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Selecting a vibration diagnostic defect 'Mechanical weakening of a structure'

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Abstract. The article describes the method of analysis of the levels of vibration signals of a support machine rotor in the radial directions and the results of its application for the diagnosis of mechanical loosening of structures of rotating machines for vibration acceleration peak values. Information about correspondence of the degree of a defect 'mechanical weakening of a structure' to the relative level of acceleration in three frequency bands is given in a table.

1. Introduction

It is known [1, 2] that the information obtained by the sensor of the absolute acceleration installed on poles of the rotor makes it possible to monitor the condition of the bearing units of the unit (Normal, Warning, Alarm), to diagnose the defects at different stages during its operation, and to draw conclusions on such defects as 'mechanical weakening of the structure'.

It should be noted that in most cases the defect appears in conjunction with the features of other types of defects on the unit: imbalance, misalignment, bearing defects, loose fit connecting the coupling halves onto the shaft that occurs for wear and violations of mounting technology, etc.

The origin of the mechanical loosening of rotating equipment is very diverse and is associated with a large number of design parameters, installation and operation.

In general all mechanical weakening can be divided into two major groups [2]:

• mechanical weakening like 'backlash', a manufacturing defect, assembly and installation of rotating equipment: free seating of rotating parts is associated with the presence of non-linearity observed in the support bearings, couplings, in adjacent parts in the support mechanisms and foundations;

• mechanical weakening of the 'mechanical weakening of the structure', which is a result of natural wear of the structural elements of the unit, resulting in the sudden disaster and the destruction of the structural elements; cracks and defects in the design and foundation arising from the operation of controlled equipment.

The division of the causes of mechanical loosening of these two groups is fairly conventional. And in fact, in both cases the characteristics of the defect in the spectrum of the vibration signal are about the same. It should be noted that this diagnostic of similarity is observed in the spectra of the vibration signals, in both qualitative and quantitative ratio.

In electrical machines vibrations occur due to deviation of the shape and arrangement of elements, bearings and other reasons. Interaction of brushes with collector plates is additional sources of vibration in the DC motor. Fan and pump vibrations are caused by the vibration of the motor structural elements and aero-hydromechanical noise. A common feature of the considered devices is the concept

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of their operations and changes of the technical condition in the most determined elements with rubbing contacts. It should be borne in mind that the technical condition of these elements depends on the dynamics and vibrations of the entire system.

The greatest effect of the application of vibration analysis and prediction takes place when it is used at all stages of the existence of active products, from manufacturing to maintenance. Diagnostic models are implemented in the form of algorithms that allow improving the processes of manufacturing and test systems and defect detection.

2. Analysis of the system of vibration parameters

The changes taking place in any system are linked to external and internal influences. Mechanical stress is generated by the static and dynamic processes, such as the vibration load, determined by the nature of the working process, the relative movement of the elements friction in kinematic pairs. An external temperature field and thermal energy released by friction also have a significant impact on the operation of the devices. It is possible to cite other examples of influences.

The changes occurring in the system may exceed permissible limits and cause decrease of parametric failures. Faults in the system can be divided into gradual and sudden. Degradation failures arise due to changes in parameters such as due processes of wear and tear; sudden faults are associated with the transition of quantitative to qualitative changes. The consequences of failure are also different. Sudden faults leads to the fact that the product cannot perform its function. The consequence of failure is output parameters of the product from the permissible values.

Parametric failures for precision tribological systems are caused by a gradual process of a changing technical condition. The basis of parametric failures is physical laws, but the diversity of factors at these frequencies is regarded as stochastic. The deep knowledge of regularity causes allows solving the problem of forecasting changes in the parameters, properties and condition of materials better. Modern science is studying patterns of changes in the properties and condition of materials at the following levels [3].

A submicroscopic level allows, based on solid-state physics, exploring the imperfections in crystals, non-equilibrium positions of the atoms in the lattice. These studies are the physical basis of solving the basic problems of the analysis of durability of materials.

