MATHEMATICAL MODEL OF BENDING VIBRATIONS OF A HORIZONTAL FEEDER-MIXER ALONG THE FLOW OF GRAIN MIXTURE
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МАТЕМАТИЧНА МОДЕЛЬ ЗГИННИХ КОЛИВАНЬ ГОРИЗОНТАЛЬНОГО ЗАВАНТАЖУВАЧА-ЗМІШУВАЧА ВЗДОВЖ ПОТОКУ ЗЕРНОВОЇ СУМІШІ

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ABSTRACT
During the grain mixture motion along the working body of the feeder, the transverse (bending) vibrations occur. To deduce a differential equation describing bending vibrations of the feeder-mixer horizontal screw, a physical model is developed; the distribution of forces acting on the element of the horizontal working body during the grain mixture movement is defined.

РЕЗЮМЕ
При русі зернової суміші вздовж робочого органу завантажувача виникають поперечні (згинні) коливання. Для отримання диференціального рівняння, яке описує згинні коливання горизонтального гвинта завантажувача-змішувача розроблена фізична модель та розподіл сил, які діють на елемент горизонтального робочого органу під час переміщення зернової суміші.

INTRODUCTION
Screw transport and technological mechanisms are widely used in various industries. The effectiveness of many sections, shops and enterprises in general depends on their reliable work. Therefore, in order to ensure the reliability and quality of the technological processes implementation by means of screw mechanisms, the dynamic stresses caused by vibration processes should be considered.

The foundations of designing and researching screw conveyors have been laid by the scholars (*Hevko B.M., 1989; Hevko B.M., Pavelchuk Yu.F., 2017; Rohatynskyi R., and others, 2012; Hevko R.B., and others, 2012; Hevko R.B., Klendiy O.M. 2014; Hevko R.B., and others, 2015; Lyashuk O.L., and others, 2015;* Tian Y., Cheng Q., and others, (2018*)*. The vibration theory has been studied in the works of (Bulgakov V., *and others,* 2017*;* Hewko B.M., and others, 2015; *Loveykin V.S., Nesterov A.P., 2002; Chen L.Q. 2009 and others; Sokil M.B., and others, 2016).* However, the dynamic stresses in screw working bodies that arise under different modes of their operation in non-resonant and resonant zones are not sufficiently studied; and the reliability of screw transport and technology systems is not assessed. Therefore, the need for further research is indisputable.

The objective of the work is to develop a mathematical model of bending vibrations of the feeder-mixer horizontal working body.

MATERIALS AND METHODS
To deduce a differential equation describing the bending vibrations of an auger of the feeder-mixer horizontal screw, along which a homogeneous grain flow moves, the following assumptions should be considered:

- the cross-sectional area, mass per unit length, rigidity of the screw-mixer (together with a cylindrical casing) are slowly variable values along the length;
- the elastic properties of the working screw, together with the casing, satisfy the close to linear law of elasticity;
- during the vibration of the working body, the resistance force is proportional to its motion velocity in degree s;

• the grain mixture flows at a constant velocity relative to the screw-mixer and does not affect its roughness;

• the normal sections of the screw-mixer remain perpendicular to its neutral axis (there is no deplanation of the cross-section);

• the deflection of separate parts of the screw-mixer occurs in the direction perpendicular to its neutral line, that is, the longitudinal displacements are neglected (deflections of the axis points of the elastic body of an arbitrary normal section occur in the horizontal plane);

These assumptions allow to unambiguously determining the position of the feeder-mixer horizontal branch by a function that describes the horizontal displacement of the screw working body neutral axis. Obviously, the specified function will depend on two variables - the linear coordinate x and the time t. The linear coordinate x will be deduced from the point of loading the grain mixture in a horizontal cylindrical casing; and the axis OX is directed along the undeformed axis of the screw. The record $y(x,t)$ denotes the deflection of the neutral axis point of the working screw with the coordinate x in the horizontal direction at an arbitrary time (Fig. 1 a). The forces acting on the conditionally specified element of the deformed screw are shown in Fig. 1 b.
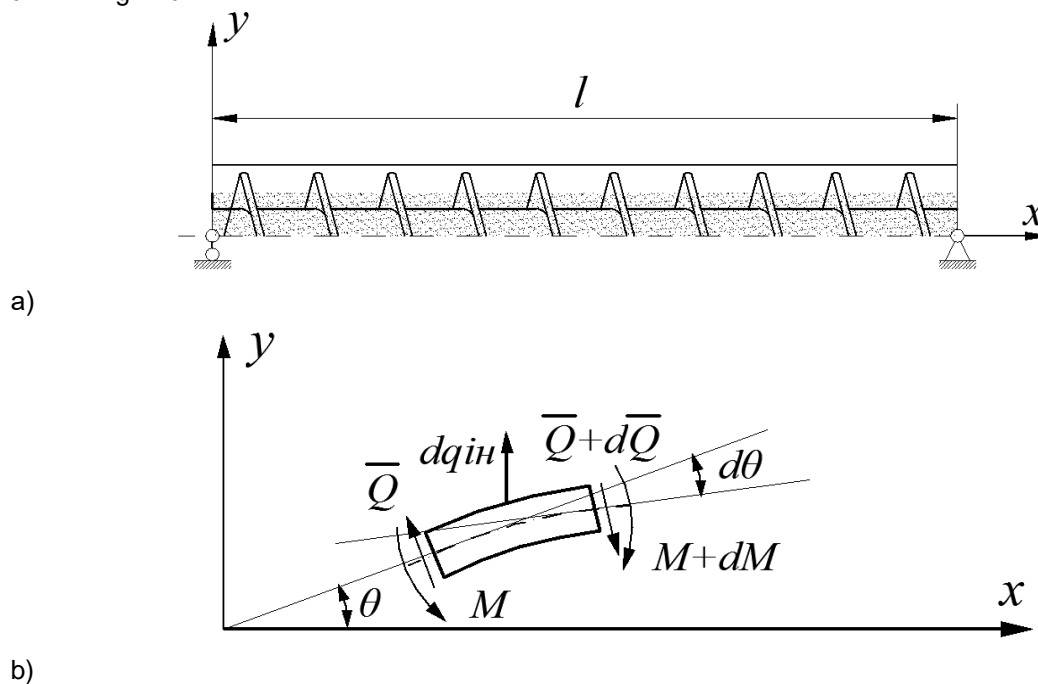
a)

