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THE POSSIBILITY OF USING STRUCTURAL DAMPING IN THE DESIGN OF A PREFABRICATED TURNING CUTTER TO REDUCE THE AMPLITUDE OF SELF-OSCILLATIONS IN THE PROCESS OF METAL CUTTING

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Abstract. A mathematical model of four mass oscillating circuit cutting machine was developed and studied in this paper. The influence of the internal friction in the joints of structural elements on the machine tool vibration amplitude and details. Effective self-oscillation amplitude blanking tool is possible by choosing the optimal parameters of the tool body frictional connection with tool holders.

Key words: self-oscillation; structural damping; surface roughness details.

Introduction

There are many ways to improve machining productivity and the quality of machined parts surfaces in machine tool and instrumentation engineering. However, the most promising way today is using dynamic processes in elastic closed technological systems. This primarily applies to self-oscillatory processes in metalworking machines. There are many ways to reduce the intensity of oscillations that are used in practice, but they can be divided into two types. The first type is aimed at increasing the resistance of the system, and the second is aimed at reducing the forces that excite oscillations. The first type includes: increasing the rigidity of the machine; damping the energy of oscillations and the use of special tools. The second type includes stabilization of the cutting force; optimization of cutting modes and the use of automatic and adaptive systems.

Problem statement

As known, oscillations arising during the cutting process negatively impact both the surface quality of the workpiece, as well as on the stability of the tool and the resource of the machine. These oscillations are divided into forced oscillations and self-oscillations. Forced oscillations are induced by the inconsistent nature of cutting forces (tearing on the machined surface), centrifugal forces (unbalanced workpieces), and periodic forces transmitted both from the machine's drive and other machines. The amplitude of these oscillations can be easily reduced by moving away from resonance (changing the spindle rotation frequency) and by isolating the machines from vibrations. A more challenging issue is reducing the amplitude of self-oscillations, the main reason for which is the ambiguity of the cutting force characteristics.

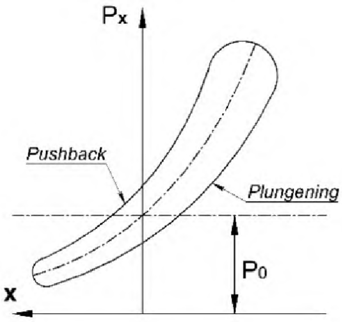

Fig. 1. Experimental dependency $P_x=f(x)$

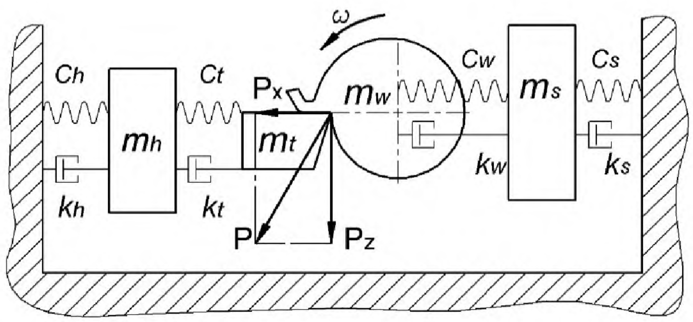

Fig. 2. Oscillatory diagram of the machine

This ambiguity leads to the formation of a hysteresis loop (Fig. 1), where the same increase in the thickness of the cut *y* corresponds to different increases in the cutting force P_x when the cutter is plunging and pushing away. When pushing back, the cutter comes into contact with a more hardened (riveted) metal than when plunging, so when pushing back, the cutting force will be greater than when plunging, with the same cut thickness value. The more sensitive the workpiece material is to the hardening, the greater the force that generates self-oscillations. And these oscillations will occur at the natural frequency of the oscillating system, i. e. in resonance.

