RESEARCH OF NON-RESONANT OSCILLATIONS OF THE "TELESCOPIC SCREW - FLUID MEDIUM" SYSTEM
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ДОСЛІДЖЕННЯ НЕРЕЗОНАНСНИХ КОЛИВАНЬ СИСТЕМИ «ТЕЛЕСКОПІЧНИЙ ГВИНТ – СИПКЕ СЕРЕДОВИЩЕ»

Lyashuk O.L. [1)], Hevko I.B. [1)], Hud V.Z. [1)], Tkachenko I.G. [1)], Hevko O.V. [1)], Sokol M.O. [2)], Tson O.P. [1)], Kobelnyk V.R.[1)], Shmatko D.Z. [3)], Stanko A.I. [1)]
[1)]Ternopil Ivan Puluj National Technical University, Ukraine;
[2)]Ternopil Volodymyr Hnatiuk National Pedagogical University, Ukraine;
[3)] Dniprovsk State Technical University, Ukraine
E-mail: oleglashuk@ukr.net
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ABSTRACT
In the article it is substantiated the value of the angular speeds of rotation of the auger screw, which leads to the breakdown of its lateral vibrations. The dependences describing the law of change of amplitude or natural frequency at slowly variable length of the telescopic screw are deduced. Based on the Van der Paul's method, in the developed system differential equations are obtained that determine the laws of change of amplitude and frequency of the wave process in the system of a telescopic propeller. It is established that for nonresonant oscillations for this system the main parameters of bending oscillations are a continuous flow of bulk medium - the screw does not depend on its small torsional oscillations and external periodic perturbation. The analysis of the given regression equations shows that to reduce the torque of the auger it is necessary to reduce the frequency of its rotation and the angle of the conveyor. The constructive diagram and the results of theoretical calculations for assessing the influence of constructive-kinematic parameters on the torque indicators of the telescopic screw conveyor are presented.

РЕЗЮМЕ
В статті обґрунтовано значення кутових швидкостей обертання гвинта шнека, що призводить до зриву його поперечних коливань. Виведено залежності, які описують закон зміни амплітуди чи власної частоти за повільно змінної довжини телескопічного гвинта. На основі методу Ван-дер-Поля, розробленої системи отримано диференціальних рівняння, які визначають закони зміни амплітуди та частоти хвильового процесу в системі телескопічного гвинта. Встановлено, що для нерезонансних коливань для даної системи основні параметри згинальних коливань є суцільний потік сипкого середовища – гвинт не залежать від малих крутильних його коливань та зовнішнього періодичного збурення. Представлено конструктивну схему результати теоретичних розрахунків для оцінки впливу конструктивно-кінематичних параметрів на показники крутного моменту телескопічного гвинтового транспортера.

INTRODUCTION
In order to achieve the required overloading distance, screw conveyors are made prefabricated, assembled and disassembled from separate parts in manual mode or with the help of hydraulic or pneumatic equipment, which makes their design quite expensive and difficult to operate. To improve the operation of screw conveyors and increase their mobility, it is advisable to use the principle of telescope in their designs. Due to the significant angular velocities of the augers in telescopic screw conveyors (up to 800 rpm and above), the existing inhomogeneous inclusions in the reloading material, the asymmetry of the telescopic screw and external perturbations lead to its oscillations, as well as to significant dynamic loads in the auger (*Rogatynskyi et al., 2019; Rogatynskyi et al., 2014; Hevko et.al., 2019*).

Some papers consider the operation mode of an inclined screw conveyor which has a screw operating element with constant parameters incorporated in it (*Pezo et. al., 2015; Mondal, 2018*). The kinematics of grain loading has been investigated based on the motion equations in a screw conveyor. The analysis of loading movement at constant high-speed mode has been analyzed.

The works of *Chen (2009), Fernandez et al., (2011), Sun et.al., (2017), Tian & Cheng, (2018*), provide research findings on flow patterns of bulk materials depending on constructional and kinematic characteristics of screw operating tools, bunker type and solid particles, as well as frictional forces.

The authors developed and patented a number of designs of screw conveyors with rotary casings and also the directions of screw conveyors operational life increase are described in some papers (*Tripathi, et. al., 2015; Roberts, 2015; Mondal, 2018).* Some research is dedicated to the solving of above-mentioned problems, namely the development of energy-saving designs of screw conveyors and choosing their the most efficient parameters and working modes *(Pylypaka et al., 2018; Rogatynskyi et al., 2019; Pylypaka et al., 2019*).

In other articles the influence of the motion of a continuous flow of a loose or viscous medium on the longitudinal or bending oscillations of elastic bodies is studied (*Sokil and Sokil 2017; Bogolyubov and Mitropolsky 1961; Stotsko et al., 2002; Stotsko et al., 2007*). Studies have shown that even a constant velocity of the bulk or viscous medium changes the basic dynamic characteristics of bending and longitudinal oscillations. The telescopic screw system rotates at a significant angular velocity, which means that even minor transverse deformations at some point in time lead to significant stresses. With the relative movement of the bulk medium along the screw, the mathematical model of bending oscillations qualitatively acquires a new form - the appearance in it of a mixed derivative of linear and temporal variables. The working telescopic propeller is an elastic body that rotates, so due to partial wedging, inhomogeneity of the environment and other reasons, it additionally performs more torsional oscillations.

Therefore, the study of these phenomena in the system telescopic screw is a bulk medium during the movement of bulk material makes it possible to pre-select such modes of operation of the conveyor, which make these processes impossible, and thus increase the service life of the conveyor.

MATERIALS AND METHODS

Based on some research results, the maximum relative linear displacements of the outer points of the screw, due to torsional vibrations, are much less linear displacement due to its bending vibrations. (*Rogatynskyi et al., 2019; Hevko et. al., 2018*). Mathematical models of bending oscillations of the studied system assume that torsional oscillations cause a small amount of periodic action on the bent. The main parameters of this action (primarily frequency) can be determined on the basis of the main characteristics of the screw: its running mass, the moment of inertia, elastic characteristics of the material (*Stotsko et al., 2000*) or by partial processing of experimental data (*Topilnysky et al., 2017; Hevko et.al., 2020*).

In some works, the mathematical model of bending vibrations of an elastic body, which rotates along a fixed axis with a constant angular velocity, provided that a continuous flow of a homogeneous medium of zero rigidity moves along it with a constant relative linear velocity, is the system of differential equations (*Hevko et.al., 2018; Hud, 2019*).

