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DETERMINING THE IMPACT OF STRUCTURAL FRICTION PARAMETERS IN THE BODY OF AN ASSEMBLED CUTTER ON THE DAMPING OF ITS AMPLITUDE AND SELF-OSCILLATIONS

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Abstract. The paper examines the potential use of dry friction in designing metal-cutting tools to dampen vibration energy. A mathematical model of the self-oscillating system "machine-fixture-tool-workpiece" has been developed and analyzed, considering structural friction within the assembly of the cutting tool. The study has investigated structural friction's impact on the cutter body's self-oscillation amplitude at the cutter tip and the workpiece surface. The paper demonstrates the possibility of effectively reducing the amplitude of self-oscillations in the system through the optimal selection of tool parameters.

Keywords: self-oscillations, structural damping, amplitude of self-oscillations, frequency of self-oscillations, assembled cutter, cutter stiffness, workpiece surface quality, tool stability.

Introduction

As is well known, self-oscillations negatively affect the surface quality of the workpiece, tool stability, and machine tool life [1;2]. Therefore, it is necessary to reduce the amplitude of these oscillations whenever possible.

These vibrations are classified as either forced or self-excited (auto-oscillations). Forced vibrations are caused by inconsistencies in cutting forces (disruptions on the machined surface), centrifugal forces (unbalanced workpieces), and periodic forces transmitted from the machine drive and other machines. The amplitude of these vibrations can be easily reduced by avoiding resonance (changing the spindle speed) and isolating machines from vibrations.

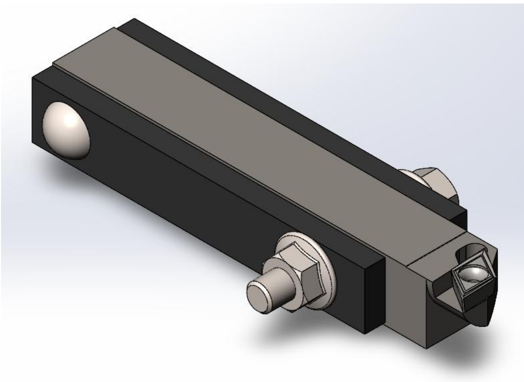
Reducing the amplitude of self-excited vibrations is a more complex issue, as they primarily arise due to the ambiguity of cutting force characteristics. Such vibrations occur when cutting certain difficult-to-machine materials (corrosion-resistant and heat-resistant steels and alloys) that tend to work-harden during cutting. They occur directly at the resonance frequencies of the technological system "machine-fixture-tool-workpiece," significantly deteriorating the machining process due to the high vibration amplitudes.

Problem Statement

This work evaluates the possibility of effectively using structural friction inside the holder of an assembled turning tool for passive damping of self-oscillation energy.

The design of the assembled cutting tool, in which this principle of vibration energy dissipation can be implemented, is shown in Fig. 1a. As seen in the figure, two metal plates measuring 65×11×4 mm are attached to the tool holder, which have dimensions of 80×12×12 mm. These plates are connected at the rear of the holder with a rivet and at the front with a bolt and nut. The M5 bolt is installed with clearance in a hole with a diameter of 6 mm. The riveted joint at the rear of the holder does not participate in the energy-damping process but serves only as a stop. The friction force between the plates and the holder is adjusted by tightening the bolt.

The cutting force applied to the tool tip during the cutting process leads to complex deformation of the tool holder (compression, bending, and twisting), resulting in potential complex relative movement of the holder's surfaces and the plates pressed against them by the bolt. The friction force between the plates and the holder is directed opposite to the movement of the holder, and the work of this friction force represents the energy used for damping possible vibrations of the tooltip.



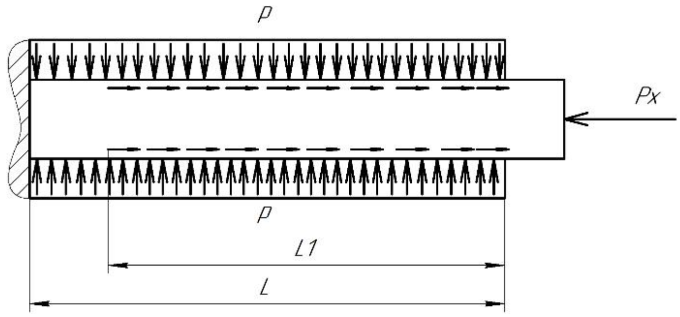


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Fig. 1. Method of damping the vibration amplitude of the tooltip
(a – structural diagram of the tool, b – an analytical model of the vibration energy damper along the X-axis).

