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DETERMINATION OF CLEARANCE IMPACT ON VIBRATIONS OF RADIAL BALL BEARING THROUGH EXPERIMENTAL TESTING

Mirjana BOJANIĆ ŠEJAT, Nicolae Florin COFARU, Milan RACKOV, Aleksandar ŽIVKOVIĆ, Ivan KNEŽEVIĆ, Ilena Ioana COFARU, Dejan LUKIĆ

Abstract: *Rolling bearings are machine elements used in shaft and axle supports to transmit loads and ensure necessary precision between parts in relative motion. They represent one of the most critical components of machines. Despite consisting of only four components (inner and outer rings, cage, and rolling elements), the static and dynamic behavior of bearings is highly complex. Research into the dynamic behavior of bearings is motivated by the desire to reduce noise and vibrations in their application, as well as to increase longevity, stiffness, operational speed, and rotational accuracy, and to develop methods for bearing diagnostics and monitoring. The dynamic behavior of bearings is governed by the dynamic behavior of their structural elements. Vibrations occurring in rolling bearings during operation are unavoidable. Clearance is of great importance for achieving satisfactory bearing performance. Clearance designed to compensate for thermal expansion of bearing elements is a source of vibrations and introduces nonlinearity in dynamic behavior. This study conducted experimental testing to observe the influence of clearance on the dynamic behavior of the radial bearing FKL 6006. Vibrations were measured using an experimental device for measuring and controlling rolling bearing vibrations, analyzing the impact of external axial load and clearance on the dynamic behavior of the radial bearing. Measurements were carried out at the Laboratory for Machine Tools, Flexible Technology Systems, and Automation of Design Processes at the Faculty of Technical Sciences in Novi Sad. During vibration measurements, bearing vibration results were recorded digitally and are suitable for further analysis.*

Keywords: *dynamic behavior, experiment, bearing, clearance, vibrations.*

1. INTRODUCTION

Based on the available literature, it can be concluded that the issue of identifying the behavior of bearings/bearing assemblies represents a highly attractive research area, as evidenced by the large number of researchers who have studied this topic over an extended period and continue to do so today. All researchers unequivocally agree that the static-dynamic behavior, along with the rotational accuracy of the bearing, are the primary indicators of the quality of this assembly. In line with these observations, the objective of this study is to determine the influence of design parameters (clearance/interference) and operational parameters (external load) on dynamic behavior, specifically the identification of operational behavior and vibration measurement of the FKL 6006 radial bearing.

Under the influence of external forces, due to the rotation of bearing elements, periodic changes in the elastic deformations of the rolling tracks occur, leading to the emergence of vibrations in the bearing elements [1-3]. Experimental investigations are conducted in laboratory conditions and operational conditions. Laboratory tests are typically carried out on special test benches designed for this purpose, where different operating regimes of the bearings are simulated. Vibration analysis in such conditions is based on the guidelines specified by ISO 1542-1-2004(E). The level of vibrations is analyzed in three frequency ranges: 50-300 Hz, 300-1800 Hz, and 1800-10000 Hz, for smaller bearings at a rotational speed of 1800 rpm. Larger bearings are tested at a rotational speed of 700 rpm, with corresponding frequency ranges of 20-120 Hz, 120-700 Hz, and 700-4000 Hz [4]. A comprehensive overview of the measurement techniques used for the vibration analysis of rolling bearings is presented by Marinescu [5], Howard [6], Harris [3], and Harnoy [7].

2. EXPERIMENT

The need for experimental testing of real systems arises from the fact that mathematical models struggle to account for certain factors related to manufacturing errors, geometric imperfections of rolling tracks (particularly waviness and roughness), and bearing component assembly. This paper presents the fundamental principles of experimental testing, describes the testing device, and showcases some measurement results.

The experimental testing was conducted on an FKL 6006 radial ball bearing (Figure 1), where bearing vibrations were measured using an experimental device for measuring and controlling rolling bearing vibrations. The study analyzed the influence of external axial load and clearance on the dynamic behavior of the radial bearing. The measurements were carried out at the Laboratory for Machine Tools, Flexible Technological Systems, and Automation of Design Processes at the Faculty of Technical Sciences in Novi Sad. During vibration measurements, the recorded vibration data is stored in digital format, making it suitable for further analysis. The testing was conducted on a sample of 30 bearings of different radial clearance classes, assuming that the geometry of the rolling elements was ideal.

Fig. 1. The tested bearing 6006

2.1 Experiment plan

According to the ISO 5753-1:2009 standard, five classes of radial clearance are specified for radial ball bearings. The geometry of the bearings used in the testing is known. The radii of the rolling paths in the cross-section perpendicular to the rolling axis of the rolling bodies, roughness, waviness, and deviation from the roundness of the rolling paths of the outer and inner rings are also known. The experiment was conducted under laboratory conditions at room temperature, following the requirements of standards ISO 554, ISO 558, and ISO 3205, thereby eliminating the influence of other potential sources of vibrations. The experimental plan is shown in Table 1 to determine the impact of design and operational parameters on the vibration amplitudes of the bearings.

