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Generic Structural Model of Machinery Vibroacoustic Signal

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Abstract. The paper deals with the challenges of the machinery vibroacoustic signal modeling. The purpose of this paper is to form a phenomenological model for the structure of vibroacoustic oscillations and signals appeared in case of malfunctions and defects. It presents particular structural models for vibroacoustic signals of some reciprocating machine malfunctions. Mathematical description of the vibroacoustic signal reveals the informative parameters of the malfunction diagnostic signs. The model enables us to define in detail the informative vibroacoustic signal parameters. The model also allows us to select the diagnostic signs of the given malfunctions and defects in certain machine parts and components. At the same time, it is considered that the signal is received in certain points on the machine taking into account the excitation sources, conditioning channel and propagation of the vibroacoustic oscillations. The model has been constructed considering the presence and interaction of the three main sources of excitation of the asset vibroacoustic oscillations, which are identified by the types of excited oscillations. It is assumed that there are damped natural, undamped induced, random broadband and narrowband vibrations. The parameters of the vibroacoustic signal structural model also take into account the properties of converting the vibroacoustic oscillation to the vibroacoustic signal by means of a piezoelectric sensor. The model shows that the vibroacoustic signal has not only the components from each source, but their mutually modulated components. We developed partial models for mechanisms of structuring the vibroacoustic signal appeared in case of malfunctions and defects in rotating parts and couplings, machine shaft and drive shaft misalignment, increased clearances and other defects and failures of the rotating and reciprocating parts and components. The proposed phenomenological structural models of the vibroacoustic signals appeared in case of malfunctions of reciprocating machines show that for the diagnostic purposes it is necessary to analyze vibroacoustic signals either in the time domain according to the cyclogram of the reciprocating machine operation or in the envelope spectrum at characteristic frequencies or the noise component. In the event of defects and malfunctions in the centrifugal rotating parts and components, there are excited the vibroacoustic signals, which are advisable to analyze in the direct spectrum or in the spectrum of the noise component envelope. The proposed models enable us to select and form scientifically grounded systems of the machine failure diagnostic signs.

INTRODUCTION

The paper deals with the challenges of machinery vibroacoustic signal modelling [1, 2, 3]. The purpose of this paper is to form a phenomenological model for the structure of vibroacoustic oscillations and signals appeared in case of malfunctions and defects. It presents particular structural models for vibroacoustic signals of some reciprocating machine malfunctions.

A reciprocating machine is a complicated gas-mechanical system. It is a powerful and multifactor vibroacoustic oscillator that has three main, statistically independent, sources [4, 5, 6, 7, 11], which, with a reasonable degree of conditionality, can be divided into:

- polyharmonic forces F_h (harmonic) induced by imbalance of rotating and moving masses;
- stochastic (noise) forces F_n (noise) induced by gas-hydrodynamic processes;
- shock exciting forces F_s induced by collision and friction between elements and parts.

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DEVELOPMENT OF GENERIC MODEL

Nonlinear interactions in the mechanical system of interacting elements lead to multiplication and superposition of all force interactions F_h, F_s, F_n with weight functions in the form of pulse responses h_n^s, h_h^s, h_s^n, h_h^n, h_n^h, h_s^h corresponding to each pair of forces. As a result, there are made up three groups of forces affecting machine elements, details, and components, which signals can be mathematically described as follows [7, 11]:

Shock forces –

$$F_{s\Sigma} = F_s + F_s F_n h_n^s + F_s F_h h_h^s; \tag{1}$$

Stochastic forces –

$$F_{n\Sigma} = F_n + F_n F_s h_s^n + F_n F_h h_h^n; \tag{2}$$

Polyharmonic forces –

$$F_{h\Sigma} = F_h + F_h F_n h_n^g + F_h F_s h_s^g. \tag{3}$$

Let us assume that in the time domain forces $F_n h_n^s$, $F_h h_h^s$, $F_s h_s^n$, $F_h h_h^n$, $F_n h_n^g$, $F_s h_s^g$ interact as convolutions of forces and impulse responses that correspond to the interaction force pairs. Let us denote these forces as F_n^s, F_h^s, F_s^n, F_h^n, F_n^h, F_s^h. Then the resultant interactions can be written as:

Shock forces –

$$F_{s\Sigma} = F_s + F_s F_n^s + F_s F_h^s; \tag{4}$$

Stochastic forces –

$$F_{n\Sigma} = F_n + F_n F_s^n + F_n F_h^n; \tag{5}$$

Polyharmonic forces –

$$F_{h\Sigma} = F_h + F_h F_n^g + F_h F_s^g. \tag{6}$$

Formally, these expressions can be represented as follows:

Shock forces –

$$F_{s\Sigma} = F_s\left(0.5 + F_n^s\right) + F_s\left(0.5 + F_h^s\right); \tag{7}$$

Stochastic forces –

$$F_{n\Sigma} = F_n\left(0.5 + F_s^n\right) + F_n\left(0.5 + F_h^n\right); \tag{8}$$

Polyharmonic forces –

$$F_{h\Sigma} = F_h\left(0.5 + F_n^g\right) + F_h\left(0.5 + F_s^g\right). \tag{9}$$

Through elementary transformations we obtained classical expressions describing the time-domain modulation processes. Thus, we obtained a phenomenological mathematical model for the processes of interaction between forces of different nature in a mechanism, which shows that there is an intermodulation of all existing forces [9, 10, 11].

Each force excites various oscillations. It can be assumed with reasonable certainty that impulse (shock) exciting forces $F_{s\Sigma}$ induce oscillations $S_{s\Sigma}$ at natural (resonance) frequencies of object, element, detail, and node casing vibrations taking into account the impulse response $h_s^s(t)$ that determines the properties and characteristics of this oscillation mode. Along with that the harmonic exciting forces $F_{h\Sigma}$ induce forced sustained polyharmonic oscillations of the vibroacoustic channel $S_{h\Sigma}$ of propagation of oscillations with the impulse response $h_h^h(t)$. Friction processes, gas-hydrodynamic forces $F_{n\Sigma}$ give rise to broadband and narrowband random vibrations of vibroacoustic channel $S_{n\Sigma}$ with impulse response $h_n^n(t)$, which, in general, can be called noise, noise-like or stochastic processes (Fig. 1).

Considering the impulse responses of the vibroacoustic channel for each oscillation mode, we may formalize the mathematical expression describing the total vibroacoustic oscillatory processes excited by all kinds of forces:

$$\|S_\Sigma(t)\| = \begin{Vmatrix} h_s^s(t) & 0 & 0 \\ 0 & h_n^n(t) & 0 \\ 0 & 0 & h_h^h(t) \end{Vmatrix} \times \begin{Vmatrix} F_s(t)\cdot[0{,}5+F_n^s(t)] & F_s(t)\cdot[0{,}5+F_h^s(t)] \\ F_n(t)\cdot[0{,}5+F_s^n(t)] & F_n(t)\cdot[0{,}5+F_h^n(t)] \\ F_h(t)\cdot[0{,}5+F_n^h(t)] & F_h(t)\cdot[0{,}5+F_s^h(t)] \end{Vmatrix}. \quad (10)$$

Thus, the three groups of the forces acting on the machine elements, details, and nodes excite vibroacoustic oscillations that can be mathematically represented as follows [7]:

$$\|S_\Sigma(t)\| = \begin{Vmatrix} S_s^s(t)\cdot[0{,}5+S_n^s(t)] & S_s^s(t)\cdot[0{,}5+S_h^s(t)] \\ S_n^n(t)\cdot[0{,}5+S_s^n(t)] & S_n^n(t)\cdot[0{,}5+S_h^n(t)] \\ S_h^h(t)\cdot[0{,}5+S_n^h(t)] & S_h^h(t)\cdot[0{,}5+S_s^h(t)] \end{Vmatrix}. \quad (11)$$

In the last expression, the following notations are used: $S_s^s(t)=F_s(t)*h_s^s(t)$, $S_n^n(t)=F_n(t)*h_n^n(t)$, $S_h^h(t)=F_h(t)*h_h^h(t)$ is a mathematical description of vibroacoustic oscillations presented as a convolution of the force functions and impulse responses of converting the forces into vibroacoustic oscillations.

The conversion of machine vibroacoustic oscillations into an electric or vibroacoustic signal is associated with another transformation of these oscillatory processes – addition of interference and superposition of the mechanical oscillation-to-electric signal conversion responses, since the oscillatory process parameters can be significantly influenced by the transmission medium, which includes the paths (elements) of oscillation propagation to the oscillation receiving place, elements of oscillation transfer to the primary transducer and the properties of the sensing element of the primary transducer (vibration sensor), which describe the sensor-machine casing system and the sensing element with the frequency responses.

