Method for automated diagnostics of the technical condition of a feed crusher

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Abstract. The material of the scientific article represents a separate stage of research on a topical topic dedicated to increasing the economic efficiency of feed mills. The solution to this problem significantly depends on the technical condition and quality of the work process carried out by technological equipment. Based on an analysis of the state of the issue on the topic under consideration, it was established that the most important and energy-intensive operation of the feed preparation process is grinding, performed in feed crushers. A study of the operating conditions of hammer crushers revealed the least reliable working element - the grinding rotor, the technical condition of which is determined by wear of the support bearings, increased vibration and imbalance. In order to substantiate the method of automated diagnostics of rotor malfunctions during operation, determine the accuracy of manufacturing crushers at factories, and assess the quality of service and repair at service enterprises, a scheme for static and dynamic loading of the rotor has been developed.

1 Introduction

An important place in the diet of animals is occupied by compound feeds containing quite a lot of easily digestible nutrients from grains of cereals and legumes. The material and technical basis of feed mills in our country is represented by feed workshops for the production and preparation of feed, which have different levels of technical equipment with automated and digitalized technological means. The economic efficiency of the operation of feed mills significantly depends on compliance with the parameters and modes of the technological process, the technical condition and quality of operation of technological machines [1, 2].

The results of research by individual scientists [3,4] have established that annually the economic efficiency of feed mills decreases by approximately 16% due to the low technical reliability of process equipment and the significant labor intensity of maintenance and repair operations. In this regard, the problem of improving methods and means of maintenance
based on automated diagnostics of technological equipment of feed shops when using them is relevant.

The production process of combined feeds includes a number of operations, each of which is performed by high-speed, dynamically loaded working machines. Based on an analysis of the state of the issue on the topic under consideration, it was established that the most important and energy-intensive operation of the feed preparation process is grinding, performed in feed crushers. A study of the operating conditions of hammer crushers revealed the least reliable working element - the grinding rotor, the technical condition of which is determined by wear of the support bearings, increased vibration and imbalance [1, 2].

One of the effective ways to reduce rapid damage to the mating parts of technological equipment units and ensure their serviceable and efficient condition is the timely and high-quality implementation of repair and maintenance operations based on the actual condition determined by technical diagnostics [1, 3].

The purpose of the study is to improve the operating efficiency of hammer crushers based on automated diagnostics of their technical condition.

2 Research methodology

The material of the scientific article represents a separate stage of research on a topical topic dedicated to increasing the economic efficiency of feed mills.

A study of the operating conditions of hammer crushers revealed the least reliable working element - a grinding rotor with hammers, the technical condition of which is determined by the wear of the hammers, support bearings, increased vibration and imbalance [1,4].

Currently, when repairing and replacing working parts, the feed crusher must be disassembled, the hammers must be deployed, and worn ones must be replaced. After replacing or rearranging worn hammers, the rotor must be carefully balanced on a special stationary stand, which requires dismantling the grinding rotor. This process leads to long downtime of the production line and significant costs for dismantling and transportation [1,4].

The most effective methods in this case are those that allow balancing of rotor mechanisms on site in their own bearing supports using electronic diagnostic tools.

The research methodology for the development of an automated diagnostic method includes several stages: observation of the conditions of use for its intended purpose, structural and investigative study of the working parts of the hammer crusher; analysis of the forces that arise in the interfaces of the grinding rotor in static and dynamic states; identification of the functional connection of the structural parameters of the technical condition of the rotary working body with diagnostic signs; conducting a laboratory experiment in order to compile a computational and experimental model of the grinding rotor in the form of regression equations [4].

3 Research results and discussion

A study of the operating conditions of hammer crushers revealed the least reliable working element - the grinding rotor with hammers, the technical condition of which is determined by wear or breakage of the grinding hammers, an increase in clearances in the support bearings, increased vibration and the amount of imbalance [3-5].

In order to substantiate the method for diagnosing rotor malfunctions during operation, determining the accuracy of manufacturing crushers at factories, assessing the quality of service and repair at service enterprises, a scheme for static and dynamic loading of the rotor has been developed. The general view of the loading of a rotary feed chopper is presented as
a dynamic system (Figure 1). On the bearing housings (support A, support B) of the rotor working body as a result of disturbing influences (imbalance $D$, centrifugal moment of inertia $I_{oz}$, artificial imbalance $D_m$), diagnostic signs are formed at the output: amplitude value of the vibration signal of the bearing housing ($A_{BC}$), angular displacement its maximum value from the reference point ($\varphi_{BC}$), as well as the amplitude value of the vibration signal caused by the imbalance of the rotary chopper ($A_{dBC}$) [3,6,7].

Fig. 1. Dynamic system “chopping rotor - support bearing”.

