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Identification of dynamics for machining systems

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Abstract. *Cutting processes are carried out in an elastic machining system, which is multi-mass with negative and positive loop control with a delay in construed mathematical models. Its behavior during the cutting process is entirely determined by dynamic properties and an adequate parameters of mathematical model is necessary to control the process. The paper proposes a method for identifying such dynamic parameters of the machining system, which include natural vibration frequencies, vibration damping coefficients, and stiffness of the replacement model of single-mass system in the direction of the machine-CNC coordinate axes.*

It is proposed to identify such parameters as a result of experimental modal analysis by impacting the elements of the tool and workpiece with an impact hammer and processing the impulse signal with a fast Fourier transform. It is proposed to adapt the results obtained to the adopted mathematical model of the machining system, presented in the form of two masses, each with two degrees of freedom, according to the equivalence of the spectrum signal power or its spectral density. The cutting force model in the form of a linearized dependence on the area of undeformed chips needs to be clarified by the coefficient using experimental oscillograms obtained during milling of a workpiece mounted on a dynamometer table. Based on the identified parameters of the machining system, a stability diagram was constructed in the "spindle speed – feed" coordinates and experiments were carried out under conditions in the zone of stable and unstable cutting. Evaluation of the roughness of the machined surface confirmed the correspondence to the location of the stability lobes diagram constructed using the identified parameters, which indicates the effectiveness of the proposed identification method.

Keywords: *machining system, identification of dynamic parameters, experimental modal analysis.*

1. Introduction

The production of machine parts in subtractive technologies is carried out on metal-cutting machines during the cutting process, which is quasi-stationary. The trend towards increasing productivity by increasing the rate of stock removal and forming reveals serious problems associated with machining dynamics. These problems can be divided into two: the first is related to the dynamics of the execution of control program commands by the drives of a CNC machine tool, and the second is the dynamics of the cutting process itself in an elastic machining system. The article discusses the second problem, since it is the most general and affects the machining results on any machine.

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Dynamic phenomena during the cutting process always lead to the occurrence of regenerative vibrations, which are the main reason for limiting productivity. Despite the importance of such problems, the dynamics of cutting processes remains poorly understood, which leads in practice to the use of empirical methods for determining acceptable machining parameters that ensure stability.

All processes occurring in an elastic machining system are analog in nature, and to analyze them using modern signal machining methods, it is desirable to obtain their digital twin. Therefore, to control the process in order to suppress regenerative oscillations, it is necessary to focus on a mathematical model that adequately describes the real processes occurring during cutting in a dynamic system. An analysis of research in this area shows that it is desirable to obtain the result with the simplest model with a minimum number degrees of freedom.

This choice always involves a trade-off between model complexity and representation accuracy. Moreover, practice shows that increasing the complexity of the model

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by increasing the number degrees of freedom does not always lead to increased accuracy. Therefore, models with one or two degrees of freedom are most often used, although the representation of the cutting process between the interaction of the tool system and the workpiece system requires a more complex two-mass model, each with two degrees of freedom [1].

After choosing the model structure, it is necessary to identify its parameters, which include the natural frequency of each mode, stiffness in the direction of the shaping axis, vibration damping coefficient and cutting force coefficients [2]. For identification, experimental methods of modal analysis are used, which usually give frequency responses that differ from the accepted structure of the model. Therefore, it is necessary to develop a way to adapt the experimental results to the adopted model, both in frequency responses and in stiffness and damping. This article is devoted to solving these problems that are relevant for designing management strategies.

2. Problem status analysis

System identification is the process of creating a mathematical model of a dynamic system based on measured data. In the identification process, at the first stage, the structure of the system, the type and order of differential equations connecting input and output quantities are adopted in accordance with its physical structure. Next, experiments are carried out to determine the parameters of the adopted structure and mathematical description. The next step is to check the resulting model for adequacy, and if the answer is insufficient, the process is repeated, starting with clarifying the structure. The process is completed when model verification produces acceptable results. Common estimation methods are least squares, instrumental variable, maximum likelihood, etc. [3]

To identify dynamic systems, the most widely used methods are modal analysis, which, based on experimental results, make it possible to determine the structure and dynamic parameters of the system [2], [3]. Such results are presented in the form of frequency responses, which are the response of a dynamic system to kinematic or force excitation by a harmonic signal. The actual task is usually to determine the Frequency Response Function (FRF) for the system of interest and then define the model by performing a modal approximation of the measured data.

To solve this problem, the machining system is usually loaded using an electrodynamic vibrator or a special impact hammer. When loaded with a vibrator, a force is created, the frequency of which is changed in the required range and the response of the system is recorded in the form of elastic movement of the machine unit of interest. The Frequency Transfer Function (FTF) is determined, for example, in the form of a Nyquist diagram on the complex plane or in the form of the real and imaginary parts of the frequency function combined in frequency [4].

