Method for determining damping coefficient, characteristic friction force in the needle mechanism

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Abstract. The article presents the development and justification of parameters of the new design of the sewing machine needle mechanism with elastic coupling, which provide an increase in speed modes of operation while reducing operating costs. As well as new scientific results obtained based on theoretical and experimental studies. A new design of the needle mechanism of a sewing machine with an elastic connection is developed and the problem of the dynamics of the needle mechanism with an elastic connection is solved by taking into account the dynamic and mechanical characteristics of the engine, inertial, elastic-dissipative parameters of the mechanism, as well as the resistance force of the sewn materials by the sewing machine. The dependences of the parameter changes in the function on the resistance force of the cross-linked materials are obtained.

1. Introduction
The current level of technology development offers the use of machines, mechanisms and working bodies that perform reciprocating, rocking or complex combined movements. Such mechanisms are used both in periodic machines (weaving, knitting, sewing, combing, shoe, etc.) and continuous action (mechanisms of removable combs of carding machines, tangle mechanisms of carding and tape machines, etc.) [1,2,3]. As a rule, machines containing such mechanisms are vibro active and require the use of vibration isolators. However, vibration isolation, protecting the Foundation and the floor of the production room from the impact of dynamic loads from the machines, does not change the size or nature of the loads in the machine itself and therefore cannot serve as a means to guarantee the normal operation of the machine. This negatively affects the process. When designing any mechanism, the designer has to make, as a rule, a compromise solution that would satisfy many, in most cases, conflicting requirements. The same situation occurs when designing the needle mechanism of a sewing machine with an energy storage device. For example, when selecting the parameters of an energy storage device, it is impossible to proceed only from the requirement of maximum reduction of inertial loads in the kinematic pairs of the drive mechanism. Here it is necessary to take into account the requirements of reliability and performance of the device itself. Obviously, these tasks are contradictory, since fulfilling the first requirement may make it impossible to fulfill the second requirement. To correctly determine the parameters of the mechanism and device, it is necessary to reasonably select all the criteria: design, technological and economic, ensuring their normal functioning [4].
2. Research methods

The dynamic analysis happened to with use the known methods of the decision of the differential equations, as well as the general methods to theories mechanism technological machines, theories of the fluctuations of the complex systems. The experimental studies on determination of the reactions in acting joint of the mechanism were conducted by method tensor. The experimental studies on optimization main parameter new mechanism were conducted by method of the mathematical planning [5].

The study used methods of mathematical statistics, higher mathematics, theoretical and applied mechanics.

As already noted, the criteria do not allow you to judge the loads in the device itself, which can cause the failure of the device elements or failure to perform the tasks assigned to it.

It is not possible to obtain the numerical value of the damping coefficient purely by calculation. The only way to get this coefficient is an experimental method. In practice, when developing a mechanism and preparing it for production, as a rule, a prototype is produced, on which scientific experiments can be carried out to obtain the missing information and clarify its individual parameters [6].

To conduct an experiment to determine the \( \eta \), we can take, in the first approximation, a spring whose stiffness satisfies the condition \( K_i = m\omega_i \) where: \( \omega_i \) - is the working circular frequency of the designed needle mechanism.

Next, select the method for measuring the load between the master and slave links. If the resistance to the movement of the needle mechanism is large enough, the experiment is performed at the operating frequency \( \omega_i \):

\[
R = kx_0 + ak + \eta\omega_i\alpha \sin(\omega_it - \varphi)
\]

It is known that the amplitude value of the variable component of the re reaction can be measured most easily on an oscillogram \( R_e \). The resulting value of \( R \) is equated to the coefficient \( \sin(\omega_it - \varphi) \)

\[
R_e = \eta\omega_i\alpha
\]

Where: \( \eta\omega_i\alpha \) - the force of resistance to the movement of the mechanism, hence: \( \eta = \frac{R_e}{\omega_i\alpha} \)

If the resistance force in the needle mechanism is small \( R_e \), then the value of the re measured on the oscillogram at \( \omega = \omega_i \), it will be determined with a large error. In this case, the experiment should be performed at the frequency \( \omega = \varepsilon\omega_i \), where \( \varepsilon = 0.8 \) or \( \varepsilon = 1.2 \).