b)

Fig. 1 - Physical model and distribution of forces acting on the element of the horizontal working body during the grain mixture displacement

Providing that:

ρ- the mass per unit length of the horizontal working body;

m - the mass per unit length of the grain mixture continuous flow, which moves relative to the working body at a constant velocity u ;

$E(x)$ and $I(x)$ - accordingly, the elasticity modulus of the working body material and the inertia moment of the cross-section relative to the neutral axis;

$M + \dfrac{\partial M}{\partial x}dx$ and $M + \dfrac{\partial M}{\partial x}dx$ - bending moments at the beginning and at the end of the conditionally specified element;

Q and $Q + \dfrac{\partial Q}{\partial x}dx$ - cross-cutting forces in the above-mentioned places of the conditionally specified element;

θ - and $\theta + d\theta$ - angles formed by a tangent to the neutral line with the OX axis at the beginning and at the end of the specified element;

$dq = q(x,t)dx$ - the resulting force component of the external forces in the plane OXZ (plane of vibrations) acting on the conditionally specified element of the working body; $q(x,t)$ - the forces intensity.

$\dfrac{\partial^2 y}{\partial t^2}$ - a projection of the acceleration of the specified element centre on the axis OY;

$\dfrac{d^2 y}{dt^2}$ - a projection on the same axis of the absolute acceleration of the grain mixture element, which coincides with the screw auger element under study at the given time;

then, for the case of small ($\sin\theta = \dfrac{\partial y}{\partial x}$, $\sin(\theta + d\theta) = \dfrac{\partial y}{\partial x} + \dfrac{\partial^2 y}{\partial x^2}dx$) bending vibrations of the working body horizontal part, the equation of "dynamic equilibrium" acquires the form

$$Q\cos\theta - (Q + dQ)\cos(\theta + d\theta) + dq_{i\text{H}} + q(x,t)dx = 0 \qquad (1)$$

The inertia force (*Chen, L.Q., 2009*) of the element $dq_{i\text{H}}$ under study together with the grain mixture is determined by the dependence

$$dq_{i\text{H}} = m\frac{d^2 y(x,t)}{dt^2}dx + \rho\frac{\partial^2 y(x,t)}{\partial t^2}dx + (m + \rho)\omega^2 \qquad (2)$$

where ω - angular velocity of the screw working body rotation.

Connecting the cross-cutting force to the bending moment (*Hevko I., 2012; Hevko I.B., 2013; Oleg Lyashuk, and others, 2016*) ($Q = \dfrac{dM}{dx}$) by the known ratio, the following formula is derived from equation (1)

$$m\frac{d^2 y(x,t)}{dt^2} + \rho\frac{\partial^2 y(x,t)}{\partial t^2} + (m + \rho)\omega^2 y + \delta\left(\frac{\partial y(x,t)}{\partial t}\right)^s =$$

$$+\frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}\left(E(x)I(x)\left(\frac{\partial^2 y(x,t)}{\partial x^2} + \mu\left(\frac{\partial^2 y(x,t)}{\partial x^2}\right)^3\right)\right)\right] = f(x, \text{y}, \theta). \qquad (3)$$

The addend $\delta\left(\dfrac{\partial y(x,t)}{\partial t}\right)^s$ in equation (3) describes the external force of the resistance, which is proportional to the motion velocity in degree s , and δ is the coefficient of proportionality at the given force during vibration of the feeder-mixer working body.

The function $f(x, \text{y}, \theta)$ is somewhat different in nature (nonlinear and periodic in θ), and it will be further considered.

