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SELF-EXCITED VIBRATIONS COUNTERACTION IN MACHINE TOOL – CUTTING PROCESS SYSTEM USING CONTROLLABLE ELIMINATOR OF VIBRATION

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In the paper, the results of simulation investigations on the application of a semi-active vibration eliminator for suppression of machining chatter are presented. First, the idea of a semi-active vibration eliminator arranged in the "machine tool - cutting process" system (MT-CP) is presented. Next, the results of the identification process of parameters of this system are shown. Properties of the mass-damping-spring system of the machine tool were estimated during the modal testing. A mechanistic model of the cutting process was built on the basis of results of cutting tests. Two control algorithms have been tested: one with the time-invariant model of disturbances (MD) and another with the adaptive model of disturbances. Estimation of the eliminator effectiveness in the suppression of self-excited vibrations in a MT-CP system is also included.

Key words: vibrostability, semi-active eliminator of vibration.

1. Introduction

Self-excited vibrations, which appear in dynamic MT-CP (machine tool – cutting process) system are very undesirable. Their negative influence on productivity, tool life and the workpiece surface quality is well known. Although a great improvement of dynamic properties in new designs of machine tools is now observed, chatter vibrations are still present in modern machining. There are few reasons of the development of cutting instability. Assuming good dynamic properties of the machine tool, the chatter can be caused by a wrong selection of cutting parameters. It is, however, a rare situation. The most common case is when in the production process, the application of a very compliant tool is necessary and/or the machined workpiece is flexible. This causes that new methods of opposing the phenomenon of instability are still being developed and improved.

There are different possible ways of solving the problem of chatter. The literature devoted to this topic is very rich. Different approaches to this problem are tested and various methods are presented. The most common, but insufficiently effective, method used to cancel out the chatter in MT-CP system is the reduction of the effect of the phase shift between the outer and the inner modulation through the variation of the spindle speed [5, 6, 11]. Some "optimal" control algorithms of these changes were developed [4, 13]. Methods of adaptive control of the feed rate are implemented too [1, 10]. A separate group of methods is based on the application of different types of vibration absorbers – passive, semi-active and active [7, 12, 14].

In this paper, investigations connected with the development and application of a controlled dynamic vibration absorber (the eliminator of vibrations), are presented. The starting point of the investigations was the statement that the weakpoint, that is "responsible" for the loss of stability in dynamic MT-CP system, is the workpiece. This statement was preceded by the analysis of a great number of modern constructions of machine tools, which apparently show usually high rigidity. Thus, a special element for modeling a compliant workpiece was designed. The flexibility of this element can be controlled, influencing dynamic characteristics of the whole MDS (mass-damping-spring) structure of the machine.

The idea of application of a controllable dynamic absorber for neutralizing the chatter is based on the fact that the frequency of chatter changes with the change of parameters of MT-CP system. The advantage of the controllable absorber, compared with the passive one, is the fact that it can change its stiffness, and can be tuned to the temporary frequency of chatter vibrations. Additionally, self-excited vibrations are very sensitive to the changes of structural parameters of the machine tool.

In order to receive reliable data for the absorber design and obtain good effectiveness of vibration minimization, a range of simulation tests and experimental investigations was considered. The results of modeling and simulations were then verified during cutting tests on the experimental stage of the work. The series of actions that were necessary to get the goal were carried out in the following order:

1. Modification of MDS structure of machine tool, through attachment of a special element simulating the behavior of the compliant workpiece.
2. Experimental investigations of MDS structure of FWD-32J milling machine.
3. Identification of parameters of the MT-CP system.
4. Design of the vibration eliminator and its realization.
5. Identification of parameters describing the model of the absorber.

6. Simulation investigations of efficiency of the absorber controlled by different control algorithms.

Individual steps of this algorithm are described in detail in the next sections of this paper.

