FEM NUMERICAL ANALYSIS OF TOOL HOLDER OF TURNING LATHE

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Abstract. Finite element modeling provides a method in understanding vibration processes. A study of a tool holder by a numerical Finite Element Method (FEM) was conducted. To analyze the modal properties of the tool holder, a 3D finite element model was developed using ANSYS software. It was found that decrease of the amplitude of a cantilever structure in vibro-impact motion depends on the location of the supports. In the experimental modal analysis, natural frequencies and mode shapes were measured for static conditions. The finite element model is validated by comparing the numerical to experimental results. The cantilever type tool holder tends to function as an effective damping element, which abates detrimental influence of unwanted vibrations. This phenomenon could be useful for cantilever type cutting tools, when it is necessary to increase the quality of the machining surface by decreasing the amplitudes of cutting edge vibrations.

Keywords: tool holder, FEM analysis, vibration.

Introduction

The stability of a cutting process directly influences the quality of a final surface. The control of the cutting process is an important problem for machining technology. Instabilities usually manifest as harmful chatter vibrations generated during cutting. Recently many studies have been conducted on this problem by focusing on the conditions of appearance of such unwanted vibrations [1; 2].

Undesirable vibrations can be reduced by either using passive or active damping control methods. The passive methods can be implemented by increasing damping constant or by modifying the structure by adding constrained layer damping, damped links, tuned mass dampers, interface damping, solid-spacer damping, etc. The passive vibration control strategy can be quite effective and desirable for its simplicity [1]. One of the approaches is smart tools which reduce forced vibrations, chatter and heat during machining. The Smart Tool proposed by B.-K. Min et al. has a fast tool servo that utilizes a piezoelectric actuator and two laser photo sensors to actively isolate the cutting insert from erroneous bar motions while rejecting cutting force disturbances. The purpose of the tool tip servo is to isolate the cutting insert from erroneous boring bar motions (i.e., deviations from a perfectly cylindrical path) while rejecting cutting force disturbance. The servo has to be fast enough to compensate for at least the forced vibration due to spindle rotation at 6000 rpm.

The majority of the studies dealing with the cutting process of the turning lathe are mainly focused on the analysis of tool wear and dynamic cutting force simulation [4; 5]. The review of this research indicates that the numerical simulation of this process is conditioned upon a number of assumptions and uncertainties as well as highly dependent on various factors such as the material properties, cutting regimes, geometric parameters of a tool, etc. [1; 2].

A method to detect the presence of cross section changes in beams and to determine their location and size has been developed based on the results of experimental modal analysis [1]. The changes of the natural frequencies and frequency response function amplitudes as a function of the cross section depths and its locations are implemented. In Ref. 2, a comparison between experimental and numerical results of the vibrating cantilever type tool structure led to the conclusion that the best results were obtained when the second flexural mode of the cantilever type tool concentrator is excited.

The implementation of structural modifications by changing the cross section at different locations that intensified higher natural vibration modes resulted in reduction of magnitude of unwanted deleterious vibrations. This suggests, that excitation of higher natural vibration modes could be advantageous, since the amplitude of higher modes becomes more intensive, the energy dissipation inside the structure material increases significantly [3] making the tool holder into a passive damper.

The main focus of this research is based on treating the tool holder of the turning lathe as a flexible circular hallow cantilever structure of which vibrations are characterized by several natural modes. The study presents several sample simulations and experimental verifications of the tool holder vibrations.

Materials and methods

The majority of analytical and numerical methods use modal identification techniques to obtain natural frequencies and mode shapes. The finite element numerical method is used in this paper. Since the cantilever structure has distributed mass, stiffness and other parameters, it is necessary to model the dynamic characteristics of such an elastic structure by including several modes of natural vibrations. The impulse force that is produced during spinning, when the cutting tool breaks away from the surface, has wide band spectral contents that are capable of exciting many natural modes of the cutting tool. The sudden movement of this structural component could be constrained by supports. The developed 2-D finite element model of impacting tool holder (Fig. 1) represents the general case of the vibro–impact system. For numerical calculations, the impacting tool is represented by a 2-D model since flexural vibration modes have a much more significant influence on the vibro-impact process in comparison to torsion modes [3].

