FORCED VIBRATIONS OF A COMPOSITE CHEVRON-TYPE FEED CYLINDER WITH TORSIONAL RESISTANCE

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Annotation

The article studies the forced vibration of a composite chevron-type feeding cylinder with torsion resistance. The procedure for calculating the torsional vibration of the feed cylinder is given. An elementary analysis of the most important parameters of the chevron type compound feed cylinder has been done and they are tabulated. The operation of a compound feed cylinder excitation torque cylinder in one period is determined. The mutual coupling between the parameters of the feeding cylinder is analysed. For convenience some given is obtained from the general laws of dynamics.

Keywords: dynamics, oscillation, amplitude, forced, excitation moment, resonance, period, momentum, solution, cylinder, torsional, period, compound.

1. INTRODUCTION.

Torsional vibrations are periodic angular oscillations of masses concentrated on the shaft, causing twisting of individual sections of the shaft. Forced torsional oscillations are defined as steady-state angular oscillations of an elastic torsional system under the action of an exciting periodic torque.

Torsional vibrations are periodic angular oscillations of masses centred on the shaft that cause individual shaft sections to twist [1]. Periodic motion has the property that it is completely reproduced after some interval of time, called the period of motion. Forced oscillation of a compound feed cylinder under the action of the harmonic moment of the feed table springs in the feed zones of spinning machines. To apply the laws of dynamics we select a compound feed cylinder in rotor spinning machines.

2. MATERIALS AND METHODS .

The essence of the design consists in the fact that the feeding cylinder of the spinning device, containing a drive shaft with a rigidly mounted sleeve on top of which are located rubber sleeve with a sleeve put on it with inclined riffles, made of a composite of two parts, with inclined riffles are arranged symmetrically in the form of a chevron. In many cases the questions of calculation of structural elements lead to the problem of strength and stability of thin-walled rods in elastic medium [2].

In this case, the chevron inclined riffles on the surface of the feeding cylinder at their interaction with the fibres at the expense of horizontal component forces of part of the fibres at its edges, move to the centre there is uniformity of their distribution along the length of the feeding cylinder. This ensures a uniform fiber density across the width of the sliver, thereby ensuring a uniform sliver feed and reducing fiber damage.

3.RESEARCH OUTCOMES AND DISCUSSIONS

The design is explained in the drawing, where in Fig. 1, a general view of the feeding cylinder of a chevron-type spinning unit with elastic dampers.

The construction of the spinning unit with feeding cylinder (see Fig. 1) consists of symmetrically arranged compound outer sleeves 1 and 2 with inclined riffles forming a chevron shape, which are fixed on the inner sleeve 4 by means of a rubber sleeve 3. The sleeve 4 is rigidly mounted on the drive shaft 5.

Practical purpose of solid mechanics - description of behaviour of real bodies under force and thermal effects [3].

The construction works as follows.

The fibre mass (cotton, wool, chemical and other types of fibre) in the form of a ribbon enters through the sealing funnel and into the feeding zone between the table (not shown in the figure) and the feeding cylinder. It is known that the discretization process consists in the separation of the sliver into separate non-contacting fibers, in the relative mixing and in their distribution over a very long length [4]. To ensure uniformity of sliver feeding along the length of the feed cylinder and to reduce the damage of fibers in the sliver, the design of the feed cylinder of the spinning unit was improved [5]. In this case, due to the chevron arrangement of the corrugations of the composite outer sleeves 1 and 2, the captured fibers are distributed uniformly along their entire length.

This is provided by the displacement of the fiber portions towards the middle from the edges of the outer sleeves 1 and 2 due to the horizontal components of the forces exerted by the riffles on the fibers. When the corrugations of the outer sleeves 1 and 2 interact with the fibre belt due to the downward force, the rubber sleeves 3 are deformed, cushioning these forces. This effectively eliminates the damage to the fibres. Studying the sampling process theoretically and by applying it in practice expands the possibilities of obtaining yarn with high quality indicators [6]. Torsion of shells of rotation at large deformations is a less studied problem of interest for analysing the performance of rubber elastic elements, in particular torus couplings [7].