2. Estimation of parameters of Machine Tool – Cutting Process system

It was noticed in the previous section of this paper, that usually the machine tool looses stability because of a compliant workpiece. To simulate such a behavior of the workpiece, a special element was added to the primary structure of the machine tool. This element, mounted on the table of the machine, is shown in Fig. 1. Construction of this element allows to change its stiffness by changing the active length of flat springs. The stiffness in the feed direction was intentionally made higher than the stiffness in other directions. After computations of the static and dynamic characteristics, the element was build and tested. In the next step of the work, the impulse tests were conducted. They helped to determine the dynamic characteristics (dynamic compliances) of the modified machine tool. Kistler 9726A20000 modal hammer and Kistler acceleration sensors were used during this test. SCADAS III analyzer was used as a front-end device for acquiring and conditioning the signals. Transfer functions were computed with the help of CADAX-NT software. Results of the tests are presented in Fig. 2. For most of the frequencies applied in the test, the y amplitudes are much greater in comparison with other directions. It is a very important conclusion, because

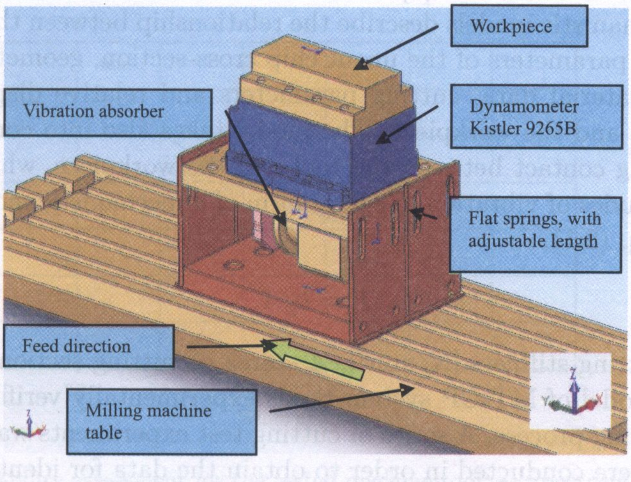

Fig. 1. Modification of machine tool MDS system by the addition of the element simulating compliant workpiece.

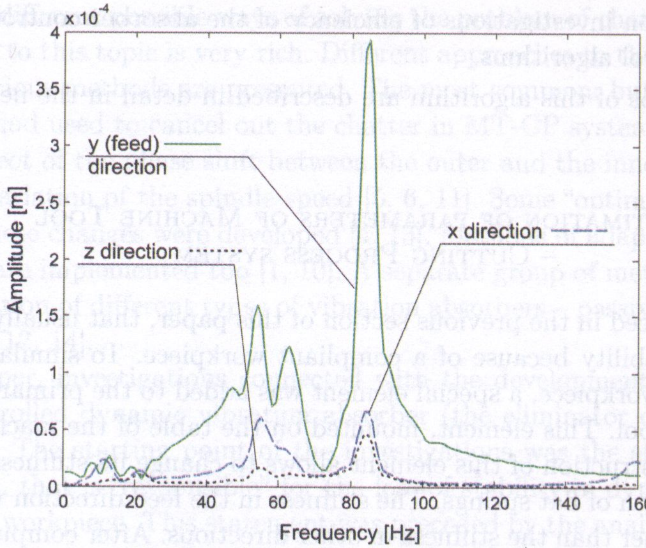

FIG. 2. Results of the impulse test of modified MDS structure of FWD 32-J machine tool.

it sanctions the reduction of MT-CP model made in the next stage of the investigations. Dynamic characteristics obtained during impulse tests were used in the process of estimating the parameters describing the model of the analyzed system.

Dynamic MT-CP system consists of a MDS system of the machine tool and the cutting process. There are different models of the cutting process – analytical, experimental and mechanistic [3]. A mechanistic model was used in these investigations. Mechanistic models describe the relationship between the cutting force and geometric parameters of the uncut chip cross-section, geometric description of the tool, material data, cutting parameters and relative displacements between the tool and the workpiece. The model takes also into consideration the effect of loosing contact between the tool and the workpiece, which takes place at high amplitudes of vibration. General form of a mechanistic model considered is shown in Eq. (2.1).

$$F = k_p A, \tag{2.1}$$

where k_p – cutting stiffness [N/mm^2], A – area of cutting section [mm^2].

