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The Substantiation of the Parameters of the KOLOSNIKOV on Elastic Supports of the Cleaner of Fiber Material

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ABSTRACT: In the article the scheme of effective design of the bars on the elastic supports of the fibrous material cleaner is shown. The theoretical basis for calculating the grate parameters on an elastic support with nonlinear stiffness and random perturbation is considered. The results of tests of the recommended design of a cleaner with grates on elastic supports are given.

KEYWORDS: Cleaner, fibrous material, grate, elastic support, oscillation, stiffness, dissipation, amplitude, frequency, raw cotton, test, effect.

I.INTRODUCTION

To reduce the damage of cotton fibers and raw cottonseeds, it is advisable to reduce the multiplicity of interaction of working organs with cotton during the primary processing of cotton. At the same time, it is important to increase the efficiency of interaction between cotton and working bodies by improving their design. We recommend a new design of the grate of a cotton cleaner from a large litter [1].

Objects and methods of research. At the same time, the recommended grate design significantly reduces frictional resistance against the side surfaces with raw cotton. In addition, the resilient sleeves 4 are eccentric in the sidewalls 3 of the grate. In this case, the elasticity of the supports will in fact be nonlinear. According to the known technique, in [2] the elastic element can be represented as a conical spring with nonlinear rigidity (see Fig. 1b).

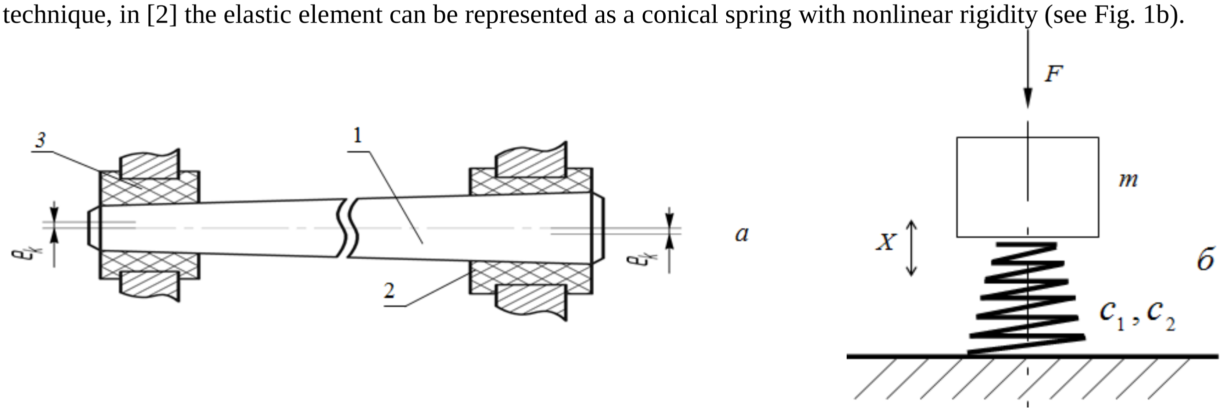

Figure 1. The scheme of conical grates on elastic supports and the design scheme where, a is a solid conical grate on elastic supports with nonlinear stiffness; b - calculated scheme of oscillations of a conical grate; 1- conical grates, 2-rubber sleeve with variable thickness, 3-shell (side, segment) grate

The elastic bushings 2 are also eccentric, have a variable thickness. It should be noted that the eccentricity position may change during operation (there are some circular motions). Therefore, the eccentricity $e_к$ and the difference in diameters of the conical grates 1 does not exceed (2,0÷3,0)10^{-3}м. $20 \cdot 10^{-3}$ $м$ is with an average grate diameter

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Results and discussion.

Fluctuation of the grate with nonlinear rigidity of the support. According to the design scheme (see Fig. 1 b), we will compose an equation describing the oscillation of the grate. It is known that the grate operates a random perturbing force from the side of the squeezed raw cotton

$$F_b = (F_b) \pm \delta(F_b) \quad (1)$$

It should be noted that the rigidity of the elastic support is non-linear and the restoring force is determined from expression

$$P = c_1 x_1 + c_2 x_1^3 \quad (2)$$

where, c_2, c_1 - the values of the stiffness coefficients of the elastic support;

x_1 - moving the grate in a vertical direction.

