SUPPRESSION OF SELF-EXCITED VIBRATION IN CUTTING PROCESS USING PIEZOELECTRIC AND ELECTROMAGNETIC ACTUATORS

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S u m m a r y

This paper describes the reduction of self-excited vibrations with controlled eliminators containing a piezoelectric actuator and an electromagnetic system. The results of numerical simulations and experimental investigations confirm the effectiveness of the eliminators. The simulations were conducted with a Matlab - Simulink system. The experimental verification was conducted at a workstation simulating the cutting process, and also on an NC machine tool.

Keywords: machine tool, vibrostability, piezoelectric actuator, electromagnetic actuator, self-excited vibration

Tłumienie drgań samowzbudnych eliminatorami z piezoelementami i elektromagnesami w procesie skrawania

S t r e s z c z e n i e

W artykule przedstawiono sterowane eliminatory do tłumienia drgań samowzbudnych podczas frezowania. W konstrukcji eliminatorów zastosowano aktuator piezoelektryczny oraz układ elektromagnesów. Omówiono wyniki symulacji numerycznych oraz badań doświadczalnych, które potwierdzają skuteczność działania eliminatorów. Badania symulacyjne wykonano w systemie Matlab-Simulink. Weryfikację doświadczalną prowadzono na stanowisku symulującym proces skrawania, a także podczas badań pracą na obrabiarce numerycznej.

Słowa kluczowe: obrabiarka, wibrostabilność, aktuator piezoelektryczny, aktuator elektromagnetyczny, drgania samowzbudne

1. Self-excited vibrations and their elimination

One of the key questions in the operation of machine tools is the vibrostability of the 'machine tool – machining' system, which to a great extent depends on self-excited vibrations. The analyses of these vibrations have lead to many concepts for counteracting this undesirable phenomenon.

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Self-excited vibrations depend on the motion parameters of the vibrating system [1]. They can occur only in the presence of a variable force that generates them. Further development of vibrations in the system results from the occurrence of internal feedback between the motion of the machine tool casing elements (mass-damper-spring system elements) and the work process.

The development of self-excited vibrations is mainly caused by trace regeneration. A flexible cutting tool starts to vibrate due to the impulse of the cutting force. On the workpiece surface a wave appears which reflects the relative motion of the tool and the workpiece. The resulting irregularities cause variations in chip thickness, which in turn affect the values of cutting force and consequently induce further vibrations in the system. Figure 1 presents the discussed variations in chip thickness.

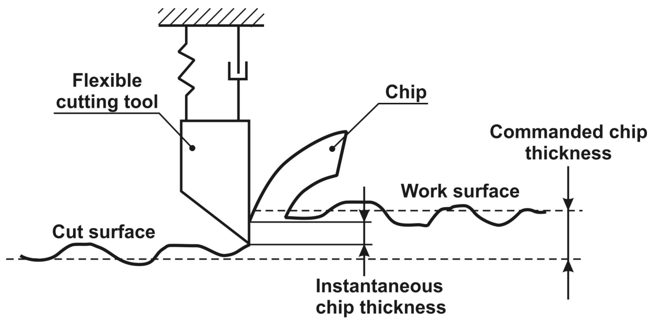

Fig. 1. Chip thickness variation due to cutter vibrations

In milling, the subsequent cutting edge cuts a wave which has been created by a previous cutting edge on the workpiece surface. This wavy surface changes the instantaneous chip thickness, which in turn results in a variation in the cutting force values and further cutter vibrations. This feedback mechanism (internal feedback) may lead to self-excited vibrations in the 'machine tool-machining' system. Depending on the relationship between the waves cut on the surface by any previous and the present cutting edge of the vibrating tool, the machining forces may greatly increase or decrease [1, 2].

One can mention several methods for the elimination of self-excited vibrations, e.g. tools with the active elimination of vibrations, a change of phase between the external and internal modulation in trace regeneration, and the change of dynamic properties of the machine tool-machining system through the use of vibration eliminators. The decrease of the effect of trace regeneration through a change in the phase shift is performed by means of a variable rotation speed of the spindle [1, 3, 4]. Another parameter that may affect the vibrostability of machining is feed speed. In the methods mentioned so far, they are mainly applied for one-edge tools (e.g. used in turning) [4, 5] For machining with multiple edge tools, system vibrostability can be enhanced by changes in tool geometry (for example the rake angle) [6] and unequal spacing of cutting plates in the mill head, which gives a similar effect to variable cutting speeds. Another method used in the reduction of self-excited vibration is the introduction of a vibration eliminator system based on the intentional/deliberate control of the energy flow in the system. Eliminators use the principle of compensation of forces or are based on the principle of additional suppression which becomes significant in the resonance of the main system and decreases the amplitude of resonance vibration [7, 8].

