THEORETICAL AND EXPERIMENTAL STUDIES OF THE SPINDLE OF A HIGH SPEED TURNING MACHINE

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Abstract: *The paper presents a new methodology for research and optimization, from point of view of a dynamical behaviour, of high speed main spindle of a turning machine tool with 6000 rpm. The theoretical studies were made by using Finite Element Method (FEM) in the advanced CAD/CAE software CATIA V5R7 and the results were validated by experimental FFT spectra and transfer function measures using Vibroport 41 – Bruel&Kjaer.*
Key words: *High Speed Spindle, High Speed Machining, Finite Elements Method, dynamic behaviour. FFT and transfer functions analysis.*

1. INTRODUCTION

The cutting moving in all the operations in the cutting surface generation process needs is characterized by static, dynamic and termic loads of the elastic systems defined by the machine-tool, clamping devices, tool and workpiece. In this elastic system appear some intermediary and stationary processes that generate difficulties for specific productivity condition, dimensional precision and surface quality. The static and transitory processes are defined by a large number of phenomena, mechanical, termical, electric, hydrodynamic and aerodynamic flows, produced in the same time and interdependence one which other. These are the features of the dynamical process for the machine-tools. A vibration is defined by the alternatives moves of a mechanical system, relative to the equilibrium state. These moves are obtained by adding or removing some forces in the mechanical system.

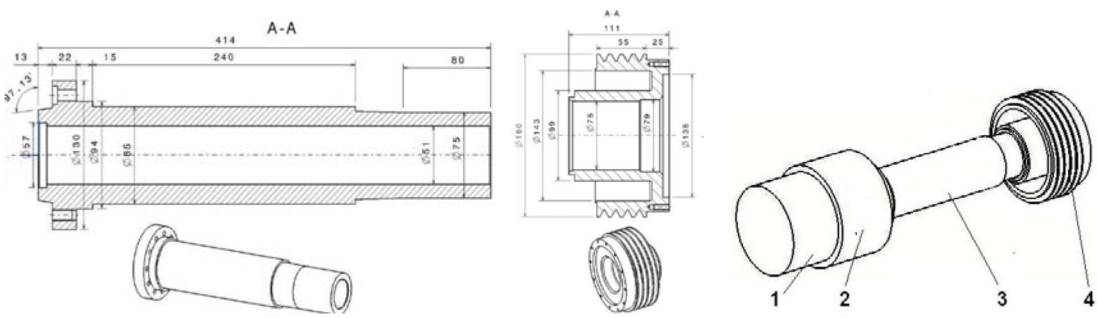
A new concept is known as the maintenances “in situ”. In this case the monitoring condition is able to determinate the state of the machine tool while it works. By monitoring a machine tool, the repairs are made only when the state of the machine is over a predefined level of mall-function (when it’s possible to generate a major damage of the machine or production flows).The signal of noise and vibration measured on the machine tool contained a large quantity of information about the internal processes. The analyses of these measures represent an easy way to appreciate the machine tool working state (vibration increases with the wear level or other damage).

2. THEORETHICAL STUDY OF THE MAIN SPINDLE

2.1.Theoretical model of the spindle

The theoretical model of a spindle was made by using CATIA V5 based on real shape of the main spindle. There were considered the entire elements witch compound the main spindle of RAMO turning machine tool. A 3D model was designed for the spindle (fig.1.c.3), belt wheel (fig.1.c.4), lathe chuck (fig.1.c.2) and work pieces (fig.1.c.1). The weight of the spindle was about 12 Kg and the belt wheel weight is about 7.5 Kg. The lathe chuck (mass about 14 Kg) and work pieces (mass about 17 Kg) were designed in a similar way. All the components were designed as 3D virtual solid model according to the technical documentation of the real pieces of the turning machine-tool (figure 1)

Another important feature for FEM model is the design of the spindle constraints, depending of the bearing type. For each type of bearing there are specific statical and dynamical features which have to be taken in account for analysis. Compared with other common bearings types, rolling bearings are used for some high performance spindle systems. These are normally made up of angular contact ball bearings because this type is specially suited for high-speed applications due to its friction and wear characteristics.



