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Dynamic damping of vibrations of a solid body mounted on viscoelastic supports

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Abstract. The study of the problem of damping vibrations of a solid body mounted on viscoelastic supports is an urgent task. The paper considers the problem of reducing the level of vibrations on the paws of electric machines using dynamic vibration dampers. For this purpose, the paw of electric machines is represented in the form of a subamortized solid body with six degrees of freedom mounted on viscoelastic supports. The *aim* of the work is to develop calculation methods and algorithms for studying the oscillations of the resonant amplitudes of a solid body mounted on viscoelastic supports. Dynamic oscillation (vibration) damping *method* consists in attaching a system to the protected object, the reactions of which reduce the scope of vibration of the object at the points of attachment of this system. Applying the D'Alembert principle, the equations of small vibrations of a solid with dampers are derived. For practical calculations, a simplified system of equations was obtained that takes into account only three degrees of freedom. Numerical calculations were carried out on a computer to determine the amplitude-frequency characteristics of the main body. Numerical experiments were carried out using the Matlab mathematical package. Considering that a solid body is characterized by vibration, as a rule, in a continuous and wide frequency range, therefore, dynamic vibration dampers are used to protect a solid body mounted on viscoelastic supports. It was found that when the damper is set at a frequency of 50 Hz, the vibration level at the left end of the frequency interval of rotary motion of the rotor-converter, decreases to 37.5 dB, and at the right end — to 42.5 dB. At a frequency of 50 Hz, the paws do not oscillate. When setting the dampers to a frequency of 51.5 Hz, the maximum vibration level does not exceed 40 dB. The optimal setting of the dampers is within the frequency of 50.60...50.70 Hz, and a two-mass extinguisher is 10–15% more efficient than a single-mass one. Thus, the paper sets the tasks of dynamic damping of vibrations of a solid body mounted on viscoelastic supports, develops solution methods and an algorithm for determining the dynamic state of a solid body with passive vibration of the object in question.

Keywords: vibration, dynamic damper, construction, viscoelastic support, shock absorber.

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Introduction

The transformation of some types of energy into others in machines and mechanisms, the transformation of forms of movement, the implementation of work processes are inevitably associated with the appearance of variable forces that generate vibration [1–3]. Vibration negatively affects the strength and reliability of machines, load-bearing structures, structures and has a harmful effect on the physiological state of people [4,5]. During the operation of electric machines, vibrations are often observed during their operation [6]. The causes of the oscillations may be disturbing forces of mechanical, electrical and aerodynamic origin. By balancing the rotor, improving the suspension and the construction of the electric machine, it is not always possible to reduce the level of vibrations to acceptable standards. Therefore, we have to find additional means to dampen unwanted vibrations [7, 8]. In order to limit vibration in various fields of technology, there are requirements and norms for its regulation. In most cases, the norms are set taking into account all the most important conditions. Since they cannot equally satisfy all the requirements, they are the result of a compromise solution [9–12].

Dynamic vibration damping consists in attaching a system to the protected object, the reactions of which reduce the scope of vibration of the object at the points of attachment of this system.

If the frequency of the disturbing force changes little, then one of the promising methods requiring the development of ways to reduce the level of vibrations is the use of dynamic dampers [13, 14]. A dynamic damper schematically represents a mass suspended on a spring and having the ability to move in one or more directions. It is known that the use of a dampener tuned to the frequency of the disturbing force makes it possible to reduce the movement of a body with one degree of freedom at this frequency and reduce the level of vibrations at frequencies close to it [15–17].

In this paper, the problem of reducing the vibration level on the paws of electric machines using dynamic vibration dampers is considered.

1. Methods

1.1. Problem statement and basic relations. For a theoretical study of the issue of reducing the level of vibrations on the paws of an electric machine, we will choose the following calculation scheme. The body and paws of the electric machine are considered quite rigid, we neglect the malleability of the rotor and bearings and the gyroscopic effect of the rotor. We will represent an electric machine in the form of a cushioned solid with six degrees of freedom. On the body of the electric machine, we will install dynamic dampers with sensitivity axes directed along the coordinate axes that are connected to the body (Fig. 1).

