RESEARCH ON RESONANT OSCILLATIONS OF THE TELESCOPIC SCREW – GRANULAR MEDIA SYSTEM CAUSED BY EXTERNAL PERIODIC FORCES

ДОСЛІДЖЕННЯ РЕЗОНАНСНИХ КОЛИВАНЬ СИСТЕМИ ТЕЛЕСКОПІЧНИЙ ГВИНТ – СИПКЕ СЕРЕДОВИЩЕ ЗУМОВЛЕНИХ ЗОВНІШНІМИ ПЕРІОДИЧНИМИ СИЛАМИ

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ABSTRACT
The article presents the results of resonant oscillations theoretical research of the telescopic screw - granular material system caused by external periodic forces. In the theoretical part a mathematical model of bending vibrations of an elastic body is described, which rotates along a fixed axis with a constant angular velocity, provided that it moves with a constant relative linear velocity along the elastic body. In the experimental part, according to the obtained equations, it has been established for the various kinematic characteristics of the system, a telescopic screw-granular media of amplitude passing through resonance at the frequency of external periodic perturbation.The results of experimental research and recommendations for the selection of constructive kinematic for telescopic screw conveyor have been presented.

РЕЗЮМЕ
У статті наведено теоретичні дослідження резонансних коливань системи телескопічний гвинт – сипкий матеріал зумовлених зовнішніми періодичними силами. В теоретичній частині описано математичну модель згинальних коливань пружного тіла, яке обертається вздовж нерухомої осі із сталою кутовою швидкістю за умови, що вздовж нього рухається зі сталою відносною лінійною швидкістю. В експериментальній частині відповідно до отриманих рівнянь, представлено для різних кінематичних характеристик системи телескопічний гвинт – сипке середовище амплітуду проходження через резонанс на частоті зовнішнього періодичного збурення. Представлені результати експериментальних досліджень та наведені рекомендації для вибору конструктивно-кінематичних телескопічних гвинтових транспортерів.

INTRODUCTION

One of the premises for the high efficiency of machine-building enterprises is to improve the existing and to introduce new competitive products, which can fully meet the needs of consumers. Screw transport mechanisms are one of the most commonly used mechanical means in agricultural production and other sectors of the economy as individual elements, as well as in other machines. According to various data, their specific gravity in loading and unloading operations of different types is 40-45%.

They are especially widely used as elements of agricultural machines when overloading agricultural loads in the field conditions. Often, the universal units for loading seeders, hoppers, reloaders and combine harvesters are complicated (the term is used for composition and decomposition) in order to obtain a significant material overload path. To achieve the required overload distance, they are decomposed - composed with the use of hydraulic or pneumatic equipment. This makes their constructions too complicated and expensive. The examples are the universal hoppers-reloaders of the companies “EGRITECH”, “Liliani MBA”, combine harvesters CASE 8120 AXIAL FLOW, New Holland CR10.90 and others. Therefore, using the telescope principle will have widespread use in the design of agricultural machinery equipped with screw conveyors.

Despite a large number of scientific works devoted to the development and research of the features of screw conveyors operation, there is a wide range of unexplored issues related to their constructive and