a. *main spindle* b. *belt wheel* c. *3D model*

Fig..1. Elements geometrical characteristic

Tabel 1. Comparison of the bearing type

	roller	hydrodynamic	hydrostatic	aerostatic	magnetic
High $n*d_m$	◕ 1)	◐	◐	●	●
High durability	◕	◕	● 2)	● 2)	● 2)
High accuracy	◕	◕	●	●	●
High rigidity	◕	◕	●	◐	◕
Easy lubrication	● 3)	◐	○	◐	○ 4)

● very favourable ◕ favourable ◐ medium ○ not favourable

Table 2 Bearing stiffness calculation based on Palmgren's formula

Type of bearing	Load case: δa = 0	δr = 0
Angular contact ball bearing	$\delta_r = \frac{0.02}{\cos\alpha} \cdot \sqrt[3]{\frac{Q^2}{Dw}}$	$\delta_a = \frac{0.02}{\sin\alpha} \cdot \sqrt[3]{\frac{Q^2}{Dw}}$
Double row linear contact cylindrical roller bearing	$\delta_r = \frac{0.0006}{\cos\alpha} \cdot \frac{Q^{0.9}}{l_a^{0.8}}$	$\delta_a = \frac{0.0006}{\sin\alpha} \cdot \frac{Q^{0.9}}{l_a^{0.8}}$
Load	$Q = \frac{5Fr}{iz\cos\alpha}$	$Q = \frac{Fa}{iz\sin\alpha}$
Number of rolling elements	z = (πDr) / (Dw + 4)	z = (πDr) / (Dw + 4)

The most important bearing feature for the high speed behaviour of the spindle is the rigidity. This is function of the bearing type as it is shown in the next table.

The bearing types for the main spindle of the RAMO turning machine-tool are three angular ball bearings at the head with lathe chuck and workpiece (fig 3.1) and one double row linear contact cylindrical roller bearing at the head with belt wheel (fig.3.2)

2.2. Finite Elements Method analysis

The principle of FEM is to approximate the unknown field function (temperature, deformation, dynamic behaviour) with a spline function so that on a small range, named finite elements, the approximation of this function would be almost the exact solution. This approximation is used considering the extreme (minimum) conditions of the function associated to the differential equations system which govern the phenomenon.

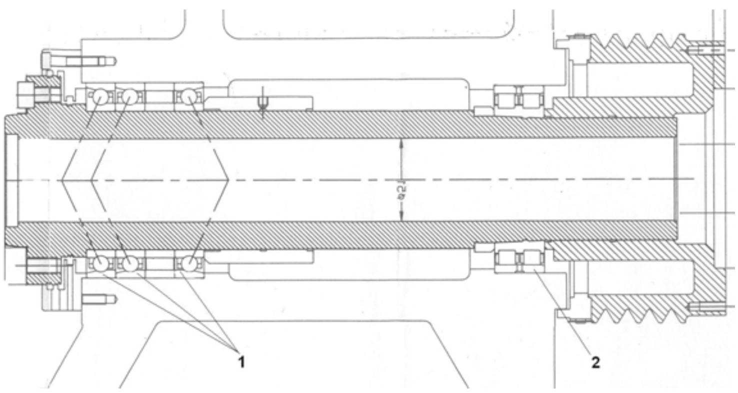
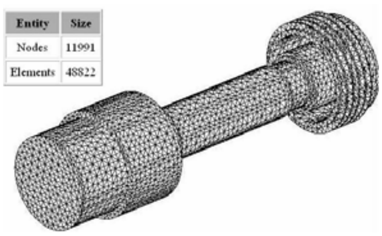
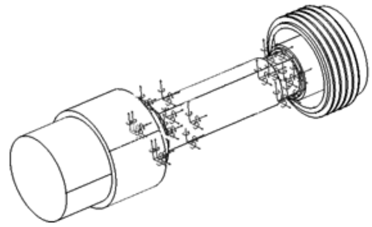


Fig.3: Main spindle bearing type:1. angular ball bearings;
2. double row linear contact cylindrical roller bearing.



