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Dynamics of the warp beam brake drive on a weaving loom for the production of natural silk fabrics

Saidvoris Rakhimhodjaev[1], *Salikh* Tashpulatov[2,3], *Gulfiya* Sobirova[1], *Marguba* Rajapova[3], and *Igor* Tyurin[4*]

[1]Tashkent Institute of Textile and Light Industry, Tashkent, Uzbekistan
[2]Namangan institute of textile industry, Namangan, Uzbekistan
[3]Jizzakh polytechnic institute, Jizzakh, Uzbekistan
[4]Kosygin University, Moscow, Russia

Abstract. In the process of producing silk fabrics, by the fact of the high elastic properties of the threads large relaxation oscillations occur in the weaving system of the loom. Therefore, it is advisable to equip looms for the production of silk fabrics with brake regulators, and the rest of the range of fabrics with main regulators. On the basis of a mechanical model of relaxation oscillations and the scheme of a braking regulator for this model the regularities of friction force change depending on the duration of immovable contact and spring tension in time are obtained. The friction force for long contact time, zero contact time and contact time are determined. The mode against rotation of a driving disk causes stability of braking in a frictional pair.

1 Introduction

There are several theories to explain the occurrence of a jump change in friction force [1-8]. However, varied these theories may be, they all boil down to a relationship between the friction force and either the time of fixed contact or the sliding velocity. In particular, the theory is based on an increase in the coefficient of rest friction as a function of the duration of stationary contact [9,10]. It is shown that the relaxation oscillations are considerably influenced by the properties of the rubbing surfaces and the rigidity of the system, and the jumps decrease as the relative velocity increases.

As the fabric develops, the main threads must be advanced in the direction from the warp to the edge of the fabric, and at the same time, the fabric formed in each cycle of the loom must be removed from the zone of its formation and wound onto a commodity roller. These operations are performed with the help of special brake regulators for the tension of the warp threads and regulators for the retraction and winding of the fabric. In brake regulators, the continuously changing moment of winding the main threads, caused by the tension of the warp threads on the beam, and the value of which is determined by the radius of the winding of the main threads, is compared with the moment of friction resistance in the friction clutch.

* Corresponding author: tyurin-in@rguk.ru

When the equality of these moments are achieved, the beam receives an impulse. The duration of the impulse is equal to the time of movement of the beam. A continuous change in the tension of the warp is converted in the system into a series of pulses and the pulse frequency is equal to the time of one revolution of the main shaft of the machine [1, 3]. During the cycle of the machine, three characteristic levels of tension can be distinguished - with a closed throat, that is, at the time of the spade, with duck surf and with the formation of a throat. The tension of the warp threads increases with the formation of a pharynx and reaches its maximum value when the duck is surfed, and decreases when the pharynx is closed. During each cycle, the beam and with it the brake pulley 2 have stops, therefore, at the beginning of the movement, the forces acting on the beam must overcome the static friction force. The static friction coefficient is an unstable value and depends on normal pressure, material of rubbing surfaces, temperature, humidity, dust content, etc. The alternation of the coefficients of friction of rest and movement leads to an abrupt movement of the beam, and as a result, a sharp drop in the tension of the warp threads, which causes a deterioration in the structure of the fabric and increases the breakage of the warp threads [2, 4].

To stabilize the coefficient of friction in the friction pairs of the brake clutch, vibrating drive discs or combinations of friction pads of viscous friction (leather pads) with dry friction (asbestos pads) on the drive disks are used [2]. The most effective use of the mode of counter-rotation of the driving disk relative to the driven disk connected kinematically with the weaving beam. During each cycle, the beam and the driven disk have stops, therefore, at the beginning of the movement, the forces acting on the beam must overcome the static friction force. In this case, the transition from static friction (friction coefficient at rest) to kinematic friction (friction coefficient in motion) does not occur smoothly, but spasmodically. In a friction pair, due to the elasticity of the contact of two bodies sliding one relative to the other, there occurs jumps during friction, explained by periodically repeating processes of the appearance and subsequent disappearance of elastic stresses (relaxation oscillations). The magnitude of the jumps (amplitude of relaxation oscillations) is determined by the intensity of the growth of the static friction force with an increase in the time of stationary contact when the beam stops, as well as by the intensity of the decrease in the sliding friction force with an increase in the speed of the relative movement of the driven disk. These oscillations have a negative effect on the braking process, disrupting the normal operation of the brake regulator [7;8]. Consequently, the uneven, spasmodic rotation of the warp can lead to drastic changes in the tension of the warp threads and, ultimately, affect the quality of the fabric being produced. Therefore, there were carried out analytical studies of the brake regulator in the work. The following tasks were set: construction of a calculation scheme for a brake controller and the determination of friction parameters on the basis of a mechanical model of relaxation oscillation; the laws of relaxation oscillations in the friction pair of the brake controller; changes in the friction force depending on the duration of the fixed contact in the friction pair of the brake controller; determination of friction forces at different contact times in the friction pair of the brake controller.

