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THE PRIORITY OF VIBRATION PROTECTION SYSTEMS WITH A FLUID COUPLING

Abstract*: The article deals with the issue of studying the dynamics of vibration protection of systems with a liquid connection. At the same time, the main goal is to derive and study the vibration equations of systems with vibration protection with a liquid connection and develop on their basis the vibration equations of systems with vibration protection with a liquid connection.*

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Introduction

One of the main problems of technology today is the protection of these systems from harmful vibrations. The method of protection of systems from external vibrations by means of dynamic extinguishers has its advantages. First, dynamic extinguishers can be used directly in the operation of machinery, and secondly, the use of dynamic extinguishers is not costly and is cost-effective.

It is known that mathematical modeling of mechanical systems can be done in different ways. We got acquainted with the simplest of them in the courses of Theoretical Mechanics and Analytical Mechanics.

However, the modeling of nonlinear mechanical systems differs from that of linear mechanical systems in its features. This is primarily due to the fact that the internal or external forces acting on the mechanical system are non-linear.

Siklik ravishda ro'y beradigan yuklanishlarda to'liq sikldagi energiya tarqalishi grafik ravishda yopiq egri chiziqning yuziga son qiymatdan teng bo'lishini (ilmiy adabiyotlarda bu yopiq egri chiziq «gisterezis tuguni» deb ataladi) N.N.Davidenkov, I.L.Korchinskiy, D.Yu.Panov, Ye.S.Sorokinlarning ishlarida uchratish mumkin.

In the works of NN Davidenkov, IL Korchinsky, D.Yu. Panov, YS Sorokin it can be seen that the energy distribution in the full cycle in cyclic loads is graphically equal to the numerical value of the closed curve (in the scientific literature, these closed curves are referred to as "hysteresis nodes").

Gisterez tugunining matematik ifodasi Ye.S. Sorokin misolida.

In the research of scientists after YS Sorokin, the relationship between the strain tensor and the stress tensor was considered linear.

Examples of effective methods for checking the dynamics of nonlinear systems are the method of small parameters, the method of harmonic linearization, the method of separation of variables, the method of averages, asymptotic methods.

Formulation of the problem.

Let us consider the problem of deriving differential equations of motion of free oscillations of mechanical systems with two degrees of freedom. We construct the differential equation of oscillation motion of the system using the Dalamber principle.

[5,7] examined the dynamics of a mechanical system consisting of a spherical solid and an incompressible fluid placed inside a spherical solid. The peculiarity of such mechanical systems is the presence of a liquid joint. The equations of motion of mechanical systems with fluid joints are represented by integro-differential equations or equations of special product.

Simplified equations of models consisting of systems of liquids and solids are formed using finite-dimensional mathematical modeling. The masses of solids and liquids can be assumed to be continuous. Let's look at dynamic extinguishers of the "spherical in-sphere" type, which are dynamic dampers consisting of a spherical solid placed in a solid spherical shell filled with liquid (Figure 1). For this type of dynamic extinguisher, the following relationship is appropriate between the relative velocity of a solid in a dynamic extinguisher and the hydrodynamic force exerted on it by a liquid [7]:

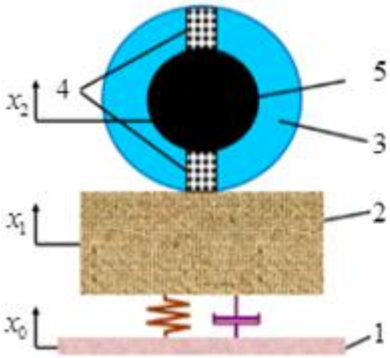

Figure 1. Hysteresis is a system that protects against vibrations with an elastic dissipative characteristic and fluid joints. 1-th basis; 2-protective object; 3-4-5 – dynamic quenching of vibrations with fluid joints.

$$F(s) = -W(s)\dot{x}_2(s); \qquad (1)$$

where W (s) is the transfer function as follows:

$$W(s) = 8\pi\nu\rho r_2^4 \varepsilon^{-3} \cdot \psi(s); \qquad (2)$$

$$\psi(s) = \frac{\chi^3 sh\chi}{24(1 - ch\chi) + 12\kappa sh\chi}; \chi = \varepsilon^2 \nu^{-1} s,$$

where e is the distance between two concentric spheres; ν and ρ are the kinematic viscosities and densities of the fluid, respectively; r2 is the radius of the inner sphere; $s = i\omega$; ω is the circular frequency of oscillations. The last relation holds for $r_2/\varepsilon \geq 10$.