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

Where:

$u(t,z), w(t,z)$ - projections of the vector of movement of the point of the central axis with the coordinate z of the telescopic screw at any time t;

Ω is angular velocity of rotation of the screw around the specified axis,

ρ_1, ρ_2 is respectively, the mass per unit length of the body and the moving medium,

EI is its stiffness to bend the screw (*Hud, 2019*)

$f\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},...,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)$ and $g\left(u,w,\frac{\partial u}{\partial t},\frac{\partial w}{\partial t},\frac{\partial u}{\partial z},\frac{\partial w}{\partial z},...,\frac{\partial^3 u}{\partial z^3},\frac{\partial^3 w}{\partial z^3},\gamma\right)$- 2π - periodic for $\gamma=\nu t+\gamma_0$ functions that describe the nonlinear components of the reducing force, resistance force and other forces, the maximum value of which is much smaller than the value of the reducing force, as indicated by a small parameter.

The dynamic process of the auger screw depends not only on the force and kinematic ($\frac{\partial\vartheta(z,t)}{\partial t}$, $\frac{\partial^2\vartheta(z,t)}{\partial t^2}$,Ω, V) factors but also on the method of fastening. In the case of slowly adjustable length and corresponding to the movement of the elastic screw in the bearings, the distance between $l(\tau)$ takes the form *(Hud, 2019)*:

$$u(t,z)\big|_{z=0}=\left.\frac{\partial^2 u}{\partial z^2}\right|_{z=0}=0,\quad w(t,z)\big|_{z=0}=\left.\frac{\partial^2 w}{\partial z^2}\right|_{z=0}=0,$$
$$u(t,z)\big|_{z=l(\tau)}=\left.\frac{\partial^2 u}{\partial z^2}\right|_{z=l(\tau)}=0\ w(t,z)\big|_{z=l(\tau)}=\left.\frac{\partial^2 w}{\partial z^2}\right|_{z=l(\tau)}=0. \tag{2}$$

In the mathematical model, the variable length of the screw, $l(\tau)=l_0+\varepsilon k_1 t$, k_1 is constant. This problem is to describe the main parameters of the bending oscillations of the auger screw, provided that the torsional oscillations are described by the dependence (*Hud, 2019)*:

$$\vartheta(t,z)=h\sin\frac{k\pi}{l(\tau)}z\cos\vartheta,\ \vartheta=(\Theta t+\vartheta_0) \tag{3}$$

where h is their amplitude; $\Theta=\frac{k\pi}{l(\tau)}\sqrt{\frac{GJ_0}{I_0}}$, - frequency; ϑ_0 - initial phase; I_0 - running moment of inertia about the neutral axis of the elastic body together with the medium; J_0 - its equatorial moment of inertia; G - shear modulus.

In this problem, it is believed that the bulk medium that moves along the screw does not change the stiffness of bending and torsion, and torsional vibrations of the screw. The most important from a practical point of view are cases of external and internal resonances. As for the resonance caused by external force factors, they can be caused by periodic action at the points of imperfect fixation or the interaction of the telescopic screw and the casing, and others. Restrictions on the amplitude of torsional oscillations of the considered dynamic system, the continuous flow of the medium-elastic screw, allow to build the solution of the boundary value problem Eq. 2, provided Eq. 3 to use the basic ideas of asymptotic methods of nonlinear mechanics to construct a solution of the undisturbed analogue of Eq. 1. (*Lyashuk et al., 2018; Hud, 2019; Hevko et al., 2019*).

$$\begin{gathered}(\rho_1+\rho_2)\frac{\partial^2 u_0}{\partial t^2}-2(\rho_1+\rho_2)\Omega\frac{\partial w_0}{\partial t}+\rho_2 V^2\frac{\partial^2 u_0}{\partial z^2}-2(\rho_1+\rho_2)I\Omega\frac{\partial^3 w_0}{\partial t\partial z^2}+\\+EI\frac{\partial^4 u_0}{\partial z^4}-(\rho_1+\rho_2)\Omega^2 u_0=0,\\(\rho_1+\rho_2)\frac{\partial^2 w_0}{\partial t^2}+2(\rho_1+\rho_2)\Omega\frac{\partial u_0}{\partial t}+\rho_2 V^2\frac{\partial^2 w_0}{\partial z^2}+2(\rho_1+\rho_2)I\Omega\frac{\partial^3 u_0}{\partial t\partial z^2}+\\+EI\frac{\partial^4 w_0}{\partial z^4}-(\rho_1+\rho_2)\Omega^2 w_0=0\end{gathered} \tag{4}$$

under boundary conditions:

$$u_0(t,z)\big|_{z=j}=\left.\frac{\partial^2 u_0}{\partial z^2}\right|_{z=0}=0,\quad w_0(t,z)\big|_{z=l}=\left.\frac{\partial^2 w_0}{\partial z^2}\right|_{z=l}=0 . \tag{5}$$

In order to determine the natural frequency of bending oscillations of the body as a function of the angular and linear velocity of the medium along the elastic body in the form *(Hevko et al., 2020)*

$$\omega_k = \Omega\left(I\kappa_k - 1\right) \pm \kappa_k \sqrt{\Omega^2 I\left(\kappa_k^2 I - 2\right) - \frac{\rho_2 V^2 - EI\kappa_k^2}{\rho_1 + \rho_2}} \tag{6}$$

The obtained ratio makes it possible to determine the influence of nonlinear forces on the dynamic process, as well as the whole set of external and internal factors on the bending oscillations of the auger. Among the analytical methods for studying linear oscillatory systems with concentrated masses, the most convenient for their practical use is the combination of the Bubnov-Galerkin method (*Rogatynskyi et al., 2019*) and Van der Pol (*Lyashuk et al., 2018*) and KBM (Krylov-Bogoliubov-Myitropolskyi) (*Lyashuk et al., 2018; Hevko et al., 2015; Hud et al., 2020*). To solve this problem, the basic idea of the first methods will be extended to the basic system of Eq. 1 under boundary conditions Eq. 2 Based on the Bubnov - Galerkin method and the principle of oscillation frequency in nonlinear systems, for the solution of the perturbed (nonlinear) boundary value problem Eq. 1, Eq. 2 in a form close to the "*k*" form of dynamic equilibrium, assume that for the perturbed case the parameters h and φ be functions of time:

$$\begin{aligned} u(t,z) &= h(t)(\cos\left(\kappa z + \omega t + \psi(t)\right) - \cos\left(\kappa z - \omega t - \varphi(t)\right), \\ w(t,z) &= h(t)(\sin\left(\kappa z + \omega t + \psi(t)\right) + \sin\left(\kappa z - \omega t - \varphi(t)\right). \end{aligned} \tag{7}$$

For complex nonlinear oscillations of the auger, more precisely for their bending component it is obtained:

$$\begin{aligned} \frac{\partial u(t,z)}{\partial t} &= -h(\omega + \frac{d\varphi}{dt})(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) + \\ &+ \frac{dh}{dt}(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)), \\ \frac{\partial w(t,z)}{\partial t} &= h(\omega + \frac{d\varphi}{dt})(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)) + \\ &+ \frac{dh}{dt}(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) \end{aligned}, \tag{8}$$

where the index k indicates the form of dynamic equilibrium.

