Bearing diagnostics of slow speed industrial rotating equipment by shock pulses

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Abstract. Fifty years have passed since the world's first patent for a widely used method of the rolling element bearings condition monitoring by ultrasonic pulse vibration (SPM-method) was registered. The method is widely used in the power industry and in other industries. During these years, several technical realizations of this method were implemented. These methods are based both on the form of the recorded single pulses analysis and on the spectral analysis of the envelope of periodic pulses. But in all implementations, there remains a problem of establishing the relationship of controlled vibration parameters with the defect severity. The paper analyses the possibility of detecting and assessing the risk of rolling element bearing defects in low-speed rotating units by vibration excited by shock pulses, the sources of which can be both the bearings and the tooth wheels. In the production of electricity such units include rotary heat exchange equipment, among the major consumers - rotary steel furnaces for different purposes. The proposed solution - a synchronous analysis of the pulse vibration of all bearings of low-speed units at medium, high and ultrasonic frequencies.

1. Introduction

The widely implemented shock pulse method uses the properties of two different physical processes that are the sources of ultrasonic vibration of the working bearing - elastic collision between the loaded elements[1,2], and abrupt changes in the structure of the loaded metal, accompanied by acoustic emission [3,4]. During the elastic collision, the degree of danger of the bearing condition is determined by the magnitude of the transmitted force pulse, which excites vibration in different frequency ranges [5,6]. When the acoustic emission appears, then the bearing condition is determined by the number of detected ultrasonic pulses [7].

Besides the ultrasonic shock pulse method [2], the most sensitive to the appearance of lubricating layer ruptures in the rotating bearing, the assessment of its condition can be made by three vibration methods.

The basis of the first method is the analysis of bearing components of vibration detected in the auto spectra of vibration measured at the control points of the diagnostic object, most often on the housing of rotation units [8-10]. This is the most accurate method of detection and evaluation of the developed defects.

The second, and most easily implemented method is the control of moments distribution of the vibration signal (acceleration) measured at bearing locations of the object. The fourth central moment of the vibration acceleration signal distribution changes significantly when the components of the
pulse origin appear in the signal [11], and the maximum of the spectral density of the measured vibration acceleration falls on the average frequencies, where the shocks are most strongly manifested in the developed defects of the contacting mechanical units, i.e., for the registration of shocks, preliminary filtering of the signal is not even required. But this method is sensitive to not so dangerous defects of gears, which often does not allow to specify the place of occurrence and the degree of danger of the defect of a particular bearing in a reducer.

The third method is based on the analysis of the magnitude and properties of the friction forces by the random vibration excited by them. The main method of analysis is spectral, which allows to identify periodic changes in the power of friction forces in the testing bearing and to separate them with periodic shock pulses on the spectrum of the envelope vibration measured at user-selected medium or high frequencies [11-14]. The method allows detecting and separating all kinds of defects of bearings and gears at an early stage of their development, but it reduces its selectivity after the development of defects to the stage of increased danger.

In the bearings with rotation speed, sufficient for a continuous lubricant layer formation between the loaded rolling surfaces, the levels of the bearing vibration components in autospectra are controlled. Ultrasonic pulse vibration is measured only to monitor the condition of the lubricant, in which, as it deteriorates, appear the inclusions of the wear products of friction surfaces, which leads to the ruptures in the lubrication layer accompanied by short pulses of forces. The shorter are the pulses, the higher is the vibration frequency by which they can be detected. When the loaded rolling surfaces move along an extended defect, there are many such ruptures, and some experts, as in the registration of acoustic emission pulses, estimate the risk of the defect by the number of detected ultrasonic pulses per time unit [2, 6, 7]. In fact, when the surface area of the defect increases, the vibration excited by the main force pulse is determined by the interaction time of the defects, and, the longer is its duration, the lower are the frequencies that are manifested in vibration. Therefore, for a more accurate assessment of the shock pulses danger, its analysis is often carried out by the third of the mentioned methods, by the spectra of the envelope vibration at lower frequencies, the optimal region of which is selected, mainly by the bearing rotational speed. The method of envelope spectrum vibration analysis provides additional opportunities to analyze the properties of friction forces, by which also the condition of the sliding friction elements can be estimated, and in the rolling element bearing such an element is a separator [13].

