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Influence of boundary conditions on the vibration modes of the smart turning tool

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1. Introduction

The paper presents analysis of smart turning tool dynamics. It is well known that turning process is often used in different metal parts manufacturing process. The growing demand for fast and low cost part manufacturing brought in to the day light vibration assisted turning systems, which have several advantages over ordinary turning process. It also well known that post treatment of the machined parts cost a lot of money.

The authors of [1] have carried out an experiment with vibration assisted turning and have obtained data which showed that usage of ultrasonic vibration reduces the roughness of the surface.

According to [2] ultrasonic turning can reduce the cutting force up to 47% compared to conventional turning.

Authors of the [3] have proposed metal cutting by using ultrasonic frequencies vibrations which are more rational compare with traditional cutting method. Research was applied in turning process. For this purpose was created the special cutting knife with ultrasonic vibration actuator of piezoceramics. Results proved theoretical research, which says that by using ultrasonic frequencies the surface of machined detail is improved.

In work [4] vibration-assisted machining tool forces were compared to the conventional turning process forces. The forces in the vibration-assisted machining were reasonably lower then those occurring in the conventional turning. It was also stated in this paper [4] that vibration-assisted machining extends tool life several times.

Ultrasonic vibrations are also used in the vibration assisted drilling process. Authors of the [5] have carried out comprehensive investigation of the vibration assisted drilling. This study was concerned with application of numerical-experimental approach for characterizing dynamic behavior of the developed piezoelectrically excited vibration drilling tool with the aim to identify the most effective conditions of tool vibration mode control for improved cutting efficiency. 3D finite element model of the tool was created on the basis of an elastically fixed pre-twisted cantilever (standard twist drill). The model was experimentally verified and used together with tool vibration measurements in order to reveal rich dynamic behavior of the pre-twisted structure, representing a case of parametric vibrations with axial, torsional and transverse natural vibrations accompanied by the additional dynamic effects arising due to the coupling of axial and torsional deflections ((un)twisting). Numerical results combined with extensive data from interferometric, accelerometric, dynamometric and surface roughness measurements allowed to determine critical excitation frequencies and the corresponding vibration modes, which have the largest influence on the performance metrics of the vibration drilling process. The most favorable tool excitation conditions were established: inducing the axial mode of the vibration tool itself through tailoring of driving frequency enables to minimize magnitudes of surface roughness, cutting force and torque. Research results confirm the importance of the tool mode control in enhancing the effectiveness of vibration cutting tools from the viewpoint of structural dynamics.

Differences between conventional and ultrasonic turning in stress distribution in the process zone and contact conditions at the tool/chip interface are investigated in the [6].

In this paper modal and harmonic analysis of the vibration assisted turning tool is presented in order to find the useful frequencies for lowering the cutting force and decreasing the surface roughness. Longitudinal and transverse turning tool vibration modes are investigated while changing the turning tool fixation areas.

2. Modal analysis

The goal of modal analysis is to determine the natural mode shapes and frequencies of a structure. The finite element method (FEM) is commonly used to perform this analysis.

Modes are inherent properties of a structure. Modes are determined by the material properties (mass, stiffness), type, configuration and boundary conditions of the structure. Each mode is defined by a natural (modal or resonant) frequency, modal damping, and a mode shape. If either the material properties or the boundary conditions of a structure change, its modes will change. At or near the natural frequency of a mode, the overall vibration shape (operating deflection shape) of a machine or structure will tend to be dominated by the mode shape of the resonance. All vibration is a combination of both forced and resonant vibrations. Forced vibrations occur due to the fact of:

- internally generated forces;
- disbalances;
- external loadings;
- ambient excitation.

Resonant vibrations typically amplify the structure response far beyond the level of deflection, stress, and strain caused by static loading.

A modal analysis determines the vibration characteristics (natural frequencies and mode shapes) of a structure or a machine component. It can also serve as a starting point for another, more detailed, transient and harmonic response analysis, or a spectrum analysis. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions.