Consider the small vibrations of a frictionless system relative to the static equilibrium position. As independent coordinates, we choose ξ_0, η_0, ζ_0 — absolute displacements of a point G of a body taken as a pole, three angles φ, ψ, θ of successive rotations of a solid body about the axes $G_{x_1}, G_{y_2}, G_{z_3}$ and ξ_l, η_j, ζ_k — absolute displacements of the quenchers' masses (Fig. 1). The characteristic of the vibration isolator is the dependence of its reaction on the elongations $\triangle l_l (l{=}1, 2, ..., N)$ of the deformable element. The deformable element is considered massless. The relationship of dynamic stiffness and elongation of the deformable element satisfies the following integral dependence [18, 19]:

$$\tilde{c}_n \varphi(t) = c_{0n} \left[\varphi(t) - \int_{-\infty}^{t} R_{cn}(t-\tau)\varphi(\tau)d\tau \right] \tag{1}$$

($\tilde{c}_n$ — operator modulus of elasticity, $\varphi(t)$ — arbitrary time function, $R_{cn}(t-\tau)$ — relaxation kernel, c_{0n} — instantaneous modulus of elasticity). The relationship of dynamic stiffness and elongation of the deformable element satisfies the physical relations for deformable massless elements of zero volume [18, 19]:

$$F_e = -c_e \triangle e = -c_e \left[1 - \Gamma_e^c(\omega_R) - i\Gamma_e^s(\omega_R)\right] \triangle e,$$

where

$$\Gamma_e^c(\omega_R) = \int_0^\infty R_{\lambda,m}(\tau) \cdot \cos\omega\tau d\tau; \quad \Gamma_e^s(\omega_R) = \int_0^\infty R_{\lambda,m}(\tau) \cdot \sin\omega\tau d\tau,$$

F_e — effort in the i-th concentrated element, $\triangle e$ — elongation of this element. Then the following notation is applied: E — instantaneous modulus of elasticity, A, α and β — dimensionless parameters. Using the Dalembert principle to derive the equations of motion, we obtain the following system of equations:

