Розроблено нову конструкцію вібраційного сепаратора барабанного типу з концентричним розміщенням сит та побудовано математичну модель опису його динаміки, яка є нелінійною, уніфікованою та параметричною. За допомогою неї визначено вплив основних конструктивних параметрів сепаратора і кінематичних показників його приводу на характеристики його руху та інтенсивність процесу сепарації. Експериментальні дослідження підтвердили достовірність отриманої моделі

Ключові слова: барабанний сепаратор, дебаланс, нелінійна модель, вібраційний привід, концентричні сита, пружинна підвіска

Разработана новая конструкция вибрационного сепаратора барабанного типа с концентрическим размещением сит и построена математическая модель описания его динамики, которая является нелинейной, унифицированной и параметрической. С ее помощью определено влияние основных конструкционных параметров сепаратора и кинематических показателей его привода на характеристики его движения и интенсивность процесса сепарации. Экспериментальные исследования подтвердили достоверность полученной модели

Ключевые слова: барабанный сепаратор, дебаланс, нелинейная модель, вибрационный привод, концентрические сита, пружинная подвеска

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MODELING THE DYNAMICS OF VIBRATORY SEPARATOR OF THE DRUM TYPE WITH CONCENTRIC ARRANGEMENT OF SIEVES

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1. Introduction

Under conditions of development of the Ukrainian market economy and present-day realities, modern industrial production integrates competitive technologies for product manufacturing. They are characterized by high productivity, relatively low energy and material costs for ensuring their operation and minimum negative impact on the environment.

Accordingly, the equipment that implements these technological processes should be reliable in operation and whose design should be sufficiently simple for repair. At maximally small dimensions, it should have the universality of application. While requiring minimal power to provide for its operational functionality, it should be distinguished by relatively low cost. The given equipment should be readjusted easily and rapidly for manufacturing another type or size of products in various industries. One of the indisputable conditions for meeting these requirements is the development of optimal designs and their estimation, determining operational modes for this equipment at the stage of its design.

We relate the technology of separation to one of such required and effective technologies. This is the process of selecting, sorting of separate components out of mixtures of various types and state by certain physical-mechanical attributes. Separation is widely used in different areas of production – mining, agriculture, food processing production, medicine. It is included as an integral link to the general technological process of manufacturing many goods, and the very quality of separation, its performance speed and efficiency significantly affect quality of the finished product.

Therefore, an important task today is to examine and develop the phenomena of separation of mixtures of different substances, and, in this context, is to devise new designs of separators, which would be distinguished by high efficiency of operation and universality. In order to resolve these tasks, there is an actual issue to study the dynamics of different types of separators by using modeling. After all, nonlinear, adequate, parametric, universal model that describes the motion of separator will simplify and speed up the process of its estimation to develop its optimal design. It will make it possible to choose its operating modes to provide for its maximum efficiency.

Present study is actually devoted to solving the problems indicated above. It proposes a new design of highly efficient separator and investigates its dynamics. The research was conducted by creating non-linear, unified and parametric model of the designed separator. The resulting non-linear mathematical model can be used for a number of vibratory processing systems at the stages of their design and operation. This will speed up the design process of the specified systems and improve their performance efficiency.

2. Literature review and problem statement

A range of designs of separators is extremely varied. They are used practically in all sectors of industry, agriculture, medicine and everyday life. That is why much attention is paid to exploring the principle of their operation, improving productivity, as well as to creating new prototypes. An effective approach to designing a new separator is to create its mathematical model, examine it and then select, by employing it, optimal parameters for future development. However, such an approach to developing new designs of separator has not gained a wide practical application. This can be explained by the fact that such studies require the use of complex mathematical apparatus, advanced mathematical methods and computing equipment. To adequately reflect all dynamic phenomena that occur in a separating system, mathematical models have to be non-linear. The linearity of models leads to the fact that they cannot reflect full extent of impact from the parameters of a would-be design on its performance efficiency. They also cannot take into account the effect of all parameters of the examined system.

The physical and mathematical models, explored up to now [1, 2], describe the possibility of constructing vibratory separators with electromagnetic actuator. In these studies, there is also a devised a mathematical model of influence of the medium layer under separation on the motion of vibratory system. However, these estimations are suitable for separators that provide directed vibrations. Vibratory separators with unbalance vibration exciter perform complex spatial motion, which is why these models cannot adequately reflect dynamics of the separator. Previous studies [3] describe dynamics of the vibratory separator only based on the simplified linear models. A theoretical description of separators was compiled based on certain assumptions [4, 5], which accounted for particular design of the separator. This is, in particular, the impact of interaction between particles of the medium and the working surface of the sieve on separation efficiency, a possibility to increase and decrease the mound of medium layer and the sieve throughput. In those cases, they considered motion of the medium layer in the sieve. In [6], a mathematical model was developed for determining parameters of vibratory screens considering the impact of inertia moments in the spring suspension, debalances and the separator housing. Much to our regret, the mathematical models proposed cannot reflect all the phenomena that occur in the separator and how they affect the separation performance. They were developed for particular designs of separators, which narrows the scope of their application.

Wide spread are the separators with a rotational motion of the sieve. Such separation is implemented in separators of the drum type [7]. In these separators, a layer of material is separated by centrifugal forces and gravity. Concentric arrangement of several sieves in a drum separator is proposed to be employed in future studies because of significant expansion of the separation area.

Stirring and transporting the mixtures largely influence determining the performance of separator. Accordingly, in order to improve these processes, it is proposed to use a vibratory drive for propelling the drum separator. It will make it possible to reach different amplitude and shape in the trajectory of motion of screening sieves. Thus a new type of separator was synthesized from two types, that is, a drum vibratory separator, which will possess advantages of the two original ones.

A vibratory separator of the drum type will represent a complex dynamic system whose adequate description can be rationally compiled only by means of nonlinear modeling. Similar type of research was conducted by the authors to examine the dynamics of machines that implement vibratory volumetric processing of goods and vibratory separators with flat sieves [8–11].