According to the main idea of the Van der Paul's method, dependencies that relate the derivatives of the desired functions are obtained:

$$\begin{aligned} -h\frac{d\varphi}{dt}(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) + \frac{dh}{dt}(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)) = 0, \\ h\frac{d\varphi}{dt}(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right) + \frac{dh}{dt}(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) = 0. \end{aligned} \tag{9}$$

Therefore, by differentiating the dependencies of Eq. 9, taking into account the derivatives of the desired functions, we are get:

$$\begin{aligned} \frac{\partial^2 u(t,z)}{\partial t^2} &= -\frac{dh}{dt}\omega(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) + \\ &+ h\omega(\frac{d\varphi}{dt} + \omega)(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)), \\ \frac{\partial^2 w(t,z)}{\partial t^2} &= \frac{dh}{dt}\omega(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)) - \\ &- h\omega(\frac{d\varphi}{dt} + \omega)(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)). \end{aligned} \tag{10}$$

The obtained dependences in the base are substituted into the system of differential equations Eq. 1 taking into account Eq. 3, it is obtained:

$$\begin{aligned} -\frac{dh}{dt}\omega(\sin\left(kz + \omega t + \varphi\right) + \sin\left(kz - \omega t - \varphi\right)) + h\omega\frac{d\varphi}{dt}(\cos\left(kz + \omega t + \varphi\right) - \cos\left(kz - \omega t - \varphi\right)) = \varepsilon\tilde{f}(h, x, \psi, \vartheta), \\ -h\frac{d\varphi}{dt}(\sin\left(\kappa z + \omega t + \varphi\right) + \sin\left(\kappa z - \omega t - \varphi\right)) + \frac{dh}{dt}(\cos\left(\kappa z + \omega t + \varphi\right) - \cos\left(\kappa z - \omega t - \varphi\right)) = 0, \end{aligned} \tag{11}$$

$$\frac{dh}{dt}\omega(\cos(\kappa z+\omega t+\varphi)-\cos(\kappa z-\omega t-\varphi))-h\omega\frac{d\varphi}{dt}(\sin(\kappa z+\omega t+\varphi)+\sin(\kappa z-\omega t-\varphi))=\varepsilon\tilde{g}(a,z,\psi,\gamma,\vartheta).$$

where $\varepsilon\tilde{f}(h,x,\psi,\gamma,\vartheta),\varepsilon\tilde{g}(h,z,\psi,\gamma,\vartheta)$ - correspond to the values according to the function:

$$\begin{aligned}&f_1\left(u,w,\frac{\partial u}{\partial t},...,\frac{\partial^3 w}{\partial z^3},\gamma\right)+2(\rho_1+\rho_2)\frac{\partial\vartheta(z,t)}{\partial t}\frac{\partial w}{\partial t}+2(\rho_1+\rho_2)I\frac{\partial\vartheta(z,t)}{\partial t}\frac{\partial^3 w}{\partial t\partial z^2}+2(\rho_1+\rho_2)\Omega\frac{\partial\vartheta(z,t)}{\partial t}u+(\rho_1+\rho_2)\frac{\partial^2\vartheta(z,t)}{\partial t^2}w\\&f_2\left(u,w,\frac{\partial u}{\partial t},...,\frac{\partial^3 w}{\partial z^3},\gamma\right)-2(\rho_1+\rho_2)\frac{\partial\vartheta(z,t)}{\partial t}\frac{\partial u}{\partial t}-2(\rho_1+\rho_2)I\frac{\partial\vartheta(z,t)}{\partial t}\frac{\partial^3 u}{\partial t\partial z^2}-2(\rho_1+\rho_2)\Omega\frac{\partial\vartheta(z,t)}{\partial t}w(\rho_1+\rho_2)\frac{\partial^2\vartheta(z,t)}{\partial t^2}u\end{aligned}\quad(12)$$

where, $u(t,z)$, $w(t,z)$, $\vartheta(t,z)$ functions are described by relations Eq. 3, Eq. 8.

If the right-hand side of the basic system of differential equations satisfies the specified condition, then this relation takes the form:

$$\begin{aligned}&\int_0^l\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)(\sin(\kappa z+\psi)+\sin(\kappa z-\psi,\gamma,\vartheta))dzd\psi=\\&=\int_0^l\int_0^{2\pi}\tilde{g}(h,z,\psi,\gamma,\vartheta)(\cos(\kappa z+\psi)-\cos(\kappa z-\psi)dzd\psi,\\&\int_0^l\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)(\cos(\kappa z+\psi,\gamma,\vartheta)-\cos(\kappa z-\psi,\gamma,\vartheta)dzd\psi=\\&=\int_0^l\int_0^{2\pi}\tilde{g}(h,z,\psi,\gamma,\vartheta)(\sin(\kappa z+\psi)+\sin(\kappa z-\psi))dzd\psi,\end{aligned}\quad(13)$$

The obtained system of differential equations Eq. 12 with conditions that follow from the main idea of the Van der Paul's method, allows determining the laws of change of amplitude and frequency of the wave process *(Hevko et al., 2020)*:

$$\begin{aligned}&\frac{dh}{dt}=-\frac{1}{2\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)(\cos(\kappa z+\psi)-\cos(\kappa z-\psi))d\psi dz,\\&\frac{d\theta}{dt}=\omega+\frac{1}{2h\pi\omega l(\rho_1+\rho_2)}\int_0^l\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)(\sin(\kappa z+\psi)+\sin(\kappa z-\psi))d\psi dz,.\end{aligned}\quad(14)$$

In the system of differential equations Eq. 14 the subintegral functions, and the right parts are periodic by arguments ψ,γ,ϑ, which can have non-resonant and resonant oscillations in the screw auger. Based on this, non-resonant oscillations of the amplitude and frequency of bending oscillations of the telescopic auger are described by the dependences *(Hevko et al., 2020)*:

$$\begin{aligned}&\frac{da}{dt}=\frac{-\varepsilon}{8\omega\pi^2 l}\int_0^l\int_0^{2\pi}\int_0^{2\pi}\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)\sin\frac{\pi}{l}z\cos\psi dzd\psi d\gamma d\vartheta,\\&\frac{d\psi_1}{dt}=\omega-\frac{\varepsilon}{8\omega\pi^2 hl}\int_0^l\int_0^{2\pi}\int_0^{2\pi}\int_0^{2\pi}\tilde{f}(h,z,\psi,\gamma,\vartheta)\sin\frac{\pi}{l}z\sin\psi dzd\psi d\gamma d\vartheta\end{aligned}\quad(15)$$

Thus, with 2π periodicity in the phase of forced vibrations and the phase of torsional vibrations of the auger screw of subintegral functions it follows that for non-resonant oscillations the system is considered where the main parameters of bending oscillations of the system are continuous flow of bulk medium. However, the amplitude of these vibrations depends on the elastic characteristics of the screw material, which affect the natural frequency, viscoelastic friction forces, the method of fastening, but also on the angular velocity of its rotation.