Then:

\[
\eta = \frac{1}{\varepsilon\omega_i} \sqrt{\frac{R_e^2}{\alpha} - \left(K - m\varepsilon^2\omega_i^2\right)^2}
\]

After determining \( \eta \) the value we substitute its value in the formula \( K = m\omega_i^2 \), \( X_0 = \alpha \left( \frac{\eta}{m\omega_i} \right) - 1 \).

If \( b > 0 \), then the elastic element must be given a preliminary deformation \( X_0 \) and this is where the selection of its parameters ends. If \( X_0 < 0 \), then the stiffness of the elastic element must be recalculated using the formula:

\[
K = \frac{1}{2} \left( m\omega_i^2 + \frac{\eta^2}{m} \right), \quad X_0 = 0
\]

If \( \eta < m\omega_i \). In this case, a preload is not needed (in practice, a preload is introduced during commissioning to compensate for the static friction forces of the driven part of the needle mechanism).
Figure 1 plots R(\xi) for various values of the dimensionless attenuation coefficient [7].

\[ \xi = \frac{\eta}{2m\omega_1} \]

Where, on the abscissa axis is the ratio of the working frequency of the needle mechanism to the natural frequency \( \omega_1 \), free oscillations of mass m on the spring. By \( KX_0 \), the solution R is indicated with the introduction of preliminary deformation of the \( X_0 \) of the spring and with [8].

\[ \frac{\omega}{\omega_1} \rightarrow 0 \]

In the mechanism of the needle movement of the point B needle bar, we will consider in a fixed system report \( X B_2 Y \). Determine the acceleration of point B of the needle bar in its extreme positions \( B_1 \) and \( B_2 \), suppose crank speed \( \alpha_k = \text{const} \). Then we get the relationship between the speeds \( v_A \) and \( v_B \).

\[ v_B = v_A \frac{OB}{OA} \]

Where: \( OB \) is the segment cut off on the straight line \( OY_1 \) which is a continuation of the connecting rod symmetry axis (figure 2) [9].

The acceleration of point B is determined by the expressions: \( y = OB \)

\[ \alpha = \omega_k \frac{dy}{dt}, \quad y = (OA + AB - x)\tan \beta, \]

Then \( \alpha \) it will be equal to:

\[ \alpha = \omega_k \left[ \frac{dx}{dt} \tan \beta + \frac{1}{\cos^2 \beta} \frac{d\beta}{dt} \right] \]
Where: \( k_1 \) - scale factor for \( y=OB \), from formula (1) it can be seen that \( OA \sin \alpha = AB \sin \beta \), 
\[
OA \cos \alpha \frac{d\alpha}{dt} = AB \cos \beta \frac{d\beta}{dt}
\]

Therefore, for the right extreme position of point B, we have:
\[
\alpha = 0; \beta = 0; \chi = 0; \frac{dx}{dt} = 0; \frac{d\beta}{dt} = \lambda \omega,
\]

Where: \( \lambda = r/l \) – coefficient determining the uniformity of the needle bar stroke. Similarly, for the left extreme position of point B we have: \( \alpha = 180^0; \beta = 0; \chi = 2r; \)
\[
\frac{dx}{dt} = 0; \frac{d\beta}{dt} = -\lambda \omega,
\]

In accordance with this, by the formula (1), we find the acceleration of the needle bar point \( B \) in the right and left extreme positions:
\[
a_n = r \omega^2 \left( 1 + \lambda \right), a_n = -r \omega^2 \left( 1 - \lambda \right)
\]

As you know, the function graph, depending on the needle bar path, has the form: \( a = f(S) \) where:
\( S = k_1 x \) - with sufficient accuracy can be mistaken for a parabola that passes through the points \( e_1(2k_1OA; a/\omega^2) \) and \( d_1(0; a_2/\omega^2) \), and the tangents to the parabola at these points intersect at \( g_1(k_1x_j; 3\lambda \omega^2 / k_1 \omega^2) \), where: \( k_1x_j \) represents the abscissa of the point of intersection of the straight line \( d_1e_1 \) with the axis of the guide (figure 3) [10].

