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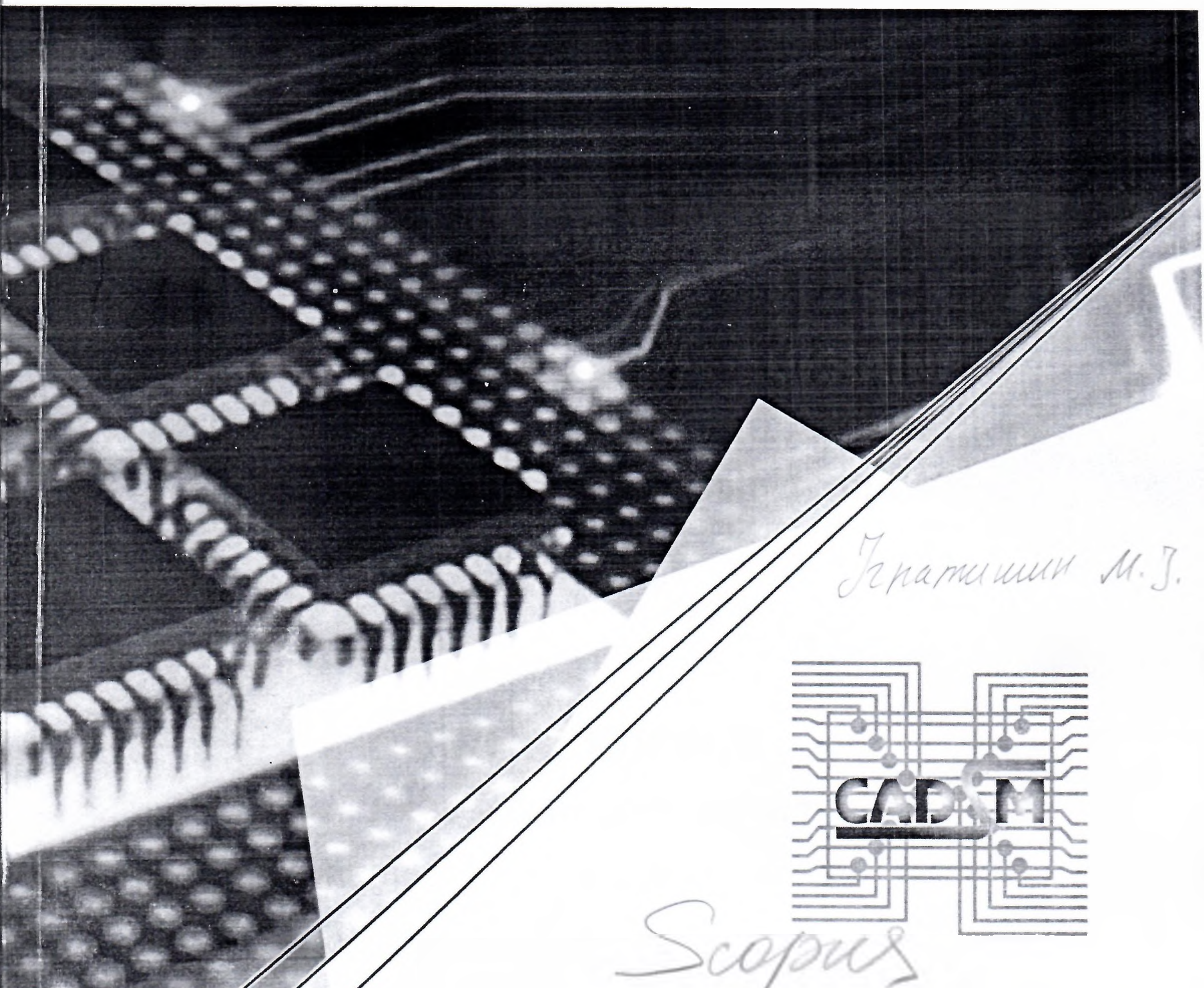

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MODELS AND METHODS FOR MICROELECTROMECHANICAL SYSTEMS

TECHNOLOGIES FOR MEDICINE

Amplitude-Frequency Response Quencher Mechanical Vibrations

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***Abstract*: This paper provides an overview of the generalized problem of optimization of mechanical quencher, considered specific problem - investigated the frequency response of a cylindrical pendulum and quencher mechanical vibrations calculated the resonant frequency quencher and calculation of mechanical parameters quencher for a given resonant frequency. The analytical value used to optimize the mechanical parameters quencher mechanical vibrations..**

***Keywords*: mechanical quencher pendulum mechanical quencher, cylindrical mechanical quencher, optimization of mechanical quencher.**

I. Introduction

Mechanical oscillation dampers are widely used in engineering. They are designed for damping vibrations of mechanisms, machinery, building structures etc. The cause of undesirable vibrations can be both man-made and natural factors such as earthquakes.

