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SELF-SYNCHRONIZATION STABILITY OF VIBRATION EXCITERS OF A TWO-MASS APPARATUS FOR PROCESSING GRANULAR MATERIALS

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Abstract. The article studies stability conditions for self-synchronization of vibration exciters in antiphase oscillation regime of working tools of an apparatus for granular materials processing. The apparatus operating regime provides grinding, attrition and mixing processing of material grains combined with a transportation process. The condition of stable antiphase rotation regime of vibration exciters' rotors has been found with implementation of integral criterion of stability of synchronous motions. The condition of stable self-synchronization rotation regime of vibrational exciters' rotors was found with the usage of integral criterion of stability of synchronous movements. The obtained relationships allow one to choose parameters of apparatus construction and to specify vibrational working tool's operating regimes that perform effective implementation of process of granular material treatment.

Keywords: two-mass apparatus, working tools walls, self-synchronization of vibrators, integral criterion of synchronization stability, granular materials

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УСТОЙЧИВОСТЬ САМОСИНХРОНИЗАЦИИ ВИБРОВОЗБУДИТЕЛЕЙ ДВУХМАССОВОГО АППАРАТА ДЛЯ ПЕРЕРАБОТКИ ЗЕРНИСТЫХ МАТЕРИАЛОВ

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Аннотация. Работа направлена на поиск условий устойчивой самосинхронизации вибраторов для противофазного режима колебаний рабочих органов аппарата для переработки зернистых сред. Режим работы аппарата обеспечивает процессы измельчения, истирания и перемешивания зерен материала, совмещенные с процессом транспортирования. С использованием интегрального критерия устойчивости синхронных движений получено условие устойчивой самосинхронизации роторов вибровозбудителей в противофазном режиме их вращения. Найденные соотношения позволяют выбирать оптимальные значения параметров конструкции аппарата и задавать режимы колебаний вибрационных рабочих органов, обеспечивающие эффективное осуществление процессов технологической переработки зернистых материалов.

Ключевые слова: двухмассовый аппарат, стенки рабочих органов, самосинхронизация вибраторов, критерий устойчивости синхронизации

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Introduction

The phenomenon of self-synchronization of unbalanced rotors of vibration exciters consists of vibration-induced interaction between them, so that they rotate synchronously at equal or commensurate speeds and with specified mutual phases [1]. This interaction is particularly important for processing of granular materials, including such stages as crushing, transportation, attrition, mixing, and others.

A key requirement for mechanical processing of a material is controlled excitation of vibration patterns. This involves generating specific directions of vibrational loads and trajectories of the machine's components. Such synthesis allows to optimize many processes.

Vibration is widely used in technologies for processing rocks, powders and other granular materials in the mining, construction, chemical and foundry industries. The main directions for improving these technological processes are reducing the energy and material consumption throughout design and production of vibration machinery.

The discovery of the self-synchronization effect led to a qualitative leap in vibration technology. Introducing this effect into engineering practices has contributed to great advances in a wide range of vibration technology in our country and abroad. Large contributions to this field were made by such outstanding scientists as Blekhman, Vaisberg, Lavrov, Ragulskis and others [1–6].

Self-synchronization of vibration exciters remains a promising direction; a class of multi-mass vibration systems has appeared, allowing to optimize various processes during operation at resonance, reduce loads on the base and implement specialized operating modes for treating granular materials using several components [1]. In view of this, multi-mass machines with self-synchronizing vibration exciters mounted on several components present an attractive option. Examples of such machines are two-mass vibration crushers, vibrating screens, equipment for beneficiation of granular materials [2, 4].

Modern systems and methods for forced synchronization control of rotors in vibration equipment allow to preset phase relationships and rotational speeds, allowing to further optimize the material processing techniques based on vibration methods [10, 12].

A promising method for vibrational processing of granular materials is impact on the material layer by the working tools of a two-mass system, forming a longitudinal channel with flat walls, with an unbalanced vibration exciter installed on each wall (Fig. 1) [9]. The rotation of the vibration exciter rotors induces plane-parallel oscillations of each wall along elliptical trajectories.

