Raharjo, Parno, Gu, J., Fan, Yibo, Gu, Fengshou and Ball, Andrew

Early Failure Detection and Diagnostics of Self-Aligning Journal Bearing through Vibro-acoustic Analysis

Early Failure Detection and Diagnostics of Self-Aligning Journal Bearing through Vibro-acoustic Analysis

Journal:	*CM 2010 and MFPT 2010*
Manuscript ID:	Draft
Topic:	Condition monitoring (CM) methods and technologies
Date Submitted by the Author:	
Complete List of Authors:	Gu, Fengshou; University of Huddersfield, SCE
Keywords:	Acoustic emission, Vibration analysis, On-line oil condition monitoring

Early Failure Detection and Diagnostics of Self-Aligning Journal Bearing through Vibro-acoustic Analysis

P. Raharjo, J. Gu, Y. Fan, F. Gu and A. Ball

School of Computing and Engineering, University of Huddersfield,

Queensgate, Huddersfield, HD1 3DH, UK

Abstract

Journal bearings are widely used to support the rotor of industrial machinery with high loads, such as steam turbines, centrifugal pump and compressors. The problem of journal bearings can cause catastrophic failures, results in huge economic loss and create high safety risks. It is necessary to develop effective condition monitoring technologies to detect and diagnose the failures at early stage. Many researchers have studied the low frequency vibration characteristics as well as the high frequency vibration and acoustics emission in the detection of journal bearing failure. However, these studies have shown relatively little information regarding to the vibro-acoustic characteristics of the self aligning journal bearings. This research focuses on understanding the full feature of vibro-acoustics in association with the influence of radial, torsion load, radial load, shaft speed and lubricant viscosity properties in order to develop an more reliable way to monitoring journal bearings.

Keyword: S*elf aligning journal bearing, vibro-acoustic monitoring, early failure detection, fundamental frequency.*

1. Introduction

The bearings that are used for supporting rotating machinery are one of the crucial elements by which safe operation of the machinery is ensured. The function of a bearing is to provide relative positioning and rotational freedom while transforming a load between two parts, commonly a shaft and it's housing [1].

Experience has indicated that most of the engine or machinery problems were caused by bearing failure. In a study of machines with greater than 100HP, there was an indication that more than 40% of motor failures were caused by a bearing problem [2]. Stephen Flood showed that 13% of total mechanical seal failure is the consequence of distress that originated with bearing problems [3]. A breakdown of typical motor failure is indicated that approximately 40% of failure is bearing related [4]. Induction machine failure surveys have investigated the most common failure mechanisms in induction machine. These have been categorized according to the main components of machine stator related faults (36%), bearing related faults (40%), rotor stator related faults (10%) and other faults (12%) [5].

In power generation, the main problem of these machines is caused by the bearing fault. A power generator with a capacity of 560 megawatt, earning an average profit of $50 per hour, can avoid loses of about $ 11,000 an hour or $264,000 for a day [6].

The bearing is relatively cheap, but this component damage will cause catastrophic failure and creates difficulties in both economic and engineering aspects. Catastrophic failure is a serious contamination or disturbance characterized by the sudden and complete loss of components or machine performance. Catastrophic failure can be the result of gradual degradation and intermittent failure. Based on the time frame and manner in which the event occurs, failure can be classified in to catastrophic, intermittent, out of tolerance, and maladjustment failure [7].

From bearing condition monitoring, early faults and symptoms can be detected, therefore catastrophic failure and more disadvantages can be avoided. This paper focuses on the investigation of journal bearing vibro-acoustics for condition monitoring, which has found only a few investigations in the literature.

2. Journal bearing and surface interaction

Journal or plain bearings consist of a shaft or journal which rotates freely in a supporting metal sleeve or shell. There are no rolling elements in these bearings. Their design and construction may be relatively simple, but the theory and operation of these bearings can be complex [8]. Journal bearings are produced in many designs and sizes to match their many uses. These can be grouped into bushing, split journal bearing, spherical or self aligning plain/journal bearing, tilting pad bearing and thrust bearing.

