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Research of Vibration Processes of Bearing Units of Mining Equipment

I.P. Egamberdiev[1], L. Atakulov[2], R.O. Muminov[3], Kh.Kh. Ashurov[4]
[1] Navoi branch of the Academy of Sciences of the Republic of Uzbekistan, Uzbekistan
[2] Navoi State Mining Institute, Uzbekistan
[3] Navoi State Mining Institute, Uzbekistan
[4] Navoi State Mining Institute, Uzbekistan, kh_ashurov@mail.ru

ABSTRACT

The article presents the relationship between the technical condition of the elements of bearing supports of mining equipment and spectral characteristics based on the analysis of the high-frequency component of the vibration signal, allowing to determine the type of defects and predict their development.

Various methods of diagnostics of rolling bearing defects are considered and the method of narrow-band analysis of spectral components is recommended for industrial development. In addition to the General level of vibration, reference spectral masks limiting the levels of vibration intensity in individual spectral bands can be used as diagnostic signs of changes in the technical condition.

As a result of the study, the dependences of vibration parameters on defects of bearing bearings were established, on the basis of which the method of in-place assessment of the technical condition of mining equipment and prediction of the dynamic quality of bearing assemblies, which improves the reliability and efficiency of their use, is justified.

Key words: vibration Signal, cage, separator, pulse, wear, spectral density, frequency, amplitude, oscillation, stiffness.

1. INTRODUCTION

The whole variety of oscillatory processes in mining machines can be considered in the form of forced and natural vibrations. As a carrier of information about the technical condition of the nodes of mining machines, these two types of oscillatory processes can be used. Moreover, forced (usually low-frequency vibrations) serve as an indicator of the manufacturing quality of machine components, while natural vibrations in the mid or high-frequency range carry information about the deterioration of the technical condition caused by processes that begin, as a rule, with contact interactions and are most often initiated by wear [3,5,6].

The vibration parameters of units with rolling bearings are largely determined by the design features of the bearing unit (for example, radial, angular contact, twin bearings) and the unit itself (for example, with the horizontal and vertical position of the rotor, the ratio of static loads, etc.).

II. PROPOSED METHODOLOGY

Vibration, depending on the nature of the forces that excite it, can be either deterministic (often periodic) or random .

The analysis of both periodic and quasiperiodic signals is based on the Fourier transform

$$S(\omega)=\frac{1}{2\pi}\int_{-\infty}^{\infty} y(t)\cdot e^{-j\omega t}dt, \qquad (1)$$

Where $S(\omega)$ - signal spectral density; $y(t)$ - time signal.

Random signals, unlike deterministic ones, have continuous spectra. In this regard, the change in the parameters of the vibration signal as a result of changes in the technical condition of the bearings is convenient to model not only by varying the ratio of spectral amplitudes A_k discrete components $S(k\omega_0)$, $k=1,2,....,n$, but also by introducing additional noise excitation $\omega_{III}(t)$ c uniform spectrum $S_{III}(t)$ in the considered frequency range:

$$x(t)=\sum_{k=1}^{n} A_k(t)\sin[k\omega_0 t+\varphi_k(t)]+\xi(t), \qquad (2)$$

Where $k\omega_0$ - narrow-band average frequency; $A_k(t)$ - random, slow compared to $T_k=\frac{2\pi}{k\omega_0}$ variable envelope of a narrow-band process; $\varphi_k(t)$ - random, slowly changing phase; $\xi(t)$ - noise (random) component, vibration.

This form of presentation of the vibration signal reflects quite well the growth of the noise component with an increase in the degree of wear of the contacting surfaces in the rolling bearings, is sensitive to deterioration of lubrication conditions or the loss of lubricity due to aging, that is, in all cases where it is necessary to take into account the appearance or change of a random component, resulting from the interaction of friction forces or shock disturbances.

The practice of diagnosing mining machines shows that vibrations include a high-frequency process, modulated in amplitude by a low frequency.

Forced vibrations of the reference node can be represented as a dynamic system with n degrees of freedom

$$[\vec{M}][\ddot{\vec{X}}]+[\vec{K}][\dot{\vec{X}}]+[\vec{C}][\vec{X}]=[\vec{G}], \qquad (3)$$

Where $[\vec{M}],[\vec{K}]$ и $[\vec{C}]$ - symmetric matrices describing the properties of inertia, stiffness and dissipation, respectively; $[\vec{G}]$ - n - dimensional vectors characterizing disturbing influences; $[\vec{X}]$ - n - dimensional output vector characterizing vibration.

