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Characterization of Acoustic Emissions from Mechanical Seals for Fault Detection

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Abstract

The application of high-frequency Acoustic Emissions (AE) for mechanical seals diagnosis is gaining acceptance as a useful complimentary tool. This paper investigates the AE characteristics of mechanical seals under different rotational speed and fluid pressure (load) for develop a more comprehensive monitoring method. A theoretical relationship between friction in asperity contact and energy of AE signals is developed in present work. 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. The results reveal the potential of AE technology and data analysis method applied in this work for monitoring the contact condition of mechanical seals, which will be vital for developing a comprehensive monitoring systems and supporting the optimal design and operation of mechanical seals.

1. Introduction

Mechanical seals are one of the most important types of shaft seals which are found on rotating machinery and can protect against leakage across much higher pressure differences than the other seals [1]. The seal is made between the very smooth, very flat faces of two rings, one is attached to and rotates with the shaft, and the other is attached to the housing and is stationary. A schematic illustration of the components of a mechanical seal is shown in Figure1.

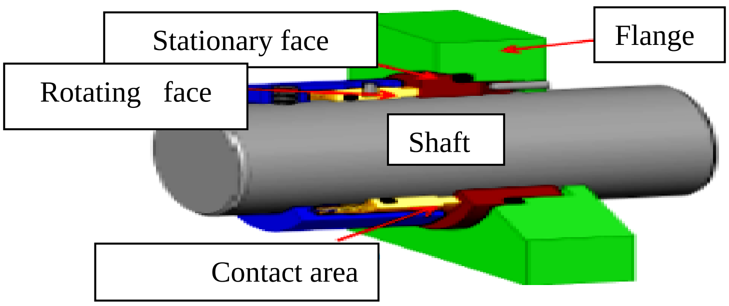

Figure 1. Schematic illustration of mechanical face seals [2]

Since AE signals carry information about the details of micro-damage process, a significant amount of research amount of research has been reported on the condition

monitoring of mechanical seals using AE technology[3]. for instance, lingard et al.[4], Hisakado et al.[5] and Hase et al.[6] investigated the correlation between AE signal characteristics such as AE count, count rate and amplitude with sliding contact. During condition monitoring of mechanical seals, today's AE systems are able to process thousands of AE events per second and record them to data storage. A number of studies have investigated the use of the time-domain andfrequency-domain techniques in processing the AE signals from mechanical seals and showed that using these techniques, it is possible to process AE signals using sophisticated computing methods [7-8]. Analysing the AE signal in the frequency domain could characterise the AE signals especially under noisy conditions, however it is often difficult to identify effective features for fault diagnosis in the frequency spectra due to the limitations mainly caused by the nature of AE events such as high frequency, attenuation, dispersion, multiple reflections and the non -linear character of AE signal during its propagation [9]. Holstein et al. [8] claimed that analysis in the frequency domain was investigated in the research on fault diagnosis of mechanical seal. However, they failed to present the analysis results to prove the effectiveness of this method. Boness and Mcbride [10] performed a comprehensive study involved the measurement of AE signals in sliding contact and reported that impact of friction surface at microscopic level (asperity contact) was the main source of acoustic emissions during pure sliding friction. They also proposed an empirical model which presents AE RMS value in terms of wear volume. Miettinen and Siekkinen [11] studied the AE response to the sliding contact behaviour of a mechanical seal on a centrifugal pump under different working conditions. They reported the possibility of detecting leakage, dry running and cavitation in face seals by measuring the RMS value of AE signal. In subsequent studies, Mba et al. confirmed that the RMS value of AE signals can be used to monitor seal condition [7]. In another study, Fan et al. developed a mathematical AE model based on elastic asperity contact and pointed out that the level of AE measurement depends on the sliding speed, the load supported by contact, the number of asperity contact, and surface topographic characteristics among others [12].
In this paper, a mathematical model of the AE generated in frictional asperity contact is developed to establish a relationship between AE RMS value and working parameters of seals (rotational speed, load and number of asperities in contact). Furthermore, the AE characteristic of mechanical seals are presented in the time domain and the frequency domain firstly and then AE feature parameters such as RMS and kurtosis values are successfully explored in associating with fluid pressure (load) and rotational speed of shaft.

