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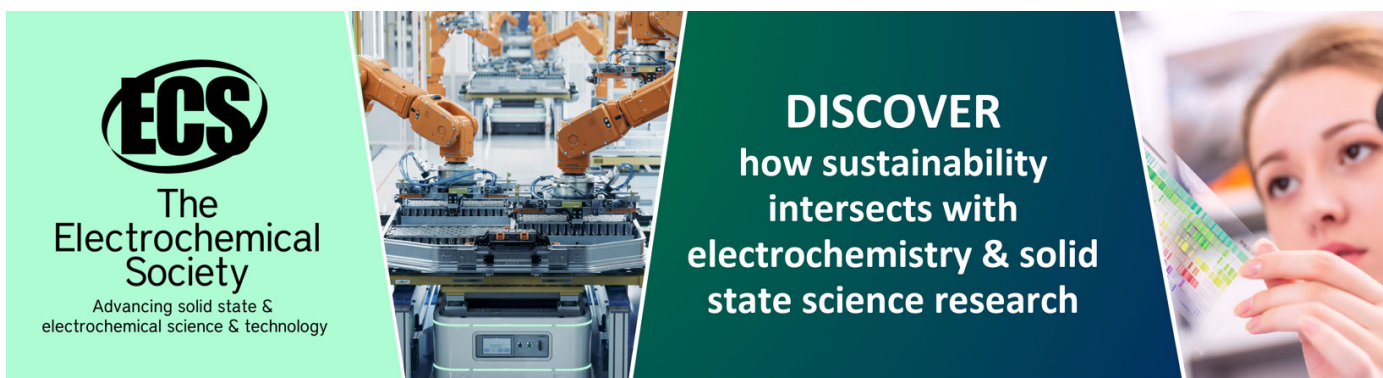
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The analysis of puncheon periodic vibrations of the ultrasonic sewing machines

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Abstract. The article deals with the research of periodic vibrations of the sewing machine's puncheon for ultrasonic welding of materials. The dynamic model of the puncheon drive mechanism is analysed, taking into account elastic-dissipative characteristics of springs that are disposed between a puncheon holder and a machine casing, a puncheon holder and an output link of leverage mechanism of the drive mechanism. Welding materials are represented as an elastic-dissipative element and a unilateral constraint element. With the use of the periodized harmonic method, the analytical solution of the nonlinear differential equation of the mathematical model of puncheon vibrations was found. The proposed analytic dependencies can be used for the research of the system's amplitude-frequency characteristics and the selection of design requirements of the driving mechanism, taking into account the standard conditions of an ultrasonic sewing machine.

1. Introduction

During the manufacture of sewn products, the technology and equipment of ultrasonic welding are widely used [1,2]. Ultrasonic welding helps to increase productivity and exclude the use of additional material (thread and hot-melt adhesive). There are two main types of equipment for ultrasonic welding – roller feed and rack feed. Roller feed machines are limited in use by linear or long radius stitches.

The utilize of rack feed ultrasonic welding machines (interruptive action machines) is limited by fabric. The interruptive action machines of ultrasonic welding include the puncheon drive mechanism [1,3]. The article [4] describes dynamic and mathematical models for the description of the puncheon's vibrations for welding fabric. Computer modeling was applied. This article is devoted to the search for an approximate analytical solution of the mathematical model for the analysis of a puncheon's periodic vibrations.

2. Simulation method

Figure 1 shows the schematic of a puncheon's drive [4]. The schematic shows the rotating motion of the crank 1 is converted into the crank motion of the slider 5. The slider 5 moves toward the puncheon's holder 8, which is firmly connected with the puncheon 12 and the stop 9. When the slider



5 contacts with the stop 9, the puncheon's holder 8 with the puncheon moves along the vertical axis. If the puncheon 12 contacts with fabric, the slider 5 moves toward the puncheon's holder 8. The spring 6 provides the necessary pressure of the puncheon's clamp 12 to fabric 13. In order to prevent impact between the slider 5 and the stop 9, the spring 11 is used. It is possible to regulate the pressure which is made by the puncheon in the fabric fixation during the welding. The regulation is made by the spring compression 6. The movement of the stop 9 which is fixed by the adjustment screw 10 toward the puncheon's holder 8 can regulate the time of contact between the puncheon and fabric during welding.

