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DEVELOPMENT AND CALCULATION OF A NEW DESIGN OF A GRITT ON ELASTIC SUPPORTS FOR A RAW COTTON CLEANER

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ABSTRACT

The article is devoted to the compilation of differential equations of forced vibrations of the grate rod on elastic supports, the theoretical solution of strength issues, the development of proposals for determining the optimal parameters of the grate and reducing the metal consumption of the machine.

Keywords: cotton fiber, grate, tubular rod, elastic support, forced vibration, beam, technological distributed load, saw-toothed drum, forced vibration, cotton cleaner from large debris.

In the world market, cotton fiber is the main product of the textile industry. According to the International Advisory Committee (ICAC), "in 2015-2016, countries such as Bangladesh, Vietnam, China, Turkey, Indonesia, and Pakistan were leading in the import of cotton fiber." During the years of independence, extensive measures were taken in the republic regarding the creation of management systems leading to an improvement in the consumer properties of cotton products in the technological process of primary processing. In these areas, in particular, in cotton ginning enterprises, depending on the primary indicators of obtaining a cotton product with the intended qualities, in improving the technique and technology of cleaning cotton from small debris, tangible results have been achieved. The strategy for the movement of the Republic of Uzbekistan in 2017-2021 includes the planned tasks of increasing the competitiveness of the national economy, reducing energy and resource consumption, and widespread introduction of energy-saving technologies in production. In the implementation of these tasks, the creation of new technologies for the primary processing of cotton, in particular the improvement of equipment and technology for cleaning from small debris and implementation in production, is one of the important tasks of the cotton industry.

To implement the tasks set in the Decree of the President of the Republic of Uzbekistan dated February 7, 2017 number PF-4947 "Strategy of action for the development of the Republic of Uzbekistan", In decisions of the Cabinet of Ministers of the Republic of Uzbekistan dated April 3, 2017 number 70, "Program for the modernization and reconstruction of cotton ginning industry enterprises", 22 December 2016 PC 2692, "Accelerated renewal of physically worn out and obsolete machines and equipment in industrial enterprises, as well as additional measures to reduce production costs" and in other lists of legal documents, this dissertation research to a limited extent deserves attention [1].

In recent years, the requirements for the quality of cotton - fiber and seeds - have increased significantly. To obtain high quality cotton fiber and seeds, the issues of improving the design of the working parts of cotton primary processing machines, especially cotton purifiers from large impurities, are being comprehensively addressed. [2].

Our industry produces various grate - saw blade cleaners. The grate plays an important role in creating the cleaning effect. Trash impurities fall out through the gaps between the grates under the influence of centrifugal force and air flow [3].

The existing grate consists of five tubular rods, two large and three small segments, connected in a certain way by welding and forms a rigid structure. Based on a generalization of grate designs and types of their cleaning by domestic technological machines, as well as foreign countries, a new grate design was developed, oscillating due to elastic-yielding supports from the action of raw cotton (A.S. No. 874776).

This work is the basis for drawing up differential equations for forced vibrations of the grate rod on elastic supports, theoretically solving issues of strength, developing proposals for determining the optimal parameters of the grate and reducing the metal consumption of the machine.

Experimental studies have shown that the prevailing dynamic load from technological resistance on the rods is the load in the radial direction of the saw drum. Processing the results of experimental studies revealed that the technological distributed load along the length of the rod has a form described by the law: $q_1 = \int S^* \sin\frac{\pi x}{L}$, and forced vibrations of the grate rod by vibration load according to the law q (x) sin θt.