The oscillations of the grate are described by the following differential equation

$$m\ddot{x} + c_1 x + \frac{c_2}{\mu} x^3 = F_0 \sin \omega\, t \quad (3)$$

where, m - reduced mass of grate; μ - constant coefficient of nonlinearity; $F_0 \sin \omega t$ - the disturbing force from the raw cotton

We seek the solution of (3) by the Galerkin method [3] in the form

$$x_1 = x_0 \sin \omega t \quad (4)$$

Substituting expression (4) into the differential equation (3) and taking the integral equal to zero, we have

where, $2\pi/10$ - period of fluctuation.

After integrating, we obtain

$$\frac{3}{4}\frac{c_2}{\mu} x_0^3 + (c_1 - m\omega^2) x_0 - F_0 = 0 \quad (5)$$

In this case, the roots of equation (5) according to the known technique [73] will be:

$$x_1 = -2r\cos\frac{\varphi}{3}; \quad x_2 = 2r\cos\left(\frac{\pi \pm \varphi}{3}\right)$$

Where, $r = sign\alpha\sqrt{|\eta|}; \; \varphi = \arccos\frac{\alpha}{r^3}$

We can select the required amplitude and frequency of the non-linear oscillations of the grate using the recommended method for specific parameter values. Consider the well-known method for solving the problem [3].

Equation (3) can be rewritten in the form

$$m\ddot{x} + m\omega^2 x = (m\omega^2 - c_1)x - \frac{c_2}{\mu} x^3 + F_0 \sin \omega t \quad (6)$$

Using the Duffing method [4], we obtain a solution as a first approximation

$$x_1 = x_0 \sin \omega t \quad (7)$$

Substituting x_1 in the first part of equation (6), we obtain the equation for calculating the second approximation:

$$m\ddot{x}_2 + m\omega^2 x = \left[\left(m\omega^2 - c_1\right)x_0 - \frac{3}{4}\frac{c_2}{\mu} x_0^3 + F_0\right]\sin \omega t + \frac{1}{4}\frac{c_2}{\mu} x_0^3 \sin 3\omega t \quad (8)$$

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We are only interested in periodic oscillations of the grate, then to exclude the secular term, condition

$$(m\omega^2 - c_1)x_0 - \frac{3}{4}\frac{c_2}{\mu}x_0^3 + F_0 = 0 \tag{9}$$

Then we can obtain the second approximation, as solutions of the differential equation:

$$m\ddot{x} + m\omega^2 x = \frac{1}{4}\frac{c_2}{\mu}x_0^3 \sin 3\omega t \tag{10}$$

The solution of the differential equation (10) is

$$x = A\sin\omega t + B\cos\omega t - \frac{c_1 x_0^3}{32\omega^2 m}\sin 3\omega t$$

The constants of integration are determined from the initial conditions:

wherein $t = \frac{T}{4}$; $T = \frac{2\pi}{\omega}$; $x = 0$; $\dot{x} = 0$

$$A = x_0 - \frac{c_2 x_0^3}{32\omega^2 m\mu};\ B = 0 \tag{11}$$

Finally, the approximate solution has the form

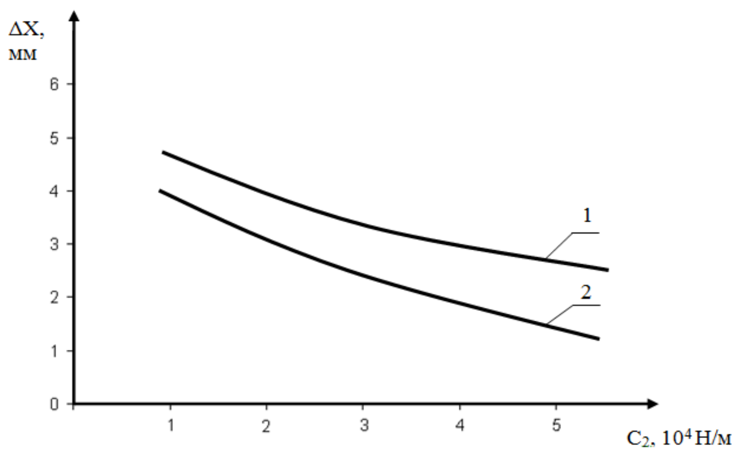
$$x = x_0 \sin\omega t - \frac{c_2 x_0^3}{32\omega^2 m\mu}(\sin\omega t - \sin 3\omega t) \tag{12}$$

In this case, the value of x_0 is determined from equation (9). Taking into account the initial values of the system parameters, the regularities of the oscillatory motion of the bars on elastic supports with nonlinear rigidity were obtained. Based on the processing of the obtained results, graphical dependences of the swing width of the grate oscillations are constructed with the variation of the average value of the rigidity of the elastic support, the mass of the grate at $\omega = 65c^{-1}$ и $\omega = 40c^{-1}$ (see Fig. 2 and Fig. 3).