Considering the impulse responses of the conversion, amplification channel $h_{tr}(t)$, we will get an expression describing the sensor output electric signal as an equivalent of the noise and periodic components [5, 7, 11] of the vibroacoustic signal (Fig. 1):

$$\|U_\Sigma(t)\| = \|h_{tr}(t)\| * \|S_\Sigma(t)\| = \begin{Vmatrix} h_{str}(t) & 0 & 0 \\ 0 & h_{htr}(t) & 0 \\ 0 & 0 & h_{ntr}(t) \end{Vmatrix} * \begin{Vmatrix} S_s^s(t)\cdot[0{,}5+S_n^s(t)] & S_s^s(t)\cdot[0{,}5+S_h^s(t)] \\ S_h^h(t)\cdot[0{,}5+S_n^h(t)] & S_h^h(t)\cdot[0{,}5+S_s^h(t)] \\ S_n^n(t)\cdot[0{,}5+S_s^n(t)] & S_n^n(t)\cdot[0{,}5+S_s^n(t)] \end{Vmatrix}. \quad (12)$$

As a result, there are formed the noise and periodic components of the vibroacoustic signal [11], which are the sums of the oscillation processes of the three main force types and their intermodulated components.

MODEL OF SHOCK FORCES

The solving of the problems in the structural modeling of the vibroacoustic oscillations associated with the shock interactions of the details at the moment of collision rests on the finding reported in the publications of N.S. Zhdanovsky, V.A. Alliluev and B.V. Pavlov [7, 11] that are based on the H. Hertz theory of contact between solids.

The rise and fall of the interaction force in the details during their collision is shown as a half cosine wave:

$$F_s\ (t) = \begin{cases} \dfrac{\pi q_0}{\tau}\cos\dfrac{\pi}{\tau}t & \text{with } -\dfrac{\tau}{2} \le t \le \dfrac{\tau}{2} \\ 0 & \text{with } |\,t\,| \ge \dfrac{\tau}{2} \end{cases}, \quad (13)$$

where q_0 is the pulse area; τ is the pulse length.

In view of the Hertz theory we can derive a formula considering the clearance between the details:

$$F_s\ (t)=\begin{cases}\dfrac{3.35}{\tau}\sqrt{2mFh_0}\cos\dfrac{\pi}{\tau}t & \text{with } -\dfrac{\tau}{2}\le t\le\dfrac{\tau}{2}\\ 0 & \text{with } \mid t\mid\ge\dfrac{\tau}{2}\end{cases}, \tag{14}$$

where m is the reduced detail mass; F is the average force acting on the details; h_0 is the clearance between the details.

The spectral density of the detail collision force impulse is defined from the following formula:

$$F_s\ (\omega)\approx 3.35\frac{2}{\pi}\sqrt{2mFh_0}\left[\cos\frac{\omega\tau}{2}\Big/1-\left(\frac{2}{\pi}\frac{\omega\tau}{2}\right)^2\right]. \tag{15}$$

MODEL OF DAMPED OSCILLATIONS AT NATURAL FREQUENCIES

Each shock excites damped detail oscillations at the natural frequencies:

$$s(t)=Ae^{-\delta t}\sin(\omega_0 t)\,, \tag{16}$$

where A is the initial amplitude of oscillation that depends on the excitation intensity and the properties of the excited details; ω_0 is the natural frequency of the detail oscillations; δ is the oscillation damping factor (depends on the detail material properties).

Oscillations are not usually excited in one detail, but at least in two interacting ones, then they are transferred through other machine parts, and the force pulse can reach other details, in which oscillations at natural frequencies can also be excited, therefore the shock action causes the excitation of a set of elementary oscillating pulses with their own amplitudes, frequencies and damping, so, in general, the force action (15) results in a set of oscillatory processes that can be described as follows:

$$s(t)=\sum_{k=1}^{\infty}A_k\sin(\omega_k t+\varphi_k)\,, \tag{17}$$

where $A_k=a_k e^{-\delta_k t}$ is the oscillation amplitude of the k-th component with natural damping δ_k; φ_k is the oscillation phase of the k-th component; ω_k is the k-th component frequency; a_k corresponds to the waveform of the detail collision impulse (14).