II. Text

Studied mechanical vibrations damper relates to two-mass mechanical systems. Such systems are actively investigated for their optimization. Sudak F.M., Voronina I.F., Altukhov D.P. [1] conducted a study in order to reduce the noise of combustion engines with the help of mechanical damper devices, they developed the design of one of the possible options for such a device. Leheza V.P., Leheza D.B., Huzenko S.V. reviewed a pendulum damper type "dumbbell" [2] and proposed to perform regulation of the natural frequency of the pendulum damper with optimum choice of structural dimensions.

Thus the study and analysis of known structures of mechanical vibrations damper, obtaining correlations linking the dynamics of linear and nonlinear oscillations [3] damper with its structural and mechanical characteristics, further formulation of optimization problem on the basis of the mathematical formulas, improving of known and the synthesis of new mechanical dampers to reduce the harmful effects of man-made and natural disasters related to mechanical destruction of machinery, vehicles and buildings are relevant. Methods of solution of applied problems of deforming solid mechanics are used in the studies of Levin E.E., Manevych A. (study of forced vibrations of cylindrical damper).

In the work of Levin E.E., Manevych A. [3] mathematical models of pendulum and cylindrical oscillation dampers were made, complex amplitude A and B of oscillations of massive body and a damper were obtained:

for a pendulum damper

$$\begin{cases} A = \left(\mu \tilde{\Omega}^2 B - iU_0\right) / \left(1 - \tilde{\Omega}^2 + i\tilde{\beta}\tilde{\Omega}\right) \\ B = i\tilde{\Omega}^2 U_0 / \left[-\left(1 - \tilde{\Omega}^2 + i\tilde{\beta}\tilde{\Omega}\right)\left(\tilde{\omega}^2 - \tilde{\Omega}^2\right) + \tilde{\Omega}^4 \mu\right]. \end{cases} \quad (1)$$

for a cylindrical damper

$$\begin{cases} A = \left(\mu \tilde{\Omega}^2 B - iU_0\right) / \left(1 - \tilde{\Omega}^2 + i\tilde{\beta}\tilde{\Omega}\right) \\ B = i\tilde{\Omega}^2 U_0 / \left[-\left(1 - \tilde{\Omega}^2 + i\tilde{\beta}\tilde{\Omega}\right)\left(\tilde{\omega}^2 - 3\tilde{\Omega}^2/2\right) + \tilde{\Omega}^4 \mu\right]. \end{cases} \quad (2)$$

where

$$\mu = \frac{m}{M+m}, \quad \tilde{\Omega} = \Omega\sqrt{\frac{M+m}{k}}, \quad \tilde{\omega} = \sqrt{\frac{g}{k} \cdot \frac{M+m}{R-r_0}}, \quad \tilde{\beta} = \frac{\beta}{\sqrt{k(M+m)}}, \quad (3)$$

m - damper weight, M - a massive body weight oscillations of which a damper absorbs, k - spring stiffness, r_0 - the radius of a cylindrical damper, R - radius of the cylindrical surface, Ω - angular frequency of forced oscillations, β - coefficient of viscous friction, Fig. 1.

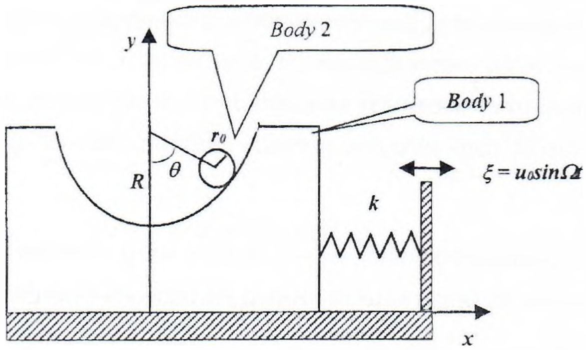

Fig. 1. The body 1 – a massive body; the body 2 is shaped like a cylinder and a damper of mechanical vibrations of a massive body.