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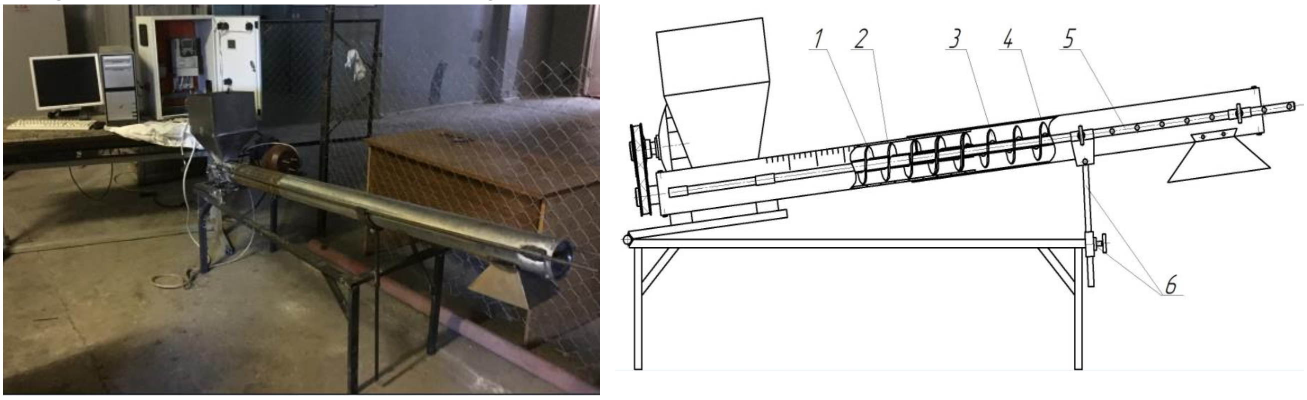
functional characteristics. Therefore, the development of telescopic screw conveyors makes it possible to improve the efficiency of the overload of loose materials. However, due to the significant angular velocities of the screw rotation in telescopic screw conveyors, the asymmetry of the telescopic screw and external perturbations, fluctuations often occur, which result in significant dynamic loads in the screw *(Hevko I.B., 2013; Lyashuk O.L., et al., 2016)*, especially in resonant cases. The works of *Lyashuk O.L. et al. (2018)* show the movement of the grain mixture along the working body of the loader. Differential equations are obtained that describe the bending fluctuations of the horizontal screw of the mixer loader.

The results of theoretical and experimental studies of simultaneous transportation and mixing of feed mixtures components on the curved lines of tubular conveyors are presented. A mathematical model is constructed that characterizes the dependence of the change of elementary work, which is performed when the elemental mass of loose material moves along the curvilinear region *(Hevko R.B., et al., 2018)*. The method of carrying out researches on determining the power indicators of different movement types of loose materials is presented. Therefore, the study of dynamic loads in the screw of a telescopic conveyor during the movement of loose material enables to choose the rational modes of operation of the conveyor, which minimize these negative phenomena and increase the life of exploitation of telescopic screw conveyor.

MATERIALS AND METHODS

To study the principle of telescopes in screw conveyors based on a patent search and analysis of scientific literature and synthesis (*Rogatynsky R.M., et al., 2014*), a pilot installation has been developed, designed, manufactured and shown in Figures 1 - 3.



a) *b)*

Fig. 1 - Stand to study the characteristics of telescopic screw conveyors: a) general view; b) structural scheme

1) screw axial motion in the axial direction of the screw section; 2) part of the casing is fixed in the axial direction; 3) screw moving in the axial direction of the screw section; 4) part of the casing moving in the axial direction; 5) guides; 6) support for adjusting the height of the material

The outer diameter of the auger in the pilot installation is 97 mm, internal diameter of the fixed pipe is 100 mm, the outer one is 107 mm, the inner diameter of the movable branch pipe is 109 mm. The movable nozzle is made of galvanized sheet, and therefore it contains a connecting seam and ovals and has uneven shapes along the entire length, which affect the speed of twisting and unwinding of the telescopic part of the screw conveyor.

Studies have shown that the biggest problem in telescopic screw conveyors is to maintain the same clearance between the casing and the spiral in different sections of the telescope, which significantly affects the time of rolling in and rolling out of the axially moving part of the auger fixed and the appearance and magnitude of the rotational bending oscillations. It has also been found that the overload performance of agricultural goods by telescopic screw conveyor does not differ from the overload performance of these materials with traditional screw conveyors.

It is established by *Sokil B.I. et al. (2010; 2011; 2012; 2017)* that there is an effect of the motion of a continuous stream of granular media on the longitudinal or bending vibrations of elastic bodies. Consequently, it can be assumed that even the movement velocity of a granular media has changed the main dynamic characteristics of bending or longitudinal oscillations, and the magnitude of the action of this media increases significantly with an increase in the relative amount of its motion.

Therefore, at a significant angular velocity of rotation of telescopic screws, even small transverse deformations at a certain moment result in significant tension.

