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# New deep groove ball bearings high frequencies vibration testing

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## 1. Introduction

The rolling bearings generate mechanical vibration in wide frequencies range: from low mechanical kinematic vibration frequencies (up to 1000 Hz) caused by bearing elements macro-defects up to high mechanical vibration frequencies (up to 10 000 Hz and greater) generated by micro-defects existing in each element. The study of high frequency vibration response due to micro-defects is the task for quality assessment of new bearings.

In general, bearing defects may be categorized as kinematic (macro) and micro-defects. Kinematic defects in the bearings dominate during long term of rotor operation. The rolling element bearings are spallings of the races or the rolling elements. This is caused when a fatigue cracks appear below the surface of the metal and propagates towards the surface until a piece of metal breaks away leaving a small pit or spall. This is fatigue failure and may be expedited by overloading or shock loading of the bearings during long time operation or improper installation. Micro-defects in new bearings are caused by manufacturing technology. Micro-defects exist in each element of new bearing and include surface roughness, waviness, misaligned races, and off-size rolling elements [1]. The analytical model is proposed to study the nonlinear dynamic behavior of rolling element bearing 6205 2RSL JEM SKF, including surface micro-defects. The stability of rotor bearing system with ball bearings is studied and emphasized that there are a lot of parameters which can act as a source of nonlinearity in such systems as: radial internal clearance, local surface micro-defects, slipping and skidding effects between the rolling elements. The basic routs to chaos in rolling bearing systems are discussed detail [2]. The vibration response has been studied mostly in the time domain and vibration frequency domain formats. The simulation of kinematic, dynamic and structural behavior in low frequency range of the rolling bearings vibration response without faults presented and analytical model is designed using Lagrange formulation and simulation results characterize the error signal in the inner raceway [3]. The novel approach to diagnose bearing faults using Artificial Neural Networks with pattern classification theory is presented in [4]. Experimentally tested and theoretically simulated diagnostic method of low speed rotors with antifriction bearings raceways micro-defects and changes in dynamic stiffness are presented in [5]. The quantification diagnosis for rolling element bearings with fully restrained background noise and extraction of the entire impulsive attenuation signal for assessing the operating condition and faulty severity of rolling bearings during all of the life cycle are presented in [6]. Bearings were tested at rotational speed of 3000 r/min and loading with permanent magnets [7]. The permanent magnet usage involves damping effect on high frequencies vibration acceleration values and changes the dynamic properties of bearing.

The primary aim of this research was to design a quantitative diagnostic experimental method for quality assessment of new with micro-defects in the races and balls deep groove ball bearings. The vibration generated by direct metallic contact of rolling elements - raceways - cages is caused by micro-defects and results in high frequency vibration acceleration of tested grease lubricated and dry bearings.

## 2. Research method and theory

To evaluate the quality of the new deep groove ball bearings elements and manufactured technology by high frequency mechanical vibration acceleration measurements, have the following acceptances:

1. New bearings do not excite low frequency kinematic vibrations as fatigue failures (e.g. spots) in the new bearing element materials do not exist.

2. The new bearings were tested with experimental kit horizontal rotor rotating in journal bearings at 1800 r/min.

3. The micro-defects of new bearings (surface roughness $R_z$, $R_{zmax}$, waviness $W_s$ and of size rolling elements) excite high frequency vibration accelerations up to 10 000 Hz and higher.

4. The mechanical vibration of bearings were experimentally tested without and with grease lubricant, with and without stationary radial preloads attached to the outer rings.

5. Bearings' absolute vibration accelerations were measured with two seismic transducers attached with 180° phase difference. The accelerometers were rigidly attached to the non-rotating outer bearing ring with screws.

6. Tested bearings contacting elements with the surfaces roughness $R_z$, $R_{zmax}$, waviness $W_s$, misaligned races and off-size rolling elements, roundness' were measured with high quality devices disassembling tested bearings, (Fig. 1). Measurements of inner raceway, outer raceway and balls roundness, waviness and roughness were provided with high accuracy measurement devices. Bearing 6205 C3P6 elements measurement results indicated the existence of micro-defects (surface roughness $R_z$, $R_{zmax}$, $R_a$ waviness $W_s$) in the new bearing elements (Fig. 1, Table).