As a first approximation, the grate rod is considered as a beam with a uniformly distributed load (from the weight of the rod), resting on five elastic supports. When drawing up the equations of motion, the following are not taken into account: excitation of the rod from impact loads perpendicular to the radial direction due to its insignificance, the influence of vibrations of other grate rods and loads in the axial direction of the rod. Taking into account the above assumptions, we have a continuous beam on elastic supports. For dynamic calculations, this beam is converted into a single-span, statically indeterminate beam by eliminating all intermediate supports and replacing their action on the beam with unknown reaction forces. The rigidity of the elastic supports C_1at points A and E are equal to each other and are defined as the reciprocal of the compliance of the rubber bushings installed at these points. The rigidities of the intermediate supports $C_{2,}$ C_3 at points B, C, D are determined taking into account the compliance of the rubber bushings and the rigidity of the beam itself . To compile a differential equation for forced vibrations of a beam on elastic supports under a vibration load, we differentiate twice the equation of the curved axis of the beam [3]

$$\mathrm{EY}\frac{\partial^4 y}{\partial x^4} + m\frac{\partial^2 y}{\partial x^2} + \sum P_k = \mathrm{q(x)}\sin\theta t \qquad (1)$$

g de $y -$ movement of the beam in the section

m $\times \frac{\partial^2 y}{\partial x^2}$- intensity of inertial forces directed upward

$\sum P_k = R_A + R_B + R_C + R_D + R_E$- forces of resistance to movement directed upward

q (x) sin (θt)vibration load causing forced vibrations of the beam
θ −angular frequency .
EU= const because the beam is homogeneous and has identical cross sections along its entire length. Due to the fact that the free vibrations of the beam decay over time due to resistance, in steady motion there will only be forced vibrations, therefore, in equation (1) the free vibrations of the beam are not taken into account.
a particular solution to equation (1) in the form:

$$У_{част}(x, т) = (x).\sin(\theta.t)$$

Substituting it into the original equation we get

$$y_4^{IV}(X) - S^4.y(X) = \frac{q(X)}{EУ} \quad (2)$$

Where $S^4 = \frac{m\theta^2}{EУ}$
The general solution to equation (2) has the form

$$У_{обш}(X) = Q \cdot A_{sx} + b \cdot B_{sx} + c \cdot C_{sx} + d \cdot D_{sx}$$

Where , B_{sx}, C_{sx}, D_{sx} _ A_{sx} - influence functions are determined according to the method proposed in [5].
Particular solution (2), if q (x) is not a polynomial

$$У_{част}(X) = \frac{q(x)}{S^4 EУ}$$

Then the general solution (complete integral) (2) will be:

$$(X) = У_{обш}(X) + У_{част}(X) = Q \cdot A_{sx} + b \cdot B_{sx} + c \cdot C_{sx} + d \cdot D_{sx} + \frac{q(x)}{S^4 EY} (3)$$

where a, b, c, d are determined by successive differentiations of equation (3), using the initial conditions:
y(0); $y'(0)$; M(0) =EJ $y''(0)$; Q(0) =EJ $y'''(0)$ [6]
Substituting the found values of a, b, c, d into equation (3) and its successive derivatives, y′(x), y′′(x)we obtain:

$$y(x) = y(0)A_{sx} + \frac{y'(0)}{S}.Bsx - \frac{Q(0)}{S^3 E.Y}Dsx - \frac{Q(0)}{S^4 E.Y}.(A_{sx} - 1) (4)$$

$$y'(x) = y(0)\,SD_{sx} + \gamma'(0).A_{sx} - Q(0).\frac{C_{sx}}{S^2 EJ} - q(0).\frac{D_{sx}}{S^3 EJ} \quad (5)$$

$$M(x) = -EYy''(x) = Q(0)\frac{Bsx}{S} + \frac{q(0)}{S^2}C_{sx} - S^2 EJy(0)C_{sx} - SEJy(0) \quad (6)$$

Due to the fact that in the beam under consideration (Fig. 1) there are discontinuities at points B, C, D where the reaction forces of the elastic supports are applied, it is therefore necessary to create separate equations of the initial parameters for each section of the beam:
I section $0 \leq X \leq L/4$

$$y_1(x) = y(0)A_{sx} + \frac{y'(0)}{S}B_{sx} - \frac{RA}{S_3 EY}D_{sx} - \frac{q+q1}{S^4 EJ}(A_{sx} - 1) \quad (7)$$

$$M_1(x) = \frac{RA}{S}B_{sx} - \frac{q+q1}{S^4 EJ}C_{sx} - S^2 EYy'(0)D_{sx} (8)$$