Analysis of the graphs shows that with increasing rigidity of the elastic support, Δx decreases by a nonlinear law, and with decreasing the ω decrease Δx becomes more intense (see Fig. 2).

With an increase in the mass of the grate, the influence ω on the decrease in Δx becomes insignificant. This is explained by the fact that with a large mass of the grate its inertia increases and the value of Δx tends to a constant value (2.0-2.4 mm). It is at these values of Δx that the cleaning effect becomes tangible, which is confirmed by the results of experiments [5].

Thus, using the proposed method, it is possible to substantiate the necessary parameters of the system, which ensure an increase in the cleaning effect of cotton cleaners, from large litter.



where, 1- at $\omega = 65s^{-1}$ 2- at $\omega = 40s^{-1}$, C_1=2,5·10^4 N/м; m=4,0 Ns^2/м

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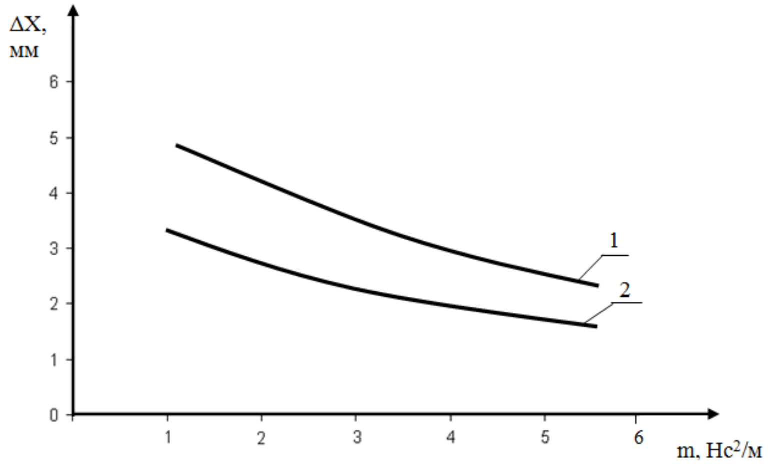
Figure 2. Dependence of the swing of the grate oscillation as a function of the rigidity of the elastic support of the cotton cleaner



where, 1-at $\omega = 65s^{-1}$ 2- at $\omega = 40s^{-1}$

Figure 3. Dependence of the swing of the grate oscillation depending on its mass.

Effect of grate parameters on the oscillation frequency. For an approximate determination of the deformation value of the grate supports, let us consider the kinetic energy of the raw cotton to be pulled together with the grate in the process of impact, transforming into the potential energy of the deformable support:

$$T = \frac{mV_y^2}{2} \qquad \Pi = \int_0^{x_{max}} \left(c_1 x + c_2 x^3\right) dx \qquad (13)$$

where: *T*- kinetic energy of raw cotton and grate; *m*- otal mass of grate and raw cotton; V_y - speed of impact of raw cotton on the grate;

$c_1 -$ linear component of the elasticity stiffness coefficient; $c = \frac{c_2}{\mu}$ - nonlinear component of the stiffness coefficient;

П- potential energy of a deformable elastic support;

From the accepted interaction condition:

$$V_y = \sqrt{\frac{2}{m}\int_0^a c_1 x dx + \int_0^a \frac{c_2}{\mu} x^3 dx} \qquad (14)$$

where, *a*- maximum value of deformation.

According to the study [6], with the nonlinear rigidity of the elastic element of a single-mass oscillatory system under conditions from $x = 0$ to $x = a$ oscillations is,

$$t = 4\sqrt{\frac{n}{\alpha}} \cdot \frac{1}{\alpha^{n-1}} \int_0^1 \frac{d\xi}{\sqrt{1-\xi^{2n}}} \qquad (15)$$

where, α and n – permanent, $n - 1,2...,; \xi = X / a$, with a restoring force equal to αx^{2n-1} .