Review of modern information sources on the subject of the paper

The Existing methods for reducing the amplitude of self-oscillations can be divided into two classes: technological and structural. Technological methods include the selection of appropriate cutting modes and tool sharpening angles [2; 4; 5]. Structural methods include increasing the resistance in the oscillating system and the use of dynamic vibration dampers [3; 6], and given the fact that self-oscillations occur in resonance, where the influence of damping is of great importance, increasing the resistance in the oscillating system is not only a significant factor in reducing the amplitude of self-oscillations but also a factor in the possibility of their occurrence in general, because if the frictional energy is greater than the excitation energy, then self-oscillations will not occur at all [1].

Objectives and problems of research

The purpose of the research is to assess the possibility of using internal friction in the joints of the elements of the “machine-fixture-tool-workpiece” (MFTW) system - structural damping to reduce the amplitude of self-oscillations, as well as to identify the most sensitive link of the MDTW system for reducing self-oscillations due to this damping.

Main Material Presentation

To study the self-oscillations in the MFTW system, we will draw up a system of differential equations for a four-mass oscillatory scheme consisting (Fig. 2) of a tool holder, cutter, workpiece, and spindle, which are connected and the machine bed by elastic links with dampers, which corresponds to the scheme of a classical lathe, the mass of the bed of which is much greater than the mass of its components, so their oscillations will be considered relative to a conditionally fixed bed. The four-mass oscillatory scheme is characterized by its versatility, since if we replace the spindle mass with the table mass and the tool holder mass with the spindle mass, it will already be a milling machine scheme, and the oscillatory scheme of the machining center can be described similarly.

To simplify, let's consider the oscillatory scheme in just one coordinate, namely – the coordinate X:

$$\begin{cases} \frac{d^2x_h}{dt^2}\cdot m_h - C_h x_h - k_h\cdot\frac{dx_h}{dt} - C_t(x_h - x_t) - k_t\cdot\left(\frac{dx_h}{dt}-\frac{dx_t}{dt}\right)=0 \\ \frac{d^2x_t}{dt^2}\cdot m_t + C_t(x_h - x_t) + k_t\cdot\left(\frac{dx_h}{dt}-\frac{dx_t}{dt}\right)+P_x=0 \\ \frac{d^2x_w}{dt^2}\cdot m_w - P_x - C_w(x_w - x_s) - k_w\cdot\left(\frac{dx_w}{dt}-\frac{dx_s}{dt}\right)=0 \\ \frac{d^2x_s}{dt^2}\cdot m_s + C_w(x_w - x_s) + k_w\cdot\left(\frac{dx_w}{dt}-\frac{dx_s}{dt}\right) - C_s x_s - k_s\cdot\frac{dx_s}{dt}=0 \end{cases}, \quad (1)$$

where x_i – the movement of the i-th element of the diagram (tool holder, cutter, workpiece, and spindle); m_i – the given mass of the i-th element; c_i – the stiffness of the i-th element; P_x – the horizontal component of the cutting force; k_i – the attenuation coefficient of the i-th element of the diagram

$$k_i = \frac{m_i\delta_i\omega}{\pi}, \quad (2)$$

where δ_i – the logarithmic decrement of the oscillations of the i-th element of the oscillatory scheme, which characterizes the rate of attenuation of the oscillatory process; ω – the angular frequency of oscillations.

The horizontal component of the cutting force according to the theory of A. P. Sokolovskoho [5] is represented as follows:

$$P_x = P_0 - ry + a_1 b\frac{\dot{x}}{V} + a_2 b\left(\frac{\dot{x}}{V}\right)^2 - a_3 b\left(\frac{\dot{x}}{V}\right)^3, \quad (3)$$

where P_0 – the value of the cutting force in the absence of vibrations (Fig. 1), x – the relative displacement between the cutter and the workpiece, r – the cutting stiffness coefficient, $r = kb$, b – the depth of cut, k – the specific cutting force (k = 2000 MPa), V – the cutting speed. The cutting constants a_1, a_2, a_3 are determined by the least squares method according to the experimental diagram (Fig. 1).