![Figure 2. Scheme of the mechanism of the needle with an elastic bond.](image)

![Figure 3. The graph of the change in the dependence of the movement of the needle on acceleration.](image)
The center of mass of the moving links of the mechanism is determined by the vector:

$$\overline{OS} = \sum_{i=1}^{3} \overline{h}_i,$$

Where: $\overline{h}_i$ - the main point of the $i$-th link. If the vectors $\overline{h}_1$ and $\overline{h}_2$ the main points of the crank and connecting rod satisfy this condition.

$$\frac{h_1}{OA} = \frac{h_2}{AB}$$  \hspace{1cm} (2)

Then we get the following equalities: $m_1OS_1 = -m_{2A}OA$ representing the condition of balance of the rotating masses, which include the crank mass $m_1$ (figure 3) and the part of the connecting rod mass $m_2$, statically reduced to the crank point A and equal to $m_{2A} = m_2 \frac{BS_2}{AB}$. Thus, to implement condition (2), the crank must be shaped so that its imbalance relative to the axis of rotation is:

$$D_1 = m_{2A}OA$$

If the latter condition is satisfied, then the center of mass of the moving parts of the mechanism will move along the axis of the guide with acceleration $\ddot{a}$. Therefore, an unbalanced force will act along the axis of the guide

$$\overline{P}(S) = -m\ddot{a}$$  \hspace{1cm} (3)

Where: $m = m_3 + m_{2B}$ represents a progressively moving mass consisting of a mass of m3 needle bar and a connecting rod mass:

$$m_{2B} = m_2 \frac{AS_2}{AB}$$

Reducing dynamic loads in kinematic pairs from the action of force $\overline{P}(S)$, it will cause dynamic loads, which can create not only the above-mentioned negative phenomena, but in some cases serve as the main obstacle to improving the performance of the machine [11].

To completely unload rotational pairs from the action of force $\overline{P}(S)$, an elastic element with this characteristic must be installed between the needle bar and the stand $Q(S)$, to meet the requirements of the conditions:

$$Q(S) = \overline{P}(S)$$  \hspace{1cm} (4)

At any crank speed.

However, strength $\overline{P}(S)$ (3) is substantially nonlinear and depends not only on the abscissa of the needle bar point B, but also on the crank speed. This creates reasons; the implementation of equality (4) meets in the general case significant structural difficulties [11].

In solving this problem, it is advisable to use the theory of the uniform best approximation of functions due to the fact that only such an approximation can guarantee the deviation of functions $\overline{P}(S)$ and $Q(S)$ with a predetermined accuracy at all intervals of the change in the abscissa S of point B of the needle bar mechanism [12,13].

Thus, the task at hand is to first approximate the polynomial function:

$$Q(S) = AS + B$$  \hspace{1cm} (5)

The first degree evenly and in the best way on a segment:
And then determine the characteristic \( Q(S) \) of the elastic element from the condition

\[
Q(S) = -Q_i(S)
\]

Coefficients \( A \) and \( B \) of polynomial (1.5) should be chosen so that the value has minimum value.

\[
E_i = \max_{0 \leq S \leq 2r} |P(S) - Q_i(S)|
\]

The polynomial \( Q_i(S) \), which gives a minimum to \( E_i \), is called the polynomial of the best uniform approximation, or a polynomial that slightly deviates from the function \( P(S) \) on the segment if \( E_i \leq \varepsilon \), where: \( \varepsilon \) - a constant value, depending on the structure and parameters of the needle mechanism. Then from formula (8) it follows:

\[
|P(S) - Q_i(S)| \leq \varepsilon
\]

For all points \( S \in [0,2r] \). In this case, the polynomial \( Q_i(S) \) on the interval (7) will uniformly approximate the function \( P(S) \) up to a value [14].

