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MODAL ANALYSIS OF BALL BEARINGS USING FINITE ELEMENT METHOD

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Abstract: *Dynamic behavior of roller bearings under the loads can be described by a mathematical model. The dynamic bearing model is modeled using the finite element method. Analysis of the dynamic behavior of the radial bearing refers to the modal analysis, ie. the determination of the natural frequencies, the main forms of oscillation and damping for the considered bearing, with different values of the clearance. This analysis is based on the simultaneous analysis of excitation and response signals in the time domain or more often in the frequency domain. In addition to FEM, it was also performed the experimental test on a FKL 6006 bearing with different clearance values.*

Within this paper, a comparison of the results of the modal analysis performed experimentally with the results obtained by the finite element method was performed. The compared results show a satisfactory deviation.
Key words: *frequency, FEM, modal analysis, modal parameters, oscillation modes*

Modalna analiza kotrljajnih ležaja metodom konačnih elemenata. *Dinamičko ponašanje kotrljajnih ležaja pod dejstvom opterećenja, može se opisati matematičkim modelom. Dinamički model ležaja je definisan primenom metode konačnih elemenata. Analiza dinamičkog ponašanja radijalnog ležaja odnosi se na modalnu analizu, tj. određivanje sopstvenih frekvencija, glavnih oblika oscilovanja i prigušenja za razmatrani ležaj, sa različitim vrednostima zazora. Ova analiza se zasniva na istovremenoj analizi signala pobude i odziva u vremenskom ili češće u frekventnom domenu. Pored MKE, sprovedeno je i eksperimentalno ispitivanje ležaja FKL 6006 sa različitim vrednostima zazora.*

U okviru ovog rada izvršeno je poređenje rezultata modalne analize koja je sprovedena eksperimentalnim putem sa rezultatima dobijenim metodom konačnih elemenata. Upoređeni rezultati pokazuju zadovoljavajuće odstupanje.
Ključne reči: *frekvencija,MKE, modalna analiza, modalni parametri, modovi oscilovanja*

1. INTRODUCTION

Rolling bearings are one of the most widely used machine elements, whose task is to serve as supports for shafts with simultaneous transmission of radial and axial forces, as well as ensuring the accuracy of their position. Ball bearings are a complex modeling system with a large number of input and output parameters and complex physical and chemical processes that occur during their exploitation. For these reasons, it is practically impossible to form a comprehensive mathematical model for the analysis of ball bearing behavior.

Based on Hertz's theory of contact and raceway control theory, Jones [1] and Harris [2] proposed a classical model for rolling bearing analysis, which is still used in bearing simulation. In analyzes, the contact angle is often considered constant, but in reality it is not. Liao [3] proposed a new model for the analysis of displacement and load in a ball bearing with a variable contact angle. Wang [4] developed a new quasi-static model for the calculation of ball bearings, which are exposed to the combined load of radial forces, axial forces and moment. Stribeck [5] was the first to investigate the load distribution for the radial load of ball bearings with zero clearance, and later extended the research to different clearances in ball bearings. Based on Stribeck's research, Sjovall et al. [6] investigated the load distribution on ball and the state of contact between the ball and the raceway under a given radial and axial preload. The clearance affects the dynamic behavior of the ball bearings. the clearance affects the creation of disturbing forces that cause oscillatory movement and vibrations of the bearing elements. The size of the radial clearance does not affect the natural frequency of the bearing [7, 8] and the clearance does not affect the frequency of rotation of the bearing elements[9].

The effects of static and dynamic bearing behavior must be predicted with great certainty at the design phase. As already mentioned, in order to obtain the most favorable construction, the defined static and dynamic model must take into account a large number of parameters that affect the behavior of the corresponding bearing [10]. Therefore, in practical application when establishing a mathematical model, more attention is paid to the parameters that affect the behavior of bearings in exploitation.

In this paper, the procedure for determining modal parameters, experimental testing and FEM model is presented. Experimental tests are one of the methods of determining modal parameters that are, for the most part, simple, fast and very importantly, non-destructive method. Based on the identified modal parameters, it is possible to obtain a mathematical model of dynamic behavior that describes the dynamic behavior of ball bearings under the action of external loads.

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2. EXPERIMENTAL MODAL ANALYSIS

Experimental modal analysis is an experimental definition of the modal parameters of a linear, time-invariant system that represent a significant basis for the analysis of the dynamic behavior of roller bearings. This analysis is based on the simultaneous analysis of excitation and response signals in the time domain or more often in the frequency domain.

