Study on vibration of the saw cylinder and its critical speed

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Abstract. The force impact of the 130 saw cylinder of gins 4DP-130 and 5DP-130 was studied. It was determined that due to the significant weight of the saw cylinder, when it is rotated at 730 rpm, its radial runout increases, leading to shaft deflection and wear of its neck for bearing fit, rapid wear of the working part of the grates in the grate and blunting of saw teeth when saw teeth touching the grate, which led to their rapid wear. Increased wear of the saw blade and grate, increases the consumption of imported saws and grates, reducing their service life. In addition, due to the increase in the radial runout of the saw cylinder, it was impossible to set the necessary gaps of 1.5-2.0 mm along the entire length between the saw cylinder and the air chamber nozzle, which did not ensure complete removal of the fiber from the saw teeth and its transportation to the fiber discharge channel. This led to a decrease in the productivity of the gin and the daily production of fiber. When studying the saw cylinder, it was revealed that with a decrease in the mass of the shaft, the deflection and vibration of it should decrease, increasing stability. In this regard, a hollow shaft for saw gin is proposed. Theoretically, the critical speed of a hollow shaft for 130 saw gins is considered. It is determined that for the stability of the saw cylinder, its critical speed must be.

1. Introduction

The saw cylinder drives the raw roller, ensuring constant operation of the saw teeth [1]. According to the technological requirements, the saw cylinder must have a high gripping ability, which is ensured by the normal state of the saw teeth, the saw blades must be rigidly fixed on the shaft, not change their position during operation, and during rotation, pass strictly in the middle of the grate gap [2].

When testing the saw cylinder of 130 saw gins in production, it showed that. due to the significant weight of the saw cylinder, when it was rotated at 730 rpm, its radial runout increased [3]. An increase in radial runout led to shaft deflection and wear of its neck for bearing fit, rapid wear of the working part of the grate in the grate and blunting of the saw teeth when the saw teeth touched the grate, which led to their rapid wear. Increased wear of the saw blade and grate, increases the consumption of imported saws and grates, reducing their service life [4]. Also, with an increase in the beating of the saw cylinder, it was impossible to set gaps of 1.5-2.0 mm along the entire length between the saw cylinder and the air chamber nozzle. Instead of the required gaps, it was necessary to set the gaps between the saw cylinder and the nozzle at 2.5–3.0 mm [5].

An increase in the distance between the saw teeth and the air chamber nozzle did not ensure complete removal of the fiber from the saw teeth and its transportation to the fiber discharge channel. The strands of fibers remaining on the teeth of the saws forget the lower part of the grate [6]. Also, when a strand of fiber is brought back into the chamber with the saw teeth, they occupy the useful volume of the front part of the tooth, thereby reducing the intensity of the capture and separation of the fibers [7]. To eliminate the bottoms of the lower part of the grate, it is often necessary to stop the genie [8].

This leads to a decrease in the productivity of the gin and the daily production of fiber. Insufficient filling of the front edge of the saw teeth with new fiber significantly reduces the performance of the gin.

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2. Materials and methods

All this shows the study of the stability of the saw cylinder. Insufficient strength and rigidity of the saw cylinder parts, as well as vibrations, can lead to a deterioration in the ginning process and damage to the fibers, therefore, its parts such as the saw shaft, saw blades, gaskets and tightening washers should be subjected to the necessary force calculations when designing [9].

Consider, for example, the calculation of the saw shaft of the gin 5DP-130 on stability during its rotation [10]. At ginneries, when ginning raw cotton of medium fiber varieties, saw gins of the brand 4DP-130 and 5DP-130 are used. In saw gins, under significant force from external loads, there is mainly its main assembly - the saw cylinder [1].

The saw cylinder consists of a shaft 1, saw blade 2, saw spacers 3, 2 side washers 4 and 2 side nuts 5 (Fig.s 1 and 2). 130 saw blades with an outer diameter of 320 mm and 129 saw blades are mounted on the shaft. On the cantilever part of the cylinder there is a concentrated mass-weight of the coupling half [2, 3].

