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VIBROACOUSTIC MONITORING OF BEARING POINTS FAILURES IN THE DRIVING SYSTEM OF THE MINING MAIN VENTILATION FANS

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Bearing points constitute the integral part of present-day machinery. They often limit in the service life of machines. As there is no access to them in the course of operation, diagnostic methods which can be applied without dismantling the machine, and particularly vibroacoustic methods based on the fact that there is a dependence between failure-free operation and the vibroacoustic condition of bearings become more and more essential. The evaluation of the technical condition of rolling and slide bearings, based on measures of a vibratory signal, made on the basis of amplitude values of discrete components occurring in the spectra of the recorded vibratory signals, has been the subject of diagnostic examinations. The study also covered the rolling bearings and slide bearings supporting the main shaft of the mine fans driving system of WPK type, designed for main ventilation in normal conditions of their operation. It is worth noting that considerably less work has been devoted to the elaboration of methods of vibroacoustic diagnosis for slide bearings than the methods for rolling bearings although slide bearings also constitute weak points in a number of machines. Many reasons account for such a situation, above all the fact that a slide bearing is characterized by low vibroactivity in comparison with that of other points of machines. This means that weak vibroacoustic signals are generated in slide bearings and it is sometimes difficult to isolate them from the noise present, in particular, in sophisticated machinery. The purpose of the examinations carried out on separated bearing points was to evaluate the range of sensitivity of the introduced, relatively simple, estimates of the recorded vibratory signals, to changes of the degree of wear in the conditions of constant loading and increasing period of operation. The obtained results, and the analysis of these results, served as the basis for elaborating the criteria for the evaluation of the technical state of bearing points. Here, such diagnostic estimates were selected that their values determined at the moment of checking, allowed to draw conclusions on the functional characteristics of the bearing points.

1. Introduction

Recent advances in technology, as well as the growing sophistication of technical equipment, set greater requirements for design engineers as regards reliability of functioning, service life, and the determination of a predicted failure-free period of operation.

In order to meet these requirements it is necessary to check often the technical state of the equipment of essential importance, by performing an intermediate examination of residual processes [1, 2]. Because of frequent operational failures of bearing points in the driving system of mine fans, designed for the main ventilation systems, the elaboration of effective methods used for a complex evaluation of the technical state of the bearing points is what matters in the course of their service.

Consequently, a proper schedule of repairs should be set up which would not be limited by the period of operation, but by the technical state of the major bearing points. Therefore, methods aimed at defining the degree of their usability for further operation, expressed in the time period, as well as the continuous improvement of methods become more and more important when checking the technical state of bearings in the fan driving systems [3, 4].

Because of the kinematics of rolling elements, bearings are divided into two types: rolling bearings and slide bearings. Vibrations of slide bearings may be caused by many factors. They depend mainly on the design of the bearing itself, on its size, manufacture accuracy, quality of the co-operating surfaces, assembly conditions, and on the design solution of the whole bearing unit.

The following mistakes are often made already when designing bearing supports for horizontal rotors:

the lack of sufficient stiffness of the body in transverse and in longitudinal direction results in high vibrations of bearing supports;

the frequency values of free vibrations of the bearing body in the transverse and in longitudinal direction, which approximate to rotational speed of the rotor may cause supplementary resonant vibrations of the support;

the asymmetric load of a bearing body accounts for the fact that apart from a vertical force which applies symmetrically a load to this body, a bending moment is also produced, and this causes vibrations of the bearing support.

The existing vibrations, together with processes of usual wear, make bearings lose their functional ability in a shorter time than that anticipated on the basis of the occurrence of wearing processes only. Therefore, there is no possibility of predicting the failure-free service time of a rolling bearing which is incorporated in a complex mechanism.

There are two processes of usual wear which take place in rolling bearings: fatigue wear and abrasive wear of working surfaces. The effect of surface abrasive wear increases with time. Continuous rolling of the rolling elements results in changes of quality of the material only, and surface fatigue itself appears as spalling in the final phase of the service life of the bearing. The process takes the form of an avalanche growth from the moment the first spalling takes place, and this causes directly a breakdown of the bearing. From the observations made so far it is evident that about 20–30% of the bearings in operation lose their functional ability due to excessive clearance, caused both by abrasive wear, and by excessive transverse vibrations [5].

Failures of slide bearings occurring in operating conditions may be divided into the following groups:

failures caused by fatigue of material,

failures caused by increased wear,

failures caused by changing of clearances and fits between a shaft and a bearing shell,

failures caused by improper lubrication.

Of all undesirable effects which are found in slide bearings of machinery, seizures leading to the formation of burrs in a bearing and, consequently, resulting in shaft keying should be considered the most deteriorative.

Some of the deteriorative processes specified, e.g., processes of wear leading to an increase of clearances, and to changes in fits, and particularly the effect of seizure, produce measurable pulses in the vibroacoustic signal of the bearing point.

2. Sources of generation of vibration effects in bearing points

The damage of bearing points occurring under operational conditions is often the consequence of defects of some elements of the bearing and of the simultaneous growth of a number of failures. Under these circumstances the process of damage may be illustrated in spectrograms showing a change in the amplitudes of a number of discrete components. In the case of practical industrial applications it is most convenient to diagnose the state of a bearing taken as a whole, by evaluating the degree of the growth of deterioration processes in the bearing, and the predicted duration of its reliable service. This is a strict, practical approach to the problem, because when a given mechanism becomes unoperational, the operating personnel is not interested in which element of the bearing has been subject to failure, but in the fact that the bearing must be replaced. In the case of a number of exceptionally simple mechanisms, there is a possibility of determining in the amplitude and frequency spectrum some discrete components corresponding to definite, characteristic operational failures [6].

2.1. Influence of operational failures on the level of rolling bearing vibrations

Any type of damage a bearing race influences, to a considerable degree, the vibroacoustic state of rolling bearings. Since the rolling elements roll between surfaces of curvilinear generators, the real motion is very complex. This is even more complex due to the friction and inertia forces, acting on the rolling elements which, depending on the location of certain mechanical damage during the motion, are a source of vibrations, the amplitudes of which come within various frequency bands [7, 8]. Figure 1 shows three typical kinds of bearing damage, producing periodical vibroacoustic pulses:

local damage of a bearing outer ring,
local damage of a bearing inner ring,
damage of one rolling element.