The processing of the material layer includes two main periods of impact from the walls on the layer:

compression of the layer by compressive forces when the walls approach each other (Fig. 1,*a*), accompanied by generation of stresses in the layer and intense abrasion of the surface of material grains during their relative movement within the layer;

decompression of the layer when the walls move away from each other (Fig. 1,*b*), accompanied by loosening of the layer and its relative movement along the working channel by microthrow motion.

Simultaneously with processing, the material is transported along the working channel.

An extensive monograph [1] considers the phenomenon of self-synchronization of dynamic systems, obtaining various conditions for stability of self-synchronization of vibration exciters in single-mass and multi-mass systems. Furthermore, a recent paper reports on synchronous anti-phase and in-phase rotation of rotors in vibration exciters of a two-mass system in a single assembly [11]. However, self-synchronization of rotors in such systems with vibration exciters located on both walls is insufficiently studied.

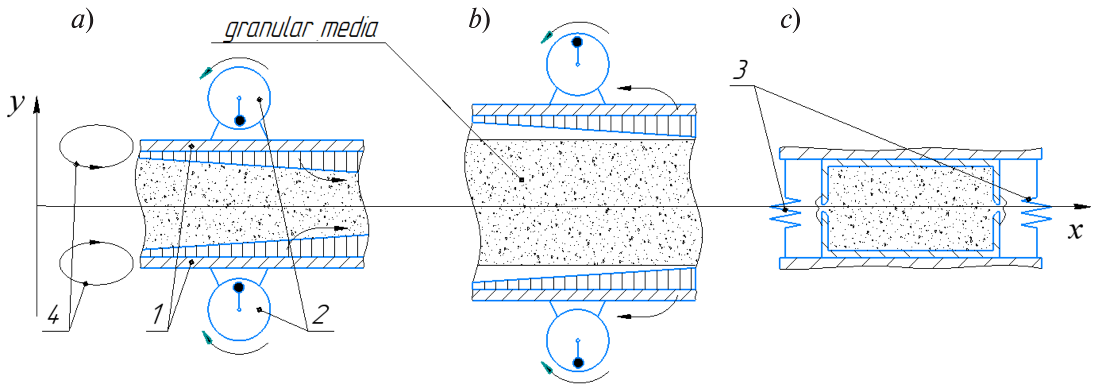

Fig. 1. Schematic for volumetric vibrational impact on layer of processed granular medium by walls of two-mass system generating synchronous anti-phase oscillations along elliptical trajectories (*4*): *a*, *b* correspond to periods of compression and decompression of the layer, respectively; *c* is the cross-section of the working channel

The figure shows walls (*1*), unbalanced vibration exciters (*2*), springs (*3*) connecting the walls; arrows indicate the movements of walls *1* and rotations of vibration exciter rotors *2*

Analysis of the operating modes in a two-mass system based on the described mathematical model allows to determine the parameters for attrition of granular material in with the highest compressive forces simultaneously with transportation. The best combination of these parameters is achieved under synchronous anti-phase oscillations of the walls along elliptical trajectories in opposite directions, maintained by self-synchronization of anti-phase counter-rotation of rotors in vibration exciters (see Fig. 1).

The goal of our study is therefore to find the optimal values of the design and operating parameters of the two-mass system, ensuring stable self-synchronization of anti-phase rotation of vibration exciter rotors in opposite directions.

The design parameters include the masses and dimensions of the walls, their moments of inertia, the stiffness of the springs and their positions along the walls. The operating parameters include the frequencies and amplitudes of the wall oscillations.

The solution to this problem should help determine the range of parameter values for the impact of the walls on the processed material, which serves as a basis for formulating recommendations for design and operation of systems for beneficiation of mineral granular materials [7].

Depending on the direction and phase difference of rotation of unbalanced rotors, the system can generate various modes of relative motion of the walls, each the most favorable for a specific type of processing procedure [8]. For example, one of the oscillation modes for attrition of material grains involving the walls moving with relative oscillation phase shifts in the transverse and longitudinal directions equal to π (Fig. 2). In this case, the walls move in one direction with co-rotation of the vibration exciter rotors (for example, clockwise). This allows to induce two types of deformations in the material layer: compression and shear deformations.

The requirements for the magnitude of compressive and shear stresses generated in the layer for attrition of the material depend on the oscillation frequency of the walls and the magnitude of the exciting force induced by vibration exciters.

