad (3)$$

Making substitutions, equation (1) becomes:

$$f_{oj} = j\frac{d}{n(D_{et} + d + j_0)^2}K \qquad (4)$$

where: j = 1, whereas maximizing their lowest frequency and the coefficient $K = \sqrt{Gg/8\pi^2\gamma}$ is dependent on the mechanical properties of the material of the spring considered known. For the analyzed springs, K = 3.63 · 104 MPa [1].

To find the maximum function possible given the relation (5), must be identified and fixed boundary conditions in terms of constructive-functional:

1. Overall conditions are given outside diameter of the barrel D_{et}, and work strain $\Delta H = H_L - H_1$, this reflected in the number of active coils in the spring:

$$D_{m\,min} = D_{et} + d + j_0 \le D_m \le D_{m\,max} \qquad (5)$$

$$D_{m\,min} = d \cdot i_{min} \qquad (6)$$

$$D_{m\,max} = d \cdot i_{max} \qquad (6')$$

where:

j_0 – is the clearance between the outside diameter of the barrel and the inside diameter of the spring coil (j_0 = 4…6 mm);

i – spring index; i = 4…16 (cold coiled springs), i = 4…10 (hot coiled springs);

Specific case it is advisable to take minimum average diameter.

2. Number of active coils of spring recovery is limited by:

$$n_{min} \leq n \leq n_{max} \tag{7}$$

where: n_{min} - minimum number of turns [2]:

$$n_{min} = 5 \tag{8}$$

n_{max} - maximum number of turns [2]:

$$n_{max} = \frac{H_0 - \delta_L}{d} n_r \tag{9}$$

n_r - number of turns for support;

Relation (9) can be written as:

$$d \prec \frac{A_1}{n_i + A_2} \tag{10}$$

where the coefficients:

$$A_1 = H_0 - \delta_L, \text{ and } A_2 = n_r \tag{11}$$

2. The condition for avoiding the phenomenon of buckling springs recovery is performed, although they are mounted on the gun barrel itself and are supported at both ends.

The spring length H_F has not yet appeared buckling phenomenon is given by the expression [2], [3]:

$$\frac{H_F}{D_{et} + d + j_0} \prec \frac{2{,}62}{\nu} \tag{12}$$

resulting from this condition:

$$d \succ \frac{H_F \nu}{2{,}62} (D_{et} + j_0) \tag{12'}$$

This calculation is performed to prevent the friction forces between the pipe and the outside diameter of the spring turns. When factor analysis is slender case $\lambda \geq \lambda_{critic}$, the arrow spring $\delta \leq \delta_{critic}$.

4. The condition of the spring stiffness is given by:

$$c_{min} \leq c \leq c_{max} \tag{13}$$

where the relation for calculating the stiffness is:

$$c = \frac{Gd^4}{8nD_m^3} \tag{14}$$

Using relation (10), the condition of rigidity expressed by the relation (9) can be written as:

$$\frac{Gd^4}{8c_{max}} \prec n(D_{et} + d + j_0)^3 \prec \frac{Gd^4}{8c_{min}} \tag{15}$$

Relation (15) can be write as:

$$\sqrt[3]{\frac{B_1 d}{n_i}} \prec \frac{D_{et} + j_0 + d}{d} \prec \sqrt[3]{\frac{B_2 d}{n_i}} \tag{16}$$

where the coefficients:

$$B_1 = G/8c_{max}, \text{ and } B_2 = G/8c_{min} \tag{17}$$

5. Resistance provided at the request of torsion spring:

$$\tau_{t\,max} \leq \tau_{at} \tag{18}$$

where the maximum torque voltage is determined by the relation [1], [2], [3]:

$$\tau_{t\,max} = \frac{8kF_n D_m}{\pi d^3} = \frac{8kF_n(D_{et} + d + j_0)}{\pi d^3} \leq \tau_{at} \tag{19}$$

Relation (19) can be write as:

$$\frac{D_{et} + j_0 + d}{d} \prec \frac{C_2}{C_1} d^2 \tag{20}$$

where the coefficients:

$C_1 = 8kF_n$ and $C_2 = \tau_{at}$.

Pairs of values n, d that satisfy stringent conditions and maximize function (4) is a curve that can be considered optimal curve of variation of the number of turns of the coil spring depending on the diameter d.

To determine this curve was applied the following strategy [4]:

1. The range of the number of turns was divided into a number of intervals $n_1, n_2, \ldots n_i, \ldots n_k$.

2. For each value we restrictive conditions turned the curve in relation to the variable d.

3. Of behavior maximized function, that function is increasing, so the corresponding value of (n_i, d_i), i = 1 ... k be the largest value of d, which satisfies the boundary conditions simultaneously.

4. We obtained a set of values (n_i, d_i), i = 1 ...k. These values have enabled tracing function d = f (n), optimal function.

To illustrate the method used in obtaining, fig. 3 shows how we value d_i for n_i chosen. It use the following notations:

$$F1_i := \sqrt[3]{\frac{B_1}{n_i} d_i} \tag{21}$$

$$F2_i := \sqrt[3]{\frac{B_2}{n_i} d_i} \tag{22}$$

$$F3_i := \frac{60 + d_i}{d_i} \tag{23}$$

$$F4_i := 0.036 d_i^2 \tag{24}$$

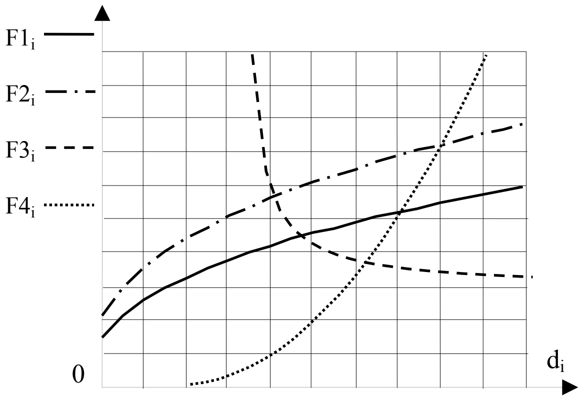

Fig.3 Representation of restrictive conditions in relation to variable d

In fig. 4 is presented graphically optimal curve obtained. Given a value for n, resulting from the curve corresponding value for the spring wire diameter d plotting graphs and mathematical modeling was performed using Professional MathCad 7 utility [5].

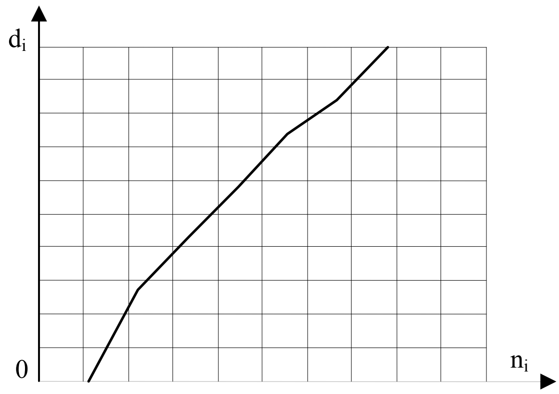

Fig.4 Representation optimal function d = f (n)

4. CONCLUSIONS

The method presented in this paper can be applied in individual cases running of elastic systems, both for their design, or test cases. Also, following the same steps of calculation, the method presented is applicable to other variants of elastic systems.

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