Object and methods of research.

The object is a mechanical system of two bodies interacting with each other through the power of pressure and friction and one of the bodies undergoes periodic perturbation of a given frequency and amplitude through an elastic link and is under the influence of dissipative forces proportional to velocity. The subject of the research is mathematical model of mechanical oscillations dampers . The methods of deforming solid bodies mechanics and mathematical analysis have been applied.

Problem

It is necessary to investigate the vibration amplitude A of a massive body, the ratio of (1) and (2), for extremum, to find analytical formulas for the calculation of the oscillations resonance frequency of a massive body with a damper.

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Imagine modules of complex amplitudes (1) and (2) as follows:

for pendulum damper

$$|A| = U_0 \sqrt{\frac{\left(\tilde{\Omega}^2 - \tilde{\omega}^2\right)^2}{\tilde{\omega}^4 + \left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 2\right]\tilde{\omega}^2\tilde{\Omega}^2 + \left\{1 + \left[2\left(2 - \tilde{\beta}^2 - \mu\right) + \tilde{\omega}^2\right]\omega^2\right\}\tilde{\Omega}^4 + \left(2\tilde{\omega}^2\mu + 2\mu - 2\tilde{\omega}^2 + \tilde{\beta}^2 - 2\right)\tilde{\Omega}^6 + (\mu - 1)^2\tilde{\Omega}^8}} \quad (4)$$

for a cylindrical damper

$$|A| = U_0 \sqrt{\frac{\left(3\tilde{\Omega}^2 - 2\tilde{\omega}^2\right)^2}{4\tilde{\omega}^4 + \left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 3\right]\tilde{\omega}^2\tilde{\Omega}^2 + \left\{9 + 4\left[\tilde{\omega}^2 - 3\tilde{\beta}^2 + 2(3 - \mu)\right]\tilde{\omega}^2\right\}\tilde{\Omega}^4 + \left\{4(2\mu - 3)\tilde{\omega}^2 + 3\left[2(2\mu - 3) + 3\tilde{\beta}^2\right]\right\}\tilde{\Omega}^6 + (2\mu - 3)^2\tilde{\Omega}^8}} \quad (5)$$

Therefore, to explore the extreme of denominator expression of ratios (4) and (5) under the root.

Results of the study

Let us find the derivatives of these denominators to $\tilde{\Omega}$ and equate them to zero. Rejecting the solution $\Omega = 0$ we obtain the equation of the sixth degree:

for a pendulum damper

$$8(\mu - 1)^2\tilde{\Omega}^6 + 6\left(2\tilde{\omega}^2\mu + 2\mu - 2\tilde{\omega}^2 + \tilde{\beta}^2 - 2\right)\tilde{\Omega}^4 + 4\left\{1 + \left[2\left(2 - \tilde{\beta}^2 - \mu\right) + \tilde{\omega}^2\right]\omega^2\right\}\tilde{\Omega}^2 + 2\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 2\right]\tilde{\omega}^2 = 0 \quad (6)$$

for a cylindrical damper

$$8(2\mu - 3)^2\tilde{\Omega}^6 + 6\left\{4(2\mu - 3)\tilde{\omega}^2 + 3\left[2(2\mu - 3) + 3\tilde{\beta}^2\right]\right\}\tilde{\Omega}^4 + 4\left\{9 + 4\left[\tilde{\omega}^2 - 3\tilde{\beta}^2 + 2(3 - \mu)\right]\tilde{\omega}^2\right\}\tilde{\Omega}^2 + 8\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 3\right]\tilde{\omega}^2 = 0 \quad (7)$$

Let us denote $\tilde{\Omega}^2 = x$ and obtain two equations of the third degree:

for a pendulum damper

$$8(\mu - 1)^2 x^3 + 6\left(2\tilde{\omega}^2\mu + 2\mu - 2\tilde{\omega}^2 + \tilde{\beta}^2 - 2\right)x^2 + 4\left\{1 + \left[2\left(2 - \tilde{\beta}^2 - \mu\right) + \tilde{\omega}^2\right]\omega^2\right\}x + 2\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 2\right]\tilde{\omega}^2 = 0$$

