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Dynamic synchronization of vibration exciters of the three-mass vibration mill

***Abstract**. The problem of self-synchronization of a three-axis vibrating mill with four unbalance vibrators, which are actuated by independent asynchronous motors and have the same angular velocities, is considered in the paper. In self-synchronization, synchronization and phase matching of the vibrator is achieved due to the vibration of the system of bodies on which the vibrator is installed, that is, by providing a dynamic connection between them.*

Streszczenie.** W artykule analizowano problem synchronizacji trzyosiowego wibracyjnego młyna z czterema niezależnymi wibratorami. Samoczynna synchronizacja została osiągnięta dzięki dynamicznemu połączeniu między wibratorami i podłożem.* ***Dynamiczna synchronizacja trzyosiowego młyna z czterema wibratorami.

Keywords: vibration exciter, imbalance, synchronization, milling chamber.
Słowa kluczowe: wibracja, młyn wibrtacyjny, synchronizacja.

Introduction

The fundamental disadvantage of single-vibration vibratory mills is that the exciter`s force is limited by the permissible loads of their bearing assemblies, as well as difficulties in assembling and disassembling them. Therefore, the creation of powerful vibration exciters is possible, first of all, by aggregation of universal simple (two-support) vibration exciters.

The aggregation of vibration exciters can significantly reduce the time and reduce the cost of design and manufacture of vibration mills, while at the same time simplify their maintenance and repair. In other cases, the use of several low-power exciters instead of one equal in power is due to the need to bridge the forced force over the vibrating working body of a vibrating mill of considerable size. Aggregation of vibration exciters can be carried out sequential, parallel, as well as mixed.

The synchronization of the rotation of the vibrators is necessary to create a continuous movement of the working body (camera) of the vibrating mill in a circular path and is provided mainly by three methods of synchronization: using special shafts, gears, chain or belt transmissions. , forced electrical synchronization (using an electric shaft system or synchronous motors) or self-synchronization. Obviously, the best way to synchronize is to self-synchronize, since it does not require additional devices that complicate the design.

In recent years, dynamic synchronization (self-synchronization) has been widely used in vibration technology. The phenomenon of dynamic synchronization consists in the fact that several artificially created or natural objects, which, in the absence of interaction, make oscillatory or rotating movements with different frequencies when long weak links are superimposed, begin to move with the same or multiple frequencies, which are in rational relations, moreover, certain phase relations between oscillations and rotations are established.

Analysis of literary sources and problem statement

One of the main parameters of vibration mills is the maximum power that is transmitted by the exciters to the load through the grinding chamber at a given amount of force, which is developed by the exciters themselves [1, 2]. It was found that the maximum power is transmitted to the load when the camera moves along a circular path, which determines the location of the vibration exciters [3]..

The creation of powerful vibration exciters is possible by aggregating simple universal (two-support) vibration exciters. This allows you to reduce the time significantly and reduce the cost of designing and manufacturing vibratory mills [4, 5, 6], while simplifying their repair and maintenance. In other cases, the use of several low-power exciters instead of one equal in power is due to the need to disperse the force over the vibrating working body with a large-sized vibrating mill.

The most optimal of the methods of synchronization is self-synchronization, in which the synchronism and phase matching of the vibration exciters is achieved by establishing a dynamic connection between them.

Analyzing the methods of aggregating vibration exciters to create the translational movement of the working body of the vibratory mill along a circular path and the necessary synchronism and phase matching, it can be mentioned that dynamic synchronization (self-synchronization) is currently widely used [7, 8, 9, 10, 11, 12].

The solution of specific applied problems of self-synchronization of vibration exciters can be most effectively performed using the integral criterion of stable synchronous movements [6].

Achieving stability of the required synchronous rotation mode of vibration exciters is achieved by combining, firstly, by attaching some additional mass to the initial supporting solid using elastic elements, and secondly, by changing the number of vibration exciters [12, 13, 14].

In the works [15, 16, 17] the technological efficiency of such vibro-installations is shown.

