THE INFLUENCE OF THE FRICTION FORCE TORQUE WITH THE TAPE ON THE ROTATION IRREGULARITY OF THE SUPPLY CYLINDER HEADSET OF THE DISCRETE ZONE OF THE SPINNING MACHINE

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Abstract. *An efficient design scheme of the supply cylinder with a rubber bushing of the recommended composition and working elements with knurled teeth in the discretizing zone of the spinning machine is given. Equations of motion of the machine unit, taking into account the driving mechanism and the characteristics of the engine flywheel, were obtained. Based on the solution of the problem, the values of the parameters of the supply cylinder are recommended.*
Keywords: *supply cylinder, rubber bushing, unit, fiber tape, friction force, torque, knurled working element, engagement, angular velocity, shaft, gasket, coverage technological resistance.*

Calculation scheme and mathematical model. As noted, the recommended supply cylinder is composed of a rubber bushing between the shaft and the grooved gasket [1, 2]. In order to separate the fibers from each other, working elements with prismatic riffles of four different variants are placed in the grooves on the surface of the garnet. In this case, the frictional force between the corrugated prismatic working elements and the fiber tape changes during one rotation of the supply cylinder. This causes the angular velocity of the headset to change. As a result, an angular acceleration is created, which ensures that the fibers move relative to each other, they become parallel to each other and lead to a quality yarn. For this purpose, it is important to determine the limit of change of the angular speed of the supply cylinder gasket depending on the thickness of the rubber bushing and the change of the friction moment between the working elements and the fiber tape. For this, we build dynamic and mathematical models of the machine unit. Figure 1 shows the calculation scheme of the machine unit.

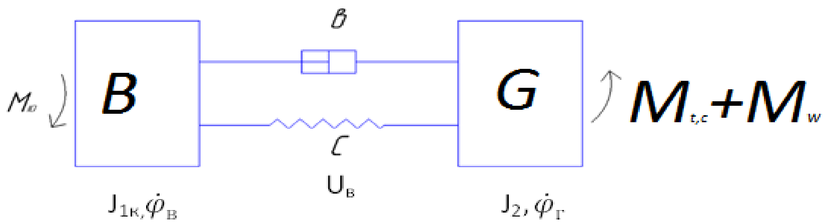


Fig.1. Supply cylinder mechanism machine unit calculation scheme

The given calculation scheme is a dynamic model of a two-mass machine unit, and the first mass represents the power supply and the shaft of the supply cylinder, while the second mass is taken into account.

Using Lagrange's II-order equation [3,4], we derive the differential equations representing the laws of motion of the supply shaft and gasket. In this case, the kinetic and potential energies of the system and the dissipative function [5, 6]:

$$T=\frac{1}{2}(J_{1k}\,\dot{\varphi}_{в}^{2}+J_2\dot{\varphi}_{g}^{2});\ П=\frac{1}{2}c(\varphi_{в}-И_{в2}\varphi_{g})^2;\ Ф=\frac{1}{2}b(\dot{\varphi}_{в}-И_{в\alpha}\dot{\varphi}_{g})^2 \quad (1)$$

here J_{1k}, J_2 - the moment of inertia of the electric drive rotor on the cylinder shaft and the moment of inertia of the first one with the working elements of the gasket bushing, respectively; $\dot{\varphi}_{в}, \dot{\varphi}_{g}$ - supply cylinder shaft and gasket angular velocities; $И_{в2}$ - transmission ratio; C, v - rubber bushing rotational uniformity and dissipation coefficients. The torque of the electric motor is taken into account through the mathematical mechanical characteristic [7, 8]:

$$M_{yu}=\frac{2M_{к}(1-\varepsilon)}{S/S_k+S_{к}/S};\qquad Ɛ=\frac{1}{\sqrt{r_a^2+r_u^2}} \quad (2)$$

hare $M_{ю}$, M_k - current of electric conductor and its critical value; S, S_k - slip in an asynchronous motor and its critical value; r_a, $r_и$ - short circuit active and inductive resistance.

