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Development of the Design and Calculation of Parameters of the Saw Cylinder with an Elastic Bearing Support Jin

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Abstract: The article provides an effective, resource-saving structural design of an elastic bearing support for saw gins. Expressions are obtained for calculating the amplitude and frequency of oscillations of the saw cylinder on an elastic bearing support. A technique for calculating the bending vibrations of a saw cylinder is provided. The problem of the dynamics of a machine unit of a two-mass gin saw cylinder system has been solved. The analytical method provides formulas for determining the laws of motion of the rotor of the engine and the shaft of the saw cylinder; The problem of the oscillatory movement of the grate on an elastic support with nonlinear rigidity by the analytical method is solved. An expression is obtained for determining the law of grate vibrations. The results of experiments obtained the dependence of changes in the amplitude of oscillations of the shaft of the saw cylinder on the elastic bearings, justified the parameters of the system. The results of production tests of the recommended design are presented.

Keywords: Saw gin, Cylinder, Shaft, Bearing, Rubber Support, Vibration, Bending, Stiffness, Amplitude, Frequency, Mass, Quality, Fiber.

I. INTRODUCTION

The main disadvantages of the existing technology and designs of cleaning fibrous materials are low cleaning effect, low productivity, as well as high damage to fibrous material. To eliminate these drawbacks, a number of designs and technologies for cleaning fibrous material have been developed. At the same time, the effect on the cleaning effect of the construction of grid is important [1].

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II. LITERATURE REVIEW

A. Analysis of the problem

Analysis of the research on saw gin was carried out mainly by ginning technology, by determining the diameter of the saw blades, the size of the grate, the working chamber, ejaculation, removal and removal of fibers, technological gaps and regulation of cotton nutrition. No studies have been carried out to develop the design of saw cylinders and their units of drive mechanisms, bearing bearings with elastic elements to substantiate their parameters based on deep dynamic theoretical and experimental studies [2]. Insufficient rigidity of the shaft of the saw cylinder can lead to unacceptable distortions of the saws in the gap between the grates, which are undesirable for the ginning process, as it can damage the fibers.

Important is the maximum reduction in the bending of the shaft of the saw cylinder by reducing its moment of inertia, ensuring balance, as well as the use of elastic bearing bearings [3, 4].

B. An effective bearing arrangement for absorbing vibrations of rotating shafts

Known bearings in which the bearing mates directly with the housing and the connecting surfaces of the rolling bearing to the housing are the outer diameter and width of the rings [5]. A drawback of existing bearings in the composition of any mechanisms and machines is the direct transmission of oscillations of the rotating shafts in the bodies of machines and mechanisms to the bodies themselves, which leads to an increase in vibration noise of the corresponding machines and mechanisms.

In another known construction, the bearing support of the shaft of contents contains a housing with a bearing mounted therein and an oval shaped elastic element of variable cross section located between the outer surface and the housing. Moreover, the major axis of the outer oval surface is aligned with the minor axis of the inner oval surface and is installed in the housing so that the axis of minimal rigidity coincides with the direction of the loading force [6]. The disadvantage of this design is its complexity and high manufacturing costs. In addition, this design provides vibration damping only in the presence of radial disturbing forces and when the disturbing force is applied at a certain angle, the vibration damping effect is reduced.

The design of the support for absorbing vibrations of the rotating shafts comprises a housing with a bearing mounted therein and an elastic element arranged in the form of a sleeve located between its outer surface and the housing. The sleeve is made, for example, of rubber of circular cross section, while the axis of the hole of the sleeve is shifted relative to its central axis in the direction opposite to the direction of action of the resultant loading force by no more than 15% of the internal radius of the sleeve [7,8]. The disadvantage of this design is not effective when acting on the shaft axial force.

The design allows damping only forces acting in a vertical plane perpendicular to the axis of the shaft. In this case, the absorption of forces in the axial direction is virtually absent.

In the process, the following forces act on a rotating shaft: driving torque, gravity, inertia from unbalanced masses, friction, technological loads, etc. The components of the resultant force will be directed both in the radial and axial directions. These forces will act cyclically on the housing 1 through the bearing 4 and the elastic bushings 2 and 3 (Fig. 1).