II section L /4$\leq X \leq L/2$

$$y_2(X) = y_1(X) + \frac{R_b}{S^3}.D_s(X - \frac{L}{4})$$

$$M_2(X) = M_1(X) + \frac{R_b}{S}.B_s(x - \frac{L}{4}) \quad (9)$$

III section L /2$< X \leq 3/4.L$

$$y_3(X) = y_2 + \frac{R_c}{S^3}.D_s(X - \frac{L}{4})$$

$$M_3(X) = M_2(X) + \frac{R_c}{S}.B_s(X - \frac{L}{2})(10)$$

IV section ¾.L< $X \leq L$

$$y_4(X) = y_3(X) + \frac{R_D}{S^3}Ds(X - \frac{3}{4}.L)$$

$$M_4(X) = M_3(X) + \frac{R_D}{S}Bs\left(X - \frac{3}{4L}\right) \qquad \text{(eleven)}$$

the unknown parameters y(0), y1(0), R_A, R_B, R_c using additional conditions imposed on movements in the direction of excluded supports. Due to the symmetry of the loads, the unknown support reactions at points D and E are equal to:$R_A = R_E,\ \ R_B = R_D$

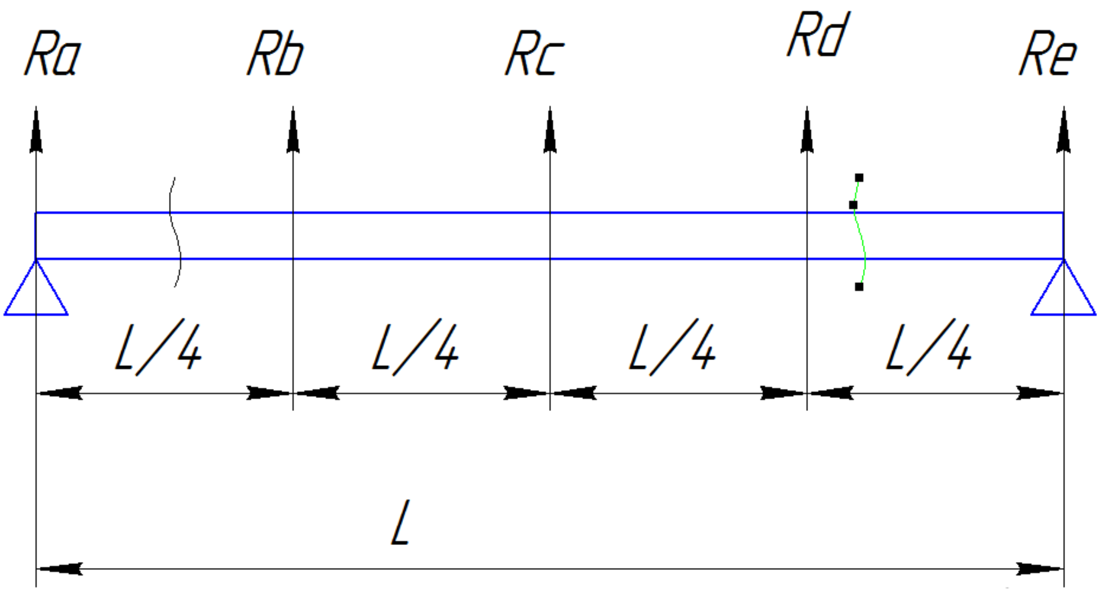

Fig.1. The force acting on the reaction of the grate supports of the raw cotton cleaner from large debris.