2. Vibration eliminators

In a system including a machine tool, handgrip, workpiece and cutting tool, it is often the workpiece that has a considerable flexibility. Machining such a workpiece may induce the undesirable effect of self-excited vibration. For the purpose of experimental studies on eliminators, a special element is installed on the milling table (Fig. 2) to simulate the properties of a workpiece with high flexibility in one direction. Controlling the height of flat springs, one obtains a variable stiffness in the flexible workpiece. Between the upper and lower plates there is a place for clamping a vibration eliminator.

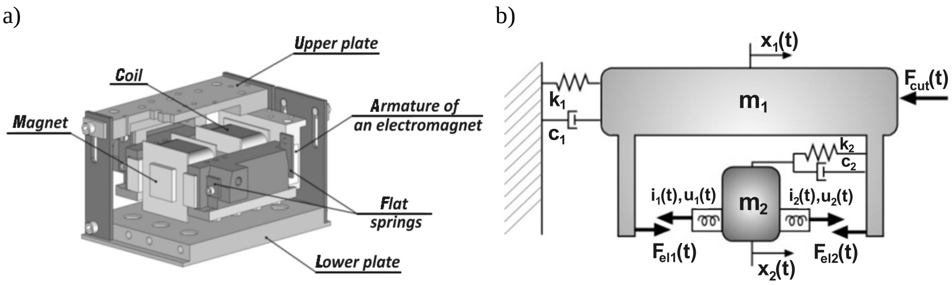

Fig. 2. Structure of an electromagnetic vibration reduction system (a) and phenomenonological model of the electromagnetic vibration eliminator (b)

This paper presents the construction of two eliminators used in the reduction of self-excited vibration. It also presents simulation studies on vibration elimination using the aforementioned systems and experimental results on a real object.

3. Eliminator with an electromagnetic system

The first of the discussed eliminators uses a electromagnet with double-sided work. The place of attachment of the eliminator was selected with the assumption that its greatest effectiveness would be obtained in the closest possible proximity to the cutting zone. Figure 2b presents a diagram with the construction of the eliminator, consisting of two electromagnets which ensure double-sided work, and flat springs for attuning to a selected frequency. The system adapts to a required frequency of vibrationsvibration, responsible for the generation of self-excited vibrations. The eliminator mass is an electromagnet which generates a force dependent on the value of the air gap between its face and the clamp, and from the value of current flowing through its windings.

In the structure of the eliminator one may isolate the following sections:mechanical – deciding upon the frequency and amplitude of free vibrations; electromagnetic – determining the effect of an actuator on the object. Figure 2 presents a diagram showing a simplified model of the vibration eliminator. Machining a workpiece, with a mass m1 and a stiffness coefficient k1, is characterized by a high level of vibrations due to its low stiffness. The workpiece shows one dominant frequency of free vibrations resulting from the parameters m_1, k_1, c_1. In order to improve machining conditions, the vibration eliminator was attached; having a mass m1, stiffness coefficient k_2, and damping coefficient c_2. These three parameters were matched to ensure the passive calibration of the eliminator to the vibration of the work. A machining process is a dynamic phenomenon which induces the frequency variation of self-excited vibrations. This variation decreases the effectiveness of the passive eliminator due to the de-calibration of the system from the designed frequency. As self-excited vibrations are sensitive to factors that interfere with their generation and development, it is best to use an active eliminator with two electromagnets working in a differential system. Their task is the generation of forces F_{el1} and F_{el2}, which move the eliminator mass m_2, and via springs k_2 induce a motion in the workpiece to ensure a reduction in vibrations m_1 caused by the cutting force $F_{skr}(t)$.

The equation of motion for the aforementioned system with two degrees of freedom may be described as follow:

$$\mathbf{M}\ddot{\mathbf{x}}(t) + \mathbf{C}\dot{\mathbf{x}}(t) + \mathbf{K}\mathbf{x}(t) = \mathbf{F}(t) \tag{1}$$

where:

$$\begin{gathered} \mathbf{M} = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix},\ \mathbf{C} = \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix},\ \mathbf{K} = \begin{bmatrix} k_1 + k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix} \\ \mathbf{F} = \begin{bmatrix} -F_{skr}(t) + F_{el1}(t) - F_{el2}(t) \\ -F_{el1}(t) + F_{el2}(t) \end{bmatrix},\ \mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} \end{gathered} \tag{2}$$

The construction of a model described by formula (1) requires the identification of an object in order to determine the necessary values of mass, stiffness and supression. A linearized model (described by equations 3 and 4) was used to model the properties of the electromagnetic actuators [9-11].