2. Mathematical model concept

2.1 Modelling friction

The basic idea behind this work is modelling the friction between the asperities which is assumed to be the main sources of AE in mechanical seals. In general, the tangential contact friction between a pair of asperities in contact can be defined by [13]:

$$F = \iint \tau \, dA \tag{1}$$

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

$$f = \tau / p \tag{2}$$

where, P is the normal pressure in a single asperity contact. Substituting τ from Equation 2 into Equation 1 and rearranging the integral yields to:

$$F = \iint \tau \, dA = \iint f \; p \, dA \;\; = f \, W \tag{3}$$

Where W is the normal load. Considering the Greenwood and Williamson models [14] for the contact of real surfaces, the probability of making contact at any selected asperity can be expressed as (see Figure2):

$$p(z) = f(z) \, dz \tag{4}$$

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

$$n = D \int_d^{\infty} f(z) dz \tag{5}$$

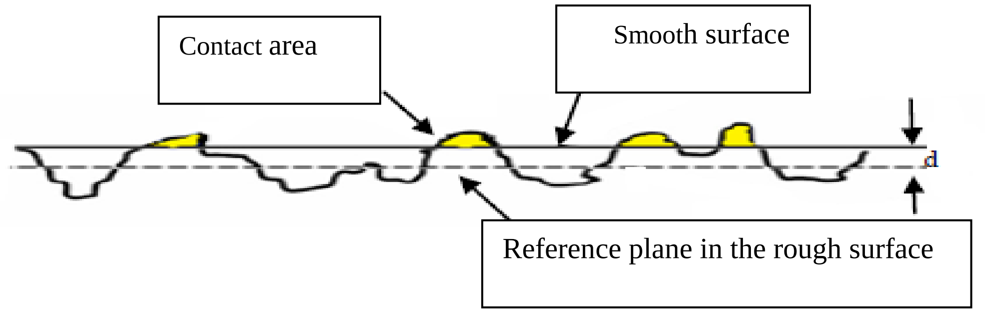

Figure 2. Greenwood and Williamson for contact model

Based on Hertz theory the maximum deflection in the contact area can be expressed as:

$$\delta = \left(\frac{W^2}{E'^2 R'}\right)^{1/3} \tag{6}$$

2.2 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 the product:

$$U_{iAE} = \int F ds \tag{7}$$

Now we calculate sliding distance S in asperity contact based on Figure 3. In this Figure, a is the Hertzian radius of the asperity contact circle given by:

$$a = \left(\frac{W R'}{E'}\right)^{1/3} \tag{8}$$

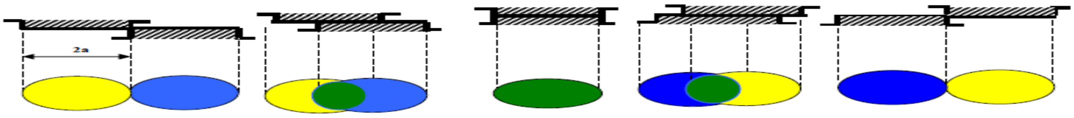

Figure 3. The concept of displacement in asperity contact

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

$$S = 2a + 2a = 4\left(\frac{W_i R'}{E'}\right)^{1/3} = 4(\delta R')^{1/2} \tag{9}$$

Substituting F from Equation (3) and extracting ds from Equation (9), Equation (7) can be rearranged as:

$$U_{iAE} = \int F ds = \int \frac{4}{3} f W \left(\frac{E}{WR}\right)^{\frac{2}{3}} dW = f\left(\frac{E}{R}\right)^{\frac{2}{3}} W^{\frac{4}{3}} \tag{10}$$

Equation (10) can be re arranged in terms of maximum deflection as follow:

$$U_{iAE} = f\left(\frac{E^4}{R}\right)^{\frac{1}{3}} W^{\frac{2}{3}} \delta \tag{11}$$

Since $\delta = Z - d$, the mean frictional work of one asperity contact is:

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

The total frictional work in the asperity contacts U_{AE} can be expressed as:

$$U_{AE} = A_0 n \overline{U}_{iAE} \tag{13}$$

where A_0 is the apparent contact area; n is the number of contacts in unit area given by Equation (5).Thus, the total frictional energy can be expressed as:

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

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

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

The total time in frictional contact can be calculated as:

$$t = \frac{S}{v} = \frac{4a}{v} = \frac{4(\delta R')^{1/2}}{v} \tag{16}$$

where v is sliding speed. Using $\delta = z - d$, the mean friction contact time is:

