

CONDITION MONITORING OF ROLLER BEARINGS USING DIFFERENT DIAGNOSTIC METHODS

Vytautas BARZDAITIS*, Vytautas ŽEMAITIS*, Kestutis ŽEBELYS*, Zigmantas POCIUS*, Pranas MAŽEIKA**

*Kaunas University of Technology, Mickėvičiaus st.37, Kaunas, LT-44244, Lithuania,

**Klaipėda University, Bijūnų st.17, Klaipėda, LT-5802, Lithuania

E-mail:vytautas.barzdaitis@ktu.lt

Summary

This paper deals with condition monitoring and failure diagnostics of spherical roller bearing on adapter sleeve running in high technology Chemical industry machines. Two methods have been used: low frequency mechanical vibration (up to 1000 Hz) and high frequency mechanical vibration (up to 16000 Hz) monitoring method and a shock pulse method (SPM, frequency 32000 Hz). Machines have been tested in at free run and fully loaded. Results have been applied to increase reliability of diagnostic methods in a case when radial clearances inside a bearing are too small and there is no enough lubricant oil supply to bearings. Radial clearances and relative vibration displacements of rotor shafts have been measured using proximity probes. The low frequency vibration excitation method is not efficient enough in failure diagnostics of bearings in comparison to high frequency and shock pulse methods. Both these methods together are efficient evaluating technical condition of heavy loaded roller bearings. The experiments have been provided with spherical roller bearings on adapter sleeve SKF 22228 CCK/W33 and SKF 22216 CJC 403.

Key words: technical condition, diagnostics, vibration, roller bearings, shock pulse method.

1. INTRODUCTION

Great importance is attached to the investigation of technical conditions of roller bearings and their failure diagnostics. It is closely related to an increase in modern machines efficiency, safety of work, maintenance cost and production quality. The paper appraises two different diagnostic methods of roller bearing defects, namely mechanical vibration and shock pulses. Mechanical vibration related to the kinematical vibration of bearing elements, i.e., to the low frequency range of vibration not exceeding 1000 Hz, and to the high frequency vibration exceeding the resonance frequencies of the analyzed bearings up to 16000 Hz [3, 4]. To investigate the intensity of kinematical frequency vibration and the bearing failure, the kinematics theory of planetary (friction) gear elements motions has been applied. When analyzing kinematics of bearing elements, the sliding phenomenon in kinematical pairs is to be evaluated. The sliding friction effect on planetary gears kinematics is many times smaller than to roller bearings and is observed by the fluctuation of gear ratio parameter. High frequency mechanical vibration is evaluated by determining the bearing vibration acceleration related to the resonance vibration frequencies of bearing elements and the rollers shocks. For example, in vibration signal analyzers Adash 4101, Pruftechnik System 2 two frequency ranges are provided: the first - from 5000 Hz to 16000 Hz in "g" units; the second frequency range is from 0.8 Hz to 16000 Hz

using vibration acceleration for the evaluation of vibration intensity.

The SPM of bearing diagnostics runs with high frequency – 32000 Hz [2]. It is the frequency of propagating vibration waves excited by bearing element (rollers) shocks, their intensity expressed by nominal units – decibels. The apparatus used for vibration measurements: Shock Pulse tester T2001 [2]; 8 mm proximity probes for relative vibration measurements (BNC TK 83, TK 84, TK 85) [4]; seismic transducers (frequency range - 0.7 ÷ 10000 Hz) and VSA (Adash A4101 PRO, Pruftechnik AG System 2) [5].

The paper deals with the analysis of the technical condition of roller bearings referring to the research experimental results. It presents the experimental investigation results of technical condition of double-row spherical roller bearings on adapter sleeve SKF 22216 CJC 403 and SKF 22228 CCK/W33.

2. STUDY ITEMS CHARACTERISTICS

This paper presents two machines. The first is a cylindrical dryer of large geometrical dimensions and mass – 2.26 m diameter and 15.5 m length used in chemistry industry as shown in Fig. 2.

