ICSV14
Cairns • Australia
9-12 July, 2007

14th International Congress on Sound & Vibration

INITIAL EXPERIMENTS WITH A FINITE ELEMENT MODEL OF AN ACTIVE BORING BAR

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Abstract

One of the most troublesome sources of vibration in metal cutting is the vibration caused by internal turning operations. A boring bar is a tool holder which is used to machine deep precise cavities inside a workpiece material. In order to perform this internal turning the boring bar usually has a large length-to-diameter ratio, and thus the boring bar vibrations are easily excited by the material deformation process during metal cutting. The vibrations are related to the lower order fundamental bending modes of the boring bar. To overcome the vibration problem an active control technique can be used. In particular, by utilizing an active boring bar with an embedded piezoceramic actuator and a suitable controller, the primary boring bar vibrations originating from the material deformation process may be suppressed with secondary "anti-" vibrations. In order to produce an active boring bar several decisions should be done, i.e. the characteristics of the actuator, the position of the actuator in the boring bar, etc. This usually implies that several prototypes of an active boring bar should be produced and tested, thus the design of an active boring bar is a tedious and costly procedure. Therefore a mathematical model which would incorporate the piezo-electric effect in order to predict the dynamic properties and the response of the active boring bar are of great importance. This paper addresses the development of a "3-D" finite element model of the system "boring bar-actuator-clamping house". The spatial dynamic properties of the active boring bar, i.e. its natural frequencies and mode shapes, as well as the transfer function between the voltage applied to the actuator and acceleration of boring bar are calculated based on the "3-D" FE model and compared to experimentally obtained estimates.

1. INTRODUCTION

Vibration is a common problem in internal turning operations performed in a machine tool. Since a boring bar usually has a geometry with a large length-to-diameter ratio, it is generally the weakest link in a machine tool. During turning the material deformation process induces a broad band excitation of the machine tool and as a result relative dynamic motion between the boring bar and the workpiece frequently occurs, often referred to as chatter. High levels of boring bar vibration result in poor surface finish, excessive wear and breakage of an insert, significant levels of acoustic noise, thus decreased productivity and degraded working environment. The investigations in this area have shown that high vibration levels are excited at the natural frequencies related to low-order bending modes of the boring bar and are dominated by the bending mode in the cutting speed direction, since in this direction the cutting force has the largest component [5, 11]. There is a continuous interest in suppressing boring bar vibrations with respect to the increase in productivity of machine tools and improvement of operator's working conditions. The attempts to reduce boring bar vibrations can be classified into two directions: control without and with redesigning the tool holder. The first direction simply implies avoiding cutting data which results in unstable turning, i.e. variation of cutting parameters in a structured manner [9, 4].

The second direction considers dynamic properties of one or several parts of the chain: insert - tool holder - clamping - machine tool or the tool holder - spindle interface with the purpose of structural modifications in order to increase system's resistance to machine tool chatter. Methods of control can be split into two groups: passive and active. In passive control the increase in dynamic stiffness is achieved by changing the static stiffness of the tool holder, for instance by using a boring bar, produced completely or partly (composite boring bar) of materials with higher modulus of elasticity such as sintered tungsten carbide [9]. Another passive control strategy is to use passive Tuned Vibrational Absorbers (TVA) to resist machine tool chatter.

An active control of turning operation is based on selective increase of the dynamic stiffness at the actual frequency of dominating bending mode and implemented with the use of feedback control schemes, since primary excitation signal is impossible to measure separately to produce feedforward reference signals due to the phenomenon of the material deformation process. Active control of boring bar vibration relying on active dynamic absorbers, inertial-mass actuators attached externally close to the tool-end, and LQ-control has been investigated [10]. Another principal direction in active control of boring bar vibrations is to use an active boring bar equipped with actuator and sensor in conjunction with a controller. The active boring bar has an accelerometer attached close to the tool-end, which measures primary excitation applied by material deformation process. The controller uses the sensor signal to produce secondary or "anti-" vibrations that are applied to the boring bar via actuator embedded into the boring bar. Due to piezoelectrical properties of actuator material secondary vibration voltage signal induces expansion of the actuator, which results in tension force and bending moment applied the boring bar to counteract primary excitation [3].