The equation of motion for an undamped system, expressed in matrix notation is [1]:

$$[M]\{\ddot{u}\}+[K]\{u\}=\{0\}, \tag{1}$$

where [M] is the mass matrix and [K] is the structure stiffness matrix, including prestress efects.

The solution of Eq. (1) has the general form:

$$\{u_i\}=\{\Phi\}_i \cos\omega_i t, \tag{2}$$

where {Φ} is the eigenvector representing the mode shape of the i-th natural frequency and ω_i is the i-th natural circular frequency. Thus, Eq. (1) becomes:

$$\left(-\omega_i^2[M]+\{K\}\right)\{\Phi\}_i=\{0\}. \tag{3}$$

Rather than outputting the circular frequencies ω_i, the natural frequencies were output:

$$f_i=\frac{\omega_i}{2\pi}. \tag{4}$$

The normalization of each eigenvector Φ_i was in respect to the mass matrix:

$$\{\Phi\}_i^T[M]\{\Phi\}_i=1. \tag{5}$$

The eigenvalues and eigenvectors are the solutions of the equation:

$$[K]\{\Phi_j\}=\lambda_j[M]\{\Phi_j\}, \tag{6}$$

where [K] is the structure stiffness matrix, {Φj} is the eigenvector, λ_j is the eigenvalue and [M] is the structure mass matrix.

3. Harmonic analysis

Harmonic response analyses are used to determine the steady-state response of a linear structure to loads that vary sinusoidally (harmonically) with time, thus enabling to verify whether or not your designs will successfully overcome resonance, fatigue, and other harmful effects of forced vibrations.

Harmonic response analysis gives for the ability to predict the dynamic behaviour of cutting tool structures. This technique is used to determine the steady-state response of a linear structure to loads that vary harmonically with time [7].

The general equation of motion for a structural system is [8]:

$$[M]\{\ddot{u}\}+[C]\{\dot{u}\}+[K]\{u\}=\{F^a\}, \tag{7}$$

where [M] is structural mass matrix; [C] is structural damping matrix; [K] is structural stiffness matrix; $\{\ddot{u}\}$ is nodal acceleration vector; $\{\dot{u}\}$ is nodal velocity vector; $\{u\}$ is nodal displacement vector; $\{F^a\}$ is applied harmonic load vector.

As stated above, all points in the structure are moving at the same known frequency, however, not necessarily in phase. Also, it is known, that the presence of damping causes phase shifts. Therefore, the displacements may be defined as:

$$\{u\}=\{u_{max}e^{i\Phi}\}e^{i\Omega t}, \tag{8}$$

where $\{u_{max}\}$ is maximum displacement; i is square root of -1; $\Omega = 2\pi f$ is imposed circular frequency (radians/time); f is imposed frequency (cycles/time); t is time; Φ is displacement phase shift.

4. Model of turning tool

Experimental model of the vibration assisted turning tool was created by faculty of Mechanical Engineering and Mechatronics of Kaunas University of Technology. The model is composed of the following main parts (Fig. 1): carbide insert *1*, turning tool *2*, tool holder *3*, two fixation bolts *4*. Langevin type piezoelectric transducer [9] which contains of horn *5*, piezoelectric rings *6* and backing *7*. Properties of the parts are listed in the Table.

The model was transferred to the CAD system Solidworks, after the dimensions were taken. The main feature of the model is the four contact areas. These areas will act as the representation of bolts used in the real model.