Self-synchronization in vibration exciters under the described synchronous anti-phase counter-rotation of their rotors is analyzed in two stages:

research of self-synchronization stability in vibration exciters of a two-mass system under load taking into account the interaction with the processed material;

research of self-synchronization stability in vibration exciters in idle motion regime.

This study focuses only on the second stage, analyzing the stability of rotation in the system's vibration exciter rotors in idle motion regime.

Calculations

To analyze the self-synchronization stability in rotors of the system's vibration exciters, we used the integral criterion for stability of synchronous motion [1], consisting of finding the condition

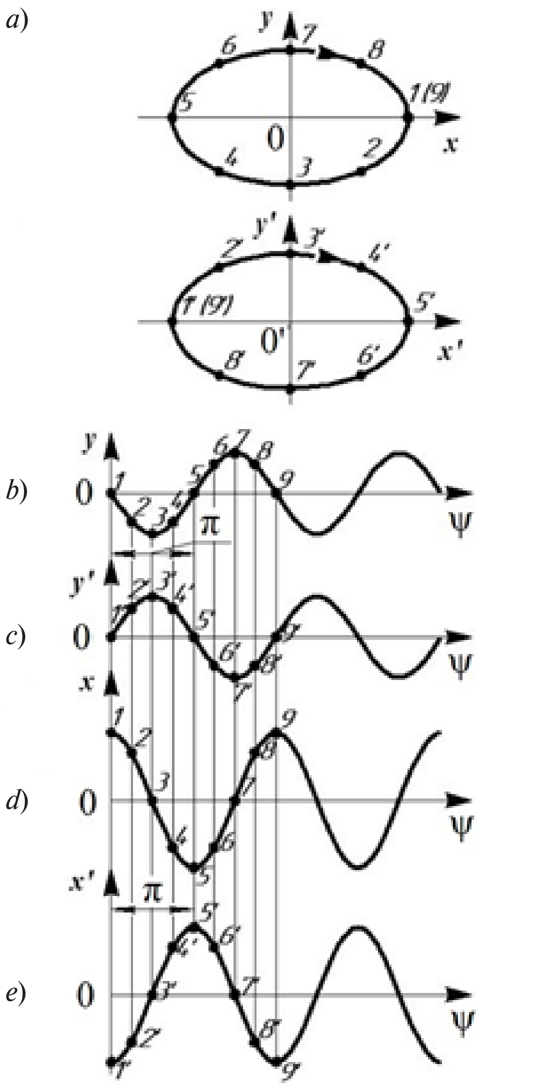

Fig. 2. Trajectories and phase diagrams for wall oscillations with phase ψ at phase shift $\varepsilon = \pi$ in longitudinal (ε_x) and transverse (ε_y) directions:
trajectories of upper and lower walls (*a*), arrows indicate their directions; vertical coordinates of lower (y') and upper (y) walls (*b*, *c*); horizontal coordinates of lower (x') and upper (x) walls (*d*, *e*); numbers of points *1–9* and *1′–9′* correspond to positions of the walls at the same points in time

for positive quadratic form of the function *D*, which is the average value of the Lagrange function of the system over the period:

$$D = \frac{\omega}{2\pi} \int_0^{2\pi/\omega} (T^{(I)} - \Pi^{(I)})dt, \qquad (1)$$

where $T^{(I)}$ and $\Pi^{(I)}$ are the kinetic and potential energies of the oscillatory subsystem.

We compose the equations for kinetic and potential energy of the two-mass mechanical system. The mathematical model for analyzing the self-synchronization stability of the rotors is shown in Fig. 3.

Two unbalanced vibration exciters in the mathematical model are placed on identical flat walls with mass *M*, each with a moment of inertia *I* relative to the CoM. The walls are placed parallel to each other and are connected by a pair of elastic springs with the total stiffness c_x and c_y in the longitudinal and transverse directions, placed at the same distances *l* relative to the wall CoM. The walls can move in a plane perpendicular to the rotation axes of the rotors. The position of the walls is determined by the absolute coordinates x_1, y_1 and x_2, y_2 of their CoM and the rotation angles ψ_1 and ψ_2 of each wall relative to the CoM. The rotors of each of the two vibration exciters, assumed to be identical, are characterized by angles φ_1 (φ_2) around the rotation axis, mass *m* of unbalanced part of the rotor, eccentricity ε and distance *r* from the rotation axis to the CoM of each wall. The subscript 1 refers to the lower wall, and 2 to the upper one.

