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Resonant oscillation of vertical working part of conveyer-loader

The experimental equipment for testing the screw loader of bulk materials with horizontal and vertical branches was designed and manufactured, which allows to determine the process productivity and power consumption according to the developed methods. Experimental equipment is equipped with laboratory equipment, which ensures the change of investigated processes in wide ranges with high accuracy in the automated control mode with the fixation of necessary research results. On the basis of mathematical models, the dependences of the angular velocity of perturbation on the physico-mechanical and geometric parameters of the system of branches and the angular velocity of SWP are obtained. It was established that the resonance dynamic stresses at significant angular velocities exceed several times the resonant stresses of a «static elastic body» (which does not rotate), which makes it possible to take into account when choosing a dynamic coefficient of strength. With the same physical-mechanical and geometric characteristics of an elastic body, the resonance for larger values of the angular velocity of its rotation takes place for a smaller frequency of external periodic perturbation.

Key words: mathematical model, amplitude, resonance, conveyer.

Introduction

The technical means of continuous transportation of friable materials are the basis of the complex mechanization of loading and unloading operations, which increase the productivity and efficiency of production processes. The specific gravity of spiral conveyors in loading and unloading works of most construction, road, agricultural, processing and other machines, which is about 40–50 %.

It is known that resonant oscillations are the most dangerous modes of operation of machines and equipment. They are characterized by a significant increase in the amplitude of oscillations, and hence dynamic loads. Therefore, these operating modes greatly reduce the operating life of the machine. If, in some cases, resonance phenomena cannot be avoided, then, by choosing the system parameters, they try to provide a minimum amplitude increase during the transition through the resonance. Therefore, the question of resonance oscillation research is of vital importance for the improvement of their structures.

Analysis of recent research and publications. The interaction of auger mechanisms with agricultural materials, as well as the choice of rational parameters of GTTM and their modes of operation, were investigated by M.P. Vasylenko [1], B.M. Gevko [2, 3], P.M. Zaika [4], R.M. Rogatinsky [5, 6], I.M. Zuyev [7], V.L. Kulikovskiy [18] and others.

The problems of determining the rational operating modes and constructive parameters of the GTTM are devoted to the work of R.L. Zenkov [8–11], A.M. Grigoriev [12–14], B.M. Gevko [3], K.V. Alferov [8], H.A. Khailis [15], V.I. Plavinsky [16], N.V. Ostapchuk [17, 18], O.R. Rogatinskaya [19], I.M. Khorolskiy [20].

Questions of substantiation of the parameters of oscillations of mechanical systems are devoted to the works of I.M. Babakov [21], O.A. Goroshko [22], P.D. Dotsenko [23], M.P. Martyntsev [24], M.A. Pavlovsky [25], S.N. Nikiforov [26], Yu.A. Mitropolsky [27], M.M. Bogolyubov [28], and others. However, issues relating to the improvement of the structures of the spiral mechanisms themselves and their working parts need further research. Investigating the resonant oscillations of the vertical working part of the conveyer loader.

Material and method

For carrying out the experimental researches of productivity of spiral loader with a working part the stand [29] was used, the general view of which is depicted in Figure 1. Stand for the study of the overload of friable materials from a horizontal to a vertical axis, made in the form of a frame 1, a cylindrical horizontal spiral working part 2 with a central shaft 3 and a vertical groove 4 with a vertical spiral working part 5. A cylindrical and horizontal groove of spiral working part 2 is rigidly mounted on the stand 6 on the frame 1. Below there is a vertical groove 4 is a perpendicular to its axis a rotary connection sleeve 7 which, in a known manner, performs the angle of rotation of the vertical groove 4. From above, at the entrance to the cylindrical horizontal spiral working part 2, a bunker 8 with a friable material 9 is rigidly installed.

The stand is equipped with a driven actuator 10 with a cylindrical horizontal spiral working part 2 and a vertical spiral working part 5 through the pass transmissions 11 and 12 and on the additional T-shaped transmission gear 13. At this productivity of the vertical spiral working part 5 should be no less than the productivity of the cylindrical horizontal working part 2, in order not to jamming the technological process. At the top of the vertical groove 4, a loading window 14 with a tray under which the tank 15 is installed for collecting friable material from the vertical section is made. In addition, at the bottom of the cylindrical end the horizontal working part 2, the initial window 16 with a crossbar for measuring the productivity of the horizontal section was made. Under the outlet there is a container 17 for collecting friable material that the horizontal section moves.