In the general case of loading a rotary chopper in its support axles A and B, inertial forces $F_a$ and $F_b$ arise, having a constant value, but a variable direction (rotating together with the rotary chopper), as well as forces $G_a$ and $G_b$, with a constant value and direction (Figure 2).

\[
\begin{align*}
F_a &= D \frac{d}{a+b} \omega^2; & F_b &= D \frac{c}{a+b} \omega^2 \\
G_a &= G \frac{b}{a+b}; & G_b &= G \frac{a}{a+b}
\end{align*}
\]  

\begin{equation}
(1)
\end{equation}

Fig. 2. Graphic representation of the forces acting on the rotary chopper during operation.

The dynamic and static forces applied to the bearing supports can be replaced by the resultant, which (for each support) can be determined by the vector expression:

\[
\bar{T} = G_{b} + F_{b} + N.
\]  

\begin{equation}
(2)
\end{equation}
As numerous scientific studies in the field of vibration diagnostics show [3, 6-8], due to changes (due to wear) in the mass of hammers, the imbalance of the rotor increases, which leads to increased vibration in bearing housings supports.

A quantitative assessment of the unbalanced state of a rotary chopper can be carried out based on an analysis of the forces acting on it during operation and obtaining equations for their connection with the kinematic and mass-geometric parameters of the rotary working body.

The magnitude of the inertial forces arising during the operation of the grinding rotor is determined by its angular velocity \( \omega \), angular acceleration \( \varepsilon \) and depends on the location of its mass relative to the axis of rotation (Figure 3):

\[
F = r \varepsilon + \omega^2 m, \tag{3}
\]

![Fig. 3. Scheme of dynamic loading of the grinding rotor supports.](image)

To increase the accuracy of the vibration diagnostic method, the process of isolating the signal, which directly characterizes the change in the technical condition of the diagnostic object, is very important [6-8]. For this purpose, we will analyze the dynamic loading of the grinding rotor supports by individual forces, as well as their combined influence.

Let us make the assumption that the elementary mass of the rotor \( dm \) moves under the influence of an unbalanced force with linear acceleration \( a \). Applying an elementary inertial force \( dF \) to it, we obtain the expression:

\[
dF = -adm. \tag{4}
\]

The inertial force \( dF \) applied to the rotary chopper, due to the counteraction of the mass \( dm \), changes its speed and gives acceleration to this mass.

Let us decompose the inertial force \( dF \) into two components, directing one of them tangentially to the trajectory, and the other along the main normal line.

It is known that the components of the inertial force have a direction opposite to tangential and normal accelerations. Then we will have the following expressions:

\[
dF = -at \cdot dm = - \left( \varepsilon \cdot \mathbf{r} \right) \cdot dm, \tag{5}
\]
As a criterion for the theoretical balance of the grinding rotary working body, we accept the equality to zero of inertial forces and their moments (dynamic equilibrium):

\[ \overline{F} = 0; \overline{M}^F = 0, \]  

where \( \overline{F} \) is the main vector of inertia forces; \( \overline{M}^F \) is the main vector of moments of inertia forces.

The magnitude of the imbalance of the grinding working body is proposed to be determined by the module of the vector of moments of inertia forces, which occurs exclusively in the dynamic mode of operation, in accordance with the expression:

\[ \overline{I}_{oz} = r \overline{D} \neq 0. \]  

In order to establish a connection between the structural parameters of the technical state of the elements of a dynamic system (input parameters) with the output (diagnostic) parameters of the oscillatory process, as well as to determine the loading mode of bearing supports, it is necessary to obtain an expression for the adjustable unbalance indicator:

\[ K_p = \frac{\Phi}{G}, \]  

where \( \Phi \) – inertial force; \( G \) is the magnitude of gravity.

A mathematical description of the oscillatory process in the bearing housing under dynamic loading can be presented as:

\[ \ddot{X} + 2\delta \dot{X} + \omega_0^2 X = \frac{1}{m}q, \]  

where \( \ddot{X} \) is acceleration; \( \dot{X} \) - speed; \( X \) - movement of the oscillating element; \( \delta \) - attenuation coefficient; \( m \) is the mass of the colliding body; \( \omega_0 \) - natural frequency.

Expression 10 represents a theoretical connection between the diagnostic parameters of vibration signals and the structural parameters characterizing the technical condition of the rotary grinder.

The spectrum of the signal \( X(\omega) \) received by the measuring transducer is characterized by the collision spectrum of parts \( Q(\omega) \) and the amplitude-frequency characteristic of the channel \( H(\omega) \). If the signal is recorded by a vibration acceleration sensor, we will have:

\[ H_a(\omega) = \frac{\omega_0}{2m\delta}. \]  

The shock impacts received by the measuring transducer are represented by a cosine pulse:

\[ q(t) = \begin{cases} \frac{\pi q_0}{\omega_0} \cos \frac{\pi}{\tau} t \text{ at } -\frac{\tau}{2} \leq t \leq \frac{\tau}{2}, \end{cases} \]  

where \( q_0 \) is the pulse area; \( \tau \) is the duration of the impact.

