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Research paper

Experimental Studies of Sliding Bearings of Locomotive Turbochargers by Their Level of Vibration

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Abstract

The diagnostic features and sources of vibration of sliding bearings of diesel locomotive turbochargers (disappearance of the negative allowance on bearing bushings, self-oscillation in the bearing, self-oscillation when the negative allowance on the bearing bushing disappears, excitation of blade harmonics, etc.) are described in brief. The results of experimental studies of diesel locomotive turbochargers equipped with cylindrical and elliptical sliding bearings are shown. The specific spectra of vibration velocity of the bearing housings (cylindrical and elliptical) in the low-frequency range are presented. Bearing frequencies have been detected in the low-frequency vibration spectrum of the turbocharger housing with cylindrical bearings, and the connection of the detected diagnostic features with defects in sliding bearings has been established. Blade frequencies of the compressor and turbines on turbochargers with cylindrical and elliptical sliding bearings have been found in the high-frequency part of the vibration spectrum. These frequencies are necessary for identifying rotor harmonics and bearing frequencies. It has been established that for a defect-free bearing of turbochargers with elliptical and cylindrical sliding bearings, the vibration velocity of the rotor harmonics and the bearing frequency should not exceed the rated values.

Keywords: *blade harmonics; frequency bands; low-frequency spectrum; self-oscillations; sliding bearings; turbocharger; vibration velocity.*

1. Introduction

The long-term practical operating of turbocharger plants shows that one of the most common causes of increased vibration is the non-conservative forces of the oil layer of the slide bearings and the loosening of the attachment of the slide bearing housing. These causes contribute to an increase in the vibrational activity of machineries due to a change in the rigidity properties of support systems and the occurrence of various types of imbalance of the rotor, which causes an increase in centrifugal forces [1]. In this case, the loss the stability of the rotor becomes possible.

To avoid the classical instability characteristics for cylindrical journal bearings, sliding bearings with movable bushings (segment bearings) are used to increase the spectrum of rotational speeds within which the rotor is stable. Decrease in margin of stability and, in some cases, the occurrence of self-oscillations, may be caused by the deterioration of the hinge support of the "pads" – pad jamming. In addition, the changed stiffness of the support when fastening of the bearing housing is loosened for low damping in the supports may lead to a decrease in the threshold rate of instability vs. that of the rigid supports. Therefore, loosening of the fastening of the bearing housing in a system operating far beyond the threshold for the instability onset (a rigid rotor system) can lead to the self-oscillations excitation in case of loss of stability and to destabilize the system.

2. Diagnostic Signs and Sources of Vibration of Sliding Bearings of Diesel Locomotive Turbochargers

2.1. Disappearance of Negative Allowance on the Bearing Bushings

The exponential distributive law of the vibration velocity spectrum of rotor harmonics is violated as level of the 2nd and sometimes 3rd rotor harmonics increases disproportionately to the level of the 1st harmonics in radial directions (Figure 1). When the negative allowance on the bushings decreases, the intensity growth, mainly of the 2nd rotor harmonics, can remain unchanged and correspond to the level of a defect-free unit. As the negative allowance (clearance) in the bushings disappears, the 2nd rotor harmonic increases and the level of the 1st rotor harmonics rises significantly. The subharmonic levels remain practically unchanged, except for the 1/3-fold frequency of the 1st rotor harmonic. An increase in the subharmonics is an optional sign, it may occur provided resonance vibrations of the housing are excited (Figure 1).

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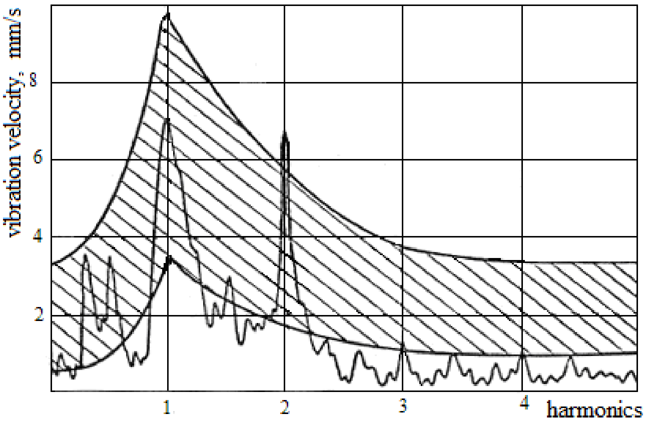

Fig. 1: Distribution of rotor harmonics levels. Disappearance of negative allowance on bearing bushings.