3. Results and discussion

On base of the analysis existing design is designed new design of the mechanism of the needle of the sewing machine with springy relationship are received equations of the moving the mechanism with springy element at kinematics closing between leading and knowledge by parts of the mechanism; is solved problem speakers mechanism of the needle with springy relationship at account dynamic and mechanical feature of the engine, inertia, springy-dissident parameter mechanism, as well as power of the resistance sutured material by sewing machine; an experimental certain parameters and nature power laden mechanism with springy element and without it; also determined rational state of working sewing machine when use the springy drive to energy [15,16].

The task of determining, in this case, is somewhat complicated by the fact that the function \( P(S) \) is not explicitly known. However, since the function \( P(S) \) is a quadratic function, it can be argued that the function \( P(S) \) has a second derivative of constant sign on the interval (6). Under this condition, the linear function (5) of the best uniform approximation on the segment (6) will represent the geometrically average parallel between the intersecting \( ed \) passing through the extreme points \( e \) and \( d \) of the parabola segment and the tangent \( l_k \) to the parabola parallel to this intersecting lines [17].

In figure 3 functions \( P(S) \) and \( Q(S) \) are constructed as an example for a planar axial needle mechanism with a geometric parameter \( \lambda = 1/5 \).

The value to wake \( \varepsilon \) up is equal to:

\[
\varepsilon = k \omega_r^2 y_{2r}
\]

where: \( y_{2r} \) - the largest difference between the ordinates of the graphs of the functions \( P(S) \) and \( Q(S) \), shown in figure 4, characterizes the absolute accuracy of the approximation of the function by the polynomial \( Q_i(S) \) on the interval (6).

4. Experiment and result

4.1. A. Experimental results and optimization of sewing machine needle parameters

Practical value recommended design of the mechanism of the needle of the sewing machine, allows increasing to capacity under qualitative technology suture material. It is recommended broadly use in sewing production [18].
We now turn to the definition of the function $Q_i(S)$. The equation of the chord passing through the points $d [0, -OA(1+)]$ and $e [2OA, OA(1-)]$, the graph of the function defined by polynomial (4), has the form (figure 4):

$$y - x + OA(1 + \lambda) = 0$$

(11)

With the value of $y = 0$ (11), we determine the abscissa of the point $q$: $x_f = OA(1 + \lambda)$

Therefore, the coordinates of the midpoint $n_1$ of the segment $eq$ is equal to:

$$\begin{align*}
x_{n1} &= \frac{OA}{2} (3 + \lambda); \\
y_{n1} &= \frac{OA}{2} (1 + 2\lambda).
\end{align*}$$

(12)

With this sequence, we find the coordinates of the midpoint $n_2$ of the segment $qd$:

$$\begin{align*}
x_{n2} &= \frac{OA}{2} (1 + \lambda); \\
y_{n2} &= \frac{OA}{2} (2\lambda - 1).
\end{align*}$$

(13)

We draw a straight line through the point’s $n_1$ and $n_2$, using formulas (12) and (13) for this:

$$y - x + \frac{OA}{2} (2 - \lambda) = 0$$

(14)

The line intersects the ordinate axis at $k (0, -y_k)$, where:

$$y_k = \frac{OA}{2} (2 - \lambda)$$

(15)

And a straight line at $l$ with coordinates: $x - 2OA = 0$

$$\begin{align*}
x_l &= 2OA; \\
y_l &= \frac{OA}{2} (2 - \lambda)
\end{align*}$$

(16)

By virtue of the well-known property of the parabola, the line (14) passing through the midpoints of the segments $eq$ and $qd$ will necessarily touch the parabola. In addition, the segment $n_1, n_2$ of straight line (14), being the middle line of the triangle $eqd$, will be parallel to the intersection of $ed$ and the function graph $P(S)$ (figure 4).