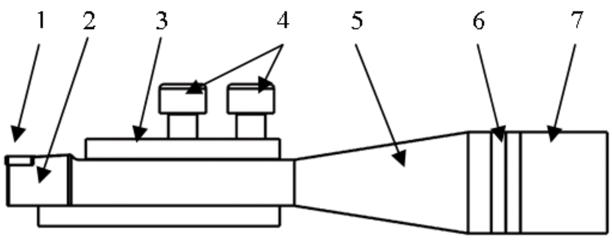

Fig. 1 Scheme of turning tool with holder: *1* – insert, *2* – turning tool, *3* – tool holder, *4* – bolts, *5* – horn, *6* – piezoceramic rings, *7* – backing

Table

Mechanical properties of the material used in FEM analysis

Component	Material	Density ρ, kg/m^3	Poisson`s ratio	Young`s modulus, GPa
Turning tool, tool holder, fixing contact areas, horn, backing	Steel C45	7850	0.33	210
Insert	Carbide	15000	0.24	640
Piezoceramic rings	PZT5	7800	0.371	66

It is known that boundary conditions are very important for the analysis results and for the dynamics of the turning tool. In our model boundary condition is the locations of the contact areas. These areas are changed during the analysis to investigate influence to tool dynamics. Because there is a lot of variations of contact areas and this would cause to solve optimization task in this paper six different sets of contact areas were analysed (Fig. 2).

The 3D model of turning tool was transferred to FEA software – ANSYS. For the modal and harmonic response analysis of the turning tool was chosen the SOLID98 finite element. In the [8] this element is described as the element which has a quadratic displacement behavior and is well suited to model irregular meshes (such as produced from various CAD/CAM systems). When used in structural and piezoelectric analyses, SOLID98 has large deflection and stress stiffening capabilities. The element is defined by ten nodes with up to six degrees of freedom at each node.

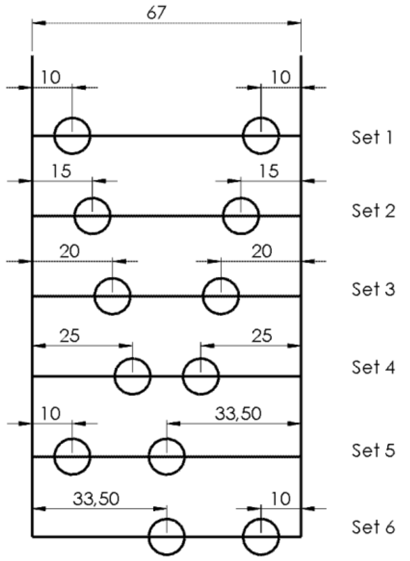

Fig. 2 Locations of contact areas positions

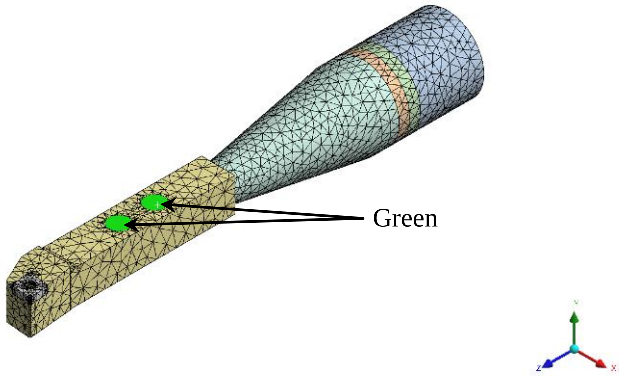

Fig. 3 Finite element model of the turning tool

In Fig. 3 the meshed turning tool with the 20042 nodes and 10925 elements is presented. Green areas are the highlighted contact areas.

5. Modal analysis results

Modal analysis of the turning tool was carried out between 0 and 40 kHz frequency. As it was stated in the [2] and [5] the longitudinal vibrations are very useful for decreasing surface roughness. It was determined first vibration assisted turning tool longitudinal vibration mode at 5.3 kHz, second at 27.6 kHz (Fig. 4).

The first transverse mode of the turning tool was determined at 9.9 kHz and the second at 35.1 kHz.

Analysed turning tool transverse and longitudinal frequency dependence from contact area position are showed in Fig. 5. We can see that modes that we are interesting most efficiently separates – differs in frequency, when fifth contact area set is used.

