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Identification of lubrication Regimes in Mechanical Seals using Acoustic Emission for Condition Monitoring

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Abstract

The quality of lubrication condition between seal faces directly affects the reliability, operating life and sealing performance of mechanical seals. Thus, the identification of lubrication regimes in face seals i.e. boundary lubrication (BL), mixed lubrication (ML) and hydrodynamic lubrication (HL) is of high importance for developing effective online condition monitoring approaches. This paper investigates the tribological behaviour and frictional characteristics of mechanical seals based on nonintrusive acoustic emission (AE) measurements. Mathematical models for AE generation mechanisms are derived based on the tribological behaviour and operating parameters of mechanical seals. They produce agreeable results with experimental data in explaining the types of AE signals observed in monitoring the face lubrication conditions. Frequency domain analysis of data shows that the viscous friction process generates more low frequency AE signals, whereas the asperity interactions show more high frequency AE. Moreover, the feasibility of using statistical parameters of the time domain data is shown to identify the lubrication regimes in face seals.

1. Introduction

Mechanical seal system mainly consists of two annular faces which are fitted around a rotating shaft. One face rotates with the shaft, and the other is attached to the device housing as schematically shown in Figure 1. The sealed fluid enters at sealing interface and distributes itself such that a thin layer of lubricant is formed.

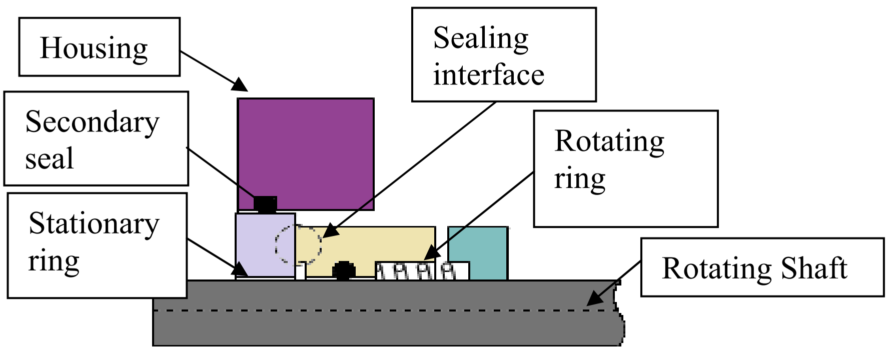

Figure 1 Schematic illustration of mechanical face seals

Frictional characteristics of sealing interface are key parameters affects the reliability, operating life and sealing performance of mechanical seals. Many researchers have investigated the frictional behaviour of seals during previous decades [1-3]. However, the history of subject dates back to beginning of 20th century, when Stribeck reported the dependency of the frictional behaviour in lubricated system as a function of velocity so-called Stribeck curve [4]. In the generalized Stribeck curve, the coefficient of friction *f* is plotted as a function of a duty parameter *G* as shown in Figure 2. Based on Stribeck curve, friction state between the rotating and stationary rings can be classified as boundary lubrication (BL), mixed lubrication (ML) and hydrodynamic lubrication (HL) depending on the operating parameters and healthy conditions.

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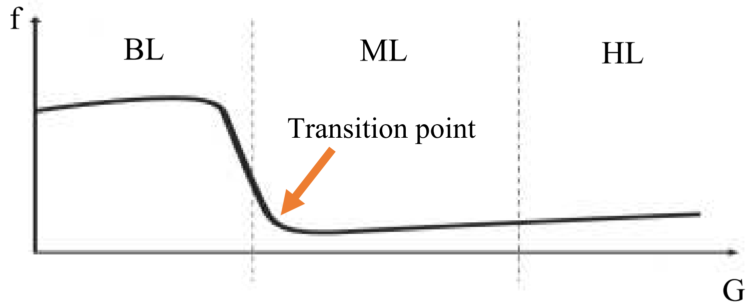


Figure 2 Generalised Stribeck curve

The dimensionless duty parameter characterizes the lubrication regime of face seals and usually is defined as Equation (1) [4-6]. Duty parameter is usually based on to judge the lubrication regime and hence conditions of mechanical seals. For instance, according to work of Wei et al. [4], Table.1 can be used to predict different lubrication condition of face seals based on the duty parameter.

$$G = \frac{\mu V_m b}{F} \tag{1}$$

where μ is the dynamic viscosity of sealing fluid; V_m is the sliding velocity on the mean face diameter; b is the width of the seal face; and F is the net closing force acting on the seal, which can be obtained by

$$F = F_{sp} + A_f(0.1B(P_f - P_a)) + 0.1P_a) \tag{2}$$

where F_{sp} is spring force; A_f is sealing interface area; P_f is sealed fluid pressure in bar; P_a is atmospheric pressure in bar and B is seal balance ratio.