In vibrational systems with nonlinear rigidity of the elastic element, there is a definite relationship between the period and the amplitude. For this reason, the term "natural frequency" is also avoided for such systems, since the frequency of free oscillations ceases to be an intrinsic parameter of the system. In this case, the recovering force

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$c_1 x + \frac{c_2}{\mu} x^3$ and therefore in (15) the n value takes the values 1 and 2. Then the period of oscillation of the grate on the elastic support with nonlinear rigidity is determined from the expression:

$$t = 4\sqrt{m}\left[\sqrt{\frac{1}{c_1}}\int_0^1 \frac{d\xi}{\sqrt{1-\xi^2}} + \sqrt{\frac{2\mu}{c_2 a^2}}\int_0^1 \frac{d\xi}{\sqrt{1-\xi^4}}\right] \tag{16}$$

where, µ - coefficient taking into account the nonlinearity of the elastic characteristic, м^2.

In the expression (15) obtained, we integrate the terms in parenthesis, while the second term is computed (integrated) by means of tables of special functions according to [5] and we obtain

$$t_k = 4\sqrt{m}\left[6{,}28\sqrt{\frac{1}{c_1}} + \frac{1{,}8541}{\alpha\sqrt{c_2^2/\mu}}\right] \tag{17}$$

For the frequency of free oscillations, taking $\rho_2 = 2\pi / T$ we have:

$$\rho_k = \frac{0{,}25a\sqrt{c_1 c_2/\mu}}{\sqrt{m}(2\pi\alpha\sqrt{c_2/\mu} + 1{,}85\sqrt{c_1}} \tag{18}$$

An analysis of the derived formula (18) shows that the natural oscillation frequency does not decrease linearly with the growth of the reduced mass of the grate with cotton.

With increasing amplitude of oscillations and stiffness coefficients c_1 and c_3, the frequency of natural oscillations changes in a nonlinear relationship. It is important to ensure that the grate oscillates in up to the resonance zone, since the natural frequency of the grate oscillation also varies depending on the values of the reduced mass, amplitude, and the nonlinear rigid characteristic of the elastic support. At numerical calculations for initial parameters are accepted:

$$m = 3{,}8 - 4{,}2\text{кг};\ c_1 = 2{,}5\cdot 10^4\ H/\text{м};\ c_2 = 1{,}2\cdot 10^{-4}\ H/\text{м};$$
$$\mu = (0{,}5 - 1{,}0)\text{м}^2;\ a = (1{,}0 - 1{,}2)\cdot 10^{-3}\ \text{м}$$

Figure 4 shows graphical dependences of the change in the relative value of the natural frequency of the grate oscillations on the increase of its reduced mass. Analysis of the graphs shows that the relative value of the natural frequency with increasing the reduced mass of the grate decreases according to a nonlinear regularity. The magnitude of the amplitude (deformation of the elastic support) does not actually affect this regularity, that is, an increase in the amplitude leads to a parallel upward shift of the regularity curve with a difference $\rho_k / \rho_н = 0{,}25 - 0{,}05$ (with an increase of a from $0{,}8\cdot 10^{-3}\ \text{м}$ до $1{,}2\cdot 10^{-3}\ \text{м}$).

Figure 5 shows the graphs of the change in the relative period of the grate oscillation on an elastic support with a nonlinear characteristic from the change in the amplitude of the natural oscillations. So with the value of the amplitude $0{,}5\cdot 10^{-3}\ \text{м}$ period of fluctuation $t_k / t_n = 1{,}9$, at $m_k = 4{,}5\text{кг}$, a at $a = 1{,}75\cdot 10^{-3}\ \text{м}$ и $m_k = 4{,}1\text{кг}$, period of fluctuation $t_k / t_n = 1{,}49$. This means that the amplitude of the grate oscillation slightly affects the period and frequency of oscillation. In this case, the mass of the conical grate includes the average mass of cotton (fly), on average, on the grate surface.