During operation of the support unit, the magnitude of the radial clearance in the bearings increases, which leads to a change in parameters:

$$[\vec{C}(s,t)];[\vec{K}(s,t)] \quad \text{and} \quad [\vec{G}(s,t)]$$

The determination of the disturbing influences and the dynamic characteristics of the support directly during operation is difficult. However, the magnitude of the deviations arising - the type and depth of the defect can be judged by the output information from the sensors installed at the specified locations:

$$\vec{X}(s,t)=L\cdot\vec{G}(s,t),$$

Where L – system operator.

Oscillations with frequencies $k\omega_1$, by the assessment of which it is possible to determine the state of the node, i.e. the presence of gaps and defects are perceived by the vibration sensor mounted on the housing of the support unit. The information received from the sensor corresponds to the accepted model of the diagnostic condition of the bearing assembly.

Taking into account the time variation of damping parameters [$\vec{C}$](t) stiffness [$\vec{K}$](t) fundamentally necessary for bearing bearings of drilling machines, where the simulation of vibrations without taking into account the change in stiffness of bearing bearings over time is incorrect. Accounting for changes in stiffness over time [$\vec{K}$](t) leads to amplitude modulation of the vibration signal, while the time dependence of the damping coefficient [$\vec{C}$](t) connected with the frequency modulation of the signal. The development of a defect causes a trend in the depth of modulation, i.e. an increase in the amplitudes of the combination frequencies with the operating time, and does not affect the amplitudes of the polyharmonic series of the main frequencies of the node excitation [**Error! Reference source not found.**,**Error! Reference source not found.**,**Error! Reference source not found.**,]. In general, the amplitude-modulated signal has the form

$$x(t)=A[1+\mu B(t)]\sin(\omega_0 t+\varphi_0), \qquad (4)$$

Where ω_0 - carrier frequency; A - amplitude; μ - modulation depth (varies from 0 to 1); $B(t)$ can be represented as

$$B(t)=\sum_{k=1}^{n} c_k \sin[k\Omega t+\varphi_k], \qquad (5)$$

Where Ω - modulation frequency ($\Omega \prec \omega_0$).

In this case, the resulting amplitude-modulated process has the form

$$x(t)=A\left[1+\sum_{k=1}^{n}\mu_k \sin(k\Omega t+\varphi_k)\right]\sin(\omega_0 t+\varphi_0), \qquad (6)$$

Where μ_k - partial modulation factor; Ω - angular modulation frequency; φ_k and φ_0 - phase shifts.

This complex process can be decomposed into the sum of simple harmonic oscillations:

$$x(t)=A\left\{\sin(\omega_0 t+\varphi_0)+\sum_{k=1}^{n}\frac{\mu_k}{2}\sin[(\omega_0+k\Omega)t+\varphi_0+\varphi_k]+\sum_{k=1}^{n}\frac{\mu_k}{2}\sin[(\omega_0-k\Omega)t+\varphi_0-\varphi_k]\right\} \qquad (7)$$

where the first term is the oscillation of the carrier frequency; second term -n of the oscillations of the upper lateral frequencies $\omega_0+k\Omega$; third term -n of the lower lateral frequencies $\omega_0-k\Omega$.

The width of the spectrum of the amplitude-modulated process is equal to twice the width of the envelope spectrum. The envelope is extracted using an amplitude detector, the output of which produces a signal of the form

$$x(t)=A\sum_{k=1}^{n}\frac{\mu_k}{2}\sin[(\omega_0+k\Omega)t+\varphi_0+\varphi_k] \qquad (8)$$

Parameters characterizing the depth of amplitude modulation of the main frequencies of forced vibrations are used as diagnostic signs of operational defects in bearing bearings of drilling machines. Sensitive diagnostic signs of local defects of contacting surfaces such as caverns from fretting, pitting, and chips are n-dimensional vectors formed from the components of the envelope spectrum in the zone of one of the forced frequencies of the defective node.

Therefore, the next step by filtering is a slowly changing oscillatory process, i.e. we get the envelope of the original high-frequency signal. Then, again performing the Fourier transform, we get the "envelope spectrum." Thus, it was possible to separate in space the frequency components associated with characteristic defects of the bearings.

