Manuscript Details

Manuscript number	YBENG_2018_82
Title	Theoretical and experimental investigation of a vibratory mill for fine grinding of grain
Article type	Research Paper

Abstract

Investigations of the developed mathematical model of a vibratory mill made it possible to find out regularities for the complex movement of its operating mechanisms and to obtain a graphic interpretation for the kinematic, power and energy characteristics of the vibratory system to be developed. On the basis of theoretical pre-requisites, a series of experimental studies were conducted that provided an opportunity to obtain amplitude-frequency, velocity and energy characteristics of the developed machine, to determine their impact upon the kinetics of the fine grinding process of the agricultural raw material. As a result, rational operating parameters of the examined machine were established with minimal energy input.

Keywords	Vibratory mill; Fine grinding; Mathematical model; Operating mode.
Manuscript region of origin	Europe

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Highlights

- Developed vibratory mill of angular oscillations, which allows to increase the grinding efficiency of raw material.
- Theoretical studies of the process of fine grinding bulk raw materials confirmed by experimental tests.
- Rational operating parameters established with minimal energy input.

Theoretical and experimental investigation of a vibratory mill for fine grinding of grain

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Abstract

Investigations of the developed mathematical model of a vibratory mill made it possible to find out regularities for the complex movement of its operating mechanisms and to obtain a graphic interpretation for the kinematic, power and energy characteristics of the vibratory system to be developed. On the basis of theoretical pre-requisites, a series of experimental studies were conducted that provided an opportunity to obtain amplitude-frequency, velocity and energy characteristics of the developed machine, to determine their impact upon the kinetics of the fine grinding process of the agricultural raw material. As a result, rational operating parameters of the examined machine were established with minimal energy input.

Key words:

Vibratory mill, Fine grinding, Mathematical model, Operating mode.

1. Introduction

Grain refinement is the oldest technology used by humanity to produce foodstuffs. The main technological task of grinding grain is to obtain a homogeneous mixture with a desired degree of grinding of the ingredients (Linch, 2005; Butkovsky, 1990). On the other hand, the refining process at the mill allows to separate components, such as the parts of the wheat grain: the endosperm containing mostly starch and proteins, the germ composed mostly of lipids and proteins and the bran containing mainly dietary fiber (Liu et al., 2015; Marquart et al., 2007). At present, there are widely used various types of mills and crushers for the production of flour for food and animal fodder (Savinyh et al, 2016, Sysuev et al, 2016; Cho et al., 2013). As example, the production of whole grain flour is mainly affected by the milling process and actually the predominant techniques for grinding whole grain flours are stone mill, roller mill, ultra-fine mill and hammer mill (Kihlberg et al., 2004; Kent and Evers, 1994). Research in the intensification of grain refinement is reflected in many works (Bulgakov et al, 2017, Sysuev et al, 2015; Rosenkranz et al., 2011; Rajamani et al., 2000); however, the operation of a vibratory mill for fine grinding has been so far studied not enough. As a result of the execution of the vibratory action, there occur a significant decrease in the coefficient of internal friction, an active increase in the specific surface area of the material when interacting with the technological filler, efficient creating of essential gradients in the rates of internal deformations in the micro- and macro-volumes of the product on condition that its structural mechanical characteristics are changed.

When implementing the fine grinding process, the vibratory action makes it possible to substantially increase the shock-erasing effect due to the possibility of technological variation of its strength and friction components; on the one hand, to increase the destruction rate of the material particles under the impact of cyclic loads, but on the other hand, – as a result of dynamic interaction between each other, it ensures their ability of active abrasion (Doumanidis et al., 2016; Janovich et al., 2016; Mori et al., 2004; Gonzalez, 1995). To the main shortcomings of the traditional machines for the production of fine and highly disperse material one should relate significant specific energy input into the processing of raw materials, low performance characteristics of the operating mechanisms due to the active abrasion of their operating surfaces, and lowering of their technological efficiency as a result of adhesion of products with increased humidity (Kaletnik, 2016, Nasir, 2005, Yaroshevich, 2011).

In order to eliminate the above-mentioned shortcomings in the intensification of the processes of grinding agricultural raw materials and to increase the operational and technological parameters of machines for their implementation, it is proposed to apply a complex vibromechanical impact, which is the purpose of this scientific work, justifying its topicality and prospects for its implementation.

The purpose of this scientific work is to develop a mathematical model of a vibratory mill and to carry out experimental studies on the substantiation of the technological parameters of its operation under the condition of an intensified grain refinement process, and to increase the operational and technological indicators of machines on the basis of a complex vibro-mechanical impact.

2. Materials and methods

An experimental vibratory mill of angular oscillations was used for the studies (Janovich, 2015; Franchuk, 1970) but the theoretical investigations were carried out on the basis of its equivalent scheme. Estimation of the energy consumption of the developed machine, its amplitude-frequency, kinetic and operational parameters was carried out on the basis of experimental studies, but the analysis of the kinematic characteristics of the vibrator (vibration acceleration, vibration velocity and vibrodisplacement) was conducted applying the developed software for commutation.

2.1. The measure chain

In order to determine the amplitude-frequency characteristics, an accelerometer analyser was developed, based on the 3-axis accelerometer LIS3DH (STMicroelectronics, USA) (Fig. 1), whose main features were: dynamically user-selectable full scales of ±2g/±4g/±8g/±16g, capability of measuring accelerations with output data rates from 1 Hz to 5.3 kHz. .

The principle of operation of this device is the following: after connecting the sensor to the surface of the grinding chamber of the mill, the drive mechanism of the machine is switched on, due to which the resulting vibrations of the grinding chamber initiate inclusion of the integrated accelerometer that starts registration of the amplitude-frequency characteristics of the machine under study. To record the rotation speed of the drive shaft, the UNI-T UT372 (Uni-Trend Technology Limited, China), wireless tachometer was used, whose main technical features were: measurement 10 to 99,999 RPM, accuracy 0.04% +/- 2dgt. The control and change of the rotational speed of the motor shaft was carried out with the help of the autotransformer AOSN-20-220-75, which is intended for operation with alternating current. To determine the energy characteristics of the machine studied, the electronic wattmeter EMF-1 was used.

The dispersion ability of the material was determined through the method of mechanical separation of the parts – by the sieve analysis on a laboratory sieve analyzer A-20 (labTime, Russia), whose main technical features were: diameter of screen 200 mm, number of screen 4 to 10, amplitude of oscillations- 2 to 4, frequency of oscillation 1500 rad·s^{-1}, granularity 0.008 to 5 mm. It is used as standard screen machine grading all kinds powder material particle. In order to determine the relative humidity of the material, a Wile 55 moisture meter was applied, which can be used to measure the relative humidity of various types of grain and seeds, the data being stored in the memory of the instrument. Equipment Wile 55 (Farmcomp Oy, Finland) can measure a wide range of moisture, grain 8 to 35% and oil seeds 5 to 25%. Wile 55 is the basic tool for precise quick measurement of grain moisture. It has an easy-to-read LCD display and automatic temperature compensation that utilizes the internal temperature sensors for both grain and the temperature of the device itself.

Besides, it is equipped with a conditional scale showing the rough values of the measurements and can be used to measure the humidity of the samples not entered into the database of the moisture meter. To determine the specific surface area of the treated material during the vibratory grinding process with the particle size being 25-1000 microns, equipment PSH-9 was used, the action of which is based on measuring the air permeability of a layer of material through which air is leaking under pressure close to the atmospheric one. Equipment PSH-9 (Company Hranat, Russia) had the following main features: range of specific surface area of the material 300 to 50000 $m^2 \cdot g^{-1}$, time of one analysis 3.5 min, accuracy 0.5%.

To analyse the quality of the crushed raw material, a laboratory sampler was applied for sampling by the point selection method. To control the supply of material, a mobile vibratory dispenser PG-2 (Company Vybrotekhnyk, Russia) was used, whose main technical features were: size of the source material 0.05 to 5 mm, productivity 400 $kg \cdot h^{-1}$, amplitude of oscillation of the tray 2 mm, bunker volume 9 dm^3.

Thus, the presented structural design for carrying out laboratory and production tests allows us to determine precisely enough the main technological parameters of the angular oscillations of the developed mill and to evaluate the quality of the material obtained within the range of experimental studies.

2.2. The developed vibratory mill

The principle of operation of the developed vibratory mill (Janovich, 2014) is the following (Figure 2): when the electric motor 1 is switched on, the rotation torque is transmitted through the elastic clutch 2 to the drive shaft 3 with unbalances 4 arranged on it, rotation of which ensures generation of a combined force and moment imbalance of the oppositely arranged (on the periphery from the two sides) in relation to the central axis 5 grinding chambers 6 filled to 75% with the grinding elements in the form of steel balls. The grinding chambers, in their turn, are interconnected by traverses (crosspieces) 7. This design solution of the machine allows to provide angular oscillations, ensuring a significant dynamic state of the technological filler (promoting active collision of the grinding elements), and, as a consequence, it provides a possibility to increase the grinding efficiency of the raw material being processed continuously and entering through the feed pipes 8 from the hopper 9. Destruction of the raw material particles takes place due to cyclic loads created by the crushing elements and their dynamic interaction with each other. When the particle size of the material to be crushed is reduced under the impact of the centrifugal forces and reversal loads, it is transported along the grinding chamber to the separating section where particles, cmaller than the separation holes of the sieve, are discharged from the mill through the pipe 10; the particles of the material that have a larger size – are crushed due to the active impact of crushing elements.

As an electric drive of the developed vibratory machine, an electric motor with a power of 0.75 kW and an operating rotation speed of 1500 min^{-1} was used. Preliminary studies of the experimental mill witnessed that its efficiency is 220 $kg \cdot h^{-1}$. To study the regularities of the process of fine grinding of bulk raw materials, it is proposed to determine the internal structure of the dynamical system under investigation and to find linear and nonlinear effects, and on this basis to formulate a mathematical model of the vibratory system (Dreizler and Ludde, 2010; Maitra and Prasad, 1985; John and Stephens, 1984,).

