ISSN 2181-8622

Manufacturing technology problems

NAMANGAN MUXANDISLIK-TEXNOLOGIYA INSTITUTI

Scientific and Technical Journal Namangan Institute of Engineering and Technology

INDEX COPERNICUS INTERNATIONAL

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Volume 9
Issue 2
2024

DEFORMATION OF THE SHAFT IN TORSION AND THE EFFECT OF TORSION ALONG WITH BENDING

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Abstract: Insufficient smoothness of the saw cylinder shaft leads to impermissible misalignments in the gap between the columns, which has a very negative effect on the grinding process. In this case, using the method of equivalent diameters developed by B.P. Jemochkin, we determine the bending and turning angles of the saw cylinder shaft.

Keywords: shaft, singularity, bending vibrations, saw cylinder, slack, technological slot, saw, colossian

Introduction. It is known that the improvement of technological machines and their working parts produced for cotton ginning enterprises in our Republic, the creation of resource and energy-efficient types and the growth of their level of competitiveness only by applying new progressive technological processes using appropriate technological equipment and developing their construction and conducting theoretical and practical research on compliance with new requirements has been emphasized in many research works. In particular, the issues of high-efficiency, resource and energy-saving, high work productivity and improvement of product quality indicators of cotton pre-processing machines have always been and remain relevant in the industry. The main working part of sawing machines consists of a sawing cylinder and a rib cage. As a result of the interaction of these two working bodies with each other, the processes of the machine work are carried out. The saw cylinder shaft is important in the grinding process. The research focuses mainly on improving resource efficiency by reducing the mass of the saw cylinder shaft, increasing machine productivity and ensuring product quality while reducing energy consumption and bending. The permissible deflection of the saw cylinder shaft is not more than 0.3-0.4 mm. It is allowed that the transverse deflection of the saws in rotation is not more than 0.15 mm, because otherwise the position of the saw will change in the slotted gap between the columns, which causes fiber damage as the fiber is pulled through the colostrums by the saw teeth. The rotation speed of the saw shaft in the existing constructions of sawing machines is 730 rpm.

In order to eliminate the indicated shortcomings, there is a need to identify the causes of low reliability, to develop new constructions that allow to increase the performance of gins while maintaining the quality of the fiber and reducing its cost.

Materials and methods. When determining the uniformity according to B.P. Jemochkin method, the shaft is considered as a stepped beam lying on two supports (Fig. 1a). If the moment of inertia J_1 moment of inertia of a beam of constant cross-section $J_0=kJ_1$ if it is replaced by a hammer and in this case all loads and reactions

$$\kappa = \frac{J_0}{J_1}$$

times, then the proposed shaft curve matches the default hammer curve.

We will use this transformation and change the existing shaft of the saw cylinder to an equivalent shaft with a hollow section. Cutting the saw shaft into separate parts with constant moment of inertia J_1, J_2 and J_3 and imposing transverse force Q and bending moments M on the edges of the section (Fig. 1.b), each step of the shaft is subjected to the external loads imposed on it and together with internal forces, we change to a shaft with a constant moment of inertia along its length.

Multiplication coefficients for each step

$$\kappa_1 = \frac{J_0}{J_1},\ \kappa_2 = \frac{J_0}{J_2},\ \text{va}\ \kappa_3 = \frac{J_0}{J_3}$$

We assume that the moment of inertia of the equivalent shaft is equal to the moment of inertia $J_0=J_1$, then

$$\kappa_1 = \frac{J_0}{J_1} = 1;\ \kappa_2 = \frac{J_0}{J_2} \langle 1;\ \kappa_3 = \frac{J_0}{J_3} \rangle 1.$$

Bending moments M_1, M_2, M_3 and transverse forces Q_1, Q_2, Q_3 are determined from the diagrams or analytically calculated. A diagram of a shaft driven by external loads and internal forces and having a constant moment of inertia J0 multiplied by the matching coefficients of sections (Fig. 1. v) is shown. Fig. 1. g shows an equivalent shaft, in which the given loads changed by ki times, additional loads and moments where the sections are touching, equal differential movements ΔQ_i and ΔM_i, and they are determined for each section.