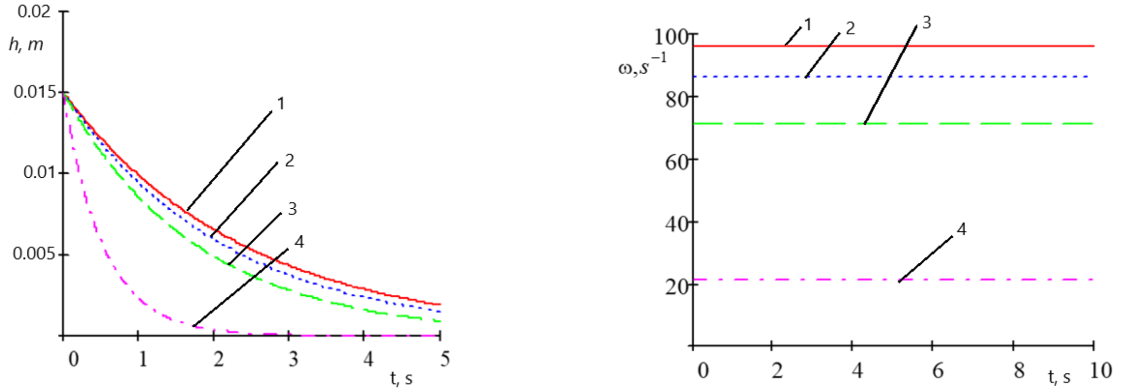
For different angular velocities of the screw rotation, the time variation of the amplitude of damped oscillations is presented, provided that the elastic properties of the material of the auger screw satisfy the nonlinear technical law of elasticity, and the viscous friction force is proportional to the speed in the degree of motion of the points.

In this case, differential equations Eq. 15 are transformed to the form:

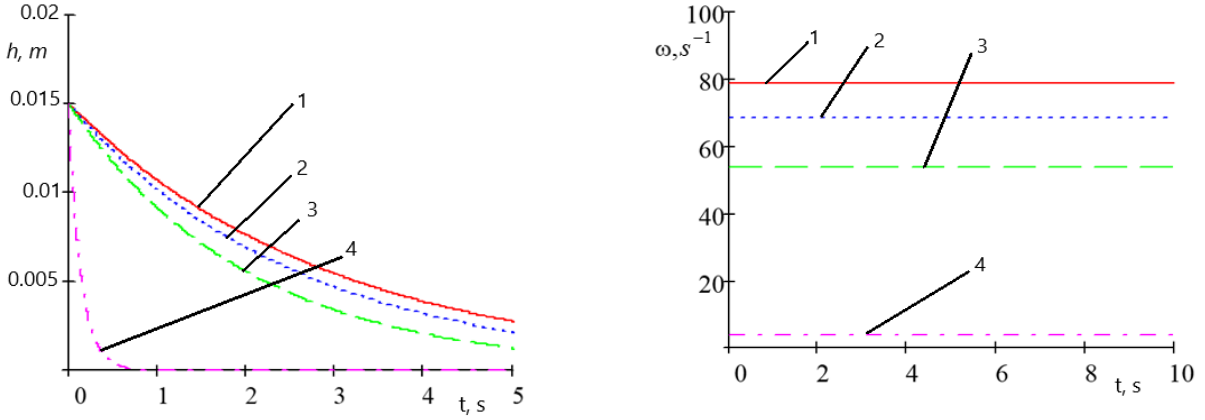
$$\frac{dh}{dt}=-\frac{k_1(\omega)^{s-1}}{(\rho_1+\rho_2)\pi}h^s;\quad\frac{d\psi}{dt}=\omega-\frac{\bar{k}_1 EI}{(\rho_1+\rho_2)}h^2-(\frac{\pi}{l})^2\frac{\rho_2}{8\omega(\rho_1+\rho_2)}V^2.\quad(16)$$

RESULTS AND DISCUSSION

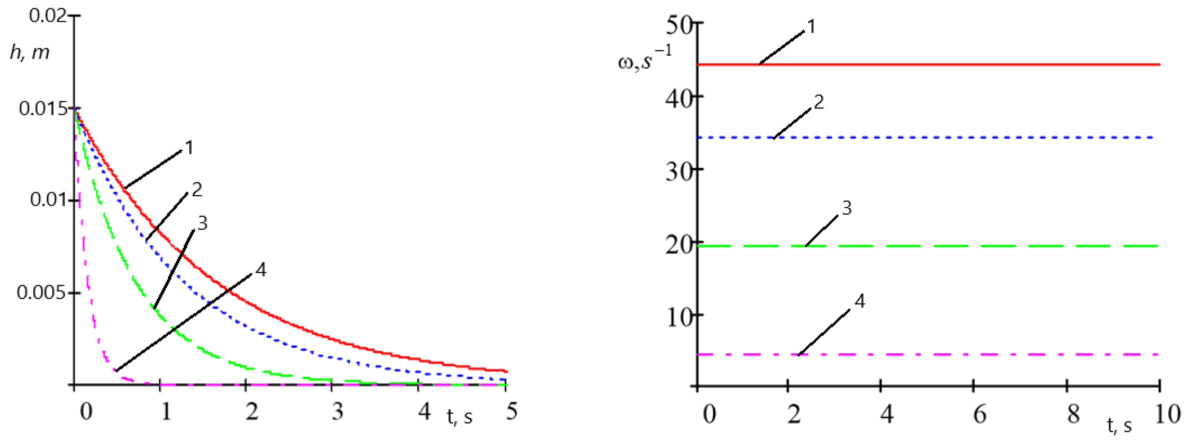
In fig. 1 and fig. 2, for different values of the parameters of the system "telescopic screw as a bulk medium" is shown the change in time of the amplitude and frequency of the damping oscillations of the screw.