When the negative allowance disappears, the bearing housing receives two impulses with rotor tilting per each rotation, and the path of the center of the rotor can be shaped as a very elongated ellipse. This defect is difficult to tell from an ovality defect of the rotor necks, which causes the double stiffness of the rotor. In this case, the second rotor harmonics [2] are also actuated, and the motion trajectory of the rotor is described by a curve called "Pascal snail".

2.2. Self-Oscillations in the Bearing

With an increased vertical clearance in the bearing [1], the exponential law of the distribution of the spectrum of the rotor harmonic levels is violated by a disproportionate increase in the subharmonics level at a 1/2-fold frequency to the rotor speed (Figure 2). Meanwhile, the level of the 1st rotor harmonic may remain low. The increase in the clearance leads to an increase in the level of the 1st rotor harmonic and the further growth of subharmonics. If a clearance is equal to or exceeds the critical one, prerequisites for the excitation of a self-oscillatory process in the oil layer of the bearing arise. In this case, the vibration levels of these components increase sharply and can reach V_e = 15 to 20 mm/s.

Self-oscillations of the rotor in the bearings may occur in the transient modes of operation of the plant. In the vibration spectrum of the compressor, a harmonic appears at a frequency equal to half the circulating frequency (harmonic of self-oscillations), (Figure 2). At the same time, the vibration velocity increases to 42 mm/s, which is 20 times higher than the level at the nominal mode and exceeds the permissible norms [6]. This phenomenon can be caused by the inertia of an individual pad in the transient mode, which creates conditions corresponding to the "jamming" of a individual pad. Calculations of stability of rotor rotation using the methodology of [1] show that in this case, self-oscillations of the rotor with a frequency equal to half the rotation rate of the rotor $1/2_{fp}$ may occur. In addition, this phenomenon can be caused by the appearance of additional hydrodynamic efforts in the bearing. It has been established that the vibration velocity spectrum here includes a number of frequencies with an integer multiplicity of the self-oscillation frequency $3/2_{fp}$, $5/2_{fp}$. After completion of the transient process, the value of the half-harmonic level corresponds to the norm. The repeated spectrograms recorded at the same points in the steady mode confirm the stabilization of the rotation of the rotor.

Thus, the transient modes excite intensive self-oscillations of the rotors of the investigated aggregates and represent a danger, since with the possible destruction of bearings. The growth of the 2nd rotor harmonic is a diagnostic sign of self-oscillations.

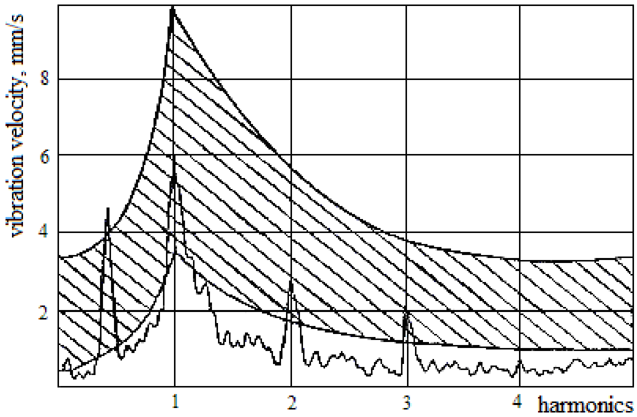

Fig. 2: Distribution of levels of rotor harmonics. Self-oscillations in bearings.

2.3. Self-Oscillations When the Negative Allowance on the Bushing Disappears

A self-oscillatory process can also occur the negative allowance on the bearing bushing disappears. The presence of this defect is accompanied by the growth of the second rotor harmonic. The peculiarity of the self-oscillating process is that the oscillation frequency does not clearly follow the reverse frequency, as well as the excitation of harmonics with an integer multiplicity to the frequency of self-oscillations.