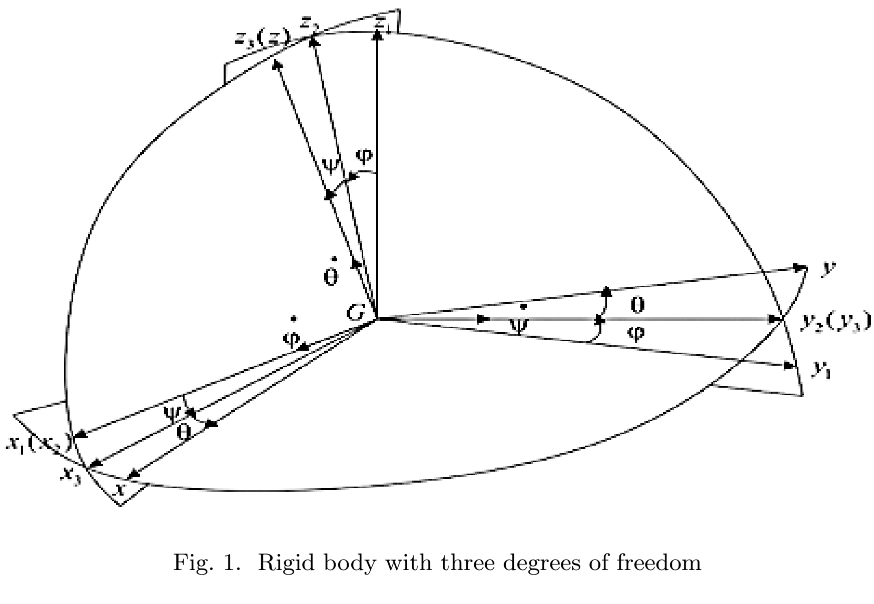

Fig. 1. Rigid body with three degrees of freedom

$$\left(M_0+\sum_1^{n_2} m_j+\sum_1^{n_3} m_k\right)\ddot{\xi}_0+\left(M_0 z_{oyt}+\sum_1^{n_2} m_j z_j+\sum_1^{n_3} m_k z_k\right)\ddot{\psi}-$$
$$-\left(M_0 Y_{oyt}+\sum_1^{n_2} m_j Y_j+\sum_1^{n_3} m_k Y_k\right)\ddot{\theta}+\sum_1^{r_1}\tilde{c}_l\left(\xi_0-Y_{al}\theta+z_{al}\psi\right)+$$
$$+\sum_1^{n_1}\tilde{k}_l\left(\xi_0-Y_l\theta+z_l\psi-\xi_l\right)=\sum_1^{s_1}F_l\sin(p_l t+\nu_l),$$
$$\left(M_0+\sum_1^{n_2} m_l+\sum_1^{n_3} m_l\right)\ddot{\eta}_0+\left(M_0 x_{oyt}+\sum_1^{n_2} m_l x_l+\sum_1^{n_3} m_k x_k\right)\theta-$$
$$-\left(M_0 z_{oyt}+\sum_1^{n_2} m_l z_l+\sum_1^{n_3} m_k z_k\right)\ddot{\varphi}+\sum_1^{n_1}\tilde{c}_j\left(\eta_0+x_{aj}\theta-z_{aj}\varphi\right)+$$
$$+\sum_1^{n_1}\tilde{k}_j\left(\eta_0+x_j\theta-z_j\varphi-\eta_j\right)=\sum_1^{s_2}F_l\sin(p_l t+\nu_l),$$
$$\left(M_0+\sum_1^{n_2} m_j+\sum_1^{n_3} m_k\right)\ddot{\xi}_0+\left(M_0 z_{oyt}+\sum_1^{n_2} m_j z_j+\sum_1^{n_3} m_k z_k\right)\ddot{\psi}-$$
$$-\left(M_0 Y_{oyt}+\sum_1^{n_2} m_j Y_j+\sum_1^{n_3} m_k Y_k\right)\ddot{\theta}+\sum_1^{n_1}\tilde{c}_l\left(\xi_0-Y_{al}\theta+z_{al}\psi\right)+$$
$$+\sum_1^{n_1}\tilde{k}_l\left(\xi_0-Y_l\theta+z_l\psi-\xi_l\right)=\sum_1^{s_1}F_l\sin(p_l t+\nu_l),$$
$$\left[J_{0x}+\sum_1^{n_1} m_l(Y_l^2+z_l^2)+\sum_1^{n_2} m_j Y_j^2+\sum_1^{n_2} m_k z_k^2\right]\ddot{\varphi}-\left[J_{0xy}+\sum_1^{n_1} m_l x_l y_l+\sum_1^{n_2} m_j x_j y_j\right]\ddot{\psi}-$$
$$-\left[J_{0xz}+\sum_1^{n_1} m_l x_l z_l+\sum_1^{n_2} m_k x_k z_k\right]\ddot{\theta}+\left[M_0 Y_{0yt}+\sum_1^{n_1} m_l Y_l+\sum_1^{n_2} m_j Y_j\right]\ddot{\zeta}_0-$$
$$-\left[M_0 z_{0yt}+\sum_1^{n_1} m_l z_l+\sum_1^{n_2} m_k x_k\right]\ddot{\eta}_0+\tilde{c}_x\varphi-$$
$$-\sum_1^{r_2}\tilde{c}_j(\eta_0+x_{aj}\theta-z_{aj}\varphi)z_{\alpha j}+\sum_1^{r_3}\tilde{c}_k(\zeta_0-x_{ak}\theta+Y_{\alpha k}\varphi)Y_{ak}-\sum_1^{n_2}\tilde{k}_j(\eta_0+x_j\theta-z_j\varphi-\eta_j)z_j-$$
$$-\sum_1^{n_3}\tilde{k}_k(\zeta_0+x_k\psi-Y_k\varphi-\zeta_j)Y_k=M_x-\sum_1^{s_2}F_j z_{Fj}\sin(\omega_j t+\nu_j)+\sum_1^{s_2}F_k Y_{Fj}\sin(p_k t+\nu_k),$$