In this case, the critical value of slip in an asynchronous electric conductor [9]:

$$S_{к}=S_{H}\left(\frac{M_{к}}{M_{н}}+\sqrt{\frac{m_{к}^2}{M_{н}^2}-1}\right);\qquad S_{H}=\frac{\omega_0-\omega_{н}}{\omega_0} \quad (3)$$

hare $M_{н}$, S_H - rated torque and slip; $\omega_{н}$ - nominal angular velocity; ω_0- angular velocity for the ideal position of the rotor [7, 9]:

The supply cylinder gasket is affected by technological resistance and friction torque:

$$M_{т,к}=M_1\pm\Delta M_1;\qquad M_{ишк1}=\left(f+\frac{lb_1}{S_0}\right)\cdot Fr_{ц} \quad (4)$$

$$M_{ишк2}=\left(f+\frac{lb_2}{2S_0}\right)\cdot Fr_{ц}$$

$$M_{ишк3}=\left[f+\frac{1}{S_0}(l_1+t)кb_3\right]\cdot Fr_{ц};$$

$$M_{ишк4}=\left(f+\frac{кlb_4}{S_0}\right)\cdot Fr_{ц}$$

$$\left[\left(G_{в}+G_{рв}+G_{г}\right)+\frac{G_{в}+G_{рв}+G_2}{2}\left(\frac{\pi\, n_{ц}}{30}\right)^2+\frac{C_{рв}C_{л}}{C_{рв}+C_{л}}\left(1-cos\frac{\alpha}{2}\right)\right]$$

hare $M_{ишк1}$, $M_{ишк2}$, $M_{ишк3}$, $M_{ишк4}$- moments of frictional forces acting on zones; $b_1 b_2, b_3, b_4$ – width of working elements; l – total length of working elements; l_1 – tooth thickness; k – number of teeth.

Taking into account the received expressions (1)-(4), and by defining the summaries of Lagrange equations corresponding to the generalized coordinates φ_v and φ_g, we create the following system of differential equations representing the motion of the machine assembly:

$$M_{yu}=\frac{2M_{к}(1-\varepsilon)}{S/S_k+S_{к}/S};\qquad Ɛ=\frac{1}{\sqrt{r_a^2+r_u^2}} \quad (5)$$

$$J_{1к}\ddot{\varphi}_{v}=M_{yu}-C(\varphi_{в}-И_{в2}\varphi_{г})-в(\dot{\varphi}_{в}-И_2\varphi_{г});$$

$$J_{в}\ddot{\varphi}_2=И_{в2}C(\varphi_{в}-И_{в2}\varphi_{г})+И_{в2}b(\dot{\varphi}_{в}-И_{в2}\dot{\varphi}_{г})-M_{тк}-(M_{ишк1}+M_{ишк2}+M_{ишк3}+M_{ишк4}) \quad (5)$$

It should be noted that the sum of the friction forces presented in the system (5) is conditionally given. During operation, their effect is carried out sequentially through a special program [11].

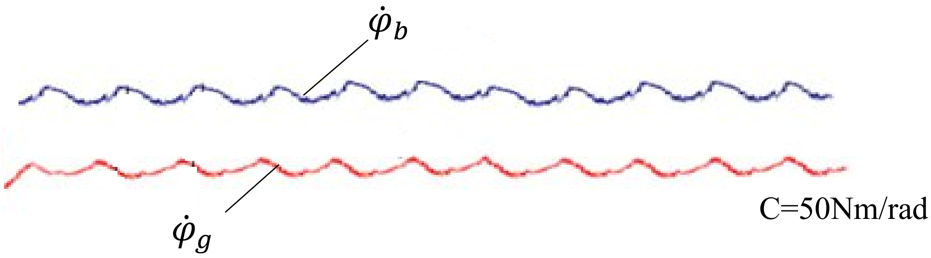
Numerical solution of the problem and analysis of the results. When obtaining the numerical solution of the obtained system (5), the values of M1 in Mtq are performed through a numerical generator [11,12]. The results obtained through experimental studies [12] are taken into account. The following initial values of the parameters were used: $n_{в}$=10 rotation/minute; $M_k=6\cdot10^{-2}$Nm; $M_n=1{,}2\cdot10^{-2}$Nm; S_k=0,31; S_n=0,26; ω_n=10,46C^{-1}; C=(25÷30)Nm/rad; b=(0,8÷1,2)Nms/rad; f=(0,15÷0,25); $r_c=(0{,}9\div1{,}2)\cdot10^{-3}$m; $И_{b2}$=1,0;

J_{1k}=0.21·10^{-2}kgm^2; J_2=0.133·10^{-2}kgm^2; C_{rv}=(20÷30) cN/mm; C_1=(3,5÷6,5) cN/mm; l'_1=0; b_1, b_2, b_3, b_4=(1,8÷2,5) ·10^{-2}м; k=(6,0÷8,0); l=(1,8÷2,2) ·10^{-2}m; g=9.81 m/s^2

The numerical solution was implemented in EHM. In this case, the law of motion of the supply cylinder was analyzed. It is important to determine the angular velocities of the supply cylinder shaft and gasket. Because the change in the angular velocity of the feed cylinder assembly changes the frictional force in contact with the fiber tape, affecting the fiber separation and transmission. It also determines the productivity and the quality of the obtained thread.

