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Issues of Stabılıty of Mechanısms of Technologıcal Processes of Textıle Machınes

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ABSTRACT: Article it is devoted dynamic research of modern textile machines.Dynamic models of real textile machines are made and solved their differential equations.Analized the received results are also recommended new actions for designing such machines.

KEYWORDS: Linear simulation,Nonlinear simulation, Full simulation

Ensuring the stability of the system (structure, machine, mechanism, technological process) is the most important task of optimal design [1].

The previous section shows that textile machines and processes can and should be considered as objects of dynamics. In this regard, it is important to determine the criteria for the applicability of methods for assessing dynamic systems to textile processes. Mechanical stability is evaluated by various methods depending on the state of the system and the specific tasks of its analysis.

The condition of static equilibrium, which is the equality to zero of the sums of projections of all forces and moments on any axis of an arbitrary coordinate system, is applicable, as a rule, only to rigid fixed structures. In the general case, the condition of static equilibrium, which is also the condition for the static stability of an immobile, non-deformable system, is represented by three pairs of equations of the form:

$$\sum_{i=1}^{n} F_i = 0 \quad \text{и} \quad \sum_{j=1}^{m} M_j = 0, \tag{1}$$

where Fi and Mj are the projections of the vectors of forces and moments on one axis of the spatial coordinate system.

If the system under consideration can be represented by a flat model, then the static equilibrium condition (1) can be written as:

$$\sum_{i=1}^{n} F_{ix} = 0; \quad \sum_{i=1}^{n} F_{iy} = 0 \quad \text{и} \quad \sum_{j=1}^{m} M_j = 0 \tag{2}$$

where Fix and Fiy are projections of forces on the corresponding axes of the Cartesian coordinate system;

M - is the moment of forces relative to any point of the coordinate plane.

Static equilibrium conditions (1) or (2) are often used in calculations of mobile systems. In this case, assuming the elements of the system to be absolutely rigid, it is sufficient, according to the d'Alembert principle, to include forces and moments of inertia forces in the composition of force actions in order to obtain the condition for the stability of a moving system in an inertial field.

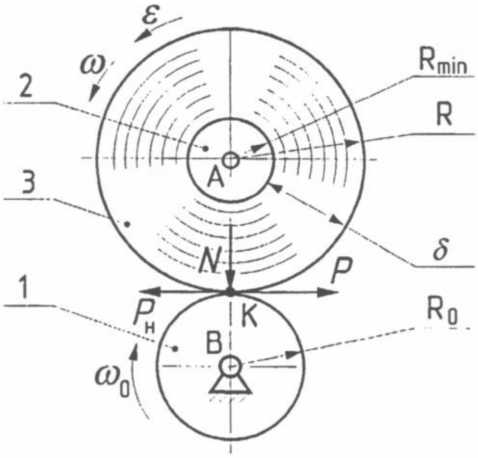
This method is widely used in the theory of machines and mechanisms. Here, it is especially important to preliminarily estimate the number of degrees of freedom of the system in order to determine the extra connections that, during the operation of the mechanism, will require compliance of the links, or, conversely, extra mobility, the presence of which significantly distorts the given patterns of movement of the working bodies and can disrupt the course of the technological process.

The wide distribution of this method, based on the equations of kinetostatics, in design practice is due to its relative simplicity and the availability of the mathematical apparatus. It, despite its limitations, allows you to solve a fairly large range of problems in textile engineering. The stationarity of the technological process, provided on the basis of the kinetostatic stability of the working mechanism, is often sufficient. In addition, using the methods of static or kinetostatic analysis, one can obtain initial data for subsequent dynamic analysis. Naturally, with all this, it is necessary to take into account the requirements for the quality of products and the modes of the technological process.

This approach in textile engineering is most suitable for the analysis and design of mechanisms for the formation of packages of various types from different materials:

- rolls of fiber in scutching units or on rough-carding machines;
- rolls of fabric on looms or finishing machines;
- warping rollers or weaving beams;
- reels or packages of another;

As an example, let's consider an abbreviated version of the structural and force analysis of the friction, winding mechanism in general and the winding mechanism of the self-twisting spinning machine PSK-225-LO in particular.