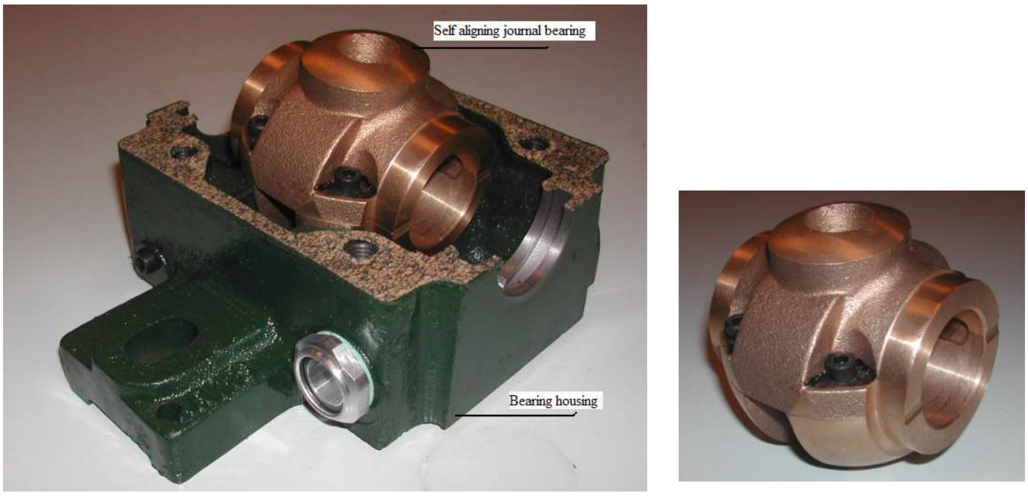

Figure 1 - Self aligning journal bearing for high speed

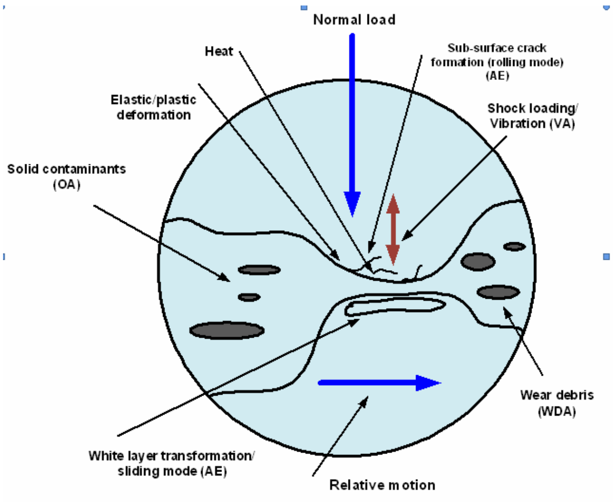

Figure 2 - Interaction between two surfaces in relative motion [9]

Figure 2 gives the interaction between two relative moving surfaces in journal bearings. As the first component contacts second component under a load condition, the approaching surface between them becomes a closely interrelated system influencing the way one component slides over another. 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. Increase in the load causes a decrease in the thickness of fluid film. If the lubrication film is too thin, asperity contact between surfaces is inevitable and wear particles are generated [10].

Techniques for condition monitoring of bearings include wear debris analysis, thermograph analysis, vibro-acoustic analysis and acoustic emission analysis. The first use of oil wear debris analysis dates back to the early 1940s by the railway companies in the Western United States. By the 1980s oil analysis formed the basis of Condition Based Maintenance in most railways in North America. Owing to the success of oil analysis in the railway industry, the American Navy used spectrometric techniques to monitor jet engines on their aircraft in the mid 1950s. Then commercial oil analysis laboratories first appeared on the scene in the early 1960s [11].

Vibration analysis is one of the most commonly used monitoring techniques in general industry. It has the advantage that it yields relevant data in quantitative format and can be operated remotely in the real-time mode and pre-set alarm limits can be triggered automatically [9].

Acoustic or sonic analysis is the measurement of sound waves caused by two or more component contacts. It is a term that is commonly used in the music recording industry, sound wave generation and human speech arena, but its application in industry with respect to monitoring bearing lubrication is relatively new. 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. Acoustic bearing analysis is intended for the lubrication technician and focuses on proactive lubrication measurements [12].

Many researchers such as Choy, F.K. et. al. [13] have studied the vibration characteristics of hydrodynamic journal bearings at low frequency ($f \leq 20$ kHz). Yoon D.J. et. al. [14] have investigated early detection of damages in journal bearing in the in the high frequency range of acoustic emission ($f \geq 100$ kHz). Rho, B.H. et. all.[15], have studied acoustical properties of hydrodynamic through nonlinear analysis including rotor imbalance.

