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**NEW EFFECTIVE STRUCTURAL SCHEME AND PRINCIPLE OF
OPERATION PARAMETERS FOR YARN STRETCHING TOOL OF RING
SPINNING MACHINE**

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***Аннотация.** The paper written about a new effective design scheme and the principle of operation of a machine for stretching yarn of a ring-spinning device based on the parameters of belt bushings. The mathematical model and the results of solving the problem of oscillations of the axis of the composite squeezing roller of the drawing apparatus of the spinning machine have been determined. The recommended values of the roller parameters are given. Based on the analysis of the research results and the conditions for obtaining high-quality yarn, the recommended parameter values for each squeezing rollers were determined.*

On a ring spinning machine, the fiber is stretched on a spinning machine and sufficient twists are given [1]. According to the existing technology, in the elongation zone, the fibrous strands are stretched due to the difference in velocities between the three rifle and three elastic-coated compression pairs, the rifles perform the corresponding twists [2].

The load on each load roller is carried out separately by means of springs mounted on a single lever. Each stretch juf T installed separately before the gaskets. To control the movement of the fibers, two bands are installed in the second elongation zone : upper and lower. The lower belt tension is done using a spring bracket. The spacing between the upper and lower bars of the strips can be changed depending on the

thickness of the product [2, 3] . The change in distance between the slats is changed by means of interchangeable supports. The disadvantage of the stretching device considered is that the fiber product is not evenly distributed in the stretching zones, moreover, the flexible interchangeable bushings do not allow the fibers in the yarns to have the same elongation [4].

The proposed improved stretching tool consists of three pairs of equipment with three rifle bottom cylinders 1,2,3 and three load rollers 4,5,6 with elastic coating on the top, and a spring loading lever 7 [5] . Elastically coated rollers 4,5,6 axis are connected by conical springs 8,9,10 (Fig. 1).

