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MODAL ANALYSIS OF MOTORIZED SPINDLE USING FINITE ELEMENT METHOD

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Abstract: *Dynamic market conditions shows contant need to significantly shorten the time cycle of the product design phase, with demands for the lowest possible price and highest possible quality. Motorized spindle is already widely used as the main spindle of modern machine tools. Predict dynamic characteristic through numerical simulations is trend among machine tool manufacturers, especially when high accuracy is involved. Finite element method (FEM) is the most commonly used method for analyzing the dynamic behavior of the main spindle. In this work, a model was developed based on the finite element method. The obtained results are compared with experimental test.*
Key words: *Modal analysis, FEM, Motorized spindle.*

Modalna analiza motor-vretena primenom metode konačnih elemenata. *Dinamični tržišni uslovi pokazuju stalnu potrebu da se značajno skrati vremenski ciklus faze projektovanja proizvoda, sa zahtevima za najnižom cenom i najvišim mogućim kvalitetom. Motor-vreteno je već pronašlo široku primenu kao glavno vreteno savremenih mašina alatki. Predikcija dinamičkih karakteristika kroz numeričke simulacije je trend među proizvođačima mašina alatki, posebno kada je u pitanju visoka tačnost. Metoda konačnih elemenata (MKE) je najčešće korišćena metoda za analizu toplotnog ponašanja glavnog vretena. U ovom radu razvijen je model zasnovan na metodi konačnih elemenata. Dobijeni rezultati su upoređeni sa eksperimentalnim testom.*
Ključne reči: *Modalna analiza, MKE, Motor-vreteno.*

1. INTRODUCTION

The machine tools behavior in exploitation is conditioned by the behavior of certain vital assemblies [1]. One of the most influential machine tool components, affecting the accuracy and productivity of machining is the spindle unit. According to Li [2], the magnitude of the natural frequency also depends on the installation method of the angular contact ball bearings. It was noticed that for the "X" bearing layout, the values of the natural frequencies increase as the number of revolutions increases, while for the "O" bearing layout with the same preload, the value of the natural frequency decreases with the increase in the number of revolutions [3] [4]. Prediction of the dynamic behaviour of machine tools in the design phase is an important step in their development.

A particularly important place is the identification of the dynamic behaviour of the main spindles, which is conditioned by the behaviour of individual elements (bearings, tool holder, tool,), as well as the influence of connections (joints) between elements.

The dynamic behaviour of the main spindle assembly directly affects the stability of the cutting process. Its properties also affect the reliability of the entire system. The problem of looking at the dynamic properties of the system mainly spindle bearings, lies in the fact that with every change in the number of revolutions, tool, tool holder, the conditions and properties of the system change.

Finding the natural frequencies affects the RPM ranges in which the machine will operate, i.e. vice versa, the spindle will be modified so that the natural oscillation frequencies are outside the standard RPM ranges at which the machine tool operates. Bearing positions, type of bearing method and type of tools and tool holders, etc. it will affect the shape of the main spindle oscillation and the reliability of the system.

The movement of the spindle tip during oscillation affects the stability of the cutting process, contributes to a lower quality of the processed surface and lower accuracy of the work piece.

2. FEM ANALYSIS OF MOTORIZED SPINDLE

An analysis of the dynamic behaviour of the motorized spindle was carried out for a freely supported main spindle with bearing. This analysis included the development and determination of natural frequencies and main modes of shaft oscillation without damping, and without taking into account damping in the material. After the evaluation of the results obtained by the FEM method, a comparison will be made with the results obtained by the experimental test.

2.1 Motorized spindle

Tested motorized spindle the GMN TSSV 90000, whose cross-section is shown in figure 1.

The shaft is fitted with two pairs of high precision ball bearings with angular contact, the front bearing is EX 12 7C1 DUL SNFA while the rear is EX 10 7C1 DUL SNFA, mounted in a "tandem" arrangement in pairs, so that the entire bearing forms an "O" arrangement.

The high-speed motorized spindle assembly is designed as an asynchronous electric motor, so the stator is mounted in the housing, and the rotor is integrated with the shaft. One of the sources of generated heat is the asynchronous motor, while the other source is the bearings. For this reason, the stator is mounted in a sleeve with a channel around it, through which the cooling fluid flows, while the cooling of the bearing is done by pressurized air and oil, i.e. oil mist flowing through the bearings and in the gap between the rotor and the stator.

