DIAGNOSIS BY VIBRATORY ANALYSIS OF ROTATING MACHINES. JAW CRUSHER CASE. THEORY AND EXPERIMENTATION

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Abstract—In this work, we propose to study the dynamic behavior of a primary jaw crusher. It shows cracks and deformations in the structure and on the flywheel. This jaw primary crusher, capacity of 400 tons / hour, is modeled and studied numerically and experimentally. Using the SolidWorks simulation software, one part of the system is treated: eccentric shaft assembly, connecting rod and moving jaw, as well as the drive pulley and the flywheel. Also, we studied the vibratory behavior of this installation practically, and this by a programming of the points of measurements of vibrations using a software of vibration analysis XPR 300. Then we took samples of vibration measurements using an Analyzer. The spectral interpretation reveals the presence of a shock on the eccentric shaft line, causing a play in the rolling bearings of the shaft. The comparison of the results of the numerical simulation obtained with those found experimentally makes it possible to conclude the appearance of a resonance of the structure by causing cracking and deformations.

Index Terms- Cracking, Crusher, Mechanical defects, Shock, Vibrations.

I. INTRODUCTION

The technique of spectral analysis of vibratory measurements makes it possible to precisely locate the mechanical or electrical failures that may arise during the operation of the rotating machines. Industrial jaw crushers are large-scale rock crushing plants. A mechanical defect in the early state may evolve over time and easily compromise the operation of these machines. It can cause other mechanical anomalies that can lead to deformation and rupture of one or more organs [1]. We study the practical case of a primary jaw crushe. It presents several mechanical faults acting at the same time: imbalance, misalignment and wear on bearing surfaces. These defects cause vibrations of an alarming level according to the international standards VDI 2056. They cause cracks on the flywheel, ... A study of the vibratory behavior of the system is carried out to determine the modal characteristics. The results of the theoretical simulation were compared with those of the experimental vibration measurements. The consequences and the severity of mechanical defects are evaluated and proposals made to the system and establish an adequate vibration prognosis.

II. MACHINE DESCRIPTION “FIG. 1-1 AND 1-2”

The equipment consists of an eccentric shaft line with a diameter of 330 mm and a length of 2712 mm. It has a pulley and a cast iron flywheel diameter 1800 mm and width 800 mm “Figure 1-2”. This shaft line is supported by four rolling bearings with cylindrical roller bearings. This system is driven by a 250 KW electric motor running at 1000 RPM (Photo 1-1).
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2.1 Choice of vibration measurement points

The objective is to know the general state of the machine. It is therefore necessary to know the state of the bearings throughout the kinematic chain, and the behavior of the shaft line carrying the crusher. We study allows the machine and we choose the vibration measurement points on the bearings “Fig. 1-3” to detect all the failures that can arise on the machine during its operation. The programming of the measurement points is established in a way that allows to capture all the frequencies of interest and to follow their evolutions in the three (03) horizontal, vertical and axial directions.

2.2 Equation of the vibratory behavior of the mechanical System

Either an eccentric shaft system carrying in both ends a driving pulley and a flywheel of the same diameter “Fig. 1-3”. The system is modeled by SolidWorks. The numerical model is obtained “Figs 2-1 and 2-2”.

The equations of motion can be expressed by [2] [5]:

\[ \begin{bmatrix} M & \{\bar{q}(t)\} + \{C\} \{\dot{q}(t)\} + \{K\} \{q(t)\} = \{F(t)\} \end{bmatrix} \]

\[ \begin{bmatrix} M & \{\bar{q}(t)\} , & \{C\} et \{K\} \end{bmatrix} \]

are respectively mass, damping and stiffness matrices. \( \{\bar{q}(t)\} \), \( \{\dot{q}(t)\} et \{q(t)\} \) are respectively the vectors of generalized accelerations, generalized velocities and generalized displacements as a function of time. \( \{F(t)\} \) is the vector of forces generalized as a function of time.

The maximum amplitude of the movement is [2]:

\[ A = \frac{a}{2 \cdot \sqrt{k.(m_r + m_p)}} \] (1)

We pose

\[ C_r = 2 \cdot \sqrt{k.(m_r + m_p)} \] (2)

\[ A < 1 \] [3] [6] [7]: a vibratory motion, if a low damping. In practice the value of is small: a steel spring \( A=0.001 \), a rubber spring: \( A=0.05 \) ....

Where \( m_r \) : rotor mass, \( m_p \) : parasitic mass, \( \omega_0 \) : natural rotor frequency, \( k \) : support stiffness (N/m), \( a \) : support damping (N/m.s).

