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THE SEARCH FOR WEAK ELEMENTS AFFECTING THE VIBROSTABILITY OF THE SYSTEM CONSISTING OF THE MACHINE TOOL AND THE CUTTING PROCESS, BASED ON SYMPTOMS OBSERVED DURING OPERATION

Key words

Machine tool, modal analysis, chatter, stability.

Summary

The paper presents a methodology for the search of the “weak element” in a dynamic system, consisting of the machine tool and the cutting process, using the methods of experimental modal analysis. The subject matter of this research was inspired by specific machining problems that occurred in the industrial operation of machine tools. It became necessary to identify the reasons of the loss of stability, establish the self-excited vibrations mechanism, indicate the “weak element” in the mass-spring-damping system, then specify the required changes in design, and verify them in practice. The paper also presents examples showing the practical effectiveness of the proposed research method.

Introduction

Modern machine tools are designed according to the strictly regulated operational criteria. This applies to both universal machine tools, as well those designed to perform specific tasks. The variety of cutting variants achieved by universal machine tools is significant. This translates into a significant number of specific configurations of body elements of the machine tool. It also makes it necessary to use a large number of various cutting tools and technological parameters of cutting. These factors, in particular their variability, significantly affect the dynamic properties of the system, consisting of the machine tool and the cutting process (MT-CP), whose block diagram is shown in Fig. 1.

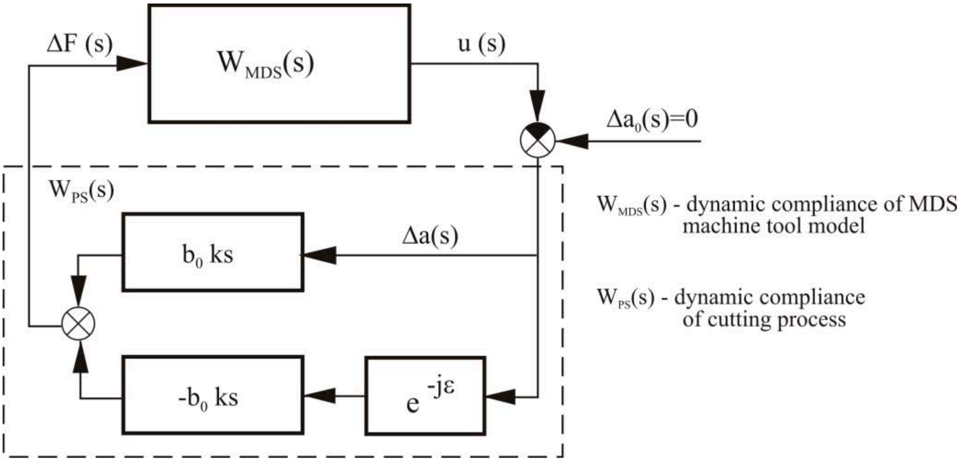

Fig. 1. Block diagram of the dynamic system consisting of the machine tool and the cutting process, including the regeneration effect [14]

Vibrostability is one of the basic operational criteria of the machine tool. The occurrence of self-excited vibrations, regardless of its source, disqualifies the machine tool and makes it impossible to perform cutting. There are several mechanisms behind this type of vibrations. Regenerative self-excited vibrations, due to the variability of geometric parameters of the cut, particularly the depth of cut, caused by the wave created on the cutting surface in the previous rotation of the tool or workpiece (regeneration effect) [1, 9, 12, 14]. The variability of these parameters results in the variation in cutting force, which under specified conditions, outcomes in the formation and development of the relative vibration of the tool and the workpiece. The maximum depth of cut, taking into account the regeneration effect, can be determined based on the following relationship [14]:

$$b_0\, k_s\, \mathrm{W_{MDS}}(j\omega) - b_0\, k_s\, \mathrm{W_{MDS}}(j\omega) e^{-j\varepsilon} = -1$$

where

$$b_{0lim} = -\frac{1}{2\, k_s\, \mathrm{Re_{MDS\,min}}}$$

where

b_0	–	limit width of cut,
k_s	–	cutting stiffness,
$\mathrm{W_{MDS}}$	–	dynamic compliance of machine tool MDS (mass--damping-spring) model,
$\mathrm{Re_{MDS\,min}}$	–	minimum of the real part of the dynamic compliance for a machine tool MDS model.

Parametric self-excited vibrations are very difficult to predict; it results from undesired configuration of the dynamic properties of the machine tool MDS system, reduced to the point of contact between the tool and the workpiece. The direction of the resultant cutting force can correspond with the main axis of stiffness ellipsoid, thereby resulting in high amplitude vibration [9, 12, 14].

The aforementioned mechanisms of self-excited vibrations are shown in Fig. 2. They can occur independently but also at the same time [14]. In the scientific literature, one can also find information about the frictional and thermo-mechanical self-excited vibrations, although they are much less frequent [14, 15].

