The increasing of fixed mining machines resource rates by diagnostic maintenance improving

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Abstract
We established the insufficient efficiency of rotary type FMM existing diagnostic methods, where the use of diagnostic information on rotor forced spatial vibrations from operational defects effect is not provided. In existing technical condition diagnostic monitoring technologies, the factor of machine working body rotation axis stability, dynamic misalignment rise, and new dynamic loads mechanism introduction are not fixed in due time. The new diagnostic features of facts registration of operational defects rise and impact on load growth (particularly axial loads) upon working body, its support and base members, which lead to their recourses sharp decrease, were developed, investigated and suggested.
Key words: ROTOR, VIBRATIONS, SUPPORT, LOAD, DIAGNOSTIC, RECOURSE

The modern development of mining machinery, particularly fixed machines, is characterized by heavy-duty machines development and implementation. The improvement of fixed mining machines (FMM) organization and technical operation methods and vibration diagnostics implementation affect significantly iron ore raw materials extraction and processing efficiency. Conducted FMM operating reliability researches by means of technical diagnostics tools including machine technical condition evaluation method by vibration signal show that in most cases FMM have working technical condition parameters when sufficient low resource rate and high repair intervention rate. The existing diagnostics method weaknesses are the following: unreliable diagnostic information obtaining method; mining machinery diagnostic model inadequate rationale; unexplored diagnostic models transmitting functions properties. Established weaknesses complicate the diagnostic information interpretation and require new technical condition diagnostic methods. There is the lack of object informative mathematical models for FMM diagnostic parameters selection, which can explain the FMM working body spatial motion. The following tasks should be solved on the base of such models: the selection of diagnostic parameters covering all possible faults, their optimum number, trifling value, minimum measurement time and maximum reliability.

As the result of conducted researches of main FMM components, inoperativeness common reasons have been established and individual machine components failure coefficient has been discovered. The statistic analysis of FMM components failure coefficients showed that additional radial and axial loads on rolls support rise under the effect of uncontrolled operational factors (installation, maintenance and repair quality, regulations and operating conditions non-observance). These loads cause FMM life characteristics 4-folds decrease from design values. The diagnostic features of life FMM decreasing defects may be found out by means of spectrum analysis, but their identification becomes untenable due to existing diagnostic model inadequacy.

The availability of new diagnostic features at low-frequency range of subharmonic components (fractional conciseness frequency to the first harmonic of the rotor) is established when FMM vibrations spectrum analysis with low resource characteristics (Fig.1). The subharmonic components spectrum is close to the known [2] but in this case, harmonic amplitudes are much higher in axial and horizontal directions.

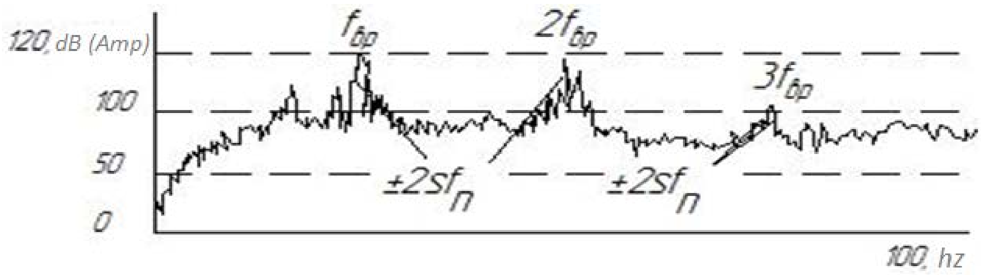
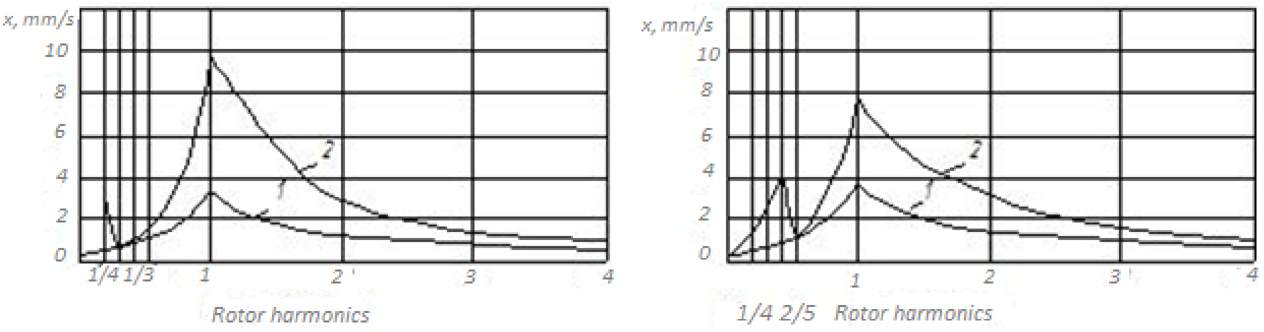