PARTIAL MODEL OF VIBROACOUSTIC SIGNAL

The result of detail collision interaction has the form of a sequence of pulses that follow one another at a frequency equal to the frequency of shocks. This pulse sequence, assuming that the excitation amplitude and the oscillation damping factor for a given node are constant, can, in general, be represented as a sequence of pulses with random initial amplitude [5, 6, 7, 11]:

$$s(t)=\sum_{n=0}^{\infty}A_n\sum_{k=1}^{\infty}A_k\sin\left[\omega_k(t-nT_0)\right], \tag{18}$$

where T_0 is the period of excitation of oscillations ω_k having amplitude A_n; ω_k is the natural oscillation frequency of the k-th source. $A_k=a_k e^{-\delta_k(t-nT_0)}$ is the oscillation amplitude of the k-th component with natural damping δ_k.

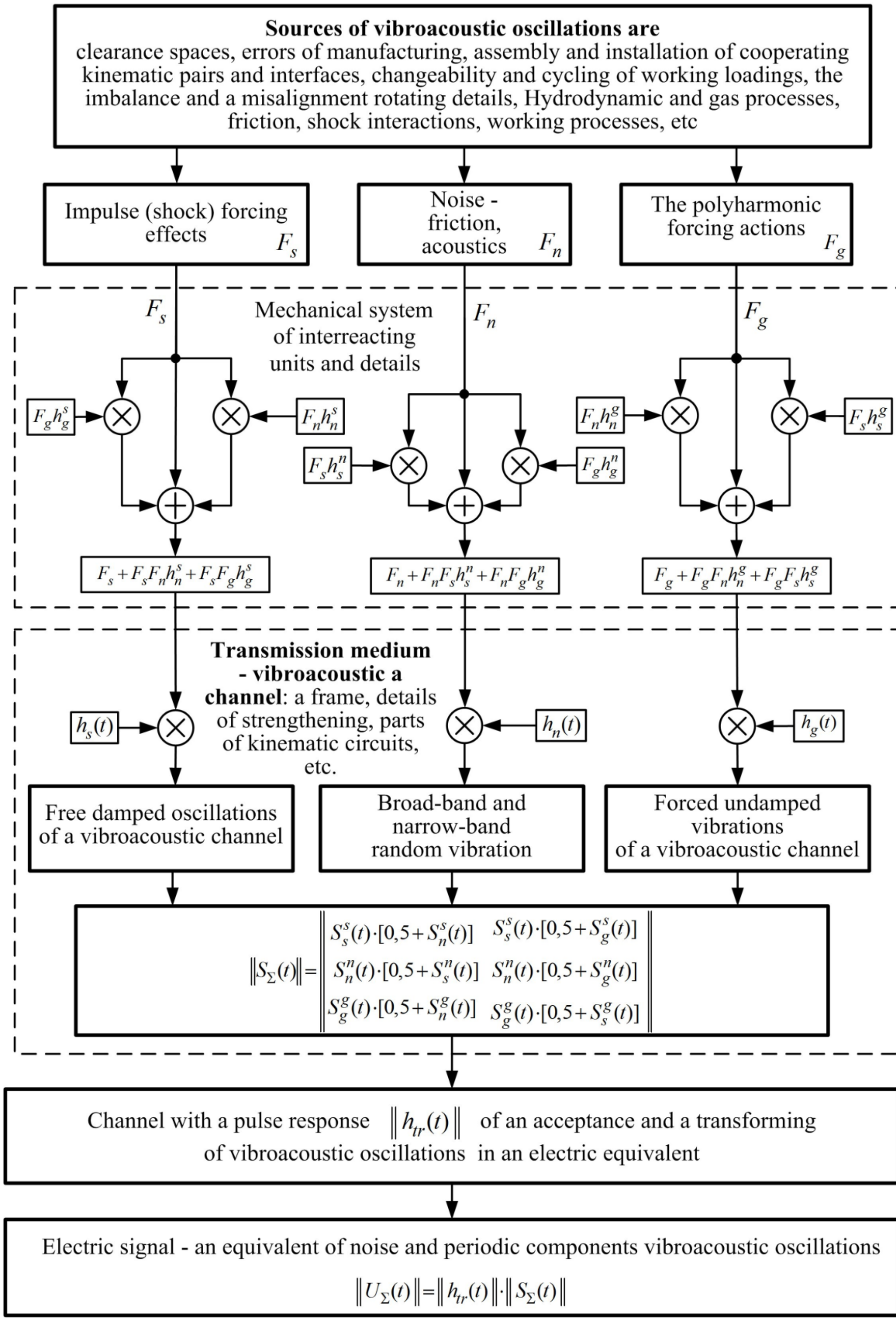

FIGURE 1. A vibroacoustic signal envelope spectrum: modulation of bearing defect frequency f_{BPFO} with the rotational frequency f_0 in case of imbalance [11]