Fig. 2 - Scale of the stand screw to study the characteristics of telescopic screw conveyors

Fig. 3 - A stand for the study of telescopic screw conveyors in a disassembled state

In study of *Fedoseev V.I. (1951)*, a mathematical model of flexural oscillations of an elastic body rotating along a fixed axis with a constant angular velocity Ω was presented. This action was provided when a continuous flow of a homogeneous medium moves along it at a constant relative linear velocity V. The proposed model of the dynamic process was presented in the form of zero rigidity, where a system of differential equations is used.

$$\begin{aligned}&(\rho_1+\rho_2)\frac{\partial^2 u}{\partial t^2}+2\rho_2 V\frac{\partial^2 u}{\partial t\partial z}-2(\rho_1+\rho_2)\Omega\frac{\partial w}{\partial t}+\rho_2 V^2\frac{\partial^2 u}{\partial z^2}-\\&-2(\rho_1+\rho_2)I\Omega\frac{\partial^3 w}{\partial t\partial x^2}+EI\frac{\partial^4 u}{\partial z^4}-(\rho_1+\rho_2)\Omega^2 u==\varepsilon f\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},\ldots,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)\\&(\rho_1+\rho_2)\frac{\partial^2 w}{\partial t^2}+2\rho_2 V\frac{\partial^2 w}{\partial t\partial z}+2(\rho_1+\rho_2)\Omega\frac{\partial u}{\partial t}+\rho_2 V^2\frac{\partial^2 w}{\partial z^2}+\\&+2(\rho_1+\rho_2)I\Omega\frac{\partial^3 u}{\partial t\partial x^2}+EI\frac{\partial^4 w}{\partial z^4}-(\rho_1+\rho_2)\Omega^2 w=\varepsilon g\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},\ldots,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)\end{aligned} \tag{1}$$

In Eq.(1) $u(t,z), w(t,z)$ - projection of the vector of moving the point of the central axis with the coordinate z of the telescopic screw at an arbitrary time t in the projections on the axis of the fixed coordinate system $OXYZ$. The axis OZ of the reference system coincides with the undrained straight line of the auger screw Ω, - the angular velocity of the screw rotation around the axis, ρ_1, ρ_2 - respectively the mass of the unit of body length and moving medium, EI - its rigidity to the screw's bend, $f\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},\ldots,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)$ and $g\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},\ldots,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)$ - 2π - periodic by $\gamma=\nu t+\gamma_0$ functions that describe the nonlinear components of the restoring force, the strength of the resistance, and other forces the maximum values of which are significantly lower than the value of the restoring force, as indicated by the small parameter ε. Below, for simplicity, we assume that these functions are polynomials in the set of variables, and from the physical content of them it results that they must be bound by the relation

$$f\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},\ldots,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)=g\left(w,u,\frac{\partial w}{\partial t},\frac{\partial u}{\partial t},\frac{\partial w}{\partial z},\frac{\partial u}{\partial z},\ldots,\frac{\partial^3 w}{\partial z^3},\frac{\partial^3 u}{\partial z^3},\gamma\right)$$

In the case of complex oscillations of the screw (a combination of bending and twisting), provided that the latter are described by a known law $\vartheta(z,t)$ (the torsion below is considered to be the simplest of their mathematical models), the system of equations (1) is transformed into a form:

$$(\rho_1+\rho_2)\frac{\partial^2 u}{\partial t^2}+2\rho_2 V\frac{\partial^2 u}{\partial t\partial z}-2(\rho_1+\rho_2)\left(\Omega+\frac{\partial \vartheta(z,t)}{\partial t}\right)\frac{\partial w}{\partial t}+\rho_2 V^2\frac{\partial^2 u}{\partial z^2}-$$

$$-2(\rho_1+\rho_2)I\left(\Omega+\frac{\partial \vartheta(z,t)}{\partial t}\right)\frac{\partial^3 w}{\partial t\partial x^2}+EI\frac{\partial^4 u}{\partial z^4}-(\rho_1+\rho_2)\left(\Omega+\frac{\partial \vartheta(z,t)}{\partial t}\right)^2 u-(\rho_1+\rho_2)\frac{\partial^2 \vartheta(z,t)}{\partial t^2}w=$$

