ISSN: 2350-0328

International Journal of AdvancedResearch in Science, Engineering and Technology

Vol. 5, Issue 12, December 2018

Design Development and Justification of Parameters of Elastic Bearing Support Shawn Jin Shaft

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ABSTRACT: The article presents the design features and the principle of operation of the elastic bearing supports for the shafts of technological machines. A method for calculating the parameters of elastic bearings for heavy shafts is given. The system parameters are determined on the basis of full-factor experimental studies. The results of the performance tests of the gin saw cylinder, the shaft of which is mounted on elastic bearing supports are presented.

KEYWORDS: Shaft, bearing, support, elastic element, oscillation, saw cylinder, rigidity, cotton contamination, moisture, rubber, fiber yield, hairiness.

I. INTRODUCTION

The main technological processes in cotton plants are cotton ginning and cleaning. In this case, the main working body of these machines is the saw cylinder. According to the results of a number of well-known studies, it was revealed that the main disadvantage of the saw cylinders of gins and cotton cleaners is high fiber and seed damage, high power consumption due to large mass, low cleaning and ginning effects, and not high productivity. Due to the large mass of saw cylinders, the shaft bends, high-frequency vertical oscillations occur, resulting in not only damage to the fibers and seeds of cotton, but also rapid failure of the bearing supports [1, 2].

Known bearings, in which the bearing mates directly with the housing, and the inherent joining surfaces of the rolling bearing to the machine body, are the outer diameter and width of rings. leads to an increase in the vibration noise of the respective machines, as well as the rapid exit of the failure of the bearing supports.

Also known is the design of the shaft support, the housing housing with the bearing mounted in it and the elastic element of an oval shape [3, 4, 5], which is interleaved between the outer surface and the housing.

The disadvantage of this design is its complexity and high costs in manufacturing. In addition, this design loses the effect of absorbing oscillations of the shaft at high speed conditions and large moments of inertia of the shafts.

Development of new designs of bearing elastic supports. We have developed a new resource-saving scheme of the bearing support for the shaft of the gin saw cylinder [6, 7, 8].

The essence of the design lies in the fact that the support for absorbing vibrations of the rotating shafts includes a housing and twin (two in each bearing) bearings mounted in it and elastic (rubber) bushings placed between their outer surfaces and the housing. The thickness of the elastic sleeve of the inner bearing of the support is made larger than the thickness of the elastic sleeve of the outer bearing of the support. At the same time, the load from the side of the rotating shaft will be distributed over the four elastic bushes of the bearing supports. Due to the lower stiffness (large thickness) of the inner bearing bushings of the bearings, which deform and absorb significant components of the oscillations of the shafts. At the same time, due to the high stiffness of elastic bushings (of smaller thickness), which are deformed to a lesser degree and will absorb part of the oscillation of the shafts and help reduce the movement of the axis of rotation of the shafts. The application of the design leads not only to the absorption of vibrations of massive shafts at high speed conditions, but also on the reliability of the support.

The support for absorbing oscillations of the rotating shafts includes a housing 1 (Figure 1), in which two elastic (rubber) sleeves 2 and 3 are fixed, in which two bearings 4 are installed. Between the bearings 4 a sleeve 6 is seated on the shaft 5. The bearings 4 are mounted on shaft 5. The thickness h_v of the elastic sleeve 2 of the inner bearing 4 is made larger than the thickness h_H of the elastic sleeve 3 of the outer bearing 4 of the bearing (h_v>h_H):

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
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$$h_v = \frac{D_{vh} - D_{vv}}{2};\ h_H = \frac{D_{hh} - D_{hv}}{2};$$

where D_{vh}, D_{vv}, D_{hh}, D_{hv} - are the respective outer and inner diameters of the inner 2 and outer 3 elastic bushings.