2.3. The mathematical model

In order to create a mathematical model of angular oscillations of the vibratory mill (based on the D'Alembert-Euler principle), an equivalent scheme was designed, characterised by five degrees of freedom (Fig. 3).

Accordingly, the differential equations of the movement of the operating mechanism of the machine are described by five independent generalised coordinates, i.e.:

ρ – the radial coordinate of positioning the grinding chamber relative to axes Oz, Ox, m;

φ – the displacement angle of the grinding chamber with respect to the vertical plane, rad;

α – the arrangement angle of the vibrodrive relative to the container, rad;

φ_1 – the displacement angle of the vibrator drive depending on the change of the position of the grinding chamber, rad;

φ_3 – the angle of rotation of the unbalance relative to its axis of rotation, rad.

O_{xz} – the fixed coordinate system;

O_1 – the centre of the container mass;

c_x, c_z – the spring stiffness in the direction of the corresponding axis, $N \cdot m^{-1}$;

F_c – the force of elasticity, N;

F_k – the centrifugal force, N.

Considering the peculiarity of the angular movement of the grinding chamber of the machine, the mathematical model of the dynamic system is formed in the polar coordinate system. The prerequisite for theoretical investigations is a search of operational and design parameters of the developed machine in which the vertical component of the vibrations of its grinding chambers will acquire its maximum value. It is possible to find out the relationship between the design and technological parameters using the initial Lagrange equations of the second kind.

The kinetic energy of the grinding chamber will be equal to:

$$T_c = \frac{1}{2}\left(m_c V_{O_1}^2 + J_c \dot{\varphi}_1^2\right). \qquad (1)$$

where m_c – the mass of the container, kg;

J_c – the inertia moment of the container, $kg \cdot m^2$;

$\dot{\varphi}_1$ – the angular velocity of the container, $rad \cdot s^{-1}$;

V_{O_1} – the velocity of point O_1 in the polar coordinate system:

$$V_{O_1}^2 = \dot{\rho}^2 + \left(\rho \cdot \dot{\varphi}\right)^2. \qquad (2)$$

In its turn, the kinetic energy of the unbalance will be equal to:

$$T_{unb} = \frac{1}{2}\left(m_{unb} V_{O_3}^2 + J_{unb} \dot{\varphi}_3^2\right), \qquad (3)$$

where m_{unb} – the mass of the unbalance, kg;

J_{unb} – the inertia moment of the unbalance, $kg \cdot m^2$;

$\dot{\varphi}_3$ – the angular velocity of unbalance, $rad \cdot s^{-1}$;

V_{O_3} – the velocity of the centre of masses of unbalance, that is determined using the polar coordinate system, where the pole to the system is located at this point. The speed V_{O_3} is then determined according to the following vector equation:

$$\overline{V}_{O_3} = \overline{V}_{O_1} + \overline{V}_{O_2O_1} + \overline{V}_{O_3O_2}\,, \tag{4}$$

where $\overline{V}_{O_1}$ – the vector velocity of point O_1 in the polar coordinate system;

$\overline{V}_{O_2O_1}$ – the vector velocity of point O_2 relatively point O_1;

$\overline{V}_{O_3O_2}$ – the vector velocity of point O_3 relatively point O_2;

In this case, the expression for the kinetic energy of the unbalance has the form:

$$\begin{aligned} T = {} & \frac{1}{2} m_c \left(\dot{\rho}^2 + \rho^2 \dot{\varphi}^2\right) + \frac{1}{2} J_c \dot{\varphi}_1^2 + \frac{1}{2} m_{unb} \big[\dot{\rho}^2 + \rho^2 \dot{\varphi}^2 + \dot{\varphi}_1^{\,2} l_2^{\,2} + \dot{\varphi}_3^{\,2} l_3^{\,2} + \\ & 2\dot{\rho}\dot{\varphi}_1 l_2 \cos\left(\varphi - \varphi_1 - \alpha\right) + 2\dot{\rho}\dot{\varphi}_3 l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right) + 2\rho\dot{\varphi}\dot{\varphi}_1 l_2 \sin\left(\alpha + \varphi_1 - \varphi\right) + \\ & + 2\rho\dot{\varphi}\dot{\varphi}_3 l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right) + 2\dot{\varphi}_1 l_2 \dot{\varphi}_3 l_3 \sin\left(\alpha - \varphi_3\right) \big] + \frac{1}{2} J_{unb} \dot{\varphi}_3^{\,2}. \end{aligned} \tag{5}$$

As a result of the partial derivatives found for the corresponding generalised coordinates, the generalised forces of the system and their functional transformations, we obtain a general form of the required Lagrange equations of the second kind that fully determine the movement regularity of the grinding chamber of the experimental mill:

$$\begin{aligned} & \left(m_c + m_{unb}\right) \ddot{\rho} \varphi_1 - m_c \rho \dot{\varphi}^{\,2} + m_{unb} [-\rho \dot{\varphi}^{\,2} + \ddot{\varphi}_1 l_2 \cos\left(\varphi - \varphi_1 - \alpha\right) + \\ & \left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right) + \dot{\varphi}_1^{\,2} l_2 \sin\left(\varphi - \varphi_1 - \alpha\right) - \left(\dot{\varphi}_1 + \dot{\varphi}_3\right)^2 \times \\ & \times l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right)] = -\left(c_x \cos^2 \varphi + c_z \sin^2 \varphi\right) \rho; \end{aligned} \tag{6}$$

$$\begin{aligned} & (m_c + m_{unb}) \rho^2 \ddot{\varphi} + 2(m_c + m_{unb}) \rho \dot{\rho} \dot{\varphi} + m_{unb} [\rho \ddot{\varphi}_1 l_2 \sin\left(\alpha + \varphi_1 - \varphi\right) + \\ & + \rho \dot{\varphi}_1^2 l_2 \cos\left(\alpha + \varphi_1 - \varphi\right) + \rho\left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right) + \\ & + \rho\left(\dot{\varphi}_1 + \dot{\varphi}_3\right)^2 l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right)] = \left(c_x - c_z\right) \rho^2 \sin\varphi \cos\varphi; \end{aligned} \tag{7}$$

$$
\begin{aligned}
&J_c\ddot{\varphi}_1 + J_{unb}\left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) + m_{unb}[\ddot{\varphi}_1 l_2^{\,2} + \left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) l_3^{\,2} + \ddot{\rho} l_2 \cos\left(\varphi - \varphi_1 - \alpha\right) + \\
&+\ddot{\rho} l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right) + \rho\ddot{\varphi} l_2 \sin\left(\alpha + \varphi_1 - \varphi\right) + \rho\ddot{\varphi} l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right) + \\
&+\left(2\ddot{\varphi}_1 + \ddot{\varphi}_3\right) l_2 l_3 \sin\left(\alpha - \varphi_3\right) - 2\dot{\rho}\dot{\varphi} l_2 \sin\left(\varphi - \varphi_1 - \alpha\right) + 2\dot{\rho}\dot{\varphi} l_3 \times \\
&\times\cos\left(\varphi - \varphi_1 - \varphi_3\right) - \rho\dot{\varphi}^2 l_2 \cos\left(\alpha + \varphi_1 - \varphi\right) - \rho\dot{\varphi}^2 l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right) - \\
&-\left(2\dot{\varphi}_1 + \dot{\varphi}_3\right)\dot{\varphi} l_2 l_3 \cos\left(\alpha - \varphi_3\right)] = -c_z\varphi_1 - m_{unb}\left[l_2 \sin\left(\alpha + \varphi_1\right) + l_3 \cos\left(\varphi_1 + \varphi_3\right)\right];
\end{aligned}
\tag{8}
$$

$$
\begin{aligned}
&J_{unb}\left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) + m_o[\left(\ddot{\varphi}_1 + \ddot{\varphi}_3\right) l_3^{\,2} + \ddot{\rho} l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right) + \rho\ddot{\varphi} l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right) + \\
&+\ddot{\varphi}_1 l_2 l_3 \sin\left(\alpha - \varphi_3\right) + 2\dot{\rho}\dot{\varphi} l_3 \cos\left(\varphi - \varphi_1 - \varphi_3\right) + \dot{\varphi}_1^{\,2} l_2 l_3 \cos\left(\alpha - \varphi_3\right) - \rho\dot{\varphi}^2 \times \\
&l_3 \sin\left(\varphi - \varphi_1 - \varphi_3\right)] = M_{unr} - m_{unb}\dot{\varphi}_3^2 l_2 l_3 \cos\left(\varphi_1 + \varphi_3\right);
\end{aligned}
\tag{9}
$$

$$
M_{unr} = \frac{2M_r\left(\omega_w - \omega_{\max}\right)\left(\omega_w - \omega\right)}{\left(\omega_w - \omega\right)^2 + \left(\omega - \omega_{\max}\right)^2};
\tag{10}
$$

where M_r – the rotation torque on the drive shaft, N·m;

M_{unr} – the moment of the support on the drive shaft, N·m;

ω, $\omega_{\max}$, ω_w – the respective initial, maximal and operating angular velocity of the drive shaft, rad·s^{-1}.

Assuming that M_r = const, we consider that $\omega = \omega_3$, ω_w = const, ω_r = const. We are looking for solutions like:

$$
\rho(t) = \rho_0 + \rho_1(t),
\tag{11}
$$

where $\rho_0 = const$ – the initial value ρ, m;

$\rho_1 = \varphi_3(t) = \omega t$ – the value of ρ as the function of time t, m;

$\varphi(t) = \omega t + \varphi_0(t)$ – the value of angle φ is due to the rotational movement of the unbalance.