a) 1– $\Omega_1 = 0 rad/s$; 2– $\Omega = 10 rad/s$; 3 – $\Omega = 25 rad/s$; 4– $\Omega = 75 rad/s$



b) 1 – $\Omega_1 = 0 rad/s$; 2– $\Omega = 10 rad/s$; 3– $\Omega = 25 rad/s$; 4– $\Omega = 75 rad/s$

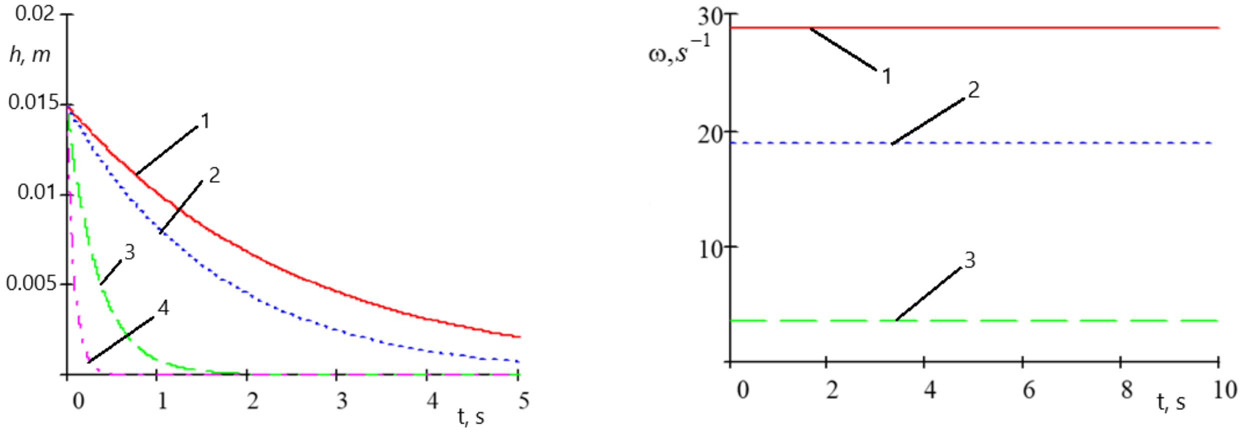


c) 1 – $\Omega_1 = 0 rad/s$; 2– $\Omega = 10 rad/s$; 3– $\Omega = 25 rad/s$; 4– $\Omega = 40 rad/s$

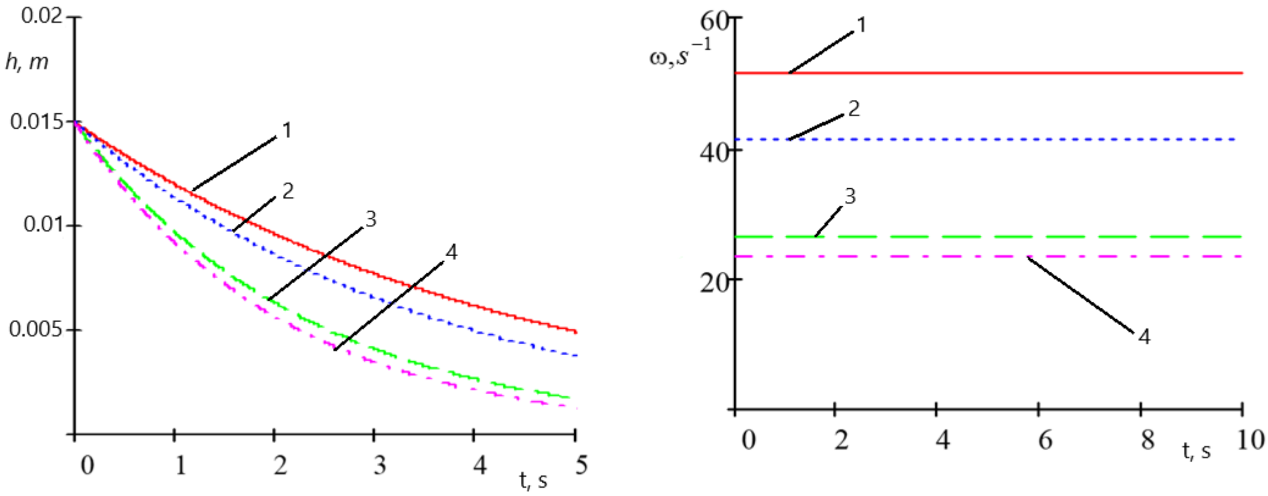
c) -------- $\Omega_1 = 0 s^{-1}$; ····· $\Omega = 10 s^{-1}$; – – – – – $\Omega = 25 s^{-1}$; –·–·– $\Omega = 40 s^{-1}$

Fig. 1 - Laws of change of amplitude of damped oscillations and natural frequency of the telescopic screw without environment ($\rho_2 = 0$ kg/m) at various angular speeds of its rotation at $I = 6 \cdot 10^{-6}$ kg m 2**;**

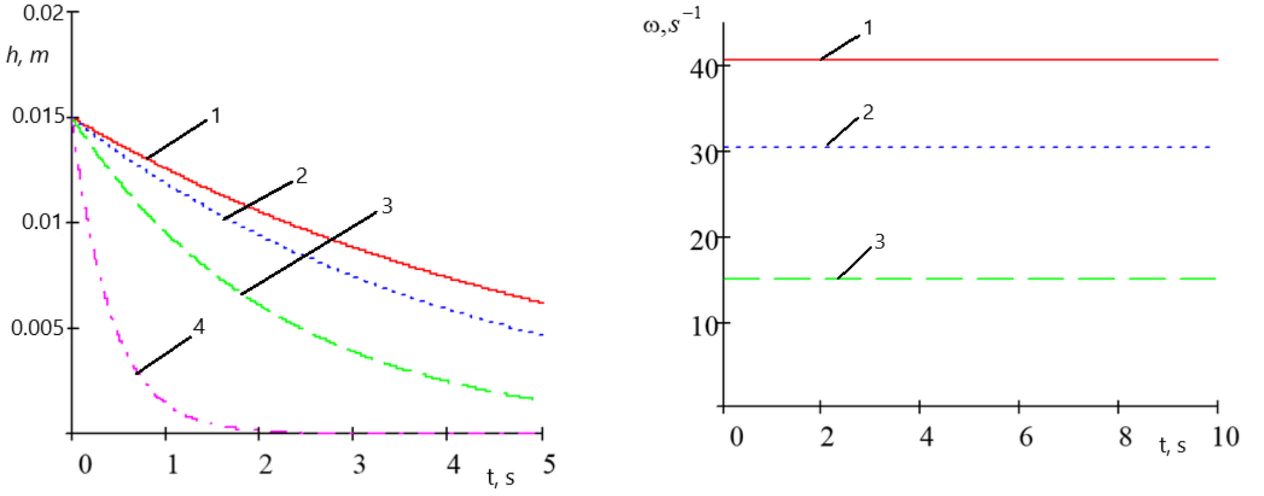
$E = 2,06 \cdot 10^{11}$ N/ m2: *a) l = 6 m;* $\rho_1 = 10$ *kg / m; б) l = 6 m;* $\rho_1 = 15$ *kg / m; в) l = 8 m;* $\rho_1 = 15$ *kg / m*