In order to ensure maximum productivity of the proposed vibratory separator of the drum type with concentric arrangement of sieves, theoretical studies of its dynamics should be conducted. They are required to clarify the impact of its parameters (in particular, geometric parameters of the design and kinematic parameters of the drive) on the intensity of the separation process of bulk mixtures. And the actual development of nonlinear model to describe the motion of separator will make it possible to implement this task. Uniformity of the developed model will allow us to use it not only for the proposed new design of separator, but also for designing a series of separators or vibratory processing machines of other types.

3. The aim and tasks of the study

The aim of the conducted studies was to construct a nonlinear model of vibratory separator of the drum type with concentric arrangement of sieves and unbalanced (eccentric) drive. This model will make it possible to explore dynamic phenomena in the separator for determining the impact of separator parameters on the factors of separating bulk mixtures in it.

To achieve the set aim, the following tasks were to be solved:

– to examine influence of key (geometric and kinematic) parameters of the created vibratory separator on the character of motion of its sieves by mathematical modeling of the separator motion – such a research should be conducted for the stationary and resonant operation modes of separator;

– to study experimentally influence of the separator parameters on the character of its motion and bulk mixture, which needs to be separated, to verify adequacy of the obtained theoretical results;

– to explore influence of the designed separator parameters on the factors of separation intensity in it.

4. Methodology and methods for constructing a mathematical model to describe dynamics of the designed separator and its experimental verification

Studies into dynamics of vibratory separator are based on the laws that govern the motion of arbitrary points of its concentric sieves and the working body that houses them, over the specified interval of separation. These laws are a totality of sets of analytical expressions, which in one form or another includes all the required parameters of separator. These expressions are the analytical solution for the compiled system of nonlinear differential equations that theoretically describes dynamics of the examined separator. They will comprise a mathematical model of its motion. Coordinates of arbitrary points of the separator whose motion needs to be examined, as well as its required performance parameters, may be introduced to these expressions. As a result, they will enable constructing the trajectories of displacement of the selected points of separator and creating their amplitude-frequency and amplitude characteristics. The basis for a mathematical model to describe dynamics of vibratory separator with concentric arrangement of sieves will be formed by analytical expressions that describe the motion of some of its "key" points. These may include, for example, coordinates of the position axis of concentric sieves of separator, its center of mass and the magnitude of rotation angle of separator around its center of mass. If these laws are available, then by relatively simple mathematical manipulations, it is possible to explore dynamics of the arbitrary part of separator.

Thus, in order to construct a mathematical model for examining the dynamics of vibratory separator, we first selected its principle and calculated scheme. A system of nonlinear differential equations, which reflects the motion of vibratory separator, is built using generalized coordinates of the Lagrange equation of second kind [12] and the Newton's laws of classical dynamics. Analytical solution of the resulting system of equations is derived by employing asymptotic methods of nonlinear mechanics [13–15], the Bogolyubov-Mitropolsky method, method of Poincaré, and a general method of perturbation theory [13].

In the process of examining the solutions found, we conducted their test on stability, in other words, to verify existence of the solution over the entire time interval in the operation of vibratory separator. For this purpose, we used a technique for testing stability with the help of equation of the Mathieu type [16, 17].

In the proposed design of vibratory separator, we employ several independent unbalanced drives. That is why it was appropriate to determine conditions for their self-synchronization according to the integral criteria for describing stability of synchronous motion [18].

In order to realize experimental studies, we used an experimental mockup model of separator, which is set into motion by two debalances that have an independent drive. With its help, we practically tested the impact of structural and kinematic parameters of separator on:

1) the amplitude of its working body vibrations;
2) the trajectory of its motion and character of vibrations.

We tested the impact of the following design parameters of separator – dimensions of the working drum, its weight, arrangement of vibration actuators and elastic suspension, rigidity of suspension, radii of debalances, and their mass. We also tested influence of angular velocities and rotation direction of the debalances as kinematic indicators. The experimental mockup model allowed us to regulate and change key parameters of separator, which affected its performance efficiency.

In order to determine experimental impact of the separator parameters on the character of vibrations of its sieves, a methodology for conducting the experiment was developed. A vibration measurement system was used as measuring and registering tools. The measuring system includes:

1) primary converter "VS-080 (Germany) – a sensor that was mounted on the vibratory separator;
2) device for processing primary information "VIBROTEST 30" made by SCHENCK (Germany).

As the recording module we used:

1) 16-bit signal converter ES-1868 (Taiwan);
2) personal computer (Intel Core II Duo, 2000 GHz, 4 GB RAM);
3) software package "Sound Forge" (USA).

5. Results of examining dynamics of vibratory separator of the drum type with concentric arrangement of sieves

5. 1. Schematic diagram of the designed drum vibratory separator with concentric arrangement of sieves

Fig. 1 shows three projections of schematic diagram of the proposed design of vibratory separator whose dynamics was examined in the present study by means of modeling.

The following designations are used in Fig. 1:

1) drum of vibratory separator;
2) concentric sieves (4 pieces);
3) driving electric motors (4 pieces);
4) independent controlled debalances;
5) the largest separated fraction;
6) the smallest separated fraction;
7) elastic suspension;
8) loading bunker;
9) system of unloading from each sieve and bottom of the drum;
10) elastic couplings;
11) frame of vibratory separator.

On this drum vibratory separator with concentric arrangement of sieves, we fix two pairs of debalances that will rotate in the planes perpendicular to the axis of the separator drum. In addition, the given pairs of debalances are set at the edges of the drum (in the area of the right and left end). The separator drum itself will have four built one-in-one concentric sieves with gaps between them for the accumulation of separated fractions, and stirring the residues of mixture that was received from the sieve closest to the axis of the drum. It is obvious that the size of the four sieve cells will decrease in the radial direction. Thus, the largest fraction will be accumulated in the middle of the drum, and the smallest – near the walls of the drum. The rest, intermediate-sized, fractions will be accumulated between the sieves. Thus, such a design of separator will be able to divide the mixture of substances that should be separated into five parts.