The difficulties in the bearing diagnostics begin as its speed decreases [15]. The main problem appears because of the complexity of the measurement and analysis of vibration autospectra of low-speed machines and mechanisms, starting at rotational speed below 120 rpm. The vibration at frequencies below 2 Hz, includes, as a rule, many extraneous components, even excited by a remotely located machines and mechanisms. And in the vibration of, for example, reducers, when there is inaccuracy of production or wear of gears, very often appear subharmonic components at frequencies several times lower than the frequencies of rotation speed of low-speed stage [8,11].

A compulsory refusal of monitoring the bearing vibrations at frequencies below about 3-5 Hz does not lead to the decrease of the bearing diagnostic quality, because there remains the possibility to measure and analyze the envelope of vibration taking into account the properties of the friction forces at the mid-frequencies. For example, Figure 1 shows the envelope spectrum of the medium-frequency vibration of the bearing unit of the gearbox output stage with 10 rpm rotation speed with high frequency resolution, which requires stable operation of the reducer for a long time. The vibration envelope spectrum enables to determine the defect type, in this case it is a wear (skew) of the output gear wheel of the mill reducer, separating the defects of mechanical gears and their bearings.
But if the bearing of a rotary heat exchanger, that is used in the production of electricity, rotates at a speed of less than 1 rev/min, the problem of its diagnosis cannot be solved either by the auto spectra of vibration nor by its envelope spectra. And to control the bearing condition of large equipment, such as converter which does not make a full turn and with a load often depending on the angle of its rotation, is not possible even by the pulse vibration at ultrasonic frequencies, because at such low speeds a continuous oil film is not formed and a pulse ultrasonic vibration in magnitude due to elastic and inelastic shocks occurs in both defective and defect-free bearings

2. Experimental results
Studies on the analysis of the possibility of assessing the condition of such bearings were carried out on the largest of the produced units with rolling bearings – converters used in metallurgical equipment for steel melting. The structure of the converter with measurement points is shown in Figure 2.

The main mode of the converter movement is its "swing" with a change of direction, but there is also a technological mode with its "scrolling" at a constant speed of rotation with several (up to 4) full revolutions of the converter at 1 rpm speed of rotation.

The first stage of work was to check the possibility of diagnostics by using the envelope spectrum of vibration measured in the scrolling mode. Figure 3 shows the envelope spectrum of the vibration previously extracted in the frequency range of 1.6-6.4 kHz.
Figure 3. The envelope spectrum of high-frequency vibration of a converter rotation support measured during its scrolling at a constant speed of rotation and its fragment up to 1 Hz. Measurement time lasts for 4 min.

As follows from the given spectrum, it is impossible to determine the frequency of shocks and estimate their magnitude by the envelope spectrum of vibration, and, accordingly, by its autospectrum. The reason is that the vibration signal, see Figure 4, consists of a pulsed component in which there is no clearly expressed periodicity, and the magnitude of the pulse due to the dependence of the load and wear of the bearing on the angle of rotation of the shaft varies randomly over a wide range.

Figure 4. The shape of the envelope of the pulse vibration, measured in the supports of the converter at a time interval during its angular displacement of 260 degrees.

Thus, to assess the condition of the converter low-speed shaft bearings by vibration of the rotation supports, only one possibility can be used – the analysis of the pulse vibration, the maximum contribution to which is generated by the medium-frequency components. It is only necessary to determine whether it is possible to use the basic mode of “swing”, in which frequency ranges to analyze the pulse vibration, and how to distinguish the pulses formed in different bearings and gears.

Since shock pulses in the measurement points on all bearing supports occur rarely and have time to decay by the time of the next one, the results of the analysis of the damped vibration from one of them are presented below. So, Figure 5 shows the synchronous vibration, from the same momentum in the three measurement points. At first glance, the source of this shock pulse is the shaft of the left gear.
Figure 5. The form of a signal excited by the same vibration pulse on the inner support of the rotating gear, near and far supports of the rotating converter.