The presence of elastic bushings 2 and 3 significantly reduces the effect of these forces on the housing 1. Moreover, the bending of the shaft 5 due to the radial components of the forces is significantly reduced. The execution of the elastic bushings 2 and 3 in the form of a truncated cone with the diameters of the bases d and D allows the absorption of axial component forces. The use of this kind of shaft support will allow, due to the absorption of vibrations of the rotating shafts, to reduce the transmission of vibrations to the frames (housings) of the respective machines and mechanisms, therefore, the vibration and noise characteristics of these machines and mechanisms are reduced to a large extent.

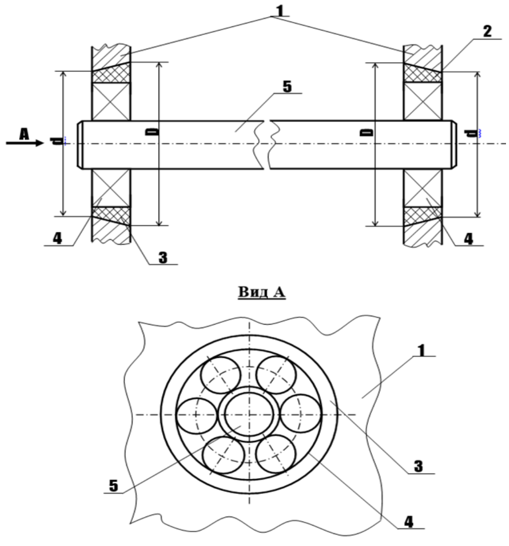

Fig. 1. Support for absorbing vibrations of the rotating shafts

The proposed support can be recommended for use as a vibration-absorbing support in the main machines for primary processing of cotton (cotton and fiber cleaners, separators, linters, roller and saw gins, etc.) in the main for the bearing support of the gin saw cylinder.

A support has been developed that provides vibration reduction of structural elements of the device when external forces are applied to the shaft, both in radial and axial directions.

III. RESEARCH METHODS

A. The influence of the parameters of the gin saw cylinder on the amplitude of oscillations

During operation of the saw gin, vibrations occur mainly due to the bending of the saw cylinder [9]. But, in the recommended design of the saw cylinder, its bend is reduced to a minimum. In this case, the vertical oscillations of the gin saw cylinder occurs due to unbalanced masses, mainly from the mass of the fiber of the gin cylinders captured and carried away by the teeth of the saws. The value of this fiber mass varies between (0.31 ÷ 0.55) kg. Given the variability of the mass of the fibers captured and dragged by the saw cylinder at n = 730 rpm, D=0.16 m, the centrifugal force varies within (409 ÷ 467) N.

According to the methodology described in [10], we determine the amplitude of oscillations of the saw cylinder vertically, taking into account the mass of cotton fiber captured by the teeth of the saw blades:

$$A=\frac{\left(m_c+m_{cf}\right)}{2\left(c_1+kc_2\right)\left|1-\frac{\omega_c^2}{\rho_0^2}\right|} \tag{1}$$

Where, $\rho_0=\sqrt{\frac{\left(c_1+kc_2\right)}{\left(m_c+m_{cf}\right)}}$

m_c - is the mass of the saw cylinder; m_{cf} - mass of cotton fiber; k-coefficient of nonlinearity of rigidity of the conical rubber sleeve of the bearing support of the shaft of the saw cylinder. c_1- linear component of the stiffness coefficient of the conical elastic bearing, kc_2 - non-linear component of the stiffness of the bearing.

The numerical solution (1) on a PC was made with the following parameter value $\omega_c=76.4\ s^{-1}$; $n=730\ rpm$; $D_c=2\ m$; $R_c=0.32\ m$; $m_c=\left(400\div500\right)kg$; $m_{cf}=\left(0.3\div0.6\right)kg$; $k=0.2\div0.6$; $c_1=\left(4.5\div6.0\right)\cdot10^4\ \frac{N}{m}$; $c_1=\left(0.5\div0.8\right)\cdot10^4\ \frac{N}{m}$;