(2) The following relations can be formed from the expression of the transmission function when low frequency oscillations occur

$$k_2 = \lim_{\omega \to 0} \mathrm{Re}\, W(i\omega) = 8\pi\rho\nu r_2^4 \varepsilon^{-3};$$

$$m_n = \lim_{\omega \to 0} \omega^{-1} \mathrm{Im}\, W(i\omega) = \frac{4}{5}\pi\rho r_2^4 \varepsilon^{-1},$$

where k_2 is the damping coefficient; m_p is the mass of liquid attached to 2 bodies.

At sufficiently large values of the dimensionless frequency ω' (in the case of almost $\omega' = \varepsilon^2 \nu^{-1} \omega \geq 100$), the transfer function (2) can be written as

bu yerda k_2 – dempferlash koeffitsiyenti; m_p – 2 jismga birikkan suyuqlik massasi.

$$W(i\omega) = k_2 \alpha(\omega) + i\omega m_n \beta(\omega);$$

here

$$\alpha(\omega) \approx \frac{1}{6} \frac{\left(\frac{\omega'}{2}\right)^{\frac{3}{2}}}{\frac{\omega'}{2} - 2\left(\frac{\omega'}{2}\right)^{\frac{1}{2}} + 2};$$

$$\beta(\omega) \approx \frac{5}{6} \frac{\left(\frac{\omega'}{2}\right)^{\frac{1}{2}} \left[\left(\left(\frac{\omega'}{2}\right)^{\frac{1}{2}} - 1\right)\right]}{\frac{\omega'}{2} - 2\left(\frac{\omega'}{2}\right)^{\frac{1}{2}} + 2}.$$

As can be seen from the last expression, this transmission function has the appearance of a half-degree fractional-rational function with respect to the frequency of the oscillations.

The solution of the problem.

The motion of the mechanical system under consideration is represented by the following matrix equation:

$$A\ddot{X} + B\dot{X} + CX = F, \qquad (3)$$

here

$$\ddot{X}=\begin{pmatrix}\ddot{x}_1\\ \ddot{x}_2\end{pmatrix},\quad \dot{X}=\begin{pmatrix}\dot{x}_1\\ \dot{x}_2\end{pmatrix},\quad X=\begin{pmatrix}x_1\\ x_2\end{pmatrix};$$

$$F=\begin{pmatrix}-W_0(m_1+m_2+m_3)\\ -W_0(m_2-m_в)\end{pmatrix},$$

Vector-columns of accelerations, velocities, coordinates and inertial forces of displacement generalized accordingly,

$$A=\begin{pmatrix}m_1+m_2+m_3 & m_2-m_в\\ m_2-m_в & m_2+m_п\end{pmatrix};$$

$$B=\begin{pmatrix}k_1\\ k_2\end{pmatrix};\quad C=\begin{pmatrix}c_1\\ c_2(1-\nu_1+i\nu_{2*})\end{pmatrix},$$

where m_1, m_2, m_3 are the masses of the base, suspension and vibration damper, respectively; mv, m m_v, m_p - mass of liquid squeezed by 2 bodies and mass of liquid attached to 2 bodies in a dynamic extinguisher, respectively; k_1, k_2 are the coefficients of viscosity of the damper and the fluid, respectively; c_1, c_2 are the coefficients of elasticity of the elastic element attached to the load and the elastic elements mounted on the base, W_0 is the acceleration of the base.