There are several theories explaining the occurrence of an abrupt change in the friction force in the friction clutch of the brake controller [4, 8]. No matter how diverse these theories are, they all boil down to the dependence of the friction force either on the time of stationary contact [4] or on the sliding speed in brake friction couplings [8]. In particular, the theory is based on taking into account the increase in the coefficient of static friction depending on the duration of the stationary contact. It was shown in [3] that the properties of rubbing surfaces and the rigidity of the system have a significant effect on relaxation oscillations, and the jumps decrease with increasing relative velocity.

2 Methods

As a research method, a mechanical model of relaxation oscillations in the friction clutch of the brake controller is used. The material for the study is the friction clutch of the brake regulator of the loom. To do this, we present the calculation scheme of the friction clutch of the brake regulator in the form of a well-known model [1, 2, 3], shown in fig. 1, which considers friction on the friction ring with a radius Rav = 85 mm. is considered. In the model specified in fig. 1 Cpr is the reduced rigidity of elastic system of filling; N is normal pressure in the friction coupling; VA1, is relative speed of the leading disk. Stiffness of elastic system is reduced to point A2 (fig. 2) of a slave disk on its average radius.

According to [5, 14], the stiffness of the elastic system in the loom

$$C = \frac{C_0 \cdot C_{тк}}{C_0 + C_{тк}}$$

where C0 is the stiffness of the main threads in the loom elastic system of fabric formation (ESFF); Cc- the stiffness of the fabric in the loom elastic system of fabric formation (ESFF).

Elastic system stiffness reduced to the mean radius (point A2) of the driven disc

$$C = \frac{C_0 \cdot C_{тк}}{C_0 + C_{тк}} \cdot \frac{\rho}{R_{cp} \cdot i} \qquad (1)$$

where $\boldsymbol{\rho}$ - is the winding radius of the warp on the beam; i -is the transmission ratio from the warp to the driven disc. Linear sliding velocity in the friction coupling at average friction radius

$$V = V_{A1} - V_{A2} \qquad (2)$$

where VA1 - velocity of point A1 of the driving disk; VA2 - velocity of point A2 of the slave disk.

The velocity of point A1 of the driven disk is a constant value.

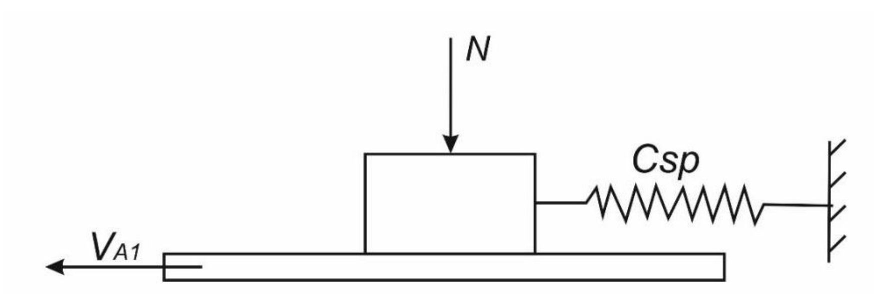

Fig. 1. Mechanical model of relaxation oscillations in the friction clutch of the brake controller.

here:

VA1 is the speed of point A1 of the drive disk;

Csp is the reduced stiffness of the elastic system, reduced to the average radius (to point A2) of the driven disk of the brake friction clutch;

N-normal pressure [2].