The grain mixture moves relative to the screw working body with a constant relative linear velocity u , so the body's inertia equals

$$m\frac{d^2 y(x,t)}{dt^2} = m\frac{\partial^2 y(x,t)}{\partial t^2} + m\frac{\partial^2 y(x,t)}{\partial x^2}(u)^2 + 2m\frac{\partial^2 y(x,t)}{\partial t\partial x}u. \qquad (4)$$

Based on the above, the following formula is deduced from equation (3)

$$\left(m+\rho\right)\left(\frac{\partial^2 y\left(x,t\right)}{\partial t^2}-\omega^2 y\left(x,t\right)\right)+2mu\frac{\partial^2 y\left(x,t\right)}{\partial t\partial x}+mu^2\frac{\partial^2 y\left(x,t\right)}{\partial x^2}+\delta\left(\frac{\partial y\left(x,t\right)}{\partial t}\right)^s+$$

$$+\frac{\partial}{\partial x}\left[\frac{\partial}{\partial x}\left(EI\left(x\right)\left(\frac{\partial^2 y\left(x,t\right)}{\partial x^2}+\mu\left(\frac{\partial^2 y\left(x,t\right)}{\partial x^2}\right)^3\right)\right)\right]=f(x,\text{y},\theta). \tag{5}$$

Considering the physical process of the system, the boundary conditions for the last equation can be presented as

$$y\left(x,t\right)\Big|_{x=l}^{x=0,}=0,\frac{\partial^2 y}{\partial x^2}\left(x,t\right)\Big|_{x=l}^{x=0,}=0. \tag{6}$$

Notes:

1. In dependence (3), the condition of nonlinear-elastic properties of the screw working body material together with the cylindrical casing is applied in the form of a nonlinear technical law of elasticity (*Mytropolskyi Y.A., Lymarchenko O.S. 1998*): $\sigma=E\left(\varepsilon_1+\mu\varepsilon_1^3\right)$ $\varepsilon_1=\dfrac{\partial y\left(x,t\right)}{\partial x}$ - relative deformation of the screw working body, μ defines the deflation of its elastic properties from the linear law and $\mu<<1$;

2. The grain mixture motion is complex. The first summand of the right-hand side of the dependence (4) expresses the inertial forces of the mixture in the transportation motion, the second - in the relative motion, and the third summand expresses Coriolis components. Based on the above, inertial forces of the grain mixture can be defined.

The influence of the whole set of parameters describing the dynamics of the system under study could be defined only by solving the boundary problems (5), (6). It is difficult to find the analytical solution even for the corresponding linear model (without considering the nonlinearly elastic properties of the working screw material). Therefore, the dynamic process of the system 'an elastic body - a continuous flow of grain mixture' at a limited velocity of the mixture should be studied. Based on the above, the proposed model of the dynamic process can be presented as

$$\frac{\partial^2 y}{\partial t^2}+\alpha^2\frac{\partial^4 y}{\partial x^4}-\omega^2 y=-\mu F\left(y,\theta,\frac{\partial y}{\partial t},\frac{\partial^2 y}{\partial x^2},\frac{\partial^3 y}{\partial x^3},\frac{\partial^4 y}{\partial x^4}\right) \tag{7}$$

$$\alpha^2=\frac{EI}{m+\rho},\quad F\left(y,\theta,\frac{\partial y}{\partial t},\frac{\partial^2 y}{\partial x^2},\frac{\partial^3 y}{\partial x^3},\frac{\partial^4 y}{\partial x^4}\right)-\text{analytical } 2\pi\text{- periodical } \theta \text{ function:}$$

$$\mu F\left(y,\theta,....,\frac{\partial^4 y}{\partial x^4}\right)=-\eta\frac{EI}{m+\rho}\frac{\partial^2}{\partial x^2}\left(\frac{\partial^2 y}{\partial x^2}\right)^2+\frac{1}{m+\rho}f(x,y,\theta)-\frac{1}{m+\rho}(mu^2\frac{\partial^2 y\left(x,t\right)}{\partial x^2})^2+2m\frac{\partial^2 u}{\partial x\partial t}u+-\delta\left(\frac{\partial y\left(x,t\right)}{\partial t}\right)^s$$

Thus, the dynamic process of the given system can be considered as the overlap of two waves of the same length with time-varying amplitudes and frequencies, that is, the first approximation of the asymptotic solution (*Mytropolskyi Y.A., Lymarchenko O.S. 1998*;) of the boundary value problem (7), (6) can be submitted as

$$y\left(x,t\right)=C_1\cos\left(\kappa x+\psi\right)+C_2\cos\left(\kappa x-\psi\right)+\mu\, y_1\left(a,x,\psi,\theta\right) \tag{8}$$

where $C_1, C_2,$ - the constant content and appearance of the waves will be set below, ψ - the phase of the indicated waves, $\mu\, y_1\left(a,x,\psi,\theta\right)$ - the perturbation of the vibration process is caused by nonlinear and other forces.