As a result of solving and simplifying the obtained equations, based on the trigonometric connection of the polar and Cartesian coordinate systems, the obtained regularity of the movement of the grinding chamber of the mill will assume the form:

$$z(t) = -\frac{m_{unb} l_3 \sin \omega t}{m_c + m_{unb}} + l_2 \sin\left(\frac{3\pi}{2} + \frac{\left(m_{unb} m_c l_2 l_3 \cos(\omega t - \alpha)\right)\cos(\omega t - \alpha)}{\left(J_c + J_{unb} + m_{unb}(l_2^2 + l_3^2)\right)(m_c + m_{unb})}\right),$$

$$x(t) = -\frac{m_{unb} l_3 \sin \omega t}{m_c + m_{unb}} - l_2 \cos\left(-\frac{\left(m_{unb} m_c l_2 l_3 \cos(\omega t - \alpha)\right)\cos(\omega t - \alpha)}{\left(J_c + J_{unb} + m_{unb}(l_2^2 + l_3^2)\right)(m_c + m_{unb})}\right). \quad (12)$$

2.4. Data analysis

The processing of the experimental data was carried out using the methodology of planning a multifactor experiment and a variational analysis of the factor space in the Microsoft Excel, Statistika 10.0, FluidLab software environments. To assess the impact of the technological parameters of the developed vibrator upon the efficiency of the grinding process of a bulk raw material on condition that energy input is minimised, a statistical method of rotatable central compositional planning of a multifactor experiment was applied (Protasov, 2005).

Among the basic criteria for the evaluation of the grinding process of a bulk raw material, there were chosen the energy consumption of the vibrator *N*, kW·h, and the specific surface area of the initial raw material *S*, $cm^2 \cdot g^{-1}$, characterised by the impact of four most significant factors that determine the kinetics of the particular treatment: vibro-acceleration *a*, $m \cdot s^{-2}$, as a complex parameter of dynamic state of the vibratory system; the diameter of the balls *d*, mm and the degree of loading of the grinding chambers *E*, % with the technological filler.

The choice of the variation ranges of the factors of functions was made in such a way that any of their aggregates, as envisaged by the experimental design, could be embodied in these intervals and did not lead to contradictions.

3. Results and discussion

Analysis of the obtained dependencies (12) in the FluidLab software environment made it possible to establish that the maximal dynamic state of the system is provided at a 290-degree angle of the vibratory drive and the angular velocity of the drive shaft 120 rad·s^{-1} (Fig. 4). With these parameters the ratio of the components of the amplitude of the oscillations is A_z/A_x = 2.5 times (Fig. 5). In this case, the displacement amplitude of the grinding container along axis Oz = 5.4 mm, along axis Ox = 2.2 mm.

Also, it should be noted that the received graphic dependences of the amplitude of oscillations from the angle of placement of the vibration drive, have a symmetric distribution relative to the x-axis. The point of their symmetry is 180 degrees.

To verify the obtained analytical results of the theory, a number of experimental studies were carried out on the efficiency of fine grinding in the developed machine. It was established that the total amplitude of oscillations A at ω = 44 rad·s^{-1} without a technological load (E=0%) and at E=50% and E=75% reaches 3.8 mm, whereas in the resonance of the investigated vibratory system in the range ω = 50-57 rad·s^{-1} the value of the required parameter without a technological filler (E=0%) is A = 4.5 mm; at E=50% A = 4.2 mm; E=75% A = 4.0 mm. After that curve A in the range ω = 63-110 rad·s^{-1} stabilises by 3.7 mm, regardless of the degree of loading of the technological filler (Fig. 6).

The obtained data made it possible to evaluate the efficiency of the developed design in the context of minimising the dissipative properties of the investigated vibratory system, the criterion for the estimation of which being the change in the magnitude of the oscillation amplitude depending on the loading of the grinding chamber of the machine with the technological filler. This trend is due to the presence of a peripherally located drive shaft with relatively oppositely placed grinding chambers of the mill, the forced force of which considerably exceeds the inertial characteristics of the technological filler. The machine is characterised by angular oscillations of the container relative to the centrally located axis. Therefore, the obtained data on the dominance of the vertical component do not contradict the concept of the development of the investigated vibrator and ensure an effective dynamic state of processing the raw material. The analysis of the dependence of the energy consumption of the mill upon its angular velocity and loading with the technological filler witnessed (Fig. 7) that the filling degree has no significant effect on the summary energy input. Thus, in the absence of a technological filler (E=0%) $N = 0.65$ kW·h; at E=50% $N = 0.68$ kW·h; at E=75% $N = 0.75$ kW·h. On the basis of the obtained results one can make a conclusion that introduction of the peripheral placement of the vibrator drive is one of the key factors for the minimisation of energy consumption in the vibratory mills of this design solution while maintaining a significant power potential in the processing of bulk materials. In order to estimate the quality indicators of the vibratory grinding process of a bulk material processed in the mill, a number of experiments were carried out to change the dispersion ability of the treated raw materials, caused by the force of the technological filler (peas, the oat and wheat grains with a moisture content of 8-11% were used in the experiments). Evaluation of the grinding efficiency was based on the determination of the specific area S, the particle size distribution ΔS, μm, and the proper specific area of the material particles, which characterises the ratio of their geometric sizes before and after the treatment. When analysing the change in the specific area of the initial fraction of wheat under the condition that a vibratory mill of angular vibrations (Fig. 8) is used, it is evident that increase in S depending on the angular velocity of the drive shaft does not differ significantly, and at $\omega = 110$ rad·s^{-1}, $t = 55$ s it is 4300-4500 cm^{2}·g^{-1}. This trend is caused by a specific form of vibration, which considerably intensifies the process of active abrasion of the particles of the material processed.

At the same time, with the increase in the specific area of the material, there occurs active decrease in the proper area of the particles of the processed material and, as a consequence, the time of technological treatment, depending on the physical-mechanical parameters of the incoming raw material. On the basis of the obtained data about the kinetics of the pea, corn and wheat grinding process at ω = 110 rad·s^{-1} (Fig. 9), one can conclude that the approximate time of their processing will be t = 45 s at the particle size is: for grinding of peas ΔS = 200-240 μm, for grinding of corn ΔS = 100-125 μm, for grinding of wheat ΔS = 50-100 μm.

Along with the obtained kinetic curves of the process under investigation, depending on the amplitude-frequency characteristics of the developed machine, there are also obtained a series of dependencies of the influence of the geometric parameters of the technological filler d upon the value of the specific area S of the crushed material (Fig. 10).

Based on the obtained data, one can conclude that the use of grinding balls with a diameter d = 10-18 mm is most expedient for coarse grinding S = 3000 cm^2·g^{-1}, but, when the degree of grinding is increased to S = 4500 cm^2·g^{-1}, most purposeful it is to use balls with a diameter d = 3-9 mm, which promotes significant increase in the number of their collisions and, as a consequence, leads to active processing of the raw material. Analysis of the dependence of the specific area of the initial material upon the degree of loading of the grinding chamber with the technological filler testified that the highest value of S = 4300-4500 cm^2·g^{-1} is observed at E = 60-75%. However, the filling percentage of the grinding chamber with the technological filler is always connected with increased energy consumption; therefore the choice of optimal operating conditions should be made by means of a compromise solution for the specific area of the initial fraction and the efficiency of the experimental machine.

In order to evaluate the efficiency of the developed machine, a series of experiments were carried out to study the productivity of the developed angular oscillation mill, depending on the kinematic characteristics of the drive shaft and the diameter of the sieve holes, under the condition of previously established operational and design parameters of the developed machine (Fig. 11). When analysing the dependence of the productivity of the vibratory mill of angular oscillations with a varying diameter d_s, it is worthwhile noting the narrow range of its values P = 100-110 kg·h^{-1} at ω = 100-105 rad·s^{-1}. However, a significant increase in the productivity P is observed increasing the angular velocity of the drive shaft ω = 110-125 rad·s^{-1}, which leads to intense transportation of the processed material in relation to the grinding chamber.

Thus, the maximal value of productivity in the operating mode of the developed mill ω = 110 rad·s^{-1} is: for d_s = 0.5 mm P = 150 kg·h^{-1}, for d_s = 1 mm P = 160 kg·h^{-1}, and for d_s = 2 mm P = 168 kg·h^{-1}. Evaluation of the investigated process of grinding of the loose raw material was carried out on the basis of the results of the analysis of the selected samples (Fig. 12).

After processing the results of the obtained data in the statistical environment Statistica 10.0, dependencies of the evaluation criteria upon the investigated factors were obtained in the form of the second-order multiple regression equations. For the vibratory mill of angular oscillations, the regression models will have the following forms, respectively for the specific surface area of the initial raw material S, cm^2·g^{-1} (13) and the energy consumption of the vibrator N, kW·h (14), and for the concerning indicators the following values: the multiple determination coefficient – D = 0.82; the coefficient of multiple correlation – R = 0.90; the criterion Fisher's – F = 5.7. Analysis of the above indicators confirms the high level of adequacy of the regression models.

$$S = -204 - 38.5 \cdot a + 584.5 \cdot d + 55 \cdot E + 1 \cdot a^2 - 80 \cdot d^2 - 0.6 \cdot E^2 + 4.1 \cdot E \cdot d, \qquad (13)$$

$$N = 987.3 - 19 \cdot a + 122 \cdot d - 6.6 \cdot E - 0.3 \cdot a^2 - 8.5 \cdot d^2 + 0.7 \cdot a \cdot d - 0.7 \cdot d \cdot E. \qquad (14)$$

The results of the graphical interpretation of the produced equations are shown respectively in Figure 13 and Figure 14.

By the results of experimental studies and on the basis of the constructed response surfaces of the obtained regression equations efficient parameters of their operation were determined. At the same time, a compromise value was found by the Cramer method in the Mathcad 15 mathematical environment. Thus, at the angular velocity of the drive shaft 110 rad·s^{-1} and the moisture content of the material 8-11%, the rational values of the vibration acceleration of the grinding chamber are 45-50 m s^{-2}, the diameter of the holes in the separation surface is 0.5 mm; the specific area of the material is 5000 $cm^2 \cdot g^{-1}$; the productivity – 220 $kg \cdot h^{-1}$; energy consumption – 0.75 kW·h, with the specific energy consumption being equal to 0.003 $kW \cdot h \cdot kg^{-1}$.