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

Now we define the total number of asperity contacts between the two surfaces as:

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

Dividing Equation (15) by Equation(17), and defining all constant parameters as K_1, the acoustic energy release rate can be expressed as:

$$\dot{U} = Nv\,(K_1 W^{2/3}) \qquad (19)$$

Supposing that a portion (k_f) of the energy released as a result of friction converts to AE pulses and the gain of the AE measurement system is (k_g) we have :

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

2.3 Frictional AE Model

Based on detailed discussion found in the work of Fan et al. [11], 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, V_{rms} in frictional asperity contact can be expressed as:

$$V_{rms} = \sqrt{KN\,v\,W^{2/3}} \qquad (22)$$

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

3. Experimental Test Rig and Test Mechanical Seal

The test rig is shown in Figure 4. The driving power is a 3.0 kW, 3-phase AC electric inductions motor and the drive shaft can be run at different speeds up to a maximum of 2800 rpm. Two John Crane Type 1648 MP pusher cartridge seals and a stainless steel tube formed a pressurized chamber. An auxiliary circulating system, was connected with the chamber to pressurize the working fluid (plain water in this research) and take away the heat generated by the friction of mechanical seal. Two WD S/N FQ36 AE sensors (ch1 and ch2) with an operating frequency range from 100 kHz to 1MHz were employed to obtain the AE signals, allowing high frequency events due to asperity contacts to be monitored. The transducers were placed directly on the cartridge of the non-driven end (NDE) seal as shown in Figure 4(right). The signal from AE sensor is amplified and acquired by a 2MHz high speed data acquisition system with 16 bit resolution.

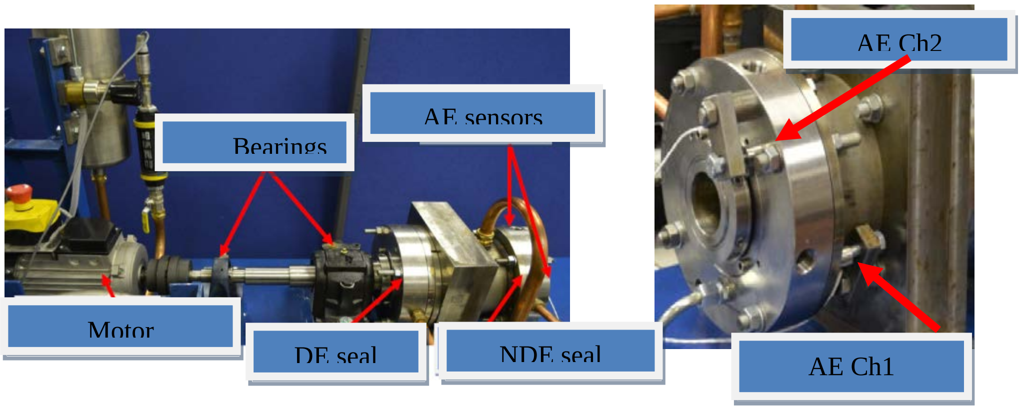

Figure 4. Layout of the test rig (left) and position of AE transducers (right)

4. Experimental Procedure

In the industrial applications, the AE signal is created not only by the contact of the mechanical seal faces, but also from other sources, such as the bearings, electrical motor and electromagnetic noises. In order to monitor the sliding contact of the mechanical seals, it is necessary to distinguish the AE signals generated by the seal faces from the background noises, which requires an appropriate experimental procedure and signal processing method. The experimental study in this work included four comparative programs: manual operation of seal (MOS), running with sensor on seal (RSS), running with sensor on flange (RSF), and test under different speeds and pressures (TDSP). In the MOS experiment, the shaft was turned manually to generate a sliding of seal faces. The purpose to carry out this test was to relate the acoustic emission event to the sliding contact of faces. In addition, MOS and RSS were used to compare the difference between AE signals picked up under static and running condition. The RSF refers to running the test on condition that the AE sensors were mounted far from seal faces as shown in Figure5. As can be seen, in RSF experiment, the position of AE sensors is far from the sealing area, consequently it is expected that no AE signal would be picked up from the sealing interface due to the quicker attenuation of higher frequency signal. Purpose to carry out this test was to evaluate the noises.