$$\left[J_{cy}+\sum_1^{n_2} m_j(x_j^2+z_j^2)+\sum_1^{n_1} m_l x_l^2+\sum_1^{n_3} m_k z_k^2\right]\ddot{\psi}-\left[J_{0yz}+\sum_1^{n_2} m_j z_j Y_j+\sum_1^{n_3} m_k Y_k z_k\right]\ddot{\theta}-$$
$$-\left[J_{0xy}+\sum_1^{n_1} m_l Y_l z_l+\sum_1^{n_2} m_j z_j Y_j\right]\ddot{\varphi}+\left[M_0 z_{0yt}+\sum_1^{n_2} m_j z_j+\sum_1^{n_3} m_k z_k\right]\ddot{\xi}_0-$$
$$-\left[M_0 X_{0yt}+\sum_1^{n_1} m_l x_l+\sum_1^{n_2} m_j x_j\right]\ddot{\zeta}_0+\tilde{c}_y\psi-$$
$$-\sum_1^{r_3}\tilde{c}_k(\zeta_0+x_{ak}\psi-Y_{aj}\varphi)x_{ak}+\sum_1^{r_2}\tilde{c}_i(\xi_0-Y_{ai}\theta+z_{ai}\psi)z_{ai}-\sum_1^{n_3}\tilde{k}_k(\zeta_0+Y_k\varphi-z_j\varphi-\zeta_k)x_k-$$
$$-\sum_1^{n_2}\tilde{k}_l(\xi_0-Y_l\theta+z_l\psi-\xi_l)z_i=M_y+\sum_1^{s_1}F_l z_l\sin(p_l t+\nu_l)-\sum_1^{s_3}F_k Y_{Fk}\sin(p_k t+\nu_k),$$
$$\left(M_0 x_{yt}+\sum_1^{n_1} m_l x_l+\sum_1^{n_3} m_k x_k\right)\ddot{\eta}_0-\left(M_0 Y_{oyt}+\sum_1^{n_2} m_j Y_j+\sum_1^{n_3} m_k Y_k\right)\ddot{\xi}_0+\tilde{c}_z\theta-$$
$$-\sum_1^{r_1}\tilde{c}_l(\xi_0+z_{al}\psi-Y_{al}\theta)Y_{al}+\sum_1^{r_2}\tilde{c}_j(\eta_0+x_{aj}\theta-z_{aj}\varphi)x_{aj}-\sum_1^{n_1}\tilde{k}_l(\xi_0-Y_l\theta+z_l\psi-\xi_l)Y_l+$$
$$+\sum_1^{n_2}\tilde{k}_j\eta_0+x_j\theta-z_j\varphi-\eta_j)x_j=M_z+\sum_1^{s_2}F_j x_{Fj}\sin(p_j t+\nu_j)-\sum_1^{s_1}F_l x_{Fl}\sin(p_l t+\nu_l),$$

$$
\begin{cases}
m_l\ddot{\xi}_l - \tilde{k}_l(\xi_0 - Y_l\theta + z_l\psi - \xi_l) = 0 & 1 \leqslant i \leqslant n_1,\\
m_j\ddot{\eta}_j - \tilde{k}_j(\eta_0 - x_j\theta - z_j\varphi - \eta_l) = 0 & 1 \leqslant j \leqslant n_2,\\
m_k\ddot{\zeta}_k - \tilde{k}_k(\zeta_0 - x_k\psi + Y_k\varphi - \zeta_k) = 0 & 1 \leqslant k \leqslant n_3.
\end{cases} \tag{2}
$$

