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An Experimental Study of Acoustic Emission Generated by Journal Bearing

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***Abstract*—In many missions of rotating machines such as compressors, engines, turbines and large centrifugal pumps use Journal bearings to support and guide rotating shafts due to the high load capacity of this type of bearing. The effect of operating conditions on heavy duty in the journal bearings will degrade machine performance, reduce life time and increase risk. The application of high-frequency Acoustic Emissions AE signals is used to carry information about the details of the micro-collision process. Furthermore, the AE for journal bearing diagnosis is gaining acceptance as a useful complementary tool for the condition monitoring. The experimental investigation has been conducted to demonstrate the effect AE characteristics of the journal bearing in different critical operating conditions. The parameters rotational speed and radial load as well as different lubricant viscosity are mainly measured. For the overcoming occurrence of the bearing failures a comprehensive monitoring method has developed between friction in asperity contact and energy release of AE signals for journal bearing. A mathematical model is applied in order to obtain an insightful understanding of the influences operating parameters on the energy of AE generated in rolling element bearing. The obtained results showed that the potential of AE technology and data analysis method applied for monitoring the asperities contact condition in journal bearings, which will be vital for developing a comprehensive monitoring system, supporting the optimal design by simulating the Stribeck curve and operation of journal bearings.**

Keywords*— *journal bearing, acoustic Emission AE, Stribeck curve, root mean square.

I. Introduction

In 1846, Stribeck reported that the friction coefficient was inversely proportional to speed. Thus, he presented the characteristic curve of the coefficient of friction versus speed. Figure1 illustrations the Stribeck curve, this shows the relationship between the coefficient of friction and bearing parameter or modulus ηN/p, where η is the absolute viscosity of the lubricant in kg/m.s, N is the shaft speed in rpm and p is the pressure on the projected area in Pa. The optimum point of the curve is when the coefficient of friction passed through a minimum point from mixed to hydrodynamic lubrication [1]. The coefficient of friction changes with lubrication regime: mixed-film lubrication and hydrodynamic lubrication as shown in Stribeck's curve [2].

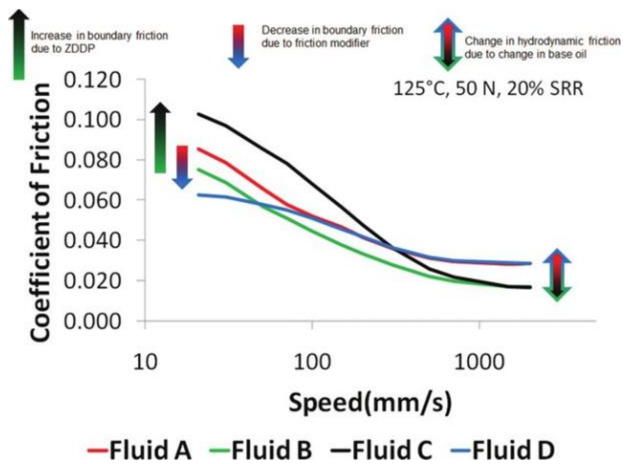


Fig 1 Stribeck curve under different types of lubricants [2]

II. Asperity Collision Excitation

In general, any journal bearing is designed to operate under the hydrodynamic regime. Due to abnormal operating conditions during transient operations or fault cases like oil leakages, oil degradations and worn surfaces might make it often operates under mixed or boundary regimes [3]. In these two regimes asperity collisions occur and result in increased relief plastic deformation energy. Fig 2 illustrates that asperity contact can be expressed as a source of plastic wave in journal bearing. It shows that one of the asperity pairs has a larger degree of collisions, which produces higher transient energy; whereas the smaller contact produces lower transient responses. The stiffness of a single asperity contact pair are built and calculated based on shoulder-shoulder asperity contact [4] as shown in Fig 3 shoulder-shoulder asperity contactFig 3.