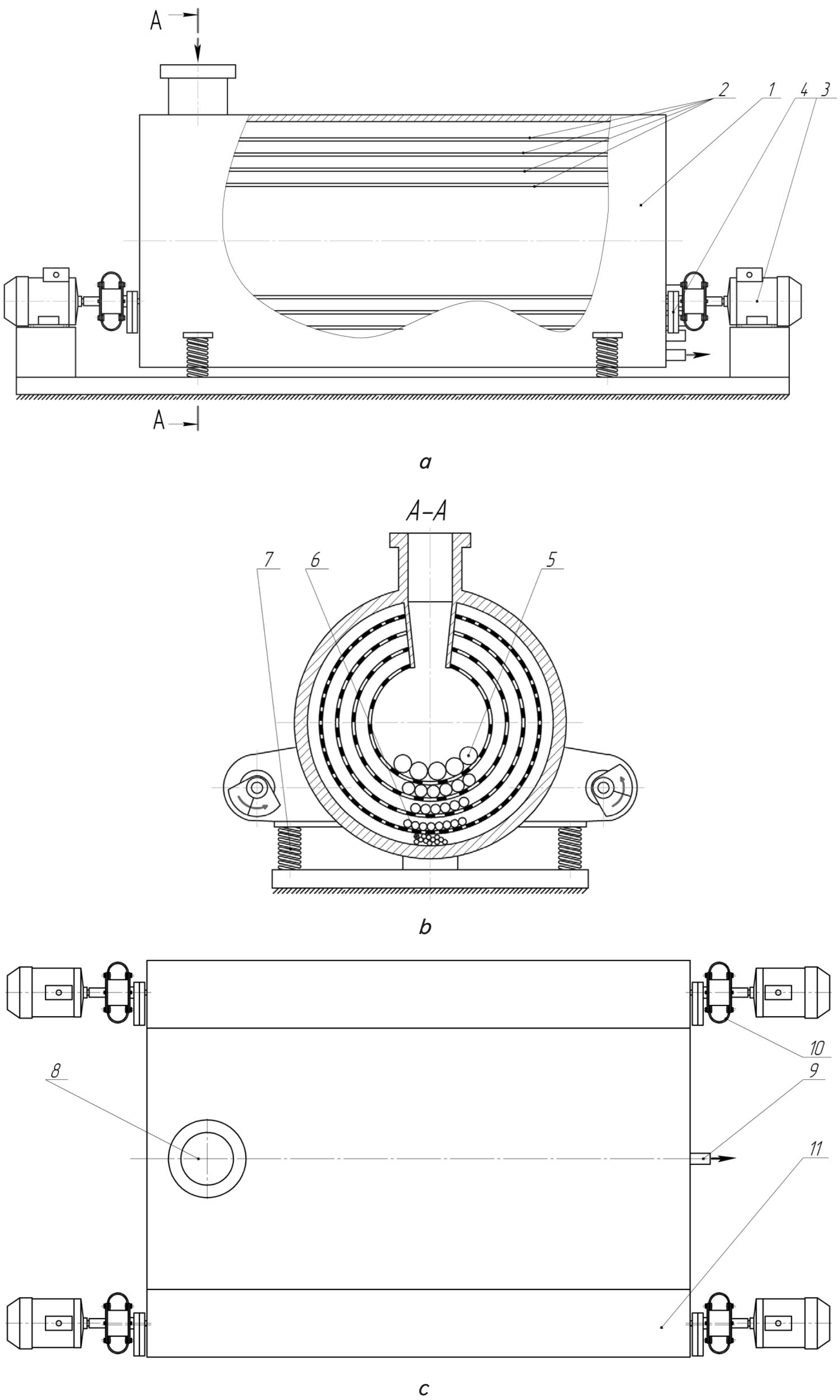

Fig. 1. Vibratory separator of the drum type with concentric arrangement of sieves and four debalances that have an independent drive: *a* — vibratory separator — main view; *b* — vibratory separator — side view (cross-section); *c* — vibratory separator — top view

5. 2. The estimated scheme of drum separator with concentric arrangement of sieves and vibration drive

Debalances of the separator drive will rotate in the plane perpendicular to the axis of the separator drum causing its flat parallel motion. In other words, the drum will be in the flat motion. Of course, it will move along its axis as well. However, these axial vibrations are minor if compared to the fluctuations that occur vertically – in the plane of cross section of the separator, so they can be neglected.

As a result, we have a mechanical system that is flat and in general will have 5 levels of freedom. In particular, the separator drum has three levels of freedom and debalance of each pair has one degree. As we consider the cross-section of separator as a whole, then this cross-section may include only one pair of debalances. Debalances perform rotating motion around its axis, which is rigidly fixed relative to the separator drum. Accordingly, to the total number of levels of freedom of the examined mechanical system we can add only 2 levels of freedom of the debalances. As a result of the given considerations, the examined system will have five levels of freedom in total.

Fig. 2 shows the developed estimated scheme of the drum separator with concentric arrangement of sieves and a vibration drive.

The following designations are used in Fig. 2:

– 1 – drum of separator, which has total mass M_K;

– I – inertia moment of vibratory separator relative to the axis, which crosses the center of mass of the separator (denoted as O_3);

– 2, 3 – left and right debalance;

– 4 – spring suspension;

– 5, 6, 7, 8 – separating cylindrical sieves placed concentrically (cell size of the sieve decreases from the inner to the outer sieve);

– 9, 10 – separated fractions of the largest and the smallest size, respectively;

– 11 – loading bunker;

– 12 – frame of the separator;

– XOY – stationary coordinate system;

– $X_1O_3Y_1$ – coordinate system that is connected to the separator drum, oscillates with it);

– ϕ – rotation angle of the separator in the course of its operation;

– c – point of the center of mass of the separator drum;

– $S=cO_3$ – distance from the center of coordinate system $X_1O_3Y_1$ to the center of mass of the separator;

– O_{D1}, O_{D2} – points around which debalances rotate;

– r_1 – radius of the left debalance (displacement of the center of mass of the left debalance relative to the axis of its rotational motion);

– r_2 – radius of the right debalance (displacement of the center of mass of the right debalance relative to the axis of its rotational motion);

– M_{D1} – weight of the left debalance;

– M_{D2} – weight of the right debalance;

– $O_{D1}L_1=k_1$, $O_{D2}L_2=k_2$, $O_3L_1=l_1$, $O_3L_2=l_2$ – geometric coordinates of location of the debalances relative to the coordinate system associated with the separator drum;

– ω_1 and ω_2 – angular velocities at which the debalances move;

– α_0 and ψ_0 – initial phases of arrangement of the debalances;

– $\alpha = \omega_1 t$ and $\psi = \omega_2 t$ – phases of the debalances in an arbitrary point of time of separator operation;

– C – total rigidity of separator suspension, C1 – magnitude of rigidity of the left suspension, C2 – rigidity of the right suspension;

– WF_1 and MQ_1 – appropriate magnitudes of separator suspension lengths at any point of time of separator operation;

– WF and MQ – appropriate magnitudes of separator suspension lengths at the initial time.

