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Modified tool structures for effective cutting

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1. Introduction

New ideas for improving performance of vibration cutting processes, carried out by the author of this paper in the field of vibration turning [1], suggested and encouraged to perform a more thorough investigation of associated dynamic phenomena. A stiffer cutting tool does not improve stability significantly. It means, that better way of passive control technique is related to the effective control of higher modes of flexible tool structures. The reduction of deleterious vibrations in the machine–tool–workpiece system is based on the excitation of a particular higher vibration mode of a tool, which leads to through intensification of internal energy dissipation in the tool material.

The metal cutting using ultrasonic frequencies vibrations is more rational by comparison with traditional cutting method [2]. For turning process research the special cutting tool with ultrasonic vibration actuator of piezoceramics was created. Results have shown that by using ultrasonic frequencies the surface of machined details improve. On the other hand, active tools are expensive and it is hard to manage them without complicated equipment.

Using passive control technique complicated hardware is not useful and the end-user does not need to introduce new handling routines. The location and the value of the passive vibration absorbers already available on the market mostly are related to response energy in the first three modes of the structures. Sharma et al [3], simulate the boring bar as a cantilever Euler-Bernoulli beam considering its first mode of vibration. The two-degree-of-freedom model was analyzed constructing the stability diagram, dependent on the bar characteristics and on the absorber parameters (mass, stiffness, damping and position). Subsequent analysis performed in this work, allowed formulating of new analytical expressions for the tuning frequency improving the behavior of the system against chatter. Miguelez et al in paper [4] deals with the application of tunable vibration absorbers for the boring operation chatter suppression. The tunable vibration absorber was composed of mass, spring and dashpot elements and boring bar modeled as a cantilever beam. The effect of spring mass was considered in this analysis. The optimum specifications of the absorber such as spring stiffness, absorber mass and its position were determined using an algorithm based on the mode simulation method. Moradi et al [5] show that stability of the cutting tool can be considerably enhanced with a higher damping ratio of the cutting tool structure, which is related to the higher vibration modes, essentially when a cutting tool of low stiffness is used. According to Vela-Martinez et al [6], a stiffer cutting tool does not improve stability significantly. It means, that better way of passive control technique is related to the effective control of higher modes of flexible tool structures. It is based on the excitation of a particular higher vibration mode of a tool, which leads to the reduction of deleterious vibrations in the machine–tool–workpiece system through intensification of internal energy dissipation in the tool material. Paper [7] proposes to extend the stability limits of the machining system by enhancing the structure's damping capability via distribution of damping within the machining system exploiting the joints composing the machine tool structure.

The main idea of the reported research work is based on treating cutting tool as a flexible structure which is characterized by several modes of natural vibrations. In such machining processes as internal turning the structural configuration of the tool resembles cantilever beam. The first vibration mode of cantilever is characterized by maximum amplitudes of free end vibrations. The establishment of structural modifications of cutting tool as flexible structure is related with intensification of the higher natural vibration modes, similar to the vibration cutting conditions, results in reduction of magnitude of unwanted deleterious vibrations generated during machining. This suggests that excitation of higher natural vibration modes could be advantageous for this purpose since it is known that as the amplitude of higher modes becomes more intensive, energy dissipation inside tool material increases significantly and thereby makes the tool a more effective damper, which positively influences the amplitudes of the workpiece or machine tool itself, providing the possibility to reduce chatter.

The paper consists of numerical and experimental parts. Finite element model of a boring tool is presented in chapter 2 as well as chapter 3 is dedicated to description of qualitative and quantitative modeling results received during simulation. Chapter 4 is dedicated for practical realization of boring tool. Chapter 5 deals with experimental investigation of modified boring tools. The paper is finalized with concluding remarks.

2. Analysis of vibro-impact cutting process

The tool vibration during cutting could be described as vibro-impact process. As any kind of tool has distributed mass, stiffness and other parameters it is necessary to consider the dynamics of such elastic structure that is characterized by several modes of natural vibrations.

