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Dynamic Turn off Vibrations and Swings on Sewing Machines

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Abstract: In this article, the forces that affect the sewing machine connections and, for these reasons, the negatives, failures that result from these vibrations and these faults are governed by the sewing quality of the machines. Dynamic models of the elastic element flexible shock absorber mechanism were developed and theoretical problems were solved, the stress characteristics and parameters of the elastic element flexible shock absorber mechanisms were reported by putting forces at different masses to determine the experimental methods.

Keywords: sewing machine, vibration, mass, shock absorber, spring, extinguisher, speed, platform, link, kinematic pair, mechanism.

The main directions of the development of light industry machinery and technology are to dramatically increase the productivity of these machines and mechanisms and to obtain a wide range of high quality products [1].

In the process of effective design of technological machines of light industry cannot be done without a dynamic effect. The number of dynamic loads will be higher in high-performance machine mechanisms. In the garment industry, great attention is paid to the development of precise technological processes that require special laboratory equipment to reduce vibration levels, accurate measuring instruments and scientific research [2].

Our research to reduce vibration and vibration in sewing machines is also dedicated to high-speed sewing machines.

The problem of acoustic factors of the production environment (vibration and noise) is one of the most important issues in the current development period.

Today, in the light industry, machines and mechanisms consisting of advanced turning, vibration and complex moving workpieces are widely used. Such mechanisms are widely used in periodic and continuous motion machines and require the use of vibration dampers, mainly because they are considered active in vibration. However, while vibration dampers protect the machine from dynamic stresses acting on the foundation, they do not change the nature and value of the stresses in the machine itself. This has a negative impact on the technological process. Increasing the speed of the mechanism and working parts to increase the labor productivity of the machine leads to an increase in the dynamic and inertial stresses in the gears. Such stresses have a negative impact on the reduction of the service life of the mechanism links and kinematic pairs, the volume and quality of the product produced. If the reduction of inertial stresses in the kinematic pairs of the mechanism is achieved, it is possible to increase the operating speed of the mechanism, as well as reduce operating costs.

Based on the above, one of the urgent tasks is to create new designs of machines and mechanisms of the sewing industry, which will increase the speed of machines and reduce dynamic stresses.

In recent years, scientists and designers have developed new effective technologies for sewing materials of various characteristics, new types of shuttle and chain rods, as well as high-efficiency equipment for sewing production. On the improvement of sewing machines V.N.Gorbaruk, S.I.Rusakov, A.I.Komissarov, N.M.Archilov, V.L.Polukhin, L.B.Reybach, O.Suziki, V.B.Sherbekov, and other scientists have conducted scientific research.

For the development of sewing machinery and technology in Central Asia Z.Tadjibaev, K.Djemanikulov, A.Juraev, S.Baubekov, K.T.Olimov, S.Sh.Tashpulatov, D.S.Mansurova, I.M.Rakhmonov and others others have made tremendous contributions. The adequacy of the technology of sewing was studied, effective designs of working bodies of sewing machines were developed. However, little research has been conducted on the creation of new mechanisms with elastic elements of working bodies with high productivity, which provide high quality of sewing, reduce stresses in the links and kinematic pairs [3].

Machines used in the sewing industry mainly use periodic mechanisms (curved polzunli, curved karomisloli, curved backstage and so on). Increasing the speed in these types of mechanisms leads to an increase in dynamicstresses in the kinematic pairs. The increase in dynamic stresses leads to disruption of the technological process, premature failure of parts and deterioration of the quality of products. In the accelerated motion of the joints of the mechanism, the force exerted by the machine on the base includes the dynamic constituents. In a stabilized order, the dynamic constituents change periodically. This means that the machine is subjected to a periodically changing force on its base. Under the influence of this force, the foundation vibrates. By taking special measures to eliminate or reduce such harmful effects, these constituents should be reduced to zero or their amplitude should be limited to the allowable values. Such a problem-solving mechanism, which is related to the dynamic design of a machine aggregate mechanism, is called a mechanism [4,5].

It is necessary to study the types of imbalances in the mechanisms. Let us consider a flat mechanism in which the starting joint rotates at a constant angular velocity of 1 (Fig. 1, a). All the remaining joints move with angular acceleration, while the centers of mass C_1, C_2, C_3 have linear oscillations.

Let the construction of the mechanism joints be symmetrical with respect to the drawing plane, as is the case with the mechanisms of most machines. In this case, the main vectors of the inertial forces of all the joints, as well as the principal moments (resulting pairs) are located in this plane.