When deriving the equations , the notation is used: G_{xyz} — coordinate system rigidly connected to the body; l, j, k — indexes showing that the element, force, moment, etc., works or acts in the direction of the axes G_x, G_y, G_z, respectively; M_0, m_j, m_k, m_l — solid masses and extinguishers, respectively; $J_{ox}J_{oy}J_{oz}J_{oxy}J_{ozx}J_{oyz}$ — moments of inertia of a rigid body relative to the coordinate system G_{xyz}; $x_{oyt}, Y_{oyt}, z_{oyt}$ — coordinates of the center of gravity of a solid body without taking into account the masses of extinguishers in the coordinate system G_{xyz}; $x_l, Y_l, ..., Y_k, z_k$ — coordinates of the quenchers' masses in the static equilibrium position in the coordinate system; $x_{al}, Y_{al}, ..., Y_{ak}, z_{ak}$ — coordinates of the attachment points of springs to a solid in the coordinate system; $x_{Fl}, Y_{Fl}, ..., Y_{Fk}, z_{Fk}$ — coordinates of points of application of external forces in the coordinate system G_{xyz}; $\tilde{c}_l, \tilde{c}_j, \tilde{c}_k$ — operator stiffness coefficients of the springs on which the body is suspended, determined by the dependence (1); $\tilde{k}_l, \tilde{k}_j, \tilde{k}_k$ — operator coefficients of spring stiffness in dampers, which are determined by the dependence (1); $\tilde{c}_x, \tilde{c}_y, \tilde{c}_z$ — operator stiffness coefficients of torsion springs located along the axes G_x, G_y, G_z, respectively, which are determined by dependence (1); F_j, F_k, F_l — amplitudes of external disturbing forces applied to the body; p_j, p_k, p_l — frequencies and phase of external forces; ν_j, ν_k, ν_l — phases of external forces; n_1, n_2, n_3 — number of extinguishers in each direction; r_1, r_2, r_3 — number of springs supporting the body; M_x, M_y, M_z — external moments acting on the body; s_1, s_2, s_3 — number of external forces; $\omega_j, \omega_k, \omega_l$ — natural frequencies.

The equations of motion (2) are derived using the Dalembert principle, possible displacements. The identity of the obtained systems is proved using Lagrange equations of the second kind.

Experiments show that the main vibrations of the machine are vertical movements of the paws. Therefore, we will compile a simplified system of equations of motion of a rigid body with dampers. We will consider only those degrees of freedom of movement of the body that give vertical movements to its points. Such movements will be the movement of the body's pole along the axis $G_x - \xi_0$ and its rotation relative to the axes G_y and $G_z - \psi$ and θ.

N_1 of dynamic dampers with sensitivity axes that are parallel to the G_x axis are installed on the solid. We will place the origin of coordinates in the center of gravity of a solid body, and we will direct the coordinate axes along the main axes of inertia of the body. Then the equations of motion will take the form

