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Detection of unbalance defect by the vibration analysis technique

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ORIGINAL RESEARCH PAPER

ABSTRACT

The main advantage of condition monitoring by analyzing vibrations on rotating machines is the possibility of detecting faults before a failure occurs leading to the unplanned stoppage of a machine.

The unbalance defect is the most common and most frequently encountered cause of vibration, it is also one of the main causes of reduction in the life of machines and it constitutes one of the main risks for Security.

This paper is devoted to the experimental study of the two types of unbalance defects, their different vibratory signatures and how to differentiate their vibratory behavior from other mechanical faults which manifests itself by the presence of a peak at the rotational frequency in the case of combined faults.

KEYWORDS

conditional maintenance, vibration analysis, unbalance defects

1. INTRODUCTION

The machining and assembly faults cause significant vibration problems for rotating machines. Manufacturers and operators seek to eliminate them to increase the life of the machines, to optimize their efficiency and to ensure their regularity of operation. One of the most important vibration problems is caused by the unbalance defect, which cause vibrations that are usually synchronous.

The unbalance is the result of a set of periodic forces generated by the differences between the centers of gravity of the various elements making up the rotor and its axis of rotation. This centrifugal force resulting from these non-concentricity or adjustment faults is proportional to the mass of material that creates this imbalance and the square of the rotation speed ($F = Mr\omega^2$) (Fig. 1) [1, 2]. It generates rotating forces, which act on the various components of the rotor, the machine anchors and the bearings until they are damaged. For this reason, it is the subject of important standards, both for its evolution and for its correction [3, 4].

This unbalance generally comes from machining defects, assembling and mounting, it is the consequence [5, 6]:

- Mechanical deterioration: loss of blades (pump), erosion or clogging (fan), etc.;
- Thermal alteration: deformation following expansion of the materials constituting the rotor or localized temperature differences.

The aim of this research is to define the unbalance defect, the types of unbalance, to treat its temporal and frequency vibratory representation theoretically and also to study its vibratory behavior experimentally on a test bench built in the laboratory to facilitate its detection.

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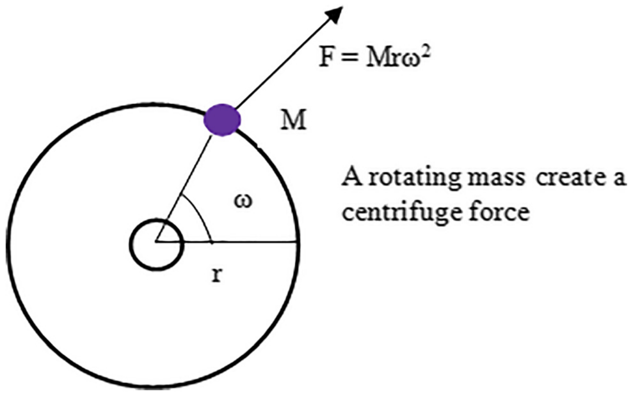

Fig. 1. Unbalance defect

2. UNBALANCE TYPES

2.1. Static unbalance

The main axis of inertia and the axis of rotation are parallel, but not coincident [7, 8] (Fig. 2).

2.2. Dynamic unbalance

The main axis of inertia forms a non-zero angle with the axis of rotation, and their intersection does not coincide with the center of the mass [9] (Fig. 3).

3. THEORETICAL STUDY

3.1. Temporal representation

The temporal representation of an unbalance is a periodic function type simple sinus [10] (Fig. 4),