Figure 2 shows the laws of change of angular velocities of the supply cylinder shaft and gasket. The laws of change of the obtained φ˙_v and φ˙_g show that with the increase of the rotational uniformity coefficient of the rubber bushing in the supply, the vibration amplitudes φ_v and φ_2 increase, which especially has a significant effect on the set movement law (Fig. 2, graphs a, b).

a



b

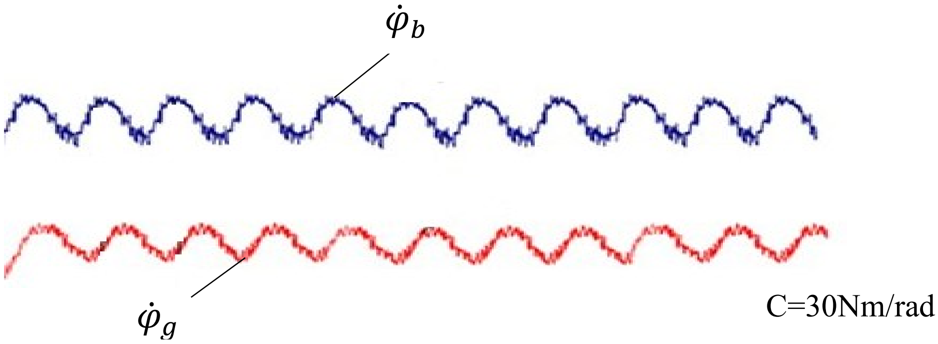


Figure 2. Laws of change of angular velocities of supply cylinder shaft and gasket

It should be noted that the influence of the torque of the friction forces in the interaction of the fibrous tape with the cylinder gasket, which provides the laws of change of $\Delta\dot{\varphi}_в$ and $\dot{\varphi}_г$, is high. As a result of these studies, Fig. 3 presents the laws of change of the angular velocities of the supply cylinder shaft and gasket depending on the change of the torque of the friction force with the fiber tape. According to the analysis of the obtained laws, with the increase in the torque of the friction forces, the angular velocity changes of the cylinder shaft and gasket, especially the values of $\Delta\dot{\varphi}_в$ and $\dot{\varphi}_г$ increase (Fig. 3, graphs a,b,v). was built.