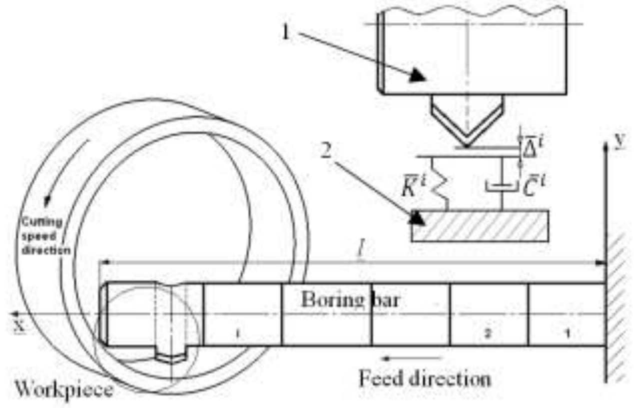

Fig. 1 Model of boring tool on the workpiece surface. *1* - boring bar with insert, *2* - workpiece

The impact interaction between elastic links is characterized by a rich spectral content of excitation impulses capable to excite a wide range of natural modes. In boring process the structural configuration of the tool resembles cantilever beam. Fig. 1 presents a computational scheme of the developed finite element (FE) model of impacting boring tool on the workpiece internal surface expressed by the rheological properties - stiffness and viscous friction. The model consists of i = 1, 2..., m finite elements.

The boring tool and workpiece interaction modeling is based on contact element approach and makes use of Kelvin-Voigt (viscoelastic) rheological model, in which linear spring is connected in parallel with a damper. The former represents the impact force and the latter accounts for energy dissipation during impact.

After proper selection of generalized displacements in the inertial system of coordinates, model dynamics is described by the following equation of motion given in a general matrix form:

$$[M]\{\ddot{y}(t)\}+[C]\{\dot{y}(t)\}+[K]\{y(t)\}=\{F(y,\dot{y},t)\}, \quad (1)$$

where [M], [C], [K] are mass, damping and stiffness matrices respectively, $\{y(t)\}$ $\{\dot{y}(t)\}$ $\{\ddot{y}(t)\}$ are displacement, velocity and acceleration vectors respectively.

Vector of interaction forces $\{F(y,\dot{y},t)\}$ between tool cutting edge and the workpiece, the components of which express the reaction of the tool cutting edge impacting workpiece and acquire the following form:

$$f_i(y_i,\dot{y}_i,t)=\overline{K}^i(\overline{\Delta}^i)-|y_i(t)|+\overline{C}^i y_i(t), \quad (2)$$

where $\bar{K}^i, \bar{C}^i$ are stiffness and viscous friction coefficients of the workpiece material, $\bar{\Delta}^i$ is distance from the cutting edge located at the i-th nodal point of the tool structure to the surface of the workpiece. In the case of considered model the assumption of proportional damping is adequate therefore internal damping is modeled by means of Rayleigh damping approach [7]:

$$[C]=\alpha_{dM}[M]+\beta_{dK}[K], \quad (3)$$

where α_{dM}, β_{dK} are mass and stiffness damping parameters respectively that are determined from the following equations using two damping ratios ξ_1 and ξ_2 that correspond to two unequal natural frequencies of vibration ω_1 and ω_2 [7]:

$$\alpha+\beta\omega_1^2=2\omega_1\xi_1;\ \alpha+\beta\omega_2^2=2\omega_2\xi_2. \quad (4)$$

The presented FE model of the vibro-impact system was implemented.

3. Numerical analysis of the tool vibrations

Vibro-impact process consists of free vibrations of the tool in the intervals between the impacts and its vibration during the impact interaction with workpiece. Therefore, profound investigation of free and impact vibration of elastic tool is essential.