The entering drying agent temperature is +400,...,+450⁰ C and output temperature is +100,...,+150⁰ C. The shell is made of carbon steel ST 50 IS 2062/DIN 17100. The shell rotates on 4 supporting cylindrical rollers contacting on both

sides (right and left) with two carrier rings. Each cylindrical roller rotates at 60 rpm in two double-row spherical roller bearing on adapter sleeve SKF 22228 CCK/W33+SMS 3128H. The shell rotates by 45 kW power electric motor via a belt driver,

gear box and spur gearing (z_5, z_6). The driving gear is subjected to the rotation torque of 6500 Nm. The overall varying mass of a dryer is 60 000 kg. One supporting cylindrical roller is subjected to force 167100 N.

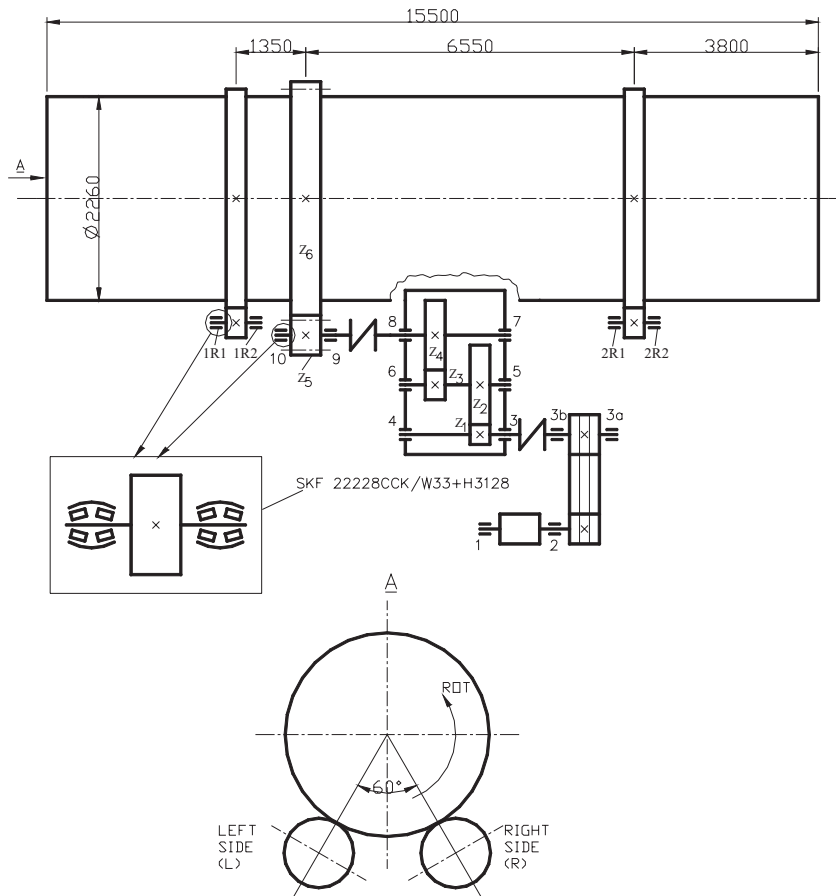


Fig.2.1. The Dryer with cylindrical shell and supplementary mechanisms: 1,2 – electric motor bearings; 3a, 3b – belt drive bearings; 3,4,5,6,7,8 – gearing box shafts; z_1, z_2, z_3, z_4 – teeth numbers of gears; 9, 10 – shell gear drive z_5 bearings; 1R1, 1R2, 2R1, 2R2 – bearings of right dryer side cylinders; ROT – shell rotation direction

The second machine is a grinding mill with two impellers: large 8 and small 16 impellers are rotated by electric motors 5 and 13 through belt drives 17 as shown in Fig. 2.2. The rotation speed of impellers is 1350 rpm with opposite rotation directions. Each impeller 8 and 16 rotates on two double-row spherical roller bearings 3, 4 and 11, 12 (SKF 22216 CJC 403). The impellers designed as hollow rollers whose both side disks are connected by pins.

3. SPECIFIC CHARACTER OF ROLLER BEARING FAILURE DIAGNOSTICS

3.1 Bearings condition monitoring using mechanical vibration frequencies

Mechanical vibrations are analyzed as low and resonance frequency vibrations. Low frequency vibrations are related to the bearing element kine-

matics which is analogous with that of planetary gearing.

Vibration frequency excited by a separator when an inner or outer bearing ring is rotating.