Reliable model of MT-CP system needs experimentally verified information about the cutting process. A series of cutting test experiments was designed and experiments were conducted in order to obtain the data for identification of the cutting process model. In the tests, the values of components of the cutting force in function of time were determined. Six-component 9265B Kistler dynamometer was used during these tests. They were conducted at different rotational speeds of the milling cutter and for different geometrical variants of the milling operation. First, the cutting parameters were set on the level, which guarantees stable machining. Then, the depth of cut was increased up to the value, when the system demonstrated the first symptoms of instability. Assessment of dynamic state of the system was conducted in several ways. During machining, the sound emission and the acceleration level of chosen elements of the MDS system were observed. This coarse estimation helped to secure machine tool elements and the tool against damage. After machining, the frequency spectrum of acceleration signals were determined. Observation of the level of the dominating component in these spectra, and setting the boundary value to the amplitude allows to determine the

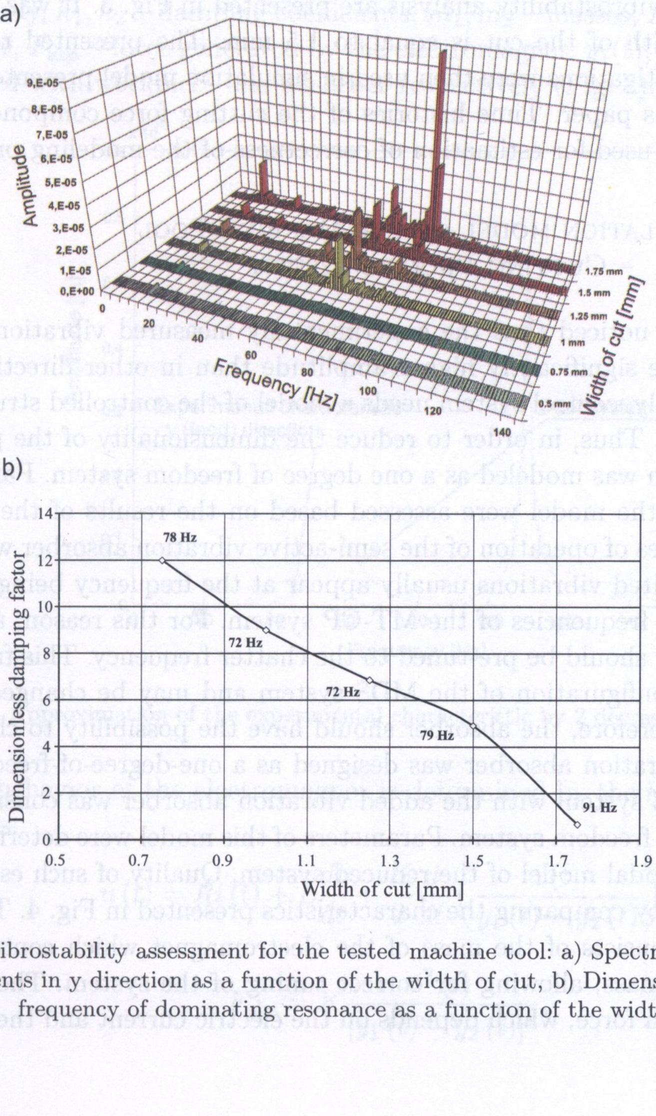

FIG. 3. Vibrostability assessment for the tested machine tool: a) Spectrum of the workpiece displacements in y direction as a function of the width of cut, b) Dimensionless damping and frequency of dominating resonance as a function of the width of cut.

limiting values of cutting parameters. Limiting parameters of cutting obtained with the help of this method are not very precise – there is a transitory area where an unambiguous estimation of the dynamic state of the cutting process is not possible – Fig. 3. This is why another approach was also proposed. Assuming that the acceleration signal, measured in the impulse tests, can be treated as the impulse response of the tested structure, it is possible to find a pole of the analyzed system that is responsible for the loss of stability. Damping connected with this pole should be negative but nonlinear nature of the structure causes that this damping is not negative. Its level is, however, significantly lower than damping connected with other poles. Estimation of frequency and damping of this pole can be performed by proper selection of parameters of digital filter, according to least square method [2].