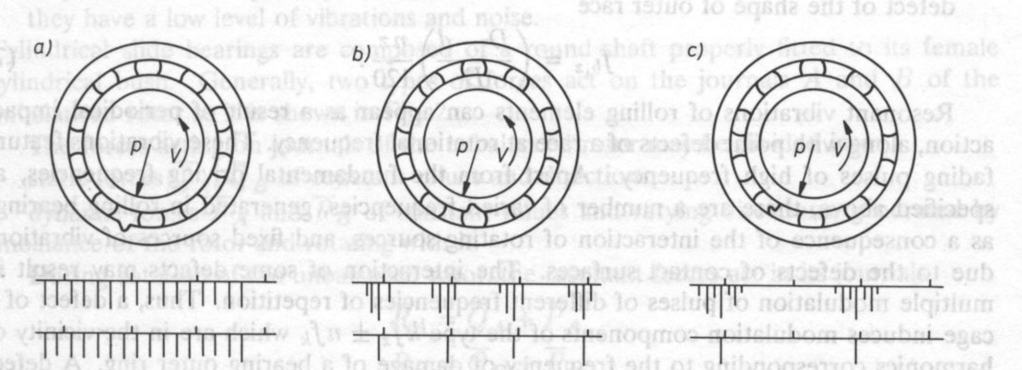

FIG. 1. Typical kinds of bearing failures causing periodical vibroacoustic pulses.

The damage causes pulse functions with repetition frequency:
in the case of damage of a bearing outer ring

$$f_z = \left(\frac{z}{2}\right) f_0 \left(1 - \frac{d}{D}\cos\beta\right) \tag{1}$$

in the case of damage of a bearing inner ring

$$f_n = \left(\frac{z}{2}\right) f_0 \left(1 + \frac{d}{D}\cos\beta\right) \tag{2}$$

in the case of damage of a rolling element

$$f_e = \left(\frac{D}{d}\right) f_0 \left[1 - \left(\frac{d}{D}\cos\beta\right)^2\right] \tag{3}$$

in the case of damage of bearing cage

$$f_k = \frac{f_0}{2}\left(1 - \frac{d}{D}\cos\beta\right) \tag{4}$$

where f_0 — frequency of ring revolution, z — number of rolling elements, β angle of force action pressure angle, d — diameter of rolling element, D–pitch diameter of a cage.

When damage of the structural components of bearings occurs and develops in the course of their service, discrete components appear in the spectrum of a working mechanism, and their frequencies oneconditioned by the kind of damage.

These defects are highly diversified and have the form of waviness of rotating races, increased clearance in a bearing housing, ovality of rolling elements and of rings, roughness, increased clearance in the seats of a cage, slips and others. The fundamental frequencies of excitation, generated as a result of the damage of roller bearings, are conditioned by the following defects:

defect of the shape of rolling elements

$$f_{e,t} = \left(\frac{D+d}{d}\right)\left(\frac{D-d}{d}\right)\frac{n}{30}, \tag{5}$$

defect of the shape of inner race

$$f_{b,w} = \left(\frac{D+d}{D}\right)\frac{nz}{120}, \tag{6}$$

defect of the shape of outer race

$$f_{b,z} = \left(\frac{D-d}{D}\right)\frac{nz}{120}. \tag{7}$$

Resonant vibrations of rolling elements can appear as a result of periodical impact action, along with point defects of a race at rotational frequency. These vibrations feature fading pulses of high frequency. Apart from the fundamental driving frequencies, as specified above, there are a number of varied frequencies, generated in rolling bearings as a consequence of the interaction of rotating sources, and fixed sources of vibrations due to the defects of contact surfaces. The interaction of some defects may result in multiple modulation of pulses of different frequencies of repetition. Thus, a defect of a cage induces modulation components of the type $kf_z \pm nf_k$ which are in the vicinity of harmonics corresponding to the frequency of damage of a bearing outer ring. A defect of a bearing inner ring induces, in turn, the frequencies corresponding to the interaction of this ring, and the bearing outer ring $kf_z \pm nf_w$, and that with the rolling elements $k(f_w - f_h)z$. This is why we may find, in the spectrum of vibration, diversified frequency components of the type:

$$\begin{aligned} &kf_w \pm n(qf_z \pm pf_w) \\ &kf_z \pm n(qf_w \pm pf_k) \end{aligned} \tag{8}$$

where the factors k, n, p, q are integers defining the order of harmonics. Other combinations of driving frequencies are possible, and these cover the rotational frequencies of rotors which, in many cases, reflect the degrees of unbalance of a rotating shaft.

One of the most common defects of assembling rolling bearings is the bevelling of rings.

Bevelling of bearing outer ring generates discrete components, corresponding to the frequencies:

$$f_{z,p} = kf_k z\,. \tag{9}$$

Bevelling of a bearing inner ring generates discrete components of the frequencies

$$f_{w,p} = k(f_0 - f_k)z\,. \tag{10}$$

Defects of rolling elements having the form of wear zones can be also a source of vibrations, the frequency of each of them is lateral to harmonics of double rotating frequency, defined by the expressions

$$f_{\substack{k,z \\ k,w}} = f_k\left[2k\frac{D}{d}\left(1 + \frac{d}{D}\right) \pm 1\right]. \tag{11}$$

Other defects of elements of a rolling bearing may also appear in the frequencies specified (9)–(11) and, in particular, defects of squeeze, scratches and the like.

The complex nature of the vibration spectrum of rolling bearings, occurrence of a great number components of modulation, cause net difficulties when examining the technical state of bearing points.

2.2. *Effect of operational defects on the level of vibrations of slide bearings*

Slide bearings differ essentially from rolling bearings. They feature the following advantages:

it is possible to design them for any loads and rotational speeds,

they have low susceptibility to errors of manufacture and to inaccuracy of assembly,

they have a low level of vibrations and noise.

Cylindrical slide bearings are composed of a round shaft properly fitted to its female cylindrical bush. Generally, two types of forces act on the journals A and B of the unbalanced rotor as it is shown in Fig. 2.

The forces acting on journals of the unbalanced rotor may be divided into:

static forces $\overline{Q}_A$, $\overline{Q}_B$ of constant values and directions;

dynamic forces $\overline{P}_A$ and $\overline{P}_B$ of constant values and varying directions, generated by unbalance of the rotor and rotating with it.

During rotation of the unbalanced rotor, the resultant forces act in its journals

$$\overline{R}_A = \overline{Q}_A + \overline{P}_A$$

$$R_B = \overline{Q}_B + \overline{P}_B\,.$$

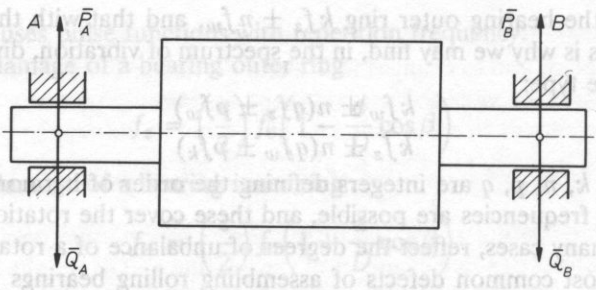

FIG. 2. Forces acting on journals of an unbalances rotor.