By introducing the differentiation operator $p=\dfrac{d}{dt}$ in the system of equations (3), we come from the system of differential equations to the following system of linear algebraic equations

$$\begin{aligned}&x_1\left[p^2(m_1+m_2+m_3)+pk_1+c_1\right]+\\&+x_2p^2(m_2-m_в)=-W_0(m_1+m_2+m_3);\\&x_1p^2(m_2-m_в)+x_2\left[p^2(m_2+m_п)+pk_2+\right.\\&\left.+c_2(1-\nu_1+i\nu_2)\right]=-W_0(m_2-m_в).\end{aligned}\tag{4}$$

Solving the system of linear equations (4) with respect to the variables x_1 and x_2, we obtain the functions of transmission of vibrations for the dynamic suppressor and the protected object

$$x_1(p)=\frac{W_0C_1(x_2,p)}{M(x_2,p)};\ x_2(p)=\frac{W_0C_2(p)}{M(x_2,p)},\tag{5}$$

here

$$\begin{aligned}M(x_2,p)&=a_4p^4+a_3p^3+a_2p^2+\\&+a_1p+1+\left(b_2p^2+\alpha_1p+1\right)(-\nu_1+i\nu_2)\end{aligned};$$

$$\begin{aligned}C_1(x_2,p)&=n_1^{-2}n_2^{-2}p^2(1+\mu_0+\mu_1-\mu_2)+\\&+n_1^{-2}(\alpha_2p+1-\nu_1+i\nu_2)(1+\mu_0+\mu_1);\end{aligned}$$

$$C_2(p)=n_2^{-2}\mu_3(\alpha_1p+1);$$

$$n_1^2=\frac{c_1}{m_1};\ n_2^2=\frac{c_2}{m_2+m_п};\ \mu_0=\frac{m_2}{m_1};\ \mu_1=\frac{m_3}{m_1};$$

$$\mu_2=\frac{(m_2-m_в)^2}{m_1(m_2+m_п)};\ \mu_3=\frac{m_2-m_в}{m_2+m_п};\ \alpha_1=\frac{k_1}{c_1};\ \alpha_2=\frac{k_2}{c_2};$$

$$a_1=\alpha_1+\alpha_2;\ b_1=\alpha_1;\ a_2=n_1^{-2}(1+\mu_0+\mu_1)+n_2^{-2}+\alpha_1\alpha_2;$$

$$b_2=n_1^{-2}(1+\mu_0+\mu_1);\ a_3=\alpha_1n_2^{-2}+\alpha_1n_1^{-2}(1+\mu_0+\mu_1);$$

$$a_4=n_1^{-2}n_2^{-2}(1+\mu_0+\mu_1-\mu_2).$$

In expression (5) we move from the variable p to the variable $i\omega$, and after simplifications we obtain the absolute values of the variables x_1 and x_2, i.e. the amplitude-frequency characteristics of the protected object and the dynamic extinguisher

$$\begin{aligned}x_1&=\frac{W_0}{|M|}\sqrt{\left(a_4\omega^2-b_2(1-\nu_1)\right)^2+b_2^2(\alpha_2\omega+\nu_2)^2};\\x_2&=\frac{W_0|\mu_3|}{|M|}\sqrt{1+\alpha_1^2\omega^2}.\end{aligned}\tag{6}$$

It can be seen from the second equation of expression (6) that in the case of m3 = 0, i.e. in the zero buoyancy state of the added load, the dynamic extinguisher has no effect on the protected object. In this case, the process of protection against vibrations is not observed. Since a2 ≠ 0, the oscillations of the protected object are not completely extinguished at any values of the parameters of the mechanical system from the first equation of expression (6). This does not happen even if there is no internal inelasticity in the internal elastic elements of the dynamic extinguisher.