Review of Modern Information Sources on the Subject of the Paper

Existing methods for reducing self-oscillation amplitude can be divided into technological and structural. Technological methods include the selection of appropriate cutting modes and tool sharpening angles [2; 6; 7]. Structural methods include increasing the resistance in the oscillating system and the use of dynamic vibration dampers [3; 4; 5; 9], 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, so self-oscillations will not occur [1; 2].

Objectives and Problems of Research

This research aims to assess the feasibility of using internal friction in the joints of the "machine-fixture-tool-workpiece" (MFTW) system - structural damping - to reduce self-oscillation amplitude. Additionally, it aims to determine the necessary parameters of a structural damper for effectively damping the vibration amplitude of the tooltip.

Main Material Presentation

As seen in Fig. 1b, the projection of the cutting force onto the X-axis primarily leads to compression of the tool holder, while other projections of the cutting force result in different deformations. The various directional deformations of the tool holder have different effects on both vibration energy damping and the formation of the geometric parameters of the machined surface.

The most significant influence on the surface parameters of the workpiece comes from tool vibrations along the X-axis (longitudinal vibrations). In contrast, tool vibrations along the Y and Z axes (transverse vibrations) have a considerably smaller impact.

Fig. 2a illustrates forming a hysteresis loop between the plate, which deforms along the X-axis, and the base to which it is pressed by contact pressure *p*. The area of this loop represents the energy dissipated during a vibration cycle. The ordinate axis plots the relative displacement of the plate's end from 0 to 1.

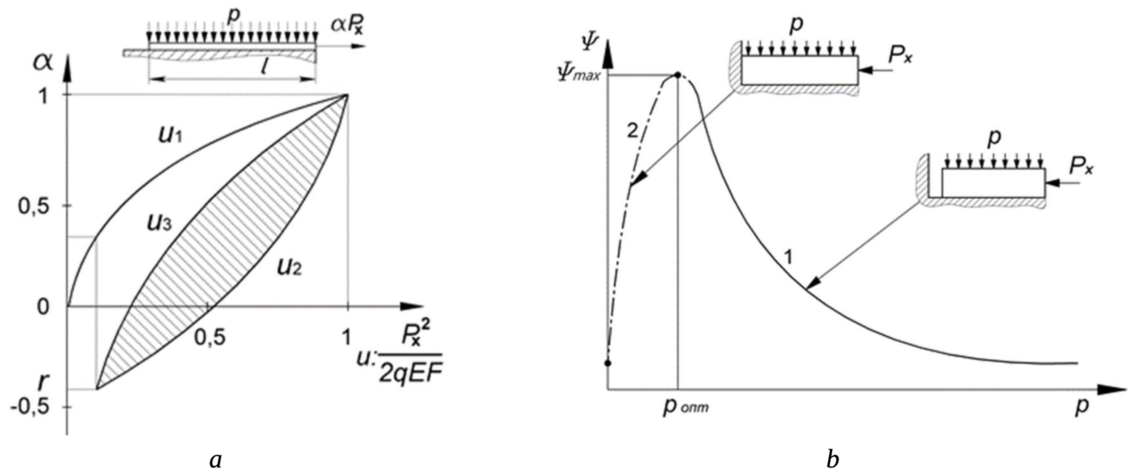


Fig. 2. Vibration Energy Dissipation per Cycle (a – Hysteresis loop, b – Dissipated energy)

Fig. 2b shows the graphical dependence of energy dissipation during the oscillation cycle on contact pressure. As the figure shows, the optimum pressure lies at the boundary of full plate shear relative to the base. In our case, this is impossible, as the rear parts of the plates are riveted to the shank, as shown by the dashed-dotted line.