Such experiments require complex equipment and cannot be implemented for all types of processing and machine configurations. Therefore, a more flexible method of experimentally determining frequency responses using an impact hammer is used. The impact hammer creates an impulse effect on the system in the desired direction, and the reaction is recorded in the form of an Impulse Frequency Response (IFR).

There are several approaches to finding modal parameters. Experimental modal analysis (EMA) has proven its effectiveness when the machine is stopped. The excitation achieved using an impulse hammer or shaker, as well as the excitation response at several locations, are measured. Alternatively, operational modal analysis (OMA) can be used to determine modal parameters during operation. Here, only the arousal responses resulting from work are measured. Modal parameters are mathematically identified based on the measured signals in both cases, but using different methods.

In paper [5] discusses the extent to which both approaches (EMA and OMA) can lead to reliable identification of machine natural frequencies during milling. For EMA, the focus is on capturing perturbation. It can be assumed that procedural forces are the most significant perturbation. However, in addition to process forces, there are other sources of excitation (for example, actuators, hydraulic and pneumatic units), which, according to this assumption, are considered part of the disturbances that have consequences for the identification of modal parameters.

The dynamics of machine tools can differ significantly from the results obtained by traditional static EMA, which leads to the demand for a method of operational identification. However, due to the lack of the ability to measure the input excitation, it is impossible to obtain the frequency response, which in the classical sense is the ratio of the output signal to the input signal as a function of frequency. Therefore, a method for estimating FRF for OMA based on random cutting excitation methods is proposed [6]. The cutting force data is used, which is the input signal, based on the results of the process simulation. In such an experiment, the cutting speed will change and in order to evaluate this effect on the cutting force, a special function is introduced into the forecast. The simulated resulting cutting force is then synchronized with the measured responses before it is used to estimate the FRF.

Realizing the importance of the dynamic properties of a machine tool for accurate process modeling and chatter prevention, most researchers come to the need to determine frequency responses using OMA. However, the problem of identifying the input excitation as a function of the cutting process remains. Therefore, a modified OMA based on transfer function is proposed, which assumes a constant mode shape for the spindle assembly to monitor changes in the natural frequency and damping ratio of the spindle. The solution based on the transfer function only considers the relative vibrations between two sensors mounted on the spindle body when the machine tool is excited by cutting forces during machining [7].

Although real-world machining systems have multiple degrees of freedom (MDOF) and some degree of nonlinearity, they can usually be thought of as a superposition of linear single degree of freedom (SDOF) models [8]. To accurately predict the dynamic behavior of their real-world counterparts, these models need to be identified, meaning that the values of the physical model's involved parameters must be found by comparing the model with the measured data of its real-world counterpart. However, procedures for bringing to equivalence MDOF with SDOF which are in mathematical model have not been developed.

To identify machining systems using OMA methods, it is proposed to measure the frequency response using periodic test signals [9]. Typically, sinusoidal signals of fixed frequencies are used, but other periodic signals can also be used, such as e.g. rectangular, trapezoidal or triangular signals.

To identify the dynamics of machining systems, it is mandatory to identify cutting force coefficients. Work [10] presents a generalized model for identifying the cutting force coefficient, applicable for processing both isotropic and anisotropic materials. The basis is a general mechanical model applicable to any machining operation. Cutting force coefficients are evaluated in the frequency domain, taking into account the dependence of fiber orientation in composites or the runout effect in isotropic alloys.

The goal of identification research is to develop digital models of all stages of parts production. The intersection of tool and workpiece along the tool path is evaluated in discrete steps, which are then used to calculate chip area, cutting load, torque, power and energy consumed by the machine tool, and detected the occurrence of vibration. The dynamics of the CNC system are included in the digital model to estimate the true tangential feed and machining cycle time [11].

Digital twins are one of the components of creating automatic machining modules and are already having a significant impact on the manufacturing industry [12]. Autonomous digital twin technologies presented also include vibration prediction and control, as well as adaptive feed rate control. The proposed Digital Twin machining system is implemented on a large-sized CNC machine designed for high-speed machining of aircraft parts.

When performing studies to identify the dynamic properties of machine tools, it must be borne in mind that during standard experiments the machine does not perform cutting, and its components do not move relative to each other. The actual spectrum and frequency range of these forces are unknown. Experimental data obtained from different types of tests clearly show the difference in dynamic compliance of the same machine tool when cutting and idling. When dynamic testing of machine tools using various types of external exciting devices, the conditions of real load and interaction of moving parts, including the cutting process itself and external sources of vibration, are not taken into account [13].