This is very useful when we want to achieve needed vibration mode and the mode‘s frequencies do not overlap each other. Such overlapping we can see in the second and third contact area fixing set.

The modal analysis of the turning tool with different contact areas has showed that (Fig. 5.) the position of the contact areas have large influence for tool dynamics. For example the eigenfrequency difference between first transverse mode in Set 1 (f = 10 kHz) and the same transverse mode but in Set 4 is two times lower (f = 4.9 kHz). The eigenfrequency of the second transverse vibration mode of the turning tool is approximately more than three times greater. Lowest eigenfrequency of the transverse vibrations second mode is then contact area Set 3 and f = 27.6 kHz. Highest vibration in the transverse mode is reached when tool vibrates with the frequencies equal to f = 39.6 kHz (Set 6).

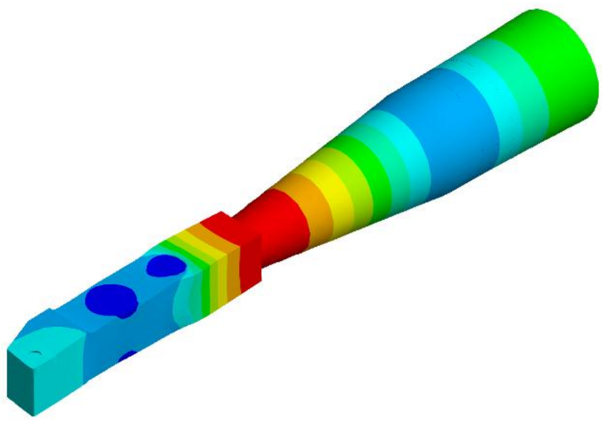

Fig. 4 Second longitudinal mode of the turning tool

Longitudinal vibrations are one of the main vibrations which are used to lower surface roughness and cutting forces and to extend tool life. In our model longitudinal vibrations are achieved in high frequencies. The range in which longitudinal vibrations occur are from f = 25 – 32 kHz. The large difference of frequencies of the first and second modes that we had in the transverse vibrations did not occur in the case with the longitudinal vibrations.

The obtained data gives the possibility to predict that exciting turning tool to according frequencies we can control the vibration modes which leads to control the useful vibrations.

The more accurate data for the frequencies needed for tool excitation could be found if the contact areas locations were find out during optimization. But this task is time consuming and further investigation should be carried out before proceeding to optimization analysis.

6. Results of harmonic analysis

The harmonic analysis of the vibration assisted turning tool was carried out in order to get quantitative results of the analysis. From the modal analysis results (Fig. 5) we can see that in the contact area Set 5 we have turning tool modes that are separated quite evenly. For this reason we have chosen the contact area Set 5 for harmonic analysis. Furthermore in the Set 5 we have good localization of the vibration modes. This feature is very important, because such good localization of the modes will enable to excite interested modes of the turning tool to get needed useful vibrations.

Harmonic analysis was carried out in the range from f = 15 – 40 kHz. The material properties and the mesh were used the same as in the modal analysis.

The results of the harmonic analysis are presented in the Fig. 6 and Fig. 7. The Fig. 6 shows turning tool tip amplitude-frequency characteristic in the axis Z along the tool holder direction, contact are Set 5. The peak amplitude