To ensure the repeatability of the results, each measurement was conducted three times. The testing began with the bearing labeled 11 and continued up to the bearing labeled 66, respectively. First, the bearing is placed on the vibration measurement device, and a minimum axial load of 200 N is applied to the outer ring of the bearing, after which the spindle rotation is activated. The device is allowed to reach a constant number of revolutions before measurements begin. After the first measurement is completed, the second and third measurements are performed without turning off the spindle rotation. The minimum axial load was adopted according to the SRPS ISO 15242-2 standard. The magnitude of the axial load is increased to a maximum of 1000 N, in steps of 100 N. Results were recorded for 810 measurements of bearing vibrations in files with the .mat extension, and the management of the measurement device and data processing was carried out using the Matlab software package.

Table 1

Experiment plan

Sample/Bearing designation	Number of measurement repetitions	Axial load [N]								
11, 12, 13, 14, 15, 16 31, 32, 33, 34, 35, 36 41, 42, 43, 44, 45, 46 51, 52, 53, 54, 55, 56 61, 62, 63, 64, 65, 66	3	200	300	400	500	600	700	800	900	1000
30 samples	3 measurements for each sample and load	9 different loads								

2.2 Measurement Equipment

Figure 2 shows the experimental device for measuring vibrations in rolling bearings. The bearing is "mounted via its inner ring onto a measuring spindle, which is connected to the shaft using a cone and threaded connection. The shaft is supported by hydrodynamic bearings and rotates at a constant speed of 1800 RPM during measurements. The outer ring remains stationary and is loaded with an axial force applied through a pneumatic cylinder.

The key component in the vibration measurement chain is an electrodynamic velocity sensor (Figure 3), which generates an output voltage whose amplitude and frequency are proportional to the vibration velocity of the observed rolling bearing. Since the signal amplitude from the electrodynamic probe is low, an amplifier is used to enhance the signal for digital processing and display.

The block diagram of the measurement and control system for bearing vibration testing is shown in Figure 4. The analog signal processing components include an amplifier and a band-pass filter. The amplifier increases the signal level from the electrodynamic velocity sensor to a level suitable for digital processing and display. An amplifier with a gain of 1500 was used, ensuring sufficient signal amplitude for digital processing".

The frequency range of interest for rolling bearing vibration testing is from 20 Hz to 10 kHz. The filter is designed to limit the signal spectrum from the amplifier to this range. The designed filter introduces relatively low attenuation of waveform oscillations within the passband and significant attenuation for oscillations outside the range (below 20 Hz and above 10 kHz), in accordance with the SRPS ISO 15242-1 standard (Rolling bearings—Methods for measuring vibration—Part 1: Fundamentals). The passband of the filter is also defined based on this standard.

Signal digitization is performed using the NI DAQ USB-6009 measurement acquisition system. The sampling frequency is 48 kHz, and the internal A/D converter has a resolution of 13 bits. This setup ensures high-quality signal acquisition from the electrodynamic velocity sensor and its digitalization for further computational processing.

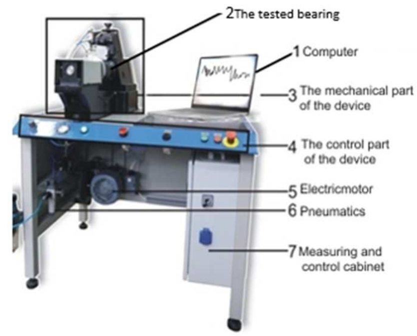

Fig. 2. Experimental device for measuring vibrations

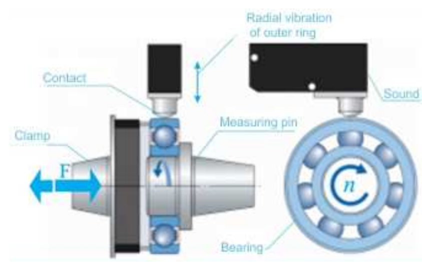

Fig. 3. Schematic representation of the principle of measurement using an electrodynamic speed sensor

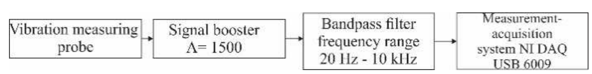

Fig. 4. Block diagram of the measurement and control system

2.3 Signal Processing

The measurement results can be presented in four characteristic frequency ranges:

- LOW
- MEDIUM
- HIGH
- PEAK

The characteristic frequency ranges depend on the spindle speed:

For Low Speeds (700 RPM)

- Low-Frequency Range: 20 – 120 Hz
- Mid-Frequency Range: 120 – 700 Hz
- High-Frequency Range: 700 – 4000 Hz
- Peak: 20 – 4000 Hz

For High Speeds (1800 RPM)

- Low-Frequency Range: 50 – 300 Hz
- Mid-Frequency Range: 300 – 1800 Hz
- High-Frequency Range: 1800 – 10,000 Hz
- Peak: 50 – 10,000 Hz