Solving this equation using expressions (11) and (12) we obtain:

\[ \dot{X}(t) = \frac{\omega_0}{2m\delta} \frac{\pi q_0}{\tau} \cos \left[ \frac{\pi}{\tau} t + \phi_0 \right]. \]  

By mathematical transformations of this spectrum \( \dot{X}(t) \) in a small frequency range \([\omega, \omega + d\omega], \) we will have the amplitude of vibration acceleration of the support bearing housing at \( \frac{\pi}{\tau} t + \phi_0 = 0 \). Then the dependence of the maximum amplitude will be:

\[ A^{ac} = k \left| T \right|, \]  

where \( k = \frac{\omega_0 \pi}{2m\delta \tau} \) is a constant coefficient for a given speed limit.
The phase of the maximum amplitude of the vibration signal will be determined by the expression:

$$\varphi = \omega_B + \frac{\tau \varphi_0}{\pi},$$  

(15)

where $\omega_B$ is the angular speed of rotation of the rotor shaft.

In order to increase the accuracy of determining the magnitude of the imbalance of the grinding rotor on the hammer crusher itself, in its own support bearings, it is necessary to first estimate the value of their total gap. To isolate the vibration signal component characterizing bearing wear, it is proposed to create a maximum disturbing effect by sharply increasing the angular velocity of the grinding rotor. The force impulse causing the rotor shaft in the bearing to tip over during acceleration will be determined by:

$$q_0 = M_R \sqrt{h^2 \varepsilon^2 + \omega^4 + 2hg \cdot \cos \alpha},$$  

(16)

where $M_R$ is the reduced mass; $h$ - radial clearance; $\alpha$ is the angle between the vertical and the direction of the force in the acceleration mode.

Taking into account (14-16), the relationship between the maximum amplitude of the vibration signal and the radial clearance in an unsteady mode will be determined:

$$A^{bc} = kM_R \sqrt{h^2 \varepsilon^2 + \omega^4 + 2hg \cdot \cos \alpha}.$$  

(17)

As a result of the theoretical studies performed, it was established [3, 5, 6, 8] that the use of expressions (14) and (17) allows us to evaluate the change in the imbalance of the grinding rotor of a hammer crusher, as well as determine the technical condition of its support bearings.

To implement the developed method of vibration diagnostics of the rotary grinder of a hammer crusher, a diagnostic and balancing stand is proposed (Figure 4).

Fig. 4. Block diagram of the diagnostic and balancing stand: 1 – rotary chopper; 2 – support bearings; 3 – support posts; 4 – stand base; 5 – electric motor; 6 – V-belt drive; 7 – elastic element; 8 – laser tachometer; 9 – tripod; 10 – strobe disk; 11 – vibration measuring transducers; 12 – spring dampers; 13 – platform; 14 – signal converter; 15 – electronic device.

The measuring complex of the proposed stand is a portable balancing device in the form of an electronic device, with the help of which it is possible to carry out balancing operations of rotary working bodies mounted on their own supports in two mutually perpendicular planes. The device kit includes: 2 vibration sensors, 2 sensors for measuring the phase of the maximum amplitude of the vibration signal, and a laptop. All control and balancing operations are carried out by the measuring complex in automatic mode. After processing the measured signals, information about: the technical condition of the support bearings, the
magnitude of the imbalance is displayed on the computer screen; weight and location of attachment of balancing elements.

The technological process of performing diagnostic operations on a diagnostic and balancing stand consists of several stages: establishing the sequence of operations; calibration of instruments and sensors; determination of the rational rotor speed, selection of the necessary frequency filtering of the vibration signal in order to obtain the maximum value of the informative signal.

4 Conclusion

Theoretical studies have established that by comparing the measured values of the maximum amplitude of vibration signals, which characterize the imbalance of the rotor working body and the wear of its support bearings, with reference (normative) values, a conclusion can be made about the technical condition of the grinding working body of hammer crushers.

The developed method of vibration diagnostics of a rotary chopper allows: control the quality of its manufacture and carry out balancing in a dynamic mode at the factory; determine the need for specific repair and maintenance actions before repair; evaluate the quality of repairs during final inspection.

The practical use of the proposed diagnostic and balancing stand will make it possible to: significantly reduce the labor intensity of diagnostic operations for the working parts of hammer crushers and increase the accuracy of measurements; automate diagnostic and balancing operations.

The presented measuring complex can be used as a portable device for periodic diagnostics of feed crushers at the workplace, as well as in built-in diagnostic systems for continuous monitoring of the technical condition of working parts without stopping the feed preparation production line.

References