Rice. 1. Friction winding pair

Friction winding mechanisms are understood as such devices in which the package being wound and the winding drum form a friction pair (Fig. 1).

Here, the winding drum 1, which is the leading link, transfers the rotational movement to the driven link - the package 2, which rests on the drum with its generatrix. In this case, the contact between the bobbin and the drum is carried out through a winding layer 3 with a thickness

b = R – Rmin,

where Rmin is the chuck radius;

R - is the radius of the package, increasing during the winding process
from Rmin to Rmax.

At a constant frequency of rotation Φ, of a winding drum with a radius Ro, the package rotation frequency during the winding process decreases:

$$\omega = \omega_0 \frac{R_0}{R}(1-\nu), \qquad (3)$$

where ν - is the slippage of the package surface relative to the drum.

As in any friction gear, the amount of slip depends on the elastic properties of the contacting surfaces and does not exceed 0.04. In order for the contact of the package with the drum to be reliable, i.e., there would be no slippage of the package (deceleration of the circumferential speed of the package compared to the circumferential speed of the drum by more than the amount of elastic slip V), it is necessary that the moment from the adhesion force P relative to the axis rotation of package A exceeded the sum of the moments of all other forces applied to the bobbin, about the same axis:

$$P \cdot R \geq P_{H} \cdot R + M_{A} + M_{k} + J_{A}\varepsilon \qquad (4)$$

where PH - is the projection of the tension of the wound thread on the plane normal to the line of contact between the package and the drum;

MA - is the moment of friction forces in the package supports;

MK - is the moment of rolling friction in the zone of frictional contact;

JK - is the moment of inertia of the masses of the package relative to axis A;

ε - is the angular acceleration of the package.

Condition (4) of frictional contact stability is suitable for stationary mode (ωo =const) and package acceleration (ε > 0). In the case of package braking, this condition is as follows:

$$P \cdot R \geq J_{A} \cdot \bar{\varepsilon} - P_{H} \cdot R - M_{A} - M_{k}, \qquad (5)$$

where: $\bar{\varepsilon}$ - packing deceleration.

Considering that $\bar{\varepsilon} = -\varepsilon$, we get:

$$P \cdot R \geq \pm(P_{H} \cdot R + M_{A} + M_{k} + J_{A}\varepsilon), \qquad (6)$$

where the "+" sign corresponds to acceleration and stationary mode,
sign "-" - deceleration mode.

The non-stationary mode of rotation of the drum and packaging is typical for winding machines, where the slipping of the bobbin on the drum is used to prevent rope winding. Here, for transient modes, such absolute values of acceleration £ are chosen so that condition (6) is guaranteed not to be observed.

for stationary mode. The tension of the wound thread depends on its properties and usually does not exceed 40% of the breaking load. To determine the remaining quantities included in (6), we express them in terms of normal pressure N: P= N:f,

where f is the sliding friction coefficient of the winding layer on the drum (usually 0.15 < f < 0.22). Then Mk. = N.k,

where k - is the coefficient of rolling friction on the line of contact between the package and the drum; at k= 0.2...1 mm we get: MA = N fAr,

where fA. — coefficient of sliding friction in package supports (trunnions); typically 0.1 < f. <0.15; r is the radius of the trunnions of the package; as a rule,

r = 2...4 mm.

In the case of using rolling bearings in package supports, r= 10...20 mm,

0.02 < fA < 0.03. Thus, the product fAr= 0.2...0.6 mm for any option. Dividing the left and right parts of (6) by R, we obtain the condition for the stability of the stationary regime:

$$P_{н} \leq N\left(f - \frac{f_{A} \cdot r + k}{R}\right) \qquad (7)$$

In friction winding mechanisms, cartridges with a radius of less than 20 mm are not used, and the rolling friction coefficient k can reach its maximum values only on soft winding machines, where low levels of thread tension Ri are maintained. Therefore, in (7) the numerical value of the fraction does not exceed 10% of the friction coefficient f, and it can be neglected. Then the steady state stability condition, suitable for design practice, will be as follows:

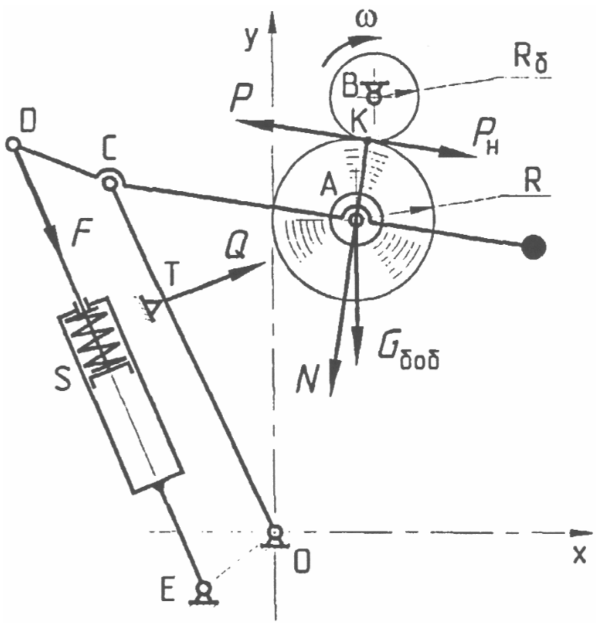
$$P_{н} \leq 0.9 \cdot Nf \qquad (8)$$

The winding mechanism of a self-twisting spinning machine of the PSK-225 family [2, 3] is much more complicated than the simplest one. It consists of three main elements:

1. Winding drum with a groove for laying out the thread along the bobbin.

2. Reel holder.

3. The bobbin withdrawal mechanism, acting through the thrust from the pneumatic chamber when the thread breaks.



Rice. 2. Scheme of the winding mechanism of a self-twisting spinning machine

The latter does not affect the winding process. Therefore, in the future, the term "winding mechanism" will be understood as the combination of the first two elements. The scheme of the mechanism in rectangular coordinates xOy is shown in fig. 2. In this case, the Ox axis is directed from the depth of the machine to its front side, and the Oy axis is directed from the bottom up.

Here, the winding drum of radius Rya rotates about the axis B in the direction u with an angular frequency Φ, contacting in the zone ks with a reel of variable radius R installed in the freely rotating plates A of the inclination ACD of the bobbin holder.

The latter is a four-link OCDE, the ED link of which is equipped with a spring-oil damper S and is a link of variable length, the CD rod serves as a slope for installing a bobbin into it. In order to fix it in the working position and regulate this position, the bobbin holder is equipped with an adjustable stop T, on which the OS post rests under the action of the damper spring S. In the OSDE four-link, the OS post plays the role of a rocker arm.

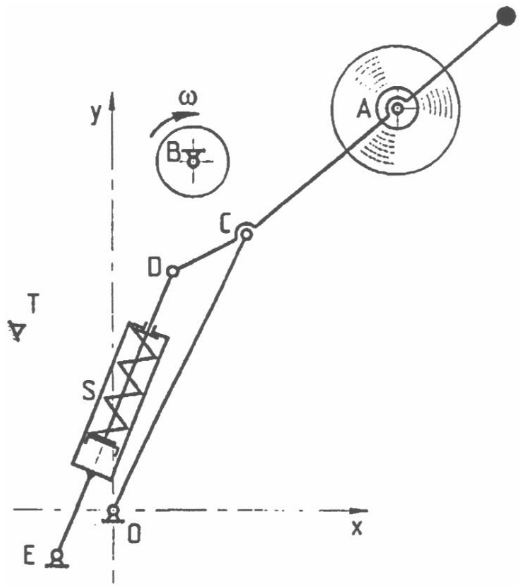
The number of degrees of freedom of the mechanism according to the well-known [3] dependence:

w = 3- n - 2-p5, -p4, (9)

where n - is the number of moving links, p5; - the number of couples of the 5th grade,

p4 - number of pairs of the 4th class.