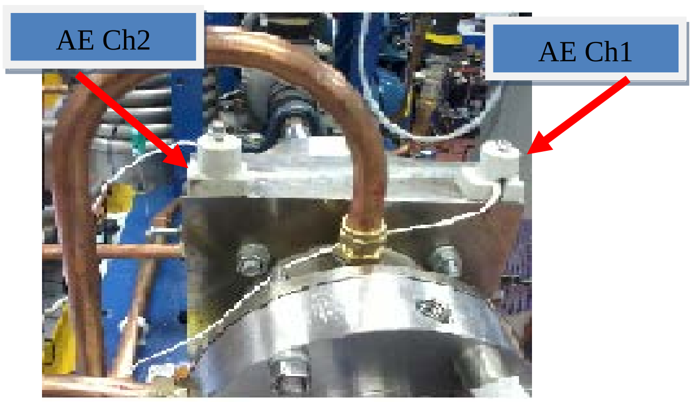

Figure 5. Position of the sensors on flange

Both the RSS and RSF tests were carried out under 4 bar (sealed pressure) and at 900 rpm to allow the seal faces to come into contact. These tests would be compared to evaluate the proper frequency band of frictional AE signal and eliminate the noise signals.The aim of TDSP program was to establish a correlation between AE activity with rotational speed and load. This was accomplished by controlled incremental load at constant speed. The TDSP experiment consisted of five different speeds: 900,1200,1500,1800 and 2100 rpm, at each speed there were three different loads (sealed pressure): 3bar, 5bar and 8bar. To evaluate the repeatability of the tests, each test has been run for three times. Through all the tests the seals were cooled with plain water. The tests were performed under near isothermal conditions and starts save at a temperature around 28°C, so that the effect of temperature on the acoustic emission responses can be reduced.

5. Results and Discussion

5.1 Frictional AE Generation

During this test the AE data acquisition system was kept on working and the shaft of the test rig was turned manually. AE events were clearly observed in the signals picked up by both AE sensors. Since it is impossible to generate noise by just turning the shaft, the AE was likely caused by the rubbing of seal faces rather than by the noise sources of the test rig. As an example, Figure 6 shows the typical raw AE signals around an event. It can be observed that only background noises were recorded before and after the turning of the shaft. This proves that the signals from the AE sensors are the acoustic emissions caused by the rubbing of seal faces. When the shaft of the test rig was turned, the amplitude of AE signals from the both sensors increased significantly and their magnitudes were similar.

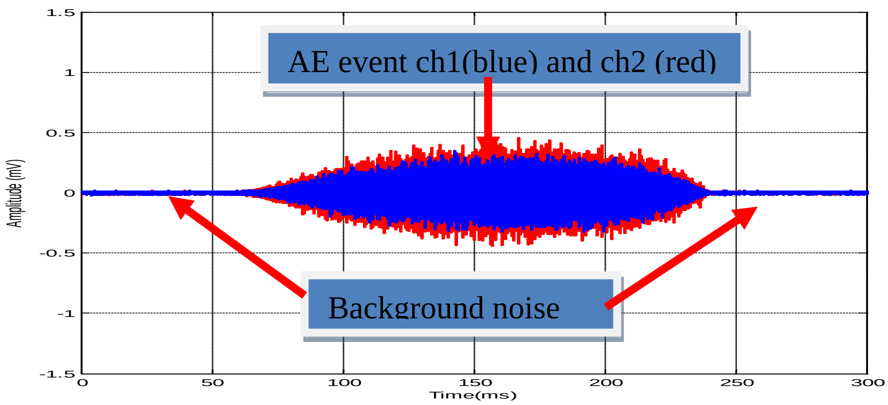

Figure 6. The raw AE signal in the MOS experiment

5.2 Identification of the AE Signals from Faces

To insight into the AE signals from seal faces, analysis in the frequency domain was carried out on experimental data and the results are shown in Figure 7-9. The AE signal from RSS test coincides very well with the signals in the MOS test. It can be seen that the frequency peaks is approximately the same but the amplitude of the former was higher than that of the latter. Compared the MOS test, the RSS test generates more background noises and burst type emissions. In RSF, the AE energy mainly concentrated in two frequency bands whose center frequencies were about 50 and 100 kHz, as shown in Figure 9. In RSS, however, in addition to the above two frequency bands, there were some higher frequency bands whose center frequencies were about

200,270,370 and 450 kHz. Based on our comparison of AE spectra, It can be concluded that the frequency of AE from the seal faces was concentrated in six frequency bands at center frequencies of about 50, 100, 200,270,370 and 450 kHz, and that the first two lower frequency bands also existed in all AE signals and signals from other sources of the test rig. These results led us to consider the frequency band around from 100 to 500 kHz to be the specific band from the seal faces. To eliminate the noise signals, we used a digital band-pass filter of the frequency band 100–500 kHz. RMS, skewness and kurtosis of the filtered signals were calculated to express the AE energy generated by seal faces and to characterize AE signals from mechanical seals.