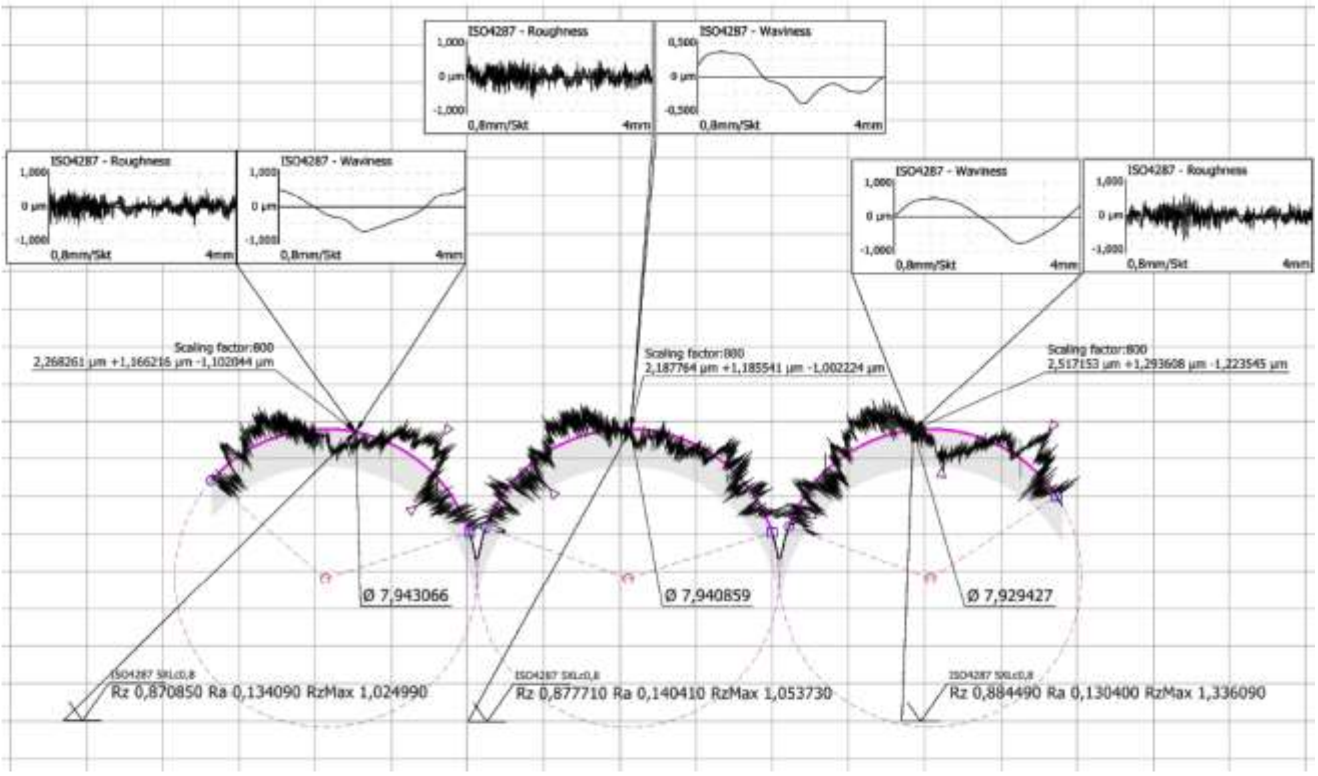



Fig. 1 Bearing balls geometric dimensions: roughness $R_a$, $R_z$, $R_{zMax}$ and waviness $W_s$ values of three balls of the tested bearing; balls' nominal diameter (7.933±0.0004 mm) with deviations: 7.943066 mm (2.266 μm); 7.940859 mm (2.18764 μm); 7.929427 mm (2.517153 μm); micro-defects (surface roughness $R_z$ 0.87085-0.88449 μm, $R_{zmax}$1.02499-1.33609 μm) and waviness $W_s$<±0.5 μm (by courtesy of Craft Bearing Corp.)

Table

The measured geometric dimensions of new tested 6205 C3P6 bearing

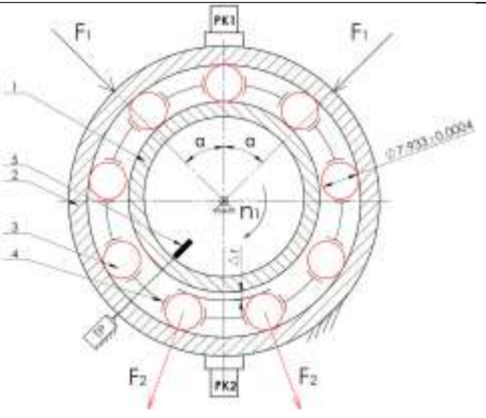
| 6205 C3P6 bearing | Parameters | Ball | Inner ring raceway | Outer ring raceway |
|---|---|---|---|---|
| | Diameter, mm | 7.933 | 30.567 | 46.433 |
| 1. | Raceway roundness and deviations, μm | 2.188<br>+1.186<br>-1.002 | 3.367<br>+1.548<br>-1.819 | 29.447<br>+14.450<br>-14.996 |
| 2. | Sphere diameter, mm | 7.941 | 8.164 | 8.459 |
| 3. | Raceway roughness, μm | | | |
| 4. | $R_z$ - mean roughness depth | 0.878 | 0.814 | 0.754 |
| | $R_{zMax}$ – maximum roughness depth | 1.054 | 1.109 | 0.938 |
| | $R_a$- average roughness | 0.1404 | 0.131 | 0.124 |

