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BENDING AND VIBRATION OF THE WORKING CYLINDER SHALL ON SAW FIBER SEPARATION MACHINES

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Abstract
In this paper, the results of research on the development and substantiation of the parameters of a resource-efficient, vibration-resistant design of a gin machine saw cylinder flexible element bearing support are presented. According to authors, it was found that the saw cylinder bends during operation, which has a negative impact on the sawing technology, causing the saw cylinder to vibrate and, as a result, the bearings to fail quickly. The use of flexible bearing supports ensures significant absorption of shaft vibrations and increases the service life of the saw cylinder and bearings
Keywords: cotton, saw gin machine, saw cylinder, vibration, bending, elastic element, base, bearing, inertia.

Introduction

Extensive research is being carried out around the world to improve the techniques and technology of primary processing of cotton and to create their scientific basis. In this regard, in particular, increase the efficiency of the ginnery, which is the main machine of ginners, increase the efficiency of work, equip the working bodies with resource-efficient structures, increase their durability, improve the operational reliability of the machine, develop mathematical models and optimize maintaining quality is important.

the issue of developing a new design of the saw cylinder with a flexible bearing base of the gin machine which is resource-saving, increases the toughness of the fiber, reduces vibration and noise, increases productivity and allows you to maximize the natural properties of the fiber has risen to the number of important issues facing the industry today.

1. Problem statement

The issues of improving the working bodies of the sawmill, saving resources, increasing the productivity of the machine, the strength of the working bodies, improving the quality of products are considered by a number of foreign and domestic scientists. However, in the known work so far, no in-depth theoretical and experimental research has been conducted to determine the causes of bending and vibration of the saw cylinder shaft in the gin machine, to develop technical solutions to reduce them. The aim of this study is to develop a resource-efficient, vibration-resistant design and substantiate the parameters of a gin machine saw cylinder bearing element bearing support.

In-depth theoretical and experimental studies to substantiate the driving mechanism and parameters of the sawing machine show that the sawing cylinder oscillates at different levels and directions during operation, and this situation has many negative consequences in the process. As a result of the research, a support structure was developed as a flexible element for the gin machine saw cylinder as an option to suppress such vibrations [1 - 6].

2. Vibrations of the saw cylinder during operation and the factors influencing them

The fact that the saw cylinder oscillates at different frequencies and amplitudes in the vertical and horizontal directions during operation, and this situation has many negative consequences in the process, has been substantiated both theoretically and practically in a number of scientific studies [3-6]. Therefore, we will not dwell on this issue. We have developed a technical solution that provides a certain degree of bending and vibration suppression of the saw cylinder, the main essence of which is that a rubber bushing of a certain thickness is mounted on the bearing of the saw cylinder shaft and mounted on the body. This measure significantly reduced the bending and vibration of the saw cylinder shaft. In these studies, we will make a theoretical analysis of the bending and vibration of the saw cylinder shaft after the installation of the flexible base.

It is known that during operation, the bending of the saw cylinder is minimized due to the deformation of the flexible base. However, at the same time, there will be small vertical oscillations in the saw cylinder. Given the technological power of ginned cotton, these vibrations are random. Cylindrical oscillations are characterized by the following differential equations:

$$m_{ц}\frac{d^2z}{dt^2} + b_n\frac{dz}{dt} + (c_1 + kc_2)z = F_0 + \delta F_0 \qquad (1)$$

where $F_0, \delta F_0$ - the average value of the force forcing the saw cylinder from the mass of cotton fiber and the force forming the random.

c_1 - linear component of rigidity coefficient, kc_2 - nonlinear component of riginity coefficient, $m_{ц}$ - mass of cylinder, b_n - dissociation coefficient of flexible support bearing.

Based on the numerical solution of (1), the laws of variation of vertical oscillations for different values of the bearing elasticity of the saw cylinder cylinder bearing shown in Fig. 1 are obtained:

An analysis of the laws of change in z and ż (Fig. 2) shows that the greater the external force, the greater the static displacement of the saw cylinder axis due to the increase in mass. When this force

is 1,2 H±(0,05÷0,12) N, the displacement reaches 1,2·10^{-3} m, and when 1,8 H±(0,1÷0,18) N the deflection is 1.5 to 10^{-3}m [7-11].

The coupling graphs in Figure 2 show that the saw cylinder mass is $5.0 \cdot 10^2$ kg and the oscillation amplitude increases from $0.3 \cdot 10^{-3}$ m to $0.252 \cdot 10^{-3}$ m as the load increases from 0.25 N to 1.8 N.

