Investigation of warp tension on looms with a new automatic brake

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Abstract. In this article, the task was set to study the existing main planetary regulator on machines of the AT type. It is proposed to modernize the design of the planetary regulator, expediently providing for the features of the production of fabric from boiled natural silk. This will allow you to choose the optimal warp tension mode and reduce the breakage of the warp threads on the loom.

Keywords: breakage, main threads, beam, planetary regulator, tension, influence, rock, friction, arises, brake, reverse, pulley, washer.

1 Introduction

The theoretical and practical material of the study of a new mechanism for tensioning and releasing the warp, installed on the machine AT 100-5M, is presented. To improve the design of the new brake, the following tasks were set:

a) ensuring the alignment of the tension of the warp threads by coordinating the cyclic oscillation of the rock with the cyclo-rama of the operation of other mechanisms of the machine;

b) the implementation of the high sensitivity of the mechanism due to the optimal friction force in the friction pair brake band-brake pulley;

c) ensuring good conditions for the process of beating the weft thread to the edge, as well as the necessary “play” of the warp;

d) maximum use of the details of the existing main planetary gear and the simplicity of the design of the new main brake.

2 Materials and methods

The design of the main release and tension mechanism that meets these requirements were created by us according to the scheme of Fig.1.

In the proposed design of the negative automatic belt brake, the following parts of the existing main planetary regulator are used: pile gear Z3, shaft 19 for pile gear Z2, differential gear (as a brake pulley) 14, pawl with bushing 21, pressure lever 5 with spring 9 and thrust 6, shaft 8 with probe 18. The rest of the parts were beaten out of the mechanism.

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New parts for the brake are brake band 13, brake lever 7, pin 12, and single-arm lever 16 with spring 15. The principal distinguishing feature of the proposed design is that the running end of the brake band is connected through springs 15 to lever 16 and shaft 8 with warp probe 18.

The new brake contains a swinging rock 1, installed by pins on the bearings of bracket 2, fixed on rock 3, freely installed in bearings 4. A pressure lever 5 is also fixed on rock 3, the end of which is pivotally connected by rod 6 to the brake lever 7, pivotally connected to shaft 8. A spiral spring 9 is freely put on rod 6, one end of which is in contact with the nut 10, and the other end rests against the fixed support 11, fixed on the machine frame. The brake lever 7 is connected to the brake band 13 using a pin 12 installed in the eyelet, which wraps around the brake pulley 14. The other end of the band through the spring 15 is connected by a one-arm lever 16. The lever 16 is rigidly fixed to roller 8 and supported by bearings 17. The other end of roller 8 is rigidly connected to probe 18.

Fig. 1. The design of the main release and tension mechanism.

On the roller 19 is fixed the stock gear Z2, which is engaged with the piled gear cast integrally with the flange of the weaving pile 20. The locking pawl 21 is rigidly fixed on the shaft 19 and is in contact with the toothed thread Z1 of the outer plane of the brake pulley. The locking pawl fixes the pulley 14 on the roller 19. The pawl 21 can be disengaged from the brake pulley using a special handle to rotate the beam from the handle. The tension of the main threads is regulated by moving the finger 12 along the eye of the brake lever 7 in one direction or another, changing the force of the spring 9 with the nut 10 and selecting the materials of the rubbing surfaces between the tape 13 and the pulley 14. During shedding and surf, the main threads experience additional tension, as a result of which the rock will sink and, overcoming the forces of the spring 9, through the lever 5 and the rod 6, turn the brake lever 7 clockwise. The latter, turning freely on roller 8, weakens the action of tape 13 on pulley 14. Under the influence of the increased tension of the warp threads, the beam turns, feeding a certain length of wound threads into the working area of the machine. The length of the main threads unwound from the beam depends on the force of the spring 15, which creates a braking moment on the pulley 14. With a decrease in the diameter of the winding of the main threads on the pile 20, the probe 18 under the action of the spring 15 turns the roller 8 and the one-arm lever 16 counterclockwise, as a result of which the force on the brake band 13 decreases, and the tension of the main threads remains.
constant. During the operation of the machine, the warp threads located in the working area are subjected to longitudinal cyclic deformations under the action of working mechanisms. As a result, a cyclically changing tension force of the warp threads will act on the beam and the movable rock. Under the influence of this force, the beam can make a rocking motion. When the brake pulley rotates in the opposite direction, the belt tension forces $S_H$ and $S_{ob}$ (Fig. 2) change places and the weight of the load required for braking increases $S_{ob}$.

