IETI Transactions on Engineering Research and Practice

http://ieti.net/TERP/

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

NUMERICAL AND EXPERIMENTAL MODAL ANALYSIS OF HIGH SPEED SPINDLE

Miloš Knežev [1, a], Aleksandar Živković [1,b], Milan Zeljković[1,c], Cvijetin Mlađenović [1,d]

[1]University of Novi Sad, Faculty of technical sciences,

[a]knezev@uns.ac.rs, [b]acoz@uns.ac.rs, [c]milanz@uns.ac.rs, [c]mladja@uns.ac.rs

Abstract The need for bigger productivity has increased drastically over the last period, to cope with these increasing demands manufacturing companies are trying to improve and speed up their machining processes. The geometric quality of high-precision parts is highly dependent on the dynamic performance of the entire machining system, which is determined by the interrelated dynamics of machine tool mechanical structure and cutting process. This paper takes a grinding high-speed motorized spindle system, as a example for obtaining a finite element model analysis. The model takes into account bearing support contact interface, which is established by spring-damper element with appropriate stiffness. Furthermore, modal analysis and critical speed were done by means of ANSYS commercial software. The proposed model has been verified experimentally by measuring the natural frequencies of the spindle, and the simulated results are compared well against the experimental measurement.

Keywords: Motorized spindle; Spindle dynamics; FEM.

1. INTRODUCTION

High-speed machining (HSM) technology has been widely used in automotive, aerospace, die making, electronics and many other industries to increase productivity and reduce production costs. This technology is mainly limited by the performance of the spindle, which has a significant influence on the machining accuracy [1]. Spindle system is one of the key components of machine tools, which will determine the machining quality directly [2]. Classically, main spindles were driven by belts or gears and the rotational speeds could only be varied by changing either the transmission ratio or the number of driven poles by electrical switches. Later simple electrical or hydraulic controllers were developed and the rotational speed of the spindle could be changed by means of infinitely adjustable rotating transformers. The need for increased productivity led to higher speed machining requirements which led to the development of new bearings, power electronics and inverter systems. The progress in the field of the power electronics led to the development of compact drives with low-cost maintenance using high frequency three-phase asynchronous motors [3]. The safety and reliability due to imperfect dynamic performance have become the primary problem of structural design and machine operation. As the number of revolutions of the machine tool main spindles is increasingly approaching first natural frequency, there is a risk of resonance [4] Therefore, the dynamic performance research of the high-speed motorized spindle has an important theoretical and practical significance [5].

2. DYNAMIC MODEL OF THE SPINDLE

The dynamic analysis of spindle bearing system plays a crucial role as it directly affects the machining, productivity as well as the quality of the product [6]. The aim of this paper is to establish

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

a FEM model of the high-speed motorized spindle. The model takes into account bearing support contact interface. Modal analysis is conducted by using ANSYS commercial software.

A schematic of the spindle-bearing assembly of a high-speed grinding motorized spindle shown in figure 1. The rotor of the spindle is supported by two pairs of angular contact ball bearings. To drive this spindle-bearing system, an integral induction motor is located between the front and rear bearings, and the maximum rotating speed of the spindle is 90000 [rpm].

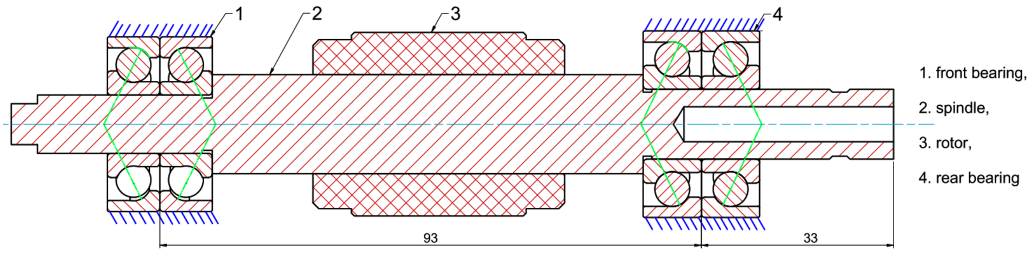


Figure 1. The spindle-bearing assembly scheme.