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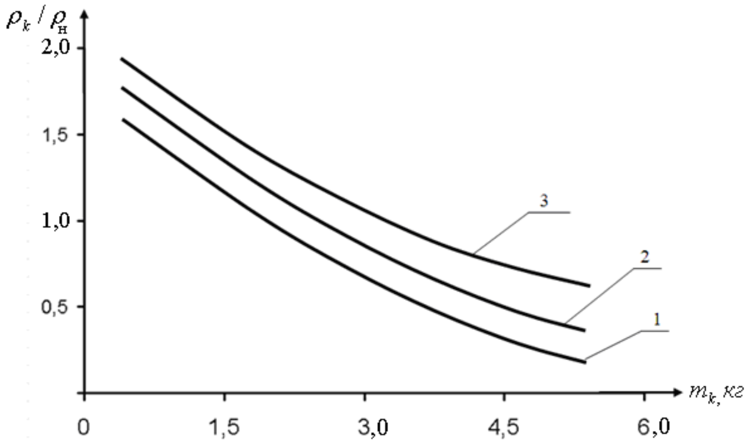
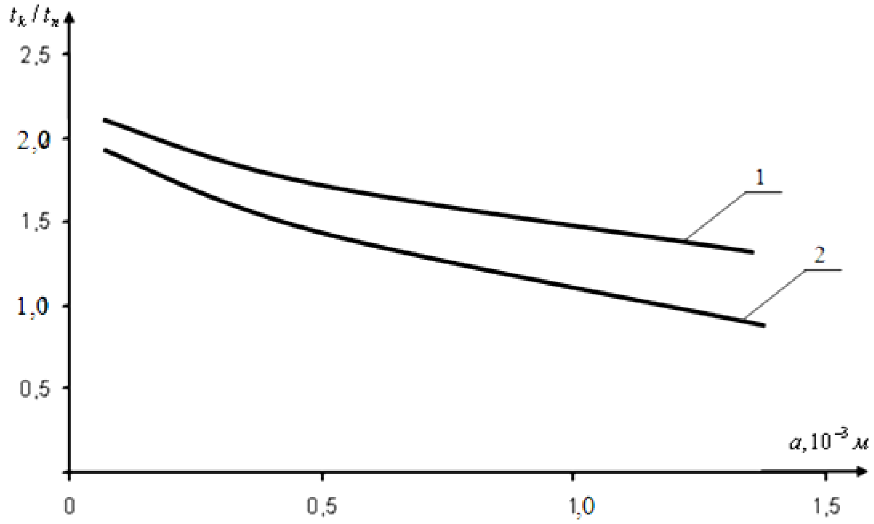


where, 1-at $a = 0{,}8 \cdot 10^{-3}$ м; 2-at $a = 1{,}0 \cdot 10^{-3}$ м; 3-at $a = 1{,}2 \cdot 10^{-3}$ м

Figure 4. Graphical dependencies changed relative values the natural frequency of the grate oscillations from the increase in its reduced mass.



where, 1 - m_k= 4,5 kg; 2 - m_k= 4,1kg

Figure 5. Graphical dependencies of the change in the relative value of the grating oscillation period from the change in the maximum amplitude.

It should be noted that the very amplitude of the natural oscillations depends on the magnitude of the deformations of the elastic support, that is, from its stiffness characteristic. Studies have shown that an increase in the rigidity of the elastic support leads to an increase in the natural frequency of the system by a nonlinear regularity. The nonlinearity of the elastic support depends on the location of the eccentric rubber bushing through which the conical grates are mounted in the casing of the cotton cleaner from the large litter.

Vibrations of the grate on elastic supports with nonlinear rigidity at random resistance from raw cotton.

In the process of cleaning the cotton (or cleaning zone) from a large litter with a sawtooth drum, captured volutes are dragged through the grate bars. In this case, each grate is cyclically interacted with the volatilization of raw cotton. That is, the loads on the part of the volutes are a disturbing force of forced oscillations of conical bars on elastic supports with nonlinear stiffness (variable thickness of the rubber bushing).

Taking into account the random function of the disturbing force from raw cotton, the nonlinearity of the restoring force of the elastic support, its dissipative characteristics, taking into account the works [6, 7, 8], one can write the equation of oscillatory motion of a conical grate in the form:

$$m\frac{d^2x}{dt^2} + в\frac{dx}{dt} + c_1 x + \frac{c_2}{\mu}x^3 = M(F_в) \pm \delta(F_в) \qquad (19)$$

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where, в is the coefficient of internal resistance of the elastic support of the grate.