Wherein

$$y\ (0) = \frac{R_A}{C_1}; y\left(\frac{L}{4}\right) = \frac{R_B}{C_2};\ \ y\left(\frac{L}{2}\right) = \frac{R_c}{C_3}; y\left(\frac{3}{4L}\right) = \frac{R_D}{C_2};(12)$$

Therefore:

$$Y(\ L\ /4) = y(0)\frac{AsL}{4} + \frac{y'(0)}{S}.\frac{B_sL}{4} + \frac{q+q_1}{S^4EJ}.(\frac{A_sL}{4} - 1) = \frac{R_B}{C_2}(13)$$

$$y(L/4) = y(0)\frac{A_sL}{4} + \frac{y^1(0)}{S}B_sL/4(14)$$

After solving the system of equations by definition and substituting their values into equations (7)-(14), it is possible to determine the displacements and bending moments in any section of the beam. Based on the results of solving equations (7) - (14) and performing strength calculations on a computer with various variations of the initial data, the operating parameters of the new grates were determined relative to existing cleaners with rigid grates. The difficult operating conditions of the grate, as well as the dependence of the nature of vibrations on many factors, show the need to create a dynamic model of the grate on elastic supports to calculate its rational parameters. A single-mass oscillatory system was chosen as a dynamic model. The nature and magnitude of technological resistance were obtained by strain gauging during experimental studies. Further processing of the experimental oscillograms made it possible to

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present them $F_{t.b.}$ as the sum of a number of harmonic forces and a random one, representing a nonlinear force. The differential equation of motion of a single-mass oscillatory system is:

$$m\dot{x} + в\dot{x} + c\dot{x} = F_{t.b.};\ \ F_{t.b} = M(F_{t.b)} \pm \delta(F_{t.b}$$

g de m - mass of the grate; C is the rigidity coefficient of the elastic support; c is the coefficient of viscous resistance of the elastic support; M($F_{t.b}$), $\delta(F_{t.b})$is the mathematical expectation of the technological disturbance on the grate and its random component. The analytical solution of this nonlinear differential equation is practically difficult, so its solution was carried out on a Pentium - IV type computer using the Maple program using the Runge-Kutta method. To determine the best dynamic parameters of the system and find the connection between them, variational studies of the initial parameters obtained experimentally were carried out:

$$m = 1{,}8\frac{Hc^2}{m};\ \ C = 12500\frac{H}{m};\ \ \ b = 50\,H\frac{c}{m};\ \ F_{t.b} = M(F_{t,b}) \pm \delta(F_{t.b})$$

$$M(F_{t,b}) = 30 + 3.97\cos 30x - 2.28\sin 30x + 0.73\cos 60x + 0.52\sin 60x$$

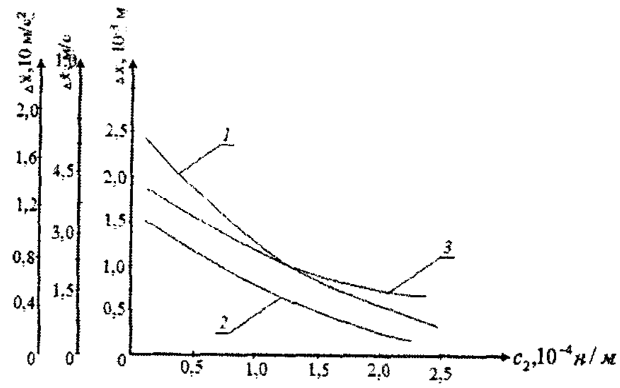
where, $\delta(F_{t.b})$is a random setting of the technological load from cotton. Let's set it in the form of a random number generator with a frequency of 34 Hz and an amplitude of 15 N.

The range of variation of the stiffness value was established based on the results of experiments: C = 6250, 18750, 25000, 37500 N/m. The importance of variation in the rigidity of the elastic sleeve is due to the fact that its value directly affects the entire dynamic process in the system, determines the occurrence of forced oscillations of a certain amplitude, which ensures the efficiency of the cotton cleaning process. In Fig. Figure 2a shows theoretical curves of changes in the displacement of the mass of the system X, its speed $\dot{X}$and acceleration $\ddot{X}$during the technological operating mode with varying the rigidity coefficient of the elastic support of the grate. Analysis of these graphical dependencies shows that the natural oscillations of a single-mass system change quite quickly (about 0.1 s) and the correspondence of the forced oscillations of the grate mass to the nature of the approximated technological load. A decrease in the amplitude of low-frequency oscillations with an increase in the rigidity coefficient of the elastic support is also clearly visible (Fig. 1).