The model of main spindle is built by using ANSYS APDL beam modeler module and the structure such as thread hole, keyhole, chamfer, fillet and so on are simplified. This features, has no effects on the analysis result. Due to simple axial-symmetric nature of the spindle system, the spindle is described by one dimensional element type with two nodes BEAM188, which is six degree element, (3 translational and 3 rotational) based in Timoshenko beam theory, it is one-dimensional line element in space and requires cross-sectional details for modelling, which is suitable for representing the stepped nature of spindle. Steel property was used in the mathematical model as follows: Young's modulus, E = 210 [GPa], density ρ= 7850 [Kg/m3], Poisson`s ratio 0.3. Stiffness of bearings is defined by equivalent spring-dumper element type COMBIN14 on every bearing location, whereby radial stiffness value of bearings is defined according to producer catalogue, for front bearing k=240 [N/µm], and rear bearing k=156 [N/µm], both in "O" arrangement, with the fixed preload. On the same locations rotational and translational degrees of freedom along the X and Z-axis was constrained. And rotational speed of spindle was given around the Z axis. The dynamic finite element model is shown on figure 2.

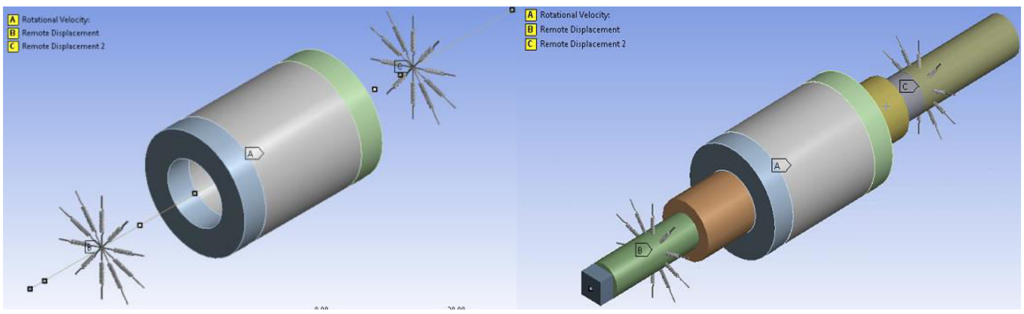


Figure 2. FE model of the spindle system

2.1. Boundary conditions

Modal analysis is the basis of dynamic analysis. The modal can be obtained by analyzing the system frequency, vibration mode, natural frequency. For modal analysis it is required to establish the

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

dynamic equation of the spindle bearing system. The dynamic equation of spindle bearing system is shown below [6]:

Equation of forced vibration:

$$m\ddot{x} + c\dot{x} + kx = F(x) \tag{1}$$

Where $F(x) = F_o \cos\omega t$

Where (m) is the mass of the system (kg), (c) is damping coefficient (Nsec/mm), (k) is the stiffness (N/mm), F(x) is the excitation vector, Fo is the constant excitation force and (x) is the displacement vector (mm).

From modal analysis the natural frequency of the spindle is found by the material properties, structure, and the damping will have very little effect on the natural frequency of spindle bearing system, therefore F(x)=0, [6]

Equation of free vibration:

$$m\ddot{x} + kx = 0 \tag{2}$$

Assuming the spindle bearing system in the simple harmonic vibration

$$x(t) = x_o sin(\omega t + \emptyset) \tag{3}$$

Where: x0 – Amplitude, ω – Angular frequency and ϕ – Phase angle. Substituting (3) in (2):

