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Forced Vibrations of the Feeding Cylinder of Pneumomechanical Spinning Machines, Having an Elastic Shell

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Abstract. The article theoretically discusses the forced vibrations of the shell of the feed cylinder of the feed zones of rotor spinning machines. For the most part, vibrations of the feed cylinder shell of the feed zones of rotor spinning machines are considered on the basis of approximate theories. The new design of the feeding cylinder, working with the springs of the table, is given in detail. A conclusion was drawn and a scientific decision was made to improve the operating efficiency of spinning machines, at the same time increasing energy efficiency based on accurate calculations.

INTRODUCTION

The essence of the discretization process is to separate the tape into individual non-contacting fibers, to mix them relatively and to distribute them over a very long length. During the discretization process, extra high thinning occurs, i.e., the tape is thinned 3000-7500 times, and in the cross section of the discrete flow with ideal separation there are 2-6 non-contacting fibers. This is the difference between discretization and extrusion [1].

In a spinning unit, the main phases include: innings, discretization, transportation, taking off and transportation by air. In the feeding area, the sliver is taken out of the pelvis and fed at a constant speed. When removing the tape from the canvas or pelvis, there is no large axial force and no deformation of the tape is observed, so there is no redistribution of the fibers in the tape along the length. During feeding, the section of the tape changes to a flat rectangular one, convenient for discretization. The tape passes through a sealing funnel, which directs it to approximately the center of the width of the feed cylinder. The sealing funnel exerts its influence primarily as an organ that gives a certain direction to the tape and limits its width. On the BD-200 machine, the cross-section of the sealing funnel is selected so that the width of the tape at the outlet does not exceed 9 mm and the thickness does not exceed 2 mm. Changing the cross-section of the tape is achieved by increasing the density of the fibers in the cross-section. The density of the fibers increases, since with a gradual decrease in the cross-section for the passage of fibers in the tape, stresses arise under the influence of elastic transverse deformations. The sealing funnel does not change the uneven distribution of the fibers. Stresses in the tape cause frictional forces on the walls of the sealing funnel, which prevent the outermost layers of fibers from moving. These friction forces act unevenly along the perimeter of the sealing funnel cross-section. To prevent hidden exhaust, it is necessary to bring the outlet section of the sealing funnel closer to the compression area of the feeding device. The tape is compressed between the feed cylinder and the table. In this case, the density of the fibers in the cross-section increases and at the same time the width of the tape increases to the width of the slot in the table.

There is a known design in which the feed cylinder is made corrugated with straight grooves (parallel to the cylinder axis). When this cylinder operates, the condition for reliable operation of the feed, without disturbing the

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uniformity of the tape, is to overcome the resistance of the friction forces between the tape and the table and create the necessary movement using the feed cylinder. At the same time, the force of clamping the tape to the table changes the friction force, and some sliding of the tape also occurs. This sliding depends on the distance between the riffles of the feed cylinder [2].

The disadvantage of this design of a corrugated feed cylinder is the uneven distribution of the friction force along the length of the cylinder, which leads to a slight lag in the movement of the tape fibers at the edges of the cylinder. Due to the rigid interaction of the feed cylinder with the fibers, they are damaged. In addition, premature failure of the cylinder due to minor damage to the riffle. In another known design of the feeding cylinder, the riffles are made obliquely to the cylinder axis and form single diamonds, installed in rows along the length of the feeding cylinder [3].

A disadvantage of the existing design is also the uneven distribution of the friction force of the fibers of the tape with a grooved cylinder and a table along the length of the cylinder. This leads to a slight lag in the movement of the tape fibers at the edges of the cylinder. In addition, damage to the tape fibers occurs due to the rigid interaction of the rhombic riffles on the tape, although the table is amortized by a spring. Also known is the feeding cylinder of the spinning device with riffles on the surface, installed on the shaft, made of a composite of an outer corrugated sleeve and an inner sleeve, connected to each other by means of a rubber sleeve made barrel-shaped along the outer surface, while the inner sleeve is rigidly mounted on the drive shaft [4].

