UDC 629.7.01

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Computer technology of elastic-damping elements of aircraft equipment unit optimization

The system of computer simulation of dynamic behavior of units of aircrafts and their optimum designing is developed. The outcomes of researches will allow to improve reliability of units and equipment of aircrafts at the stage of designing, it is essential to reduce the number of bench and flight tests.

Amortization suspensions of equipment should provide the necessary protection of equipment from vibration and shock while maintaining its efficiency throughout the operational period.

On the basis of the review of the literature it is established that the optimization of the mass of such objects was carried out by simplified methods, not relying on nonlinear mathematical programming [1 – 4].

The suspension of the air conditioning unit of the aircraft is considered as an object of optimization. This system is characterized by an asymmetric mass distribution and bonds arrangement, as shown in Fig. 1. The mechanical links are the attachment points in the form of shock absorbers and the docking points with pipes or bellows. We shall assume that the block is a rigid body. The shock absorbers shown in Fig. 2, we shall assume as optimized elements.

The problem of the real constructions synthesizing, as a rule, is modeled by nonlinear systems of partial differential equations with many functional links and constraints. To solve the problem, a hybrid adaptive method of extremum search was applied [5]. It consists in the fact that some set of optimization methods functions in a structural connection. Each method contributes to the process of finding the extreme point.

The problem of reducing of the vibration insulators weight and the overloads of the unit is considered. Its own inertial characteristics do not change, but generalized suspension parameters (the stiffness coefficients *k* and friction forces *f*) are variable.

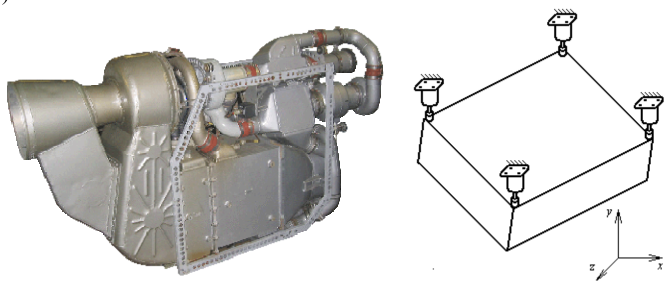

Fig. 1. Aircraft air conditioning unit and its design model

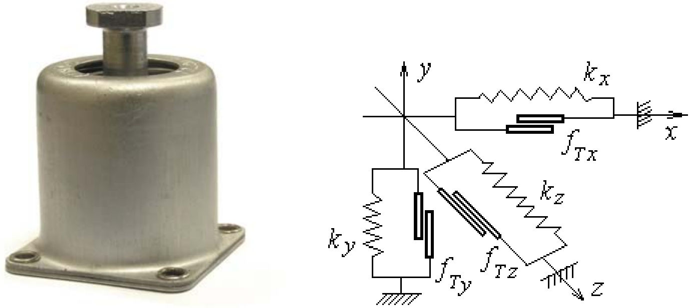

Fig. 2. Shock absorber and its design model

Restrictions are imposed on the design dimensions and strength of shock absorbers. The latter is due to shear stresses in the springs. Since the spring stiffness depends on the diameter of its wire and the shear stresses depend on it, the permissible rigidity is expressed in terms of the permissible shear stress. The elastic and damping forces in the supports are generally nonlinear. Suspension consists of four supports, attached to the unit by hinges. Each support is modeled by three linearly elastic springs and three dry friction dampers located along the coordinate axes. The weight of the suspension, which is a function of the aim, is composed of the mass m_k of the elastic and the mass m_c of the damping elements.

A linearly resilient member is generally provided in the form of a spring. Its rigidity is proportional to d^4 and inversely proportional to R^3, where d is the diameter of the wire, and R is the radius of the spring. The spring mass is proportional to the product d^2R. Assuming that when the spring is varied, the ratio d / R is constant, we find that the mass of the spring

$$m_k = Ak^3 .$$

We assume that when the dimensions of the shock absorber vary, the mass of rubbing parts is proportional to the frictional force. Then there will be

$$m_c = Bf$$

The analysis task establishes the relationships between the parameters of the object for solving the synthesis problem. As a result of the solution, we obtain the most rational relationships of these parameters, which ensure the fulfillment of the necessary criteria. In this case, in order to determine the overloads of the unit and the efforts in the connections, it is necessary to determine the law of its movement. As a result of solving the equations of motion, we find the overloads n and compare them with the allowable value [n]. The rest of the restrictions are assigned from constructive considerations and regulatory requirements.