Figure 1. Spectrum of (vertical) vibrations changes for FMM when there are evidences of dynamic misalignment caused by asymmetric operating changes of system rotor - bearing - body compliance

The parameters of power loads on rotor and its support vary for FMM where significant life reduction was fixed, primarily working body support. This occurs because of operational defects, which cause unit dynamic misalignment, body deformation etc. We detected subharmonic and rotor harmonic parameters ratios, which are typical for technically sound unit, distortion of vibroacoustic signal as part of diagnostic information properties of these defects.

For example, subharmonics of 1/2, 1/3, 1/4, 2/5 multiplicity of rotor speed and vibration of second and third rotor harmonic frequency (Fig.2) are detected in FMM vibration spectrum if any operating defects. The friction bearing oil layer nonconservative forces give rise to subharmonics of 1/2 multiplicity of rotor speed. It has been established that such rotor machine behaviour takes place if any mentioned defects when support spatial position changing up to body part distortion similar to established regularities in papers [1].



a) body distortion in thrust and radial bearing area of FMM body;

b) body distortion in mounting bearing area.

Figure 2. The regularity of vibration speed amplitude distribution of rotor harmonics and subharmonics when unit FMM bodies distortion: **1** – defect-free unit; **2** – body distortion.

In addition, the dynamic misalignment and body distortion diagnostic features (Fig.3) appear in the spectrum if any active subharmonics; the diagnostic model is described in paper [2].

The introduction of such a combination of uncontrolled operational defects by modern diagnostic methods primarily may be caused by asymmetric support hardness, installation faults, support operational damage, support systems characteristics change, critical to operating speed ratio, vibrations deformation degree of FMM rotor body, and also by bodies nonuniform heat deformations. All these cause uncontrolled spatial variation, loads growth, and there regularities have not been investigated yet.

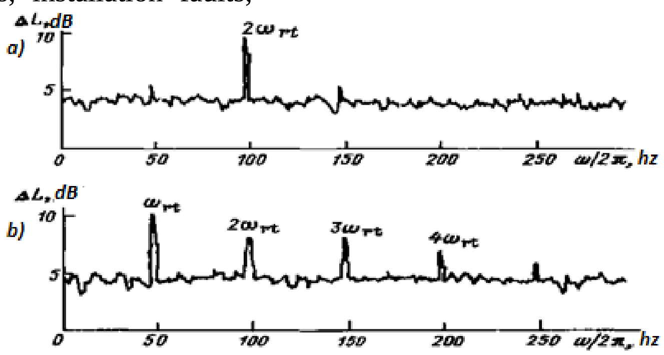


Figure 3. The envelope vibration spectra of working body support when unit dynamic misalignment: a) rolls centershift; b) rolls spindles break

Thuswise, in papers [2, 3, 4], the forces acting on the rotating rotor on the ideal support in coordinates, which is rigidly connected with them, are determined from rotor rotation equation

$$m_{rt}\,\omega_{rt}^2\left(e_{fix}+\Delta e\right)-c_{rt}\,\Delta e=0 \qquad (1)$$

When support rigidity final value C_i, the sum of forces acting on support is of the form of (1).

At that, the rotor deformation value is considered including bending vibration eigenfrequency on fixed support (first critical frequency):

$$\Delta e=e_{fix}\,\omega_{rt}^2/(\omega_0^2-\omega_{rt}^2) \qquad (2)$$

The mass centre vertical vibrations of considered rotor are diagnostic feature of rotor and its support loads growth in traditional diagnostics. These vertical vibrations are characterised by shift, which equation is of the form

$$y(t)=\left[e_{fix}\omega_0^2/(\omega_0^2-\omega_{rt}^2)\right]\,\cos\,(\,\omega_{rt}\,t-\varphi\,) (3)$$

In this diagnostic model however, the emphasis is placed rotor vertical displacement amplitude determining, which coincides with all the standards requirements of technical condition determining. And the attention is not paid to that the rotor rotation axis is shifted by a certain angle under asymmetric support rigidity when inertia axis positional stability. In this case, the diagnostic features area is distorted.

As a result, we can make a conclusion that FMM rotor executes more complicated spatial variations, which are not considered in assumed diagnostic models of fixed rotor machines [2, 3, 4, 5].