As can be seen from the analysis of this graphical dependency, there is an optimal contact pressure value relative to the shear force of the plate, and any deviation from this value results in reduced energy dissipation of the oscillations.

To solve the system of differential equations describing the four-mass system of the metal-cutting machine tool [5], it is necessary to determine the value of the damping coefficient of the cutter's oscillations as the ***i***-th element of the overall system of equations:

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

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.

$$k_i = \frac{m_i\delta_i\omega}{\pi}, \tag{2}$$

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

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

$$\psi = \frac{\Psi_n}{\Pi_n} = \frac{A_n^2 - A_{n+1}^2}{A_n^2} \approx \frac{k}{m} T \approx 2\delta, \qquad (3)$$

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

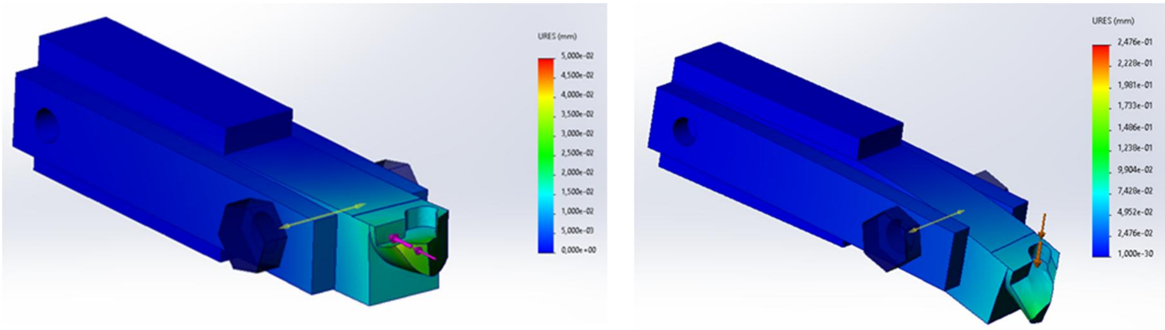
$$\Psi_n = \frac{P_x^3 (1-r)^3}{12qEF}, \qquad (4)$$

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), r – load cycle asymmetry coefficient.

$$r = \frac{P_{min}}{P_{max}}. \qquad (5)$$

$$\Pi_n = \frac{2P_x^2}{3qEF}. \qquad (6)$$

As for the transverse bending modes of the tool's vibrations, damping along the Z-coordinate occurs in a similar manner, except that the tool undergoes bending displacements relative to the plates (Fig. 3).



a *b*

Fig. 3. Deformations of the Tool Under Load from Cutting Force Components (a – Compression, b – Bending)

As seen in Fig. 3a (with the cutting plates hidden), the compression of the tool holder is not centered. This is due to the displacement of the cutting plate above the center.

However, the situation is fundamentally different in the case of tool holder bending in the horizontal plane (Y-axis) (Fig. 4a).

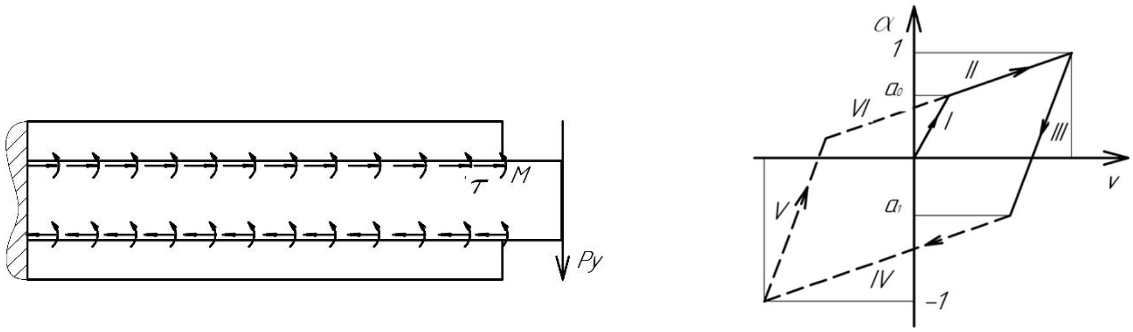


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Fig. 4. Vibration Energy Dissipation per Cycle
(a – Analytical model of the vibration energy damper along the Y-axis, b – Hysteresis loop)

As seen in the figure, shear stresses τ and bending moment M arise between the tool holder and the plates, which are evenly distributed along the entire contact length between the plates and the tool holder. As a result, the dependence of dissipated energy on the cutting force has a linear character (Fig. 4b).