The displacement of the unit during oscillating motion is limited by the dimensions of the niche in which it is installed, the maximum stroke of the shock absorber rod and the permissible movements of the compensators flanges connecting the unit to the pipelines. In our calculations, the restriction of displacement along the longitudinal axis of the aircraft $[u_x] = 8$ *mm* is accepted.

Permissible displacement along the vertical and transverse axes $[u_y] = [u_z]$=5 *mm*. The permissible rotation angles around these axes are $[\varphi] = 0.02$ *rad*, $[\psi] = [\theta] = 0.04$ *rad*. For the calculation, the values $A = 0.8{\cdot}10\text{-}17\ s^6 / kg^2$, $B = 0.1{\cdot}10\text{-}10\ s^2 / m$, which are chosen on the basis of the static calculation of the springs under the influence of inertial forces at the initial value of the overload are adopted. Acceptable overload $[n] = 2$ and characteristics of the prototype of the suspension, which corresponds to the column Y_0 of table, are adopted.

Table

Results of optimization

	Y_0	Y_C	Y_K	$Y_П$	Y_Σ
$k_x/10^3$	134	152	027	55	70
$k_y/10^3$	134	141	290	238	261
$k_z/10^3$	134	123	041	66	67
$\beta{\cdot}10^3$	4,17	5.65	2.04	2.99	3.48
a_1	5.00	4.85	2.96	3.99	3.57
a_2	3.00	2.87	2.50	3.43	2.59
a_3	4.00	3.95	2.74	3.23	3.07
a_4	2.00	1.89	2.08	1.82	2.14
m	15.18	13.55	14.00	16.08	14.00
n	2.28	2.28	1.48	1.60	1.53

The initial parameter vector Y_0=X_0 consists of the stiffness coefficients k_x, k_y, k_z, the ratio of the damping coefficient to the stiffness coefficient β, the relative values a_1, a_2, a_3, a_4 characterizing the structural composition of the suspension of the unit.

The vibration of the airframe of the aircraft, from which the unit should be protected, is given in the form of a spectrum of overloads. In the frequency range from 1 *Hz* to 22 *Hz*, the vibration overload is constant and equal to 1, in the interval from 22 *Hz* to 60 *Hz* it increases linearly from 1 to 7 and then to the highest frequency of 2000 *Hz*, it remains equal to 7.

For the vector Y_0 indicated in the table, the first optimization step was performed without taking into account the overload n. It gave the vector Y_1. The optimization process as a whole is constructed as a sequence of solutions to the problems of minimizing the mass m and overload n. The next stage of optimization is carried out with minimization of the overload n, but without the restriction of the mass m. As a result, the vector Y_2 is obtained. Omitting the results of intermediate calculations, we give in the table the vector Y_C, the significant digits of the parameters of which differ from the previous one only in the third digit. Note that the optimization process in the stages where the aim function was mass was limited to such allowable overload, which was achieved as its minimum in the previous stages with the aim function in the form of overload.

As a result of the optimization, the weight of the suspension elements and the overload of the suspended unit are reduced. In the design under consideration, when the restrictions on overload are removed, the total mass can be brought to

3.96 *kg*. When studying the influence on the process of searching for an extremum of various constraints (geometric parameters, forces in shock absorbers, etc.), it is established that the active restriction is the displacement of the unit u_x. By this criterion, the optimization process comes to the boundary of the domain of admissible solutions and then moves along this boundary.

When protecting the unit from short-term exposure, similar results were obtained. The active restrictions are the movement of u_x and the overload of the unit. The first value of the mass in the allowable region was 27.84 *kg*. Alternately optimizing the mass and overload, received a design mass of 14.00 *kg*. Parameters of the task at the end of optimization are given in column Y_K.

Effectively minimize mass can only in combination with minimizing overload. Therefore, it is natural to use multicriteria optimization. Calculations were carried out with two variants of the goal function. The first

$$F_{\Pi} = m \cdot n ,$$

the second

$$F_{\Sigma} = m/m_{in} + n/[n] ,$$

where m_{in} is the initial value of the shock absorber mass. The optimization results are shown in the rows of the table Y_{Π} and Y_{Σ}, respectively.

When using the product m and n as the aim function, the mass of the suspension of the unit is reduced to 16.08 *kg*. When the aim function was the sum of m and n, the weight of the suspension was 14.00 *kg*.

Thus, it was possible to obtain the optimal Pareto [7] vector of parameters, when further improvement of one of the criteria can only lead to the deterioration of the other. The weight value decreased in comparison with the original weight by 8.4%, and the overload by 25%, which indicates a significant reserve in the existing design.

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