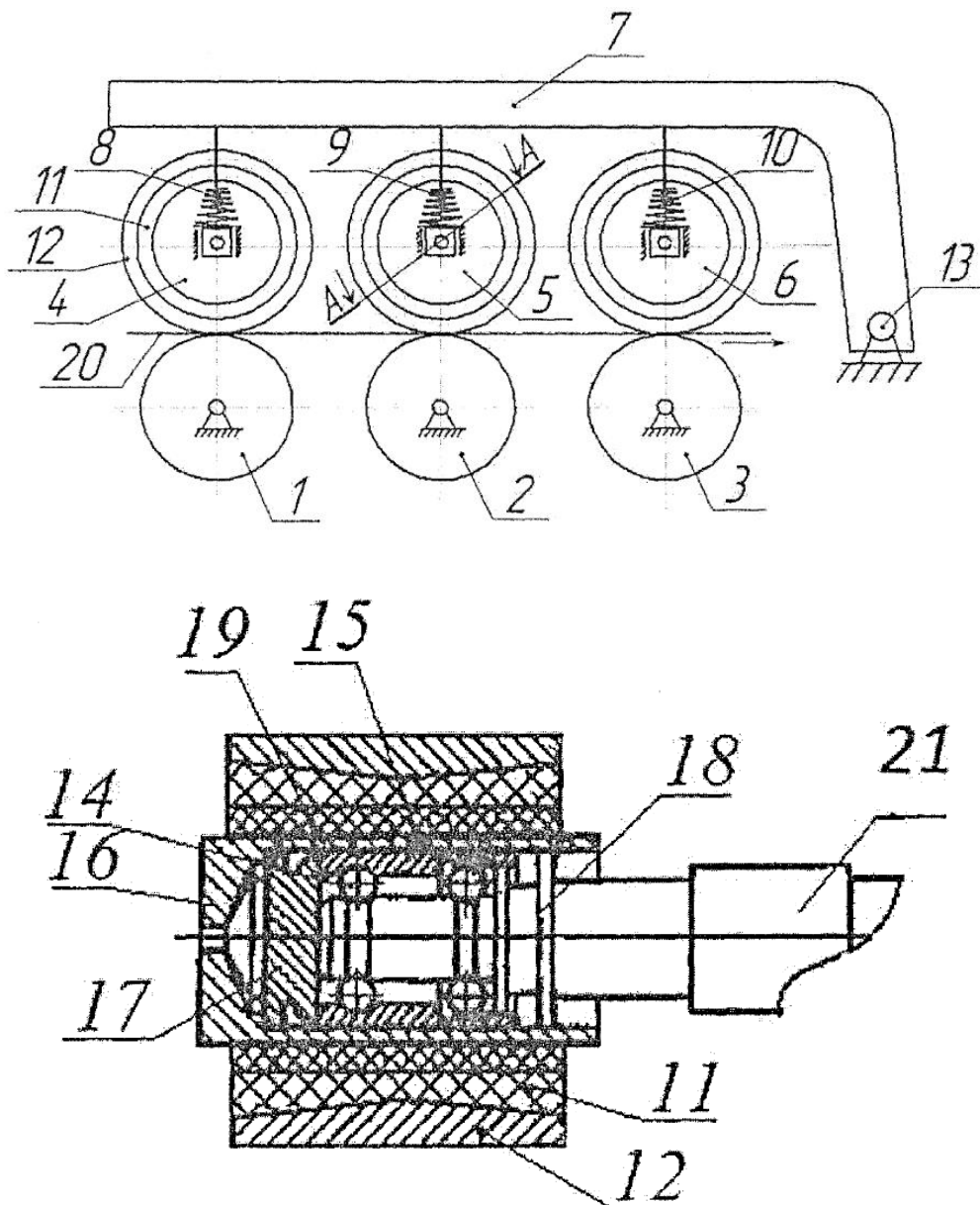


Figure 1. Schematic of an elongated tool with a rolled coating flexible structure.

The springs have 8,9,10 virgins, selected in such a way that their unity increases in the direction of the spindle, and $C_1 < C_2 < C$ [5].

We construct a differential equation that represents the oscillations of the axis of the flexible bushing shaft of each loader with a loading device:

$$(m_y + m_p + m_b + m_l)x'' + (b_1 + b_2 + b_l)\dot{x} + \frac{c_1 c_2 c_l}{c_1 c_l + c_2 c_l + c_1 c_2} x = F_0 \sin \omega t \quad (1)$$

where F_0 is the amplitude and frequency of change of the external force acting on the roller shaft from the ω -pin.

$c_1 c_2$ -rubber and outer bushing virginity coefficients, the coefficient of virginity of ω -pin

b_1, b_2, b_l are the dissipation coefficients of the rubber and outer bushings and the plug, respectively.

m_y -target mass; m_p -rubber bushing mass; m_l is the mass of the elongated plug in the part corresponding to the width of the roller.

The solution of the obtained differential equation (1) using the existing analytical method we obtain the following expression:

$$x = \frac{F_0 \sin(\omega t + \beta)}{\sqrt{\left[\frac{c_1 c_2 c_l}{(m_y + m_p + m_b + m_l)(c_1 c_l + c_2 c_l + c_1 c_2)} - \omega^2 \right] + \left[\frac{(b_1 + b_2 + b_l)\omega}{m_y + m_p + m_b + m_l} \right]^2}} \quad (2)$$

