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# THE PROBLEM OF PASSIVE AND ACTIVE VIBRATION PROTECTION OF DISSIPATIVE-INHOMOGENEOUS MECHANICAL SYSTEMS WITH A FINITE NUMBER OF DEGREES OF FREEDOM

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Kinematic excitation (Fig. 1 ) is used for vibration protection of technical devices, technological apparatus, machines, i.e., electronic devices and equipment that are very sensitive to vibrations of devices installed on moving objects.

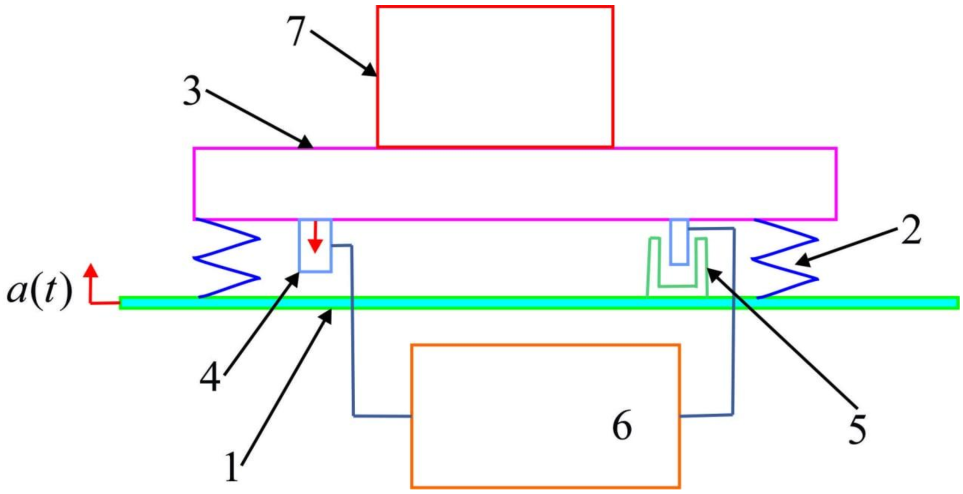


Fig. 1. One-dimensional diagram of an active vibration protection system.

1 – plate (base); 2 – springs (deformable elements); 3 – intermediate plate; 4 – acceleration sensor; 5 – viscous element; 6 – control unit; 7 – vibration protection object; $a(t)$ – base displacement

In active vibration protection systems (Fig. 1 ), the control object will be the vibration protection object, and passive elements are included in the system as springs, dampers or their combinations. If the system also includes a servo drive, this means that the system has a regulator.

The input signals will be displacements, velocities and accelerations of points, angular displacements, angular velocities, angular accelerations, forces and stresses , etc.

$$[M]\{\ddot{X}\}+[B_2]\{\dot{X}_2\}+[B_1]\{\dot{X}_1\}+[C_2]\{X_2\}+[C_1]\{X_1\}=\{F(t)\}+\{\eta(t)\};$$

$$L\frac{di}{dt}+Bl(\dot{x}_3-\dot{x}_2)+Ri=(u_{bx}-u_{dp})K_1; \qquad (1)$$

$$u_{bx}=K_0K_{dp}x_1; \quad u_{dp}=K_{dp}(x_3-x_1),$$

Here

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$$[M]=\begin{matrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{matrix},\ [C_1]=\begin{matrix} c_1 & 0 & 0 \\ 0 & c_2 & 0 \\ 0 & 0 & c_3 \end{matrix},\ [C_2]=\begin{matrix} 0 & 0 & 0 \\ 0 & c_2 & 0 \\ 0 & 0 & 0 \end{matrix},\ [B_1]=\begin{matrix} b_1 & 0 & 0 \\ 0 & b_2 & 0 \\ 0 & 0 & b_3 \end{matrix}$$

$$[B_2]=\begin{matrix} 0 & 0 & 0 \\ 0 & b_2 & 0 \\ 0 & 0 & 0 \end{matrix},\ \{X\}=\begin{matrix} x_1 \\ x_2 \\ x_3 \end{matrix},\ \{X_1\}=\begin{matrix} \dot{x}_1-\dot{x}_2 \\ \dot{x}_2-\dot{x}_1 \\ \dot{x}_3-\dot{x}_2 \end{matrix},\ \{X_2\}=\begin{matrix} 0 \\ x_2 \\ 0 \end{matrix},$$

Here $i$ – current strength; $u$ – voltage; $L$ , $R$ – inductance and resistance; $\ell$ – conductor length; $K_{dp}$ – displacement sensor gain; $K_I$ – tracking system gain, $B\,\ell\,i$ – electromechanical force ;

The proposed research method with servo links represents a new direction, which is based on the application of the model as a system, which is built using multi-pole elements, which made it possible to apply automatic control methods to vibration protection systems and electromechanical systems.

Methods of vibration protection and vibration isolation of machines and devices are developing to transform vibration protection systems into controlled systems.

Active vibration protection devices (AVD) have a lower frequency range limit of ≈ 2 Hz, as well as a maximum vibration suppression coefficient of 35 to 40 dB, which is available at a frequency of ≈ 10 Hz.