The measured signal is first processed through a digital signal processing module to extract information within these four frequency ranges. Through digital signal processing, the effective vibration values in the desired frequency bands are obtained, along with the amplitude values at characteristic frequencies of the tested bearing. As previously mentioned, the raw signal from the sensor is initially amplified and filtered. This amplified and filtered signal is then fed into the measurement acquisition card. The first element in digital signal processing is a module for efficient calculation of the Fourier Transform:

$$X_k = \sum_{n=0}^{N-1} x_n e^{-i2\pi k \frac{n}{N}}; k = 0,1,...,N-1 \quad (1)$$

where x_n represents the samples of the input signal, X_k are the samples of the spectral density (which are complex numbers), and N is the number of points at which the Fourier Transform is calculated. To obtain the spectral density of amplitudes, it is necessary to compute the modulus of equation (1), i.e.:

$$|X_k| = \sqrt{\mathrm{Re}\{X_k\}^2 + \mathrm{Im}\{X_k\}^2} \quad (2)$$

To calculate the root mean square (RMS) value of vibrations in the desired frequency bands, it is necessary to convert the spectral density of amplitudes into the spectral density of RMS values, i.e.:

$$|X_k|_{RMS} = \frac{|X_k|}{\sqrt{2}} \quad (3)$$

The boundaries of the desired frequency bands are denoted as f_{lo}, f_{med}, f_{hi} and f_{end} within the three frequency ranges (LOW, MEDIUM, HIGH) over which the RMS value is calculated:

$$\begin{aligned} f_I &\in [f_{lo}, f_{med}] \\ f_{II} &\in [f_{med}, f_{hi}] \\ f_{III} &\in [f_{hi}, f_{end}] \end{aligned} \quad (4)$$

The effective values for these three frequency ranges are calculated as:

$$\begin{aligned} RMS_I &= \sqrt{\left|X_{f_{lo}}\right|^2 + \left|X_{f_{lo+1}}\right|^2 + ... + \left|X_{f_{med}}\right|^2} \\ RMS_{II} &= \sqrt{\left|X_{f_{med}}\right|^2 + \left|X_{f_{med+1}}\right|^2 + ... + \left|X_{f_{hi}}\right|^2} \\ RMS_{III} &= \sqrt{\left|X_{f_{hi}}\right|^2 + \left|X_{f_{hi+1}}\right|^2 + ... + \left|X_{f_{end}}\right|^2} \end{aligned} \quad (5)$$

The values obtained using relation (5) represent the columns of the effective vibration values LOW, MEDIUM and HIGH. The PEAK value represents the maximum amplitude of vibrations in the spectrum and is calculated as:

$$X_{peak} = \max\{|X_k|\}, k = 0,1,...,N-1 \quad (6)$$

During the experimental examination of self-vibrations, the vibration signal is collected in the time domain with appropriate sensors. This kind of signal does not provide much information about the vibration characteristics, so it is necessary to transform it into the frequency domain (Figure 5). The signal is transformed by applying a fast Fourier transform (Fast Fourier transform - FFT), breaking it down into components of different frequencies. Each of the components has its own frequency, amplitude and phase angle. Each device, as well as the machine, i.e. the elements of which it is composed, generate vibrations, during rotation, at characteristic frequencies. To analyze the vibration signal in the frequency domain, it is necessary to know the characteristic frequencies of all elements, so based on the obtained frequencies it is possible to determine which element of the machine or bearing generates vibrations.

The reason for moving from the time domain to the frequency domain, using the fast Fourier transformation, is to clearly observe the frequency of vibrations that correspond to characteristic phenomena in the frequency domain.

3. RESULTS

In the following, the recorded signals are analyzed using FFT in three characteristic areas according to the standard, namely 50-300 Hz, 300-1800 Hz and 1800-10,000 Hz, for the analysis of smaller sized bearings. Figures 6 - 10 left show signal diagrams in the frequency domain, for the value of the radial clearance G_r=0 μm, for an axial load of F_a = 200-1000 N, with a step of 200 N. Figures 6-10 right, show the results of the RMS vibration speed analysis for all areas, for the same bearings as in figures under left. Figures 11 – 15 left show signal diagrams in the frequency domain, for the value of the radial clearance G_r = 30 µm, for an axial load of F_a = 200-1000 N, with a step of 200 N. Figures 11 – 15 right show the results of the RMS vibration speed analysis for all areas, for the same bearings as in figures left. By analyzing the signal in the frequency domain, it can be determined at which frequencies the dominant amplitudes occur, as well as which element of the bearing causes them. The measurement results can be used to determine the quality class of the bearing.