In order to determine the modal parameters of the observed ball bearings, ie. natural frequency, modal stiffness and damping coefficient, it is necessary to experimentally define the function of the frequency response of the ball bearing signal based on the input signal (excitation) and the output signal (in this case acceleration). Figure 1 shows an experimental model for determining the bearing frequency response function, which consists of an acceleration sensor (1), which measures the oscillation on the outer bearing ring, and an excitation hammer (2), which excites the bearing.
The excitation hammer and the acceleration sensor are connected to the A/D card (3), which sends the collected data directly to the computer (4).

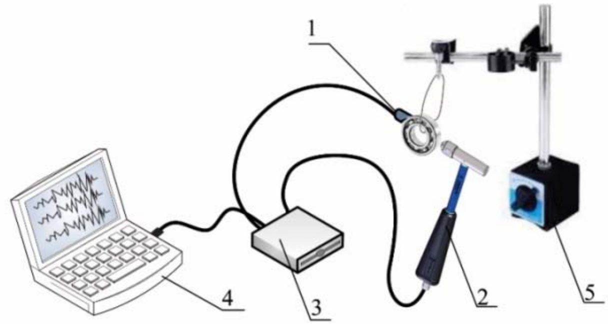

Fig. 1. Schematic of the experiment to determine the FRF bearings

The signal collected by the acceleration sensor and the hammer signal are sent via A/D card to a PC, where they are stored in tabular form using a specific software system. By creating an algorithm in the program system, fast Fourier transform (FFT) of the obtained signals and determination of the frequency response function of the observed system are enabled. The system frequency response function is displayed as a real and an imaginary part. It should be noted that when defining the real and imaginary part of the bearing frequency response function, signal filtering below 50 Hz and above 10 kHz was applied in order to remove unnecessary noise from the recorded signals, and to obtain as clear diagrams as possible.

2.1 Experimental determination of frequency response function and modal parameters of ball bearings

Experimental modal analysis with impulse excitation force is the simplest method for determining the frequency response function, and as such is most often used to define the modal parameters of the bearing. The transfer function determined by measuring for one mode of oscillation, for the "real" and "imaginary part". The maximum value of the amplitude is not at $\omega/\omega_n=1$, but at lower values, and the attenuated natural frequency has a value $\omega_d=\omega_n\sqrt{1-\xi^2}$. The same applies to the minimum value of the transfer function $\omega_d=\omega_n\sqrt{1+\xi^2}$. According to [11] , the modal damping coefficient can be determined from the real part of the transfer function as:

$$2\xi=\frac{\left(\frac{\omega_2}{\omega_1}\right)^2-1}{\left(\frac{\omega_2}{\omega_1}\right)^2+1} \quad (1)$$

When the recorded data in the time domain is transformed into a frequency domain using fast Fourier transform, the Frequency Response Function (FRF) is obtained between the point where the response of the system (i) was measured and the point where the excitation force (j) acted, which can be shown as follows:

$$FRF_{ij}(\omega)=\frac{X_i(\omega)}{F_j(\omega)} \quad (2)$$

Also, it should be noted that in most cases it is necessary to obtain the displacement X as the response of the system. However, if the measurement is performed with acceleration sensors, the previous equation has the form [12]:

$$FRF(\omega)=\frac{\ddot{x}}{F(\omega)}=\frac{A(\omega)}{F(\omega)} \quad (3)$$

The conversion of acceleration into displacement can be performed by dividing the signal collected by the accelerometer by $(i\omega)^2$ and-ω^2, respectively.

The frequency response function of the system, obtained in the previously described way, is complex, ie it consists of a real G and an imaginary H part:

$$FRF(\omega)=G+iH \quad (4)$$

The real and imaginary part of its frequency response function can be calculated based on the following expressions:

$$\mathrm{Re}\left(\frac{X}{F}\right)=\frac{1}{k}\left(\frac{1-r^2}{(1-r^2)^2+(2\cdot\zeta\cdot r)^2}\right) \quad (5)$$

$$\mathrm{Im}\left(\frac{X}{F}\right)=\frac{1}{k}\left(\frac{-2\cdot\zeta\cdot r}{(1-r^2)^2+(2\cdot\zeta\cdot r)^2}\right) \quad (6)$$

where is: $r=\frac{\omega}{\omega_n}$, at $\omega=\omega_n$ (the moment when resonance occurs) the imaginary part of the transfer function has value $H=-1/(2\xi k)$, then is

$$k=\frac{-1}{2\varsigma H} \quad (7)$$

Forasmuch it is its own frequency $\omega^2=k/m$, modal mass m can be expressed as:

$$m=\frac{k}{\omega_n^2} \quad (8)$$

where is: ξ - damping coefficient; k - modal stiffness, where H is the minimum of the imaginary part FRF; m - modal mass.