The chromatogram of shear stresses between the tool holder and the side plates, which are pressed by the contact pressure created by the clamping bolt, is shown in Fig. 5a. The vector diagram of the contact pressure between the plates and the tool holder is presented in Fig. 5b.

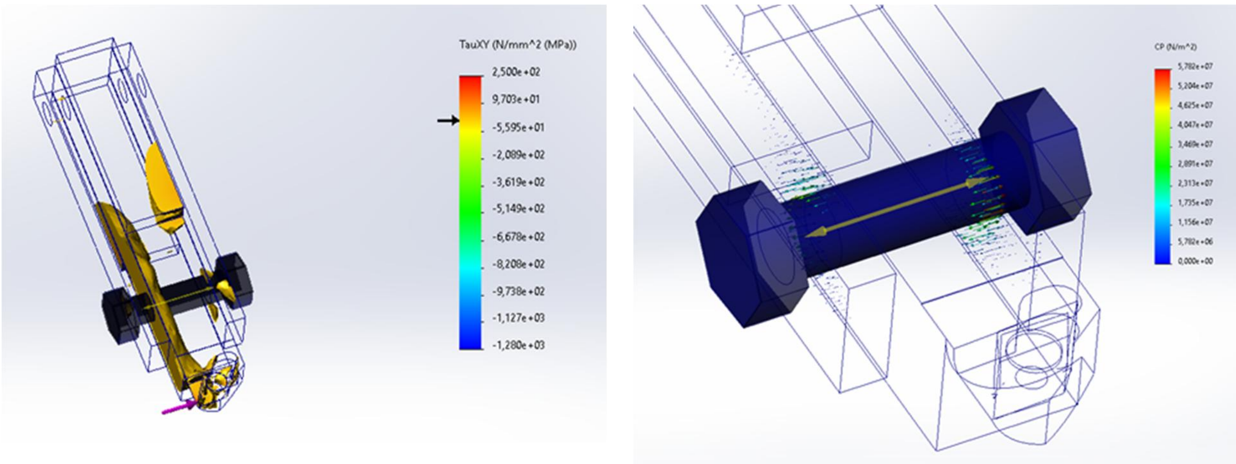

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Fig. 5. Calculated Parameters of the Tool (a – Shear stresses; b – Contact pressure)

As seen in Fig. 5a, the shear stresses are evenly distributed along the length of the side plate; however, the plates are not equally loaded. This is due to the displacement of the tooltip closer to the front (more heavily loaded) plate. The contact pressure values on both plates (Fig. 5b) are identical.

The dissipated energy is calculated using the following formula:

$$\Psi_n = \frac{8fpl^3}{Eh^2}\left(P_y - \frac{4fpbh}{3}\right), \tag{7}$$

where E is Young's modulus; b and h are the height and thickness of the plate, respectively.

The potential energy value is determined as follows:

$$\Pi_n = \frac{P_y^2 l^3 (4-3\alpha_0^2)}{48EJ}, \tag{8}$$

where J is the moment of inertia of the tool holder's cross-section.

Due to the transverse hole in the front part of the tool holder, the tool's stiffness decreases, which may negatively impact the vibration amplitude. The stiffness values for the solid tool holder are: C_x=63.29×10^6 N/m, C_y=12.27×10^6 N/m, C_z=14.64×10^6 N/m.

For the drilled tool holder, the stiffness values are: C_x=40.98×10^6 N/m (-35%), C_y=4.03×10^6 N/m (-40%), C_z=8.85×10^6 N/m (-67%). As we can see, the stiffness of the tool with a drilled holder has significantly decreased.

The damping energy parameters in the tool design were determined using equations (2–8).

