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Stability Of Nonlinear Vibrations Of Plate With Dynamic Absorber Under Kinematic Excitations

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Abstract

In this article is about the study of stability of nonlinear vibrational motion of a plate under the influence of kinematic excitation with a dynamic absorber. Elastic dissipative properties of hysteresis type material of plate and dynamic absorbing element materials are based on the Pisarenko-Boginich hypothesis [1]. Stability conditions of the vibrations of plate were obtained and numerical calculations were performed to determine the borders and fields of stability for different values of the system parameters, and analytical conclusions were given.

Keywords stability, kinematic excitations,

Let us examine whether stability of the vibrational motion of the plate with dynamic absorber under the kinematic excitations. In order to determine the preference conditions through analytical solutions, we substitute nonlinear functions that represent the dissipative properties of the plate material and the dynamically absorbing element material with linear complex functions using harmonic linearization [2].

Differential equations of motion of the plate and the dynamic absorber are given as follows: [2]:

$$\begin{gathered}\ddot{x}_{ik} + (N_1 + jN_2 sign(\omega))p_{ik}^2 x_{ik} - (r_1 + jr_2 sign(\omega))d_{3ik}u_{ik0}x_2 = -d_{ik}W_0; \\ u_{ik0}\ddot{x}_{ik} + \ddot{x}_2 + (r_1 + jr_2 sign(\omega))n^2 x_2 = -W_0,\end{gathered} \tag{1}$$

where x_{ik} are displacements of ik points of plate; x_2 is displacement of dynamic absorber relative to the point of plate on which it is installed; $j^2 = -1$; $n = \sqrt{\frac{c}{m}}$ - natural frequency of the dynamic absorber; c, m are stiffness and mass of the elastic damping

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element of the dynamic absorber, respectively; ρ, h are density and thickness of the plate respectively; $d_{ik} = \frac{d_{1ik}}{d_{2ik}}$; $d_{1ik} = \iint_s u_{ik} dxdy$; $d_{2ik} = \iint_s u_{ik}^2 dxdy$; $d_{3ik} = \frac{c}{\rho h d_{2ik}}$;

$u_{ik} = u_{ik}(x, y)$ are natural modes of the plate;

$u_{ik0} = u_{ik0}(x_0, y_0)$ are quantities of natural modes of the plate at a point (x_0, y_0) which dynamic absorber is installed;

$N_1 = 1 - (c_0 + T_1)\eta_1 - (T_2 + T_3)\nu_1$; $N_2 = (c_0 + T_1)\eta_2 + +(T_2 + T_3)\nu_2$;

$\eta_1, \eta_2, \nu_1, \nu_2$ are harmonic linearization coeffisients of dissipative characteristics of the material of the plate;

$r_1 = 1 - \theta_1 (D_0 + F)$; $r_2 = \theta_2 (D_0 + F)$; $F = \sum_{n^*=1}^{s_1} D_{n^*} x_{2a}{}^{n^*}$;

θ_1, θ_2 are harmonic linearization coeffisients of dissipative characteristics of the material of the elament of dynamic absorber;

D_0 and D_{n^*} $(n^* = 1, \dots, s_1)$ are parameters that depend on material of dynamic absorber and are determined by experiment [3].

$$T_1 = \frac{3D}{d_{2ik}\rho h p_{ik}^2} \sum_{i_1}^{r} \frac{c_{i_1} x_{ika}^{i_1} h^{i_1}}{2^{i_1}(i_1+3)} \left[\iint_s u_{ik} \left(\frac{\partial^2}{\partial x^2} \left(\alpha_{11} |\alpha_{11}|^{i_1} \right) + \frac{\partial^2}{\partial y^2} \left(\alpha_{22} |\alpha_{22}|^{i_1} \right) \right) dxdy \right];$$

$$T_2 = \frac{6D(1-\mu)}{d_{2ik}\rho h p_{ik}^2} \sum_{i_2}^{s_2} \frac{k_{i_2} x_{ika}^{i_2} h^{i_2}}{2^{i_2}(i_2+3)} \left[\iint_s u_{ik} \left(\frac{\partial^2}{\partial x \partial y} \left(\alpha_{33} |\alpha_{33}|^{i_2} \right) \right) dxdy \right];$$

$$T_3 = \frac{2D(1-\mu)}{d_{2ik}\rho h p_{ik}^2} k_0 \iint_s u_{ik} \frac{\partial^2}{\partial x \partial y} (\alpha_{33}) dxdy;$$