In the analysis of the stability of the rotor motion in sliding bearings, the bearings are assumed to be mounted on elastically compliant supports. Nonlinear hydrodynamic forces occurring in load-carrying oil layers of the bearings are found from the solutions of the Reynolds equation as functions of the equilibrium position corresponding to the stable state of the system and determined by the relative eccentricity. The hydrodynamic forces in the bearing can be linearized, i.e. are expressed via the stiffness and damping coefficients in the dimensionless form. In this case, the rigidity of the support has the form

$$K_o = (K_v \cdot K_k)/(K_v + K_k)\,, \tag{1}$$

where K_v, K_k - stiffness coefficients of bearing bushing and housing, respectively.

The guaranteed negative allowance between the bushings and the shell ensures the fulfillment of the condition $K_v >> K_k$ and hence $K_o \approx K_k$. However, when the negative allowance disappears and the clearance Δ_v appears between the bushing and the bearing housing, the value of the stiffness rate of the bearing bushing changes and depends on the clearance in the bearing. Using the method of replacing the nonlinear characteristic of a conditional linear [3], an equivalent stiffness rate is introduced from the conditions of equality of potential energies of elastic couplings at maximum displacements. Analysis of stability and non-stationary oscillations of the “rotor-bearing” system on compliant supports is carried out in accordance with the levels of motion by investigation of the characteristic equation. The relative stiffness and damping rates with guaranteed negative allowance on the bearing bushing are, respectively, K_o = 3.15; C_o = 1.8. In this case, the rotor stability conditions are not violated. When the negative allowance in the bearing bushings disappears, the relative stiffness and damping rates are, respectively, K_o = 0.94; C_o = 0.1, and in this case the rotor stability conditions are preserved. However, with a decrease in the relative eccentricity of the bearing journal corresponding to the stable equilibrium position, the rotor becomes unstable as the antifriction surface of the bearing bushing is wearing off. Thus, the increase in the level of the half rotor harmonic indicates the self-oscillations in the bearings. The simultaneous growth of the half and the 2nd rotor harmonics is a diagnostic sign of the self-oscillations of the bearing caused by the disappearance of the negative allowance on the bearing bushings.

2.4. Excitation of Blade Harmonics

The theoretical definition of unsteady aerodynamic forces caused by vortex traces allows for identifying the main regularities and finding only the criteria dependencies, because determining the disturbing forces in modes with blades of arbitrary shape, streamlined by a flow with vortex traces is very difficult. It should be taken into account that at very high process frequencies characteristic of turbocompressors, even at small Mach numbers of the main flow, the compressibility impact must not be neglected, which will also require experimental corrections. At the same time, it may turn out that at very high frequencies and high degree of turbulence, the process of displacement of the vortex traces inside the lattice does not depend significantly on the absolute Mach number, the Reynolds number, and the Strouhal number calculated from the speed of sound. Therefore, in evaluating the disturbing forces, a connection is established between the heterogeneity of the flow and the magnitude of the non-stationary force. Suppose that the non-stationary aerodynamic force is proportional to the gas density, the characteristic ripple velocity in the V_o trace, the relative velocity at the output from the wheel, and the step of the impeller of compressor blade. Then the expression for the force P takes the form

$$P = (G \cdot V_o / Z) \cdot \sin\beta_2 \,, \tag{2}$$

where G - weight consumption of gas flowing through the supercharger; Z - number of impeller blades; β_2 - the output slope angle of the relative velocity.

The resulting expression (2) describes the non-stationary aerodynamic force acting on the housing from one channel formed by two adjacent blades of the compressor wheel. Expression (2) represents an approximate expression for P. However, since the value of V_o can be determined only from experiment, then expression (2) represents some characteristic pulsation of the amount of motion of the gas element passing through the wheel of the supercharger. It is natural to assume that the nonstationary force is proportional to the pulsation of the momentum. In addition, it is logical to assume that all unsteady velocities are proportional to V_o. In this case, the multiplication ($G \cdot V_o$) is interpreted as a value which is proportional to the perturbed pressure, while the unsteady force must be proportional to the pressure pulsation [3].