Microscopic properties of materials are based on the study of the processes occurring in a small area. Primarily, this refers to the interaction of the surface layers of the friction elements in the presence of lubricants and surfactants. A macroscopic level is based on the change of the integral parameters characterizing elements in general (sizes, elastic characteristics, coefficient of friction, etc.). The analysis of processes at the macroscopic level allows one to generalize the physical laws of changes of the technical condition of parts and to investigate the relationship dynamics of systems with the destruction processes.

In the study of the dynamics of systems of particular importance is the analysis of the natural oscillations. The importance of research of vibration parameters of the systems is determined by their technical condition, as well as the change in the parameters of elements containing rubbing contacts (bearings, current-carrying elements, gears, etc.), depending on their natural oscillations. The forces, which excite vibration and noise of machines, appliances and devices, by their nature may be of mechanical, magnetic and aerodynamic origin. In accordance with this division mechanical, magnetic and aerodynamic vibration is produced. The source of mechanical vibration and noise are unbalanced rotating or oscillating parts, bearings, gears, current-carrying nodes, such as a brush-collector, and other elements. Unbalance elements cause oscillations with frequencies that are multiples of the frequency of rotation or stationary traffic. The amplitude of the driving forces is proportional to the square of the frequency and imbalance.

The main reasons for support oscillations are cycled stiffness in rotation or other stationary motion and geometric imperfections of their contacts and the mating surfaces. Parameters of the driving forces depend on the working conditions, dimensions and elements of technological errors. Magnetic vibration and noise occur due to periodic changes in the electromagnetic forces in the air gap of the

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electromagnetic system. The sources of vibration and noise of the aerodynamic origin are the moving parts of machinery. Vibrations from these factors interact with each other, and as a result vibrations occur in a wide range of frequencies.

More effective methods that provide control of a functioning and allow one to define proper disturbance, are the methods of technical diagnostics, the basic content of which is to examine the rationale and methods of indirect measurements of hidden parameters of devices affecting the nature of its functional behavior. The existing diagnostic methods are generally based on the analysis of the main operating parameters of the object. The object model and signals indicating of the object status as an element of the system does not fully reflect the internal processes and their changes, but only some have conclude a formal link between the input and output object. In this model, it is impossible to diagnose the unobservable internal processes therefore a special diagnostic model of the technical condition of the system and a change of the parameters of the object is required. It is necessary to choose the best information signals to establish the relationship between object as an element of the system and the model for diagnosis.

The main properties of the object as a member of the system are characterized by operator *L*, which connects input and output signals U_1(t) and U_2(t), and also accounts for dependence U_2(t) of disturbing factor *U*(t), generated by its own internal processes. The quality of operation depends not only on the design parameters, but also on perturbation *U*(t), which changes over time and can cause failure of the parametric system.

Changing the technical condition can be controlled by changing the natural oscillations of *z*(t) (vibration), which are generated by internal processes. In this case, as a rule studies are limited by the establishment of diagnostic models. The disadvantage of this approach is the lack of communication between *U*(t) and *z*(t).

It is necessary to create such a model for the diagnosis, which makes it possible to record the changes in internal processes and also establishes a link between the disturbing factor of *U*(t) and natural oscillations *z*(t).

As the block diagram model is illustrated in figure 1, the main parameter is r, which binds *U*(t) and *z*(t).Parameter r is defined by geometric characteristics deviations from the nominal values, technological uncertainties and other fluctuations. The relationship between *U*(t) and *r*, *z*(t) is set by operator *T*, and between r and *z*(t) – by *W*.