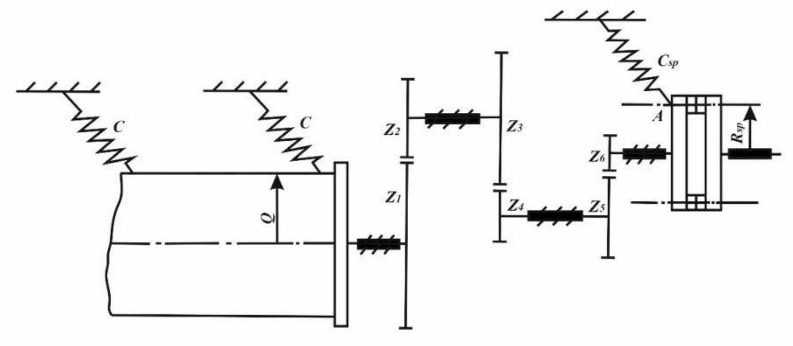

Fig. 2. Design scheme for a mechanical model of a braking controller.

here:
C-stiffness of the elastic threading system on the loom;
Csp is the reduced rigidity of the elastic system, reduced to the average radius (to point A2) of the driven disc brake friction clutch;
radius of winding the warp on the pile;
Z1-Z6 - gears;
Rav is the average radius of the clutch ring [13].

If we denote in Fig. 2 the magnitude of spring tightness by λm mm. and spring stiffness Cm, N/mm., then the normal pressure for the adopted model of Fig. 1 is equal

$$N=\lambda_m \cdot C_m \quad (3)$$

Hence, the friction force for the dynamic model

$$F_0=\lambda_m \cdot C_m \cdot f_0 \quad (4)$$

here:
f_0 is the coefficient of friction at zero contact time.
It is known [3, 4] that the value of friction force depends on the duration of stationary contact and is the following function of time

$$F(t) = F_\infty - (F_\infty - F_0) \cdot \exp[-\delta \cdot t] \quad (5)$$

here:
F_∞ is the friction force at an infinitely long time of a fixed contact;
F_0 - friction force at zero contact time;
δ is the aftereffect rate or coefficient characterizing the bond strengthening rate; t is the duration of the fixed contact.
In [6, 7, 8, 11, 15] it is indicated that at low speeds of mutual sliding in the friction system there are jumps, which disappear when the speed increases, besides the amplitude and frequency of jumps depends on the sliding speed, mass of the ram and rigidity of the system. It is noted, in particular, that at a uniformly low velocity of the plane the ram will make the relaxation oscillations according to the sawtooth law (Fig.3) around the equilibrium position, determined by the relation

$$C_{пр} \cdot a = F_0 \quad (6)$$

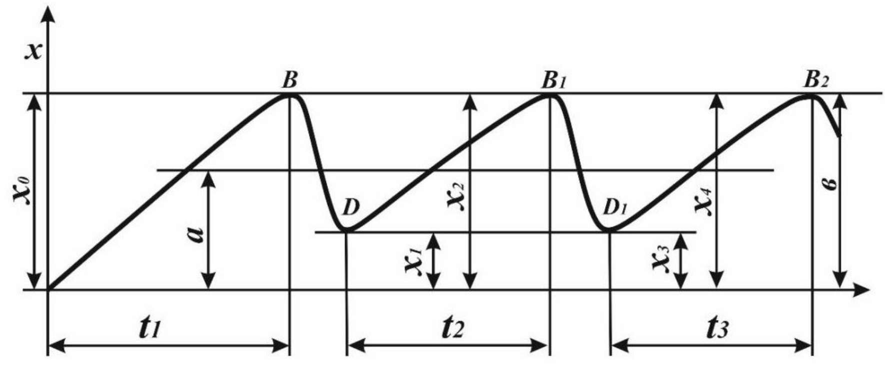

Fig.3. Relaxation oscillations according to the sawtooth law of the slider [6, 15]

Consider the process of steady-state relaxation oscillations in the selected model of Fig. 3. The ram moves under the action of spring force and sliding friction force arising between the ram and the moving plane. Up to the moment t1 (Fig.3) the rod moves together with the plane at velocity VA1 (point B on the curve), stretching the spring. At time t1 the rod begins to move relative to the plane (curve BD) until its velocity reaches the value indicated by point D on the curve. The ram will have a velocity equal to VA2 at the moment when the value of x = x1 is determined by the following equation

$$x_1 = 2a - x_0$$

since

$$a = \frac{x_0 - x_1}{2}$$

Further movement of the slider along the plane is impossible because at this point the frictional force changes its sign and its magnitude is greater than the elastic force of the spring. The slide will then start moving with the plane again (section DB1) until the spring tension force is again equal to the friction force at some value of x = x2. In order to find the value of x2 and time t2 it is necessary to solve the equation

$$F(t_2) = C_{пр} \cdot (x_1 + V_{A1} \cdot t_2) \quad (7)$$

here:

x1 + VA1 - t2 = x2 - is the amount of spring deformation at which the friction force takes its maximum value (the moment of a new stall).