In the existing designs of the main band brakes, the running end of the brake band is attached to a fixed point (for a larger force $S_H$), so only a lower tension force acts on the brake lever. Fig. 2: The belt tension forces.

In the presence of tangential friction forces, the tension of the tape in the sections of the elementary arc will be $S + dS$ and $S$. In this case, the projection of forces on the direction $\alpha N$.

\[
(S + \alpha S)\sin \frac{\alpha \varphi}{2} + S\sin \frac{\alpha \varphi}{2} - \alpha N = 0
\]

\[
(S + \alpha S)\cos \frac{\alpha \varphi}{2} + S\cos \frac{\alpha \varphi}{2} - \mu \alpha N = 0
\]

\[
\sin \frac{\alpha \varphi}{2} \approx \frac{\alpha \varphi}{2} \cos \frac{\alpha \varphi}{2} \approx 1
\]

Taking $\sin \frac{\alpha \varphi}{2} \approx \frac{\alpha \varphi}{2} \cos \frac{\alpha \varphi}{2} \approx 1$ and discarding the infinitesimal higher order ($\alpha S \alpha \varphi$) we get

\[
\alpha S \varphi = \alpha N; \\
\alpha S = \mu \alpha N
\]

\[
\int \frac{\alpha S}{S} = \int \mu \alpha \varphi \text{ or } \ln S = \mu \varphi + C \\
\text{At } \varphi = 0
\]
when \( S = S_c \delta \), \( \ln S = \mu \phi \); \( \log S/S_c = \mu \phi \).  

\[
S_H = S_c \delta^n \mu \phi
\]

where \( \phi \) — pulley wrap angle, rad; \( l \) is the base of the natural logarithm, equal to 2.718…

The greatest specific pressure between the belt and the brake pulley occurs at the point of application of belt tension \( S_H \) (Fig. 3).

Fig. 3. The greatest specific pressure between the belt and the brake pulley

\[
P = S_H - S_c \delta
\]

\[
S_H = S_c \delta^n \mu \phi
\]

\[
S_H = \frac{p \cdot e^{\mu \phi}}{e^{\mu \alpha - 1}} S_c \delta = \frac{p}{e^{\mu \alpha - 1}}
\]
where \( S_H \) and \( c_\delta S \) — the force of the incoming and outgoing end of the tape;

\( e \) is the base of the natural logarithm;

\( \mu \) — coefficient of friction;

\( \alpha \) — pulley wrap angle, rad;

\( P \) — circumferential force on the surface of the pulley.

To determine the relationship between the tensions \( S_H \) and \( c_\delta S \), we select an elementary arc (Fig. 3) corresponding to the angle \( \varphi \), which is the distance from the point under consideration to the point where the tape runs off the pulley. With an increase in the force \( S_H \), the return rotational motion of the beam decreases, thereby reducing the compensation of the warp tension.

The automatic main band brake developed by us (author's certificate No. 681128) was tested in the production of satin (satin) weave fabric [1,2]. Let us determine the tension of the warp threads on a loom with a negative automatic main band brake. During the operation of the loom, the tension of the warp threads consists of static and dynamic components.

\[ K = K_{st} + K_{dyn} \]

To determine the static component of the tension, consider the forces acting on the links of the main brake (Fig. 4).

From the equilibrium condition of the beam, we have

\[ \sum_{m=1}^{2} \left( K_s - \varphi r - H_z - \frac{z}{z} \right) = 0 \]

Then

\[ N = Q_T f + Q_H f + H_z \frac{z}{z} + \frac{z}{z} \cos \gamma \]

\[ K = \frac{Q_T f r + Q_H f r + H_z \frac{z}{z}}{\varphi + f r \cos \gamma} \]

Fig. 4.
Here the mass of the main threads; $Q_T$

$Q_H$ - the mass of the empty beam; $H$

$f$ - coefficient of friction in the supports of the weaving beam. From the equilibrium condition of the brake pulley (Fig. 4, b)

$\sum m_{z_1} = H z_1 r_1 + TR_{uc} - t R_{uc} = 0$

$H z_1 = \frac{(T-t) R_{uc}}{R_{uc}} = \frac{t (\frac{f}{\alpha} - R_{uc})}{R_{uc}}$

$t = c \lambda$

$\lambda = \lambda - \Delta \lambda; \Delta \lambda = \frac{1}{2} (\tilde{n} - \tilde{n})$

$\lambda = \lambda - \frac{1}{R} (\tilde{n} - \tilde{n})$

$t = \left[ \lambda - \frac{1}{R} (\tilde{n} - \tilde{n}) \right]$

$K_d = \frac{Q_T f r + Q_H f r}{\tilde{n} + f r \cos v} + \frac{C R_{uc} R_z (\frac{f}{\alpha} - \frac{1}{R})}{R_z (\tilde{n} + f r \cos v)} (\tilde{n} - \tilde{n})$
The dynamic component of the thread tension is due to the inertial resistance of the moment of inertia. With an average diameter of a weaving beam, the moment of inertia is 0.05 m, which will lead to a decrease in the static tension of the main threads.