In the analysis of kinematics, the rolling elements are assumed to roll over bearing rings races without sliding friction, though, under heavy dynamic loads or the unloaded bearings, sliding friction is unavoidable in kinematical pairs. For this reason, the measured vibration frequencies excited by the defective bearing elements do not agree with the theoretical calculation results. Mechanical vibration excited by a faulty separator has low frequency – approximately half the synchronous frequency of rotor rotation.

Vibration frequencies excited by inner and outer ring rolling races. In the case of horizontal and vertical rotors, the number of loading cycles of a bearing fastened on a centre shaft, which appears when the balls are shocking inner and outer ring

rolling races at the contacting points, depends on the rotating ring. When the inner bearing ring is rotating and the outer ring is fixed in the support, the shaft transmits the load to the inner ring first

and then by its rollers to the outer ring and the bearing support. In static it is assumed that in a case of a radial loading only one side of the bearing in the rotor is non-uniformly loaded.

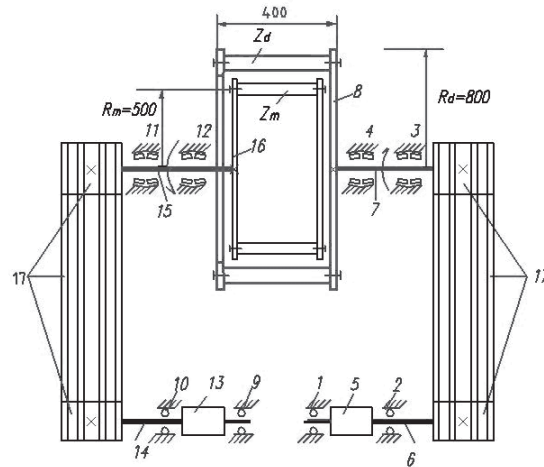


Fig.2.2. Grinding mill machine: 1,2,9,10 – electric motors bearings; 3,4,11,12 – driven shafts rotors bearings; 5,13 – electric motors; 6,14 – driving shafts; 7,15 – driven shafts; 8,16 – large and small impellers (z_d , z_m - number of large and small impeller rollers and R_d , R_m - radiuses of impellers, respectively), 17 – belt drives

.Under dynamic forces, however, at each moment all balls in a bearing are loaded as, for example, in the supports of angular contact ball bearings subjected to radial and axial forces [1]. In dynamics the angle of a bearing element contact is α and is not equal to the initial angle α_0 . It is estimated when analyzing the vibrations in high-precision rotor supports with precise bearings and heavily loaded bearings. The variation of a contact angle is caused by dynamic forces related to the tolerances of bearing element geometrical dimensions, limited stiffness of their steel which, under loading, does not ensure the stable position of a shaft rotating axis. Therefore, theoretically obtained frequencies, as a rule, do not agree with the measured ones under the natural conditions of the bearing rotation. The excited vibration intensity and the frequencies of bearing elements may vary for the uniform machines among the bearings of the same sort, because, in addition to outer parameters and bearing assemble into units technology, the vibration amplitudes depend on the accuracy of the manufacture of each roller bearing element, roughness of their surface, the cage design, lubricant quality and a lubrication method.

Vibration frequencies excited by balls and rollers - defective rolling elements. The vibration frequencies of defective rolling elements are frequently calculated by assuming that during a revo-

lution of a defective ball it strikes once to an inner and outer bearing ring races by its defective spot. Here, a gyroscope effect subjecting the revolving ball is not taken into consideration because, as practice has shown, a defective ball spot touches the rolling races not periodically, it does it at varying time intervals thus impeding the diagnostics of a ball bearing fault.

High frequency vibration are related to the resonance frequency of a bearing outer ring, acting as an element with tightness fastened on a bearing support, as it has been examined in an angular contact ball bearing [3].

3.2. Bearing diagnostics with an shock pulse method (SPM)

This method is considered to give the precise information on the technical condition of machine roller antifriction bearings in a special vibrations data format. The essence of SPM may be illustrated by a sample of a metal ball which, when rebounding, strikes a metal plate, Fig.3.1. During the shock the colliding molecules generate a pressure wave spreading in both bodies. After the shock in the next collision stage, both bodies begin vibrating in mechanical vibration frequency which is the function of colliding bodies' masses and stiffness.