The effect of the unbalance on the operation of both bearings may be defined by means of the factors of the unbalance (8)

$$\varepsilon_A = \frac{P_A}{Q_A}, \quad \varepsilon_B = \frac{P_B}{Q_B}$$

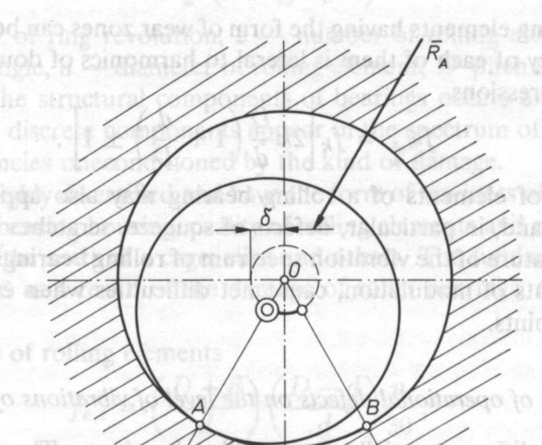

FIG. 3. A trajectory of the centre of a bearing journal in the case of one-sided wear of a bearing bush.

The centre of the journal displaces, effecting an oscillating motion over the circular arc of radius δ, what has been marked in Fig. 3. The arc AB is a geometric locus of points of contact of the journal with the bearing bush. It may be assumed that in the case considered, the centre of the journal effects oscillatory motion with angle amplitude α_0. As a result of these vibrations, wear of the journal occurs over the full circumference, whereas the bearing bush wears only on the arc AB over the length

$$L = \frac{r\pi}{90}\alpha_0$$

where r — radius of the bush chamber, α_0 — angle measured as degree.

As the journal becomes worn, the mass eccentricity of the rotor increases as a result of the force $\overline{R}_A$ increase, and this in turn, causes more intensive wearing of the journal and of the bush. Therefore, wear of the journal and the bush advances with time, and this results in a continuous increase of unbalanced centrifugal forces, and in uninterrupted impairing of the vibroacoustic state of the machine. From the experimental examinations made so far, and referring to the vibration diagnosis of slide bearings, it appears that increased clearance at untight fit of the bearing produces polyharmonic vibrations with frequencies which are a multiple of half frequency of the rotor revolution $k \,.\, f_0/2$.

In the vibration spectrum the half subharmonic of the rotation frequency is often higher than the level of noise disturbances of 20–25 dB [9]. Distinguishing the type of damage in slide bearings may be carried out on the basis of the exact reading of a component of frequency, constituting 42–48% (and not exactly 50%) of the frequency of shaft rotation. The occurrence of this frequency is a typical indication of the instability of journal vibrations in the oil layer, which finally reduces considerably the service life of a slide bearing. The natural frequency of the bearing point $f_{w,w}$ is also of essential importance, and its extraction from the summary signal, by means of a narrow-band filter, will make it possible to obtain a narrow-band signal $x(t)$, the time realizations of which differ very much from each other, as regards the normal state of the bearing, and the damaged one.

The normal state of the bearing is characterized by "stable" time realizations without any evident pulse components, whereas the appearance of seizure causes a sudden increase of amplitude of a narrow-band signal, related to a given type of defect.

The so-called factor of excess [10] is of great significance in the qualitative and quantitative evaluation of changes in the vibratory signal, as conditioned by the progress of deterioration of the slide bearing

$$E_k = \frac{\mu_4}{\delta^4} - 3 \tag{12}$$

where δ is the variance of the signal, μ_4 — central moment of the fourth order.

The specified quantities are determined by the expressions

$$\delta^2 = \lim_{N\to\infty} \frac{1}{N} \sum_{i=1}^{N} [x_i(t_j) - m(t_j)]^2 , \tag{13}$$

$$\mu_4 = m_4 - 4m_3m_1 + 6m_2m_1^3 - 3m_1^4 , \tag{14}$$

where

$$m_p(t_j) = \lim_{N\to\infty} \frac{1}{N} \sum_{i=1}^{N} x_i^p(t_j)$$

and m_p stands for the initial moment of the pth order.

In an ideal case, when a vibrational signal is of normal distribution then $E_k = 0$.

From a series of our examinations it appears that in the normal state of a slide bearing, the value of the excess coefficient averaged over the time set of instantaneous values is $\overline{E}_k = 0.04$, whereas in the case of a defect, for example, such as the burr of a sliding bearing mounted in a reduction gear $\overline{E}_k = 5.0$. It is also evident that the excess coefficient can serve as an important diagnostic indicator, by means of which it is possible to determine both the moment when the initial state is impaired, and the current technical state. Defects of a slide bearing can also be evaluated by means of the parameters characterizing the depth of phase modulation of the forced vibrations of the bearing point. The coefficient of the n-dimensional vector rising may be used as a diagnostic measure of wear of a slide bearing. The components of the vector are the differences of the values

$$S_n(f_i, \Delta f) = \sum_{i=1}^{n} |S_{xi}^{0}(f_i, \Delta f) - S_{xi}(f_i, \Delta f)|, \tag{15}$$

where $S_{xi}^{0}(f_i, \Delta f)$; $S_{xi}(f_i, \Delta f)$ — respective amplitudes of the harmonic components of the function of spectral power density corresponding to the rotational speeds of the bearing in normal state and in worn state.

An analysis of results of experimental examinations carried out in the field of vibroacoustic diagnosis of slide bearings has indicated that the simplest and most reliable algorithm of the diagnosis of progressive burr in contact surfaces of slide bearings are quite analogous to the algorithms of diagnosis of the state of seizure of the gear wheels mesh [11]. Using these algorithms, it is possible to diagnose the process of seizure of slide bearings already in the phase of its formation.

3. Description of the subject of examinations

Fans are machines designed to compress and to force through, gas where the total head produced by fans does not usually exceed 10.000 N/m^2.

According to the function, the mine fans are divided into:

main ventilation fans, also called main fans,

auxiliary fans used, first of all, for local ventilation.

Three-phase asynchronous compact motors are usually applied to drive fans.

Both centrifugal fans and axial-flow fans are used as main mine fans. When comparing centrifugal fans with axial-flow ones, it should he stated that centrifugal fans are simpler, less noisy, easier to make and are thus cheaper than axial-flow fans.

The evaluation of the technical state of rolling bearings and slide bearings of main ventilation fans drive systems of the type WPK-2.1 and WPK-3.3 with shafts of rotational speed of 600 r.p. m. and 500 r.p. m. respectively, were the subject of diagnostic examinations. The diagram of the drive system of the fans of WPK type is shown in Fig. 4. When carrying out vibroacoustic measurements, the location of the measuring points, accordingly situated on the housing of a bearing point, is of great importance. The arrangement of measuring points on the housing of the bearing points 3, 4 is shown in Fig. 5. The measuring points fixed in this way cover both axial vibrations, and radial vibrations as related to the main shaft.