The phase difference \( \phi \) sera will be expressed by:

\[ \tan(\phi) = \frac{2 \cdot A \cdot \left( \frac{\omega}{\omega_0} \right)}{1 - \left( \frac{\omega}{\omega_0} \right)^2} \] (3)

The different schemes are [2]:

- When the rotation speed of the rotor is lower than the resonance frequency of the support, then: the assembly behaves as if the bearings were very rigid.
- When the rotation speed of the rotor is higher than the resonance frequency of the support, then the assembly behaves as if the bearings were very flexible, the inertia forces in the system are greater than the elastic forces exerted by the suspension that can be neglected, then: for \( \omega_0 << \omega \), we have:
\[
\frac{X}{|r|} = \frac{1}{(m_r + m_p)} \quad (4)
\]

This shows that if the vibration sensors measure the movement \(X\) of the bearings as being representative of the eccentricity \(|e|\), then the ratio of the parasitic mass to the mass of the rotor will be minimized to obtain a higher sensitivity.

When the speed of the rotor coincides with the natural (clean) frequency of the support, the inertia and elastic forces are in phase opposition and have the same amplitude. The amplitude of the radial displacement is then related to the damping term \(A\). If this damping is small, the dynamic effect can produce significant radial displacements.

### 2.3 Résultats de simulation

The results of the simulation were summarized (Table 1). We start with the lower natural frequencies, where the first and second modes represent the bending mode. From the third to the eighth mode, one has the modes of bending while the ninth mode represents the mode of torsion. We are interested in the lower frequencies which are the most important because they have an influence on the dynamic behavior of the structure. In particular the first and the second mode of vibration, where the modal manifestations of the behavior of the system begin and which relate respectively to the frequency 126.91 Hz and 126.98 Hz. These last two frequencies correspond to the thirty-eighth (38) harmony of the ray comb which is 126.66 Hz, whose base frequency is 03.34 Hz “Figs 3-1 and 3-2”, measured respectively on the two bearings No. 03 and 04 (Fig 1-1).

For this purpose, this mode of flexion relative to the first mode of vibration constitutes a real risk on the system. Its natural frequency (126.91 Hz) can easily coincide with the 38 th harmony (126.66 Hz) generated by a shock caused by any game when rotating the system. Unfortunately, these two frequencies coincide which explains the cracks appearing on the hub of the flywheel “figs 3-7 and 3-8”. We have modified the rotational speed of the electric motor, and consequently the speed of rotation of the eccentric shaft which represents our studied system and which will be of 234.6 rpm corresponding to the frequency of 03.91Hz. Automatically, with this change of speed, all the frequency components of the collision line comb will change and move away from Eigen frequencies, except the 32nd and 34th harmony, respectively relative to 125.31 Hz and to 133 , 14 Hz. This operation made it possible to slow down the evolution of the cracks in the hub of the flywheel.

![First mode of rotor vibration](image1)

![Second mode of rotor vibration](image2)

![Third mode of rotor vibration](image3)

![Fourth mode of rotor vibration](image4)

![Fifth mode of rotor vibration](image5)
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<table>
<thead>
<tr>
<th>n° mode</th>
<th>Fréquence naturelle en (Rad/s)</th>
<th>Fréquence naturelle en (Hz)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>797.4</td>
<td>126.91</td>
</tr>
<tr>
<td>2</td>
<td>797.83</td>
<td>126.98</td>
</tr>
<tr>
<td>3</td>
<td>849.99</td>
<td>135.28</td>
</tr>
<tr>
<td>4</td>
<td>850.25</td>
<td>135.32</td>
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<tr>
<td>5</td>
<td>850.48</td>
<td>135.36</td>
</tr>
<tr>
<td>6</td>
<td>850.7</td>
<td>135.39</td>
</tr>
<tr>
<td>7</td>
<td>2092.5</td>
<td>333.04</td>
</tr>
<tr>
<td>8</td>
<td>2093.3</td>
<td>333.15</td>
</tr>
<tr>
<td>9</td>
<td>2575.2</td>
<td>409.86</td>
</tr>
<tr>
<td>10</td>
<td>2576.3</td>
<td>410.03</td>
</tr>
</tbody>
</table>

Table N°1: List of eigenfrequencies.

III. COMMENTS [4]

The follow-up by the vibratory analysis method started on July 09, 2015. The spectral interpretation reveals the presence of several mechanical anomalies: a wear defect on the bearing surface in the two bearing Nos. 03 and 04 bearing the eccentric rotor of the crusher, as well as a bearing fault in these two bearings. Also, we detected the presence of a misalignment between these two levels “Figs. 3-1 and 3-2”.