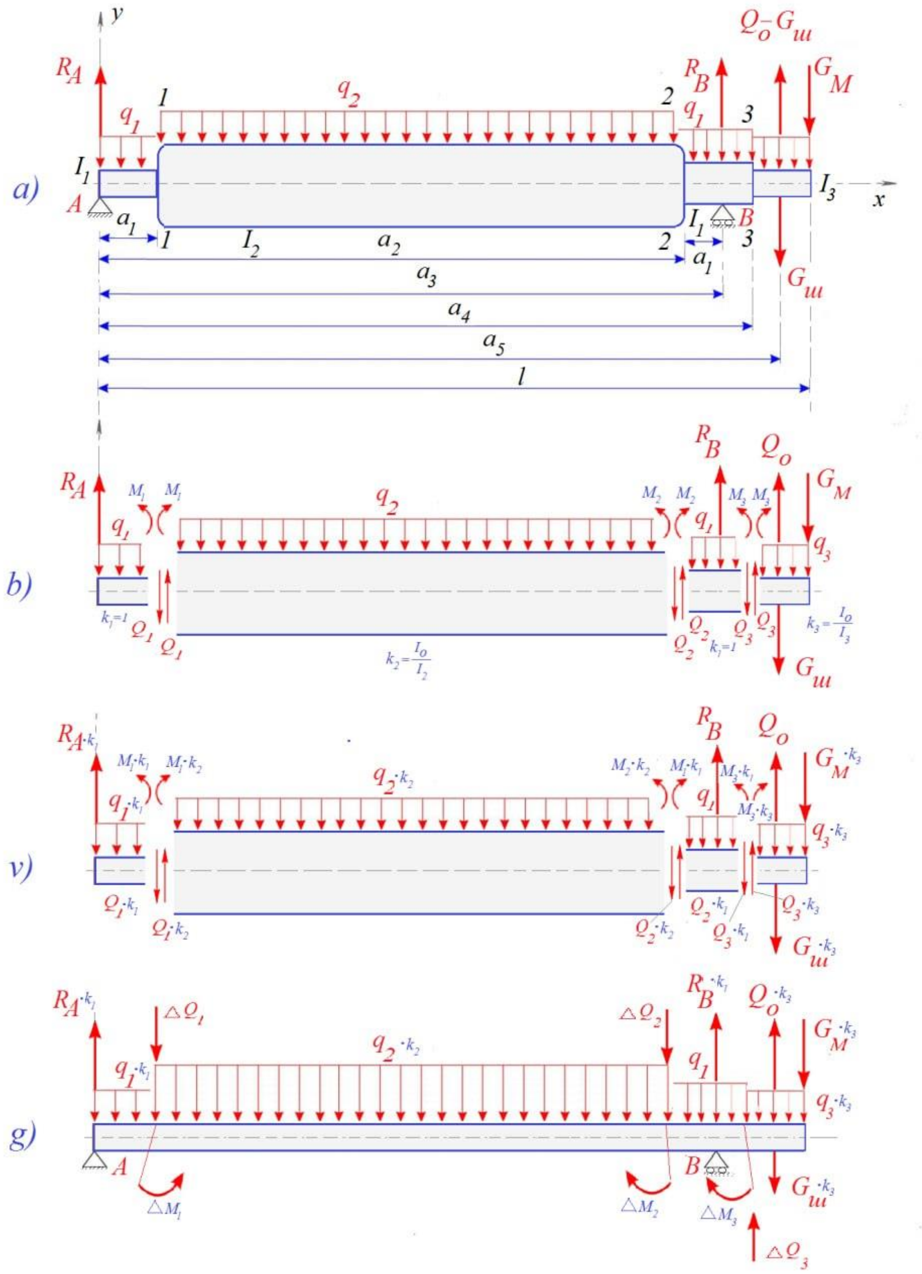

Figure 1. Scheme for determining the static stiffness of the saw cylinder shaft.

Shaft bends determined by this method

x=168,4 sm; y=-0,0064 sm;

x=87,4 sm; y=-0,0585 sm;

x=40,5 sm; y=-0,039 sm;

x=6,4 sm; y=-0,0065 sm.