III. RESULT ANALYSIS

This is evidenced by the good convergence of the experimentally obtained spectral frequencies with the results of calculations of the characteristic "bearing frequencies" (Table 1.).

We adopted a set of harmonics in the spectrum of the envelope of the high-frequency signal and the partial modulation coefficients as a sign of the technical state: modulation frequency $k\Omega$ determines the type of defect, and the modulation depth - the degree of development of defects.

Table 1: Calculation of characteristic frequencies of bearings of mining equipment

№	m, kg	n, pcs.	d, mm	D, mm	B, mm	Z, pcs.
3620	13	2	100	215	73	14
3628	36,2	2	140	300	102	14
42244	39	1	220	400	65	20
46244	36	1	220	400	65	16
8260Г	44,2	1	300	420	95	22
42152	30,9	1	260	400	65	21
32314		5	70	150	35	12
66314	3,2	4	70	150	35	14
3620	$D_{тк}$, mm	$F_{об}$, Hz	$F_{тк}$, Hz	$F_с$, Hz	$F_в$, Hz	$F_н$, Hz
3628	31	10	24,26	8,02	27,72	112,28
42244	32	2	5,05	0,8	16,8	11,2
46244	40	2	8,82	1,12	21,12	26,88
8260Г	44	2	6,91	0,86	18,24	12,04
42152	44	2	8,06	0,88	24,64	19,36
32314	40	2	8,12	0,88	23,52	18,48
66314	20	50	120	20,46	354,5	245,5
3620	25,4	50	102,5	19,23	430,78	269,2

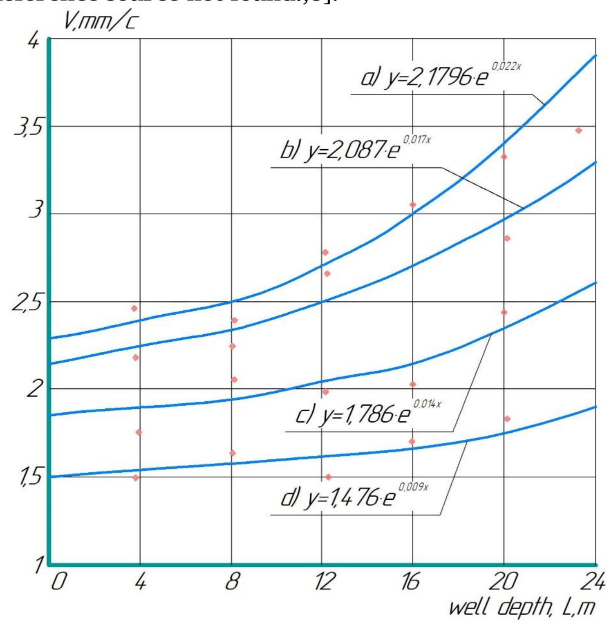
Note: $f_{вр}$ – shaft speed; $f_в$ – rolling frequency of rolling elements along the inner ring; $f_н$ – rolling wheel rolling frequency on the outer ring; $f_{тк}$ – rotation frequency of rolling elements; f_c – separator speed;

A direct study of vibration changes in the support nodes under various operating conditions of mining equipment was not carried out, as this is associated with great difficulties of a technical and organizational nature. It is difficult, for example, to transmit signals from sensors mounted on rotating parts of equipment, complicated by the need to conduct an experiment in a production environment.

In order to find out the laws of vibration formation on the rotational feed mechanism of the СБШ-250 machine at various operating modes in real production conditions, a series of experiments was carried out.

The vibration measurement data of the support units operating under the same conditions, taking into account the ergodic property of random processes during material processing, were recorded on the VIBXPERT II instrument, which allows for the frequency analysis of the vibration signal. To convert mechanical vibrations into digital form, a piezoelectric accelerometer with a wide frequency range was used. Piezo accelerometers have the smallest measurement error and have a fairly wide frequency range from 5 Hz to 25 thousand Hz. The mass of the sensor is small in relation to the mass of the studied objects and therefore does not affect the vibrational characteristics.