Based on Lagrange's theorem on the uniformity of functions, by performing complete differentiation operations in equations (6), by performing transformations, by calculating the products of stationary amplitudes ω, to determine the presence of vertical experiments in the considered functions under the following conditions not difficult

$$\lim_{\omega\to\omega_*}\frac{\partial x_1}{\partial\omega}=\infty;\quad \lim_{\omega\to\omega_*}\frac{\partial x_2}{\partial\omega}=\infty,$$

From the last equations (6) we create the conditions for the presence of verticals between the attempts transferred to the graph of the function.

$$\begin{aligned}&\gamma_1^2+\gamma_1^2+(\beta_1\gamma_1+\beta_2\gamma_2)\left(2\nu_1+x_2\frac{d\nu_1}{dx_2}\right)+(\beta_1\gamma_2-\beta_2\gamma_1)\times\\&\times\left(2\nu_2+x_2\frac{d\nu_2}{dx_2}\right)+\left(\beta_1^2+\beta_2^2\right)\left(\nu_1\frac{d(\nu_1x_2)}{dx_2}+\nu_2\frac{d(\nu_2x_2)}{dx_2}\right)=0,\end{aligned}\tag{7}$$

here

$$\gamma_1=a_4\omega^4-a_2\omega^2+1;\ \gamma_2=a_3\omega^3-a_1\omega;$$

$$\beta_1=b_2\omega^2-1;\ \beta_2=b_1\omega.$$

When these equations are fulfilled, vertical attempts appear on the graphs of the amplitude-frequency characteristics of the system. The presence of vertical attempts indicates that not one, but two or three amplitude values can correspond to certain frequencies on the amplitude-frequency characteristic graph. In this case, the amplitudes of the system's vibrations indicate that there are amplitudes of vibrations that do not actually occur. In this case, by slowly changing the frequencies of the system, the amplitude of the oscillations changes sharply when it reaches the frequency at which the vertical motions are present, that is, the amplitude function acquires the characteristic of a "jump" type.

Taking expression (7) as a quadratic form with respect to the variables b1, b2, g1, g2, we construct the following matrix consisting of the coefficients of this quadratic form:

$$\begin{Vmatrix} a_{11} & a_{12} & a_{13} & a_{14} \\ a_{21} & a_{22} & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ a_{41} & a_{42} & a_{43} & a_{44} \end{Vmatrix}$$

$$a_{11}=1;\ a_{12}=0;\ a_{13}=\nu_1+\frac{x_2}{2}\frac{d\nu_1}{dx_2};\ a_{14}=-\nu_2-\frac{x_2}{2}\frac{d\nu_2}{dx_2};$$

$$a_{21}=0;\ a_{22}=1;\ a_{23}=\nu_2+\frac{x_2}{2}\frac{d\nu_2}{dx_2};\ a_{24}=\nu_1+\frac{x_2}{2}\frac{d\nu_1}{dx_2};$$

$$a_{31}=\nu_1+\frac{x_2}{2}\frac{d\nu_1}{dx_2};\qquad a_{33}=\nu_1\frac{d(\nu_1x_2)}{dx_2}+\nu_2\frac{d(\nu_2x_2)}{dx_2};$$

$$a_{34}=0;$$

$$a_{32}=\nu_2+\frac{x_2}{2}\frac{d\nu_2}{dx_2};\ a_{41}=-\nu_2-\frac{x_2}{2}\frac{d\nu_2}{dx_2};\ a_{43}=0;$$

$$a_{42}=\nu_1+\frac{x_2}{2}\frac{d\nu_1}{dx_2};\ a_{44}=\nu_1\frac{d(\nu_1x_2)}{dx_2}+\nu_2\frac{d(\nu_2x_2)}{dx_2};$$

It is not difficult to verify that the third-order principal diagonal minor of this matrix can have a negative sign if certain conditions are met. This means that there may be instability oscillations between the oscillations of the system under consideration.

By constructing the amplitude-frequency characteristics of the system determined by equations (6) for some values of the system parameters, we construct graphs of the predominance intervals of the stationary oscillations of this system in some special cases (7).