Based on the above, we determine the moment of resistance. According to the consistency condition

$$\tau_{\max} = \frac{M_0^{\max}}{W_\rho} \le [\tau]$$

Based on this;

$$W_{\rho} = \frac{M_b^{\max}}{[\tau]} = \frac{10000kg \cdot sm}{1500kg \cdot sm^2} = \frac{100}{15} = 6{,}6sm^3$$

Polar moment of inertia

$$J_p = \frac{\pi d^4}{32} = 0{,}1 \cdot 10^4 = 1000sm^4$$

$$G = 8 \cdot 10^5 \ \ kg/sm^2.$$

$$\varphi = \frac{M_b^{\max} \cdot l}{G \cdot J_p} = \frac{10000 \cdot 300}{8 \cdot 10^5 \cdot 1000} = 3.7 \cdot 10^{-3} rad$$

If the cylindrical steerer working in torsion acts as a shaft, in order to lighten it, in order to reduce its bending and save resources, the middle part is carved while keeping the permissible bending; in that case, the shaft takes the form of a groove, that is, it is called a hollow shaft. In our prohibitions, this issue is considered as the main issue. Hollow shafts do not reduce the torsional resistance much because the main stress is in the flange of the cross section, decreases in the middle parts and is zero in the center.

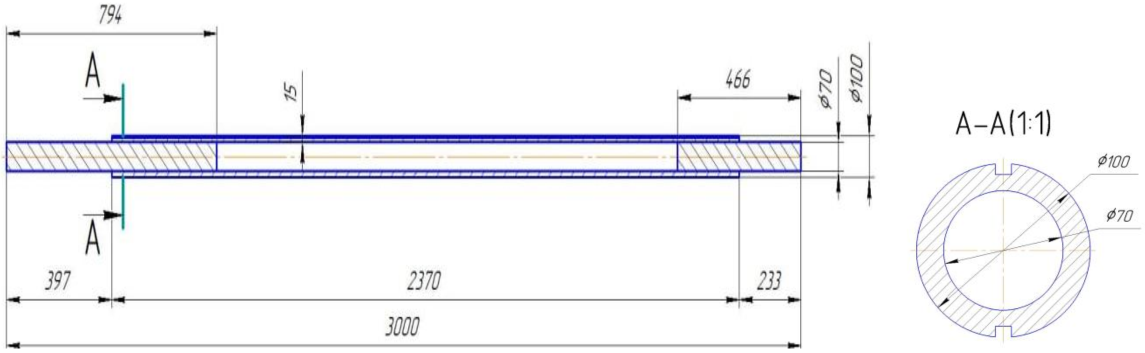

Figure 2.a. Proposed Hollow Shaft Design.

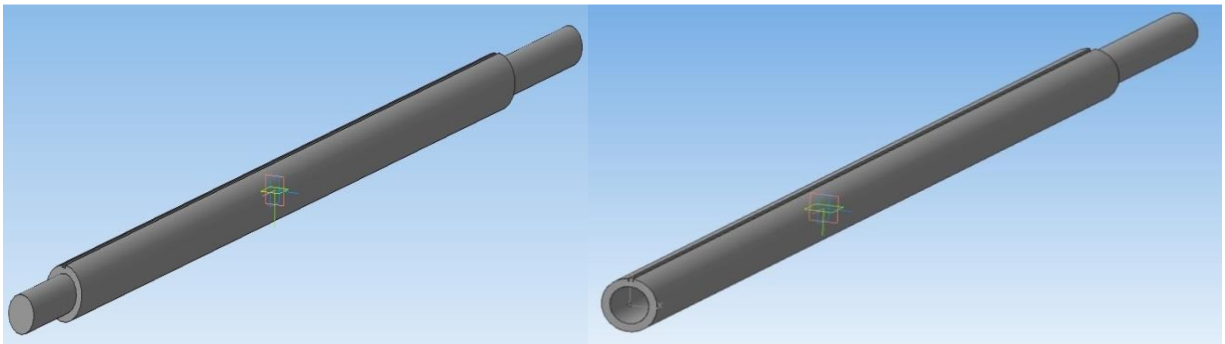

Fig. 2. b. 3D view of the proposed hollow shaft construction.