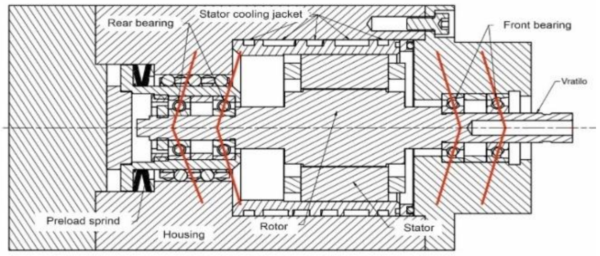

Fig. 1. Cross section of GMN TSSV 90000 motorized spindle assembly

2.2 FEM modal analysis

Through experimental tests (Figure 3), it is possible to generate the frequency response function (FRF) or the system transfer function (TF). The basic idea is to excite the machine structure with a force of a certain frequency at a certain place, and to monitor the dynamic response of the system at the same or some other place.

In general, the assembly of the motorized spindle is complex for simulation, due to the nonlinearity of the system itself, as well as the complexity of mechanical and thermal phenomena. In order to examine the influence of certain parameters on the dynamic behaviour of the main spindle, a dynamic numerical model of the spindle was developed in the paper (figure 2). An isoparametric hexahedron (SOLID 187) was used for the discretization of FEM spindle model 27606 of them.

The analysis of dynamic behavior was performed without the effect of the excitation force (modal) and with the effect of the excitation force on the top of the spindle (harmonic), in order to define the amplitude-frequency characteristics of the considered main spindle for certain natural frequencies and given excitation. The force (220 N) is given at the top of the spindle, while the spindle responses are observed at points 1, 2, 3, 4 and 5 (Figure 2)

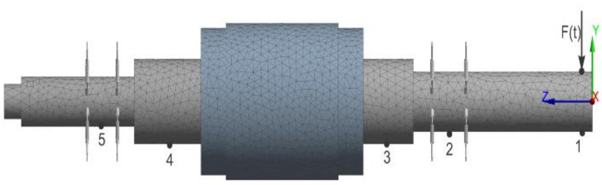

Fig. 2. Discretized model of shaft

3. RESULTS AND DISCUSION

The defined frequency response function obtained by the numerical model (Figure 3), i.e. its real and imaginary part, is used to calculate the modal parameters of the modal stiffness (k) and modal mass (m) system in the form:

$$k=\frac{-1}{2\varsigma H} \quad (1)$$

$$m=\frac{k}{\omega_n^2} \quad (2)$$

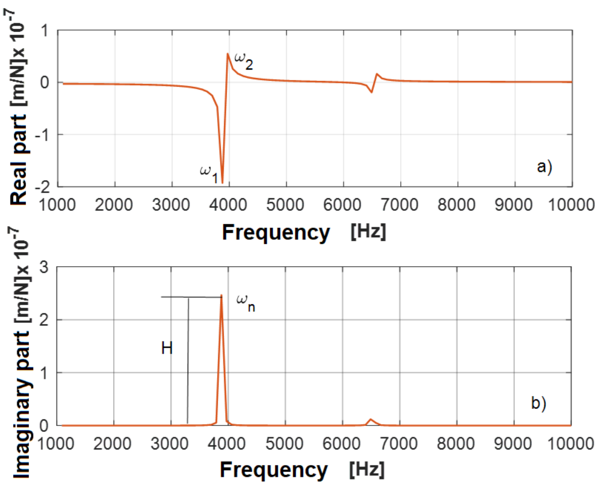

Fig. 3. a) Real; b) imaginary part of the frequency response function at point 1 for a freely supported main spindle

Figure 4 shows the first form of oscillation of a freely supported shaft.