$$(k - \omega^2 M)\emptyset = 0 \tag{4}$$

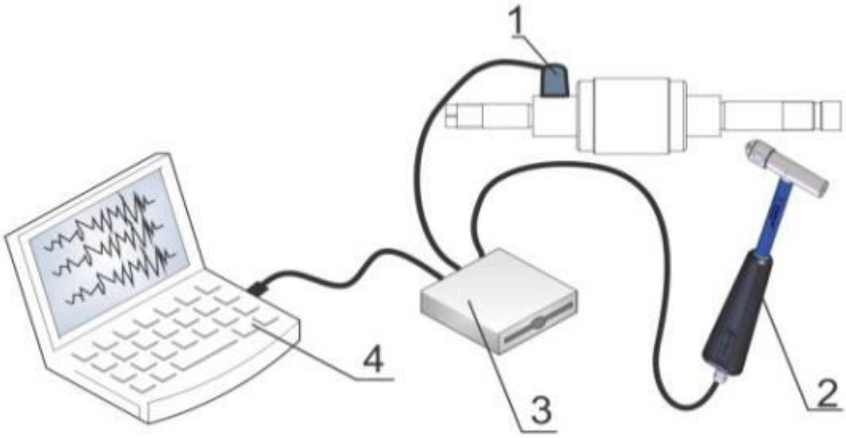
The formula (4) is used to calculate the modes of spindle bearing system where ϕi (i = 1,2,3, …., n).

2.2. Experimental setup

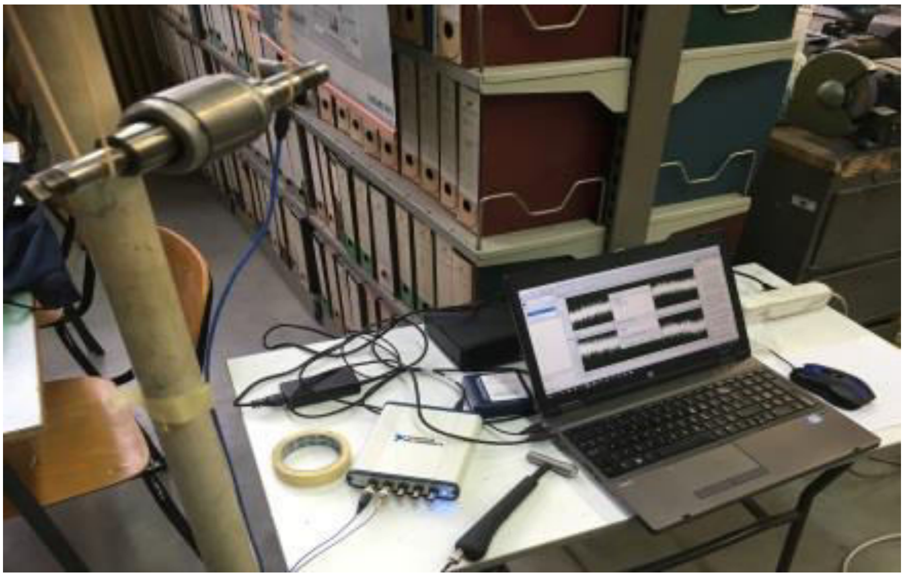
In order to verify the numerical model, impact tests was performed on the spindle. The impact force was provided by the hammer Bruel & Kjaer type 8206 and the vibrations of the system were captured using the accelerometer PCB 352C33 as shown on figure 3. The electrical signals from the sensor were transferred to the A/D converter, which converts them into digital signals. These were processed and analyzed by the data acquisition software and the final FFT graphs were plotted, the FFT provides values of natural frequencies.

IETI Transactions on Engineering Research and Practice
http://ieti.net/TERP/

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004



a)



b)

Figure 3. Experimental setup a) Scheme 1. sensor, 2. impact hammer, 3. A/D converter, 4. computer b) physical experiment

3. RESULTS AND DISCUSION

3.1. Modal analysis

The analysis of dynamic characteristics of the spindle system includes natural frequencies. Figure 4 shows the first six natural frequencies of spindle. The spindle was not constrained along its longitudinal translational direction and also along its axis of rotation. This translational and rotational freedom was the reason for the first natural frequency is close to zero i.e., rigid body mode. Therefore, from modal analysis it is clear that the critical frequencies are well beyond the working speed. In [8] states that the maximum rotating speed cannot exceed 75 [%] of its critical speed. For first mode the

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

working speed are 117252 [rpm], which is around 1/3 above the working range of spindle, 0 – 90000 [rpm].