$$
\begin{aligned}
&M_0\ddot{\xi}_0 + \sum_1^{r_1}\tilde{c}_l\left(\xi_0 - Y_{al}\theta + z_{al}\psi\right) + \sum_1^{n_1}\tilde{k}_l\left(\xi_0 - Y_l\theta + z_l\psi - \xi_l\right) = \sum_1^{s_1}F_{1l}e^{-ip_lt},\\
&J_{oy}\ddot{\psi} + \sum_1^{r_1}\tilde{c}_l\left(\xi_0 - Y_{al}\theta + z_{al}\psi\right)z_{ai} + \sum_1^{n_1}\tilde{k}_l\left(\xi_0 - Y_l\theta + z_l\psi - \xi_l\right)z_l = \sum_1^{s_1}F_{2l}z_{Fl}e^{-ip_lt},\\
&J_{oz}\ddot{\theta} + \sum_1^{r_1}\tilde{c}_l\left(\xi_0 - Y_{al}\theta + z_{al}\psi\right)Y_{al} - \sum_1^{n_1}\tilde{k}_l\left(\xi_0 - Y_l\theta + z_l\psi - \xi_l\right)Y_l = -\sum_1^{s_1}F_{3l}Y_{Fl}e^{-ip_lt},\\
&m_l\ddot{\xi}_l - \tilde{k}_l\left(\xi_0 - Y_l\theta + z_l\psi - \xi_l\right) = 0, \quad 1 \leqslant l \leqslant n_1,
\end{aligned} \tag{3}
$$

where $F_{1l} = F_le^{i\varphi_1}$, $F_{2l} = F_le^{i\varphi_2}$, $F_{3l} = F_le^{i\varphi_3}$, φ_1, φ_2, φ_3 — phase shifts of external loads. Similarly, the phase shift of the elements of the mechanical system is taken into account.

1.2. Solution methods. Suppose that a perturbing force $F_{0l}e^{-i\varphi_1}$, $l = 1, 2, ..., L$ acts on a solid, where L is the number of external loads.

If the integral term is given on a finite segment $[0, t]$,

$$\tilde{c}_n[\varphi(t)] = c_{0n}\left[\varphi(t) - \int_0^t R_{cn}(t-\tau)\varphi(\tau)d\tau\right], \quad n = (i, j, k) \tag{4}$$

then the natural oscillations of the mechanical system (1) are considered. We take the integral term in (4) small. Then $\varphi(t) = \psi(t)e^{-i\omega_R t}$, where $\psi(t)$ is a slowly changing function of time, ω_R is a real frequency. We will replace the ratios (4) approximations of the form [19, 20]

$$\bar{c}_n[\varphi] = c_{0j}\left[1 - \Gamma_j^c(\omega_R) - i\Gamma_j^s(\omega_R)\right][\varphi], \tag{5}$$

where

$$\Gamma_n^c(\omega_R) = \int_0^\infty R_n(\tau)\cdot\cos\omega_R\tau d\tau; \quad \Gamma_n^s(\omega_R) = \int_0^\infty R_n(\tau)\cdot\sin\omega_R\tau d\tau,$$

cosine and sine — Fourier images of the relaxation kernel of the material. As an example of a viscoelastic material, we take the three-parameter Rzhanitsyn-Koltunov relaxation kernel: $R_n(t) = A_n e^{-\beta_n t}/t^{1-\alpha_{jn}}$. On the influence function $R_n(t-\tau)$ the usual requirements of integrability, continuity (except $t = \tau$), sign-definiteness and monotonicity are imposed:

$$R > 0, \quad \frac{dR(t)}{dt} \leqslant 0, \quad 0 < \int_0^\infty R(t)dt < 1.$$

When solving the problem of natural oscillations, there are no external loads and natural frequencies are determined at given values of physico-mechanical and geometric parameters.

If there are vibration effects on the body, then the resonant frequencies are set and the amplitude-frequency characteristics of various points of the mechanical system are constructed.

We will look for a solution to the problem of forced oscillations in the form:

$$\begin{pmatrix}\xi_0\\ \psi\\ \theta\\ \xi_l\end{pmatrix} = \begin{pmatrix}\Sigma_0\\ \Psi\\ \Theta\\ \Sigma_l\end{pmatrix} e^{-\omega t}, \tag{6}$$

where ω is a given real value. And when solving the problem of natural oscillations $\omega = \omega_R + i\omega_I$ — a complex unknown quantity (frequency) that needs to be determined.