The representation of the solution in the form (8) must satisfy the boundary conditions (6). Therefore, $C_1=-C_2=a$, and for the periodic by ψ and θ function $y_1\left(a,x,\psi,\theta\right)$, the ratios must be satisfied

$$y_1\left(a,x,\psi,\theta\right)_{|x=0}=y_1\left(a,x,\psi,\theta\right)_{|x=l}=0 \tag{9}$$

In addition, if we impose on a periodic by ψ and θ function a condition of absence in its representation by harmonics ψ the first component

$$\int_{0}^{2\pi} y_1\left(a,x,\psi,\theta\right) \begin{Bmatrix} \cos\psi \\ \sin\psi \end{Bmatrix} d\psi = 0 \tag{10}$$

then, the parameter will be considered the amplitude of the direct or reverse wave. The wave number κ is related to the frequency Ω of the dynamic process by the dispersion ratio

$$\Omega^2 - \alpha^2\kappa^4 + \omega^2 = 0 \tag{11}$$

The representation of the solution in the form (8), as well as the boundary conditions (9), allow determining the set of values of the wave number $\kappa_k = \dfrac{k\pi}{l}, k = 1, 2, ...$. Simultaneously, the obtained dispersion ratio determines the frequency Ω of the process as a function of a wave number in the form

$$\Omega = \sqrt{\frac{EI}{m+\rho}\kappa^4 - \omega^2} \; .$$

Considering the obtained values κ_k, the eigenfrequencies spectrum of the working body of the conveyor horizontal branch is obtained

$$\Omega_k = \sqrt{\frac{EI}{m+\rho}\left(\frac{k\pi}{l}\right)^4 - \omega^2} \tag{12}$$

Typically, in nonlinear systems, a dynamic process with a frequency close to the main frequency ($k = 1$) of the system frequency spectrum *(Mytropolskyi Y.A., Lymarchenko O.S. 1998)* is developed; therefore, the unperturbed motion of the considered conveyor branch is defined by the dependence

$$y(x,t) = a\left[\cos(\frac{\pi}{l}x + \sqrt{\frac{EI}{m+\rho}\left(\frac{\pi}{l}\right)^4 - \omega^2}\,t + \psi) - \cos(\frac{\pi}{l}x - \sqrt{\frac{EI}{m+\rho}\left(\frac{\pi}{l}\right)^4 - \omega^2}\,t - \psi)\right] \tag{13}$$

or

$$y(x,t) = 2a\sin\frac{\pi}{l}x\cos(\sqrt{\frac{EI}{m+\rho}\left(\frac{\pi}{l}\right)^4 - \omega^2}\,t + \psi) \tag{14}$$

The ratios (13) and (14) do not contradict each other; only for (14), the parameter a will be nothing more than a half of the vibrations amplitude.

In Fig. 2, the dependence of the internal vibrations frequency Ω_k of the feeder-mixer horizontal working body on the angular velocity of the working body rotation at different values of the grain mixture mass per unit length (the elastic modulus of the material - $E = 2,06 \cdot 10^{11} N/m^2$, $l = 6 \div 10\,m$, $m = 0 \div 40\,kg/m$, $\rho = 30\,kg/m$) is presented:

Obviously, the higher are the angular rotational velocities of the working body of the feeder-mixer horizontal branch, the lower is the actual frequency of its vibrations.

Moreover, the decay velocity from the angular rotational velocity is higher provided the values of the mass per unit length of the grain mixture are, For a non-perturbed case in (13) or (14), the parameters a and ϕ are stable; in perturbing motion larger they are variables in time, and the laws of their changes are determined by the right-hand side of equation (7) - the nonlinear-elastic properties of the auger screw material, the physical-mechanical and kinematic parameters of the grain mixture and the speed of its movement.
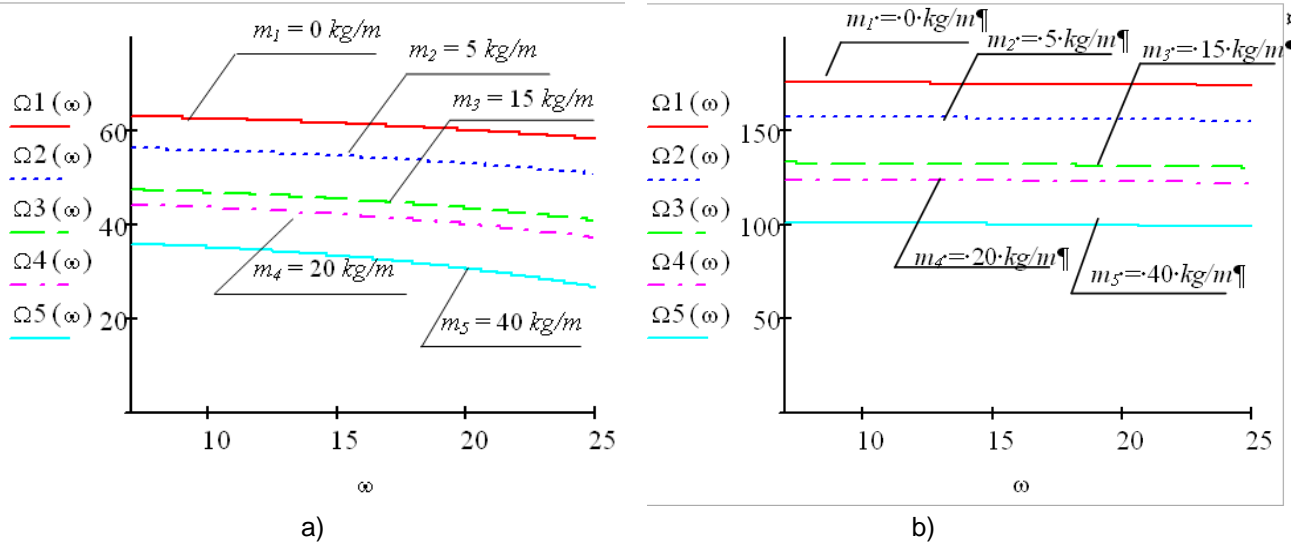
Fig. 2 - Dependence of the eigenfrequency on the working body length on the angular velocity of its rotation
a) $l = 10$ m; b) $l = 6$ m

