Optimization of Damping Parameters of Self-Oscillations of Multifrequency Oscillatory System

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***Abstract* – Developed and researched mathematical model four-mass scheme autoclave machine tools. Influence of parameters of internal friction is analysed in the joints of structural elements machine-tool on relative amplitude of vibrations the cutter and detail. The ambiguousness of influence of parameters of damping energy of vibrations is shown on relative amplitude of the vibrations a cutter and detail.**

Keywords – self-oscillation, structural damping oscillations amplitude, frequency oscillations, the quality of the workpiece surface, the stability instrument.

I. Introduction

The existent methods of reduction of amplitude of self-excited oscillations can be broken up on two classes – technological and construction. To the technological methods it follows to take such as a choice of the corresponding modes of cutting and angle of sharpening of instrument [5]. To the construction is an increase of resistance in the oscillating system and use of dynamic vibration dampers [4], and taking into account a that circumstance, that self-excited oscillations take place in resonance, where influence of fading plays a large value, increase of resistance the oscillating system is not only a ponderable factor in reduction of amplitude of self-excited oscillations but also factor of possibility of their uprise in general, in fact if energy of friction will be anymore from energy of excitation, then auto-oscillation in general not will not arise up.

II. Main Material Presentation

For research the self-oscillations in the MDTD system, we will compile a system of differential equations of the four-mass oscillation scheme, which consists of (fig. 1) from the support, the cutter, the part and the spindle, which are interconnected with the base of the machine by elastic bonds with the damps corresponding to the classical scheme lathe, mass frame which is much greater than the mass of its individual components, so they consider fluctuations relative to conventionally fixed frame. Four-mass oscillatory scheme is characterized by its versatility, since if you replace the mass of the spindle with the weight of the table, and the mass of the support on the mass of the spindle, then this will be the scheme of the milling machine, similarly, you can describe the vibrational scheme of the machined centre.

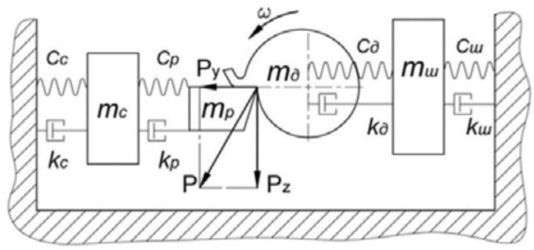


Fig. 1. Four-mass oscillation scheme of the machine tool

For simplify, consider the oscillation scheme in only one coordinate, namely – Y:

$$\frac{d^2 y_p}{dt^2} m_p + c_p(y_c - y_p) + k_p\left(\frac{dy_c}{dt} - \frac{dy_p}{dt}\right) + P_y = 0 \quad (1)$$

$$\frac{d^2 y_\partial}{dt^2} m_\partial - P_y - c_\partial(y_\partial - y_ш) - k_\partial\left(\frac{dy_\partial}{dt} - \frac{dy_ш}{dt}\right) = 0$$

$$\frac{d^2 y_ш}{dt^2} m_ш + c_\partial(y_\partial - y_ш) + k_\partial\left(\frac{dy_\partial}{dt} - \frac{dy_ш}{dt}\right) -$$

$$-c_ш y_ш - k_ш \frac{dy_ш}{dt} = 0$$

where y_i – the movement of the i-th element of the scheme (a support, a cutter, a part and a spindle); m_i – the mass of the *i*-th element; c_i – stiffness of the i-th element; P_y – horizontal component of cutting force; k_i – coefficient fading of the i-th element of the scheme.

$$k_i = \frac{m_i d_i w}{p}, \quad (2)$$

where δ_i – logarithmic decrement of oscillations of the i-th element of the oscillation scheme, which characterizes the temperature of the fading oscillation process; ω – angular frequency oscillation.

The horizontal component of the cutting forces on the theory Sokolovsky A. [5] is represented as follows:

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

where P_0 – value of the cutting force in the absence of vibrations, y – relative displacement between the cutter and the detail, r – coefficient of rigidity of cutting, $r = kb$, b – cutting depth, k – specific force of cutting (k = 2000 MPa), V – cutting speed Constant cutting a_1, a_2, a_3 – are determined by the least squares method by the experimental diagram.

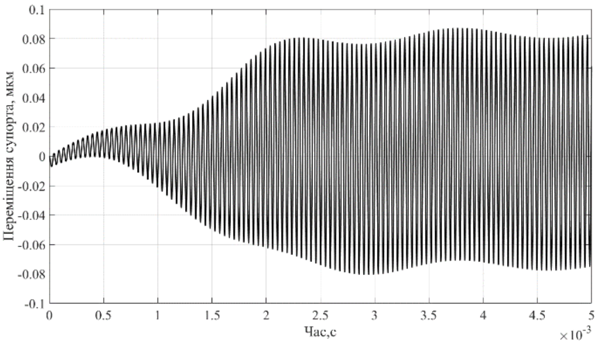
For the case of our oscillation scheme (Fig. 1), f. (3) will take the form of:

Constant component of cutting force P_0 will decrease in the compilation of the system of differential equations (1), that is, we obtain oscillations relative to the constant component.