$c_{i_1} (i_1 = 0, \dots, r)$ and $k_{i_2} (i_2 = 0, \dots, s_2)$ are parameters that depend on material of the plate and are determined by experiment [3];

$$\alpha_{11} = \frac{\partial^2 u_{ik}}{\partial x^2} + \mu \frac{\partial^2 u_{ik}}{\partial y^2}; \ \alpha_{22} = \frac{\partial^2 u_{ik}}{\partial y^2} + \mu \frac{\partial^2 u_{ik}}{\partial x^2}; \ \alpha_{33} = \frac{\partial^2 u_{ik}}{\partial x \partial y};$$

μ is the Poisson ratio; $D = \frac{Eh^3}{12(1-\mu^2)}$ is the cylindrical stiffness of the plate; E is the Young modulus; p_{ik} are natural frequencies of the plate; ω is frequency of the system; $W_0 = W_0(t)$ is the acceleration of motion of the support.

The stability will be examined by Lyapunov's method [4] of the motion of the plate which is protected from vibrations under the kinematic excitations. For this purpose, we will seek solutions of differential equations of the system which is protected from vibrations as following [5]:

$$x_{ik} = x_{ika}(t)\cos(\omega t + \beta_{ik}(t));$$
$$x_2 = x_{2a}(t)\cos\left(\omega t + \beta_2(t)\right), \tag{2}$$

where $x_{ika}(t)$, $x_{2a}(t)$, $\beta_{ik}(t)$, $\beta_2(t)$ coefficients are the amplitude values of the variables x_{ik} and x_2, respectively, and are slowly changing functions.

We will explore the expression of the base acceleration in the system of differential equations (1) for harmonic excitations as follows:

$$W_0 = \varepsilon\xi \cos \omega t, \tag{3}$$

where $\varepsilon\xi$ is amplitude of harmonic excitations; ε is small parameter.

We calculate the corresponding derivatives of expressions (2) and putting them and (3) the expression of the base acceleration into (1) the system of equations, having the following system of first order differential equations:

$$\dot{x}_{ika} = \frac{1}{2\omega}\left(d_{ik}\varepsilon\xi sin\beta_{ik} - p_{ik}^2 x_{ika}N_2 + d_{3ik}u_{ik0}x_{2a}(r_1 sin\varphi + r_2 cos\varphi)\right);$$
$$\dot{\beta}_{ik} = \frac{1}{2\omega x_{ika}}\left(d_{ik}\varepsilon\xi cos\beta_{ik} + x_{ika}(p_{ik}^2 N_1 - \omega^2) + d_{3ik}u_{ik0}x_{2a}(r_2 sin\varphi - r_1 cos\varphi)\right);$$
$$\dot{x}_{2a} = \frac{1}{2\omega}\left(b_2\varepsilon\xi sin\beta_2 - r_2 x_{2a}b_3 - p_{ik}^2 u_{ik0}x_{ika}(N_1 sin\varphi - N_2 cos\varphi)\right); \tag{4}$$
$$\dot{\beta}_2 = \frac{1}{2\omega x_{2a}}\left(b_2\varepsilon\xi cos\beta_2 + b_3 x_{2a}r_1 - x_{2a}\omega^2 - p_{ik}^2 u_{ik0}x_{ika}(N_1 cos\varphi + N_2 sin\varphi)\right),$$

where $b_1 = n^2 d_{ik} + u_{ik0}d_{3ik}$; $b_2 = 1 - u_{ik0}d_{ik}$; $b_3 = n^2 + u_{ik0}^2 d_{3ik}$, $\varphi = \beta_2 - \beta_{ik}$.