With the aim of explanation of new loads sources, which reduce machine life characteristics, let us consider FMM rotor as flexible beam on two yielding supports, each of which has double-beat system, which is fulfilled in body construction. In this case, complicated rotor spatial variations are formed under asymmetric rotor and body support rigidity. These rotor spatial variations form the axial loads and thus the diagnostic features reflecting the impact of operating defects, which reduce support components life.

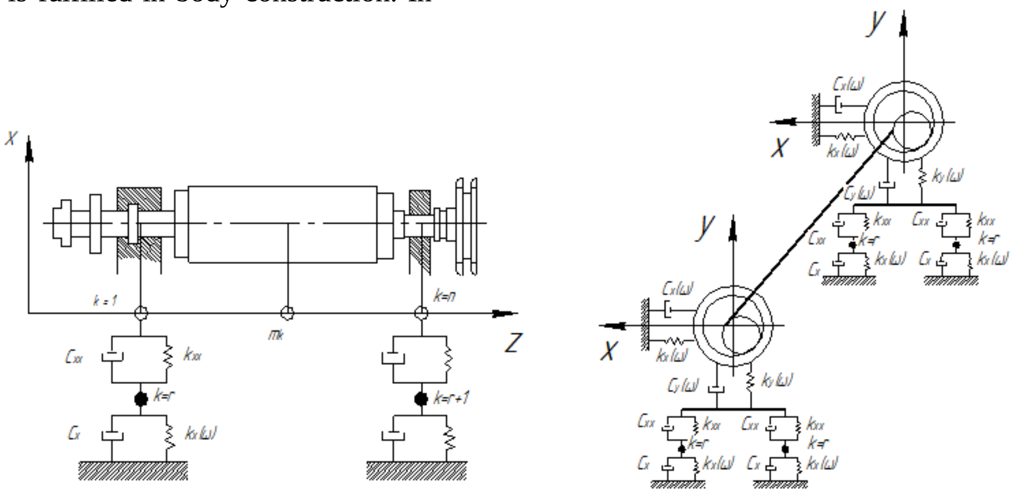


Figure 4. The pattern of FMM working body balance on four supports and explanation of its spatial variations cause under asymmetric support rigidity

The explanation of rotor spatial variations from the operating loads effect including asymmetric support rigidity defects is shown in mathematical model [5].

Having applied the rotor C centre-of-mass motion theorem and also angular momentum change at relative motion towards the centre of inertia theorem to rotor motion in coordinates *OXYZ* (in coordinate system $CX_CY_CZ_C$, which axes are parallel to the axes of coordinate system *OXYZ* and the coordinate origin moves progressively with rotor C centre of mass, Fig. 6), the rotor motion equation is obtained:

$$\begin{cases} M\begin{bmatrix}\ddot{x}_c\\ \ddot{y}_c\end{bmatrix}+\begin{bmatrix}Cl_{xx} & Cl_{xy}\\ Cl_{yx} & Cl_{yy}\end{bmatrix}\begin{bmatrix}\dot{x}_l\\ \dot{y}_l\end{bmatrix}+\begin{bmatrix}Kl_{xx} & Kl_{xy}\\ Kl_{yx} & Kl_{yy}\end{bmatrix}\begin{bmatrix}x_l\\ y_l\end{bmatrix}+\begin{bmatrix}Cr_{xx} & Cr_{xy}\\ Cr_{yx} & Cr_{yy}\end{bmatrix}\begin{bmatrix}\dot{x}_r\\ \dot{y}_r\end{bmatrix}+ \\ \qquad \begin{bmatrix}Kr_{xx} & Kr_{xy}\\ Kr_{yx} & Kr_{yy}\end{bmatrix}\begin{bmatrix}x_r\\ y_r\end{bmatrix}=Mu\Omega^2\begin{bmatrix}\sin\Omega t\\ \cos\Omega t\end{bmatrix}\ ; \\ \begin{bmatrix}B\ddot{\varphi}+A\Omega\dot{\beta}\\ -B\ddot{\beta}+A\Omega\dot{\varphi}\end{bmatrix}+\begin{bmatrix}Cl_{yx} & Cl_{xy}\\ Cl_{xx} & Cl_{yy}\end{bmatrix}\begin{bmatrix}\dot{x}_l\\ \dot{y}_l\end{bmatrix}\frac{l}{2}+\begin{bmatrix}Kl_{yx} & Kl_{yy}\\ Kl_{xx} & Kl_{xy}\end{bmatrix}\begin{bmatrix}x_l\\ y_l\end{bmatrix}\frac{l}{2} \\ \qquad -\begin{bmatrix}Cr_{yx} & Cr_{yy}\\ Cr_{xx} & Cr_{xy}\end{bmatrix}\begin{bmatrix}\dot{x}_r\\ \dot{y}_r\end{bmatrix}\frac{l}{2}-\begin{bmatrix}Kr_{yx} & Kr_{yy}\\ Kr_{xx} & Kr_{xy}\end{bmatrix}\begin{bmatrix}x_r\\ y_r\end{bmatrix}\frac{l}{2} \\ \qquad +\begin{bmatrix}Cml_{xx} & Cml_{xy}\\ Cml_{yx} & Cml_{yy}\end{bmatrix}\begin{bmatrix}\dot{x}_l\\ \dot{y}_l\end{bmatrix}+\begin{bmatrix}Kml_{xx} & Kml_{xy}\\ Kml_{yx} & Kml_{yy}\end{bmatrix}\begin{bmatrix}x_l\\ y_l\end{bmatrix} \\ \qquad +\begin{bmatrix}Cmr_{xx} & Cmr_{xy}\\ Cmr_{yx} & Cmr_{yy}\end{bmatrix}\begin{bmatrix}\dot{x}_r\\ \dot{y}_r\end{bmatrix}+\begin{bmatrix}Kr_{xx} & Kr_{xy}\\ Kr_{yx} & Kr_{yy}\end{bmatrix}\begin{bmatrix}x_r\\ y_r\end{bmatrix}=\begin{bmatrix}0\\ 0\end{bmatrix} \end{cases}$$