The mechanism can occupy two stable positions - working (Fig. 2.) and filling (Fig. 3). In addition, if the contact of the OS rack with the stop T is disturbed, an unstable working position is possible, shown in Fig.



Rice. 3. Winding mechanism in filling position

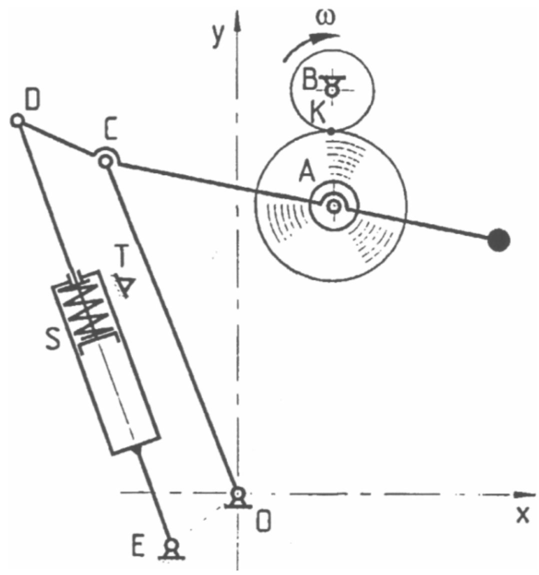


Fig.4. winding mechanism in unstable working position

Hereinafter, the term “working position” will be understood as such a position in which the process of winding the thread on the bobbin continues with the given technological parameters (thread tension, the pattern of its layout along the bobbin, etc.).

Obviously, if the contact K of the bobbin with the drum is broken, the mechanism goes into a non-working position, in which the winding process continues only for a short time due to the inertia of the bobbin. These positions are one filling and two unstable working positions, to which the mechanism can move from the working positions shown in fig. 2 and 4 when turning the slope clockwise.

Degrees of freedom of the mechanism
Table

The determination of the number of degrees of freedom of the mechanism for all five of the listed positions is summarized in Table 1, from which it can be seen that the transition from the working position to the non-working position is carried out without changing the number of degrees of freedom of the mechanism, which means that under the appropriate conditions, a stationary mode of sequential alternation of the working and non-working positions, which is nothing more than fluctuations in the inclination of the ACD around the C axis, i.e. — bobbin vibration. Thus, the structure of the mechanism does not prevent the bobbin from vibrating when it is wound.

Схема			Рис. 2.7		Рис. 2.8	Рис. 2.9	
Положение			рабочее	нерабочее	заправочное	рабочее	нерабочее
Исходные данные	Подвижные звенья	Число n	6	5	5	6	5
		Перечень	барабанчик, бобина, OC, CD, DS, SE	бобина, OC, CD, DS, SE	бобина, OC, CD, DS, SE	барабанчик, бобина, OC, CD, DS, SE	бобина, OC, CD, DS, SE
	Пары 5-го класса	Число p_5	7	6	6	7	6
		Перечень	A, B, C, D, E, O, S	A, C, D, E, O, S	A, C, D, E, O, S	A, B, C, D, E, O, S	A, C, D, E, O, S
	Пары 4-го класса	Число p_4	2	1	0	1	0
		Перечень	K, T	T		K	
Результаты	Степени свободы	Число W	2	2	3	3	3
		Перечень	1. Вращение бобины A. 2. Подвижность демпфера S.	1. Вращение бобины A. 2. Подвижность демпфера S.	1. Вращение бобины A. 2. Подвижность демпфера S. 3. Вращение стойки OC.	1. Вращение бобины A. 2. Подвижность демпфера S. 3. Вращение стойки OC.	1. Вращение бобины A. 2. Подвижность демпфера S. Вращение стойки OC.

Switching the mechanism from the working position to the filling position and back is carried out by the operator. Three degrees of freedom of the mechanism make it easy to manipulate the reel in the filling position. If the reel and the mechanism are carelessly placed in the working position, the latter may be in an unstable working position and remain in it indefinitely with the drum stationary. If the latter rotates, then under the influence of the winding force P, the mechanism moves to a stable working position.