$$=\varepsilon f_1\left(u,w,\frac{\partial u}{\partial t},...,\frac{\partial^3 w}{\partial z^3},\gamma\right)$$

$$(\rho_1+\rho_2)\frac{\partial^2 w}{\partial t^2}+2\rho_2 V\frac{\partial^2 w}{\partial t\partial z}+2(\rho_1+\rho_2)\left(\Omega+\frac{\partial \vartheta(z,t)}{\partial t}\right)\frac{\partial u}{\partial t}+\rho_2 V^2\frac{\partial^2 w}{\partial z^2}+$$

$$+(\rho_1+\rho_2)\frac{\partial^2 \vartheta(z,t)}{\partial t^2}u=\varepsilon f_2\left(u,w,\frac{\partial u}{\partial t},...,\frac{\partial^3 w}{\partial z^3},\gamma\right) \quad (2)$$

From the form of the right-hand side of the differential equation (2) it results that in the system under consideration, resonant phenomena are possible due to external periodic forces or torsional oscillations of the telescopic screw itself. The second case will be called internal resonance, and - the first - external. Simpler resonant oscillations have been caused by external periodic forces, so let's consider them first.

The condition for the existence of a given type of resonance is the existence of a rational connection between the frequencies of the internal oscillations of the telescopic screw system and the granular media and the frequency of the external periodic perturbation, i.e. $m\nu \neq n\omega_k$

The dispersion ratio determines the intrinsic frequency of bending vibrations of the body as a function of the angular and linear velocity of the medium along the elastic body in the form:

$$\omega_k=\Omega(I\kappa_k-1)\pm\kappa_k\sqrt{\Omega^2 I\left(\kappa_k^2 I-2\right)-\frac{\rho_2 V^2-EI\kappa_k^2}{\rho_1+\rho_2}} \quad (3)$$

Simultaneously, the obtained ratio serves as the basis for solving a more complex problem - the determination of the influence of nonlinear forces on the dynamic process, as well as the whole set of external and internal factors on bending fluctuations of a screw. Dependence (3) indicates at the same time the simplest way to avoid external resonance: for the given magnitude of the external periodic perturbation, the rotation frequency of the screw propeller should be chosen from the condition:

$$m\left[\Omega(I\kappa_k-1)\pm\kappa_k\sqrt{\Omega^2 I\left(\kappa_k^2 I-2\right)-\frac{\rho_2 V^2-EI\kappa_k^2}{\rho_1+\rho_2}}\right]\neq m\nu .$$

The ratios that determine the laws of amplitude variation and frequency of the wave process are as follows:

$$\frac{da}{dt}=-\frac{1}{2\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\tilde f(a,z,\psi,\gamma,\vartheta)(\cos(\kappa z+\psi)-\cos(\kappa z-\psi))d\psi dz,$$

$$\frac{d\theta}{dt}=\omega+\frac{1}{2a\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\tilde f(a,z,\psi,\gamma,\vartheta)(\sin(\kappa z+\psi)+\sin(\kappa z-\psi))d\psi dz,. \quad (4)$$

In the system of differential equations (4), the integral functions, and therefore the right parts of it are periodic in terms of arguments ψ,γ,ϑ, which means that non-resonant and resonant oscillations may occur in the screw auger. As for the former, they have in the conditions $\omega\neq\nu$ and $\omega\neq\Theta$ ratios between which there is no rational connection between their own frequency ω and frequency of external perturbation ν or frequency of torsional oscillations Θ. For the first approximation of non-resonant oscillations, the amplitude and frequency of flexural oscillations of a telescopic screw have been described by dependencies:

$$\frac{da}{dt} = \frac{-\varepsilon}{8\omega\pi^2 l}\int_0^l\int_0^{2\pi}\int_0^{2\pi}\int_0^{2\pi} \tilde{f}(a,z,\psi,\gamma,\vartheta)\sin\frac{\pi}{l}z\cos\psi dz d\psi d\gamma d\vartheta,$$
$$\frac{d\psi_1}{dt} = \omega - \frac{\varepsilon}{8\omega\pi^2 al}\int_0^l\int_0^{2\pi}\int_0^{2\pi}\int_0^{2\pi} \tilde{f}(a,z,\psi,\gamma,\vartheta)\sin\frac{\pi}{l}z\sin\psi dz d\psi d\gamma d\vartheta \tag{5}$$