In fig. 1. shows the dependence of the intensity of vibration of the bearing bearings of the rotator and the support node from the depth of the well. Their processing was carried out according to a technique developed on the basis of the theory of random functions [**Error! Reference source not found.**,**Error! Reference source not found.**,**Error! Reference source not found.**,8].



a) support unit, 120 rpm
b) support unit, 60 rpm
c) rotator, 120 rpm
d) rotator, 60 rpm

Figure 1: The dependence of the intensity of vibration of the bearing on the depth of the well.

An increase in drilling depth leads to an increase in load, which in turn leads to an increase in vibration intensity. From the graph of Fig. 1. it can be seen that when drilling up to 12 m, the vibration amplitude increases linearly, and after 15 m its sharp increase begins, and this leads to premature failure of the drilling rig parts (bearings).

From the experimental dependences it can be seen that the vibration intensity directly depends on the rotational speed of the drill string and its length.

Statistical Analysis of Random Functions - $A_0(t)$ allowed to obtain their following probabilistic characteristics: m_a, m_v — mathematical expectations; D_A – variance; σ_a - standard deviations; v_a - coefficient of variation; A_{min}, A_{max} – maximum and minimum values of random functions; $K_A = m_A + 3\sigma_A / m_A$, dynamic factors.

The results of statistical analysis showed that dispersions increase with increasing length of stavas D_A. These parameters increase significantly if the bit rotation frequency is increased to 120 min^{-1}. Figure 2 shows the dependences of the dispersion on the length of the drill string and the bit rotation speed.

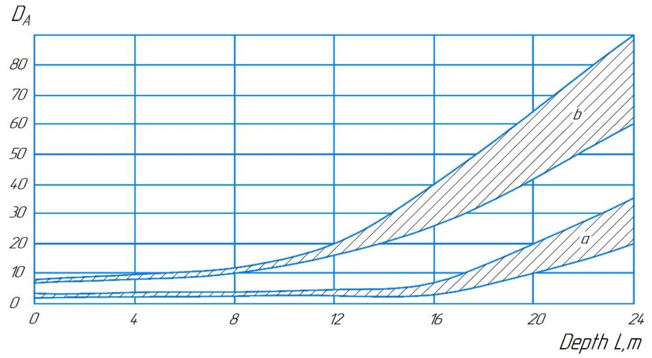


Figure 2: Dependences of the dispersion DA for the reference node on the drilling depth and operating conditions of the machine: a - 60 rpm; b - 120 rpm.

The given dependences have a critical point at which their steepness sharply changes. Before and after this point, the statistical values of the dependencies $D_a=f(l)$ and $K_a=f(l)$ for two values n_c (60 and 120 min^{-1}) approximated fairly well by linear functions. If we fix these critical points on the abscissa, we get: for n_c=60 min^{-1} $l_к$= 15 m, at n_c=120 min $^{-1}$ $l_к$=12 m. Also with increasing n_c dispersion dispersion region grows D_p, determined according to several parallel experiments. The dispersion value varies from 5 to 20% depending on the length and frequency of rotation of the stav. An increase in the length of the drill string leads to an increase in the absolute values of the variance D_A and its dispersion regions.

The experiment showed that forcing the operating modes of the machine and increasing the depth of the well (length of the stave) lead to an increase in the dynamics of the system represented by random functions A*(t).* Decrease in expectation m_A. and the occurrence of rebounds of the bit from the bottom lead to a decrease in machine productivity precisely because the operating modes are forced (dynamic paradox) [**Error! Reference source not found.**]. And in this sense, increasing the speed of the bit is unpromising, it is better to increase the axial force.

In fig. 3. shows the spectra of the envelope of the high-frequency vibration of the bearing support of mining machines for various types of defects. The spectrograms clearly show characteristic diagnostic features (harmonics with frequencies that are multiples of rotational frequencies), indicating increased wear of the separator, rolling elements - a malfunction of the unit.

For the formation of the envelope spectrum, a prerequisite is the choice of the frequency range, and comparable in power components of different nature should not fall into this range. Most of the defects in bearings that occur during their manufacture and use affect the frequency spectrum of their vibrations. We have established frequency ranges in which it is most effective to control defects that arise and develop during the operation of mining machines.