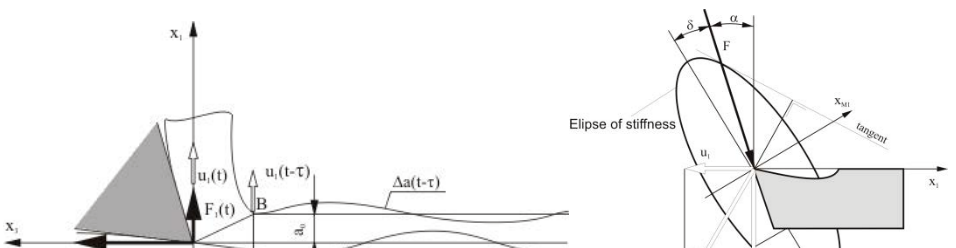

Fig. 2. The mechanism of self-oscillation: (a) due to the regenerative effect; (b) due to mode coupling effect, based on the example of turning [14]

The activation self-excited vibrations in the MT-CP system may result from the presence of a "weak element" caused by:

- Design or construction error,
- Assembling error,
- Improper mounting of the workpiece in a chuck,

- The use of interchangeable parts with incorrectly selected dynamic properties,
- The use of tools with significant overhang (high compliance),
- The high compliance of the workpiece (thin walled work-piece),
- Changes in the dynamic properties of the machine tool due to the wear of its elements during normal operation, and
- Deliberate changes to the MDS system of the machine tool, not supported by relevant model studies.

A common feature of mentioned reasons is that they cannot be identified before the production of the machine and its use in industrial conditions. One solution to this problem could be the performance of large-scale approval testing after the production of the machine tool. However, it seems economically unreliable, given the variability of factors affecting the stability of machining. As a result, in the case of the aforementioned negative factors, the user is forced to perform a number of actions designed to achieve stable machining conditions. There are many options here. First of all, it is possible to select appropriate parameters of cutting which would ensure the stability of the machining. The effective use of this method, however, requires the knowledge of the position of the stability lobes [13], enabling the selection of the optimal rotational speed of the tool or workpiece and the depth of cut. This, in turn, requires the implementation of appropriate procedures related to the modelling of the cutting process and the dynamics of the MDS model of the machine tool [3, 4, 15], which is usually not possible for machine tool users. Another method is to optimize the tool path, which would minimize the cutting force generated during machining. This is now made possible by advanced computer-aided manufacturing systems. One can also use tools with built-in dampers or various “smart” solutions, for example, active chuck systems [11]. However, they are usually very expensive and do not solve the problem completely. Besides the previously mentioned methods, it is very important to detect a “weak element” in the MT-CP system revealed during the operation of the machine and then to remove its negative influence. Therefore, it is reasonable to develop effective research methodology, enabling the discovery and localization of the “weak element” in the MT-CP system. It should be followed by the indication of the actions aimed at restoring the vibrostability of the machine tool.

1. Methodology of a „weak element” determination in the MT-CP system in operating conditions

A modern machine tool is a complex mechatronic system. Numerous interactions between its components and functional units influence its performance and determine its range of use. If the stages of design and construction involve computer modelling and a comprehensive simulation analysis, then there is a good chance of obtaining satisfactory characteristics of the designed MT-CP system. If this is also accompanied by the identification research in prototypes, it is possible to make appropriate changes in the design of the machine tool, improving its dynamic properties and thus its stability.

There is a considerable volume of research on the mathematical modelling of the MT-CP system [4, 9, 10], experimental research in the field of static [2, 6] and dynamic properties [5], and predicting the vibrostability [7, 8], and other aspects of machine tool use. Despite this, very often the stages of design and construction of machine tools are carried out intuitively, based on the engineering experience of the designer. This approach to design does not always lead to optimal solutions, and if changes in design are necessary, it is not possible to predict their effect. Therefore, modelling and computer simulations play an important role in shaping the dynamic properties of the MT-CP system at the design stage.

However, in practice, information on the dynamic properties of the machine tool in operation are not known to the user, and the cause in the problems in machining, including self-excited vibrations, needs to be identified fast. This situation occurs mainly when the machine loses its desired properties in operation, due to the occurrence of certain factors. In this case, the user needs to take specific actions focused on removing the causes of the instability of MT-CP system, in the shortest time and lowest cost possible. When assuming that the physical and mathematical model of the machine tool in question is not available, it is proposed to use experimental modal analysis to identify the "weak element" in the MDS system as the cause of the formation and development of self-excited vibrations in the MT-CP system.

The algorithm for identifying a "weak element" in the MT-CP system in operational conditions with the use of the experimental modal analysis is proposed as follows:

1. Analysis of factors associated with the machining technology:
   a. Verification of the selection of technological parameters of machining,
   b. Determination of machining cases at which vibrostability is lost,
   c. Validation of the clamping of the workpiece, cutting tool, its overhang,
   d. Verification of the correctness of the machining program.
2. Analysis of the MDS system in the machine tool:
   a. Analysis of the MDS system structure and the evaluation of its technical condition,
   b. The determination of design changes and modifications made prior to the difficulty of machining,
   c. Identification of possible failures and their sources that may have occurred during operation,
3. Identification of the self-excited vibration mechanism:
   a. Visual assessment of unstable machining marks on the surface of the workpiece,

   b. Determination of the frequency at which the machine tool vibrostability is lost,
   c. The determination of spatial configuration of the machine tool body components at which instability occurs
4. Plan of experiment:
   a. Analysis of the machine tool in terms of the selection of excitation type (impact test, use of exciters, operational modal analysis);
   b. The analysis of the machine tool body in order to perform a proper selection of the locations of measurement points;
   c. The choice of reference point, excitation points, and directions;
   d. The selection of proper transducers respect to response level and frequency range;
   e. Establishing the parameters of signal processing, availability of the algorithms of modal model estimation.
5. Experiment:
   a. Machining tests - estimating of self-oscillation frequency and amplitude of vibration levels on the selected elements of the MDS system;
   b. Comprehensive modal tests for all measuring points distributed on an object in order to determine the vibration modes and animation of the motion for each of the natural frequencies;
   c. Complementary tests, detailing the unambiguity of the identification of the “weak element” of the MDS system in the machine tool.
6. Conclusion:
   a. Comparative analysis of vibration amplitude levels on the selected elements of the machine tool body,
   b. Spatial interpretation of vibration including the significance of vibration amplitude levels,
   c. The creative interpretation of information acquired in the course of the entire test - the indication of the “weak element” and to proposing changes to the MDS system in the machine tool that will remove the cause of the loss of vibrostability.