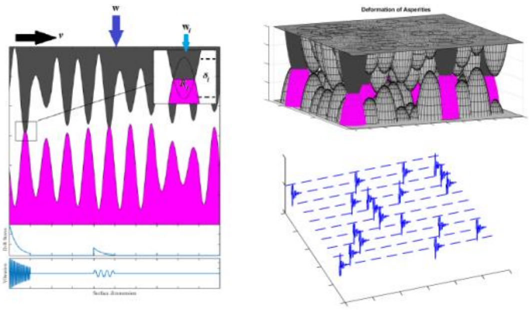


Fig 2 Asperity of deformation caused by collision [4]

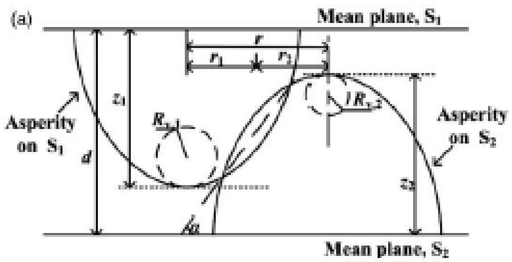


Fig 3 shoulder-shoulder asperity contact [5]

III. MATHEMATICAL MODEL CONCEPT

A. ModellingFriction

Modelling the friction between the asperities collisions which is the main sources of AE in journal bearing. In general, the tangential contact friction F between a pair of asperities in contact can be obtained [6] as:

$$F = \iint \tau dA \qquad (1)$$

Where τ is shear stress at the asperity contact and A is the area of one asperity in contact. The friction coefficient f of a single asperity can be expressed as:

$$f = \frac{\tau}{P} \qquad (2)$$

Where P is the normal pressure in a single asperity contact. Substituting by τ from the equation (2) into the equation (1) and rearranging the integral yields to:

$$F = \iint \tau dA = \iint fpdA = f\,W \qquad (3)$$

Where W is the normal load. Considering the Greenwood and Williamson models [7] for the real contact surfaces, the probability of making contact at any selected asperity can be expressed as (see Fig 4):

$$p(z) = f(z)dz \qquad (4)$$

If the number of asperities per unit area is D, the expected number of contacts in any unit area is:

$$n = D\int_d^\infty f(z)fz \qquad (5)$$

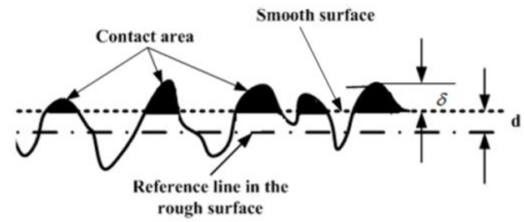


Fig 4 Greebwood and Williamson for contact model [7]

Based on Hertz [6] [8] theory and on this model the maximum deflection δ in the contact area can be expressed as:

$$\delta = (\frac{w^2}{E'^2R'})^{1/3} \qquad (6)$$

B. Friction Energy Release Rate Model

The frictional work done by friction force F on a point that moves a sliding distance S in the direction of tangential sliding contact is:

$$U_{iAE} = \int Fds \qquad (7)$$

Now, it can calculate sliding distance S in asperity contact based on Fig 5. From that figure, a represents the Hertzian radius of the asperity contact circle and given by:

$$a = (\frac{w\,R'}{E\prime})^{1/3} \qquad (8)$$

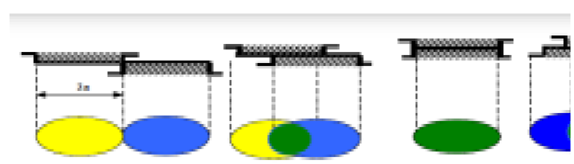


Fig 5 The concept of displacement in asperity contact [9]

Thus, sliding distance S in asperity contact can be expressed as:

$$S = 2a + 2a = 4(\frac{w\,R'}{E\prime})^{1/3} = a(\delta R')^{1/2} \quad (9)$$

Substituting by F from Equation (3) and extracting ds from Equation (9) into Equation (7) hence it can be rearranged as:

$$U_{iAE} = \int Fds = \int \frac{4}{3} fw(\frac{E}{wR})^{2/3}\,dw =$$
$$f(\frac{E}{R})^{2/3}w^{4/3} \qquad (10)$$

