VIBRATION DIAGNOSTICS OF ANTIFRICTION BEARING DAMAGES DURING INSTALLATION AND ASSEMBLY OF HEAVY ROTORS

V.M. Shabaev, A.S. Kazantsev (JSC "Gasturboservice", Tyumen)
M.K. Leontiev, I.V. Garanin (Moscow Aeronautical Institute, Moscow)
V.A. Karasev (Central Institute of Aviation Motors (CIAM), Moscow)

During installation and assembly of heavy rotors with antifriction bearings, for example in oversize gas turbine engines, there can occur local damages of rollers, races and a cage-carrier because of swashes, impacts, or overforced installation. Damages appear in form of chips, grooves, dents, or shear rupture. Also, metal buckling is possible in the damage zones. Once a rotor is installed, such defects can not be detected unless disassembly of machine is performed. During exploitation, these defects can cause further damages of bearings, making an operation process dangerous. That is why the development of diagnostics technologies of such assembly damages of bearings without disassembling the machine is very much of current interest. Let us examine the results of experimental research on diagnosability of damages in heavy rotors' bearings by the methods of vibration diagnostics and with the aid of accelerometers' signals placed on the outer side of case-shaped parts way from the bearing and under condition of slow rotor spinning.

Subject of Research

We investigated roller bearing of the rotor VD of gas turbine engine DZ-59L2 installed in an assembly-room of a factory on VM3000 balancing stand (Fig. 1).

Specifications of roller bearing of the rotor VD of gas turbine engine DZ-59L2

Bearing type 6.32132 BT2
Number of rollers 24
Diameter of a roller, mm 20
Length of a roller, mm 20
Operating length of a roller, mm 18
Size variation of a roller according to Technical Demands, mm . . . . < 0.002
Weight load of a bearing, kg 619
Radial clearance after fitting of an inner race, mm . . . . 0.04…0.099
Outer race fitting . . . . loose
Bearing lubrication . . . . liquid

Outer race of the bearing is located in a box in form of a liner. The liner is fixed by single-sided support on a flat bearing (cradle) of the stand which has the form of 30 mm thickness plate. On this plate, accelerometers are placed vertically. To reach the measurement point a signal has to pass through several connections from the bearing. This is quite a complicated way with a length around 0.5 mm. Main bearing frequencies have following values: FTF=0.45f_{ROT}, BPFI=13.2f_{ROT}, BPFO=10.8f_{ROT}, BSF=4.95f_{ROT}, where FTF is rotation frequency of a cage-carrier; BPFI is passing frequency of rollers on the inner race; BPFO is passing frequency of rollers on the outer race; BSF is roller rotation frequency; f_{ROT} is rotor spinning frequency.

Defects of the bearing

Experiments were performed on the same new good state bearing. In this bearing, a new roller was replaced with a damaged one. Damaged rollers were taken from an exploited bearing. All of them were worn on the edges. These rollers were also additionally damaged analogously to what can happen in the process of installation. Images of such damages and their description according to the measurement data before and after testing are presented in Fig. 2.

Measuring equipment and software

We employed accelerometers of the following types: KD-41, VK-315, RA-032, and others. For primary treatment there was used vibration equipment of the "National Instruments" – electronic board NI-4472. The electronic board has eight simultaneously respondent channels, each having 24-digit analog-to-digital converter (ADC). For record and processing of vibration signals we employed VibroNET R2.1 software system [1] written in LabVIEW 6.1. Duration of temporal realizations at the established regimes reached 30…40 s. Sampling frequency amounted 10 kHz for every channel.

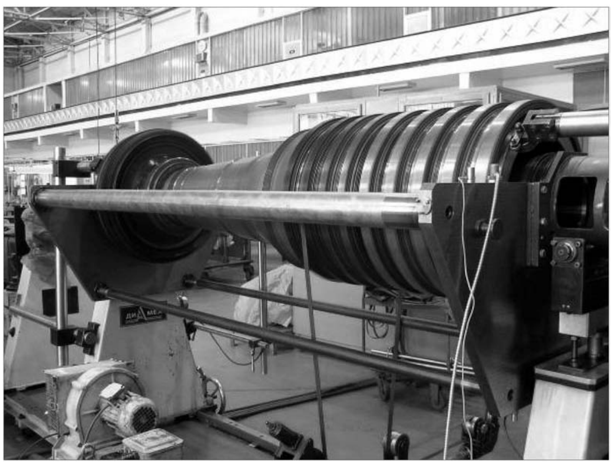


Fig. 1. General view of the rotor on a balanced stand from the side of the tested bearing. Also, there can be seen accelerometers on the stand cradle