Two external preload radial forces $F_1 = 100$ N values were each loaded at the stationary outer ring. $F_1$ added symmetrically reference to the first accelerometer PK1 location in order to measure bearing's outer-inner raceways contacts with balls excited vibration (Fig. 2). These forces guarantee permanent dynamic contact between rolling elements at PK1 transducer location. Radial external forces $F_1$ values were limited by minimum frictional moment ($M = \sim 0.5\ \mu\ F_1\ d = 3.75$ N mm) in the bearing elements ($\mu = 0.0015$ is the coefficient of friction for the deep groove ball bearing, $d$ is bearing ball diameter). The second accelerometer PK2 measures bearing outer ring raceway contact with balls excited vibration. The balls are in permanent contact with outer raceway at the PK2 attached location. The existing nominal radial internal clearance $\Delta r$ between inner raceway and balls are guaranteed by this loading scheme. Such vibration measurement method with two accelerometers and external loading allow separating inner raceway micro-defects excited vibration from outer raceway micro-defects excited vibration. The vibration measurement excited by the inner raceway and balls contacts in practice is complicated. Each bearing was tested when two accelerometers were attached at outer ring with screws in two conditions: lubricated and dry bearing.

When bearing's inner ring rotates at $n_1 = 1800$ r/min rotational speed around rotor axis, and outer ring is fixed; each ball rotates together with cage about rotor axis at $n_{cage} = 714$ r/min rotational speed. Each ball mass $m_r$ generates eccentric inertia force $F_2 = 0.383$ N which guaranties permanent contact of balls with outer ring raceway.

$$F_2 = \frac{1}{2} m_r\ d_m\ \omega_{cage}^2 = \frac{1}{2} 0.00355 \cdot 0.0385 \cdot 74.84^2 = 0.383\,\text{N},$$

where $m_r = 0.00355$ kg is the mass of the ball *4* (Fig. 2); $\omega_{cage} = 74.84$ 1/s, cage with balls angular velocity about shaft axis, cage radius of rotation ½ $d_m = 1/2$ 0.0385 = = 0.01925 m, $d_m = (d + D)/2 = 0.0385$ m is the mean diameter of bearing, $d = 0.025$ m is nominal bore diameter, $D = 0.052$ m is nominal outside diameter, $d_r = 7.933$ mm is ball diameter, $z = 9$ is the number of balls.




PK1, PK2 transducers and mark for TP locations with preload forces $F_1$ and balls eccentric forces $F_2$

Experimental kit photo with tested bearing and accelerometers PK1, PK2

Fig. 2 Bearing 6205 C3P6 vibration acceleration measurement scheme and rotor kit photo: two accelerometers PK1, PK2; preloads $F_1$ forces with angle $\alpha$; $F_2$ are the balls eccentric inertia forces, inner ring *1*, outer ring *2*, balls *3*, cage *4*, non-contact tachometer probe with TP mark on the shaf *5*; $\Delta r$=13-18 μm is the radial internal clearance, $n_1$= =1800 r/min is rotational speed of the shaft with bearing's inner ring

The balls are in permanent contacts with inner and outer raceways at the same time at the accelerometer PK1 attached location. These dynamic contacts between bearing elements generate high frequency vibration acceleration proportional to micro-defects values existing in both raceways and balls. Preloading forces $F_1$ pushed outer ring through balls to inner ring eliminating radial internal clearance $\Delta r$ at the transducer PK1 location. At the same time, the radial internal clearance $\Delta r$ between balls and inner raceway exists during rotation at the PK2 transducer location, because ball mass eccentric $F_2$ force pushes each ball towards the outer raceway (Fig. 2). The balls at the accelerometer PK2 attached location are in permanent contacts only with the outer raceway, but not with the inner raceway. The high frequency vibration generated by these contacting elements with micro-defects describes manufacturing technological quality of the outer ring raceway and balls. The radial internal clearance of the tested 6205 C3P6 bearings according ISO 5753:1991 are of 13 - 28 μm value.

It was possible to evaluate micro-defects separately of both inner and outer raceways and balls through vibration measurements using both output vibration signals from PK1 and PK2 transducers. Algebraically subtracting vibration responses signals acquiring from the PK1 and PK2 accelerometers and synchronizing signals pickups by tachometer probe TP from rotation speed measurement device it is possible to evaluate technological quality of outer and inner raceways separately.