We bring the whole system of inertial forces to the center A (Fig. 1, b), so that the whole system is a common general vector:

$$\overline{\Phi}_{\Sigma} = \sum_{1}^{n} \Phi_i$$

Concentration on the general head moment:

$$M_{\Phi\Sigma} = \sum_{2}^{n} M_{\Phi i} + \sum_{2}^{n} M_A(\overline{\Phi}_i)$$

where n is the number of moving joints of the mechanism (Fig. 1, $a = n = 3$). Since $w = const$, $M = 0$, $M(\Phi) = 0$.The dynamic components of the base load F and M are numerically equal to the total general vector Φ and the total principal moment $M_{\Phi\Sigma}$of the system of inertial forces of all moving joints of the mechanism:

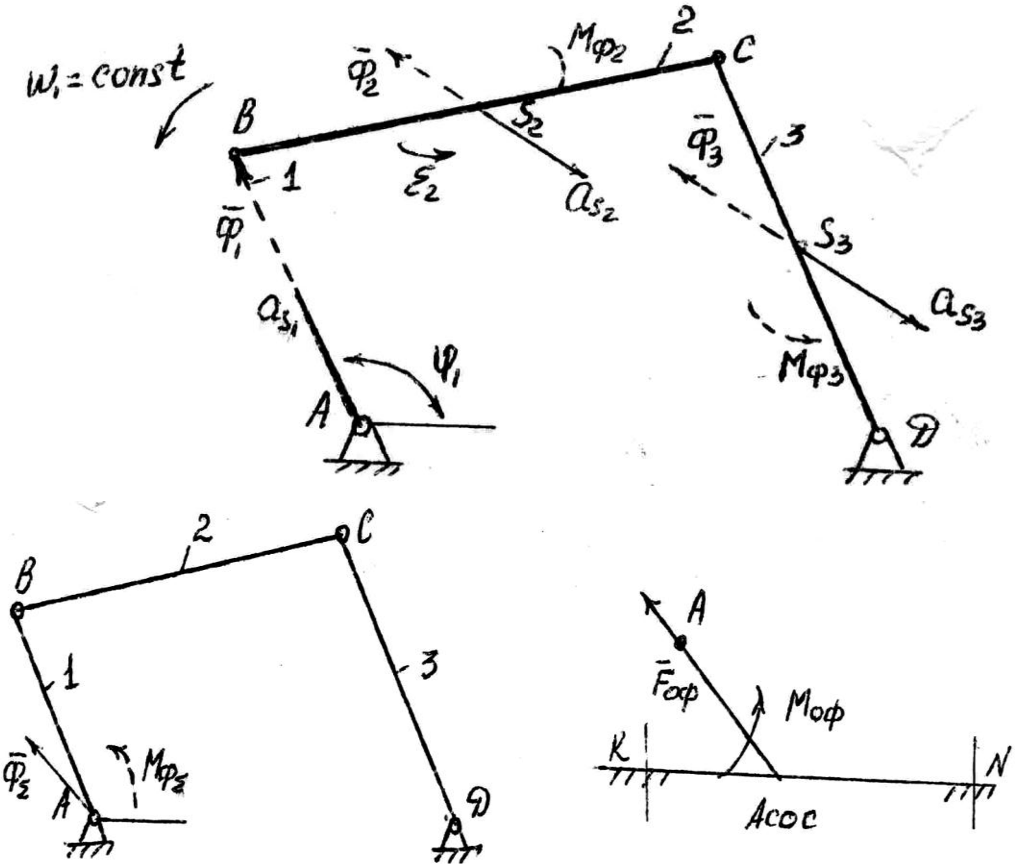

Figure -1: Types of imbalances in mechanisms

It should be noted that the forces loading the base are applied to the place where the machine body is attached to the base (in the places marked by Fig. 1 *K* and *n*). Therefore, Φ and M are pure computational quantities that represent only the sum of the results of the dynamic effect of the mechanism on the base. If the mechanism is the main vector of inertial force, $\overline{\Phi}_{\Sigma} \neq 0$, then such a mechanism is called a statically unbalanced mechanism. If $\Phi_{O\Phi} \neq 0$, but $M_{\Phi} \neq 0$, then such a mechanism is called a moment-unbalanced mechanism [6].