The presented methodology seems to be complete but may be modified in certain cases. For example, there may be a problem of the insufficient length of cables in the case of large machine tools, or the problem of isolating the machine tool from ambient influences when surrounded by other machinery and equipment. However, the main conclusion is that the modal research carried out in consultation with the manufacturer and the user allows for the unambiguous identification of the causes of self-excited vibrations, an indication of “weak element” of the MDS system, and the determination of required specific structural modifications or changes in manufacturing technology. Examples of the effective application of the presented methodology are shown in Section 2.

2. Examples of applications

First of all, the search for the "weak elements" in the system consisting of the machine and the cutting process, based on the symptoms during operation, requires the existence of a critical situation in which machining loses stability. The aforementioned actions of the user, often carried out in consultation with the manufacturer of the machine tool or tools, can bring desirable results. However, it often happens that the introduction of intuitive changes can have undesirable effects. In points 2.1 - 2.3 the real problems of machining were described, and demonstrated the effectiveness of the methodology proposed in Section 1 and the conclusion based on the results of experimental modal analysis.

2.1. Regenerative self-excited vibrations in turning process

The practical effectiveness of the methodology proposed in Section 1 for the search of "weak elements" in the MT-CP system has been demonstrated in examples, which resulted from the need to solve specific operational problems. The first example relates to the research whose aim was to identify a "weak element" of the MDS system of lathe, responsible for the formation of self-excited vibrations during intense turning – Fig. 3. These vibrations appeared for a very short overhang of the workpiece and short overhang of the tool.

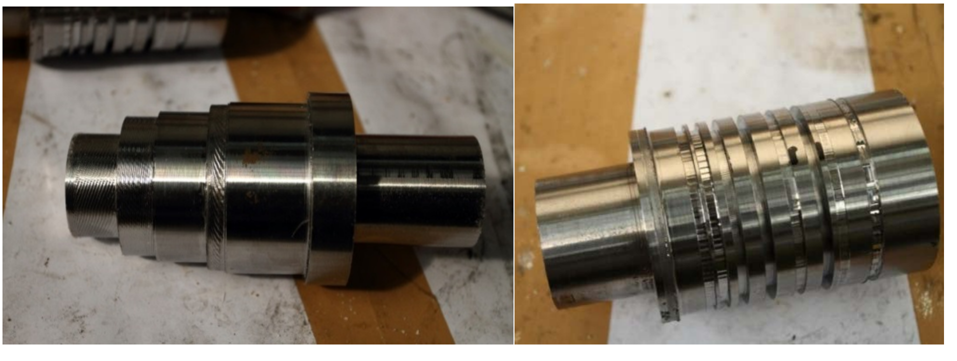

Fig. 3. Marks of unstable machining: (a) in longitudinal turning b) in cutting

The machine tool worked properly in its original structural set-up. Then, it was modernized: Slidings were replaced by rolling-element guideways in the connection between the main body of the lathe and a tool turret. The spindle and several components of the power transmission system were replaced as well. These actions caused by an unspecified change in the dynamic properties of the MDS system of the lathe. It should be emphasized that these changes were made intuitively based on the assumption that they would improve the dynamic properties and hence vibrostability. In fact, it turned out that the changes were unfavourable from the point of view of stability. In order to determine the reasons, the research methodology presented in Section 1 was used.

The nature of self-excited vibrations was identified as regenerative vibrations (chatter), and based on the multiple implementation of the unstable cutting process at different speeds, its frequency was determined to be 300 Hz. Then the location of the measurement points and excitation points were chosen based on lathe structure study. Accelerations and forces signals were measured at the certain points of the MDS system of the machine tool – Fig. 4.

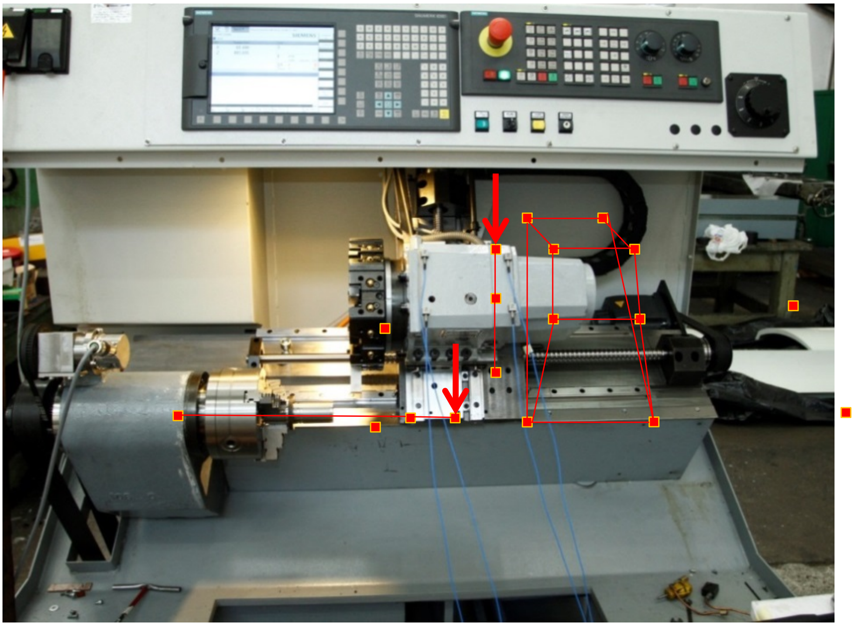

Fig. 4. A view of the studied lathe with the schematic indication of selected measurement points and the points of force application