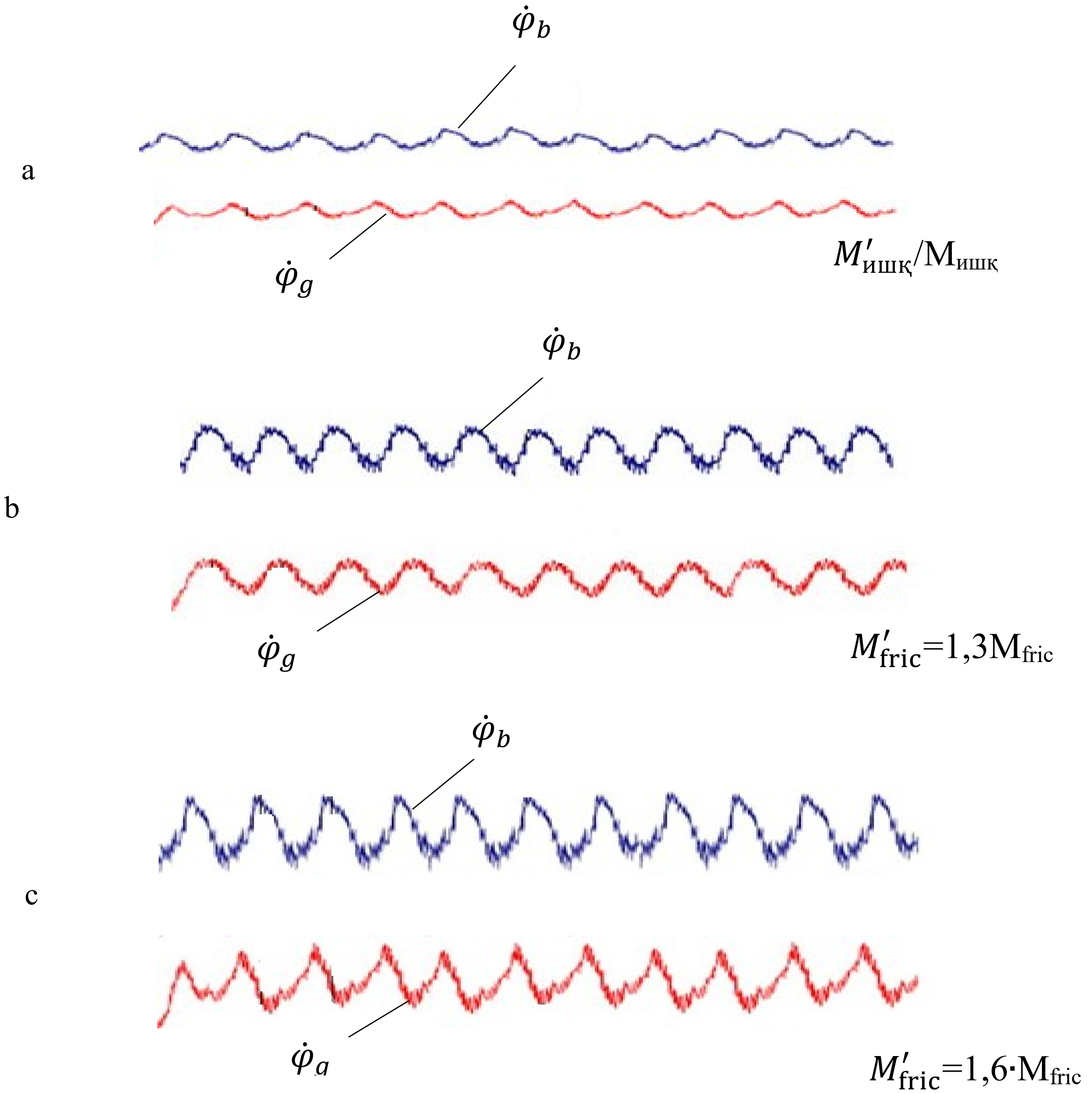
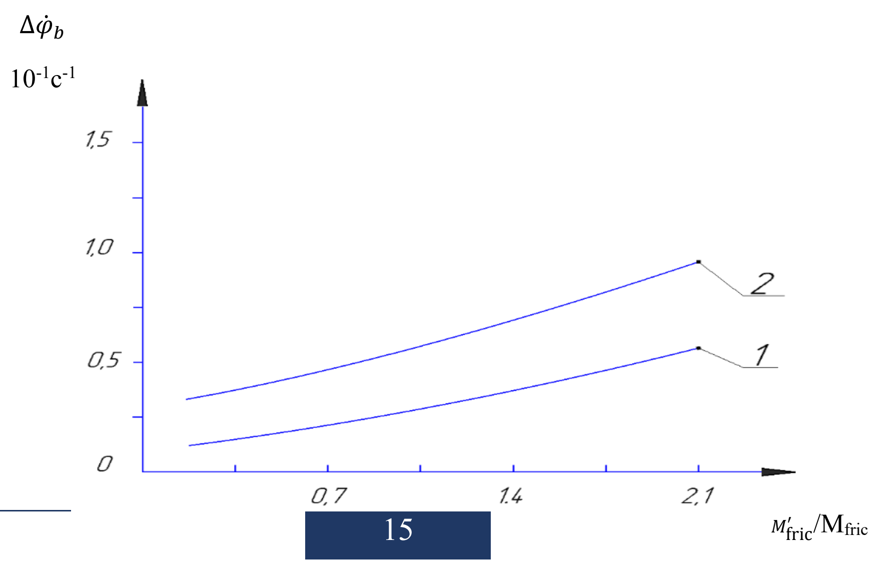


Figure 3. The laws of change of the angular velocities of the supply cylinder shaft and gasket depending on the change of the torque of the friction force with the fiber tape

Fig. 4 shows the graphs of dependence of angular velocities of the discretizing zone of the cylinder shaft and gasket on the relative change of the torque of the friction force with the fiber tape.