RESULTS

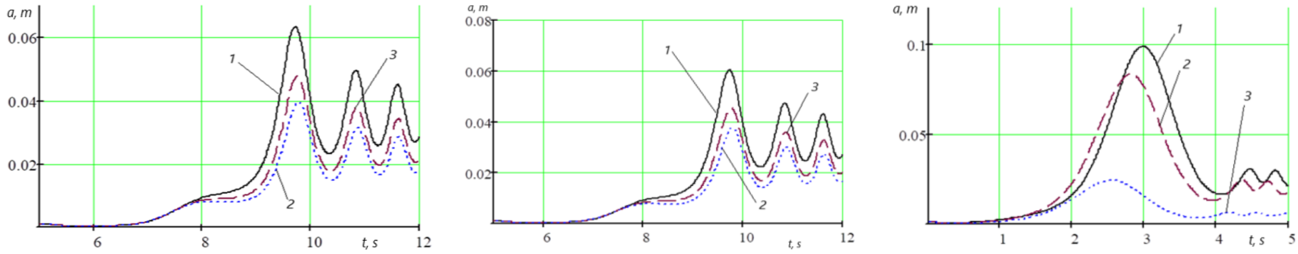
To study the process of resonance passage for the base, we adopt the system of differential equations (5) and the property of the resonant oscillations, which is as follows: the amplitude of the resonance passage essentially depends on the phase difference between the proper and forced oscillations. For the case under consideration this is $\phi = \psi - \gamma$ (here and below for simplicity only the case of the main resonance is considered). Entering the given parameter in the system of differential equations (4), after averaging over the phases of forced and torsional oscillations we obtain:

$$\frac{da}{dt} = -\frac{1}{8\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\int_0^{2\pi} \tilde{f}(a,z,\phi+\gamma,\gamma,\vartheta)(\cos(\kappa z+\phi+\gamma)-$$
$$-\cos(\kappa z-\phi+\gamma))d\gamma d\vartheta dz,$$
$$\frac{d\gamma}{dt} = \omega - \nu + \frac{1}{8a\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\int_0^{2\pi} \tilde{f}(a,z,\phi+\gamma,\gamma,\vartheta)(\sin(\kappa z+\phi+\gamma)+$$
$$+\sin(\kappa z-\phi+\gamma))d\gamma d\vartheta dz. \tag{6}$$

The system of equations (6) for the case of the above nonlinearly elastic law of the telescopic screw material and the viscoelastic forces of resistance and mono-harmonic periodic perturbation is transformed into a form:

$$\frac{da}{dt} = -\frac{k_1(\omega)^{s-1}}{(\rho_1+\rho_2)\pi}a^s + \frac{h}{\pi\omega}\cos\gamma\,; \tag{7}$$
$$\frac{d\gamma}{dt} = \omega - \nu - \frac{\bar{k}_1 EI}{(\rho_1+\rho_2)}a^2 - (\frac{\pi}{l})^2\frac{\rho_2}{8\omega(\rho_1+\rho_2)}V^2 + \frac{h}{\pi a\omega}\sin\gamma$$

Below, in accordance with the obtained equations, a telescopic screw is shown for various kinematic characteristics of the system - the granular media of amplitude passing through resonance at the frequency of external periodic perturbation. Figure 4.