Taking into account the two electromagnets working in a differential system (Fig. 2b), the following relationships were assumed for the electromagnet continuous:

$$F_{el1} = \kappa \left(\frac{i_0}{x_{s0}} \right)^2 + k_i i(t) + k_s x_s(t) \tag{3}$$

$$F_{el2} = \kappa \left(\frac{i_0}{x_{s0}} \right)^2 - k_i i(t) - k_s x_s(t) \tag{4}$$

where:

$$\kappa = \frac{\mu_0 A z^2}{4}, \quad k_i = 2\kappa \frac{i_0}{x_{s0}^2}, \quad k_s = 2\kappa \frac{i_0^2}{x_{s0}^3}, \; L_0 = \frac{2\kappa}{x_{s0}} \tag{5}$$

μ_0 – air magnetic permeability, A – surface area of the core surface, z – number of coil windings, i – current flowing in the coil, x_s – air gap.

The attached eliminator influences the amplitude of the workpiece motion both during stable and unstable machining. Such a solution enables an increase in the values of machining parameters at which stability is lost, and in the case of stable work – an improvement in the quality of the work surface.

Control of the vibration eliminator was carried out by a system using a suboptimal pole placement design (PZP) [12]. The quality of control was measured using two indicators: efficient value for the speed of vibration of the moving workpiece (RMS) described by formula (6), and maximum machining width d at which vibrations of the workpiece $x_1(t)$ do not cause a loss of contact between the cutting edge and the workpiece.

$$RMS = \sqrt{\frac{1}{\tau} \int_{t-\tau}^{t} \left[\dot{x}_1(t) \right]^2 \mathrm{d}t}, \quad \tau = \frac{60}{n \cdot l_{\mathrm{zeb}}} \tag{6}$$

where: n – spindle rotation speed, l_{zeb} – number of cutting edges.

Figures 3 and 4 show examples of RMS indicators for the proposed system and maximum machining widths with active and inactive systems of eliminator control.

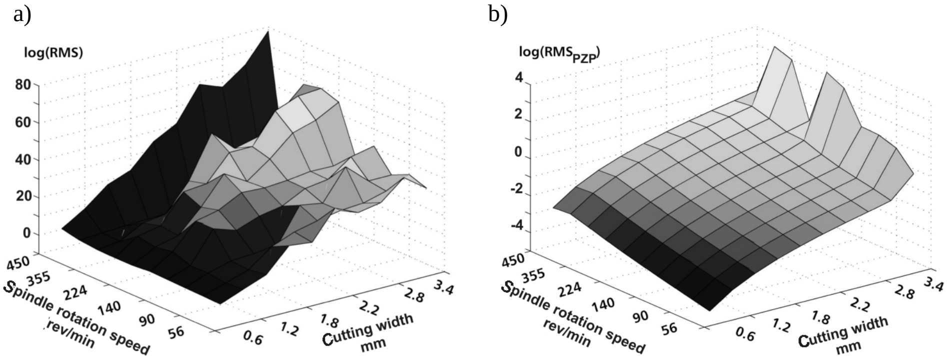

Fig. 3. RMS criterion for machining: a) without the control system, b) with an eliminator controlled by a PZP system

The analysis of RMS quality criterion (Fig. 4) shows a distinct improvement in maximum machining widths by 50% over the entire range of rotation speeds, in comparison with the system without eliminator control.

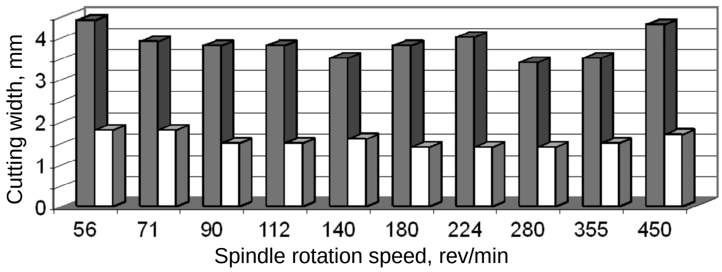

Fig. 4. Maximum cutting width with the active (grey) and inactive (white) PZP control system

Figure 5 presents examples of plots of the characteristic signals during simulation of the machining variant n = 112 rev/min, d = 0.6 mm.

4. Piezoelectric eliminator

In the construction of another version of a vibration eliminator, a piezoelectric actuator was fixed to the underside of the upper plate (Fig. 6a).

The eliminator masses are electromagnets together with a casing, attached to the underside of the upper plate on four flat springs. The actuator and the eliminator mass are connected by a special clamp. The piezoelectric actuator PST-1000-16-150-vs25 manufactured by Piezomechanik GmbH has the following parameters: maximum elongation: 150 µm, length: 147 mm, capacity: 1400 nF, stiffness: 50 N/µm, maximum load: 15 kN [13, 14].