The research was performed using a test set consisting of a PCB 356A32 acceleration transducers with a sensitivity of approximately 10 mV/(m/s^2) and a measuring range of about 500 m/s^2, with the weight of 6 grams. Excitation of vibration was performed with the use of the Kistler 9726A20000 modal hammer with the sensitivity 0.23 mV/N. Signal acquisition and processing were performed using the Difa SCADAS III, with 24-bit analog to digital transducer cards and anti-aliasing filters. During the tests, the individual machine axes were positioned and held in an equilibrium position by a drive (not brakes) in order to match the dynamic behaviour of the machine during machining as closely as possible.

Then, a comparative analysis of frequency transfer functions was performed based on signals of the exciting force and accelerations, in order to indicate the elements with the dominant levels of the amplitudes at the selected frequencies. This was necessary because the animation of vibrations does not allow the assessment of the significance of movement but only of its form. The workpiece was the element that had the dominant amplitude compared to other elements of the MDS system of the machine tool. However, it was not possible to definitely conclude that the workpiece was a "weak element," because self-excited vibrations were observed for machining carried out with the minimum overhang, which meant that the stiffness of the workpiece was very high compared to the stiffness of other components of the MDS system. Therefore, it was necessary to perform modal testing. A test was carried out, consisting in excitation of vibrations with a modal hammer on the workpiece and tool sides of the branch, respectively. This procedure allowed obtaining satisfactory coherence function. Then, using the Polymax algorithm of modal parameter estimation, and after the rejection of non-orthogonal modes of vibration, the modal model of the lathe was established – Tab. 1.

Table 1. Modal model of lathe

Pole	Frequency [Hz]	Modal damping [%]
1	129.669	2.87
2	168.015	1.92
3	**200.255**	**1.70**
4	**256.319**	**1.64**
5	274.777	2.08
6	**370.611**	**1.46**
7	390.630	2.83
8	432.158	2.03
9	487.611	3.84
10	584.230	2.02
11	695.737	3.31
12	768.738	2.28
13	799.037	2.47

The modes of vibrations were determined for each pole, allowing the interpretation of the vibratory motion of the object at a certain frequency. Animation was generated to show the movement of the machine tool at each natural frequency. Three modes of vibration were identified with a possible negative effect on the vibrostability of the machine tool, characterized by the significance of motion and a significant value of the amplitude of the dynamic compliance function:

- Mode 3 at 200 Hz, characterized by significant antiphase vibration of the workpiece and the tool;
- Mode 4 at 256 Hz, with a significant amplitude of vibration of the workpiece; and,
- Mode 6 at 370 Hz, also manifested by the significant vibration of the workpiece.

The study revealed that the higher compliance to self-excited vibrations was not caused by the changes in the guide connection but by element used to mounting of spindle. The modification of this structural element of the MDS system of the lathe effectively removed the cause of self-excited vibrations.

2.2. Parametric self-oscillation in thread milling

The second example of applying the proposed methodology for the effective search of "weak element" in the MDS system of the machine tool referred to the determination of the reason of intense self-excited vibrations generated during thread milling – Fig. 5. It was provisionally assumed that the reduction of the vibrostability of the lathe MDS system was caused by previously made modifications.

First, the analysis involved factors associated with the thread milling technology used. It was checked for the proper application of technological parameters of cutting, machining program, the tool, its chucking and overhang value, as well as the quality of the workpiece clamping. No cause with a possible negative effect on the vibrostability of cutting process could be found. Then, the marks of machining on the workpiece were examined and correlated with the frequency of generated self-excited vibrations.

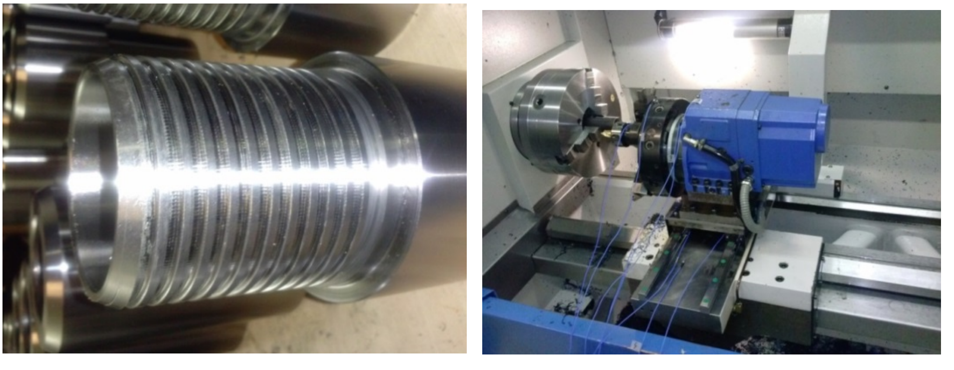

Fig. 5. a) Path of thread milling in unstable conditions; b) The general view of the machine with mounted sensors

The frequency of 1032 Hz and its harmonics, dominant in the obtained spectrum of accelerations, is very high. In most cases, self-excited vibrations are manifested at frequencies close to the frequency corresponding to one of the structural modes of machine tool body machine. Body elements are made of cast iron and usually vibrate in the range below 300 Hz. A detailed comparison of the vibration levels led to the conclusion that at a frequency of 1032 Hz, which the largest amplitude of the vibrations, can be observed on the upper surface of the tool turret and on the steel base to which the entire turret was attached. Attempts at turning with a very similar tool with an almost identical setting of the MDS system elements, differing only in the side of tool approach, resulted in stable machining. Vibrations in the work-piece branch of the force chain were negligible. Even when self-excited vibrations were observed, elevated levels of vibration were not observed on the spindle elements.