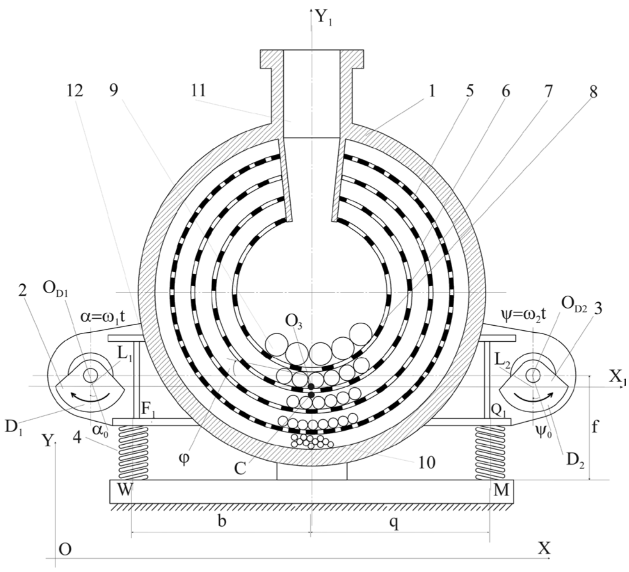

Fig. 2. Estimated scheme of the drum vibratory separator with concentric arrangement of sieves

5. 3. A mathematical model of examining the dynamics of separator

Dynamics of the examined designed separator can be rather accurately described by using a constructed system of differential equations (it is nonlinear and parametric):

$$\begin{cases} \ddot{x}_c + \omega^2 x_c = \varepsilon f_x(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0); \\ \ddot{y}_c + \omega^2 y_c = \varepsilon f_y(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0); \\ \ddot{\phi} + \omega_\phi(t)\phi = \varepsilon f_\phi(\phi, \dot{\phi}, \ddot{x}_c, \ddot{y}_c). \end{cases} \quad (1)$$

The following designations are used in system (1):

– x_c, y_c – coordinates of the center of mass of the separator drum with sieves in time t (time of separator operation);

– φ – magnitude of rotation angle of the separator during its operation;

– ω – frequency of natural oscillations of the drum;

– ε – the so-called low parameter;

– f_x, f_y, f_φ – nonlinear expressions that include parameters of the separator;

– $\omega_\varphi(t)$ – the so-called frequency of "circular" vibrations of the separator drum, which takes into account the debalance drive.

According to the asymptotic methods of nonlinear mechanics, the solutions to system (1) are as follows:

$$\begin{aligned} x_c &= x_0 \sin(\omega t + \alpha_x) + \varepsilon \chi_x\left(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0\right) + \\ &+ \varepsilon^2 \chi'_x\left(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0\right) + \ldots, \\ y_c &= y_0 \sin(\omega t + \alpha_y) + \varepsilon \chi_y\left(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0\right) + \\ &+ \varepsilon^2 \chi'_y\left(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0\right) + \ldots, \end{aligned} \quad (2)$$

where $\varepsilon = 1/M << 0$ (M is the oscillatory weight of separator); $\chi_i^j\left(\phi, \dot{\phi}, \ddot{\phi}, \omega_1 t + \alpha_0, \omega_2 t + \psi_0\right)$ are the functions that are defined by the asymptotic methods.

In order to obtain the final solution, we used only the first two members of the series – the first represents solution for the so-called unperturbed system (motion of separator excluding the drive), and the other one takes into account the motion of separator with enabled drive. Effect of the next members of the series on the adequacy of solution is irrelevant.

Eventually, the analytical expressions that describe motion of the separator drum during its operation:

$$\begin{aligned} x_c &= x_0 \sin\left(\sqrt{\frac{c}{M}} t + \alpha_x\right) + \\ &+ \frac{\varepsilon}{\sqrt{\frac{c}{M}}} \int_0^t f_x(\phi^*, \dot{\phi}^*, \ddot{\phi}^*, \omega_1 t + \alpha_0, \omega_2 t + \psi_0) \sin\left(\sqrt{\frac{C}{M}}(t - u)\right) du, \end{aligned} \quad (3)$$

$$\begin{aligned} y_c &= y_0 \sin\left(\sqrt{\frac{c}{M}} t + \alpha_y\right) + \\ &+ \frac{\varepsilon}{\sqrt{\frac{c}{M}}} \int_0^t f_y(\phi^*, \dot{\phi}^*, \ddot{\phi}^*, \omega_1 t + \alpha_0, \omega_2 t + \psi_0) \sin\left(\sqrt{\frac{C}{M}}(t - u)\right) du, \end{aligned} \quad (4)$$

$$\phi = \phi_0 \cos\left(\omega_0 t + \theta(t)\right). \quad (5)$$

In expressions (3)–(4),

$$f_x(\phi^*, \dot{\phi}^*, \ddot{\phi}^*, \omega_1 t + \alpha_0, \omega_2 t + \psi_0)$$

and

$$f_y(\phi^*, \dot{\phi}^*, \ddot{\phi}^*, \omega_1 t + \alpha_0, \omega_2 t + \psi_0)$$

are the non-linear functions with parameters of vibratory separator.