At the same time, the simplest analysis shows that the components at the natural frequencies of different rotation supports are distinguished in the signal. The spectra of damped oscillations in the considered supports constructed on the interval of 0.06 sec. from the front of the shock pulse are shown in Figure 6.

Figure 6. The natural oscillations spectrum measured on different bearings of the reducer, excited by a specific shock pulse in a particular rotation support.

The spectra are observed both - common to all supports natural oscillation frequencies (frequencies in the frequency range of 3000 Hz and 3500 Hz), and individual for one support (2000 Hz), which are most strongly manifested only in the support, in which there was a specific shock pulse. This allows to localize the place of the shock pulse formation by vibration at natural frequencies. According to the given spectra, the source of the shock is the left support of the converter rotation (point №3).

In addition to the middle frequencies range, the shock pulse excites the natural oscillations of the supports at both high and ultrasonic frequencies. As not only in the transmission of vibration, but also in the transmission of a force pulse its components at different frequencies have different attenuation, the vibration, excited by it at high frequencies in the absence of high-q resonances, should be higher in the support where this pulse is excited.

So, Figure 7 shows the ultrasonic component of damped oscillations excited by the same pulse in the same rotation supports.
As seen on the above figure, the ultrasonic pulse is more powerful on the near support of the converter, which is the source of this shock pulse. It would seem that the above signals only confirm that the SPM-method of ultrasonic pulse vibration signal control and its comparison with absolute and relative thresholds is effective in identifying the fault bearing and assessing the degree of its danger, including in low-speed bearings, but this conclusion is not true.

Firstly, when analyzing the low-speed gear stages bearings ultrasonic vibration in time, as a rule, a large number of synchronously appearing pulses are detected, which have neither medium-frequency nor low-frequency components. This mode of ultrasonic vibration excitation is a characteristic of a very short pulse, and as the pulse power is proportional to the square of the rolling surface velocity and in low-speed bearings it is close to zero, it is impossible to estimate the bearing condition in the absence of pulse vibration components at medium and low frequencies. Secondly, experimental studies contradict the standard representation of diagnostic equivalence of one extended defect and a large group of small defects located within the boundaries of such extended defect. That is why the estimation of the bearing condition of an extended defect by the total power of small pulses of ultrasonic vibration excited by a group of small defects often leads to a mistake.

Therefore, the optimal way to analyze the rolling element bearings condition of low-speed converter drive is the analysis of pulse vibration at medium frequencies, as the mass and size parameters of low-speed converter drive are the highest, and the pulse vibration of high-speed drive without significant defects do not prevent its analysis. The identification of the defective unit - the source of pulse vibration can be carried out by synchronously measured ultrasonic vibration. The main problem of synchronous analysis of mid-frequency and ultrasonic vibration can be ultrasonic background due to the lubrication breaks of low-speed bearings, masking simultaneously controlled pulse vibration in different frequency bands.

Thus, in reducers with low-speed stages, it becomes possible for each shock pulse, simultaneously manifested in all rotation supports, to determine the place of its action either by the magnitude of the response in the ultrasonic vibration in different rotation supports, or by the individual frequencies of the excited resonances at medium frequencies.

Further, all synchronously recorded shock pulses are sorted into groups at the place of origin, for which the vibration of all bearing units is measured synchronously, and the pulses of each group are compared with the state thresholds individually determined for each rotation support.

3. Conclusions
The concrete conclusions are as follows:

- Monitoring the bearing condition of low-speed equipment, including those that work with variable load, based on the spectral analysis of vibration is often ineffective.

![Figure 7. The form of the ultrasonic (25-50 kHz) component of the vibration signal excited by the same pulse in the inner support of the rotating gear, supports near and far to the converter.](image)
- At bearing speeds below 2-3 rpm and bearings of "swinging" mechanisms, another method is also ineffective for assessing their condition - the envelope (power) spectral vibration analysis in any of frequency regions.
- To assess the condition of such bearings by vibration, the most effective method is the analysis of pulse vibration of bearing units in the mid-frequency band, where the main contribution is made by strikes in defective friction units.
- When monitoring the pulse vibration of low-speed bearings it is desirable to carry out synchronous measurements of the shape of the vibration pulses at all points and in several frequency bands to sort their sources by measurement points and then to make an independent decision for each source.

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