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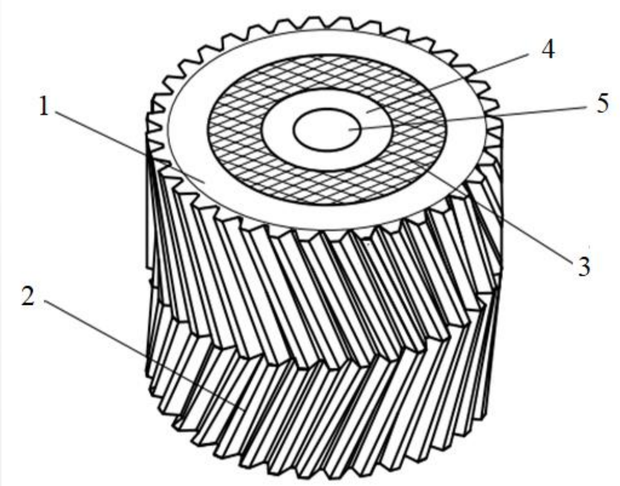

Fig.1. Feed cylinder of the spinning unit

Oscillatory motion of a composite feed cylinder of the chevron type under the action of a harmonic torque

$$M_{tor} = M_{tor}^{a}\sin(k\omega t + \beta_{tor})$$ (1)

and the moment of resistance acting on the oscillating mass are determined by the formula ,

$$M_{\eta} = -\eta\frac{d\varphi}{dt}$$ (2)

Where η – is the damping factor that takes into account all types of resistance; φ – is the angular displacement; t – is the current time.

It is described by the equation

$$I\frac{d^2\varphi}{dt^2} + \eta\frac{d\varphi}{dt} + c\varphi = M_{tor}^{a}\sin(k\omega t + \beta_{tor})$$ (3)

Starting from the discussion that $\eta_{кр} = 2I\omega_c$ c and that $D = \frac{\eta}{\eta_{кр}}$, we obtain

$$\ddot{\varphi} + \frac{2\eta\omega_c}{2I\omega_c}\varphi + \omega_c^2\varphi = \frac{M_{tor}^{a}}{I}\sin(k\omega t + \beta_{tor})$$ *(4)*

Or it can be written in the following form

$$\ddot{\varphi} + 2\omega_c D\dot{\varphi} + \omega_c^2\varphi = \frac{M_{tor}^{a}}{I}\sin(k\omega t + \beta_{tor})$$ (5)

The total solution of this equation is the sum of the total solution φ of the equation and the partial solution Starting from φ , we obtain that

$$\varphi = e^{-\frac{\eta}{2I}t\left(C_1^I cos\omega_d t + C_2^I sin\omega_d t\right)} + \varphi_{pr} \quad (6)$$

Where the first term of this equation characterises the free oscillation of the feed cylinder and the second term φ_{pr}_h the steady-state forced oscillation.

The partial solution φ_{pr} of the inhomogeneous torsional vibration equation of the feed cylinder (5) will be found in the form usual for equations whose right-hand side contains trigonometric functions sinα or cosα,namely in the form

$$\varphi_{pr} = \boldsymbol{\mathcal{B}sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi})$$

(7)

where $\boldsymbol{\beta_\chi}$– is the phase angle of the harmonic of the disturbing torque of the compound feed cylinder; $\boldsymbol{\varepsilon_\chi}$- is the phase shift between the amplitude of the forced angular oscillations of the compound chevron-type feed cylinder and the amplitude of the exciting periodic torque; $\boldsymbol{\mathcal{B}}$ –is the amplitude of the torsional oscillations of the chevron-type feed cylinder.

Therefore, only the steady-state motion defined by the second term, i.e., the partial solution of (7), is of practical interest.