where Cl_y, Kl_y, Cr_y, Kr_y $(\mathrm{i,j} = \{x, y\})$ – damping and stiffness coefficients for forces calculation in left and right bearings;

$Cml_y, Kml_y, Cmr_y, Kmr_y$ $(\mathrm{i,j} = \{x, y\})$ – damping and stiffness coefficients for moments calculation in left and right bearings;

A - axial moment of rotor inertia;

B - equatorial moment of rotor inertia;

$x_c = \frac{x_l + x_r}{2}$ and $y_c = \frac{y_l + y_r}{2}$ – rotor centre of mass coordinates (P. C in Fig. 5).

The system solution of shafting and body general vibrations, which become diagnostic features of radial loads growth, for working body axis coordinates at point k determining, is of the form:

$$x_k = \overline{x}_k \exp(i\omega t)\,;(k = 1,2,\dots,n);$$
$$y_k = \overline{y}_k \exp(i\omega t)\,;(k = n+1,\dots,2n);$$

where x_k, y_k – the complex amplitude of mass system vibration characterized by vibration module and phase.

For new axis loads investigation, let us consider the lateral vibrations equation in plane xz when spatial displacement according Fig. 4:

$$\frac{\partial^2}{\partial z^2}\left[EJ(z)\frac{\partial^2 x(z,t)}{\partial z^2}\right] = q_x(z,t),$$
$$\frac{\partial^2}{\partial z^2}\left[EJ(z)\frac{\partial^2 y(z,t)}{\partial z^2}\right] = q_y(z,t),$$

where E – roll material elasticity modulus; ; z — coordinate along the roll axis; $J(z)$ — roll inertia equatorial moment at the point z; $x(z,t)$, $y(z,t)$ — rotor displacement in horizontal and vertical plane; $q_x(z, t)$, $q_y(z, t)$ - distributed load including all the forces applied at point z at time point t.

The forces $q_x(z,t)$, $q_y(z,t)$ from rotor and its support operating defects impact cause rotor spatial displacements within the vibration cycle shown in Fig. 5.

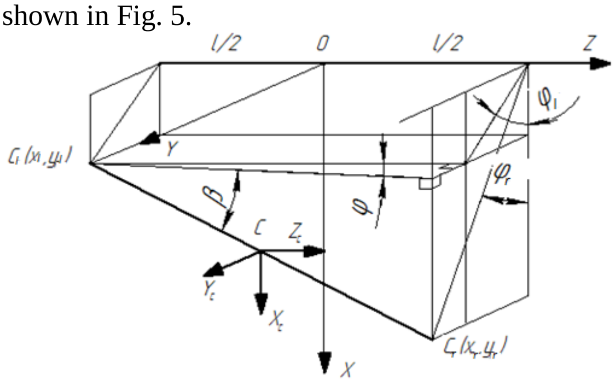


Figure 5. Values, which determine rotor position in space

Consequently, in accordance with spatial displacement [5], axial force F_z act on rotor and its support. These forces are proportional to rotor centre of mass displacement projections in Z-axis direction by the formula:

$Y_z = Y\sin\beta$, $X_z = X\sin\varphi$

where β, φ - Resal angles (Fig. 6) are determined within small quantity of the first order by formulas $\beta = \frac{x_r - x_l}{l}, \varphi = \frac{y_r - y_l}{l}$ [5].

