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IDENTIFICATION OF SPATIAL DYNAMIC PROPERTIES OF THE BORING BAR BY MEANS OF FINITE ELEMENT MODEL: COMPARISON WITH EXPERIMENTAL MODAL ANALYSIS AND EULER-BERNOULLI MODEL

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Abstract

In metal cutting the boring operation is known to be one of the most troublesome regarding vibration. Boring bars are frequently subjected to vibrations originated from the load applied by the workpiece material deformation process. These vibrations are easily excited due to the boring bars general geometric dimensions, i.e. large length to diameter ratio. Large overhang is usually required to perform internal boring operation and as a consequence the vibration may frequently reach extremely high levels, which result in a poor surface finish, reduced tool life and annoying noise level in the working environment. The vibration problem is directly related to the first bending modes of a boring bar. Therefore investigations of the boring bar's spatial dynamic properties are of a great importance. The results from experimental modal analysis show that a conventional analytical approach - calculation of boring bar eigenfrequencies using an Euler-Bernoulli model - results in rough estimates. This can be explained by existing nonlinearities introduced e.g. in the areas of contact between the boring bar and the clamping bolts as well as the clamping house, which is not considered in the analytical model where the boring bar instead is assumed to be rigidly clamped. Therefore the estimation of the eigenfrequencies and eigenmodes of a boring bar based on a 3-D finite element model of the clamped boring bar incorporating contact between the bar and the bolts respective the clamping house is a more beneficial strategy. This paper addresses the estimation of the boring bar's first eigenfrequencies and corresponding eigenmodes based on the 3-D finite element model. The results are compared with results obtained both from experimental modal analysis and an analytical Euler-Bernoulli model.

INTRODUCTION

In the modern manufacturing industry the problem of noise and vibration in the internal turning operation, in particular, remains to be considerable. In internal turning operations boring bar vibrations frequently arise under the random excitation applied by workpiece material deformation process at the frequencies correlated to the first eigenfrequencies of the boring bar which correspond to the fundamental bending modes. Investigations in this area [6, 7] showed that boring bar vibrations are dominating at the boring bar's eigenfrequency in the cutting speed direction since in this direction cutting force has the largest component. In order to reduce the productivity and working environment degrading vibration problems in internal turning the boring bar vibrations should be suppressed. This can be achieved by using different techniques, e.g. active and passive control [1, 8]. The level of success of utilizing methods for the reduction of tool vibration is closely related to the knowledge of the dynamic properties of the tooling structure - the interface between the cutting tool or insert and the machine tool - involved [1].

Estimation of the dynamic properties of boring bars can be done using different approaches, the most common are experimental modal analysis, analytical modeling (Euler - Bernuolli and Timoshenko models) and numerical modeling (finite element model).

A previous investigation concerning dynamic properties of the boring bar showed that an analytical Euler-Bernoulli model overestimates eigenfrequencies of the boring bar compared to the results obtained by experimental modal analysis [5]. This fact was explained by difference between the boundary conditions used in the analytical model, where the boring bar was assumed rigidly clamped, and the actual nonlinear boundary conditions of the boring bar, i.e. contact between clamping house, bolts and the boring bar. The examinations of the boring bar response in a continuous boring operation revealed also existing nonlinearities in the system "boring bar - clamping house" [6]. Better correspondence between eigenfrequencies obtained experimentally and analytically can be achieved by linearizing the actual boundary conditions - reducing the gap between the boring bar and clamping house [5]. However if more accurate estimate is desired without the changing the system the boring bar can not be considered rigidly clamped, which means that modifications has to be done in the Euler-Bernoulli model or another model should be considered.

The approach of identification of the boring bar's spatial dynamic properties based on numerical modeling, in particular 3-D finite element model, reveals itself to be more advantageous. It gives a possibility to create boring bar and clamping house models which follow the geometry of the real objects. It allows combining these two models and including nonlinear boundary condition, i.e. contact between the boring bar, bolts and clamping house.

The paper addresses identification of spatial dynamic properties of the boring bar using experimental modal analysis, Euler-Bernoulli model and 3-D finite element model with the focus on the last model. The results obtained using three approaches are compared and discussed.

MATERIALS AND METHODS

The experimental setup used in the experimental modal analysis, physical properties of the boring bar material and the methods of identification of the boring bar spatial dynamic properties are described in this section.