4. Conclusions

A mathematical model of a vibratory mill has been developed as a foundation of the theory of angular oscillations, which made it possible to obtain a functional dependence of the movement of its grinding chambers and to establish that the maximal dynamic state of the technological filler and the surface treatment is ensured at an angular velocity of the drive shaft of 120 $rad \cdot s^{-1}$ and the angle of the vibrodrive placement of 290 degrees. With these parameters the ratio of the amplitude components of oscillations is 2.5 times.

The conclusions of theoretical studies of the process of fine grinding bulk raw materials using a vibratory mill of angular vibrations have been confirmed by experiments. It has been established that at 120 $rad \cdot s^{-1}$ and the angle of placement of the vibration drive equal to 290 degrees, the limits of the rational operating parameters of the vibratory machine will have the following values: vibro-acceleration 45-50 $m \cdot s^{-2}$ the vibration intensity 16 $m^2 \cdot s^{-3}$ the grinding chambers 50-60% with a diameter of the crushing balls 3-5 mm. It has also been established that these technical solutions will make it possible to produce a material with a specific surface of 5000 $cm^2 \cdot g^{-1}$ at a productivity of 220 $kg \cdot h^{-1}$ and specific energy consumption of 0.003 $kW \cdot h \cdot kg^{-1}$.

Acknowledgements

This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors. The Authors equally contributed to the present study.

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Figure captions

Figure 1. General view of the device: 1 – microport for attaching the sensor of the accelerometer; 2 – power battery; 3 – memory card; 4 – power button; 5 – adaptive microport for reading data; 6 – accelerometer housing; 7 – accelerometer

Figure 2. General view of the vibratory mill of angular oscillations: 1 – electric motor; 2 – flexible coupling; 3 – shaft; 4 – unbalances; 5 – central axis; 6 – grinding chamber; 7 – traverses (crosspieces); 8 – feed pipes; 9 – hopper; 10 – discharge pipe; 11 – bearing supports; 12, 13 – bearing units assemblies; 14 – frame

Figure 3. Equivalent scheme of the vibratory mill of angular oscillations

Figure 4. Kinetic characteristics of the grinding chamber of the mill, depending on the angle of placement of the vibratory drive α

Figure 5. Dependence of the amplitude of oscillations A on the angle of placement of the vibrator activator α: a) A_x, horizontal component of the oscillation amplitude; b) A_z, vertical component of the oscillation amplitude; c) A, ratio of the aforesaid components of the amplitude

Figure 6. a) Total amplitude of oscillations A and b) vibration acceleration α of the machine as function of the angular velocity ω of the drive shaft and the loading of the technological filler E: 1) without a technological filler, E=0%; 2) at E=50%; 3) at E=75%;

Figure 7. Dependence of energy consumption N of the mill of angular oscillations upon the angular velocity ω and loading of the technological filler E: 1) without a technological filler, E=0%; 2) at E=50%; 3) at E=75%

Figure 8. Dependence of the specific area of the wheat particles upon the angular velocity of the drive shaft ω and time t being in the grinding chamber of the mill: 1) ω = 90 rad·s^{-1}; 2) ω = 100 rad·s^{-1}; 3) ω = 110 rad·s^{-1}

Figure 9. Dependence of the particle size of the processed material upon the time of its being in the grinding chamber at ω = 110 rad·s^{-1}: 1) grinding of peas; 2) grinding of corn; 3) grinding of wheat

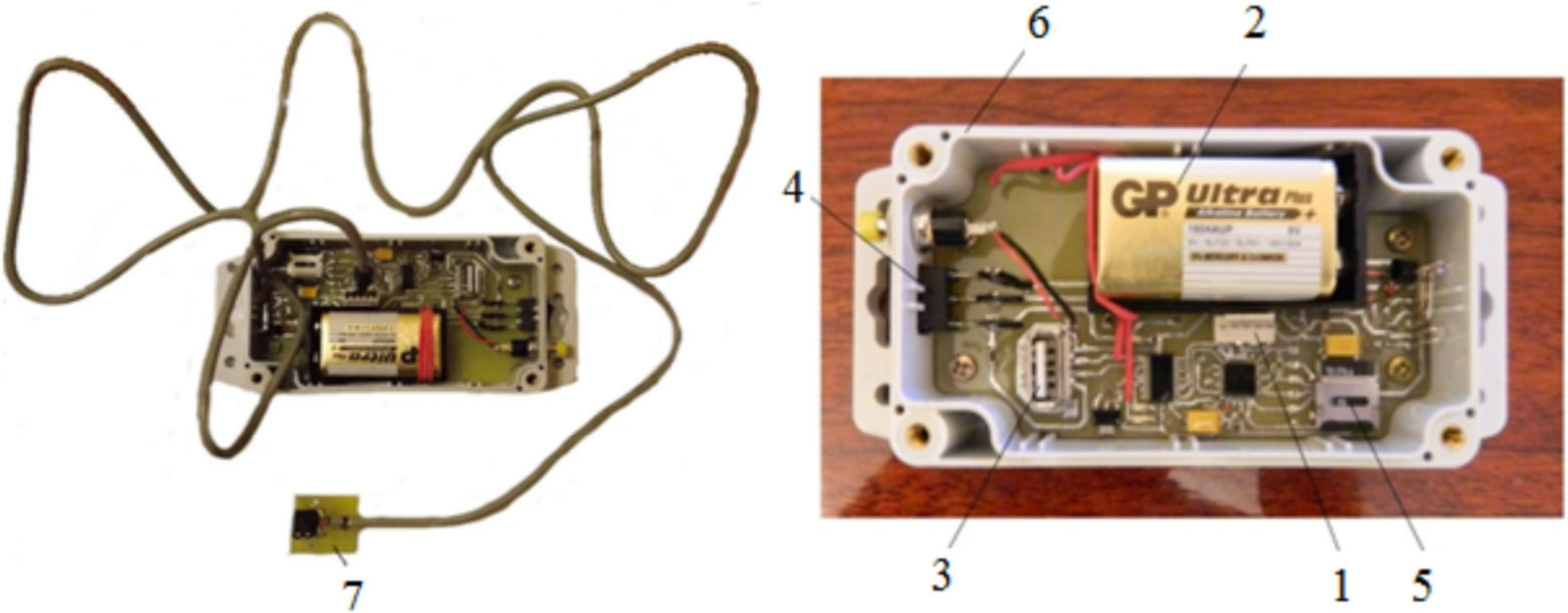
Figure 10. Dependence of the specific area of the wheat particles processed in the mill of angular vibrations depending on time t and diameter d of the crushing balls: 1) 50 s; 2) 40 s; 3) 30 s; 4) 20 s; 5) 10 s

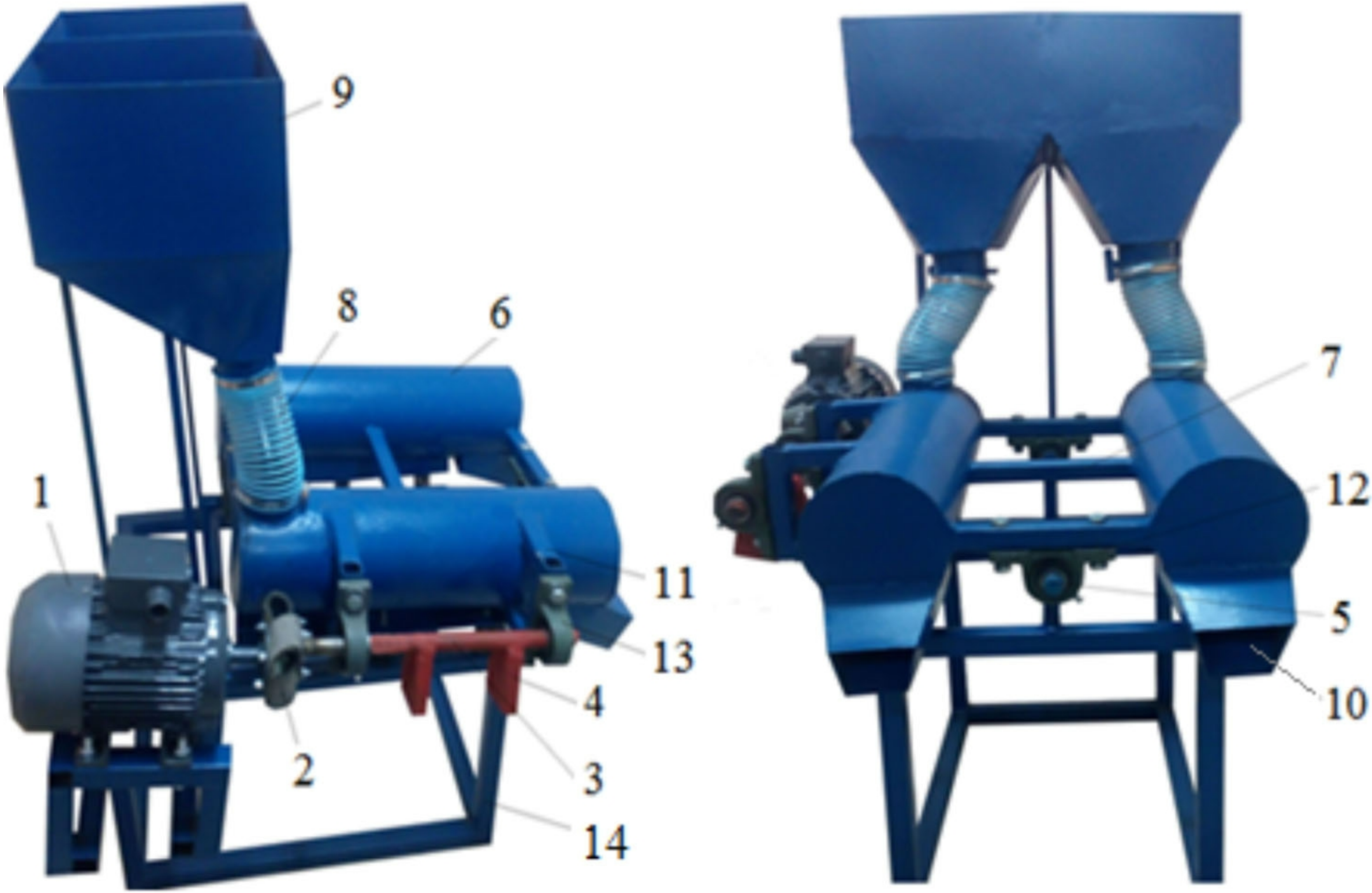
Figure 11. Dependence of the output of the angular oscillation mill upon the angular velocity of the drive shaft ω and the diameter of the screen holes d_s: 1) d_s= 0.5 mm; 2) d_s= 1.0 mm; 3) d_s= 2.0 mm