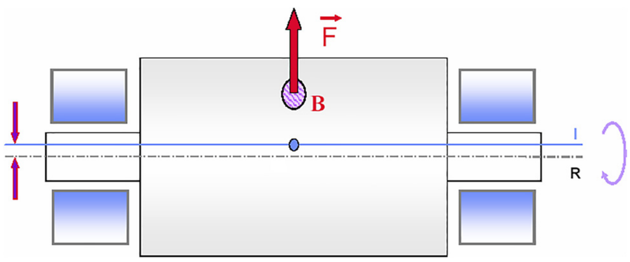

Fig. 2. Static unbalance

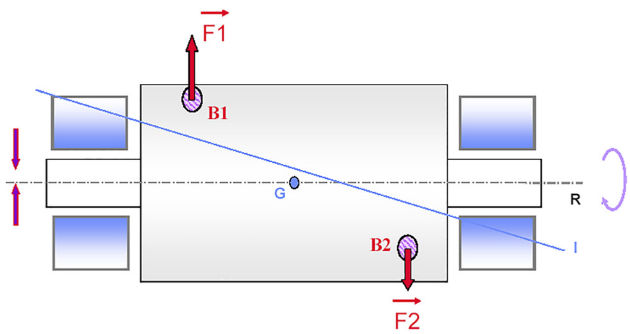

Fig. 3. Dynamic unbalance

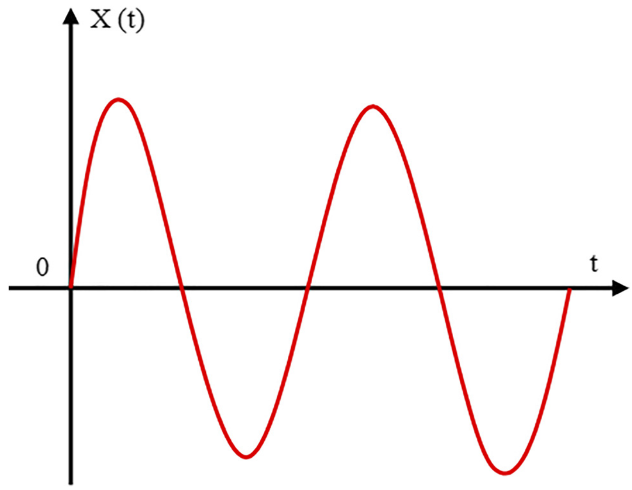

Fig. 4. Temporal representation of a simple sinus

$$X(t) = \sin(2\pi f_0 t). \tag{1}$$

3.2. Frequency representation

The frequency representation of a function periodic simple sinus is as follows (Fig. 5):

$$X(f) = j/2\,[d(f + f_0) - d(f - f_0)]. \tag{2}$$

4. EXPERIMENTAL STUDY AND ANALYSIS OF RESULTS

The experimental study of the two types of static and dynamic unbalance is carried out by a global analysis, a spectral analysis and a phase analysis. The measurements are produced with a vibration analyzer and two accelerometers, on a test bench built in the laboratory. The test bench consists of an electric motor with two output shafts on which two disks are mounted. The motor has a rotation speed of 3000 RPM (so the rotation frequency $f_r = 50\,\text{Hz}$) (Fig. 6).

4.1. Static unbalance

To create an unbalance defect, a mass must be added to a disc 1 (Fig. 6).

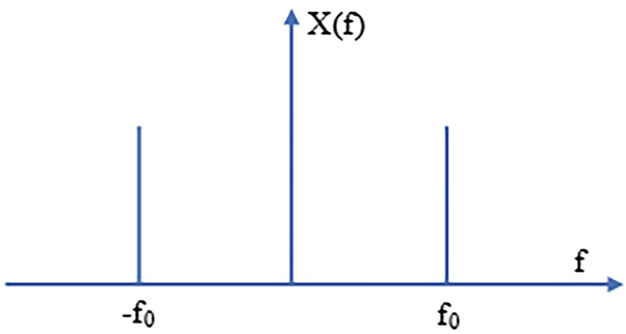

Fig. 5. Frequency representation of a simple sinus

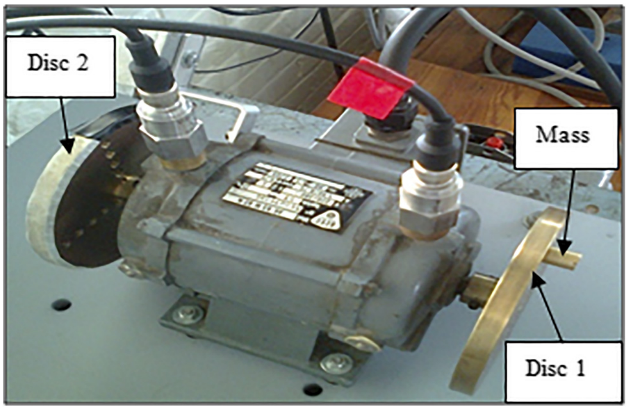

Fig. 6. Image of the test bench

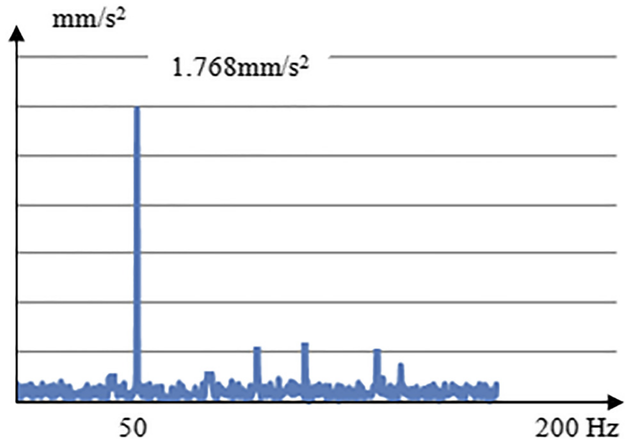

Fig. 7. Spectrum for static unbalance recorded

4.1.1. Global analysis.

- Without defect: The global vibration level is 0.652 mm/s;
- With unbalance defect: The global vibration level is 1.943 mm/s.