The system of equations (3)–(5) is rather cumbersome, which is why it is not represented in the expanded form. As an example, we shall demonstrate only a few of its embedded functions:

$$\omega_0 = \sqrt{\frac{-C_1\left(b^2 + af\right) - C_2\left(q^2 + af\right) + g\left(M_K S - M_{D1} k_1 - M_{D2} k_2\right)}{M_K S^2 + I + M_D\left(r^2 + l^2 + k^2\right)}}, \quad (6)$$

$$a = L + f - \frac{Mg}{C}, \quad (7)$$

$$\begin{aligned} \theta(t) &= \frac{B}{4 \cdot \phi_0}\left(l\left(\frac{\cos(\omega_2 t + \psi_0)}{\omega_2} - \frac{\cos(\omega_1 t + \alpha_0)}{\omega_1}\right) + \right. \\ &\left. + k\left(\frac{\sin(\omega_2 t + \psi_0)}{\omega_2} - \frac{\sin(\omega_1 t + \alpha_0)}{\omega_1}\right)\right) + C^*, \end{aligned} \quad (8)$$

$$\ddot{\phi}=-\omega_0^2\left(1+B\left(\cos\frac{(\omega_1+\omega_2)t+\alpha_0+\psi_0}{2}\times\right.\right.$$

$$\left.\left.\times\left(\sin\frac{(\omega_1-\omega_2)t+\alpha_0-\psi_0}{2}+\cos\frac{(\omega_1+\omega_2)t+\alpha_0+\psi_0}{2}\right)\right)\right)\cdot\phi, \quad (9)$$

$$B=\frac{4M_D r}{M_K S^2+I+M_D\left(r^2+l^2+k^2\right)}, \quad (10)$$

$$\dot{\phi}=-\phi_0\sin\left(\omega_0 t+\theta(t)\right)\times$$

$$\times\left(\omega_0+\frac{B}{2\phi_0}\left(\cos\frac{(\omega_1+\omega_2)t+\alpha_0+\psi_0}{2}\times\right.\right.$$

$$\left.\left.\times\sin\frac{(\omega_1-\omega_2)t+\alpha_0-\psi_0}{2}+\cos\frac{(\omega_1-\omega_2)t+\alpha_0-\psi_0}{2}\right)\right). \quad (11)$$

System (3)–(5) allows us to describe dynamics of the center of mass of vibratory separator. Expressions that will make it possible to find and describe a trajectory of motion of arbitrary point of vibratory drum separator with concentric arrangement of sieves:

$$x_{i3}=x_c+x_i\cos\phi(t)-y_i\sin\phi(t),$$

$$y_{i3}=y_c+x_i\sin\phi(t)+y_i\cos\phi(t), \quad (12)$$

where x_{i3}, y_{i3} are the coordinates of the point of separator whose motion it is necessary to examine, relative to the XOY coordinate system; x_i, y_i are the coordinates of the point of separator whose motion it is necessary to examine, relative to the $X_1O_3Y_1$ coordinate system.

In the process of constructing a mathematical model, we tested stability of the obtained solutions (3)–(5) for system (1). The result indicates that within the operational and design parameters of the examined separator, the resulting system of analytical dependences will describe dynamics of the separator over the entire time interval of its operation.

5. 4. Exploring impact of the separator parameters on the amplitude of vibrational motion of the drum with sieves

The next step of the studies was to explore the magnitude of impact of the separator parameters on the amplitude of vibrational motion of the drum with sieves. In this case, we examined stationary and transitional modes of the separator operation. These studies were carried out based on the obtained mathematical model that describes dynamics of vibratory separator of the drum type and the system of automated mathematical calculations MathCAD. It was accepted that the amplitude of vibrations of the separator drum is a key factor for determining intensity of the separation process.

We used the following parameters of separator for conducting the studies:

$r_1=r_2=0{,}065$ m – debalances radii (magnitudes of the debalances eccentricity);

$\omega_1=\omega_2=95$ s^{-1} – angular velocities of debalances;

$M_{D1}=M_{D2}=2.5$ kg – masses of the debalances;

$k_1=k_2=0.003$ m; $l_1=l_2=0.65$ m – location parameters of the rotation axes of debalances;

$\alpha_0=\psi_0=0^o$ – initial angles of the debalances position;

L=0.2 m – length of suspension springs;

b=q=0.6 m – location parameters of the separator elastic suspension;

S=0.1 m – distance from the drum axis to the center of its mass in the initial moment of time;

$M_K=45$ kg – weight of separator drum;

$C_1=C_2=12.0$ kN/m – magnitudes of rigidity of the separator suspension springs (from the left and right sides);

I=6.5 kg·m^2 – inertia moment of the separator drum.

The charts below (Fig. 3–5) show the impact of parameters of vibratory separator of the drum type – radii of the debalances, their mass, rotation angular velocities – on the amplitude and character of its oscillations. Graphical dependences were built for the relative time interval of separator operation that lasted for 1–2 s for the stationary mode of its operation. The picture of oscillations for other time intervals is similar.

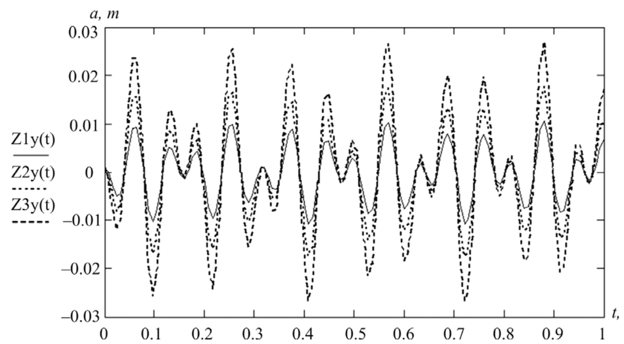

Fig. 3. Impact of the debalances radii on the magnitude of oscillation amplitude of the separator drum at Z1 – $r_1=r_2=0.045$ m, Z2 – $r_1=r_2=0.09$ m, Z3 – $r_1=r_2=0.135$ m