Therefore, the Equation (10) can be rearranged as well in terms of maximum deflection as follows:

$$U_{iAE} = f(\frac{E^4}{R})^{1/3}w^{2/3}\delta \qquad (11)$$

Since δ = Z-d, thus the mean frictional work of one asperity contact is:

$$\bar{U}_{iAE} = f\left(\frac{E^4}{R}\right)^{\frac{1}{3}} \frac{\int_d^\infty w^{\frac{2}{3}}(z-d)f(z)dz}{\int_d^\infty f(z)dz} \qquad (12)$$

The total frictional work at the asperity contacts UAE can be expressed as:

$$U_{AE} = A_0 n \bar{U}_{iAE} \qquad (13)$$

Where A0 is the apparent contact area; n is the number of contacts in unit area as given by Equation (5). Thus, the total frictional energy can be expressed as:

$$U_{AE} = A_0 n\, f\left(\frac{E^4}{R}\right)^{\frac{1}{3}} \frac{\int_d^\infty w^{\frac{2}{3}}(z-d)f(z)dz}{\int_d^\infty f(z)dz} \qquad (14)$$

Substituting by n from Equation (5) into Equation (14) will result in:

$$U_{AE} = A_0 D\, f\left(\frac{E^4}{R}\right)^{\frac{1}{3}} \int_d^\infty w^{\frac{2}{3}}(z-d)f(z)dz \qquad (15)$$

The total time in frictional contact can be defined and calculated as:

$$t = \frac{S}{v} = \frac{4a}{v} = \frac{d(\delta R\prime)^{1/2}}{v} \qquad (16)$$

Where v is sliding speed. By using δ = z −d, the mean friction contact time is:

$$t = \frac{4\int_d^\infty R'^{\frac{1}{2}}(z-d)^{\frac{1}{2}}f(z)dz}{v\int_d^\infty f(z)dz} \qquad (17)$$

Now, it can define the total number of asperity contacts between the two surfaces as follows:

$$N = A_0 D \int_d^\infty f(z)dz \qquad (18)$$

Dividing Equation (15) by Equation(17), and defining all constant parameters as K1, the acoustic energy release rate can be expressed as:

$$\dot{U} = Vv(K_1 w^{2/3}) \qquad (19)$$

Supposing that a portion (kf) of the energy released as a result of friction converts to AE pulses and the gain of the AE measurement system is (kg) then [9]:

$$\dot{U}_{AE} = K_f K_g N v(K_1 w^{\frac{2}{3}}) \qquad (20)$$

C. Frictional AE Model

Based on detailed discussion found in the work of Fan et al. [10], the RMS value of the AE signal excited by the contact friction can be expressed as:

$$V_{rms} = \sqrt{R\,\dot{U}_{AE}} \qquad (21)$$

Where R is the electrical resistance of the AE measuring circuit. Substituting Equation (20) into Equation (21) and defining all constants with K, Vrms in frictional asperity contact can be expressed as:

$$V_{rms} = \sqrt{KNvw^{2/3}} \qquad (22)$$

Based on the Equation (22), the RMS value of AE signal increases with load, rotational speed and number of asperities in contact [9].

IV. ACOUSTIC EMISSIONS SIGNALS

The main source of acoustic emission signals is mainly generated by sliding contact of friction surfaces at the microscopic level (asperity contact) Boness and McBride [11]. Acoustic Emission (plastic energy release) carries evidence about micro structure deformation, many researchers stated on the condition monitoring field using AE technology [12]. AE technique is able to process and record thousands of AE event per second. Count rate and amplitude with sliding contact have been investigated using time and frequency domains to carry information about the details of the micro-damage process. Frequency domain analysis could characterize AE signals, nevertheless AE proceedings, such as high frequency, attenuation, dispersion, multiple reflections and the non-linear character which is often difficult to identify effective features for fault diagnosis. Miettinen and Siekkinen [6] and Mba et al. [13] confirmed that the time domain statistic parameters of AE signals can be used in condition monitoring technique. They reported that the possibility of detecting leakage, dry running and cavitation by measuring the RMS value of AE signal of a mechanical seal on a centrifugal pump.