If the outer radius of the section is R, and the inner radius is r (Fig. 2.a.), the integral defining Ip is written as follows.

$$I_p = 2\pi \int_r^R \rho^3 d\rho = \frac{\pi}{2}\left(R^4 - r^4\right)$$

(1)

$$I_p = \frac{\pi}{32}\left(D^4 - d^4\right) \approx 0{,}1\left(D^4 - d^4\right);$$

from this, the resistance moment is equal to the following expression;

$$W_p = \frac{I_p}{\rho_{\max}} = \frac{\pi\left(R^4 - r^4\right)}{2R} = \frac{\pi\left(D^4 - d^4\right)}{16D}$$

(2)

We see that the moment of inertia of the pole and the moment of resistance have a certain value for each cross-section and depend on the dimensions of the cross-section.

$$M_b = 9736\frac{75}{730} = 1000 N \cdot m$$

And the strength limit

$$[\sigma] = \frac{\lfloor \sigma_{oq} \rfloor}{n}$$

is equal to the following expression. Here n=2.5 for plastic area. For 45XN

$$[\sigma] = \frac{7500}{2{,}5} = 3000 kg / sm^2$$

And the test voltage

$$[\tau] = 0{,}5 \div 0{,}6[\sigma]$$

subject to condition. Accordingly

$$[\tau] = 0{,}5 \cdot 3000 = 1500 kg / sm^2$$

we accept that.

As we know, a saw gin cylinder consists of a toothed saw disk that enters the shaft groove, seals between the saws, washers, and a compression nut.

We considered the rotation of the shafts under the influence of torques and their transverse rotation relative to each other above in the problems related to the calculation of shafts. The twist angle from equation (1).

$$\frac{d\varphi}{dx} = \frac{M_b}{GI_\rho}$$

(3)

if we integrate over x:

$$\varphi_x = \frac{M_b}{GI_p}x + c$$

originates. We find the optional constant (c) from the condition that the fixed section of the shaft does not move. When x=0, the section does not move, that is, Therefore c=0; means;

$$\varphi_x = \frac{M_b}{GI_p} x$$

If the length of the shaft is l, then the largest turning angle is at the defined section of this length. That is, it is equal to the following expression.

$$\varphi = \frac{M_b l}{GI_p}$$

The angle of twist, for a given torque, depends on the amount of the multiplier GIp. We call this quantity the unit of torsion. In this case, GI_P reflects the influence of the physical properties of the material of the torsion shaft and the cross-sectional dimensions on the torsional deformation. The issue of determining the angle of rotation has twofold importance. First, when solving static indeterminate problems, the additional equation is determined by checking the deformation; secondly, and most importantly, determining the torsion angle is necessary to check the integrity of the shafts.

To ensure the normal operation of the machines, the angle of rotation should not exceed a certain amount. For example, for machine shafts operating under normal conditions, the permissible twist angle

$$[\varphi] = 0{,}3^0$$

should be. For machine shafts operating under variable load.

$$[\varphi] = 0{,}15^0 - 0{,}25^0$$

should be.

It is necessary to determine the dimensions of the shafts not only from the condition of strength, but also to satisfy the condition of uniformity:

$$\varphi = \frac{M_b \cdot l}{GI_\rho} \le [\varphi] \cdot l \quad \text{or} \quad \frac{M_b}{GI_\rho} \le [\varphi] \tag{4}$$

We calculate the moment of polar inertia for the cross-section of the hollow shaft and the moment of resistance in torsion. When calculating long shafts, bus hart is often preferable to the strength requirement.

$$W_\rho = \frac{\pi D^3{}_t}{16} - \frac{\pi D^3{}_i}{16};$$

In the current design, the rotational speed of the saw shaft is 730 rpm.

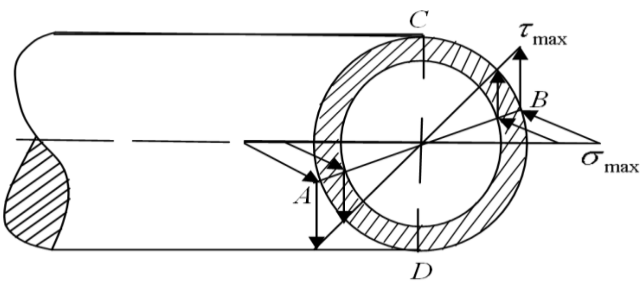

Fig. 3. Stresses acting on the cross-sectional surface of a hollow shaft.