The present paper focuses on the vibro-acoustic characteristic of the self aligning journal bearing. Vibro-acoustic characteristic of self aligning journal bearings are investigated through time-domain and frequency-domain analysis of the radial, torsion load, speed and viscosity. Vibro-acoustic of journal bearings may also be caused by contact between in the journal bearing and oil friction as well.

2.1 Schematics of self-aligning journal bearing

The schematic structure of a self-aligning journal bearing is presented in Figure 3. Radial load consists of an external load including weight of the rotor and unbalance load applied in the vertical direction (y-axis).

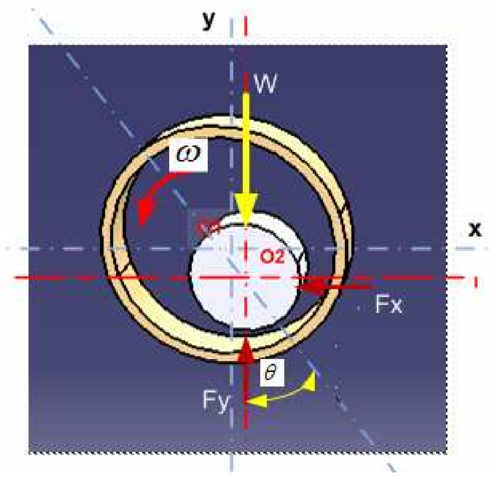

Figure 3 - Journal bearing schematic structure

The Reynold equation for pressure distribution in a journal bearing in steady condition may expressed as [16].

$$\frac{\partial}{\partial x}\left(h^3.\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(h^3.\frac{\partial p}{\partial z}\right)=6.\eta.V\frac{\partial h}{\partial x}+12.\eta.\frac{\partial h}{\partial t} \quad (1)$$

If the radius is denoted by R and the relations $x=R\theta$ and $V=\omega R$ are used, then the Reynold equation can be expressed as :

$$\frac{1}{R^2}.\frac{\partial}{\partial \theta}\left(h^3.\frac{\partial p}{\partial \theta}\right)+\frac{\partial}{\partial z}\left(h^3.\frac{\partial p}{\partial z}\right)=6.\eta.\omega\frac{\partial h}{\partial \theta}+12.\eta.\frac{\partial h}{\partial t} \quad (2)$$

Where: η is viscosity dynamic of lubricant (Pa.s), V is linear velocity of shaft (m/s), R is radius of shaft (m), c is radial clearance (m), ε is eccentricity ratio, and φ is contact angle (rad). For a long journal bearing, the pressure distribution is given in the following equation [17].

$$p(\theta)=\frac{6.\eta.V.R}{c^2}.\frac{\varepsilon(2+\varepsilon.\cos\varphi).\sin\varphi}{(2+\varepsilon^2)(1+\varepsilon.\cos\varphi)^2} \quad (3)$$

2.2 Vibration of self aligning journal bearings

Vibration of a self-aligning journal bearing is the response to forces caused by moving parts in the bearing including the external load, the mass of the rotor and the rotor imbalance. The elasticity system of a self-aligning journal bearing consists of a load, the rotor mass, mass unbalance, oil stiffness, damping coefficient and oil bearing mass. Vibration on the bearing system can be modelled using Jeffcot rotor model. The simplest lateral rotor vibration (LRV) can be described as the rotor orbital motion in an x-y plane as shown in Figure 4. It is connected to the bearing and the bearing housing through a linear spring and damper which can be excited by the time-varying radial unbalance force.