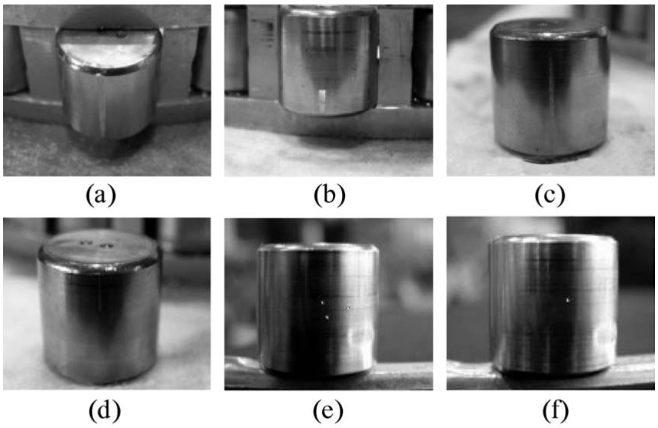


Fig. 2. Damages of rollers according to the measurement data. (a) Flattened surface along the whole length of the roller; 0.015…0.025 mm in depth. (b) Small flattened area on the part of roller length; 0.018 mm in depth, 4.5 mm in length. (c) Groove along the whole length of the roller; 0.046 mm in depth, 0.64 mm in width. (d) Small groove on the part of roller length; 0.023 mm in depth, 5 mm in length, 0.3…0.5 mm in width; buckling of material at the edge of a groove before testing 0.063 mm, after testing 0.018 mm. (e) Three sharp dents + one sharp dent; 0.3…0.5 mm in diameter; 0.064 mm in depth; material buckling near the sharp dent – 0.006 mm; single sharp dent is placed circumferentially at the angle of 215° relatively to the zone of three sharp dents. (f) Single sharp dent; 0.5 mm in diameter; 0.053 mm in depth

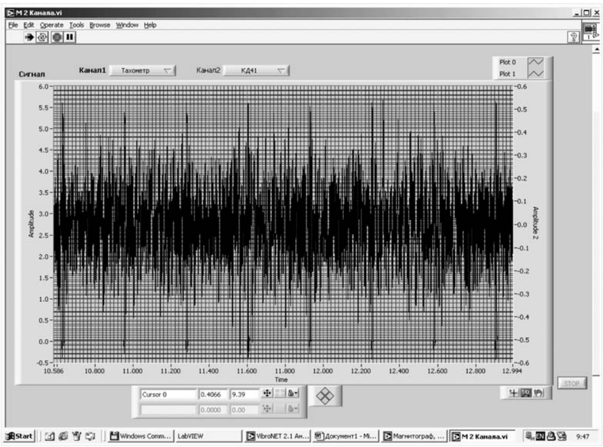


Fig. 3. Oscillogram of a good state bearing. Regime 1

Kinds of testing

Experimental research was pursued in two stages. The first stage included preliminary testing for evaluation of testing conditions and assessment of vibrational, acoustic and electromagnetic interference from the factory equipment. Passing of weak vibration signals from the good state bearing to the measurement point was examined too. At the second stage bearing signals were measured under condition of 180 min^{-1} and 240 min^{-1} frequency of rotor spinning (regime 1 and 2 correspondingly). Accuracy of rotation frequency maintenance reached 1%.

Analysis of interference from the factory equipment

Accelerometers and equipment appeared to be sensitive to an influence of the operating factory equipment. The biggest influence was detected on frequencies below 10 Hz, and on 50 Hz frequency and its harmonic components.

Passing of vibration signals from the bearing to the measurement point

Passing of weak vibration signals from the bearing to the point of vibration measurement was examined by the impact test on no-rotor stand. We tapped on cylindrical surface of a single roller by an 80 g wooden brick from the distance of 30 mm. The roller was installed in the lower part of the outer bearing race. Impact frequency approximately equaled 1 hit per second. In process of tapping of the roller there were distinguished several resonance zones (Table 1). Maximal amplitude of hit response on resonance frequencies exceeded the level of continuous spectrum 5–10 times.