The ram then performs a new oscillatory movement which, at a value of x = x3 = 2 \\ mathbit {a} - x2, will change into a uniform movement along with the plane. Then, at some value of x4, the slider will once again break away. It may happen that the sequence of values x0, x2, x4..... will tend to some value x = c other than \mathbit{a}, i.e. a relaxation oscillation will be established. If this sequence tends to a value x = a, then the jumps will stop, the sample will stand still and the spring tension force will be balanced by the frictional force of the plane sliding on the sample.

Let us determine the conditions under which relaxation oscillations are possible in the friction model adopted, that is, in the friction coupling of the brake regulator.

If the oscillations are established, it should be assumed that a = c, and the coupling moment of the slider with the plane x = 2a-c. For a known velocity of the plane VA1, the time of movement of the ram coupled to the plane will be determined by the expression

$$t = \frac{2b - 2a}{V_{A1}} \quad (8)$$

wherefore

$$b = \frac{V_{A1} \cdot t}{2}$$

On the other hand, at the moment of stall, the friction force F(t) is equal to the spring tension P(t) = Cpr-in, for a certain time t we have the equation

$$F(t) = P(t) = C_{пр} \cdot в \, (\frac{V_A \cdot t}{2} + a) = C_{пр}$$

3 Results and Discussions

Let's draw a graph of change of friction force depending on duration of fixed contact F(t), as well as a graph of change of spring tension P(t) in time (fig. 4). At time t = 0, both graphs have a common point m, since F0 = Cpr -a. This point will be the only one if the angle β between the tangent to the curve F(t) and the abscissa axis at t = 0 is smaller than the angle α of the slope of the line

$$P(t) = (\frac{V_A \cdot t}{2} + a) \, C_{пр}$$

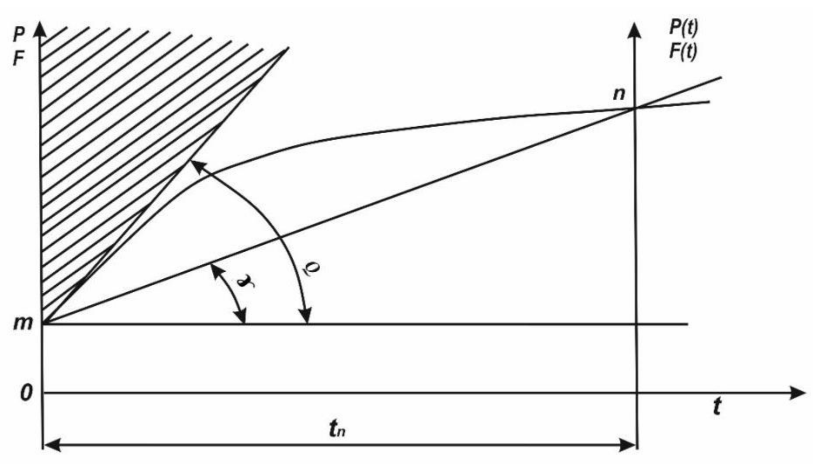

Fig.4. A graph of the change in the friction force and spring tension in time in the brake regulator clutch.

here: P (t) is the spring tension in the brake regulator clutch; F (t) is the friction force in the brake regulator clutch.

In this case there will be no relaxation oscillations. The area indicated is shaded in the graph.

If, however, the line P(t) crosses the graph F(t) also at the point n with abscissa t, the relaxation oscillations will occur.