**Methods for solving the problem of natural and forced vibrations ( passive and active vibration protection) of dissipative-inhomogeneous mechanical systems**

When considering natural oscillations, the right-hand side is identically equal to zero. We will seek a solution in the form:

$$q_j = A_j e^{-i\omega t}, \qquad j = 1 , \ldots, N ,$$

Where $\omega = \omega_R + i\omega_I$ – complex natural frequency. The problem is reduced to a complex algebraic eigenvalue problem of the form:

$$\sum_{k=1}^{N} (A_k (C_{jk}(\omega_R) - \omega^2 a_{jk})) = 0, \qquad j = 1,2,\ldots,N , \ ( 2 )$$

with a nonlinearly entering complex parameter $\omega_R$ . The characteristic equation of problem ( 2 ) is solved numerically, using the Muller method. As an initial approximation, a solution close to ( 2 ) of the corresponding conservative problem is adopted. In this case, the determinant of system ( 2 ) at each iteration of the Muller method is calculated using the Gauss method with the selection of the main element by rows and columns. Thus, the solution of the problem of natural oscillations using the Muller method does not require the disclosure of its determinant.

We will seek the solution to the problem of forced oscillations of the system ( 2 ) in the form:

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$$q_j = A_j e^{-i\lambda t}, \qquad j = 1,\dots,N \text{ , ( 3 )}$$

Where $A_j$ – the sought complex amplitudes. The problem of steady-state forced oscillations is reduced to a system of non-homogeneous algebraic equations:

$$\sum_{m=1}^{N} (C_{jm}(\lambda) - \lambda^2 a_{jm}) A_m = f_j + \eta_j(q_э),$$

where the solution is carried out by the Gauss method. The result of solving the problem of forced oscillations is obtaining the amplitude-frequency characteristics (AFC) of the mechanical system. If the mechanical system has one degree of freedom, then the equation of motion of the system is written as:

$$y'' + \omega^2 y - \int_{-}^{t} R(t - S_1) y(S_1) dS_1 - C \sin pt = -\eta_1(q_э) \sin pt, \qquad (4)$$

where $p$ is the frequency of the external influence. The particular solution of equation ( 4 ) has the form:

$$d_1 \cos p_1 t + d_2 \sin p_1 t = y(t),$$

Where

$$d_1 = \frac{(-c + \eta_1(q_э))\omega^2 \Gamma_S(p_1)}{\omega^2(1 - \Gamma_C(p_1)) - p_1^2 \;\; ^2 + \omega^4 \Gamma_S^2(p_1)};$$

$$d_2 = \frac{(c - \eta_1(q_э)) \;\; \omega^2(1 - \Gamma_C(p_1)) - p_1^2}{\omega^2(1 - \Gamma_C(p_1)) - p_1^2 \;\; ^2 + \omega^4 \Gamma_S^2(p_1)};$$

$$F_C(p_1) = \int_0 R(\tau) \cos p_1 \tau d\tau; \quad F_S(p_1) = \int_0 R(\tau) \sin p_1 \tau d\tau.$$

When the viscous properties of deformable elements are taken into account through viscous friction, then in matrix form relative to the matrix – column of $\{X\} = colon(x_1, \dots, x_n)$ integro-differential equations (IDE) ( 4 ) take the form

$$[M]\{\ddot{X}\} + [C]\{\dot{X}\} - \int_{-}^{t} [R(t-\tau)]\{X(\tau)\}d\tau + [K]\{X\} = \{f\} + \{\eta(q_э)\}, \qquad (5)$$

where [ $M$ ] is a positive definite matrix, the elements of which denote concentrated masses, [ $C$ ] is the matrix of damping coefficients of deformable bodies and [ $K$ ] is the matrix of elements of rigidity characteristics of deformable bodies (square symmetric matrix), $[R(t-\tau)]$ – viscosity matrix without mass elements. The disturbing force is denoted by the column $\{f\}$vector .

The above matrices have a physical meaning, i.e. $M_{jk}$ , $C_{jk}$ and $K_{jk}$ – respectively, the elements of the mass, damping and stiffness matrix. All matrices are quadratic, $j$ – number of lines, $k$ – the number of columns. The elements of this equation ( 2 ) are obtained from the system of differential equations ( 1 ).

**Equation of motion of vibration protection systems with a finite number of degrees of freedom.**

The equations of vibrations of machine elements can be written in the form:

$$[M]\{\ddot{\varphi}\}+[C]\{\varphi\}+[B]\{\dot{\varphi}\}=\{F(t)\}, \ (6)$$

where $\{\varphi\}$ is the matrix of generalized coordinates, [ $M$ ], [ $B$ ], [ $C$ ] are the matrices of inertial, dissipative and elastic loads, respectively; $F(t)$ is the matrix of external loads.

For active vibration protection systems, its system of equations can be represented by differential equations:

$$\{\dot{x}\}=\{F(t)\}+[A]\{x\}, \ (7)$$

where $x$ is the state matrix; $[A]$ is a square matrix.

If a complex system is considered, then [ $B$ ] and [ $C$ ] may be "dense" matrices.

The reactions of servolinks can be determined by structural diagrams that are equivalent to automatic control systems. This approach can also be used for systems for which mass is taken as an inertial element of a rigid body.

It is known that it is impossible to be distracted from the implementation of servolinks . They can be implemented not passively, not by simply touching bodies, but by mechanisms that generate various generalized forces, for example, mechanical, electromechanical, pneumatic, hydraulic, electrohydraulic, etc.

**Conclusions.**

methods of solving the problem and the algorithm of combined vibration protection of a mechanical system with a finite number of degrees of freedom based on the constraint of the mechanical system by servo links have been developed . Based on the analysis of the obtained numerical results, it has been established that the efficiency of including active vibration protection in the support, in parallel with the passive one, is determined by the gain in the feedback circuit. It has been substantiated that an appropriate choice of the gain can achieve unlimitedly high efficiency, however, with large values of the gain in the feedback circuit, the system may lose stability.

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