The numerical model consists of i = 1, 2..., m linear beam elements located in a single layer and j = 1, 2..., k motion limiters or supports ($0 < k < 2m$) that are located at i = 1,2,...,m nodes. Each beam element has two nodes with three degrees of freedom (DOF) at each: displacement in x- and y-axis directions and rotation in xy plane. The model is meshed manually with a number of finite element m equal to 50, resulting in 150 total DOFs. Impact modeling is based on a cantilever approach and makes use of the Kelvin-Voigt (viscoelastic) rheological model, in which a linear spring is connected in parallel with a damper–the former represents the impact force and the latter accounts for energy dissipation during the impact [7].

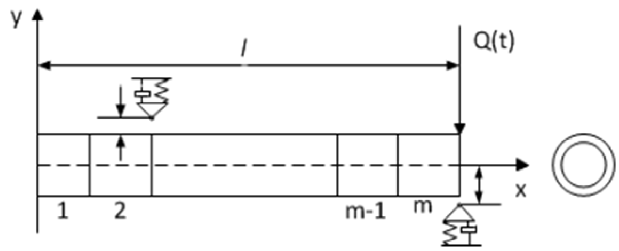

Fig. 1. **Schematic of 2-D finite element model of impacting cantilever**

Model dynamics is described by the following equation of motion given in a general matrix form [7]:

$$[M]\{\ddot{y}(t)\}+[C]\{\dot{y}(t)\}+[K]\{y(t)\}=$$
$$\begin{cases}\{Q(t)\}, if \{\overline{\Delta}^i_j\} > \{y_i(t)\} \cap \{\overline{\Delta}^i_j\} < \{y_i(t)\} \cup f_i(y_i,\dot{y}_i,t) \geq 0; \\ \{Q(t)\}+\{F(y_i,\dot{y}_i,t)\}, if \{\overline{\Delta}^i_j\} \leq \{y_i(t)\} \cup \{\Delta^i_j\} \geq \{y_i(t)\} \cap f_i(y_i,\dot{y}_i,t) < 0,\end{cases} \tag{1}$$

where $[M]$ – mass matrix;
$[C]$ – damping matrix;
$[K]$ – stiffness matrix;
$\{y(t)\}$ – displacement vector;
$\{\dot{y}(t)\}$ – velocity vector;
$\{\ddot{y}(t)\}$ – acceleration vector;
$\{Q(t)\}$ – vector representing the external forces acting on the structure. This vector is used purely as a mechanical load during simulations;
$f_i(y_i,\dot{y}_i,t_i)$ – reaction of the impacting structure;
$\{F(y,\dot{y},t)\}$ – vector of impact interaction between the cantilever microstructure and the support;
Δ^i_j – distance from the i-th nodal point of the structure to the j-th surface of the support located at the corresponding nodal point.

The mass and stiffness matrices are expressed respectively as shown in equations (2) and (3) [5-7].

The obtained results indicate a process of free vibrations of a circular hollow cantilever structure. In order to analyze the modal properties of the tool holder, a 3D finite element model of the hollow circular cantilever was developed using ANSYS software. Numerical finite element models were constructed using the information of geometry and material properties of the structure. The required parameters were chosen to be representative of a real tool holder from the standards [8]. For the numerical analysis, the cantilever with the following parameters has been analyzed: outside diameter is 33.7 mm, inner diameter is 30.5 mm length is 300 mm and cross sectional area is 373.2 mm^2. A cantilever with similar geometric and material parameters was also chosen for experimental study. Tool holders are usually made of high strength steel, so the material parameters for numerical simulation were chosen accordingly as (DIN 1.2210) steel. Its modulus of elasticity is $2 \cdot 10^{11}$ N·m^{-2}, density is 7870 kg·m^{-3}, moment of inertia is 41898.279 mm^4. After creating a geometric model of the holder, the finite element mesh was generated. The structure is meshed by m=50 finite elements and the rigid support is located at the free end of the cantilever (Fig. 1).