Results of the vibrostability analysis are presented in Fig. 3. It was decided, that the limit width of the cut is equal to 1.5 mm. The presented results of experimental investigations were then used in simulation model presented in the then section of this paper. Time histories of the cutting force components and accelerations were used for estimation of correctness of the modeling process.

3. Simulation model of the Machine Tool – Cutting Process system

It was already noticed that the experimentally measured vibrations in the feed direction have significantly higher amplitude than in other directions (see Fig. 2). Additionally, control system needs a model of the controlled structure as simple as possible. Thus, in order to reduce the dimensionality of the problem, the MT-CP system was modeled as a one degree of freedom system. Parameters m_1, h_1 and k_1 of the model were assessed based on the results of the impulse tests. The principles of operation of the semi-active vibration absorber were then proposed. Self-excited vibrations usually appear at the frequency being close to one of the natural frequencies of the MT-CP system. For this reason, a passive vibration absorber should be pre-tuned to the chatter frequency. This frequency depends on the configuration of the MDS system and may be changed during machining [9]. Therefore, the absorber should have the possibility to change its parameters. A vibration absorber was designed as a one-degree-of-freedom system, and the MDS system with the added vibration absorber was considered as the two-degrees-of-freedom system. Parameters of this model were determined on the basis of the modal model of the reduced system. Quality of such estimation could be assessed by comparing the characteristics presented in Fig. 4. The mass of the absorber consists of the mass of the electromagnet which contains also a supplementary mass, allowing for correct tuning of the system. The electromagnet generates a force, which depends on the electric current and the air gap.

This enables us to change the stiffness through the control of the electromagnetic force (or current) and makes possible tuning of the vibration eliminator to the actual chatter frequency. Mathematical model of MT-CP system with the added absorber is described by the following equations:

$$\text{(3.1)} \qquad \ddot{y}_1(t) = \frac{1}{m_1}\Big[(-k_1 - k_2)\, y_1(t) - k_2 y_2(t) - (h_1 + h_2)\,\dot{y}_1(t) - h_2 \dot{y}_2(t) + F_{\mathrm{abs}} + F_{\mathrm{cut}} \Big],$$

$$\text{(3.2)} \qquad \ddot{y}_2(t) = \frac{1}{m_2}\Big[-k_2 y_1(t) + k_2 y_2(t) - h_2 \dot{y}_1(t) + h_2 \dot{y}_2(t) - F_{\mathrm{abs}} \Big],$$

where k_1, k_2 – spring coefficients of the elements modeling MDS and the absorber respectively, h_1, h_2 – damping coefficients, m_1, m_2 – masses, F_{cut} – machine cutting force, F_{abs} – force generated by the electromagnet, $y_1(t), y_2(t)$ – coordinates connected with workpiece and absorber respectively (Fig. 2).

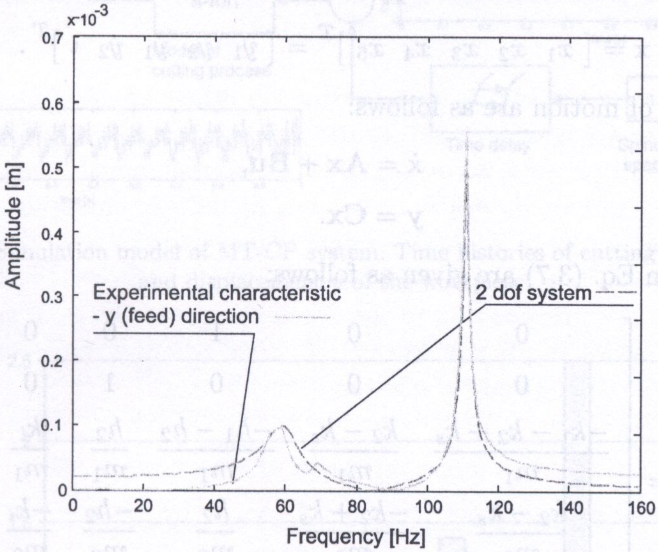

FIG. 4. Approximation of the experimental characteristic by 2 degree of freedom model.