The disadvantage of this device is significant damage to the fibers in the extreme positions of the cylinder due to the lag in the movement of the fibers along the edges of the tape due to the high friction of the fibers on the table. In this case, the fiber density will be greatest towards the edges of the tape along the length of the cylinder. In the known design of the feed cylinder, uniform supply of the tape along the length of the feed cylinder is ensured by the fact that the outer surface of the corrugated sleeve is made concave, while the difference in diameters in the middle and along the edges of the sleeve is 2.0 mm. The disadvantage of this design is damage to the fibers in the extreme zones of the cylinder due to the reduced gap between the table and the surface of the cylinder. In another known feed cylinder of a spinning device, the working surface is made in the form of splines curved along a helical line, having protrusions and depressions, and the surfaces of the protrusions have grooves. At the same time, several options have been proposed for the design of feed cylinders with curved corrugated strips on the surface of the feed cylinders of the spinning device [5].

The disadvantage of this design is that during operation the fibers shift only in one direction due to the screw arrangement of the corrugated protrusions (slot). In this case, the greatest load falls on only half the length of the cylinder, and the other half of the cylinder remains underloaded. This can cause significant damage to the fibers of the tape fed to the sampling area. In addition, if an elementary part of the riffle fails, the supply cylinder itself becomes unusable. An approach to constructing a mathematical model based on the fact that initial irregularities in the shape of a thin-walled cylindrical shell triggers the internal interaction of low-frequency bending vibrations with high-frequency radial vibrations was proposed by G.S. Leizerovich [6-9].

To describe the motion of the shell, classical displacement equations are used [10,11,12].

Nonlinear vibrations of a double-curvature viscoelastic composite shell with an elastic middle layer and a magnetorheological layer were studied in [13].

Nonlinear vibrations of composite three-layer shells of double curvature with a piezoelectric layer are considered in [14].

To derive the equations of motion, high-order shear theory and von Karman’s theory of geometrically nonlinear deformation are used. The nonlinear dynamics of a double-curvature hollow shell with a honeycomb core having a negative Poisson's coefficient under the influence of an explosion is studied in [15].

Geometrically nonlinear forced vibrations of a cylindrical three-layer shell are modeled in [16].

MATERIALS AND METHODS

The most important problem in obtaining high-quality yarn is to ensure uniform supply of the tape along the length of the feed cylinder and reduce damage to the fibers in the tape, as well as increase the service life and maintainability of the cylinder. The problem is solved by improving the design of the feed cylinder of the spinning device, using combined forms of riffles and making the cylinder composite with an elastic element. The essence of the design is that the supply cylinder is made composite in the form of a cylinder having through longitudinal prismatic grooves in the axial direction with a trapezoidal cross-section, into which prismatic parts identical in shape to the grooves are installed. On the outer plane they have riffles made in six versions, installed sequentially in the grooves of the cylinder. Moreover, in the first option, the teeth have an inclined, bent shape without riffles; in the second option, the teeth have an inclined, bent pyramidal shape (figs. 1 and 2). In the third option, the shape of the teeth in two different combinations is a pyramid with a flat, without riffles surface. In the fourth option, the shape of the teeth in two different combinations is also a pyramid with truncated pyramids. Subsequent continuity also continues in the fifth version. In this version, the shape of the teeth has the appearance of a truncated pyramid with smooth surfaces without riffles. In the most recent, sixth variant, the teeth have a completely truncated pyramidal shape. The design is illustrated in the drawing, where Fig. 1 is a general view of the feed cylinder of a chevron-type spinning device with elastic shock absorbers. The design consists of a composite feed cylinder 1, a cut semicircle 2, a prismatic groove 3, a shaft 4 with keys 5 and an elastic rubber shell 6. Its structure is indicated by the numbers 7 and its geometric shape is divided by the numbers 8, 9, 10, 11, 12, 13. We give some information about the shape of the teeth of the proposed feed cylinder. In the prismatic groove 6, teeth 8 are installed, having an inclined bent shape without riffles, teeth 9 have an inclined bent shape. The feed cylinder teeth 10 have two different pyramid combinations with flat, non-grooved surfaces. After this we will see teeth 11, shaped like a pyramid and a truncated pyramid. The teeth are marked with the numbers 12 and have the shape of a truncated pyramid with smooth surfaces without riffles. The very last tooth 13 has a completely truncated pyramidal shape.