To find the influence of these factors on the process dynamics, a function $y_1(a, x, \psi, \theta)$ in the asymptotic representation (8) should be developed. As the screw working body can be exposed to external periodic perturbation by frequency v, then the dangerous resonant vibrations may occur provided $pv \approx q\Omega_k$ (p and q are relatively simple numbers). This fact is a prerequisite for considering two cases of perturbed vibrations of the horizontal working body: resonant ($pv \approx q\Omega_k$) and non-resonant ($pv \neq q\Omega_k$). First, a simpler non-resonant case should be considered, for which, as shown in (*Mytropolskyi Y.A., Lymarchenko O.S., 1998; Hevko I.B., 2013; Oleg Lyashuk, 2016; Sokil M.B., and others, 2016, Sun X.X.; Meng W.J., Yuan Y., 2017*), the amplitude and frequency of the process must be determined by the ratio:

$$\frac{da}{dt} = \mu A_1(a) + ...,$$
(15)

$$\frac{d\psi}{dt} = \Omega + \varepsilon B_1(a) + ...,$$

The unknown functions $A_1(a)$ and $B_1(a)$ are found in the following way: the representation (15) should satisfy the original equation (7) and boundary conditions (6) with the accurate size of order μ inclusively. Therefore, by differentiating on independent variables x and t (7), the formula for the first approximation is deduced:

$$\frac{\partial y}{\partial t} = \mu A_1(a) \big(\cos(\kappa x + \psi) - \cos(\kappa x - \psi) \big) - a \big(\Omega + \varepsilon B_1(a) \big) \big(\sin(\kappa x + \psi) + \sin(\kappa x - \psi) \big)$$

$$+ \mu \left(\Omega \frac{\partial y_1(a, x, \psi, \theta)}{\partial \psi} + \frac{\partial y_1(a, x, \psi, \theta)}{\partial \theta} v \right),$$
(16)

$$\frac{\partial^2 y}{\partial t^2} = -a \omega_k^2 \big(\sin(\kappa x + \psi) + \sin(\kappa x - \psi) \big) -$$

$$-2\mu \big(A_1(a) + a B_1(a) \Omega \big) \big(\cos(\kappa x + \psi) - \cos(\kappa x - \psi) \big) +$$

$$+ \varepsilon \left(\frac{\partial^2 y_1}{\partial \psi^2} \omega^2 + \frac{\partial^2 y_1}{\partial \theta^2} v^2 + 2 \frac{\partial^2 y_1}{\partial \psi \partial \theta} \Omega v \right),$$
(17)

$$\frac{\partial y}{\partial x} = -a\kappa\left(\sin\left(\kappa x + \psi\right) - \sin\left(\kappa x - \psi\right)\right) + \mu\frac{\partial y_1}{\partial x},$$

$$\frac{\partial^2 y}{\partial x^2} = -a\kappa^2\left(\cos\left(\kappa x + \psi\right) - \cos\left(\kappa x - \psi\right)\right) + \mu\frac{\partial^2 y_1}{\partial x^2}$$

$$\frac{\partial^3 y}{\partial x^3} = a\kappa^3\left(\sin\left(\kappa x + \psi\right) + \sin\left(\kappa x - \psi\right)\right) + \varepsilon\frac{\partial^3 y_1}{\partial x^3},$$

$$\frac{\partial^4 y}{\partial x^4} = a\kappa^4\left(\cos\left(\kappa x + \psi\right) - \cos\left(\kappa x - \psi\right)\right) + \varepsilon\frac{\partial^4 y_1}{\partial x^4}.$$

(18)

Substituting in the base equation (7) the above dependences after the equalization of the coefficients at μ, a differential equation for determining the desired functions $A_1(a)$, $B_1(a)$ is deduced.