Radius \( R = 18 \) cm, \( l = 5 \) cm, \( r = 0.068 \) m, \( Z = 0.5 \) m, \( f_2 = 0.62 \) kg (6.2 N), \( L = 0.135 \) cm, \( l = 0.06 \). The calculated values of changes in the length of the weaving beam are given in Table 1.

### Table 1

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<th>( R )</th>
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Day of simplification of formula:

\[ K_{cm} = \frac{0.01 Q_T}{q+0.01 \cos \psi} + \frac{11.45 \lambda_0}{0.066(qx+f_1 r \cos \psi)} - \frac{1.14(q_{max} - q_2)}{0.066(qx+f_1 r \cos \psi)} \]

\[ K_{st} = \frac{\tilde{n}_{max} - \tilde{n}_x}{q+0.01 \cos \psi} + \frac{11.45 \lambda}{0.066(qx+f_1 r \cos \psi)} = \frac{103.0.01}{0.25} + \frac{11.45 \cdot 0.05}{0.066 \cdot 0.25} = 4.12 + 34.5 = 8.62 \text{ kg} = \frac{386.2H}{2} \]

\[ K_{st} = \frac{0.01 \cdot 40}{0.18} + \frac{11.45 \cdot 0.05}{0.066 \cdot 0.15} - \frac{1.14(0.25 \cdot 0.17)}{0.066 \cdot 0.18} = 2.22 + 48.06 - 7.66 = 42.62 \text{ kg} = 426.2H. \]

\[ K_{st} = \frac{11.45 \cdot 0.05}{0.066 \cdot 0.11} - \frac{1.14(0.25 - 0.11)}{0.066 \cdot 0.11} = 81.7 - 22.8 = 42.62 \text{ kg} = 589H. \]
Inertia reduced to the axis of the weaving beam.

In the general case, the dynamic component is determined by the formula known in the literature:

\[ K_{din} = \frac{J_{pr} \varepsilon}{\hat{n}} \]

where,

- \( J_{pr} \) is the moment of inertia reduced to the axis;
- \( \varepsilon \) - angular acceleration of the beam, determined empirically or from its equation of movement;
- \( \hat{n} \) is the radius of the weaving beam.

The dynamic component of the warp tension is not a constant value and changes according to a complex law.

To approximate the calculations of the dynamic component of the warp tension, we will take according to the theory of Prof. V. A. Gordeev, the movement of the beam is uniformly accelerated during the acceleration period and uniformly slowed down during the braking period. We will consider the length of the base, twisted from the beam for each cycle of the machine, as a constant value. In this case, the dynamic component of the warp tension is determined by the well-known formula:

\[ 2 \alpha \rho \Delta L \rho \frac{\alpha}{\sqrt{n}} = \frac{J_{pr} \alpha}{\hat{n}} \]

where,

- \( \Delta L \) - the length of the warp, twisted about the pile for each cycle of the machine;
- \( \alpha \rho \) - moment of inertia of the mass of the beam with the warp;
- \( n \) - the number of revolutions of the main shaft of the machine per minute;
- \( \rho \) - the radius of winding the warp on the pile;
- \( \alpha \) - the fraction of time of one revolution of the main shaft during which the beam moves.

Consequently, the dynamic component of the warp tension is proportional to the square of the number of revolutions of the main shaft of the machine, the moment of inertia of the beam and is inversely proportional to the square of the winding radius, the square of the time of the beam movement.

The length of the warp that is wound from the beam for the cycle of the machine can be determined by the well-known formula:

\[ L = \frac{1}{y_R} \left( 1 - \frac{1-\alpha}{100} \right) \]

where

- \( y_R \) is the weft density of the fabric;
- \( \alpha \) - working out the base.