When using vibrators, it must be kept in mind that since it is attached to a structure, in experimental studies mass is added to the dynamic system being studied [14]. This leads to errors in determining frequency responses.

The review of research in the field of dynamics of machining systems shows that, as a rule, they are represented by second-order dynamic models with one or two degrees of freedom. However, practical results from EMA or OMA yield frequency responses that correspond to multi-mass dynamic systems [4]. Therefore, when modeling machining systems, the problem arises of adapting the obtained experimental results to a SDOF model, which will be equivalent to the real system in terms of the experimental frequency response. In addition, the problem arises of identifying the stiffness, vibration damping coefficient and cutting force coefficient. The article presents the authors' experience in solving such pressing problems of identifying the machining system of a milling CNC-machine.

3. The aim and objectives of the study

The purpose of this work is to create a methodology for identifying the dynamic parameters of machining systems using the example of a CNC milling machine, which is based on experimental modal analysis, experimental results of determining the components of cutting force, stiffness and reduction to an equivalent SDOF model, which will allow assigning a cutting mode that ensures maximum performance with minimal vibration.

To achieve the goal, it is necessary to solve the following problems:

– construction a diagram of an experimental modal analysis of the machining system of a milling machine in its representation in the form of a model with two masses, each of which has two degrees of freedom;

– develop a methodology for determining the dynamic parameters of a model based on the equivalence of its frequency response to the results of experimental modal analysis;

– conduct experimental testing of the developed methodology and check its effectiveness.

4. The study materials and methods

The object of the study is the machining system of a CNC milling machine, and the subject of the study is the dynamic properties of the system, on the basis of which a vibration-free cutting mode is determined, providing the highest possible productivity with the required quality. The cutting process is considered during its implementation in a machining system, when to determine its components it is necessary to use the dynamic responses of the elastic system and the results of EMA.

To implement the methodology for identifying the dynamics of the machining system, experimental modal research schemes were drawn up using an impact hammer

Impact Hammer Model 086D05, Multicomponent sensor MCS10, a dual-beam storage oscilloscope model XDS 3202E, accelerometer RSV 353B15, amplifier ClipX BM40 and corresponding signal processing programs and their representations in digital format. To process digital signals of modal tests, a special program was compiled, which provides Fast Fourier Transform (FFT) and determination of the natural frequency and oscillation damping coefficient of an equivalent SDOF dynamic system.

The identification results obtained were used to solve the problem of determining the vibration-free cutting mode during end milling on a CNC machine [15], which confirmed the effectiveness of the proposed identification technique.

5. Results of identification machining system for end milling

5.1. Model of machining system

A mathematical model of the milling process is necessary to calculate the vibration-free cutting mode, optimize the process and assumes the preliminary identification of all components, including the cutting process and the dynamic parameters of the elastic system. The mathematical model of the machining system for 2D milling must take into account the closedness of the cutting process, control by two inputs (cutting depth and feed) and machining along the traces. The cutting process involves two dynamic systems of the tool and the workpiece, which interact through the cutting process (Fig. 1).

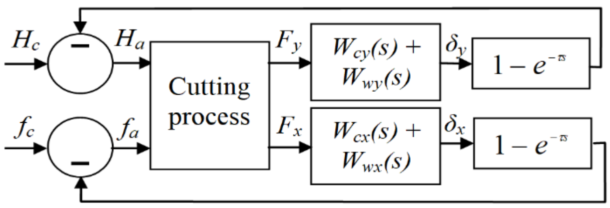


Fig. 1. Blok-diagram of machining system

Machining along the trace in two coordinates is represented by a delay link, where τ is the tooth passing period, s is the Laplace operator. The cutting process is described by dependencies that involve modeling by numerical methods [16]:

$$F_y = \sum_{j}^{z} \sum_{i=1}^{n} C_P a_i^k \delta B_i \mathrm{Cos}\left(\varphi + \varphi_z j - \gamma - \beta\right), \tag{1}$$

$$F_x = \sum_{j}^{z} \sum_{i=1}^{n} C_P a_i^k \delta B_i \,\mathrm{Sin}\left(\varphi + \varphi_z j - \gamma - \beta\right), \tag{2}$$

where C_P, k are the empirical coefficient and exponent, a_i is the cutting thickness, δB_i is the width of the elementary section along the Z coordinate, γ is the rake angle of the mill cutter flute, z is the number of cutter tooth, n is the number of sections along the milling width.

From formulas (1) and (2) it is clear that the calculation is performed by representing the cutter as a set of elementary cylindrical cutters along the milling width along the Z coordinate. The cutting thickness for each elementary cutter is determined by the well-known formula:

$$a_i = f_t \,\mathrm{Sin}\,\varphi_i \,, \tag{3}$$

where f_t is the feed per cutter tooth, φ_i is the angle of the cutting arc in the corresponding section.