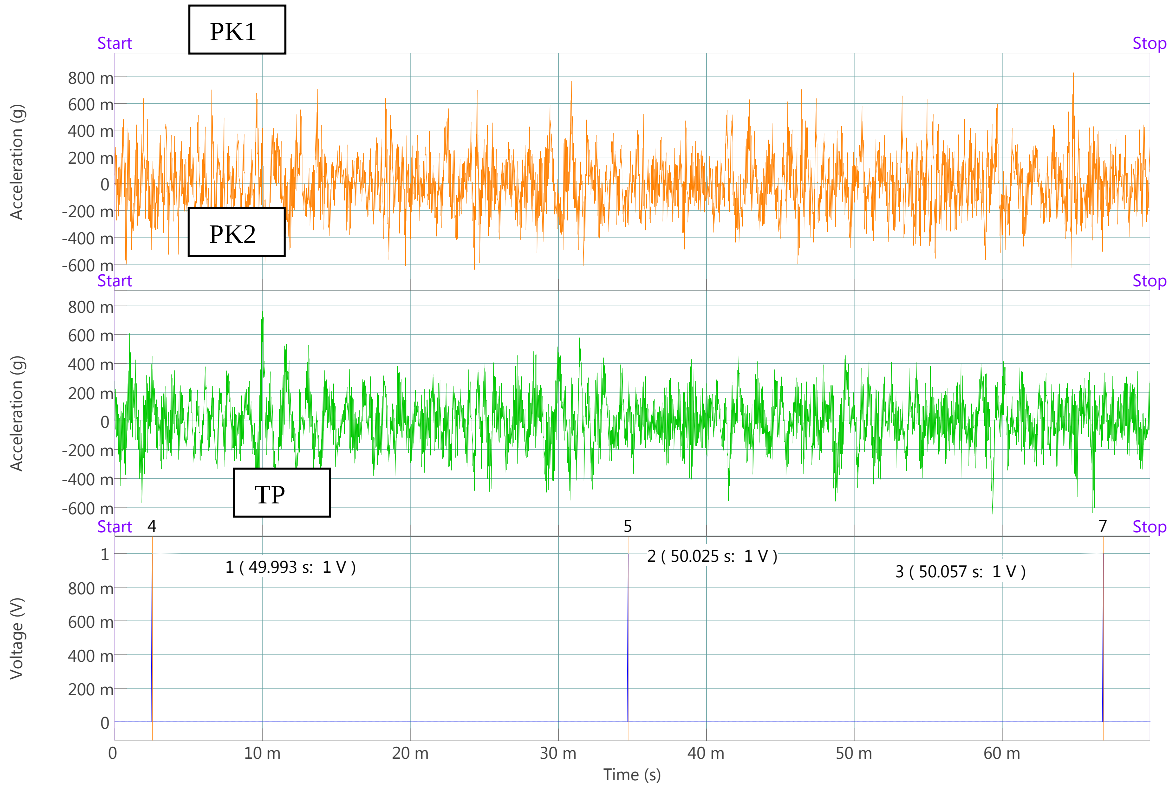
## 3. Experimental testing of 6205 C3/P6 bearings vibration

Vibration measurements of deep groove ball bearings 6205 C3P6 were pursued with accelerometer of 787A type (WR USA, dynamic sensitivity of 100 mV/g, resonance frequency of 22 000 Hz, frequency response of ±3 dB at frequency interval of 0.7-10 000 Hz), vibration signal analyzer OROS Mobi-Pack OR-36 (FR), software OROS NVGate V7.00 and OROS ORBIGate V3.00 (FR), oscilloscope ScopiXII OX7104 (Metrix, FR). The new bearing was tested during each experimental run it was lubricated with standard grease and without grease (dry run). The high frequency vibration acceleration data formats as time plots and spectra were used together to analyze vibration response. Handing weighting window method was chosen for vibration evaluation and FFT transformation of original acceleration signal.

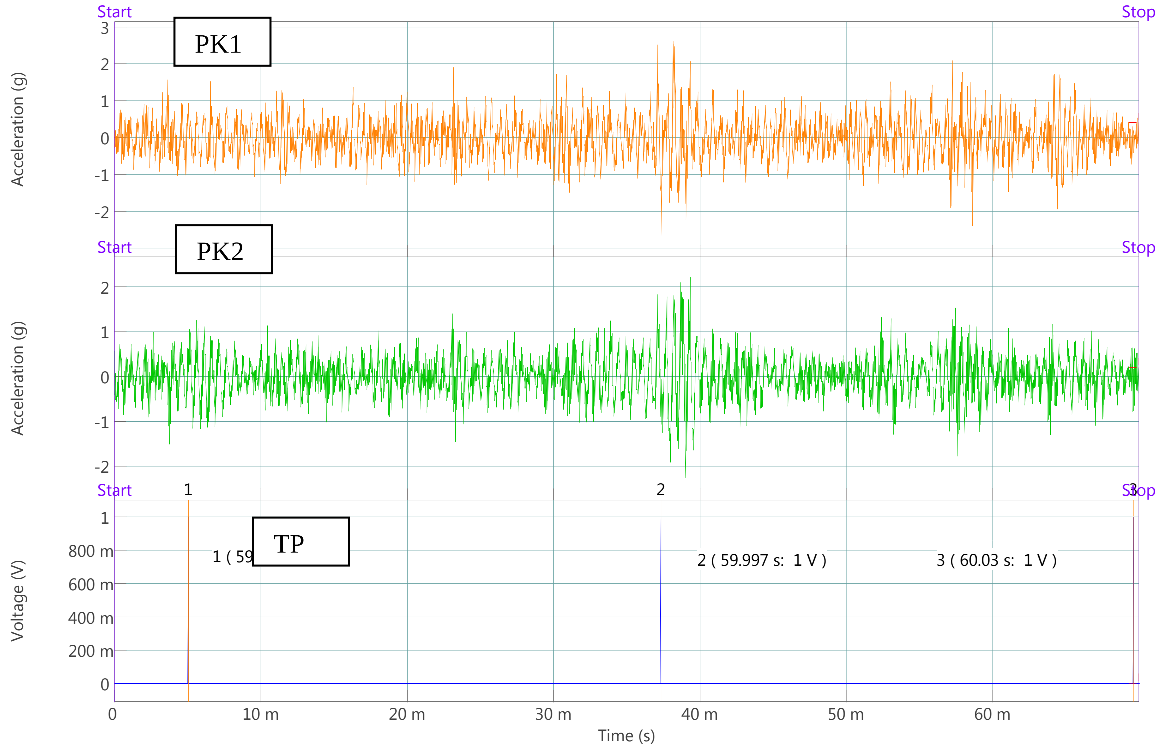
Bearing vibration was measured when acting with preloading forces $F_1$ and vibration acceleration-time base plots shown in (Fig. 3). TP signals indicated each full rotation time (0.0333 s) of bearing inner ring. The maximum acceleration magnitude value of the bearing with grease lubricant reached $a_{p\text{-}pmax}$ = ~1.2 m/s$^2$ and was ~2.5 times less in comparison with dry bearing vibration acceleration $a_{p\text{-}pmax}$ = ~3 m/s$^2$ (Fig. 3, a, b).