By differentiating (5) with respect to t and assuming t = 0, we obtain

δ-(F∞ - F0), which is tgβ. It is in turn obvious that tgα at t = 0 after differentiating (14) the straight-line equation

$$P(t) = (\frac{V_A \cdot t}{2} + a) \cdot C_{пр} \quad \text{will be equal to} \quad \frac{C_{пр} \cdot V_A}{2}$$

The condition of existence of relaxation oscillations will be written tg(α) tg(β)

$$\text{or as } \delta(F_\infty - F_0) > \frac{C_{пр} \cdot V_A}{2}, \qquad V_A < V_{кр} = \frac{2(F_\infty - F_0)\delta}{C_{пр}} \tag{9}$$

here
δ is the coefficient characterizing the rate, bond hardening or rate of consequence, sec^{-1}.

The value of δ depends on the materials of the rubbing pairs and the specific pressures at which sliding occurs, and is determined experimentally. If inequality (8) is satisfied for all values of the parameters of our mechanical system, then stable spasmodic rotations of the beam arise in the friction clutch of the brake controller, which leads to drastic changes in the tension of the warp threads and which can affect the quality of the produced fabric. Let us show that during the production of crepe de chine fabric, relaxation oscillations can occur in the friction coupling of the brake regulator.

According to [5, 14] we accept the stiffness of elastic filling system, C = 140 N/mm, therefore, from expression (1) the reduced stiffness to the middle radius of the driven disk Cpr = 10,1 N/mm. When the spring in the friction clutch is tightened λ_m = 12 mm, the friction forces are determined from expression (4).

The friction force at long contact times

$$F_\infty = 60 \cdot \lambda_m \cdot f_\infty = 242 \ H.$$

The friction force at zero contact time

$$F_0 = 60 \cdot \lambda_m \cdot f_0 = 184 \ H.$$

The friction force corresponding to the duration of contact

$$F(t) = 60 \cdot \lambda_m \cdot f(t) = 191 \ H.$$

The values of friction coefficients were obtained experimentally:

$$\boldsymbol{f_\infty} = 0{,}336; \ \boldsymbol{f_0} = 0{,}256; \ \boldsymbol{f_t} = 0{,}264.$$

For the given fabric assortment, the time of fixed contact in the friction coupling equals 0.25 sec.

Let's rewrite equation (5) and solve with respect to speed of after action

$$F(t) = F_\infty - (F_\infty - F_0) \cdot \exp[-\delta \cdot t]$$

$$\delta = \frac{ln(F_\infty - F_0) - ln(F_\infty - F_{(t)})}{t}$$

Substituting the numerical values, we get:

$$\delta = \frac{ln58 - ln51}{0{,}25} = \frac{4{,}06044 - 3{,}93173}{0{,}25} = 0{,}52$$

Determine critical sliding velocity in the friction coupling

$$V_{кр} \frac{2(\boldsymbol{F_\infty} - \boldsymbol{F_0}) \cdot \boldsymbol{\delta}}{\boldsymbol{C}_{кр}} = \frac{2 \cdot 58 \cdot 0{,}58}{10{,}1} = 6 \text{ мм/sec}$$

The linear velocity of point A2 of the driven disc was determined experimentally.

$$V_{A2}^{max} = 4 \text{ мм/sec}$$
$$V_{A2}^{min} = 1{,}6 \text{ мм/sec}$$

There were used a complex of measuring equipment in the experiment and consisted of an oscilloscope, a power supply unit, a strain gauge amplifier, a shunt store and additional resistances, and sensors that perceive signals of various parameters. The influence of the coefficient of friction at rest and the coefficient of friction in the movement of the brake regulator on the conditions for the release of the warp from the beam was determined.

A tach generator was used to assess the rotation of the beam and the driven disk of the brake regulator under dynamic conditions. The tension of the warp threads was measured in the "rock-lamella" zone using overhead sensors. Determination of quantitative and qualitative characteristics of the friction coefficient at rest and the friction coefficient in motion in the friction regulators were carried out on a stand developed by us. At the stand, we have the opportunity to obtain characteristics for any friction pair used in brake regulators on looms, and the working conditions of the friction pair are close to the working conditions of the friction pair on the loom. Analysis of the results shows that with an increase in normal pressure, the friction coefficient at rest increase in motion. The coefficient of friction in motion decreases as the relative speed of the brake regulator disc increases. The friction coefficient at rest is greater than the friction coefficient in motion. The condition of the base release in the brake regulators is influenced by the coefficient of friction at rest and the coefficient of friction in motion in the friction pair.