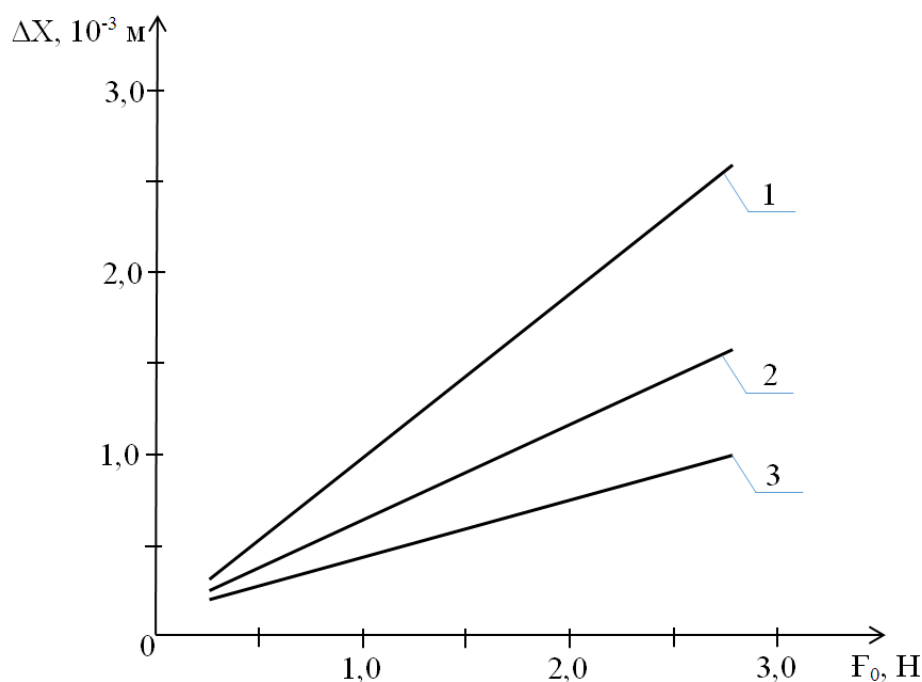
where, β is the spatial shift of the impact force.

To obtain a numerical solution of the problem, the initial calculated values of the parameters were obtained in the following intervals: $m_y = (0.1 \div 0.22)$ kg; $m_p = (0.02 \div 0.07)$ kg; $m_b = (0.05 \div 0.08)$ kg; $m_l = (0.003 \div 0.005)$ kg; $c_1 = (2.5 \div 4.0) \cdot 10^3$ N/m; $c_2 = (1.0 \div 1.5) \cdot 10^3$ N/m; $c_l = (0.08 \div 0.2) \cdot 10^3$ N/m; $b_1 = (0.35 \div 0.45)$ Ns/m; $b_2 = (0.25 \div 0.3)$ Ns/m; $b_l = (0.05 \div 0.11)$ Ns/m, $F_0 = (1,0 \div 3,0)$ N.

Based on the numerical solution of the problem, the law of vibration of the axis of the elongated tool shaft on the ring spinning machine was obtained in different

parameters, as a result of which graphs of the interaction of system parameters were constructed. Based on the analysis of the graphs shown in Figure 2, it was found that the oscillation coverage of the roller axis of the first loading elastic element increases linearly with the amplitude of the impact force. In particular, when the amplitude of the impact force increases from 0.29 N to 2.84 N, the vertical oscillation coverage of the third loading roller axis increases in a linear connection from $0,18 \cdot 10^{-3}$ m to $1,06 \cdot 10^{-3}$ m, while the vibration of the second loading roller axis increases. coverage increases from $0,25 \cdot 10^{-3}$ m to $1,52 \cdot 10^{-3}$ m.

Respectively, we can see that the oscillation coverage of the first loading roller axis increases from $0,31 \cdot 10^{-3}$ m to $2,51 \cdot 10^{-3}$ m (Fig. 2, graphs 1,2,3). Given the experimental results and the unevenness of the elongated shaft, the impact force is in the range of $(2.3 \div 3.0)$ N, the oscillation coverage of the first loading roller axis $(1,7 \div 2,4) \cdot 10^{-3}$ m, the second roller shaft $(1,3 \div 1,65) \cdot 10^{-3}$ m and the third roller axis vibration coverage is recommended to be in the range $(0,7 \div 1,1) \cdot 10^{-3}$ m..



$$1 - \Delta X_1 = f(F_0); \quad 2 - \Delta X_2 = f(F_0); \quad 3 - \Delta X_3 = f(F_0);$$