The behavior of the electromagnet is determined by the following system of equations:

$$\text{(3.3)} \qquad u(t) = Ri(t) + L\frac{di}{dt} + \frac{K}{2}\frac{d}{dt}\left(\frac{i(t)}{y_1(t) - y_2(t)}\right),$$

$$\text{(3.4)} \qquad F_{\mathrm{abs}} = K\frac{i^2(t)}{[y_1(t) - y_2(t)]^2},$$

$$K = \frac{N_c \mu_0 A_r}{4} \left[\frac{\mathrm{Vsm}}{\mathrm{A}}\right], \tag{3.5}$$

where N_c – number of turns of the winding, A_r – cross-sectional area of the core [m^2], R, L, u – resistance, inductance and supply voltage respectively.

Parameters m_2, h_2 and k_2 of the vibration absorber were tuned during computational investigations. Then a design of a new tuned eliminator of vibration. To check if the parameters which were used during simulation investigations are correct, identification measurements of the absorber were conducted. The static characteristic, showing the generated force as a function of the air gap and the control current, was obtained during these experiments. Knowledge of this characteristic enables the linearization process and allows to estimate the range of needed and allowable changes of the control parameters.

Equations (3.1)–(3.3) can be rewritten in the state-space notation. Displacements and velocities of the workpiece y_1, dynamic absorber y_2 and the control electric current were chosen as the state variables:

$$\mathbf{x} = \begin{bmatrix} x_1 & x_2 & x_3 & x_4 & x_5 \end{bmatrix}^T = \begin{bmatrix} y_1 & y_2 & \dot{y}_1 & \dot{y}_2 & i \end{bmatrix}^T. \tag{3.6}$$

Equations of motion are as follows:

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u}, \\ \mathbf{y} &= \mathbf{C}\mathbf{x}. \end{aligned} \tag{3.7}$$

Matrices in Eq. (3.7) are given as follows:

$$\mathbf{A} = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 \\ \dfrac{-k_1 - k_2 + k_s}{m_1} & \dfrac{k_2 - k_s}{m_1} & \dfrac{-h_1 - h_2}{m_1} & \dfrac{h_2}{m_1} & \dfrac{k_i}{m_1} \\ \dfrac{k_2 - k_s}{m_2} & \dfrac{-k_2 + k_s}{m_2} & \dfrac{h_2}{m_2} & \dfrac{-h_2}{m_2} & \dfrac{-k_i}{m_2} \\ 0 & 0 & \dfrac{2k_i}{L_0} & \dfrac{2k_i}{L_0} & \dfrac{2R}{L_0} \end{bmatrix};$$

$$\mathbf{B} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ \dfrac{1}{m_1} & 0 \\ 0 & 0 \\ 0 & \dfrac{1}{L_0} \end{bmatrix}; \qquad \mathbf{C} = [1]. \tag{3.8}$$

where:

$$L_0 = \frac{2K}{y_1(t) - y_2(t)} \left[\frac{\mathrm{Vs}}{\mathrm{A}}\right],$$

$$k_i = \frac{2Ki_0}{(y_1(t) - y_2(t))^2} \left[\frac{\mathrm{N}}{\mathrm{A}}\right], \qquad k_s = \frac{2Ki_0^2}{(y_1(t) - y_2(t))^3} \left[\frac{\mathrm{Vs}}{\mathrm{m}^2}\right].$$

Model of the MT-CP system used during simulations is presented in Fig. 5. Simulations were carried out for different rotational speeds of the milling cutter and different variants of the cutting operation.