Based on the numerical solution of the problem, graphical dependences of the system parameters were constructed. An increase in the angular velocity of the gin saw cylinder leads to an increase in its oscillation amplitude according to a nonlinear regularity (Fig. 2). So, when changing ω_c from 68 s^{-1} to 80.2 s^{-1} the amplitude of the oscillations increases from $0.205\cdot10^{-3}$m to $0.61\cdot10^{-3}$m when the mass of cotton fiber is captured and dragged by the teeth of the saw cylinder 0.35 kg, and when $m_{cf}=0.75\ kg$ the amplitude A increases from $0.42\cdot10^{-3}$m to $1.409\cdot10^{-3}$m. Considering the results of experimental studies, the maximum bending of the shaft of the saw cylinder taking into account the elastic bearing support is within $\left(0.30\div0.38\right)\cdot10^{-3}\ m$. To ensure these values, the recommended values are: $m_{cf}=\left(0.35\div0.4\right)kg$; $\omega_c=\left(7.4\div7.8\right)\cdot10\ s^{-1}$.

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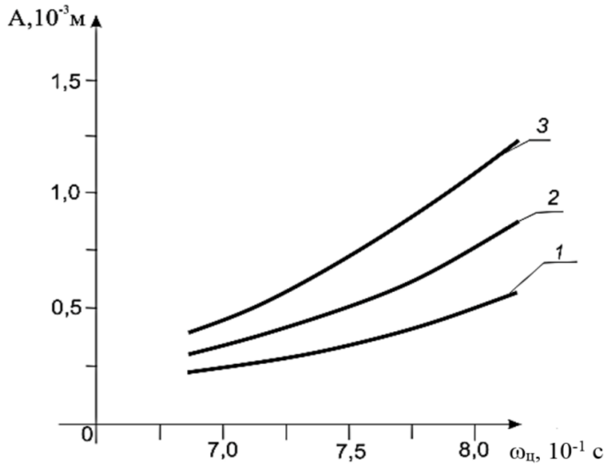
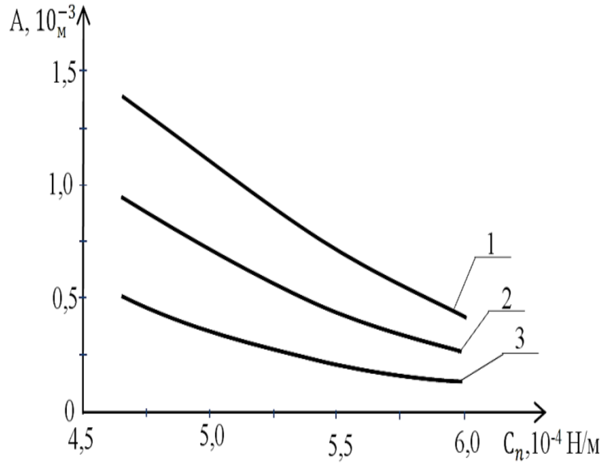
It is important to study the influence of the rigidity of the elastic supports on the amplitude of oscillation of the saw cylinder. In fig. 2. b graphical dependences of the change in the amplitude of oscillation of the saw cylinder on the variation of the stiffness coefficient of the elastic rubber bearing support of the gin saw cylinder are presented.

An analysis of the constructed graphical dependencies shows that with an increased stiffness coefficient of the elastic bearing support from $4.61 \cdot 10^4$ H/m до $6.0 \cdot 10^4$ H/m the amplitude of oscillation of the saw cylinder decreases from $1.315 \cdot 10^{-3}$ m до $0.39 \cdot 10^{-3}$ m according to a nonlinear regularity with $\omega_c = 80.0\ s^{-1}$; and *k=0.6*. With a decrease in ω_c and coefficient to *k=0.2* and $\omega_c = 70.0\ s^{-1}$; , the amplitude of oscillations of the saw cylinder decreases to $0.11 \cdot 10^{-3}$ m.



where, 1- at $m_{cf} = 0.35\ kg$ 2- at $m_{cf} = 0.55\ kg$ 3- at $m_{cf} = 0.75\ kg$

patterns of change in the amplitude of oscillations of the saw cylinder with an elastic conical bearing of the bearing from a change in the angular velocity of the saw cylinder;



where, 1- at *k=0.6;* $\omega_c = 80.0\ s^{-1}$; 2- at k=0,4; $\omega_c = 76\ s^{-1}$; 3- at *k=0.2;* $\omega_c = 70.0\ s^{-1}$;

patterns of change in the amplitude of oscillations of the saw cylinder with an elastic conical bearing of the bearing from a change in the coefficient of total stiffness of the elastic support.