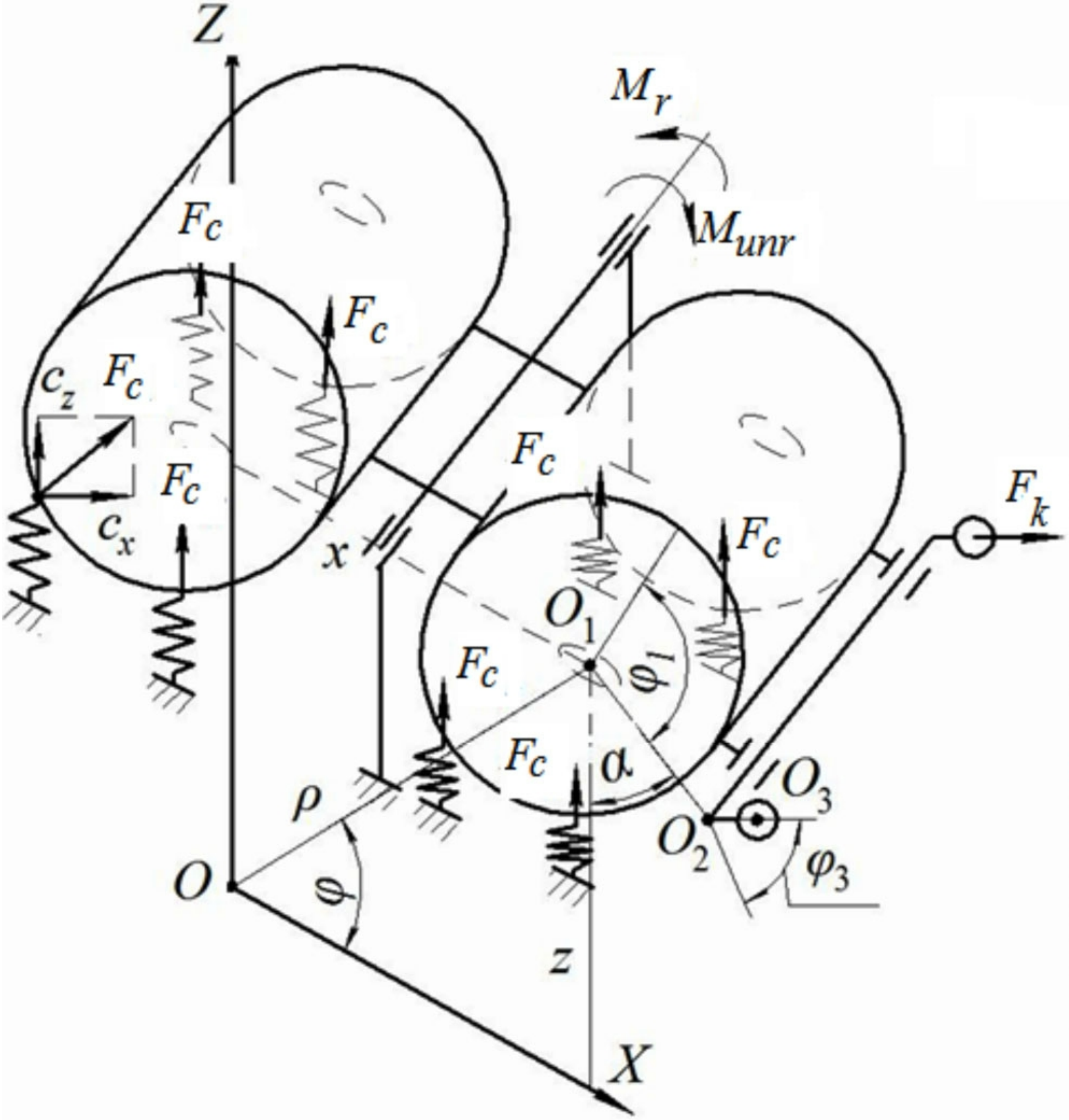
Figure 12. The obtained samples of the raw material before and after treatment: a) corn; b) peas; c) rye; d) wheat

Figure 13. Specific area of the material S as a result of its processing in the vibratory mill of angular oscillations depending on: a) degree of loading with the technological filler E and vibration acceleration α; b) degree of loading with the technological filler E and the diameter of the crushing bodies d

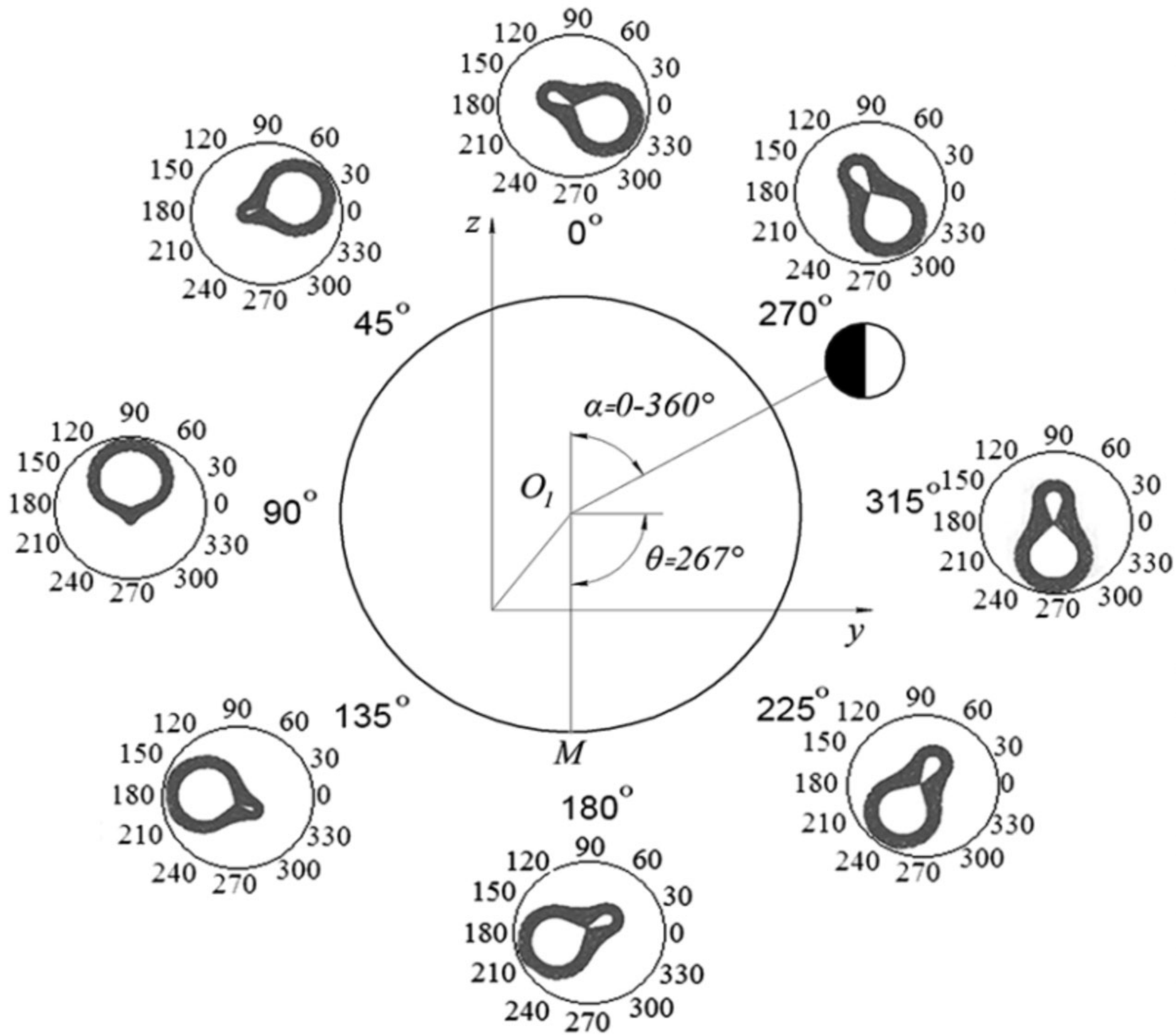
Figure 14. Energy consumption of the vibratory mill of angular oscillations depending on: a) diameter of the crushing bodies d and vibration acceleration α; b) degree of loading with the technological filler E and the diameter of the crushing bodies d