The design of an active boring bar is dependent on particular application and thus is a complex, time-consuming and costly process. Since the implementation of an active boring bar is based on the redesign of a standard boring bar, several prototypes are produced and tested before a product is completed. The development time and cost can be reduced by means of numerical modeling, i.e. using a "3-D" finite element model of an active boring bar. Such model can be used e.g. for prediction of dynamic properties of the active boring bar. This problem addresses the process of development a "3-D" finite element model of the system "boring bar - actuator - clamping house". Estimates of the first two natural frequencies and the corresponding mode shapes have been produced based on the initial "3-D" finite element model and experimental modal analysis of the active boring bar. Estimates of control path frequency response functions, i.e. frequency response function between the voltage applied to the actuator and the acceleration in the position of the error accelerometer in the cutting speed direction have been produced for data from numerical simulations using the "3-D" finite element model and for experimental data from the active boring bar.

2. MATERIALS AND METHODS

Spatial dynamic properties of the boring bar were estimated experimentally by means of experimental modal analysis. The experimental setup as well as methods for estimation of dynamic properties of the boring bar is presented in this section.

2.1. 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, 1 Ling Dynamic Systems shaker v201, 1 Gearing & Watson Electronics shaker v4, 2 Brüel & Kjær 8001 impedance heads, HP VXI E1432 front-end data acquisition unit, PC with IDEAS Master Series version 6.

The boring bar was excited in the cutting speed direction and cutting depth direction by two shakers (see Figure 1). Spatial motion of the boring bar was measured by 14 accelerometers glued with distance 25 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.

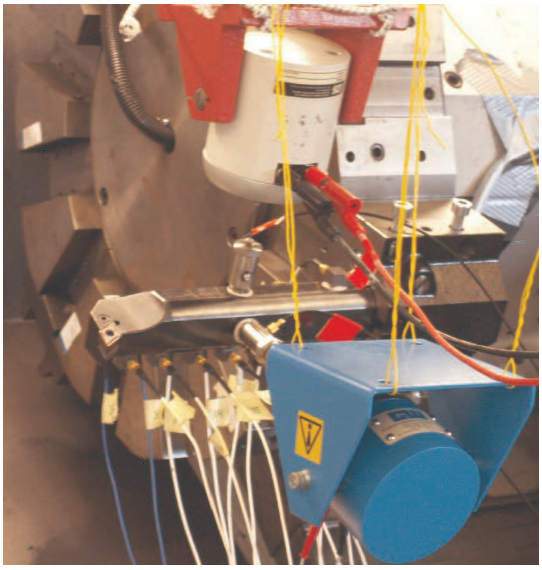


Figure 1. Setup for the experimental modal analysis

2.2. Physical properties of the boring bar and actuator materials

The boring bar used in experiments and modeling is standard boring bar S40T PDUNR15 F3 WIDAX. It is made of material 30CrNiMo8: Young's elastic modulus $E = 205\ GPa$, density $\rho = 7850\ kg/m^3$, Poisson's coefficient $\nu = 0.3$. A piezoelectric actuator made of PZT-5H piezoceramic material was utilized in the experiments [8].

2.3. Experimental Modal Analysis

In the concept of experimental modal analysis the boring bar is modeled as a system with multiple-degrees-of-freedom. The spatial dynamic properties of the boring bar were identified using the time-domain polyreference least squares complex exponential method [6], based on the impulse response function matrix estimate $[h(t)]$.

$$[h(t)] = \sum_{n=1}^{N} \left(A_n e^{\lambda_n t} + A_n^* e^{-\lambda_n t} \right) \tag{1}$$

Where $A_n = Q_n \{\psi\}_n \{\psi\}_n^T$ and $\lambda_n = 2\pi(-f_n \zeta_n + j f_n \sqrt{1-\zeta_n^2})$; $\{\psi\}_n$-$N \times 1$ mode shape vector, ζ_n-modal damping ratio, f_n-undamped system's eigenfrequency, Q_n-modal scaling factor, N - number of degrees-of-freedom. The impulse response function matrix estimate $[h(t)]$ is produced based on Inverse Fourier Transform of the receptance matrix $[H_r(f)]$ [6].

2.4. System Identification

The estimate of the receptance matrix $[H_r(f)]$ was obtained using $\hat{H}_1(f)$ estimator, which corresponds to the case, when contaminating noise is assumed to corrupt the response signal. The $\hat{H}_1(f)$ estimator is given by:

$$\hat{H}_1(f) = \frac{\hat{P}_{yx}^{PSD}(f)}{\hat{P}_{xx}^{PSD}(f)} \tag{2}$$

Where the power spectral densities of the excitation and response signals $PSD_{xx}(f)$ and $PSD_{yy}(f)$ as well as the cross-power spectral density $PSD_{yx}(f)$ can be obtained using e.g. Whelch's method [7].