We will also consider the static balancing process as available. In designing the mechanism

$\overline{\Phi}_{\Sigma} = 0$

the mechanism of special measures taken to fulfill the condition is static balancing. It should be noted that at the same time the condition $M_{\Phi} \neq 0$is not intended to be fulfilled. Consequently, a statically balanced mechanism has no dynamic effect on its base in the form of any force .However, such a mechanism generally has a dynamic effect in the form of torque ($\overline{F}_{\Phi} = \overline{\Phi}_{\Sigma} = 0$).It is known from theoretical mechanics that: M_{Σ} is the mass of the system of all moving joints of the mechanism; a_c is the acceleration of the center of mass of this system. Hence, the condition is fulfilled only when a_c=0, which can occur only when the center of mass of the moving joints of the mechanism does not shift C.

Thus, static balancing is such that as a result, the center of mass of the moving joints of the working mechanism remains stationary. This can be achieved by the method of alternating masses.

In static balancing, only the main vectors of inertial forces of the joints are taken into account, and the main moments of inertial forces are not taken into account.

The static balances of the hinged four-joint mechanism (Fig. 2, a) are given by the lengths of the moving joints $л_1, л_2, л_3$ and their states of mass $м_1, м_2, м_3$ and centers of mass C_1, C_2, C_3

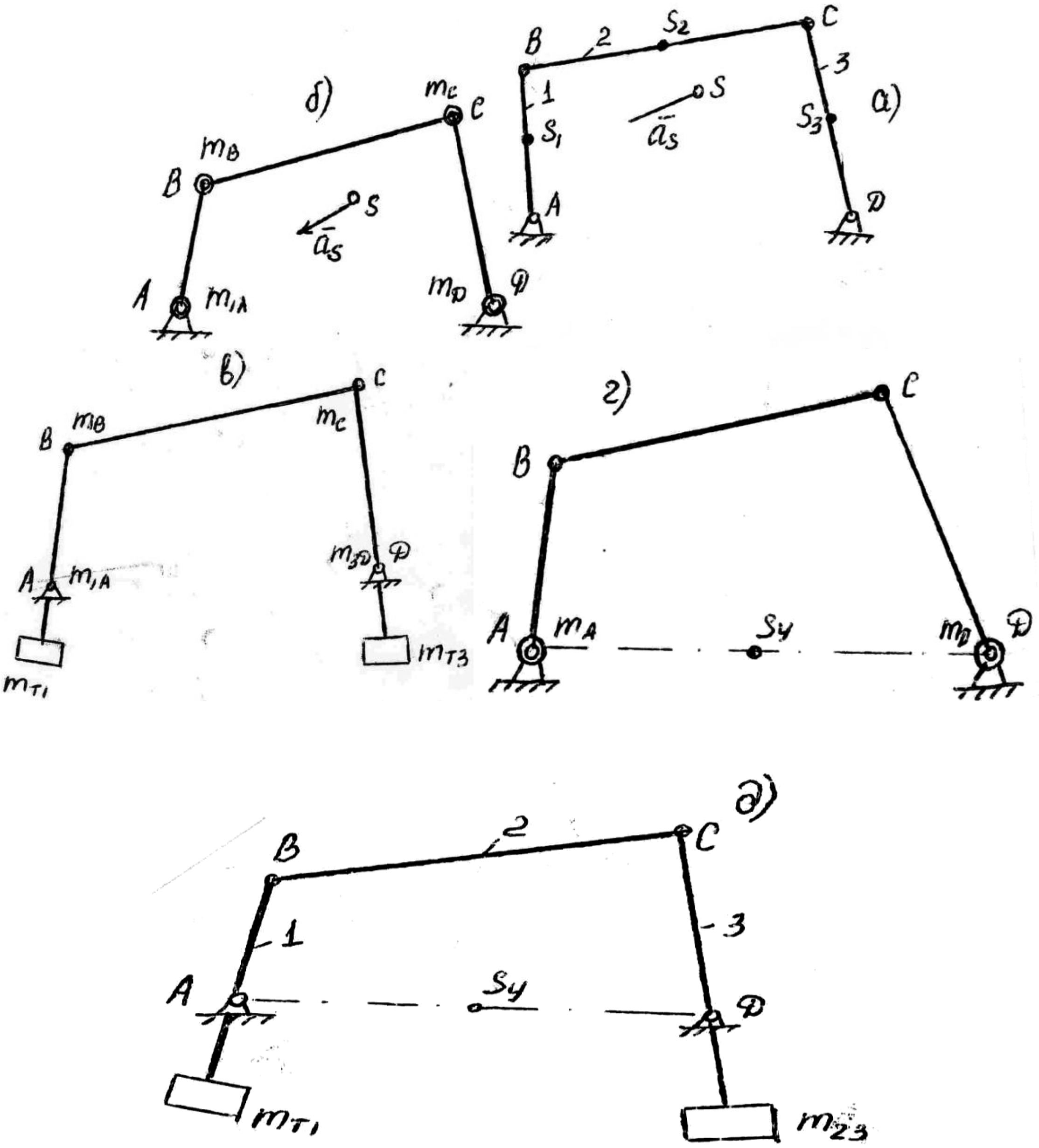

Figure -2: Static balancing

We replace each joint with two accumulated masses:

$м_{1A} = м_1 л_{БС} / л_1$,$м_{1Б} = м_1 л_{AC1} / л_1$, $м_{2Б} = м_2 л_{C2} / л_2$

$м_{2C} = м_2 л_{БC2} / л_2$,$м_{3C} = м_3 л_{ДC3} / л_3$, $м_{3Д} = м_3 л_{CC3} / л_3$

We combine the masses at points B and C:

$м_Б = м_{1B} + м_{2B}$,$м_C = м_{2C} + м_{3C}$

So, the given mechanism is exchanged with three masses located at points A, B, C, Д. The center of mass of the system C is located where the moving joints of the given mechanism 1, 2, 3, the center of mass of the system. When the mechanism is operating, the center of mass moves with acceleration C, which means that the given mechanism (Fig. 2, a) is not statically unbalanced.

For joints 1 and 3 ($м_Б$, $м_{Tu}$) we set posangi (forming masses) and so that the centers of mass of the system fall on the fixed points A and Д (Fig. 2, б). To do this, the following relationships must be met:

$м_{Tu}р_{Tu}= м_Бл_1$,$м_{T3}р_{T3}= м_Cл_3$

We combine the masses located at joints 1 and 3.

$м_A = м_{uA} + м_Б + м_{Tu}$,$м_Д = м_{3Д} + м_C + м_{T3}$

Thus, the sung on which the posangi are installed, the given mechanism can be replaced by two fixed and mass systems. Therefore, the center of mass S of this system, therefore, the center of mass of the given mechanism and theposangi are fixed (Fig. 2, г, д). This means that static balancing of the given mechanism has been performed. It is necessary to determine from the equation that the posangi and their masses and values are given to the dimensions.

Thus, the method of substituting masses is as follows: each joint of the mechanism must be replaced by two cumulative masses, then inserted posangi (excitatory masses) and combine them with alternating masses, so that the center of these combined masses is finally fixed at the stationary points of the mechanism.

Dynamic extinguishers are used to reduce machine vibration and impact force on the platform. Consider the theory of dynamic quenching of vibration. Let a machine with mass m1 be mounted on springs on the platform. We define the sum of the stiffness coefficients of the springs as k_1, the mass of the dynamic damper of the vibration as m_2, and the stiffness coefficient of the springs installed in the machine space as k_1.

Our goal is to select the mass m_1 (Figure 3).

We write the equation of system oscillations.

$m_1\ddot{x}_1 + k_1x_1 + k_2(x_1-x_2) = F \sin wt$

$m_2\ddot{x}_2 + k_2(x_2-x_1) = 0$ (1)

Or:

$m_1\ddot{x}_1 + (k_1 + k_2)x_1- k_2x_2 = F \sin wt$

$m_2\ddot{x}_2 + k_2x_2 - k_1x_1 = 0$,

Where$F=mw^2r$– amplitude of forces acting on the body of the sewing machine by mechanisms, X_1 – m_1 mass shift, m_2– mass of unbalanced parts of the machine.

We determine the special solution of the system of equations from the form

$x_1=a\sin wt$,$x_2=b\sin wt$

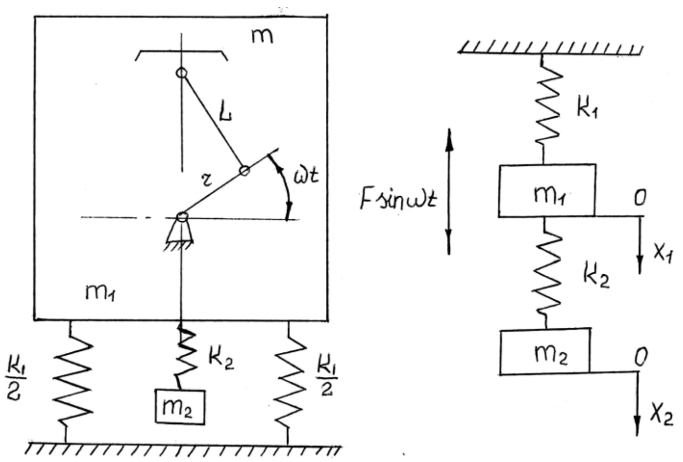

Figure- 3: Schematic of mass calculation of sewing machine links

Substituting these expressions into Equation (1), we obtain the following

$(k_1 + k_2 - m_1w^2)a - k_2b = F,$

$-k_2a + (k_2 - m_2w^2)b = 0$ (2)