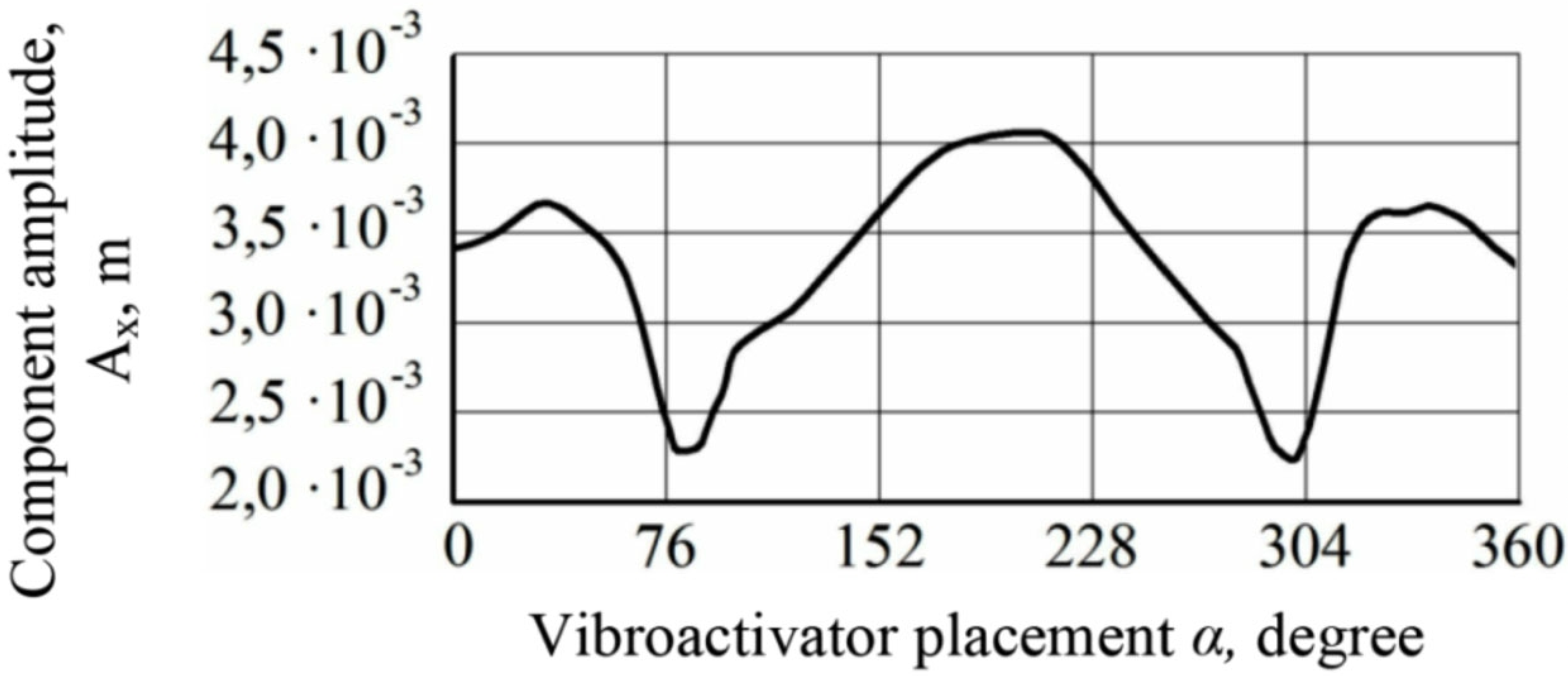

6
2
4
GP Ultra Plus
3
1
5
7


9
8
6
1
11
13
4
2
3
14
7
12
5
10

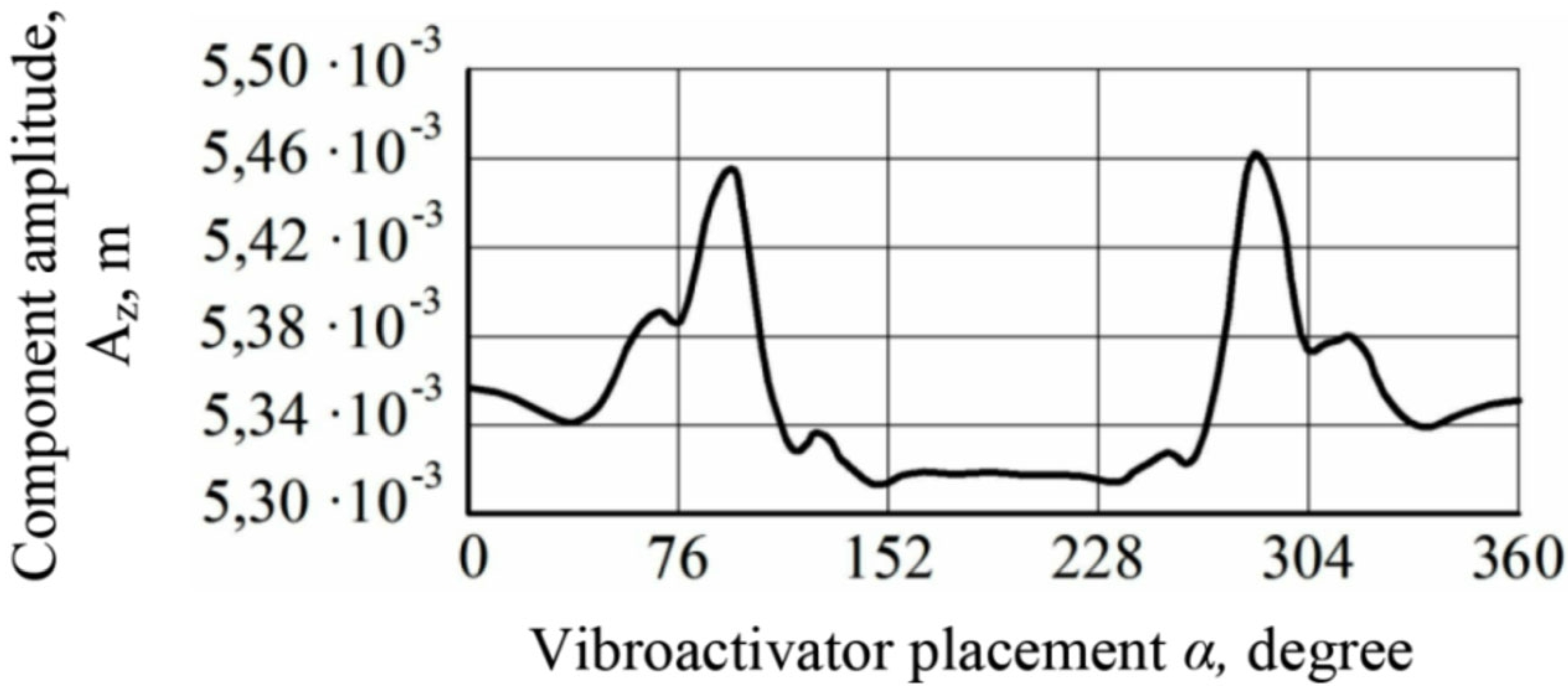


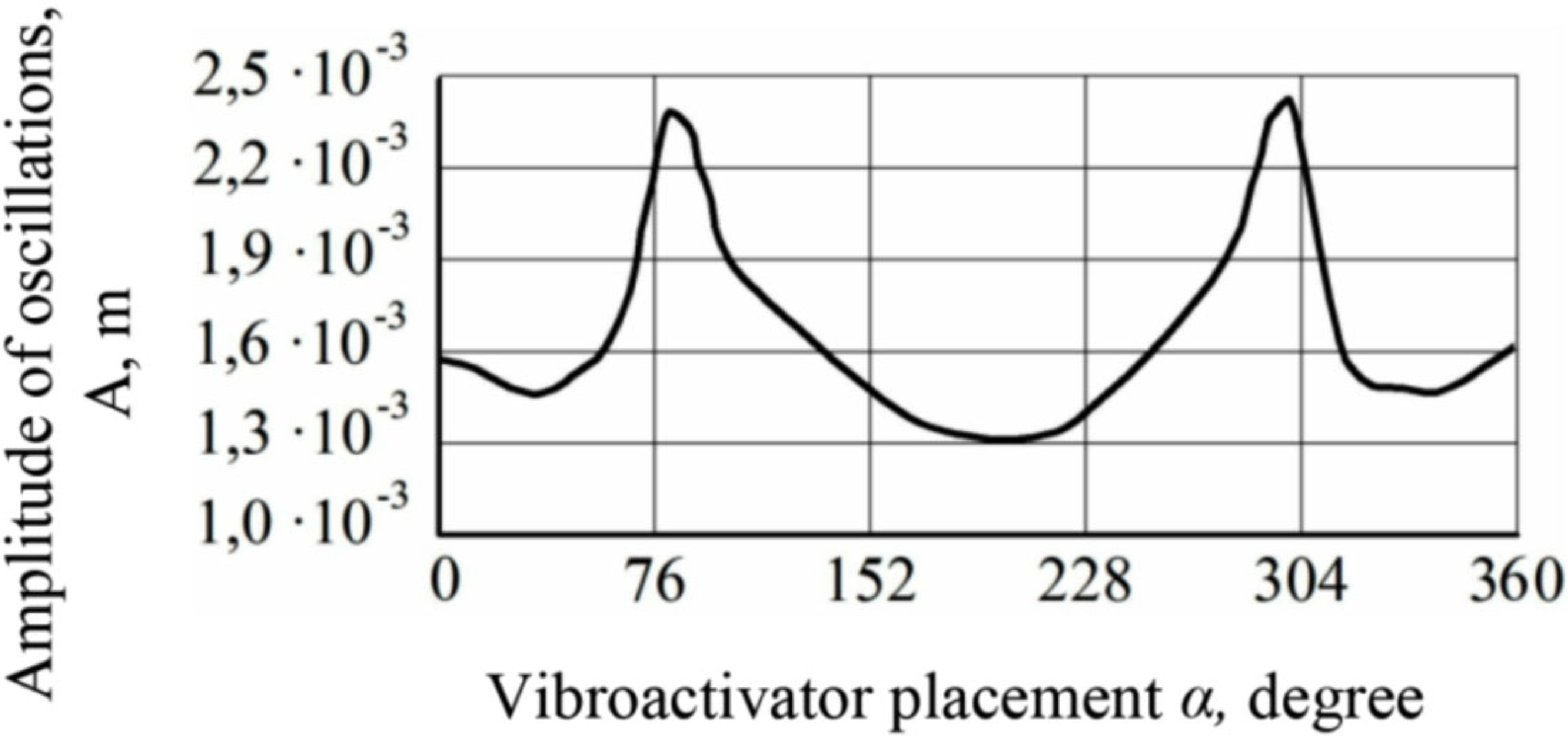
Z
M_r
M_{unr}
F_c
F_c
c_z
F_c
F_c
c_x
F_c
F_k
x
F_c
O_1
φ_1
F_c
F_c
α
O_3
ρ
O_2
φ_3
O
φ
z
X