Based on the aforementioned considerations and measurements, the self-excited vibrations were identified as parametric (mode coupling) – Figure 2b. Such vibrations are observable when the resultant cutting force direction coincides with the direction of the greatest compliance of the MDS system of the lathe, measured at the point of contact of the tool and workpiece. The determination of stiffness ellipsoid mentioned in the Section 1 would require a much more complex research program, and it was not feasible due to time constraints and available machine tool accessories. Parametric oscillations occur at a frequency lower than one of the natural frequencies, while regenerative ones are generated at a frequency greater than one of the natural frequencies [3]. This conclusion could be drawn only based on the impact test performed in the subsequent step, the results of which are shown in Figure 6.

Fig. 6 shows the results of the impact test for force applied to the tool. The dominant resonance can be seen at the frequency of about 1129 Hz, similar to that determined dominant frequency spectrum of the signal recorded during self-excited vibrations. In addition, the analysis of the various transfer functions allowed concluding that the greatest amplitudes of vibration occurred in the upper part and the base of the tool turret, on a steel element to which they were attached rolling elements in guideway connection. The force acting on the housing of the tool turret showed weak coherence between the designated signals, similar to the situation when the force was applied to the lower part of the guide. This demonstrates the good properties of the damping elements made of cast iron. Detailed analysis of the experimental data and the analysis of the MDS system of the machine tool led to the identification of the steel element, used to attach the rolling element, as the “weak element” – Fig. 7.

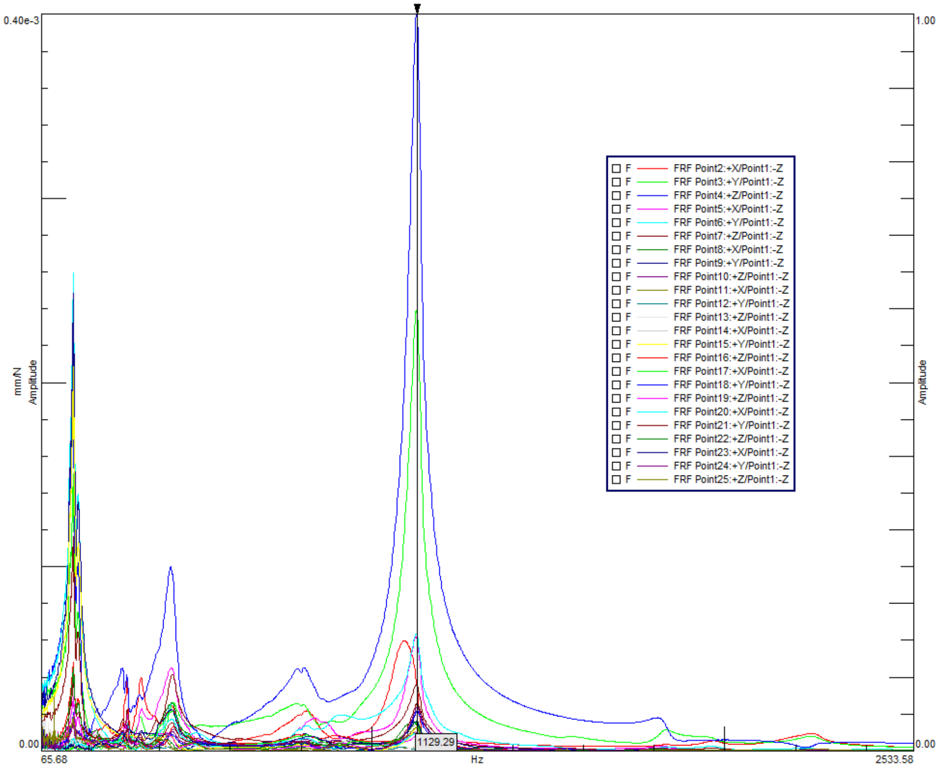

Fig. 6. Amplitude-frequency characteristics obtained in the pulse test for force acting on the tool