These issues are discussed in more detail in [5], where it is shown that the measured blade harmonics (diagnostic sign) allow identifying rotor harmonics of turbochargers in conditions of high noise level (diesel operation). At the same time, power and consumption are characteristics of turbocharger operating conditions and need to be brought to reference values, and efficiency is a diagnostic sign of the state of the flow section and affects the growth of the vibration velocity amplitude of the blade harmonic of housing oscillations.

Thus, not only the measured vibration characteristics, but also measured temperature at the entrance to the turbocharger, the barometric pressure, the rotor speed or the rotor harmonics of the vibration velocity, the gas consumption and the turbocharger technical condition rate should be available to obtain the reduced blade harmonic.

3. Allocation of a Useful Signal and Detection of Defects by the Level of Vibration Velocity

3.1. General

Vibration tests of turbochargers (TC) were carried out on locomotives in the Osnova depot of the Southern Railway. The measurements were carried out using the Brüel & Kjær equipment according to the standard diagram shown in Fig. 3. Turbochargers with cylindrical sliding bearings and turbochargers with elliptical sliding bearings (6 TC) installed on diesel locomotives of the Osnova depot before TP-2 and after TP-2 were tested, and rheostat testing was performed.

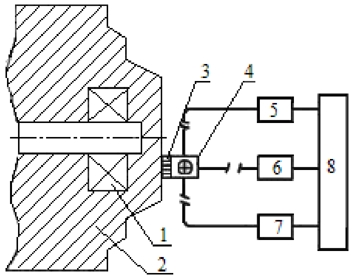

Fig. 3: Standard measuring diagram: 1 – bearing of the unit; 2 - the housing of the unit; 3 - sensor mounting magnet; 4 - three-component sensor 4321; 5, 6, 7 – charge amplifiers of three channels of the device for vertical (5), transverse (6) and axial (7) directions; 8 - Brüel & Kjær meter.

In the first phase, vibration processes were measured using primary information converters (vibration sensors), matching devices (charge amplifiers), and recording in the operating long-term memory.

The quality of the diagnosis depends not only on the type of vibration processes being measured, but also on the diagnosis mode, the location of the sensors (Figure 4), and the parameters of the measuring equipment. Therefore, the vibration sensors for additional monitoring were installed on the diesel engine housing, and the tests were conducted in two idling modes and four characteristic modes with the loaded diesel engine. The main requirements for primary converters – ensuring a given linear dependence between the output signal and the level of the measured parameter in a given operating frequency range – have been fulfilled in the entire frequency range.

The article presents characteristic vibration signals at the 8th (8 CP), 12th (12 CP), 15th controller positions (15 CP).

Analysis of the vibrations of the bearing housings established that the diesel is the most powerful exciter of oscillations. A state-of-the-art technique of spectral and coherent analysis has been used to allocate the diagnostic features of vibration activity of the turbocompressor able to recognize the characteristic frequency components under conditions of strong interference (diesel vibration).

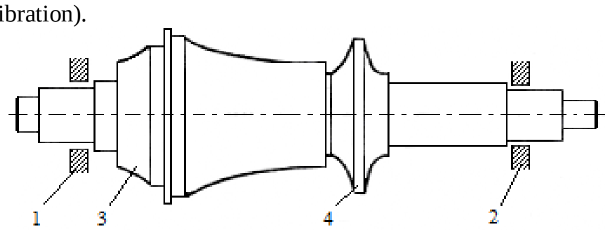

Fig. 4: List of vibration check points for turbochargers 1 - journal and thrust bearing (compressor side); 2 - supporting bearing (turbine side); 3 - the compressor impeller; 4 - the turbine impeller.