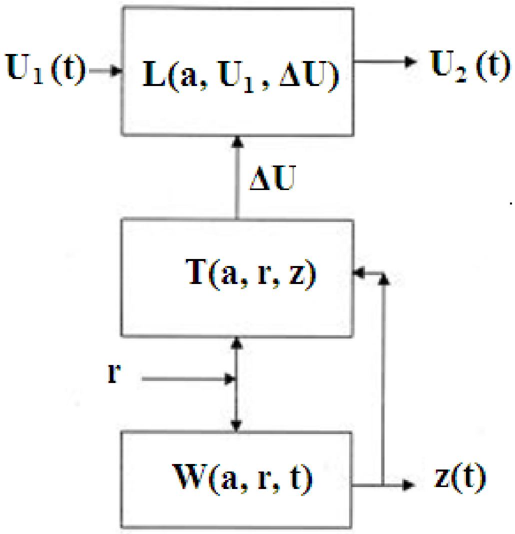

Figure1. A block diagram of a diagnostic device model.

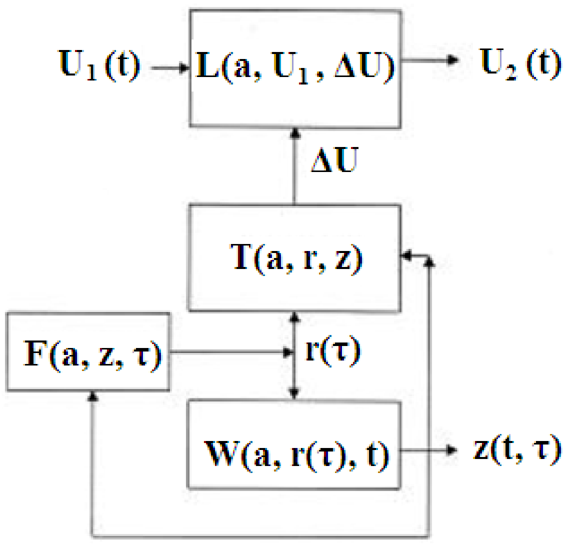

Figure 2. A block diagram of a general diagnostic device model.

This model is not complete, as parameter r operates in long-term system transient. Changing of r is not only aging, but also the dynamic vibration effects; it is slow compared to the vibration and the fluctuation of the main operational indicators; it is a vibrorheological process. The relationship between the change and vibration is set by the operator F (Figure 2).

Thus, a generalized model has two types of characteristic processes: a rapid (time t) – vibration and fluctuation in performance and a slow of parameters. Fast processes determine the quality of functioning at a given moment of time, and slow – parametric reliability of the system. Selection of diagnostic signal must be carried out so as to be sufficiently informative for evaluation of the vector r, its variations, and hence to estimate z(t) and U(t).

Defects mechanisms eliciting vibrodiagnostic control techniques are presented in table 1.

Table 1. Defects detected by the mechanisms of vibrodiagnostic control methods.

Defect	Explanation
Imbalance	Rotor, impeller, the system 'rotor motor – coupling – the impeller', etc.
Misalignment of shaft	Fracture and mixing of shafts. The curved shaft.
No rigid attachment	Cracks in the frame or body. Loosening of the foundation or mounting bolts. 'Soft foot'.
Uneven defects of motor	Air gap between the stator and the rotor. Damage of the stator windings or insulation. The eccentricity of the rotor. Broken or loose bars in the cage. Looseness stator windings. Voltage unbalance in phases, etc.
Defects of drive sleeve	Weakening of landing on the shaft. Uneven torque of transmission engagement elements.
Defects of compressor, pump, fan cavitation	Surging. Stall. Faulty blades, etc.
Defects of gear	Wear gear tooth. Broken tooth. Shock engagement. Shaft misalignment.
Defects of belt wear.	Easing tension. Misalignment of the pulleys. The eccentricity of the pulley. Resonance of belt etc.
Defects of plain bearings	Wear. Elliptical shaft neck. Forcing the oil film, oscillations, etc.
Defects of rolling bearings	Defects of the rolling elements, the separator, and the inner and outer rings. All defects in manufacture, installation and wear. As a lubricant.

There are dynamic forces in each machine. These forces are a source of the noise and vibration damage mechanism nodes. Vibrational forces acting in units of machines may have very different nature and a variety of frequency components.