When loads mechanism in [5] studying, it is recommended to improve the differential equation system of shafting vibrations for the consideration of lubricant film reactive forces effect in the bearing F_k. These forces lead to axial components increasing when roll position skewing.

$$m_r\ddot{x}_r + \beta_r\dot{x}_r - F_k(x) + K_x(\omega)x_r + C_x(\omega)\dot{x}_r = 0$$
$$m_r\ddot{y}_r + \beta_r\dot{y}_r - F_k(y) + K_x(\omega)y_r + C_y(\omega)\dot{y}_r = 0$$

(r = 1,2,…,n)

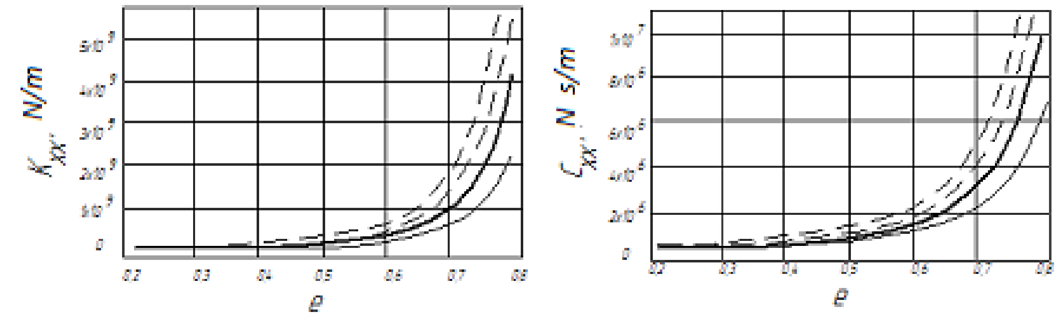
where $K_x(\omega)$, $K_y(\omega)$, $C_x(\omega)$, $C_y(\omega)$ – stiffness and damping coefficients for body bearing part, n – bearings number.

Bearing hydrodynamic forces near the stable balance position may be linearized, i.e. expressed through the stiffness K_{xx}, K_{xy}, K_{yy}, K_{yx} and damping C_{xx}, C_{xy}, C_{yy}, C_{yx} coefficients (Fig. 4).

$$F_k(x) = -\left(K_{xx}x + C_{xx}\dot{x} + K_{xy}y + C_{xy}\dot{y}\right)$$
$$F_k(y) = -\left(K_{yy}y + C_{yy}\dot{y} + K_{yx}x + C_{yx}\dot{x}\right)$$
$$x = x_k - x_r\,;\, y = y_k - y_r$$

where x_r, y_r – bearing body displacement.

We must pay attention to that rotor bearings axis loads increase lubricant film reactive forces in the bearing F_k. These reactive forces depend significantly on bearing skewing rate, since the values K_{xx} and C_{xx} are changed in accordance with diagrams shown in Fig. 6.



Figur $)2^0$, γ =

The loads F_z, F_k are in power dependence on axis forces Y_z, X_z level. These loads are not taken into account in the design calculations and according to conducted researches, they cause the FMM life reduction to 75% of the design values.

Conclusions

1. As can be seen from the above, if there is significant amount of regulatory, methodological and technical FMM technical condition assessment tools, the existing diagnostic methods insufficient efficiency has been established. It consists in explanation absence of high intensity repair interference (planned and unplanned) in FMM working process, and which is most important, the rotor type FMM low resource indicators.

2. The significant working body support axis load, which is not considered in design calculations, has been fixed when FMM operating. The most typical defects list has been established; more adequate FMM technical condition diagnostic model has been proved; these defects diagnostic features list has been suggested for FMM low resource indicators explanation.

3. The impact assessment of operational defects on the technical condition, resource parameters, working efficiency, FMM hidden defects affecting the stability of the working body rotation axis, the dynamic misalignment, dynamic loads forming mechanism, has been determined on the basis of the FMM technical condition diagnostic model analysis and accumulated experience. It has been established that the dynamic load level growth depends significantly on the elastic inertial and damping characteristics parameters changes of the rotor system bearings, this leads to FMM lifetime characteristics reduction by 75%.

4. The conducted researches results show the method of emergency outages preventing and equipment life increasing when using of more accurate diagnostic models and more correct operational defects identification, carrying out of equipment components dynamic properties flexible correction in maintenance regulation and corresponding defects correction.

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