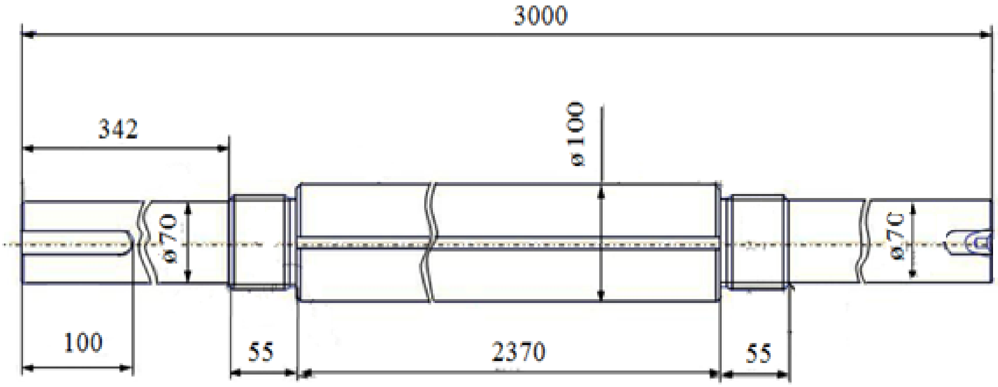

Fig. 1. Working drawings of a solid shaft of gin 5DP-130.

Shaft length 3000 mm, shaft weight 162 kg, weight of one saw blade - 0.57 kg, weight of one saw blade - 0.24 kg, weight of one washer - 2.6 kg, weight of one nut - 2.6 kg. Then the total mass of the saw cylinder is 275.4 kg.

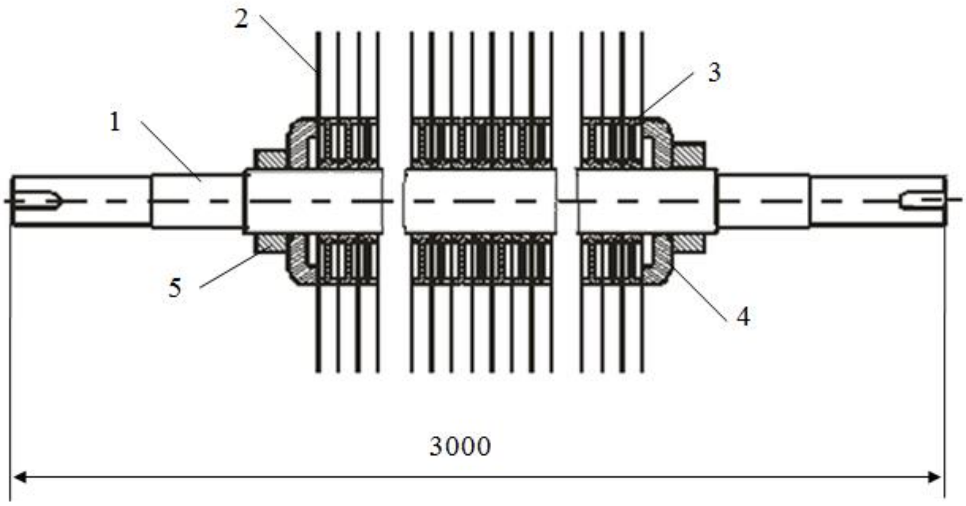

Fig. 2. Scheme of the saw cylinder of a genie brand 5DP-130: 1-shaft, 2-saw gin, 3-saw blades, 4-washer, and 5-nut.

To determine the stability of the shaft rotation, we assume that the center of gravity of the disk, due to the presence of eccentricity, does not coincide with the axis of rotation of the shaft and rotates in a circle. Then a centrifugal force will act on the disc, tending to bend the shaft [5, 6]:

$$C = \omega^2 my \tag{1}$$

and the elastic force of the shaft, opposing the centrifugal force:

$$F = \frac{1}{\alpha} y \quad (2)$$

where, m is the mass of the disk, y is the deflection of the shaft at the location of the disk, and α is a coefficient depending on the stiffness of the shaft, the location of the bearings and the load between the bearings.