2.2 Experimental modal analysis for ball bearing 6006

Experimental modal analysis is performed by measuring the bearing response to the excitation force generated by the excitation hammer at a point located in the middle of the outer surface of the outer ring for the bearing, which is supported on a magnetic holder, by means of an elastic rubber band. The measurement is performed using an acceleration sensor. Figure 2 shows the procedure of experimental modal analysis.

Modal analysis was performed for bearings 6006 with clearance values of 10, 20, 30 and 40 μm. Figure 3 shows the appearance of the real and imaginary part of the FRF for the tested bearing and the gap values of 20 μm. It can be seen from the figures that the tested bearing has only one dominant mode in the observed frequency range, so the modal parameters of the bearing are determined only for that mode, based on relations (7) and (8). The calculated results are shown in Table 1.

Fig. 2. Experimental modal analysis of ball bearings

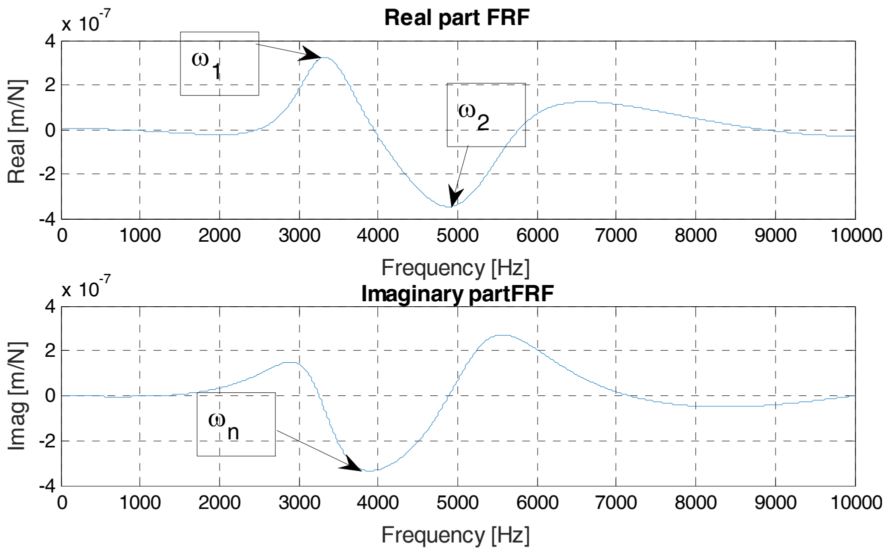

Fig. 3. Real and imaginary part of FRF for ball bearing 6006 and clearance value G_r = 20 μm

Clearance	ω_1	ω_2	ω_n	ζ	*k [N/m]*	*m [kg]*
10	*3296*	*4853*	*4267*	*0,36*	*0,22 x 10^7^*	*0,119*
20	*3337*	*4866*	*3919*	*0,38*	*0,19 x 10^7^*	*0,123*
30	*3384*	*4889*	*4121*	*0,35*	*0,20 x 10^7^*	*0,117*
40	*3323*	*4708*	*3995*	*0,33*	*0,18 x 10^7^*	*0,113*

Table 1. Modal parameters calculated based on experiment and FRF application

3. MODAL ANALYSIS OF BEARING OBTAINED BY FEM MODEL

In order to examine the modal parameters, a numerical model of ball bearings was developed, modeled in a general purpose software system, based on the finite element method (FEM). Modal analysis was performed for different clearance values (0, 10, 20, 30 and 40 μm) for the case of a freely suspended bearing, in order to compare the results with the experimental tests. Figure 4 shows the main forms of vibrations of the observed bearing for a radial clearance of 20 μm.