Fig. 2. Graphical dependence of the amplitude of oscillations of the saw cylinder

To ensure oscillations of the saw cylinder within $(0.30 \div 0.38) \cdot 10^{-3}\ m$, taking into account the elastic bearing conical shape, it is recommended: $c_p = (5.4 \div 6.0) \cdot 10^4\ \frac{N}{m}$; $\omega_c = (7.4 \div 7.8) \cdot 10\ s^{-1}$. $k = 0.2 \div 0.4$.

IV. THEORETICAL RESEARCH

A. The results of the calculation of the bending of the gin saw cylinder

In the process, the bending of the saw cylinder occurs due to the large mass of the cylinder, the strength of the existing raw chamber, as well as unbalanced masses, including the fibers captured by the teeth of the saw blades.

The scheme of the gin saw cylinder is shown in Fig. 3. Given that when assembling a saw cylinder with a length l, the radius of the disks R requires the necessary compression force. Moreover, according to the results of [9,10], we write the expression for determining the bending stiffness of the saw cylinder:

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

where C is the bending stiffness of the saw cylinder; h_g – Thickness of disks; N – function of the impact of friction between saws; λ_q – function of the effect of elastic deformation; E_b, F_b – modulus of elasticity and cross-sectional area.

Consider the process of bending vibrations of the saw cylinder. To solve the problem of bending the saw cylinder, the Hamilton methods and the second-order Lagrange equations were taken into account [11]. In this case, saws and discs are distributed according to the necessary regularity:

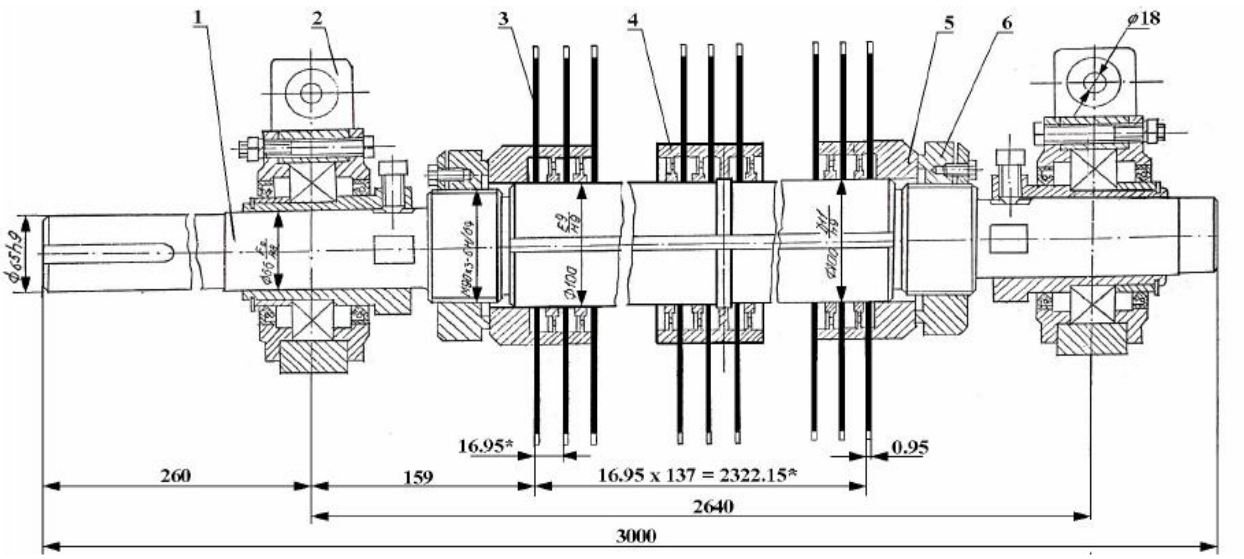


Fig. 3. Saw cylinder layout

fmpi (z)=ρp Fpi(z) and fmni (z)=ρn Fni(z)

If the number of drank j, then the mass distribution function will be $(j-1)$.

The replacement function according to [12] will have the form:

$$f_m = \frac{\rho_n j l_p F_p}{L} + \frac{\rho_n (j-1) l_n F_n}{L} \quad (3)$$

Where l_p, l_n - is the thickness of the saws and discs.