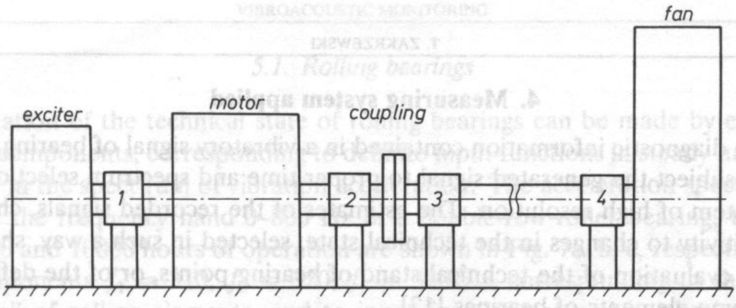

FIG. 4. Kinematic diagram of a driving system of mine fans. Designations: *1*, *2* — slide bearings, *3*, *4* — rolling bearings.

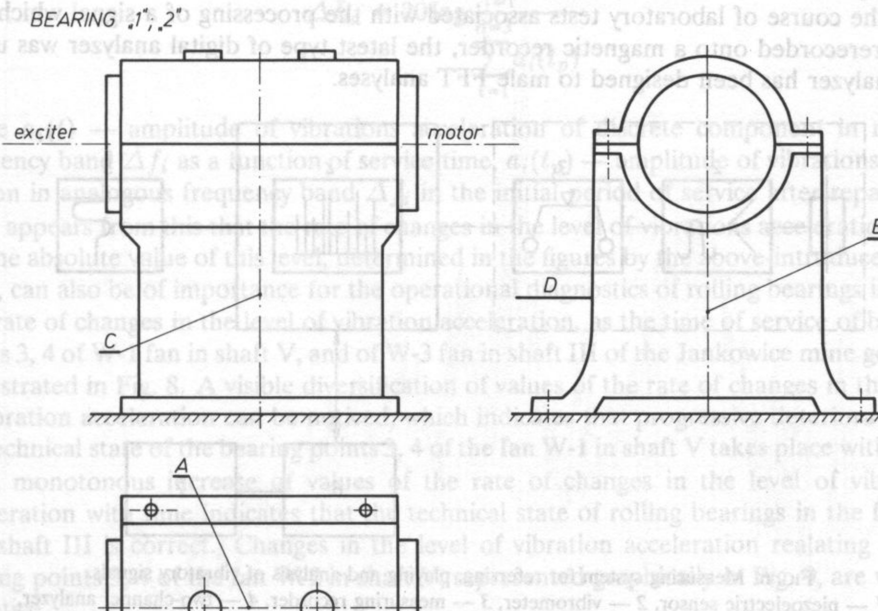

FIG. 5. Arrangement of measuring points on the body of slide bearings.

4. Measuring system applied

To obtain diagnostic information contained in a vibratory signal of bearing points, it is necessary to subject the generated signal to proper time and spectrum selection, using an analyzing system of high resolution. The estimates of the recorded signals, characterized by high sensitivity to changes in the technical state, selected in such a way, should be the basis for the evaluation of the technical stand of bearing points, or of the defined typical failures of some elements of bearings [12].

A laboratory measuring system designed for recording, analyzing and processing of measuring data is shown in Fig. 6. Signals from a piezoelectric sensor (1) were recorded by a measuring recorder (3) via a vibrometer (2); the system constituted a separate whole, and served to record directly the vibration parameters.

The further part, including the recorder, formed a laboratory system which incorporated a two-channel analyzer made by Bruel and Kjaer of the type 2034/4/, a digital recorder of the type 2313/5/. A computer and a plotter formed an autonomous system, necessary to calculate the number of estimates for the evaluation of the dynamic state of bearing points.

In the course of laboratory tests associated with the processing of a signal which has been prerecorded onto a magnetic recorder, the latest type of digital analyzer was used. This analyzer has been designed to male FFT analyses.

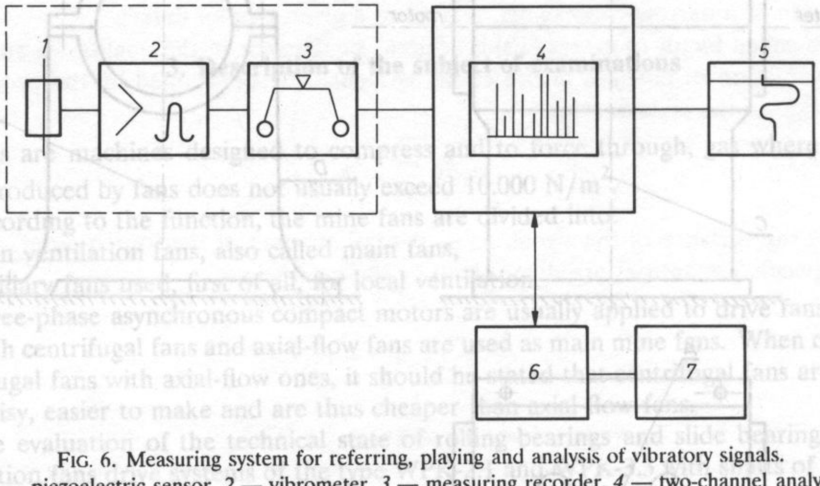

FIG. 6. Measuring system for referring, playing and analysis of vibratory signals.
1 — piezoelectric sensor, *2* — vibrometer, *3* — measuring recorder, *4* — two-channel analyzer, *5* — digital recorder, *6* — computer, *7* — plotter.

5. Measurement results and their analysis

The subject of measurement analysis were discrete estimates, formed on the basis of the distribution of the characteristic frequencies polyharmonic spectrum of the vibration acceleration, corresponding to the occurring most frequently failures of bearing points. The correct elaboration of a method used to investigate the technical state of bearing points consists in the calculation of such vibroacoustic estimates which could be the ground for the evaluation of changes in service characteristics of rolling bearings and of slide bearings as well.

5.1. *Rolling bearings*

An estimation of the technical state of rolling bearings can be made by extraction of the discrete components, corresponding to definite input functions in steady narrow bands of frequency in the spect rum of vibration acceleration. The acceleration spectra or radial vibrations in the frequency hand 0–800 Hz of a double-row roller bearing, type 2234H, after 50, 2000 and 10000 hours of operation are shown in Fig. 7a, b, c, respectively. Three discrete components $f_1 = 310$, $f_2 = 320$, $f_3 = 330Hz$, corresponding to the frequency of "flickering" of rolling elements, and to input functions caused by the interaction of rolling elements and bearing raceways, are predominant. In the case of bearing points comprising rolling bearings (nos 3, 4 according to Fig. 4), the averaged value of vibrations acceleration within the frequency band 300–330 Hz from the interval of minimum values within the range of changes of 20%, has been assumed as a reference level. Taking into account the assumed reference level, a relative evaluation parameter of the technical state of rolling bearing has been introduced

$$\Delta L_a = 20 \log \frac{\sum_{i=1}^{n=3} a_i(t)}{\sum_{i=1}^{n=3} a_i(t_p)} \tag{16}$$

where $a_i(t)$ — amplitude of vibrations acceleration of discrete component in narrow frequency band Δf_i as a function of service time, $a_i(t_p)$ — amplitude of vibrations acceleration in analogous frequency band Δf_i in the initial period of service after repairing.