Expression (18) generally is a sequence of pulses having random initial amplitude. A sample analysis of such sequences in the spectral domain can be found in [5, 6, 7, 11, 12, 13, 15]. In the particular case, the amplitude A_n does not depend on n and then it can be written as:

$$s(t) = A\sum_{n=0}^{\infty}\left\{\sum_{k=1}^{\infty} A_k \sin\left[\omega_k (t - nT_0)\right]\right\}. \tag{19}$$

The sequence spectrum for the k-th source follows the formula:

$$S(\omega) = \frac{A}{2}\cdot\frac{1}{\sqrt{\delta^2 + (\omega-\omega_k)^2}}\cdot\frac{\sin\frac{n\omega T_0}{2}}{\sin\frac{\omega T_0}{2}}. \tag{20}$$

In reciprocating machines, the period of excitation corresponds to the crankshaft speed harmonics and subharmonics [18, 19]. Informative is the phase of pulse excitation with respect to one of the known crankshaft angle-of-rotation positions. Therefore, a more general form of representation (18) is as follows:

$$s(t) = \sum_{n=0}^{\infty} A_n \sum_{m=1}^{M} A_m \left\{\sum_{k=1}^{\infty} A_k \sin\left[\omega_k\left(t - n\left(T_0 + \Delta t_m\right)\right)\right]\right\}, \tag{21}$$

where Δt_m varies from 0 to T_0 as m varies from 1 to M; the value of M corresponds to the number of shock excitations per machine operation cycle; $A_k = a_k e^{-\delta_k((t-n(T_0+\Delta t_m))}$ is the oscillation amplitude of the k-th component with natural damping δ_k. For a reciprocating compressor, this cycle normally corresponds to one revolution of the shaft, i.e. the rotational speed.

The parameters of the detail shock interaction are influenced by various factors, which, from the vibroacoustic oscillation generation viewpoint, must first include the force excitations associated with the polyharmonic forces F_h and friction F_n. In view of the said features of excitation of oscillatory processes in an object at collision, in order to simplify the formal mathematical description, it is advisable to make some assumptions and write (21) as follows:

$$S_s^x = S_s^x(t) = F_s(t) * h_s^x(t) = \sum_{n=0}^{\infty} A_n^x \sum_{m=1}^{M} A_m^x \left\{\sum_{k=1}^{\infty} A_k^x \sin\left[\omega_k\left(t - n\left(T_0 + \Delta t_m\right)\right)\right]\right\}, \tag{22}$$

where $h_s^x(t)$ is the impulse response that describes the conversion of force $F_s(t)$ into oscillatory processes of elastic waves at a given point under given conditions (x); Δt_m varies from 0 to T_0 as m varies from 1 to M; the value of M corresponds to the number of shock excitations per machine operation cycle; $A_k = a_k e^{-\delta_k((t-n(T_0+\Delta t_m))}$ is the oscillation amplitude of the k-th component with natural damping δ_k; A_n is the amplitude of the oscillation source having frequency ω_k at cycle n with period T_0. For a reciprocating compressor, this cycle normally corresponds to one revolution of the shaft, i.e. the rotational speed, and for a four-stroke combustion engine – two revolutions of the shaft.

Figure 2 shows a sample time waveform of a vibroacoustic signal, which demonstrates the shock pulse sequences per a revolution of the shaft and their change from revolution to revolution. The analysis of a vibroacoustic signal time waveform received from a sensor mounted on the main bearing of a type 2M10-11/42-60 reciprocating compressor (Fig. 2) shows that a strong shock occurs about three degrees before the bottom dead center (BDC). The second shock occurs after the bottom dead center at the moment of closing the valves in the discharge cavity on the side of the crosshead. The increased dynamic bearing loads are due to a failure of the discharge valve in this discharge cavity, and also due to hydraulic impacts. In addition, the signal shows the moments of opening the valves in both discharge cavities.