Vibration signals form a good state bearing

Oscillograms of unfiltered signals from a good state bearing generally have noisy kind (Fig. 3).

At the same time, the signals' spectra clearly show components with frequencies BSF, BPFO and their harmonic waves (Fig. 4).

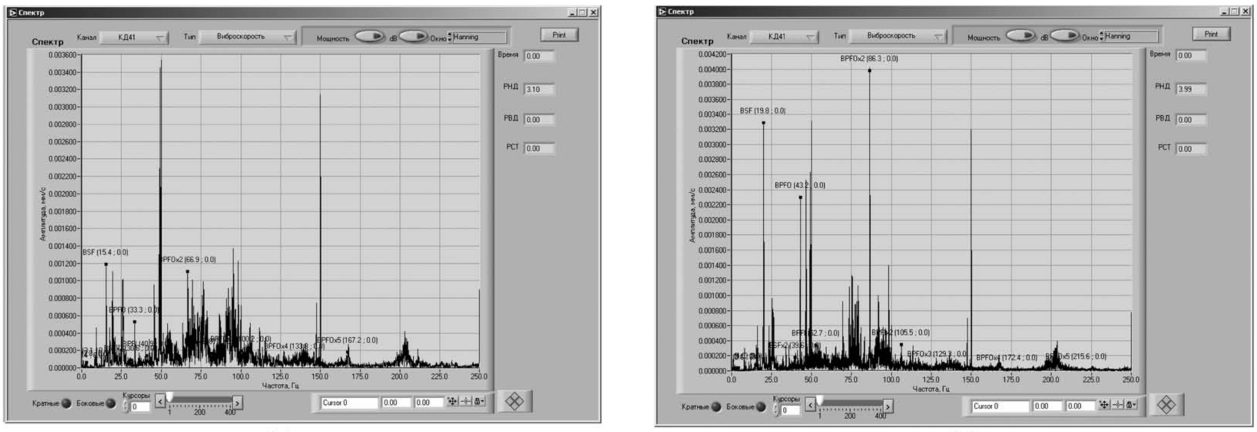
On the frequency 2BPFO the amplitude is higher, than on the frequency BPFO. It should also be mentioned that the frequency BSF aligns with the 5th harmonic wave of the rotor spinning frequency and for that reason it shows up more than other bearing frequencies.

Vibration signals from the bearings with damaged rollers

Flattened surface along the whole length of a roller (Fig. 5). As it was expected, the signal created by a bearing damage represents periodic sequence of a set of several impulses. Maximal signal amplitudes reached (0.4…0.8)g at regime 1 and (1.0…1.8)g at regime 2. An interval between neighbor impulses equals 1/2BFS; the cycle of set repeating is equal to one turn of a cage-carrier, i.e. 1/FTF; number of impulses in the set – 6 (sometimes 7).

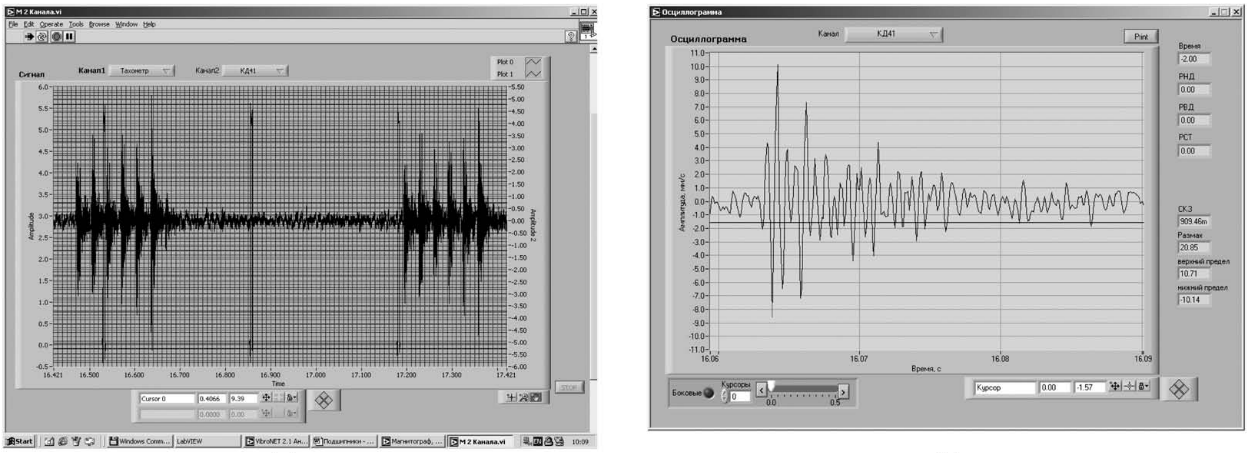
1. Resonance zones in the spectra of signals that appears when tapping the roller