3.2. Acoustic Vibration Tests. Spectral and Coherent Analysis

3.2.1. Cylindrical Sliding Bearings

Figures 5 and 6 shows the characteristic low-frequency vibration spectrum of the bearing housing up to 400 Hz (15 CP and 8 CP) In the spectrum multiple rotor harmonics of the diesel engine are present: $f_{r.d}$ = 14 Hz (TC No. 1.15 CP) and $f_{r.d}$ = 9.4 Hz (TC No. 2, 8 CP), which prevents from analyzing turbocharger vibrations. The coherence function (Figure 5) shows that the signal at the allocated frequencies is coherent and the rotor harmonics are stable.

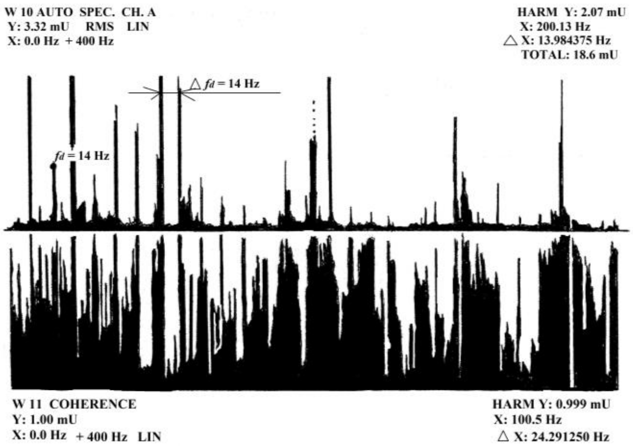

Fig. 5: Low-frequency spectrum of vibration velocity of a turbocharger (TC No. 1, 15 CP).

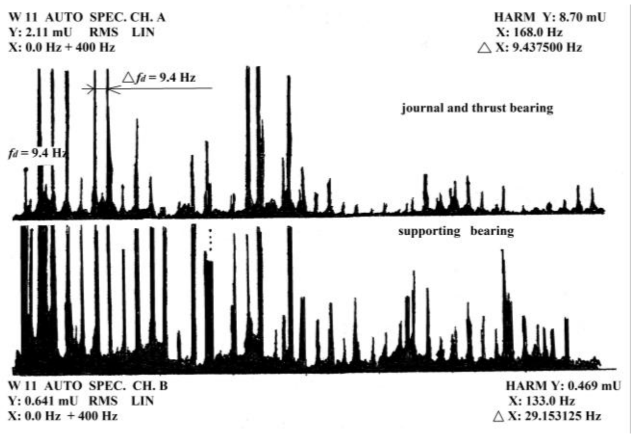

Fig. 6: Low-frequency spectrum of vibration velocity of a turbocharger (TC No. 2, 8 CP).

In experimental materials on vibration diagnostics of defects of sliding bearing [4], it is shown that an increased clearance in case of a loose fit of the bearing causes excitation of polyharmonic oscillations with frequencies that are multiples of half the rotor speed $k_{fp}/2$, where k = 1, 2, 3 …. In the vibration spectrum, the half-subharmonic of the rotation rate should exceed the noise level of the interference by 20 dB. In addition, it is proposed to differentiate the appearance of the defect that occurs in the slide bearing by accurately counting the frequency of the newly formed component that is less than f_p. If the occurring frequency is 42 to 48% of rotation rate of the rotor, this is a typical indication of the instability of the oscillations of the rotor on the oil film, characterizes the appearance of vortices in the lubricating layer, which results in a serious service life decrease of the sliding bearings.

The main difficulty in detecting self-oscillating modes of the turbocharger is the identification of half rotor harmonics (bearing frequencies) and rotor harmonics of the vibration spectrum of the bearing housings. This is due to the fact that the operating rotations are not measured and only the rotation ranges of the turbocharger rotor and diesel crankshaft are indicated for each position of the controller in the manufacturer's certificate. In addition, at different values of the diesel engine power, the rotation rate range of the rotor can be much wider than the one provided in the manufacturer's certificate.

At the 8th CP, the operating rate of the rotor of turbocharger No. 2 varies from 135 to 168 Hz. Identification of the rotor harmonic f_p from the vibration spectrum of the bearing housings in this frequency range is impossible (Figure 6).