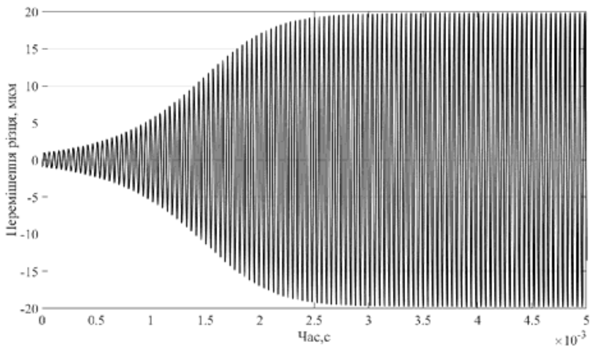
For the case of not very significant viscous resistance, when $k_i / (2m_i) < w$, the logarithmic decrement of oscillations is defined as the natural logarithm of the ratio of two neighbouring amplitudes of fading oscillations [3]:

$$d_i = \ln\frac{A_n}{A_{n+1}} = \frac{k_i}{2m_i}T = \frac{p\,k_i}{m_i w}, \qquad (4)$$

where T – period of oscillations; T=2π/ω.



a



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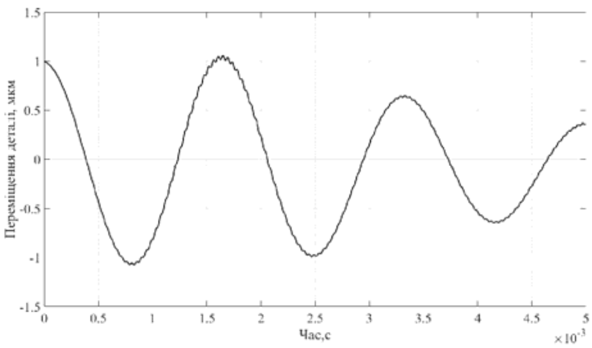

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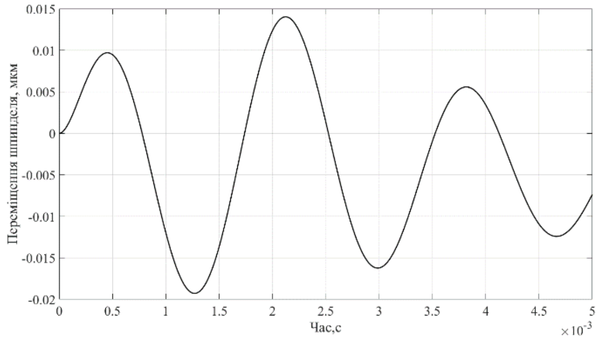

d

Fig. 2. Self-oscillation of working parts of the machine and detail with low damping

We find numerical solutions of the system of differential equations (1) for values m_p = 0,1 kg; $m_д$ = 1 kg; m_c = =$m_ш$ = 25 kg; c_p = 2×10^9 N/m; $c_д$ = 1×10^7 N/m; $c_ш$ = c_c = =1,5×10^8 N/m and we will build their time graphic dependencies. Coefficient the fading will be determined from the known values of the logarithmic decrements of oscillations: δ_p = 0,01 (for steel 45); $\delta_д$ = 0,005 (steel 14X17H2) δ_c = $\delta_ш$ = 0,15 (taking into account the existing structural damping elements in compounds lathe [4]).

Solutions of the system of differential equations (1) are presented in Fig. 2. In Fig. 2 (a), present the transfer of support in time, in Fig. 2, b – moving the cutter, in Fig. 2, g – moving of the part, and in Fig. 2, d – spindle.

As can be seen from the drawings, at the time t = 0, the coordinates (movement) of the support, the cutter and the spindle are equal to zero, and the displacement of the part is equal to 1 microns. This is the excitation of the system. And if the system was stable, then the oscillations of its elements (masses) caused by this excitement would have extinguished to zero. Since the system is unstable, then it generate self-oscillation. The oscillation amplitude of the support is less than 0.1 microns, the cutter is 20 microns. Oscillation of the part and spindle – even smaller. The relative amplitude of the oscillations of the cutter and the part is determined mainly by the oscillations of the cutter and is 20 microns. From the analysis of rice. 2 it is possible to set the frequency of oscillations. Period T = 4.405 × 10-5 s. Then the oscillation frequency f is equal to 22.7 kHz, and the angular frequency ω is 142628 rad/s, which approximately corresponds to the highest frequency of oscillations of the quadratic oscillation system with the parameters adopted for the study of the system.

Changing the material of the tool bit holder (steel 45, δ_p = 0,01) on gray cast iron GCI 20 (δp = 0,05) we get a slightly lower (30%) amplitude fluctuations details, which will improve the quality of the workpiece surface [1] and the stability instrument.

Conclusion

Use of the tool bit holder from alloys of increased damping leads to a decrease in the high-frequency component of the amplitude of self-oscillations system MDTD. However, for some sub-optimal parameters of the system, this can lead to low-frequency self-oscillations with significantly higher amplitudes, the elimination of which is an extremely difficult task.

References

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