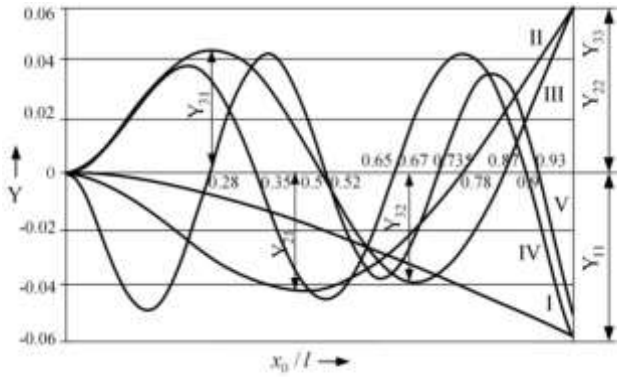

Fig. 2 The modes of natural transverse flexural vibrations of tool. x_0/l denotes ratio between the distance x_0 from the anchor of the tool and its whole length l, Y_{ij} - maximum amplitudes of the flexural: index i - number of vibration mode, j - sequence number of the maximum amplitude point with respect to the anchor point. *I* – the first transverse mode, *II* – the second transverse mode, *III* – the third transverse mode, *IV* – the fourth mode, *V* – the fifth mode

The modes of natural transverse vibrations of elastic cantilever beam-shaped tool presented in Fig. 2 consist of transverse displacement Y. Of the whole range of natural vibrations, the first five modes are distinguished (*I, II, III, IV, V*) which in the intersection with the axis line form nodal points marked by numbers that express the ratio between the distance x from the fixing site of the cantilever beam-shaped tool and its whole length l. The letter Y_{ij}, denote the values of the maximum amplitudes (deflections) of the flexural modes. As it is known, the first vibration mode of a cantilever is characterized by maximum amplitudes of free end vibrations as the amplitudes of each higher mode gradually decrease.

Identically to the process of free impact vibrations of cantilever released from statically deformed position bouncing the support the process of free impact vibrations of boring tool is simulated. The main purpose of such simulation is to imitate the cutting process which is characterized by the wide frequency range of cutting forces during chip formation when the tool contacts with elastically recovered surface of the workpiece.