Fig.3.1. Scheme of a ball shock on a plate, amplitude A of a pressure wave and a signal

Subjected to the shock wave the contact transducer measuring shock pulses is vibrating at a certain natural frequency. The transducer touches the bearing support or the bearing itself during vibration monitoring. The transducer separates a high frequency shock pulse signal A , its electronic system amplifiers that signal filtering the low frequency mechanical vibration. The intensity of shock pulses is evaluated by decibels - the values of dBm and dBc as parameters [2].

3.3. Dryer's bearings vibration evaluation

The bearing condition monitoring technology is applied with proximity probes pioneered by Bently Nevada Corporation (USA) in the last decades [1]. This technology has been used for evaluation the technical condition of heavy duty low rotation rotor bearings providing measurements of rotors shaft vibration and radial shaft displacements S_{pp} , as shown in Fig.3.2.



Fig.3.2. Measurement of shaft vibration and radial shaft displacement with proximity probes

This method is based on monitoring shaft displacement and vibration displacement with respect to a bearing support for low and high power machines bearings. It is shown here that this method is

also applicable for diagnostics of rotors rotating at very low speeds.

Rotation frequency of cylindrical roller is 48 rpm. The vibration displacement peak-to-peak amplitude is $S_{pp}=424 \mu\text{m}$ and characterizes the dynamic cylindrical roller position data. The running speed $1X=0.80 \text{ Hz}$ frequency displacement amplitude is dominant in the spectrum, Fig.3.3. The vibration displacements to multiples ($2X, 3X, \dots$) of running speeds are insignificant values. For example, $2X=1,6 \text{ Hz}$ frequency vibration displacement amplitude comprises only 30% of $1X$ frequency amplitude value. The measured dynamic displacement $424 \mu\text{m}$ value shows the changes of the stiffness characteristics two roller bearings on adapter sleeves. The cylindrical roller displacement data shows changes in the average position of the roller during one rotation cycle. These data do not characterized rapidly changing data because they are effectively low-pass filtered. The roller displacement data shows eccentricity of the cylindrical roller rotating in the two roller bearings. The measured eccentricity value is $103 \mu\text{m}$ including changes in the static radial load and stiffness characteristics of the cylindrical roller rotor.

The absolute vibration spectra are shown in Fig. 3.4. In the absolute vibration velocity spectrum it is possible to see low frequency harmonics, e.g. 8.75 Hz frequency as one line rollers of the double row roller bearing passing inner ring trace frequency. 12.5 Hz frequency represents double row rollers passing outer ring trace frequency. The 15 Hz/1.2mm/s frequency represents double row rollers passing inner ring trace frequency. Rollers rotation frequency 5,2 Hz vibration is low.

These frequencies have small displacement amplitudes in the relative vibration spectrum in comparison with $1X$ frequency displacement amplitude as was shown in Fig. 3.3.

Despite the fact that high frequency vibration intensity is low, as shown in Fig. 3.4, but acceleration data format is informative in bearing periodic condition monitoring method.

The relative vibration displacement measurement method is powerful in diagnostics of low speed rotors together with high frequency absolute acceleration method and SPM method. Technical condition of dryer's cylindrical roller bearings according measurements data is satisfactory.