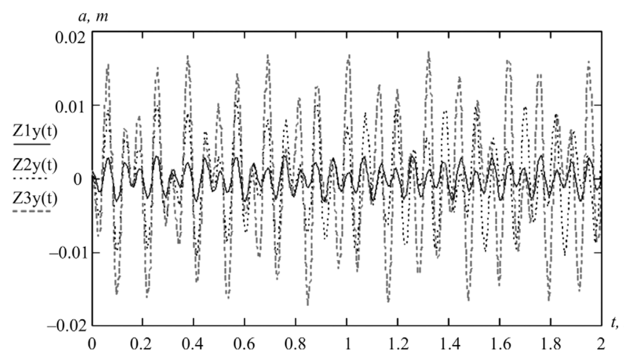

Fig. 4. Impact of the debalances masses on the magnitude of oscillation amplitude of the separator drum at Z1 – $M_{D1}=M_{D2}=1$ kg, Z2 – $M_{D1}=M_{D2}=2.5$ kg, Z3 – $M_{D1}=M_{D2}=4$ kg

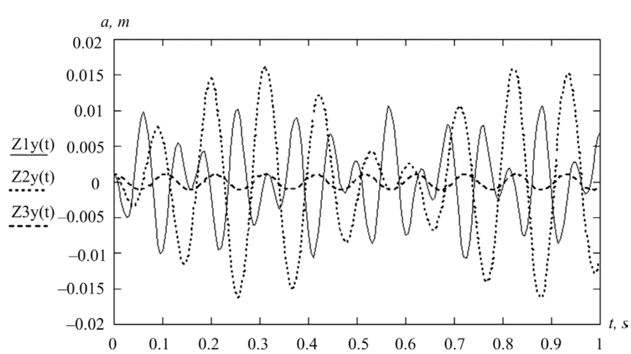

Fig. 5. Impact of the debalances rotation angular velocities on the magnitude of oscillation amplitude of the separator drum at Z1 – $\omega1=95$ s^{-1}, $\omega2=-95$ s^{-1}, Z2 – $\omega1=50$ s^{-1}, $\omega2=-50$ s^{-1}, Z3 – $\omega1=140$ s^{-1}, $\omega2=-140$ s^{-1}

The next step of the studies was constructing direct graphical dependences of parameters of the examined separator of the drum type on the amplitude of its oscillations when one variant was variable while the others were fixed. As an example, certain dependences on the key parameters of separator are shown in Fig. 6–8.

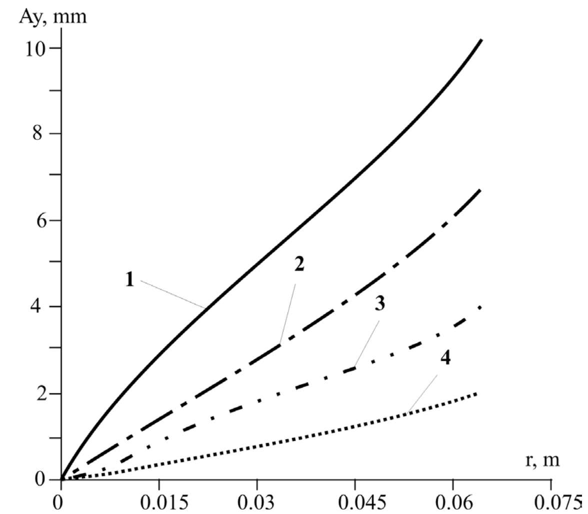

Fig. 6. Dependence of oscillation amplitude of vibratory separator on the radius of debalances: 1 – mass of debalances 8 kg; 2 – 6 kg; 3 – 4 kg, 4 – 2 kg

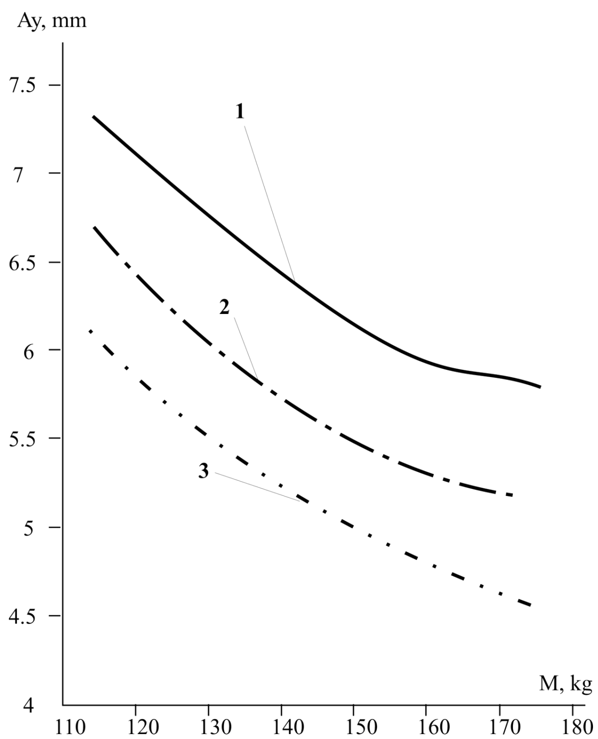

Fig. 7. Dependence of oscillation amplitude of vibratory separator on the oscillatory mass at the debalances angular velocity $\omega_1=\omega_2=110\ s^{-1}$: 1 – suspension rigidity 10 kN/m; 2 – 13 kN/m; 3 – 16 kN/m

Based on the integral criterion of determining stability conditions for the synchronous motion of independently driven debalances of separator, we established conditions of their synchronized oscillations. It is determined that:

a) debalances rotate synchronously and in antiphase in one direction at a total weight of separator drum with sieves and the separated mixture to 78.43 kg;

b) if oscillatory mass exceeds 78.43 kg, the debalances will rotate synchronously to each other and in phase in one direction;

c) when the debalances rotate in opposite directions, the phenomenon of self-synchronization will be observed only under condition that the total weight of oscillatory system does not exceed 78.43 kg. In this case, the debalances will move synchronously and in antiphase.