To identify the cutting process, it is necessary to focus on the experimental results that are obtained when milling a workpiece installed on the Multicomponent sensor MCS10. Since the cutting force model is based on a mechanistic approach, only two values C_P, k need to be identified. Such experiments are constructed according to a well-known planning scheme with the results assessed by the average value of the cutting force, taking into account the intermittency of the cutting process [16].

It is advisable to represent the dynamic systems that make up the machining system in the form of SDOF elastic systems with one degree of freedom along each coordinate. In this case, the transfer functions of the cutter and workpiece system are represented at each coordinate.

Transfer functions of the “cutter” system:

$$W_{cy}\left(s\right) = \frac{1/k_{cy}}{\dfrac{s^2\delta_{cy}}{\omega_{cy}^2} + 2\xi_{cy}\dfrac{s\delta_{cy}}{\omega_{cy}} + \delta_{cy}},$$

$$W_{cx}\left(s\right) = \frac{1/k_{cx}}{\dfrac{s^2\delta_{cx}}{\omega_{cx}^2} + 2\xi_{cx}\dfrac{s\delta_{cx}}{\omega_{cx}} + \delta_{cx}}, \tag{4}$$

where k_{cy}, k_{cx} – rigidity of the system along the Y axis and X axis, ω_{cy}, ω_{cx} – frequency of natural vibrations along the Y axis and X axis, ξ_{cy}, ξ_{cx} – vibration damping coefficients along the Y axis and X axis, δ_{cy}, δ_{cx} – elastic displacements along the axis Y and X axis.

Transfer functions of the “workpiece” system:

$$W_{wy}\left(s\right) = \frac{1/k_{wy}}{\dfrac{s^2\delta_{wy}}{\omega_{wy}^2} + 2\xi_{wy}\dfrac{s\delta_{wy}}{\omega_{wy}} + \delta_{wy}},$$

$$W_{wx}\left(s\right) = \frac{1/k_{wx}}{\dfrac{s^2\delta_{wx}}{\omega_{wx}^2} + 2\xi_{wx}\dfrac{s\delta_{wx}}{\omega_{wx}} + \delta_{wx}}, \tag{5}$$

where k_{wy}, k_{wx} – rigidity of the system along the Y axis and X axis, ω_{wy}, ω_{wx} – frequency of natural vibrations along the Y axis and X axis, ξ_{wy}, ξ_{wx} – vibration damping coefficients along the Y axis and X axis, δ_{wy}, δ_{wx} – elastic displacements along the axis Y and X axis.

Equations of closure of the machining system according to Fig. 1:

$$\begin{cases} H_a = H_c - \left(\delta_{cy} + \delta_{wy}\right)\left(1 - e^{-\tau s}\right) \\ f_a = f_c - \left(\delta_{cx} + \delta_{wx}\right)\left(1 - e^{-\tau s}\right) \end{cases}, \qquad (6)$$

where H_a, H_c are the actual and commanded cutting depth in the Y coordinate, f_a, f_c are the actual and commanded feed in the X coordinate.

Thus, to identify the dynamic parameters of the machining system presented in (4) and (5), it is necessary to conduct an EMA and reduce the actual results to equivalent single-mass systems.

5.2. Experimental Modal Analysis

To carry out the modal analysis, an experimental technique was used based on the study of a dynamic system in a static state separately for the cutter system and the workpiece system using an impact hammer.

Experimental modal analysis of the workpiece system was performed on a workpiece mounted on the MCS10 dynamometer table (Fig. 2). An accelerometer is attached to the workpiece, connected through an amplifier to the input of the oscilloscope. The impact hammer is also connected to the second input of the oscilloscope, and the impact is performed in the direction of one of the coordinate axes. The result of recording the received signals on the oscilloscope screen is shown: the accelerometer signal is indicated by line 1, the impact hammer signal is indicated by line 2.