RESULTS

The fibrous mass (cotton fiber) in the form of a tape enters through the sealing funnel into the supply zone between the table (not shown in the figure) and the feed cylinder 1. In this case, the tape (fibrous mass) is compressed between the table and parts 7, which have different designs 8,9,10,11,12,13, which are installed in grooves 3 sequentially with a step of each through three grooves 6. With this arrangement of parts 7, the friction force between the surfaces of parts 7 and the fibrous tape increases successively due to an increase in protrusions (shoulders) on the surface of parts 7. Moreover, such a zone of influence of parts 7 on the fibrous material is repeated cyclically depending on the step between the parts 7 installed in grooves 3. The elastic rubber shell 6 provides shock absorption of the surfaces 8,9,10,11,12,13 of parts 7 on the fibrous material. This ensures uniform fiber density across the width of the tape, thereby ensuring uniform feeding of the tape, as well as reducing fiber damage.

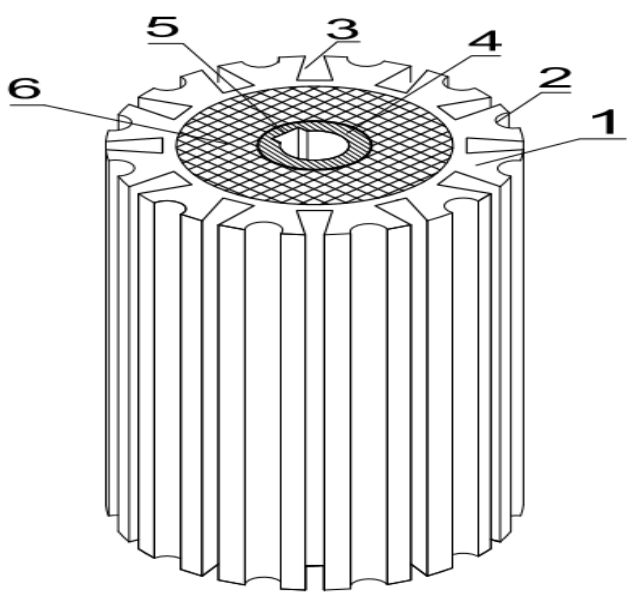

FIGURE 1. Feeding cylinder having through longitudinal prismatic grooves in the axial direction.

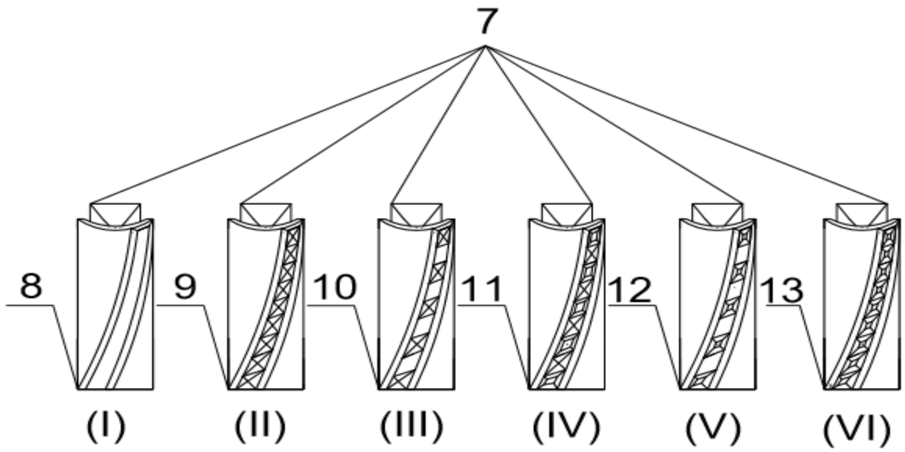

FIGURE 2. Geometric shape of feed cylinder teeth.

Forced vibrations of the shell of the feed cylinder in spinning practice are most often excited kinematically. Here, the lower end of the shell is fixed motionless, and the upper end is connected to the teeth of the supply cylinder and performs a given periodic movement with a period τ.