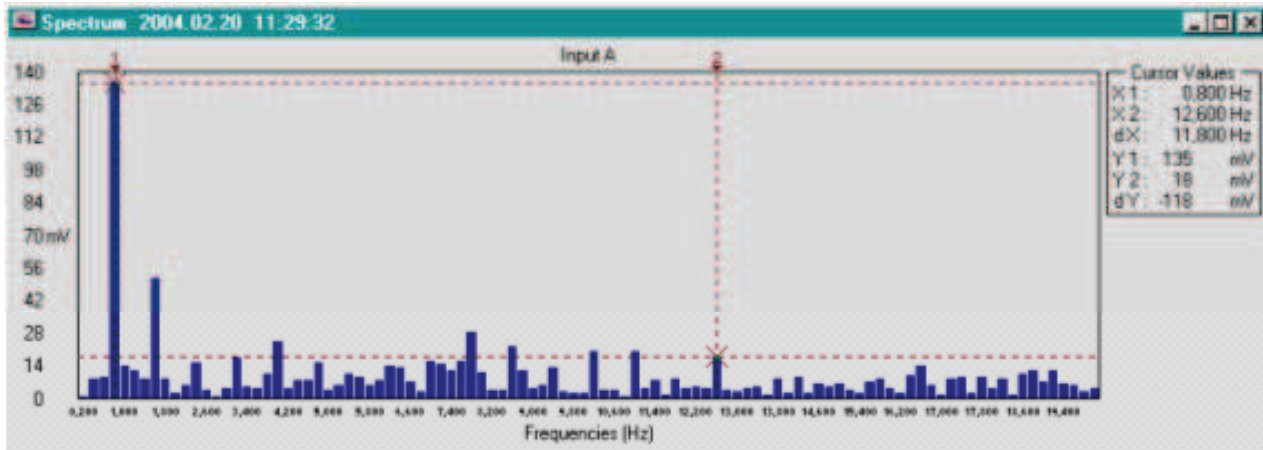


Fig.3.3. Relative vibration displacement spectrum plot of the right side cylindrical roller; unfiltered vibration is measured in radial direction; $S_{pp}=424 \mu\text{m}$, $\text{SPM-dBm/dBc}=18/2$

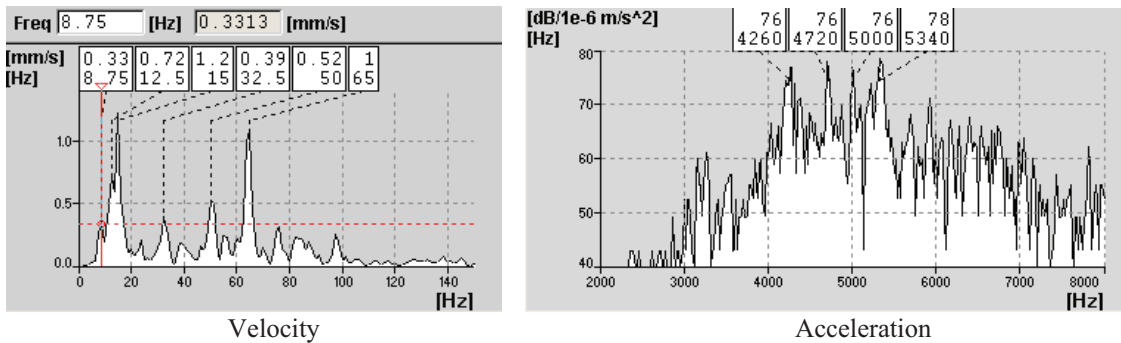


Fig. 3.4 Cylindrical roller bearing absolute vibration velocity and acceleration spectra of dryer machine

3.4. Grinding mill's bearings vibration evaluation

Applying the above-mentioned methods and means the experimental researches of roller bearings have been carried out with an electromechanical drive whose kinematical scheme is given in Fig. 2.2. To perform the experimental tests SPM tester T2001, vibration signals analyzers System 2 and A4101ENG, and software DDS 2000 have been used [2, 5]. The results have been obtained by running a heavy-loaded trapezoidal belt drive. Elimination of gear drives has enabled us to separate more accurately high frequency vibration excited by a faulty bearing. The drives of an analyzed machine are subjected to external mechanical vibration excited by the shocks of other machine loads. These shocks increase the vibration amplitude of the equipment being measured 2-3 times without changing the vibration frequencies.

The application of SPM for determining the technical condition of the 3rd and 4th bearings is shown in Fig. 3.5: upper graphs dBm and dBc are of the 3rd bearing, the lower ones – of the 4th. The machine to be tested was new, never used before. During the first run (2002.11.11) the technical condition of the 3rd (35/25) and the 4th (39/33) bearings fell to the caution region according to SPM dBm/dBc parameters, whereas the technical condi-

tion of roller bearings was satisfactory. After a month of continuous exploitation the technical condition of both bearings worsened. The critical limit was fixed in 2002.12.23 when dBm/dBc parameters of both bearings were: 68/38 – of the 3rd bearing, 54/36 – of the 4th one. The machine was stopped, the bearings were revised, the joint of the outer ring was replaced and the lubricant was replaced by a higher quality one having some molybdenum. Having taken those preventive measures in 2003.02.24 dBm/dBc parameters were: 41/33 – of the 3rd bearing, 38/28 – of the 4th.

The radial clearances in the 3rd and 4th bearing supports were measured by contact less measuring transducers of vibration displacement and radial displacement. The radial clearance of the 3rd bearing was found to have decreased and reached from 0.002 mm to 0.004 mm, while the radial clearance of the 4th bearing reached 0.016 mm. These results made it possible to reduce the tightening and dynamic bearing stiffness as well as to increase the radial clearances. However, the bearing having been not disassembled, the tightening of the inner bearing ring on a centre shaft had been not reduced.