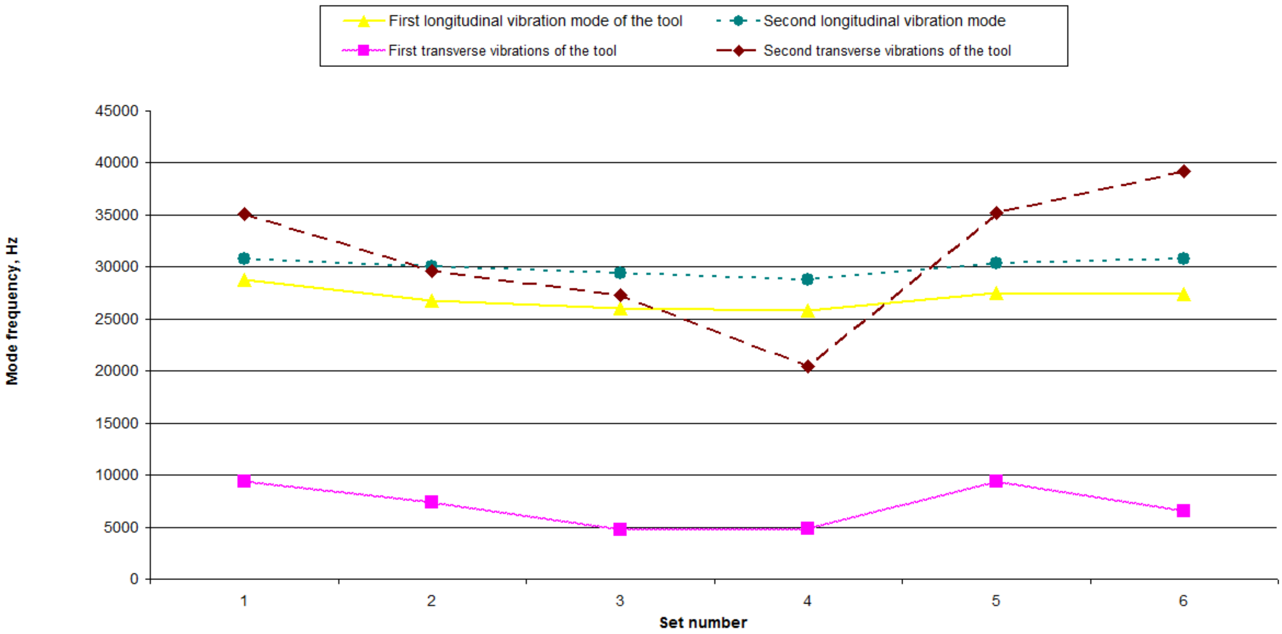

Fig. 5 Vibration mode dependence from the contact area Set number

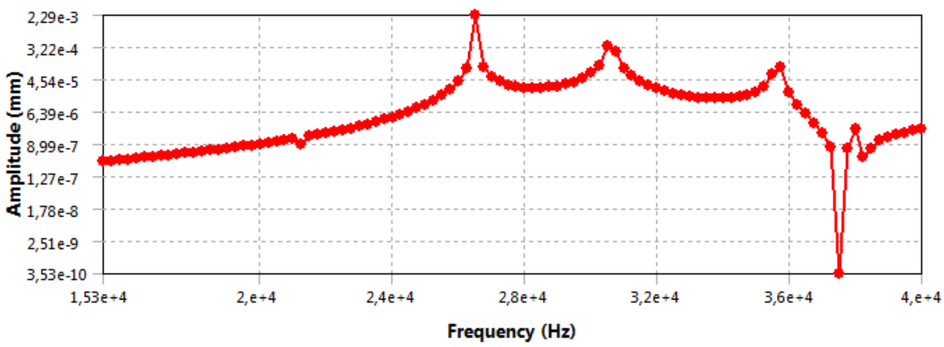

Fig. 6 Amplitude-frequency characteristic of the tool tip displacement in axis Z direction, Set 5