$$[M] = \rho \cdot A \cdot L \begin{bmatrix} \frac{1}{6} & 0 & 0 & \frac{1}{3} & 0 & 0 \\ 0 & \frac{13}{35} & \frac{11}{210} \cdot L & 0 & \frac{9}{70} & -\frac{13}{420} \cdot L \\ 0 & \frac{11}{210} \cdot L & \frac{1}{105} \cdot L^2 & 0 & \frac{13}{420} \cdot L & -\frac{1}{140} \cdot L^2 \\ \frac{1}{6} & 0 & 0 & \frac{1}{3} & 0 & 0 \\ 0 & \frac{9}{70} & \frac{13}{420} \cdot L & 0 & \frac{13}{35} & -\frac{11}{210} \cdot L \\ 0 & -\frac{13}{420} \cdot L & -\frac{1}{140} \cdot L^2 & 0 & -\frac{11}{210} \cdot L & \frac{1}{105} \cdot L^2 \end{bmatrix} \tag{2}$$

$$[K] = E \cdot \begin{bmatrix} \frac{A}{L} & 0 & 0 & -\frac{A}{L} & 0 & 0 \\ 0 & 12 \cdot \frac{I}{L^3} & 6 \cdot \frac{I}{L^2} & 0 & -12 \cdot \frac{I}{L^3} & 6 \cdot \frac{I}{L^2} \\ 0 & 6 \cdot \frac{I}{L^2} & 4 \cdot \frac{I}{L} & 0 & 6 \cdot \frac{I}{L^2} & 2 \cdot \frac{L}{I} \\ -\frac{A}{L} & 0 & 0 & \frac{A}{L} & 0 & 0 \\ 0 & -12 \cdot \frac{I}{L^3} & -6 \cdot \frac{I}{L^2} & 0 & 12 \cdot \frac{I}{L^3} & -6 \cdot \frac{I}{L^2} \\ 0 & 6 \cdot \frac{I}{L^2} & 2 \cdot \frac{L}{I} & 0 & -6 \cdot \frac{I}{L^2} & 4 \cdot \frac{L}{I} \end{bmatrix} \tag{3}$$

where E – modulus of elasticity, N·m^{-2};
A – cross sectional area of beam, mm^2;
L – length of element – displacement, m;
ρ – density of cantilever material, kg·m^{-3};
I – second moment of inertia, mm^4.

To perform the experimental modal analysis of the circular hollow cantilever, the beam was installed in the bench (grips) of a planers lathe. The vibrations of the lathe components were measured by PULSE software developed by Brüel&Kjær together with the PULSE data acquisition module 3580, the impact hammer 8206 for creating known or controlled excitation and type 4370 piezoelectric charge accelerometers.

The experimental modal analysis was performed by exciting the circular hollow cantilever that was supported at the free end. The chosen length of the hollow cantilever is 350 mm, the outer diameter is 33.7 mm and the wall thickness is 1.5 mm. The experiment modal analysis was performed at the ASU laboratory.

Results and discussion

The mode shapes provide important information on the vibration response characteristics. The four calculated natural modes are shown in Fig. 2.