The following variants should be highlighted from the mechanical forces of nature[4]:

• centrifugal forces determined by unbalance of rotating parts;

• kinematic forces determined, as a rule, by smooth irregularities of interacting surfaces and above all the friction surfaces;

• parametric forces defined by variable component stiffness rotating assemblies or supports rotation;

• friction force that cannot always be regarded as mechanical, but almost always they are the result of cumulative action of multiple snatching with deformation (elastic) contact of asperities on the surfaces of friction;

• type of impact forces arising from the interaction of individual elements of friction, accompanied by their elastic deformation.

The following variants should be highlighted because of the forces of the electromagnetic origin in electric mechanisms:

• magnetic forces determine the change of magnetic energy in a certain limited space, usually limited by the length of the site of the air gap;

• electrodynamics forces determine the interaction of the magnetic field with an electric current;

• magnetostrictive forces, determine the effect of magnetostriction, i.e. a change in linear dimensions of the magnetic material by the magnetic field.

From the aerodynamic forces the origin should be highlighted: the lifting force, i.e. pressure force on the body, for example, impeller blades moving in a stream or flow are; frictional force at the boundary of the flow and stationary parts of the machine (the inner wall of the pipeline and the like mechanism); pressure pulsations in the flow, determined by its turbulence. The origin of hydrodynamic forces is substantially of the same nature as in the gas atmosphere, but it adds to the pressure pulsations due to cavitation, which in certain conditions can arise in the liquid flow.

The temporary implementation of vibration carries a large amount of information that is invisible to the naked eye. Some of this information may consist of very weak components, the value of which is less than even the thickness of the line graph. To circumvent the limitations of the analysis in the time domain, in practice one usually applies frequency or spectral analysis of the vibration signal. When selecting the vibration frequency domain used for diagnostics of machines and equipment it must respond to different properties of different vibration frequencies. At high frequencies, the vibration becomes of the wave nature of the spectrum of a little line, little (at first glance) information, but small enough power to drive the measured vibration.

The dynamic forces excite vibrations in machines, either directly or forces excite the noise, and the noise – vibration of the body. Vibration, depending on the nature of the forces may be either deterministic (most periodical) or random.

To maximize the dynamic range of the measurement parameter onemust use the vibration (displacement, velocity or acceleration), which provides the most uniform spectrum. Usually this parameter is the vibration velocity, but in general, it depends on the type of machine [5].

In the analysis of the spectrum, there are three groups of vibration components: harmonics, asynchronous components and sub-harmonics. Harmonics are peaks at frequencies that are multiples of the frequency of the action (rotation speed) of a technical complex: they can be used to draw conclusions about the imbalance, misalignment or loose connections [5]. Nonsynchronous components occur at frequencies of no multiple frequency cycle: the analysis of the components of this group allows one to detect defects, such as elements of bearings and belts. Subharmonic components are below the rate of rotation. They can be caused by such phenomena as whirlwinds in the oil bearing wedge, increased friction between elements, defects in the belt drive, the excessive weakening of the compounds or a knock on the car.

The most important are normally one or more harmonics of the fundamental frequencies. The reason is the presence of harmonic distortion of the wave oscillations with the fundamental and non-sinusoidal periodic motion. Harmonic frequencies may coincide with resonant frequencies of certain parts or elements of the designs and significantly increase and form a major source of acoustic noise or dynamic forces imparted by other mechanical elements.

A very popular method of analyzing the vibration activity of the complex in the frequency domain, is the method of the envelope. The spectrum of the vibration signal generated by nonlinear vibrational forces loses a considerable amount of information: it is quite difficult to recognize an amplitude located close in frequency. Moreover, even at a high frequency resolution it is difficult to detect modulation of the strongest components of the vibration at frequencies close to the frequency of rotation. The spectrum of the envelope allows one to select random non-periodic signal components (shock pulses) and therefore control the sliding surface.