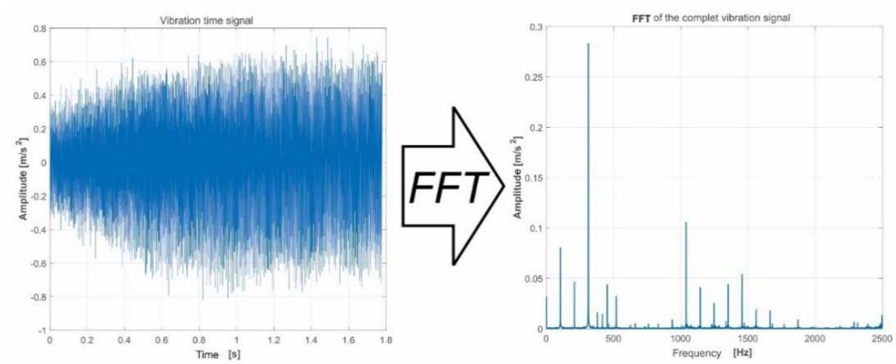

Fig. 5. Transformation of vibration signals from time to frequency domain [8]

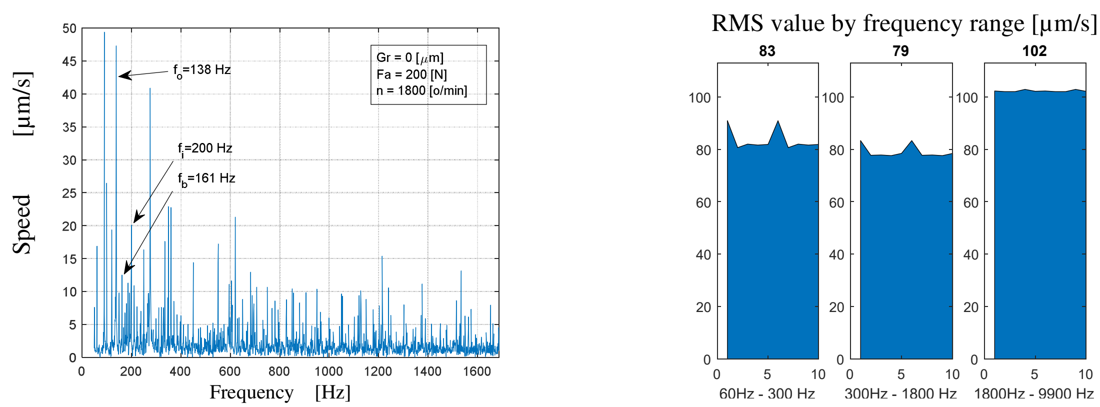

Fig. 6. Frequency domain vibration analysis for G_r=0 µm, F_a=200 N

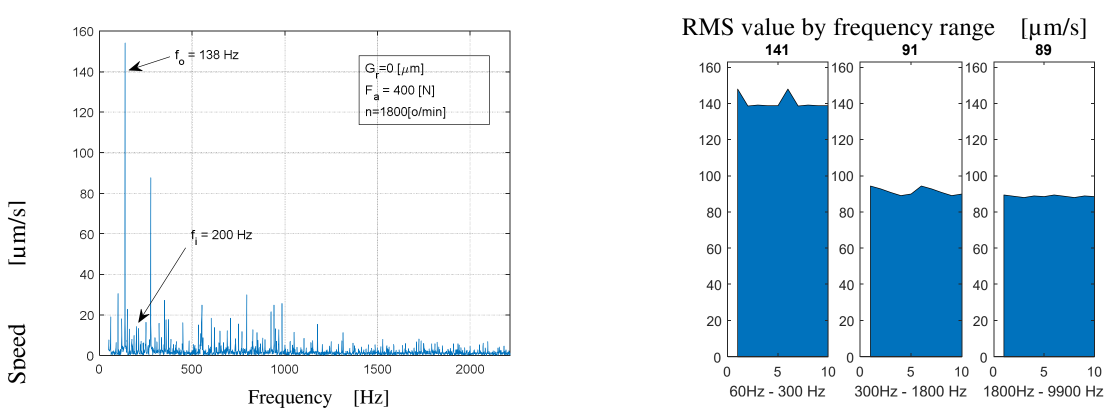

Fig.7. Frequency domain vibration analysis for G_r= 0 µm , F_a=400 N

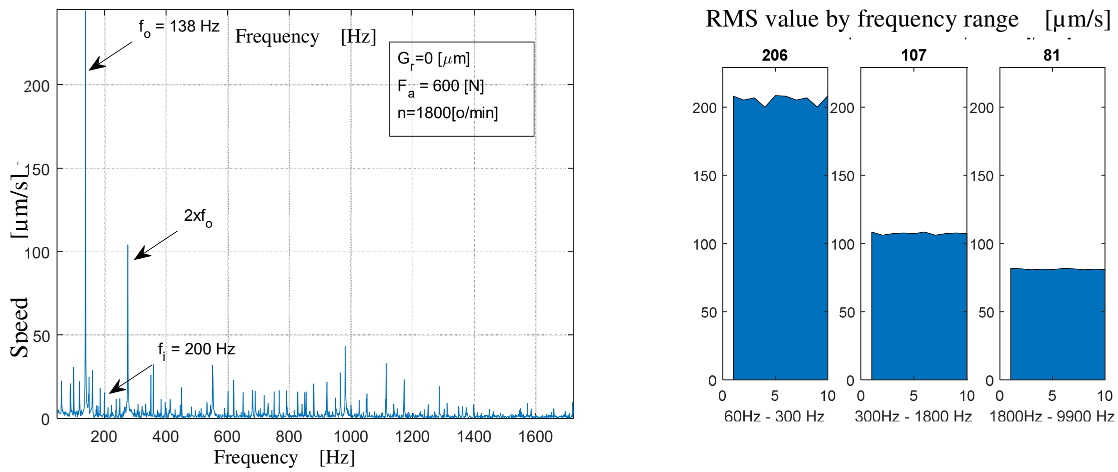

Fig. 8. Frequency domain vibration analysis for G_r=0 µm, F_a=600 N

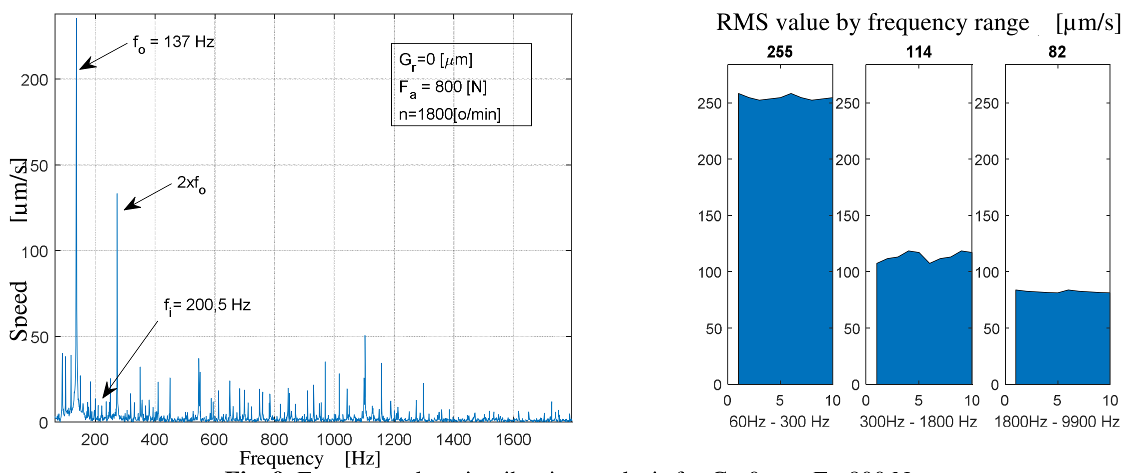

Fig. 9. Frequency domain vibration analysis for G_r=0 µm, F_a=800 N

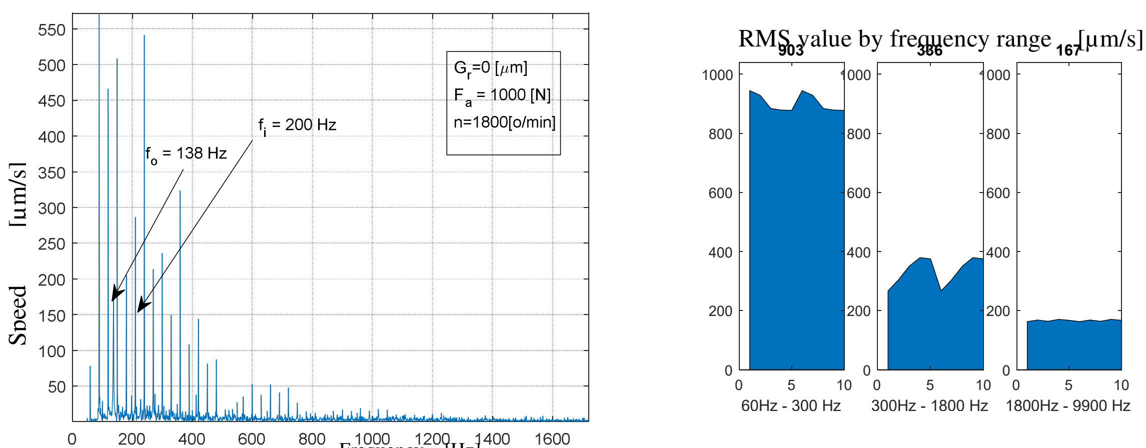

Fig. 10. Frequency domain vibration analysis for G_r=0 µm, F_a=1000 N

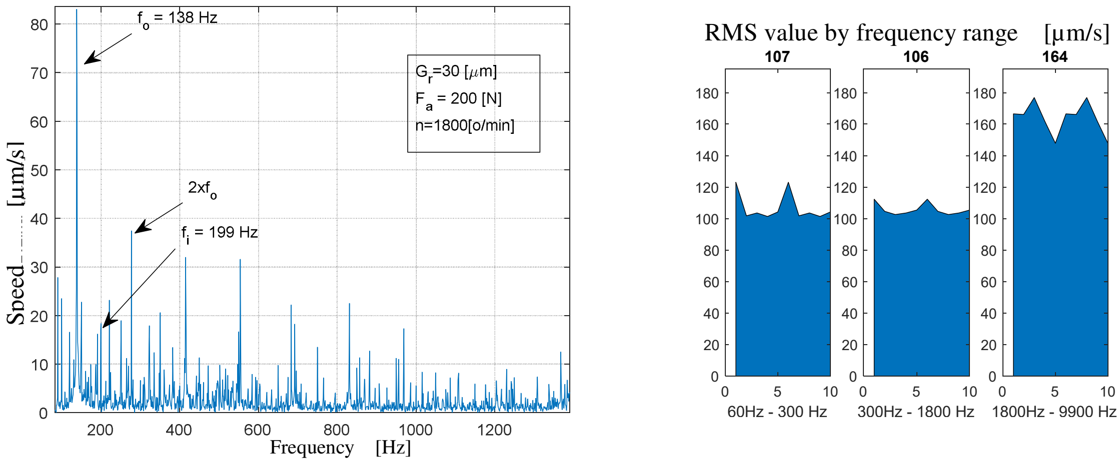

Fig. 11. Frequency domain vibration analysis for G_r=30 μm, F_a=200 N

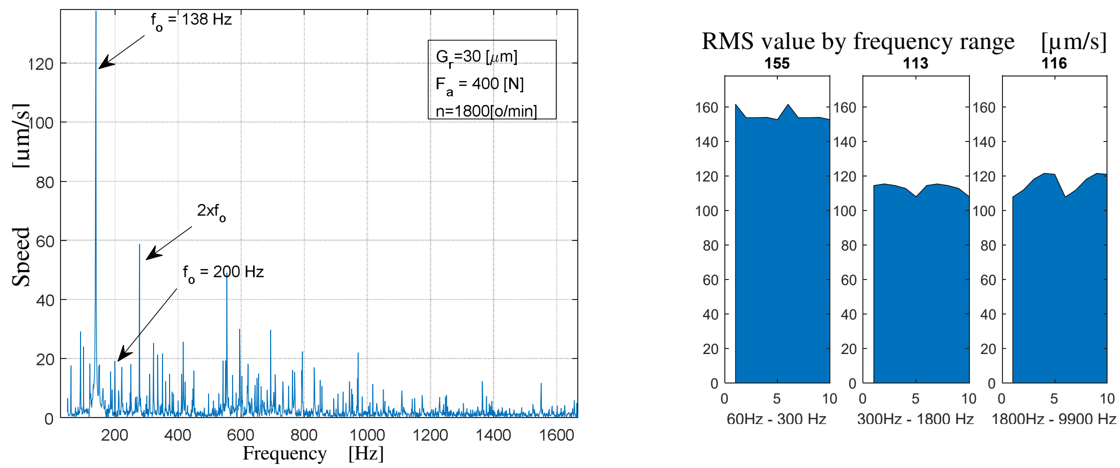

Fig. 12. Frequency domain vibration analysis for G_r=30 μm, F_a=400 N

