Optimal synthesis of the impulsive resonant two-mass vibro-impact systems

Volodymyr Gursky, Sergiy Murashev, Rostyslav Gogol

Department of Mechanics and Automation Engineering (MAE), Lviv Polytechnic National University, UKRAINE, Lviv, S. Bandery street 12, E-mail: gursky@polynet.lviv.ua

Abstract* – *The article solves the applied problem of the technological efficiency increase of resonant two-mass vibrating machines with an impulsive electromagnetic drive based on optimization synthesis and frequency analysis of asymmetric piece-linear elastic description. The value of speed of working mass is accepted for an objective function. Synthesis is based by frequency coefficient and relation of eigenfrequencies. The rational resilient parameters of asymmetric piece-linear elastic description settle accounts after the optimum values of the proper coefficients. The availability of the poly-frequency spectrum in the resonant mode of machine, operated by impulsive electromagnetic perturbation, is confirmed by the dynamic analysis and numerical system modeling.

Key words – mechanical resonant system, vibratory screen, vibro-impact system, electromagnetic drive, two-mass oscillation system.

I. Introduction

Technologically effective vibration machines and tent-bed dynamic tests used in machine-building, build, processing, mountain industries for intensification of productions. The vibroshock machines are more technological effective [1].

Thus, due to various approaches and methods of calculation try to attain some compromise both from the point of view functioning of machine and from the side of realization of process of treatment. Therefore, to get the rational variant of machine from the point of view complex energy efficient is an important question.

II. Raising of task

The optimization synthesis of asymmetric piece-linear elastic description, based by the analysis of influence of resilient parameters on the value of speed of worker mass is done.

III. Exposition of basic material

Resonant vibratory screen (fig. 1) is presented by two-mass oscillation system with a working 1 and reactive mass 2, connected by a resilient link with the set of cylinder springs 3, previous force is regulated screw-bolt connection 4. Machine is suspended by springs 5 on a working mass. Resilient link 3 has asymmetric description (fig. 2) by different inflexibility of cylinder springs (the spring of c1 works in positive direction, in negative – c2.). Electromagnetic force has an impulsive view.

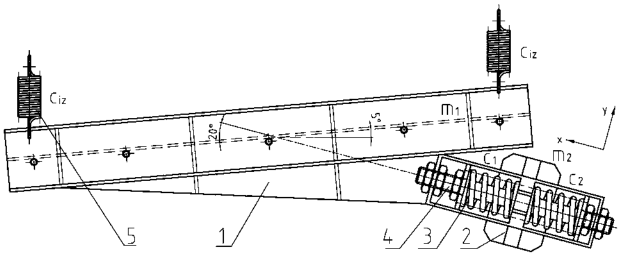

Рис. 1. The two-mass construction of the resonant vibro-impact screen

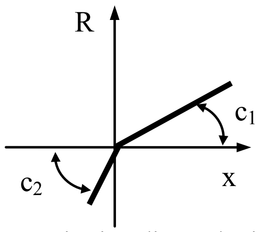

Fig. 2. Asymmetric piece-linear elastic description

Electromechanical model of the two-mass vibro-impact vibratory system with an impulsive electromagnetic drive has a view:

$$\begin{cases} \dfrac{k}{2\left(\delta_0-\left(x_1(t)-x_2(t)\right)\right)}\cdot \dot{i}(t)+\left[r+\left[\left(1-\Phi(i(t))\cdot r_d^{<+>}+r_d^{<->}\right)\right]+\dfrac{k\cdot\left(\dot{x}_1(t)-\dot{x}_2(t)\right)}{2\left(\delta_0-\left(x_1(t)-x_2(t)\right)\right)^2}\right]\cdot i(t)=U_0\sin(\omega t); \\ m_1\ddot{x}_1(t)+\begin{cases} c_1\cdot\left(x_1(t)-x_2(t)\right)+b_1\cdot\left(\dot{x}_1(t)-\dot{x}_2(t)\right), & \textbf{if } x_1(t)-x_2(t)\ge 0 \\ -c_2\cdot\left(x_1(t)-x_2(t)\right)-b_2\cdot\left(\dot{x}_1(t)-\dot{x}_2(t)\right), & \textbf{if } x_1(t)-x_2(t)< 0 \end{cases}=\dfrac{k\cdot n}{4}\left[\dfrac{i(t)}{\delta_0-\left(x_1(t)-x_2(t)\right)}\right]^2; \\ m_2\ddot{x}_2(t)-\begin{cases} c_1\cdot\left(x_1(t)-x_2(t)\right)+b_1\cdot\left(\dot{x}_1(t)-\dot{x}_2(t)\right), & \textbf{if } x_1(t)-x_2(t)\ge 0 \\ -c_2\cdot\left(x_1(t)-x_2(t)\right)-b_2\cdot\left(\dot{x}_1(t)-\dot{x}_2(t)\right), & \textbf{if } x_1(t)-x_2(t)< 0 \end{cases}=-\dfrac{k\cdot n}{4}\left[\dfrac{i(t)}{\delta_0-\left(x_1(t)-x_2(t)\right)}\right]^2. \end{cases} \quad (1)$$

Model (1) is a system of nonlinear differential equations of second order, where values of electrical and mechanical parameters:

$i(t)$ – current;

$x_1(t)$ – displacement of a worker mass;

$x_2(t)$ – displacement of a reactive mass;

$\Phi(i(t))$ – Heaviside function of current;

$r_d^{<+>}$, $r_d^{<->}$ – resistance of diode in direct and reverse direction ($r_d^{<+>} >> r_d^{<->}$);

$k=\mu_0 S w^2$; $\mu_0=4\pi\cdot 10^{-7}$ N/A^2 – permeability of air;

$n=2$ – amount of parallel included electromagnets;

$S=2{,}784\cdot 10^{-3} m^2$ – area of surface of the electro-magnet's poles;

$w = 650$ – amount of spool's coils;
$r = 18\Omega$ – active resistance of spool;
$\delta_0 = 0{,}006\text{m}$ – nominal value of air-gap;
$U_0 = 100V$ – nominal value of voltage;
$\omega = 314s^{-1}$ – angular velocity of voltage;
$b_1 = \gamma \cdot c_1 / \omega$, $b_2 = \gamma \cdot c_2 / \omega$ – coefficients of viscid friction;
$\gamma = 0{,}04$ – coefficient of internal non-elastic's of spring's material;
$m_1 = 110kg$ – inertial value of a worker mass;
$m_2 = 18kg$ – inertial value of a reactive mass.