The quality of the frequency response function estimate can be checked using the coherence function $\hat{\gamma}_{yx}^2(f)$ given by [2]:

$$\hat{\gamma}_{yx}^2(f) = \frac{|\hat{P}_{yx}^{PSD}(f)|^2}{\hat{P}_{xx}^{PSD}(f)\hat{P}_{yy}^{PSD}(f)} \tag{3}$$

2.5. "3-D" Finite Element Model

The finite element model of the system "boring bar - actuator - clamping house" was built using commercial finite element software MSC.MARC. The initial "3-D" finite element model of the clamped boring bar includes separate models of a boring bar and a clamping house interacting by means of frictionless contact, a model of a piezoelectric actuator and enables the capability of calculating response at any node of the "3-D" mesh grid.

The spatial dynamic properties of the system undamped "boring bar - actuator - clamping house" were calculated based on equation of dynamic equilibrium, i.e.

$$[\mathbf{M}]\{\mathbf{w}(t)\} + [\mathbf{K}]\{\mathbf{w}(t)\} = \{\mathbf{0}\}, \quad (4)$$

Where $[\mathbf{M}]$ is the system's $N \times N$ mass matrix, $[\mathbf{K}]$ is the system's $N \times N$ stiffness matrix, $\mathbf{w}(t)$ is the space and time dependent $N \times 1$ displacement vector, N is the number of degrees of freedom of the finite element model. The modal analysis was conducted by using the Lanczos iterative method in the of MSC.MARC software [1]. The dynamic behavior of the system was also examined by means of transient analysis using the Single-Step Houbolt transient operator [1].

2.5.1. Model of the system "boring bar - clamping house"

The model of the system "boring bar - clamping house" consists of two sub-models: a sub-model of the boring bar and a sub-model of the clamping house, the sub-models are connected in terms of touching contact. As a basic finite element a tetrahedron was chosen, as the most convenient element to describe the geometric shape of the boring bar and clamping house. The following boundary conditions were used: the nodal displacements on the surfaces of the clamping house, which correspond to surfaces of the real clamping house attached to turret, in $x-$, $y-$ and $z-$ directions are set to zero. The finite element model of the system "boring bar - clamping house" is shown in Fig. 2 a).

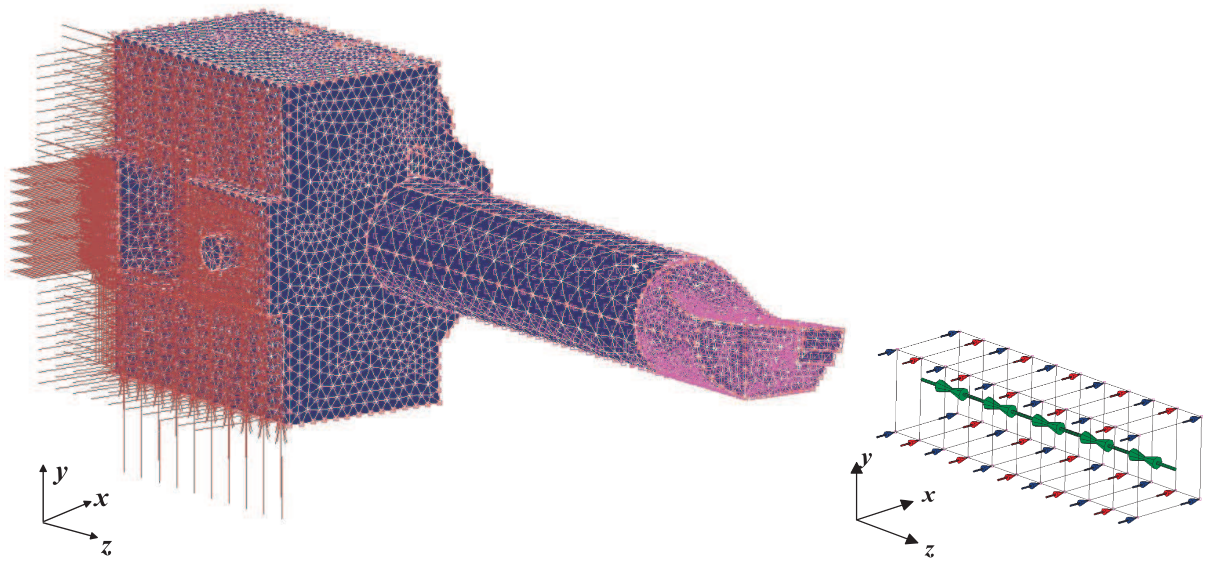
2.5.2. Model of the actuator

A "3-D" finite model of a piezoelectric actuator was developed. Thin layers of electrically active ceramic material connected in parallel were modeled as a stack of 8-nodes bricks with piezoelectrical properties, i. e. besides three translational degrees of freedom each node has a fourth degree of freedom - electric potential. The scheme of a "3-D" finite element model of the actuator is shown on Fig. 2 b).