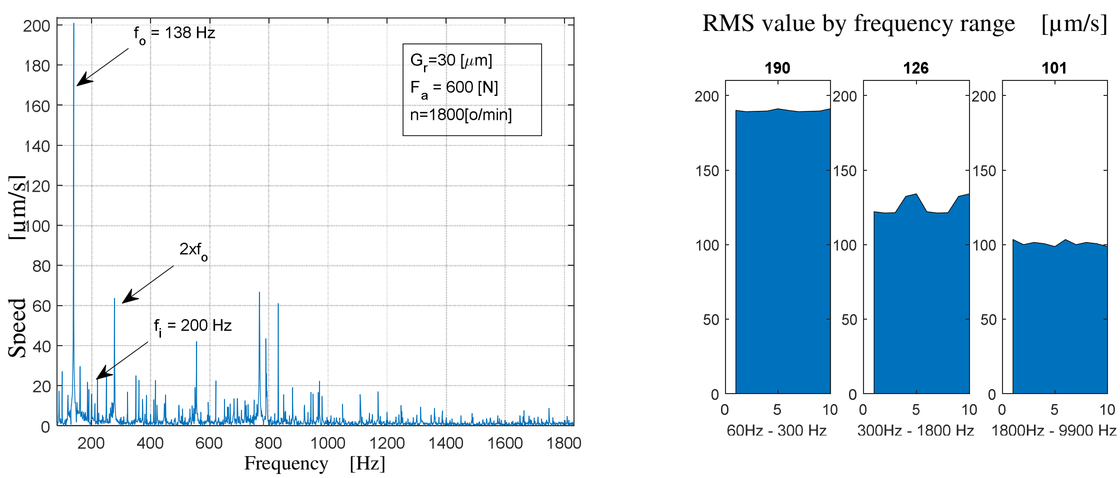

Fig.13. Frequency domain vibration analysis for G_r=30 μm, F_a=600 N

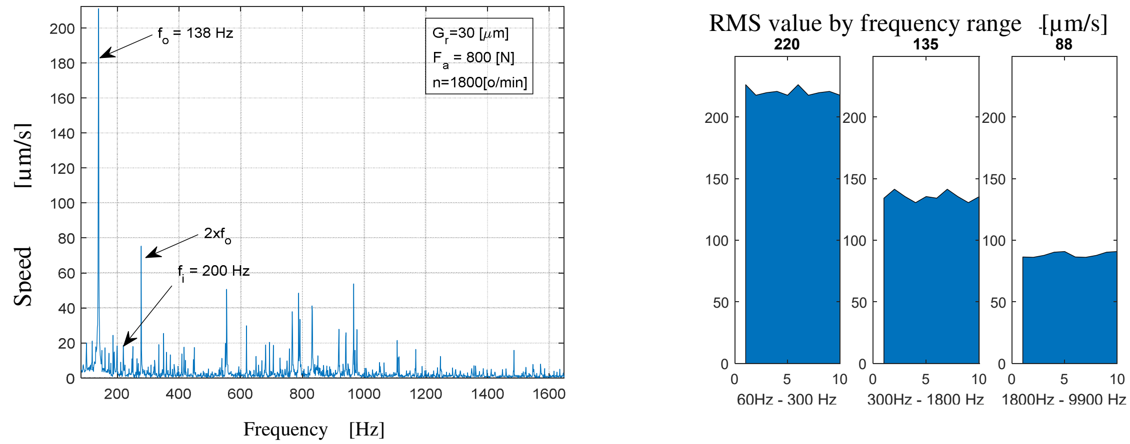

Fig.14. Frequency domain vibration analysis for G_r=30 μm, F_a=800 N

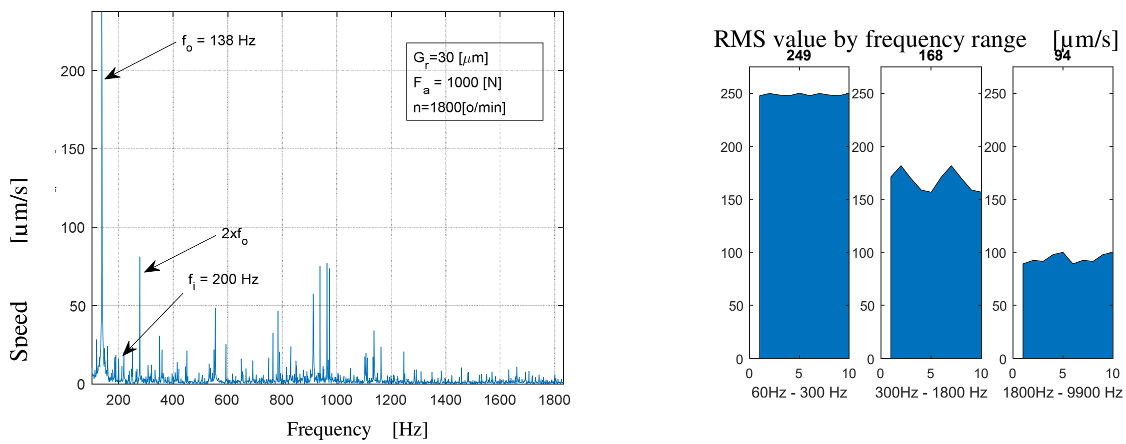

Fig.15. Frequency domain vibration analysis for G_r=30 μm, F_a=1000 N

4. DISCUSSION AND CONCLUSION

Analyzing the obtained RMS values of the amplitude of the vibration speed from figures 6b - 10b and figures 11b - 15b, it can be concluded that there is an increase in the RMS value in the field of low and medium frequencies with an increase in the axial load. In the field of high frequencies, the RMS value decreases with an increase in the axial load up to a load value of 800 N. At a load of 1000 N and in the field of high frequencies, there is an increase in the RMS value, for both bearings, that is, for both values of the radial clearance (G_r = 0 μm and G_r = 0 μm).