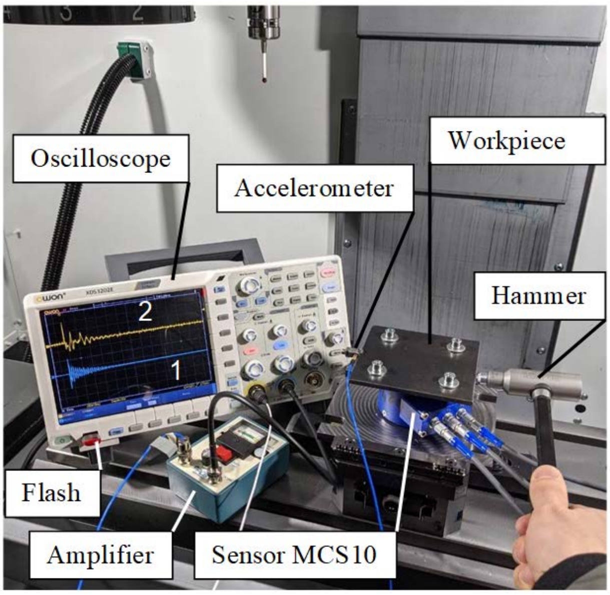
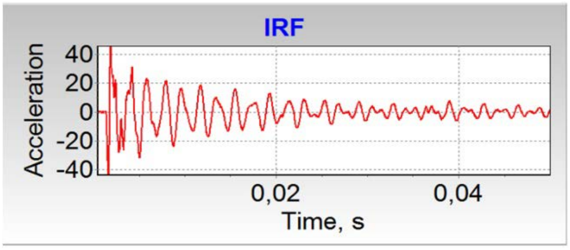
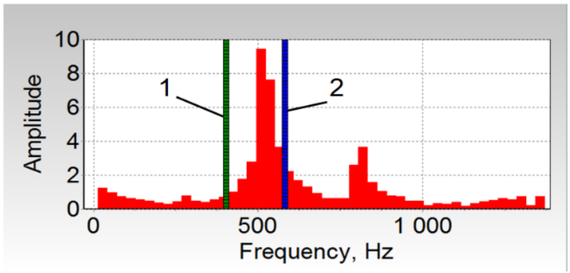


Fig. 2. Experimental setup for determining the IRF of the “workpiece” system

To determine the IRF in the direction of another axis, the location of the accelerometer and the direction of the impact are changed. This is how an experimental modal analysis of the “workpiece” system was carried out in a static state. The oscilloscope generates a digital file that can be saved to flash memory in *.txt format. Such a file provides sufficient information to identify a dynamic system (Fig. 3 *a*). In addition, using the second channel of the oscilloscope, you can observe the impulse response of the impact hammer, which is important for selecting a tip to excite the required frequency spectrum in the machining system under study.



a



b

Fig. 3. Results of the modal experiment system “Workpiece”: *a* – IRF; *b* – FRF

Using the FFT, a discrete spectrum of the TFT signal was constructed in the form of amplitudes of the frequency response (Fig. 3 *b*). It can be seen that the dynamic machining system is multi-mass and can be represented as a variety of single-mass systems. To identify the necessary parameters of a single-mass system adopted in the mathematical model (5), one can use the principle of energy equivalence. Energy of the discrete spectrum according to the results of EMA:

$$E = \sum_{i=1}^{n}\left(\omega_i^2 A_i^2\right)\Big/2\,, \qquad (7)$$

where ω_i, A_i is the frequency and amplitude of each harmonic in the discrete spectrum, n is the number of selected harmonics.

Then the frequency of natural oscillations of a single-mass system, equivalent by energy:

$$\omega_{SDOF} = \sqrt{\frac{2E}{A_a^2}}\,, \qquad (8)$$

where $A_a = \left(\sum_{i=1}^{n} A_i\right)\Big/n$ is the average amplitude of the discrete spectrum.

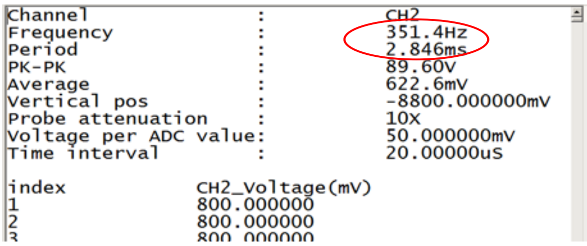
This frequency (402.3 Hz) for the “workpiece” system is shown by line 1 in Fig. 3 *b*.

In addition, given the somewhat stochastic nature of the process, it is useful to estimate the frequency of the replacement mass using the spectral density of the power signals. With this approach, the frequency of the replacement mass is:

$$\omega_{SDOF} = \frac{2S}{\pi Am_a}, \quad (9)$$

where $S = \sum_{i=1}^{n} \omega_i Am_i$, Am is integer part of the amplitude of each harmonic, $Am_a = \sum_{i=1}^{n} Am_i$.