In additional studies, Fan et al. [10] [14] stated that the level of raw Acoustic Emission measurement depends on the sliding operating conditions such as speed, the load supported by contact, and the number of asperity collisions, as well as the surface topographic characteristics among others [10].

V. EXPERIMENTAL PROCEDURE

The test rig of journal bearing consisted two self-aligning spherical journal bearings (SA35M), electrical motor and load system have been exerted radial load on the shaft supported between bearings as shown in Fig 6.

Moreover, AE measurement instrumentation transducers were mounted. Also, an encoder and a pressure sensor are placed in order to measure the output rotating speed and radial load, respectively. The journal bearing has been tested to validate the plastic energy release by investigating acoustic emission response. The experiments were carried out to test three different lubrication types under different radial loads and different rotating speed. This model demonstrates a clear correlation between AE Root Mean Square (RMS) value and sliding speed, contact load and number of contact asperities. To benchmark the proposed model, a mechanical seal test rig was employed for collecting AE signals under different operating conditions. Then, the collected data was processed using time domain and frequency domain analysis methods to suppressing noise interferences from mechanical system for extracting reliably the AE signals from mechanical seals.

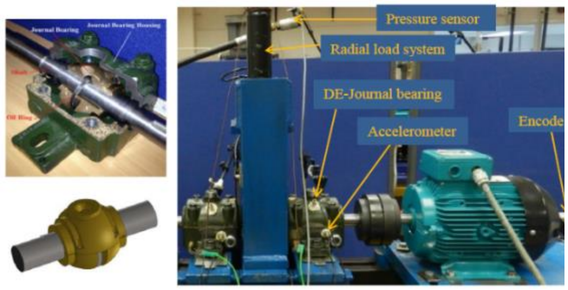
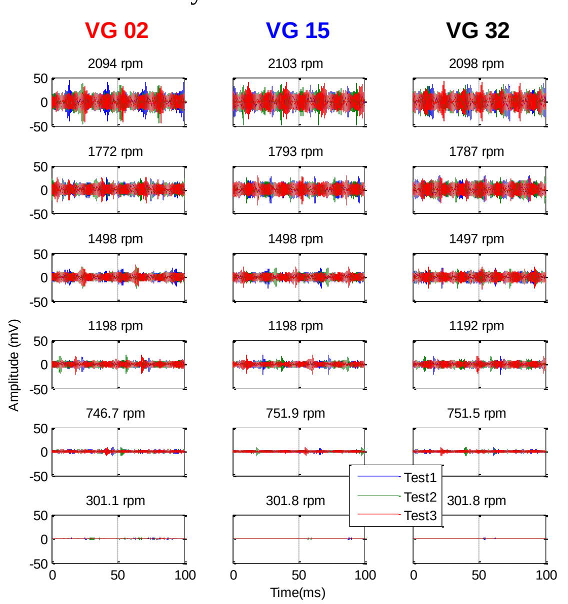


Fig 6: Self-aligning journal bearing companies and test rig [15]

VI. Results and Discussion

A. Time Domain and Frequency Domain Results

The investigation of time and frequency domains features of AE is simplified presented the energy release of a journal bearing as shown in Fig 7. Time domain features show that the elastic energy is increased while rotating speed increases. Fig 7 illustrations the AE power spectrum of journal bearing at multiple operating conditions. Also, it shows that spectrum signatures obviously appearance frequency of high speed. The high - frequency band might be associated with asperity-asperity smashes, so asperity collusions release more power during shaft rotating with high speed. However, time domain and frequency domain could not recognize the differences between different oils easily.



