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Early Failure Detection and Diagnostics of High Speed Self Aligning Journal Bearing

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ABSTRACT

Because of their high load carrying capacity and low cost, journal bearings are widely used to support the rotor of industrial machinery with high loads, such as steam turbines, centrifugal compressors and pumps. However, sudden catastrophic failure journal bearings can result in huge economic loss and high safety problems. It is necessary to develop effective condition monitoring technologies to detect and diagnose the failures at early stage and avoid such catastrophic failure. Previous researchers have studied the low frequency vibration characteristics as well as the high frequency vibration and acoustics emission for the early detection of journal bearing failure. However, these studies give relatively little information to the vibro-acoustic characteristics of high speed self aligning journal bearings. This paper focuses on the condition monitoring of high speed self aligning journal bearings through vibro-acoustic analysis.

Keywords: S*elf aligning journal bearing, vibro-acoustic analysis, early failure, fundamental frequency.*

1. INTRODUCTION

Bearings are used for supporting rotating machinery are one of the crucial elements by which the safe operation of the machinery is ensured. The goal of bearings is to provide relative positioning and rotational freedom while transferring speed [1].

Experience indicates most of the machinery problems are caused by bearing failure. The study of machine of 75kW (100 HP) or more, indicate that more than 40% of motor failures were caused by bearing problems [2]. Induction machine failure surveys have found the most common failure mechanism in induction machine categorized according main components are machine bearing related faults (40%), stator related faults (36%), rotor stator related fault (10%) and other faults (12%) [3].

Bearings are relatively cheap, but if damaged will cause catastrophic failure and serious disadvantages both in economic and engineering terms. Catastrophic failure can be as a result of gradual degradation and intermittent failure. Based on the time frame in which the event occurs, failure can be classified as catastrophic, intermittent, out of tolerance, and maladjustment [4]. By bearing condition monitoring early failure can be detected and then catastrophic failure can be avoided.

2. JOURNAL BEARING AND SURFACE INTERACTION

The journal bearing consists of a shaft, also called a journal, and a supporting component, which may be a shell around the shaft called a sleeve, a half shell that the shaft fits into, two half shells (top and bottom parts) or a multipart shell. Journal bearings are produced in many designs and sizes to match their many uses. These can be grouped as bushing, split journal bearing, spherical or self aligning plain/journal bearing, tilting pad bearing and thrust bearing. This paper focuses on high speed self aligning journal bearings. The bearing is provided with an oil ring lubrication system.

Today, journal bearings are used in many rotating machineries such as steam turbines, generators, compressors, internal combustion engines, and ship propulsion shafts because journal bearings are superior to rolling element bearings in vibration absorption, shock resistance, quietness, and long life.

All these characteristics come from the journal bearing principle of supporting a shaft by a thin oil film. Also, the smaller outside diameter of a journal bearing compared with a rolling element bearing is often beneficial to designers [5].
Figure 1 represents the interaction contact between two surfaces in relative motion that may be used to detect and bearing diagnostic.

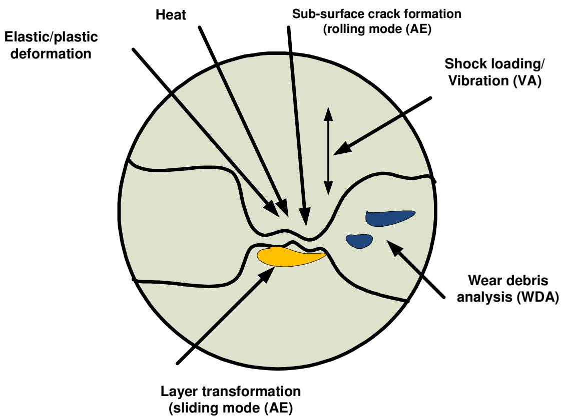

Figure 1- Contact interaction between two surfaces in relative motion [6]

The journal bearing consist of two paired components in contact with each other. When the first comes under loading, the surfaces between become closer and this influences the way one surface slides over the other. As sliding starts, any fluid lubricant will shear and the viscous response to surface discontinuities will produce pressure in the fluid. If the pressure is sufficient to balance the applied load, sliding will occur with no solid contact between the surfaces. Increasing the load further causes the fluid film to decrease in thickness and cause high spots on the surfaces to meet and create particle wear [7]. Vibration analysis is one of the most commonly used monitoring techniques and has the advantage that it yields relevant data in a quantitative format, and can be operated remotely in the real-time mode. Pre-set alarm limits can be triggered automatically, based on signal velocity levels, or alternatively as a result [6].
Acoustic analysis is similar to vibration analysis; however, its focus is not to detect causes for rotating equipment failure by measuring and monitoring vibrations at discrete frequencies and recording data for trending purposes. [8]. Many researchers have studied the vibration characteristics of hydrodynamic journal bearings at low frequency ($f \leq 20$kHz), acoustic emission and acoustical properties of hydrodynamic in the high frequency range ($f \geq 100$kHz) [9].
Vibro-acoustic characteristic of a self aligning journal bearing have been investigated through time based domain and frequency domain analysis of the radial, torsion load and speed.