Nevertheless, after two months of nonstop run the bearings technical conditions deteriorate. In 2003.04.23 dBm/dBc parameters determined by means of SPM were: 63/43 – of the 3rd bearing, 36/25 – of the 4th. The bearings having been lubri-

cated, dBm/dBc values diminished considerably: 36/27 – of the 3rd bearing, 27/16 – of the 4th. The tests proved the necessity of grease lubrication of the bearings periodically every 150-200 h nonstop run. As seen from the plots, the bearings technical condition according to SPM even having reached the warning zone, the machine was running and according to the monitoring data of 2003.06.26 the parameters of the 3rd bearing were 49/29, those of the 4th were 45/21, i.e. the technical conditions of both bearings were stable.

The absolute vibration velocity and acceleration spectra of the 3rd bearing shown in Figs 3.6 a, b indicate that in the spectra of vibration velocities of kinematical excited frequencies there are no vibration frequencies excited by bearing elements. The amplitude of vibration speed over 100 Hz is low, lower than 0.25 mm/s. The vibration speed amplitude of synchronous vibration frequency 22.5 Hz attains 1.7 mm/s and belongs to the vibrations excited by rotor unbalance. The vibration of 2000 Hz

and 4000 Hz frequency are prevailing in the vibration acceleration spectra, their source being the roller bearing elements, not the vibration excited by grinding mill impeller rollers (8 grinding rollers of a small impeller x 10 grinding rollers of a large impeller x 22.5 Hz , synchronous rotation speed of a rotor system = 1800 Hz). The vibration acceleration amplitudes up to 2000 Hz are low, lower than 1 m/s². The comparison of vibration acceleration of the 3rd and 4th bearings reveals that the intensity of the 3rd bearing vibration is much greater. It is due to the design scheme of the 8th impeller of the rotor: the 4th bearing located closer to a heavy impeller and is stronger subjected to the gravitation force of massive impeller. While the 3rd bearing is subjected to this force vertically upwards the force acts vertically downwards. Dynamic forces, acting from further machines through an insufficiently stiff support frame system are also acting the rotor