In the turbochargers with cylindrical sliding bearings the rotor harmonics were identified: f_p = 232 Hz (Figure 7, TC No. 1) and f_p = 272 Hz (Figure 8, TC No. 2) at the fifteenth controller position (15 CP).

In the vibration spectrum of the turbocharger, half-rotor harmonic of ½ f_p = 116 Hz (Figure 7) is recorded which considerably exceeds the amplitude of the rotor harmonic. For example, the half-rotor harmonic is 7.23 mm/s, and the rotor harmonic is 1.2 mm/s.

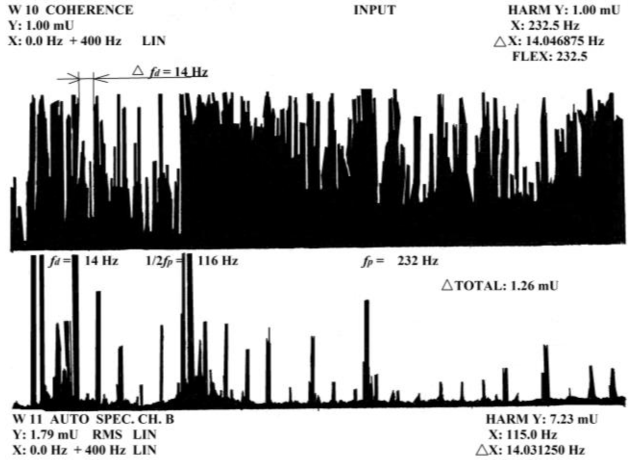

Fig. 7: The coherence function and autospectrum of turbocharger vibration velocity (TC No. 1, 15 CP).

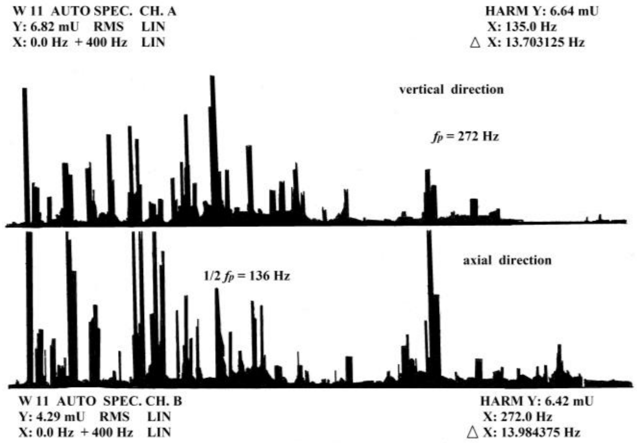

Fig. 8: Self-oscillation of the turbocharger rotor (TC No.2, 15 CP).

The coherence function shows the best coherence equal to unity near the half rotor harmonic, within a wide frequency range (Figure 7), which is due to the self-oscillations of the rotor in the bearings. Self-oscillating rotation modes of the rotor are found on almost all tested turbochargers. The amplitude of the half rotor harmonic of the vibration spectrum 1/2 f_p = 136 Hz is 6.64 mm/s and is commensurate with the amplitude of the diesel harmonic in the vertical direction equal to 6.82 mm/s (Figure 8). The rotor harmonic of the turbocharger meanwhile is 2.47 mm / s. Similar values are recorded along the axial direction (Figure 8).

The rated value of the rotor harmonic of turbocharger No. 2 is within the range f_p = 200 to 270 Hz. It manifests itself on the spectrum as a peak at a frequency of 200.5 Hz (Figure 5) and is difficult to tell from the vibration of the diesel engine. Bearing frequency equal to f_p = 0.5, f_p is slightly manifested. In the vibration spectrum with a defect in the sliding bearing, the bearing frequency f_p = 1/2 f_p = 136 Hz (TC No. 2) is much larger than the rotor harmonics of the turbocharger and diesel engine, and is a diagnostic sign of self-oscillations and, consequently, destruction of the bearing (Figure 9). The rotor harmonics f_p = 272 Hz are much less than half harmonics.

Rotor harmonics of turbochargers were identified by means of vibration of aerodynamic origin (blade frequencies).