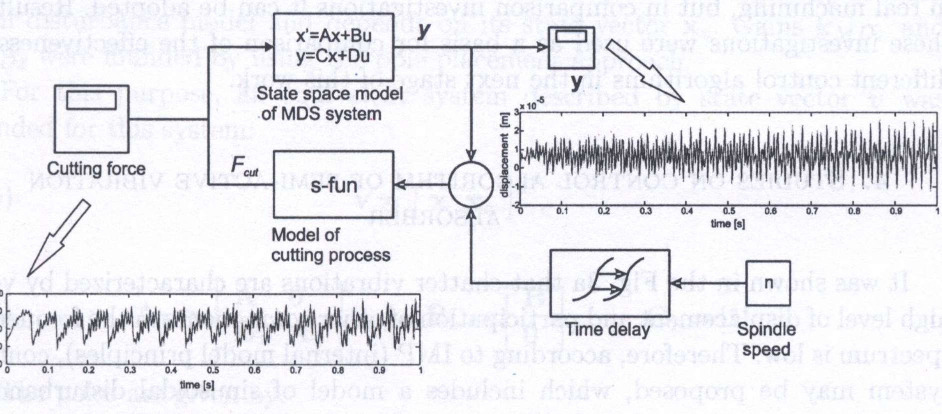

FIG. 5. Simulation model of MT-CP system. Time histories of cutting force F_{cut} and displacement y of the workpiece.

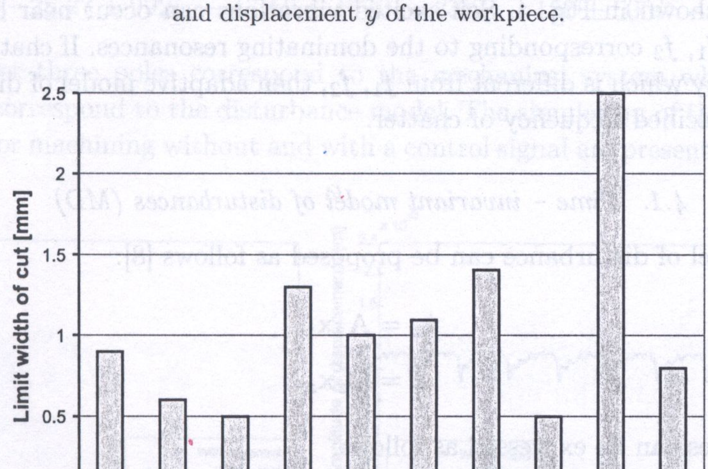

FIG. 6. Limit width of cut calculated from the model of MT-CP system for a range of cutter speeds.

The output of the "Cutting process" block in the presented diagram estimates the value of the cutting force on the basis of parameters of the cutting conditions, geometry of the tool and cutting arrangement (Fig. 6). It allows to detect moments when the tool and the workpiece are not in contact. A series of model computations made for each available from the gear-box rpm has been evaluated. The results agree with the data taken from experiments and are used when comparing the efficiency of the tested control algorithms. Time histories of displacement signals were analyzed and limit values of cutting parameters were searched. Vibrostability of the process was marked by loosing of contact between the tool and the workpiece. It is obvious that such a situation cannot be accepted in real machining, but in comparison investigations it can be adopted. Result of these investigations were used as a basis for comparison of the effectiveness of different control algorithms in the next stage of this work.

4. Studies on control algorithm of semi-active vibration absorber

It was shown in the Fig. 3a that chatter vibrations are characterized by very high level of displacement and participation of other components in the frequency spectrum is low. Therefore, according to IMP (internal model principles), control system may be proposed, which includes a model of sinusoidal disturbances. Dynamic characteristic of the MDS system of machine tool with added vibration absorber is shown in Fig. 4. Self-excited vibrations can occur near one of the frequencies f_1, f_2 corresponding to the dominating resonances. If chatter occurs at a frequency which is different from f_1, f_2, then adaptive model of disturbance is used to specified frequency of chatter.