$$L\left(\frac{\partial^2 y_1}{\partial\psi^2}, \frac{\partial^2 y_1}{\partial\theta^2}, ..., \frac{\partial^4 y_1}{\partial x^4}\right) = \frac{\partial^2 y_1}{\partial\psi^2}\Omega^2 + \frac{\partial^2 y_1}{\partial\theta^2}v^2 + 2\frac{\partial^2 y_1}{\partial\psi\partial\theta}\Omega v + \alpha^2\frac{\partial^4 y_1}{\partial x^4} + \omega^2 y_1 =$$

$$= 4a\Omega A_1(a)\sin\frac{\pi x}{l}\sin\psi + 4a\Omega B_1(a)\sin\frac{\pi x}{l}\cos\psi - -u^2 a\left(\frac{\pi}{l}\right)^2\sin\frac{\pi x}{l}\cos\psi +$$

$$+2ua\Omega\frac{\pi}{l}\cos\frac{\pi x}{l}\sin\psi + F(a, \psi, x).$$

(19)

If a function $y_1(a, \psi, \vartheta, x)$ is periodic by an argument ψ, then its partial derivatives are the same by the noted argument. Consequently, the left-hand side of differential equation (19) does not contain the summands that are proportional to $\sin\psi$ and $\cos\psi$. This is the basis for finding the functions $A_1(a)$, $B_1(a)$

$$A_1(a) = -\frac{1}{p}\frac{1}{4\Omega\pi^2}\int_0^l\int_0^{2\pi}\int_0^{2\pi} F(a, x, \psi, \theta)\sin\frac{k\pi}{l}x\sin\psi\, dx\, d\psi\, d\theta,$$

$$B_1(a) = -\frac{1}{p}\frac{1}{4\Omega\pi^2 a}\int_0^l\int_0^{2\pi}\int_0^{2\pi} F(a, x, \psi, \theta)\sin\frac{k\pi}{l}x\cos\psi\, dx\, d\psi\, d\theta,$$

(20)

where $p = \dfrac{l}{2}$.

As the elastic body material satisfies the nonlinear technical law of elasticity (*Owen, Philip J., Cleary, Paul W., 2010; Sun X.X.; Meng W.J., Yuan Y., 2017*) then the resistance to the motion is proportional to the velocity, and the right-hand sides of the equation (17), that is, the functions $A_1(a)$, $B_1(a)$ acquire the form

$$\mu A_1(a) = -\frac{\overline{\delta}}{m+\rho}\left(\Omega\right)^{s-1}a^s,$$

$$\mu B_1(a) = -\frac{3\mu}{32}\frac{\pi^2}{l^2}\frac{a^2}{\Omega} - \left(\frac{\pi}{l}\right)^2\frac{m}{m+\rho}\frac{u^2}{8\Omega}.$$

(21)

RESULTS

In Fig. 3, the dependence in time of the amplitude and frequency of nonlinear vibrations of the feeder-mixer horizontal working body at its various geometric dimensions, the angular velocity of the working body rotation, the grain mixture mass per unit length, and the speed of its transportation are presented.
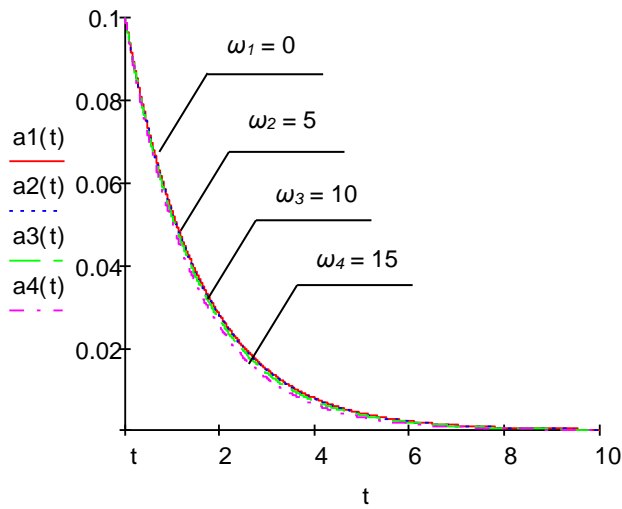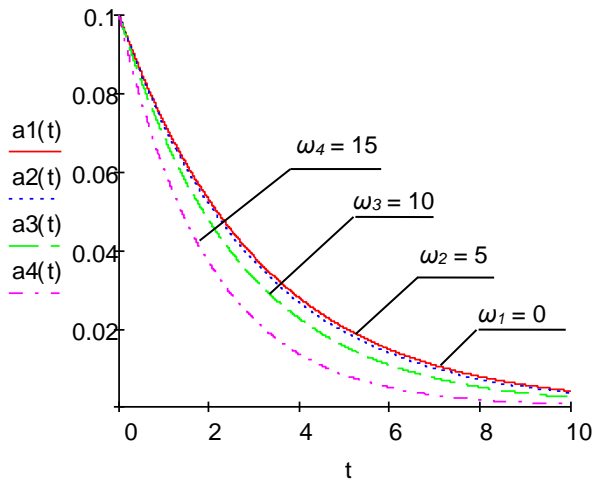
$$\rho = 10 kg/m, \, m = 0, kg/m \, I = 6 \cdot 10^{-6} m^2, \, E = 2.06 \cdot 10^{11} N/m^2, \, l = 10m, u = 10m/s$$
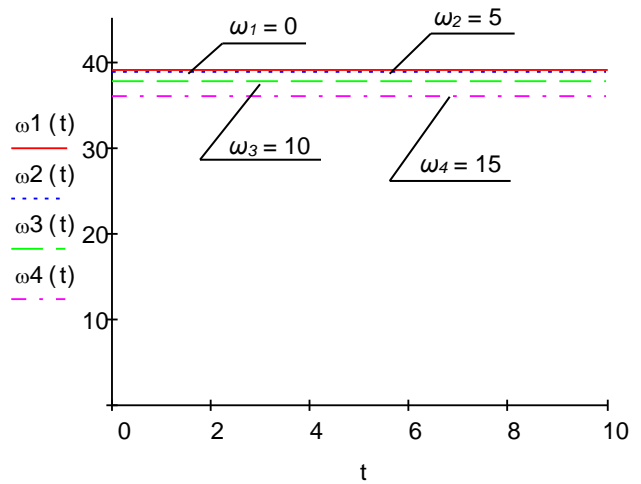
a)   *b)*