a) $1-\Omega_1=0 rad/s;\ 2-\ \Omega=10 rad/s;\ 3-\ \Omega=25 rad/s;\ 4-\Omega=28 rad/s$



b) $1-\ \Omega_1=0 rad/s;\ 2-\Omega=10 rad/s;\ 3-\ \Omega=25 rad/s;\ 4-\Omega=28 rad/s$



c) $1-\Omega_1=0 rad/s;\ 2-\Omega=10 rad/s;\ 3-\Omega=25 rad/s;\ 4-\Omega=38 rad/s$

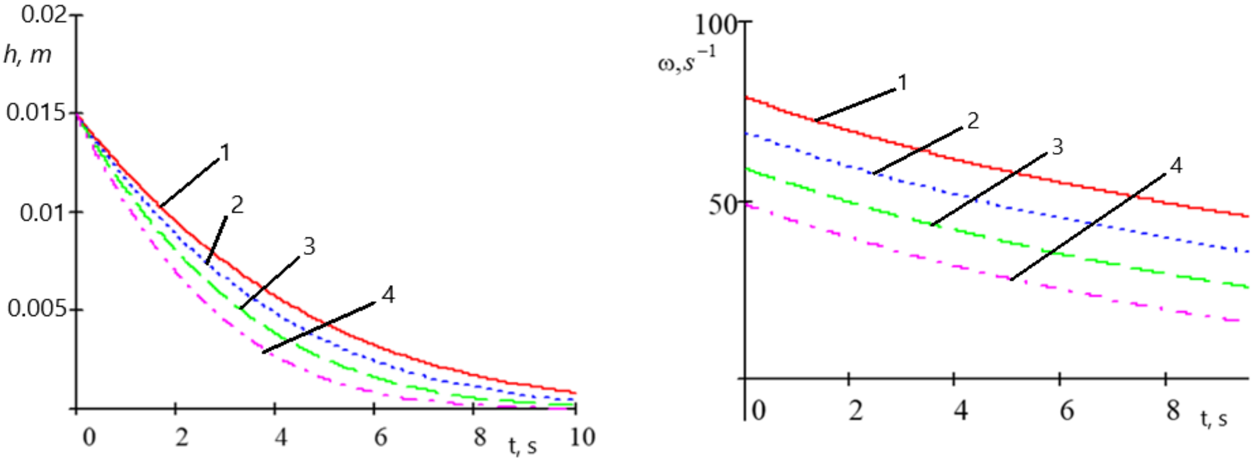
Fig. 2 – Laws of change of amplitude of damped oscillations and natural frequency of the telescopic screw at different angular velocities of its rotation and different quantities of relative motion of the medium at

ρ_1 =15 kg/m; $I = 6\cdot10^{-6}$ kg m ²; V = 5 m/s; $E = 2.06\cdot10^{11}$ N/m²:

a) ρ_2 =20 kg/m*; l* = 8 m*; b)* ρ_2 =20 kg / m*; l* =6 m*; c)* $\rho_2 = 40$ kg/m*; l* =6 m

In fig. 1 and fig. 2, it can be seen that the failure of oscillations occurs at Ω = 100 rad/s (Fig. 1.a), Ω= 80 rad/s (Fig. 1.b) and Ω = 45 rad/s (Fig. 1.c); Ω = 30 rad/s (Fig. 2.a), Ω = 55 rad/s (Fig. 2.b) and Ω = 40 rad/s (Fig. 2.c). It is established that at certain values of angular velocities of the auger screw the failure of its transverse oscillations occurs for larger values of running mass of the propeller.

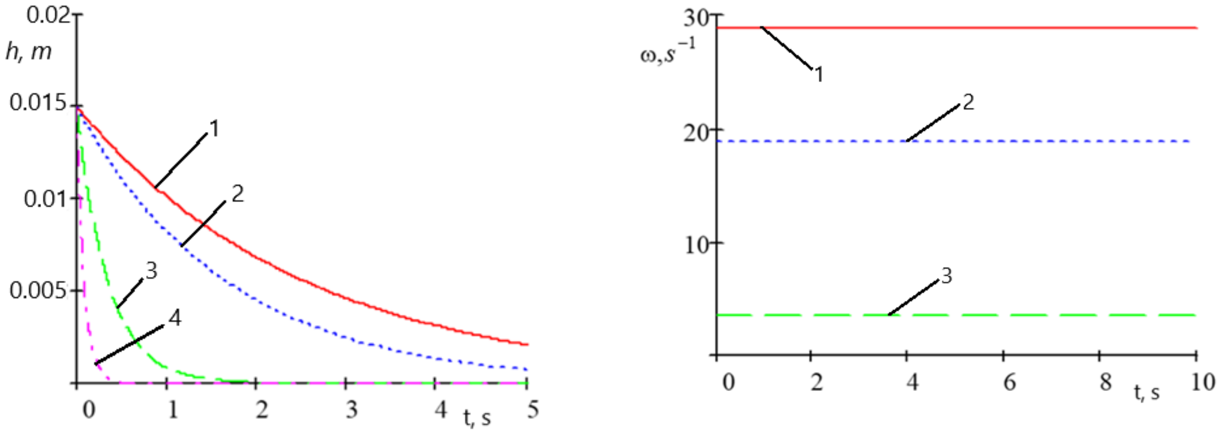
The obtained results reflect the dynamic processes in the screw under the condition of slowly changing its length. The obtained dependence Eq.16 describes the law of change of amplitude or natural frequency at slowly variable length of the telescopic screw. Figure 3 shows the laws of change of these parameters for a telescopic propeller without a bulk medium, and in Fig. 4 with bulk medium.