The moment of inertia reduced to the beam axis is determined by the formula:

\[ J_{pr} = J_H \rho + J_G \left( \frac{W_{G}}{W_{H}} \right) = J_H \rho + \left( J_{Z_1} + J_{Z_2} \right) i^2 \]

where

- \( J_H \rho \) - moment of inertia of the beam with the warp;
- \( J_G \) - moment of inertia of the loads;
- \( J_{Z_1} \) - moment of inertia of the brake pulley;
- \( J_{Z_2} \) - moment of inertia of the root gear.
\[ J_i = \frac{nH_j}{\gamma} (\mathcal{g} - R_w) \]

where \( H \) is the distance between the beam flanges; \( \gamma \) - specific winding density; \( \mathcal{g} \) - acceleration of gravity; \( R_w \) - beam trunk radius.

Substituting the value of equations (10), (11) into (9), we obtain

\[ K_{\text{din}} = \frac{n + \alpha}{\bar{n}} \left[ J_H + J_G + \left( J_{Z_2} + J_{Z_3} \right) \right] \]

The dynamic component of the tension of the warp threads during the operation of the weaving beam changes with a decrease in the radius and moment of inertia of the weaving beam.

The results of changing the dynamic component are given in Table 2.

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\[ K_{\text{tot}} = \frac{Q_T f_1 r + Q_H f_1 r}{\bar{Q} + f_1 r \cos \nu} + \frac{[\lambda_0 - \frac{f_2}{2} (q_{\text{max}} - q_x)] (f_2 \alpha - 1) R_{w2} (1 + \frac{\alpha_0}{100}) n^2}{P_y 900 \bar{q}^2 \alpha_i^2} \]

\[ + \left[ J_H + J_G + \left( J_{Z_2} + J_{Z_3} \right) \right] \]

As can be seen from Fig. 5, the dynamic component is significantly reduced in size when the weaving beam is triggered. When substituting equations (6) and (13) in (1), we find the total tension of the warp threads in the weaving process.

\[ K_{\text{tot}} = \frac{Q_T f_1 r + Q_H f_1 r}{\bar{Q} + f_1 r \cos \nu} + \frac{[\lambda_0 - \frac{f_2}{2} (q_{\text{max}} - q_x)] (f_2 \alpha - 1) R_{w2} (1 + \frac{\alpha_0}{100}) n^2}{P_y 900 \bar{q}^2 \alpha_i^2} \]

\[ + \left[ J_H + J_G + \left( J_{Z_2} + J_{Z_3} \right) \right] \]

The estimated data of the total tension of the warp threads for fabric production art. 002 Uz on a loom with a modernized tension and release mechanism are given in Table 3 and in Fig. 5.

Analysing the results of research on the operation of the new mechanism of release and tension, we can draw the following conclusions:

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\[ K_{\text{tot}} = \frac{Q_T f_1 r + Q_H f_1 r}{\bar{Q} + f_1 r \cos \nu} + \frac{[\lambda_0 - \frac{f_2}{2} (q_{\text{max}} - q_x)] (f_2 \alpha - 1) R_{w2} (1 + \frac{\alpha_0}{100}) n^2}{P_y 900 \bar{q}^2 \alpha_i^2} \]

\[ + \left[ J_H + J_G + \left( J_{Z_2} + J_{Z_3} \right) \right] \]

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1. The tension of the main threads on looms with a new negative brake is whiter and more uniform when the weaving needle is triggered, which should have a positive effect on the quality technological process in the direction of reducing the breakage of the warp threads and improving the quality of the fabrics produced art.
2. The main negative brake developed by us has a significant advantage over the planetary regulator, since its design and adjustment are simple, and the use of a transmission mechanism in the brake increases the reliability of its operation by reducing the load on the brake band. The use of a swinging rock in the negative brake also helps to equalize the tension of the main threads during the cycle of the loom, it is especially advisable to use it in the production of plain weave fabrics.

Figure 4 shows a diagram of a swinging rock and the action of the spring $F_0$ in the conditions of equilibrium of the moments of forces relative to the lever of rotation of the sub-rock $O_3$.

Let rock 1 rotate in the middle bearings of the bracket at a distance $\alpha$ from the center of rotation of the rock $O_3$. In this case, the angle $\beta_1$ (with a full weaving beam) is equal to the angle $\beta_2$ (finishing the warp from the beam). The common side of angles $\beta_1$ and $\beta_2$ with bracket 2 is $90^\circ$.

From the condition of equilibrium of the tightening of the spring, the tension of the base and the total weight of the moving system, we find the force $K_{st1}$ (the calculation did not take into account the friction of the rock and rock in the bearings).