With grease and preloads $F_1 = 100$ N bearing vibration acceleration magnitudes are 2-2.5 times greater at accelerometer PK1 measurement location in comparison with PK2 data, as ball shocks at outer and inner raceways were more intensive in comparison with outer raceway at PK2 accelerometer location.

The vibration acceleration spectra acquired from acceleration-time signals (Fig. 3, a, b) are shown in Fig. 4, a, b. Outer and inner ring raceways and balls stochastic vibration were damped by the grease. Vibration acceleration spectra (Fig. 4, a) indicated low frequency 700 Hz vibration acceleration amplitudes in the spectra of both PK1 and PK2 transducers output signals. This frequency was generated by $z = 9$ balls rotating about their own axes, but not of the micro-defect. The vibration acceleration spectra of dry bearing indicated dominating high frequency of 2 750 Hz and 1825 Hz (Fig. 4, b). The high frequency of 2750 Hz vibration acceleration at 1.4 m/s$^2$ amplitude was measured with PK1 accelerometer indicated the first resonance frequency of the outer ring. The vibration acceleration frequency of 1825 Hz indicated outer raceway and balls dynamic shocks excited vibration when bearing was tested without grease (Fig. 4, b).



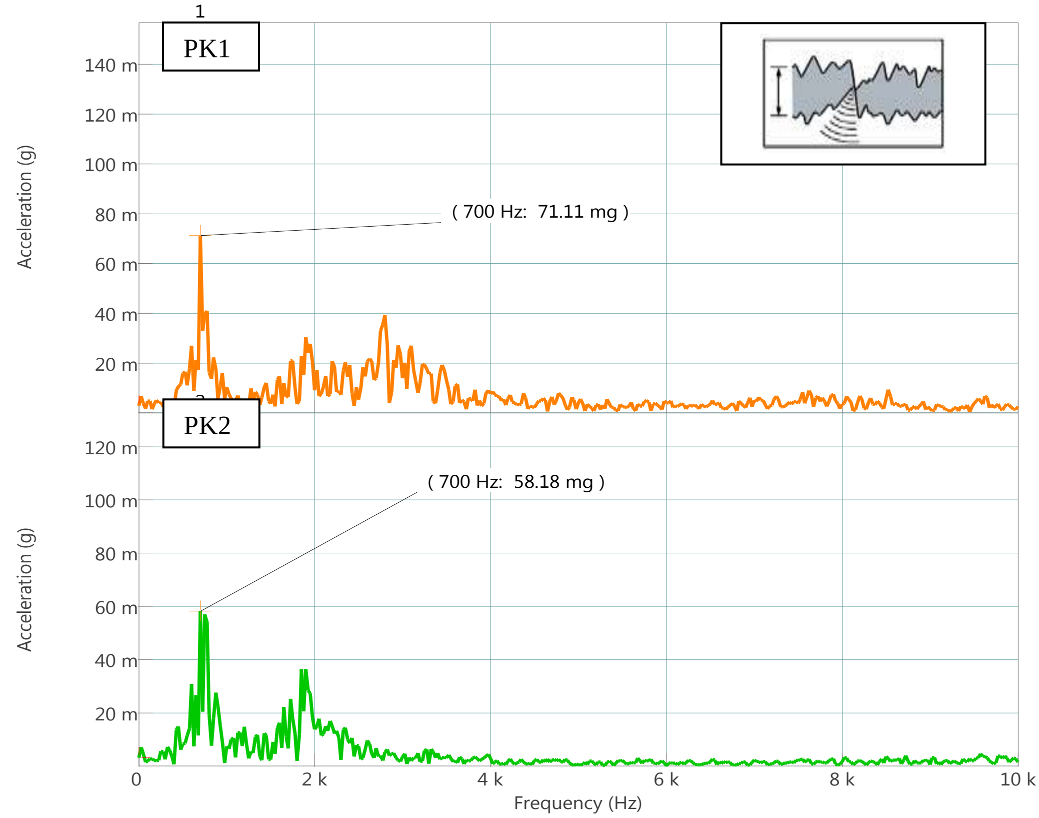

a




