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EXPERIMENTAL MODAL ANALYSIS OF A MILLING MACHINE SPINDLE -TOOL HOLDER - CUTTER ASSEMBLY

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Abstract: *In this article are presented results of the experimental modal analysis performed on the main shaft assembly, spindle - tool holder - cutter of a CNC milling machine. Knowing accurately the dynamic behavior of a CNC milling machine, the modes of vibration, the frequencies and the damping coefficients, we can ensure optimal use of it to values close to maximum cutting speed and maximum feed rates. Also it is possible to avoid the unstable milling domains .*
Keywords: *experimental modal analysis, CNC, spindle, tool holder, cutter, main shaft.*

1.INTRODUCTION

In milling operation, vibrations are the main cause that limit the establishment of technological parameters of processing, cutting speed and feed rate, near the maximum allowed values stated in the user guide of the milling machine. In practice it is found that, to avoid the unstable domain in the milling process, cutting speed should be limited with 5% to 15% compared to the optimal values, which leads to low productivity.

To determine the extent to which the dynamic behavior of spindle-tool holder-cutter influence the vibrations during the milling process, was made the experimental modal analysis. The purpose of this analysis is to obtain equivalent dynamic parameters. With these parameters we can calculate the stability diagram and we can define the stable and unstable milling domains.

In technical literature there are many studies on vibration that occur in machine tools during cutting process, like the one presented in [3].The modal analysis methods, analytical and experimental are described in [2], [4].

Modal analysis can be applied if the system studied is observable, linear and time invariant and respect Maxwell's reciprocity condition. It was considered that the entire main shaft, the spindle, tool holder and cutter assembly satisfy all these conditions.

2. THE ANALITICAL MODAL ANALISYS

The purpose of analytical modal analysis is to determine the behavior of physical systems studied its modes of vibration. To achieve this it is necessary to know the characteristics of components and how they interact with each other using a physical model complemented by a mathematical model [11],[15]. Analytical modal analysis connects the physical system with the experimental modal analysis.

Considering the particular case of roughing milling, the cutting depth is equal to the secondary cutter diameter; we assumed that mainly the cutter vibrates in one direction, the direction of advance. We chose the reference system Ox axis along with the feed direction.

In Figure 1 is presented the simplified physical model of the spindle-tool holder-cutter assembly.

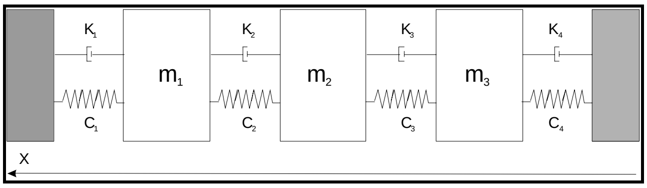

Fig.1 The simplified physical model of the spindle-tool holder-cutter assembly

The general form of representation of this kind of mechanical system by differential equations is:

$$M.\begin{bmatrix} \frac{d^2x_1}{dt^2} \\ \frac{d^2x_2}{dt^2} \\ \frac{d^2x_3}{dt^2} \end{bmatrix} + C.\begin{bmatrix} \frac{dx_1}{dt} \\ \frac{dx_2}{dt} \\ \frac{dx_3}{dt} \end{bmatrix} + K.\begin{bmatrix} x_1(t) \\ x_2(t) \\ x_3(t) \end{bmatrix} = \begin{bmatrix} f_1(t) \\ f_2(t) \\ f_3(t) \end{bmatrix} \qquad (1)$$

where:
M - matrix of inertia
C - viscous damping matrix
K - the stiffness matrix

For the particular case of milling operation, x1, x2, and x3 are the deformations on the X direction at the level of spindle, tool holder and cutter. The f1, f2, f3 are the expressions of external forces. Determination of the system parameters is done using the transfer matrix. Laplace transform applied to equation (1) leads to:

$$(M.p^2 + C.p + K) = F(p) \qquad (2)$$

where:

$$X(p) = L[X(t)] \qquad (3)$$

$$F(p) = L[F(t)] \qquad (4)$$

$$H(p) = p^2.M + p.C + K \qquad (5)$$

Equation (2) becomes:

$$H(p).X(p) = F(p) \qquad (6)$$

Transfer matrix is the inverse matrix H:

$$H^{-1}(p) = \frac{H^*(p)}{|H(p)|} \qquad (7)$$

$H^*(p)$ is the complex conjugate matrix of the H matrix. The final solution is:

$$X(p) = H^{-1}(p).F(p) \qquad (8)$$

From transfer matrix result the equivalent dynamic parameters. The precision with which they are determined depends on the physical model from which to start and accuracy of data input. While mass components can be determined very precisely, damping factors and elastic constants are determined approximately.

The results reported in [13] shows that errors due to the theoretical estimation of the modal parameters are over 10%. To improve accuracy in [10] is made a very detailed analysis, including the main shaft tool holder interface. According to [5], analytical models cannot correctly describe the behavior of real structures because the approximations made in joints are large and not always justified so, the solution is found in experimental modal analysis.

3. THE EXPERIMENTAL MODAL ANALISYS

Experimental modal analysis, presented in detail in [12] aims to determine modal parameters by measuring the structural response to a well determined disturbance.

The main methods of excitation of the structure are electrodynamics vibrator method and impact hammer method, each with advantages and disadvantages [6], [7], [8].

Details of the impact hammer method, applied to study different types of structures are described in [1]. In [9] is presented the particular case when the studied structure is a milling machine and the accelerometer is mounted at the end of the cutter. The impact hammer applies an impulse at the point diametrically opposite to accelerometer.

Steps to be taken in experimental modal analysis are data acquisition, determination of frequency response function, extracting modal parameters, validation and data presentation.

3.1 The frequency response function

Frequency response function is the method most often used for extracting modal parameters. To determine the frequency response function we have to rewrite equation (1) in the frequency domain. We applied the Fourier transform:

$$X(j.\omega) = H^{-1}(j.\omega).F(j.\omega) \qquad (9)$$

$$\begin{bmatrix} X_1(j\omega) \\ \cdot \\ \cdot \\ X_q(j\omega) \end{bmatrix} = \begin{bmatrix} H_{11}(j\omega) & . & . & H_{1s}(j\omega) \\ \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot \\ H_{q1}(j\omega) & . & . & H_{qs}(j\omega) \end{bmatrix} . \begin{bmatrix} F_1(j\omega) \\ \cdot \\ \cdot \\ F_s(j\omega) \end{bmatrix} \qquad (10)$$

where:
ω - angular frequency
H - frequency response function matrix

$$p = j.\omega \qquad (11)$$

$$j = \sqrt{-1} \qquad (12)$$

$$H_{ij}(\omega) = \frac{X_i(\omega)}{F_j(\omega)} \qquad (13)$$

$$H_{ij}(\omega) = \mathrm{Re}_{ij}(\omega) + j\,\mathrm{Im}_{ij}(\omega) \qquad (14)$$

In experiments, we used a single input signal, applied with an impact hammer and recorded a single output signal by using an accelerometer. This method is called Single Input Single Output. For this particular case we have the following frequency response function:

$$H_{ij}(\omega) = H(\omega) = \frac{X(\omega)}{F(\omega)} \qquad (15)$$

$$\operatorname{Re} H(\omega) = \frac{k - m.\omega^2}{(k - m.\omega^2)^2 + c^2.\omega^2} \qquad (16)$$

$$\operatorname{Im} H(\omega) = -\frac{c.\omega}{(k - m.\omega^2)^2 + c^2.\omega} \qquad (17)$$

$$A(\omega) = \frac{1}{\sqrt{(k - m.\omega^2)^2 + c^2.\omega^2}} \qquad (18)$$

$$\phi(\omega) = -\frac{c.\omega}{k - m.\omega^2} \qquad (19)$$

where:
c- damping coefficient
k - elasticity coefficient
m - mass
A(ω) - the amplitude of vibration at the angular frequency ω