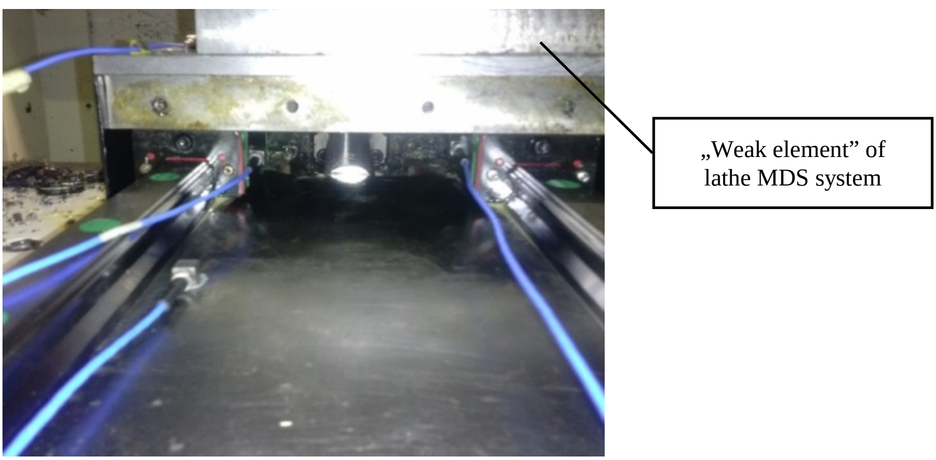

Fig. 7. “Weak element” of lathe MDS system

In consultation with the manufacturer, it was established that the original connecting element was made of cast iron. The replacement of sliding guides into rolling guides required the modification of the element; and, in order to lower the cost of the product, it was made of steel (as no additional casting was needed). The result was an unintended negative change in the dynamic properties of the entire MDS system of the machine tool.

The formulated conclusion allowed the manufacturer to take specific measures to stiffen and increase the damping of the element. The changes brought about the complete elimination of the problem of chatter. Therefore, it was shown that a seemingly small change in the MDS system resulted in the loss of its vibrostability for a particular relative configuration of the resultant cutting force and the stiffness ellipsoid, and also resulted in the activation of chatter mechanism.

2.3. Self-excited vibrations in the process of milling on the machining centre

Another example of the application of the methodology indicating the “weak element” in the machine tools based on symptoms observed during operation was to the search for the causes of self-oscillation during milling performed with the use of the fixed-axis machining centre. This problem was revealed in the marks of unstable machining, on one of the side surfaces of the workpiece, resulting from a particular spatial configuration of movements of individual body elements – Fig. 8.

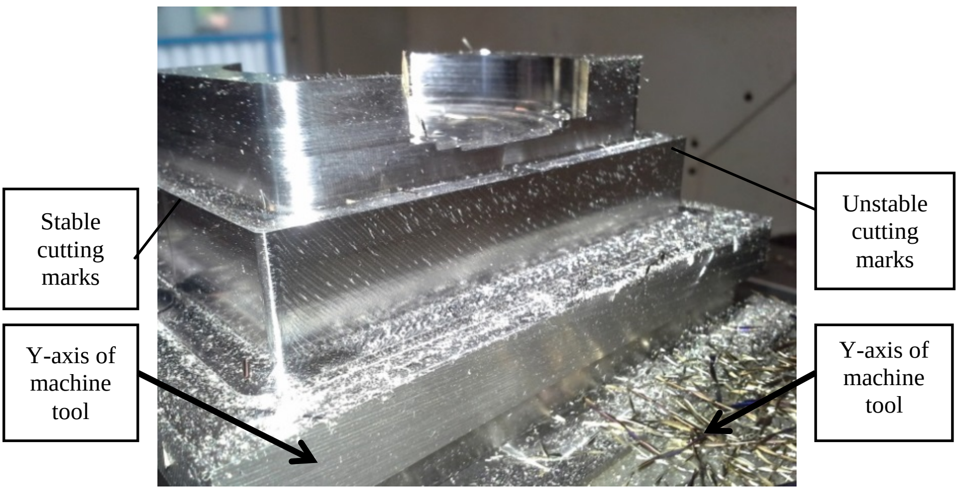

Fig. 8. Machining marks on the walls of the tested object (machining stability dependent on the direction of machining)

The machined surfaces with marks of vibration were examined prior to the research. It was agreed that probably the loss of the MDS vibrostability occurred via the regeneration effect. In order to unambiguously confirm the causes of vibration, an experimental research program consisting the following was initiated:

- Cutting test with the use of the cuboid object, with a tool with a small overhang: In this test, the depth of cut was a_p = 18 mm, while the incremental width of cut in subsequent tests was a_e = 0.1–1 mm, and the aim of this study was to identify the frequency of chatter.
- Impact test: the purpose was to determine the dynamic characteristics of the machine and to find the resonant frequency with the value close to the recorded frequency of self-excited vibrations. This was to enable the selection of spindle speed that would ensure machining stability.
- Cutting with the changed rotational speed: This stage of the study was to confirm the thesis that the cause of the vibration lies in the phenomenon of regeneration effect.
- Cutting and impact test using a tool with a large overhang: This stage was to determine the dynamic properties of the machine tool with a tool clamped in the spindle, with increased compliance and select machining technology.
- Modal analysis of the milling machine: The aim was to identify the “weak element,” i.e. a structural element, that is responsible for reducing the dynamic stiffness of the machine tool.

In the first place, it was determined if the vibrations were related to the tool or workpiece. A comparison of the amplitudes of vibration showed the dominance of vibrations on the tool. Then the self-excited vibrations frequency was determined by comparing the frequency spectra for the machining with incremental milling width. The observed frequencies of harmonic vibrations resulted from the spindle speed and the number of cutting edges. The frequencies of self-excited vibrations were related to the frequency corresponding to the structural form of vibrations (and do not depend on the frequency of cutting). The frequency of chatter was 1166 Hz, which is confirmed by the signal spectrum of vibration acceleration, measured on the workpiece – Fig. 9.

Next, the impact test was performed with the purpose of correlating the chatter frequency with the frequency corresponding to the structural mode of the machine tool. Fig. 10 shows the FRF’s (frequency response functions) for the X and Y axes of the machine tool, which shows that the mode of vibration at 1079.79 Hz is likely responsible for the formation of chatter.