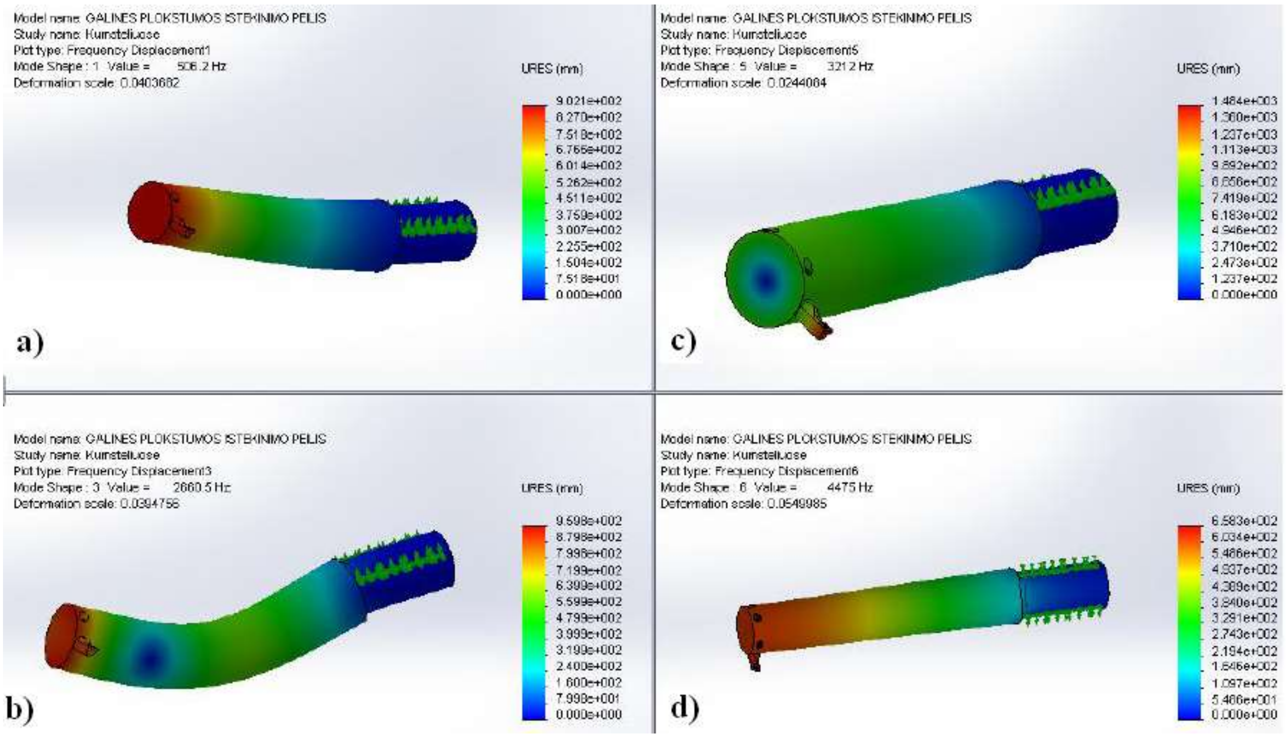

Fig. 3 The modes of natural vibration of boring tool: a) the first natural transverse vibration mode, b) the second natural transverse vibration mode, c) the first natural rotational vibration mode, d) the first natural longitudinal vibration mode

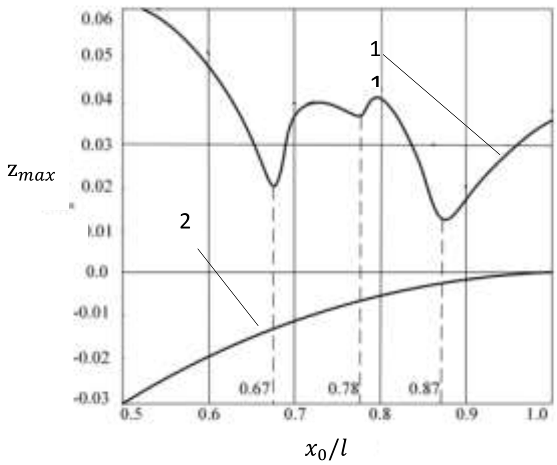

Fig. 4 Dependence of dimensionless rebound amplitude *1* of the tool $z_{max} = y_{max}/l$ on the position of the cutting edge expressed as a ratio between the distance x_0 from the anchor of the tool and its whole length l, where *2* – position of the workpiece

Fig. 3 represents different vibration modes of the boring tool. Natural frequencies depend on structural parameters by changing which it is possible to approach transverse and rotational frequencies by changing structural parameters of boring tool.

Fig. 4 presents the dimensionless dependence of the maximum amplitude of the post-impact rebound $z_{max} = y_{max}/l$ on the position of the tool cutting edge where the smallest rebound amplitudes are obtained when the cutting edge is located at points coinciding with x_0/l = 0.87 or x_0/l = 0.67. A slight decrease in the rebound amplitude is also observed at x_0/l = 0.78. The lower curve in Fig. 4, that asymptotically approaches the axis line, corresponds to the deflection of the free end of the tool during the impact with the workpiece. According to Fig. 2 points x_0/l = 0.87 and x_0/l = 0.67 coincide with particular points of the third mode of transverse vibrations of cantilever, when the point x_0/l = 0.78 with the nodal point of the second mode.