Since the frequency response function (15) is expressed in terms of compliance, X (m) / F (N) and the actual measurement is made in terms of inertia a (m/s2) / F (N), according to [14], it must be applied the transformation:

$$H_{com}(\omega) = \frac{1}{\omega^2(1+\zeta^2)^2} \cdot \frac{a(\omega)}{F(\omega)} \qquad (20)$$

where:
H_{com} - frequency response function matrix expressed as compliance
ζ - damping factor

The same transformation applies to equations (16) - (19). Frequency characteristics are not always accurate since are influenced by external electrical signals that overlap the useful electrical signal. As a result, it was necessary to use algorithms to reduce errors.

4. EXPERIMENTAL MODAL ANALYSIS OF THE SPINDLE-TOOL HOLDER CUTTER ASSEMBLY

Method of impact hammer consists in striking the studied structure using a calibrated hammer fitted with an electric sensor. The block diagram of the experiments is shown in Figure 2.

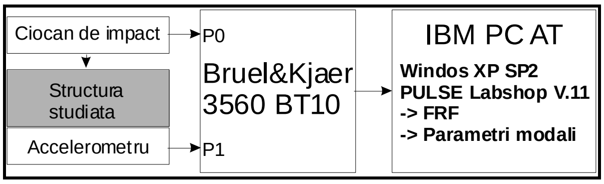

Fig.2 Block diagram of experimental modal analysis performed with the Bruel & Kjaer 3560BT 10

Figure 3 shows attachment of the accelerometer on the main shaft. The instrumented hammer strikes in a point placed approximately diametrically opposite with the accelerometer.

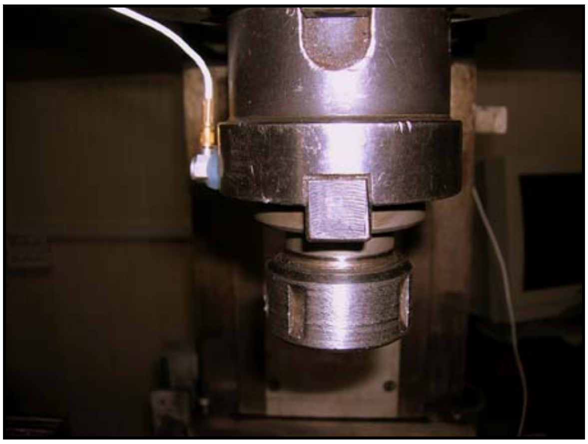

Fig.3 Accelerometer Securing on the main shaft

4.1 The hardware components in the modal analysis experiment

The hardware required to perform the experimental modal analysis are a FUS 22 CNC milling machine, an accelerometer, an impact hammer, constant current source, an analog to digital converter and an IBM PC-AT compatible computer.

We used the Dytran 5800B3 type impact hammer. Depending on the frequency we want to be explored, the hammer is equipped with a corresponding tip. We can use three types of impact tips. The 6250 A aluminum tip is useful in the frequency range from 0 - 10 kHz, the 6250 P, hard plastic tip is used in the frequency range of 0-5 kHz and the 6250PS soft plastic tip is useful in the frequency range of 0-2 kHz

The accelerometer used was Dytran 3035B2G type. The mass of the accelerometer must respect the restriction stated in [1]:

$$m_{str} > 10.m_{acc} \quad (21)$$

where:
m_{str} - the mass of the structure
m_{acc} - the mass of the accelerometer

The accelerometer is mounted on the structure to be tested with DYTRAN 6273 wax.This limits the frequency range of the accelerometer. Depending on the thickness of the wax, the useful frequency range is 0-2500 Hz.