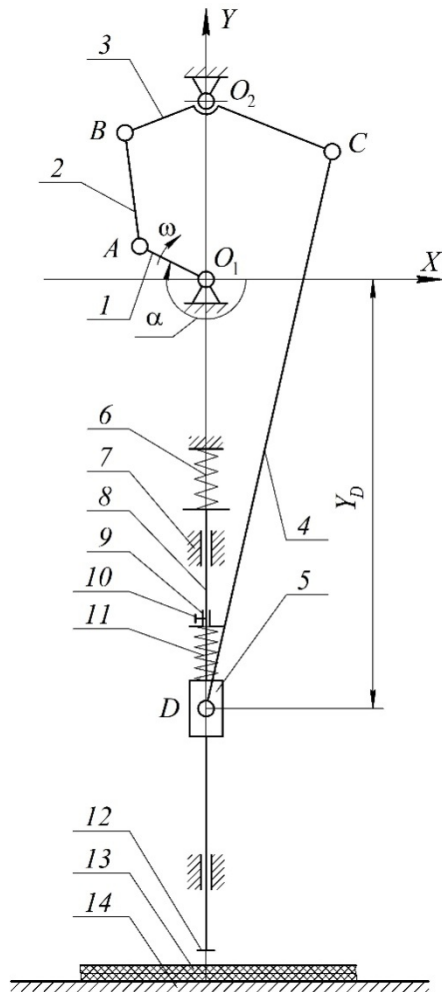


Figure 1. The schematic of the mechanism.

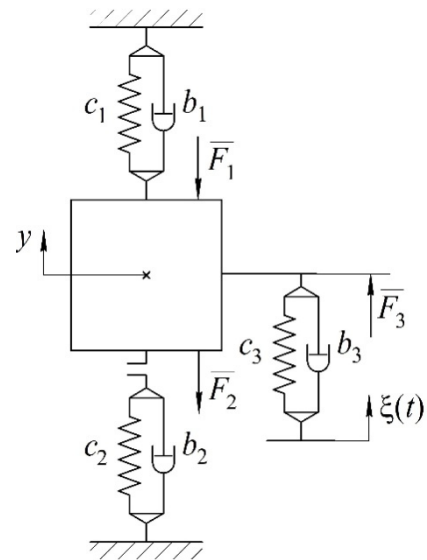


Figure 2. The dynamic model.

Figure 2 shows the dynamic model of the system [4] (puncheon 12, puncheon holder 8, the stop 9 are considered to be one perfectly rigid body with the mass m – “puncheon” which moves along the vertical axis; connecting spring 6, welding materials 13, spring 11 are demonstrated by the elastic-dissipative elements with the stiffness coefficient c_i and b_i resistance, $i = \overline{1,3}$ respectively; unilaterial constraints of the “puncheon” with fabric is shown in the “unilaterial constraint” element $\xi(t)$ - slider movement 5. The mathematical model of the system has the form:

$$m\ddot{y} = -F_1 - F_2 + F_3 - G, \quad (1)$$

$$\text{for } t = 0: y(0) = \xi(0), \quad \dot{y}(0) = \dot{\xi}(0),$$

where y – generalized coordinate, $y = \xi(t) - \Delta$, Δ – the deformation of the spring II ; F_1 – force from the side of the spring 6 ; F_2 – equally-effective distributed force from the welding materials as a result of deformation under “puncheon” by y (if $y < 0$ “puncheon” digs into the material, if $y \geq 0$ “puncheon” moves above the material); F_3 – the force of the spring II where the unit of deformation is the following $\Delta = \xi(t) - y$ during the relative motion of the slider 5 and the puncheon’s holder 8 ; G – “puncheon’s” gravity force ($G = mg$, m – “puncheon’s” mass).