For a shaft of constant cross section, mounted on two supports, and the disc is located in the middle of the span:

$$\alpha = \frac{l^3}{48EJ} \quad (3)$$

To determine the mass of the shaft of the existing saw cylinder gin 5DP-130:

$$m_B = 3.14\,\rho R_d^2 L_B + m_o = 3.14\ \cdot\ 7850 \cdot\ 0.052\ \cdot\ 2.37 + 15.2 = 146.04 + 15.2 = 161.24\,kg$$

where, $\rho = 7850 \frac{kg}{m^3}$ - steel density, $R_d = 0.05\,m$ - shaft radius, $L_B = 2.37m$ - the length of the shaft seat where saw blades, saw blades, washers and nuts are installed, $m_o = 5.93 + 9.27 = 15.2\,kg$ - trunnion mass, $m_{2B} = 275.46 - m_o = 275.46 - 15.2 = 260.26\,kg$ - shaft seat weight.

3. Results and discussion

The use of solid shafts with large masses in the saw cylinder 130 saw gins, when rotated at 730 rpm, generates large vibrations during shaft deflection, which reduces the stability of the shaft. With a decrease in the mass of the shaft, the deflection and vibration of it should decrease, increasing stability.
In this regard, let us consider the critical speed of a hollow shaft for saw gin brand 5DP-130. The design dimensions of the hollow shaft 130 of the saw gin are shown in Fig. 3.

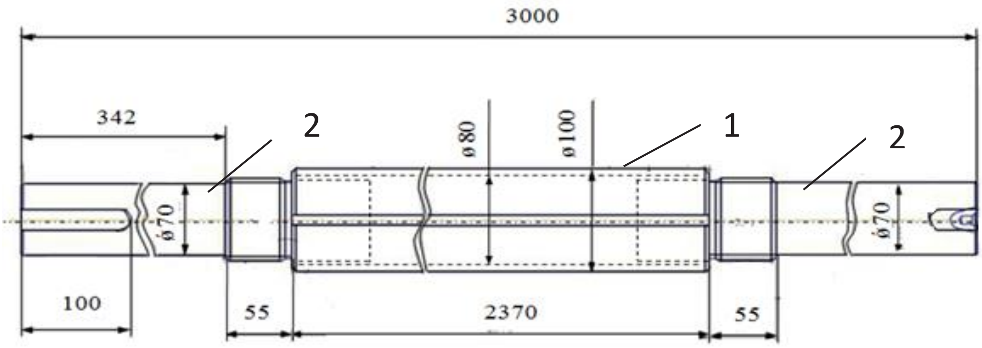

Fig. 3. Structural dimensions of a hollow shaft with pins of a genie brand 5DP-130: 1 - hollow shaft, 2 - pins.

To determine the masses of the hollow shaft together with the trunnions:

$$m_n = 3.14\rho(R_d^2 - R_c^2)L_B + m_{o1} = 3.14 \cdot\ 7850\ \cdot(0.05^2 - 0.04^2)\cdot 2.37 + 43.1 = 52.57 + 43.1 = 95.7\,kg$$

$m_{C1} = 3.14\rho R_{C1}^2 L_{C1} = 3.14\ \cdot 7850 \cdot\ 0.045^2\ \cdot 0.52 = 25.6\,kg$ - mass of the first trunnions, $m_{C2} = 3.14\rho R_{C2}^2 L_{C2} = 3.14\ \cdot 7850 \cdot\ 0.045^2\ \cdot 0.355 = 17.5\,kg$ - mass of the second trunnions.

$$m_{o1} = 25.6 + 17.5 = 43.1\,kg$$

where, $R_d = 0.05\,m$ - shaft outer radius, $R_{BH} = 0.04\,m$- shaft inner radius, and $L_B = 2.37m$ - hollow shaft length.