When we determine the characteristic equation of the system, normal equations of the system will be varied. Consequently, if we varied the system of differential equations of the normal form (4), it can be written with respect to variations. As a result, the characteristic equation of the system is determined as follows [6]:

$$\lambda_1^4 + S_1\lambda_1^3 + S_2\lambda_1^2 + S_3\lambda_1 + S_4 = 0, \tag{5}$$

where $S_1 = -(a_{11} + a_{22} + a_{33} + a_{44})$;
$S_2 = a_{22}a_{44} + a_{33}a_{44} + a_{22}a_{33} + a_{11}(a_{22} + a_{33} + a_{44}) - a_{24}a_{42} - a_{23}a_{32} - a_{34}a_{43} - a_{12}a_{21} - a_{13}a_{31} - a_{14}a_{41}$;
$S_3 = a_{11}(a_{24}a_{42} + a_{23}a_{32} + a_{34}a_{43} - a_{22}a_{44} - a_{33}a_{44} - a_{22}a_{33}) + a_{22}(a_{34}a_{43} - a_{33}a_{44}) + a_{23}(a_{32}a_{44} - a_{34}a_{42}) + a_{24}(a_{42}a_{33} - a_{32}a_{43}) + a_{12}a_{21}(a_{33} + a_{44}) + a_{13}a_{31}(a_{22} + a_{44}) + a_{14}a_{41}(a_{22} + a_{33}) - a_{12}(a_{24}a_{41} + a_{31}a_{23}) - a_{13}(a_{21}a_{32} + a_{34}a_{41}) - a_{14}(a_{31}a_{43} + a_{21}a_{42})$;

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$S_4 = a_{11}(a_{22}(a_{33}a_{44} - a_{43}a_{34}) + a_{23}(a_{34}a_{42} - a_{32}a_{44}) + a_{24}(a_{32}a_{43} - a_{42}a_{33})) + a_{12}(a_{21}(a_{33}a_{44} - a_{43}a_{34}) + a_{23}(a_{34}a_{41} - a_{31}a_{44}) + a_{24}(a_{31}a_{43} - a_{41}a_{33})) + a_{13}(a_{21}(a_{32}a_{44} - a_{42}a_{34}) + a_{22}(a_{34}a_{41} - a_{31}a_{44}) + a_{24}(a_{31}a_{42} - a_{41}a_{32})) - a_{14}(a_{21}(a_{32}a_{43} - a_{42}a_{33}) + a_{22}(a_{33}a_{41} - a_{31}a_{43}) + a_{23}(a_{31}a_{42} - a_{41}a_{32}))$;
$a_{11} = -p_{ik}^2\frac{\partial(x_{ika}N_2)}{\partial x_{ika}}$; $a_{12} = (\omega^2 - p_{ik}^2 N_1)x_{ika}$; $a_{13} = \left(\frac{\partial(x_{2a}r_1)}{\partial x_{2a}}sin\varphi + \frac{\partial(x_{2a}r_2)}{\partial x_{2a}}cos\varphi\right) d_{3ik}u_{ik0}$; $a_{14} = (r_1 cos\varphi - r_2 sin\varphi)x_{2a}d_{3ik}u_{ik0}$;
$a_{21} = \left(p_{ik}^2\frac{\partial(x_{ika}N_1)}{\partial x_{ika}} - \omega^2\right)\frac{1}{x_{ika}}$; $a_{22} = -p_{ik}^2 N_2$; $a_{23} = \left(\frac{\partial(x_{2a}r_2)}{\partial x_{2a}}sin\varphi - \frac{\partial(x_{2a}r_1)}{\partial x_{2a}}cos\varphi\right)\frac{d_{3ik}u_{ik0}}{x_{ika}}$; $a_{24} = (r_2 cos\varphi + r_1 sin\varphi)\frac{x_{2a}d_{3ik}u_{ik0}}{x_{ika}}$;
$a_{31} = \left(\frac{\partial(x_{ika}N_2)}{\partial x_{ika}}cos\varphi - \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}sin\varphi\right)p_{ik}^2 u_{ik0}$; $a_{32} = (N_1 cos\varphi - N_2 sin\varphi)x_{ika}p_{ik}^2 u_{ik0}$; $a_{33} = -b_3\frac{\partial(x_{2a}r_2)}{\partial x_{2a}}$; $a_{34} = (\omega^2 - b_3 r_1)x_{2a}$;
$a_{41} = -\left(\frac{\partial(x_{ika}N_1)}{\partial x_{ika}}cos\varphi + \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}sin\varphi\right)\frac{p_{ik}^2 u_{ik0}}{x_{2a}}$; $a_{42} = (N_2 cos\varphi - N_1 sin\varphi)\frac{x_{ika}p_{ik}^2 u_{ik0}}{x_{2a}}$; $a_{43} = \left(b_3\frac{\partial(x_{2a}r_1)}{\partial x_{2a}} - \omega^2\right)\frac{1}{x_{2a}}$; $a_{44} = -b_3 r_2$; $\lambda_1 = 2\omega\lambda$.