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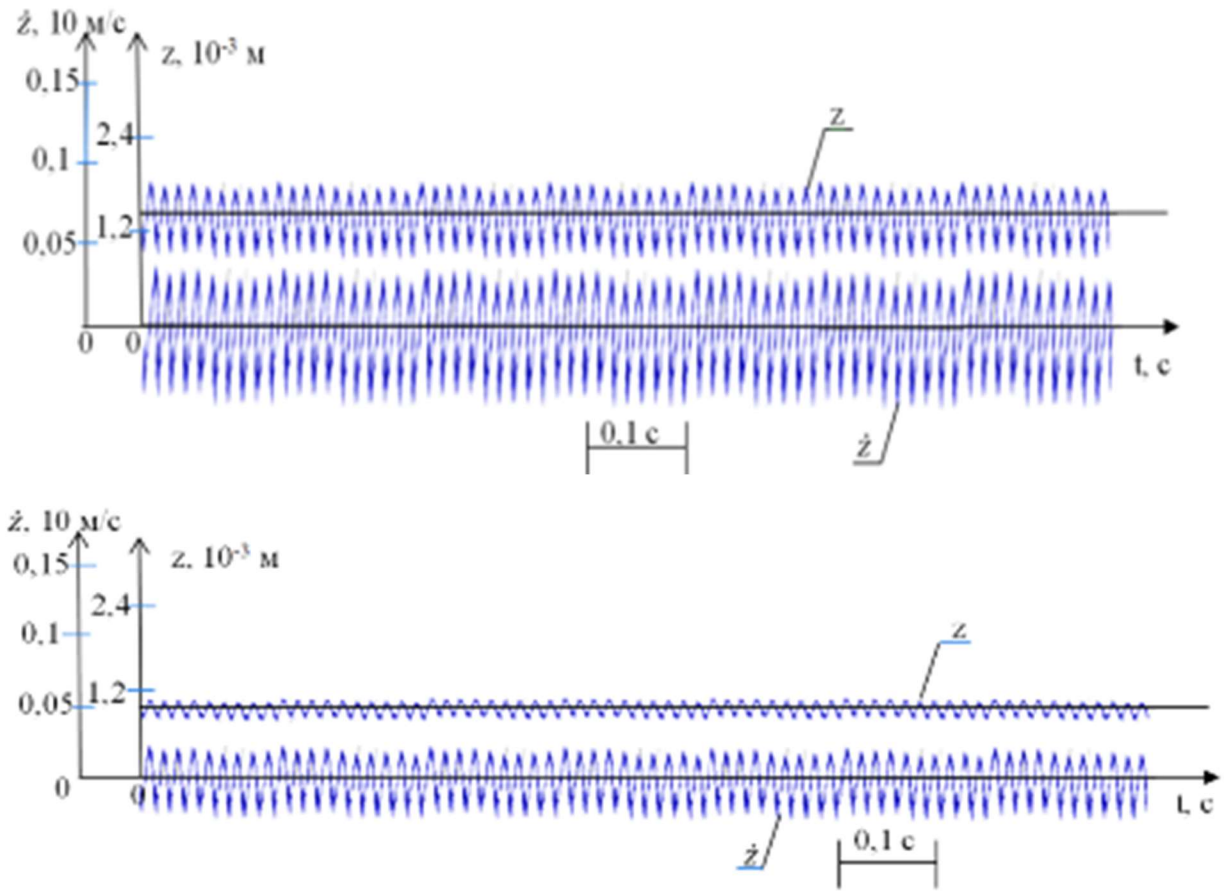

Figure 1. The law of variability of vertical oscillations of the saw cylinder at different values of elastic stiffness of the saw cylinder bearing

If the mass of the saw cylinder is reduced to $3.75 \cdot 10^2$ kg, the amplitude of the saw cylinder shift is from $0.4 \cdot 10^{-4}$ m to $0.71 \cdot 10^{-3}$ m. Therefore, it is advisable to take $m_ц \leq (3{,}75 \div 4{,}25)$ and $F_0 \leq (1{,}0 \div 1{,}2)N$ to ensure that the amplitude of vibration of the saw cylinder does not exceed $0.3 \cdot 10^{-3}$ m. The results of the study also show that an increase in the coefficient of elasticity of the bearing's flexible bushing leads not only to a decrease in the oscillation amplitude z and (z)˙, but also to an increase in the oscillation frequency of the cylinder.