Solution (19) presents a certain difficulty by analytical methods. The solution can be carried out by approximate methods. To carry out the computer experiment, the problem is solved numerically on a PC using standard programs. The following calculated values of the parameters were taken into account:

$$m = 4{,}0 Hc^2 / м;\ c_1 = 2{,}5\cdot 10^4 H/м;\ c_2 = 0{,}12\cdot 10^4 H/м;\ в = 60\ Hc/м;\ \mu = 1{,}0 м^2;$$

$$M(F_k) = 19{,}67 + 0{,}98\sin(x + 55^0 12') + 7{,}83\sin(2x + 112^0 14') +$$
$$+ 1{,}8\sin(3x + 103^0 23') + 3{,}37\sin(4x + 4^0 39') +$$
$$+ 6{,}96\sin(5x + 93^0 24') + 2{,}7\cos 6x$$

From the analysis of experimental data and processing by mathematical statistics it was determined the mathematical expectation of the perturbation force from cotton to grate and its possible variations in both frequency and amplitude. As a result of the implementation of the mathematical model of the vibrating system of the grate of a cotton cleaner from a large litter on a PC with the variations in parameters, graphic dependencies were obtained.

Figure 6 shows the fragment of displacement, speed and acceleration of a conical grate on an elastic support with a nonlinear restoring force at $m = 3{,}0 Hc^2 / м\ и\ c_1 = 2{,}5\cdot 10^4 H/м$, $c_2 = 1{,}2\cdot 10^4 H/м$, $M(F_в) = 12{,}5 H$, $\delta F_в = (0{,}8 \div 1{,}1)H$. It should be noted that the frequency of the grate oscillation is (40…55) Gts. In this case, the high-frequency component of the grate oscillation is (147÷178) Gts.

The low-frequency component of the frequency of the forced oscillations corresponds to the rotation frequency of the saw cylinder of the UCC unit, and the high-frequency component corresponds to the number of bars in the section. From Fig. 2.8 it can be seen that under forced oscillations of a conical grate the grate deviates by an average of $X_{cp} = (1{,}4 \div 1{,}6)\cdot 10^{-3} м,$ and the swing of the oscillations at the calculated values of the parameters is $\Delta X = (1{,}8 \div 2{,}1)10^{-3} м.$

For cylindrical bars on elastic supports according to [9], the swing range is $\Delta X = (2{,}2 \div 2{,}5)\cdot 10^{-3} м.$ Comparison of the results shows that in the proposed construction of a conical grate on (20÷25)%, the amplitude of the oscillations decreases due to the nonlinear rigid characteristic of the elastic support.

Similarly, the values $\dot{X}$ and $\ddot{X}$. The range of the velocity oscillations reaches from 0.6 m / s to 1.25 m / s, and the amplitude of the oscillations of the accelerations with the calculated parameters of the system varies within (6,5÷10) m/s^2. The frequency of vibration of speed and acceleration corresponds to the high-frequency component of the technological load from cotton. Figure 7 shows the graphs of the variation in the range of displacements, velocity, and acceleration from the increase in the mass of the grid grate. It is known that as the mass of the oscillatory system increases, a large force is required for its perturbation, then, as the mass increases, the amplitude of the oscillations of the conical grate decreases. It is especially important that the intensity of the decrease in the amplitude ΔX, $\Delta\dot{X}$ and $\Delta\ddot{X}$ of the oscillations decrease with increasing mass. This is due to the nonlinear rigid characteristic of the elastic support. With increasing load on the grate, the intensity of deformation of the elastic support decreases, which leads to a decrease in the amplitude of the oscillations of the grate (see Fig. 7).