4. Practical realization of self-exiting tool structure

The practical issue of given simulation results could be the modified boring tool structure (Fig. 5) in which the cutting edge is located at the distance 0.87*l* from the anchor point of the tool [8]. The main purpose of this invention is an improvement of cutting conditions by decreasing tool vibration amplitude and consequently increasing frequency. For example, a desire of being able to perform a cutting operation into pre-drilled holes in a workpiece limits the diameter or cross-sectional size of the boring bar during boring when the vibrations are a cumbersome part of the manufacturing process [9]. Usually a bo-

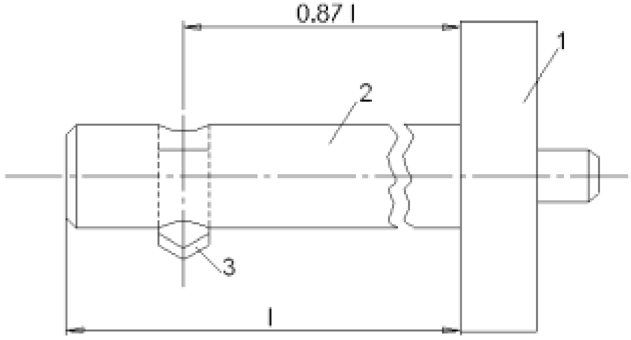

Fig. 5 Modified boring tool structure with cutting edge located in the nodal point of the third natural vibration mode at the distance 0.87*l* from the anchor point. *1* - tool holder, *2* - boring bar, *3* - insert

ring bar is comparatively long and slender, and is thereby more sensitive to excitation forces. Vibrations usually dominate by the first resonance frequency in either of the two directions of the boring bar.

This process usually is not stationary. The vibrations of the boring bar affect the result of machining and surface finish in particular. The tool life is also likely to be influenced by vibrations. The tool structure consists of the end part *1* for the fixation in machine-tool spindle and the cantilever part *2* of length *l* constant cross-section tool holder. At the distance 0.87*l* the cutting insert *3* is fixed. When the tool is cutting the variable force excites vibro - impact motion. As cutting insert is located in the nodal point of the third mode of flexural vibrations of the cantilever shaped structure the third mode of construction vibrations is predominated. This mode is distinguished by lower amplitudes and higher frequencies resulting in vibration cutting regime. Intensification of the vibration energy dissipation in the tool holder material decreases not only the amplitudes of tool, but also the amplitudes of workpiece and machine-tool vibrations.

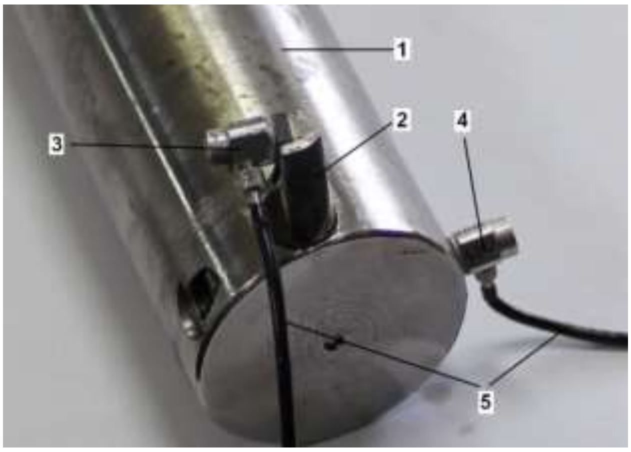

Fig. 6 Practical realization of modified tool structure: *1* - boring tool holder, *2* - cutting insert, *3* - sensor for radial and *4* - for tangential vibrations measurements, *5* - cables

Practical realization of the modified boring tool is presented in Fig. 6 On the boring tool holder *1* the cutting insert *2* could be fixed at the distance, which coincides with the nodal points of the higher modes of flexural vibrations in radial direction. As vibrations usually dominate in either of the two directions of the boring bar two sensors-accelerometers – KD91 (RFT, Germany) *3* and *4* are placed in the two perpendicular planes. Sensor *3* is attached in the same plane as cutting part and is capable to measure vibrations in radial direction as well as sensor *4* in perpendicular direction for tangential direction to cutting surface measurements. The experimental study was carried out with intention to demonstrate tool vibrations. Workpiece from steel 37 was machined using identical cutting parameters with conventional and modified tools: feed f = 0.14 mm/rev, spindle rotation n = 310 rpm, cutting depth a_p = 0.25 mm (Fig. 7).