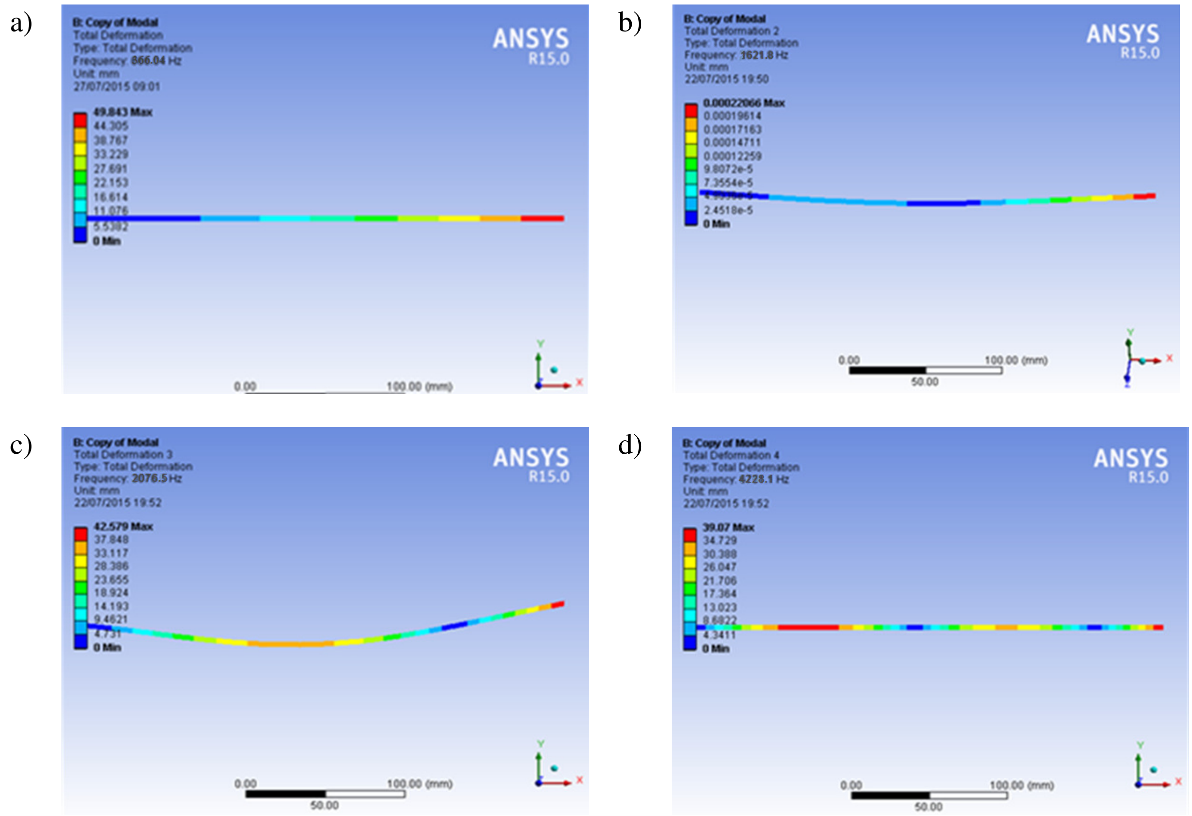

Fig. 2. **Mode shapes of tool holder:** a – first mode shape; b – second mode shape; c – third mode shape; d – fourth mode shape

Fifteen modes of vibration of cantilever beam were calculated during the FEM analysis. The natural frequencies of the rest of the calculated modes obtained from FEM are presented in Table 1.

The obtained numerical results of this analysis show that the amplitudes of free vibration from the impact can be markedly decreased at the free end of the cantilever where the support is located. This is due to the dynamical intensification of the higher modes that increase the cantilever vibration frequency and decrease the vibration amplitudes.

This numerical simulation could be useful for cantilever type cutting tools, when it is necessary to increase the quality of the machining surface by decreasing the amplitudes of the cutting edge vibrations.

Table 1

Natural frequencies of tool holder

Mode number	**Numerical frequency, Hz**	**Mode number**	**Numerical frequency, Hz**
1	366.0	9	13109
2	1622	10	14217
3	2077	11	16128
4	4228	12	16554
5	6751	13	18352
6	7865	14	18984
7	10488	15	20077
8	12682	-	-

In the experimental modal analysis, natural frequencies and mode shapes were measured for static conditions using accelerometers and an impact hammer. The excited frequencies were recorded for static conditions with no load. During the machining process, it is difficult to measure vibration response. The measurements were obtained using the impulse response generated by the impact hammer and the impact force measured by the sensor located at the hammer tip. Only the first two modes were identified experimentally. The frequency response function of the tool holder is shown in Fig. 3.