3. Results and discussion

From a temporary weakening of the vibration signal when the mechanical contains a significant non-stationary, non-periodic component, it is possible to conclude that even from the turn to turn the waveform can vary. It is quite difficult to diagnose because it is often fickle, there are occasional peaks and shifts.

There is a large number of peaks from the collisions, which have, at first glance, a chaotic character (Figure 3). This form of temporary signal usually accompanies all the defects of the 'backlash' or 'mechanical weakening of the structure' [1].

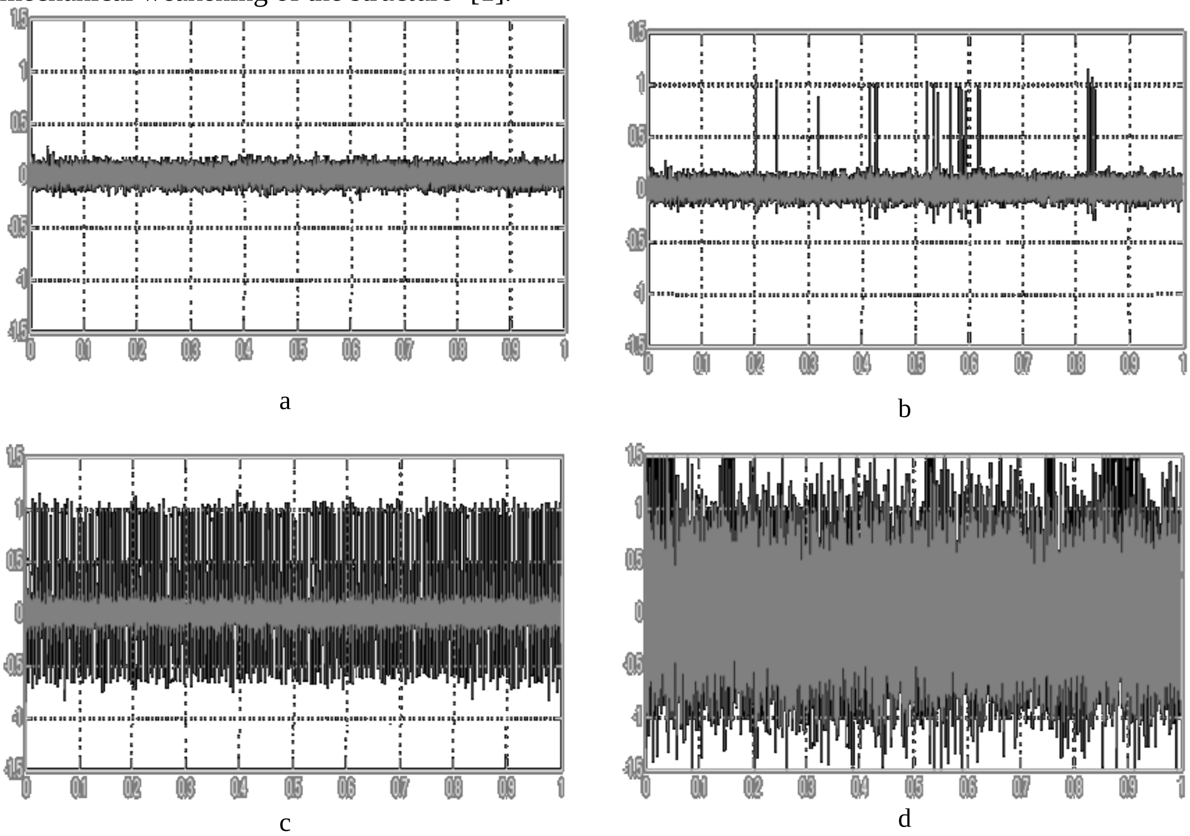

Figure 3. A time signal corresponding to: a – the absence of the defect, b – the emergence of the defect, c – the progression of the defect, d – fixing of the bug.

In order to improve the reliability of defect determination 'Mechanical design facilitation' by the search parameter and the vibration frequency range, sensitive to changes in the stiffness of the rotor poles, we can propose an additional diagnostic feature.