It is enough that the real part of the root of characteristic equation (5) must be negative sign for being stable of vibrations of vibroprotected system. According to the Hurwitz criterion it is possible to show that the real part of the characteristic equation is negative. Hurwitz criterion is following for given system:

$$S_1 > 0,\ S_2 > 0,\ S_3 > 0,\ S_4 > 0,\ S_1S_2S_3 - S_1^2S_4 - S_3^2 > 0, \qquad (6)$$

The following general conclusion is relevant for the inequalities (6) obtained by Lyapunov's method:

These inequalities allow to investigate borders, conditions and fields of stability of vibrations of the hysteresis type plate and the dynamical absorber at different values of the system parameters.

If the dynamic absorber has elastic characteristics in the system (6) the $S_{1*}, S_{2*}, S_{3*}, S_{4*}$ coefficients will be as following:

$S_{1*} = p_{ik}^2\left(N_2 + \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right)$;

$S_{2*} = \left(N_1\frac{\partial(x_{ika}N_1)}{\partial x_{ika}} + N_2\frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right)p_{ik}^4 - \left(N_1 + \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}\right)\omega^2 p_{ik}^2 + \omega^4 + \left(n^2 + u_{ik0}^2 d_{3ik} - \omega^2\right)^2 + \left(N_1 + \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}\right)p_{ik}^2 u_{ik0}^2 d_{3ik}$;

$S_{3*} = \left(N_2 + \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right)\left((n^2 - \omega^2)^2 + n^2 u_{ik0}^2 d_{3ik}\right)p_{ik}^2$;

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$S_{4*} = \left(N_1 \frac{\partial(x_{ika}N_1)}{\partial x_{ika}} + N_2 \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right)(n^2 - \omega^2)^2 p_{ik}^4 - \left(N_1 + \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}\right) p_{ik}^2 (n^2 - \omega^2)\omega^2(\omega^2 - n^2 - u_{ik0}^2 d_{3ik}) + \omega^4 (\omega^2 - n^2 - u_{ik0}^2 d_{3ik})^2.$

As a result in (6), $S_{1*} > 0$, $S_{3*} > 0$ inequalities are always relevant. $S_{1*} = p_{ik}^2 \left(N_2 + \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right)$ is positive because of $p_{ik}^2 > 0$, $N_2 + \frac{\partial(x_{ika}N_2)}{\partial x_{ika}} > 0$ and N_2 determines the energy dissipation in the material of the plate depending on the amplitude and is a positive growing function and $N_2 x_{ika}$ is also positive. For S_{3*} all variables are positive so that $S_{3*} > 0$.

The following inequality, written in general terms, can express the remaining three inequalities:

$$\left(N_1 \frac{\partial(x_{ika}N_1)}{\partial x_{ika}} + N_2 \frac{\partial(x_{ika}N_2)}{\partial x_{ika}}\right) X^2 - \left(N_1 + \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}\right) XY + Y^2 + Q^2 > 0 \qquad (7)$$

For $S_{2*} > 0$,
$X = p_{ik}^4$; $Y = \omega^2$; $Q = (\omega^2 - n^2 - u_{ik0}^2 d_{3ik})^2 + \left(N_1 + \frac{\partial(x_{ika}N_1)}{\partial x_{ika}}\right) p_{ik}^2 u_{ik0}^2 d_{3ik}$;
For $S_{4*} > 0$,
$X = (\omega^2 - n^2)p_{ik}^2$; $Y = (\omega^2 - n^2 - u_{ik0}^2 d_{3ik})\omega^2$; $Q = 0$;
For $S_{1*}S_{2*}S_{3*} - S_{1*}^2 S_{4*} - S_{3*}^2 > 0$,
$X = p_{ik}^4$; $Y = 2\omega^2 - n^2 - u_{ik0}^2 d_{3ik}$; $Q = 0$.