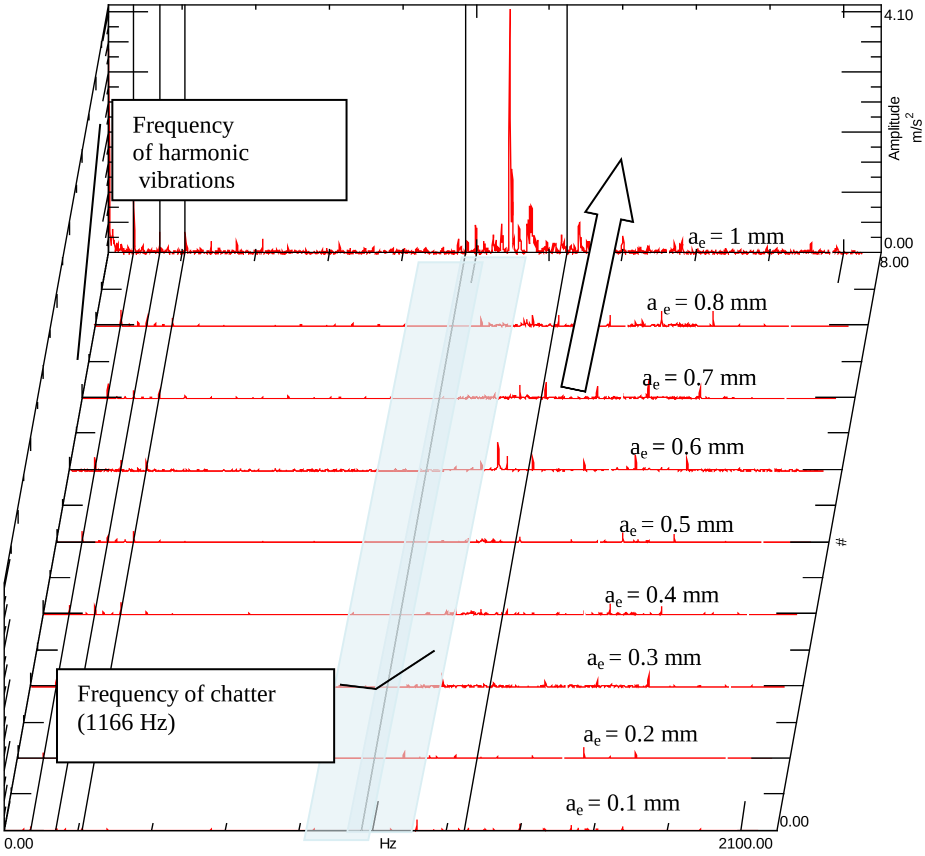

Fig. 9. Spectra of the vibration acceleration signal with a variable width of cut, as recorded on the spindle

Based on the known structural frequency, the rotational speed of the tool was selected to ensure the stability of machining. It can be determined based on the theory of chatter (temporal relations between the waves of primary and secondary rotation of tool or workpiece [14]. Vibrostability was obtained at the speed of 1477 rpm. In the next step, a cutting test with the pre-determined speed was performed. The observed significant difference in the levels of vibrations recorded for the same depth of cut (18 mm) and the same width of cut (1 mm), but at different speeds (1400 and 1477 rpm), confirmed the chatter was generated via the regeneration effect.

In the next step, a test was performed for the tool with a high compliance that is used for molding shells machining. The instability of machining (high levels of vibration) was observed already for very small sections of cut (a_p = 0.15 mm, a_e = 0.1 mm). Based on the recorded vibration signals, it was determined that the chatter frequency in this case was equal 389.95 Hz – Fig. 11b.