Due to the fact that the spring of the damper S provides sufficient force for winding in the area of contact between the bobbin and the drum, the transition to any non-working position from the corresponding working position is possible only in an oscillatory mode in the presence of factors that repel the bobbin from the drum, which can be the inertia forces of the bobbin and slope, arising during the oscillatory process, and irregularities in the shape of the installation of the bobbin, which, if they are large enough, can become kinematic or force perturbing effects on the oscillatory system, which is, due to the presence of an elastic link - the spring of the damper S, the "bobbin - bobbin holder" complex.

Thus, with the correct shape and installation of the bobbin, i.e. - in the absence or insignificant magnitude of disturbing influences, the mechanism during the winding process can only be in one, stable working position, where two states are possible - static and stationary:

The statistical state is characterized by the fact that there is no rotation of the drum and reel, and the tangential force P (see Fig. 3.) is the rolling friction force of the reel on the drum and can be directed both to the left and to the right tangentially to the reel.

The stationary mode occurs when the drum rotates. In this case, the force P is the winding force, which ensures the rotation of the bobbin and the tension of the wound thread. Here, the force P is directed to the left tangentially to the bobbin and is the difference between the sliding friction force of the bobbin along the drum and the projection of the thread tension force on a plane normal to the axis of rotation of the bobbin . Obviously, to bring the bobbin into rotation, it is necessary that the moment from the force P is not less than the moment of rolling friction.

The condition for the bobbin to be in a stable working position is the contact of the OS post with the stop T, i.e. the presence of the reaction of the stop on the O post.

This condition for any of the two modes is determined from the equation of the moments of all forces about the O axis, which can be written in the following form:

- for static mode

$$\sum M_O = \pm N\cdot h_N + G_{боб}\cdot h_{G_{боб}} \pm G_п\cdot h_{G_п} - G_c\cdot h_{G_c} - F\cdot h_F \pm P\cdot h_P + Q\cdot h_Q = 0; \quad (9)$$

- for stationary mode

$$\sum M_O = \pm N\cdot h_N + G_{бoб}\cdot h_{G_{бoб}} \pm G_п\cdot h_{G_п} - G_c\cdot h_{G_c} - F\cdot h_F - P\cdot h_p + P_н\cdot h_p + Q h_Q = 0, \qquad (10)$$

where N - is the pressure force of the drum on the reel,

P - winding force (for stationary mode) or friction force (for static mode),

PH - is the projection of the tension force of the thread on a plane normal to the axes bobbin and drum,

Gbob. - bobbin weight,

Gp - slope weight,

Go - rack weight,

F - is the tension force of the spring,

Q - is the reaction of the emphasis on the rack,

h - shoulders of forces (indices correspond to forces).

In these dependencies, taking into account the above, one can notice:

- for the statistical mode P=N·k/R;
- for stationary mode P = N.f,

Where k - is the coefficient of rolling friction,

f - is the coefficient of sliding friction.

It is obvious that the minimum thrust reaction value can be obtained from (2.10) when the resultant P. - P = N * f is directed to the right, which can occur when moving from a stable working position (Fig. 2.) to an unstable one (Fig. 4.). The maximum stop reaction value is obtained from (10) in the absence of thread tension, when P= N:f is directed to the left. Thus, the limit values of the stop are determined from expressions:

$$Q = \frac{G_c\cdot h_{G_c} + F\cdot h_F \mp G_п\cdot h_{G_п} - G_{бoб}\cdot h_{G_{бoб}}}{h_Q} \mp N\cdot\frac{h_N}{h_Q} \pm N\cdot f\cdot\frac{h_p}{h_Q}, \qquad (11)$$

where replacing ,

$$M_0 = G_c\cdot h_{G_c} + F\cdot h_f \mp G_п\cdot h_{G_п} - G_{бoб.}\cdot h_{G_{бoб.}}, \qquad (12)$$

we get:

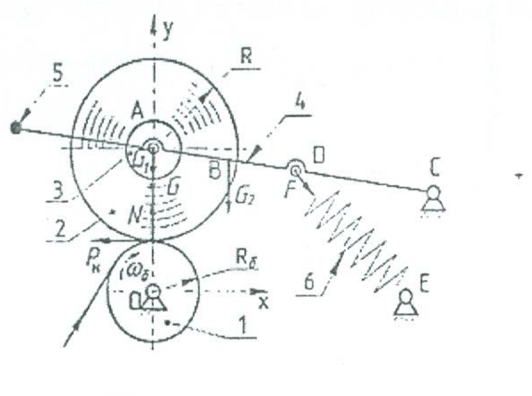
$$Q_{max} = \frac{M_0 \mp N\cdot h_N}{h_Q} + N\cdot f\cdot\frac{h_p}{h_Q}, \qquad (13)$$

$$Q_{min} = \frac{M_0 \mp N\cdot h_N}{h_Q} - N\cdot f\cdot\frac{h_p}{h_Q}, \qquad (14)$$

In both cases, the condition for a stable working position will be Q > 0.

In a stable working position, the OS post is motionless, which makes it possible to determine the minimum force of normal pressure from the condition of static equilibrium of the slope relative to its axis of rotation C. This calculation can also be applied to simpler winding mechanisms equipped with a spring loading system, as in the case of the upper one (Fig. 5.), and at the lower (Fig. 6.) location of the bobbin in relation to the drum.

In any case, such winding mechanisms consist of a winding drum 1 with a radius Rb, rotating with a constant angular frequency ωb, on which a reel rests, consisting of a winding layer 2 and a cartridge 3, which has the possibility of free rotation on the axis A of the slope lever 4, equipped with a handle 5 of the withdrawal bobbin by turning relative to axis C, and loaded at point D with force F from spring 6, which provides the necessary pressure H from the side of the drum to the bobbin.



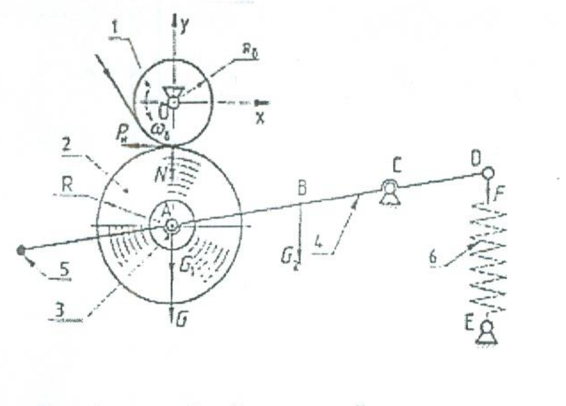


Fig. 5. the simplest winding mechanism Fig. 6. the simplest winding mechanism top bobbin bottom bobbin

In the statistical mode, the balance condition of the winding mechanism is determined by the zero sum of the moments about the C axis of all forces applied to the reel and the slope, with the exception of inertia forces, from which the normal pressure of the drum on the reel is determined as follows:

$$N = \frac{F\cdot h_F - P_н\cdot h_{P_н} \pm (G\cdot h_G + G_1\cdot h_{G_1} + G_2\cdot h_{G_2})}{h_N}, \qquad (15)$$

Where the "+" sign corresponds to the top location of the bobbin, the "-" sign - bottom.

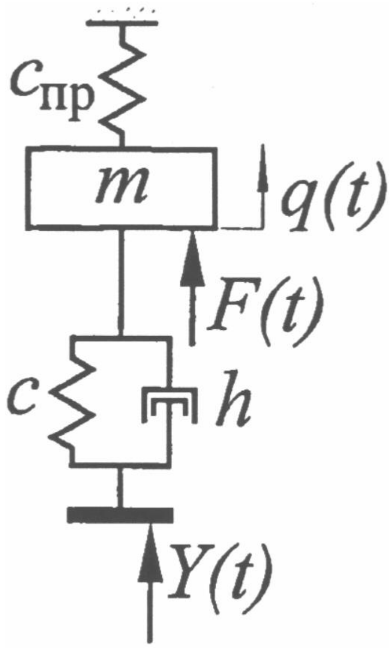
Here, all weights can be represented in terms of the masses of the corresponding elements and reduced to the axis of rotation of the bobbin:

$$G=\left[m_0+m_n\cdot\frac{l_{BC}}{l_{AB}}+\pi\left(R^2-R_{\min}^2\right)H\rho\right]g, \tag{16}$$

where G - is the total weight of the mechanism reduced to axis A;

To analyze the dynamic stability of machines, it is necessary to analyze them from the point of view of dynamics.