Parameter	Resonance frequency, Hz					
	260	700	760	1,050	2,500	4,300
Amplitude, mm/s	0.0002	0.00017	0.0004…0.0007	0.00015…0.0006	0.0003	0.0002
Amplitude, g	0.033	–	–	–	–	–



(a) (b)

Fig. 4. Spectrum of a good state bearing signal.
(a) Regime 1. (b) Regime 2



(a) (b)

Fig. 5. Oscillogram of a signal from the bearing with damaged flattened surface along the whole length of the roller. Regime 1.
(a) Interval between index marks of rotation frequency sensor is equal to one rotor turn.
(b) Stretched oscillogram of an impact response signal

In the interval between impulses there are detected several (12…15) damped oscillations with the frequency around 1000 Hz, what can be clearly viewed on a stretched oscillogram (Fig. 4, (b)). These oscillations are totally attenuated till the next impulse. This is characteristic for a dynamic system’s response to impact under condition of comparatively low damping, when duration of an impact is considerably shorter, than that of oscillation period of the very system, and an impact frequency is low.

Fig. 6 (a) shows a spectrogram of a vibration signal from the bearing with a damaged roller for regime 2.

Amplitudes of the main resonance constituents of the spectrum are stated in Table 2.

Table 3 shows measurement results on low frequencies.

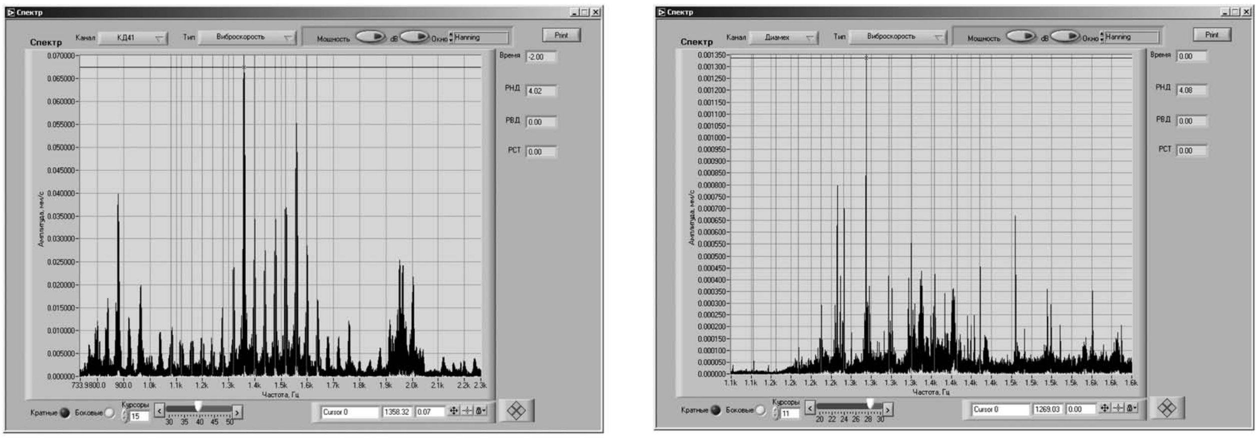
Groove along the whole length of a roller. Oscillograms and spectra of this kind of damage are analogous to those of flattened surface damage type along the whole length of a roller.

Three sharp dents + one sharp dent. In this kind of damage the amplitude of the main resonance component of spectrum on frequencies 1250…1280 Hz reached 0.0012 mm/s at regime 1 and 0.0013 mm/s at regime 2 (See Fig. 5 (b)).

In the spectrum of the resonance zone there become apparent side combinational spectrum components with frequency interval changing proportionally to the frequency of rotor spinning (Table 4). Under such conditions, the ratio $\Delta f/2BSF$ is approximately equal to the ratio of angles on the right and on the left between the damage zones on the circumference of a roller 145°/215° =0.67.