The displacement x_{disp} of this elastic shell of the feeding cylinder can be represented as a Fourier series

$$x_{disp} = c_0 + c_1 \sin(\omega t + \varphi_1) + c_2 \sin(2\omega t + \varphi_2) + \cdots \quad (1)$$

Where $\omega = \frac{2\pi}{\tau}$, and the stiffness coefficient is known.

Let's compose the equation of motion of the element dz of the table spring

$$m_0 dz \frac{\partial^2 x}{\partial t^2} = \frac{\partial N}{\partial z} dz \quad (2)$$

Where m_0 is the mass per unit length. The normal force in the section N turns out to be related to the longitudinal deformation $\varepsilon = \frac{\partial x}{\partial z}$ by Hooke's law. For a uniaxial stressed state, the spring has the form:

$$N = EF \frac{\partial x}{\partial z} \quad (3)$$

Where F is the cross-sectional area of the table spring.

After substitution the value N :

$$\frac{\partial^2 x}{\partial z^2} - \frac{1}{a^2} \frac{\partial^2 x}{\partial t^2} = 0 \quad (4)$$

Where $a = \sqrt{\frac{EF}{m_0}}$

We assume that the spring is homogeneous $m_0 = \rho F$,, where ρ is the density of the rubber shell of the supply cylinder

$$a = \sqrt{\frac{E}{\rho}}$$

We look for displacement in the form:

$$x(z,t) = u_0(z) + \sum_{i=1}^{\infty} u_i(z)\sin(i\omega t + \varphi_1) \tag{5}$$

Substituting this expression into equation (4), we obtain for each of the functions $u_i(z)$ an ordinary differential equation:

$$u_i + \frac{i^2\omega^2}{a^2} u_i = 0 \tag{6}$$

Where $u_0 = b_0 z + e_0;$

$$u_i = b_i sin\frac{i\omega z}{a} + e_i cos\frac{i\omega z}{a} \tag{7}$$

Since the end of the spring of the feed cylinder table $z = 0$ is fixed, then:

$$\mathrm{e_i} = 0(\mathrm{i} = 0, 1, \ 2 \ldots)$$

And the displacement of any point of the spring has the form:

$$\mathrm{x(z,t)} = \mathrm{b_0 z} + \sum_{\mathrm{i=1}}^{\infty} \mathrm{b_i sin}\frac{\mathrm{i\omega z}}{\mathrm{a}}\mathrm{sin(i\omega t} + \varphi_\mathrm{i}) \tag{8}$$

In particular, for the loaded end of the spring z=h we obtain

$$\mathrm{x(H,t)} = \mathrm{b_0 H} + \sum_{\mathrm{i=1}}^{\infty} \mathrm{b_i sin}\frac{\mathrm{i\omega H}}{\mathrm{a}}\mathrm{sin\ (i\omega t} + \varphi_\mathrm{i}) \tag{9}$$

Comparing this expression with formula (1), we find:

$$\mathrm{b_0} = \frac{\mathrm{c_0}}{\mathrm{H}}; \quad \mathrm{b_i} = \frac{\mathrm{c_i}}{\mathrm{sin}\frac{\mathrm{i\omega H}}{\mathrm{a}}}$$

Thus, the movement of any point of the spring under a given law (1), the movement of its end is determined by the expression

$$\mathrm{x(z,t)} = \mathrm{c_0}\frac{\mathrm{z}}{\mathrm{H}} + \frac{\mathrm{c_1}}{\mathrm{sin}\frac{\mathrm{\omega H}}{\mathrm{a}}}\mathrm{sin}\frac{\mathrm{\omega z}}{\mathrm{a}}\mathrm{sin(\omega t} + \varphi_1) + \frac{\mathrm{c_2}}{\mathrm{sin2}\frac{\mathrm{\omega H}}{\mathrm{a}}}\mathrm{sin2}\frac{\mathrm{\omega z}}{\mathrm{a}}\mathrm{sin(2\omega t} + \varphi_2) + \cdots.. \tag{10}$$

If $sin\frac{i\omega H}{a}$ goes to zero, then the corresponding term in formula (10) increases without limit. Resonance occurs if the harmonic component of the movement of the end of the spring is often:

$$i\omega = \mathcal{R}\frac{\pi a}{H} = \mathcal{R}\pi\sqrt{\frac{EF}{m_0 H^2}} \tag{11}$$