The analytic forms of the forces are the following (1):

$$\begin{aligned} F_1 &\approx F_{10} + c_1 y + b_1 \dot{y}, \\ F_2 &= \begin{cases} F_2^* & \text{if } y < 0 \text{ and } F_2^* < 0, \\ 0 & \text{if } y \geq 0 \text{ or } F_2^* \geq 0, \end{cases} \\ F_3 &= F_{30} + c_3 \Delta + b_3 \dot{\Delta}, \end{aligned} \quad (2)$$

where F_{10} , F_{30} – forces which correspond to the initial preload of the springs 6 and II in installation position; $F_2^* \approx c_2 y + b_2 \dot{y}$.

3. Results

Differential expression (1), taking into account (2), is an absolutely non-linear expression. To analyze the periodic solution (1), we will use the method of harmonic periodization [5]. Kinematic external action $\xi(t)$ is shown as Fourier series, where we will apply the first harmonic:

$\xi(t) \approx \xi_{10} + \xi_{1c} \cos(\omega t) + \xi_{1s} \sin(\omega t)$. Based on the notation made, differential equation (1) can be rewritten as:

$$\ddot{y} + k^2 y + 2n\dot{y} + f_2(y, \dot{y}) = A_0 + A \sin(\omega t + \delta), \quad (3)$$

where $k^2 = (c_1 + c_3)/m$, $2n = (b_1 + b_3)/m$, $f_2(y, \dot{y}) = F_2(y, \dot{y})/m$, $A_0 = f + k_3^2 \xi_{10}$, $f = (-F_{10} + F_{30} - mg)/m$, $k_3^2 = c_3/m$, $A = \xi_{1c} \sqrt{k_3^4 + 4n^2 \omega^2}$, $\xi_{1c} = \sqrt{\xi_{1c}^2 + \xi_{1s}^2}$, $2n_3 = b_3/m$, $\tan \delta = (k_3^2 \xi_{1c} + 2n_3 \xi_{1s} \omega) / (k_3^2 \xi_{1s} - 2n_3 \xi_{1c} \omega)$.

The solution by the method of harmonic linearization of equation (3), and the non-linear function $f_2(y, \dot{y})$ can be presented as following:

$$\begin{aligned} y &= a_0 + a \sin \omega t, \\ f_2(y, \dot{y}) &\approx f_0 + q(y - a_0) + r\dot{y}, \end{aligned} \quad (4)$$

where $f_0(a_0, a)$, $q(a_0, a)$ and $r(a\omega)$ – coefficients of harmonic linearization [5].

Taking into account (4) on the basis of equation (3) we get the equation to trace the resonance curve:

$$k^2 a_0 + f_0 = A_0, \quad (5)$$

$$|a| = \frac{\xi_1 \sqrt{k_3^4 + 4n_3^2 \omega^2}}{\sqrt{(-\omega^2 + k^2 + q)^2 + (2n + r)^2 \omega^2}}.$$

According to (5) the equation of backbone curve and the maximum curve can be presented as follows:

$$\begin{aligned} -\omega^2 + k^2 + q &= 0, \\ |a_{\max}| &= \frac{\xi_1 \sqrt{k_3^4 + 4n_3^2 \omega^2}}{(2n + r)\omega}. \end{aligned} \quad (6)$$

4. Conclusion

To trace the resonance curve, the backbone curve, and the maximum curve, we need coefficients of harmonic linearization, which depend on [5] the undetermined values a_0 and a . Thus, the equations (5) and (6) should be considered as simultaneous equations, where the solution is possible with applying the numerical approach [6]. Based on the equations, we can analyze the influence of dynamic model characteristics on the resonance curve. It can help to choose elastic-dissipative characteristics of the connecting spring and make the conclusion about the influence of kinematic external action on the stabilization of puncheon's clamping force on the fabric during welding.

References

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