For the case of our oscillatory scheme (Fig. 2), Eq. (3) will take the form:

$$P_x = -r(x_t - x_w) + a_1 b\frac{\left(\frac{dx_t}{dt}-\frac{dx_w}{dt}\right)}{V} + a_2 b\left(\frac{\left(\frac{dx_t}{dt}-\frac{dx_w}{dt}\right)}{V}\right)^2 - a_3 b\left(\frac{\left(\frac{dx_t}{dt}-\frac{dx_w}{dt}\right)}{V}\right)^3$$

The constant component of the cutting force P_0 will be reduced when the system of differential equations (1) is solved, i. e., we will get oscillations relative to the constant component.

For the case of relatively negligible viscous resistance, having $\frac{k_i}{2m_i} < \omega$, the logarithmic decrement of oscillations is defined as the natural logarithm of the ratio of two adjacent amplitudes of damped oscillations:

$$\delta_i = ln\frac{A_n}{A_{n+1}} = \frac{k_i}{2m_i}T = \frac{\pi k_i}{m_i\omega}, \quad (5)$$

where T – the period of oscillation; $T = 2\pi/\omega$.

Lets find the solutions of the system of differential equations (1) for the values m_t = 0.1 kg; m_w = 1 kg; m_h = m_s = 25 kg; $c_t = 2\times10^9$ N/m; $c_w = 1\times10^7$ N/m; $c_s = c_h = 1.5\times10^8$ H/m. The attenuation coefficient is determined from the known values of the logarithmic decrements of oscillations $\delta_w = \delta_t = 0.04$ (for steel 45); $\delta_h = \delta_s = 0.15$ (taking into account the existing structural damping in the joints of the lathe elements [3]), then according to Eq. (2) k_t = 180 Ns/m; k_w = 1800 Ns/m; $k_h = k_s = 1.7\times10^5$ Ns/m.

Since we are studying primary self-oscillations (the absence of traces of self-oscillations on the surface of the part from the previous pass), an excitation must be introduced into the system to cause sel-f-oscillations. Such an excitation can be the presence of a small cavity (pore) in the material of the part or a solid inclusion of the same size. Let us assume, as an excitation, the deviation of the part in the initial conditions by a value of 1 micron [3].

It should be noted beforehand, and this is confirmed by theoretical and experimental studies [6], that if the system is stable, the excitation will lead to the damping of oscillations, but if it is unstable, self-oscillations will be generated in the system. The stability of an oscillating system depends on its parameters, workpiece material, and cutting modes. Since we are investigating the effect of structural damping on the amplitude of self-oscillations, we will select the system parameters and cutting modes in such a way that it is not stable.

The solutions of the system of differential equations (1) are shown in Fig. 3. In Fig. 3, *a* – shows the movement of the tool holder in time, Fig. 3, *b* – the movement of the cutter, Fig. 3, *c* – the movement of the workpiece, and in Fig. 3, *d* – the spindle