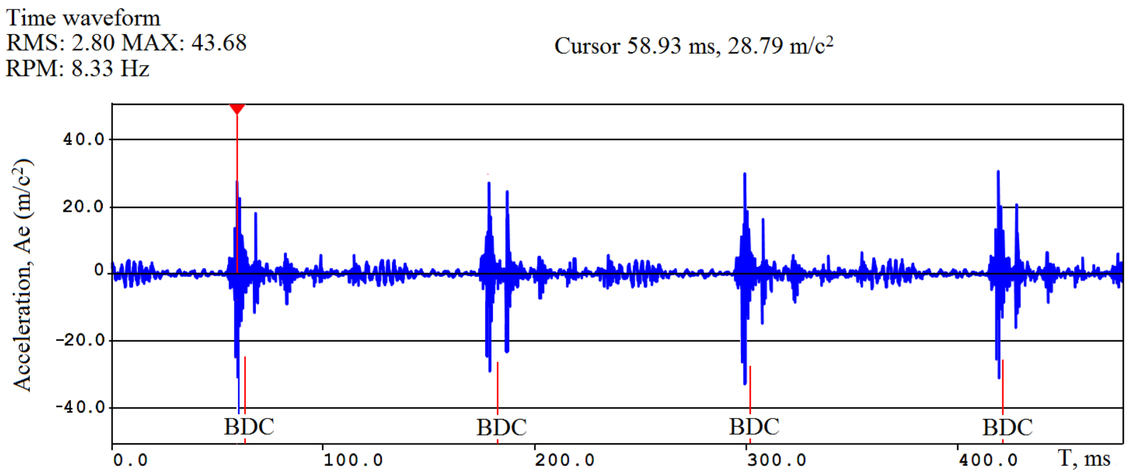

FIGURE 2. A vibroacoustic signal from a sensor mounted on the main bearing

A variety of classes of defects and malfunctions in various machines and mechanisms come from the imbalance of the rotating parts and components. In this case, various defects appear at the unbalanced mass rotation speed and integral multiple speed harmonics and subharmonics. A separate class of defects associated with rolling bearings more often occurs at speeds that are non-integral multiples of the rotational speed.

The imbalance of the moving and rotating masses is the source of the inertia forces F_h and are generally described as follows [5, 6, 7, 11]:

$$F_h(t) = F_1 \sin(\Omega_1 t) . \tag{23}$$

The sources of the stochastic component are described by narrowband noise with center frequency ω_{ξ_k}, which parameters depend on the machine structural features [5, 6, 7, 11]:

$$F_n = F_n(t) = \sum_{k=1}^{\infty} A_{\xi_k}(t) \cos[\omega_{\xi_k} t + \varphi_{\xi_k}(t)] , \tag{24}$$

where classical envelope $A_{\xi_k}(t)$ and phase $\varphi_{\xi_k}(t)$ are random functions that slowly change (in scale ω_{ξ_k}) over time.

In view of $S_{s\Sigma}(t) = S_s^s(t) \cdot [0,5 + S_h^s(t)]$ and (23), (24) can be written as:

$$S_{s\Sigma} = \left[\sum_{n=0}^{\infty} A_n^s \sum_{m=1}^{M} A_m^s \left\{ \sum_{k=1}^{K} A_k^s \sin[\omega_k (t - n(T_0 + \Delta t_m)] \right\} \right] \times \left[0,5 + [F_1 \sin(\Omega_1 t) * h_h^s] \right] . \tag{25}$$

This formula particularly confirms the commonly found in practice modulation of shock frequencies (for example, the frequencies of rolling bearing defects) with the rotational speed of unbalanced mass Ω_1 (Fig. 3).

Here, as an example, the frequencies of rolling bearing defects are presented [16], since, in the vast majority of cases, their values are non-integral multiples of the shaft speed and are easily detected both in the direct spectrum and in the spectrum of the vibroacoustic signal envelope.

