690. Passive optimal tool structures for vibration cutting

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Abstract. Higher surface quality of the machined workpiece is obtained when the cutting tool is excited at high-frequency vibrations that are superimposed on its continuous movement. Traditionally excitation of high-frequency vibrations at the tool cutting edge needs special equipment, which poses challenges for implementation of such vibrational cutting tool in industrial environment. More effective could be the passive way with respect to structural changes of the tool. The main idea is based on the intensification of higher vibration modes of a cutting tool as a flexible structure. Higher modes are characterized by higher frequencies and lower vibration amplitudes, which induces vibrational cutting effect that results in a better quality of the machined surface. Furthermore, intensification of a higher mode increases the magnitude of internal energy dissipation inside tool material and thereby makes the tool a more effective damper, which positively influences the amplitudes of workpiece or machine tool itself, providing the possibility to reduce chatter.

Keywords: vibration mode, numerical analysis, vibro-impact motion, optimal design.

1. Introduction

In manufacturing industry vibrations induced by metal cutting are of great concern. Such operations as drilling and turning are facing complex vibration-related problems. Vibration problem has a considerable influence in metal cutting on important factors like productivity, production costs, etc. Possibility to control vibration process in different equipment is one of the approaches for increasing efficiency during machining operation. Search for more effective cutting methods revealed that machining quality could be improved by decreasing tool vibration amplitudes. For many years the usual practice for the improvement of surface quality of machined parts was based on stiffening the structure of machine tool, fixing devices and the tool itself by means of increasing the cross-sections of the structural elements. However, in some cases the latter approach does not provide the required final result or is impossible. Vibration cutting process, so called ultrasonically assisted manufacturing process, is a technique for improving machining operations, where high-frequency vibrations (approximately of 20 kHz) with amplitude of 10 micrometers are superimposed on the continuous movement of the cutting tool. Compared with conventional machining this technique allows significant improvements in machining intractable materials such as hard metal alloys, brittle plastics, high-strength aerospace alloys, composites and ceramics. The high accuracy in vibration cutting is the result of reduction in elastic deformation of both the cutting tool and workpiece, as well as reduction in cutting heat and work material adherence to the cutting edge of the tool [1]. Vibration cutting sets new standards for machining time, contour accuracy and surface quality.

Vibration drilling takes place when ultrasonic vibration is superimposed on the relative cutting motion between a drill bit and the workpiece being drilled. A reduction in cutting forces, an increase in penetration speed, and elimination of burrs are among the main benefits of vibration drilling. It was found that the application of ultrasonic vibration significantly increased the penetration rate of the drills. In some cases, the penetration rate was increased by a factor of four depending on the rotational speed [2]. Deep hole machining is one of the most complex manufacturing processes. Adverse conditions of chip formation, problems of chip removal, low stiffness of tool and its special design, and the impossibility of observing the tool during machining make the deep hole drilling one of the most difficult operations. The main feature for holes of small diameter machining is the difficulty of removal of chips from the cutting region. Naturally, the chip removal becomes much easier in case of small crushed chip. Among the methods of chip fragmentation the vibration drilling is the most efficient one [3]. As the drill vibrates in torsion, it lengthens and shortens periodically, causing a wavy surface formation on the bottom of the drilled hole [4]. In [5] a drill bit tool is considered as two-degree of freedom system that can vibrate in the axial and torsional directions. The dynamical system is considered with parametric excitation where the torsion motion plays an important role, as it essentially affects the process of chip formation.