Substituting expression (7) into equation (5), we obtain

$$-\mathcal{B}k^2\omega^2 sin\left(k\omega t + \beta_\chi - \varepsilon_\chi\right) + 2\omega_c D\mathcal{B}k\omega cos\left(k\omega t + \beta_\chi - \varepsilon_\chi\right) + \omega_c^2\mathcal{B}sin\left(k\omega t + \beta_\chi - \varepsilon_\chi\right) = \frac{M_{tor}^{a}}{I}\sin(k\omega t + \beta_{tor}).$$

Or, after conversion, it can be written in the following forms.

$$-\mathcal{B}k^2\omega^2 sin\left(k\omega t + \beta_\chi\right)cos\varepsilon_\chi + \mathcal{B}k^2\omega^2 cos\left(k\omega t + \beta_\chi\right)sin\varepsilon_\chi + 2\omega_c D\mathcal{B}k\omega cos\left(k\omega t + \beta_\chi\right)cos\varepsilon_\chi + 2\omega_c D\mathcal{B}k\omega sin\left(k\omega t + \beta_\chi\right)sin\varepsilon_\chi + \omega_c^2\mathcal{B}sin\left(k\omega t + \beta_\chi\right)cos\varepsilon_\chi - \omega_c^2\mathcal{B}cos\left(k\omega t + \beta_\chi\right)sin\varepsilon_\chi = \frac{M_{tor}^{a}}{I}\sin(k\omega t + \beta_{tor}).$$

The resulting equation must be satisfied at any t. This is possible only if the sum of the coefficients at $\boldsymbol{sin}\left(k\omega t + \boldsymbol{\beta_\chi}\right)$ and $\boldsymbol{cos}\left(k\omega t + \boldsymbol{\beta_\chi}\right)$ separately equals the similar coefficients in the right-hand side

$$\begin{cases} -\mathcal{B}\mathrm{I}k^2\omega^2\cos\varepsilon_\chi + 2\mathrm{ID}\omega_c\mathcal{B}k\omega\sin\varepsilon_\chi + \mathcal{B}\mathrm{I}\omega_c\cos\varepsilon_\chi = \mathrm{M}_{tor}^{a}; \\ \mathcal{B}\mathrm{I}k^2\omega^2\cos\varepsilon_\chi + 2\mathcal{B}\mathrm{ID}\omega_c k\omega\cos\varepsilon_\chi - \mathcal{B}\mathrm{I}\omega_c\sin\varepsilon_\chi = 0. \end{cases}$$

(8)

From the second equation of the system (8) (since $\boldsymbol{\mathcal{B}I \neq 0}$) we find

$$tg\boldsymbol{\varepsilon_\chi} = \frac{\boldsymbol{2D\omega_c k\omega}}{\boldsymbol{\omega_c^2-(k\omega)^2}} = \frac{\boldsymbol{2D(\frac{k\omega}{\omega_c})}}{\boldsymbol{1-(\frac{k\omega}{\omega_c})^2}}$$

(9)

Using the solution of (9) and the relations $\boldsymbol{sin\varepsilon_\chi} = \frac{\boldsymbol{tg\varepsilon_\chi}}{\sqrt{\boldsymbol{1+tg^2\varepsilon_\chi}}}$ and и $\boldsymbol{cos\varepsilon_\chi} = \boldsymbol{1}/\sqrt{\boldsymbol{1 + tg^2\varepsilon_\chi}}$, we find from the first equation (8):

$$\boldsymbol{\mathcal{B}} = \frac{\mathbf{M_{tor}^{a}}}{\mathbf{I}}\frac{\mathbf{1}}{\mathbf{(\omega_c^2-k^2\omega^2)cos\varepsilon_\chi+2D\omega_c k\omega sin\varepsilon_\chi}} = \frac{\mathbf{M_{tor}^{a}}}{\mathbf{I}}\frac{\mathbf{1}}{\sqrt{\mathbf{(\omega_c^2-k^2\omega^2)^2+4D^2\omega_c^2k^2\omega^2}}}$$

(10)

Substituting the values of $\mathcal{B}$ from (10) into (7) we find that the forced oscillations are carried out according to the law

$$\varphi_{pr} = \frac{M_{tor}^{a}}{I}\frac{\mathbf{1}}{\sqrt{(\boldsymbol{\omega_c^2} - \boldsymbol{k^2\omega^2})^2 + \boldsymbol{4D^2\omega_c^2k^2\omega^2}}}\boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi})$$

$$= \frac{M_{tor}^{a}}{I}\frac{\mathbf{1}/\boldsymbol{\omega_c^2}}{\sqrt{[\mathbf{1} - \left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2]^2 + \boldsymbol{4D^2}\left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2}}\boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi})$$