The frequency (581.2 Hz) determined by formula (9) for the "workpiece" system is shown by line 2 in Fig. 3 *b*. The final decision in the simulation can be made as an average, taking into account the third frequency value calculated by the oscilloscope and presented in the digital data file (Fig. 4). Given some discrepancies in the data, experiments must be carried out several times and the average result must be taken into the model.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Frequency               :        351.4Hz
Period                  :        2.846ms
PK-PK                   :        89.60V
Average                 :        622.6mV
Vertical pos            :        -8800.000000mV
Probe attenuation       :        10X
Voltage per ADC value:           50.000000mV
Time interval           :        20.00000uS

index           CH2_Voltage(mV)
1               800.000000
2               800.000000
3               800.000000
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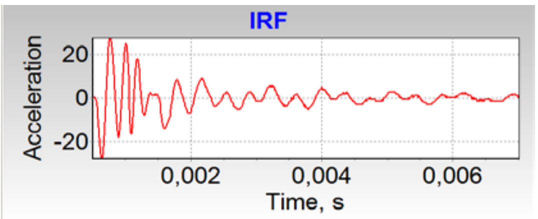
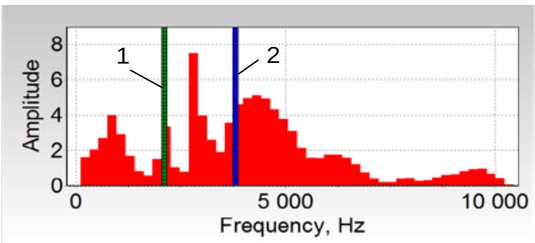
Fig. 4. Digital impulse response file

For the experimental modal analysis of the "cutter" system, the accelerometer was mounted on a milling cutter, and the hammer was also struck in two directions to obtain impulse responses, which were used to perform a modal analysis of the dynamic system (Fig. 5).



a



b

Fig. 5. Results of the modal experiment "cutter" systems: *a* – IRF; *b* – FRF

As one would expect, the spectrum has shifted to the region of high frequencies, and the frequencies of the replacement equivalent mass for model (4) are determined by the same formulas as for the "workpiece" system. The frequency based on spectrum energy equivalence is 2103.4 Hz (line 1 in Fig. 5 *b*), and the frequency calculated from spectral density is 3803.4 Hz (line 2 in Fig. 5 *b*).

Oscillation damping coefficients are calculated from the impulse response using the well-known formula [15]:

$$\xi = \frac{\ln\left(A(t)/A(t+T)\right)}{T}, \quad (10)$$

where $A(t)$ is the amplitude of the IRF at time t, $A(t+T)$ is the amplitude of the IRF at time $t+T$, where T is the period.

To identify a dynamic machining system, it is necessary to determine the stiffness of both the "cutter" system and the "workpiece" system (see formulas (4) and (5)). The determination of all stiffnesses was carried out experimentally according to the scheme for measuring the stiffness of the "cutter" system, presented in Fig. 6. For measurements, a dial indicator with a division value of 0.001 mm is used, which measures the elastic displacement on the cutter. Loading is performed by manually moving the CNC system of the machine tables through the encoder when the cutter is in contact with the workpiece during transverse (*X*-axis) and longitudinal (*Y*-axis) feed. The magnitude of the force is indexed on the computer screen to which the outputs of the dynamometer amplifiers are connected.

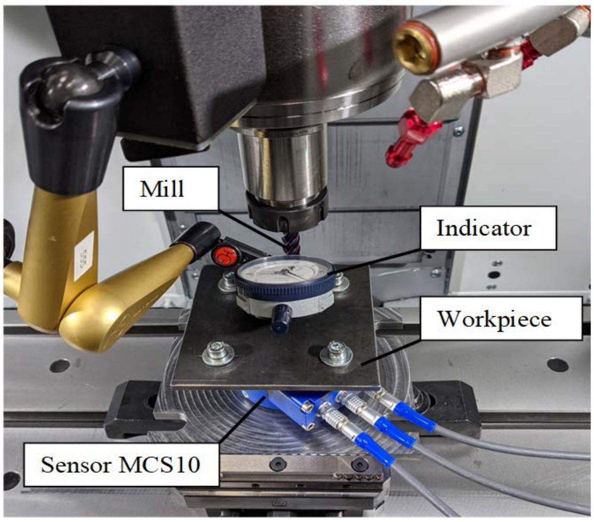


Fig. 6. Measuring schemas of workpiece stiffness

Loading was carried out to a force of 1000 N in both directions, while the stiffness was calculated using a linear relationship as the ratio of force to elastic displacement recorded on the indicator. To measure the stiffness of the "workpiece" system, the indicator was placed in contact with the workpiece, and loading was performed according to the same scheme.

Identification of the entire machining system in accordance with the block diagram of Fig. 1 also involves determining the coefficient in the cutting force formula. To do this, you can first use the data from reference books [16] with subsequent refinement based on the results of experiments [17].