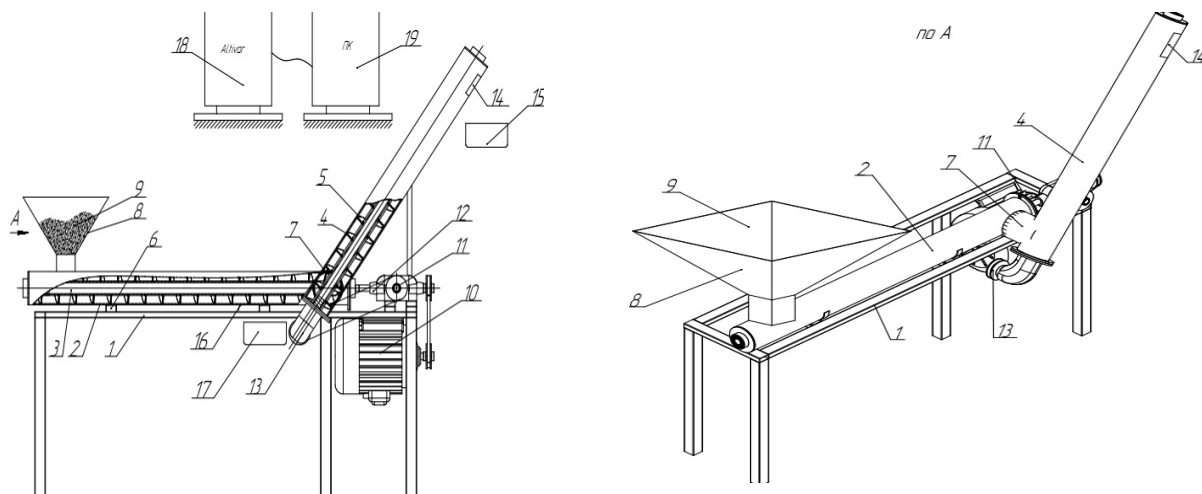


Figure 1. Stand to investigate the overload of friable materials from the horizontal to the vertical line

In addition, the steps of the cylindrical horizontal spiral working part are evenly enlarged in the direction of exit of material from it.

In addition, the stand is equipped with control devices Altivar 7.1 and a personal computer 19 for measuring power, kinematic and technological parameters.

The work of the stand to study the overload of friable materials from a horizontal to a vertical branch is carried out as follows. To study the performance of the horizontal section, open the switch under the capacity of 17, fix the time and determine the performance. To establish the performance of the horizontal and vertical sections, the shutter is closed on the horizontal section and all the friable material is transported through window 14 into a container 15 that weighs and performs the analysis.

The design of the test facility (Fig. 2) includes a spiral-mixer driven from a personal computer (PC) through a frequency converter (Altivar series) 3. The spiral mixer consists of a frame, with the possibility of axial rotation and change of the angle of inclination of the body, in which there is a working part driven by a

three-phase asynchronous electric motor (AIP90L4Y3), is located in relation to the horizon through the support. In the case the bunker and loading and unloading holes are fixed — exits, in which a recessed pipe is installed. The engine is equipped with a sensor for motor shaft rotation frequency (E40S6–10Z4–6L-5) 13.

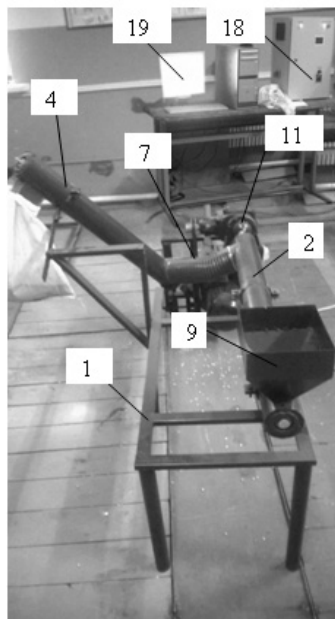


Figure 2. The general view of the spiral loader

It is known [14] that resonant modes of operation of machines and equipment are the most dangerous. They are characterized by a significant increase in the amplitude of oscillations, and hence dynamic loads. Therefore, these operating modes greatly reduce the operational terms of machines (we are not talking about special machines principle of operation of which is based on resonant phenomena). If, in some cases, it is not possible to avoid resonant phenomena, then try at the expense of choosing the system parameters to provide the minimum value (growth) of the amplitude during the transition through the resonance. Where it follows that the study of the resonant phenomena of the horizontal working body of the loader mixer has not only theoretical, but also practical value.