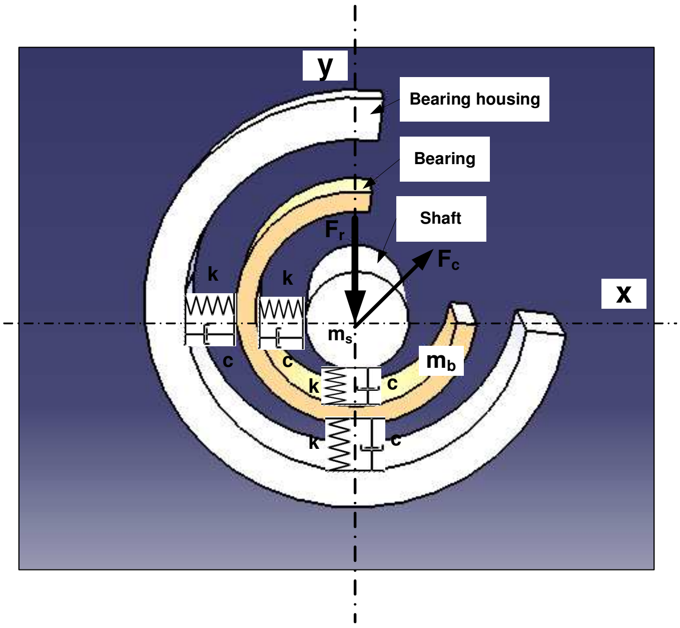

Figure 4 - Modification simplest LRV model that can handle x-y plane orbital [18].

For a system with multiple degrees of freedom, the motion equations in the y direction may be written in the following matrix notation.

$$[M]\{\ddot{y}\}+[C]\{\dot{y}\}+[K]\{y\}=\{f_y(t)\} \tag{4}$$

where M is mass matrix, C is damping matrix and K is stiffness matrix.

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

$$f_y(t)=F_r+W+m\omega^2 e\sin(2\pi t)+F_y \tag{5}$$

Where W is the gravity force; e is the eccentricity of shaft and f is the speed of shaft in Hz. f In the x direction, the motion equations may be written in the following matrix notation.

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{f_x(t)\} \tag{6}$$

The external bearing load in the x direction:

$$f_x(t)=m.\omega^2.e.\cos(\omega.t)+F_x \tag{7}$$

The reaction force of the oil film F_y and F_x are calculated by integrating the oil pressure to the bearing surface as follows: [16]

$$F_x=\int_0^A p.\sin\theta.dA$$

$$F_y=\int_0^A p.\cos\theta.dA \tag{8}$$

Where: F_y denotes the force of reaction in y direction (N); F_x is the force of reaction in x direction (N); p is the pressure of oil (Pa) and θ is the angular coordinate (rad).

2.3 Vibration transmission

In the journal bearing, vibration wave transmits in the radial direction from the component to outside through lubrication (fluid), bearing housing (solid) and air.

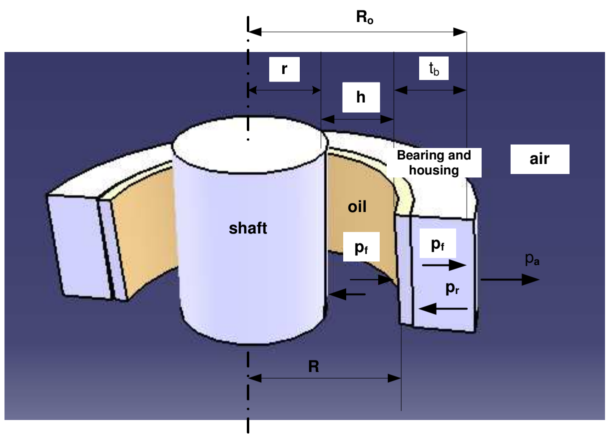

Figure 4 - The journal bearing vibro-acoustic wave transmits [16].

As shown in Figure 4, the relationship between oil pressure fluctuation P_f (Pa), pressure reflected wave P_r (Pa) and pressure transmitted wave P_a (Pa) may be written as [16].

$$p_a = p_f \cdot \frac{2.Z_s}{Z_b + Z_a} \tag{9}$$

Acoustic impedance of bearing Z_b (kg/m^2.s) is proportional to bearing density ρ_b (kg/m^3) and sound speed in bearing c_b(m/s) .

$$Z_b = \rho_b.c_b \tag{10}$$

Acoustic impedance of air Z_a (kg/m^2.s) is proportional to air density ρ_a (kg/m^3) and sound speed in air c_a(m/s) .

$$Z_a = \rho_a.c_a \tag{11}$$

3. Experimental study

To understand more about the vibro-acoustics of journal bearings an experimental study was carried out on the journal bearing test rig, as shown in Figure5, which consists of two self-aligning journal bearings at the drive end (DE) and non drive end (NDE). The rig is driven by a 3-phase induction motor controlled by a Siemens Micro Master Controller so that the rig can run at different speeds. The radial load is controlled by a hydraulic system and torque load is applied by the dynamic brake at the end of the test rig.