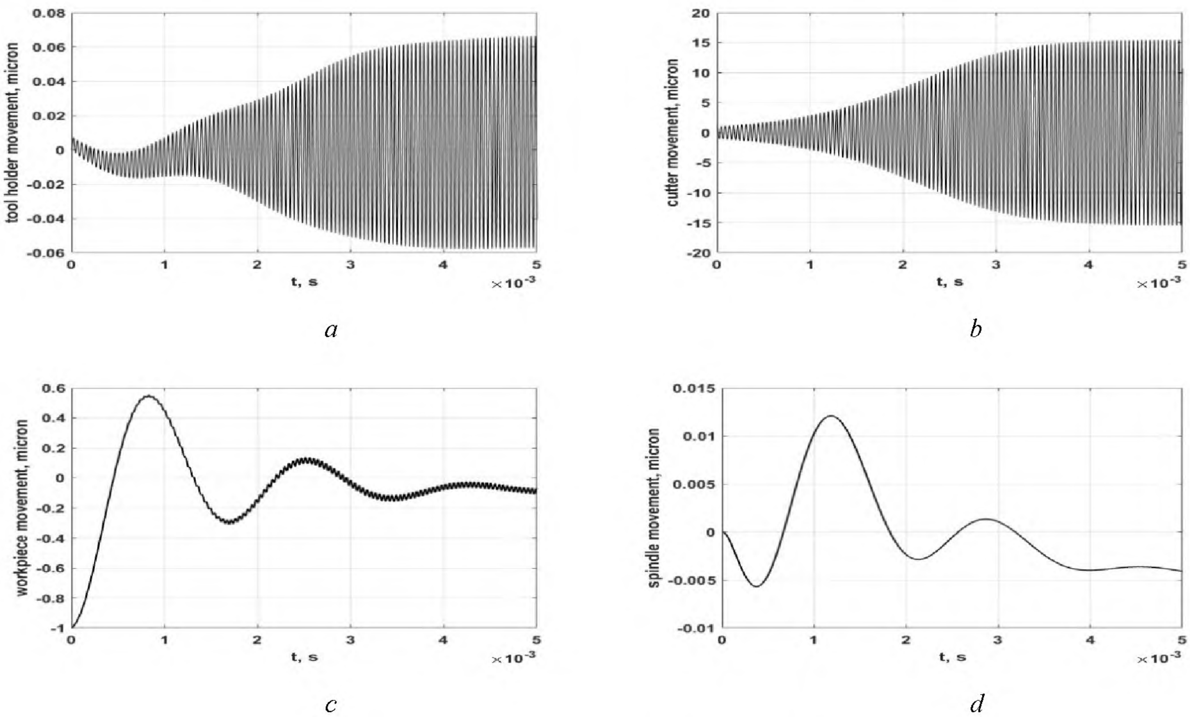

Fig. 3. Self-oscillations of machine tools and workpiece

As you can see from the figures, at time $t = 0$, the coordinates (displacements) of the tool holder, cutter, and spindle are zero, and the displacement of the workpiece is 1 micron. This is the excitation of the system. And if the system was stable, then the oscillations of its elements (masses) caused by this excitation would decay to zero. Since the system is unstable, self-oscillations are generated in it. The vibration amplitude of the tool holder is about 0.06 microns, and that of the cutter is 15 microns. Oscillations of the part and spindle are even smaller. Some asymmetry of oscillations is caused by the asymmetrical nature of the curve in Fig. 1.

From the analysis of Fig. 3, the oscillation frequency can be determined. The period $T = 4.405 \times 10^{-5}$ s. Then the oscillation frequency f is 22.7 kHz, and the angular frequency ω is 142628 rad/s, which approximately corresponds to the highest oscillation frequency of the four-mass oscillatory system with the parameters used to study the system.

Fig. 4 shows the graphical dependence of the surface roughness of a workpiece under mutual vibrations of machine elements depending on the logarithmic decrement of vibrations of its elements, namely $\delta_h = \delta_s = 0.15$, and δ_w varied in the range from 0.04 to 0.1 (dashed-dotted curve) and $\delta_t = 0.04 \div 0.1$ (solid curve).

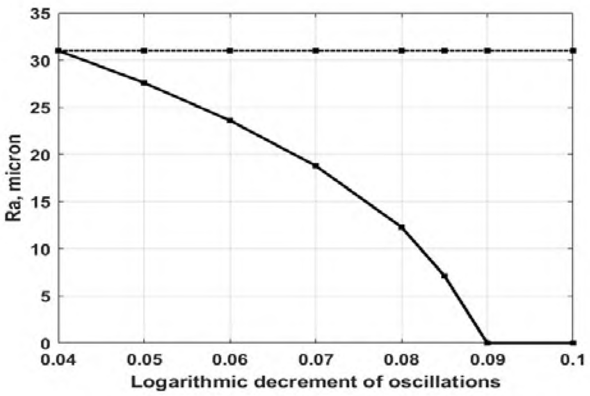

Fig. 4. Calculated dependence of the surface roughness of a part on the logarithmic decrement of oscillations

As you can see from the figure, it is not advisable to introduce structural damping at the point where the part is mounted in the spindle (or on the table). Instead, structural damping between the cutter and the tool holder, for example, in a tool holder or in the body of the cutter (tool), is an effective means of combating self-oscillations.