From the condition of existence of resonant oscillations, which for the case of the main bending working part can be written in the form:

$$\nu \approx \Omega - \frac{3\mu \pi^2 a^2}{32 l^2 \Omega} + \left(\frac{\pi}{l}\right)^2 \frac{m}{m + \rho} \frac{u^2}{8\Omega}. \quad (1)$$

Thus, the resonance phenomenon for different values of the velocity of the grain mixture, its linear mass, and the various angular velocities of the rotation of the horizontal working part will occur at different frequencies of external perturbation. As noted above, the real effect of small amplitudes of transverse vibrations of the horizontal box of the mixer loader on its own frequency is negligible. From the above, it follows that the dominant role of entering the resonance is played by: the angular velocity of rotation of the working part, the speed of the grain mixture and its mass. Thus, for different values of these parameters, the amplitude of the transition through the resonance will take different values. In addition, as shown in [14], the amplitude of passage through the resonance depends on the phase difference between the proper and the forced oscillations, in our case $\varphi = \psi - \theta$. The first resonance approximation for the above-described boundary-value problem will be sought in the form of an asymptotic representation, but in contrast to the nonresonance case, the amplitude in the transition through the resonance is determined by the relation of the form

$$\frac{da}{dt} = \mu A_1(a, \phi), \quad (2)$$

$$\frac{d\varphi}{dt} = \Omega - \nu + \mu B_1(a, \varphi), \quad \varphi = \psi - \theta.$$

The problem is to determine this type of functions, taking into account (1), (2), for the first approximation, satisfied the basic equation. Acting in the same way as for a nonresonant case, taking into account (3), we find

$$\begin{aligned} \frac{\partial y}{\partial t} &= \mu A_1(a, \phi) (\cos(\kappa x + \psi) - \cos(\kappa x - \psi)) - \\ &- a(\Omega + \mu B_1(a, \phi)) (\sin(\kappa x + \psi) + \sin(\kappa x - \psi)) + \mu \frac{\partial y}{\partial \theta} \nu + \mu \frac{\partial y}{\partial \psi} \Omega \\ \frac{\partial^2 y}{\partial t^2} &= \mu \frac{\partial A_1(a, \phi)}{\partial \phi} (\Omega - \nu) (\cos(\kappa x + \psi) - \cos(\kappa x - \psi)) - 2\mu \Omega A_1(a, \phi) (\sin(\kappa x + \psi) + \sin(\kappa x - \psi)) + \\ &+ \frac{\partial^2 y}{\partial \psi^2} \Omega^2 + \frac{\partial^2 y}{\partial \theta^2} \nu^2 + 2 \frac{\partial^2 y}{\partial \theta \partial \psi} \nu \Omega. \end{aligned} \quad (3)$$

The resulting ratio let you capture the differential equation that connects the unknown function as

$$\begin{aligned} L\left(\frac{\partial^2 y_1}{\partial \psi^2}, \frac{\partial^2 y_1}{\partial \theta^2}, \dots, \frac{\partial^4 y_1}{\partial x^4}\right) &= aV^2 \left(\frac{\pi}{l}\right)^2 \sin \frac{\pi}{l} x \cos \psi - \\ &- 2V \frac{\pi}{l} \cos \frac{\pi}{l} x \cos \psi + F(x, a, \psi, \theta) + \mu \sin \frac{\pi}{l} x \times \\ &\times \left(\cos \psi \left(-\frac{\partial A(a, \phi)}{\partial \phi} (\Omega - \nu) + 2a\Omega B \right) + \sin \psi \left(a \frac{\partial B(a, \phi)}{\partial \phi} (\Omega - \nu) + 2A(a, \phi) \Omega \right) \right). \end{aligned} \quad (4)$$

Thus, for a resonance case, the function must be a solution of the equation and satisfy homogeneous boundary conditions if represented as their form

$$y_1(x, a, \psi, \theta) = \sum \sin \frac{k\pi}{l} x Y_{1k}(a, \theta, \psi). \quad (5)$$