The data collected experimentally are tabulated below. Table 2 shows the measurement results for different radial clearance values by RMS area. The obtained data from the mentioned table are then shown graphically in Figure 16, where you can see the dependence of the RMS value of the vibration speed in the characteristic areas on the size of the radial clearance. In the area of medium and high frequencies, the increase of the radial clearance causes a slight increase in the RMS value of the vibration speed, while in the area of low frequencies, there is an intense decrease in the RMS value.

Table 3 shows the measurement results for different axial load values by RMS area. Then those results are shown graphically in Figure 17. From the table and from the diagram, it can be concluded that with the increase of the axial load in the area of low frequencies, it causes an intense increase in the amplitude of the vibration speed. Furthermore, in the area of medium frequencies, there is a slight increase in amplitude. In the area of high frequencies, there is a decrease in the amplitude of the vibration speed, with an increase in the axial load.

Table 2

Dependence of the RMS value of the amplitude of the vibration speed on the radial clearance

Radial clearance		6	7	10,5	12,75	15	17,25	19,5	21,75	24	26,25
RMS by area	I	114	113	108	105	102	99	97	94	92	89
	II	80	80	81	82	82	83	83	84	85	86
	III	118	119	121	122	123	124	124	125	126	127
Radial clearance		28,5	30,75	33	35,25	37,5	39,75	42	44,25	46,5	48,75
RMS by area	I	87	86	83	81	79	77	75	73	71	69
	II	87	88	90	91	93	94	96	98	101	103
	III	127	127	128	129	129	129	130	130	131	131