When the saw cylinder rotates, a centrifugal force *C* is generated, which tends to bend the shaft, and an elastic force *F*, which resists the centrifugal force and tends to return the shaft to its original position. Under the action of two forces, vibrations of a rapidly rotating shaft occur. The shaft after deflection can return to its original position if *F* > *C*; then we can assume that the rectilinear position of the shaft will be stable [7, 8].
Equating the values of *F* and *C*, we get:

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$$\omega_k^2 m y = \frac{1}{\alpha} y \qquad (4)$$

where,

$$\omega_k = \frac{1}{\sqrt{\alpha m}},\ rad/sec \qquad (5)$$

and the critical number of revolutions:

$$n_k = \frac{30}{\pi}\frac{1}{\sqrt{\alpha m}} \qquad (6)$$

To calculate the shaft for deflection, we will construct its design scheme (Fig. 4).

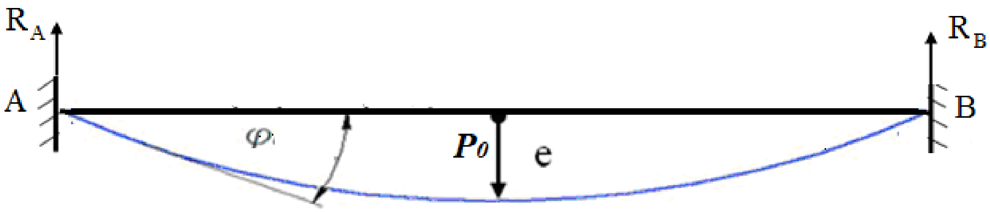

Fig. 4. Calculation scheme of the shaft for deflection.

We determine the critical number of revolutions of the proposed shaft:

$$n_K = \frac{30}{\pi}\frac{1}{\sqrt{\alpha m}} = \frac{30}{3.14}\frac{1}{\sqrt{0.485\cdot 10^{-6}\cdot 227}} = \frac{30\cdot 10^3}{33} = 909\ rpm$$

where,

$$\alpha = \frac{L^3}{48EJ} = \frac{3^3}{48\cdot 2\cdot 10^{11}\cdot 0.579\cdot 10^{-5}} = 0.485\cdot 10^{-6}$$

$m = m_{c1} + \frac{1}{2}m_n = 179 + \frac{1}{2}96 = 227$ kg, $m_{c1} = m_c - m_n = 275 - 96 = 179\ kg$, $E = 2\cdot 10^{11} P\alpha$ - Young's modulus, L=*3 m* - total length, $D_1 = 0.1m, D_2 = 0.08m,\ J = 0.579\cdot 10^{-5}\ m.$
The presence of disc eccentricity is taken into account by the formula:

$$C = \omega^2 m(y + e) \qquad (7)$$

We calculate the shaft for deflection according to the calculation scheme (Fig. 5).

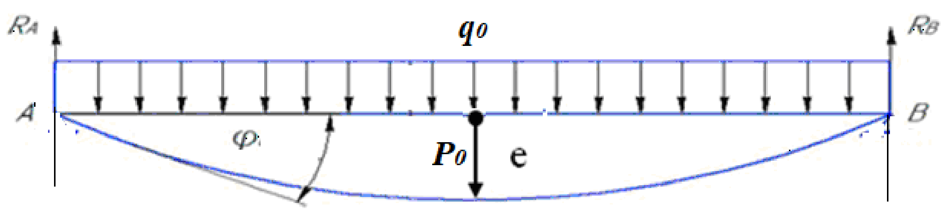

Fig. 5. Calculation scheme of the shaft for deflection.

In this case, the deflection will be distributed according to the law:

$$y = q_0 x^2 (x - L)^2 / 24EJ$$

Assuming $x = L/2,\ q_0 L = 1,$ we obtain an expression for the compliance coefficient (inverse stiffness coefficient):

$$\alpha = \frac{L^3}{8} \cdot 48EJ = \frac{0.9709 \cdot 10^{-6}}{8}\ m/n$$

Then we have:

$$n_K = \frac{30}{\pi}\frac{1}{\sqrt{\alpha m}} = \frac{30}{3.14}\frac{2\sqrt{2}}{\sqrt{0.9709 \cdot 10^{-6} \cdot 227}} = 2\sqrt{2} \cdot 643.55 = 1820\ rpm$$