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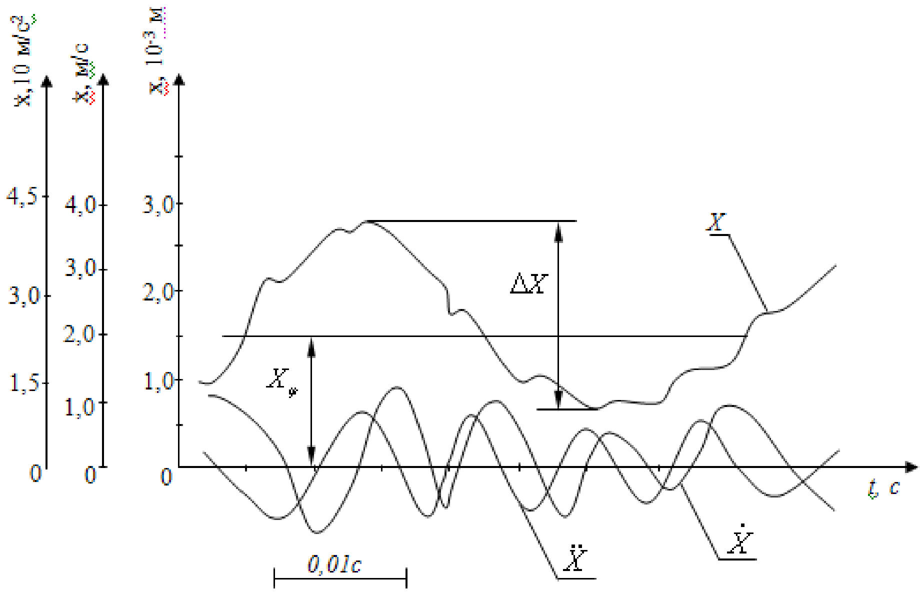
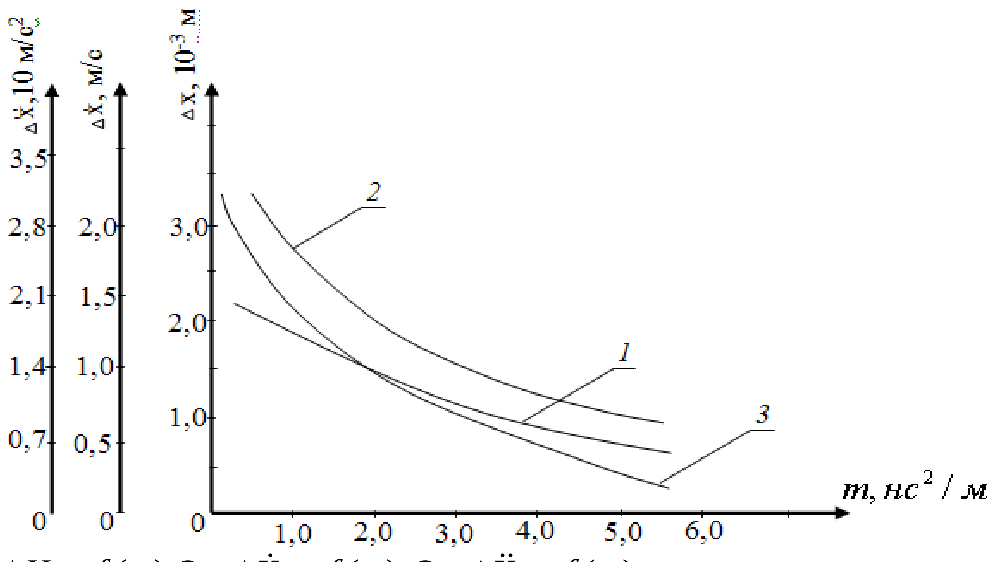


at $m = 3{,}0 нс^2/м$; $c_1 = 2{,}5\cdot 10^4 н/м$ $c_2 = 1{,}2\cdot 10^4 н/м$ $M(F_в) = 12{,}5$ H $\delta F_в = (0{,}8 \div 1{,}1)$ H

Figure 6. Changes in the displacement, speed, acceleration of the conical grate on an elastic support with random perturbation.



where, $1 - \Delta X = f(m); 2 - \Delta \dot{X} = f(m); 3 - \Delta \ddot{X} = f(m);$

at $c_1 = 2{,}5\cdot 10^4 H/м;\ c_2 = 1{,}2\cdot 10^4 H/м;$ deviations $\delta\Delta X, \delta\Delta\dot{X}, \delta\Delta\ddot{X} = (10 \div 12)\%$

Figure 7. Graphic dependence of the change in the swing amplitude of the displacement, velocity, and acceleration of the conical grate on the change in the mass of the grate.