3. VIBRATION AND PRESSURE IN SELF ALIGNING JOURNAL BEARING

Vibration of a self-aligning journal bearing is its response to imposed forces caused by moving parts in the bearing including the mass of the rotor and the rotor unbalance. A self-aligning journal bearing is assumed to be an elastic system consisting of load, the rotor mass, mass unbalance, oil stiffness, damping coefficient and oil bearing mass.

If **M** is the mass matrix, **C** the damping matrix and **K** the stiffness matrix of a multi degree of freedom systems , the motion equation may be written in the following matrix notation.

$$[M].\{\ddot{y}\}+[C].\{\dot{y}\}+[K].\{y\}=\{f_y(t)\} \qquad (1)$$

The bearing load in y direction consists of external radial force and unbalance force.

$$f_y(t) = F_r + W + m.\omega^2.e.\sin(\omega.t) + F_y \tag{2}$$

In the x direction, the equation of motion may be written as:

$$[M].\{\ddot{x}\} + [C].\{\dot{x}\} + [K].\{x\} = \{f_x(t)\} \tag{3}$$

The external bearing load in the x direction:

$$f_x(t) = m.\omega^2.e.\cos(\omega.t) + F_x \tag{4}$$

The reactive forces of the oil film F_y and F_x are calculated by integrating the oil pressure over the bearing surface as follows [10]:

$$F_x = \int_0^A p.\sin\theta.dA \tag{5}$$

$$F_y = \int_0^A p.\cos\theta.dA \tag{6}$$

When viscosity dynamic of lubricant η (Pa.s), linear velocity of shaft V (m/s), radius of shaft R (m), radial clearance c (m), eccentricity ratio ε, and contact angle φ (rad), for long journal bearing, the pressure distribution is given in the following equation [11].

$$p(\theta) = \frac{6.\eta.V.R}{c^2}.\frac{\varepsilon(2+\varepsilon.\cos\varphi).\sin\varphi}{(2+\varepsilon^2).(1+\varepsilon.\cos\varphi)^2} \tag{7}$$

4. EXPERIMENT RESULT AND DISCUSSION

Experimental work is carried out on the journal bearing test rig is connected to an electric motor through a flexible coupling. The electrical motor is a Clarke Motor 4HP/3.0kW, 3 phase induction motor controlled by Siemens Micro Master Controller. The test rig consisted of two self-aligning journal bearings with diameter 35 mm, length 76 mm, and radial clearance 20 micron as bearing test. An accelerometer (YD-5-4251) measured vibration and a microphone (BAST YG 201-07065) measured the acoustic noise, Millmax 32 is used as a lubricant. The test was carried out at constant 50% of the torsion load and constant radial load 65.4N, at 50, 60, 70, 80, 90, and 100% of maximum speed. An 8-channel data acquisition system (YE 6230B) was employed to record all the measurements at a sampling rate of 96kHz. In addition a speed-up transient test was also conducted to identify the frequency content of the vibration signals.

Waveforms illustrated in Figure 2 show typical vibration responses from the DE (drive end) journal bearing under different speed at 50% torsion load. It can be seen that the amplitudes of vibration has a slightly increases with speed increase. The amplitude increasing may be due to the effect of an unbalance of the shaft system. This analysis can be confirmed by spectrum analysis.

Figure 3 shows the spectrum in the frequency domain. It shows that bearing vibration mainly dominates under 100Hz. In particular, the spectrum shows that frequency of the first, second and third harmonics depend on shaft speed or fundamental frequency these are 12.5Hz, 17.5Hz and 22.25Hz. This figure also shows that vibration amplitudes exhibit gradual increases with different speed.