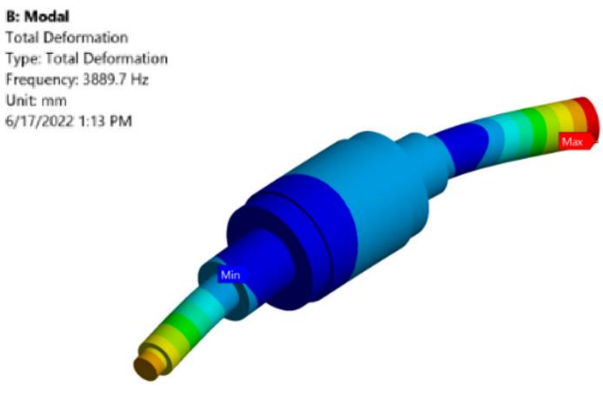

Fig. 4. First natural frequency and form of oscillation for freely supported shaft

During the mathematical modelling of the dynamic behaviour of the coefficient value, the damping was varied in a very wide range from 0.003 to 0.3 for the case of the action of the excitation force on the top of the spindle. During experimental testing, the damping coefficient values were accurately calculated using the frequency response function.

Figure 5 shows the first three forms of oscillation, i.e. the first three natural oscillation frequencies of the freely supported main spindle without consideration of damping.

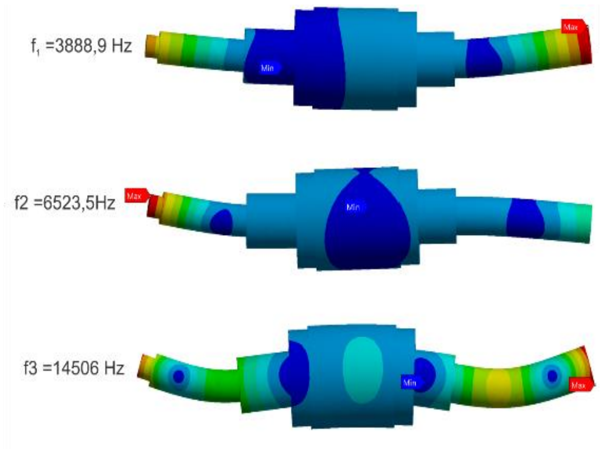

Fig. 5. The first two forms of oscillation and natural frequencies

In the analysis with the excitation force at the top of the spindle, the frequency ranges up to 10 [kHz] was observed because the first natural frequency is twice as high as the rotation frequency of the motor-spindle, so frequencies above 10 [kHz] are not critical.

For different values of the relative damping coefficient (ξ). Comparisons of natural frequencies for the first two oscillation modes determined by FEM modeling and experimental testing are shown in Table 1. It should be noted here that the first two natural frequencies are the same in all considered points, only the amplitudes at these frequencies differ.

Exp no.	**Narural frequencies**	**FEM model [Hz]**	**Experiment [Hz]**
1	f_1	3887.2	3838
2	f_2	6580.3	6525

Table 1. Comparison of the first two natural frequencies (f_1 and f_2) obtained by FEM modeling and experimental testing

In addition, the width of the top of the resonance curve at the height as:

$$\alpha = \frac{\sqrt{2}}{2} \tag{3}$$

is used in the literature to determine the damping coefficient according to the relation:

$$\xi_2 = \frac{\Delta_f}{2f_r} \tag{4}$$

where Δf represents the frequency difference at the height of the curve, while f_r is the maximum value of the frequency (Figure 7).

The values of natural frequencies and relative damping coefficients determined according to the previously described methods are shown in Table 2.

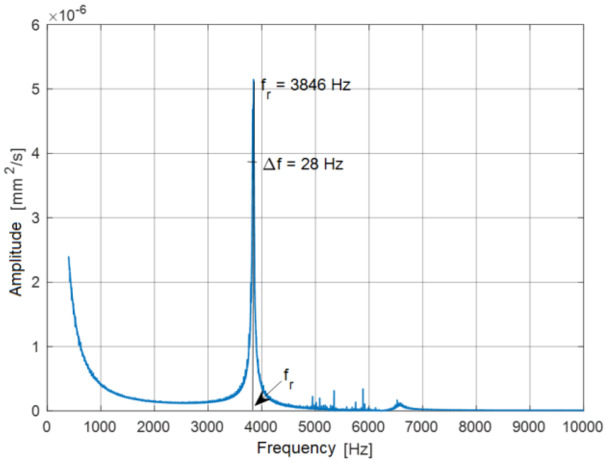

Fig. 6. Amplitude-frequency characteristic of the considered spindle at measuring point 1

	Mesuring point 1
f_r [Hz]	3846
f_{11} [Hz]	3826
f_{12} [Hz]	3852
f_{1n} [Hz]	3840
ξ_1 [-]	0,003386
ξ_2 [-]	0,003251

Table 2. Values of natural frequencies and relative damping coefficient

Table 3 shows the other modal parameters determined by the frequency response function.