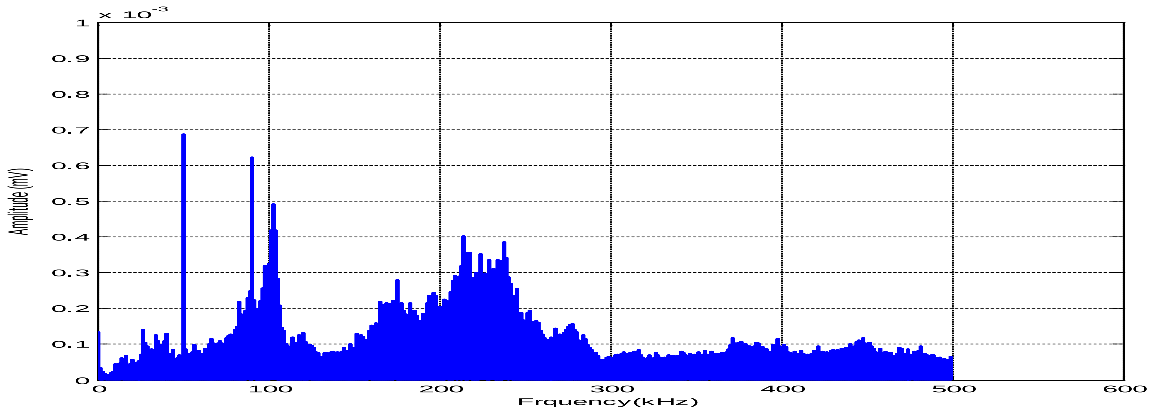

Figure 7. The AE spectrum in MOS test for ch1

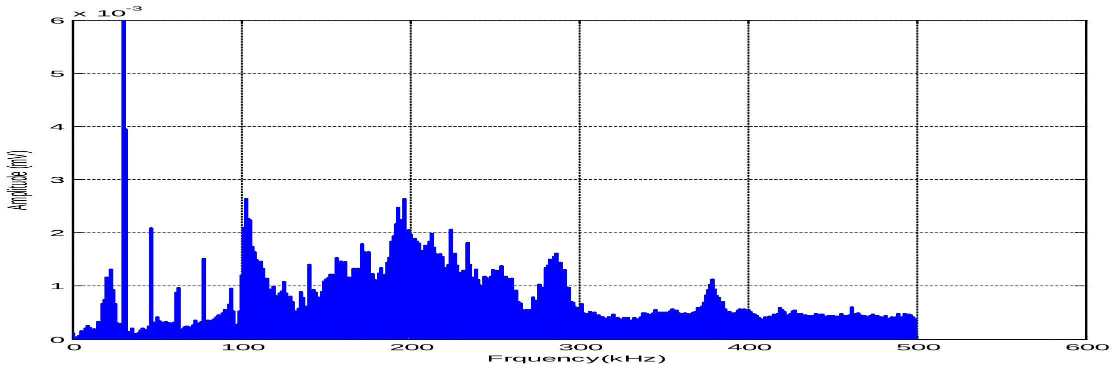

Figure 8. The AE spectra in RSS test for ch1

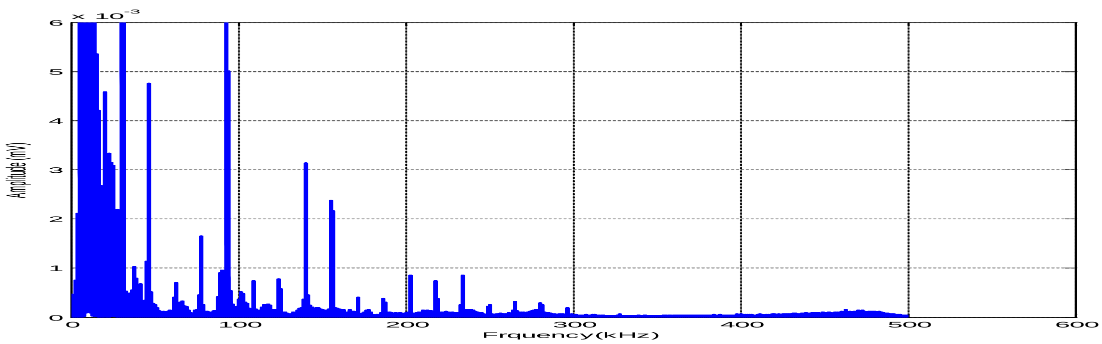

Figure 9. The AE spectra in RSF test for ch1

5.3 AE Waveform

Figure 10 shows the amplitude of the AE time domain signals of the NDE mechanical seal when operated at 1500 rpm rotational speed with three loads; 3 bar, 5 bar and 8 bar . Three tests were performed. As can be seen, the AE signals in three tests show no significant difference in