b

Fig. 3 Vibration acceleration time base plots during two full rotations of inner raceway with grease lubricated (a) and without grease lubricant (b) bearing with preloads $F_1$=100 N: PK1 transducer output upper and PK2 transducer output lower plots, TP is tachometer probe output

With grease but without preloads $F_1 = 0$ N vibration acceleration magnitudes are ~3 times less in comparison with loaded bearings and shown in Fig. 5, a and dominated low frequencies 325 Hz. Vibration intensity is of the same values measured with accelerometers PK1 and PK2. The vibration acceleration spectra of dry bearing indicated the dominant harmonics of high frequencies in the range from 1700 up to 2800 Hz including outer ring resonance frequency (Fig. 5, b). These high frequency vibration accelerations are generated by shocks of outer and inner raceways and balls micro-defects, were characterized by maximum roughness depth $R_{zMax}$. The low frequency magnitudes were negligible. The vibration acceleration frequency 1700-2750 Hz indicated outer raceway and balls dynamic shock frequencies, resonance when bearings were tested without grease (Fig. 5, b).

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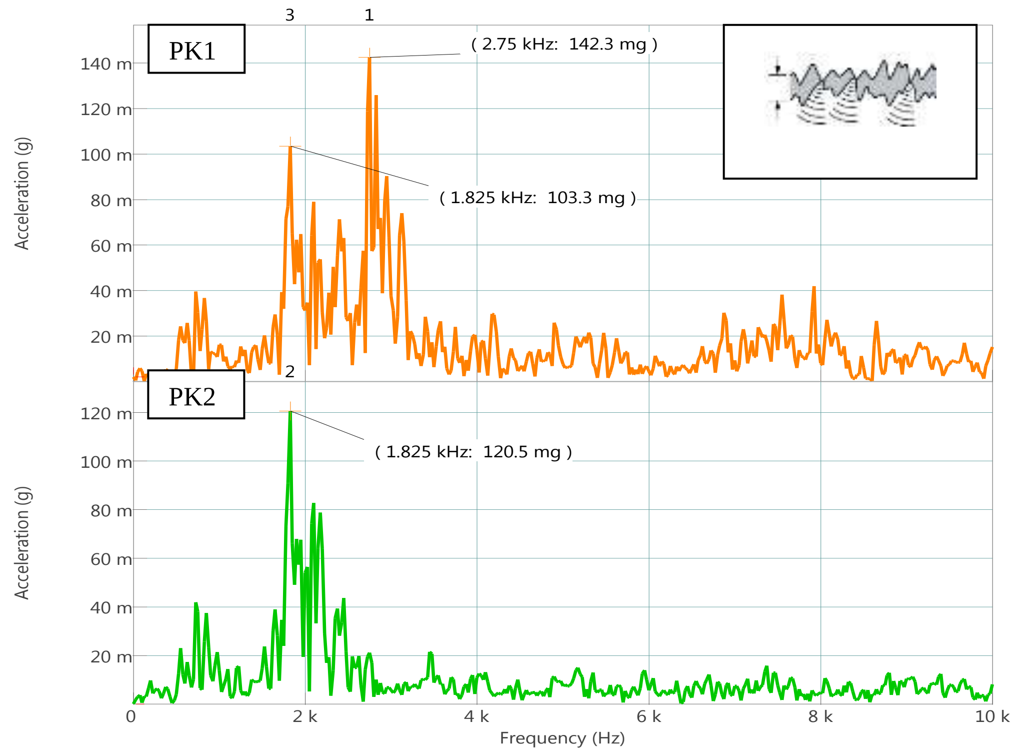
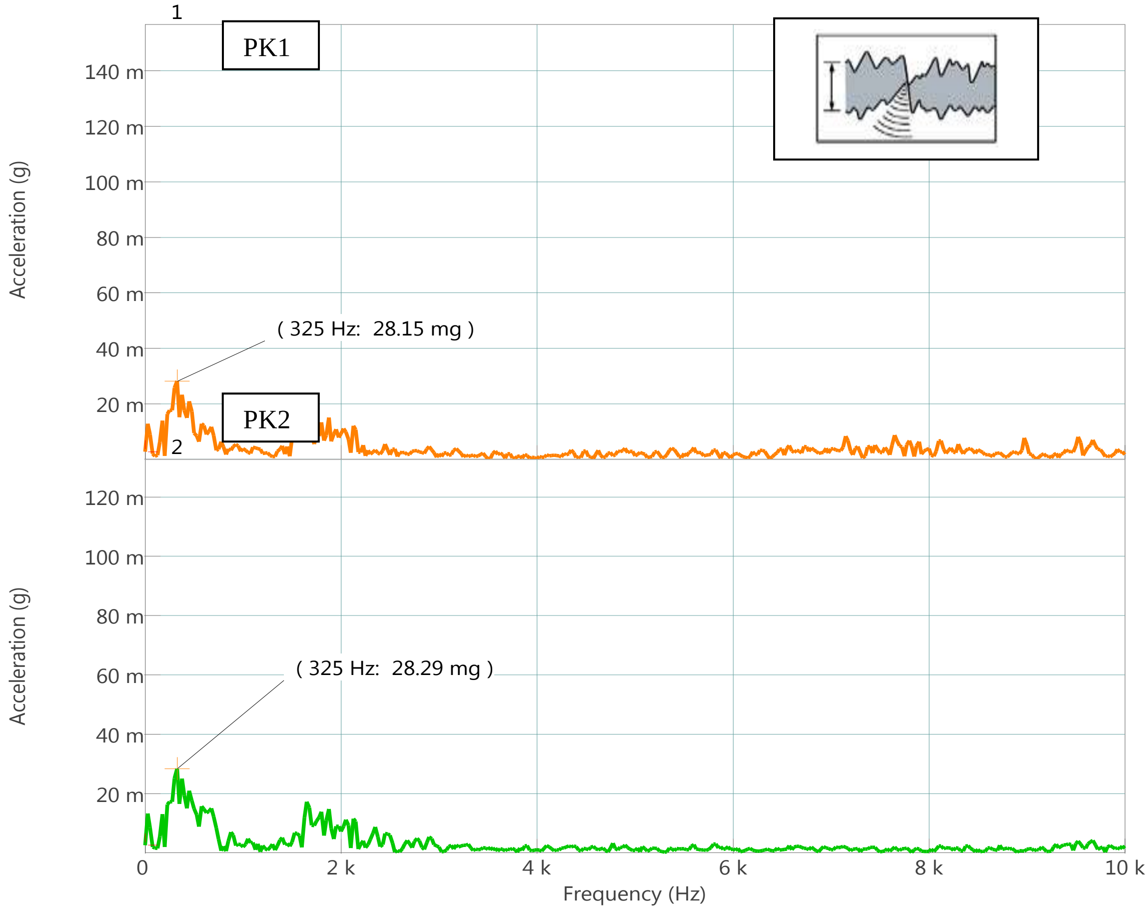