In this case, the hardness of the disk package is determined according to [11] with $c=2NR^2$

then the potential and kinetic energy during bending vibrations of the saw cylinder will be:

$$U = \frac{1}{2}\int_0^L 2NR^2\left(\frac{\partial^2 u}{\partial z^2}\right)^2 dz;$$

$$T = \frac{1}{2}\int_0^L \left(\frac{\rho_n j l_p F_p}{L} + \frac{\rho_n (j-1) l_n F_n}{L}\right)\left(\frac{\partial u}{\partial t}\right)^2 dz \quad (4)$$

Where U, T - is the potential and kinetic energy during bending vibrations; u -displacement of the disk pack from the geometric axis Z in the direction of OX.

Using the method described in [13], we obtain:

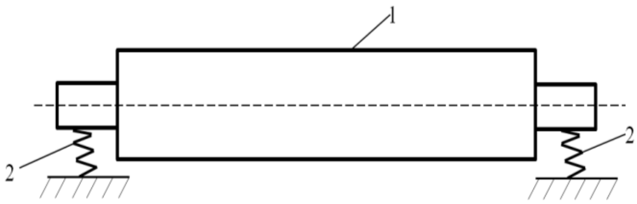
$$\frac{\partial^2}{\partial z^2}\left(2NR^2\frac{\partial^2 u}{\partial z^2}\right) + \left(\frac{\rho_n j l_p F_p}{L} + \frac{\rho_n (j-1) l_n F_n}{L}\right)\frac{\partial^2 u}{\partial t^2} = 0 \quad (5)$$

To solve (5) subject to the conditions

$$u(z_1,0) = f(z)\frac{\partial u}{\partial z}(z_1,0) = f(z)$$

Moreover, the boundary conditions depend on the form of their connection (*z=0; z=e*). When installing the saw cylinder on elastic bearings (see Fig. 4), bending vibrations are significantly reduced.

When installing the saw cylinder on the elastic bearings, the bending moment will approach zero, that is, the saw cylinder bends slightly.



1-saw cylinder, 2-elastic support

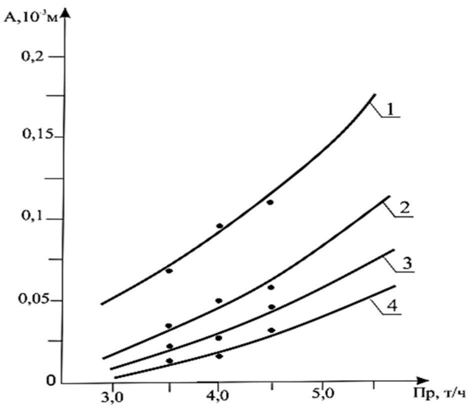
Fig. 4. The installation scheme of the saw cylinder on elastic supports

Using the above methodology and the calculation results presented in [14], we note that in the existing design, the saw cylinder bends to the maximum *(0.3÷0.5)·10*$^{-3}$*m*, and the bend of the saw cylinder mounted on the elastic bearings is 10 times less than, in the gray design, up to *(0.031÷0.053)·10*$^{-3}$*m.*

V. EXPERIMENTAL RESEARCH

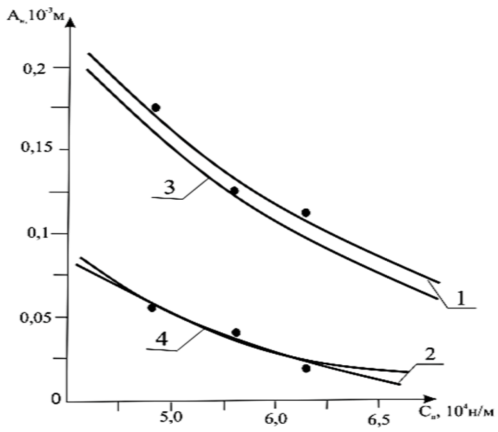
A. Analysis of the results of an experiment to determine the bending of the shaft of a gin saw

The gin saw cylinder is massive and rotates at a frequency of *730 rpm*. Therefore, there is significant bending of the shaft of the saw cylinder. At the same time, technological gaps between the saw blades and grates are violated in the middle zone of the saw cylinder, which can lead to significant damage to the fibers and seeds of cotton, reducing the service life and productivity of the machine. Therefore, it is important to study the bending vibrations of the shaft of the saw cylinder.