each column. It can be concluded that the repeatability of AE signal was good, furthermore in each row, when the load increases (from right to left) the amplitude of AE signal increases.

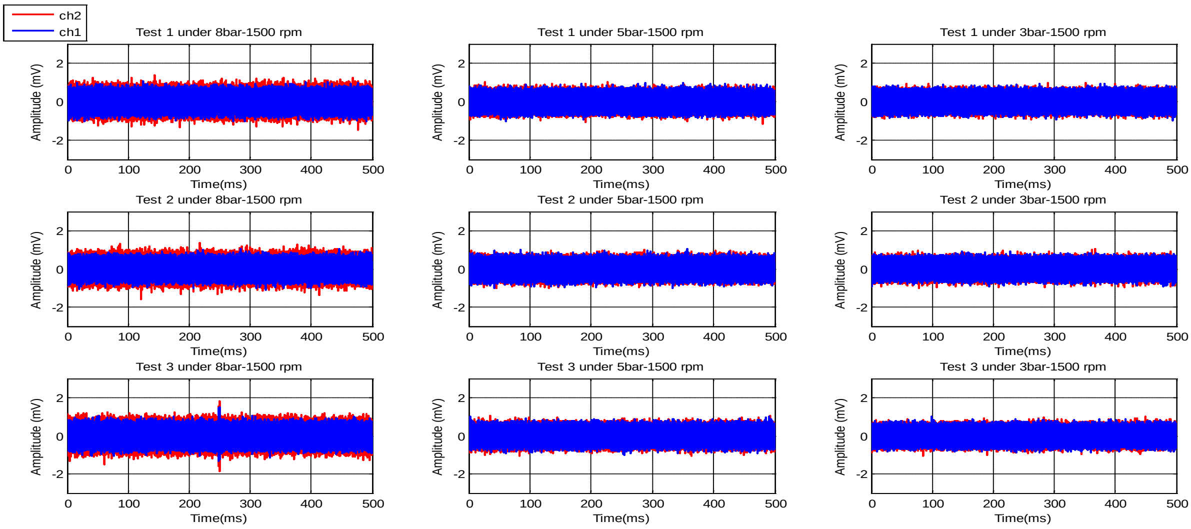

Figure 10. AE signal for NDE mechanical seal

5.4 AE spectrum

Figure11 shows the spectrum of AE row signals shown in Figure 10. As can be seen in each row, when the load increases (from right to left) the peak value also increases due to more asperity contact which clearly shows the effect of load on the frictional AE signals.