To convert analog data to digital we used the hardware platform Bruel & Kjaer type 3560 BT 10. The computer used was a Toshiba L 30 T 10, with an Intel Dual Core T2080, 1.73 GHz and 1 GB RAM.

4.2 The software components of experimental modal analysis experiment

The Bruel & Kjaer 3560 hardware platform uses PULSE LabShop V 11 software which is dedicated to modal analysis measurements.Using the project templates from PULSE LabShop, we design a specific graphical user interface for particular Dytran accelerometer and hammer used. Output data are presented both as ASCII text files and in graphical form.There are also obtained: frequency response function (15) with its real component (16) and imaginary component (17), amplitude (18) and phase (19).

4.3 Performing the experiments and determination of modal parameters

We conducted 10 measurements of modal analysis for each Kennametal cutter, tool holder and spindle. In Figures 4-12 are presented the frequency response functions and coherence functions exported by PULSE software.The frequency response function has three distinct maximum. Profile peaks and the large distance between them associated with small damping factor shows that the vibration modes are easy to medium coupled.

Calculations of modal parameters, natural frequency, damping factor and the residue are made by PULSE software according to a proprietary internal algorithm owned by Bruel & Kjaer .For further calculations and interpretations they are saved in an ASCII file In Tables 1, 2 and 3 are presented the modal parameters for Kennametal spindle, tool holder and cutters. The coherence function for cutters was consistently equal to 1 in all cases, throughout this frequency range because impact hammer and accelerometer are well fitted for structural studied.

Table1.

The modal parameters for spindle

Spindle	Frequency (Hz)	Damping factor -	Modal mass -
FUS22	350	3.11	140.96
ISO40	696	9.39	3.787

Table2.

The modal parameters for tool holder

Tool holder	Frequency (Hz)	Damping factor -	Modal mass -
ISO40	344	3.88	13.39
ISO40	696	9.39	3.787

Table3.

The modal parameters for cutters

Cutter D (mm)	Frequency (Hz)	Damping factor -	Modal mass -
6	318.8	4.75	7.4234
6	940	3.75	0.1460
6	1297	1.48	0.0326
8	315.6	6.55	3.2244
8	803.1	10.4	0.2885
8	1509	2.56	0.0440
10	318.8	5.11	6.5944
10	834	9.8	0.4780
10	1609	2.84	0.0556
12	315.6	5.56	6.8925
12	809.4	8.14	0.6359
12	1656	1.85	0.1449
14	318.9	4.71	6.8640
14	843.8	7.28	0.5558
14	1706	3.72	0.0918

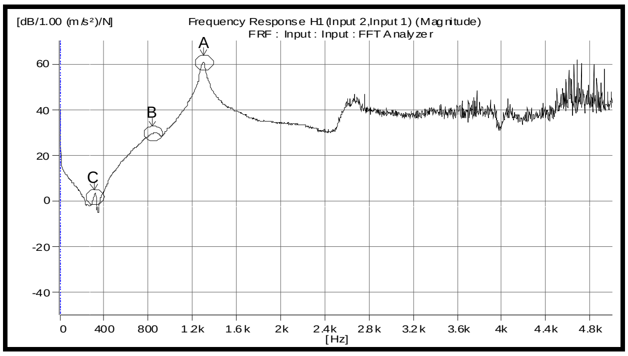

Fig.4 Frequency response function for the cutter with 6 mm diameter.