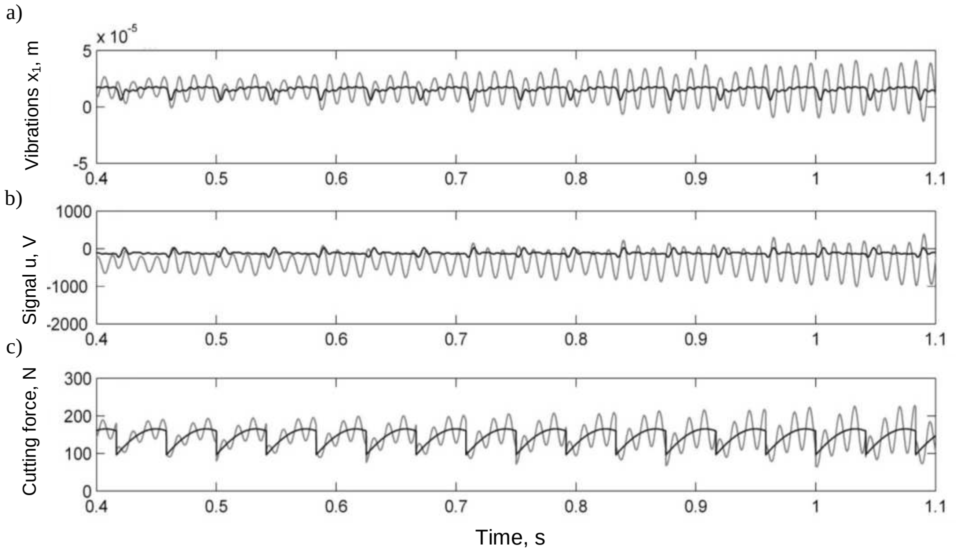

Fig. 5. Numerical simulation of the machining with n = 180 rev/min and depth of cut d = 1.6 mm (black – with eliminator controlled by the PZP system, grey – system without eliminator control): a) $x_1(t)$ – vibrations of the workpiece (according to Fig. 2c), b) $u(t)$ – signal controlling the piezoelectric actuator, c) $F_{cut}(t)$ – cutting force

Figure 6b shows a phenomenonological model of the vibration eliminator with a piezoelectric actuator.

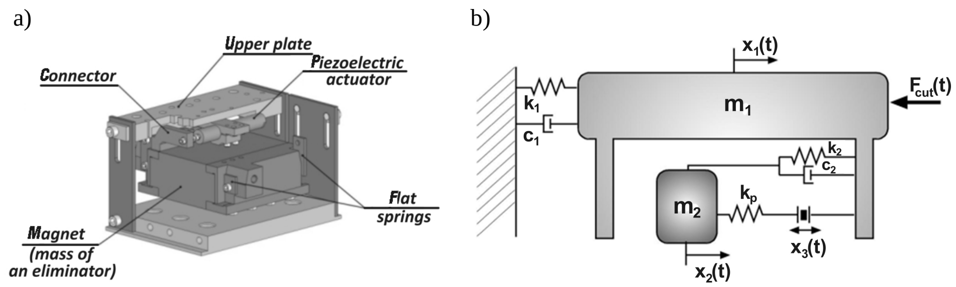

Fig. 6. Construction of the vibration eliminator using a piezoelectric actuator (a), phenomenological model of the piezoelectric vibration eliminator (b)

The equation that describes the motion of the eliminator is as follows:

$$\mathbf{M}\ddot{\mathbf{x}}(t) + \mathbf{C}\dot{\mathbf{x}}(t) + \mathbf{K}\mathbf{x}(t) = \mathbf{F}(t) \tag{7}$$

where:

$$\mathbf{M} = \begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix},\ \mathbf{C} = \begin{bmatrix} c_1 + c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix},\ \mathbf{K} = \begin{bmatrix} k_1 + k_2 + k_p & -k_2 - k_p \\ -k_2 - k_p & k_2 + k_p \end{bmatrix}$$
$$\mathbf{F} = \begin{bmatrix} -F_{skr}(t) + F_p(t) \\ -F_p(t) \end{bmatrix},\ \mathbf{x} = \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} \tag{8}$$

In this system, the force generated by the piezo element is defined by the equation:

$$F_p(t) = k_p x_3 \tag{9}$$

where: k_p – the stiffness of the piezo element together with the clamping system, x_3 – the motion/displacement generated by the actuator without loading it.

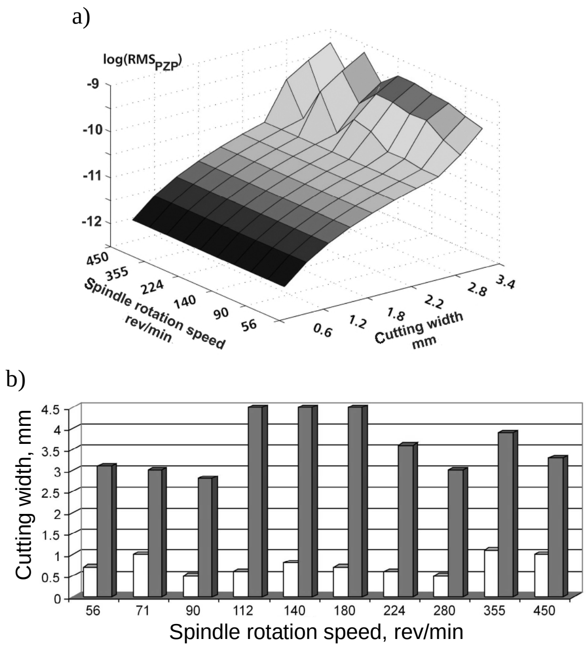

Fig. 7. RMS criterion for machining with an eliminator controlled by a PZP system (a) and maximum cutting width with the active (grey) and inactive (white) PZP control system (b)