5.3. Approbation of results

The results of the identification of the machining system were used in modeling the milling process to determine the chatter-free cutting mode in the end milling operation. The chatter-free mode was determined from the stability lobes diagram, which was designed automatically, similar to the procedure presented in [18]. In the stability lobes diagram, part of which is shown in Fig. 7, the cutting modes corresponding to the chatter-free mode (points 1 and 3) and the mode in the unstable cutting zone (point 2) are selected. All areas are processed at the same feed rate of 700 mm/min, but with different spindle speeds: point 1 – 900 rpm, point 1 – 1400 rpm, point 1 – 1800 rpm.

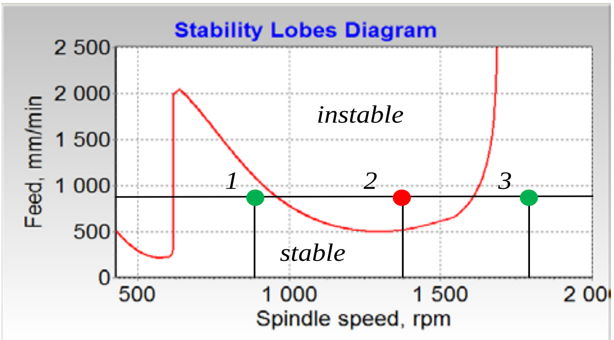


Fig. 7. Experimental modes on the stability diagram

In such modes, one side of a square workpiece installed on the machine table along with a dynamometer was mashined (Fig. 2, Fig. 6) to comply with all the conditions under which identification was carried out. Cutting was carried out in three sections of the workpiece side with stops to change the mode. The vibration level was assessed based on the roughness of the treated surfaces. The experimental results are presented in Fig. 8, which shows a photo of the machined side of the workpiece and profilograms of the corresponding surfaces.

It is known that the roughness profile of a machined surface consists of deterministic and random components. The deterministic component is determined by the geometric interaction of the cutter blade and the workpiece in the process of relative movements in accordance with the cutting mode, and the random component (chatter) is determined by the level of vibration. The results of profile measurements and evaluation of the Ra parameter show that when machining with a cutting mode corresponding to point 2 (unstable cutting area), an anomalous increase in this parameter is observed (Fig. 8). So, if we assume a linear dependence of Ra on the spindle speed, then when machining with mode at point 2, this parameter should be 2.75 μm, but in fact it is 1.5 times larger. This effect entirely depends on the location of the diagram on the "spindle speed – feed" plane and is completely determined by the dynamic parameters of the machining system. Thus, the adequacy of the identification studies performed has been experimentally proven.

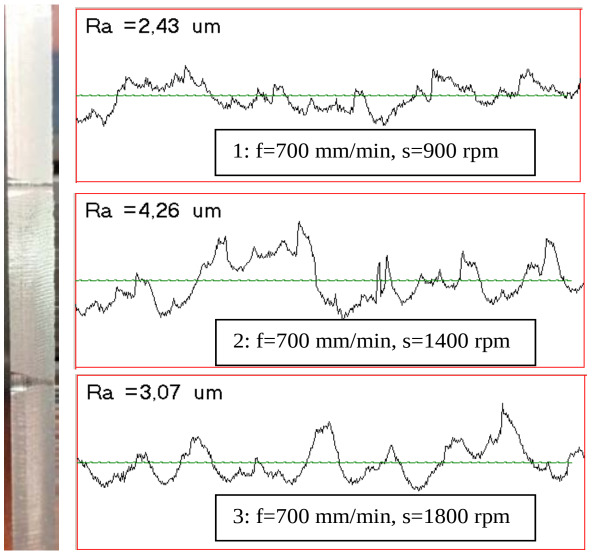


Fig. 8. Roughness of machined surfaces

6. Discussion on identifying the dynamic parameters of the processing system

The control strategy for end milling, like any other cutting process, is based on a mathematical model of the processing system. The mathematical model represents a closed elastic system with negative feedback for elastic shear and positive feedback for lagging argument for depth of cut and feed. An elastic system is usually composed of single-mass dynamic systems [2], [3], interconnected, and the cutting process itself is described by a mechanistic model [17].

All machining systems are prone to the occurrence of regenerative vibrations, which hinders the increase in productivity at the required quality [1] and one of the most effective control methods is to ensure a chatter-free cutting mode based on the stability lobes diagram (SLD). The SLD divides the entire range of possible values of spindle speed and feed into two zones: stable and unstable cutting. The required cutting mode in a stable zone is easily realized when machining on a CNC machine.

However, to construct a stability diagram, it is necessary to identify the following parameters of the machining system: frequency of natural vibrations of the single-mass model, stiffness, vibration damping coefficient, and coefficient of the linearized cutting force model. All these parameters must be identified in the direction of two or three axes, depending on the 2D or 3D milling scheme.