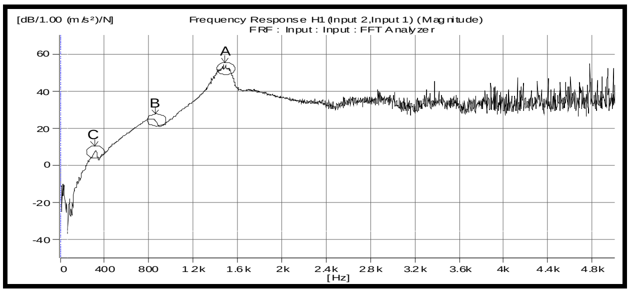

Fig.5 Frequency response function for the cutter with 8 mm diameter

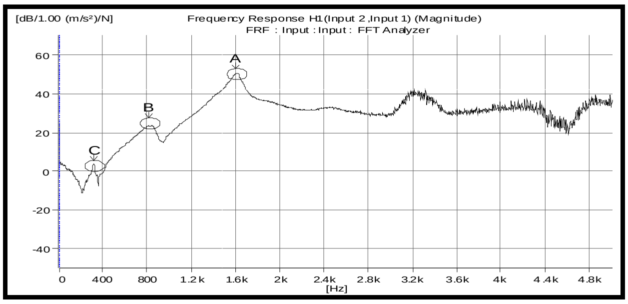

Fig. 6 Frequency response function for the cutter with 10mm diameter

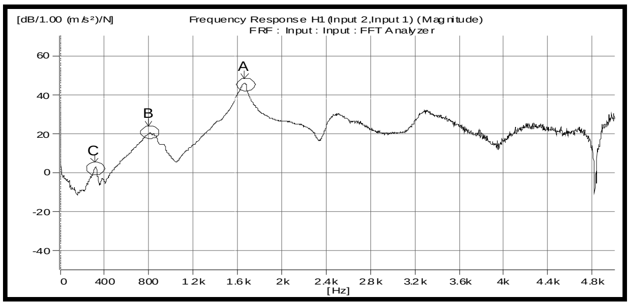

Fig.7 Frequency response function for the cutter with 12 mm diameter

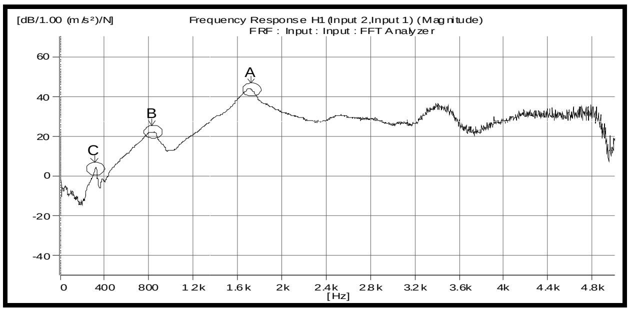

Fig.8 Frequency response function for the cutter with 14 mm diameter

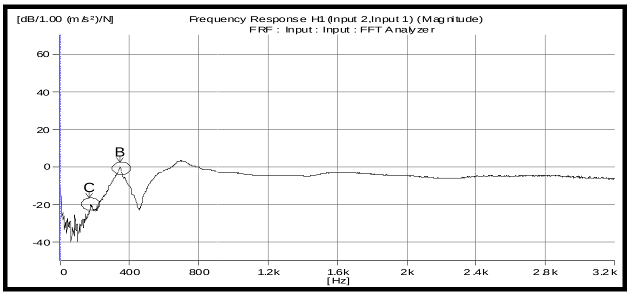

Fig.9 Frequency response function for the tool holder

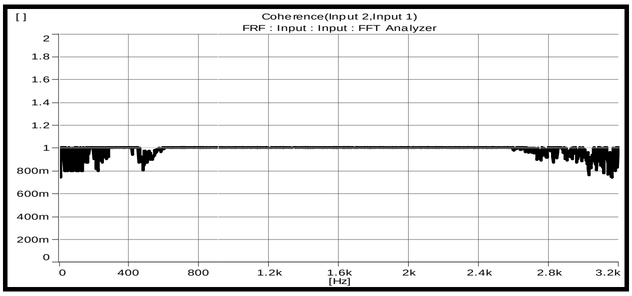

Fig.10 Coherence function for the tool holder

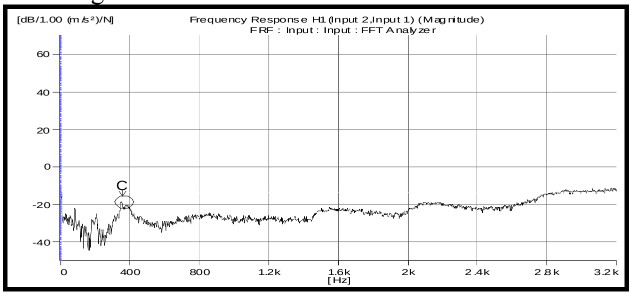

Fig.11 Frequency response function for the spindle

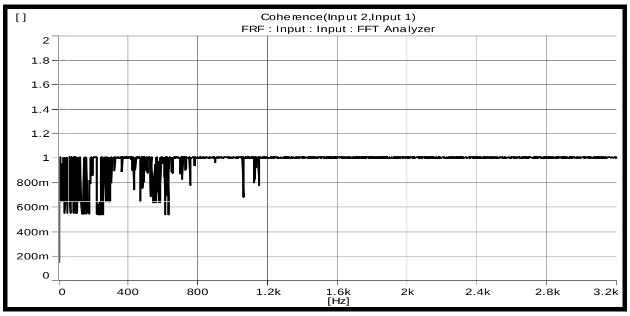

Fig.12 Coherence function for the spindle

V. CONCLUSIONS

Experimental modal analysis of spindle-tool holder-cutter permitted the determination of modal parameters for each element of the assembly. Analysis of the data from Table 3 shows as expected, that cutters vibrates at frequency dependent on the diameter. The influence of rigidity is decisive at higher frequencies of vibration: large diameter cutters, has a high rigidity and vibration frequency increases with increasing diameter.

Since the modes of vibration of the spindle and tool holder occur at low frequency we can conclude that the cutter vibration is coupled with these modes of vibration. Cutter vibration modes observed at 315.6 Hz, 318.8 Hz, 318.9 Hz are due to the spindle. Cutter vibration modes observed at 803.1 Hz, 809.4 Hz 834 Hz 843.8 Hz and 904 Hz are mainly due to the tool holder. Cutter vibration modes observed at 1297 Hz, 1509 Hz, 1609 Hz, 1656 Hz and 1706

Hz are intrinsic cutter vibrations, slightly influenced by tool holder and spindle.