b

Fig. 4 Vibration acceleration spectra with grease lubricant (a) and without grease lubricant (b), with preloads $F_1$=100 N: PK1 transducer output upper and PK2 transducer output lower plots

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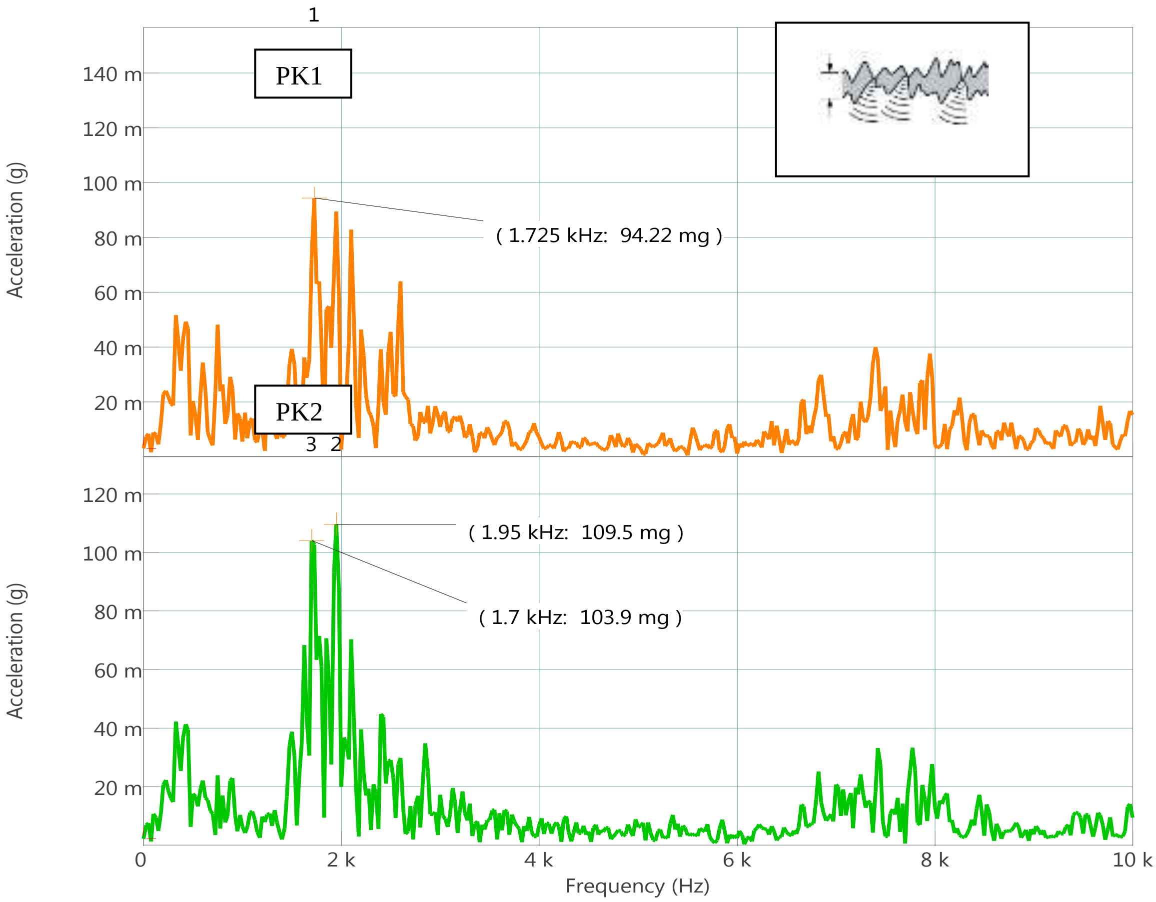