The piezoelectric system of vibration elimination is described by the model as a structure analogous to the model of the electromagnetic system. Below are the results of the numerical simulations for the pole placement system.

The RMS indicator for the proposed system and the maximum machining widths with an active and inactive eliminator control system are shown in Figures 7a and 8, with examples of numerical simulation plots in Fig. 8.

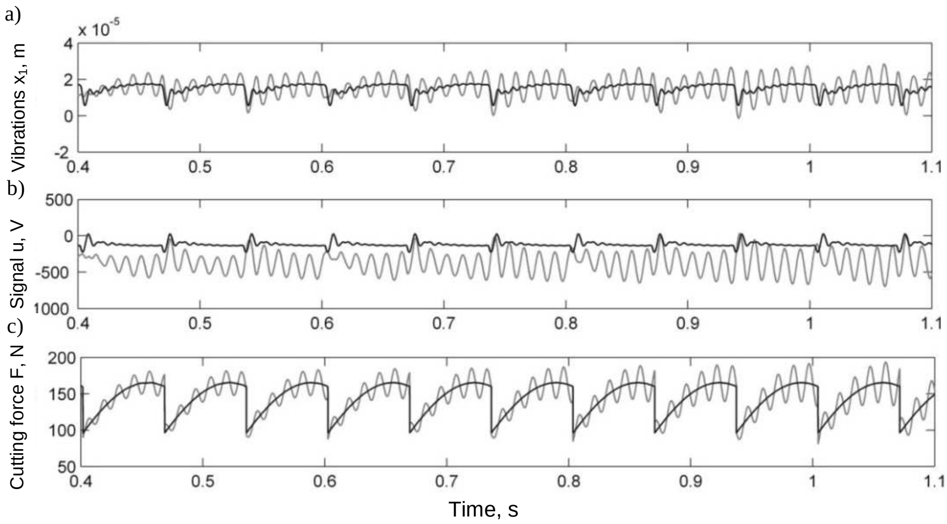

Fig. 8. Numerical simulation of the machining with n = 90 rev./min and depth of cut d = 1.0 mm with an eliminator controlled by a PZP system (black), and without eliminator control (grey): a) $x_1(t)$ – vibrations of the workpiece (according to Fig. 6b), b) $u(t)$ – control signal, c) $F_{cut}(t)$ – cutting force for machining

5. Examination at the workstation simulating the machining process

The effectiveness of the self-excited vibration eliminator was verified in two stages. First, laboratory tests were performed at a workstation simulating the machining process (Fig. 9a).

During the experiment with the workstation, machining was simulated by an electrodynamic inductor TIRAvib5200-M with an amplifier TV5220 (Fig. 9). Figure 10 presents the diagram of the cutting force in the numerical model and its realization on an electrodynamic inductor.

Cutting force was measured using a Kisler 5kN type U9A tensometric sensor. The evaluation of the vibration level of the workpiece and eliminator was performed using two laser heads for velocity measurement included in the Polytec PSV-400 3D vibrometer. The registration of signals, modelling cutting force and controlling inductor and eliminator were carried out on a dSpace 1104 signal processor control card with a sampling rate of 10 kHz.

a) b)

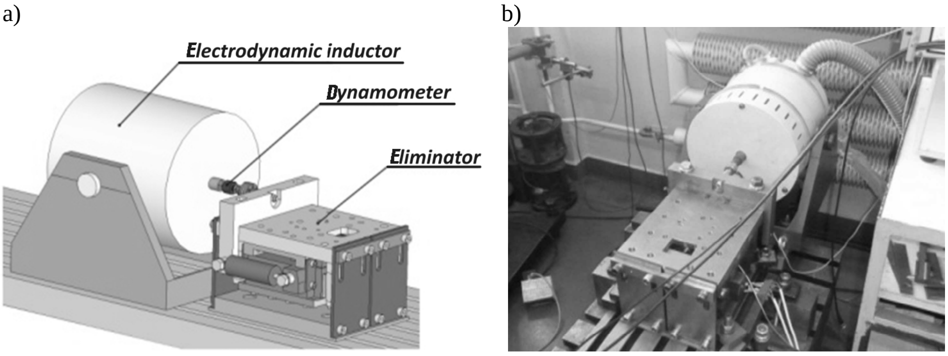

Fig. 9. The workstation for the examination of a system reducing the vibrations of a workpiece, with a simulated machining process: a) CAD model, b) eliminator with an inductor