There are three independent principal directions in which vibration cutting can be applied during turning process: feed or horizontal direction, direction of cutting velocity (or tangential), and radial direction [6]. Using vibration turning some modern materials could be treated by universal lathe [7, 8]. Possible advantages of applying vibration simultaneously both in tangential and radial directions have also been explored. Efficient vibration cutting, related to the substantial decrease in cutting forces, as an improvement in surface finish up to 50 % compared with conventional turning occurs when vibration is applied in the direction of the cutting velocity, but great influence on this process has the variation of workpiece diameter, as during the finishing operations. The reason for this is that the conventional cutting process has been transformed into a high-frequency vibro-impact process, which, in turn, increases the dynamic stiffness of the lathe-tool-workpiece system as a whole and improves the accuracy of turning. Also, abolishing the built-up-edge through the application of high-frequency vibration at low cutting speeds helps to reduce the surface roughness [9]. However, application of vibration along the feed direction enables the cutting parameters used in manufacturing industry for most materials to be reached independently of the workpiece diameter. Therefore tool vibration in the feed direction seems to be more suitable for industrial vibration turning requiring high levels of productivity and it is named as sweep cutting [9].

Studies, carried out by the research group of the author in the field of vibration turning [10], suggested new ideas for improving performance of cutting processes and encouraged to perform a more thorough investigation of associated dynamic phenomena.

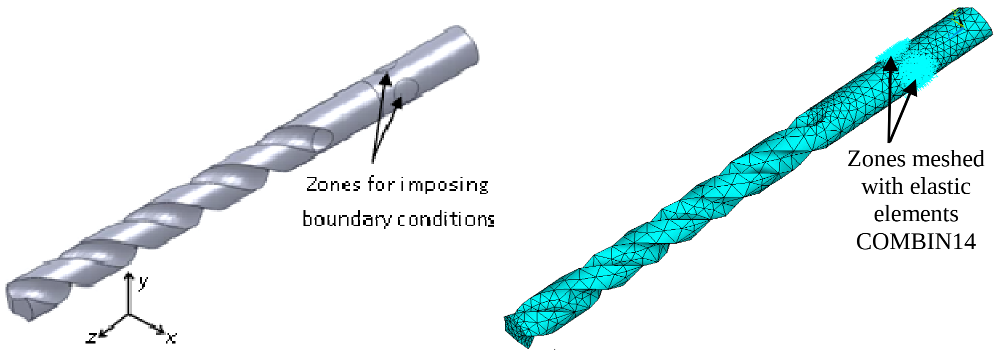
The main idea of the reported research work is based on the establishment of special conditions for cutting tool as flexible structure for intensification of the higher natural vibration modes. Reduction of magnitude of unwanted deleterious vibrations generated during machining may result in a better quality of the treated surface. 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.

2. Optimization of structures conforming to vibration laws

The aim of optimization is to select such geometrical parameters that would correspond to the technical characteristics of the structure and give a minimum value to a certain quality functional or target function [11].

In optimal design, first of all the constraints, geometrical and structure performance should be distinguished. The target functions, most frequently used in optimal design, express the structure mass minimized with respect to the constraints of a prescribed vibration frequency. It is desirable to design a structure, with the natural frequencies which would not fall in a certain interval ($\omega_{\min}$, $\omega_{\max}$).

The twist drill structure could be considered as a cantilever, but accomplishing torsion vibrations (Fig. 1). We will minimize the cantilever structure mass at a prescribed frequency of torsional vibrations. The most applicable method for the optimization could be the gradient projection in state variables space method of non - linear programming. For this purpose constraints are expressed by the form of inequalities. While applying this method we use the information only about the first derivatives or gradients. Let's denote design variables as A_1, A_2,…A_n or in vector form $A = [A_1, \quad A_2, \quad ..., \quad A_n]^T$, where A_i - are cross-sections of the FE of cantilever depending on its FE diameters d_i. During optimization we consider that the cross-sections A_i are constrained by the value $\underline{A_i}$ from below, where $i=1, 2,\ldots,n$.