(8)

for a cylindrical damper

$$8(2\mu - 3)^2 x^3 + 6\left\{4(2\mu - 3)\tilde{\omega}^2 + 3\left[2(2\mu - 3) + 3\tilde{\beta}^2\right]\right\}x^2 + 4\left\{9 + 4\left[\tilde{\omega}^2 - 3\tilde{\beta}^2 + 2(3 - \mu)\right]\tilde{\omega}^2\right\}x + 8\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 3\right]\tilde{\omega}^2 = 0 \quad (9)$$

Let us solve the cubic equation obtained by trigonometric formulas of Viet [5]. Constants of cubic equation

$$ax^3 + bx^2 + cx + d = 0 \quad (10)$$

let us denote:

a pendulum damper

$$a = 8(\mu - 1)^2,\ b = 6\left(2\tilde{\omega}^2\mu + 2\mu - 2\tilde{\omega}^2 + \tilde{\beta}^2 - 2\right), \quad (11)$$
$$c = 4\left\{1 + \left[2\left(2 - \tilde{\beta}^2 - \mu\right) + \tilde{\omega}^2\right]\omega^2\right\},\ d = 2\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 2\right]\tilde{\omega}^2$$

a cylindrical damper

$$a = 8(2\mu - 3)^2,\ b = 6\left\{4(2\mu - 3)\tilde{\omega}^2 + 3\left[2(2\mu - 3) + 3\tilde{\beta}^2\right]\right\}, \quad (12)$$
$$c = 4\left\{9 + 4\left[\tilde{\omega}^2 - 3\tilde{\beta}^2 + 2(3 - \mu)\right]\tilde{\omega}^2\right\},\ d = 8\left[\left(\tilde{\beta}^2 - 2\right)\tilde{\omega}^2 - 3\right]\tilde{\omega}^2$$

Further

$$Q = \frac{\left(\frac{b}{a}\right)^2 - 3\frac{c}{a}}{9},\ R = \frac{2\left(\frac{b}{a}\right)^3 - 9\frac{bc}{a^2} + 27\frac{d}{a}}{54}. \quad (13)$$

$$S = Q^3 - R^2,\ \varphi = \frac{1}{3}ar\cos\left(\frac{R}{\sqrt{Q^3}}\right)$$

The roots of the cubic equation (10):

$$x_1 = -2\sqrt{Q}\cos(\varphi) - \frac{b}{3a},\ x_2 = -2\sqrt{Q}\cos\left(\varphi + \frac{2}{3}\pi\right) - \frac{b}{3a}. \quad (14)$$
$$x_3 = -2\sqrt{Q}\cos\left(\varphi - \frac{2}{3}\pi\right) - \frac{b}{3a}$$

Thus, the stationary points of the denominator expressions under the root (4) and (5) shall be in accordance with frequency:

$$\tilde{\Omega}_1 = \sqrt{-2\sqrt{Q}\cos(\varphi) - \frac{b}{3a}}, \quad (15)$$
$$\tilde{\Omega}_2 = \sqrt{-2\sqrt{Q}\cos\left(\varphi + \frac{2}{3}\pi\right) - \frac{b}{3a}},\ \tilde{\Omega}_3 = \sqrt{-2\sqrt{Q}\cos\left(\varphi - \frac{2}{3}\pi\right) - \frac{b}{3a}}$$

For values $\tilde{\omega} = 1.2$, $\mu = 0.1$, $\tilde{\beta} = 0.05$ and $U_0 = 0.05$ two frequencies have been calculated corresponding to the resonant fixed points:

for a pendulum damper

$$\tilde{\Omega}_1 = 0.932,\ \tilde{\Omega}_2 = 1.357, \quad (16)$$

for a cylindrical damper

$$\tilde{\Omega}_1 = 0.883,\ \tilde{\Omega}_2 = 1.147. \quad (17)$$

Let us solve the cubic equation obtained by trigonometric formulas Cardano.