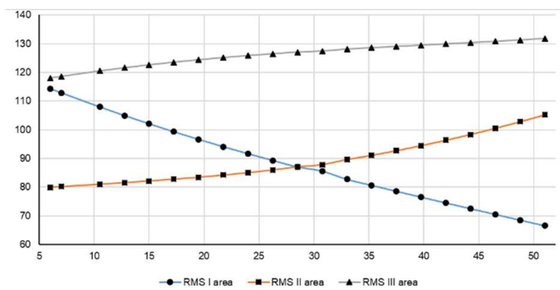

Fig. 16. Dependence of the RMS value of the amplitude of the vibration speed on the radial clearance [9]

Table 3

Dependence of the RMS value of the amplitude of the vibration speed on the axial load

Axial load		200	240	280	320	360	400	440	480	520	560
RMS by area	I	113	121	129	137	146	154	162	171	179	188
	II	80	82	84	87	89	91	94	97	99	102
	III	119	116	114	112	109	107	105	103	102	100
Axial load		600	640	680	720	760	800	840	880	920	1000
RMS by area	I	196	204	212	220	227	234	240	247	253	258
	II	105	108	111	114	117	120	123	126	130	133
	III	98	98	96	95	93	92	91	91	90	98

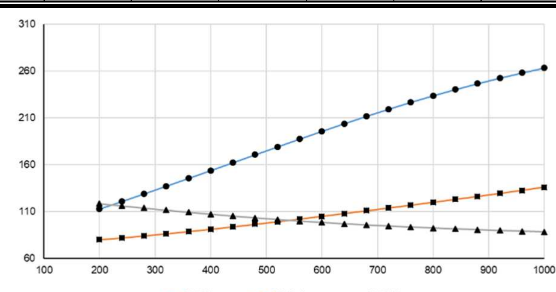

Fig. 17. Dependence of the RMS value of the amplitude of the vibration speed on the axial load [9]

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Determinarea impactului jocului asupra vibrațiilor lagărului radial cu bile prin testări experimentale

Lagărele cu rostogolire sunt elemente de mașini utilizate în suporturile de arbori și osii pentru a transmite sarcini și a asigura precizia necesară între piesele aflate în mișcare relativă. Ele reprezintă una dintre cele mai critice componente ale mașinilor. Deși sunt formate doar din patru componente (inel interior, inel exterior, colivie și elemente de rostogolire), comportamentul static și dinamic al lagărelor este extrem de complex. Cercetările privind comportamentul dinamic al lagărelor sunt motivate de dorința de a reduce zgomotul și vibrațiile în utilizarea acestora, precum și de a crește longevitatea, rigiditatea, viteza de operare și precizia rotațională, precum și de a dezvolta metode pentru diagnosticarea și monitorizarea lagărelor. Comportamentul dinamic al lagărelor este guvernat de comportamentul dinamic al elementelor lor structurale. Vibrațiile care apar în lagărele cu rostogolire în timpul funcționării sunt inevitabile. Jocul este de mare importanță pentru obținerea unei performanțe satisfăcătoare a lagărului. Jocul proiectat pentru a compensa dilatarea termică a elementelor lagărului este o sursă de vibrații și introduce neliniaritate în comportamentul dinamic. Acest studiu a efectuat teste experimentale pentru a observa influența jocului asupra comportamentului dinamic al lagărului radial FKL 6006. Vibrațiile au fost măsurate utilizând un dispozitiv experimental pentru măsurarea și controlul vibrațiilor lagărelor cu rostogolire, analizând impactul sarcinii axiale externe și al jocului asupra comportamentului dinamic al lagărului radial. Măsurătorile au fost efectuate în Laboratorul pentru Mașini-Unelte, Sisteme Tehnologice Flexibile și Automatizarea Proceselor de Proiectare din cadrul Facultății de Științe Tehnice din Novi Sad. În timpul măsurătorilor de vibrații, rezultatele vibrațiilor lagărului au fost înregistrate digital și sunt potrivite pentru analize ulterioare.

Mirjana BOJANIĆ ŠEJAT, PhD, Assistant with a PhD, Faculty of Technical Science, Department of Mechanization and Design Engineering, bojanicm@uns.ac.rs, +38121-485-2362, +38166-001-485.
Nicolae Florin COFARU, Dr. Eng, Professor, Lucian Blaga University of Sibiu, Industrial Engineering and Management, nicolae.cofaru@ulbsibiu.ro, +40722463027
Milan RACKOV, PhD, Full Professor, Faculty of Technical Science, Department of Mechanization and Design Engineering, racmil@uns.ac.rs, +381-21485-2358, +38164-1532-267.
Aleksandar ŽIVKOVIĆ, PhD, Full Professor, Faculty of Technical Science, Department of Production Engineering, acoz@uns.ac.rs, +381-21485-2352, +38164-5978-329.
Ivan KNEŽEVIĆ, PhD, Assistant Professor, Faculty of Technical Science, Department of Mechanization and Design Engineering, ivanknezevic@uns.ac.rs, +381-21485-2381, +38163-7283-889.
Ileana Ioana COFARU, PhD, Full Professor, University Lucian Blaga, ioana.cofaru@ulbsibiu.ro.
Dejan LUKIĆ, PhD, Full Professor, Faculty of Technical Science, Department of Production Engineering, lukicd@uns.ac.rs, +381-21485-2331, +38169-5048-311.