Measurement Equipment and Experimental Setup

The experimental modal analysis was carried out on a MAZAK 250 Quickturn lathe. It has 18.5 kW spindle power, a maximum machining diameter of 300 mm and 1007 mm between the centers. The following equipment was used to carry out experimental modal analysis: 14 PCB 333A32 accelerometers, 2 Ling Dynamic Systems shakers v201, 2 Brüel & Kjær 8001 impedance heads, HP VXI E1432 front-end data acquisition unit, PC with IDEAS Master Series version 6.

Experimental modal analysis was performed on the boring bar clamped in the clamping house with four bolts in the direction corresponding to the cutting speed direction. The boring bar was simultaneously excited in the cutting speed direction and cutting depth direction by two shakers (see Fig. 1). To measure spatial motion of the boring bar the accelerometers were glued with distance 28 mm from each other starting at 25 mm from the free end of the boring bar: 7 accelerometers in the cutting speed direction and 7 accelerometers in the cutting depth direction.

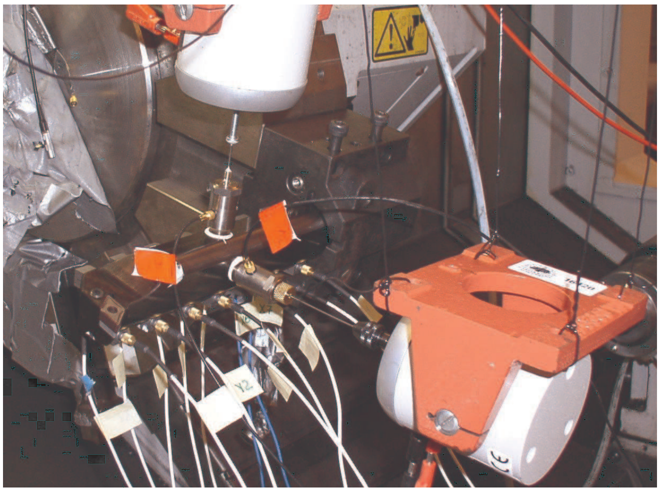

Figure 1: Setup for the experimental modal analysis

Physical properties of the boring bar material

The boring bar used in experiments and modeling is a standard boring bar S40T PDUNR15 F3 WIDAX. It is made of 30CrNiMo8 material with following physical properties: Young's elastic modulus $E = 205\ GPa$, density $\rho = 7850\ kg/m^3$, Poisson's coefficient $\nu = 0.3$.

Analytical Euler-Bernoulli model

Since the boring bar is long and slender, an analytical Euler-Bernoulli model can be used to estimate its natural frequencies and mode shapes. In the Euler-Bernoulli model the boring bar is considered to be the system with distributed mass and infinite number of degrees of freedom. This classical beam model considered only transverse beam vibrations and ignores the shear deformations and rotary inertia. These affects are described by Timoshenko beam model. However, if only the first modes are of interest the discrepancy between results obtained using both models is negligible, thus the Euler-Bernoulli model can be used in this application (see Fig. 2). The boring bar's bending motion in the cutting speed direction can be described by the following equation (bending motion in the cutting depth direction is described by the same equation, where I_x is replaced by I_y) [2]:

$$\rho A \frac{\partial^2 w(z,t)}{\partial t^2} + \frac{\partial^2}{\partial z^2}[EI_x \frac{\partial^2 w(z,t)}{\partial z^2}] = f(z,t), \tag{1}$$

The boring bar mode shapes can be calculated from Eq. 1 by setting $f(z,t) = 0$, which corresponds to the case of free vibrations and using following boundary conditions:

$$EI_x \frac{\partial^2 w(z,t)}{\partial t^2}|_{z=0} = 0, \frac{\partial}{\partial z}[EI_x \frac{\partial^2 w(z,t)}{\partial z^2}]|_{z=0} = 0, w(z,t)|_{z=l} = 0, \frac{\partial w(z,t)}{\partial z}|_{z=l} = 0. \tag{2}$$

Where ρ - density of boring bar material, A - boring bar's cross section area, I_x and I_y - cross-sectional area moments of inertia about "x axis" and about "y axis", E - Young's elastic modulus, $w(z,t)$ - bending deformation, $f(z,t)$ - external excitation force. The area and cross-sectional moments of inertia were calculated based on geometric dimensions of the boring bar cross section (see Fig. 2). The following properties were used in Euler-Bernoulli model calculations, here it was assumed that cross section area and cross-sectional moments of inertia are constants: $l = 0.2\ m$, $A = 1.1933 * 10^{-3}\ m^2$, $I_x = 1.1386 * 10^{-7}\ m^4$, $I_y = 1.1379 * 10^{-7}\ m^4$.

