Basis of the parameters of the spin extractor tool on the interned spinning machine

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Abstract. The article presents an effective new constructive scheme and the principle of operation of the exhaust device of the ring spinning machine. 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.

1 Introduction

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 (loading) 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 joint is 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 yarn to have the same elongation [4].

2 Schematic of an effective yarn stretching tool

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...
3 Analysis of the oscillations of the loading roller shaft of the stretching tool

\[ (m_y + m_p + m_b + m_l)v'' + (b_1 + b_2 + b_l)v + \frac{c_1c_2c_l}{c_1c_l + c_2c_l + c_1c_2}v = F_0 \sin \omega t \]
The solution of the obtained differential equation (1) using the existing analytical method we obtain the following expression:

\[ \chi = \frac{F_0 \sin(\omega t + \beta)}{\sqrt{\left(\frac{c_1 c_2 c_1}{m_j + m_p + m_b + m_l} + c_1 c_2 c_1 \omega^2 \right)^2 + \left(\frac{b_1 + b_2 + b_1}{m_j + m_p + m_b + m_l}\right)^2}} \]

where, 

\[ m_j = (0.1 \pm 0.22) \]

\[ m_p = (0.02 \pm 0.07) \]

\[ m_b = (0.05 \pm 0.08) \]

\[ m_l = (0.003 \pm 0.005) \]

\[ c_1 = (2.5 \div 4.0) \cdot 10^3 \text{ H/m} \]

\[ c_2 = (1.0 \div 1.5) \cdot 10^3 \text{ H/m} \]

\[ b_1 = (0.35 \div 0.45) \text{ Hc/m} \]

\[ b_2 = (0.25 \div 0.3) \text{ Hc/m} \]

\[ b_1 = (0.05 \div 0.11) \text{ Hc/m} \]

To obtain a numerical solution of the problem, the initial calculated values of the parameters were obtained in the following intervals:

\[ (0.08 \div 0.2) \cdot 10^3 \text{ H/m} \]

\[ (1.0 \div 3.0) \text{ N} \]

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 \[ 0.18 \cdot 10^{-3} \text{ m} \] to \[ 1.06 \cdot 10^{-3} \text{ m} \], while the vibration of the second loading roller axis increases. Coverage increases from \[ 0.25 \cdot 10^{-3} \text{ m} \] to \[ 1.52 \cdot 10^{-3} \text{ m} \].

Respectively, we can see that the oscillation coverage of the first loading roller axis increases from \[ 0.31 \cdot 10^{-3} \text{ m} \] to \[ 2.51 \cdot 10^{-3} \text{ m} \] (Figure 2, Figures 1, 2, 3). Given the experimental results and the unevenness of the elongated shaft, the impact force is in the range of \( (2.3 \div 3.0) \text{ N} \), the oscillation coverage of the first loading roller axis \( (1.7 \div 2.4) \cdot 10^{-3} \text{ m} \), the second roller shaft \( (1.3 \div 1.65) \cdot 10^{-3} \text{ m} \) and the third roller axis vibration coverage is recommended to be in the range \( (0.7 \div 1.1) \cdot 10^{-3} \text{ m} \).
Fig. 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 $2.27 \times 10^{-3}$ when the applied mass of the first roller is increased from $0.6 \times 10^{-3}$ N/m to $3 \times 10^{-3}$ N/m when the applied mass is 0.15 kg. Decreases in nonlinear regularity from 3 m to $1.23 \times 10^{-3}$ m. If the given mass is taken as 0.2 kg, the vibration coverage is reduced from $2.5 \times 10^{-3}$ m to $0.67 \times 10^{-3}$ m. To ensure that the vibration coverage for the first roller does not exceed $(0.7 \div 1.1) \times 10^{-3}$ m, it is recommended that the coefficient of virginity of the rubber bushings be in the range $(1.68 \div 2.7) \times 10^{3}$ N/m. The recommended coefficients for the rubber bushings of the second and third loading rollers, respectively, are in the range $C_{k2} = (1.85 \div 2.8) \times 10^{3}$ N/m and $C_{k3} = (2.6 \div 3.75) \times 10^{3}$ 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 $(2.0 \div 5.0)$ in the range of $c_{1}$.
Here, 1  

\[ m_k1 = 0.15 \text{ kg}; \]

2  

\[ m_k2 = 0.2 \text{ kg}; \]

\( a \) - graphs of the dependence of the oscillation coverage of the axis of the loading shaft on the coefficients of virginity of the first shaft flexible bushings

Here, 1  

\[ m_k2 = 0.160 \text{ kg}; \]

2  

\[ m_k2 = 0.2 \text{ kg}; \]

\( b \) - graphs of the vibration coverage of the loading roller shaft on the virginity coefficients of the first roller flexible bushings

Here, 1  

\[ m_k3 = 0.165 \text{ kg}; \]

2  

\[ m_k3 = 0.22 \text{ kg}; \]

\( c \) - graphs of the dependence of the oscillation coverage of the loading roller shaft on the virginity coefficients of the first roller flexible bushings.

Fig. 3. Graphs of dependence of virginity coefficients of rubber bushings.
Figure 4 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 4a, the coefficient of virginity of the flexible loading bushings of the first loading roller increased from $0.3 \times 10^{-3}$ N/m to $2.8 \times 10^{-3}$ N/m and the specific oscillation frequency of the roller was 1, when $m_{k1} = 0.185$ kg. Increases in linear regularity from $6.1 \cdot 10^{-1}$ to $7.7 \cdot 10^{-1}$. When the applied mass of the roller is $0.115$ kg, its axial vertical oscillation frequency increases in the range of $6.05 \cdot 10^{-1}$ to $2.02 \cdot 10^{-1}$. It is recommended that the specific vibration frequency of the first roller be in the range of $(1.1 \div 1.8) \cdot 10^{-1}$ to ensure the recommended $C_k = (1.68 \div 2.7) \cdot 10^{-3}$ N/m values.

Correspondingly, according to the analysis of Figure 4b, when the coefficient of virginity of the flexible rollers of the composite roller is increased from $0.31 \cdot 10^{-3}$ Н/м to $3.8 \cdot 10^{-3}$ Н/m, the vertical specific oscillation frequency of the axis is from $5.1 c^{-1}$ to $29 c^{-1}$. 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 $2.7 c^{-1}$ to $21.7 c^{-1}$. Similarly, when the specified mass of the third roller is taken as $0.22$ kg, its specific oscillation frequency $f_1 = (6.5 \div 32) c^{-1}$ when the applied virginity coefficient increases to $C_k = (0.75 \div 5.8) \cdot 10^{-3}$ Н/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 $C_k2 = (1.85 \div 2.8) \cdot 10^{-3}$ Н/м and $C_k3 = (2.6 \div 3.75) \cdot 10^{-3}$. It is recommended that the recommended values be in the range $f_2 = (14 \div 24) c^{-1}$ and $f_3 = (14 \div 24) c^{-1}$.
4 Conclusion
References