The electrostatic boundary conditions are shown in Fig. 2 b), blue arrows show the nodes with applied negative or zero potential and red arrows correspond to the nodes with applied positive potential. Furthermore, the large green arrows show the material orientations within the finite elements. The material orientation in this case determines the direction of current flow inside the finite element.

3. RESULTS

Spatial dynamic properties of the active boring bar estimated by means of experimental modal analysis and the initial "3-D" finite element model are presented in this section. Control path frequency response functions between the voltage applied to the actuator and the acceleration at the error sensor position in the cutting speed direction measured experimentally and calculated numerically are also presented.



a) b)

Figure 2. a) The "3-D" finite element model of the system "boring bar - clamping house" and b) the sketch of a "3-D" finite element model of an actuator.

3.1. Natural frequencies and mode shapes

The estimated and calculated first two natural frequencies are summarized in the Table 1; the respective relative damping ratios were also estimated based on the modal model see Eq. 1 as $\zeta_1 = 1.044\%$ and $\zeta_2 = 1.244\%$.

Table 1. Natural frequencies.

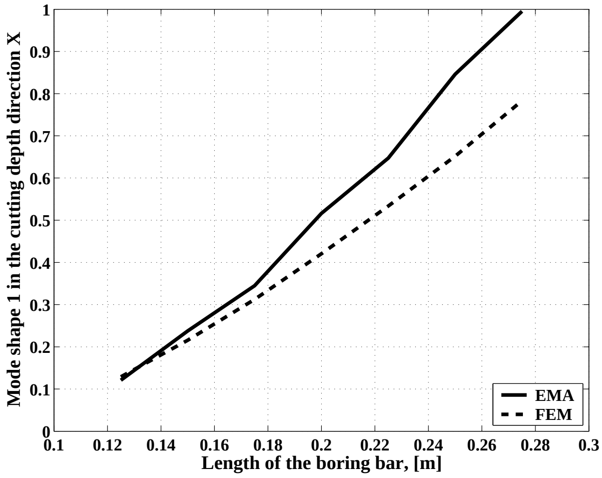
Model	Mode 1, [Hz]	Mode 2, [Hz]
EMA	501.638	520.852
Finite Element	496.232	529.046

The first two mode shapes estimated by experimental modal analysis and calculated based on finite element model are compared in Figure 3.

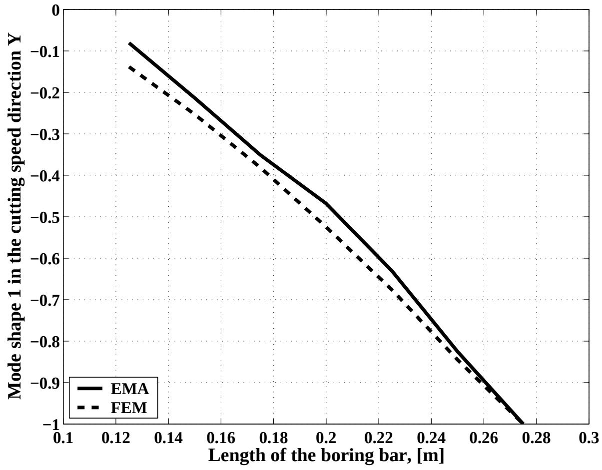
The simulated and measured frequency response functions between the voltage applied to the actuator and the acceleration at the point of the active boring bar error sensor position (25 mm form the tool tip) and corresponding coherence functions are presented in Fig. 4 a) and 4 b).

4. CONCLUSIONS

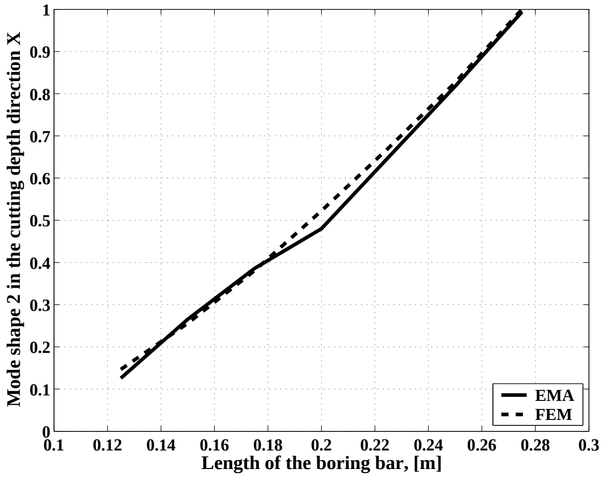
The presented results indicate that the initial "3-D" finite element model allows a rough prediction of dynamic behavior of the active boring bar. Thus, there is a discrepancy between the natural frequencies estimated by experimental modal analysis and the corresponding natural