The total tension of the main threads acting on the rock consists of a static and dynamic component:

$$K_{tot} = K_{st} + K_{din}$$

Let us determine the dynamic component of the warp tension. Let the tension increase by $\Delta K$ during the formation of the shed by a certain point in time. In this cement warp tension will be

$$K_{tot} = K_{st} + \Delta K$$

Fig. 5. Estimated data of the total tension of the warp threads of conferences 431, 06033 (2023) ITSE-2023 https://doi.org/10.1051/e3sconf/202343106033
At the same time, with an increase in the tension of the base by $\Delta K$, under the rock, overcoming the resistance of the spring $F$, it will turn through an angle $\phi$. Therefore, the tension of the warp at the moment will be

$$K_{tot} = K_{st} + \Delta K - (\alpha + r)C\phi$$

$$F = F_0 - \phi l_{C2}$$

The increase in warp tension $\Delta K = \lambda_1 C_1$ can be approximately replaced by a trigonometric function:

$$p = \frac{\pi}{T}$$

$$MK_{tot} = [K_{st} + \lambda_1 C_1 + C\sum_{n=1}^{N} \lambda_n \sin(npt + \delta_n) - (\alpha + r)C \phi \alpha \cos \beta - \delta_n]$$

$$MF = [F_0 + C \cos \beta - \phi l_{C2}]$$

$$J_{cpu} = K_{cpu} \cos \beta - F_0 - G L + \left[\lambda_1 C_1 + C\sum_{n=1}^{N} \lambda_n \sin(npt + \delta_n) - (\alpha + r)\phi l_{C2}\cos \beta - \phi l_{C2}\right]$$

where $\lambda_1$, $\lambda_2$, ... $\lambda_n$, $\delta_1$, $\delta_2$, ... $\delta_n$, $t$, $\phi$, $\alpha$, $\cos \beta$, $\lambda_1 C_1$. The circular frequency of the cyclic deformation of the elastic filling system is determined by the equation:

$$\omega = \frac{2\pi}{T}$$
\[ \varphi = \varphi f + h + \sum_{n=0}^{\infty} \sin (n \pi t + b) \]

\[ h_n = \frac{\lambda_c C \alpha \cos \beta}{J_{np}} \]

\[ \varphi = V + U = A \cdot f + \beta \cos ft + \frac{h_n}{f^2} - h_n \]

\[ \sin (n \pi t + b) + \ldots + h_n/(f^2 - (n \pi)^2) \sin ? (n \pi t + b) \]

\[ \varphi = \frac{h_n}{f} + \sum_{n=0}^{\infty} \left( \frac{-\alpha (a + r) \cos \beta}{L C - C \alpha (a + r) \cos \beta} \right) \sin (n \pi t + b) \]

\[ \varphi = \frac{\lambda C \alpha \cos \beta}{L C - C \alpha (a + r) \cos \beta} + \sum_{n=0}^{\infty} \frac{\lambda C \alpha \cos \beta \sin (n \pi t + b)}{(a + r) \cos \beta - (n \pi)} \]

\[ K_{tot} = K_{st} + \Delta K = \frac{(a + r) C \lambda \alpha \cos \beta}{L C - C \alpha (a + r) \cos \beta} + \sum_{n=0}^{\infty} \frac{(a + r) \lambda \alpha \cos \beta \sin (n \pi t + b)}{L C - C \alpha (a + r) \cos \beta - J_{np}} \]

\[ \Delta K = K_3 \]

Fig. 6.
thrust b will go down. The angle of coverage of the brake pulley by the tape will increase, and the release of the warp from the beam will decrease. The tension of the warp threads will equalize again. Thus, the tension of the main threads is equalized in the process of their operation from the weaving beam.

3 Conclusions

1. Studies and long-term observations of the work of looms AT-100-5M during the production of fabric from boiled natural silk showed that, in addition to relative humidity and temperature in the workshop, the mechanism of release and tension of the warp affects the breakage.

2. An increase in the tension of the warp during the operation of the weaving needle by 72.5% increases the breakage of the warp threads by 25%, as a result of which the quality of the produced fabrics deteriorates. The main reason for the increase in the breakage of the main threads is the imperfection of the main regulator.

3. It is advisable to modernize the existing designs of the planetary regulator, providing for the peculiarities of fabric production from boiled natural silk. This will allow you to choose the optimal warp tension mode and reduce the breakage of the warp threads on the loom.

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