(a) (b)

Fig. 1. Twist drill models: a) SolidWorks model of a pre-twisted cantilever with the delineated zones where the structure is imposed with appropriate boundary conditions, b) ANSYS finite element model with the designated zones that are meshed with elastic link elements (zones of spring elements)

The target function of structure optimization could be expressed as:

$$\Phi(A) = \min \rho a(A_1 + A_2 + ... + A_n) . \qquad (1)$$

If the mesh of FE is chosen of constant length, the target function could be rewritten in following form:

$$\Phi(A) = \min(A_1 + A_2 + ... + A_n) .$$

The cross-section of the structure is constrained from below, so it is necessary to constrain the cross-sections of all FE:

$$\Psi_1(A) = 1 - \frac{A_1}{\underline{A_1}} \le 0;$$

$$\Psi_2(A) = 1 - \frac{A_2}{\underline{A_2}} \le 0;$$

....

$$\Psi_n(A) = 1 - \frac{A_n}{\underline{A_n}} \le 0. \qquad (2)$$

As the fixed natural frequency of the structure is not less than ω^* so the constraints are also applied to it:

$$\Psi_{n+1}(\omega) = 1 - \frac{\omega}{\omega^*} \leq 0, \quad (3)$$

where ω^* - given frequency of the structure, ω - natural frequency of the mode of vibrations under investigation.

By solving the eigenproblem:

$$K(A)\theta = \xi M(A)\theta, \quad (4)$$

where $K(A)$, $M(A)$ - the stiffness and mass matrices of the structure, θ - the matrix of eigenvectors in torsion, we obtain eigenvalue vector ξ.

As equation (4) is homogenous to the eigenvector, it is possible to express $\theta^T M\theta = 1$. Equation (4) describes structure by its eigenvalues. Natural frequency can be found by solving expression $\omega = \sqrt{\xi}$.

During calculations the constraints activity should be verified at each iteration. For this purpose (2) and (3) expressions are constrained by the constants ε^A and ε^ω. Then we obtain:

$$\Delta\Psi_i(A) = 1 - \frac{A_i}{\underline{A}_i} \leq -\varepsilon^A, \quad i = 1, ..., n;$$
$$\Delta\Psi_{n+1}(\omega) = 1 - \frac{\omega}{\omega^*} \leq -\varepsilon^\omega. \quad (5)$$

Constraints like $\varepsilon^A < \Psi_i(A)$ are called active and should be considered, because they are almost violated. If they are not regarded the oscillations could occur during other iterations.

During each iteration the matrix of active constraints gradients is constructed. The matrix columns of design variables active constraints are identified from expression:

$$e^i = \frac{\partial \Psi_i^T}{\partial A}, \quad (6)$$

where i - the index of violated constraints $\Delta\Psi_j$, j=1, 2, … , n.

The components of vector e^i are the sensitivity coefficients of corresponding design variables. Based on the design variables, these vectors define the derivatives of the target function and the constraints. They are useful for the designer because they help to identify the impact of design variables on the target function and the constraints. If component e^i is positive, then Ψ_i grows with the increase of A_j. If e^i is negative – A_j reduces Ψ_i. The magnitude of sensitivity coefficients e^i informs the designer which design variables have a significant impact on Ψ_i and vice versa.

When the number of violated constraints is 1, …, n, the columns of the matrix acquire the following form:

$$e^1 = \begin{bmatrix} -1/\underline{A}_1 \\ 0 \\ 0 \\ \ldots \\ 0 \end{bmatrix}, e^2 = \begin{bmatrix} 0 \\ -1/\underline{A}_2 \\ 0 \\ \ldots \\ 0 \end{bmatrix}, \ldots, e^n = \begin{bmatrix} 0 \\ 0 \\ \ldots \\ 0 \\ -1/\underline{A}_n \end{bmatrix}.$$

The gradient of state variable $\Delta\Psi_{n+1}$ is defined by dependency

$$e^{iT} = \left[\frac{\partial}{\partial A} \left\{ \theta^{iT} K(A) \theta^i \right\} - \omega \frac{\partial}{\partial A} \left\{ \theta^{iT} M(A) \theta^i \right\} \right] \frac{\partial \Psi_i}{\partial \omega}. \quad (7)$$