Further

$$p = -\frac{b^2}{3a^2} + \frac{c}{a},\ q = \frac{2b^3}{27a^3} - \frac{bc}{3a^2} + \frac{d}{a},\ Q = \left(\frac{p}{3}\right)^3 + \left(\frac{q}{2}\right)^2. \quad (18)$$
$$\alpha = \sqrt[3]{-\frac{q}{2} + \sqrt{Q}},\ \beta = \sqrt[3]{-\frac{q}{2} - \sqrt{Q}}$$

The roots of the cubic equation (10):

$$y_1 = \alpha + \beta,\quad y_2 = -\frac{\alpha + \beta}{2} + i\frac{\alpha - \beta}{2}\sqrt{3}, \quad (19)$$
$$y_3 = -\frac{\alpha + \beta}{2} - i\frac{\alpha - \beta}{2}\sqrt{3}$$
$$x_1 = y_1 - \frac{b}{3a},\quad x_2 = y_2 - \frac{b}{3a},\quad x_3 = y_3 - \frac{b}{3a}$$

Thus, the stationary points of the denominator of expressions under the root (4) and (5) shall be in accordance with frequency:

$$\tilde{\Omega}_1 = \sqrt{x_1},\quad \tilde{\Omega}_2 = \sqrt{x_2},\quad \tilde{\Omega}_3 = \sqrt{x_3} \quad (20)$$

For values $\tilde{\omega} = 1.2$, $\mu = 0.1$, $\tilde{\beta} = 0.05$ and $U_0 = 0.05$ two frequencies corresponding to the resonant fixed points have been calculated:

for a pendulum damper

$$\tilde{\Omega}_1 = 0.932,\ \tilde{\Omega}_2 = 1.357, \quad (21)$$

for a cylindrical damper

$$\tilde{\Omega}_1 = 0.883,\ \tilde{\Omega}_2 = 1.147. \quad (22)$$

Calculations carried out by the formulas of Viet and Cardano matched, confirming their correctness and absence of non-physical solutions.

Figure 2 shows the constructed amplitude-frequency characteristics with the help of the software package MathCad, corresponding to the relation (4) and (5) and consistent with those obtained [1].

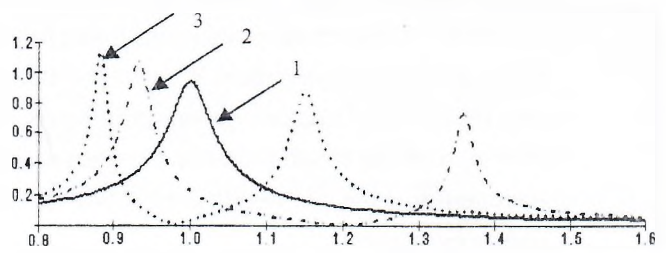

Fig. 2. Amplitude frequency characteristics: 1- a massive body without a damper, 2-a pendulum damper, 3-a cylinder damper.

Using the obtained relation let us set the task to shift the resonance peaks of the cylindrical oscillation damper to the left, i.e. towards lower frequencies, and calculate the value $\widetilde{\omega}, \mu, \widetilde{\beta}$ and the corresponding value of *m, R, в*, which follow from the formula (3). Note that, as follows from formula (3), values $\widetilde{\omega} = 1.2, \mu = 0.1, \widetilde{\beta} = 0.05$ meet such a constructive and mechanical parameters of a damper M = 20 kg, m = 2,222 kg, R = 1.2 m, r0 = 0,134 m, в = 2,806 Ns / m k = 142,075 N / m, g = 9.81 m/s2.

For the calculation $\widetilde{\omega}, \mu, \widetilde{\beta}$ we apply operators of software package MathCad.

The algorithm:

Let us shift resonant frequencies to the left $\widetilde{\Omega}_1 = 0.85$, $\widetilde{\Omega}_2 = 1.1$ at $\widetilde{\Omega}_3 = 1.026$

Then $\widetilde{\Omega}_1^2 = 0.722$, $\widetilde{\Omega}_2^2 = 1.21$, $\widetilde{\Omega}_3^2 = 1.053$.

The initial values $\omega := 1.2$, $\mu := 0.1$, $\beta := 0.05$

Given

$$\frac{6 \cdot [4 \cdot (2 \cdot \mu - 3) \cdot \omega^2 + 3 \cdot [2 \cdot (2 \cdot \mu - 3) + 3 \cdot \beta^2]]}{8 \cdot (2 \cdot \mu - 3)^2} = -(\widetilde{\Omega}_1^2 + \widetilde{\Omega}_2^2 + \widetilde{\Omega}_3^2)$$

$$\frac{4 \cdot [9 + 4 \cdot [\omega^2 - 3 \cdot \beta^2 + 2 \cdot (3 - \mu)] \cdot \omega^2]}{8 \cdot (2 \cdot \mu - 3)^2} = \widetilde{\Omega}_1 \cdot \widetilde{\Omega}_2 + \widetilde{\Omega}_1 \cdot \widetilde{\Omega}_3 + \widetilde{\Omega}_2 \cdot \widetilde{\Omega}_3$$