This directly implies the statement stated above that the average parallel between the intersecting $ed$ and the tangent $kl$ to the parabola parallel to this intersecting line really represents a graph of the linear function $Q_i(S)$ that implements the best uniform approximation to the function $P(S)$ on the segment (6).

To define the function $Q_i(S)$ in explicit form, it is enough to write the equation of a line passing through the midpoints $s$ and $q$ of the segments $el$ and $kd$.

Taking into account formulas (15) and (1.16), we find the coordinates of the point’s $s$ and $q$:

$$\begin{align*}
x_s &= 2OA; \\
y_s &= \frac{OA}{2} \left(1 - \frac{\lambda}{4}\right); \\
x_q &= 0; \\
y_q &= -OA \left(1 + \frac{\lambda}{4}\right)
\end{align*}$$

(17)

From the equation of a line passing through points (17):
\[ y = x - OA \left( 1 + \frac{\lambda}{4} \right) \]  

(18)

Thus, the coefficients of polynomial (5) have values:

\[ A = (m_3 + m_{2h}) \omega^2; \quad B = -(m_3 + m_{2h})r \omega^2 \left( 1 + \frac{\lambda}{4} \right) \]  

(19)

We note that the deviation \( E_1 \) determined by formula (8) is realized at three points (figure 4).

\[ Q_1(0) = Q_1(S_k) = Q_1(2r) = E_1, \]

Where: \( S_k \) is the abscissa of the point of tangency of the parabola with the line (14). Substituting expression (19) into equality (7), we obtain the equation of characteristic of the elastic element of the mechanism [12].

**Figure 4.** The graph of the change in the force of action \( P(S) \) from the characteristics \( Q(S) \) of the elastic element.

**Figure 5.** The graph of the imbalance of the force \( P(S) \) and force \( Q(S) \).
\[ Q(S) = -(m_3 + m_{2B})\omega^2 \left[ S - r\left(1 + \frac{\lambda}{4}\right) \right] \]  

(20)

From formula (20) we find the stiffness of the elastic element in N/m:

\[ C = (m_3 + m_{2B})\omega^2 \]

The parameter \( S_p \) of the elastic element, at which the force \( Q(S) = 0 \), is determined by the equation:

\[-(m_3 + m_{2B})\omega^2 \left[ S_p - r\left(1 + \frac{\lambda}{4}\right) \right] = 0,\]

The decision we get \( S_p = r\left(1 + \frac{\lambda}{4}\right) \), and so, the zero point of the characteristic of the elastic element is offset relative to point C by a value \( \Delta = r\lambda / 4 \). Thus, the length of the elastic bond in the Free State will be determined by the following expressions

\[ S_B = S_p + S_p. \]

Based on the foregoing, it can be concluded that reducing dynamic loads in the kinematic pairs of the needle mechanism allows increasing the reliability and durability of the links, reducing the frequency of oscillations of the working bodies, and increasing productivity, maintaining the quality of products on light industry sewing machines.

5. Conclusion

It was experimentally determined that at a speed of the main shaft of 4500 rpm to ensure high performance and the lowest load of supports and materials at the level of 4.5 mm, the stiffness coefficient of the elastic element of the mechanism should be 12.5 N/mm. An analysis of the experimentally obtained values of the acting loads in kinematic pairs, both in the existing needle mechanisms and in the recommended mechanism with elastic coupling, established that the difference in the acting reaction forces in the mechanism of the needle with an elastic element is 2.0-2.5 times less than in the existing one the mechanism. The expected economic effect of the introduction of a modernized sewing machine with a needle mechanism with elastic bond is 12176 sums per sewing machine per year.

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