It appears from this that the rate of changes in the level of vibrations acceleration, and not the absolute value of this level, determined in the figures by the above-introduced estimate, can also be of importance for the operational diagnostics of rolling bearings in fans. The rate of changes in the level of vibration acceleration, as the time of service of bearing points 3, 4 of W-1 fan in shaft V, and of W-3 fan in shaft III of the Jankowice mine goes by, is illustrated in Fig. 8. A visible diversification of values of the rate of changes in the level of vibration acceleration can be noticed, which indicates that progressive deterioration of the technical state of the bearing points 3, 4 of the fan W-1 in shaft V takes place with time.

A monotonous increase of values of the rate of changes in the level of vibration acceleration with time indicates that the technical state of rolling bearings in the fan W-3 in shaft III is correct. Changes in the level of vibration acceleration realating to the bearing points 3, 4 of the fan W-2 in shaft V, represented graphically in Fig. 9, are worthy of notice.

An almost exponential increase in values of the rate of changes in vibration acceleration of the bearing point 4 testifies to the progressive deterioration of rotational dynamics of rolling elements and bearing raceway, leading in immediate failure of the bearing. The repair done on the driving system of thin fan confirmed that fact, proving that the increase in values of the level of vibration acceleration was caused by spallings of the rolling elements, and by transverse cracks of the bearing raceway.

The increase of level of the acceleration rate of $\Delta L_a = 11$ dB (Fig. 9) occurring in the final phase after the service period of 4 thousand hours, was the direct reason for turning off the fan. A failure hazard could be expected at any moment.

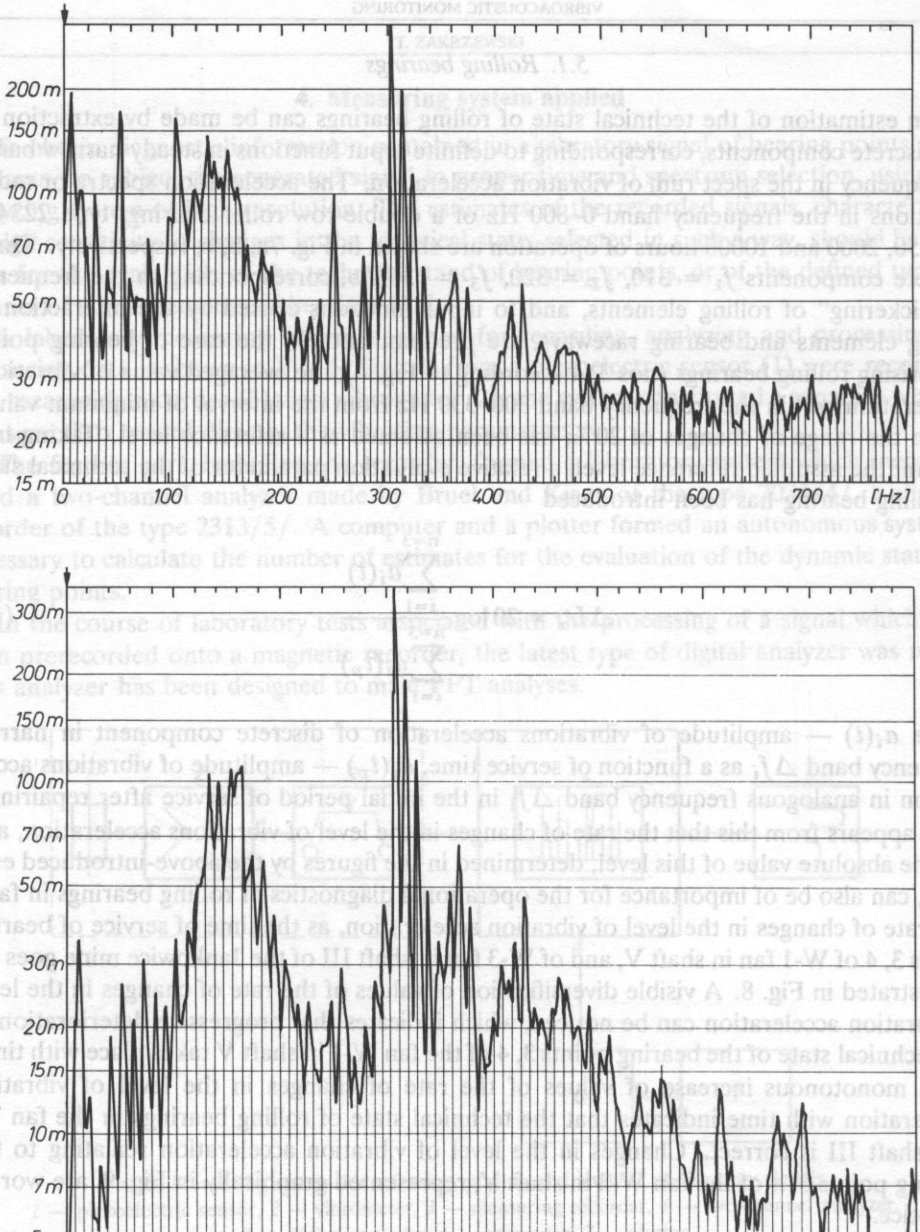

FIG. 7a. Acceleration spectrum of radial vibrations measured for a rolling bearing after a service period of t_E = 50 hours frequency range of analysis 0–800 Hz.

b. Acceleration spectrum of radial vibration measured for a rolling bearing after a service period of t_E = 2000 hours frequency range of analysis 0–800 Hz.

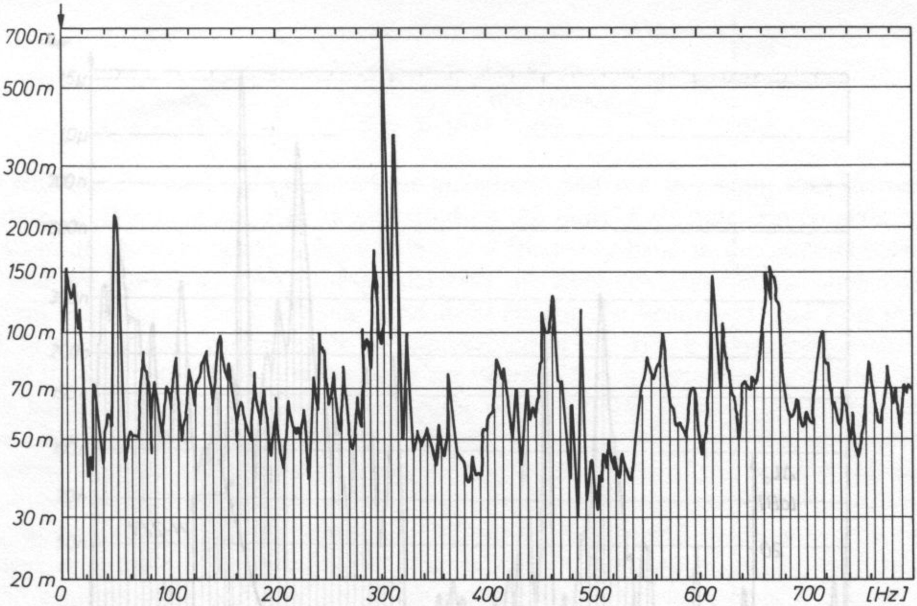

FIG. 7c. Acceleration spectrum of radial vibrations measured for a rolling bearing after a service period of $t_E = 10000$ hours frequency range of analysis 0–800 Hz.