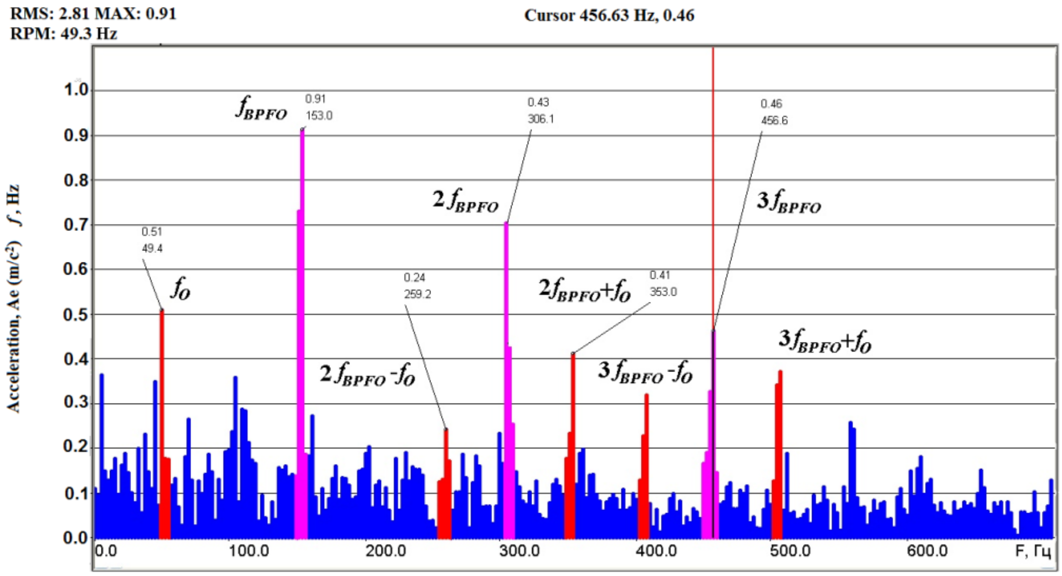

FIGURE 3. A vibroacoustic signal envelope spectrum: modulation of bearing defect frequency f_{BPFO} with the rotational frequency f_0 in case of imbalance

Similarly, we developed the models of the signals appeared in the event of defects and malfunctions in the valves of a reciprocating compressor, rotating parts, couplings, shaft misalignment, increased clearances, loosening, weakening, and others.

CONCLUSION

Thus, we developed the phenomenological structural models of the vibroacoustic signals appeared in case of malfunctions of reciprocating machines, which show that for the diagnostic purposes it is necessary to analyze vibroacoustic signals either in the time domain according to the cyclogram of the reciprocating machine operation or in the envelope spectrum at characteristic frequencies or the noise component [9, 10, 11, 12]. In the event of defects and malfunctions in the centrifugal rotating parts and components, there are excited the vibroacoustic signals, which are advisable to analyze in the direct spectrum or in the spectrum of the noise component envelope. During the diagnostics, an important role is played by the level of the components, whose values are normalized depending on the machine shaft weight, size and speed [12, 14, 17]. Such an approach provides for the real-time condition monitoring of machines and mechanisms that allows us not only to determine their technical state [1, 2, 3, 19], but to detect defects and malfunctions in machinery parts and components [11, 13, 18].

REFERENCES

1. V. N. Kostyukov and A. V. Kostyukov, "Real-time Condition Monitoring of Machinery Malfunctions" in International Conference on Oil and Gas Engineering, OGE-2015, Procedia Engineering 113 (2015), pp. 316-323.
2. V. N. Kostyukov, A. V. Kostyukov and S. N. Boychenko, "Intelligent Machinery Condition Monitoring based on Adaptive Measurements" in Oil and Gas Engineering (OGE-2017), Procedia Engineering 152, (2016), pp. 527-530.
3. A. V. Kostyukov, S. N. Boychenko, A. V. Shchelkanov, and E. A. Burda, "Multi-method automated diagnostics of rotating machines" in *Oil and Gas Engineering (OGE-2017),* AIP Conference Proceedings 1876, 020081-1-020081-6 (2017).
4. Clarence W. de Silva, Vibration and Shock Handbook (CRC Press, 2005), 1872 p.
5. R. B. Randall, Vibration-based Condition Monitoring: Industrial, Automotive and Aerospace Applications (Wiley, 2011), 308 p.