In the course of work, the following forces act on the rotating shaft: driving torque, gravity, inertial forces from unbalanced masses, friction forces, technological loads, etc. These forces will act cyclically on the housing 1 through bearings 4 and elastic sleeves 2 and 3. Availability elastic sleeves 2 and 3 significantly reduce the effect of these forces on the housing 1. The magnitude of the bending of the shaft 5 due to the radial components of the forces is significantly reduced. The implementation of elastic sleeves 2 and 3 with different thickness allow gradual absorption of oscillations of the shaft 5 and the necessary reduction

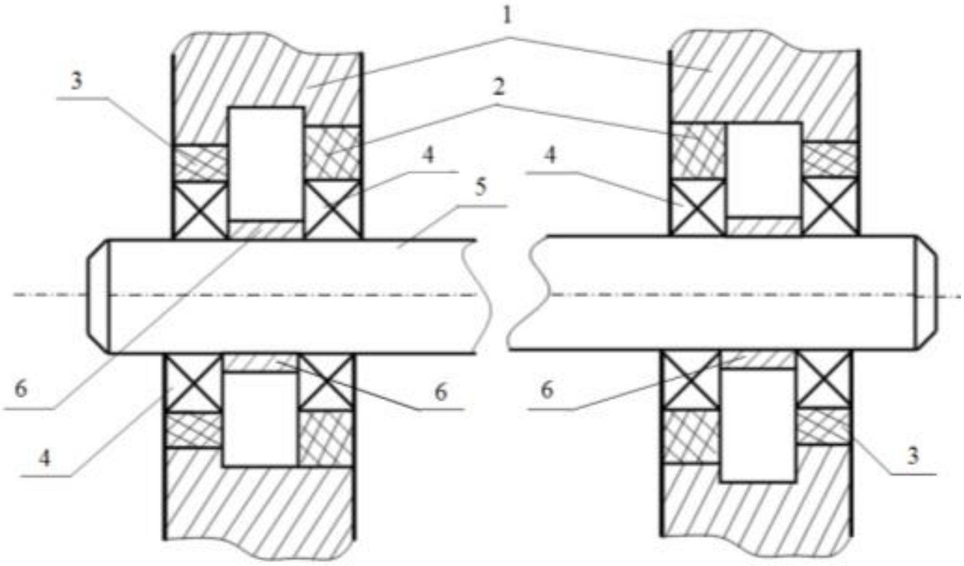

Fig. 1 - Support for the absorption of vibrations of rotating shafts

of the displacement axis of rotation of the shaft 5 due to the deformation of the sleeves 2 and 3, especially with large masses of the shafts 5 and high speed modes.

In this case, the load from the side of the rotating shaft 5 will be distributed over the four elastic bushes 2 and 3 (with two supporting shafts having twin bearings 4 with elastic bushes 2 and 3). Due to the lower stiffness of the elastic sleeves 2 (large thickness h_v) of the internal elastic sleeves 2, which are deformed and absorb a significant part of the vibrations of the shaft 5. At the same time, due to the large stiffness of the elastic sleeves 3 (smaller thickness h_H), they will absorb the necessary part of the vibrations of the shafts 5, and also allow the reduction of the mixing axis of rotation of the shaft 5.

According to the stiffness, thickness and marks of the rubber bushings 2 and 3 can be selected according to the value of the moment of inertia of the shaft and speed modes.

The design allows efficient vibration absorption rotating massive shafts at high speeds, resulting in reliable system operation.

If both radial and axial forces occur during rotation of the shafts of technological machines, it is expedient to perform the elastic supports of conical shape (see Fig. 2).

The support for absorbing vibrations of the rotating shafts comprises a housing 1 in which fixed elastic sleeves 2, 3 of a truncated conical shape are installed. Elastic sleeves 2 and 3 are made of oil resistant rubber. The diameters d of the smaller bases of the elastic sleeves 2 and 3 are installed in the housing 1 in opposite directions and are on the outside of the housing 1. The diameters D of the large bases of the elastic sleeves 2 and 3 are installed in the housing 1 towards each other and are the inner side of the housing 1.

In the holes of the elastic sleeves 2 and 3 mounted bearings 4, the holes of which are installed shaft 5.