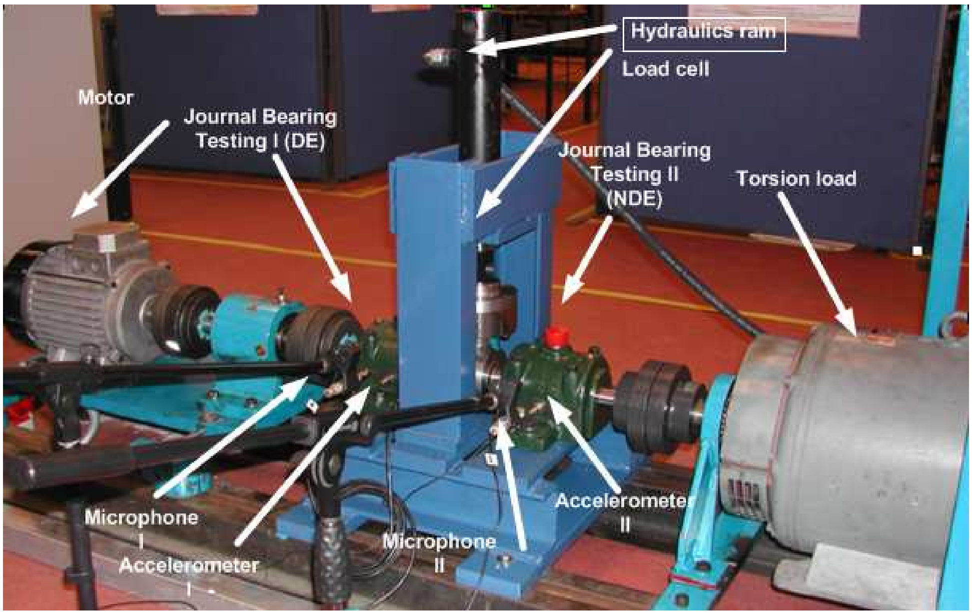

Figure 5 - Test facilities

The test rig is equipped with a load cell (CL-YB-11/2t) for the measurement of the radial load, two accelerometers (YB-3-8254) to measure vibration on bearing housing and two microphones (BAST YG 201-07065) to measure the acoustic noise, an AE transducer (SN AJ64) for AE measurement and an encoder (HENGSTLER 0 527 177) to measure the speed of the test rig. An 8-channel high speed data acquisition system (YE 6230B) was employed to record all the measurements at a sampling rate of 96kHz.

A vibro-acoustic test was conducted at 1480rpm (24.63 Hz) as the fundamental frequency (ff), with a constant radial load of 65.4N. Different torsion loads were applied at 10%, 20%, 30%, 40%, 50%, and 60% of the maximum torque. In addition a speed-up transient test was also conducted to identify frequency contents of the vibration signals.

4. Result and discussion

4.1 Vibration under constant speed operation

Waveforms illustrated in Figures 6 show typical vibration responses from the DE (drive end) journal bearing under different torsional load but at speed of 1480rpm. It can be seen that the amplitudes of vibration has a slightly increases with load increase. Based on the analysis in section 3, this small increase may be due to the effect of an investable eccentricity of the shaft system. Although the test was under constant speed and radial load, the radial forces incurred by the eccentricity will be increase with torsional load and thus the vibration responses. This analysis can be confirmed by spectrum analysis.

Figure 7 shows the vibration spectra in the frequency domain. It shows that the bearing vibration contents are mainly dominated by low frequency components. In particular, the spectrum shows distinctive peaks at shaft frequency 24.63Hz and its higher order harmonics, detailed in Figure 8. Nevertheless, Figure 7 shows that vibration amplitudes exhibit gradual increases with torsional load at the three shaft components: 24.63 Hz, 49.13 Hz and 73.75Hz, indicating that the eccentricity is the main source for the vibration increase.