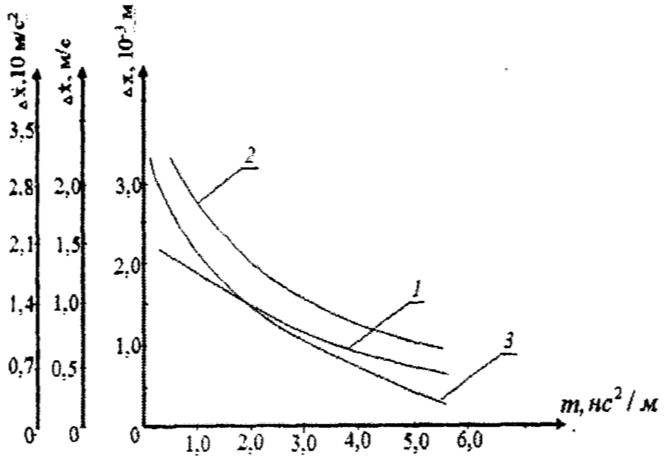
Considering that significant changes in the amplitude of oscillations of the grate lead to exceeding the limits of the gap between the sawn drum and the grate, the best values of the rigidity coefficient of the elastic support, providing the required amount of gap, are

$$\text{WITH} = (1{,}8 \div 2{,}5) * 10^4\ \frac{\text{Н}}{\text{М}}.$$

The range of variation of the values of the dissipation coefficient characterizing the elastic support was established based on the results of the experiment for rubber brand NO-68 (c = 25; 75; 100; 150 $\text{Н} * \text{с/м}$).Dissipation is taken into account due to the ability to absorb free vibrations and part of the forced vibrations. Figure 2 b shows the dependences of the change in system parameters , , X when $\dot{X}$ changing $\ddot{X}$ with Recommended values of the dissipation coefficient (80 ÷ 100) $\text{Н}\ \frac{\text{с}}{\text{М}}$. The results of experimental studies showed 3, 4 that the grate on an elastic base experiences loads from raw cotton pulled by a saw drum 2 times ÷ 3,5less relative to the load on the grate with a rigid support (serial version).



A)



b)

Rice. 2. Theoretical curves of changes in the displacement of the mass of the system X, its speed Ẋand acceleration Ẍduring the technological operating mode with varying the stiffness coefficient (a) and the dissipation coefficient (b) of the grate.

This, in turn, allows for the possibility of reducing the metal consumption of the grate through the use of hollow tubular rods. From the analysis of graphs of varying the mass of the grate (0.9; 2.7; 3.6; 5.4 N, c^2/м)it is clear that with increasing mass the amplitude of oscillations increases, but a decrease in the frequency of oscillations is observed. The best results from the analysis of studies are m = (1, 2÷ 1,8) Н$\frac{c^2}{м}$.

The importance of taking into account the technological loads from cotton on the grate rods is explained by the fact that their value varies widely, depending on the supply of cotton (machine performance) and has a random nature of the impact, speed and acceleration of the grate as a function of the technological load from raw cotton. The determining component of the load fluctuation is a low frequency of 5 Hz, which corresponds to the rotation frequency of the serrated drum of the cleaner. High frequency the component 34 Hz is due to the uneven supply of raw cotton. At the design load in the system, the grate oscillates with a small amplitude, 0.8. ÷ 1,0 мм.Increasing the load to 60 N leads to an increase in the oscillation amplitude of the grate to 1.8 mm ÷ 2,2 . But, at the same time, the increase in vibration speed is insignificant, 0.6 ÷ 0,8m/s. With a further increase in load, in practice, raw cotton is usually slaughtered between the grate and the saw drum. Thus, the developed design of a grate on elastic supports for a cotton cleaner from large debris with recommended rational parameters ensures an increase in the cleaning effect and productivity of the machine.

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