0°
45°
90°
135°
180°
225°
270°
315°
z
y
O1
M
α=0-360°
θ=267°

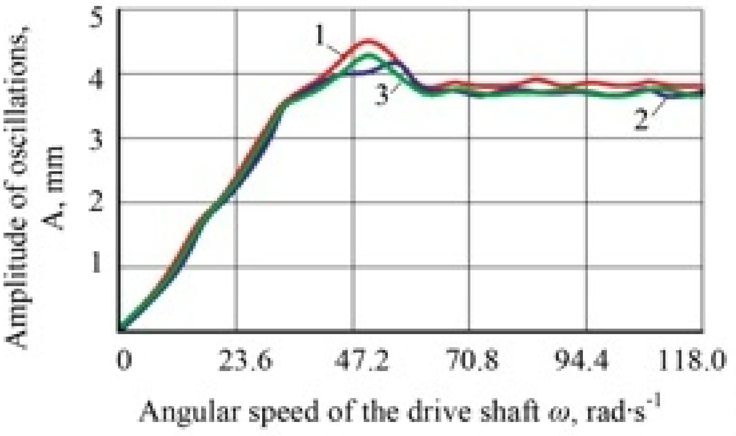


Component amplitude, A_X, m
4,5 ·10^-3
4,0 ·10^-3
3,5 ·10^-3
3,0 ·10^-3
2,5 ·10^-3
2,0 ·10^-3
0
76
152
228
304
360
Vibroactivator placement α, degree

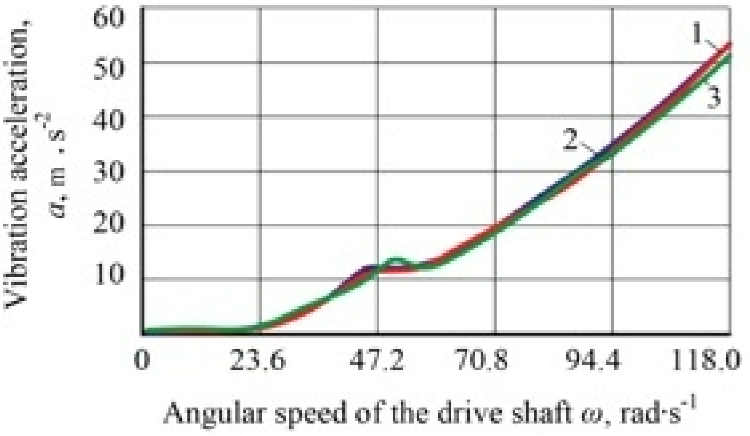

Component amplitude,
A_z, m
$5{,}50 \cdot 10^{-3}$
$5{,}46 \cdot 10^{-3}$
$5{,}42 \cdot 10^{-3}$
$5{,}38 \cdot 10^{-3}$
$5{,}34 \cdot 10^{-3}$
$5{,}30 \cdot 10^{-3}$
0
76
152
228
304
360
Vibroactivator placement α, degree



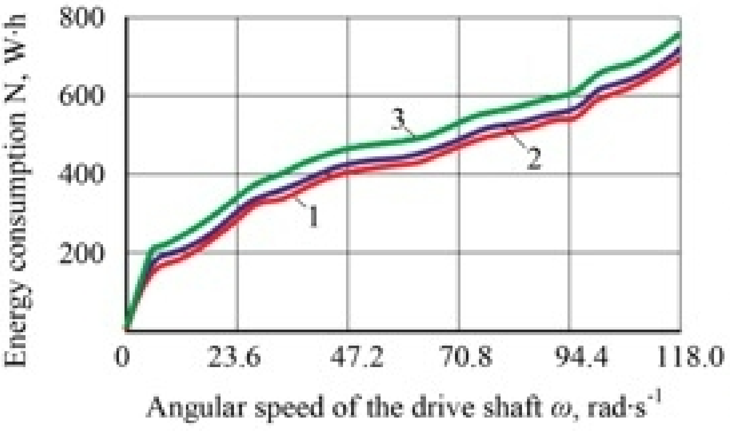
Amplitude of oscillations,
A, m
2,5 ·10-3
2,2 ·10-3
1,9 ·10-3
1,6 ·10-3
1,3 ·10-3
1,0 ·10-3
0
76
152
228
304
360
Vibroactivator placement α, degree



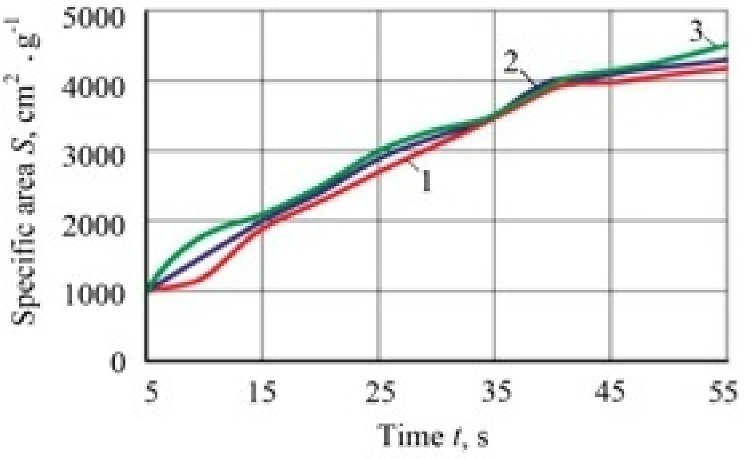
Amplitude of oscillations, A, mm
5
4
3
2
1
0
23.6
47.2
70.8
94.4
118.0
1
2
3
Angular speed of the drive shaft ω, rad·s-1



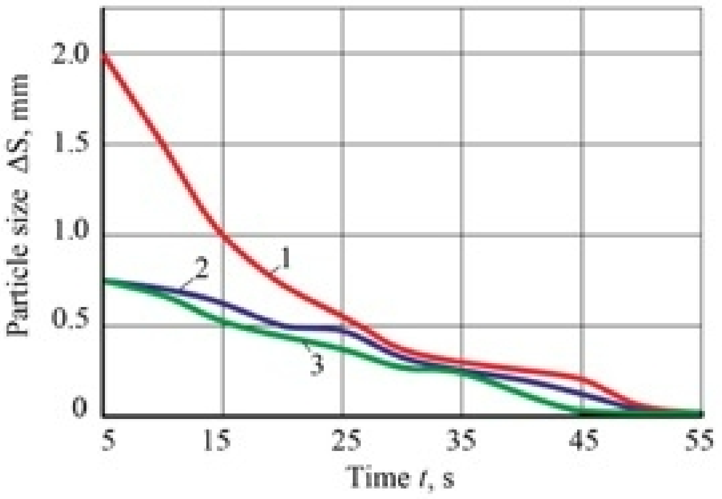
Vibration acceleration,
a, m · s⁻²
60
50
40
30
20
10
0
23.6
47.2
70.8
94.4
118.0
Angular speed of the drive shaft ω, rad·s⁻¹
1
2
3

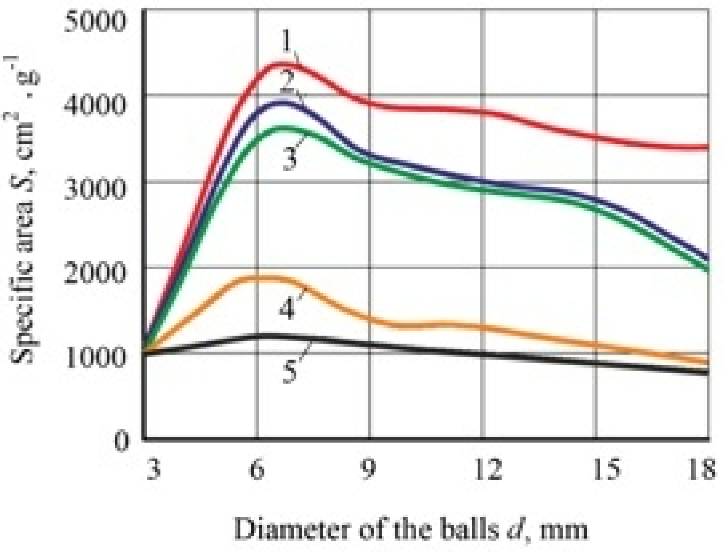


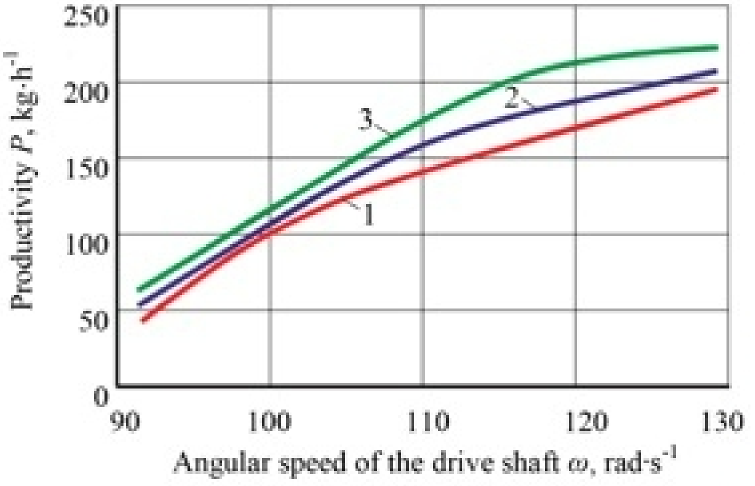
Energy consumption N, W·h
800
600
400
200
3
2
1
0
23.6
47.2
70.8
94.4
118.0
Angular speed of the drive shaft ω, rad·s-1



