Health Monitoring of Rotating Rolling Bearings by using Vibration Analysis

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Abstract— Vibrations are found almost everywhere in rotating machines. Machines life is lessened due to unbalancing which is further the cause of machine vibration and unbalanced rotor that generates more vibrations and excessive force. Some mechanical issues like mass unbalance, coupling misalignment, mechanical loosening etc. are caused by the vibrations in rotating machine and it's been proven that 85% of these problems of moving machine are resolved by balancing and alignment and they reduce the life of machine component.

Key words: Vibration Analysis, Health Monitoring

I. Introduction

A. Fault Related Terminologies

1) Unbalance

The formal definition of unbalance is described as "that condition which exists in a rotor when vibratory force or motion is imparted to its bearings as a result of centrifugal forces."

2) Heavy Spot

Heavy spot is the common term used in balancing to signify the unbalance spot in the rotor. A measurable mass on a rotor is usually not the norm. The heavy spot is actually the mass and location sum of all the distribution of non-uniform masses on the rotor.

3) Synchronous or 1X

The same speed as rotor. The "X" is equivalent to a mathematical multiplication symbol. Thus, 1X can be read as "1 times rotor speed."

4) High Spot

In 1X vibration, the high spot is the location on the shaft directly under the vibration probe when the shaft makes its closest approach to that probe. It corresponds to the positive peak of the vibration waveform.

5) Runout

Runout is the total linear displacement of a rotor. A dial indicator is normally used to measure the outside diameter of the rotor in the radial direction, which will give the runout. However, unbalance and runout are not directly related. A rotor can have zero total runout and be well out of balance. Also, a perfectly balanced rotor can show a great deal of runout. The key fact to learn is that many maintenance.

6) Eccentricity

Eccentricity is the off-center displacement of the center of gravity of a rotor. Unbalance is directly proportional to eccentricity. Equation below shows this relationship. This equation is for flawlessly balanced and completed round rotors.

$$U = eM$$

where,
U = unbalance
e = eccentricity, displacement of the center of gravity
M = rotor mass

An important lesson is not to confuse eccentricity with runout. The total rotor indicator measurement on the outer diameter is considered the runout, while the eccentricity is how much the rotor's center of gravity is displaced from the center.

II. Experimental Procedure

The Machinery Fault Simulator (MFS) -Lite will be initially configured to collect data on both the inboard and outboard rotor bearings using single plane centre-hung balanced rotor. No weights will be installed on the rotor. The MFS must be properly aligned. Data will be collected at various speeds to capture the baseline vibration of a well-balanced machine in a centre-hung condition. A trial weight of 5.05 gram will then be added to the rotor to introduce unbalance and data will be collected at various speeds. The machine will then be reconfigured to an over hung condition with a single balance rotor. No weights will be installed on the rotor. Data will be captured at various speeds to capture the baseline vibration of a well-balanced machine in an overhung condition. A trial weight of 5.05 gram will then be added to the rotor to introduce unbalance and data will be collected at various speeds.