In this case, the coefficients of its decomposition $Y_{1k}(a, \theta, \psi)$ are bound by differential equations

a) for $k = 1$

$$\begin{aligned} \frac{\partial^2 Y_{11}}{\partial \psi^2} \omega^2 + 2 \frac{\partial Y_{11}}{\partial \psi \partial \theta} \nu \omega + \nu^2 \frac{\partial^2 Y_{11}}{\partial \theta^2} + (\alpha^2 \left(\frac{\pi}{l}\right)^4 + \omega^2) Y_{11} &= \\ = aV^2 \left(\frac{\pi^2}{2l}\right) \cos \psi + \frac{1}{p} \int_0^l \sin \frac{\pi}{l} x F(a, x, \theta, \psi) dx + \\ + \left(\cos \psi \left(-\frac{\partial A(a, \phi)}{\partial \phi} (\Omega - \nu) + 2a\Omega B \right) + \sin \psi \left(a \frac{\partial B(a, \phi)}{\partial \phi} (\Omega - \nu) + 2A(a, \phi) \Omega \right) \right). \end{aligned} \quad (6)$$

b) for $k \neq 1$

$$\begin{aligned} \frac{\partial^2 Y_{1k}}{\partial \psi^2} \omega^2 + 2 \frac{\partial Y_{1k}}{\partial \psi \partial \theta} \nu \omega + \nu^2 \frac{\partial^2 Y_{1k}}{\partial \theta^2} + (\alpha^2 \left(\frac{k\pi}{l}\right)^4 + \omega^2) Y_{1k} &= \\ = aV^2 \frac{(k\pi)^2}{2l} \cos \psi + \frac{1}{p} \int_0^l \sin \frac{k\pi}{l} x F(a, x, \theta, \psi) dx. \end{aligned} \quad (7)$$

In the same way as for a nonresonance case, the conditions for the absence of a function in the schedules $y_1(a, x, \psi, \theta)$, and hence in $Y_{1k}(a, \psi, \theta)$ ($k \neq 1$), the first harmonics ψ allow us to obtain relations that determine the right-hand side of the dependences (6)

$$(\Omega - \nu) \frac{\partial A}{\partial \phi} - 2a\Omega B = \frac{1}{p} \frac{1}{4\pi^2} \sum_s e^{is\phi} \int_0^l \int_0^{2\pi} F(a, x, \psi, \theta) \sin \frac{k\pi}{l} x e^{-is\phi} \cos \psi dx d\psi d\theta \quad (8)$$

$$a \frac{\partial B}{\partial \phi} (\Omega - \nu) - 2A\Omega + V^2 \frac{\pi^2}{l^2} = \frac{1}{p} \frac{1}{4\pi^2} \sum_s e^{is\phi} \int_0^l \int_0^{2\pi} F(a, x, \psi, \theta) \sin \frac{k\pi}{l} x e^{-is\phi} \cos \psi dx d\psi d\theta. \quad (9)$$

In the case when a spiral mixer operates a periodic perturbation which does not depend on its deflection, and small nonlinear forces are determined as for the case of their own oscillations, differential equations in the resonance region can look

$$\frac{da}{dt} = -\frac{\bar{\delta}}{m + \rho} (\Omega)^{s-1} a^s - \frac{2\mu H}{\pi(\Omega + \nu(t))} \cos \varphi \quad (10)$$

$$\frac{d\phi}{dt} = \Omega - \nu - \left(\frac{\pi}{l}\right)^2 \frac{mu^2}{8\Omega(m + \rho)} + \mu \left(\frac{2H}{\pi(\Omega + \nu(t))a} \sin \phi - \frac{3}{32} \frac{\pi^2 a^2}{l^2 \omega} \right).$$

Resonant phenomena of the working part, adversely affect the technological process of loading or moving the grain mixture, while reducing the life of the machine. To avoid it and to investigate the influence of a whole range of parameters on the process of its passage, it is necessary to construct for this case the solution of the mathematical model of the process

$$L_1(\xi, \zeta) = -\delta \left(\frac{\partial \xi}{\partial t}\right)^s - k_1 \frac{\partial^2}{\partial x_1^2} \left(\frac{\partial^2 \xi}{\partial x_1^2}\right)^3 + h \cos \theta, \quad (11)$$

$$L_2(\xi, \zeta) = -\delta \left(\frac{\partial \zeta}{\partial t}\right)^s - k_1 \frac{\partial^2}{\partial x_1^2} \left(\frac{\partial^2 \zeta}{\partial x_1^2}\right)^3 + h \cos \theta.$$