4.1. Time – invariant model of disturbances (MD)

The model of disturbance can be proposed as follows [8]:

$$\begin{aligned} \dot{\mathbf{x}}_z &= \mathbf{A}_z \mathbf{x}_z, \\ \mathbf{z} &= \mathbf{C}_z \mathbf{x}_z. \end{aligned} \tag{4.1}$$

State matrices can be expressed as follows:

$$\mathbf{A}_z = 2\pi \begin{bmatrix} 0 & -f_1 & 0 & 0 \\ -f_1 & 0 & 0 & 0 \\ 0 & 0 & 0 & -f_2 \\ 0 & 0 & f_2 & 0 \end{bmatrix} \quad \mathbf{C}_z = [\mathbf{1}]. \tag{4.2}$$

And state vector is given in the form:

$$\mathbf{x}_z = \mathrm{col}\left[\ \cos(2\pi f_1 t)\ \ \sin(2\pi f_1 t)\ \ \cos(2\pi f_2 t)\ \ \sin(2\pi f_2 t)\ \right]. \tag{4.3}$$

Designed compensator is defined by:

$$\begin{aligned} \dot{\mathbf{x}}_z &= \mathbf{A}_z\mathbf{x}_z + \mathbf{B}_z\mathbf{x}, \\ u &= \mathbf{k}_{MDx}\mathbf{x} + \mathbf{k}_{MDz}\mathbf{x}_z. \end{aligned} \tag{4.4}$$

The control input u is a sum of two signals. The first one depends on $\mathbf{x}$ and provides the ability of pole placement. The second one is an actual signal taken from disturbance model and depends on its state vector $\mathbf{x}_z$. Gains $\mathbf{k}_{MDx}$ and $\mathbf{k}_{MDz}$ were founded by using the pole placement approach.

For this purpose, an equivalent system described by state vector $\boldsymbol{v}$ was founded for this system:

$$\mathbf{v} = \left[\ \mathbf{x}\ \mathbf{x}_z\ \right]^T, \tag{4.5}$$

$$\mathbf{A}_1 = \begin{bmatrix} \mathbf{A} & \mathbf{0} \\ \mathbf{B}_z & \mathbf{A}_z \end{bmatrix}, \qquad \mathbf{B}_1 = \begin{bmatrix} \mathbf{B} \\ \mathbf{0} \end{bmatrix}, \qquad \mathbf{C}_1 = [\mathbf{1}]. \tag{4.6}$$

The poles are given by:

$$s = [-25.27 \pm 992i\ \ -17.59 \pm 467i\ \ -3.08\ \ -150 \pm 500i\ \ -150 \pm 923i]. \tag{4.7}$$

The first three poles correspond to the mechanical system while the last two poles correspond to the disturbance model. The simulation of the workpiece vibration for machining without and with a control signal are presented in Fig. 7.

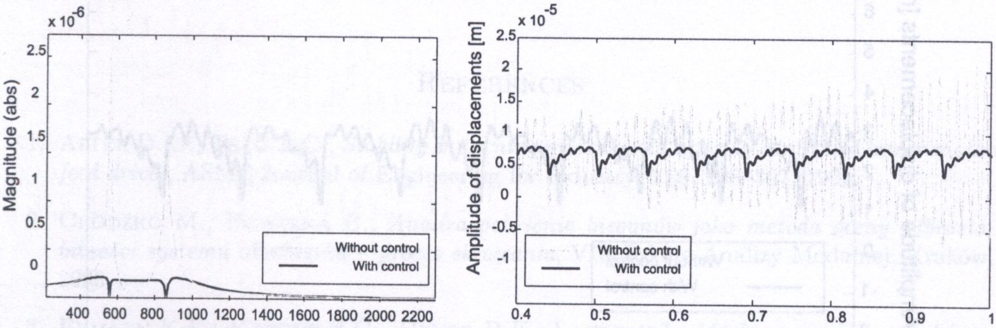

FIG. 7. Results of simulation of MT-CP system using the time – invariant model of disturbance control a) Bode diagram, b) time-dependent displacements of the workpiece.

4.2. Adaptive model of disturbances

One of the main phenomenons responsible for chatter in machine tools is the time delay between inner and outer modulation.

$$y_s(t) = y_1(t) - y_1(t-\tau) \approx A_s \sin(\omega t + \alpha), \tag{4.8}$$

where $\tau = 60/l_z n$, l_z is the number of edges of the mill.