2. Amplitudes of the main resonance components of the spectrum

Frequency, Hz	Vibration velocity, mm/s	
	Regime 1	Regime 2
875	0.018	0.04
1350	0.024	0.065



(a) (b)

Fig. 6. Spectral recording of vibration velocity. Regime 2. Side harmonic components with following steps.
(a) ~ 39 Hz (2BSF). (b) 28 Hz

3. Measurement results on low frequencies

Regime	Vibration velocity, mm/s							
	f_{ROT}	FTF	BSF+5f_{ROT}*	2 BSF	BPFO	2 BPFO	BPFI	50 Hz
1	0.038	0.02	0.12	–	0.006	–	–	0.04
2	0.07	0.1	0.04	–	0.014	–	0.01	0.03

* Presence of components ±FTF

4. Side combinational spectrum components

Parameter	Regime 1	Regime 2
Frequency interval between spectrum components, Δf, Hz	20.5	28
Δf /2BSF	0.68	0.7
Δf /FTF	15.5	17.5
Δf / f_{ROT}	6.83	7

All this indicates that relative peripheral position of damages on the surface of a roller can be determined by an interval between side combinational spectrum components in high-frequency resonance zones.

Other damages. In case of damages like short flattened surface on the part of roller length or short groove on the part of roller length, the signals practically do not differ from those of a good state bearing by oscillogram pattern or spectrum. This results not only from their considerably lesser vibration activity comparing to the flattened or groove damage along the whole length of a roller, but also with domed run-out of a roller. Roller’s diameter decreases 20 μm and more exceeding maximal contact deformation of roller and races. For that reason considerable part of damage length is not involved in contact.

In case of a damage of one sharp dent type, signals also practically do not differ from those of a good state bearing by oscillogram pattern or spectrum. This is determined by a very small vibration activity of the damage in form of single dent without buckling of material.

It should also be mentioned that low-frequency spectra (on frequencies FTF, BSF, BPFO, BPFI and their harmonic components) practically did not differ from the spectra of a good state bearing whatever were the roller’s damages.

Summing up spectra analysis results it can be concluded that spectrum analysis of vibration signal in the zone of relatively high-frequency informational resonances of the construction is informative for evaluation of dimensions and positioning of roller damages in the form of quite lengthy flattened surfaces and grooves, as well as dents with material buckling.

Band-pass filtering application

The range of frequencies from 1.0 to 1.5 kHz is the main resonance zone, which becomes apparent only in case of impacts inside a bearing.

There were pursued band-pass filtering and comparing of the derived values of vibrational acceleration in the mentioned diapason for a good state bearing and variously damages bearings. The results of the derived values comparison are presented in Fig. 7.

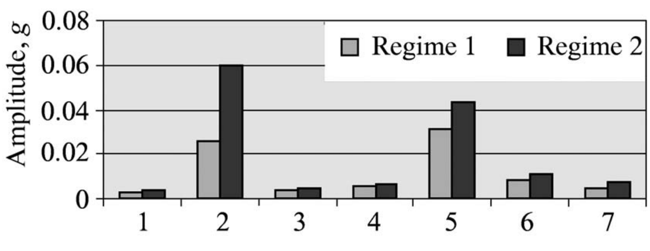


Fig. 7. Amplitudes of vibration acceleration in the frequency range 1.0…1.5 kHz:
1 – good state bearing; 2 – flattened surface along the whole length of a roller; 3 – short flattened surface on the part of the length of a roller; 4 – short groove on the part of the length of a roller; 5 – groove along the whole length of a roller; 6 – three sharp dent + one sharp dent; 7 – one sharp dent