Provided:

$$p\mu \approx q\Omega_k \quad (12)$$

Similarly, as for the horizontal working part of the conveyor-loader, the differential equations describing the laws of variation of the amplitude of oscillations in the transition through the main resonance acquire the form

$$\begin{aligned} \frac{da}{dt} &= \frac{\mu}{4\pi l_1 \Omega} \left(\int_0^l \int_0^{2\pi} \bar{f}_1(a, x_1, \varphi + \theta, \theta) (\cos(\kappa x_1 + \varphi + \theta) - \cos(\kappa x_1 - \varphi + \theta)) d\theta dx_1 \right), \\ \frac{d\phi}{dt} &= \Delta - \frac{\varepsilon}{4\pi l_1 \Omega a} \left(\int_0^l \int_0^{2\pi} \bar{f}_1(a, x, \varphi + \theta, \theta) (\sin(\kappa x + \varphi + \theta) + \sin(\kappa x - \varphi + \theta)) d\theta dx_1 \right), \end{aligned} \quad (13)$$

where Δ — the imbalance between the frequencies of own and forced oscillations, thus $\Delta = \Omega - \mu$, and $\bar{f}_1(a, x_1, \varphi + \theta, \theta)$ correspond to the values of the right-hand sides of equations (11) under the condition that $\xi(x_1, \phi)$ and $\zeta(x_1, \phi)$ take the main values in the asymptotic representation of the solution.

Results

Figure 3 and Table represent the value of the amplitude of transverse oscillations during the transition through the main resonance for various numerical values of the parameters of the investigated system and the angular velocity of the rotation of the working part.

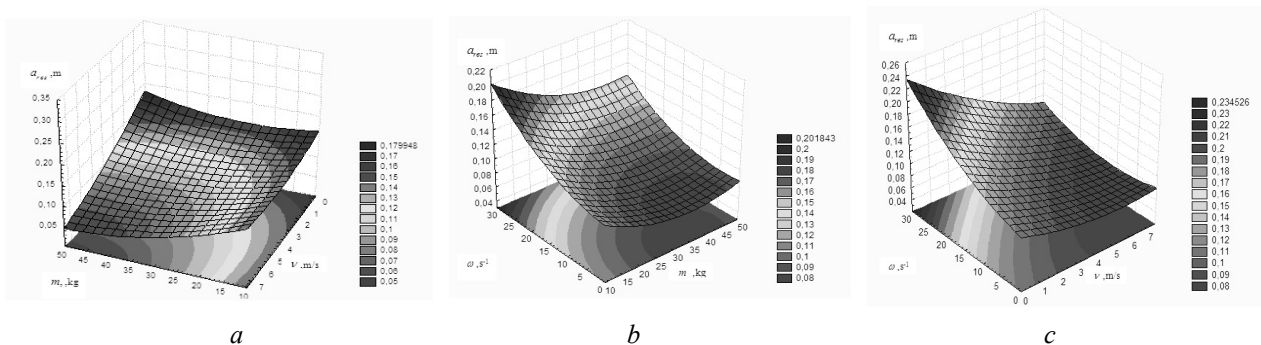


Figure 3. Graphic dependencies of the value of the growth factor of the amplitude when passing through the resonance due to the rotation of the working part

Different numerical values of the parameters of the investigated system and the angular velocity of the rotation of the working part represent the value of the amplitude of the transverse vibrations during the transition through the main resonance

m	ρ	l	w	Ω_{pez}	V = 0		V = 5 m/s		V = 7.5 m/s	
					a_{pez}	$\xi = \frac{a_{\text{pez} \omega=0}}{a_{\text{pez} \omega \neq 0}}$	a_{pez}	$\xi = \frac{a_{\text{pez} \omega=0}}{a_{\text{pez} \omega \neq 0}}$	a_{pez}	$\xi = \frac{a_{\text{pez} \omega=0}}{a_{\text{pez} \omega \neq 0}}$
kg/m	kg/m	m	s ⁻¹	s ⁻¹	m		m		m	
10	10	8	0	37.754	0.1121	–	0.1092	–	0.1023	–
10	10	8	10	27.754	0.1504	1.3427	0.1493	1.3672	0.1336	1.3056
10	10	8	15	22.754	0.1814	1.6184	0.1697	1.5540	0.1504	1.4702
10	10	8	20	17.754	0.2318	2.1240	0.2210	2.0238	0.1523	1.4438
20	10	8	0	31.285	0.1419	–	0.1291	–	0.1183	–
20	10	8	10	21.285	0.2211	1.5674	0.1813	1.4041	0.1451	1.2265
20	10	8	20	11.285	0.2381	1.6881	0.2232	1.7289	0.1961	1.6577
20	10	8	25	6.285	0.3306	2.3447	0.2987	2.3137	0.2510	2.127
10	10	6	0	64.134	0.0621	–	0.0601	–	0.0582	–
10	10	6	10	58.134	0.0731	1.1771	0.0681	1.1331	0.0622	1.0687
10	10	6	20	48.134	0.0884	1.4235	0.0841	1.3993	0.0763	1.3109
10	10	6	30	38.134	0.1101	1.7729	0.1036	1.7238	0.0943	1.6209
30	10	6	0	48.178	0.0938	–	0.0821	–	0.0634	–
30	10	6	10	38.178	0.1263	1.3475	0.1023	1.2434	0.0701	1.056
30	10	6	25	23.178	0.1984	2.1151	0.1381	1.6821	0.0893	1.4085
50	10	6	0	35.337	0.1436	–	0.0788	–	0.0411	–
50	10	6	10	29.337	0.1456	1.0139	0.0843	1.0698	0.0464	1.017
50	10	6	25	14.337	0.3061	2.1362	0.09210	1.1689	0.0493	1.023