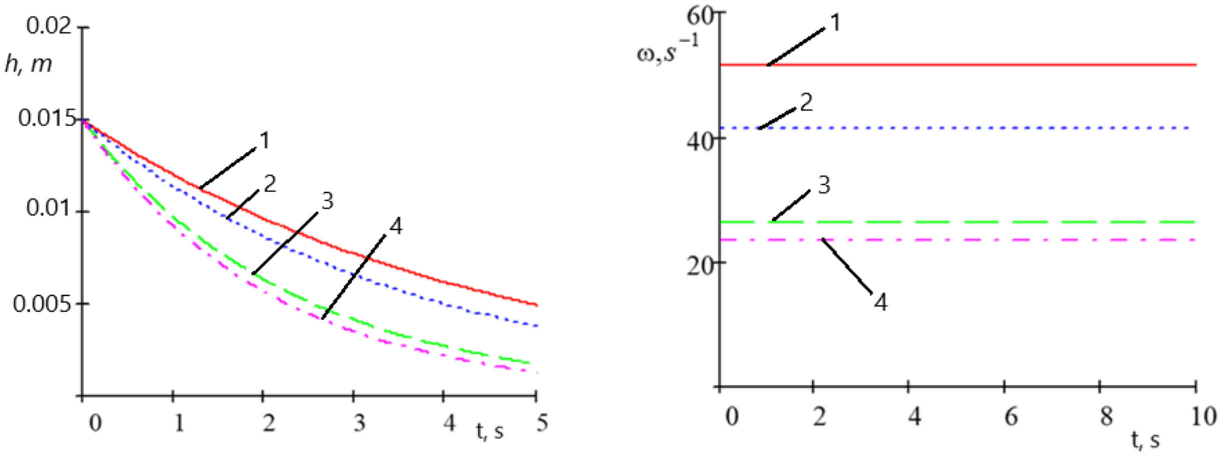
a) $1-\Omega_1=0\,\text{rad/s};\ 2-\Omega=10\,\text{rad/s};\ 3-\Omega=25\,\text{rad/s};\ 4-\Omega=30\,\text{rad/s}$

Fig. 3 – Change of amplitude-frequency characteristic of the telescopic screw at slowly variable length

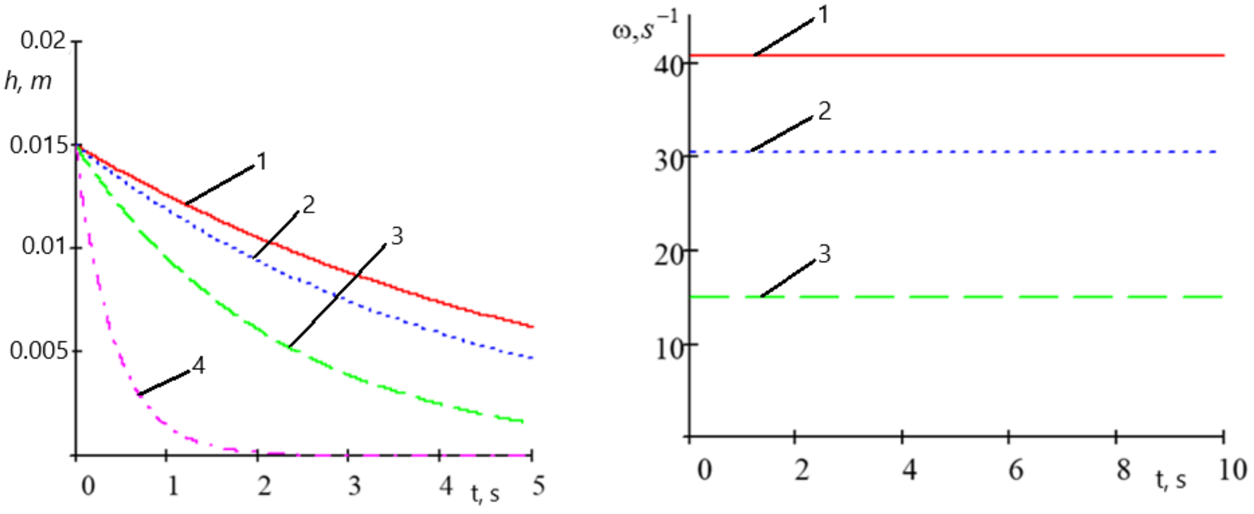
at: $\rho_1=15$ kg/m; $\rho_2=0$ kg/m; $I=6\cdot10^{-6}$ kg·m²; $E=2.06\cdot10^{11}$ N/m²; v = 5 m/s; l = 6 + 0.2t



a) 1– $\Omega_1=0\,rad/s;\ 2-\Omega=10\,rad/s;\ 3-\Omega=20\,rad/s;\ 4-\Omega=30\,rad/s$



b) 1– $\Omega_1=0\,rad/s;\ 2-\Omega=10\,rad/s;\ 3-\Omega=20\,rad/s;\ 4-\Omega=30\,rad/s$



c) $1-\Omega_1=0 rad/s;\ 2-\Omega=10 rad/s;\ 3-\Omega=20 rad/s;\ 4-\Omega=30 rad/s$

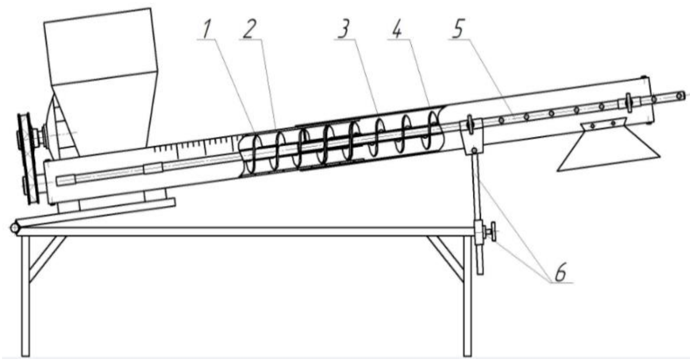
Fig. 4 – Change of amplitude-frequency characteristic of the system "telescopic screw as a bulk medium" at different quantities of movement of the last and variable length of the screw at $\rho_1=15$ kg/m; l = 6 + 0.2t;

V=5 m/s; $I=6\cdot10^{-6}$ kg·m ²; $E=2.06\cdot10^{11}$ N/m²: *а) ρ₂* =15 kg/m; *б) ρ₂* =20 kg/m; *в) ρ₂* =30 kg/m

The presented graphical dependences show that with the elongation of the telescopic screw the frequency of its oscillations decreases over time, and the main results obtained for the case of its constant length can be used for the case of slowly variable length. However, from a theoretical point of view, it is also important to study the system in the so-called resonance zone, or more precisely when passing the resonance.