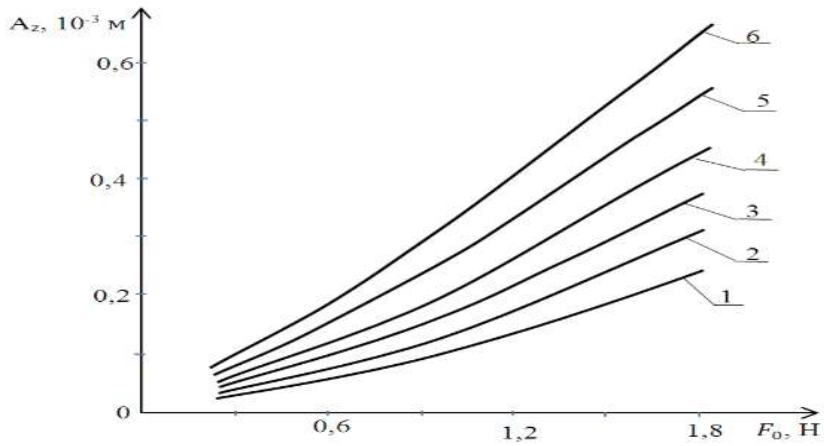

Figure 2. Graphs of the dependence of the change in the vibration amplitude of the saw cylinder displacement on the impact force of the cotton raw material

1-$m_ц$=5,0·10^{2}kg; 2-$m_ц$=4,75·10^2 kg; 3-$m_ц$=4,5·10^2; $m_ц$=4,25·10^{2}kg; 5-$m_ц$=4,0·10^{2}kg; 6 - $m_ц$=3,75·10^{2}kg.

When the force rises to 1.8 N, the value of A_z decreases from 0.58 10^{-3}m to 0.22 10^{-3}m according to the nonlinear law. Accordingly, the oscillation amplitude of the saw cylinder speed also decreases nonlinearly, and the coefficient of elasticity of the saw cylinder cylinder bearing elastic element support increases (Fig. 2) [11-14].

The recommended values of the coefficient of virginity of the bearing elastic support $c_1 = (5{,}0 \div 5{,}5) \cdot 10^4$ Н/м; $c_2 = (0{,}1 \div 0{,}12) \cdot 10^4$ Н/м, ensure that the oscillation amplitude is in the range $Az = (0{,}09 \div 0{,}14) \cdot 10^{-3}$ м, $A\dot{z} = (0{,}08 \div 0{,}11)$ м/с.

3. Bending of the jin machine working shaft

During operation, the bending of the cylinder shaft occurs mainly due to the size of the cylinder mass, the impact force of the raw material shaft in the existing working chamber, as well as unbalanced masses, forces caused by pulling fibers by saw teeth.

When assembling a saw cylinder of length l, it is necessary to compress the discs of radius R, and this requires a certain force. In addition, using the conventional method, the bending stiffness of the saw cylinder shaft is determined from the following expression:

$$C = (1 + h_g)(N + \lambda_q E_b F_b)R^2 \qquad (2)$$

where, C- the bending stiffness of the saw cylinder shaft; h_g - the thickness of the discs; N- function of the friction effect between the saws; λ_q - function of elastic deformation effect; $E_b F_b$ - elastic modulus and cross-sectional area.

Using the above methodology and calculation results, it should be noted that in the current design, if the maximum deflection of the saw cylinder is (0,3÷0,5)·10^{-3} m, the deflection of the saw cylinder with a flexible support bearing is (0,031÷0,053)·10^{-3} m or more accurately, the deflection is reduced by 10 times [2-7].

In considering the dynamics of the machine unit with an saw cylinder mechanism, the technological resistance of cotton was taken into account in the following form according to the results of experimental studies:

$$M_c = \mathrm{M}_{\text{п.ц}} \pm \delta(\mathrm{M}_{\text{п.ц}}) \qquad (3)$$

where, $\mathrm{M}_{\text{п.ц}}$ – the average value of the moment of resistance in the saw cylinder shaft during the ginning process; $\delta(\mathrm{M}_{\text{п.ц}})$ – occurring at the expense of the change in resistance during the firing process $\mathrm{M}_{\text{п.ц}}$ - the random component of the moment.

Using the second-order Lagrange equation, the differential equation representing the motion of a machine unit involving a gin saw cylinder is as follows:

$$\frac{dM_{g\text{в}}}{dt} = 2M_k\omega_c - 2M_k p\frac{d\text{ц}_{g\text{в}}}{dt} - \omega_c S_k M_{gb};\ J_n\frac{d^2\dot{\text{ц}}_{\text{п.ц}}}{dt^2} = M_{g\text{в}} - \left[M_{\text{п.ц}} \pm \delta(M_{\text{п.ц}})\right] \quad (4)$$

Where: M_{gb}, $\mathrm{M}_{\text{к}}$ – torque on the drive shaft and its critical value; $\frac{d\text{ц}_{g\text{в}}}{dt}$ – angular velocity of the drive rotor; ω_c - the rotational frequency of the network; p – the number of pole pairs of the motor; S_k – critical value of displacement, J_n- the moment of inertia given.