Figure 2. Graph of the dependence of the vertical vibration coverage of the axes of the elongated tool rollers on the change in the amplitude of the elongation resistance of the plug

Figure 3 shows graphs of the change in vibration coverage of the axes of the elongated rollers relative to the stiffness of the flexible bushings. The coupling graphs are obtained at the calculated value, when the outer bushing coefficient of the inner bushing of the first loading roller changes, the inner bushing coefficient does not change. This procedure is the same for rollers with all three loading components. The graphs in Figure 3 are constructed based on the given virginity coefficients of both rubber bushings, i.e., where the virginity has a nonlinear rigid characteristic. Based on the analysis of the graphs in Figure 3a, the oscillation coverage of the roller axis is $2.27 \cdot 10^{-3}$ when the applied mass of the first roller is increased from $0.6 \cdot 10^{-3}$ N / m to $3 \cdot 10^{-3}$ N / m when the applied mass is 0.15 kg. Decreases in nonlinear regularity from 3 m to $1.23 \cdot 10^{-3}$ m. If the given mass is taken as 0.2 kg, the vibration coverage is reduced from $2.5 \cdot 10^{-3}$ m to $0.67 \cdot 10^{-3}$ m. To ensure that the vibration coverage for the first roller does not exceed $(0.7 \div 1.1) \cdot 10^{-3}$ m, it is recommended that the coefficient of virginity of the rubber bushings be in the range $(1.68 \div 2.7) \cdot 10^3$ N / m. The recommended coefficients for the rubber bushings of the second and third loading rollers, respectively, are in the range of $C_{k2}=(1,85 \div 2,8) \cdot 10^3$ N / m and $C_{k3}=(2,6 \div 3,75) \cdot 10^3$ N / m.

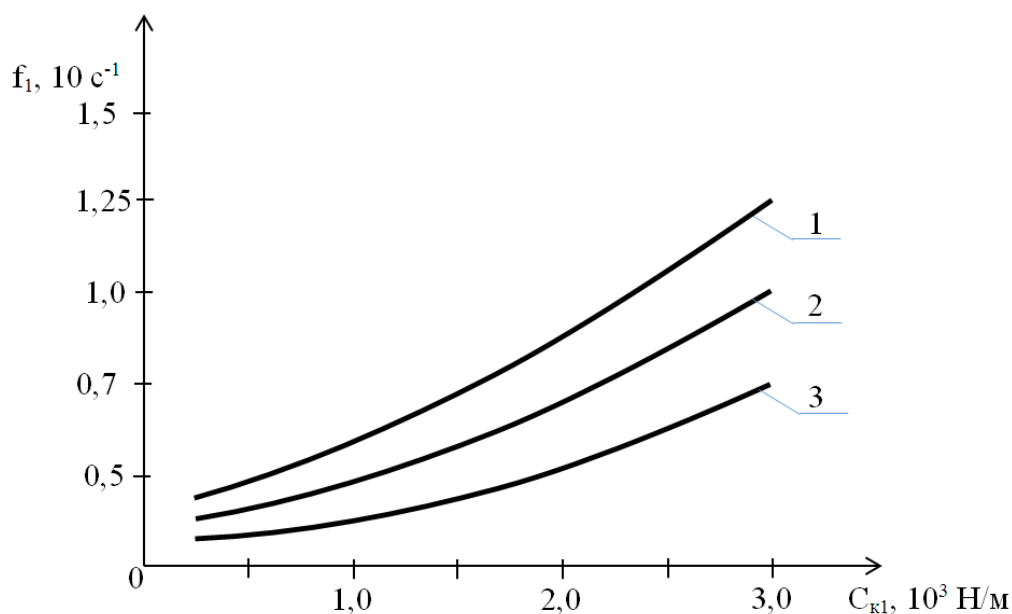
It is known that the frequency of specific vibrations with the frequency of change of force acting on the load compressing the rollers leads to resonance modes, the unevenness of the mutual compression of the fibers. Therefore, it is important to determine the limits of variation of specific and forced oscillation frequencies.