a



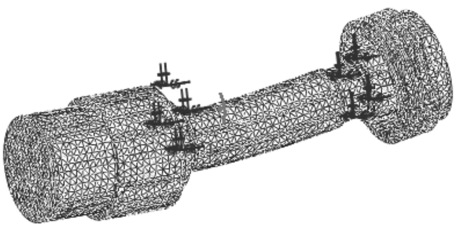
b

Fig.4. Spindle: a. discretization b. Model constraints

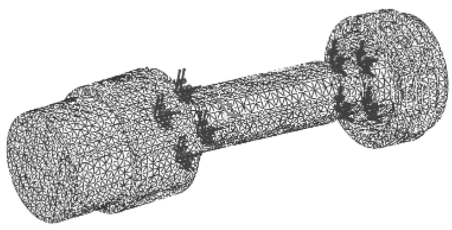
FEM modelling has application in all domains, elements represents mathematical approximation of the geometrical model, so the simulation of the behaviour will be possible. In this case for the theoretical study of the dynamical behaviour (frequency case) of the spindle it was used for discretization 48822 tetrahedral elements and 11991 nodes (figure 4.a) In the FEM analysis of the spindle dynamical behaviour the bearings were defined like constrains with displacement, depending of calculated rigidities. The calculus was made based on the formula shown in the table 2 for each type of bearings. In figure 4.b it is presented the constraints applied on the FEM model. The bearings stiffness was defined as a displacement on the three axes. The rotations on x and y axes were locked.

The analysis in frequency case was made for ten modes of vibration. Some of these modes are only for the element of the assembly main spindle and workpiece. As the matter in fact the vibration modes are calculated and it is possible to appear the entire modes in the same times. In the figure 5 there are presented the most important four modes which are important for our study.

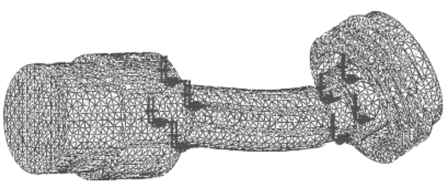
It was obtained a 140.39 Hz frequency (fig.5.a), for the second mode 512.98 Hz frequency (fig.5.b), for the third mode 3 a 904.75 Hz frequency (fig.5.c), and for fourth mod a 1255.1 Hz frequency (fig.5.d).



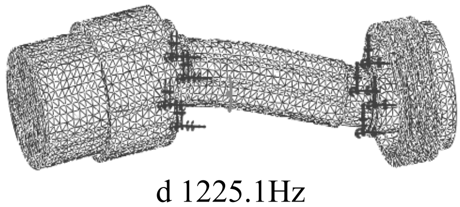
a 140.39 Hz



b 512.98 Hz



c 904.75 Hz



d 1225.1Hz

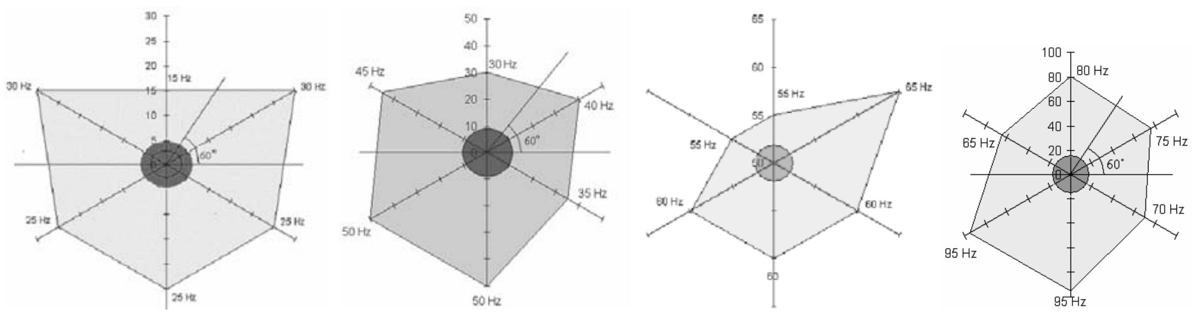
Fig.5 Vibration modes

3. EXPERIMENTAL STUDIES

Dual – channel FFT analyzers are often used for machinery analysis. Auto and cross correlation, determining the amount of agreement in the same and different signals respectively obtained from two sources and compared in the time domain, are occasionally used for machinery analysis. Coherence is a measure of the contribution from a specific source within a complex signal containing excitation from multiple sources. Coherence is calculated in the frequency domain and is frequently employed to isolate the source of a vibration signal.