From the condition of equality *F* = *C*, we write equations (7) in the form:

$$y = \frac{e\omega^2 m}{\frac{1}{\alpha} - \omega^2 m} = \frac{e}{\frac{\omega_K^2}{\omega^2} - 1} \tag{8}$$

To determine its numerical value:

$$y = \frac{e\omega^2 m}{\frac{1}{\alpha} - \omega^2 m} = \frac{e}{\frac{\omega_K^2}{\omega^2} - 1} = \frac{1.1}{\frac{909^2}{730^2} - 1} = \frac{1.1}{0.55} = 2\ mm$$

Investigating equation (8), it can be noted that at $\omega_K < \omega$ the directions of deflection and eccentricity coincide, and at $\omega_K > \omega$ the deflection $y < 0$ and the directions of deflection and eccentricity are opposite. For the $\omega \to \infty$ $y = -C$ case, i.e. the center of gravity of the disc is on the axis of rotation.

It is known that in working machines, supports are pliable; then the center of gravity of the disk will rotate around a circle with a radius of y_0, and the center of the support will rotate around a circle with a radius:

$$y_0 = \alpha_0 R = 2 \cdot 10^{-9} \cdot 0.45 = 0.9 \cdot 10^{-10} m$$

where, α_0- support compliance; for rolling bearings of shafts with a diameter of 60-80 mm,

$$\alpha_0 \approx (1 \div 3) \cdot \frac{10^{-6}\ cm}{kg} = (1 \div 3) \cdot 10^{-6} \cdot 10^{-2} \frac{m}{10N} = 2 \cdot 10^{-9}\ \ m/n$$

$R = \frac{1}{2}C$ - reaction in supports from centrifugal force.

At the moment of loss of stability of the shaft:

$$C = \omega_K^2 m y \tag{9}$$

or

$$C = \frac{1}{\alpha}(y - y_0) \tag{10}$$

$$C = \frac{1}{\alpha}(y - y_0) = \frac{1}{0.48 \cdot 10^{-6}}(2.0 \cdot 10^{-3} - 0.9 \cdot 10^{-10}) = 0.427 \cdot 10^{-7}$$

Replacing the value y_0 in formula (10), we get:

$$C = \frac{1}{1 + \frac{1}{2}\frac{\alpha_0}{\alpha}} \cdot \frac{1}{\alpha} y = \frac{1}{1 + \frac{1}{2}\frac{2 \cdot 10^{-9}}{\frac{0.48 \cdot 10^{-6}}{8}}} \cdot \frac{1}{0.48} 2 \cdot 10^{-3} = 0.4 \cdot 10^{-2} \tag{11}$$

From formulas (9) and (11):

$$\omega_K = \frac{1}{\sqrt{\alpha m}} \cdot \frac{1}{\sqrt{1 + \frac{1}{2}\frac{\alpha_{00}}{\alpha}}} = \omega_{KJ} \frac{1}{\sqrt{1 + \frac{1}{2}\frac{\alpha_{00}}{\alpha}}} = \frac{10^3}{10.49} \frac{1}{\sqrt{1 + \frac{1}{2}\frac{2 \cdot 10^{-9}}{\frac{0.48 \cdot 10^{-6}}{8}}}} = \frac{1 \cdot 10^3}{10.7} = 1 \cdot 10^2\ sec^{-1} \tag{12}$$

where, ω_{KJ} - critical angular velocity of a shaft on rigid supports.

4. Conclusions

The designs of the saw cylinder of 130 saw gins were studied. The influence of the design and the increase in radial runout on the performance of the genie, on the quality of the fiber and the consumption of electricity and spare parts are determined. To reduce radial runout, it is proposed to reduce the weight of the cylinder by using a hollow shaft. When using a hollow shaft for the stability of the saw cylinder, the critical speed of the cylinder was theoretically studied. As a result, the study determined that at the critical angular velocity $\omega_K < \omega$ the directions of deflection and eccentricity coincide, and at $\omega_K > \omega$ the deflection and the directions of deflection and eccentricity are opposite. For the $\omega \to \infty$ $y = -C$ case, i.e. the center of gravity of the disc is on the axis of rotation.

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