We substitute (6) into the system (2) and exclude Σ_i. We obtain an algebraic system of three equations with respect to three oscillation amplitudes Σ_0, Ψ, Θ. The absolute displacement of an arbitrary point of the body with coordinates Y_B and z_B is given by the expression

$$\xi = \xi_0 + z_B\Psi - Y_B\Theta = \left[\Sigma_0 + z_B\Psi - Y_B\Theta\right]e^{-\omega t} = \Sigma e^{-\omega t}.$$

Each damper, when adjusted to the frequency of the disturbing force, acts on the point of the solid on which it is installed. For a body with three degrees of freedom at the tuning frequency, it is necessary to have three dampers that are not located on the same straight line.

The software package «MAPLE-18» [19, 20] was used to calculate the amplitude-frequency characteristics. The compiled algorithm allows calculations for various forces of the imbalance of the angles between them in different planes, the masses of the dampers and their location, the settings of the dampers and the viscoelastic properties of the viscoelastic element.

2. Results and discussions

The viscoelastic properties of the material are described using a three-parameter relaxation kernel [21–23]:

$$R_{cj}(t) = R_{cl}(t) = R_{ck}(t) = A_p e^{-\beta_p t}/t^{1-\alpha_p},$$
$$R_{kj}(t) = R_{kl}(t) = R_{kk}(t) = A_g e^{-\beta_g t}/t^{1-\alpha_g}.$$

The approach to optimizing the parameters for a viscous friction damper differs from the case of a damper without damping. To obtain the optimal parameters of the damper in the works [24,25], the properties of a linear system with a single damper were used. It was found out that one damper is effective for a mechanical system with one degree of freedom.

The amplitudes of displacements of the center of mass of a body with three degrees of freedom depending on frequency are investigated. The results are obtained in dimensionless parameters, taking into account the damper and without taking into account the damper. For the calculation, the case of four dampers mounted on the paws was chosen, $A_p = 0.01$, $\beta_p = 0.05$, $\alpha_p = 0.1$ $A_g = 0.001$, $\beta_g = 0.025$, $\alpha_g = 0.05$.

$$Y_{gl} = \frac{Y_l}{Y_{01}}, \quad Z_{gl} = \frac{z_l}{z_{01}}, \quad \eta_{ml} = \frac{m_l}{M_0}, \quad \eta_{kl} = \frac{k_l}{c_1}, \quad c_1 = c_2,$$
$$\eta_{mi} = 0.025, \quad \eta_{ki} = 0.65, \quad Y_{gi} = 1, \quad Z_{gi} = 1.$$

In Fig. 2 the amplitude-frequency characteristics of the displacements of the center of mass of the main mass without absorbers and with absorbers are given. It can be seen that four absorbers mounted on the paws effectively reduce the amplitudes of movements.

In Fig. 3 shows the amplitudes of oscillations of the main mass when passing through the resonance *1* — with a one-mass absorber, and *2* — with a two-mass absorber. It is established that a two-mass absorber is 10–15% more effective than a one-mass absorber.

Calculations of a specific example were made for a body weighing 350 kg. For the calculation, the case of four absorbers weighing 9 kg each mounted on paws was chosen $A_p = 0.01$, $\beta_p = 0.05$, $\alpha_p = 0.1$ $A_g = 0.001$, $\beta_g = 0.025$, $\alpha_g = 0.05$, $c_{01} = c_{02} = 2510$ Н/м, $k_{01} = k_{02} = k_{03} = 1500$ Н/м.

In Fig. 4 the results of calculating the amplitude-frequency response in decibels for a solid without absorbers (dotted curve) and with absorbers (solid and dashed curve) are presented. Along the ordinate axis, the vibration level in decibels is postponed $W_{\text{db}} = 20\lg(w/w_0)$, where w is— acceleration of the body point, $w_0 = 2.8\cdot10^{-4}$ m/s. The relative frequency of the disturbing force $\omega_{01} = \omega/\Omega_1$ is postponed along the abscissa axis, where ω is the frequency of the disturbing force, Ω_1 is the frequency of vertical vibrations of a solid on shock absorbers ($\Omega_1 = 18.79$ Hz).