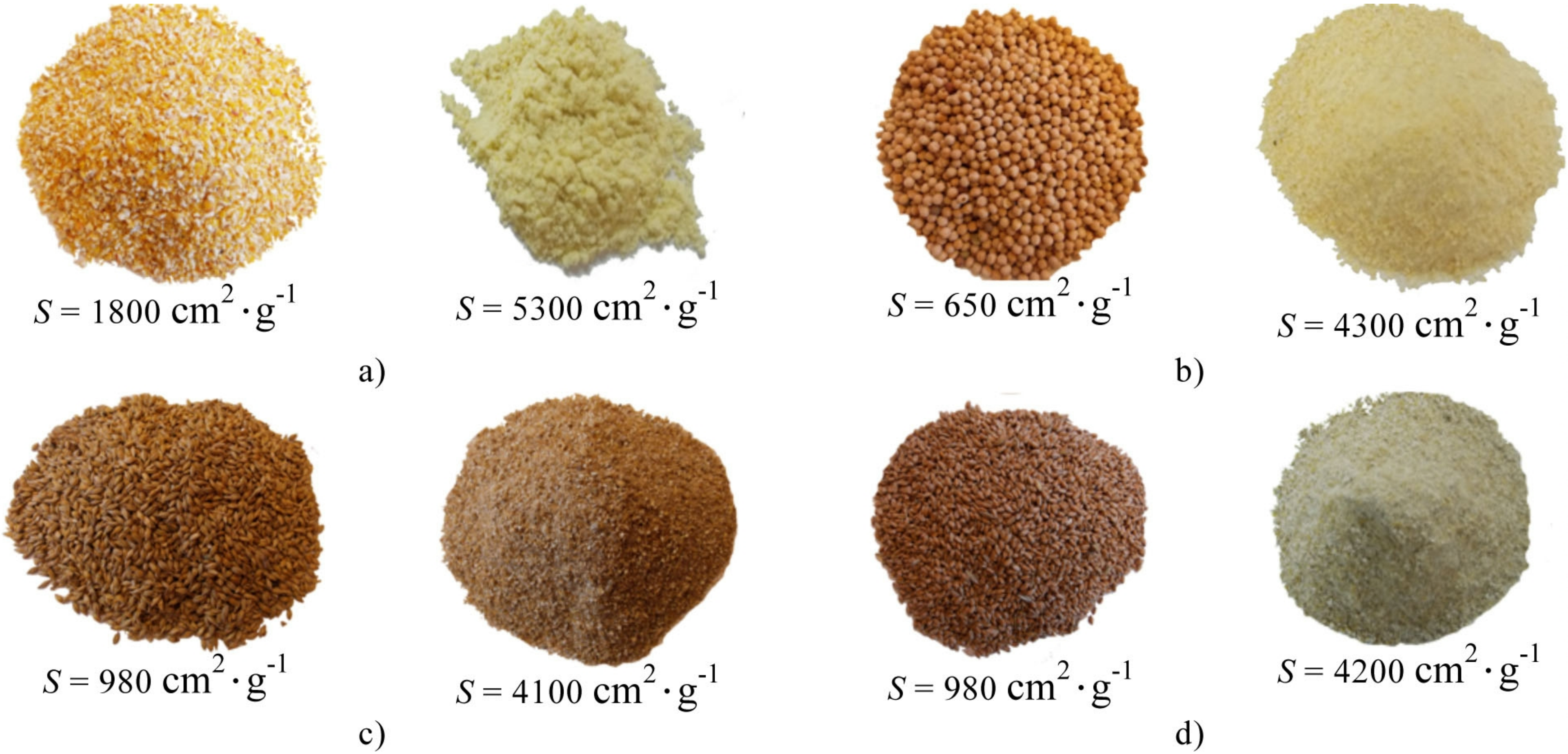
Specific area S, cm² · g⁻¹
5000
4000
3000
2000
1000
0
5
15
25
35
45
55
Time t, s
1
2
3


Particle size ΔS, mm
2.0
1.5
1.0
0.5
0
1
2
3
5
15
25
35
45
55
Time t, s

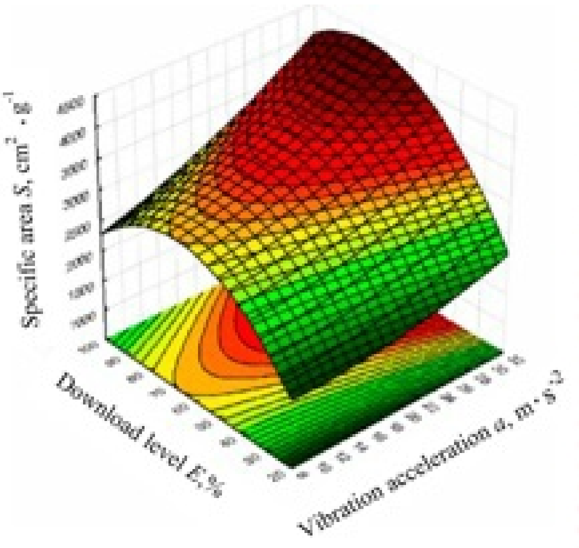

5000
4000
3000
2000
1000
0
Specific area S, cm², g⁻¹
1
2
3
4
5
3
6
9
12
15
18
Diameter of the balls d, mm



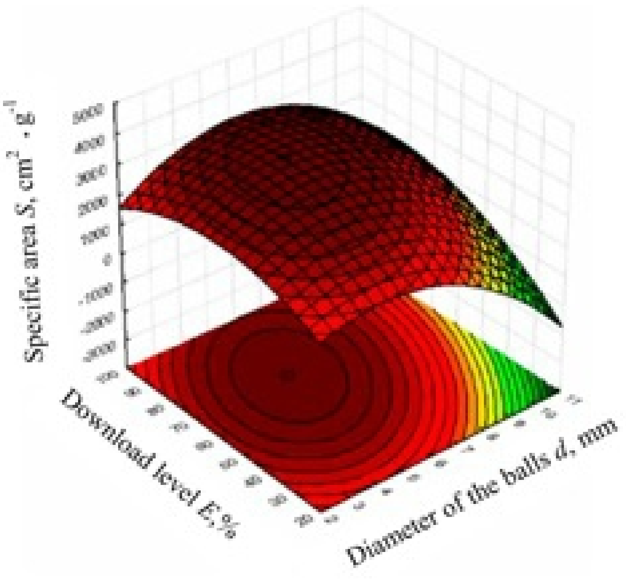
250
200
150
100
50
0
90
100
110
120
130
Productivity P, kg·h-1
Angular speed of the drive shaft ω, rad·s-1
1
2
3



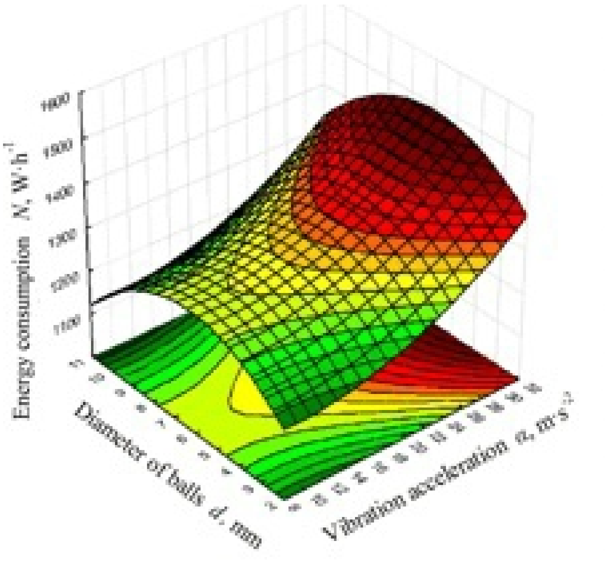
S = 1800 $cm^2 \cdot g^{-1}$
S = 5300 $cm^2 \cdot g^{-1}$
a)
S = 650 $cm^2 \cdot g^{-1}$
S = 4300 $cm^2 \cdot g^{-1}$
b)
S = 980 $cm^2 \cdot g^{-1}$
S = 4100 $cm^2 \cdot g^{-1}$
c)
S = 980 $cm^2 \cdot g^{-1}$
S = 4200 $cm^2 \cdot g^{-1}$
d)



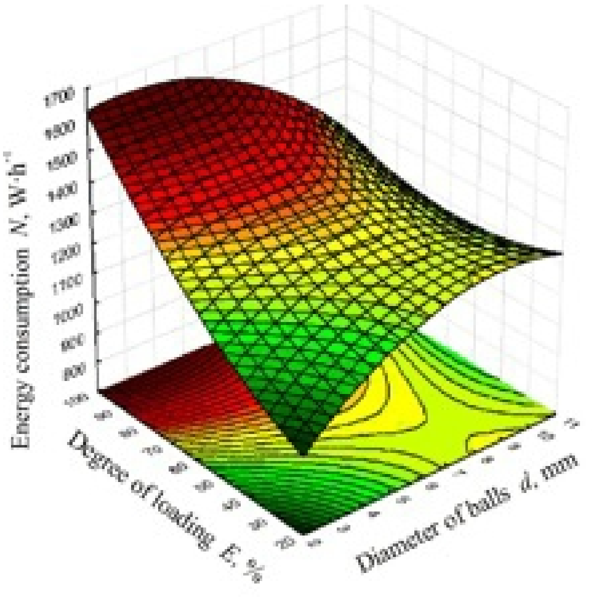
Specific area S, cm^2 · g^{-1}
Download level E,%
Vibration acceleration a, m · s^{-2}



Specific area S, cm^2 · g^{-1}
Download level E, %
Diameter of the balls d, mm



Energy consumption N, W·h⁻¹
Diameter of balls d, mm
Vibration acceleration a, m·s⁻²



Energy consumption N, W·h⁻¹
Degree of loading E, %
Diameter of balls d, mm