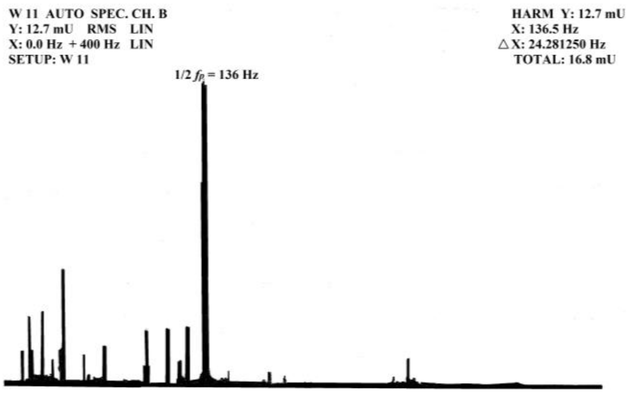

Fig. 9: The low-frequency spectrum of vibration velocity of turbocharger No. 2 (defect in the bearing).

The main reason for the excitation of such a vibration is the circumferential variation of the flow velocity. Each blade of a turbine or compressor wheel receives as many impulses per rotation as there are obstacles. As the impulses generated by each blade are summed and transferred to the turbocharger housing, the vibration of the housing with frequencies f_m and f_k multiplied by the number of working blades, respectively, of the turbine Z_m and the compressor Z_k [5] is excited.

$$f_m = k \cdot f_p \cdot Z_m; f_k = k \cdot f_p \cdot Z_k; k = 1,2,3... \tag{3}$$

The rated blade speed of the compressor wheel for turbocharger No. 1 with cylindrical bearings at a rotation rate of f_p = 232 Hz and the number of compressor blades $Z_{l.k.}$ = 16 is $f_{l.k}$= 3,712 Hz. In the high-frequency portion of the spectrum at the mode f_p = 232 Hz, the compressor blade frequency $f_{l.k}$= 3,712 Hz is present and a half and rotor harmonics are observed. At the mode f_p = 272 Hz (TC No. 2), the frequency $f_{l.k}$ = 4,352 Hz corresponding to the rated value is always present in the spectrum. The blade frequency of the turbine does not occur in the vibration spectrum (Figure 10). Thus, turbine and compressor blades are detected in the vibration velocity spectrum, which is a diagnostic sign of the technical condition of the blades.

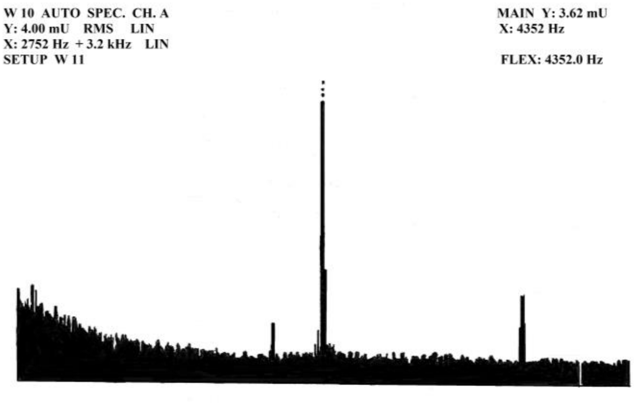

Fig. 10: High-frequency spectrum of vibration velocity of a turbocharger (TC No.2).

3.2.2. Elliptical Sliding Bearings

Figure 11 shows autospectra oscillations of the bearing housing of the 6TK turbocharger with elliptical sliding bearings at 15 CP. When the number of compressor blades $Z_{l.k}$ = 18 and the value of the blade frequency f_k = 5,792 Hz, the rotor harmonic is 321 Hz and is present in the spectrum as a discrete ejection at a frequency of 321 Hz (Figure 11). Half rotor harmonic was not detected in the vibration spectrum of the 6TK turbocharger in all modes (controller positions), while the harmonics of the diesel f_d are 6 mm/s and significantly exceed the amplitude of the rotor harmonic f_p which is 0.5 mm/s [7, 8, 9].