Energy dissipation in the process of damped oscillations can be characterized by the absorption coefficient, which is equal to the ratio

$$\psi = \frac{\Psi_n}{\Pi_n} = \frac{A_n^2 - A_{n+1}^2}{A_n^2} \approx \frac{k}{m} T \approx 2\delta, \quad (6)$$

where Ψ_n – the energy dissipated during the nth cycle of oscillations, and Π_n – the potential energy at the beginning of the same cycle.

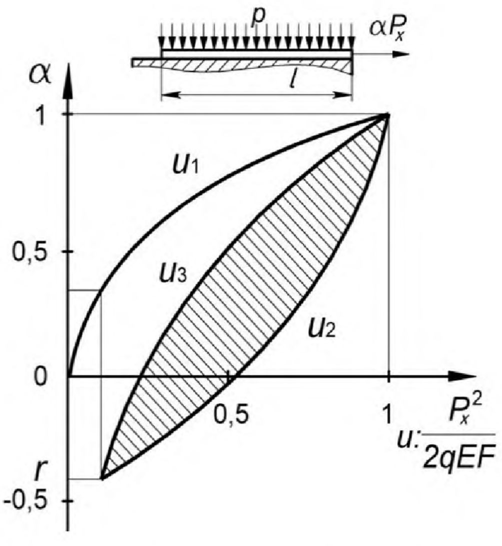

Fig. 5. Hysteresis loop Ψ

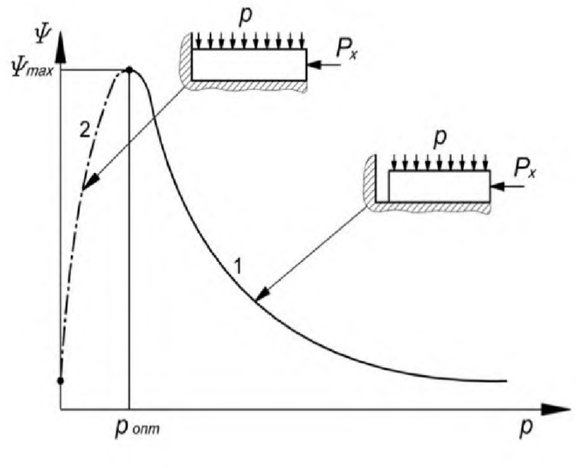

Fig. 6. Graph of the dependence $\Psi = f(p)$

In the case of cyclic loading of a thin plate with a force αP_x, pressed by a uniform pressure p against a rigid base of length l (Fig. 5), the plate will not shift relative to the base until the condition

$$P_x < fpbl, \quad (7)$$

where f – the friction coefficient, b – the width of the plate, α – a dimensionless load parameter. At the maximum load, α = 1, and at the minimum load, α = r, where $r = Px\ max_{x\ min}$ – the characteristic of the cycle asymmetry. For our case, $P_{0p_{xmax}}$;$P_{0p_{xmin}}$;where P_0 – the cutting force P_x during steady-state cutting (without vibrations, Fig. 1), A_p – the amplitude of oscillations of the cutter (Fig. 3, b).

Fig. 5 shows the process of forming a hysteresis loop between the plate and the base, the area of which is the energy dissipated during the oscillation cycle. The relative displacement of the plate end from 0 to 1 is plotted along the ordinate axis. In this case, the complete displacement of the plate relative to the base is impossible due to the fulfillment of condition (7), i. e., the displacement will occur only at a certain length of the plate, and the larger this displacement length is, the larger the area of the hysteresis loop will be (Fig. 5), and hence the energy dissipated. The value of this energy can be calculated by the formula.

$$\Psi = \frac{P_y^3(1-r)^3}{12qEF}, \qquad (8)$$

where $q = fpb$ – the limiting value of the friction force (per unit length of the plate or cutter clamped in the tool holder); E – the elastic modulus of the material of the plate (cutter body); F – the cross-sectional area of the plate (cutter body).