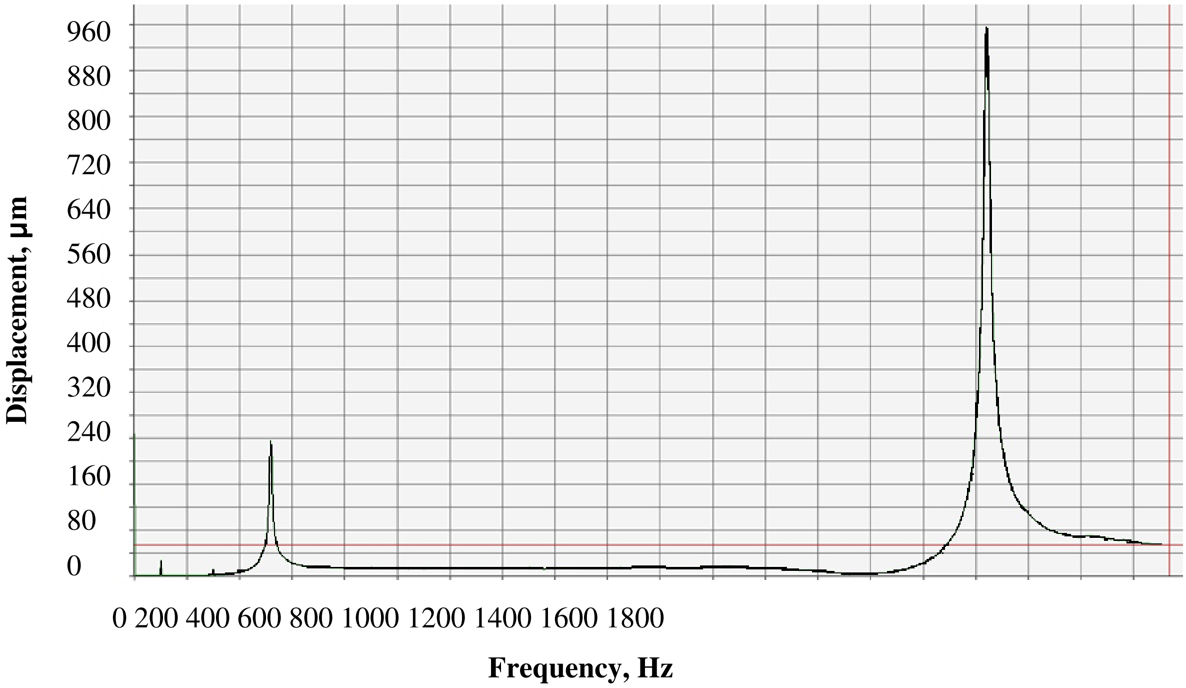

Fig. 3. **Frequency response function (FFT) of tool holder**

Having the frequency response the function, modes, modal frequencies and modal damping of the tool holder of the turning lathe can be obtained.

By comparing the numerical and experimental results it was determined that the numerical model is adequate. This model could be useful for cantilever type cutting tools, when it is necessary to increase the quality of the machining surface by decreasing the amplitudes of the cutting edge vibrations.

Conclusions

1. A numerical vibration analysis by the FEM and experiments were conducted of a tool holder of a turning lathe. It was found that the decrease of the vibration amplitude of the cantilever structure tool holder in the process of vibro-impact motion can be significantly influenced by the location of the support. This is due to dynamic intensification of the higher modes that increases the cantilever vibration frequency and decreases the vibration amplitudes.
2. By comparing the numerical and experimental results, it is concluded that the presented numerical model is adequate. This model could be useful for cantilever type cutting tools, when it is necessary to increase the quality of the machining surface by decreasing the amplitudes of the cutting edge vibrations.

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