In this case, the modal mass is approximately equal to the mass of the outer ring. The next two modes are the modes of tilt of the ring around the Z axis, ie the bearing moves from the radial plane (Figure 4b), where both frequencies coincide, too. In any case, the amplitude of the natural frequency (observed on the outer ring) is very small due to the small angular stiffness, so this natural frequency can hardly be registered in the experimental measurement. The next two modes are the oscillations of the inner ring along the X and Y axes (Figure 4c), with a very small difference between the natural frequencies (less than 2%). In this case, the amplitude of the outer ring is almost equal to 0, which indicates that these modes will be difficult to detect during experimental tests. The next eleven modes are the so-called rolling body oscillation modes. Each individual mode of rolling elements has corresponding natural frequencies that differ very little (about 1%) (Figure 4d). Also, in this case, the amplitude of the outer ring is equal to 0, so these modes will be difficult to notice when measuring.

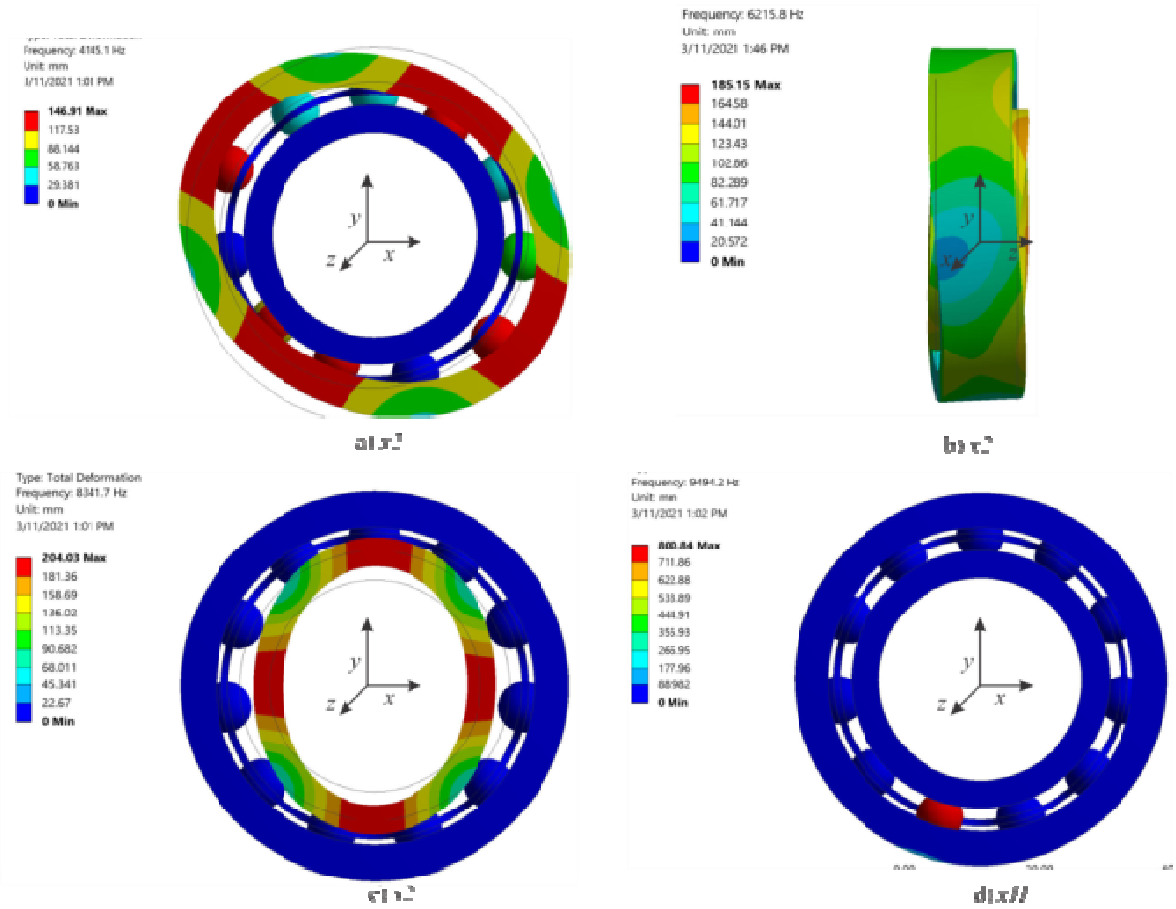

Fig. 4. Radial bearing oscillation modes

Figure 5 shows the amplitude-frequency diagram obtained during FEM modeling of a radial bearing, where it is seen that the dominant natural frequency that occurs due to the oscillation of the outer ring of the bearing. Also, the natural frequencies of the second mode can be observed on the diagram, the amplitude of the vibration of the rings lies around the Z axis (ω_{02} = 6348 Hz), as well as the amplitude of the vibration of the rolling bodies (ω_{o3} = 9284 Hz). The modal parameters of bearings based on FEM modeling were determined by the frequency response function, as in the experimental test, as shown in the previous section. Since the considered bearings have one dominant mode in the observed frequency range, regardless of the clearance, the modal parameters are determined only for that one mode.