In the Laboratory of Theory of Mechanisms and Machines of Vinnitsa National Agrarian University, a schematic diagram of a vibrating mill with a spatial-circular trajectory of technological environment motion was developed [5, 18, 19].

The working body of the vibrating mill (Fig. 1) contains closed grinding chambers 4, which are filled with grinding bodies (balls) by means of a reloading 1 and a transition trench 6. Under the action of a vibrating action, the technological environment (grinding bodies and material) 5 in the left grinding chamber 4 rises vertically through the trough 1 and flows over the transport tray 2 into the right grinding chamber. The crushed material is sieved through the unloading grate 3 and unloaded from the mill. In this case, the grinding bodies are diverted to the front of the right grinding chamber in the loading area of the material. In the front of the right grinding chamber due to the continuous passage on the chute 1 and the transport tray 2 is formed an increased volume of technological environment. Crushed material also comes here. In the back of the grinding chamber, due to the transition of part of the process environment into the left grinding chamber, a reduced volume of process environment is formed. In the left grinding chamber due to the selection of technological environment in the front part produces a reduced volume, and in the rear, due to the influx of technological environment from the right grinding chamber produces an increased volume. Thus, in the interconnected grinding chambers provides spatial-circulation movement of the technological environment in a closed screw trajectory.

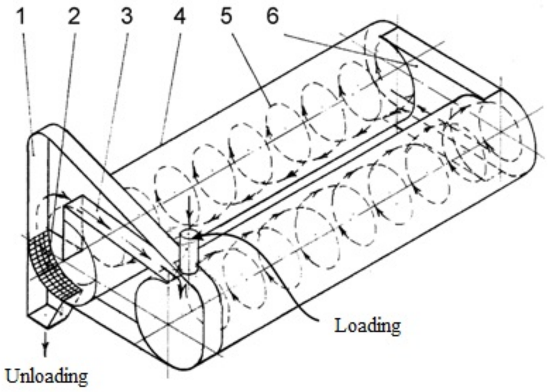

Fig. 1. Scheme of the executive body of the vibration mill with the spatial-circular trajectory of the technological environment: 1 – reloading chute; 2 – transport tray; 3 – unloading grid; 4 – grinding chamber; 5 – the trajectory of movement of the technological environment (grinding bodies and material); 6 – transitional gutter.

Due to the spatial-circulation movement of the technological environment, which is provided by the vibration action, this mill design provides continuous grinding, excluding pasive zones. This allows to significantly reduce the specific energy consumption for the technological grinding operation. However, to ensure efficient operation of the proposed design of the vibrating mill, it is necessary to set the criteria for stable motion of the technical system, which will ensure the dynamic synchronization of independent vibrators of the machine.

Purpose and tasks of research

The purpose of the research is to investigate the scheme of a vibratory mill with a U-like working chamber aggregated in parallel with simple two-support vibration exciters and establish criteria for stable movement.

The main task about the self-synchronization of mechanical vibration exciters is to determine the conditions under which all vibration exciter rotors rotate with the same average angular velocities beyond the absolute value, $\left|\varphi_s^*\right| = \omega\ (s = 1...k)$ despite the possible difference in their parameters and the forces acting on them, and the supporting bodies undergo oscillations with a period $T = 2\pi / \omega$.

Materials and methods

Solutions to specific applications of self-synchronization of vibration exciters can be most effectively performed by using the integral criterion of persistent synchronous movements. According to this criterion, the phases of rotation of vibration exciters in stable synchronous movements correspond to the minimum points of some function *D* of these phases, which is called potential. For problems important for practice (including the one under consideration), it turns out that the potential function is equal to the average over the period value of the Langrange function L of the vibrational part of the system:

(1) $$D = \frac{\omega}{2\pi}\int_0^{2\pi/\omega} L dt = \frac{\omega}{2\pi}\int_0^{2\pi/\omega}(T - P)dt;$$

where *T*, *P* are the kinetic and potential energies of the vibrational part of the system, respectively.