A. Frequency Response

1) Centre-Hung Rotor Configuration

2) Vibration spectra at 16 Hz

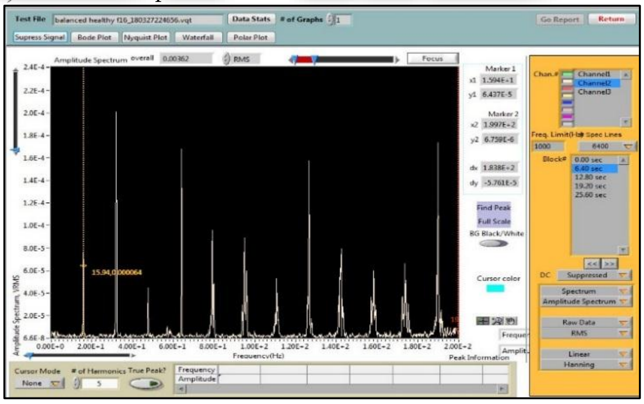

Fig. 3.1: Vibration spectrum at DE of balanced rotor at rotor speed of 16 Hz

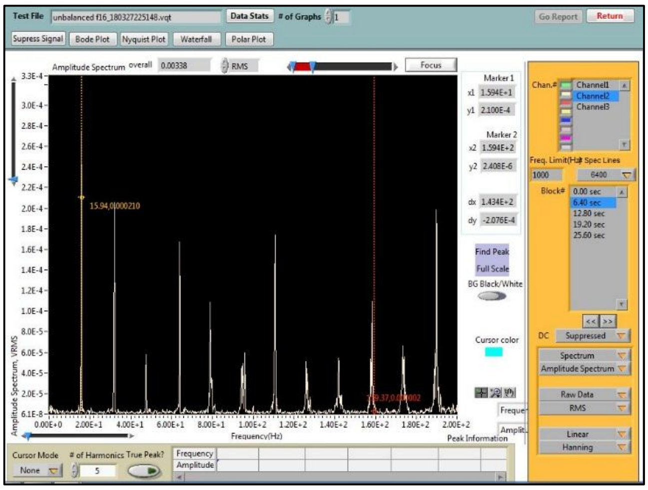

Fig. 3.2: Vibration spectrum at DE of unbalanced rotor at rotor speed of 16 Hz

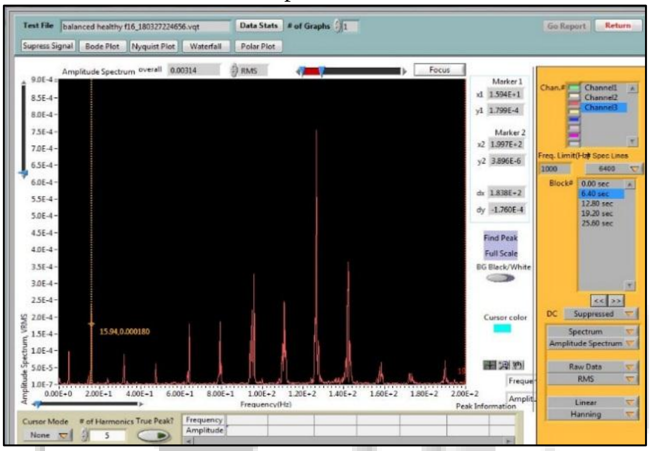

Fig. 3.3: Vibration spectrum at NDE of balanced rotor at rotor speed of 16 Hz

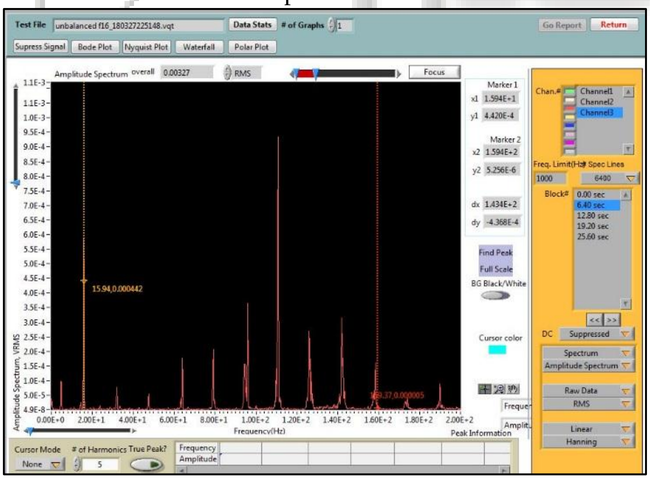

Fig. 3.4: Vibration spectrum at NDE of unbalanced rotor at rotor speed of 16 Hz

3) Single Plane Balancing

Single plane balancing is the method utilized to correct unbalance in one plane. This method is suitable for static unbalance but cannot help a machine facing a moment and dynamic unbalance. This can also be used for long rotor of one plane having majority of unbalance. Any rotor that is classified as a thin rotor can be single plane unbalanced. Also, single plane balancing may be used for long rotors when a majority of the unbalance is in one plane. The unbalance masses on the long rotors do not have to be in the same plane, only on the same side of the circumference of the rotor. Fig 5.1. Illustrates basic unbalance of a thin rotor. To balance the heavy spot, one correction weight must be placed 1800 opposite to the heavy spot. This is the most basic method of single plane balancing.

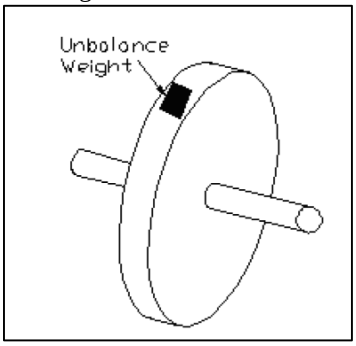

Fig. 5.1: Thin rotor with single plane unbalance

Fig. 5.2 shows a long with two unbalance weights in two planes. These can be treated as one combined weight in the location shown by the dashed lines since both weights lie in the same circumferential position.

Overhung rotors and motor pulleys are considered special cases when it comes to balancing. However single plane balancing for these types, including long rotors, is nevertheless useful. The correction weight must be added near the plane of the heavy spot or center of gravity for this method to be effective. The majority of overhung rotors can be balanced using single place balancing. Fig.5.3 shows a typical overhung rotor configuration.

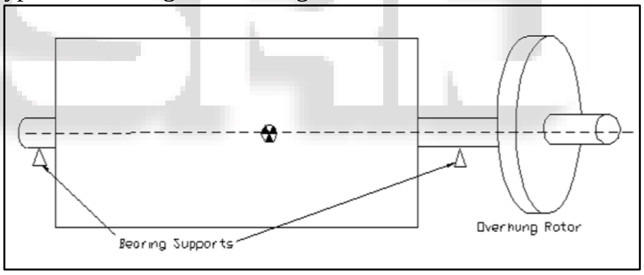

Fig. 5.3: A typical configuration of an overhung rotor

III. Conclusion

The experimental results for single plane balancing of centerhung and overhung condition obtained using Machinery Fault Simulator –Lite were compared with the analytical method (Influence coefficient method) and the results were found to be in good agreement with each other which the correctness and robustness of the experiment using MFS-Lite.

The experimental results for two plane balancing of centerhung and overhung condition obtained using Machinery Fault Simulator –Lite were compared with the analytical method (Influence coefficient method) and the results were found to be in good agreement with each other which the correctness and robustness of the experiment using MFS-Lite.

There is an increase in vibration amplitude with increase in rotor speed.Higher vibration amplitude at Non-drive end (NDE) bearings has been observed as compared to vibration amplitude at drive end bearings (DE).The balancing speed is however limited by the flexibility of the rotor and supports, it must remain well below the lowest rotor critical speed.

A linear relationship has been assumed between vibration and the unbalance force to calculate the correction weight. This proportionality relationship is assumed to hold all the way down to zero vibration. But the problem is that the mechanical world is linear only in a limited range.

A. Scope For Future Work

We can also consider flexible rotor instead of rigid rotor for balancing as a future work. The experimental setup can also be used for other fault diagnosis in future such as misalignment, cracked shaft etc. The healthy bearings can be replaced with defective bearings to study the effect of unbalance on vibration amplitude.

In the present work, rolling element bearings have been used. Other type of bearings can also be used as a future work.

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