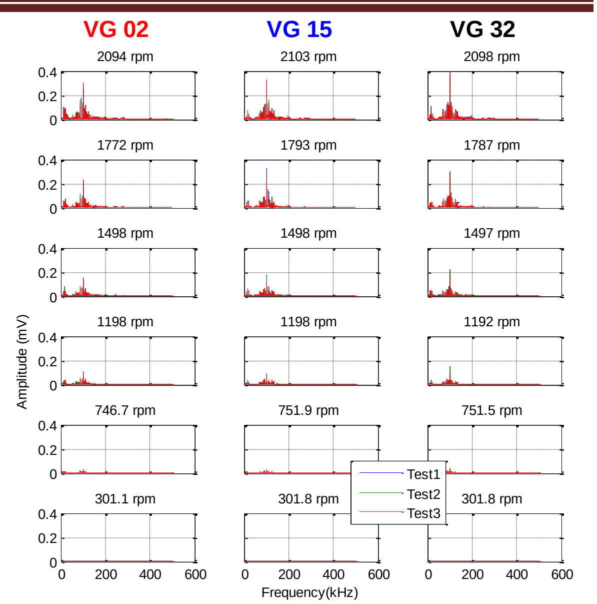
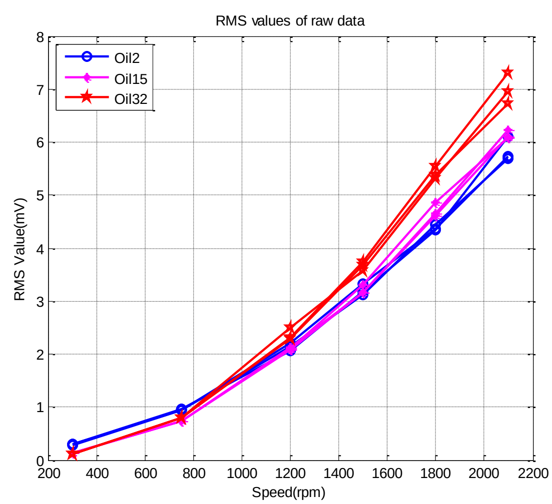


Fig 7: AE signals from a journal bearing

B. RMS Values

RMS values of AE signals provide an overall track statistics of the journal bearing situation. Fig 8 shows the overall RMS values of raw vibration data which measured from AE time domain and AE high frequency domain. RMS values show that high speed of rotating machine release more energy. Moreover, at low speed the high viscosity has less AE energy. On the other hand, at high speed the high viscosity has more AE energy. During changing rotating speeds, the RMS feature twisting based on oil type. This phenomenon in some way presents Stribeck curve at mixed-film lubrication and hydrodynamic lubrication regimes.



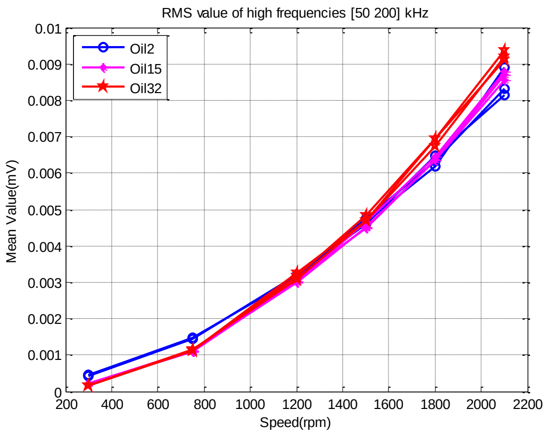


Fig 8: AE RMS value and rotational speed for journal bearing

VII. Conclusion

This paper was combined with two established signal processing methods: time domain and frequency domain analysis. Lubricant film thickness in Journal bearings rises the shaft to moderate asperity-asperity smash by separating the shaft and the bearing surfaces. The effect of lubricant type and rotating speed in the journal bearings will degrade machine performance. Acoustic Emissions AE signals successfully carry information about the details of the micro-collision process during different parameters mathematically and experimentally. Also, RMS values of the acoustic emission produced frictional asperity collision and operating parameters such as rotational speeds, loads and oil types have been established mathematically and measured experimentally. AE analysis successively monitoring the asperities contact condition in journal bearings. High rotating speed, always has high elastic energy releasing by asperity contacts. This study consecutively simulating the Stribeck curve by evaluating RMS value of time and high frequency domains. Likewise, AE RMS values might present the optimum operating conditions at exchanging values.

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