Since $\boldsymbol{\omega_c^2} = \boldsymbol{c/I}$, we finally obtain

$$\varphi_{pr} = \frac{M_{tor}^{a}}{\boldsymbol{c}}\frac{\mathbf{1}}{\sqrt{[\mathbf{1} - \left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2]^2 + \boldsymbol{4D^2}\left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2}}\boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi})$$

or for clarification it can be written in the following forms

$$\varphi_{pr} = \beta_{d.g.}\varphi_0\boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi}) = \boldsymbol{\mathcal{B}_k sin}(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi})$$

(11)

Where $\beta_{d.g.}$ is the dynamic gain coefficient of the compound feed cylinder and it is determined from the following formulae

$$\beta_{d.g.} = \frac{\mathbf{1}}{\sqrt{[\mathbf{1} - \left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2]^2 + \boldsymbol{4D^2}\left(\frac{\boldsymbol{k\omega}}{\boldsymbol{\omega_c}}\right)^2}}$$

(12)

$\mathcal{B}$ – is the amplitude of forced oscillations of the compound feeding cylinder of chevron type; $\varphi_0 = M_{tor}^{a}/\mathrm{c}$ static deflection of the system under the action of the moment M_{tor}^{a}..

It follows from (11) that the amplitude of forced torsional oscillations of the compound feed cylinder depends on the static deflection of the system φ_0 under the action of the moment M_{tor}^{a} and the dynamic force coefficient $\beta_{d.g.}$

$$\boldsymbol{\mathcal{B}_k} = \beta_{d.g}\varphi_0$$

(13)

It follows from equation (11) that if the forced oscillations occur with frequency $k\omega$ of the harmonic of the excitation torque, and the excitation torque is equal to

$M_{tor} = M_{tor}^{a}sin(k\omega t + \beta_\chi)$, then in this case the deviation is $\varphi_{(\div)} = \mathcal{B}_k sin(k\omega t + \beta_\chi - \varepsilon_\chi)$, i.e., the deviation in time lags behind the momentum by a constant value ε_χ.

Let us establish the relationship between the moments included in the equation of forced oscillations. For this purpose, we differentiate expression (11) and, substituting it into equation (5), we obtain:

$$-I\mathcal{B}_k k^2\omega^2\boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{cos\varepsilon_\chi} + I\mathcal{B}_k k^2\omega^2\boldsymbol{cos}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{sin\varepsilon_\chi} + \boldsymbol{2\omega_c DI\mathcal{B}_k k\omega}[\boldsymbol{cos}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{cos\varepsilon_\chi} + \boldsymbol{sin}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{sin\varepsilon_\chi}] + \boldsymbol{\omega_c^2 I\mathcal{B}_k sin}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{cos\varepsilon_\chi} - \boldsymbol{\omega_c^2 I\mathcal{B}_k cos}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{sin\varepsilon_\chi} + \boldsymbol{M_{tor}^{a} sin}(k\omega t + \boldsymbol{\beta_\chi})\boldsymbol{sin\varepsilon_\chi} = \mathbf{0} \qquad \textbf{(14)}$$

In equation (14), we substitute $\omega_c^2 I = \frac{c}{I} I = c$, and also $D = \frac{\xi}{\xi_{кр}} = \frac{\xi}{2I\omega_c^2} = \frac{\xi}{2c}$. Moreover, taking into account that in this equation the sums of the terms with $\boldsymbol{sin}(k\omega t + \boldsymbol{\beta}_\chi)$ and with $\boldsymbol{cos}(k\omega t + \boldsymbol{\beta}_\chi)$ separately must equal zero, we obtain a system of two equations characterizing the forced oscillations of the feed cylinder:

$$\begin{cases} M_{tor}^{a} + I\boldsymbol{\mathcal{B}}_k k^2\omega^2 \boldsymbol{cos\varepsilon_\chi} - \xi\boldsymbol{\mathcal{B}}_k k\omega \boldsymbol{sin\varepsilon_\chi} - c\boldsymbol{\mathcal{B}}_k \boldsymbol{cos\varepsilon_\chi} = \boldsymbol{0} \\ I\boldsymbol{\mathcal{B}}_k k^2\omega^2 \boldsymbol{sin\varepsilon_\chi} + \xi\boldsymbol{\mathcal{B}}_k k\omega \boldsymbol{cos\varepsilon_\chi} - c\boldsymbol{\mathcal{B}}_k \boldsymbol{sin\varepsilon_\chi} = \boldsymbol{0} \end{cases}$$