(2) The system identifier is as follows.

$$D(w^2) = \begin{vmatrix} k_1 + k_2 - m_1w^2 & -k_2 \\ -k_2 & k_2 - m_2w^2 \end{vmatrix} = (k_1 + k_2 - m_1w^2)(k_2 - m_2w^2) - k_2^2$$

We determine the coefficients a and b:

$$a = \frac{1}{D(w^2)} \begin{vmatrix} F & -k_2 \\ 0 & k_2 - m_2w^2 \end{vmatrix} = \frac{F\left(k_2 - m_2w^2\right)}{D(w^2)}$$

$$b = \frac{1}{D(w^2)} \begin{vmatrix} k_1 + k_2 - m_1w^2 & F \\ -k_2 & 0 \end{vmatrix} = \frac{Fk_2}{D(w^2)}$$

It can be seen from the generated expressions that if $k_2 - m_2w^2 = 0$ then the $b = -\frac{F}{k_2}$ machine does not have any vibrations and the platform is affected only by the weight of the machine. The base of the machine is affected by the following force.

$w^2 = \frac{k_2}{m_2}$, $D(w^2) = -k_2^2$ also

$k_2x_2 = -F \sin wt$

A negative sign in the relationship means that the machine body is affected only by the force generated by the deformation of the spring. This force is opposite to the direction of force acting on *Fsinwt*, and its value is zero.

From this expression we find the coefficient of stiffness of the spring k_2

$$k_2 = \frac{F}{x_{2\max}}$$

Where: $x_2 = x_{2max}$is the maximum compression of the spring.

We find the mass of the vibration damper from the following equation.

$$m_2 = \frac{k_2}{w_2} = \frac{F}{w^2 x_{2\max}}$$

In general, dynamic extinguishers are used to reduce machine vibration when w - the operating frequency is approximately equal to the specific frequency of the "elastic support machine" system, that is:

$$P_1 = \sqrt{\frac{k_1}{m_1}}$$

Instead of $k_2 - m_2 w^2 = 0$ in the expression $w = \sqrt{\frac{k_2}{m_2}}$ can be written.

So, $\sqrt{\frac{k_1}{m_1}} = \sqrt{\frac{k_2}{m_2}}$

$D(w^2) = 0$ the frequency equation is as follows.

$m_1 m_2 w^4 - [(k_1 + k_2)m_2 + k_2 m_1]w^2 + k_1 k_2 = 0$

Dividing the equation by k1 and k2, we obtain

$$\frac{m_1 m_2}{k_1 k_2} w^4 - \left[\left(1 + \frac{k_2}{k_1}\right)\frac{m_2}{k_2} + \frac{m_1}{k_1}\right] w^2 + 1 = 0$$

Or: $c^4 - \left(2 + \frac{m_2}{m_1}\right)c^2 + 1 = 0$

There are: $c = \frac{w}{P_1} = \frac{w}{P_2}$

From this equation it is possible to determine the resonant frequencies depending on the parameter.

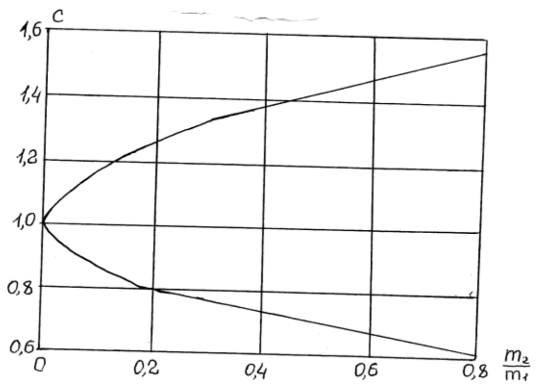

Figure-4: Graph of change of s with respect to m

The graph of the change of S with respect to mis shown in Figure 4

It is clear from the graph that the change in the mass m1 of the extinguisher should affect the resonant frequencies. When the ratio is small, the quenching effect is not significant, but the resonant frequencies are closer to the fundamental frequencies of the system [9,10].

Conclusion

As a result of the research, the law of motion of the elastic element elastic shock absorber at different speed regimes of the sewing machine was analyzed. A dynamic model of an elastic element flexible shock absorber mechanism was developed and the equation of motion of a dynamic system was derived. In order to suppress the vibrations caused by harmful forces in the links of sewing machines, calculations were carried out with different masses.

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