In inequality (7), there is a quadratic form with respect to X, Y, and $Q > 0$. For this quadratic form to be positive, the following conditions are sufficient:

$$(N_1 - \frac{\partial(x_{ika}N_1)}{\partial x_{ika}})^2 + 4N_2 \frac{\partial(x_{ika}N_2)}{\partial x_{ika}} < 0. \qquad (8)$$

This obtained condition (8) allows to determine the stability, border and field of stability of the nonlinear stationary vibrations of hysteresis type plate with linear elastic characteristic dynamic absorber at different values of the system parameters.

Numerical calculations and analysis of results. The results were obtained from polymer material epoxyurethane *KDU-2* as a material of a 45-sheet steel plate with two hinged sides and free with two sides. Numerical calculations were performed on the following values of constructive parameters [7].

$h = 0.001\,м$; $a = b = 0.19\,м$; $E = 2.08 \cdot 10^{11} N/м^2$; $G = 0.8 \cdot 10^{11} N/м^2$;
$\mu = 0.3$; $\rho = 7810\,kg/м^3$; $x_0 = a/2$; $y_0 = b/2$; $(i = k = 1)$;
$u_{ik}(x,y) = u_{11}(x,y) = sin\frac{\pi x}{0.19} sin\frac{\pi y}{0.19}$; $d_{1ik} = d_{111} = 4 \cdot 0.19^2/\pi^2$;
$d_{2ik} = d_{211} = 0.19^2/4$;
$\eta_1 = 142.717 x_{11a} + 22660477 x_{11a}^2 - 1.74149 \cdot 10^{12} x_{11a}^3$;

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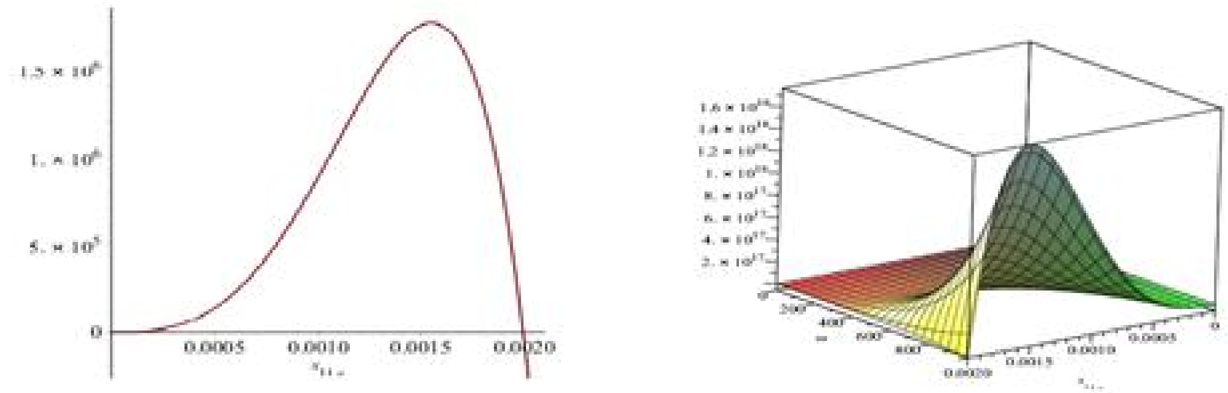
$\eta_2 = 60.57097 x_{11a} + 9617405 x_{11a}^2 - 27.389243 \cdot 10^{11} x_{11a}^3;$
$\nu_1 = 0.613245 x_{11a} - 62.7862 x_{11a}^2 + 4240.29 x_{11a}^3;$
$\nu_2 = 0.26027 x_{11a} - 26.6473 x_{11a}^2 + 1799.635 x_{11a}^3; p_{11} = 847.48\ c^{-1}; K = \frac{m}{\rho abh}.$



a) **b)**

Fig.1-a),b) Graphics are $S_{1^*} = p_{ik}^2 (N_2 + \frac{\partial (x_{ika} N_2)}{\partial x_{ika}})$ **and** $S_{3^*} = (N_2 + \frac{\partial (x_{ika} N_2)}{\partial x_{ika}})((n^2 - \omega^2)^2 + n^2 u_{ik0}^2 d_{3ik}) p_{ik}^2$ **from conditions of stability of vibrations of plate with hysteretic characteristics and dynamic absorber with elastic characteristics**