$$\rho = 10, kg/m, \, m = 30 kg/m, \, I = 6 \cdot 10^{-6} m^2, \, E = 2.06 \cdot 10^{11} N/m^2, l = 10m, \, u = 10m/s$$
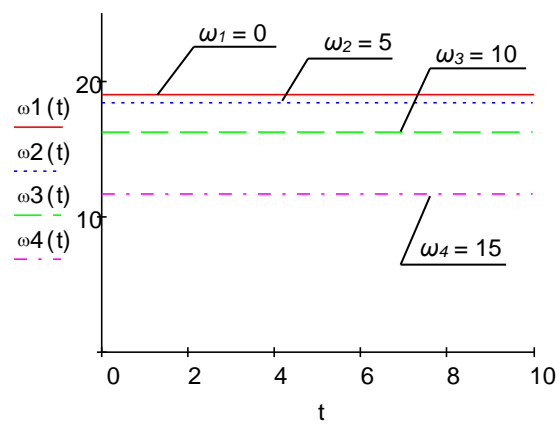
c)   *d)*
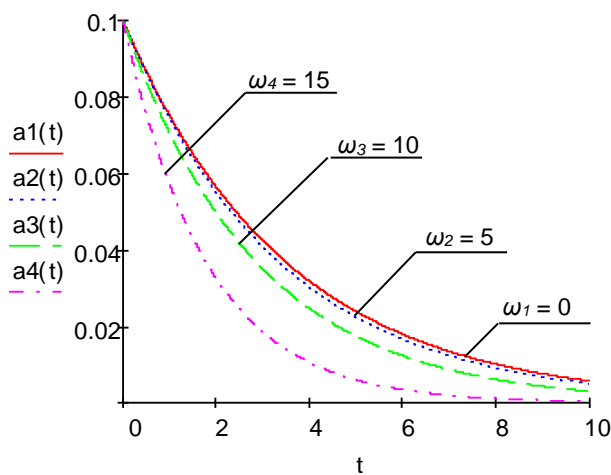
$$\rho = 10 kg/m, \, m = 40 kg/m, \, I = 6 \cdot 10^{-6} m^2, \, E = 2.06 \cdot 10^{11} N/m^2, l = 10m, \, u = 10m/s$$
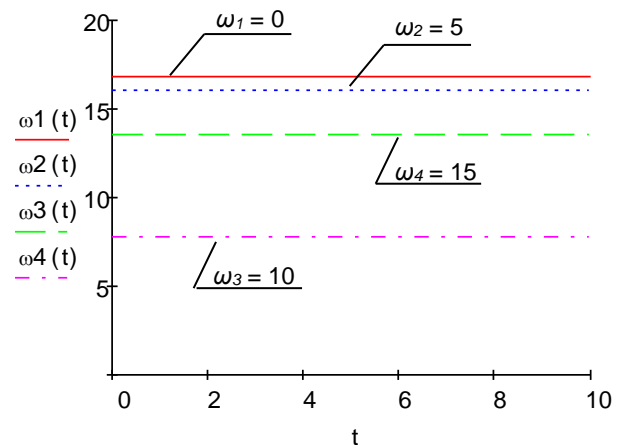
e)   *f)*
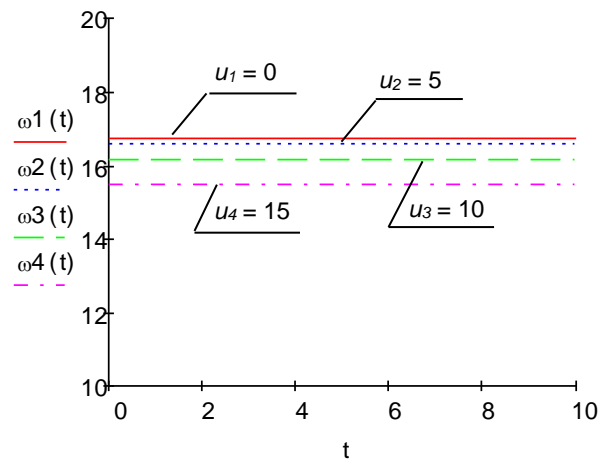
**Fig. 3 - Laws of variation in time of the amplitude and frequency of bending vibrations
of the feeder-mixer horizontal branch**
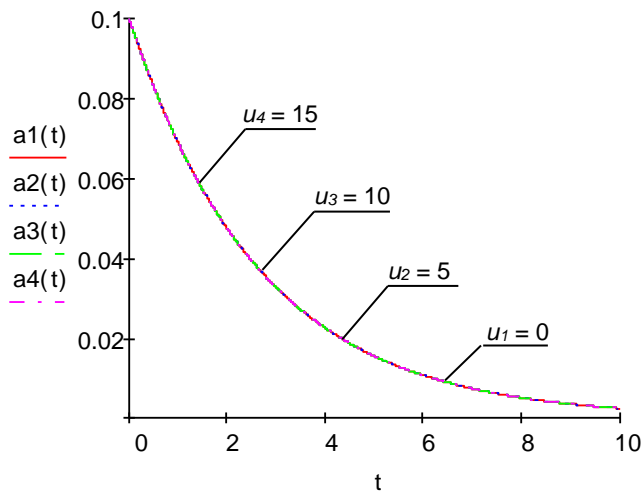
$$\rho=10kg/m, m=30kg/m, I=6{\cdot}10^{-6}m^2, E=2.06{\cdot}10^{11}N/m^2, \omega=10rad/s$$