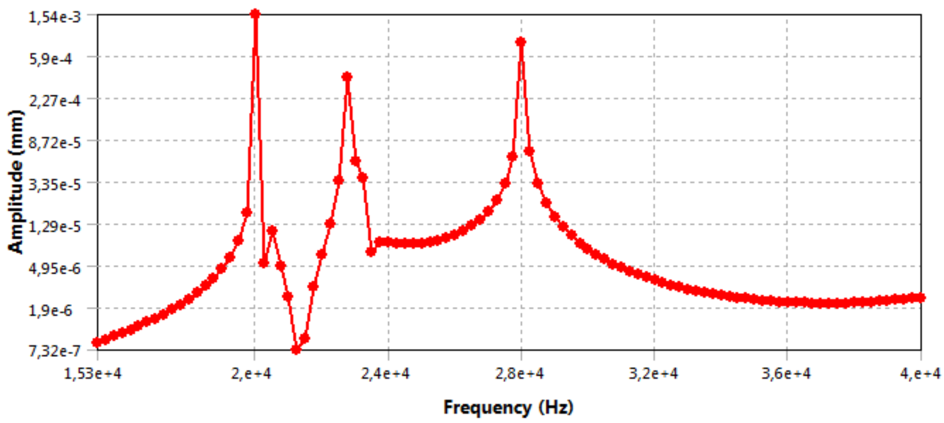

Fig. 7 Amplitude-frequency characteristic of the tool tip displacement in Z direction for Set 4

reaches in when the tool is excited with the frequency equal to 26.5 kHz and the displacement is 2.29×10^{-3} mm. This peak amplitude corresponds to the first longitudinal vibration mode of the tool. Second peak occurs at the frequency equal to 30.5 kHz and the displacement is 3.63×10^{-4} mm. At same frequency as the second peak (Fig. 6) in the Fig. 5 we see second longitudinal vibration mode of the tool. From this comparison we can state that exciting tool to according frequencies useful displacements of the vibration assisted turning tool can be achieved.

Three peaks of tool amplitude (Fig. 7) were observed when tool was excited harmonically using contact area set 4. First peak $f = 20$ kHz and the displacement is 1.54×10^{-3} mm corresponds to second transverse vibration mode. The last two peaks occurring respectively 22.8 and 28 kHz represents first and second longitudinal vibration modes of the turning tool.

Harmonic analysis of the vibration assisted tool showed that using correct excitation frequencies we can achieve useful displacement of the turning tool tip.

Further work will be carried out in order to check the computer analysis data with the real experiment data.

7. Conclusions

1. FEM analysis of the turning tool holder with the vibration amplifier was modeled with six sets of different locations of contact areas positions.
2. Turning tool modal analysis has showed transverse and longitudinal vibrations frequency dependence from location of contact areas position. Found sets where vibration mode and the mode‘s frequencies overlap and do not overlap each other.
3. Turning tool harmonic analysis has showed that the peak amplitude of the tool longitudinal vibration reaches in when the tool is excited with the frequency equal to 26.5 kHz, and the tool tip displacement is 2.29×10^{-3} mm. This peak amplitude corresponds to the first longitudinal vibration mode of the tool.

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KRAŠTINIŲ SĄLYGŲ ĮTAKA IŠMANIOJO TEKINIMO PEILIO VIRPESIŲ MODOMS

R e z i u m ė

Straipsnyje pateikta aukštojo dažnio virpesiais žadinamo tekinimo peilio tikrinių virpesių (modų) analizė, atlikta įtvirtinus jį skirtingais būdais. Nustatyta, kuriais būdais įtvirtinus tekinimo peilį modos lokalizuojasi ar dengia viena kitą. Taip pat atliktas periodinio tekinimo peilio atsako į periodinį poveikį tyrimas. Atlikus šį tyrimą su trimis skirtingais būdais įtvirtintais tekinimo peiliais, nustatyta, kad įtvirtinus peilius penktuoju būdu vyrauja jų išilginių virpesių moda, kurios amplitudė yra didžiausia.

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INFLUENCE OF BOUNDARY CONDITIONS ON THE VIBRATION MODES OF SMART TURNING TOOL

S u m m a r y

The paper presents analysis of vibration modes of a high-frequency assisted turning tool. The analysis was performed with different methods of tool contac area. Turning tool contact area set was found in which tool modes overlap or localize. Also it was conducted periodic turning tool response analysis to periodic impact. The analysis was caried out with three diferent contact area sets and it was concluded that when using fifth conctact area set longitudinal turning tool modes are dominant and the frequency amplitude is the biggest.

Keywords: smart tool, mode control, harmonic response.

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