The equations of translational motion of the walls have the form:

$$-M\ddot{x}_1 + F\cos\varphi_1 - c_x(x_1 - x_2) = 0; \qquad (2)$$

$$-M\ddot{x}_2 + F\cos\varphi_2 - c_x(x_2 - x_1) = 0; \qquad (3)$$

$$-M\ddot{y}_1 - F\sin\varphi_1 - c_y(y_1 - y_2) = 0; \qquad (4)$$

$$-M\ddot{y}_2 - F\sin\varphi_2 - c_y(y_2 - y_1) = 0, \qquad (5)$$

where $F = m\varepsilon\omega^2$ is the amplitude value of the exciting force of one vibration exciter.

The laws of rotation for the rotors can be written as follows:

$$\varphi_1 = \sigma_1(\omega t + \alpha_1),\ \varphi_2 = \sigma_2(\omega t + \alpha_2), \qquad (6)$$

where σ_1, σ_2 are the counterclockwise (+1) and clockwise (−1) directions of rotation of the rotors; α_1, α_2 are the initial phases of their rotation.

The strains of the left and right springs connecting the walls under rotation at angles ψ_1 and ψ_2 relative to the CoM (Fig. 4) are expressed as follows:

$$\Delta_{left} = -l\sin\psi_2 + l\sin\psi_1,\ \Delta_{right} = l\sin\psi_2 - l\sin\psi_1. \qquad (7)$$

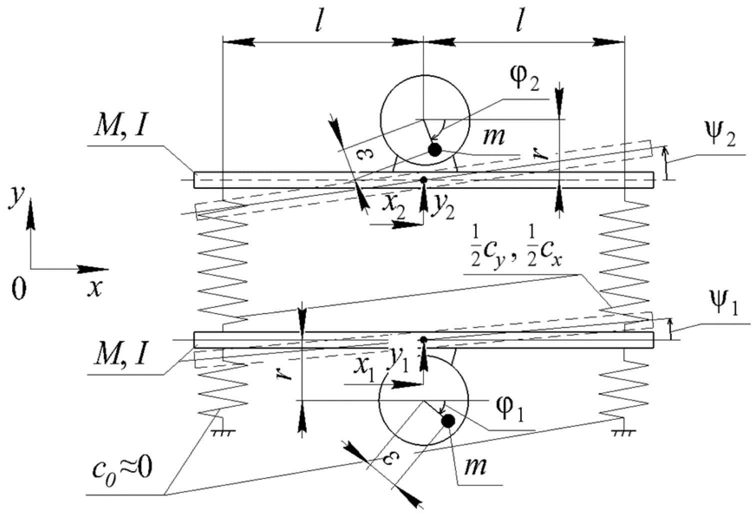

Fig. 3. Mathematical model for synchronous anti-phase oscillations of walls in two-mass system: mass of each wall M; moment of inertia I of each wall; stiffnesses c_x, c_y of the springs in the longitudinal and transverse directions; distances l from centers of mass (CoM) of the walls to the springs; coordinates x_i, y_i of wall CoM; rotation angles ψ_i of each wall relative to CG; rotation angles φ_i of the rotors around the rotation axis; eccentricity ε and distance r from the rotation axis of the rotors to wall CGs (subscripts 1 and 2 correspond to the lower and upper walls, respectively)

a) *b*)