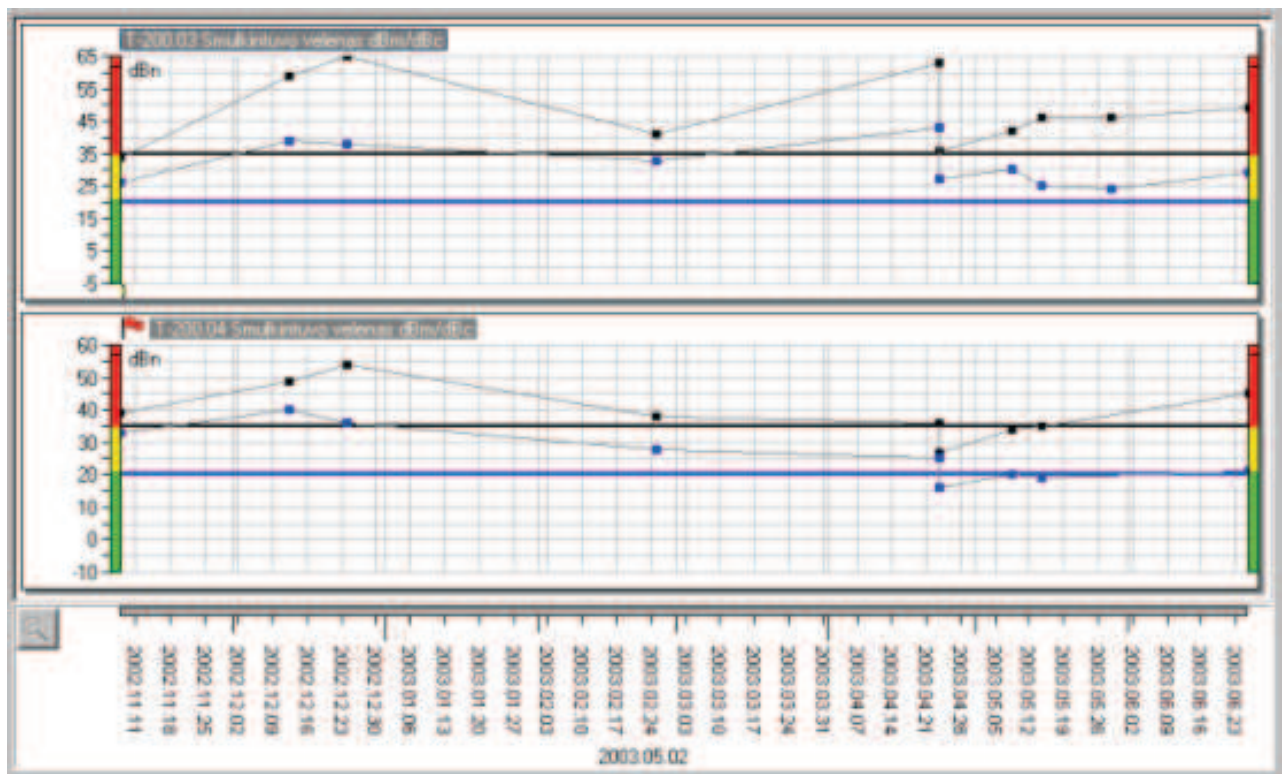


Fig.3.5. SPM parameters dBm/dBc plots versus time of bearings SKF 22216 CJC 403 during nonstop 7 months operation of new grinding mill machine

In order to diagnose these bearings the vibration velocity spectrum, Fig. 3.8 a provides no insufficient information comparing it with that of vibration acceleration spectra format, Fig. 3.8 b.

Table 1 gives the summary of the results of 3rd bearing vibration monitoring. It proves that both a classical vibration measurement method using the frequencies interval up to 16000 Hz and SPM yield similar results evaluating the technical condition of a bearing. The frequencies of kinematical excitation

cannot be used in diagnostics as the framed support structures of the equipment are not stiff and they transmit the vibration excited by adjacent machines thus aggravating the diagnostics of the bearing technical condition according to kinematical frequencies even in the initial stage of their running-in.

Table 2 presents the results of the 3rd and 4th bearings of a small impeller monitoring which confirm that for this type of equipment both bearing

diagnostic methods SPM and that of mechanical high frequency vibration are equally informing.

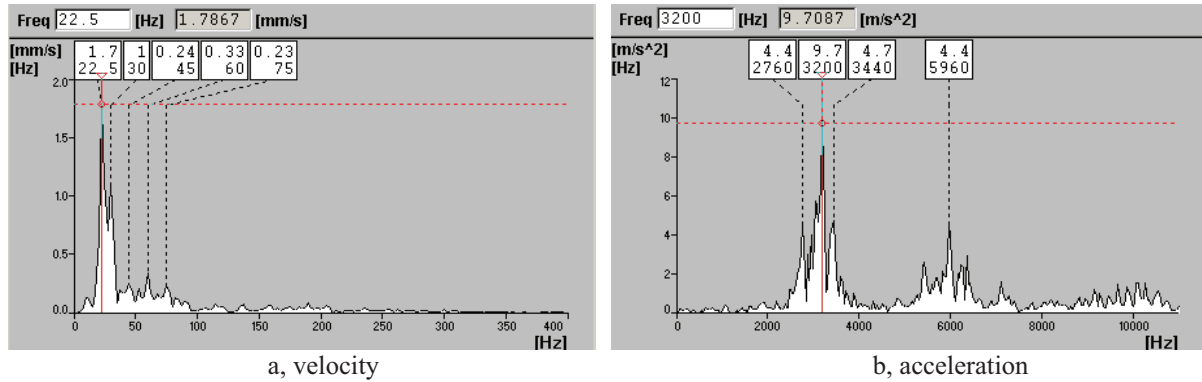


Fig. 3.6 The new grinding mill machine large impeller rotor vertical vibration velocity and acceleration spectra of the overloaded 3rd bearing

Table 1 The 3rd bearing of the large impeller vibration intensity measurements results acquired by various methods

| Bearing | 10-1000 Hz, V_{rms}/V_{0-p} | 5-16 kHz, a_{rms}/a_{0-p} | 0,8-6000Hz, a_{rms}/a_{0-p} | SPM, dBm/dBc |
|----------|----------------------------------|--------------------------------|----------------------------------|-----------------|
| 1 | 2 | 3 | 4 | 5 |
| 3rd V | 6.96 mm/s | 0.49g | 18.7 m/s² | 49/29 |
| | 11.0 mm/s | 1.18g | 44.0 m/s² | |
| 3rd H | 2.78 mm/s | 1.13g | 19.9 m/s² | |
| | 5.26 mm/s | 2.68g | 51.6 m/s² | |