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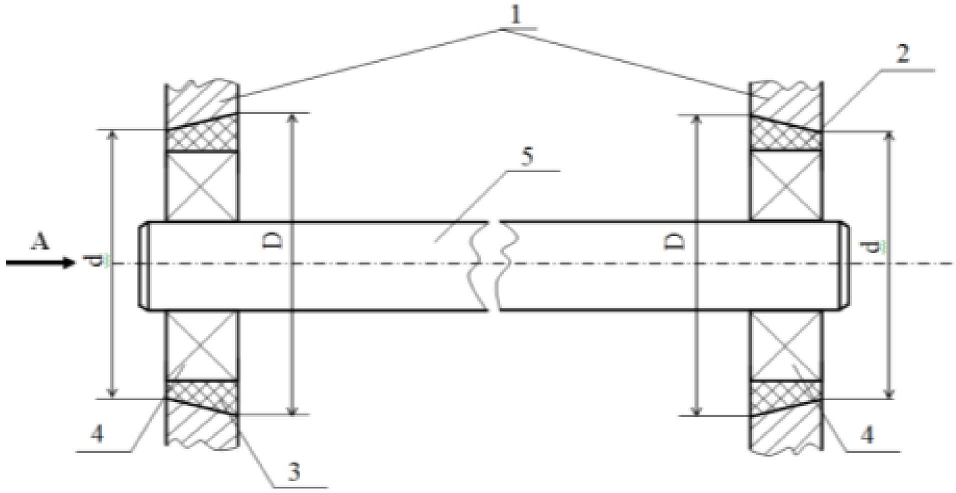
In the course of work, the following forces act on the rotating shaft: driving torque, gravity, inertial forces from unbalanced masses, friction forces, technological loads, etc. The components of the resultant force will be directed both in the radial and axial directions. These forces will act cyclically on the housing 1 through the bearing 4 and the elastic



sleeves 2 and 3.

View A-A

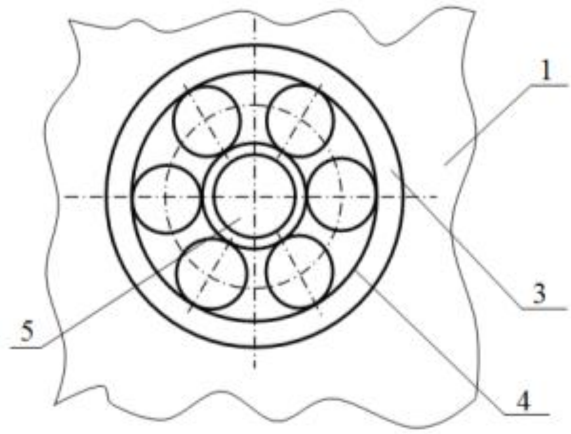


Fig. 2-support for the absorption of vibrations of the rotating shafts

The presence of elastic sleeves 2 and 3 significantly reduce the effect of these forces on the housing 1. In this case, the amount of bending of the shaft 5 due to the radial components of the forces is significantly reduced. The implementation of the elastic sleeves 2 and 3 in the form of a truncated cone with the diameters of the bases d and D allow the absorption of the axial components of the forces.

The use of this kind of shaft support will, due to the absorption of oscillations of rotating shafts, reduce the transmission of vibrations to the frames (housings) of the respective machines and mechanisms, therefore, the vibration noise characteristics of these machines and mechanisms are largely reduced.

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

The method of calculating the parameters of the elastic support shaft of the saw cylinder. Gin saw cylinders are massive, their weight reaches to half a ton. During the rotation of the saw cylinder with a frequency of 730 rev / min due to unbalanced masses, as well as from cotton trapped by the saw teeth, significant centrifugal forces arise. In dragging on average (0.3 ÷ 0.5) kg of cotton (fiber) at the same time saw cylinder (with producer 5.0 ÷ 6.0 t / h), rotating with a frequency of 730 rev / min and a radius of 0.16 m centrifugal force:

$$P_s = m_x \cdot (\frac{\pi n_s}{30})^2 \cdot \frac{D_s}{2} \quad (1)$$

where m_x is the mass of simultaneously trapped cotton (fiber); n_s= 730 rpm; D_s-diameter of the cylinder.