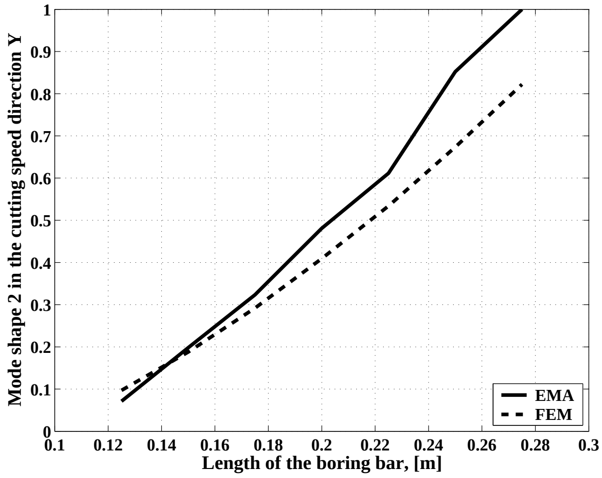
a)



b)



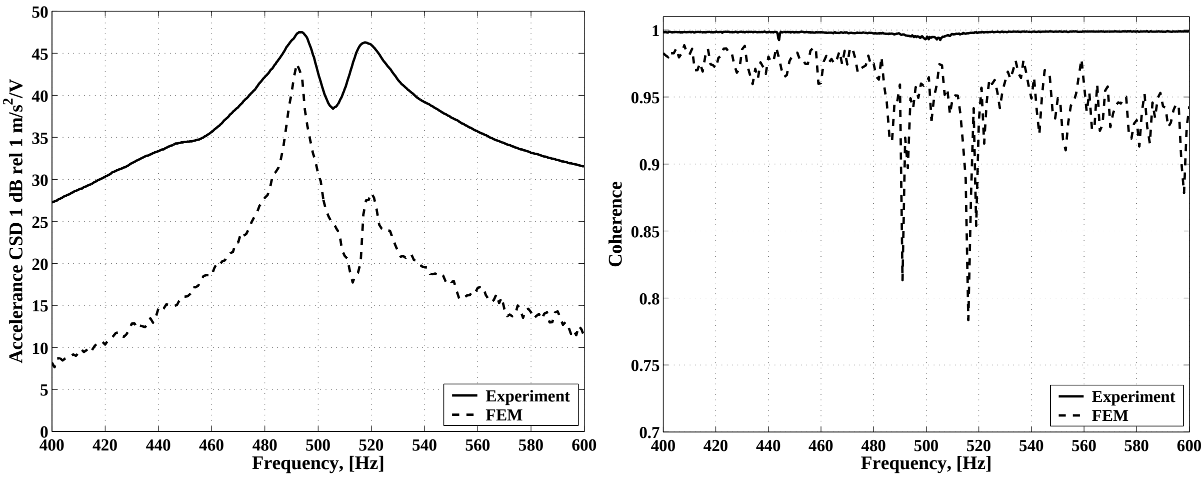
c)



d)

Figure 3. 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 and finite element model correspondingly.

frequencies calculated based on the finite element model see Table 1. These differences may be explained by the imperfection of geometrical model of the boring bar and clamping house, differences in the actual material properties and the ones used in the FE modeling, etc. Estimated and calculated mode shapes are well correlated see Fig.3. Comparison of the frequency response functions between the actuator voltage and accelerations in the cutting speed direction obtained experimentally and calculated numerically see Fig. 4 shows that the FE model provide an accelernace function who's magnitude is approximately in the same order as the magnitude of the actual active boring bar accelerance. It can be noticed also that the FE model gives stiffer approximation of the system. This can be explained by the "clamped" boundary conditions used for the clamping house in the model, since in reality the attachment of the clamping house to the turret as well as the attachment of the turret to the slide etc., can not be considered completely rigid and thus introduce flexibility in the model. This results in an experimentally estimated



a) b)

Figure 4. a) Accelerance between the voltage over the actuator and acceleration in the point of the error sensor position in the cutting speed direction and b) corresponding coherence functions estimated experimentally and obtained based on the finite element model.

accelerance function with a slightly higher overall level compared to the corresponding accelerance function produced based on the FE model. However it is possible in future to improve the accuracy by for instance incorporating damping into the FE model, e.g. by modeling nonlinear friction between the surfaces of the clamping house and the boring bar.

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