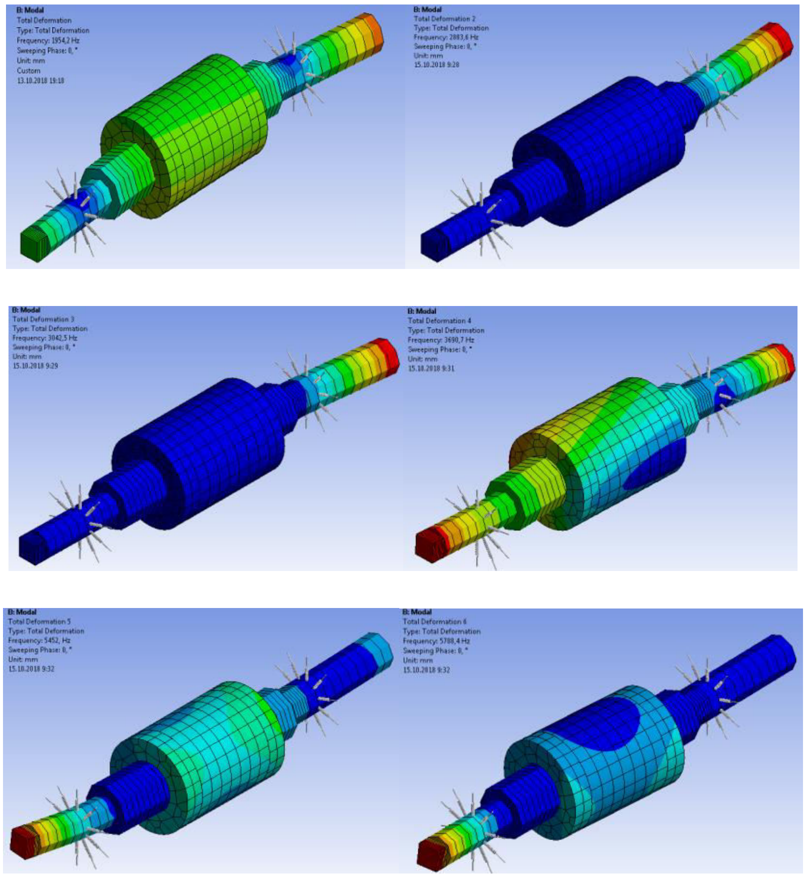


Figure 4. First six mode shapes (respectively) for spindle system.

3.2. Critical speed and Campbell Diagram

In this paper, in order to prove previous notion i.e. investigate the dynamic behavior of the spindle at operating speeds, the Campbell diagrams, critical speeds are obtained. The numerical analyzes are performed considering speeds ranging from 0 to 90000 [rpm]. Figure 5 shows the Campbell diagrams obtained by the FEM model, where could be seen that the spindle is stable in whole range of revolution.

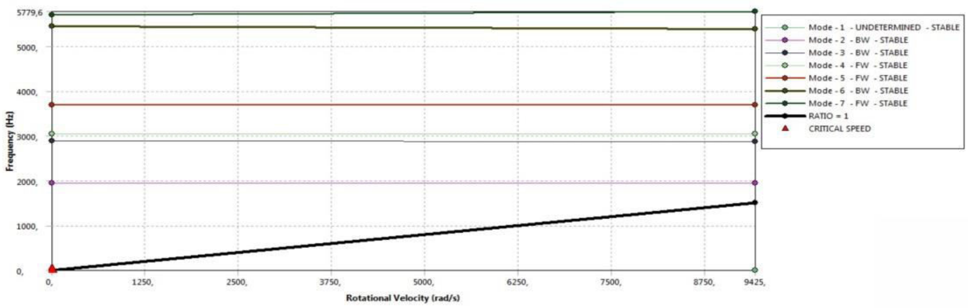


Figure 5. Campbell diagram from FEM model.