(15)

Let us determine the work done by the excitation torque excitation torque $\mathrm{M}_{tor} = \mathrm{M}_{\mathrm{tor}}^{\mathrm{a}}\sin(\mathrm{k\omega t} + \beta_\chi)$ on angular displacement $\varphi_{\mathrm{pr}} = \mathcal{B}_{к}\sin(\mathrm{k\omega t} + \beta_\chi - \varepsilon_\chi)$ for the period of momentum change $T = \frac{2\pi}{k\omega}$..

Elementary work by the excitation torque for one oscillation period is equal to

$$dW_{в} = M_{кр} d\varphi$$

(16)

From (16) , substituting the values $M_{tor} = \mathrm{M}_{tor}^{\mathrm{a}} sin(k\omega t + \beta_\chi)$, and $\varphi = \mathcal{B}_{к} k\omega cos(k\omega t + \beta_\chi - \varepsilon_\chi)dt$, we find the work of the exciting moment for one oscillation period $\frac{2\pi}{k\omega}$:

$$W_{\mathrm{exc}} = \int_0^{\frac{2\pi}{k\omega}} M_{кр} d\varphi = \mathrm{M}_{кр}^{\mathrm{a}} \boldsymbol{\mathcal{B}}_{к} k\omega \int_0^{\frac{2\pi}{k\omega}} \boldsymbol{sin}(k\omega t + \boldsymbol{\beta}_\chi) cos(k\omega t + \boldsymbol{\beta}_\chi - \boldsymbol{\varepsilon}_\chi)\boldsymbol{dt} = \mathrm{M}_{кр}^{\mathrm{a}} \boldsymbol{\mathcal{B}}_{к} k\omega \left[cos\boldsymbol{\varepsilon}_\chi \int_0^{\frac{2\pi}{k\omega}} \boldsymbol{sin}(k\omega t + \boldsymbol{\beta}_\chi) cos(k\omega t + \boldsymbol{\beta}_\chi)\boldsymbol{dt} + \boldsymbol{sin\varepsilon_\chi} \int_{\boldsymbol{0}}^{\frac{2\pi}{k\omega}} \boldsymbol{sin^2}(k\omega t + \boldsymbol{\beta}_\chi)\,\mathrm{dt} \right].$$

(17)

Solving the integrals in square brackets, we find:

$$\cos\varepsilon_\chi \int_0^{\frac{2\pi}{k\omega}} \sin(k\omega t + \beta_\chi)\cos(k\omega t + \beta_\chi)dt = \frac{\cos\varepsilon_\chi}{k\omega}\int_0^{2\pi} \sin(k\omega t + \beta_\chi)\cos(k\omega t + \beta_\chi)d(k\omega t) = 0;$$

$$\sin\varepsilon_\chi = \int_0^{\frac{2\pi}{k\omega}} \sin^2 2\omega t dt = \frac{\sin\varepsilon_\chi}{k\omega}\int_0^{2\pi} \frac{1 - \cos 2k\omega t}{2} d(k\omega t) = \pi\sin\varepsilon_\chi$$

Substituting the calculated integrals into (17), we determine the work of the excitation torque for one period

$$W_{\mathrm{exc}} = \pi \mathrm{M}_{кр}^{\mathrm{a}} \mathcal{B}_{к} sin\varepsilon_\chi$$

(18)

Part of this energy is dissipated in the form of heat (as a result of resistance forces), and part of it increases the stock of kinetic energy W in the system, and hence the amplitude of oscillations$\mathcal{B}_{к}$.