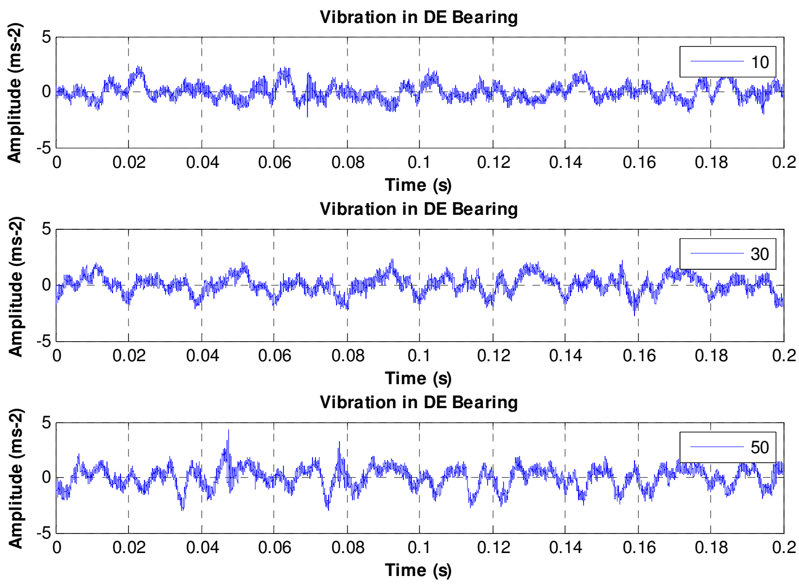

Figure 6 - Time domain vibration response at DE bearing

However, several distinctive components at frequencies: 100Hz, 200Hz and so on cannot be understood as the bearing system is not expected to have such sources. Therefore, a speed up transient test is performed to identify these components.