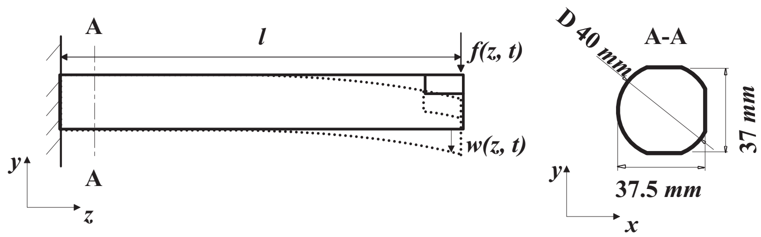

Figure 2: The dimensions of the boring bar used in Euler-Bernoulli model

The following notation for the coordinate system is used in Fig. 2: x - cutting depth direction, y - cutting speed direction, z - feed direction.

The boring bar's eigenfrequencies and eigenmodes are calculated from Eq. 1. The solution of Eq. 1 is found using separation-of-variables procedure and boundary conditions in the form of Eqs. 2.

Experimental Modal Analysis

In experimental modal analysis the boring bar is considered as multiple degree-of-freedom system. This approach uses measurement data of excitation force applied to the boring bar and the boring bar's responses collected simultaneously. Experimental modal analysis allows to identify system's eigenfrequencies, mode shapes, relative damping coefficients based on the modal model which is described by [3]:

$$[H(f)] = \frac{1}{2\pi}\sum_{n=1}^{N}\frac{Q_n\{\psi\}_n\{\psi\}_n^T}{jf-(-f_n\xi_n+jf_n\sqrt{1-\xi_n^2})}+\frac{Q_n^*\{\psi^*\}_n\{\psi^*\}_n^T}{jf-(-f_n\xi_n-jf_n\sqrt{1-\xi_n^2})} \quad (3)$$

Where N-number of degrees-of-freedom, $[H(f)]$ is $N \times N$ receptance matrix, $\{\psi\}_n$-$N \times 1$ mode shape vector, ξ_n-modal damping ratio, f_n-undamped system's eigenfrequency, Q_n-modal scaling factor. The estimate of receptance matrix is calculated based on power spectral density and cross-power spectral density estimates of boring bar response signals and excitation force signals correspondingly.

The first two boring bar eigenfrequencies and mode shapes in cutting speed and cutting depth direction were calculated using polyreference least squares complex exponential method [3]. The orthogonality of the extracted mode shapes was checked using Modal Assurance Criterion [3]:

$$MAC_{kl} = \frac{|\{\psi\}_k^T\{\psi\}_l|}{(\{\psi\}_k^T\{\psi\}_k)(\{\psi\}_l^T\{\psi\}_l)} \quad (4)$$

Finite Element Analysis

In the finite element analysis the system with distributed mass and infinite numbers of degrees-of-freedom is approximated by the system with large but finite number of degrees-of-freedom. The undamped dynamic motion of the system "boring bar-clamping house" can be described by the following equation:

$$[M]\{\ddot{\mathbf{w}}\} + [K]\{\mathbf{w}\} = \{\mathbf{f}\}, \quad (5)$$

Where $[M]$ is mass matrix of the system, $[K]$-stiffness matrix of the system, $\{\mathbf{w}\}$ is time and space-dependent displacement vector and $\{\mathbf{f}\}$-time and space-dependent force vector.

The finite element models of the boring bar and clamping house were developed using the MSC.MARC [9] (see Fig. 3). The finite element model of the boring bar follows the geometry of the real boring bar. The tetrahedron with 10 nodes and quadratic shape functions was used as a basic element for the boring bar model [9]. For simplicity the clamping house and clamping bolts were modeled as one body. Also in order to reduce complexity of the calculations the tetrahedron with 4 nodes and linear shape functions was used as a basic element for the model of clamping house with bolts [4, 9]. Usage of linear finite elements will not affect the accuracy of the results for boring bar's eigenfrequencies, because the clamping house eigenfrequencies are at significantly higher frequencies. The surface of the clamping house which corresponds to the surface of the real clamping house attached to the turret was rigidly clamped. In the compete "boring bar - clamping house" model the contact between the sub-models of the boring bar and the clamping house was modeled without friction. The system's undamped eigenfrequencies and corresponding mode shapes were calculated from Eq. 5 by setting $\{\mathbf{f}\} = 0$ using Lanczos iterative method [4, 9].