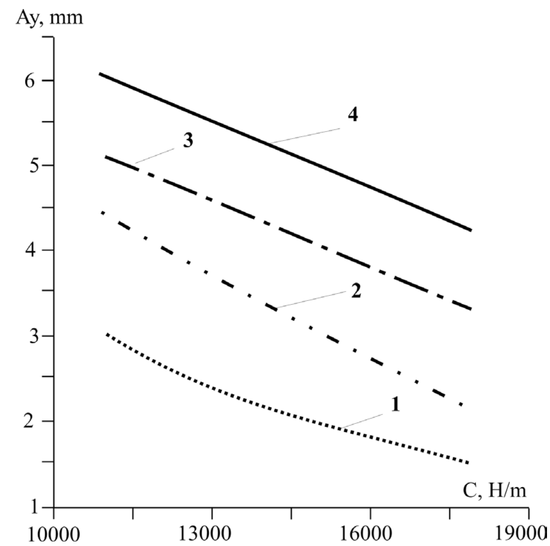

Fig. 8. Dependence of oscillation amplitude of vibratory separator on the magnitude of rigidity of the suspension springs: 1 – magnitude of the debalance radius (eccentricity) 0.015 m; 2 – 0.03 m; 3 – 0.045 m; 4 – 0.06 m

5. 5. Experimental verification of results of theoretical studies into dynamics of separator of the drum type with a vibration drive

As a result of conducting experimental studies, we obtained a number of vibrograms. They reflect the motion trajectory of certain, preliminary chosen, points of the front wall of the separator drum. Trajectories are received for the vertical and horizontal motion directions of points at different design and kinematic parameters of the examined separator. In addition, using the derived vibrograms, we obtained a spectrum of amplitude-frequency characteristics in the motion of the separator drum. This, in turn, enabled determining the impact of components of the separator drum perturbations that predetermine this very picture of its motion on the magnitude of amplitude of its oscillations by their frequency. Determining, by the amplitude-frequency characteristics, the dependences of oscillation amplitude of the drum on the frequency of disturbance components allows us to establish their significance in the impact on the separator motion character. As follows from the obtained dependences, predominant is the frequency of external disturbance – rotation frequency of the debalances (in the examined case, 22.5 Hz at revolution number 1350 RPM). The character of drum motion is also affected by the separated mixture motion in sieves (lower frequencies), but the frequency of their oscillatory motion and the amplitude of corresponding harmonics are lower.

At the next stage, design and kinematic parameters of separator of the drum type with a vibration drive, by which the experiment was conducted, were introduced to the resulting nonlinear mathematical model presented above. For the same time interval of stationary work of vibratory separator, we built theoretical vibrograms for the motion of its points. A comparison of the results obtained indicate adequacy in the representation by received mathematical model for a vibratory separator of the drum type of real dynamical processes in it.

6. Discussion of results of examining dynamics of separator of the drum type with concentric arrangement of sieves and a vibration drive

The factors of uniformity and adequacy are key in the study of dynamics of mechanical systems by the modeling method. If the built model of mechanical system proves to be inadequate, then the feasibility of conducting theoretical research by employing it does not make sense – the results will be incorrect. On the other hand, when devising the models, one should considere such a concept as the uniformity of the model. In practice, there are certain groups of mechanical systems that have common features. These, for example, include separators (with a vibration drive) with the same quantity of sieves in the drum or the debalances, the same type of suspension, etc. That is why, for these types of systems, it is expedient to develop a parameterized model that will implement key operational properties of all the separators of the selected group. Then it is possible to set to zero appropriate parameters of the constructed model, such as the magnitudes of weight of the drum, if there are several of them, or mass of one of several driving debalances. It is also possible to carry out variation of the corresponding parameters of the system's geometry, etc. As a result of such manipulations, the uniform model will reflect influence of parameters on the intensity factors in the operation of one or another actual vibratory separator from the group under consideration. Such model, in this case, can be successfully applied to a group of vibratory processing systems of another type. The indicated approach to the creation and application of models of vibratory mechanical systems will make it possible to reduce the time that would be required for their design. This is explained by the fact that there will be no need to develop from scratch, for example, a new model of vibratory separator (which will be designed). Designers will employ the models of separators that are unified, which were devised in advance. Of course, providing for the fulfillment of the given factors in the design of models of this particular type is a difficult task. However, the application of simulation when designing and operating separators and other vibratory mechanical systems is very convenient and practical. Solving this problem will enable even the automated design of vibratory processing mechanical systems.

The resulting mathematical model of the developed separator is rather universal, as it can describe not only separators of the aforementioned structural design. A several subtypes of designs of the given drum separator with concentric arrangement of sieves and a vibration drive are possible, which can be described by this model. It might be also applied to study dynamics of the separator whose sieves with cells of different size are arranged in sequence.

Many key parameters of separator, which affect its operation intensity, can be easily changed. Here we include radius, angular velocity, mass of the debalances and rigidity of suspension. Thus, the radii of debalances and their mass can be changed by using regulated debalances on separator. Angular velocity of the drive – by using direct current drive motors or induction motors (with with a phase rotor), or by changing current frequency. Rigidity of the suspension can be adjusted by changing rigidity of the suspension springs or their number, or by using pneumatic suspension. That is why the aforementioned parameters can be considered basic by the impact on the amplitude magnitude. Influence of geometric parameters of the designed separator and geometry of fixing the debalances should be considered at the stage of its design.