To confirm the theoretical assumptions, an experimental setup was designed and manufactured, which is shown in Fig. 5 (*Hud et. al., 2019*).

a)



b)

Fig. 5 – 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

In the experimental setup, the outer diameter of the auger is 97 mm; inner diameter of the fixed branch pipe - 100 mm; external - 107 mm; the inner diameter of the movable pipe is 109 mm. The movable pipe is made of galvanized sheet, and therefore it contains a connecting seam and ovals and irregularities along the entire length, which affected the speed of twisting and untwisting of the telescopic part of the screw conveyor.

To determine the influence of structural and kinematic parameters (auger speed, auger elongation length and angle of the conveyor) on the torque of the auger drive during transportation of corn, wheat and feed, full-factor experiments were performed and regression equations were derived:

- during transportation of corn:

$$T_{(n_s,l,\gamma_1)} = -12.91 + 1.18\cdot10^{-2} n_s + 17.33 l + 3.8\cdot10^{-2}\gamma_1 - 4.46\cdot10^{-3} n_s l + \\ +2.9\cdot10^{-6} n_s^2 - 1.73 l^2 - 4.6\cdot10^{-4}\gamma_1{}^2; \tag{17}$$

- during transportation of wheat:

$$T_{(n_s,l,\gamma)} = -13.59 + 1.23\cdot10^{-2} n_s + 18.77 l + 3.99\cdot10^{-2}\gamma_1 - 4.68\cdot10^{-3} n_s l + \\ +3.05\cdot10^{-6} n_s^2 - 1.84 l^2 - 4.83\cdot10^{-4}\gamma_1{}^2; \tag{18}$$

- during transportation of compound feed:

$$T_{(n_s,l,\gamma)} = -11.53 + 1.05\cdot10^{-2} n_s + 15.95 l + 3.412\cdot10^{-2}\gamma_1 - 4\cdot10^{-3} n_s l + \\ +2.63\cdot10^{-6} n_s^2 - 1.53 l^2 - 4.12\cdot10^{-4}\gamma_1{}^2. \tag{19}$$

The analysis of the given regression equations shows that to reduce the torque of the auger it is necessary to reduce the frequency of its rotation and the angle of the conveyor. Based on the regression equations Eq. 17-19, the graphical dependences of the torque value are constructed, which are shown in Fig. 6.

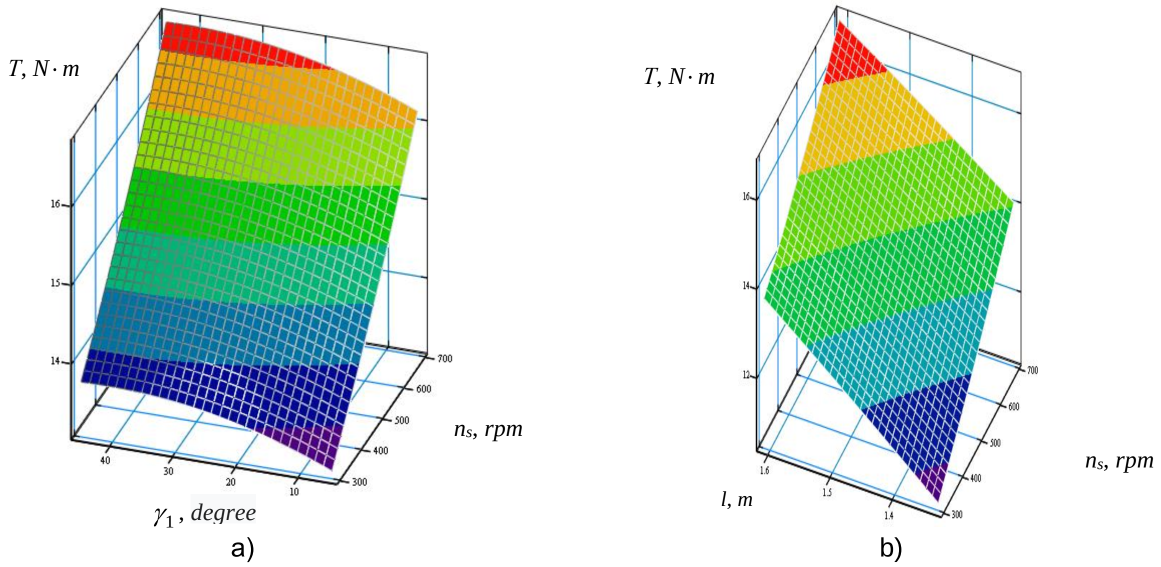


Fig. 6 – Response surface (a, b) dependence of the torque on the drive of the telescopic screw conveyor during corn overload from: a) $T = f(n_s;\gamma_1)$ at $l = 1.61$ m; b) $T = f(n_s;l)$ *at* $\gamma_1 = 45$ degrees

From (fig. 6) the analysis of the given regression equations shows that to reduce the torque of the auger it is necessary to reduce the frequency of its rotation and the angle of the conveyor. It is experimentally established that the biggest problem in telescopic screw conveyors is to maintain the same gap between the casing and the spiral in different sections of the telescope, which significantly affects the rolling of the movable part of the auger on the stationary and the appearance and magnitude of torsional and bending vibrations.

CONCLUSIONS

It has been established that at certain values of the angular velocities of rotation of the auger screw, its transverse oscillations are disrupted: for large values of the linear mass of the screw or medium, the disruption of oscillations takes place at lower angular velocities; for auger screws of large lengths, the breakdown of oscillations takes place at lower angular speeds of rotation. Dependences are derived that describe the law of variation of the amplitude or natural frequency for a slowly variable length of the telescopic screw. It was found that with lengthening of the telescopic screw the frequency of its oscillations comes with time, and the main results obtained for the case of its constant length can be used for the case of slowly variable length.

It is seen that with increasing auger speed, auger elongation length and conveyor angle, the torque on the auger drive increases and the maximum torque of 17.51 N.m is reached during wheat transportation. The maximum torque on the drive of the telescopic screw conveyor, auger for transportation of corn and compound feed is 16.75 N.m and 15.02 N.m, respectively, and the minimum - 9.94 N.m and 8.93 N.m. Increasing the speed of the auger n_s, from 300 rpm up to 700 rpm leads to an increase in torque on the auger drive to 35%. The increase in the angle of the conveyor from 5 to 45 degrees gives an increase in torque to 4.1%, and increasing the length of the elongation of the auger from 1.33 to 1.61 m leads to an increase in torque by 24.4%.

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