1-$\Delta\dot{\varphi}_в$=f(M'_{fric}/M_{fric});

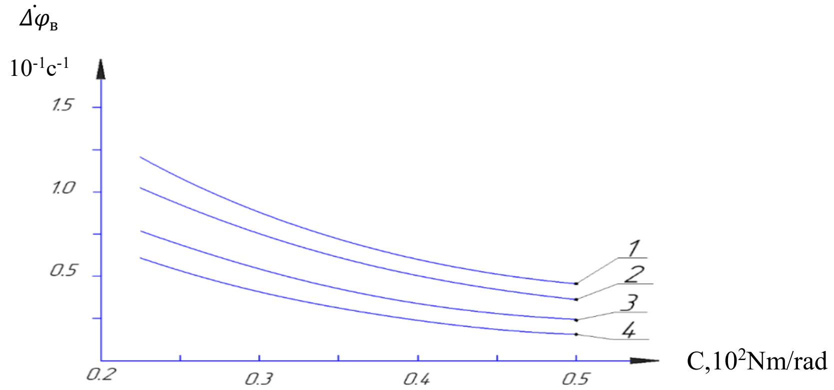
2-$\Delta\dot{\varphi}_2$=f(M'_{fric}/M_{fric})

Figure 4. Graphs of the dependence of the discretization zone on the relative change of the torque of the friction force with the fiber tape of the angular velocities of the supply cylinder shaft and gasket

The analysis shows that the average values of the moment of friction forces in the zones compared to the calculation values, i.e., M'_{fric}/M_{fric} when the values are increased from 0.35 to 2.0, the angular velocity of the shaft of the supply cylinder $\Delta\varphi_v$ increased from 0.12·10-1 s^{-1} to $0.46\cdot10^{-1}$ s^{-1} in a nonlinear pattern, the angular velocity range of the headset is slightly higher, increasing from $0.28\cdot10^{-1}$ s^{-1} to $1.15\cdot10^{-1}$ s^{-1} in nonlinear coupling.

The main reason for this is due to the corresponding effect of the rubber bushing. Therefore, to ensure $\Delta\varphi_2$=values in the range of (0.12÷0.2)s^{-1}, i.e., the values of the moment of friction forces to separate and sufficiently move the fibers of the fiber tape M_{fric}/M_{fric} ≥(1.2÷1 ,5) is suitable for the purpose.

Fig. 5 presents graphs of dependence of angular velocities of the discretizing zone of the cylinder shaft and gasket on the variation of the vibration ranges on the coefficients of rotation of the rubber bushing.



1,2-$\Delta\dot{\varphi}_в$ =f(C); *3,4-$\Delta\dot{\varphi}_2$ =f(C);*

1,3-f=0,11; *2,4-f=0,15*

Figure 5. Graphs of the dependence of the discretization zone on the angular velocity of the cylinder shaft and gasket, the variation of the vibration ranges, and the coefficients of the rotation uniformity of the rubber bushing

As mentioned above, the rubber bushing of the supply cylinder has a sufficient influence on the rotational uniformity of the angular velocities of the rotating masses, i.e., the roughness coefficient, including the coefficient of rotational uniformity of the rubber bushing from $0.25\cdot10^2$ Nm/rad to $0.5\cdot10^2$ Nm/rad when the angular velocity of the supply cylinder shaft increases when the range of oscillation f = 0.11 $\Delta\varphi_v$ values decrease from $0.51\cdot10^{-1}$ s^{-1} to $0.12\cdot10^{-1}$ s^{-1}, the values of $\Delta\varphi_2$ 0 decreases from $.63\cdot10^{-1}$ s^{-1} to $0.2\cdot10^{-1}$ s^{-1}.

Accordingly, when increasing f = 0.11, the values of $\Delta\varphi_v$ decrease from $0.87\cdot10^{-1}$ s^{-1} to $0.39\cdot10^{-1}$ s^{-1}, while the values of $\Delta\varphi_v$ are sufficiently high, 1, It decreases from $18\cdot10^{-1}$ s^{-1} to $0.46\cdot10^{-1}$ s^{-1} in nonlinear coupling. In this case, it is desirable that the rotational uniformity coefficient of the bushing is within the range of C≤(0.25÷0.3) 10^2Nm/rad in order to ensure the recommended values of the angular velocity of the supply cylinder.

Summary. A mathematical model of the machine assembly of the discretizing zone of the spinning machine, the supply cylinder mechanism was obtained. Based on the analysis of the obtained motion laws and connection graphs, the values of the system parameters were determined.

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