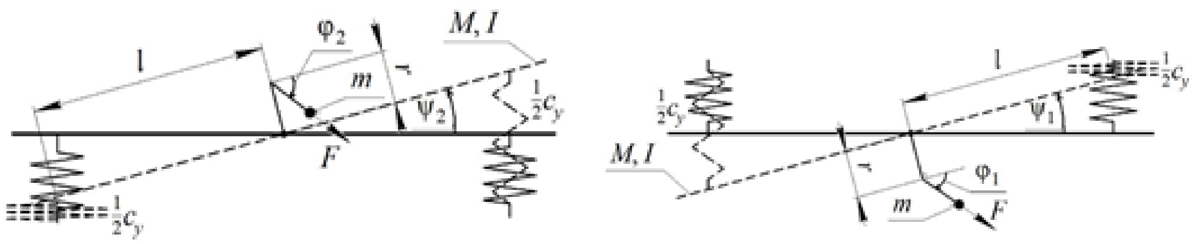

Fig. 4. Models used to determine the deformations of springs and moments arising upon rotation of upper (*a*) and lower (*b*) walls relative to the CoM

Eqs. (7) do not take into account the deformations of elastic elements in the transverse direction because they are small compared with longitudinal deformations.

Taking into account expressions (7), each wall of the system is subjected to moments from rotational inertia, exciting force of the vibration exciter, and deformations of the right and left springs (see Fig. 4). The equations of rotational motion of the walls relative to the CoM have the form:

$$-I\ddot{\psi}_1 + Fr\cos\varphi_1 - \frac{c_y}{2} l\Delta_{left} + \frac{c_y}{2} l\Delta_{right} = 0; \quad (8)$$

$$-I\ddot{\psi}_2 - Fr\cos\varphi_2 - \frac{c_y}{2} l\Delta_{right} + \frac{c_y}{2} l\Delta_{left} = 0. \quad (9)$$

Because the rotation angles of the walls ψ_1 and ψ_2 are small ($\sin\psi_1 \approx \psi_1$, $\sin\psi_2 \approx \psi_2$), they can be rewritten as follows:

$$-I\ddot{\psi}_1 + Fr\cos\varphi_1 + c_y l^2(\psi_2 - \psi_1) = 0; \quad (10)$$

$$-I\ddot{\psi}_2 - Fr\cos\varphi_2 + c_y l^2(\psi_1 - \psi_2) = 0. \quad (11)$$

We search for a solution for steady-state forced oscillations of the walls in the form

$$x_1 = A_1 \cos\varphi_1 + A_2 \cos\varphi_2; \tag{12}$$

$$x_2 = B_1 \cos\varphi_2 + B_2 \cos\varphi_1; \tag{13}$$

$$y_1 = C_1 \sin\varphi_1 + C_2 \sin\varphi_2; \tag{14}$$

$$y_2 = D_1 \sin\varphi_2 + D_2 \sin\varphi_1. \tag{15}$$

After substituting solutions (12) and (13) into the equations of motion of system (2) and (3), we determine the unknown constants A_1, A_2, B_1 and B_2:

$$A_1 = \frac{F(c_x - M\omega^2)}{(M\omega^2)^2(1-\lambda_x^2)},\ A_2 = \frac{Fc_x}{(M\omega^2)^2(1-\lambda_x^2)}; \tag{16}$$

$$B_1 = \frac{F(c_x - M\omega^2)}{(M\omega^2)^2(1-\lambda_x^2)},\ B_2 = \frac{Fc_x}{(M\omega^2)^2(1-\lambda_x^2)}, \tag{17}$$

where $\lambda_x = \frac{p_x}{\omega}$, $p_x = \sqrt{\frac{2c_x}{M}}$.

Evidently, the parameter p_x in the expressions obtained determines the natural frequency of the walls in the longitudinal direction for an equivalent mechanical model for the two-mass system with identical masses M, connected by a spring with the stiffness c_x (Fig. 5,*a*).

Substituting expressions (14) and (15) for transverse displacements of the walls into equations of motion (4) and (5), we similarly determine the unknown constants C_1, C_2, D_1 and D_2:

$$\tilde{N}_1 = \frac{F(M\omega^2 - c_y)}{(M\omega^2)^2(1-\lambda_y^2)},\ \tilde{N}_2 = \frac{-Fc_y}{(M\omega^2)^2(1-\lambda_y^2)}; \tag{18}$$

$$D_1 = \frac{F(M\omega^2 - c_y)}{(M\omega^2)^2(1-\lambda_y^2)},\ D_2 = \frac{-Fc_y}{(M\omega^2)^2(1-\lambda_y^2)}, \tag{19}$$

where $\lambda_y = \frac{p_y}{\omega}$, $p_y = \sqrt{\frac{2c_y}{M}}$.