Calculated values of asynchronous drive parameters
4A280M8У3 brand electric generator parameters: $P_{gb} = 75$ kW,
$n_{gb} = 730\ RPM;\ \omega_{gb} = 76{,}4\ c^{-1}, f_c = 50$ Hz, $\mathrm{P} = 4;\ \cos\varphi = 0{,}85;\ \omega_k = 220\ B;\ M_{\text{н}} = 674{,}5\ nm;\ \mathrm{M}_{\text{к}} = 1349\ km,\ S_{\text{н}} = 0{,}02;\ S_{\text{к}} = 0{,}075;\ J_n = 1{,}29\ kgm^2$.

The analysis of the obtained differential equation (3) showed that As M_c increases, $\dot{\varphi}_{\text{п.ц}}$ decreases in a nonlinear pattern and $M_{\text{п.с}}$ increases accordingly. Thus, as the average value of the technological resistance force moment increases from 2.8·10^2 Nm to 7.92·10^2 Nm, the average value of the saw cylinder speed increases from 0.739·10^2 s^{-1} to 0.631·$10^2 s^{-1}$. In this case, the moment of inertia of the saw cylinder is taken as 1.05 kg · m^2 [4-8].

In this case, the average value of $\mathrm{M}_{\text{п.ц}}$ rises from 1,07˙10^2 Nm to 4,07˙10^2 Nm. This is explained by the fact that for small values of the moment of inertia of the saw cylinder, the effect of the resistance force on the change in $\dot{\varphi}_{\text{п.ц}}$ is significant. When the moment of inertia $J_n = 1{,}5\ kg \cdot m^2$, the angular velocity of the saw cylinder decreases to 0,592˙$10^2 s^{-1}$, the torque increases to 7,108˙

10^2 Nm. It should be noted that a significant decrease in the angular velocity leads to a decrease in the efficiency of cotton ginning. Therefore, the recommended parameter values are:

$J_n = (1{,}0 \div 1{,}25)\ kg \cdot m^2$; $\mathrm{M_c} \leq (4{,}5 \div 6{,}5) \cdot 10^2$ Nm,

Where, $\dot{\text{ц}}_{\text{п.ц}} \geq (6{,}7 \div 7{,}0) \cdot 10\ s^{-1}$.
It is known that the greater the moment of inertia of the working body, the lower the coefficient of unevenness of the angular velocity. To provide $\delta \leq (0{,}08 \div 0{,}09)$ for the case under consideration It is recommended to choose $J_n = (1{,}0 \div 1{,}25)\ kg \cdot m^2$.

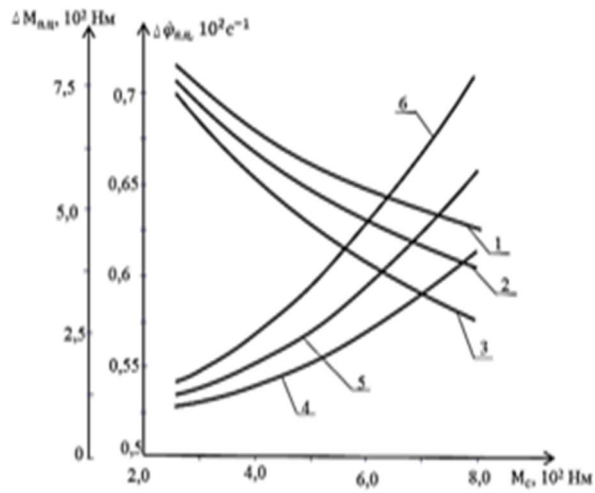

Figure 1. The change in torque and angular velocity of the gin saw cylinder shaft under the influence of raw cotton