	Modal parameters	
	k = [N/μm]	m [kg]
Measuring point	30,8	0,206

Table 3. Other modal parameters for $\xi 1 = 0.003386$

Figures 7 to 10 show the amplitude-frequency characteristics for the other considered points on the motor-spindle.

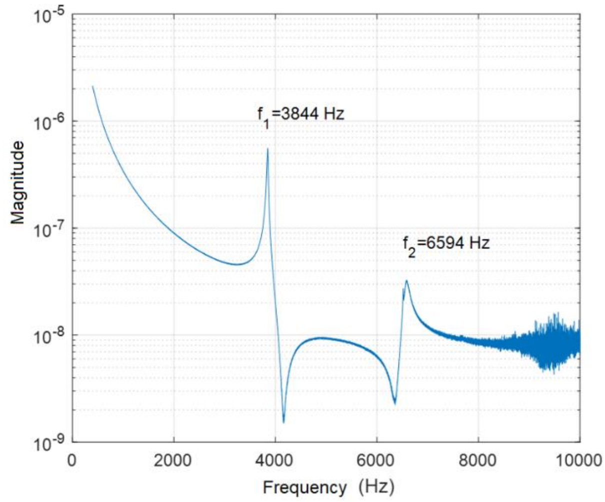

Fig. 7. Amplitude-frequency characteristic of the considered spindle at measuring point 3

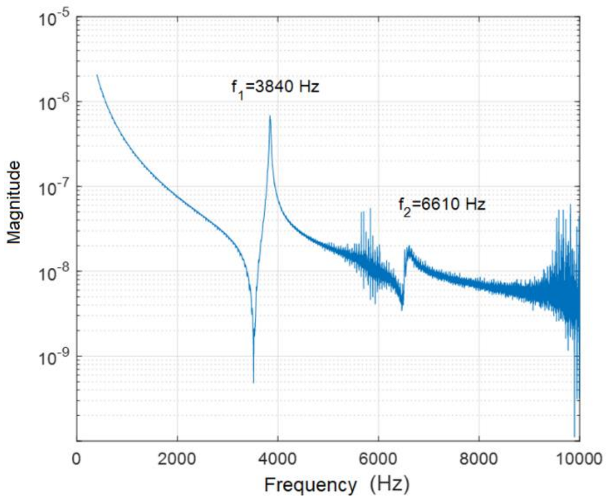

Fig. 8. Amplitude-frequency characteristic of the considered spindle at measuring point 4

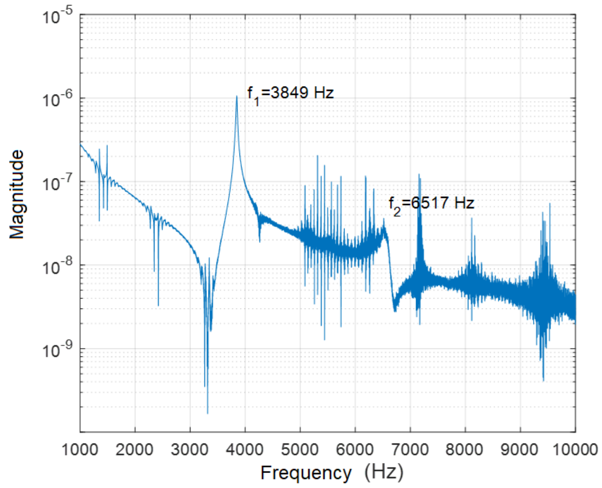

Fig. 9. Amplitude-frequency characteristic of the considered spindle at measuring point 5

4. CONCLUSION

By comparing the values of the characteristic natural frequencies (Table 1), it can be seen that the deviation between the frequencies determined by FEM modelling and experimental testing, on the first mode is 1.2 [%], which is also the most dominant.

Based on this, it can be concluded that FEM modelling gives satisfactory results, especially considering that the difference between FEM modelling and experimental testing at the second natural frequency is only 0.8 [%].

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