Similar to the oscillations of the walls in the longitudinal direction, the parameter p_y in the expressions obtained characterizes the natural frequency of the wall oscillations in the transverse direction for the equivalent mechanical model of the system consisting of two masses M connected by a spring with the stiffness c_y (see Fig. 5,*b*).

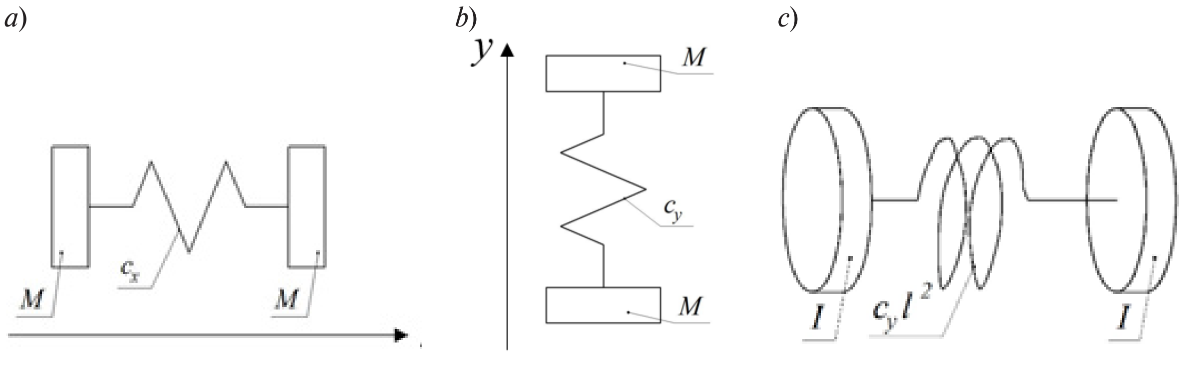

Fig. 5. Equivalent mechanical models for free oscillations of walls in longitudinal (*a*) and transverse (*b*) directions; equivalent mechanical model for their rotational oscillations (*c*)

We also search for solutions to the equations of rotational oscillations of the walls relative to the CoM as harmonic functions of the form

$$E_1 = \frac{Fr(c_y l^2 - I\omega^2)}{(I\omega^2)^2(1-\lambda_c^2)},\ E_2 = \frac{-Frc_y l^2}{(I\omega^2)^2(1-\lambda_c^2)}; \quad (22)$$

$$F_1 = \frac{Frc_y l^2}{(I\omega^2)^2(1-\lambda_c^2)},\ F_2 = \frac{-Fr(c_y l^2 - I\omega^2)}{(I\omega^2)^2(1-\lambda_c^2)}, \quad (23)$$

where $\lambda_{\tilde{a}} = \frac{p_c}{\omega}$, $p = \sqrt{\frac{2c_y l^2}{I}}$.

The obtained frequency p_c of free rotational oscillations of the walls corresponds to the equivalent mechanical model of the two-mass system with the same moments of inertia I connected to each other by a spring with the torsional stiffness $c_y l^2$ (see Fig. 5,*c*).

In accordance with the accepted mathematical model (see Fig. 3), kinetic and potential energies of oscillatory motion of the walls have the form:

$$T^{(I)} = M\frac{(\dot{x}_1{}^2 + \dot{y}_1{}^2)}{2} + M\frac{(\dot{x}_2{}^2 + \dot{y}_2{}^2)}{2} + I\frac{(\dot{\psi}_1{}^2 + \dot{\psi}_2{}^2)}{2}, \quad (24)$$

$$\Pi^{(I)} = \frac{1}{2}c_y\Delta_{left}{}^2 + \frac{1}{2}c_y\Delta_{right}{}^2 \approx c_y l^2(\psi_1{}^2 + \psi_2{}^2 - 2\psi_1\psi_2). \quad (25)$$

The condition for stable self-synchronization of the rotors is that the averaged Lagrange function of the system taking the form [1] be positive definite:

$$D = \langle T^{(I)} - \Pi^{(I)}\rangle = \frac{\omega}{2\pi}\int_0^{\frac{2\pi}{\omega}} (T^{(I)} - \Pi^{(I)})dt. \quad (26)$$