At the same time, from the calculation data P_s= 467 N. This force, acting cyclically on the cylinder supports, causes intense wear of bearings that quickly fail.

In static mode, the amount of deformation of the elastic supports under the bearings of the saw cylinder is determined from the expression:

$$a_{st} = \frac{G_s}{2C_n} \quad (2)$$

whereG_s is the weight force of the saw cylinder with bearings; C_n - support stiffness coefficient.

In the process of work from an unbalanced mass of cotton (fiber) captured by the teeth of the cylinder saw, there is a centrifugal force:

$$P_s = m_x \omega_s^2 R_s \quad (3)$$

where mx is the mass of trapped cotton (fiber) with the saw teeth; ω_s - the angular frequency of the saw cylinder; R_s-the radius of the cylinder.

While the amplitude of oscillation of the saw cylinder vertically according to (2) has the form:

$$a = \frac{a_{st}}{\left|1 - \frac{\omega_s^2}{P_0^2}\right|} \qquad P_0 = \sqrt{\frac{C_s}{m_s + m_x}}, \quad (4)$$

Where m_s -is the mass of the cylinder.

According to the calculated initial parameters of the system, the amplitude of the vertical oscillations of the saw cylinder reaches (0.05÷0.14)•10^{-3} m.

The amplitude of the reaction force of the bearings of the gin saw cylinder is proportional to the amplitude of the deformation of the elastic element:

$$R_{вит} = C_{ы} \cdot a = \frac{R_{ые}}{\left|1 - \frac{\omega^2}{P_0^2}\right|} \quad (5)$$

Тогда динамический коэффициент системы определяется из выражения:

$$K_{вит} = \frac{R_{вит}}{R_{st}} = \frac{1}{\left|\frac{C_s - \omega_s^2 (m_s + m_x)}{C_s}\right|} \quad (6)$$

By choosing the necessary values of the parameters $C_ц$, $\omega_ц m_ц$them$_x$, you can control the value of the dynamic coefficient. In this case, choosing the required values of stiffness of the elastic support, the mass of the cylinder and the performance of the gin, it is possible to minimize (dampen) the oscillations of the saw cylinder in the vertical direction, which leads to an increase in the service life of bearings.

The substantiation of the parameters of the bearing support full-factor experiments.

The recommended bearing support with elastic shock absorbers was installed in the DP-9 genie. In addition, the recommended support is installed in a cotton cleaner from coarse litter. When this rubber bushings obtained by vulcanization. The experiments were conducted with different rubber grades with different circular stiffness. Additional

ISSN: 2350-0328
International Journal of AdvancedResearch in Science, Engineering and Technology
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fluctuations and depreciation of saw cylinders allow an increase in their resource, an increase in gin productivity, a cleaning effect. Bearings of brand 216 were used.

Full-factor experiments were carried out in the production of cotton plants.

Input factors are presented in table 1.

Table 1. Input Factor Values

Factors	Meanings				Difference
		-1	0	+1	
Cotton moisture, %	X_1	9	9.5	10	0.5
Brandtires10^6N/m	X_2	2	2,5	3,0	0,5
Weediness,%	X_3	5	5.5	6	0.5