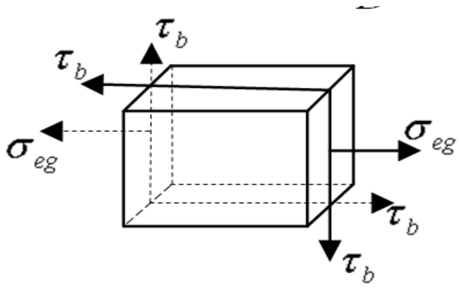

Fig.4. A cube-shaped element in a shaft parallel and perpendicular to the cross-sectional surface under investigation.

First, we calculate the normal stress caused by bending in this section, and then add the tangential stress caused by twisting. For a circular cross-section, plane bending occurs regardless of the plane in which the bending moment lies. Therefore, it does not matter in which plane the bending moment lies. In our case, the bending moment lies in the horizontal plane.

In that case, the greatest normal stress for the section we are looking at is created at points A and B at the ends of the horizontal diameter. We know that the maximum tangential stress generated by torsion occurs at the extreme points of the cross section. At points A and B, the tangential stresses will have the greatest value. Thus, the largest normal and tangential stresses are formed at points A and B in the section.

At points C and D at the ends of the vertical diameter, the greatest tangential stress from torsion and the greatest tangential stress from bending are added algebraically. But the value of these stresses is very small compared to the stresses at points A and B, so they can not be taken into account, because when checking the strength of the material, it can be confirmed by appropriate calculations that the state of dangerous stress for it is at points A and B. A cube-shaped element parallel and perpendicular to the cross-sectional surface of the shaft under investigation at a location near points A or B separates (2-fig.)

A tangential stress acts on the quadrilateral of this cube; two opposite vertical sides of these legs are also affected by normal stresses. Its other two oils are not affected by voltages. Therefore, the isolated element is in a flat voltage state. In such cases, in order to check the strength of the material, we first determine the principal stresses and and then check their values according to the theory of strength. That is;

$$\sigma_{1,2} = \frac{1}{2}\left\{\left(\sigma_x + \sigma_y\right) \pm \sqrt{\left(\sigma_x - \sigma_y^2\right) + 4\tau_{xy}^2}\right\} \tag{5}$$

We use the following expression. However, instead of normal and tangential stresses in it, we put values of normal and tangential stresses in bending and twisting.

Results and discussion. The production of shafts is not a very complicated process, that is, the technology of preparation does not require a lot of labor and resources. The shaft of any machine is one of the most important parts because it works with heavy loads and in most cases at high speeds. It is known from the researches that with the increase in the length of the shaft, its stiffness decreases based on theoretical and practical experiments, i.e. it was observed that it bends more than the permitted one under the influence of loads.

Usually, shafts are used to transmit rotational motion over long distances within the limit of their construction. In mechanical engineering, shafts are widely used in various mechanisms of transport and agricultural machines, water vessels, metal cutting machines, mining and textile machines, especially in cotton ginning enterprises, mainly due to heavy size and work process. if we get it, relatively resource-intensive shafts are used. Long-dimension shafts include shafts whose length is ten times greater than their diameter [62]. The rotating shafts are mounted on the box parts of the mechanisms through sliding or rolling bearings. In most cases, the working shafts of textile machines consist of simple and high-precision surfaces with a straight cylindrical axis of rotation, surfaces with a slot or key slot, grooved, flanged parts. In preparation, all these types of surfaces are embodied in lathe spindles.

In particular, in order to ensure the strength of the shafts of the jin machines, the issues of finding the stress generated on the surface of the dangerous section from the force or moment acting on it and comparing it with the permissible value were not considered, as well as the construction of the shafts in the condition arising from the strength conditions. the possibility of lightening and in the research, we will develop a lightened construction of technological machine shafts of cotton ginning enterprises in the condition that their strength is ensured, and we will consider issues of maintaining product quality indicators by increasing machine productivity, reducing energy and resource consumption.

Conclusion In order to reduce the bending of the shafts of the long, heavy and fast-rotating parts of the technological machines, the longitudinal groove structure of the lightened shafts was developed, including for the saw cylinder, on the basis of which the technological gap between the saw cylinder and the fiber separation brush is sufficiently maintained, the fiber output is increased, pollution and power consumption are reduced.

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