The vector of the target function can be defined as follows:

$$e^0 = \frac{\partial \Phi}{\partial A}, \quad (8)$$

or in our case we get:

$$e^0 = \begin{bmatrix} 1 \\ 1 \\ \dots \\ 1 \end{bmatrix}.$$

Lagrange multipliers, denoted by vectors λ_1 and λ_2, are defined by the matrix of active constrained gradients (5) and the vector of target function gradients (7):

$$\begin{aligned} e^T W^{-1} e \lambda_1 &= -e^T W^{-1} e^0; \\ e^T W^{-1} e \lambda_2 &= -\Delta\overline{\Psi_n}, \end{aligned} \tag{9}$$

here W – the matrix of design variables; $\Delta\overline{\Psi}$ – the vector of active and violated constraints.

The vectors of variations of design variables are identified based on Lagrange multipliers:

$$\begin{aligned} \delta A^1 &= W^{-1}(e^0 + e\lambda_1); \\ \delta A^2 &= -W^{-1} e \lambda_2, \end{aligned} \tag{10}$$

where δA^1 corresponds to the decrease direction of the target function under constraints and δA^2 corresponds to the required correction of the constraints.

It is not difficult to find parameter γ, used for estimating the size of the step:

$$\gamma = -e^{0T} \delta A^1 / 2\Delta\Phi , \tag{11}$$

here $\Delta\Phi$ is the variation of the target function, estimated by the expression:

$$\Delta\Phi = -\alpha \sum_{i=1}^{n} A_i ,$$

here α is the coefficient of the target function decrease in percents.

The step of target function decrease corresponds to the percentage reduction of target function in every iteration.

Further, the vector multiplier η is estimated:

$$\eta = \lambda_1 + 2\gamma\lambda_2 .$$

If all components of vector η, corresponding to active constraints, are not negative, then the solution satisfies Kun-Taker conditions [11]. Otherwise, if some components of η_i are negative it means that the target function $\Phi(A)$ acquires the value which is bigger than the minimum and the results can be improved by discarding corresponding constraints. This leads to the reduction of a number of active constraints, thus a new matrix e is composed and the multiplier η, corresponding to the remaining active constraints, is defined. The process is repeated until all η components become positive.

Variation δA is estimated from the expression:

$$\delta A = -\frac{1}{2\gamma} \delta A^1 + \delta A^2 .$$

And the new vector of the design parameters acquires the following appearance:

$$A^1 = A^0 + \delta A .$$

It is worth noticing the fact that when A^0 satisfies all constraints, no other variation exists at point A^0 which would violate constraints and reduce Φ (A). It means that A^0 is the relative minimum point in non-linear programming task.

Using this algorithm the optimal configuration of the cantilever beam-shaped structure is obtained with respect to the second torsional vibration frequency (Fig. 2). It can be considered as a periodical structure, where minimum and maximum cross-sections vary every 1/3 length of cantilever.

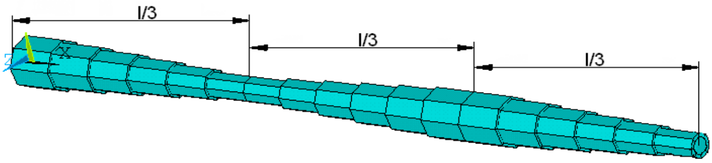


Fig. 2. Optimal configuration of the cantilever beam-shaped structure obtained for the given second torsional vibration frequency could be considered as periodical structure in which minima and maxima cross-sections vary every 1/3 length of cantilever

At a prescribed second frequency of transversal vibrations, the structure mass is minimized in an analogous way as for torsional vibrations. Fig. 3 illustrates an optimal structure obtained for the prescribed second transverse mode of the cantilever.