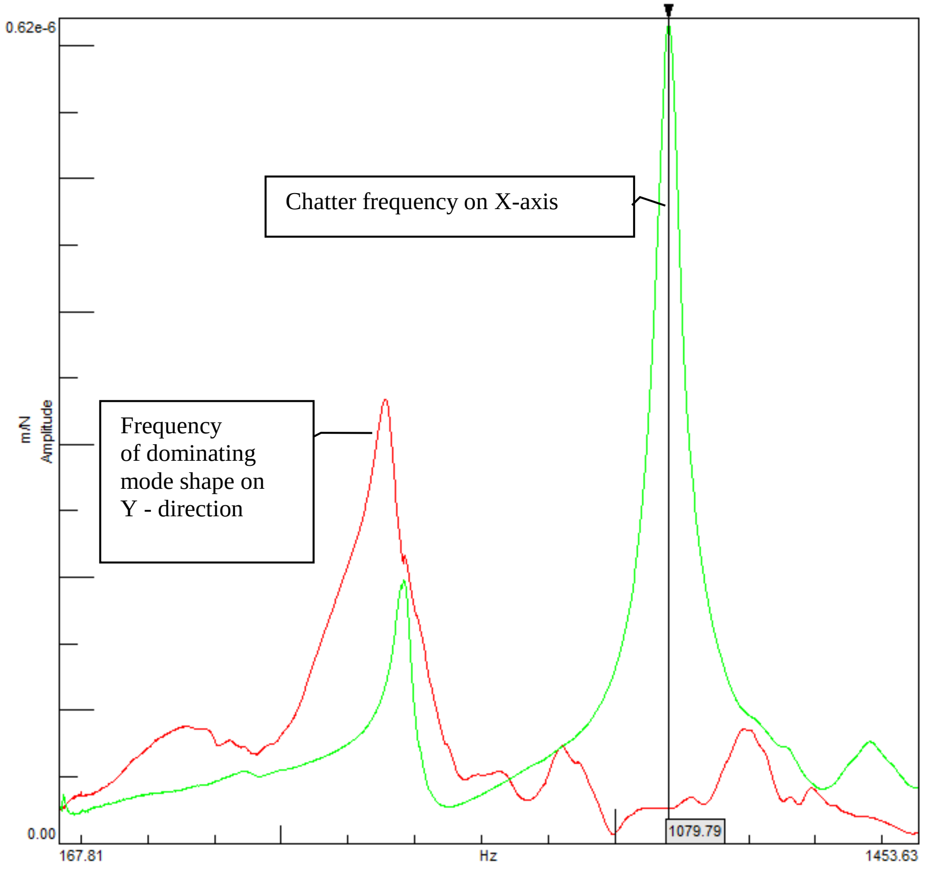

Fig. 10. Frequency response functions of the machine tool with a clamped machine tool with a small hangout, determined on X and Y axes of the machine tool

Based on the frequency response function of the machine tool with a long overhang of tool, it was determined that the chatter may be caused by the modes of vibrations at two frequencies (377 Hz, 391 Hz). This makes it difficult to choose the stable rotational speed. The frequency of chatter, and therefore the rotational speed ensuring the stability of machining, will therefore vary depending on the change of the tool.

Selection of stable cutter speed, individually for each tool, would require impact testing for each individual tool. The implementation of this idea in an industrial environment would be very difficult and sometimes impossible. It was therefore necessary to indicate the structural element whose change would have the greatest impact on the improvement of the dynamic properties of the machine tool, and thereby increase its vibrostability. The modification of the element should increase the stability of machining, irrespective of the cutting tool.

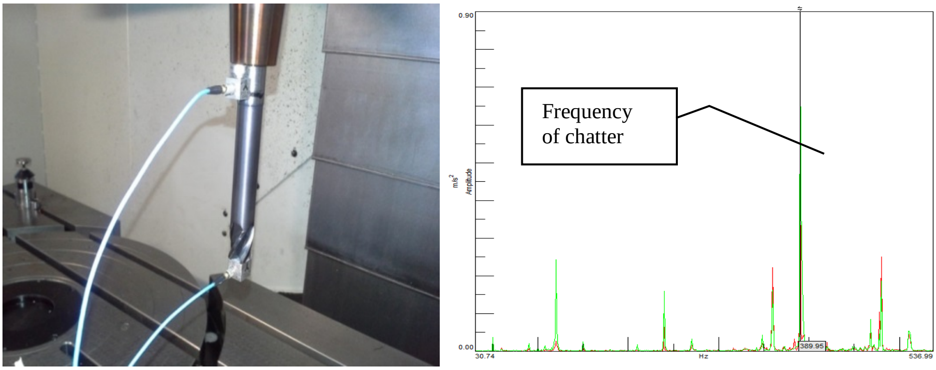

Fig. 11. a) Tool with a long overhang, ready for the impact test; b) the spectrum of the acceleration signal obtained during unstable machining with the use of the tool with a large overhang

For this purpose, based on the modal test, the modes of vibration with the greatest impact on the relative vibration of the tool and workpiece were determined. The frequencies of these modes are summarized in Tab. 2.

Table 2. Modal model of 5-axis machining centre

Pole	**Frequency [Hz]**	**Modal damping [%]**
1	116.386 Hz	5.32
2	171.233 Hz	3.55
3	426.137 Hz	3.28
4	448.682 Hz	2.10
5	870.980 Hz	1.93
6	885.739 Hz	0. 85
7	994.418 Hz	4.65
8	1153.778 Hz	0.18

Detailed analysis of the animation of the modes of vibration, amplitudes of displacements, and phase relationships allowed the identification of the "weak element", namely, the drive of the B-axis of the milling centre. These conclusions formed the basis of modifications performed in the structural system of machine tool.

Summary

The paper presents a methodology for the search of a "weak element" in the mass-damper-spring system of the machine tool under operating conditions, using experimental modal analysis. It should be noted that this methodology might be mainly used when the user, not knowing the dynamic properties of the machine tools, encounters the problem of intense chatter, which they are not able to solve. It should be emphasized that each case is different, which should be considered in developing any research program. Yet, the examples of the practical application of the developed methodology (Section 2) indicate its potential in improving the vibrostability of machining. This can be done by selecting a "weak element" of the system and to identify the MDS machine design changes, the introduction of which will lead to an improvement in its dynamic properties.

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Poszukiwanie słabych ogniw w systemie obrabiarka–proces skrawania ze względu na jego wibrostabilność, na podstawie symptomów eksploatacyjnych

Słowa kluczowe

Obrabiarka, analiza modalna, drgania chatter, stabilność.

Streszczenie

W artykule przedstawiono metodologię poszukiwania „słabego ogniwa" w dynamicznym systemie, składającym się z obrabiarki oraz procesu skrawania, z użyciem metod eksperymentalnej analizy modalnej. Prace te zostały zainspirowane koniecznością rozwiązania rzeczywistych problemów obróbkowych, które miały miejsce w przemysłowej eksploatacji obrabiarek różnego typu. Koniecznym stało się zidentyfikowanie przyczyn utraty stabilności obróbki oraz wskazanie „słabego ogniwa" w układzie masowo-dyssypacyjno-sprężystym obrabiarki. W konsekwencji określono zakres wymaganych zmian konstrukcyjnych oraz zweryfikowano ich skuteczność. W artykule wykazano również praktyczną efektywność zaproponowanej metodyki prowadzenia badań obrabiarek.