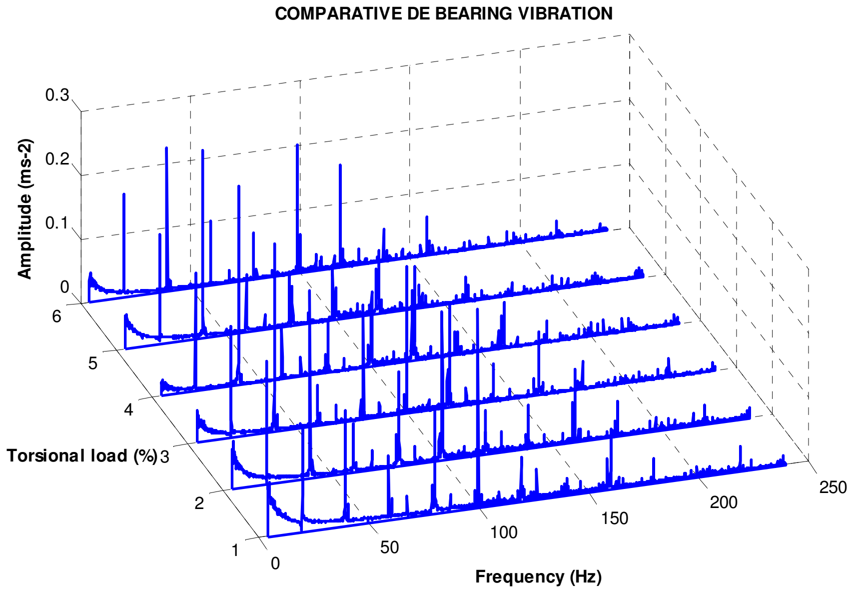

Figure 7- Frequency domain vibration response at DE nearing with torsional loads

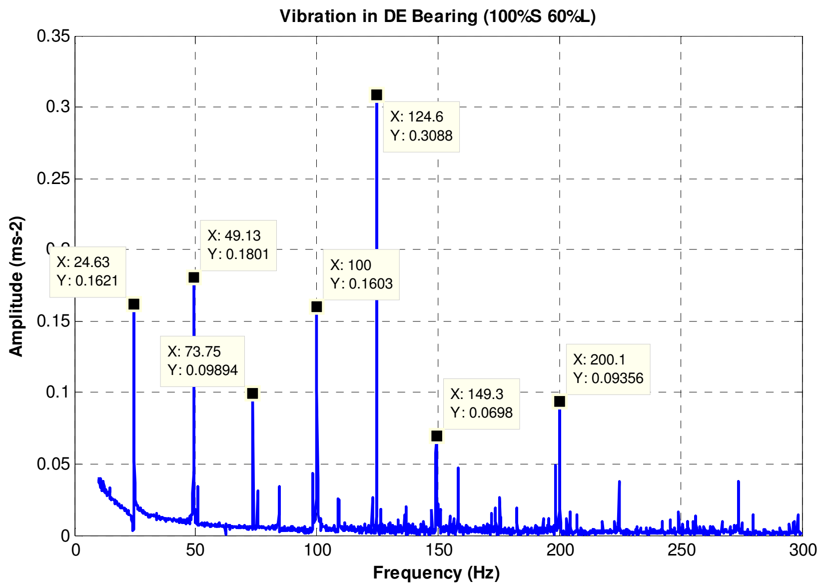

Figure 8 - Frequency domain vibration response at DE bearing

4.2 Vibration under speed transient operation

Figure 9 and 10 shows the results of transient vibrations from a speed-up test under low and high load respectively. They are obtained by applying the Shot Time Fourier transform(STFT) method to the transient signals. From figure 9 it can be seen that the 100Hz and 200Hz components appears when the driving motor start to run whereas the shaft related components; 24.63 Hz, 49.13 Hz and 73.75Hz do not show up until 10seconds late. This indicates that the 100Hz and 200Hz vibration comes from the electrical motor. In particular, the supply imbalance between the three phases is the source of this type of vibration. In addition, Figure 9 also shows that the supply imbalance vibration at 100Hz and 200Hz are clearly modulated by the shaft components, showing the nature of vibration due to the interactions of electromagnetic forces.

Moreover, it has observed that the vibration increase with shaft speed is much more significant compared with that with torsional load, discussed in section 4.1. This is particularly consistent with that predicted by the models.

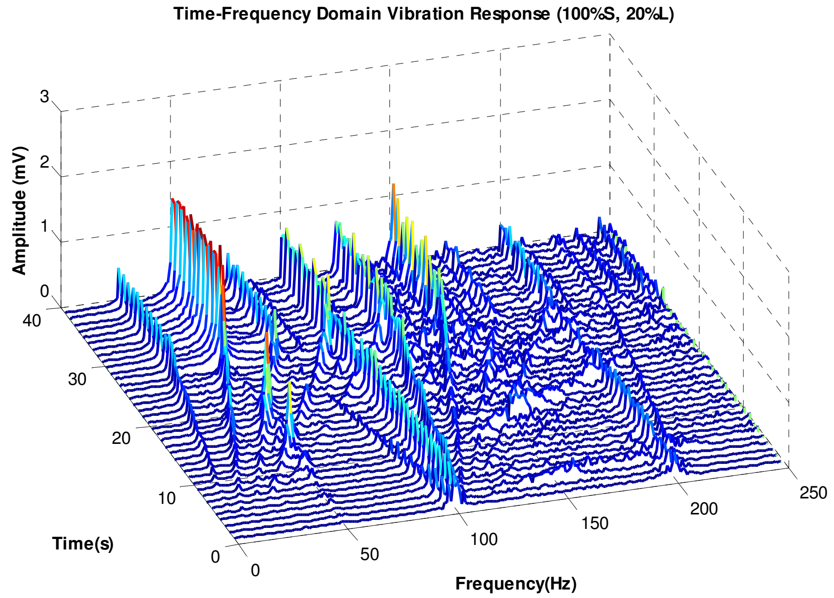

Figure 9 - Time and frequency analysis of vibration responses during speed-up transient at low load

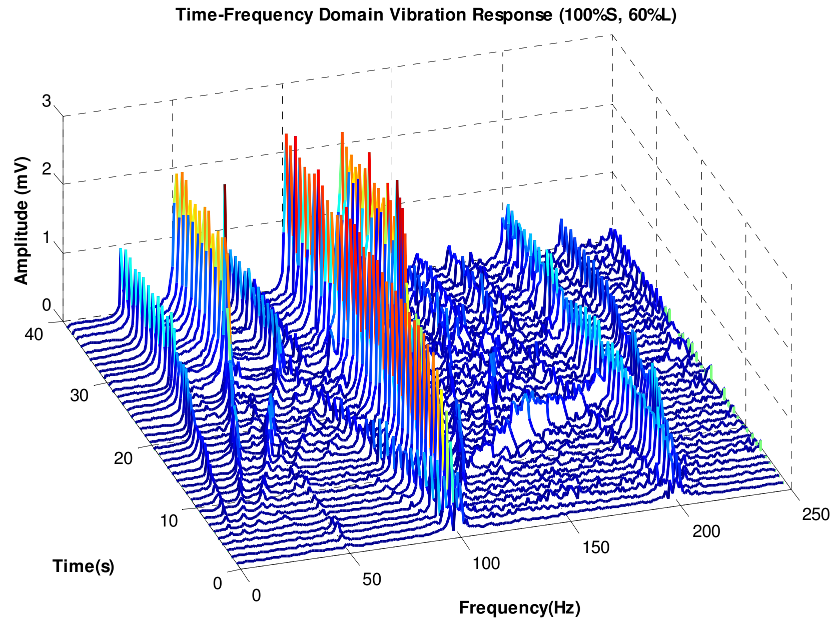

Figure 10 - Time and frequency analysis of vibration responses during speed-up transient at high load