b

Fig.5 Vibration acceleration spectra with grease lubricant (a) and without grease lubricant (b) bearing without preloads $F_1$=0 N: PK1 transducer output upper and PK2 transducer output lower plots

## 4. Conclusions

1. Lubricated and dry new bearings high frequency vibration is stochastic and correlated with $R_{zMax}$ – maximum roughness depth of bearings elements surfaces. Lubricated bearings high frequencies vibration intensity is ~2-2.5 times smaller in comparison with dry bearings. This issue is valid for transducers PK1 and PK2 measurement data.

2. The high frequency (1500-2800 Hz) vibration acceleration damped by grease lubricant when lubricated bearings loaded with external forces $F_1 = 100$ N and without external forces $F_1 = 0$ N. When dry (unlubricated) bearings loaded with external forces $F_1 = 100$ N and without external forces $F_1 = 0$ N the high frequency vibration acceleration amplitudes dominated in the spectra.

3. The resonance frequency 2750 Hz of outer ring dominated in the vibration acceleration spectra when measured with PK1 and tested bearing is loaded with external forces $F_1 = 100$ N but without grease lubricant.

4. It is recommended to test dry bearings (without lubricant) loaded with external forces for evaluation of quality of new bearings elements using high frequencies vibration. Such method separates the high frequency vibration generated by the outer raceway from inner raceway micro-defects generated vibration.

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NAUJŲ RADIALIŲJŲ RUTULINIŲ VIENAEILIŲ GUOLIŲ AUKŠTOJO DAŽNIO VIRPESIŲ TYRIMAS

R e z i u m ė

Antifrikciniai guoliai yra neatskiriama daugiamasė rotorinė sistema susidedanti iš riedėjimo elementų, vidinio, išorinio žiedų ir separatoriaus, perduodanti judesį ir apkrovas nuo rotoriaus iki mašinos korpuso. Naujų guolių, be kinematinių defektų, elementų paviršiai, apibūdinami šiurkštumu $R_z$, $R_{zmax}$, banguotumu $W_s$ ir formos nuokrypiais, žadina aukšto dažnio stachostinius virpesius. Darbas skirtas naujo vienaeilio rutulinio radialiojo riedėjimo guolio 6205 2RSL C3/P6 kokybės eksperimentiniams tyrimams, atskiriant išorinio žiedo riedėjimo takelio-rutuliukų ir vidinio žiedo riedėjimo takelio-rutuliukų mikrodefektų žadinamus aukšto dažnio virpesius. Metodas pagrįstas suteptų ir sausų guolių virpesių tyrimu, keičiant apkrovas.

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NEW DEEP GROOVE BALL BEARINGS HIGH FREQUENCIES VIBRATION TESTING

S u m m a r y

The rolling bearing is a multi-body rotating system with rolling elements, inner, outer rings and cage transmitting motion and load from rotor to the bearing supports and machine body. The existence in the new bearing elements of micro-defects (surface roughness $R_z$, $R_{zmax}$, waviness $W_s$ and of size rolling elements) generate stochastic high frequency vibration. It is important for manufacturing technology to examine quality of new bearings elements. This paper considers a new deep groove ball bearings single row 6205 2RSL C3/P6 experimental vibration testing method for quality assessment of new bearing separating micro-defects in inner raceway from micro-defects in outer races. This method is based on high frequency vibration measurements of lubricated and dry bearings with varied loads.

**Keywords:** deep groove ball bearing, micro-defects, high frequency mechanical vibration, quality assessment, dry and grease lubrication, radial internal clearance.

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