Figure 5 presents photocopies of oscillograms with a record of measured values of the brake regulator with a movable drive disc in which for each cycle of work experience in the friction pair there is only a coefficient of friction in motion. On photocopies of the oscillators: 1 - the speed of the driven disc; 2 - the speed of the loom, 3 - the tension of the fabric; 4 - the tension of the threads of the base; 5 - marker of the position of the main shaft of the machine in the zero line.

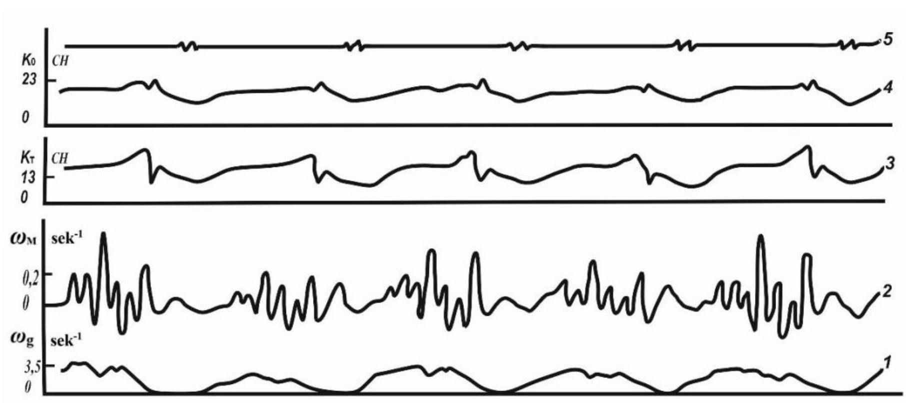

Fig.5. Photocopies of oscillograms with measured values of brake regulator with movable drive disc.

here: 1 - speed of the driven disc; 2 - speed of weaving manure; 3 - tension of fabric; 4 - tension of the threads of the base; 5 - marker of the position of the main shaft of the machine in the zero line.

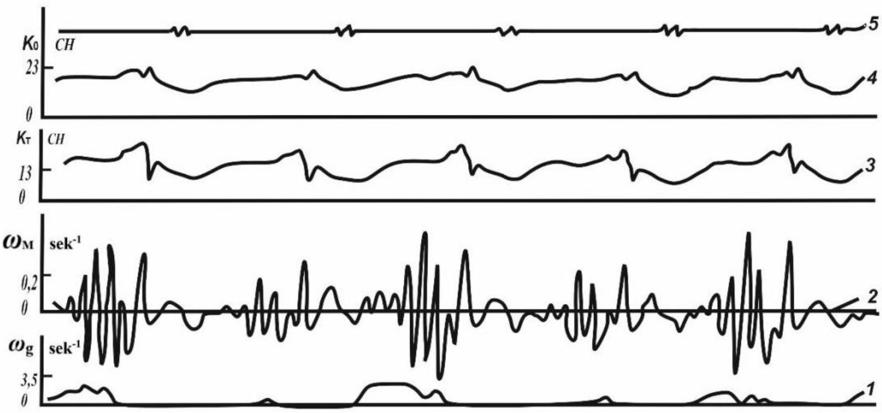

Fig. 6. Photo copies of oscillograms with measured values of brake regulator with fixed drive disc. here: 1 - speed of the driven disc; 2 - speed of weaving manure; 3 - tension of fabric; 4 - tension of the threads of the base; 5 - of the position of the main shaft of the machine in the zero line.

In the weaving loom brake regulator in the friction coupling, the driven disc receives motion from the drive and is in counter rotation with respect to the slave disc which is kinematically coupled to the weaving loom. From the calculated diagram of the brake regulator to the mechanical model (Fig. 2) shows that the speed of the point A2, driven disk - variable, as it depends on the angle of rotation of the weaving mill. The value δ depends on materials of rubbing pairs and specific pressures, at which sliding occurs, and is defined experimentally. If inequality (11) is carried out for all values of parameters of our mechanical system, then in the friction coupling of the disk basic brake steady jump turns of the warp appear, that leads to sharp changes of tension of warp threads and that can affect quality of the fabric produced. Relaxation oscillations in the friction coupling of the brake regulator can occur during fabric production. Let's substitute the velocity values of point A2 of the slave disc in equation (2).

When $V_{A1}= 0$, the master disc is stationary and the friction coupling conditions are characterized by unstable friction moments, the nave has a jump motion, that is,
$V_{кр}>V_A^{max}>V_A^{min}$.