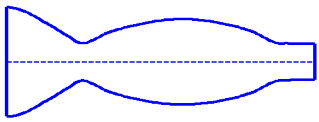


Fig. 3. Optimal cantilever at the second natural frequency of transverse vibrations

From sketch in Fig. 3, it is not difficult to measure the distances from the minimum cross-sections to the fixing site. In our case, the minimum cross-section of the structure, which is optimal with respect to the second transverse mode, is located at the distance corresponding to the 0.24*l* from the fixing site in the left side.

3. Identification of dynamical properties of the optimally shaped cantilever

Suppose we excite torsional vibrations of the cantilever. As we see from Fig. 4, the frequency range of torsional vibrations of constant cross-section structure of the diameter d = 10 mm and length of l = 100 mm could be described by the sequence of natural frequencies of torsional vibrations at 9 kHz, 20 kHz, 30 kHz, etc.

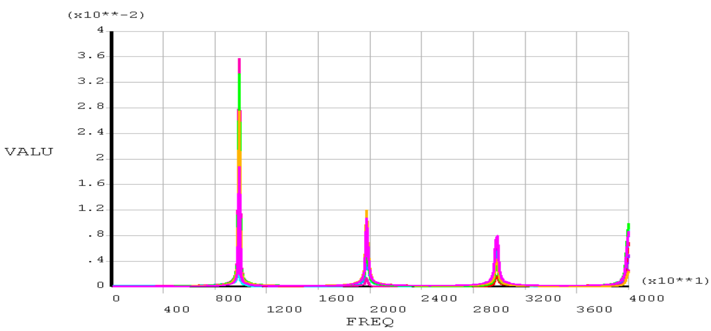


Fig. 4. Frequency range of torsion induced vibrations of constant cross-section structure of the diameter d = 10 mm and length of l = 100 mm, characterized by the sequence of natural frequencies of torsion at 9 kHz, 20 kHz, 30 kHz, etc

The intensity of vibrations changes with the cantilever of optimal configuration (Fig. 5). The excitation of such structure by the wide frequency range, as occurs during cutting, gives leap of torsional vibrations on the second mode at 28 kHz frequency, when the amplitudes of the first mode of optimal structure at 6 kHz frequency are minimal. Another phenomenon is observed during simulation, which is related to reduction of lateral vibrations of the rotating structure. It could be explained by the increase of the dynamic stiffness of the structure due to the more intensive rotational deformations of FE cross-sections. This phenomenon could be useful for long tools structures with strict constraints for lateral deviations during cutting.

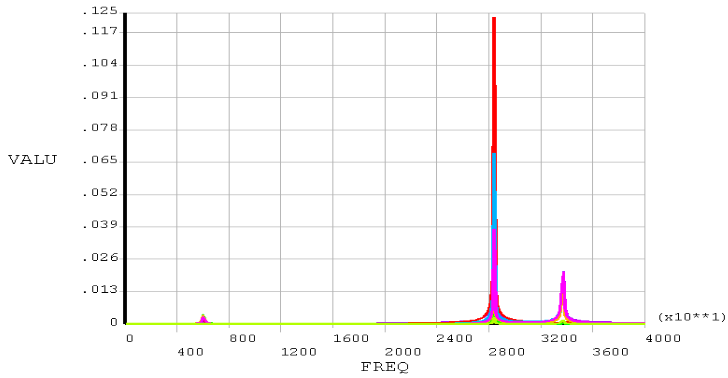


Fig. 5. Frequency range of torsion induced vibrations of optimal configuration cantilever beam-shaped structure obtained for the given second frequency of torsional vibrations with the pronounced second torsional mode at frequency of 28 kHz, when the first mode of this structure at 6 kHz is minimal

4. Invented passive optimal drill

Structure optimized with reference to the established criteria and constraints may turn out to be irrational for practical use, especially for drilling tools. For instance, if the technological requirements are neglected and not included in the constraints, there is a possibility that the developed structure will be optimal in terms of mass, but irrational in the technological aspect. For this purpose the rational structure from the technological point of view of drilling tool has been proposed (Fig. 6) and patented [12]. This structure, as the optimal one in Fig. 6, has periodically changing cross-sections, which coincide with the structural elements of twist drill such as shank 1, neck 2 and body 3, when the outer surfaces of these parts are neither convex nor concave, but cylindrical. As the drill vibrates in torsion, it lengthens and shortens periodically in axial direction, thus inducing vibration cutting regime. In paper [13] a new force model of torsional-axial and transverse vibration for drilling has been proposed, which is validated experimentally. Simulation of torsional-axial vibration is conducted by means of Bayly's model, which is based on the fact that when the twist drill „untwists", it extends in length.