Substituting solutions (12)–(15), (20), (21) for wall displacements into expressions for kinetic and potential energies (24) and (25) and calculating integral (26) for one period of the system's oscillations, we obtain the following form for the averaged Lagrange function:

$$D = \frac{M}{2}\omega^2(\sigma_1\sigma_2(C_1C_2 + D_1D_2)) + \frac{I}{2}\omega^2(F_1F_2 + E_1E_2) - $$
$$-c_y l^2(E_1E_2 + F_1F_2 - 2(E_1F_2 + E_2F_1))\cos(\alpha_1 - \alpha_2) = N\cos(\alpha_1 - \alpha_2), \quad (27)$$

where $N = \frac{M}{2}\omega^2(\sigma_1\sigma_2(C_1C_2 + D_1D_2)) + \frac{I}{2}\omega^2(F_1F_2 + E_1E_2) - c_y l^2(E_1E_2 + F_1F_2 - 2(E_1F_2 + E_2F_1))$.

For self-synchronization of vibration exciter rotors to be stable under oppositely directed synchronous rotation of the rotors, i.e., at zero initial phase difference $\alpha_1 - \alpha_2$ and at $\sigma_1 = +1$, $\sigma_2 = -1$, the quadratic form of the averaged Lagrange function D must be positive definite. This is ensured when it has a rough minimum and the multiplier N in expression (27) is negative, i.e.,

$$-\frac{M}{2}\omega^2(C_1C_2 + D_1D_2) + \frac{I}{2}\omega^2(F_1F_2 + E_1E_2) - c_y l^2(E_1E_2 + F_1F_2 - 2(E_1F_2 + E_2F_1)) < 0. \quad (28)$$

After substituting the constants C_1, C_2, D_1, D_2, E_1 and E_2 from the expressions (18), (19), (22) and (23) into this inequality, we obtain a condition for stable self-synchronization of vibration exciters in the given mode, allowing to optimize the design and operating parameters of the two-mass system:

$$\frac{l^2r^2(c_y l^2 - 3M\omega^2)}{(I\omega^2)^2(I\omega^2 - 2c_y l^2)} > \frac{M\omega^2 - c_y}{M\omega^2(M\omega^2 - 2c_y)^2}. \quad (29)$$

Let us analyze the expression obtained for oscillations of the walls below resonance, defined by the relations $c_y > 3M\omega^2$ and $2c_y l^2 > I\omega^2$.

In this case, the following inequality must hold true for self-synchronization of vibration exciters:

$$\frac{c_y - M\omega^2}{M\omega^2(M\omega^2 - 2c_y)^2} > \frac{l^2 r^2 (c_y l^2 - 3M\omega^2)}{(I\omega^2)^2 (2c_y l^2 - I\omega^2)}. \tag{30}$$

It can be seen from here that the stability of the given oscillation mode can be increased by increasing the frequency ω of the exciting force and the moment of inertia *J* of the wall, decreasing the mass *M* of the wall, the distance *l* from the wall CoM to the points where the springs are attached and the distance *r* from the wall's center of mass to the axis of rotor.

The stability condition for synchronous anti-phase oscillations of the walls is satisfied automatically for oscillations below resonance for which the relations $c_y > M\omega^2/2$ и $c_y l^2 > I\omega^2$ hold true, since the left-hand side of inequality (29) is always negative and the right side is positive.

Numerical values of the design parameters based on the solution of inequality (29) can be selected by any suitable numerical method, for example, the iteration method.

Conclusion

We found a condition for stable self-synchronization of vibration exciters in a system with two flat walls generating synchronous anti-phase oscillations along elliptical trajectories in opposite directions. The obtained expression allows to optimize the design and operating parameters of the system (frequency of the exciting force, mass of the walls and their moments of inertia, distances from the CoM of the walls to the points where the springs are attached and the distance from the CoM of each wall to the axis of the vibration exciter rotor) at the design stage to ensure the required oscillation mode of the walls, combining crushing of the material grains with transportation.

Further research will be aimed at finding the conditions for stable self-synchronization of rotors in vibration exciters of two-mass systems in operating modes with relative phase shifts and rotation directions providing optimal parameters for attrition and compression procedures in granular materials.

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