The vibrations of conventional tool, when cutting insert is located at the free end of the boring bar (*1*-line), are characterized by higher amplitudes and lower frequency than the vibrations of passive tool, modified by fixing cutting insert in the place of third mode nodal point at the distance 0.87 l (*2*-line).

As is indicated in Fig. 7 during boring operations with modified tools the higher modes of transverse vibrations of cutting tools are expressed by higher frequencies and by the few times reduced vibration amplitudes which generate vibration cutting effect.

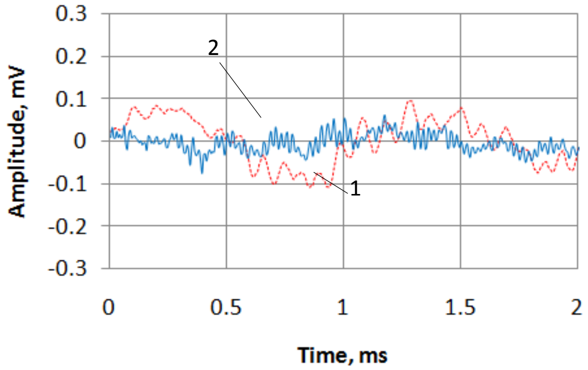

Fig. 7 Boring bar vibrations in direction of radial cutting force in the case of conventional *1* and modified *2* tool with cutting insert fixed in nodal point of the third mode of flexural vibrations at 0.87*l*

As cutting insert is located in the nodal point of the third mode of flexural vibrations of the cantilever shaped structure the third mode of construction vibrations is predominated, which is distinguished by lower amplitudes and higher frequencies resulting in vibration cutting regime.

5. Experimental research of boring tool structure with selected ratio of torsion and transverse vibration frequencies

Other possibility to modification of boring tool is related to the dimensional change of the tool holder parameters coinciding the first frequency of torsion to the second one of transverse vibrations. For intensification of the second mode of flexural vibrations of boring bar the cutting insert should be placed at the distance 0.78*l*, which coincides with the nodal point of the second mode of flexural vibrations in radial direction. Fig. 8 illustrates distinguished increase of tangential vibrations amplitudes, when cutting insert is fixed at the point 0.78*l* (*2*-line) accordingly to vibration amplitudes of conventional tool (*1*-line).

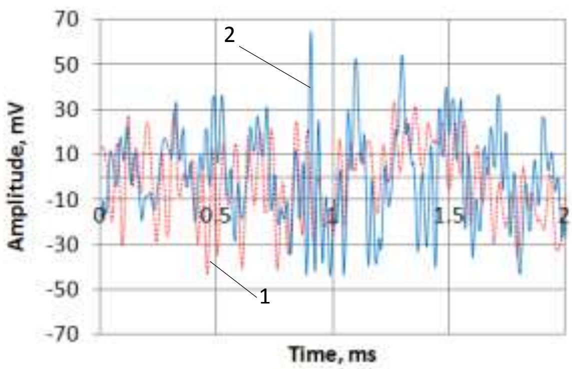

Fig. 8 Boring bar vibrations in direction of tangential cutting force case of conventional *1* and modified *2* tool with cutting insert fixed in nodal point of the second mode of flexural vibrations at 0.78*l*

It means that this is the rotational resonance case, which could be useful for the reduction of transverse vibrations in radial direction. The coincidence of two natural frequencies initiates the intensification of the first mode of rotational vibrations and the second mode of flexural as well as dissipation of energy in the tool holder material (2-line) decreasing consequently the transverse vibration amplitudes.