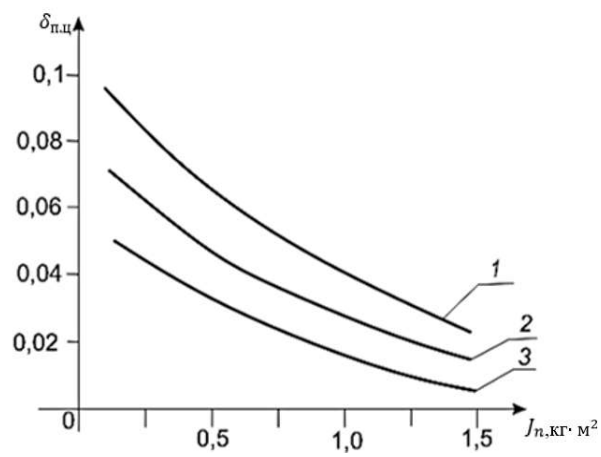

Figure 2. The effect of the change in the coefficient of inequality of the angular velocity of the saw cylinder on the moment of inertia of the saw cylinder

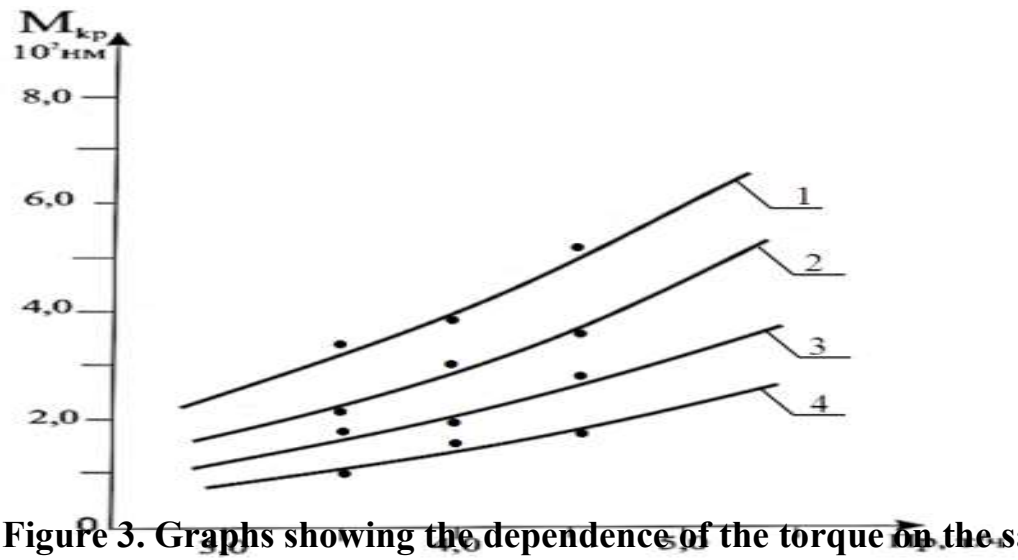

Figure 3. Graphs showing the dependence of the torque on the saw cylinder shaft on the change in machine productivity

As a result of the analysis of the results of the application of modernized saw gin machine cylinder in the production of flexible bearing support at Turakurgan and Mingbulak ginneries, the mass fraction of impurities and defects in the fiber compared to the existing structure is 0.3%, fiber mechanical damage 0.4%, seed fineness 0, It was found that it decreased by 3% and fiber output increased by 0.2%, the service life of bearings and machine body parts increased by 4.5 times, and the noise in the shaft rotation was significantly reduced [9-11].

CONCLUSION

1. A mathematical model of the vertical vibration of the saw cylinder was obtained, taking into account the stiffness of the flexible bearings and the technological resistance of the spinning cotton. Based on which it was found that the displacement of the vertical vibrations of the saw cylinder depends on the change in the speed of the cylinder and the amplitude of vibration of the saw cylinder depends on the force from the spinning cotton.
2. It was found that increasing the coefficient of virginity of the bearing bearing flexible element leads not only to a decrease in the amplitude and velocity of the saw cylinder vibration, but also to an increase in the vibration frequency of the saw cylinder. This in turn also reduces the static deformation value of the bearing base flexible element. The vertical displacement and velocity of the saw cylinder due to the change in the coefficient of rotation of the flexible bearings leads to a change in the amplitude of oscillations.
3. When the bending of the saw cylinder is studied analytically, the maximum bend of the saw cylinder in the existing structure (0.3 ÷ 0.5) is 10^{-3}m, and in the proposed design of the base of the flexible element the bend is (0.03 ÷ 0.053) $\cdot 10^{-3}$ m or the bend showed a decrease up to 10 times.

4. As a result of tests conducted at ginneries, when using flexible bearing structures, the mass fraction of impurities and defects in the fiber compared to the existing structure was 0.3%, mechanical damage to the fiber was reduced by 0.4%, seed hair was reduced by 0.3% and fiber yield was 0.2%. e, the service life of bearings and body parts increased by 4.5 times.

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