The synchronous signal y_s, carries information on the current disturbance frequency ω. To counteract effectively the vibrations, the system with time-invariant model of disturbances has been supplemented by a model that matches itself automatically the dominant vibration frequency ω. The model employs naturally the signal y_s as a source of information about the current disturbance frequency. A disturbance model for one frequency can be written in the form of a transfer function [8]:

$$U_{III}(s) = \frac{k_{AMD1}s + k_{AMD2}\omega}{s^2+\omega^2} Y_1(s). \tag{4.9}$$

After applying the inverse Laplace transform to Eq. (4.9) the signal in time-domain for the adaptive model can be obtained.

$$\begin{aligned} U_{III} = {} & [k_{AMD1}\cos(\omega t+\alpha) + k_{AMD2}\sin(\omega t+\alpha)] \int_0^\infty \cos(\omega t+\alpha)\, y_1(\tau)\, d\tau \\ & + [k_{AMD2}\cos(\omega t+\alpha)] - k_{AMD1}\sin(\omega t+\alpha) \int_0^\infty - \sin(\omega t+\alpha)\, y_1(\tau)\, d\tau. \end{aligned} \tag{4.10}$$

Two synchronous signals shifted by $\pi/2$ must be available. The lacking signal is obtained by differentiating y_s and matching the magnitude to the y_s level.

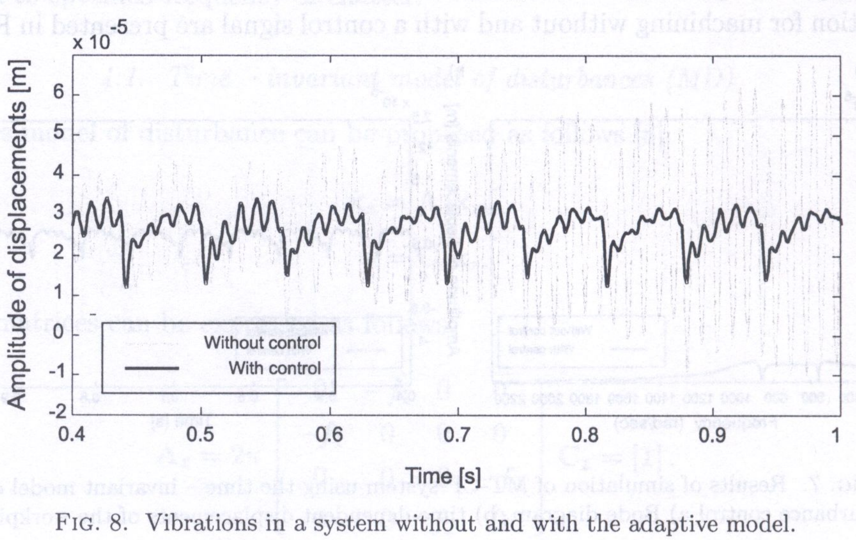

FIG. 8. Vibrations in a system without and with the adaptive model.

The values of k_{AMD1} and k_{AMD2} were chosen experimentally. Figure 8 shows vibrations of a workpiece where an automatically tuned model of sinusoidal disturbances (AMD) is used.

5. Conclusions

In the paper, an adaptive model of disturbance design methods for the semi-active absorber control was presented.

Both the presented control algorithms MD and AMD assure a significant increase of limit parameters of the cut, when the machining is stable. The final results of conducted investigations are presented in Table 1. In the next stage of work, experimental investigations of the presented control algorithms will be conducted.

Table 1. Results of effectiveness of proposed control algorithms.

Spindle speed	Width of cut [mm]		
	w/o control	for system controlled by	
[rpm]		MD	AMD
56	0.9	3.3	4
71	0.6	3.6	4.2
90	0.5	3.6	4.3
112	1.3	3.6	4.3
140	1	3.3	4.4
180	1.1	3.2	4.6
224	1.4	3.8	4.4
280	0.5	3.2	4.6
450	0.8	3	4.5

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