An increase in the global vibration level is noticed after the creation of the unbalance defect.

4.1.2. Spectral analysis. In order to have a good resolution on the vibration analyzer and to facilitate the detection of defects, it is necessary to choose the frequency band used and the number of spectral lines:

- Frequency band: the desired defect belongs to the low frequency domain, so the maximum frequency (f_M):

$$f_M \geq 4 \cdot f_r, \tag{3}$$

$$f_M \geq 200\,\text{Hz}; \tag{4}$$

- Number of spectral lines: allowing the distinction between the two closest peaks. In this case it is the 0.5 frequency of rotation (f_r) peak of mechanical gap and frequency of rotation (f_r) peak of unbalance,

$$\Delta f = f_r - 0.5 \cdot f_r; \tag{5}$$

- The number of spectral lines is obtained by the relation:

$$\text{NLS} \geq \frac{f_M \cdot 8}{\Delta f} = 64. \tag{6}$$

For the analyzer used the number of spectral lines is taken as $NLS = 100$ spectral lines.

After setting the parameters of the vibration analyzer, a spectral analysis is carried out by installing an accelerometer on a radial direction. The measured spectrum is shown in (Fig. 7).

The spectrum (Fig. 7) shows the appearance of peak of the frequency of rotation ($f_r = 50$ Hz) and its multiples with the peak $1{\cdot}f_r$ is the largest. This is because the amplitude of the signal of an unbalance defect will be maximum when the mass is at the top of the rotor and minimum when it is at the bottom, at each revolution of the rotor. The vibration signal is therefore periodic with a period ($1{\cdot}f_r$).

4.1.3. Phase analysis. The unbalance is not the only defect, which manifests itself in the vibration field by the presence of a peak at the rotation frequency; it is also the case of a damaged tooth of a gear, loosening defect, mechanical gap, curved shaft defect, etc., which makes it difficult to detect the imbalance fault in the industrial case where the machine is constituted of different components. For this, it is necessary to use a phase analysis which makes it possible to differentiate:

- Faults inducing by rotational forces as in the unbalance defect;
- Faults inducing by directional stresses as in the loosening defect, mechanical gap, etc.;
- Phase analysis between two radial measurements located at 90° on the same bearing (Fig. 8);
- Phase analysis between two radial measurements located at 180° on the two bearings (Fig. 9).

The phase analysis (Fig. 8) shows that there is a 90° phase shift, which means that it is a fault induced by a rotary force (unbalance).

The phase analysis (Fig. 9) shows that there is no phase shift (0°) between the two radial measurements located at 180° on the two bearings. This is because the both bearings

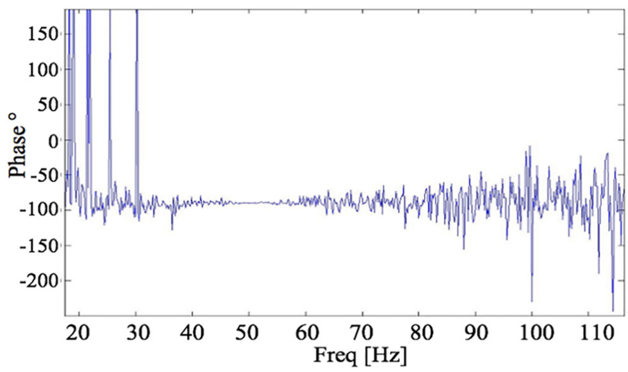

Fig. 8. Phase shift between two radial measurements located at 90° on the same bearing

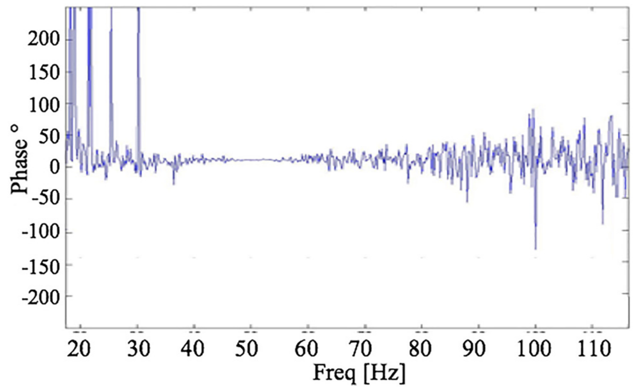

Fig. 9. Phase shift between two radial measurements located at 180° on the two bearings

supporting the rotor will be subjected to the centrifugal force due to the unbalance at the same time.