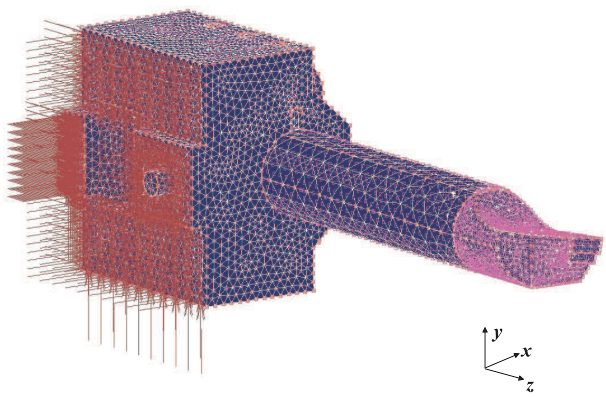

Figure 3: The 3-D finite element model of the system "boring bar - clamping house"

RESULTS

The first two eigenfrequencies and damping ratios calculated from Euler-Bernoulli model, estimated by experimental modal analysis and the finite element model are presented in Table 1, the corresponding mode shapes components in the cutting depth and cutting speed direction are shown in Fig. 4.

Table 1: Calculated eigenfrequencies and estimated damping ratios

Model	Mode 1		Mode 2	
	Frequency,[Hz]	Damping ratio,[%]	Frequency,[Hz]	Damping ratio,[%]
Euler-Bernoulli	698.12	-	698.33	-
Modal	497.494	1.112	525.832	1.571
Finite Element	533.129	-	558.256	-

As a quality measure of the mode shape estimates, obtained from experimental modal analysis and the 3-D finite element model, the Modal Assurance Criterion was used. The MAC - matrix for the modes 1 and 2 calculated according to Eq. 4 is:

$$[MAC]_1 = \begin{bmatrix} MAC_{MA_1,MA_1} & MAC_{MA_1,MA_2} \\ MAC_{MA_2,MA_1} & MAC_{MA_2,MA_2} \end{bmatrix} = \begin{bmatrix} 1.000 & 0.007 \\ 0.007 & 1.000 \end{bmatrix} \tag{6}$$