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

3.2. Experimental results

The experimental vibrational analysis of the spindle was used for validation of the numerical model, procedure is discussed in previous chapter. From the experimental impact tests, the natural frequency of the spindle was measured and found value of first was 1907 [Hz], as shown on figure 6. A difference of around 2 [%], between the simulation and the experimental results of the spindle setup is acceptable.

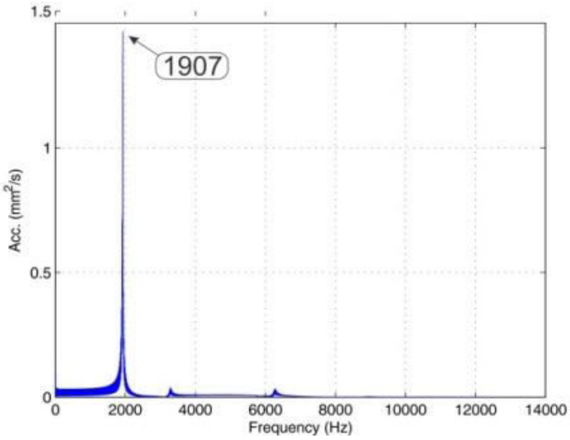


Figure 6. FFT plot for spindle.

4. CONCLUSION

Spindle structure and bearings, plays an important role in failure of spindle bearing system. The mathematical model of the spindle system using beam elements was developed using, finite element method in ANSYS Workbench software using BEAM 188 for model, and COMBIN 14 element for modeling the bearing.

This approach was chosen because of its major advantage, which is that dynamic condition can be easily integrated with complex geometries and physical conditions, especially for any further analyses of the influence of temperature or static behavior of the spindle.

Impact test was carried out on spindle, with force provided by hammer and displacement were captured using accelerometer. Data acquisition system was used to measure and analyze the output from the accelerometers, and natural frequencies of the system were found from the FFT analysis.

On the other hand this paper presents a method for calculating natural frequencies, modal shapes. Additionally, the simulation errors of the natural frequencies are around 1 [%], which in turn confirms the effectiveness of the simulated model proposed in this paper.

Acknowledgements

In this paper some results of the project: Contemporary approaches to the development of special solutions related to bearing supports in mechanical engineering and medical prosthetics – TR 35025, carried out by the

2018, Volume 2, Issue 2, 27-33, DOI 10.6723/TERP.201812_2(2).0004

Faculty of Technical Sciences, University of Novi Sad, Serbia, are presented. The project is supported by the Ministry of Science and Technological Development of the Republic of Serbia.

References

[1] Shuyun J., and Shufei Z., 2010, A modeling approach for analysis and improvement of spindle-drawbar-bearing assembly dynamics, *International Journal of Machine Tools and Manufacture*, 50, p.p. 131–142.
[2] Guanghui L., Jun H., Wenwu W., and Yanhui S., 2018, Investigation on the influence of interference fit on the static and dynamic characteristics of spindle system. *The International Journal of Advanced Manufacturing Technology*, Vol. 99, no. 5-8, p.p. 1953.
[3] Deping L., Hang Z., Zheng T., and Yufeng S., 2011, Finite Element Analysis of High-Speed Motorized Spindle Based on ANSYS, *The Open Mechanical Engineering Journal*, Vol. 5, p.p. 1-10.
[4] Zeljković M., Živković A., and Tabaković S., 2012, Computer and experimental aidd analysis of hihg-speed spindle assembly dynamic behaviour, *1st International scientific conference COMETa*, University of East Sarajevo, Faculty of Mechanical Engineering, p.p. 63-73.
[5] Li C., Ding Y., and Lu B., 2009, Development and Key Technology in High Speed Cutting, J. Qingdao Technol. Univ., Vol. 30, p.p. 7-16.
[6] Ayush A., and Himel R., 2018, Static and Dynamic Analysis of Lathe Spindle using ANSYS, *International Journal of Applied Engineering Research*, Vol. 13, No. 9, p.p. 6994-7000.
[7] Guo C., Bai L., Zheng B., Pan Y., 2013, Spindle Static and Dynamic Characteristics Analysis of Precision CNC Turning Center, *Advanced Materials Research*, Vol. 619, p.p. 47-50.