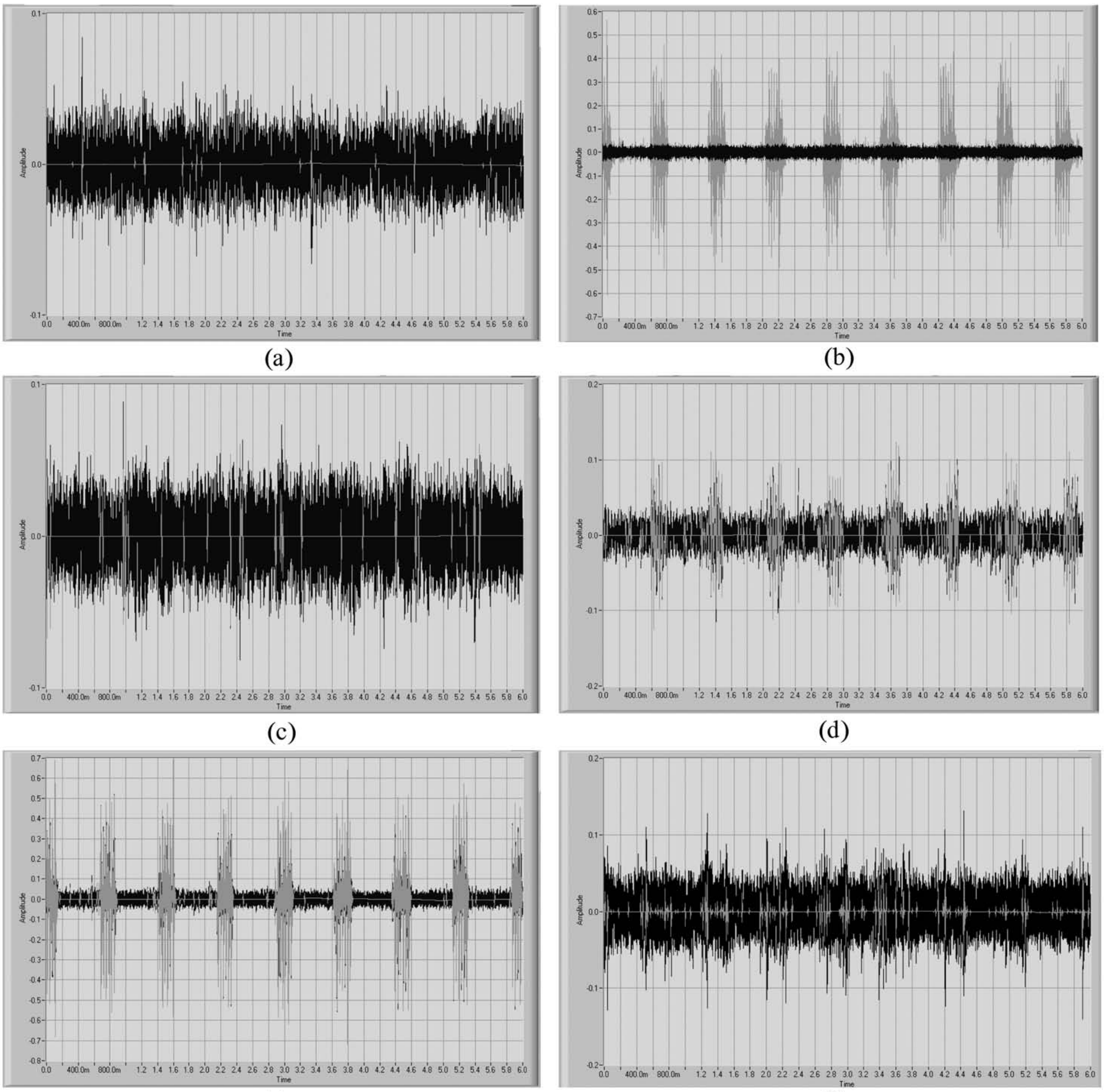
It can be clearly seen that:

- there is a significant different between the signals of a good state bearing and those of the bearing with types of damages like long flattened surface on the part of the roller length, or long groove on the part of the roller length and three sharp dents + one sharp dent;
- there is an insignificant difference between the signals of a good state bearing and those of the bearing with types of damages like short flattened surface on the part of the roller length, or short groove on the part of the roller length and one sharp dent;
- increase of rotor spinning frequency increases the difference between signals of a good state bearing and a bearing with a damaged roller.

According to the results of the band-pass filtering it can be assumed that such filtering of a vibration signal can be effective in the zone of relatively high-frequency informational resonances of construction in evaluation of dimensions of roller damages in the form of quite lengthy flattened surfaces and grooves, as well as dents with material buckling.

Wavelet analysis

The method of wavelet analysis [2] is effective in enhancing impulse components of signals with strong noise interference. As expected, wavelet analysis proved itself useful in detection of impacts from slight bearing damages, such as short grooves and little dents. Fig. 8 represents the results of processing of a signal obtained during the testing of a variously damaged bearing. According to recommendations of the work [3] there was applied wavelet transformation employing the Daubechies function of the type "db02". Parameters of the wavelet were automatically optimized. Impulse components of vibration signals obtained in process of wavelet filtering are shown in light color. It can be seen that the number and amplitude of signal impulses after wavelet transformation and even in case of slight roller damages are considerably larger, than from a good state bearing.