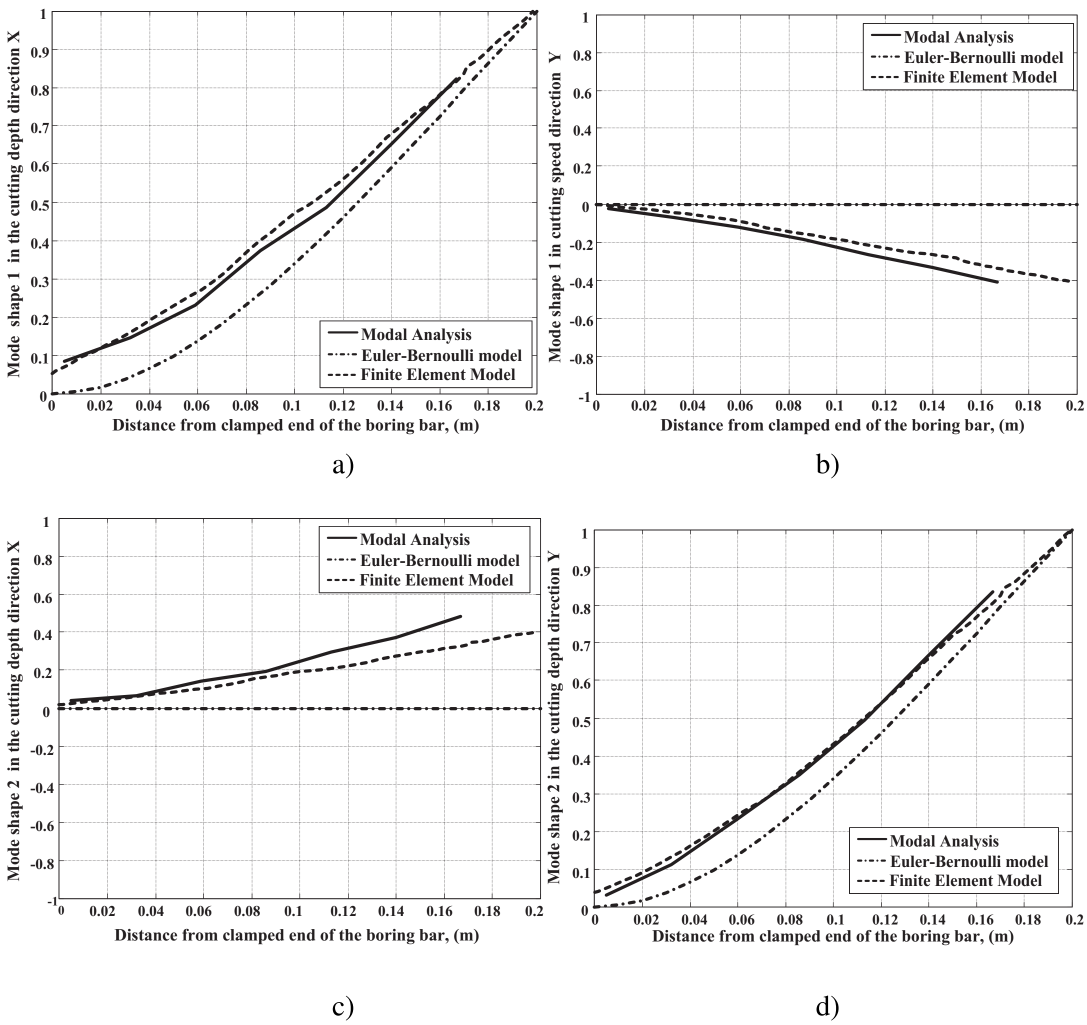

Figure 4: a) Component of mode shape 1 in the cutting depth direction b) component of mode shape 1 in the cutting speed direction, c) component of mode shape 2 in the cutting depth direction d) component of mode shape 2 in the cutting speed direction estimated by experimental modal analysis, Euler-Bernoulli model and finite element model correspondingly.

$$[MAC]_2 = \begin{bmatrix} MAC_{FEM_1,FEM_1} & MAC_{FEM_1,FEM_2} \\ MAC_{FEM_2,FEM_1} & MAC_{FEM_2,FEM_2} \end{bmatrix} = \begin{bmatrix} 1.00000 & 0.00001 \\ 0.00001 & 1.00000 \end{bmatrix} \tag{7}$$

Where MA_1, MA_2 mode shape estimates obtained from experimental modal analysis, FEM_1, FEM_2 - mode shapes calculated from the 3-D finite element model.

To get a quantitative measure of correlation between mode shapes estimated by experimental modal analysis and 3-D finite element model cross-MAC matrix was calculated:

$$[MAC]_3 = \begin{bmatrix} MAC_{FEM_1,MA_1} & MAC_{FEM_1,MA_2} \\ MAC_{FEM_2,MA_1} & MAC_{FEM_2,MA_2} \end{bmatrix} = \begin{bmatrix} 0.97656 & 0.02193 \\ 0.00738 & 0.96177 \end{bmatrix} \tag{8}$$

CONCLUSIONS

From the results of identification of spatial dynamic properties of the boring bar it can be observed that an analytical Euler-Bernoulli model overestimates the boring bar's eigenfrequencies compared to experimental modal analysis results (see Table 1). Furthermore, the estimates of the mode shapes obtained using experimental modal analysis demonstrate that the first two boring bar's mode shapes are orthogonal and rotated relative cutting speed and cutting depth directions, which is not the case for the mode shapes calculated using the Euler-Bernoulli model, where there is no angle between estimated mode shape and corresponding directions (see Fig. 4). These differences in estimates can be explained for instance by contact nonlinearity in the real system: between boring bar, bolts and clamping house, which is not considered in the analytical model, etc.

It can be noticed that results obtained based on 3-D finite element model are more accurate: the eigenfrequencies are closer to ones estimated by experimental modal analysis and mode shapes are rotated respective cutting speed and cutting depth direction (see Table 1 and Fig. 4, Eq. 8). Therefore the 3-D finite element model is preferable to the analytical Euler-Bernoulli model because it allows to describe the geometry of the real system and incorporate approximation of actual boundary conditions, i.e. contact between the surfaces of the boring bar, clamping bolts and clamping house. However there is still discrepancy in results obtained from 3-D finite element model and experimental modal analysis, which can be explained by: imperfection of geometrical model of the boring bar, bolts and clamping house, differences between actual material properties and ones used in simulations, uncertainty in measurements. The results obtained using the 3-D finite element can for instance be improved by incorporating the affect of mass loading of the boring bar by 14 accelerometers and 2 impedance heads used in the experimental modal analysis.

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