The practical technique for researching devices with self-synchronizing vibration exciters is characterized by the following sequence of actions:

- the equations of small vibrations of the bearing bodies are added with the assumption that the shafts of the vibration exciters rotate uniformly, according to the law:

(2) $$\varphi_s = \omega t + \alpha_s,$$

where α_s are unknown initial phases of rotation, *ω* is the angular velocity of synchronous rotation;

- by these equations the law of motion of the bearing bodies is determined, which corresponds to constant forced oscillations (this is a potential function);

- the value of phase displacements α_s, which can correspond to stable synchronous movements and conditions of stability, are found from the conditions of minimum potential function.

The grinding chamber (Fig. 2) of mass M_1 is considered to be an absolutely rigid body mounted on a fixed base with the help of sufficiently soft elastic elements ($c_0 \approx 0$). On the camera, two identical additional solid bodies are installed symmetrically with respect to the vertical axis using elastic elements with rigidity *c*, the mass of each is equal to M_2. The camera is driven into vibration using four unbalanced vibration exciters, each of which is equipped with an individual drive. The generalized coordinates of the oscillatory system are: *x*, *y* and *φ* – coordinates that correspond to the horizontal, vertical and angular movements of the container, and x_1, y_1 and x_2, y_2 – coordinates that correspond to the horizontal and vertical movements of the first and second additional bodies.

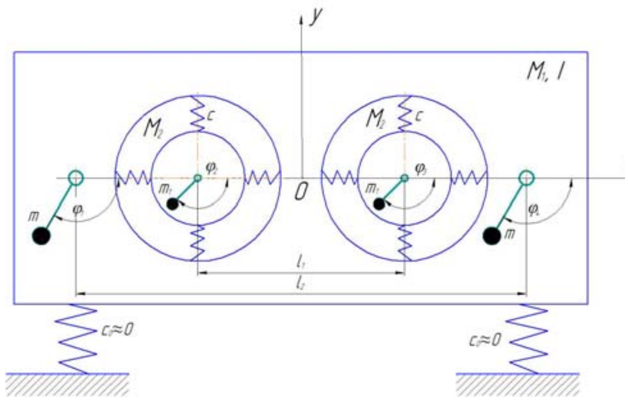

Fig. 2. Three-mass vibratory mill with four vibration exciters (кinematic scheme)

Thus, in this article, a theoretical study of the steady motion of a three-mass four-vibration mill is carried out and an analysis is carried out with specific technological and dynamic parameters. These parameters were obtained and substantiated by the results of experimental studies [20] of the pilot industrial model of the MV-400 vibrating mill (Fig. 3). This mill was developed in the laboratory of the theory of mechanisms and machines of Vinnitsa National Agrarian University.

The analysis of the conditions for the minimum of the function *D* by the analytical method is rather complicated due to the large bulkiness of the coefficients. Therefore, the studies were carried out numerically using a PC and the Maple V software package.

Fig. 3. Vibrating mill (MV-400)

Research results

The equations for the kinetic and potential energy of the vibrational part of the system have the form:

$$T=\frac{1}{2}\begin{bmatrix} M_1\left(\dot{y}^2+\dot{x}^2\right)+I\dot{\varphi}^2 \\ +M_2\left(\dot{x}_1^2+\dot{x}_2^2+\dot{y}_1^2+\dot{y}_2^2\right) \end{bmatrix}; \tag{3}$$

$$P=\frac{1}{2}c\begin{bmatrix} \left(x-x_1\right)^2+\left(x-x_2\right)^2+\left(y-l_1\varphi-y_1\right)^2 \\ +\left(y+l_1\varphi-y_2\right)^2 \end{bmatrix}. \tag{4}$$