While the rotational frequencies of the rollers are mainly $(120 \div 180)$ rpm in existing spinning machines, the vibration frequencies of the roller shafts are reciprocating. in the range of $(2.0 \div 5.0) s^{-1}$.

Figure 3 shows the dependence of the change in the specific vertical vibration frequencies of the loading rollers of the spinning machine stretching tool on the coefficients of elasticity given by the flexible bushings. According to the analysis of the graphs in Figure 3a, the coefficient of virginity of the flexible loading bushings of the first loading roller increased from $0.3 \cdot 10^{-3}$ N/m to $2.8 \cdot 10^{-3}$ N/m and the specific

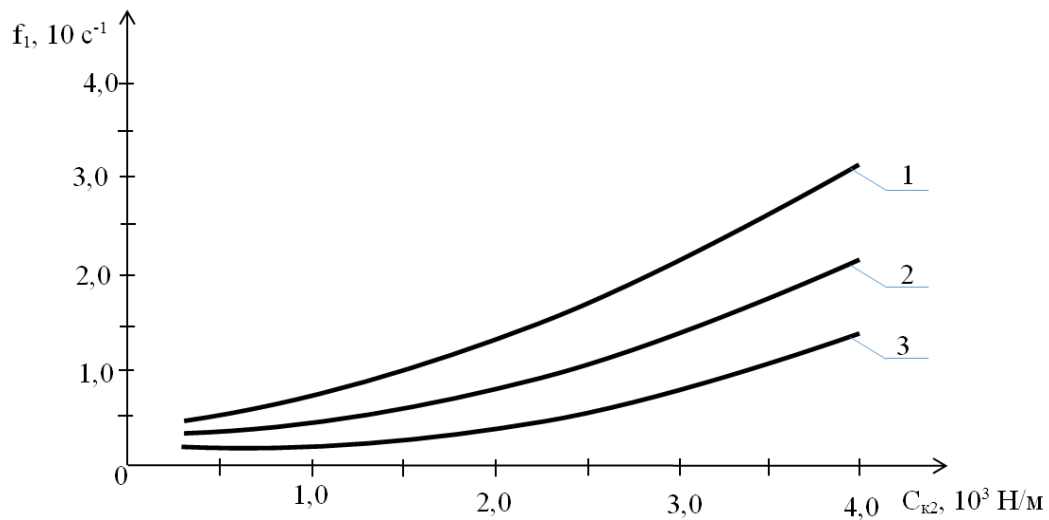
oscillation frequency of the roller was 1, when $m_{k1} = 0.185$ kg. Increases in linear regularity from 6 s^{-1} to 7.7 s^{-1} . When the applied mass of the roller is 0.115 kg, its axial vertical oscillation frequency increases in the range of 6.05 s^{-1} to 2.02 s^{-1} . It is recommended that the specific vibration frequency of the first roller be in the range of $(1.1 \div 1.8) \text{ s}^{-1}$ to ensure the recommended $C_k = (1.68 \div 2.7) \cdot 10^3 \text{ N/m}$ values.

Correspondingly, according to the analysis of Figure 3b, when the coefficient of virginity of the flexible rollers of the composite roller is increased from $0,31 \cdot 10^3 \text{ N/m}$ to $3.8 \cdot 10^3 \text{ N/m}$, the vertical specific oscillation frequency of the axis is from 5.1 s^{-1} to 29 s^{-1} . increases in nonlinear regularity. In this case, the specified mass of the roller was taken as 0.13 kg. If $m_{k2}=0,20$ kg, the specific oscillation frequency increases from $2,7 \text{ s}^{-1}$ to 21.7 s^{-1} . Similarly, when the specified mass of the third roller is taken as 0.22 kg, its specific oscillation frequency $f_1=(6,5 \div 32) \text{ s}^{-1}$ when the applied virginity coefficient increases to $C_k = (0.75 \div 5.8) \cdot 10^3 \text{ N/m}$. Varies in the range of 1. Based on the analysis of the obtained results, the vertical specific oscillation frequencies of the axes of the second and third rollers were adjusted to ensure that $C_{k2}=(1,85 \div 2,8) \cdot 10^3 \text{ N/m}$ and $C_{k3}=(2,6 \div 3,75) \cdot 10^3$. it is recommended that the recommended values be in the range $f_2=(14 \div 24) \text{ s}^{-1}$ and $f_3=(14 \div 24) \text{ s}^{-1}$.