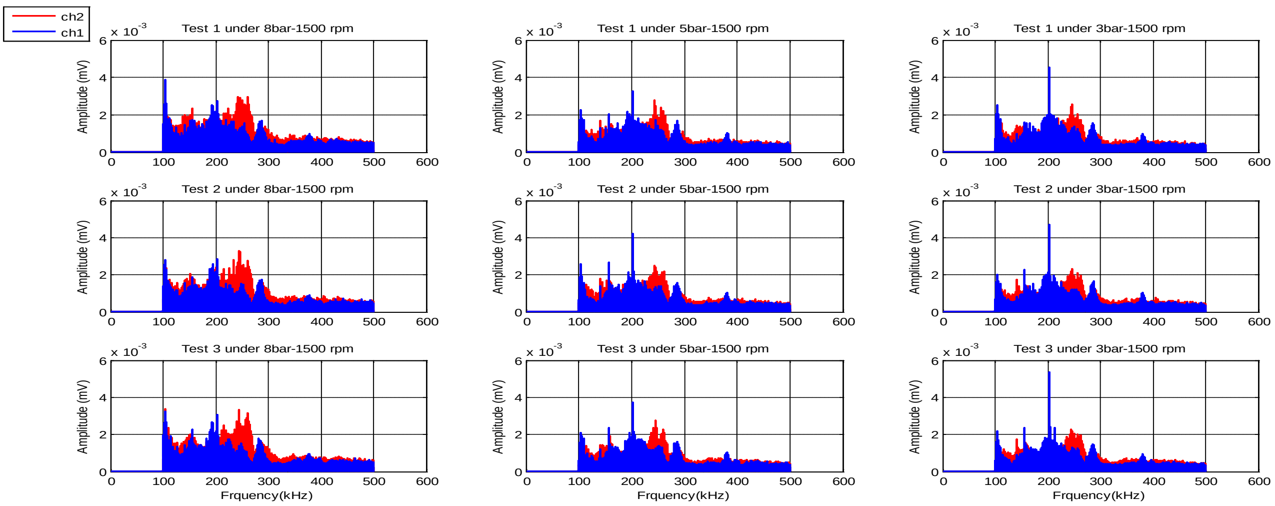

Figure 11. AE spectrum for NDE mechanical seal

5.5 Speed and Load Characteristics

The correlation between rotational speed and AE RMS values for NDE mechanical seal is presented in Figure 12. Based on the proposed model in section2, the relationship between AE RMS value and contact load is a square root trend of ($W^{1/3}$), however the AE RMS value is proportional to the square root of sliding speed. From here, it can be concluded that the contact load has less influence on AE RMS value compared with sliding speed. As can be seen in Figure 12, the AE RMS value increased slightly with load (the curves are very close to each other) and is higher for the higher load conditions. Furthermore, the AE RMS value increased incrementally with the sliding speed. As can be seen the slope of each curve is relatively high however the curves have the same trend which gives good evidence that the effect of sliding speed is more than contact load. These results are in good agreement with the proposed model.