Self-oscillations during the cutting process were studied by solving the system of differential equations for a four-mass oscillatory system (1), which consists of the tool post, the tool, the workpiece, and the spindle. Following the classical lathe model, these components are connected to the machine bed through elastic connections with dampers [5].

The research results: tool self-oscillation amplitudes are presented in Fig. 6.

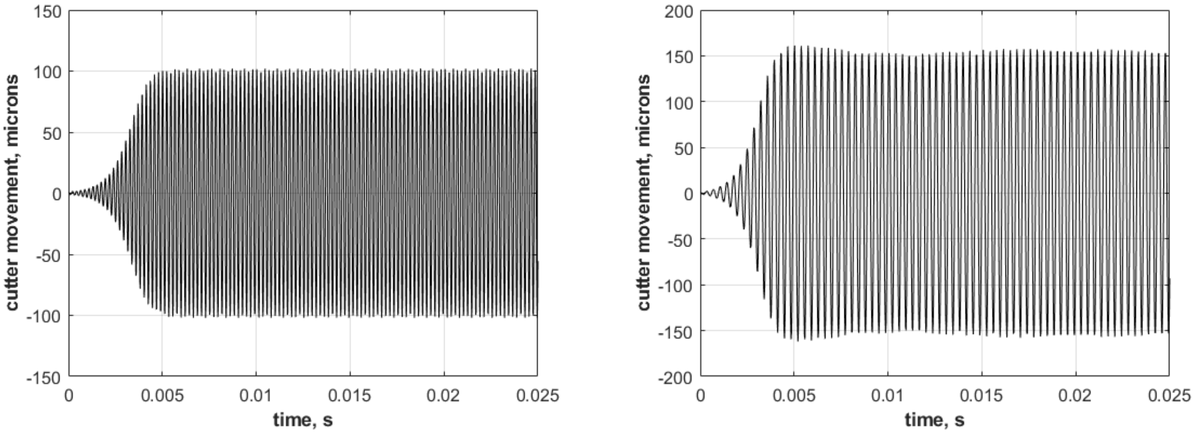

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Fig. 6. Amplitude of tool self-oscillations (a – with a solid tool holder, b – with a drilled tool holder)

As seen in the figure, the oscillation amplitude of the tool with a drilled holder increased by 1.5 times, while the oscillation frequency decreased, which is characteristic of a reduction in the stiffness of the self-oscillatory system.

The tightening of the bolt at the front of the assembled tool should create a friction force between the plates and the tool holder, proportional to the tightening force.

Fig. 7 presents the tool's self-oscillation amplitudes for different bolt-tightening forces.

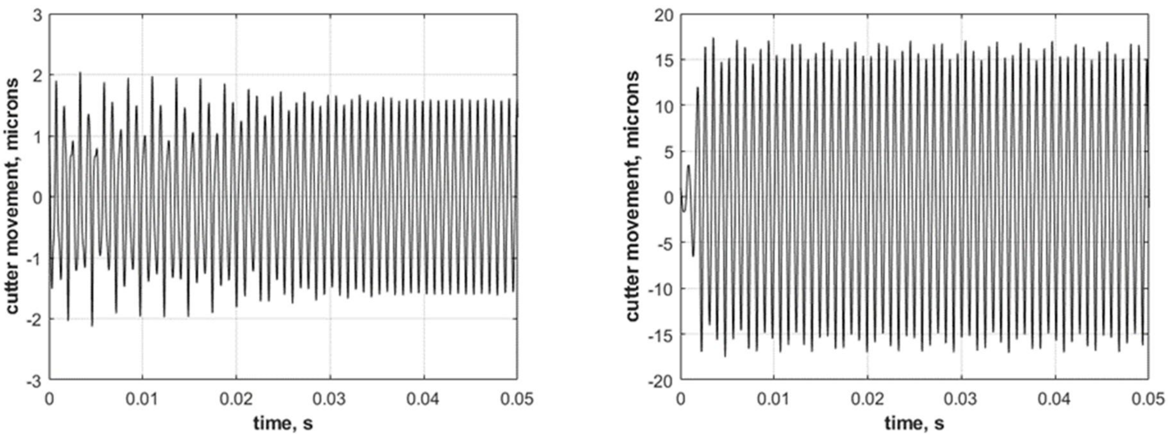

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Fig. 7. Amplitude of tool self-oscillations at different bolt tightening forces (a – 50 N; b – 500 N).