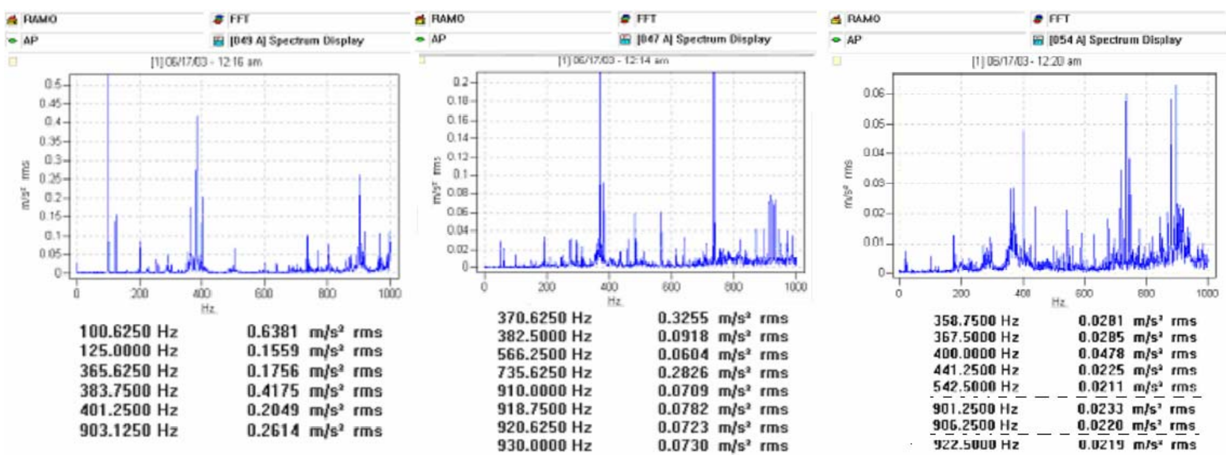
For the present study it was used an FFT analysis and impact measures for dynamic behaviour using Vibroport 41 (Schenck - BRUEL&KJAER).

The transfer functions were made on 6 direction of excitation (the accelerometer mounted on 0^o, 60^o, 120^o, 180^o, 240^o, 300^o, on lathe chuck and the impact direction on the same angle), using an accelerometer and impact hammer measures. The diagrams of variation with the angular position of the spindle are presented in figure 6. For dynamical behaviour in the real work there were made measures for FFT analysis at 300, 600, 700, 900, 1200, 1500, 1800, 2400, 3000, 4500, 6000 rpm (Figure 7)..



a. first frequency b. 2^{th} frequency c. 3^{th} frequency d. 4^{th} frequency

Fig.6 Spindle frequency variation with angular position



a. n=6000 n/min b. n=3000 n/min c.n=300 n/min

Fig.7. FFT spectre

4. CONCLUSION

The results of the theoretical study are validated with the experimental measures. All the frequencies obtained for the FEM analysis are the values close enough to the real frequencies measured. The differences between the theoretical and real values of frequency variation in different angular position of the spindle (fig.6) prove an assembly error and problems in the execution of the spindle bearings. This situation is quite difficult because the lowest stiffness is on the direction of the cutting force. Especially the spindle bore in the headstock has important deviations from the cylindrical form. In consequence this situation causes stiffness variation in different directions.

An FFT spectrum has an extensive area, from 100 to 1000 Hz, with several components with important amplitudes. In this domain of frequencies it is important to have a good equilibration of the spindle, especially for the high speed. The level of the amplitude vibration is very high at the following frequencies: 100, 383 and 903 Hz. Taking into account a 383 Hz frequency, the amplitude of the vibration corresponding at 365 and 401 Hz will be attenuate because it is situated in the same zone of resonance. In machine tool domain it is important to know dynamic behavior in different stages of machine utilization.

The paper considers dynamical behavior and mechanical signature of the spindle in order to have theoretical and real FFT spectra and transfer function of the assembly.

5. REFERENCE

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6 RETURN ADDRESS

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