where, 1-serial version of the bearing support of the saw cylinder; 2-when using an elastic bearing support from the rubber brand-7317; 3-when using an elastic bearing support from the rubber brand-10-220; 4-when using an elastic bearing support from the rubber brand-6308-MKShchIS;

Fig. 5. Graphic dependences of the change in the amplitude of bending vibrations of the shaft of the saw cylinder on the increase in gin productivity



where, 1,2-experimental dependencies; 3,4-theoretical dependencies; 1,3-at ω_c =75 s^{-1}; 2,4-at ω_c=65 c^{-1}

Fig. 6. Dependences of the change in the amplitude of the bending vibrations of the shaft of the saw cylinder on the increase in the stiffness coefficient of the elastic bearing bearings for various values of the angular velocity of the gin saw cylinder

The use of elastic bearing bearings significantly reduces bending vibrations of the shaft. Based on the processing of experimentally obtained oscillograms, graphical dependences of the change in the amplitude of the bending vibrations of the shaft of the saw cylinder on the increase in gin productivity were constructed (Fig. 5 and Fig. 6).

Analysis of the graphs in Fig. 5 shows that an increase in gin productivity from $3.5\ t/h$ to $4.5\ t/h$ leads to an increase in the amplitude of bending vibrations of the saw cylinder shaft in the serial version of the bearing support from $0.077 \cdot 10^{-3} m$ to $0.179 \cdot 10^{-3} m$. When using an elastic support made of 6308-TMKSCHIS rubber, the amplitude of the bending vibrations of the saw cylinder reaches $0{,}071 \cdot 10^{-3} m$, and when using rubber of the brand 7317, the amplitude A reaches $0.022 \cdot 10^{-3} m$. To ensure the amplitude value of the bend by more than $A \le (0.02 \div 0.03) \cdot 10^{-3}\, m$, it is recommended to use the rubber grade 10-220 as an elastic bearing support for the saw cylinder.

In this case, it is important to determine the stiffness coefficient of the elastic support on the value of the bending of the saw cylinder when the gin productivity changes.

Analysis of the graphs in Fig. 6 shows that with increasing speed of the saw cylinder from 65 c^{-1} to 76.4 c^{-1} the amplitude of bending vibrations increases to $(0.045 \div 0.052) \cdot 10^{-3}\, m$, Therefore, the recommended parameter values are: $\omega_c = (74 \div 77)\, s^{-1}$; $A = (0.02 \div 0.03) \cdot 10^{-3}\, m$, $c \ge (6.2 \div 6.6) \cdot 10^4\, N/m$.

VI. RESULTS AND DISCUSSION

A. Results of production tests of a saw cylinder with an elastic gin bearing

In the cotton mill, tests were carried out with the new recommended designs of bearing bearings for saw cylinders equipped with rubber elements. Tests have shown high reliability and stability of the gin saw with an elastic bearing support compared to the existing gin version. The amount of defects decreased on average by 0.5%, the mechanical damage to the seeds decreases by 0.2%, the seed flow decreases by 1.2%, the fiber yield increased by 0.4% drank an hour. This is because rubber bearing pads reduce the bending of the cylinder shaft, and technological clearances are maintained. At the same time, the resource of bearings and its housing increased by 4.0–4.5 times, and noise significantly decreased. Due to the high-frequency oscillations of the saw cylinder, the effect of fiber removal from cotton seeds increases, seed descent decreases, etc.

VII. CONCLUSIONS

Effective structural schemes have been developed for the elastic bearing bearings of gin saw cylinders, which allow significant absorption of shaft vibrations, which reduces damage to fibers and seeds of cotton, increases the service life of bearing bearings and the saw cylinder. Formulas are obtained for calculating the amplitude and frequency of oscillations of the saw cylinder. Graphical dependences of the change in the amplitude of oscillation of the saw cylinder on the change in its rotational speed and stiffness coefficient of the elastic bearing are constructed. Analytical methods have been used for calculating the bending of a saw cylinder taking into account the elasticity of bearing bearings. Using the method of tensometry, regularities of the change in the torque on the shaft of the gin saw cylinder are obtained. Graphical dependences of the change in torque on the shaft of the saw cylinder on the change in the performance of the gin are constructed. The main design parameters are recommended.

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