The results of high load are show the similar characteristics but the amplitude at 100Hz and 200Hz much higher because of more electrical current are used to speed up the system and hence more sever imbalance will be resulted.

Although the dominated bearing vibration is below 100Hz, it may be influenced by the higher order modulation component due to supply imbalance. 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 characterising the vibration from oil whirl. It is one of the most common problems and often leads to the failure of bearing system.

5. Conclusion and further work

Based on theoretical study and experimental evaluations, it is found that vibro-acoustics of a journal bearing is influenced by a number of factors including bearing radial and tortional loads, shaft speed and lubricant characteristics. Furthermore the vibration responses are also influenced significantly by distant vibration sources such as driving motor. Therefore, future research will forces on developing more effective methods in measuring vibro-acoustics, in processing the vibration signals and in selecting diagnostic features.

References:

1. Ugural, A.,C., Mechanical Design, An Integrated Approach, McGraw-Hill Professional
2. Schoen, R.,R., Habetler, T.,G., Kamran, F. , and Bartheld, R., G., Motor bearing damage detection using stator current monitoring, IEEE Trans. On Industry Apllication, Vol. 31, No. 6, pp. 1274-9, Nov/Dec 1995.
3. Flood, S., Mechanical Seal Reliability, What Realistically can be Achieved, IMechE, London April 2007.
4. Shreve, D., H., Integrated Condition Monitoring Technologies, IRD Balancing LLC, 2003.
5. Sin, M.,L., Soong, W., L., and Ertugrul, N., Induction Machine On-Line Condition Monitoring And Fault Diagnosis, A Survey, University of Adelide.
6. http://www.maintenanceworld.com/Articles/calpine/Power-Company-failure.htm (Power company foresees failures).
7. Kreith, F.,The CRC Hand Book of Mechanical Engineering, CRC Press Inc, USA, 1998.
8. Scott, R., , Machinery Lubrication, A Noria Publication May / June 2009,.
9. Roylance, B.J, Machine Failure and its avoidance-what is tribology's contribution to effective maintenance of critical machinery?, Proceeding Institute of Machinery Engineering, Volume 217, Part J, Engineering Tribology, J05302, IMech, 2003.
10. Glaeser A, W, Ericson, R, C, Dufrane, K, F, *Tribology, The Science of Combating Wear, Lubrication Engineering*, Journal of The Society of Tribologists and Lubrication Engineers, November, 1992.
11. Oil Analysis History, http://www.bobistheoilguy.com/whatisoilanalysis.htm.
12. Gribble, J, Acoustic Analysis for the Rest of Us, Machinery Lubrication, January-February 2006.
13. Choy, F.K., Braun, M. J. And Hu, Y., Nonlinear transient and frequency response analysis of a hydrodynamic journal bearing, Trans. ASME, J. Tribology, 1992.
14. Yoon, D.J., Kwon, O.Y., Jung, M.H. and Kim, K.W.. Early detection of damages in journal bearing by AE monitoring, J. Acoustic Emission, 1995, 13(1),1-10.
15. Rho B.H, Kim K.W, Acoustical properties of hydrodynamic journal bearings, Tribology International, 36, 61-66, 2003.
16. Rho, B-H., Kim, D-G., Kim, K-W, Noise analysis of oil lubricated journal bearings, Mechanical Engineering Science, Proceeding Institution Mechanical Engineer, Vol 217, Part C, 2003.
17. Stolarski, T.A., Tribology in Machine Design, Butterworth Heinemann, London, 2000.
18. Adams,M.,L., Rotating Machinery Vibration, From Analysis to Troubleshooting, Marcel Dekker Inc. New York, 2001.
19. Collin H Hansen, Fundamental of Acoustics, University of Adelaide, South Australia.