The differential equations of motion of the vibration unit with uniform rotation of the exciters will have the form:

$$\begin{cases} M_1\ddot{y}+c(y-y_1)+c(y-y_2)= \\ -F\left[sin(\omega t+\alpha_1)+sin(\omega t+\alpha_4)\right]; \\ M_1\ddot{x}+c(x-x_1)+c(x-x_2)= \\ F\left[cos(\omega t+\alpha_1)+cos(\omega t+\alpha_4)\right]; \\ I\ddot{\varphi}+cl_1(l_1\varphi+y_1)+cl_1(l_1\varphi-y_2)= \\ Fl_2\left[sin(\omega t+\alpha_1)-sin(\omega t+\alpha_4)\right]; \end{cases}; \tag{5}$$

$$\begin{cases} M_2\ddot{x}_1-c(x-x_1)=F_1\,cos(\omega t+\alpha_2); \\ M_2\ddot{x}_2-c(x-x_2)=F_1\,cos(\omega t+\alpha_3); \\ M_2\ddot{y}_1-c(y-y_1)+cl_1\varphi=-F_1\,sin(\omega t+\alpha_2); \\ M_2\ddot{y}_2-c(y-y_2)-cl_1\varphi=-F_1\,sin(\omega t+\alpha_3); \end{cases}. \tag{6}$$

where $F=mr\omega^2$, $F_1=m_1r_1\omega^2$ – the driving force that develops by each vibration exciter (mr ; m_1r_1 — masses and eccentricities of vibration exciters mounted respectively on the container and additional bodies).

The determinants for calculating the frequencies of free vibrations that correspond to two independent groups of homogeneous equations that correspond to system (5) and (6) can be represented in the following form [15]:

$$\begin{vmatrix} 2c-M_1p^2 & -c & -c \\ -c & c-M_2p^2 & 0 \\ -c & 0 & c-M_2p^2 \end{vmatrix}=0; \tag{7}$$

$$\begin{vmatrix} 2c-M_1p^2 & -c & -c & 0 \\ -c & c-M_2p^2 & 0 & cl_1 \\ -c & 0 & c-M_2p^2 & -cl_1 \\ 0 & cl_1 & -cl_1 & 2cl_1^2-Ip^2 \end{vmatrix}=0. \tag{8}$$

According to (8) write the frequency equation:

$$\left(c-M_2p^2\right)\left(M_1M_2p^2-2cM_2-cM_1\right)p^2=0. \tag{9}$$

As a result, we find:

$$p_1\approx 0,\, p_2^2=\frac{c}{M_2},\, p_3^2=\frac{c}{M^*}, \tag{10}$$

where $M^*=\dfrac{M_1M_2}{2M_2+M_1}$.

Taking into consideration the higher order of the determinant (8), as a result we have, the very large computational complexity that correspond to the natural frequency determined on a PC using the standard Maple V computer software package:

$$p_4\approx 0,\, p_5^2\approx 0,\; p_6^2=\frac{c}{M^*},\, p_7^2=\frac{c\left(I+2M_2l_1^2\right)}{M_2I}. \tag{11}$$

The solution of differential equations (5) and (6) which corresponds to a constant forced oscillation can be written in the form:

$$\begin{cases} x=A_1\left[cos(\omega t+\alpha_1)+cos(\omega t+\alpha_4)\right] \\ +A_3\left[cos(\omega t+\alpha_2)+cos(\omega t+\alpha_3)\right]; \\ x_1=A_2\left[cos(\omega t+\alpha_1)+cos(\omega t+\alpha_4)\right] \\ +A_4\,cos(\omega t+\alpha_2)+A_5\,cos(\omega t+\alpha_3); \\ x_2=A_2\left[cos(\omega t+\alpha_1)+cos(\omega t+\alpha_4)\right] \\ +A_5\,cos(\omega t+\alpha_2)+A_4\,cos(\omega t+\alpha_3); \\ \varphi=-A_{10}\left[sin(\omega t+\alpha_1)-sin(\omega t+\alpha_4)\right] \\ -A_5\left[sin(\omega t+\alpha_2)-sin(\omega t+\alpha_3)\right]; \\ y=-A_1\left[sin(\omega t+\alpha_1)-sin(\omega t+\alpha_4)\right] \\ -A_3\left[sin(\omega t+\alpha_2)+sin(\omega t+\alpha_3)\right]; \\ y_1=A_6\,sin(\omega t+\alpha_1)+A_7\,sin(\omega t+\alpha_4) \\ +A_8\,sin(\omega t+\alpha_2)+A_9\,sin(\omega t+\alpha_3). \end{cases}. \tag{12}$$