Tool holder vibration mode observed at 344 Hz is due to the spindle. Tool holder vibration mode observed at 696 Hz is the intrinsic tool holder vibrations, slightly influenced by the spindle.

Spindle vibration mode observed at 350 Hz is the intrinsic spindle vibration, not influenced by tool holder nor cutter.

When we perform the experiments on the tool holder and on the spindle, we observe that the electrical output signal from the accelerometer is too low and this lead to a small signal / noise ratio and also in small coherence function with large deviations from unit, according to figures 10 and 12.

The experimental results obtained from modal analysis are the input data to a mathematical model designed to calculate the stable milling domains. Knowing the stable and unstable milling domains we can select the optimum milling technological parameters, which increase productivity and lower production costs**.**

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Analiza modală experimentală a ansamblului arbore principal-portsculă-freză al unei maşini de frezat CNC

Rezumat: *În acest articol sunt prezentate rezultatele analizei modale experimentale efectuate pe ansamblul arbore principal - portsculă - freză al unei ma□ini de frezat CNC. Ştiind exact comportamentul dinamic al unei maşini de frezat CNC, modurile de vibratie, frecvențele şi coeficienții de amortizare, putem asigura o utilizare optimă a acesteia la valori apropiate de viteza maximă de a□chiere şi de avans. De asemenea, este posibil să se evite domeniile instabile de frezare.* ***Cuvinte cheie:*** *analiză modală experimentală, CNC, arbore principal, portsculă, freză.*

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