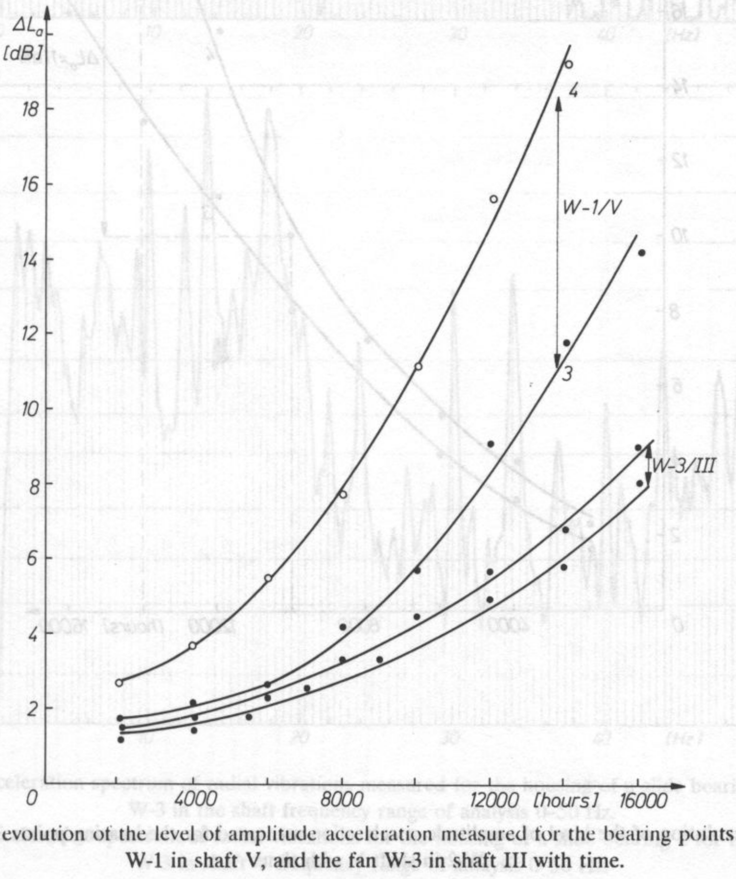

FIG. 8. Time evolution of the level of amplitude acceleration measured for the bearing points 3, 4 of the fan W-1 in shaft V, and the fan W-3 in shaft III with time.

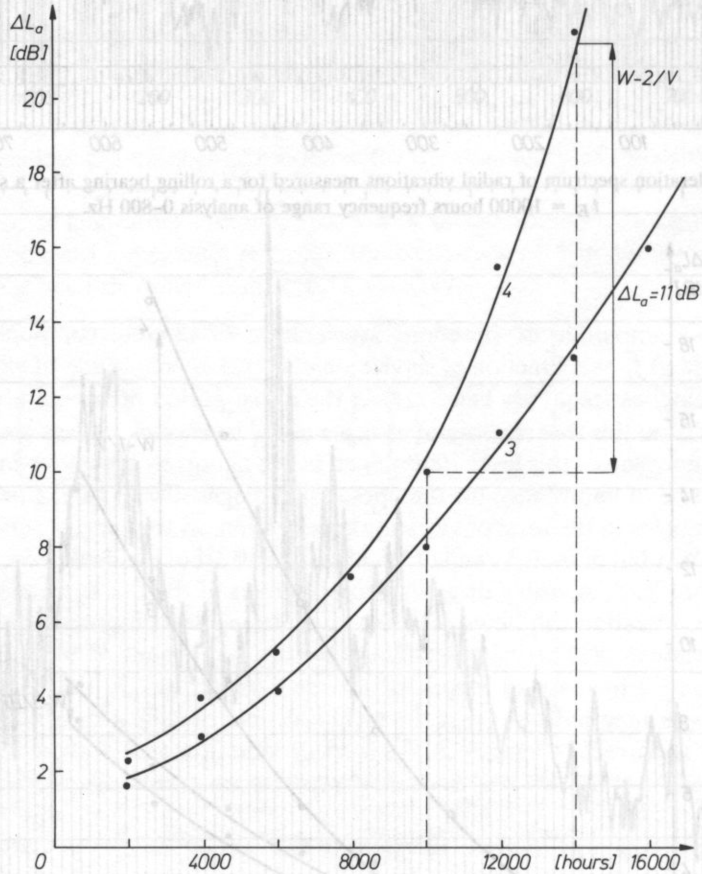

FIG. 9. Time evolution of the level of amplitude acceleration measured for the bearing points 3, 4 of the fan W-2 in shaft V with time.