Discussion

The obtained results allow asserting the following: the resonant amplitudes depend on the speed of the grain mixture, the angular speed of rotation of the working screw and the rate of change (in the resonant zone) of the frequency of the external periodic perturbation:

- for smaller values of the linear mass of the grain mix, the working screw and the higher speed of the transition through the main resonance, the value of the resonance amplitude is smaller;
- the growth rate of the amplitude in the transition through the resonance for the working part which rotates in comparison with the «stationary» its position is less for the case of more relative movement of the grain mixture.

Conclusion

On the basis of mathematical models, the dependences of the angular velocity of perturbation on the physico-mechanical and geometric parameters of the system of branches and the angular velocity of SWP are obtained. It is established that for larger values of the angular velocity of rotation of a working part, the resonant frequency value is smaller at $L=8$ m, $\Omega=17-40$ s⁻¹. For working parts of greater length, the amplitude of the transition through the resonance is greater than 10^{-25} s⁻¹ $L=8$ m and increases from 0,1121 to 0,2311 m. With an increase in the relative velocity of the transfer of the grain mixture leads to a decrease in the amplitude of the transition through the resonance at $m=50$ kg, the speed within 5...7,5 m/s decreases within 0,1023...0,0701 m. Design of screw loader designs, stand equipment and experimental installation using the Altivar 71 frequency converter and Power Suite v.2.5.0 software allowed to carry out a complex of experimental studies.

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Винттік жұктіегіштің қозғалмалы жұмыс бөлігінің резонанстық тербелісі

Шашылатын нысандар үшін қолданылатын жұктіегіштің, өндіріс үдерісін және энергия шығынын белгілі әдіспен анықтауға мүмкіндік беретін, жылжылмалы бөлігі дайындалған. Эксперимент жүзінде жасалған бөлік зерттелетін үдерістің өзгерісін автоматты түрде аса жоғары дәлдікпен бақылап отыруға мүмкіндік береді. Математикалық әдістер көмегімен қарастырып отырған құрылымның бұрыштық жылдамдығының жүйенің физика-механикалық және геометриялық параметрлеріне тәуелділігі анықталған. Жоғарғы бұрыштық жылдамдықта резонанстық динамикалық кернеу дененің

резонанстық статистикалық серпімділік кернеуінен бірнеше есе артып құрылғыға қауіп төндіруі мүмкін. Бұл жағдайды құрылғының динамикалық қауіпсіздігін есептегенде еске алу керек.

Кілт сөздер: математикалық модель, амплитуда, резонанс, транспортер.

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Резонансные колебания мобильного рабочего органа винтового загрузчика

Спроектировано и изготовлено экспериментальное оборудование для испытания винтового загрузчика сыпучих материалов с горизонтальной и вертикальной ветками, которое позволяет определять производительность процесса и энергозатраты согласно разработанных методик. Экспериментальное оборудование оснащено лабораторным оборудованием, обеспечивающим изменение исследуемых процессов в широких диапазонах с высокой точностью в автоматизированном режиме управления с фиксацией необходимых результатов исследования. На основе математических моделей получены зависимости угловой скорости возмущения от физико-механических и геометрических параметров системы веток и угловой скорости винтового рабочего органа. Установлено, что резонансные динамические напряжения при значительных угловых скоростях вращения в несколько раз превышают резонансные напряжения «статического упругого тела» (которое не вращается), что предопределяет учёт динамического коэффициента запаса прочности. При тех же физико-механических и геометрических характеристиках упругого тела резонанс при больших значениях угловой скорости его вращения имеет место для меньшей частоты внешнего периодического возмущения.

Ключевые слова: математическая модель, амплитуда, резонанс, транспортер.

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