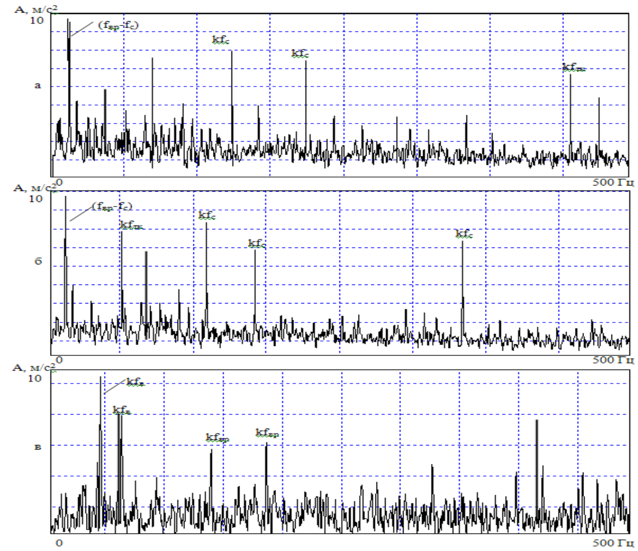


Figure 4: Envelope spectrum of high frequency vibration bearings of the rotator

a - wear of rolling elements; b - wear of the separator; c - uneven wear (shells on the treadmill of the inner ring).

An analysis of the data shows that the vibrational state of mining machines significantly depends on the operational parameters. To increase the efficiency of the use of mining equipment, in-place monitoring of the technical condition of the bearing units according to the results of measurements of vibration parameters is necessary.

The vibration spectrum of mining machines includes low frequencies in the region of 2-12 Hz, medium frequencies of 20-60 Hz, as well as high-frequency vibrations caused by processes occurring in the bearings themselves. Frequency ranges were established (rotator F = 4-6 kHz, reference node F = 6-8 kHz, compressor F = 8-10 kHz, pump F = 8-10 kHz), in which it is most efficient to control defects that arise and develop during the process operation.

IV. CONCLUSION

The interrelations of the technical condition of the elements of the bearing bearings of mining machines and spectral features, established on the basis of the analysis of the high-frequency component of the vibration signal, make it possible to determine the type of defects and predict their development. In particular, it was found that an increase in the radial clearance is characterized by an increase in the frequency components $k \cdot f_2$ in the envelope spectrum (f_2 – shaft rolling frequency through rolling elements, k = 1, 2, 3, … - harmonic number). For bearings of the rotator and the support unit №3620, №3628, №8260G, №42244 and №46244, these frequencies are respectively 57.4 Hz; 24.3 Hz; 38.8 Hz; 35.2 Hz, 27.6 Hz.

Based on theoretical and experimental studies in complex mechanical systems, a method for analyzing the spectral data of vibrodiagnostics has been developed, which allows taking into account the destructive effect of stochastic processes in complex technical systems and provides recognition of developing defects in mining equipment not available to traditional methods.

REFERENCES

1. Gerike B.L., Abramov I.L., Gerike P.B. Vibrodiagnostics of mining machinery and equipment. // - Kemerovo, 2007.-S. 167.
2. Barkov A.V., Barkova N.A. Vibration diagnostics of machinery and equipment. SPb .: SPbGMTU, 2004, 156 p.
3. Egamberdiev I.P. Monitoring the technical condition of drilling rigs. // Mountain Information and Analytical Bulletin. - 2007 - No. 8.– S. 189-192.
4. Egamberdiev I.P. - Justification of the method for assessing the technical condition of drilling rigs. Diss. for the degree of Ph.D. - M., Moscow State Mining University, 2008.
5. Renata Klein. “Comparison of methods for separating vibration sources in rotating machinery”, Mechanical Systems and Signal Processing, March 29, 2017, Elsevier.
6. Devendiran S, Manivannan K. Vibration based condition monitoring and fault diagnosis technologies for bearing and gear components — A review. International Journal of Applied Engineering Research, 2016, 11 (6): 3966–3975.
7. Moazen Ahmadi A, Petersen D, Howard C. A nonlinear dynamic vibration model of defective bearings — The importance of modeling the finite size of rolling elements. Mechanical Systems and Signal Processing, 2015, 52–53: 309–326.
8. Sh.N. Yaxshiyev, Kh.Kh. Ashurov, A.J. Mamadiyarov Dynamics of Spindle Assembly of Metal-Cutting Machine // International Journal of Engineering and Advanced Technology (IJEAT) ISSN: 2249 – 8958, Volume-9 Issue-3, February 2020. 3121-3125.