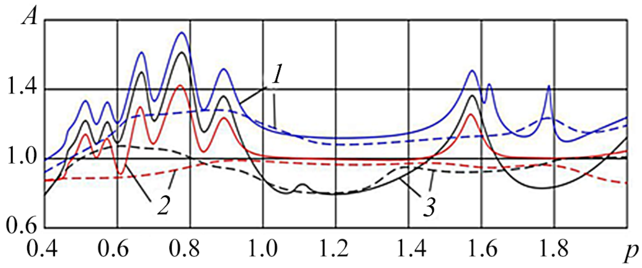

Fig. 2. Amplitude-frequency characteristics of displacements of the center of mass of the main mass (dotted curve — without absorbers and solid — with absorbers)

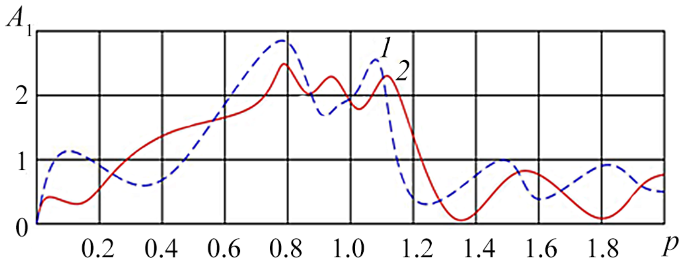

Fig. 3. The amplitude-frequency characteristics of the displacements of the center of mass of the main mass (*1* — with one-mass absorber, *2* — with two-mass absorber)

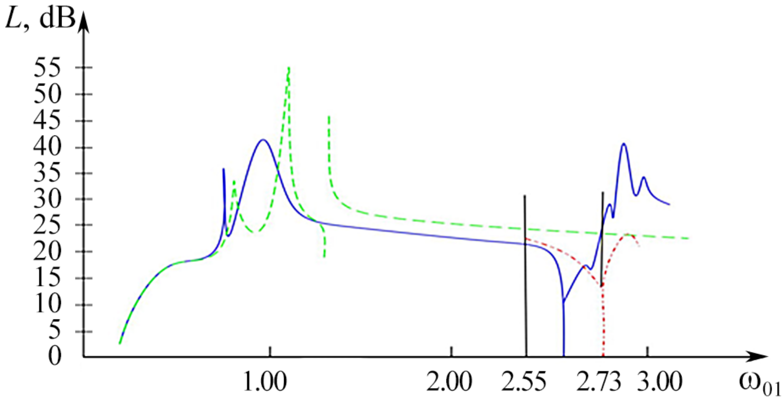

Fig. 4. Amplitude-frequency characteristics of a rigid body with three degrees of freedom: dashed curve — without absorbers and solid curve — with absorbers

At a given level of forces from the rotor imbalance $F_{01} = 0.756$ kg and $F_{02} = 0.767$ kg, the vibration level in the frequency range $\omega = 50$ Hz($\omega_{01} = 2.66$) turned out to be 45.5 dB. This is close to the maximum allowable level of 46 dB.

Conclusion

Based on the results of the research, we have made the following conclusions:

- the rotational motion frequency of the converter rotor ranges from 48 to 51.25 Hz ($2.55 \leqslant \omega_{01} \leqslant 2.73$);
- when setting the absorber to a frequency of 50 Hz, the vibration level decreases to 37.5 dB at the left end of the interval and to 42.5 dB at the right end of the interval;
- at a frequency of 50 Hz, as it follows from the theory, the paws do not oscillate;
- when setting the absorbers to a frequency of 51.5 Hz, the maximum vibration level does not exceed 40 dB;
- the optimal setting of the absorbers is in the region of 50.6...50.7 Hz;
- it has been found that a two-mass absorber is 10–15% more effective than a one-mass absorber.

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