It is possible that to show nonlinear vibrations of plate which is protected from vibrations will be stable when the roots of characteristic equations are negative sighn, namely S_{1^*}, S_{3^*} inequalities are always relevant from Fig.1-a),b). The graph of the condition S_{1^*} shows the numerical values of the amplitude intervals corresponding to their area of positive values. From the graph of condition S_{3^*} it is possible to specify numerical values of amplitudes and frequency intervals corresponding to the sphere of positive values. This will allow us to practically check further conditions of the Hurwits criterion.

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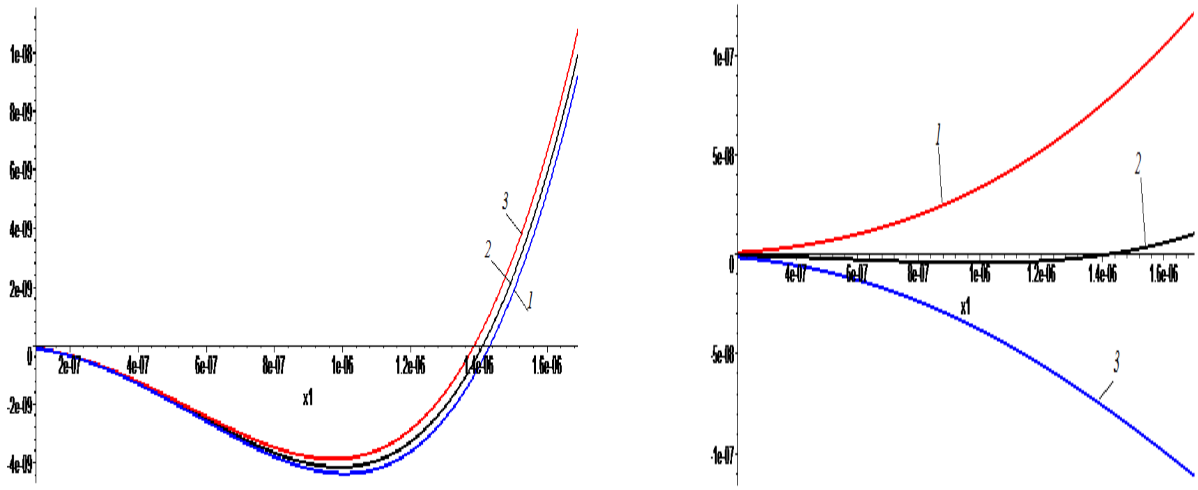

a) **b)**

Fig.2-a),b) Changing of conditions of stability at different values of $\nu_1, \nu_2, \eta_1, \eta_2$ parameters.

In Fig.2,a) is described that condition of stability of nonlinear vibrations of vibroprotected system when $\nu_1 = 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Blue), $\nu_1 \neq 0, \nu_2 = 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Red), $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Black). From these graphs it is possible to determine the range of amplitudes that corresponds to the system's stable and unstable vibration. $[2 \cdot 10^{-7}; 1.3 \cdot 10^{-6}]$ interval of amplitude belongs to stable vibrations, $[1.5 \cdot 10^{-6}; 2 \cdot 10^{-6}]$ interval of amplitude belongs to unstable vibrations. In addition we can see that from the Fig.2,a) for the interval $[1 \cdot 10^{-7}; 1.5 \cdot 10^{-6}]$ of amplitude $\nu_1 = 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Blue) is general interval of amplitude of nonlinear vibrations of plate which is protected from vibrations, $\nu_1 \neq 0, \nu_2 = 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Red) and $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Black) are situated in $\nu_1 = 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Blue).