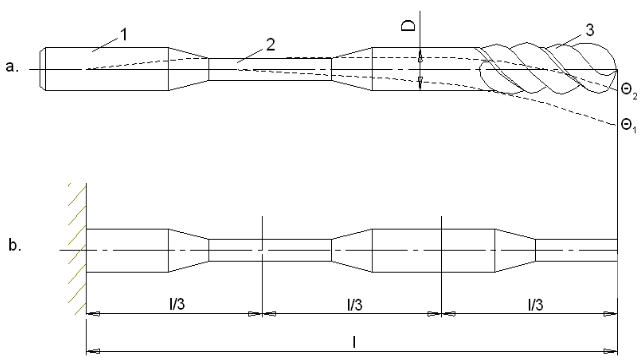


Fig. 6. Twist drill structure: a) passive close to optimal cantilever configuration, b) for the given second torsional vibration frequency with periodically changing minimum and maximum cross-sections diameters varying every 1/3 of cantilever length

5. Testing results

4-component dynamometer platform KISTLER 9272 was used for measuring the magnitudes of axial cutting force and torque that are generated during the drilling process. Cylindrical workpieces were mounted on the clamping device of the dynamometer, while the latter was installed on the desk. Cutting force and torque during drilling operations were measured, registered, the signal was transmitted to the computer, where a special software was used for signal analysis (Fig. 7). From Fig. 7 slight decrease of cutting force and cutting moment is visible for similar to the optimal cross-section drill (Fig. 6). Moreover, the decrease of amplitudes of the first mode of torsional vibrations of that drill is evidently expressed by cutting force oscillations, when the higher frequency second mode generation leads to more stable soundless cutting regime. It confirms the results of paper [2]: application of ultrasonic vibration significantly increases the penetration rate of the drills.

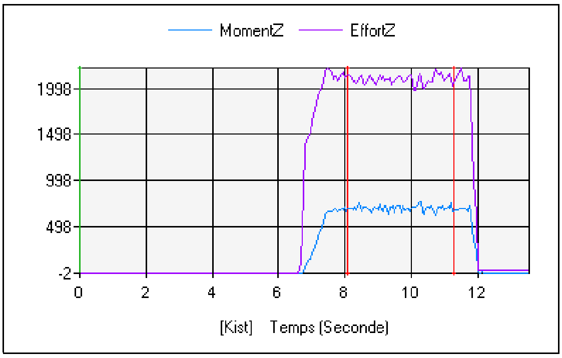


Fig. 7. Cutting force and torque variation during drilling operations with the drill having configuration, which is characterized by a cross-section that is close to the optimal one

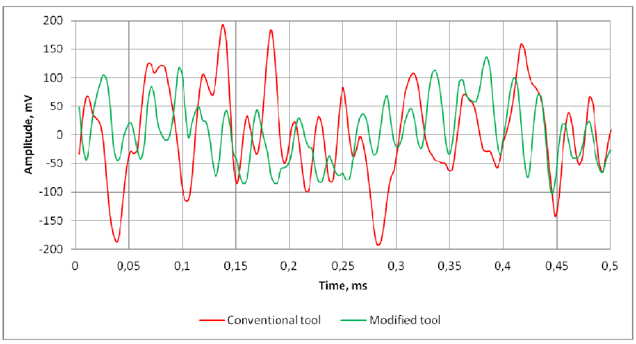


Fig. 8. Vibration curves of cutting part of the conventional (in red) and passive (modified) optimal (in green) turning tools