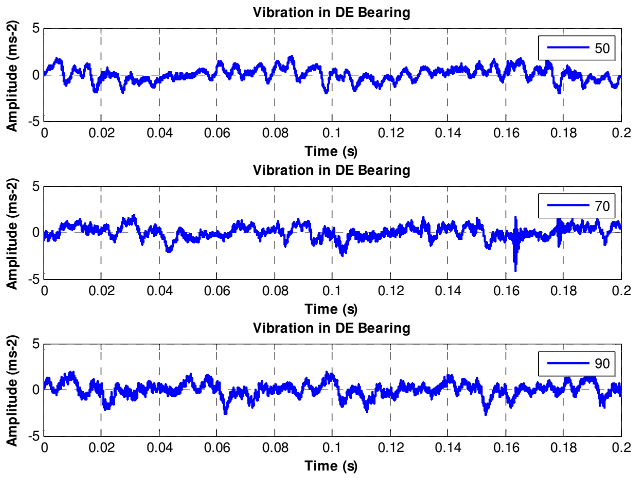

Figure 2- Time-domain vibration response at DE bearing

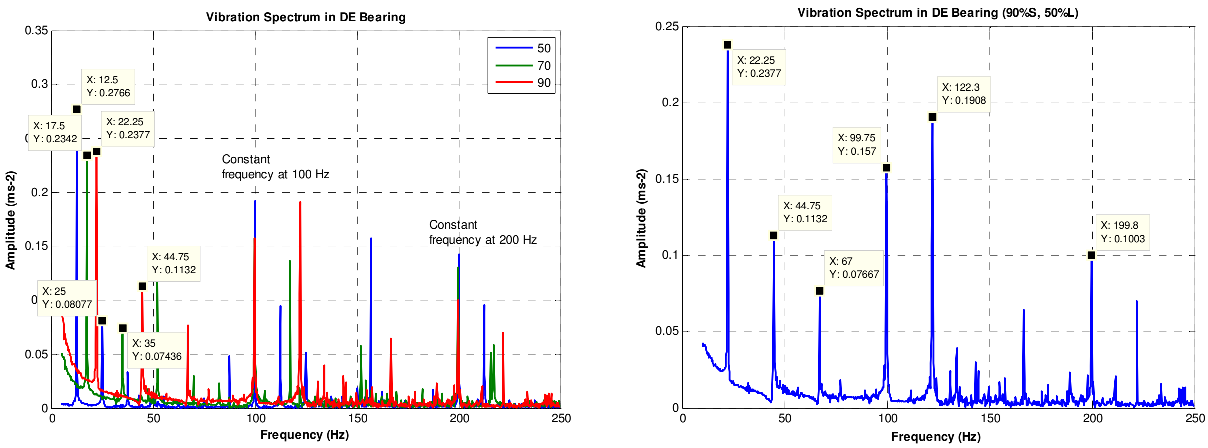

Figure 3- Frequency-domain vibration response at DE bearing

However, interesting spikes occurred in the amplitude at frequencies of 100Hz and 200Hz. To identify the source of these components a speed-up transient test was performed.

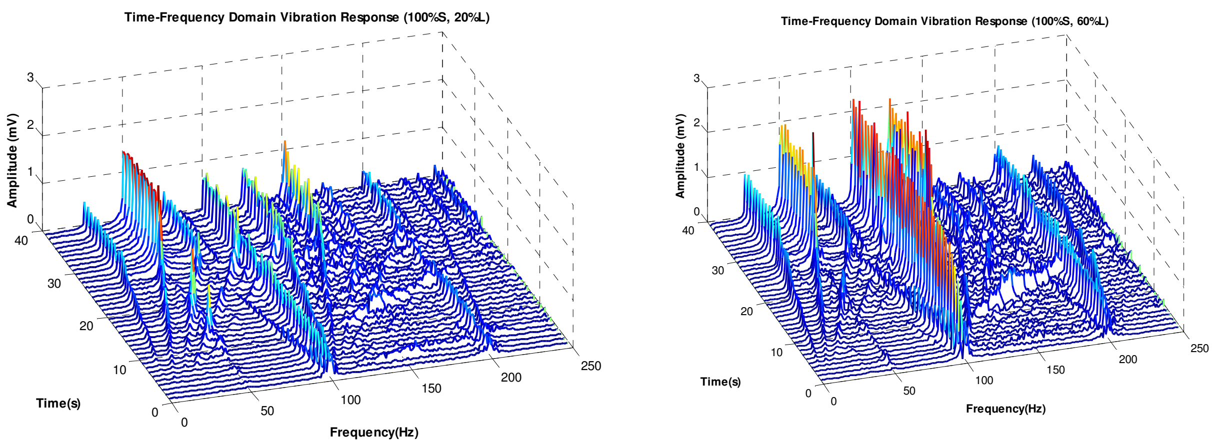

Figure 4 - Time and frequency analysis of vibration responses during speed-up transient at low and high load

The results of the transient test at high and low loads shows similar characteristics, but the amplitude at 100Hz and 200Hz spikes were much higher at higher load because more electrical current is required to drive the system and more severe imbalance will be created.

Although the dominated bearing vibration is below 100Hz, it may be influenced by the higher order modulation component due to supply imbalance and rubbing. Therefore, to suppress the imbalance will be one of the tasks in carrying more accurate study of bearing vibration. In addition, an investigation will be also carried out in characterizing the vibration from oil whirl and dry whip. It is one of the most common problems and often leads to the failure of the journal bearing system.

5. CONCLUSION AND FURTHER WORK

Based on theoretical and experimental analysis, it can be concluded that vibro-acoustics of a self aligning journal bearing is influenced by a several of factors including bearing radial and torsion loads, shaft speed, imbalance and lubricant system characteristics . Furthermore the vibration responses are also influenced by other vibration sources such as driving motor. Therefore, future investigation will emphasize developing more effective signal processing methods for analysing vibro-acoustic signals .

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