$$\frac{8 \cdot [(\beta^2 - 2) \cdot \omega^2 - 3] \cdot \omega^2}{8 \cdot (2 \cdot \mu - 3)^2} = -\widetilde{\Omega}_1 \cdot \widetilde{\Omega}_2 \cdot \widetilde{\Omega}_3$$

$$\begin{pmatrix} \omega \\ \mu \\ \beta \end{pmatrix} := Find(\omega, \ \mu, \ \beta). \quad \begin{pmatrix} \omega \\ \mu \\ \beta \end{pmatrix} = \begin{pmatrix} 1.13 \\ 0.117 \\ 0.171 \end{pmatrix}$$

The end of the algorithm.

Let us calculate the structural and mechanical parameters of the cylindrical damper that correspond to the new value, $\widetilde{\omega} = 1.13$ $\mu = 0.117$ $\widetilde{\beta} = 0.171$

$$m = M\frac{\mu}{1-\mu} = 20\frac{1{,}13}{1-1{,}13} = 2{,}639\text{kg}$$

$$R = \frac{M+m}{\omega^2} \cdot \frac{g}{k} + r_0 = \frac{20+2{,}639}{1{,}13^2} \cdot \frac{9.81}{142{,}075} + 0{,}134 = 1{,}358\text{m}$$

$$\beta = \widetilde{\beta}\sqrt{k(M+m)} = 0{,}171\sqrt{142{,}075(20+2{,}639)} = 9{,}723 \text{ Ns / m}$$

Other mechanical and design parameters of the damper remain constant M=20 kg, r_0=0.134m, k=142.075 N/m, g=9,81 m/s^2.

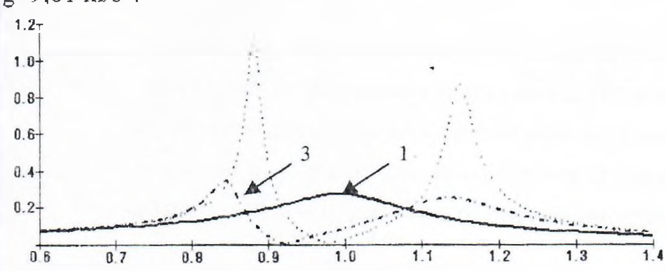

Fig.3. Amplitude frequency characteristics: 1- a massive body without damper; 2- a cylinder damper with parameters $\widetilde{\omega} = 1.2$, $\mu = 0.1, \widetilde{\beta} = 0.05$; 3 – a cylinder damper after changing parameters $\widetilde{\omega} = 1.13, \mu = 0.117, \widetilde{\beta} = 0.171$.

Figure 3 shows the constructed amplitude frequency characteristics: of a cylindrical oscillation damper before changes in mechanical parameters ($\widetilde{\omega} = 1.2$, $\mu = 0.1, \widetilde{\beta} = 0.05$), and after changing parameters ($\widetilde{\omega} = 1.13, \mu = 0.117, \widetilde{\beta} = 0.171$).

III. Conclusion

As can be seen from Fig. 2 a pendulum damper shifts one resonant frequency to the left, another to the right because of resonance frequency of the body without a damper, cylindrical damper acts similarly. It distances left peak further to the left and right a little closer to the right.

Fig. 3 shows that the change in mechanical and structural parameters of a damper led to a shift in its resonant frequency to the left and amplitude also decreased. Analytical formulas (11) - (14) obtained in this work represent an algorithm for calculating the resonant frequency of the body with the dampers and can be applied:

1. to determine the parameters $\widetilde{\omega}$, μ, $\widetilde{\beta}$ of the damper of mechanical vibrations with predetermined values $\widetilde{\Omega}_1$ and $\widetilde{\Omega}_2$;
2. according to calculated parameters $\widetilde{\omega}$, μ, $\widetilde{\beta}$ further structural and mechanical characteristics of the damper can be optimized: m - mass damper, r_0 - the radius of a cylindrical absorber, R - the radius of the cylindrical surface, Fig. 1;

for more effective damping it is expedient to change the shape of the absorber [4] leaving unchanged mass m, make dampers in a form of dumbbell with a moment of inertia that will provide the necessary parameters $\widetilde{\omega}, \mu, \widetilde{\beta}$.

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