In Fig.2,b) is described that condition of stability of nonlinear vibrations of vibroprotected system when $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 = 0, \eta_2 \neq 0$ (Blue), $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 = 0$ (Red), $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Black). From these graphs it is possible that $[1 \cdot 10^{-7}; 1.7 \cdot 10^{-6}]$ interval of amplitude belongs to unstable vibrations of plate when $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 = 0$ (Red) and stable vibrations of plate when $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 = 0, \eta_2 \neq 0$ (Blue). That interval belongs to stable and unstable when $\nu_1 \neq 0, \nu_2 \neq 0, \eta_1 \neq 0, \eta_2 \neq 0$ (Black).

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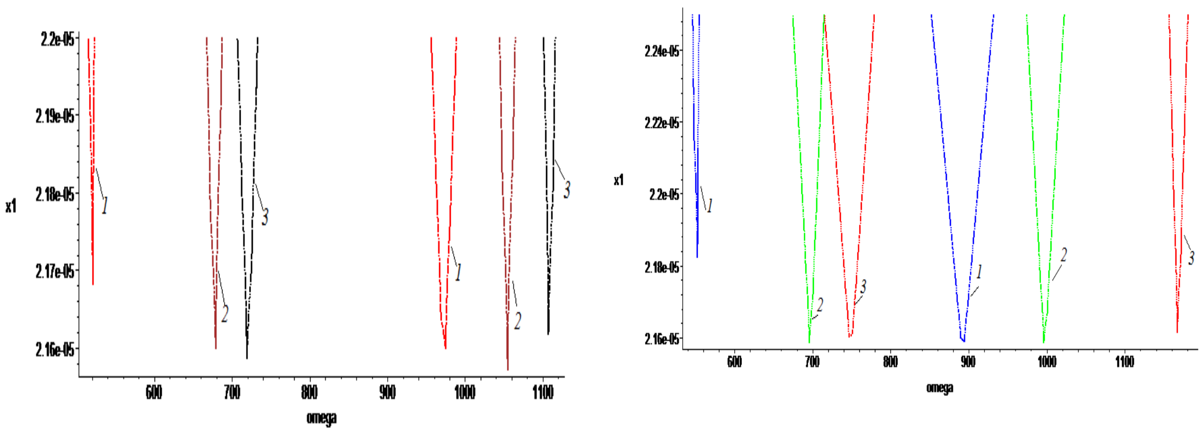

a) b)

Fig. 3-a),b). Changing of borders of stability at different values of K and c parameters.

In Fig.3,a) is described that the variation of the borders and fields of stability of the nonlinear vibrations of the plate according to the ratio of dynamic absorber mass to the plate mass when $c = 10^5$ N/$м$. If the ratio $K = 0{,}04$ (Black), $K = 0{,}05$ (Brown), $K = 0{,}1$ (Red) is increasing the borders and fields of unstable vibrations will move from right to left along the frequency axis. From this we can conclude that low-frequency vibrations in the range of amplitudes are considered to have unstable vibrations, that is, the field of stability can be clearly specified.

In Fig.3,b) is described that the variation of the borders and fields of stability of the nonlinear vibrations of the plate according to the stiffness of dynamic absorber when $K = 1/30$. If $c = 10^{4.5}; 10^{4.8}; 10^5$ N/$м$ (Blue; Green; Red) is increasing the borders and fields of unstable vibrations will move from left to right along the frequency axis. Hence, it is concluded that high-frequency vibrations in the amplitudes range are considered to cause unstable vibrations.

Conclusion. Borders and fields of stability of the plate's nonlinear vibrations which is protected from vibrations vary depending on the dynamic absorber's natural frequency, namely meaning that the decrease in this parameter relative to the high intensity of the dynamic absorber will result in the loss of stability boundaries and fields at low frequencies, that is, unstable vibrations. As the natural frequency increases, borders and fields of stability are shifted from right to left along the frequency axis, and high frequency vibrations cause unstable vibrations.

At sufficiently small values of the parameter representing the dissipative properties of the elastic damping element of the dynamical absorber, the priority limits and areas of the system are within the borders and fields of stability achieved at relatively large parameter values.

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The borders and fields of stability achieved in relatively large values are the borders and fields of general stability.

From the foregoing, it can be concluded that the obtained stability conditions allow the correct selection of the dynamic absorber's frequency, dissipative characteristics and other constructive parameters to ensure the stability of the linear vibrations of the plate, which is protected from vibration, and the high efficiency of the dynamic absorber.

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