Figure 2 shows m_1=1; m_2=0,15; m_3=0,004; m_v=0,07; k_1=0,01; ξ=1; ρ=1000; ν=10^{-4}; r_2=0,02; ε=0,002;

$$k_2=8\pi\rho\nu r_2^4\varepsilon^{-3}=0{,}1;\ m_n=\frac{4}{5}\pi\rho r_2^4\varepsilon^{-1}=0{,}2$$

c_2=1000; c_1=1000N; graphs of the amplitude-frequency characteristic of the mechanical system under consideration and the resonance curve for the following values of the system parameters. The parametric values of KDU-2 polymer material are obtained as follows

$\mu_1=164.4053,\ \mu_2=-12354.58,\ \mu_3=320537.86,$

*E=2.02*10*^{8}

From the graphs of the amplitude-frequency characteristic generated for the mechanical system under consideration, it is clear that the bending of the skeletal line is a characteristic of mechanical systems with imperfect elastic characteristics. There are such values for the frequency of oscillations ō that these values correspond to three values of oscillations. The two extreme values of these three values are the values that can actually occur, and the one in the middle that does not occur, i.e., the amplitude in the middle is the nostivor amplitude.

Another aspect that interests us is how the amplitude-frequency characteristic changes with the change in the parameters of the mechanical system under consideration. In Figure 3, the mass ratio parameter of the amplitude-frequency characteristic μ0 = 0.01; 0.08; Graphs for 0.2 values are described. It can be seen from the graphs that the increase in the mass ratio brings the maximum amplitude frequencies of the system closer together. The values of the noustivor frequencies of the mechanical system also converge, approaching the anti-resonance frequency.

Results and Discussions

The above addresses the issue of optimal adjustment of a vibration protection system with a fluid joint. As a result, the following relationship between the parameters of the mechanical system is appropriate:

$$n_2=\frac{\sqrt{1+\mu_0+\mu_1-\mu_2}}{1+\mu_0+\mu_1}n_1. \tag{8}$$

the vibration protection system with the fluid joint is adjusted.

Figure 4 shows a graph of priority intervals for a properly adjusted vibration protection system. In this case, the second field of the resonance curves corresponds to the noustivolic field of stationary oscillations of the mechanical system under consideration. The external sector is a priority area. As can be seen from the figure, the field of instability starts at a non-zero value of the amplitude and expands as the amplitude of the oscillations increases.

For simple amplitude correlations of energy dissipation in an elastic damping element, the coefficients ν_1 and ν_2 can be described as polynomials that depend on the amplitude of the oscillations. Suppose that the linearization coefficients ν_1 and ν_2 are described as a function of the amplitudes as follows.

$$\nu_1=\sigma_1x_2^n;\ \nu_2=\sigma_2x_2^n. \tag{9}$$

For connections of this type, Figure 5 shows the change in the predominance intervals of the stationary oscillations of the mechanical system under consideration depending on the parameter s1 / s2. As you can see, as this parameter increases, the system's nostalgia ranges. From this figure it can be concluded that the predominance of the mechanical system under consideration depends in many respects on the influence of the parameters n1 and n2. As these two parameters have a stronger effect on system oscillations, the predominance of oscillations changes.

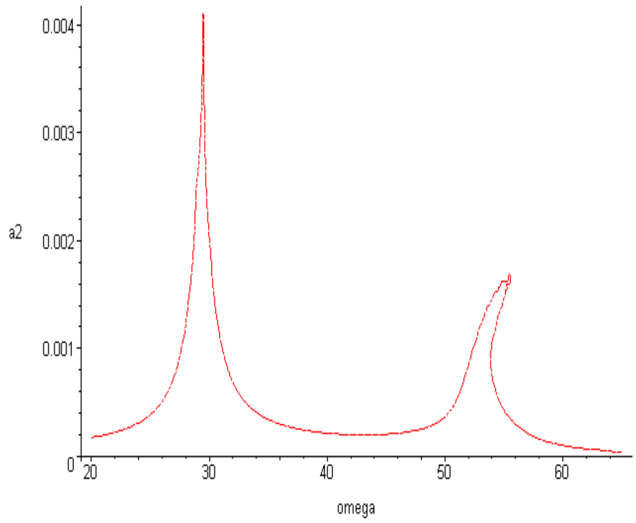

Figure 2. Amplitude-frequency characteristic of systems protected from vibrations with elastic dissipative characteristics of hysteresis with fluid joints