Here is denoted: $A_1=\dfrac{bF}{ba+2c^2}$; $A_2=\dfrac{cF}{ba+2c^2}$; $A_3=\dfrac{cF_1}{ba+2c^2}$; $A_4=\dfrac{F_1\left(ba-c^2\right)}{b\left(ba-2c^2\right)}$; $A_5=\dfrac{F_1c^2}{b\left(ba-2c^2\right)}$;

$$A_6=\frac{cF\left[\left(db-2c^2l_1^2\right)-l_1l_2\left(2c^2-ab\right)\right]}{\left(db-2c^2l^2\right)\left(2c^2-ab\right)};$$

$$A_7=\frac{cF\left[\left(db-2c^2l_1^2\right)-l_1l_2\left(2c^2-ab\right)\right]}{\left(db-2c^2l_1^2\right)\left(2c^2-ab\right)};$$

$$A_8=\frac{F\left[a\left(db-2c^2l_1^2\right)+c^2\left(2c^2-ab\right)\right]}{\left(db-2c^2l_1^2\right)\left(2c^2-ab\right)};$$

$A_9=-\dfrac{c^2F_1\left(al_1^2-d\right)}{\left(db-2c^2l_1^2\right)\left(2c^2-ab\right)}$; $A_{10}=-\dfrac{Fl_2b}{2c^2l_1^2-db}$;

$A_{11}=-\dfrac{F_1cl_1}{2c^2l_1^2-db}$; $a=2c-M_1\omega^2$; $b=c-M_2\omega^2$; $d=2cl_1^2-I\omega^2$.

To solve the problem, we use the integral stability criterion, and as a potential function, we can take the average value of the Lagrange function of the vibrational system over a period. The period average value of the Lagrange function in the calculation of equations (1) for kinetic and potential energy and solutions (12):

$$D=\frac{\omega}{2\pi}\int_0^{\frac{2\pi}{\omega}}(T-P)dt=D_1\,cos(\alpha_1-a_4)+D_2\,cos(a_1-a_2)$$
$$+D_3\,cos(a_1-a_3)+D_4\,cos(a_2-a_3)$$
$$+D_3\,cos(a_2-a_4)+D_2\,cos(a_3-a_4)+C_1,$$

where C_1 – value independent of angles α_s; $D_1 = m\varepsilon\omega^2\left(A_3 + \frac{A_{10}l_2}{2}\right)$; $D_2 = m\varepsilon\omega^2\left(A_3 - \frac{A_{11}l_2}{2}\right)$; $D_3 = m\varepsilon\omega^2\left(A_3 + \frac{A_{11}l_2}{2}\right)$; $D_4 = \frac{m_1\varepsilon_1\omega^2}{2}\left(A_5 - A_9\right)$.

Equating the derivatives to zero $\partial D / \partial \alpha_s$, we obtain an equation for determining the values of the constants α_s in possible synchronous movements.

Moreover, since the problem of self-synchronization is considered, we equate one of the phases (for example α_4) to zero.

(13) $$\begin{cases} D_1 \sin\alpha_1 + D_2 \sin(\alpha_1 - a_2) + D_5 \sin(a_1 - a_3) = 0; \\ D_2 \sin(\alpha_1 - a_2) - D_4 \sin(\alpha_2 - a_3) - D_3 \sin a_2 = 0; \\ D_4 \sin(\alpha_2 - a_3) - D_2 \sin\alpha_3 + D_3 \sin(\alpha_1 - a_3) = 0. \end{cases}$$

Equation (13) can satisfy the following five phase combinations α_s:

(14) $\alpha_1^* = \alpha_2^* = \alpha_3^* = 0$;

(15) $\alpha_1^* = 0; \alpha_2^* = \alpha_3^* = \pi$;

(16) $\alpha_1^* = \alpha_2^* = \pi; \alpha_3^* = 0$;

(17) $\alpha_1^* = \pi;\ \alpha_2^* = \alpha_3^* = 0$;

(18) $\alpha_1^* = \alpha_2^* = 0; \alpha_3^* = \pi.$

The condition for the stability of synchronous movements, in accordance with the integral criterion, is the condition for the minimum potential function.