(a) (b) (c) (d) (e) (f)

Fig. 8. Results of processing of a signal obtained during the testing of a variously damaged bearing:
(a) Without defects. (b) Flattened surface along the whole length. (c) Short flattened surface on the part of the length of a roller. (d) Short groove on the part of the length of a roller. (e) Groove along the whole length of a roller. (f) Three sharp dent + one sharp dent

Resume

It is experimentally proved that vibration diagnostics method can be quite effective in detection of local roller damages that are possible in process of installation and assembly of a bearing as part of a rotor and that have the form of flattened surfaces, grooves, and also dents (sharp dents).

It is shown that various diagnostic characteristics of the local roller damages are contained in oscillogram and spectrum of accelerometer signal obtained in process of slow rotor spinning.

Damage criteria can be determined by the band-pass filtering of a signal within the range of informational resonances. The best result in enhancement of a roller damage signal is achieved by the wavelet filtering that allows to detect even slight damages of a bearing roller. It can be concluded that complex signal processing according to the mentioned algorithms is useful for detection and evaluation of the bearing damages.

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TRANSFORMATION OF SPECTRUM AND FORM OF SIGNALS IN THE CHANNELS OF ACOUSTIC EMISSION DIAGNOSTIC SYSTEMS

V.A. Gumenyuk, N.A. Kazakov, E.V. Nesmashny
(Krylov Central Shipbuilding Research Institute, Saint-Petersburg)

Analytic research of transformation of acoustic emission (AE) signals in the channels of AE-systems forms one of the most complicated problems in electro-acoustics because of wideband and omnirange character of acoustic emission.

These aspects were stated in the first Russian AE monograph [1], but only on the level of qualitative conceptions. More recent attempt to generalize signals' analysis on the base of the spectral approach [2] was left unfinished. According to the authors, the reason came from impossibility of measuring and complexity of calculation of response characteristics, or amplitude-frequency characteristics (AFC) of emission and spreading of AE-signals in AE-control separate channels. The situation did not really change even after publication of 8-volume reference book on NDT and technical diagnostics in 2005 [3].

Nevertheless, such generalization is essential for the development of methodology and means of contemporary AE-diagnostics on the base of employment of wideband AE-systems. The latter allows to increase the number of information–bearing parameters of AE-signals (spectrum, analog form, etc.) [4, 5] in order to develop multiparameter criteria of estimation of danger level of growing defects, of noise filtration equipment and of improvement of accuracy of defect location.

In fact, AE-diagnostics represents indirect electro-acoustic measurements of mechanical displacements in the radiation point according to the parameters of electric AE–signal. Scheme of these measurements shown in [1], counts 4 main channels of oscillations' transformation in the general channel of measurements:

1) mechanoacoustic channel, in which mechanical potential energy comes out in the form of an impulse of stress waves (depression waves) as a result of impulse transition of medium surrounding the defect into a new balanced state;

2) acoustic channel (wave guide) that transmits acoustic AE-signal from the radiation point to the point of receiving – AE receiver (AER);

3) acoustoelectric channel (AER) that transforms AE-signal from acoustic form into electric form;

4) electrical channel of amplification, processing, and registration of electric AE–signal.

Initial signals pass through the mentioned channels in consecutive order. Input of every next channel puts pressure on the output of the preceding one.

Spectrum of an output signal of any channel is the spectrum of its input signal multiplied by complex AFC of this channel [2]. Therefore, spectrum of a signal that passed through the number of channels is equal to the product of an input signal spectrum of the first channel (in the sequence of considered channels) by the product of complex AFC of all channels including the first. For example, spectrum of an electric signal S_4 on the covering of piezoelectric element of AER, i.e. at the input of the 4th (electric) channel under the known spectrum S_1 at the input of the 1st (mechanoacoustic) channel and known complex AFC H_1, H_2, and H_3 in accordance with the 1st, 2nd, and 3rd considered channels, equals to

$$S_4 = S_1 H_1 H_2 H_3. \qquad (1)$$

Each amplitude–frequency characteristic represents the ratio of signals' complex amplitudes at the output and at the input of a channel and in physical sense it is its ampli-