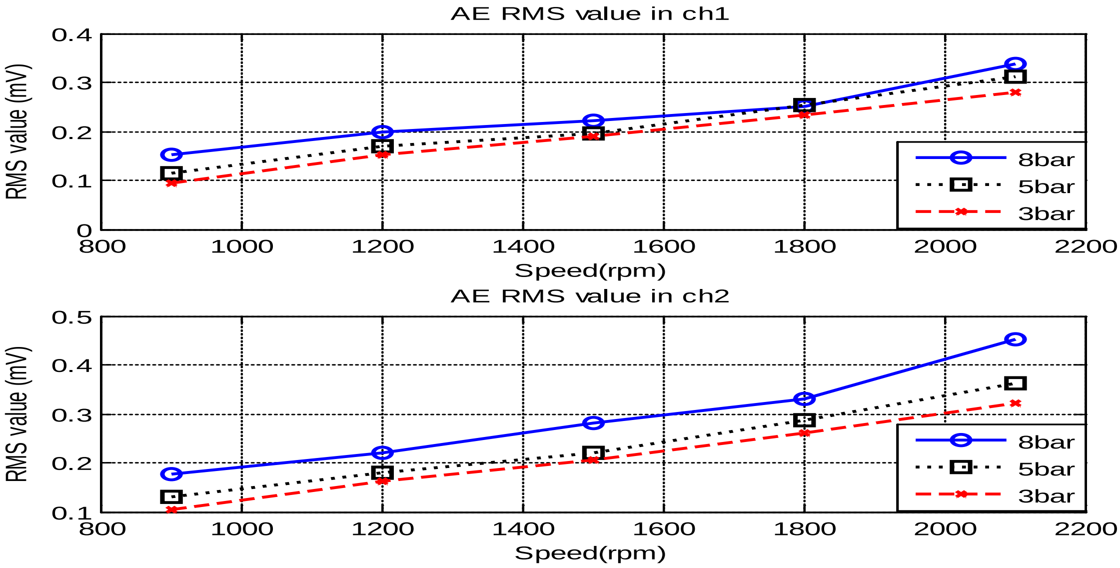

Figure 12. AE RMS value and rotational speed for NDE Mechanical seal

5.6 The variation of AE Kurtosis under different operating conditions

Figure 13 gives kurtosis of AE signals measured during TDSP experiment. It is noted that the value shown in this Figure is the difference between the kurtosis of measured AE signal and Gaussian distribution. As discussed in Section 2, the height distribution of asperities is an important property to describe the characteristic of rough surface. As it is assumed that acoustic emissions only generate at the asperity contacts, the distribution of AE signal will be an indication of distribution of asperity contacts. Hence if contact only occurs at some peak asperities, the distribution of AE signal has a kurtosis greater than 3. The more asperities contact each other, the more kurtosis of the AE signal generated is close to 3 to achieve Gaussian distribution. As can be seen in Figure13, for the experiments conducted at low speeds (900 rpm), kurtosis peaked under 5 bar and dropped to the lowest value under 3 bar. When the sealed pressure was 8 bar, kurtosis value lays in between the 5 bar and 3 bar. Since kurtosis at low speeds varied randomly with the sealed pressure, it does not support the results presented by mathematical model that the AE activity increases with number of asperity contacts. A plausible explanation is that at low speeds only the hydrostatic lubrication activated and kurtosis is not sensitive enough to the change of asperity contacts under hydrostatic lubrication conditions. However, Kurtosis of the AE signals measured at speeds higher than 1200 rpm only varied in a narrow range around zero. This is because of the hydrodynamic effect at higher speed, which helped to establish good and stable lubricant films in the range of tested pressure. These results prove the effect of number of contact asperities on distribution of AE signal as presented in mathematical model.

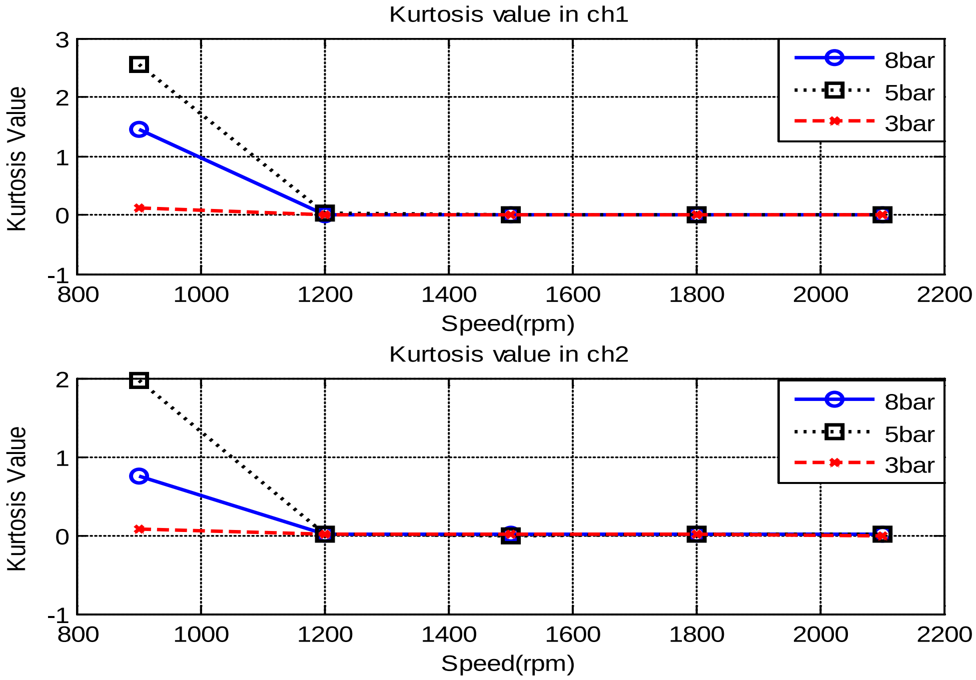

Figure 13. AE Kurtosis value and rotational speed for NDE Mechanical seal

6. Conclusion

This paper was combined with two established signal processing methods: time domain and frequency domain analysis in an effort to produce a powerful combination that in common can distinguish AE feature parameter such as RMS and kurtosis in mechanical seals. Based on obtained results AE characteristics are directly correlated to the load (sealed pressure) and rotational speed. It successfully demonstrated the AE energy level is higher for the higher load and speed conditions due to higher rate of frictional energy release from asperity contact. The variation of Kurtosis value shows the effect of number of contact asperities on distribution of AE signal. These results are in good agreement with mathematical model developed in this work. Based on the proposed model, AE RMS value increases with fluid pressure (load), rotational speed and number of asperities in contact. The results reveal the promising potential of AE technology and data analysis method applied in this work for monitoring the contact condition of mechanical seals and may help in the design and operation of seals.

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