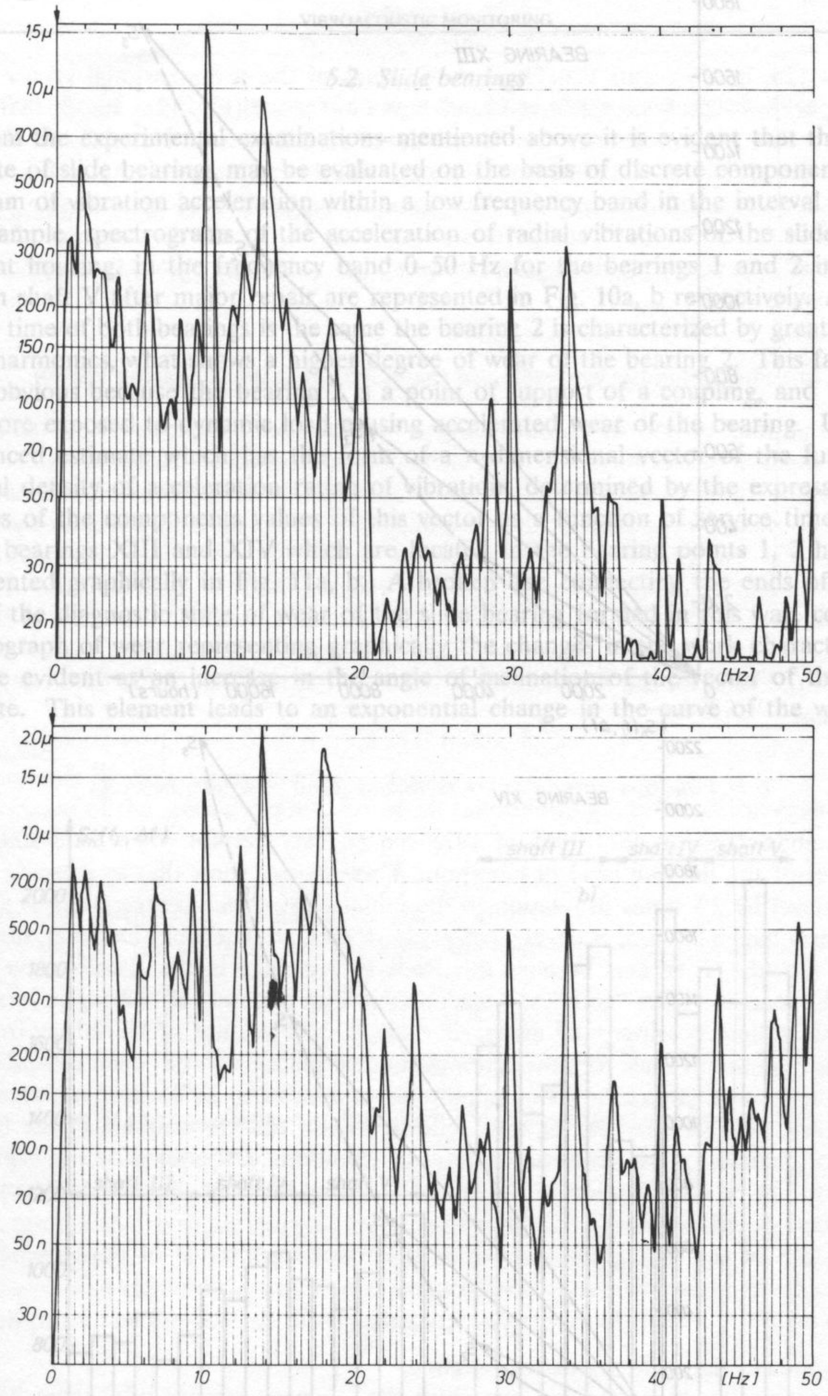

FIG. 10a. Acceleration spectrum of radial vibrations measured for the housing of a slide bearing for the fan W-3 in the shaft frequency range of analysis 0–50 Hz.

b. Acceleration spectrum of radial vibrations for the housing of a slide bearing 2 for the fan W-3 in shaft V frequency range of analysis 0–50 Hz.

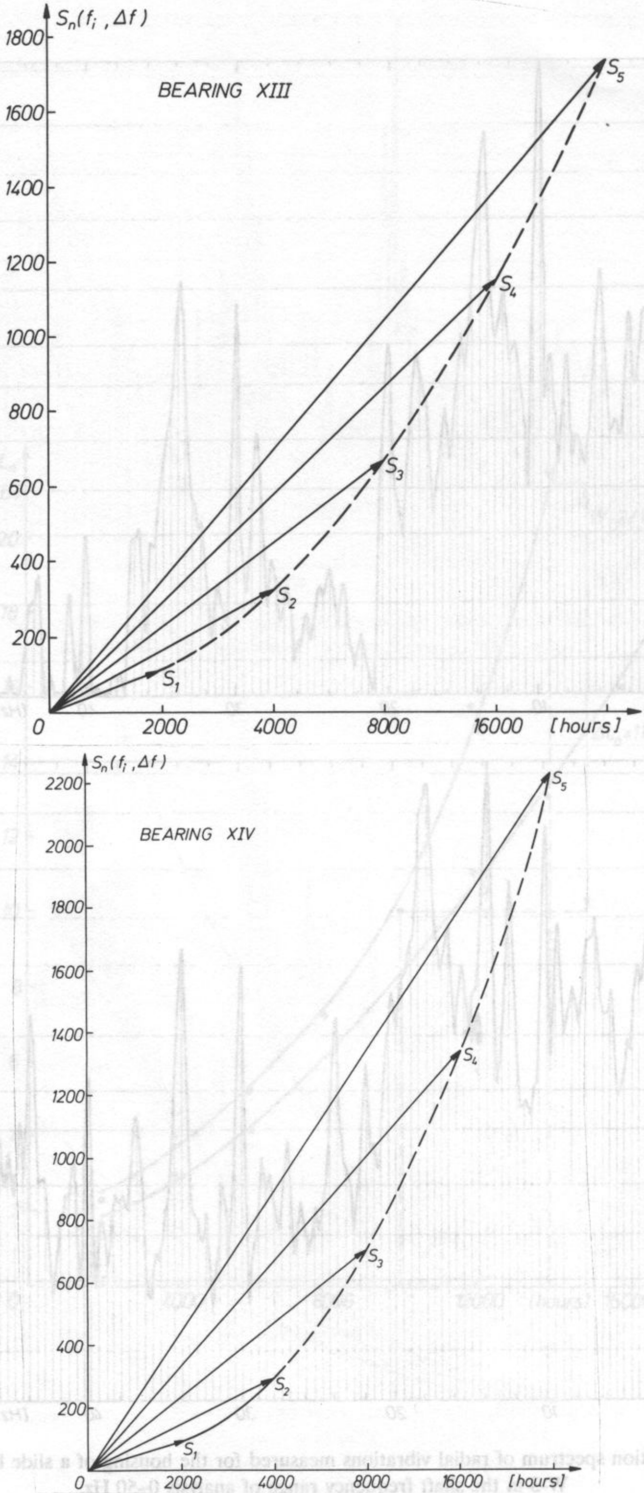

FIG. 11a. Hodograph of wear of the slide bearing XIII.
b. Hodograph of wear of the slide bearing XIV.

5.2. *Slide bearings*

From the experimental examinations mentioned above it is evident that the technical state of slide bearings may be evaluated on the basis of discrete components of the spectrum of vibration acceleration within a low frequency band in the interval 0–50 Hz. For example, spectrograms of the acceleration of radial vibrations of the slide bearing moment housing, in the frequency band 0–50 Hz for the bearings 1 and 2 in the fan W-3, in shaft V after major repair are represented in Fig. 10a, b respectively. Although service time of both bearings is the same the bearing 2 is characterized by greater amplitudes harmonics, what shows a higher degree of wear of the bearing 2. This fact seems to be obvious because the bearing 2 is a point of support of a coupling, and therefore it is more exposed to dynamic load causing accelerated wear of the bearing. Using the introduced estimate which has the form of a n-dimensional vector of the function of spectral density of acceleration rating of vibrations determined by the expression (13), changes of the components values of this vector as a function of service time relating to the bearings XIII and XIV which are located in the bearing points 1, 2 have been represented graphically in Fig. 11a, b. A broken line connecting the ends of the vectors of the diagnostic state of wear of the slide bearing formed in this way, constitutes a hodograph of wear representing graphically the changes in the work character which become evident as an increase in the angle of inclination of the vector of the technical state. This element leads to an exponential change in the curve of the war hodograph.