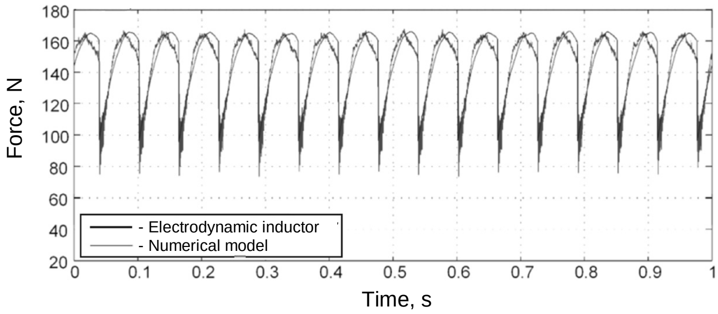

Fig. 10. Machining force in the numerical model and its realization on a TIRA inductor

The electromagnetic and piezoelectric systems of vibration elimination were examined on a workstation simulating the cutting process to evaluate the usefulness of eliminators for vibration suppression during machining. At a fixed rotation speed, cutting simulation was performed for 30 s or until the development of a self-excited vibrations with an excessive amplitude. During the experiment, the cutting width was subject to step changes every 2 s, from 1.0 to 5.0 mm with a 0.2 mm step. The plot of speed and motion were recorded for the workpiece and eliminator, along with parameters characteristic for the given system of vibration reduction system. In the electromagnetic system, the parameters were current and output voltage of the amplifier, and in the piezoelectric system it was the output voltage of the amplifier.

This experiment was performed for all available rotation speeds of the machine tool (i.e. 56-450 rev/min) and the most effective regulation systems [15].

Results for the selected machining variants are presented in subsequent subsections.

5.1. Examination of an electromagnetic system on a workstation simulating a machining process

The effectiveness of the electromagnetic vibration elimination system on a workstation simulating a machining process is illustrated in the Fig. 11, with the results of an off-on-off test (eliminator was on for 1 to 4 s). The figure presents the following parameters:

- motion of the workpiece,
- current in the electromagnets, 1 (black) and 2 (grey),
- effective value of RMS speed.

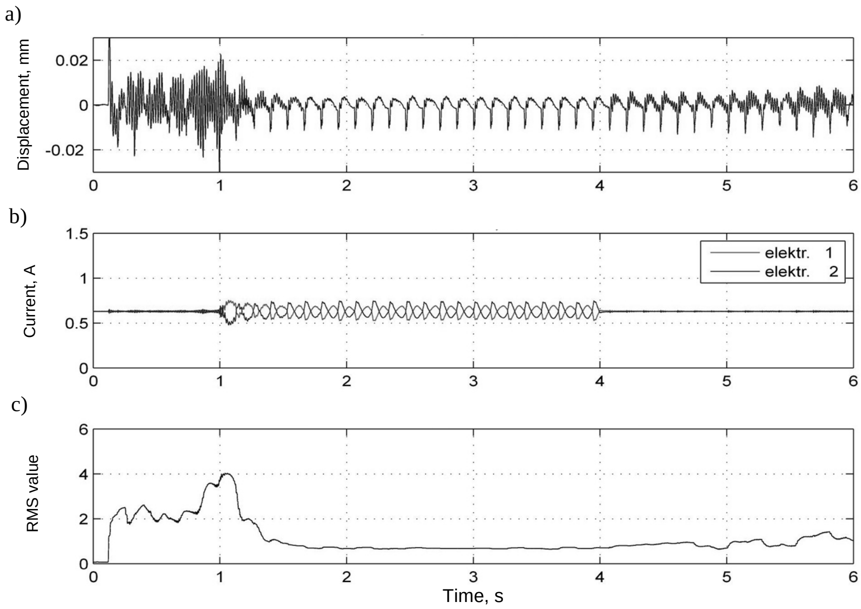

Fig. 11. Simulation of cutting process from switch off and switch the electromagnetic eliminator for the rotatinal speed n = 56 rpm and depth of cut doc = 2.1 mm: a) motion of the workpiece, b) current in the electromagnets, c) effective value of RMS speed

5.2. Examination of a piezoelectric system on a workstation simulating a machining process

Figure 12 presents the results of an off-on-off test. The figure presents the following parameters:

- motion of the workpiece,
- output voltage of a piezoelectric amplifier,
- RMS effective value of RMS speed.