Figure 8 shows the graphical dependencies of the displacement, speed, and acceleration of conical bars on elastic supports with nonlinear stiffness when the load is varied from raw cotton. With increasing resistance from cotton from 19,7H to 60H (average value), the movement of the grate increases from 0,65 $\cdot 10^{-3} м$ to $3{,}2 \cdot 10^{-3} м$. At the same time, the speed of oscillations increases according to a nonlinear regularity up to 2,45 м/с, and the acceleration rises to 21м/с^2. In this case, the deviations δx, $\delta\dot{x}$ and $\delta\ddot{x}$ depending on the random component of the

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load are within (8,0÷10)%. To prevent the loss of volatility between the bars due to the large amplitude of the oscillations of the bars and the reduction of the technological gap between the saw drum and the grate bars, the amplitude of the conical grates according to the results of the experiments should not exceed $(0{,}8 \div 1{,}2)\cdot 10^{-3}\,м$. In this case, deviations $\delta x, \delta \dot{x}, \delta \ddot{x} = (8{,}0 \div 10)\%$ $c_1 = 2{,}5\cdot 10^4\, н/м$ $c_2 = 1{,}5\cdot 10^4\, н/м$

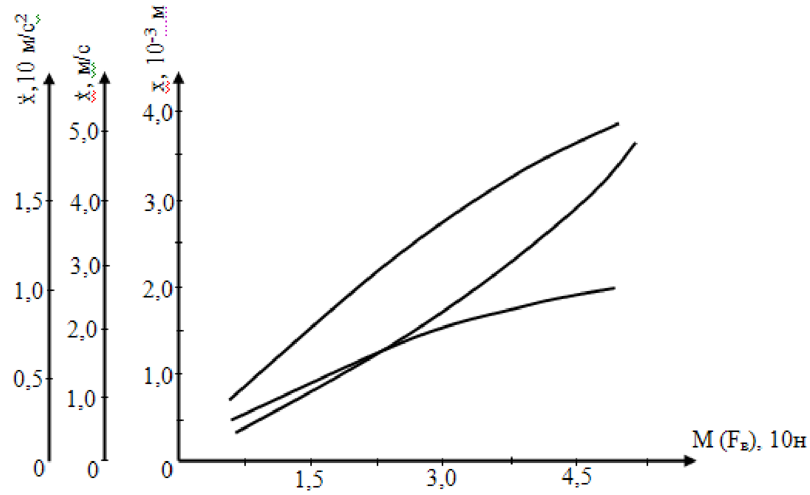

Figure 8. Dependences of changes in displacement, velocity and acceleration of conical grates in the function of resistance against cotton.

Therefore, to ensure the necessary vibration amplitudes of conical bars, it is advisable to choose resistance from raw cotton in the range (25÷35) *H*, which correspond (5,0÷7,0) T / h in the cotton cleaning machine UHK.

The results of comparative tests of the fibrous material cleaner with the recommended grate. Based on the results of the full-factor experiment, the following optimal values for the parameters of the large-scale cleaning zone were recommended: the speed of the serrated drum - 300 min^{-1};

cone-shaped grate -0,015; the rigidity of the elastic support (stamp rubber)– HO – 68 ($c_1 = 3{,}0\cdot 10^4\, H/м$; $c_2 = 1{,}6\cdot 10^4\, H/м$.) [11]. Comparative tests were carried out in production conditions during testing. The recommended construction of a grate with conical grates on elastic supports showed high reliability and stability of operation. The test results showed that the cleaning effect in comparison with the existing version of the grate increases in the average by 8.11%, The mechanical damage to seeds is reduced by 1,09%, The mechanical damage to seeds is reduced by 0,113%. This is explained by the fact that when raw cotton is interacted with a vibration-proof conical grate, raw cotton is additionally shaken, increased

Table 1
Results of comparative production tests

Indicators at %	**After the cleaner with an experienced grate on elastic supports in the 1st line of the UHK**	**After the cleaner with the serial grate in the 2nd UHK lines**
Original cotton cotton		
Humidity	8,7	8,7
Weediness	4,2	4,2
After cleaning, the cleaning effect		
Weediness	67,95	59,84
raw cotton		
mechanical damage to seeds	1,41	1,83
free fiber	2,07	3,16
	0,107	0,22

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Conclusions. Vibrating grates are recommended on the elastic supports of the fibrous material cleaner. On the basis of theoretical studies, regularities of the grate oscillation are obtained, graphic dependences of parameters are constructed, the best parameters are proved on the basis of their analysis. Experimental studies have justified the effectiveness of using the recommended grate.

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