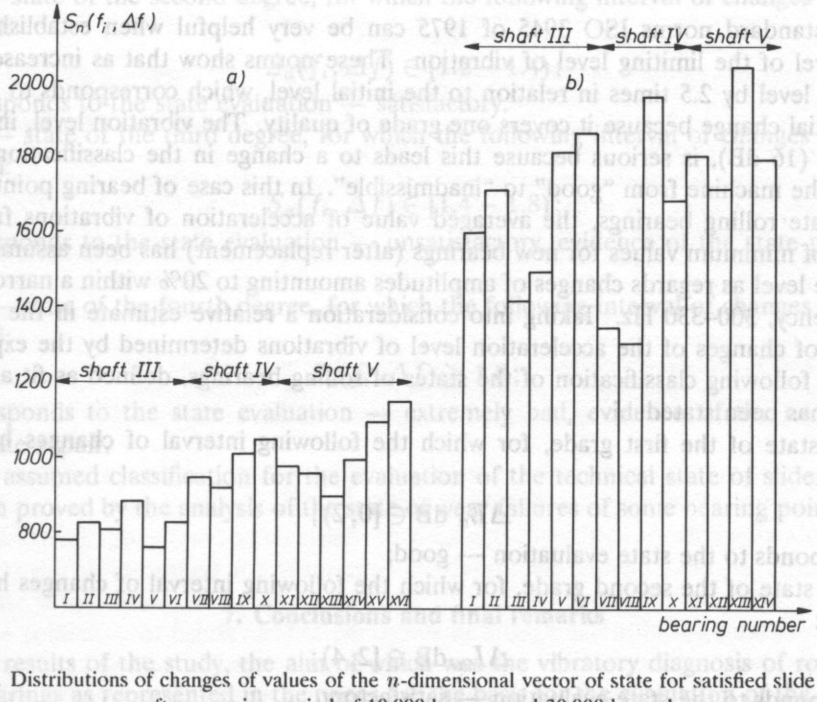

FIG. 12. Distributions of changes of values of the n-dimensional vector of state for satisfied slide bearings after a runing period of 10.000 hours a and 30.000 hours b.

Figure 12a, b represents distributions of values of the n-dimensional vector of the state of specified slide bearings after 10.000 hours of service (Fig. 12a) and 30.000 hours, respectively. A great diversification in the value of this vector, especially in the case of the service duration of 30.000 hours, proves the deterioration of the technical state of some slide bearings. When taking into account the introduced measures to evaluate the technical state of slide bearings, it is possible to distinguish tbe bearings which are characterized by a considerable degree of wear and thus, to present fans from approaching a state nearing failure.

6. Criteria for the evaluation of the technical state of bearing points

From the point of view of vibration diagnostics, the standards determining the criteria for the evaluation of machines and their elements with respect to the vibratory characteristic belong to the most important standards. To make the criteria useful, it is necessary to formulate them on the basis of the experimental data obtained as a result of vibration measurements. These measurements should be carried out under definite conditions of airflow in ventilation ducts. The criteria for the evaluation of the dynamic state of rolling bearings have been elaborated with regard to the numerical values of the rate of changes in the level of vibration acceleration, whereas in the case of slide bearings, distributions of changes in the value of the n-dimensional vector of the technical stand have been taken into account.

6.1. Vibratory criteria of the technical stand of rolling bearings

The standard norms ISO 3945 of 1975 can be very helpful when establishing the initial level of the limiting level of vibration. These norms show that as increase of the vibration level by 2.5 times in relation to the initial level, which corresponds to 8 dB, is an essential change because it covers one grade of quality. The vibration level, increased six times (16 dB), is serious because this leads to a change in the classification of the state of the machine from "good" to "inadmissible". In this case of bearing points which incorporate rolling bearings, the averaged value of acceleration of vibrations from the interval of minimum values for new bearings (after replacement) has been assumed for a reference level as regards changes of amplitudes amounting to 20% within a narrow band of frequency, 300–330 Hz. Taking into consideration a relative estimate in the form of the rate of changes of the acceleration level of vibrations determined by the expression (16), the following classification of the states of rolling bearings, defined as fit and unfit for use, has been stated:

I — state of the first grade, for which the following interval of changes has been assumed:

$$\Delta L_a \text{ dB} \in [0, 2);$$

it corresponds to the state evaluation — good;

II — state of the second grade, for which the following interval of changes has been assumed:

$$\Delta L_a \text{ dB} \in [2, 4);$$

it corresponds to the state evaluation — satisfactory.

III — state of the third grade, for which the following interval of changes has been assumed:

$$\Delta L_a \, \mathrm{dB} \in [4, 6);$$

it corresponds to the state evaluation unsatisfactory, equivalent to the state preceding failure.

IV — state of the fourth degree, for which the following interval of changes has been assumed:

$$\Delta L_a \geq 6 \, \mathrm{dB};$$

it corresponds to the state evaluation — inadmissible, evidence of the necessity of immediate repair.

6.2. Vibratory criteria of the technical stand of slide bearings

On the basis of the introduced n-dimensional vector of the technical stand of slide bearings, and obtained therefrom time distribution of values of this vector, a relative classification of the states of slide bearings, defined as fit and unfit for use, has been stated:

I — state of the first degree, for which the following interval of changes has been assumed

$$S_a(f_i, \Delta f) \in [0.8 - 1.2);$$

it corresponds to the state evaluation — good.

II — state of the second degree, for which the following interval of changes has been assumed:

$$S_a(f_i, \Delta f) \in [1.2 - 1.4);$$

it corresponds to the state evaluation — satisfactory.

III — state of the third degree, for which the following interval of changes has been assumed:

$$S_a(f_i, \Delta f) \in [1.4 - 1.8);$$

it corresponds to the state evaluation — unsatisfactory, evidence of the state preceding failure.

IV — state of the fourth degree, for which the following interval of changes has been assumed:

$$S_a(f_i, \Delta f) \geq 1.8;$$

it corresponds to the state evaluation — extremely bad, evidence of the necessity of immediate repair.

The assumed classification for the evaluation of the technical state of slide bearings has been proved by the analysis of the state of wear failures of some bearing points under repair.

7. Conclusions and final remarks

The results of the study, the aim of which was the vibratory diagnosis of rolling and slide bearings as represented in the paper are the basis for the evaluation of the technical state of drive systems of mine fans in the course of their operation. The introduced estimates of a vibratory signal in the form of the rate of changes of the acceleration level of vibrations, as well as the n-dimensional vector of the state, made it possible to set up. A relative classification of the technical state of both rolling and slide bearings. This classification made it possible to state the points characterized by values exceeding the limite of the introduced estimates, and thus, to identify the technical state of bearing nearing failure. When using the introduced diagnostic measures it is possible to develop and construct an automatic system of control of the technical state which allows for a diversified evaluation of the technical state of bearing points.

The now-applied procedure of operation of many machines in the conditions of continuous running is based on a system of scheduled preventive repairs. It is quite often that a repair of a machine is carried out too early, and it seldom happens that it is effected too late, leading to a failure. The possibility of carrying out a repair, based on the knowledge of the real technical state of the machine is an alternative. This repair, conditioned by the technical state of the machine, shortens considerably the duration of repair, and allows to predict the time and range of the necessary repair. This results in a considerable reduction of operating costs of machines and equipment.

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