6. C. M. Harris and A.G. Piersol Harris,' Shock and Vibration Handbook (Fifth Edition, McGraw-Hill Professional, 2002), 1457 p.
7. V. N. Kostyukov and A. P. Naumenko, "The Piston Compressor: The Methodology of the Real-Time Condition Monitoring" 2012 Journal of Physics: Conference Series, Vol. 364, Issue 1, pp. 012130.
8. V. N. Kostyukov and A. P. Naumenko, "Condition monitoring of reciprocating machines" in *Proceedings of the 22rd International Congress on Condition Monitoring and Diagnostic Engineering Management* (COMADEM 2009) (Spain, San Sebastián, 2009), pp. 113-120.
9. A. P. Naumenko "Real-time condition monitoring of reciprocating machines" in *The sixth international conference on condition monitoring and machinery failure prevention technologies* (CM2009/ MFPT2009) (Irish, Dublin, 2009), pp. 1202-1213.
10. V. N. Kostyukov and A. P. Naumenko, "System for Condition Monitoring of Reciprocating Machines" in *Proceedings of the 23rd International Congress on Condition Monitoring and Diagnostic Engineering Management* (COMADEM 2010) (Japan, Nara, 2010), pp. 265-272.
11. A. P. Naumenko, "Diagnostics and Condition Monitoring of Piston Compressors" in *The Seventh International Conference on Condition Monitoring and Machinery Failure Prevention Technologies* (CM2010/MFPT2010) (England, Stratford-upon-Avon, 2010), pp. 1-11. ISBN 978-1-901892-33-8.
12. A. P. Naumenko, "Modern methods and means of on-line monitoring of parameters and real-time health monitoring of piston machines" in *8th International Conference on Condition Monitoring and Machinery Failure Prevention Technologies* (2011) (British Institute of Non-Destructive Testing. Printed by Curran Associates, Inc. 2011), Vol. 1 of 2. pp. 809-821. ISBN: 978-1-61839-014-1.
13. V. N. Kostyukov and A. P. Naumenko, "Designing and operation experience of real-time monitoring systems" in *9th International Conference on Condition Monitoring and Machinery Failure Prevention Technologies 2012* (CM 2012 AND MFPT 2012) (British Institute of Non-Destructive Testing. Printed by Curran Associates, Inc. 2012). Vol. 1 of 2, pp. 1053-1060. ISBN: 978-1-62276-433-4.
14. V. N. Kostyukov and A. P. Naumenko, "Regulatory and methodological support of piston compressors vibration analysis monitoring" in *Proceedings of the 26th International Congress on Condition Monitoring and Diagnostic Engineering Management.* (COMADEM 2013) (June 11-12, 2013) (Finland, Helsinki, 2013). – 7 p. ISBN 978-952-67981-0-3-1.
15. V. N. Kostyukov and A. P. Naumenko, "Technology of piston compressors real-time diagnostics and monitoring" in *11th International Conference on Condition Monitoring and Machinery Failure Prevention Technologies* (CM2014/MFPT2014) (British Institute of Non-Destructive Testing. Printed by Curran Associates, Inc. 2014) pp.580-590. ISBN: 978-1-63439-505-2.
16. V. N. Kostyukov, A. P. Naumenko, S. N. Boychenko and I. S. Kudryavtseva, "Diagnostics of rolling bearings by the parameters of the characteristic function" in *12th International Conference on Condition Monitoring and Machinery Failure Prevention Technologies* (CM 2015/MFPT 2015) (The British Institute of Non-Destructive Testing. Printed by Curran Associates, Inc. 2015), pp. 246-249. ISBN: 978-1-5108-0712-9.
17. V. N. Kostyukov and A. P. Naumenko, "Standardization in the sphere of vibrodiagnostic monitoring of piston compressors" in *International Conference on Oil and Gas Engineering, OGE-2015*. Procedia Engineering 113 (2015), pp. 370 – 380. doi: 10.1016/j.proeng.2015.07.290.
18. V. N. Kostyukov and A. P. Naumenko, "Diagnostics and condition monitoring of piston compressors" in The International Journal of Condition Monitoring. (The British Institute of Non-Destructive Testing, 2015). Vol. 5, Issue 3, October 2015, pp.18-24(7). doi: http://dx.doi.org/10.1784/204764215816195892.
19. V. N. Kostyukov and A. P. Naumenko, "Real-time monitoring on vibration- based diagnostics of reciprocating compressors" in *1st World Congress on Condition Monitoring* (WCCM 2017) (British Institute of Non-Destructive Testing. Printed by Curran Associates, Inc. 2017) vol. 1 pp. 350-356 ISBN: 978-1-5108-4475-9.