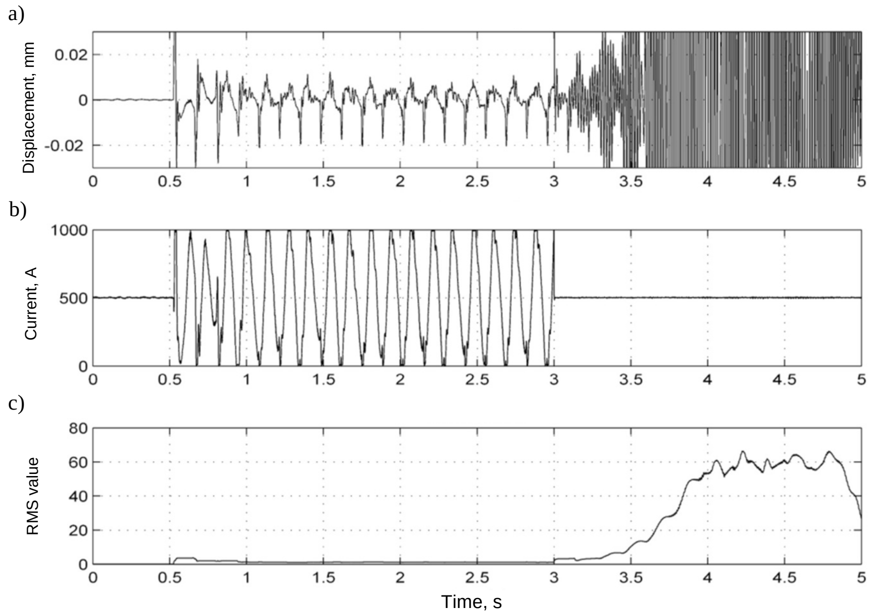

Fig. 12. Simulation of machining a workpiece with an eliminator on and off, controlled from the spindle rotation speed at 56rev/min: a) motion of the workpiece, b) current in the electromagnets, c) effective value of RMS speed

The performed experiments confirm the effectiveness of electromagnetic and piezoelectric systems of vibration reduction during machining. Figures 11 and 12 show an improvement in cutting conditions and decreased workpiece vibrations in machining according to RMS criteria and time plots of workpiece motions.

6. Experimental verification of vibration elimination systems

The positive results of vibration reduction obtained on a workstation simulating the machining process using an electromagnetic inductor were verified by work on a numerically controlled JAFO FYN-50 machine tool. Figure 13 shows a machine tool with an attached workpiece and electromagnetic eliminator.

The experiment used a 22 mm NFPa cutting blade at rotation speeds ranging from 56 to 450 rev/min, and depth of cut from 2 to 4.5 mm. The selected results of the performed measurements are presented below.

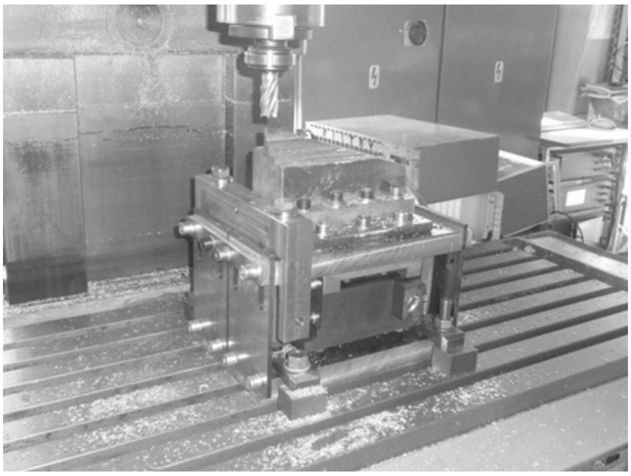

Fig. 13. JAFO FYN-50 cutter used for the examination of effectiveness of vibration elimination

6.1. Experimental verification of the electromagnetic system

Figure 14 shows the following:

- speed of the workpiece,
- motion of the workpiece – fragment representing the moment of turning on and off,
- current in electromagnets, 1 (black) and 2 (grey).

The electromagnetic system of vibration reduction enabled an effective work with a greater machining width. The general decrease in vibrations, distinctly visible on an RMS indicator, and a descrease in the speed of the workpiece, are reflected by a enlarged fragment of the motion diagram.

6.2. Experimental verification of the piezoelectric system

Figure 15 shows the results for two spindle rotation speeds. The diagrams present the following:

- speed of the workpiece vibrations,
- motion of the workpiece – fragment representing the moment of turning on and off,
- output voltage of the amplifier that controls the piezoelement,
- calibrated effective value of the RMS speed.