As a result of the research conducted, it was established:

– value of the amplitude magnitude of the oscillatory motion of vibratory separator is in nonlinear dependence on the mass of the drum with sieves and bulk mixture and decreases with the increase in the latter. For example, with the growth of oscillatory mass from 115 to 175 kg (by 1.52 times), the oscillation amplitude decreased by 1.19 times, in other words, by 1.25 mm. Present research was conducted for the system with suspension rigidity 10 kN/m and angular velocity of the debalances 70 s^{-1};

– oscillation amplitude of vibratory separator is also in nonlinear dependence on the angular velocity of the driven debalances and increases at its reduction. Thus, for example, with a decrease in angular velocity of the rotational motion of the driven debalances from 150 to 70 s^{-1} (by 2.14 times) – the value of amplitude of vertical oscillations increased by 1.53 times, in other words, by 2 mm. The research was conducted for the separator with suspension rigidity of 16 kN/m and oscillatory mass of 140 kg;

– oscillation amplitude of vibratory separator is in nonlinear dependence on the rigidity magnitude of suspension system. It decreases with increasing rigidity. For example, with the growth of rigidity magnitude of the elastic suspension from 10 kN/m to 16 kN/m (by 1.6 times) – the value of magnitude of the drum vertical vibration amplitude decreased by 1.28 times, in other words, by 1.4 mm. The research was conducted for the separator with oscillatory mass of 80 kg and angular velocity of the debalances at 110 s^{-1};

– oscillation amplitude of vibratory separator is in nonlinear dependence on the radius of the driven debalances and increases with its growth. Thus, for example, for the debalance of 8 kg, when increasing the radius from 0.015 to 0.06 m (by 4 times), the amplitude increases by 3.4 times, that is, by 6.7 mm;

– oscillation amplitude of vibratory separator is also in nonlinear dependence on the mass of debalances of the drive. It grows with an increase in the mass of debalance. Thus, for example, a increase in the mass of debalance from 2 to 8 kg (by 4 times) leads to the increase in oscillation amplitude of the drum by 6.8 times, in other words, by 5.8 mm. Here we examine the separator with radius of the driven debalance at 0.045 m;

– the possibility to change radius and mass of the debalances in the operation of separator and significance of their influence on the intensity of the separation process is very important. It should be emphasized that it is possible to consider significance of impact on the amplitude of separator oscillating motion from the static moment of debalances, which is the product of the debalance mass by its radius. However, studying separate influence of the mass of debalance and its radius is required for the debalances, which consist of several moving parts. This can be explained by the fact that these magnitudes can vary independent from each other in the process of establishing the separation modes;

– with an increase in the distance between rotation centers of the driven debalances, the oscillation amplitude of separator decreases, and in this case, the dependence is nonlinear;

– with an increase in the distance between suspension springs in the cross-section of separator, which is perpendicular to the drum axis, the amplitude of its oscillations increases with other parameters unchanged.

In addition, in the course of research, we selected a criterion by which we determine operation intensity of vibratory separator and select its parameters. In particular, it demonstrates that the operation intensity of vibratory system is proportional to the product of square of the oscillation amplitude of the drum by the magnitude of angular velocity cube of the drive [19]. Thus, the values of estimated parameters of the designed separator of the drum type with a vibration drive will be as follows:

a) magnitude of rigidity of the separator suspension – total 2·13=26 kN/m;

b) angular velocity values of debalances of the drive is 150 s^{-1}. Oscillation amplitude of the separator in this case will have a lower value than that at angular speed 110 s^{-1} or angular velocity 70 s^{-1}. However, the product of square of the oscillation amplitude by the magnitude of angular velocity cube of the drive in this case, according to the selected criteria for estimating the separator operation, will be the largest;

c) mass of debalances – 4–8 kg;

d) radius magnitude of debalances – 0.015–0.07 m;

e) distance between elastic supports – 1.25–1.55 m;

f) fixing the debalance nodes at a distance of 0.35 m from the bottom of separator drum.

In addition, debalances of the separator must rotate in one direction that will provide the most intensive stirring and circulation of the separated mixture, in the sieves.

By analyzing the obtained experimental and theoretical results, one can argue that the mathematical model of separator describes the process of its work with sufficient accuracy. This is evidenced by:

a) the existence of equivalence in the shapes of received vibrograms (theoretically and experimentally);

b) proportionality in the magnitudes of theoretical and experimental amplitudes of the examined points of separator. A mean relative error for vertical coordinates of the given points is 11.4 %, for horizontal coordinates – 15.2 %.

Therefore, this model is suitable for practical application to study the impact of kinematic and design parameters of separator with a vibration drive on the character of its motion to improve intensity of the process of vibratory separation.

7. Conclusions

1. We constructed a nonlinear mathematical model for the developed new design of vibratory separator of the drum type with concentric arrangement of sieves and a debalance drive. The model is unified and parametric. Its structure in character format includes geometric and kinematic parameters of the separator. This model allowed us to explore influence of the separator parameters on the character of motion of its sieves (the drum of separator) during its operation. It was established which parameters of the separator provide for the motion of sieves with maximum continuous tossing and stirring the mixture in them. The given character of the sieve motion ensures the fastest process of separation and unloading of the separated fractions out of the separator. We relate to the indicated parameters: angular speed and weight of debalances, the magnitude of their eccentricity, rigidity of suspension, the magnitude of oscillatory mass (of the drum with sieves and the mixture in it), geometric characteristics of the separator. The latter include position of the suspension and debalances.

2. The adequacy of the constructed nonlinear mathematical model for vibratory separator of the drum type was experimentally confirmed in terms of reflecting the oscillatory processes in it. A mean relative error when comparing theoretical and experimental results does not exceed 15.2 %. It is established that the formation of motion character of the separator sieves and separator as a whole is determined by the dominant frequency of external disturbance – frequency of debalance rotation. The character of oscillatory motion of the separator is also affected by the motion of mixture in the sieves, though this impact is not significant.

3. It was found that angular velocity, mass of debalances, the magnitude of their eccentricity, rigidity of suspension, weight of the drum with sieves and the mixture in it, the position of suspension and debalances most significantly affect oscillation amplitude of the drum with sieves. In this case, we recommend accepting oscillation amplitude of separator combined with angular velocity of the separator drive as the factors that determine operation intensity of the separator. It was established that the oscillation amplitude of separator with the aforementioned parameters is in nonlinear dependence. In addition, the magnitudes of mass of the debalance, its eccentricity (radius), rigidity of suspension can be easily changed by assigning the separation modes in the separator operation.

Research into various dynamic processes of vibratory mechanical systems, in particular separators, based on mathematical modeling is a relevant applied task. It enables exploring comprehensive impact of the separator parameters on its operation intensity. This makes it possible, at the design stage of separator, to reduce expensive experimental research, and to automate the process of designing. Results received in the present study might be applied in order to develop and operate vibratory processing systems of different types.

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