The elementary work of the moment of resistance is equal to

$$M_\epsilon = -\epsilon \frac{d\varphi}{dt} = -\epsilon \mathcal{B}_к k\omega cos\left(k\omega t + \beta_\chi - \varepsilon_\chi\right)$$

(the minus sign means that $M_\epsilon$is directed against the torsional velocity). Elementary work of the resistance moment

$$dW_{exc} = M_\epsilon d\varphi = \epsilon \mathcal{B}_k^2 k^2 \omega^2 cos^2\left(k\omega t + \beta_\chi - \varepsilon_\chi\right)dt$$

from where the work of the resistance moment for the oscillation period is determined

$$\boldsymbol{T} = \frac{2\pi}{k\omega}$$

$$W_{exc} = \boldsymbol{\epsilon \mathcal{B}_k k^2 \omega^2} \int_{\boldsymbol{0}}^{\frac{2\pi}{k\omega}} \boldsymbol{cos^2}\left(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi}\right)\boldsymbol{dt}$$

$$= \boldsymbol{\epsilon \mathcal{B}_k k^2 \omega^2} \int_{\boldsymbol{0}}^{\frac{2\pi}{k\omega}} \frac{\boldsymbol{1} + \boldsymbol{cos2}\left(k\omega t + \boldsymbol{\beta_\chi} - \boldsymbol{\varepsilon_\chi}\right)}{\boldsymbol{2}}\boldsymbol{dt}$$

Since $\cos 2\left(\mathrm{k\omega t} + \beta_\chi - \varepsilon_\chi\right) = \cos\left(2\mathrm{k\omega t} + 2\beta_\chi\right)\cos\varepsilon_\chi + \sin\left(2\mathrm{k\omega t} + 2\beta_\chi\right)\sin\varepsilon_\chi$, the integral splits into three integrals whose solution is of the form

$$\int_0^{\frac{2\pi}{\mathrm{k\omega}}} \frac{1}{2}\mathrm{dt} = \frac{\pi}{\mathrm{k\omega}};$$

$$\cos 2\varepsilon_\chi \int_0^{2\pi} \frac{\cos\left(2\mathrm{k\omega t} + 2\beta_\chi\right)}{4\pi\omega} \mathrm{d}(2\mathrm{k\omega t}) = 0;$$

$$\sin 2\varepsilon_\chi \int_0^{2\pi} \frac{\sin\left(2\mathrm{k\omega t} + 2\beta_\chi\right)}{4\pi\omega} \mathrm{d}(2\mathrm{k\omega t}) = 0.$$

Since the first integral in square brackets is equal to $\pi/k\omega$, and the other integrals are zero, the work of the resistance moment for the period of oscillations is

$$W_\epsilon = \pi \boldsymbol{\epsilon \mathcal{B}_k^2} k\omega \quad (19)$$

The previously obtained dependence (16) is easily found from the work equality at the excitation momentum resonance

$$M_{tor} = \mathrm{M}_{tor}^{a} \sin(k\omega t + \beta_\chi)$$

and the drag torque $M_\epsilon = -\epsilon \frac{d\varphi}{dt}$

Dependence of torque on parameters of stiffness coefficient and dynamic coefficient of composite feeding cylinder of chevron type at spinning of rotor spinning machines is given in fig.2.