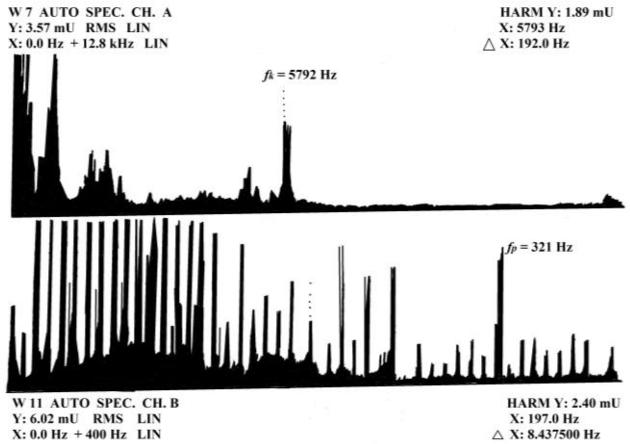

Fig. 11: High-frequency (f_k = 5,792 Hz) and low-frequency (f_p = 321 Hz) vibration velocity spectrum of the 6TK turbocharger (15 CP).

The blade frequencies of the turbine in the vibration spectrum of the 6TK turbocharger housing are absent, however, a discrete ejection of the turbine blade frequency is observed in the spectrum of sound pressure (turbocharger noise). For example, at the 12th CP of the 6TK turbocharger, there are frequency emissions f_p = 250 Hz; f_k = 4,500 Hz and f_n = 9,232 Hz in the noise spectrum (Figure 12).

When the number of turbine blades is $Z_{l.t}$ = 37, the frequency of 9,232 Hz corresponds to the turbine blade frequency and is manifested in the noise of the turbocharger and is absent in the vibration spectrum of the housing. The absence of the turbine frequency in the vibration spectrum is due to the insufficient frequency resolution of the vibration sensor.

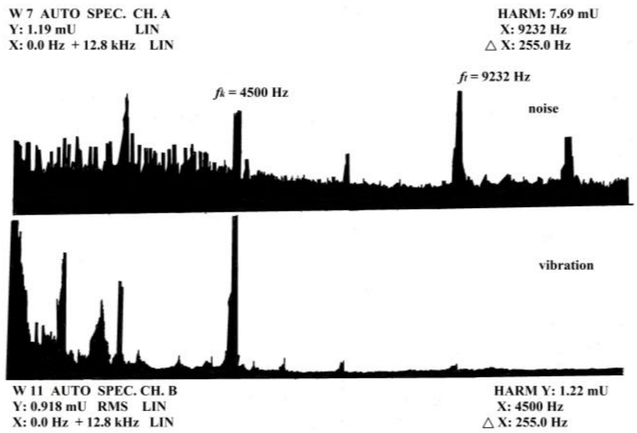

Fig. 12: Low-frequency noise spectra of vibration of a 6TK turbocharger (12 CP).

Thus, rotor harmonics of the 6TK turbocharger, which were blocked by a source of strong ambient noise – diesel harmonics f_d, have been identified using the turbine and compressor blade frequency. Half rotor harmonic in the noise and vibration spectrum of the 6TK turbocharger could not be detected in the tests of a large batch of engines. Elliptical sliding bearings are obviously stable and self-oscillations (vibrational defects) of the rotor in the sliding bearings are not observed [6].

4. Conclusions

Thus, bearing frequencies (semi-harmonic) have been detected in the low-frequency vibration spectrum of the turbocharger housing with cylindrical sliding bearings, and the association of the detected diagnostic signs with defects in sliding bearings is established. In the high-frequency portion of the vibration spectrum, the blade frequencies of the compressor and turbines are found for both types of turbochargers, necessary to identify rotor harmonics and bearing frequencies. It has been established that for a defect-free bearing (elliptic or cylindrical), the vibration velocity of the rotor harmonic and the bearing frequency should not exceed 1 mm/s [7, 8, 9]. If self-oscillations occur (cylindrical bearings), the vibration velocity of the bearing frequency is 6.7 mm/s [7, 8, 9] (Figure 8) and exceeds the levels of rotor and diesel harmonics. Self-oscillations of the rotor of a turbocharger with elliptical sliding bearings have not been detected.

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