From the analysis of Fig. 5 and eq. (8), it can be concluded that the clamping pressure of the plate affects the dissipation energy in the joint. The lower the pressure, the greater the scattering energy, but when the pressure is reduced, condition (7) must be met. It is this graphical dependence that is shown in Fig. 6 as a solid curve 1. To prevent the displacement of parts of the prefabricated cutter when the maximum value of the dissipation energy is reached, the moving part of the cutter should be rested against a fixed support. This situation is shown by the dashed curve in Fig. 6. So, as can be seen from Fig. 6, there is an optimal value of pressure on the cutter, which corresponds to the maximum value of the dissipation energy.

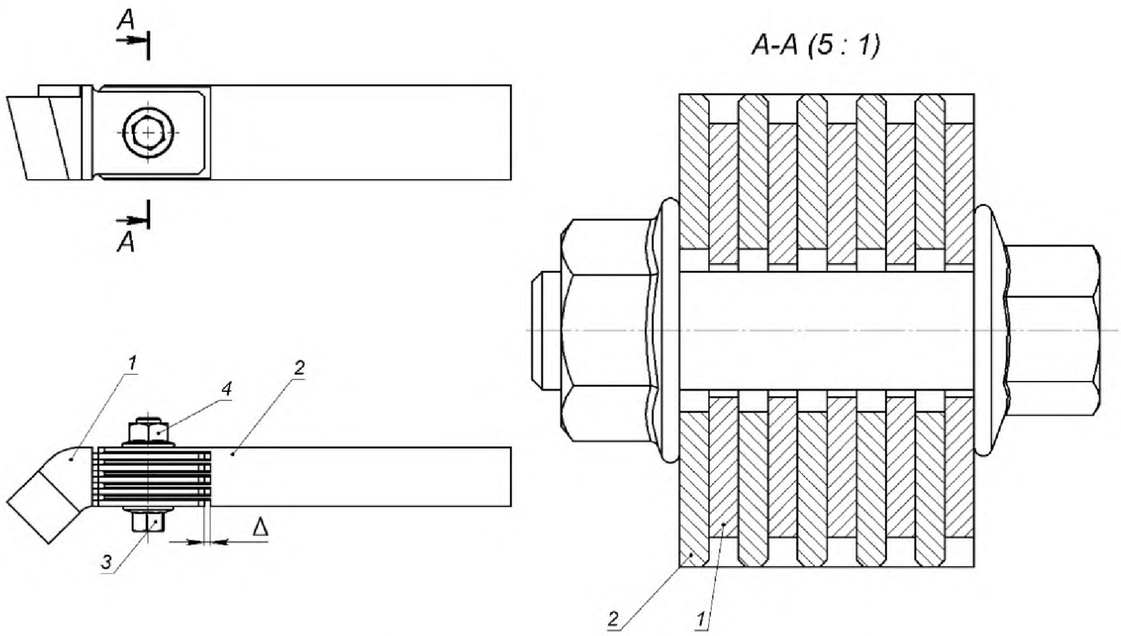

Fig. 7. Constructional diagram of a damper cutter

The structural diagram of the damper cutter is shown in Fig. 7. The cutter holder of such a cutter is made of two separate parts - conditionally movable (1) and fixed (2), the comb parts of which are compressed by bolt 3 and nut 4. At relatively low loads (for example, during finishing of a workpiece), a gap Δ is possible between the conditionally movable (1) and fixed (2) parts of the toolholder, which corresponds to line 1 in Fig. 6. At high cutting force values, this gap may be absent (curve 2, Fig. 6). The value of the friction force in the toolholder is regulated by tightening bolt 3.

Conclusions

Dissipation of oscillation energy in the tool mounting point or in the tool itself can prevent self-oscillations during metal cutting or significantly reduce their amplitude.

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