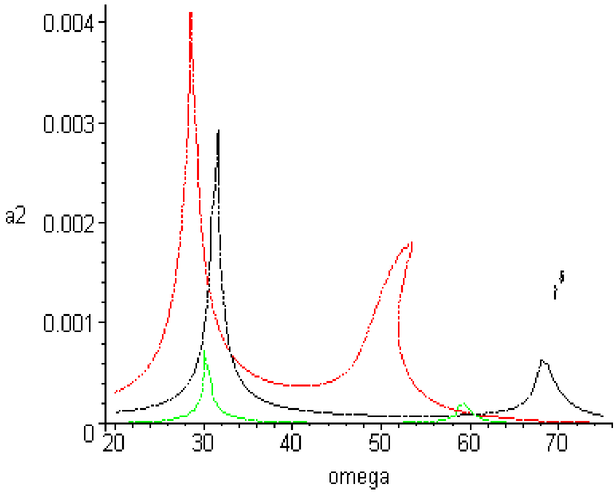

Figure 3. Amplitude-frequency characteristic of a system protected against vibrations for values of mass ratio μ0 = 0.01; 0.08; 0.2

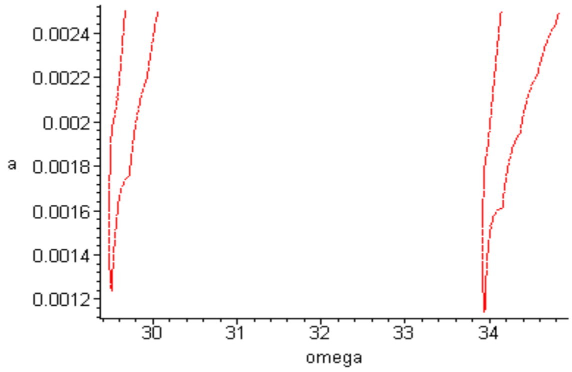

Figure 4. Priority intervals of the vibration protection system.

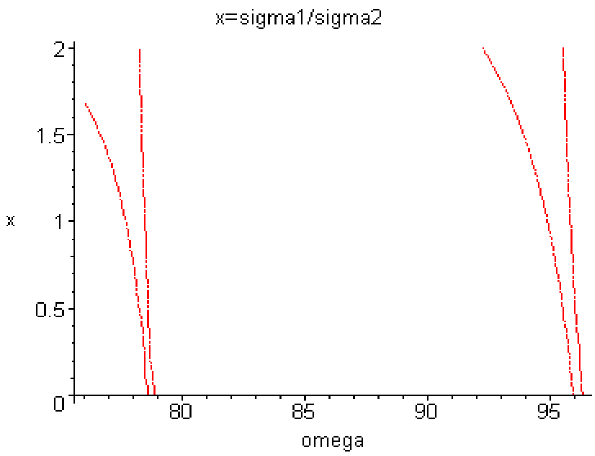

Figure 5. Variation of the system priority interval depending on the parameter σ_1/σ_2

Conclusion

It has been shown that the vibrations of a vibration-protected system cannot be completely extinguished at any value of the system parameters and the external excitation power frequencies. This conclusion is shown to be valid even in the absence of nonlinear characteristics in the mechanical system under consideration. The condition for the verticality of the attempts to plot the amplitude-frequency characteristic of the system was found. The presence of these vertical attempts suggests that noustivor amplitudes may occur between system oscillations. A graph of priority intervals is provided to ensure that the vibration protection system is properly adjusted. In this case, the internal area of the resonance curves corresponds to the noustivolic area of the stationary oscillations of the mechanical system under consideration. The external sector is a priority area. There are two invariant points of amplitude-frequency characteristic for different values of parameters for the system protected from oscillations under consideration. The resulting reaction for the parameters of the system protected against vibration is a necessary condition for the optimal adjustment of the protected system. It has been shown that the predominance of the mechanical system under consideration depends in many respects on the influence of the parameters n1 and n2. The field of instability starts from a non-zero value of the amplitude and expands as the amplitude of the oscillations increases..

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