When $V_{A1} \neq 0$, the driven disc is in counter-rotation with respect to the slave disc. In this case, $V_{кр}<V_A^{max}<V_A^{min}$ and friction clutch working conditions are typical for the developed brake regulator design, i.e. the stability of friction moment in friction unit and uniform movement of the stock. With change of assortment of the fabrics produced and speed modes of the loom, and also with the purpose of exclusion of influence on coefficient of a friction in movement of speed of sliding in a frictional coupling we recommend to establish the minimal relative speed of a leading disk $V_{A1}= 25\text{-}V_{cr} = 150$ mm/sec.

The oscillogram of the recording with the movable drive (Figure 5) shows that only the coefficient of friction in motion occurs during each cycle of the machine in the friction pair. The coefficient of friction in motion positively affects the release of the base from the warp beam, that is, the movement of the driven disc (curve I). For each turn of the main shaft there is a even release of the base, as evidenced by the curves of the movement of disc 1 and warp beam 2. Consequently, the filaments of the base are less tense and we can expect an improvement in the structure of the fabric. If the friction pair has a coefficient of friction at rest and a coefficient of friction in motion when the drive disc is stationary (Figure 6), the rotation of the driven disc is uneven. The maximum rotation of the driven disc at one turn of the main shaft is replaced by the minimum at the next rotation of the main shaft of the machine. The oscillation amplitude of the warp is greater, which ultimately affects the tension of the warp threads and the structure of the fabric being produced. In addition, during the cycle of the machine, the beam makes large torsional vibrations (curve 2). The occurrence of torsional vibrations can be explained by the following circumstance. When processing natural silk, the threads themselves have a higher elasticity than, for example, threads made from cotton yarn. At the moment of beating the weft thread, the main threads, due to the forces of elasticity, turn the beam forward at a certain angle, that is, they deform the trunk of the beam, the square (axis) of the beam, etc. After the beating the base tension drops sharply, and the warp beam begins to move at a high speed in the opposite direction, enhancing the tension of the base threads, that is, during the period of opening the yaw the threads of the base experience additional loads, and the amplitude of the tension of the strands depends on the amplitude of the warp. The fluctuations of the beam have a damped character. Consequently, when weaving fabrics from natural silk with a non-movable driving disc of the brake regulator, we have large torsional vibrations from cycle to cycle of the loom, as a result of which the base filaments experience additional fluctuations during the opening of the shed, which leads to the destruction of the structure of the warp threads, as a consequence of the breakage of the base threads.

4 Conclusions

It is advisable to equip looms with brake regulators when producing silk fabrics, because they provide a more even tension of the warp threads, both for the fabric formation cycle and for the shedding cycle. The advantage of the brake regulators is the high accuracy of the warp feed, that is, the mechanism is very sensitive to such factors as the heterogeneity of the weft thread, the different degree of deformation of the warp threads when they are worked into the fabric. There proposed a stand that simulates the operation of a friction clutch of a brake regulator, and a method for determining the quantitative and qualitative characteristics of friction materials for friction units of a brake regulator of looms. There obtained experimental dependences of the friction coefficient at rest and the coefficient of friction in motion on the sliding speed and specific pressure in the friction clutch of the brake controller. It has been established that with an increase in the specific pressure in the friction pair (ferrado-ferrado), the friction coefficient at rest and the friction coefficient in motion increase. The friction coefficient in motion decreases with increasing relative sliding speed. The coefficient of friction at rest is greater than the coefficient of friction in motion, and the difference increases with increasing relative sliding speed. In the friction clutch of the brake regulator, during the transition from static friction (friction coefficient at rest) to kinematic friction (friction coefficient in motion), relaxation oscillations occur, leading to an abrupt motion of the weaving beam. Conditions are obtained and recommendations are given under which these spasmodic movements of the weaving beam are eliminated. To stabilize the moment of resistance on the warp beam, it is necessary to use a brake regulator with a movable drive disk, in which only a stable coefficient of friction in motion takes place in the friction clutch. The conditions under which relaxation fluctuations occur in the mechanical friction model, namely, in the friction clutch of the brake regulator, have been determined. The friction force is set for a long contact time, with zero contact time and contact resistance. The anti-rotation mode of the drive disc ensures the stability of braking in the friction pair of the brake regulator.

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