Using the same principals of design the modified structure of turning tool has been prepared similar to the optimal cantilever for the given second natural frequency of transverse vibrations (Fig. 3) with decreased cross-section at the distance 0.24*l* from fixing site. The turning experiments with aluminum, steel and stainless steel shows decrease by 10 percent of surface roughness, and more stable vibrations during cutting (Fig. 8). Presented passive tool structure during cutting self-excites at the higher resonance frequencies (in green) with respect to conventional turning tools (in red). The behavior of modified passive tool during cutting is characterized by the increased magnitude of internal energy dissipation inside tool material 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.

6. Conclusions

Passive means related to structural changes of the cutting tool are proposed for excitation of high-frequency vibrations during drilling and turning. The approach is based on consideration of the 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 to the case of vibration cutting, and decreases vibration amplitudes of the cutting part, thereby assuring improvement in surface finish. The structural changes of tools and possibilities to excite higher modes are related to the modification of tools structure so as to approach the optimal one with respect to the given higher mode of natural vibrations. Some presented tool designs are patented and have been tested by users with positive feedback. This, in turn, has important practical implications because the proposed approach of tool mode control is relatively simple to implement in industrial environment as it does not require sophisticated control devices.

Acknowledgements

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References

[1] **Murakawa M., Jin M.** Turning of beta titanium alloys by means of ultrasonic vibration. Trans. NAMRI/SME, 1998, 26, 153.

[2] **Astashev V. K., Babitsky V. I.** Ultrasonic Processes and Machines Dynamics, Control and Applications. Berlin: Springer, 2007.

[3] **Stephenson D. A., Agapiou J. S.** Metal Cutting Theory and Practice. New York: Marcel Decker, 1997.

[4] **Baylay P. V., Metzler S. A., Shaut A. J., Young S. G.** Theory of Torsional Chatter in Twist Drills: Model, Stability Analysis and Composition Test. ASME J. Manuf. Sci. Eng., Vol. 123, 2002, p. 552-561.

[5] **Voronov S. A., Gouskov A. M., Kvashnin A. S., Butcher E. A., Sinha S. C.** Influence of Torsional Motion on the Axial Vibrations of Drilling Tool. Transactions of the ASME, Vol. 2, 2007, p. 58-64.

[6] **Nath C., Rahman M., Andrew S. S. K.** A study on ultrasonic vibration cutting of low alloy steel. J. Mater. Processing Technol., 2007, p. 159-165.

[7] **Babitsky V., Kalashnikov A., Meadows A., Wijesundra A.** Ultrasonically assisted turning of aviation materials. J. Mater. Processing Technol., 2003, Vol. 132, p. 157-167.

[8] **Mitrofanov A., Babitsky V., Silberschmidt V.** Finite Element Modelling of Ultrasonically Assisted Turning of Inconel 718. J. Mater. Processing Technol., 2004, p. 233-239.

[9] **Sharma V. S., Dogma M., Suri N. M.** Proc. IMechE, Part B: J. of Engineering Manufacture, Vol. 222.

[10] **Ostasevičius V., Gaidys R., Rimkevičienė J., Daukševičius R.** An approach based on tool mode control for surface roughness reduction in high-frequency vibration cutting. Journal of Sound and Vibration, Vol. 329, Issue 23, 2010, p. 4866-4879.

[11] **Haug E. J., Arora J. S.** Applied optimal design: mechanical and structural systems. New York: Wiley, 1979.

[12] **Ostasevičius V. et. al.** Twist drill with variable cross-section. SU invention No. 1357151.

[13] **Imani B. M., Moosavi S. G.** Time domain simulation of torsional-axial and lateral vibration in drilling operation. Proceedings of International Conference on Application and Design in Mechanical Engineering, Penang, 2009, p. 9G1–7.