DISCUSSION

A new design has been developed for the composite feed cylinder of the spinning unit, which has different tooth shapes, which can be changed depending on the type of fiber (natural, chemical, etc.). An expression has been obtained that allows us to determine the laws of vertical oscillations of the elastic shell of the feeding cylinder. Graphic changes in the amplitude of oscillations of the elastic shell are obtained depending on the change in the rigidity coefficient of the elastic shell and the amplitude of oscillations of the disturbing force from the supplied sliver of the spinning device. The required values of the parameters of the feed cylinder are recommended to ensure the necessary quality indicators of the resulting yarn.

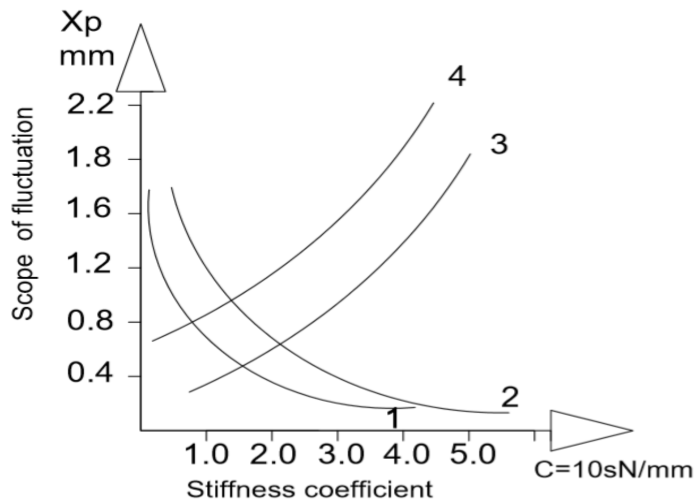

FIGURE 3. Dependence of changes in the amplitude of oscillations of the elastic rubber shell of the feed cylinder of the spinning unit depending on changes in the rigidity coefficient of the elastic sleeve and the amplitude of the disturbing force. 1.2 - range of fluctuations depending on the stiffness coefficient; 3.4-range of oscillations depending on the amplitude of the disturbing force.

$$1.2 - X_p = f(C_0);\ 3.4 - X_p = f(F_0);\ 1.4 - \text{when } F_1 = 35\ \text{sN};\ 2.3 - \text{when } F_1 = 45\ \text{sN};$$

TABLE 1. Numerical values of some data for an accurate representation of the graphical dependence are given in.

No.	Designations	Parameter names	Numerical values or their extreme limits
1	$\mathcal{R}$	Integer, that is if it coincides with one of the natural frequencies of the shell of the supply cylinder with fixed ends.	$1 \div 6$
2	H	Feed cylinder shell thickness	4 mm
3	E	Modulus of longitudinal elasticity of the elastic shell of the feeding cylinder	$8\ \mathrm{N/mm^2}$
4	F	Shell cross-sectional area	$21 \div 48\mathrm{m}m^2$
5	m_0	A parameter that depends on the density of the feeding cylinder shell material. $m_0 = \rho\frac{F\pi Di}{H}$	56700kg/mm
6	c_0	Periodic external load received from the feed table springs	$24 \div 28$ N
7	u_0	Amplitude function that determines the shape of the oscillation $u_0 = \sin\frac{\mathcal{R}\pi z}{l}$	$0.007 \div 0.45$
8	b_0	$b_0 = \frac{c_0}{H}$	$6 \div 7$
9	ρ	Density of feed cylinder rubber shell material	$150 \div 250\ \mathrm{kg/m^3}$

CONCLUSION

Theoretically discussing the forced vibrations of a composite feed cylinder, we can say that an increase in the density of fibers in the feed tape leads to an increase in the vertical movements of the elastic rubber shell of the feed cylinder. Considering that the deformation of the elastic shell in the radial direction does not exceed 0.5-0.6 mm (due to the unevenness of the fiber density in the tape), the recommended values for the rigidity coefficient of the elastic shell of the feeding cylinder are$\frac{(25-35)sN}{mm}$ when changing $\mathrm{F}_0 = 8.0 - 12$ sN.

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