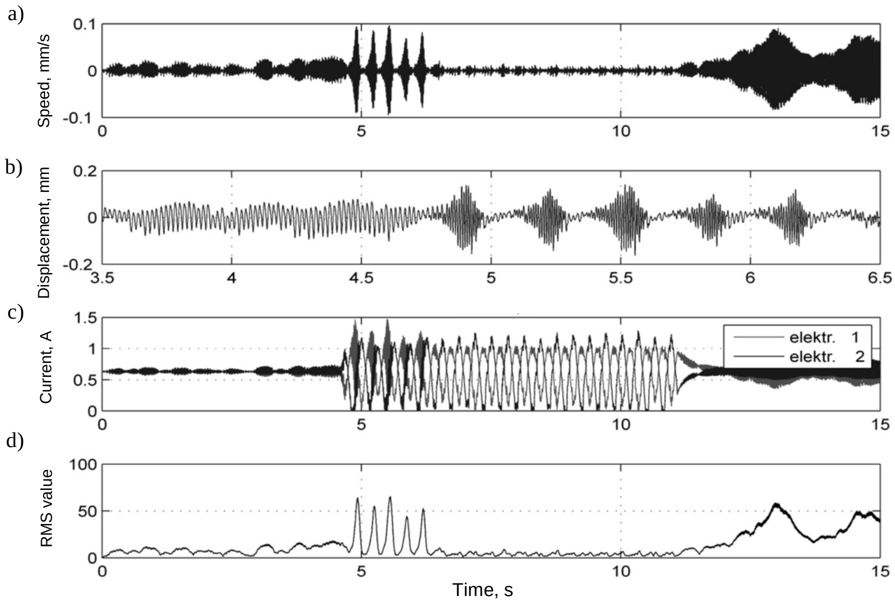

Fig. 14. Simulation of machining a workpiece with an eliminator on and off, at the spindle rotation speed at 224 rev/min, depth of cut d = 2 mm, feed p_m = 30 mm/min: a) speed of the workpiece, b) displacement of the workpiece, c) current in electromagnets, d) effective value of RMS speed

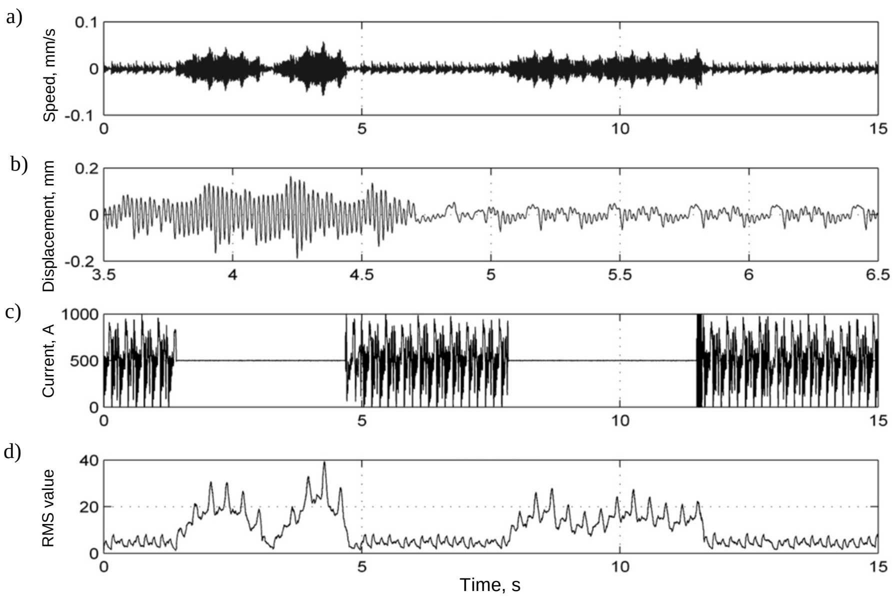

Fig. 15. Machining the workpiece with the eliminator on and off, at a spindle speed of 224 rev/min, depth of cut d = 2 mm, feed speed p_m = 60 mm/min: a) speed of the workpiece, b) motion of the workpiece, c) current in electromagnets, d) effective value of RMS speed

The piezoelectric system of vibration reduction constructed in Figure 6a shows an effective action over a wide range of machining parameters. The use of a piezoelectric actuator strongly modifies the dynamic properties of the workpiece, due to the high stiffness of the piezo-stack. A passive eliminator offsets the frequency of the workpiece, but the parameters of the applied piezo element ensure an effective decrease in vibration level. Very good properties of the piezo element with regard to higher frequencies make it more useful for workpieces with higher free vibration frequencies than electromagnetic systems.

7. Conclusions

The presented systems of vibration elimination using electromagnets and piezo elements enable a reduction of vibration level and more effective work with a greater machining width over a wide range of changes in machining parameters. A general decrease in vibrations is distinctly observable using RMS criterion and workpiece feed speed.

An important problem of the electromagnetic system is the question of "attaching" the electromagnet with a clamp, especially in the first stage of machining, when not all the edges cut along the entire length.

The application of a piezoelectric actuator strongly modifies the dynamic properties of the object – a passive eliminator offsets the workpiece frequencies.

The high dynamics of the piezoelement enables more efficient work at higher frequencies than the electromagnetic system. Additionally, the application of a piezoelectric actuator enables a more compact construction of the system, but is connected with a considerable cost (piezoelement or a high voltage amplifier).

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