Table 2 The 3rd and 4th bearings of the small impeller vibration intensity measurements results acquired by various methods

| Bearing | 10-1000 Hz, V_{rms}/V_{0-p} | 5-16 kHz, a_{rms}/a_{0-p} | 0,8-16000 Hz, a_{rms}/a_{0-p} | SPM dBm/dBc |
|----------|----------------------------------|--------------------------------|------------------------------------|----------------|
| 1 | 2 | 3 | 4 | 5 |
| 3rd V | 1.22 mm/s | 1.03g | 13.8 m/s² | 34/28 |
| | 2.91 mm/s | 2.44g | 32.5 m/s² | |
| 3rd H | 6.05 mm/s | 0.39g | 10.5 m/s² | |
| | 9.37 mm/s | 0.92g | 24.8 m/s² | |
| 4th H | 0.83 mm/s | 0.75g | 8.80 m/s² | 36/26 |
| | 1.87 mm/s | 1.77g | 20.8 m/s² | |

4. CONCLUSIONS

1. Monitoring of mechanical vibration of heavy-loaded slowly rotating (about 1 Hz) antifriction roller bearings is considerably more efficient when using the absolute vibration accelerations which vary in the frequency range up to 16000 Hz (LIN frequency interval in vibration signals analyzer Adash4101) if compared to the kinematical frequencies (up to 1000 Hz) method.
2. Both methods - SPM using vibration intensity parameters dBm/dBc and that of high frequencies - make it possible to obtain similar quantitative and qualitative parameters of roller bearings suitable to diagnostics. It concerns dynamically loaded horizontal rotors.

3. Measurement of radial clearances by proximity probes for determining the relative displacement of a centre shaft with respect to the bearing outer ring has indicated an insufficient magnitude of a radial clearance of 0.002-0.004 mm when the radial clearance in the 4th bearing of the same rotor reached 0.016 mm as in the bearings of the other reliably working grinding mill.
4. The defect in the 3rd bearing of a large impeller has developed because of too stiff tightness of the 3rd bearing which increased with an increase in temperature. Additionally, this bearing has been working with too much lubricant which has increased dynamic loads.
5. During monitoring the lack of lubricant was disclosed in the bearing. Having lubricated the defective bearings, their SPM parameter de-

crease for a short period: in the 3rd bearing from 63/24 to 36/27 and in the 4th bearing from 36/25 to 27/16. However, after 1-2 h of continuous operation the vibration went up to the initial high level. The defective bearing was replaced by a new one.

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Professor Habil. Dr. Vytautas BARZDAITIS, b. 1946. Head of Mechatronics laboratory. Scientific interests: Modelling and Simulation of Mechatronics systems using Bond Graph approach; Design and Research of machine condition monitoring, protection, fault

diagnostics and expert systems



Associate professor Dr. Vytautas ŽEMAITIS, b. 1943. Faculty of Mechanical Engineering Mechatronics laboratory. Trends of scientific activity: Modeling and Simulation of Rotors with journal and antifriction bearings, especially precision rotors with gas bearings.



Senior research scientist Kęstutis ŽEBELYS, b. 1947. Mechatronics laboratory. Scientific interests: Condition monitoring, fault diagnostics and reliability evaluation of machines. Diagnostics of antifriction bearings with Shock Pulse Method – SPM, dynamic

balancing of rigid and flexible rotors



Associate professor Dr. Zigmantas POCIUS, b. 1942. Department of Engineering Mechanics. Trends of scientific activity: Research and Stabilization of dynamic parameters of mechanical systems, vibration isolation devices of machines, active vibration control of gyroscopes rotating systems

tems



Assistant professor Pranas MAŽEIKKA, b. 1978. Graduated from Klaipėda University Faculty of Marine Engineering. Scientific interest: Modeling and Simulation of rotors with flexible supports. Vibration Monitoring and diagnostics of Marine technique, identification of vibration sources.

Dynamic balancing of critical rotors.