$\rho_1 = 15$ media; 1 - $\rho_2 = 10$; 2 - $\rho_2 = 15$; 3 - $\rho_2 = 20$; velocity 10; ; $\Omega = 15$

a)

$\rho_1 = 15$ media; 1 - $\rho_2 = 10$; 2 - $\rho_2 = 15$; 3 - $\rho_2 = 20$; velocity 10; $\Omega = 20$

b)

at various angular velocities of media rotation $\rho_1 = 15$; $\rho_2 = 10$; the angular velocity is 1 - $\Omega = 10$; 2- $\Omega = 15$; 3 - $\Omega = 20$

c)

Fig. 4 - Changing the amplitude of the telescopic screw system is a granular media at the transition through the resonance due to external periodic perturbation

The obtained dependences show that for: a) the larger values of the particle mass of the media, the amplitude of the transition through the resonance is less; b) for larger values of the angular velocity of the body, the actual frequency of bending vibrations is smaller while the amplitude of the transition through the resonance is smaller; c) the amplitude is less at a higher speed of transition.

According to the research results, the graphical dependences of engine power N on the rotation speed of the working body n, the angle of the conveyor α and the length of transportation are constructed. It is established that the power N has a fairly clearly expressed linear nature of the growth of N with increasing n.

On the basis of the multivariate experiment, a regression dependence has been obtained to determine the effect of n, α and L on the magnitude of the power N during grain transportation of corn, barley and mixed fodder: $300 \le n \le 700$ (rpm); $1{,}33 \le l \le 1{,}61$ (m); $5 \le \gamma \le 45$ (deg.).

- during corn transportation:

$$N_{(n,l,\gamma)} = -2.22\cdot10^{-2} - 4.94\cdot10^{-4}n - 2.76\cdot10^{-2}l + 9.25\cdot10^{-4}\gamma + \\ +9.29\cdot10^{-4}nl + 1.58\cdot10^{-6}n\gamma + 9.25\cdot10^{-7}n^2 + 3.37\cdot10^{-2}l^2 - 1.83\cdot10^{-5}\gamma^2; \quad (8)$$

- during wheat transportation:

$$N_{(n,l,\gamma)} = -2.13\cdot10^{-2} - 5.09\cdot10^{-4}n - 2.85\cdot10^{-2}l + 9.38\cdot10^{-4}\gamma + \\ +9.64\cdot10^{-4}n\,l + 1.65\cdot10^{-6}n\gamma + 9.75\cdot10^{-7}n^2 + 3.52\cdot10^{-2}l^2 - 1.93\cdot10^{-5}\gamma^2; \quad (9)$$

- during mixed fodder transportation:

$$N_{(n,l,\gamma)} = -2.68\cdot10^{-2} - 4.18\cdot10^{-4}n - 2.06\cdot10^{-2}l + 8.25\cdot10^{-4}\gamma + \\ +8.21\cdot10^{-4}n\,l + 1.4\cdot10^{-6}n\gamma + 8.25\cdot10^{-7}n^2 + 3.01\cdot10^{-2}l^2 - 1.65\cdot10^{-5}\gamma^2. \quad (10)$$

Using Statistica-6.0 software for PCs, we constructed a graphical reproduction of common regression models in the form of quadratic response surfaces and their two-dimensional power sections N as a function of two variable factors $x_{i(1,2)}$ with a constant unchangeable level taking into account the third factor $x_{i(3)} = const$.

The analysis of the given regression equations shows that the main factors affecting the increase in drive power are: factors x_1, x_2, (n, l) and combinations of these factors. An increase in the value of factor x_3 (γ) leads to an increase in power by 4.2% (Fig. 5). Moreover, an increase in the value of factor x_2(l) leads to an increase in power by 9.8%.

In general, to reduce power, it is necessary to reduce the speed of the screw and the angle of the conveyor inclination. The graphical values of the power dependence results obtained using Mathcad 2000 Professional based on the analysis of regression equations are shown in Fig. 5.

Figure 5 shows the response surfaces of the change in the values of n from the simultaneous change of two factors: a) $N = f(n,\alpha)$; b) $N = f(n,l)$; c- $N = f(\alpha,l)$.