Only the modified numerical methods are befit for untiing of the mentioned system of «hard» differential equalizations, particulary built-in in the software product of MATHCAD of BDF-function, Adams, ADAMSBDF, Radau.

Rational energy-saving resonance mode of operations of the system will be, when $\omega_0 = \omega / z$ with adjusting $z = 0{,}96\ldots0{,}98$. We needed for this purpose to calculate the value of eigenfrequency of vibrations ω_0, whot is the function of the proper resilient parameters:

$$\omega_0 = \frac{2\omega_{01} \cdot \omega_{02}}{\omega_{01} + \omega_{02}}. \tag{2}$$

where eigenfrequency are

$$\omega_{01} = \sqrt{\frac{c_1(m_1 + m_2)}{m_1 m_2}}, \ \omega_{02} = \sqrt{\frac{c_2(m_1 + m_2)}{m_1 m_2}}.$$

Thus stiffnes coefficients c_1 and c_2 remains unknown.

The rational values of eigenfrequencies are determined for correlations

$$\omega_{01} = \Theta \frac{\omega}{z}, \text{ a } \omega_{02} = \Lambda \omega_{01}. \tag{3}$$

where synthesizing parameters of model (1) is accepted to name *frequency coefficient* Θ and *relation of eigenfrequencies* Λ.

The condition (2) to providing of the resonance mode likes as

$$\omega_0 = \frac{2 \cdot \omega \cdot \Theta \cdot \Lambda}{z(\Lambda + 1)}, \tag{4}$$

or such as $\dfrac{2 \cdot \Theta \cdot \Lambda}{\Lambda + 1} = 1$.

Then, a rational ratio betweenness of synthesizing parameters will be purchase a kind

$$\Theta = \frac{\Lambda + 1}{2\Lambda}. \tag{5}$$

The value of speed of working mass V_1 is utilized for the estimation of influence synthesizing parameters.

Finding the optimum relation $\langle \Lambda \rangle$ of eigenfrequencies and proper frequency coefficient for the system $\langle \Theta \rangle$ is essence of optimization task. That's provide maximal value of the worker mass speed $V_{1j}(\Lambda_j) \to \max$.

The optimal stiffness parameters will be calculated by synthesis coefficients

$$c_1 = \frac{m_1 m_2}{m_1 + m_2} \cdot \left(\langle \Theta \rangle \frac{\omega}{z} \right)^2,$$
$$c_2 = \frac{m_1 m_2}{m_1 + m_2} \cdot \left(\langle \Lambda \rangle \frac{\langle \Theta \rangle \omega}{z} \right)^2. \tag{6}$$

Discrete change of parameter $\Lambda_j = 0{,}5 \cdot j$, $j = 1..12$ for untiing of optimization task is fixed.

Graphics dependences for an objective function $V_{1j}(\Lambda_j)$ and *frequency coefficient* $\Theta(\Lambda_j)$ are built on fig. 3. The maximal value of speed of working mass will get at such values of coefficients:

$$\langle \Lambda \rangle = 2, \ \langle \Theta \rangle = 0{,}75 = \frac{3}{4}.$$

Thus, a condition (4) is carried out.

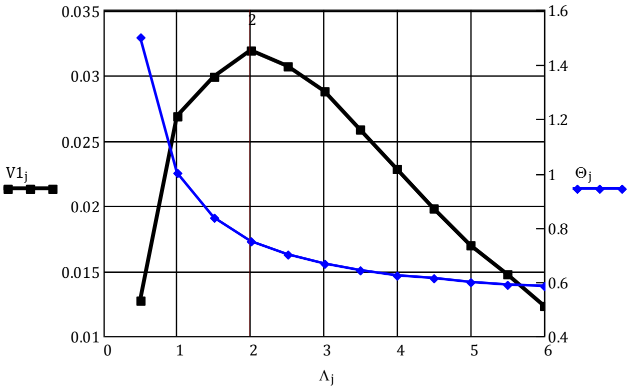

Fig. 3. 2D plot function of velocity working mass

Now we can to calculate the optimal stiffness parameters (6).

Conclusion

The optimization synthesis of stiffness parameters piece-linear elastic description of the vibro-impact system with an impulsive electromagnetic drive is conducted by a numerical analysis. The rational values of coefficients of resiliency are got, coming from the frequency analysis of the system and result of decision of optimization task. Thus, an increase of efficiency the rate of movement of working mass is 19 %, by with the use of linear resilient description of kind $c_1 - c_1$.

References

[1] I.I. Nazarenko, Yu.O. Baranov, T.F. Shcherbyna. Vykorystannya elektromahnitnykh vibratoriv na zminnomu strumi v udarno-vibratsiynykh systemakh [Using of electromagnetic vibrators with an alternating current in the shock vibratory systems], Teoriya i praktyka budivnytstva. Zbirnyk naukovykh prats'. – 2007, Vyp. 3. – Theory and practice of building. Collection of scientific labours, vol. 3, pp. 38-46, 2007.