For our case, the winding mechanism, as a dynamic system, is reduced to a single-mass model with lumped parameters (Fig. 7.) As a generalized coordinate q, a radial direction from the center of the drum to the center of the bobbin is taken. The reduced mass of the mechanism m is concentrated at the center of mass of the bobbin. The mass reduction is carried out according to (16) without multiplying the right side by the gravitational acceleration g. The reduced stiffness of the spring system is determined through the stiffness of the spring Cp and the length of the arms hF and hN (see Fig. 6).



$$c_{\text{пр}}=c_{\text{п}}\frac{h_F^2}{h_N^2}. \tag{17}$$

Rice. 7. Dynamic model of the winding mechanism

The bobbin m is under the action of a force disturbance F(t) and a kinematic disturbance Y(t) transmitted to the oscillating mass through a layer of wound yarn with stiffness and viscosity h.

If we assume that the oscillations of the bobbin are sufficiently small, then for each radius of the bobbin R, the stiffness, viscosity and parameters of the perturbing influences (amplitude and frequency) will be constant, which allows us to consider the oscillatory system as linear, described by the following dynamic equilibrium equation:

$$m\ddot{q}+h\dot{q}+\left(c+c_{\text{пр}}\right)q=F(t)+c\cdot Y(t). \tag{18}$$

Depending on the adopted design solutions, each of the terms on the right side of (18) is one harmonic function or the sum of several, determined by their amplitudes and frequencies. In this case, the frequencies of all terms can be expressed in terms of the winding speed. Based on the principle of superposition [4, 5], we obtain the solution of the linear equation (18) in the form:

$$q=\sum_{k=1}^{n}B_k\sin\left(p_kt+\theta_k+\varphi_k\right), \tag{19}$$

where k - is the disturbance index;

H- is the number of disturbances;

Pk and Ɵk - are the frequency and phase of the kth disturbing action;

Bk and φ_k are the amplitude and phase of system oscillations at the kth disturbing influence:

$$B_k = \frac{A}{\sqrt{(c + c_{\text{пр}} - mp_k^2)^2 + h^2 m^2 p_k^2}}; \tag{20}$$

$$\varphi_k = \operatorname{arctg} \frac{hmp_k}{c + c_{\text{пр}} - mp_k^2}. \tag{21}$$

Due to the fact that the phase ratio of the disturbing influences Ѳk changes during the winding process due to a change in the diameter of the bobbin and its slippage relative to the drum, the phase ratio of the components of the oscillatory process also turns out to be variable, including such that all components (19) turn out to be in phase in one of the extremes. This coincidence will correspond to the oscillatory process with the maximum possible amplitude.

Obviously, to ensure the stability of the technological process, it is necessary to keep in mind this particular case. For each speed of the technological process, it is determined by a simple sum of the amplitudes of all terms calculated according to (20). Calculations by formulas (21) and (19) turn out to be unnecessary for the winding mechanism.

The maximum possible amplitude of the oscillatory process is a function not only of the package radius, but also of the winding speed. There is a linear relationship between v the maximum possible amplitude of oscillations B and the maximum change in normal ΔN, pressure due to these oscillations:

$$\Delta N = B \cdot \tilde{c}, \tag{22}$$

where $\tilde{c}$ - is the dynamic stiffness of the system at the line of contact between the bobbin and the drum.