4.2. Dynamic unbalance

To create a dynamic unbalance fault, two masses are added, the first on disc 1 and the second on disc 2 (Fig. 6).

4.2.1. Spectral analysis. The spectral measurement on a radial direction of the dynamic unbalance (Fig. 10), gave a similar result to the static unbalance spectrum (Fig. 7).

4.2.2. Phase analysis.

- Phase analysis between two radial measurements located at 90° on the same bearing (Fig. 11);
- Phase analysis between two radial measurements located at 180° on the two bearings (Fig. 12).

The phase analysis between two radial measurements located at 90° on the same bearing (Fig. 11), gave a similar result to the static unbalance phase shift (Fig. 8).

The phase analysis (Fig. 12) shows that there is a phase shift of 180° between the two radial measurements located at 180° on the two bearings, because the two bearings supporting the rotor will be subjected to centrifugal forces in an alternating manner. The phase difference (around 180°) between the measurements taken at the same point on two consecutive bearings is therefore indicative of a dynamic unbalance.

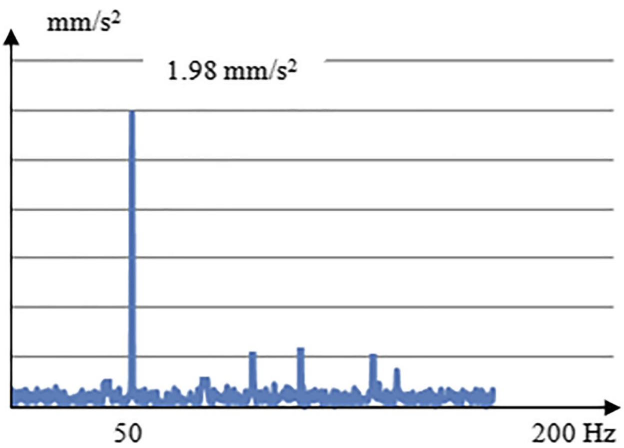

Fig. 10. Spectrum for dynamic unbalance recorded

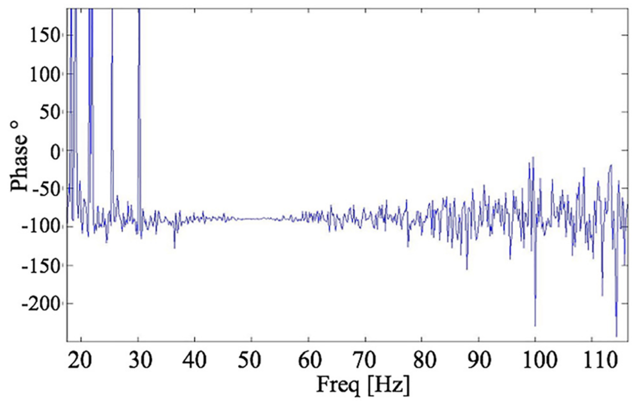

Fig. 11. Phase shift between two radial measurements located at 90° on the same bearing

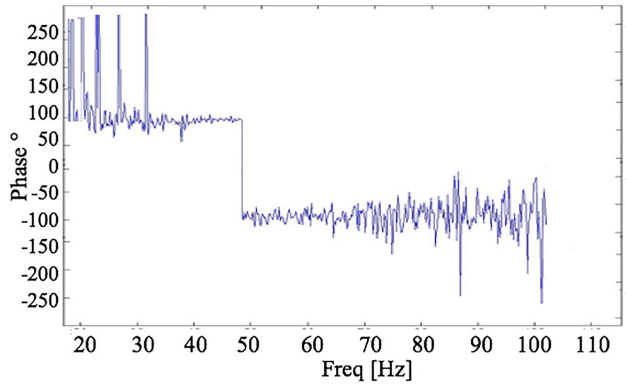

Fig. 12. Phase shift between two radial measurements located at 180° on the two bearings

5. CONCLUSION

In this paper, several analyses were carried out, a global analysis, a spectral analysis and a phase analysis, to treat the unbalance defect. It was found that the unbalance defect is detected by

- An increase in the global low frequency level compared to the reference state;
- A high amplitude component at the rotor's rotational frequency in the radial direction, sometimes in the axial direction in the case of cantilevered rotors;
- No or very few harmonics;
- No sub harmonics;
- An amplitude that can vary greatly with the speed of rotation;
- A phase shift of approximately 90° between two components corresponding to orthogonal radial measurement points on the same rotor bearing;

- The unbalance will be qualified as static or dynamic according to the phase analysis between two radial measurements located at 180° on the two bearings, if the phase is 0° it is a static unbalance requiring only a monoplane balancing, if the phase is 180° it is a dynamic unbalance requiring a multiplane balancing.

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