g)   *h)*

$$\rho=10kg/m, u=7.5m/s, I=6{\cdot}10^{-6}\ m^2, E=2.06{\cdot}10^{11}N/m^2, \omega=10rad/s$$
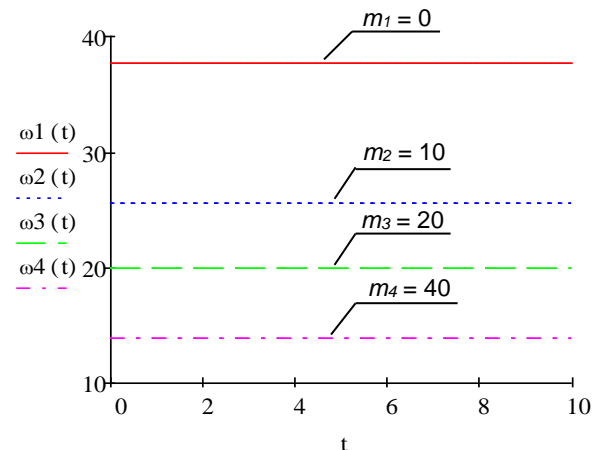
i)   *j)*

**Fig. 3 - Laws of variation in time of the amplitude and frequency
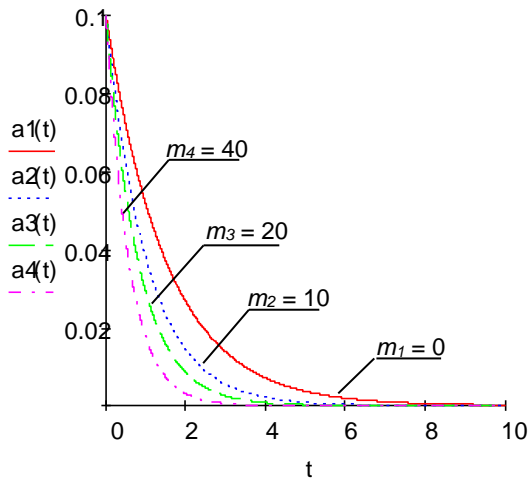of bending vibrations of the feeder-mixer horizontal branch**

CONCLUSIONS

Based on the analytical and graphical results, the following conclusions should be made:

- the higher are the angular rotational velocities of the feeder-mixer working body and the values of the grain mixture mass per unit length, the lower is the actual frequency of the horizontal branch with the grain mixture; at the same time, the rate of closing the amplitude is higher;

- if the velocities of the grain mixture motion along the feeder-mixture horizontal branch are higher, the actual frequency of vibrations is somewhat lower;

- the effect of the amplitude variation in time on the change in actual frequency of vibrations of the horizontal branch with the grain mixture is insignificant. The above can be justified by the fact that the mathematical model is considered with a small non-linearity.

- the results are very important in the study of more complex - resonant vibrations of the feeder-mixer working body.

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