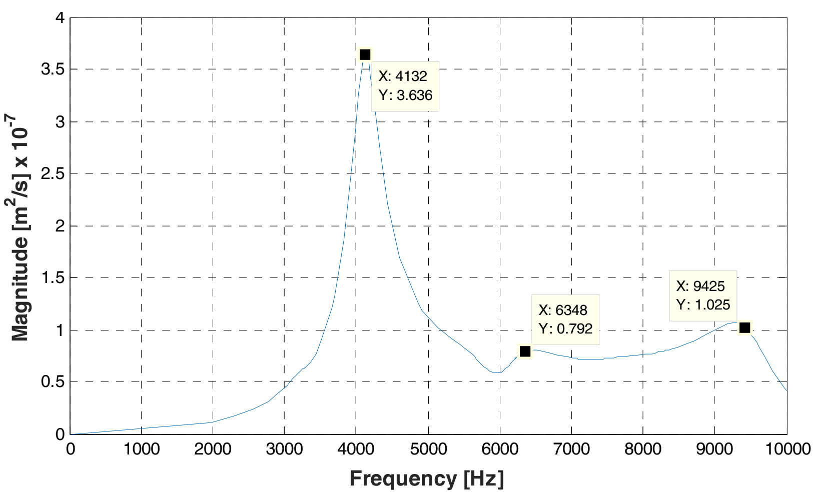

Fig. 5. Amplitude-frequency characteristic of radial bearing determined by FEM modelin

Figure 6 shows the real and imaginary part of the frequency response functions for considering bearings with a clearance of 20 µm. The comparison of natural frequencies for the first three modes of oscillation determined by experimental testing and FEM modeling is shown in Table 2.

Table 3 shows the calculated modal parameters for the observed bearings with a clearance of 20 µm and their comparison with experimentally determined data.

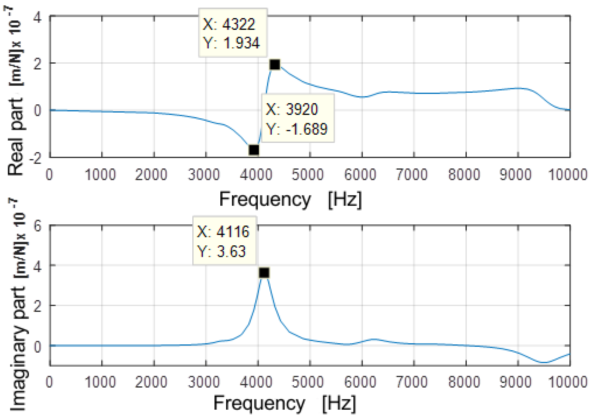

Fig. 6. Real and imaginary part of the frequency response function

Natural frequencies	*Experiment [Hz]*	*FEM model [Hz]*	*Deviation*
ω_1	*3450*	*4132*	*-16,51 %*
ω_2	*6379*	*6348*	*0,49 %*
ω_3	*9146*	*9284*	*-1,49%*

Table 2. Comparison of natural frequencies determined experimentally in relation to the obtained by FEM modeling for G_r = 20 µm

Modal parameters	*Experiment*	*FEM model*	*Deviation*
ω_n *[Hz]*	*3919*	*4116*	*-4,79%*
Kx [N/µm]	*19*	*21*	*-9,52%*

Table 3. Comparison of modal parameters determined by experiment and by FEM model

4. DISCUSSION AND CONCLUSIONS

By comparing the values of the characteristic frequencies from Table 2, it can be seen that the largest deviation between the frequencies determined by FEM modeling and experimentally, are on the first mode, which is also the largest. However, as the maximum deviation is about -16%, it can be concluded that FEM modeling gives satisfactory results, especially since the difference between FEM modeling and experimental testing at the frequency fn (Table 3) is only -4.79%. A comprehensive analysis of the results also revealed that the change in clearance in this way of testing (free support bearing) has a negligible impact on the change of natural frequencies, as well as on the change of modal parameters.

Based on the previous diagrams and Table 6.2, it can be concluded that the clearance does not affect the natural frequencies, as well as the modal parameters of the radial bearing.

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