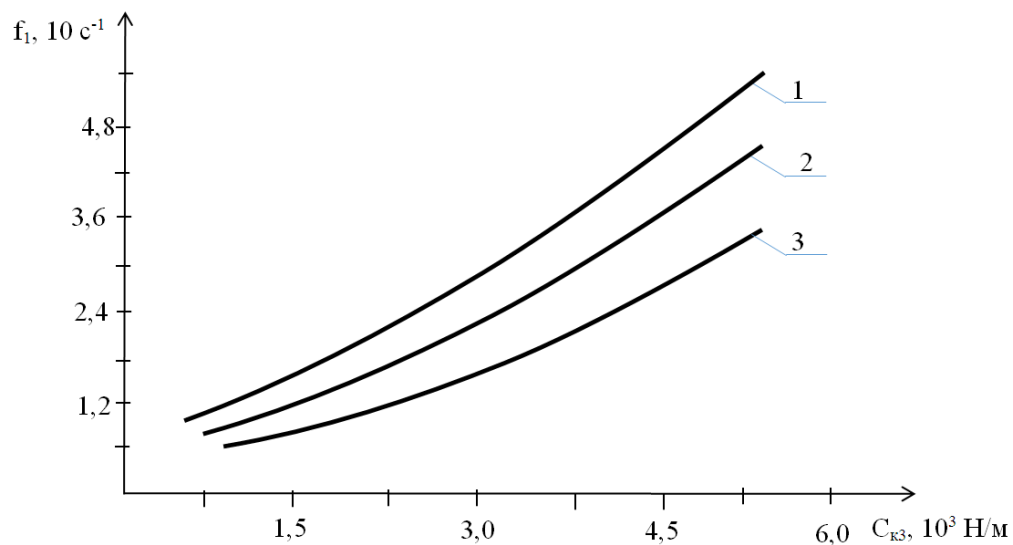
here; 1- $m_{k1}=0,115$ kg; 2- $m_{k1}=0,14$ kg; 3- $m_{k1}=0,185$ kg

Figure *a* shows the dependence of the specific vibration frequency on the bushing coefficient of the bushing of the axis of the first component shaft of the elongated tool



here; 1- $m_{k2}=0,13 \text{ kg}$; 2- $m_{k2}=0,16 \text{ kg}$; 3- $m_{k2}=0,20 \text{ kg}$

Figure *b* shows the dependence of the specific vibration frequency on the bushing coefficient of the second component shaft of the elongated tool



here; 1- $m_{k3}=0,14 \text{ kg}$; 2- $m_{k3}=0,17 \text{ kg}$; 3- $m_{k3}=0,22 \text{ kg}$

Graphs of the dependence of the frequency of specific vibrations on the axis of the third component shaft of the elongated tool in Figure *c*

Figure 3. Graphs of the dependence of the change of specific vertical vibration chatotas of loading rollers on the coefficients of elasticity of flexible bushings

Conclusion. A differential equation and an analytical solution representing the oscillations of the axis of the loading rollers of the ring spinning machine are obtained. A graph of the dependence of the vertical oscillation coverage of the shafts of the elongated tool shafts on the change in the amplitude of the resistance to elongation of the shaft and on the stiffness of the flexible bushings was constructed. Graphs of the dependence of the vertical specific vibration frequencies of the axes of the loading rollers on the change of the coefficients of virginity given by their rubber bushings are obtained. The recommended values of the parameters are based.

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