The greatest difficulty is in identifying the natural frequency of oscillations of the replacing equivalent single-mass model. Manipulation procedures using the experimental modal analysis method make it possible to solve such a problem [4], [6], [7]. Since the machining system model is represented as two masses, each with two degrees of freedom, interacting through the cutting process, modal analysis was performed for each mass separately. The experiment is carried out using an impact hammer and an accelerometer attached to the system under study, and the impulse response was recorded in a storage oscilloscope.

The main task is to adapt the resulting frequency response to a single-mass model. To solve this problem, two methods are proposed in the work: to carry out adaptation according to the power of the experimental signal and according to its spectral density (8) and (9). In addition, the oscilloscope automatically determines the frequency of the main harmonic during FFT of the IRF. As practice shows, all these three frequencies differ from each other and depend on the impact conditions, therefore, to obtain reliable information on modal analysis, it is necessary to carry out the experiment multiple times to obtain a stable result.

When measuring stiffness, an original testing method was proposed directly using the CNC system control of the machine. The system was loaded by drives along each coordinate using manual displacement encoders, and the force was recorded by a dynamometer. The stiffness is determined come relation entre this values.

The presented study uses the method of experimental modal analysis and requires the use of fairly complex measuring equipment and computer programs. Therefore, in practice, the use of such a progressive method for determining the cutting mode from the stability diagram is hindered. The authors see the development of this technique in the use of OMA [5], as well as in the creation of a special program for automatic on-line determination of all those necessary for modeling the dynamic parameters of the machining system. Such a program built into the computer of the CNC machine rack will be able to quickly adjust the cutting mode assigned during preparation of the process in the direction of increasing productivity with the required quality.

7. Conclusion

1. A technique has been developed for identifying the dynamic parameters of the machining system, which provides for the experimental determination of natural oscillation frequencies, oscillation damping coefficients, and stiffnesses of the "cutter" and "workpiece" systems in the direction of all coordinate axes. A method is also proposed for measuring the stiffness of the system when it is loaded by machine drives with control of manual movement encoders and recording of the force by a dynamometer installed on the machine table. The coefficient of the linearized dependence of the cutting force on the area of undeformed chips is refined based on the experimental results of the components of the cutting force measured by a dynamometer.

2. It is proposed to adapt the frequency responses obtained as a result of EMA and FFT to the parameters of a single-mass model with two degrees of freedom. An adaptation approach is presented based on the equivalence of the model and the real system in terms of spectral power and its spectral density.

3. The adequacy of the proposed method was confirmed as a result of milling the workpiece with modes corresponding to stable and unstable cutting, which were determined from a stability diagram constructed with system parameters identified using the developed method. Measurements of the roughness of the machined surfaces fully confirmed the validity of the proposed solutions: the roughness of the surface machined under the regime in the unstable zone of the diagram showed an excess of 1.5 times from the expected value compared to the roughness of the surfaces machined under the regime in the stable cutting zone.

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Ідентифікація динаміки обробних систем

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Анотація. *Процеси обробки різанням здійснюються в пружній обробній системі, яка є мультимасовою з негативними та позитивними із запізненням зворотними зв’язками за заданими режимами різання. Її поведінка у процесі різання цілком визначається динамічними властивостями, а для управління процесом необхідна адекватна математична модель. У роботі запропоновано методику ідентифікації таких динамічних параметрів обробної системи, до яких відносяться частоти власних коливань, коефіцієнти загасання коливань, жорсткості заміщаючої моделі в напрямку осей координат верстата. Такі параметри запропоновано ідентифікувати в результаті експериментального модального аналізу впливом на елементи інструменту та заготовки ударним молотком та обробці імпульсного сигналу швидким перетворенням Фур’є. Отримані результати запропоновано адаптувати до прийнятої математичної моделі обробної системи, представленої у вигляді двох мас, кожна з двома ступенями свободи, за еквівалентністю потужності сигналу спектра або його спектральної щільності. Модель сили різання у вигляді лінеаризованої залежності від площі недеформованої стружки потребує уточнення коефіцієнта за експериментальними осцилограмами, отриманими при фрезеруванні заготовки, встановленої на столі динамометра. За ідентифікованими параметрами обробної системи було побудовано діаграму сталості в координатах “швидкість шпинделя – подача” та проведені експерименти при режимах у зоні сталого та несталого різання. Оцінка шорсткості обробленої поверхні підтвердила відповідність розташування діаграми, побудованої за ідентифікованими параметрами, що свідчить про ефективність запропонованої методики ідентифікації.*

Ключові слова: *система обробки, ідентифікація динамічних параметрів, експериментальний модальний аналіз.*