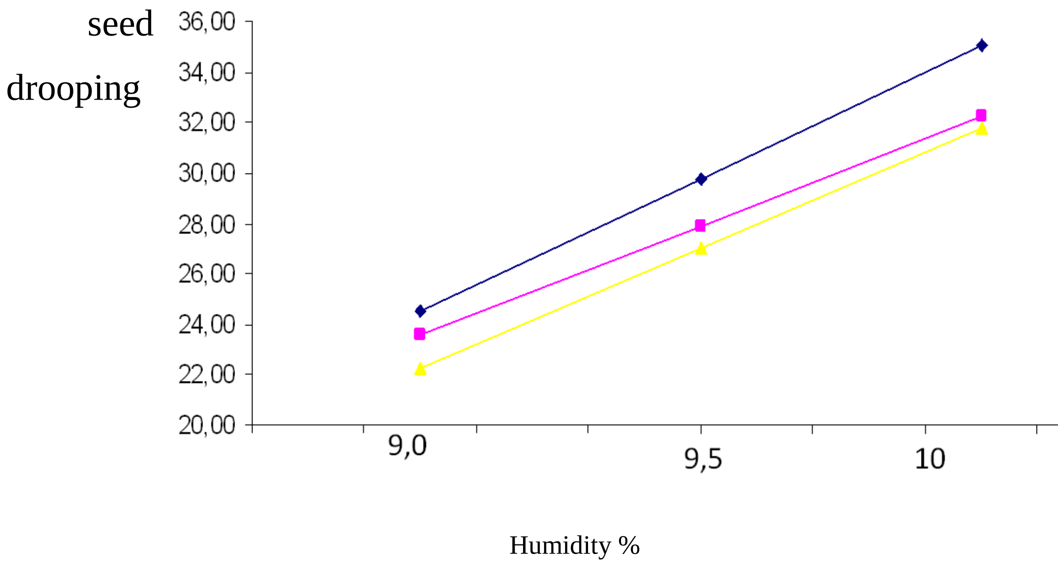
The input factor X_1 is the moisture content of cotton, the limits of variation are from 9% to 10%. The input factor X_2 is the hardness of rubber, the limits of change are 60·103 Nm/rad to 90 · 103 Nm/rad. The input factor X_3 is cotton infestation, the limits change 5% to 6%. For the output factor selected cotton seed pubescence.

The regression equation is finally obtained.

$$Y = 27{,}9 + 3{,}57x_1 - 1{,}72x_2 + 0{,}57x_3 - 0{,}5x_1x_2 + 0{,}35x_1x_3 + 0{,}35x_2x_3 + 0{,}375x_1x_2x_3 .$$

On the basis of the numerical solution of the problem, the obtained graphical

the dependences of the change in the pubescence of seeds on the change in humidity, the stiffness coefficient of the elastic bearing supports and the contamination of the original cotton, which are shown in Fig.3.



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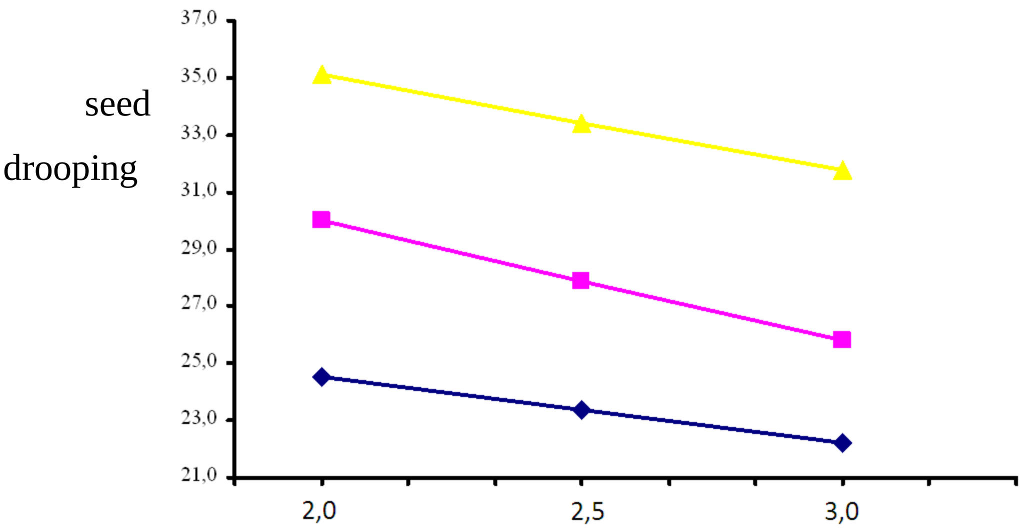
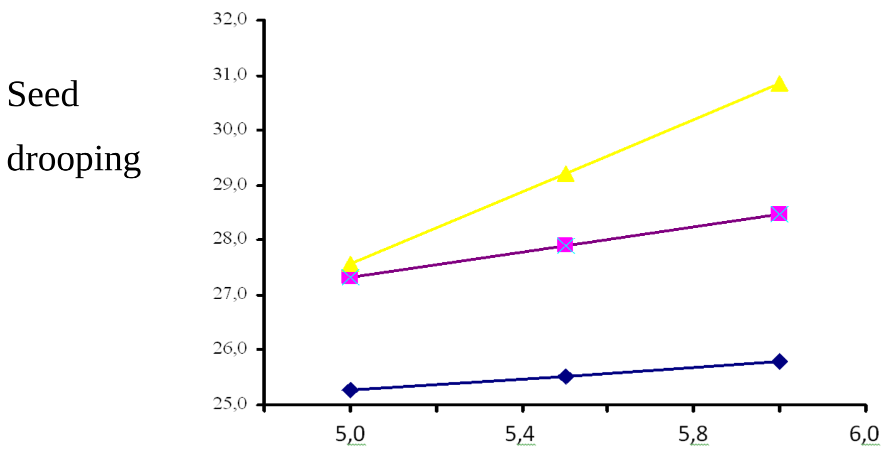