Taking into account the deviation of the normal pressure, the conditions for the stability of the technological process in the dynamic mode can be obtained from (7) and (8):

$$P_{\text{н}} \le (N - \Delta N)\left(f - \frac{f_{\Lambda} \cdot r + k}{R}\right) \tag{23}$$

$$P_{\text{н}} \le 0.9 f (N - \Delta N) \tag{24}$$

The section describes the effect of shifting the position of dynamic equilibrium with respect to the position of static equilibrium in the direction of reducing the force effect of viscosity. For friction winding mechanisms, this effect is manifested in a decrease in the width of the line of contact between the bobbin and the drum. That is, the displacement is directed towards the separation of the reel from the drum and leads to an actual decrease in normal pressure. The relative displacement, as experiments show, depends linearly on the magnitude of the load and frequency (in our case, on normal pressure and rewinding speed). Thus, taking into account the bias effect, (22) is reduced to the form:

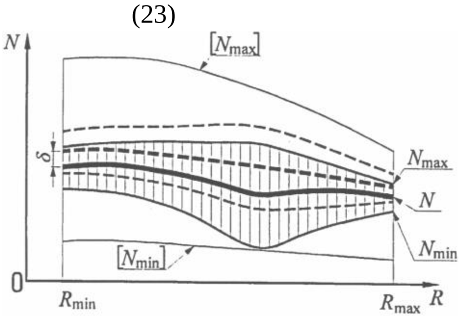
$$\Delta N = B \cdot \tilde{c} \cdot (1 \pm \delta), \tag{25}$$

where δ - is the relative displacement of the position of dynamic equilibrium from the position of static equilibrium.

Next, it is necessary to compare the ranges of changes in the bobbin pressure on the drum (from Nmin to Nmax) with the limit levels (from [Nmin], which determines the technological contact, to [Nmax], which is limited by the conditions of the bobbin strength and winding structure) for all values of the bobbin radius (from Rmin to Rmax), as shown in Fig. 8., where the dotted lines correspond to the range of changes in normal pressure without taking into account the displacement, and the solid lines correspond to δ it, taking it into account:

$$[N_{\min}] + \Delta N \le N \le [N_{\max}] - \Delta N. \tag{26}$$

Thus, if a sustainable technological process is understood asprocess that ensures the release of products of regulated quality in a given quantity, then to ensure this stability, it is necessary to create such dynamic conditions that the minimum force or geometric contact of the working bodies with the processed material necessary for the process to proceed is not violated under any conditions



Rice. 8. Changing the normal pressure of the drum on the reel

To do this, it is necessary to reduce the harmful dynamic effects on the product that occur during large-amplitude vibrations. Therefore, in addition to ensuring the static, structural, kinematic and dynamic stability of a mechanism or machine, it is necessary to ensure such a combination of design parameters and technological modes so that the latter flow away from resonant zones.

Summing up the overall assessment of textile machines and processes as objects of dynamics, we can note:

• the proposed classification of mechanical and technological systems of textile equipment, which includes not only direct technological interactions of working bodies with textile material, but also related effects, is a methodological basis for a correct analysis of the stability of technological systems and processes and for setting and solving inventive engineering problems;

• the stability of the technological process is determined, first of all, by the continuity of the technological contact of the working body with the material being processed;

• to ensure the stability of the technological process, it is necessary to carry out a structural, force and dynamic analysis of the system under the maximum possible impacts, including the impacts from the processed material.

For design practice, three stages of dynamic simulation can be recommended:

1. Linear modeling. It is carried out on fully linearized dynamic models with the inclusion of a textile material in them as an active link. In this case, the viscoelastic properties of the material are considered unchanged. The purpose of this analysis is to determine the resonant frequency spectrum. If the operating frequency range is at least twice as far from the resonant ones, then the remaining two stages of modeling can be omitted.
2. Nonlinear modeling. The viscoelastic properties of the material are chosen according to experimental data. The purpose of the stage is the spectrum of resonant frequencies. The criterion for the sufficiency of modeling is the distance from resonance by at least 40%.
3. Full simulation. All types of non-linearities are taken into account. The analysis is carried out, as a rule, by simulation methods.

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