The predominant component related to the rotation frequency of 03.34 Hz appeared, indicating the presence of an imbalance on the rotor carrying the crusher, related to the misalignment. Initially, these failures generated vibrations of an overall level of 04.64 mm / s and 04.61 mm / s respectively on the two levels N° 03 and 04 in the horizontal direction, considered acceptable with reference to international standards VDI 2056 (Table No. 02), and according to the trend curves “Figs. 3-3 to 3-6”. We recommended to change the bearings and bearings N° 03 & 04 and to control the alignment between these two bearings. The recommendations were not considered, resulting in cracks in the flywheel “Figs 3-7 and 3-8”. To this end, our concerns have focused on the evolution of the crack related to the global level of instantaneous vibrations, because the equipment operates in unstable conditions relating to the dimensions of the rocks to be crushed and the presence of foreign bodies. These concerns led us to proceed on the one hand to the monitoring of this installation by the vibration analysis technique. Another recommendation was made to change the rotational speed of the eccentric shaft by changing the rotational speed of the electric motor. The basic frequency became 3.34 Hz to 03.91 mm / s. This made it possible to avoid as much as possible the appearance of one of the peaks located at 126.66 Hz corresponding to the 38th peak of the vibratory shock generated by the mechanical defect on the system “Figs.3-1 and 3-2” which coincides with one of the Eigen frequencies of 126.91 Hz (Table 1).

The spectral interpretation of the vibration measurements taken on June 20, 2017 allows detecting the presence of several mechanical defects through: Presence of a shock on all bearings of the shaft carrying the eccentric shaft “Figures 4-1 and 4-4”. There is a line comb of components of order 1 and 2, 3, 4 ..., whose base frequency is 0.91 Hz corresponding to the speed of rotation of 234.6 rpm. This is a game on the rolling bearing in the two bearings No. 03 & 04 carrying the eccentric shaft.
This failure is caused by the presence of a misalignment between these two levels, as indicated by the second harmony of the line comb spectra taken horizontally and vertically in these two levels, corresponding to the frequency of 0.78 Hz, whose amplitude is 0.42 mm / s on the bearing No. 03 in the horizontal direction “Fig.4-1”, and 0.04 mm / s on the bearing No. 04 in the horizontal direction also “Figure 4-3”. Also, we detected the occurrence of a chipping-type bearing fault, generating a Kurtosis level of 0.751 mm / s on bearing No. 03 in the vertical direction, and 11.80 on the N bearing. ° 04 in the vertical direction too. It should be noted that a signal with a Kurtosis> 3 is represented by a narrower distribution dominated by the presence of abnormally high peak amplitudes as is the case in the presence of repeated shocks [5].

All these detected failures hampered vibrations of an overall level of 0.581 mm / s on the bearing No. 03 on the receiving pulley side, and 0.551 on the bearing No. 04 on the flywheel side with cracks that have crossed almost the entire diameter of the hub (Table No. 02). Following this analysis of vibrations, or this problem persists since: July 09, 2015, we insisted on the change of bearings and bearings, because these failures have caused cracking on the flywheel that can be spread to to the eccentric shaft, where the damage will be more serious and expensive.

<table>
<thead>
<tr>
<th>Intervention date</th>
<th>Overall vibration level in (mm / s) on the bearing No. 03</th>
<th>Overall vibration level in (mm / s) on the bearing No. 04</th>
</tr>
</thead>
<tbody>
<tr>
<td>09.07.2015</td>
<td>0.46, 04,61</td>
<td>04,61</td>
</tr>
<tr>
<td>29.09.2015</td>
<td>04,52</td>
<td>04,98</td>
</tr>
<tr>
<td>20.06.2017</td>
<td>05,81</td>
<td>05,51</td>
</tr>
</tbody>
</table>

Table 2: History of interventions

CONCLUSION

For this practical case treated, the machine has several mechanical defects. The imbalance defect caused by a shock on the shaft line caused wear on the bearing surface in the bearings. This caused other
anomalies: lack of bearings and cracks on the hub of the flywheel. The theoretical modal analysis has enriched the vibration diagnosis, by identifying the specific characteristics of the system that further aggravate the operating conditions of the installation. The rotation speed of the eccentric shaft, related to the base frequency, was changed. As a result, all the frequency components that coincide with the natural frequencies of the system have been avoided in order to limit the propagation of cracks.

REFERENCES


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