Support stiffness coefficient 10^6 N/m



Contamination %

Fig. 3-Influence of humidity, stiffness coefficient of support and cotton weediness on seed drooping

Based on the analysis of the obtained graphical dependencies, it can be noted that to increase cotton fiber yield, that is, reduce the pubescence (fibrousness) of seeds after the ginning process, the best input parameters are considered to be: humidity not more than (8.5÷9.0)%; stiffness coefficient of rubber bearing supports (2.5÷3.0)•106 N/m; cotton contamination (4.5÷5.0)%.

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THE RESULTS OF PRODUCTION TESTS OF GIN WITH AN ELASTIC BEARING SUPPORT SHAFT OF THE SAW CYLINDER.

Production tests of the recommended design of saw cylinders with elastic bearing supports were carried out in terms of cotton factories.

The process of cotton ginning is significantly influenced by moisture and the initial contamination of raw cotton. During the tests, the humidity and initial contamination of the compared gins were maintained in the same range. Analyzes were carried out at the factory laboratory (table 2).

Tests were carried out with new recommended designs of bearing supports for saw cylinders, equipment with rubber elements. Tests have shown high reliability and stability of the saw gin with an elastic bearing support compared to the existing version of gin. The sum of defects decreased by an average of 12%, mechanical damage to seeds is reduced by 1.22%, free fiber in raw cotton is reduced by an average of 0.112%, seed drooping is reduced by 0.5%, fiber yield increased by 0.5 kg per hour . This results from the fact that on rubber pillows of bearings eliminate a bend of a shaft of the cylinder, technological gaps are maintained. At the same time, the life of the bearings and its housing has increased 2.5 times, the noise has significantly decreased. Due to the high-frequency oscillations of the saw cylinder, the effect of removing fibers from cotton seeds is increased, the lowering of seeds has decreased, etc.

Table 2. Comparative technological indicators of gin with an elastic pillow support

Indicator sin %	After gin with recommended bearing	After the serial gin
Original Cotton - Raw		
Humidity	9,5	9,5
Weed in essafterginning	3,85	3,85
Cotton fiber weediness	2,8	3,3
Mechanical damage to seeds	1,89	3,11
Fiber yield (kgsaw / hour)	9,7	9,2
Seed drooping	10,5	11,0
Free fiber	0,107	0,219

Note: the experiments were carried out in triplicate. The table shows the average values of indicators.

Findings. Developed new effective design elastic bearing supports. The method of calculating the amplitude and dynamic coefficient of the system is recommended. Experiments substantiate the parameter of the elastic bearing support of the sawgin shaft.

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