Consider the possibility of stabilizing synchronous-in-phase rotation of two main vibration exciters using two additional vibration exciters on elastically suspended bodies [15, 17]. Taking into account the results of solving the problem [10], it is envisaged to obtain a positive effect upon rotation of additional vibration exciters in the antiphase relative to the main one (second group of solutions (7)). In this case, the last inequality will look like this: solution [15, 14], it is envisaged to receive a positive effect, when additional vibration exciters rotate in anti-phase on the main (15). In this case, the last inequality will have the following form:

(19) $$\begin{vmatrix} -D_1 + D_2 + D_3 & -D_1 & -D_3 \\ -D_2 & D_2 + D_3 - D_4 & D_4 \\ -D_3 & D_4 & D_2 + D_3 - D_4 \end{vmatrix} > 0 .$$

Then we come to the following condition for the stability of the considered motion:

(20) $$\begin{cases} -D_1 + D_2 + D_3 > 0; \\ D_5 > 0; \\ D_2 + D_3 > 0; \\ or \\ \frac{1}{\lambda_2^2 - 1} > 0; \\ D_6 > 0; \end{cases},$$

where:

(21) $D_5 = -(D_2 + D_3)(D_1 + D_4) + 2D_2D_3 + D_1D_4 + D_3^2$;

(22) $D_6 = -(D_2 + D_3)(D_1 + D_4) + 2D_2D_3 + 2D_1D_4$.

A graphical solution of these conditions for the second group of phase combinations is shown in Fig. 4.

Graphing was performed using the standard Maple V software package and using known input parameters that were experimentally justified earlier [20]: $M_1 = 120$ kg; $M_2 = 20$ kg; $l_2 = 0.325$ m; $l_1 = 0.16$ m; $m\varepsilon / m_1\varepsilon_1 = 0.2$; $I = 7.6$ kg·m^2; $Ml_2^2 / I = 1.44$; $\lambda_2 = \frac{P_2}{\omega} M = M_1 + M_2$.

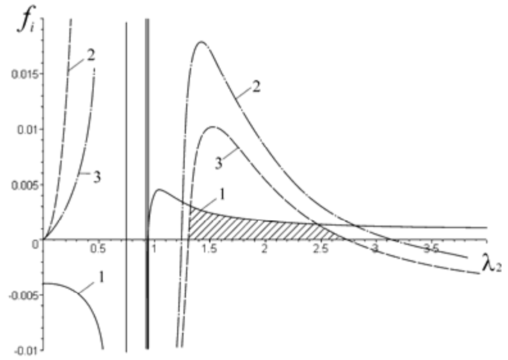

Fig. 4. Function graph for determining the area of synchronous motion of vibration exciters $(Ml_2^2 / I = 1.44) - f_i(\lambda_2)$

Thus, the successful use of self-synchronization of more than two vibration exciters is possible when additional exciters are installed on elastically suspended bodies; such exciters can stabilize the synchronous-in-phase rotation of two main vibration exciters. Note that the obtained research results are consistent with [12] (where only the final result is presented in the form of charts of solving the Fortran problem for a system with specific parameters) and in a certain content they are supplemented and refined.

Conclusions

The considered dynamic circuit can provide self-synchronization of vibration exciters, while all the symmetrical vibrations of the vibration mill are possible, which are of practical interest.

The resistance limit of synchronous rotations of vibration exciters is established, which allows you to choose the optimal installation parameters and its operation modes.

The presented dynamic scheme makes it possible to change the phases of rotation of pathogens by changing their operating frequency (rotation).

The stability of one or another mode of exciters rotation depends primarily on the elastic suspension rigidity of additional bodies and the frequency of exciters rotation.

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