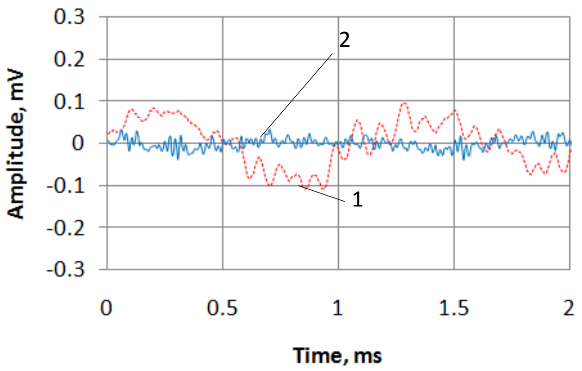

Fig. 9 Boring bar vibrations in direction of radial cutting force in the case of conventional *1* and modified *2* tool with cutting insert in nodal point of the second mode of flexural vibrations at 0.78*l*

Fig. 9 confirms this statement, because the vibration amplitudes of boring bar decrease few times (*2* - *l*ine) when cutting insert is located at the distance 0.78*l*, in the nodal point of second mode of transverse vibrations of boring bar. The coincidence of two natural frequencies initiates the intensification of second mode of transverse vibration amplitudes and consequently dissipation of energy in the tool holder material (*2* - line).

6. Conclusions

Modified means related to tools structural changes are proposed for excitation of high frequency vibrations during boring. Cutting tool as a flexible structure that is characterized by several modes of natural vibrations and intensification of some of them. Intensification of the higher vibration modes increases tool vibration frequency, which becomes similar as during vibration cutting, and decreases tools cutting part vibration amplitudes assuring improvement of surface finish. The experimental settings demonstrate better tool behavior, decrease of vibration amplitudes, justifying numerical analysis results and presumptions for controlling tool vibrations during cutting process, with modifications made for the structure of boring tool, on the basis of computational results. The structural changes of tools and possibilities to excite higher modes are related to the modification of tools structure by fixing cutting insert in the nodal point of laterally vibrating tool holder or approaching the first frequency of torsion of the tool holder to the second one of transverse vibrations. This, in turn, has important practical implications since the presented approach of modified tool mode control is relatively simple to implement in industrial application as it does not require sophisticated control devices.

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MODIFIKUOTOS ĮRANKIO GEOMETRIJOS EFEKTYVIAM PJOVIMUI

R e z i u m ė

Šiame straipsnyje aptariami tyrimų rezultatai, kurie remiasi prielaida, jog įrankio lanksti geometrija yra charakterizuojama keliomis savųjų virpesių modomis. Ištekinimo proceso metu įrankio aukštesnio dažnio virpesių formų susižadinimas lemia mažesnę įrankio virpėjimo amplitudę, o tai laidžia pasiekti geresnę apdirbamo paviršiaus kokybę bei padidinti energijos sklaidą įrankio medžiagoje, ko pasėkoje pats įrankis tampa efektyviu slopintuvu teigiamai veikiančiu įrankis – ruošinys sistemos virpėjimą pjovimo proceso metu. Padidėjęs pjovimo efektyvumas taip pat siejamas ir su galimybe išvengti įrankis – ruošinys sistemos rezonanso zonų.

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MODIFIED TOOL STRUCTURES FOR EFFECTIVE CUTTING

S u m m a r y

The main idea of the reported research work is based on treating cutting tool as a flexible structure characterized by several modes of natural vibrations, structural modification of which leads to the self-excitation of higher vibration modes during cutting. The intensification of the higher natural vibration modes results in higher frequencies and lower vibration amplitudes of cantilever type cutting tools generating vibration cutting effect, that gives the better quality of the treated surface. Consequently the intensification of the higher modes increases the magnitude of internal energy dissipation inside tool material and thereby makes the tool, without any passive vibration absorber, a more effective damper, which positively influences the amplitudes of workpiece or machine tool itself, providing the possibility to reduce chatter. Increase in cutting effectiveness related with the intensification possibility of cutting regimes avoiding the tool-workpiece resonance zones.

Keywords: flexible structures, vibration modes, vibro-impact motion.

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