As seen in Fig. 7a, tightening the bolt creates a friction force that significantly reduces self-oscillation amplitude due to the dissipation of oscillation energy. However, further increasing the bolt tightening force (Fig. 7b) increases oscillation amplitude.

Theoretical studies of the mathematical model show that increasing the contact pressure between the plates and the tool holder reduces the energy dissipation of oscillations. Consequently, this results in an increase in oscillation amplitude, which fully correlates with the graphical dependence presented in Fig. 2b

Conclusions

This study investigates the effect of structural friction within the body of a modular cutting tool on the damping of its self-excited vibration amplitude. The analysis confirms that an optimal selection of tool parameters, particularly the contact pressure between the inserts and the tool holder, allows for the effective dissipation of vibrational energy. This leads to a reduction in self-excited vibration amplitude, which positively impacts the surface quality of the machined part and enhances tool durability.

The research results demonstrate that the primary damping mechanism is associated with generating shear stresses and hysteresis losses in the contact zone between the inserts and the holder. The analysis indicates an optimal contact pressure at which energy dissipation is maximized. Identifying this optimal pressure is crucial for improving the tool's performance and stability during machining.

It has also been established that a transverse hole in the tool holder reduces its stiffness, which should be considered in tool design. The obtained results can be used to improve the design of metal-cutting tools to reduce vibrations and enhance the machining process.

References

[1] Y. Altintas, *Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design*, 2nd ed. Cambridge, UK: Cambridge University Press, 2012.

[2] Y. Altintas, M. Eynian, and H. Onozuka, "Identification of dynamic cutting force coefficients and chatter stability with process damping", *CIRP Annals - Manufacturing Technology*, vol. 57, pp. 371-374, 2008.

[3] Yu. Ya. Novitskyi, P. R. Dmyterko, "Optymizatsiia parametriv demfuvannia avtokolyvan bahatochastotnoi kolyvalnoi systemy" ["Optimization of parameters of damping of self-oscillations of a multifrequency oscillatory system"], *Visnyk Natsionalnoho universytetu "Lvivska politekhnika"*, Ser. Optymizatsiia vyrobnychykh protsesiv i tekhnichnyi kontrol u mashynobuduvanni ta pryladobuduvanni, no. 867, pp. 51–55, 2017. [in Ukrainian]

[4] Y. Novitskyi, M. Novitskyi, and Y. Lytvyniak, "Features of vibration dampers application in the design of vibration-resistant metal cutting tools", *Ukrainian Journal of Mechanical Engineering and Materials Science*, vol. 10, no. 1, pp. 38–44, 2024.

[5] M. Novitskyi, Y. Novitskyi, and A. Slipchuk, "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", *Ukrainian Journal of Mechanical Engineering and Materials Science*, vol. 9, no. 4, pp. 44–50, 2023.

[6] Y. M. Vnukov, S. I. Djadja, M. V. Kuchuhurov, and E. V. Kondratyuk, "The device for self-excited vibrations researching in turning", Ukrainian Patent UA101906, 2015.

[7] Y. M. Vnukov, M. V. Kuchuhurov, S. I. Djadja, and others, "Method and device for researching the regenerative self-excited vibrations when turning", in *Cutting & Tool in Technological System*, International Scientific-Technical Collection, Kharkov, vol. 83, pp. 42–54, 2013.

[8] A. Slipchuk and M. Novitskyi, "Definition of the geometric parameters of the undeformed chip at the cut-in stage when machining an external gear using the power skiving method", *Ukrainian Journal of Mechanical Engineering and Materials Science*, vol. 9, no. 4, pp. 50–59, 2023.

[9] V. Stupnytskyy, Y. Novitskyi, Y. Novitskyi, and S. Xianning, "Comprehensive system for simulation of vibration processes during the titanium alloys machining", *Archive of Mechanical Engineering*, vol. 70, no. 1, pp. 85–105, 2023.