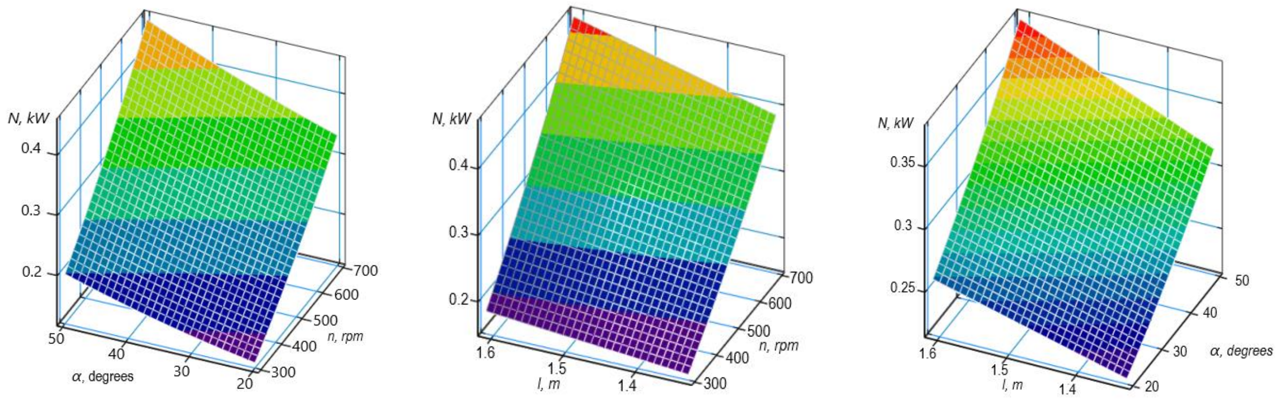


Fig. 5 - Response surfaces of the change in the values of *n* from the simultaneous change of two factors: a - $N = f(n,\alpha)$**; b -** $N = f(n,l)$**; c-** $N = f(\alpha,l)$

It can be seen from the figures that with an increase in the screw speed, length, extension of the screw and the angle of inclination of the screw conveyor, the power on the screw drive increases, and the maximum power of 1.29 kW is achieved when transporting wheat.

The maximum power on the screw drive of the telescopic screw conveyor for transporting corn and mixed fodder is 1.23 kW and 1.10 kW, respectively, and the minimum is 0.31 kW and 0.28 kW, respectively. Increase in screw speed nш, from 300 rpm. to 700 rpm leads to an increase in power on the auger drive by 3.14 times.

Studies have shown that the biggest problem in telescopic screw conveyors is to maintain the same clearance between the casing and the spiral in different sections of the telescope, which significantly affects the time of rolling in and rolling out of the axially moving part of the auger fixed and the appearance and magnitude of the rotational bending oscillations.

CONCLUSIONS

In order to improve the performance of screw conveyors, which are to ensure the transport of loose materials, increase the load capacity, as well as improve their serviceability, the telescopic screw conveyor is developed.

The theoretical calculations of the resonance phenomenon are caused by external periodic forces or torsional vibrations of the telescopic screw itself. The differential equations of the bending oscillations of an elastic body rotating along a fixed axis with a constant angular velocity Ω are deduced, provided that a continuous flow of a homogeneous capacity is used along it with a constant relative linear velocity V .

A telescopic screw- granular media of the amplitude passing through resonance at the frequency of external periodic perturbation is presented for various kinematic characteristics of the system:

a) As a result of the conducted research it has been established that for higher values of the granular media, the amplitude of the transition through the resonance is lower;

b) It has been established that for higher values of the angular velocity of the body, the natural frequency of bending oscillations is smaller. At the same time, the amplitude of the transition through the resonance is smaller;

c) It has been established that at a higher speed of transition through resonance the amplitude is smaller.

On the basis of the multivariate experiment, a regression dependence has been obtained to determine the influence of the rotation speed of the working body n, the angle of the conveyor α and the length of transportation l on the power N when transporting grain material, from the analysis of which it is found that the dominant factor affecting the value of N is the angle of inclination α = 30 degrees, hereinafter the values n =500 rpm and lengths of transportation l =1.47 m.

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