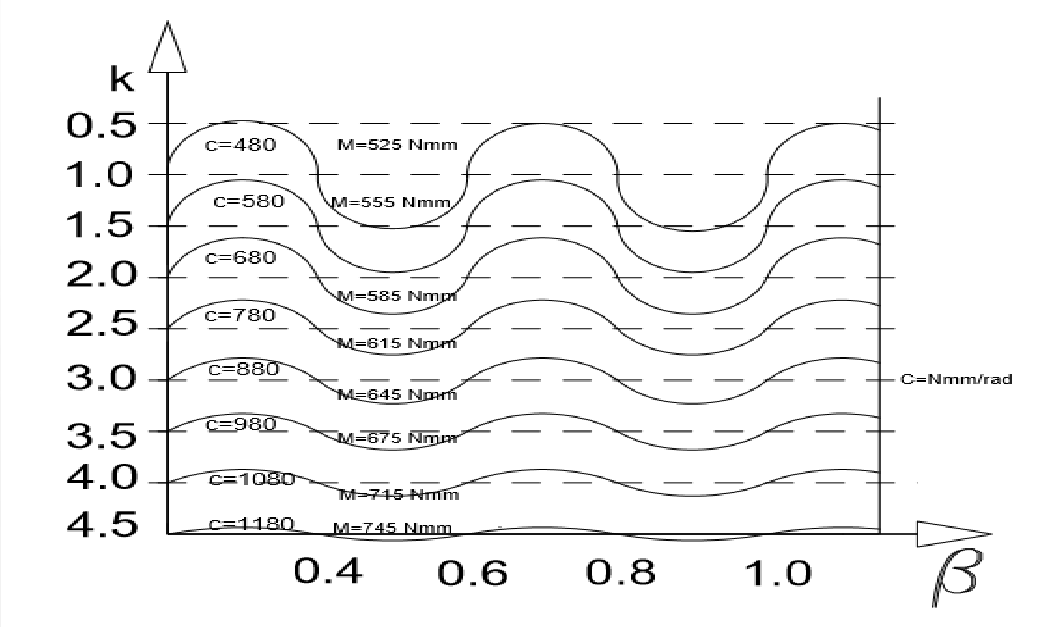

Fig.2. Graphical dependence of torque on the parameters of stiffness coefficient and dynamic coefficient of the chevron-type feed cylinder under torsion.

4.Discussion.

In some cases, vibrations interfere with normal operation and directly affect the strength of the structure, gradually preparing for post-fracture failure.

In such cases the theory can point out ways to reduce harmful fluctuations. Alongside this, it allows the justification and optimisation of technological processes in which oscillating ones are used purposefully.

The reference and result data in the study of forced vibrations of composite chevron type feed cylinder with torsional resistance are given in

Table 1.

№	Designations	Parameter name	Numerical values or their limit of boundaries
1	$\boldsymbol{E}$	Modulus of longitudinal elasticity of cylindrical sleeve of sovtava feeding cylinder of chevron type	$2 \div 6 \frac{N}{mm^2}$
2	$\boldsymbol{\mu}$	Poisson's ratio for the elastic sleeve material of the composite feed cylinder	0.47
3	$\boldsymbol{k}$	Order of the resonant harmonic	$\mathbf{1 \div 2}$
4	$\boldsymbol{\omega}$	Angular velocity of the chevron-type compound feed cylinder	$\mathbf{2}$
5	$\boldsymbol{\omega_c}$	Circular frequency of natural oscillations	$\mathbf{2900 \div 9000}$
6	$\boldsymbol{\epsilon}$	Damping factor	$0.1 \div 0.5$

		$\boldsymbol{\epsilon = 0.41 I_0 \omega;}$ $\boldsymbol{\epsilon = (I_0 \omega_д \delta)/\pi;}$	
7	$\boldsymbol{\delta}$	Logarithmic decrement of the attenuation	$\boldsymbol{\delta = 0.5 \div 0.8}$
8	D	Dampening $D = \frac{\boldsymbol{\delta}}{2\pi}$	$\boldsymbol{0.78 \div 125}$
9	I_p	Polar moment of inertia of the feed cylinder $I_p = \frac{\pi d^4}{32}$	$15000 mm^4$
10	$\boldsymbol{G}$	Shear modulus of the chevron-type compound feed cylinder $\boldsymbol{G = \frac{E}{2(1+\mu)}}$	$\boldsymbol{0.68 \div 2.1 \frac{N}{mm^2}}$
11	c	Torsional stiffness coefficient of the feed cylinder; c $= \frac{GI_p}{l}$	$480 \div 1500 \frac{Nmm}{rad}$
12	$\boldsymbol{I}$	Moment of inertia of the feed cylinder$\boldsymbol{I = 0.5 mR^2}$	$\boldsymbol{0.165\ kgmm^2}$

5. CONCLUSION: Theoretical discussion of the forced oscillation of the composite feeding cylinder of chevron type at torsion it can be said that the increase in the density of fibers in the fed belt, leads to an increase in the vertical displacement of the elastic rubber sleeve of the feeding cylinder.

Taking into account that deformation in radial direction does not exceed 0.5-0.6 mm (from non-uniformity of fiber density in the belt), the recommended values of stiffness coefficient of elastic sleeve of the feeding cylinder under torsion should be from c $= 480 \div 7800 \frac{Nmm}{rad}$.

With these parameters, quality yarns can be produced in rotor spinning machines, depending on the structural analysis of the fiber sliver.

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