Generally, based on rotor speed *fr*, there are three frequency bands:

• low-frequency (LF) – from 0 to *fr*;
• midrange (MF) – fo *b* from up to 10 • fr;
• radio frequency (RF) – 10 • *fr* to 10 kHz.

Diagnosis Symptoms of separate units of aggregates are often determined by the analysis of the parameters that were previously under the control operations are not regulated and considered to be secondary.

To select the setting of the vibration produced as a diagnostic measure let us define vibration displacement S, vibration velocity V and acceleration W on the two pillars of the rotor with variable stiffness in three frequency bands (LF, MF and HF), in the standard views magnitude, RMS and Peak, respectively. Thus the fob rotor speed was 50 Hz, which corresponds to the most common speed electric drive motor of 3000 rev/min. The analysis of the obtained vibration parameters values is produced by normalization of up to 1: a parameter in each frequency band is divided into maximum (Table 2).

Reducing the stiffness of support reduces the relative value of acceleration in the low and mid ranges (Table 3). The relative values of vibration velocity and vibration displacement are less informative. Diagnostic indication must meet several requirements: uniqueness, accessibility, ease of measurement, information, and manufacturability. To do this, check that the identification of signs of change will maintain the integrity of the rotor imbalance.

Table 2. Sensitivity of vibration parameters to change of the stiffness of supports(I – the first pillar of the rotor, II – the second pillar of the rotor).

Frequency range, Hz	Stiffness of pillar	S_I, Swipe	S_{II}, Swipe	V_I, RMS	V_{II}, RMS	W_I, Peac	W_{II}, Peac
2…50	maximum	1.0	1.0	1.0	1.0	0.1	0.1
50…500		0.1	0.1	0.3	0.3	0.7	0.7
500…10000		0.0	0.0	0.0	0.0	1.0	1.0
2…50	middle	1.0	1.0	1.0	1.0	0.1	0.1
50…500		0.1	0.1	0.3	0.3	0.6	0.6
500...10000		0.0	0.0	0.1	0.1	1.0	1.0
2…50	minimum	1.0	1.0	1.0	1.0	0.1	0.1
50…500		0.1	0.1	0.3	0.3	0.5	0.5
500…10000		0.0	0.0	0.1	0.1	1.0	1.0

To do this, at a constant frequency of the rotor back (fr = 50 Hz) produced by acceleration measurements W for balanced and unbalanced rotor on two pillars with variable stiffness in three frequency bands (LF, MF and HF). The analysis of the obtained vibration parameters values is produced by normalization of up to 1: a parameter in each frequency band is divided into maximum (Table 3). Balancing of the rotor acceleration value is reduced by an average of 15 %.

Table 3. Sensitivity of vibration parameters to change the stiffness of supports (I – the first pillar of the rotor, II – the second pillar of the rotor).

Frequency range, Hz	Stiffness of pillar	Unbalancing		Balancing	
		W_I, Peac	W_{II}, Peac	W_I, Peac	W_{II}, Peac
2…50	maximum	0.1	0.1	0.1	0.1
50…500		0.7	0.7	0.7	0.7
500…10000		1.0	1.0	1.0	1.0
2…50	middle	0.1	0.1	0.1	0.1
50…500		0.6	0.6	0.6	0.6
500...10000		1.0	1.0	1.0	1.0
2…50	minimum	0.1	0.1	0.1	0.1
50…500		0.5	0.5	0.5	0.5
500…10000		1.0	1.0	1.0	1.0

It has been found that the magnitude of the imbalance does not affect the relative values of the acceleration (Table 3).

Conclusion

1. It is advisable to carry out measurements in the low, mid and high frequency bands.

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2. It has been empirically found that the reduction in rigidity leads to a reduction of the relative values of acceleration in LF and MF bands.

3. The magnitude of rotor imbalance does not affect the relative values of acceleration on all frequency bands.

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