Table 1. Duty parameter in different lubrication regimes [4]

Friction regime	Duty parameter
Hydrodynamic Lubrication	$G > 1\times10^{-6}$
Mixed Lubrication	$5\times10^{-8} < G < 1\times10^{-6}$
Boundary Lubrication	$2\times10^{-8} < G < 5\times10^{-8}$

The reliability of mechanical seals depends on its ability to maintain a thin fluid film in the sealing interface while simultaneously minimizing the duration and extent of mechanical contact between the mating surfaces. This is achieved when seals are working in transition point between HL to ML region which is the minimum point of Stribeck curve as indicated in Figure 2. However when the operating conditions are becoming more severe, i.e. higher loads and lower sliding speeds, the sealed fluid is not able to fully separate the surfaces and asperity contact takes place at microscopic level between the mating surfaces. The severity of contact depends on the operational conditions such as load, velocity, temperature, type of lubrication as well as the surface roughness [7]. In such a situation, the seal will not work in HL regime and ML or even BL conditions are dominant which can finally result in the failure of the seal. Thus, identification of lubrication regimes in mechanical seals is of high importance for developing effective online condition monitoring approach. Many researchers have spent a lot of time to evaluate the lubrication regime in the laboratory, but few are available in the operation due to the very small gap between seal surfaces which makes it very difficult to monitor the condition of lubrication. In 1967, the lubricant film has been measured by Astridge and Longfield [8] and Chu and Cameron [9] via the approach of capacitance and resistance. In 1984, Etsion [10] used the method of eddy current to monitor the mechanical seal film thickness. Anderson et al. [11] used ultrasonic sensors to monitor the seal face contact. A piezo-electric transducer was placed on the stationary face sent an ultrasonic pulse and received the reflection. When contact between the faces commences, the reflection coefficient drops. Later. Reddyhoff et al. [12] also estimated the film thickness based on ultrasonic measurements. Most of the mentioned approaches cannot be used in industry for reliable on-line condition monitoring of mechanical seals, because some require modifying the device structure, others have very expensive cost. This paper presents the results of an experimental study in which the mechanical seal behaviour and lubrication regimes of the faces are examined based on nonintrusive AE measurements. Time domain and frequency domain techniques are applied to measured AE signals, which reveals AE signal features can be effective diagnostic tool to differentiate the lubrication regimes in seals.

2. AE Models for Different Frictional Processes

AE is a term used for characterising transient elastic waves generated due to a rapid release of strain energy caused by the interaction of two media in relative motion. Since AE signals carry information about the details of frictional process, a significant amount of research has been carried out in this area as reported by Towsyfyan et al. [13]. In this section the sources of AE in seals is first reviewed and then AE signal characteristics and modelling is discussed.

2.1 AE sources

The amount of AE energy released during the frictional process depends primarily on the source of the generated stress energy. Based on the operational conditions, two main mechanisms contribute to AE energy in mechanical seals as schematically shown in Figure 3. In a properly maintained HL regime, the main source of AE is viscous friction due to shearing lubricant at sealing interface. These emissions are expected to result in continues type of AE waves. By reducing duty parameter to ML regime, the interaction and collision between asperities generates high frequency AE signals. Because of the fact that the roughness height is usually very small in mechanical seals, moreover, asperity peak height distribution follows the Gauss distribution, thus AE continues signal is still generated. However, when frictional contact occurs in large asperity peaks, it is expected to generate burst type AE responses due to more frictional energy released in short duration of time. A comprehensive mathematical model should include all the above discussion.

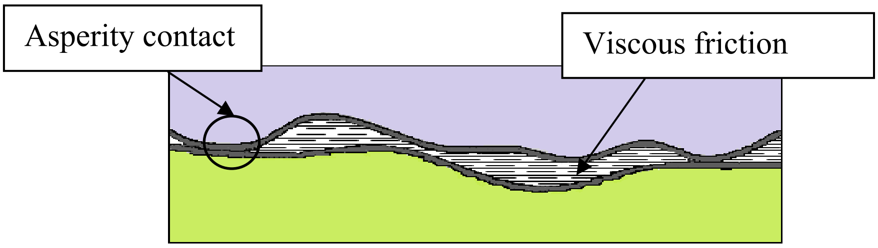


Figure 3 Schematic illustration of AE sources in the sealing interface

2.2 AE signal characteristics and modelling

In time domain techniques, statistical parameters such as mean value, RMS, skewness and kurtosis are usually used to characterize AE signals. For instance, Boness and Mcbride [14] performed a comprehensive study involved the measurement of AE signals in sliding contact and proposed an empirical model which presents RMS value of AE signal in terms of wear volume. Miettinen and Siekkinen [15] 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. [16] confirmed that RMS value of AE signal can be used to monitor seal condition. In another study, Fan et al. [17] developed a mathematical 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. To date most published work, Towsyfyan et al. [18] develops a theoretical relationship between frictional asperity contact and RMS value of AE signal to evaluate the energy of direct asperity contact under sliding friction. This model is based on the frictional work done by friction force F on a point that moves a sliding distance S in the direction of tangential asperity contact ($\int F\,ds$) and can be expressed as follow:

$$V_{rms} = \sqrt{K_I \; f \; N \; V \; W_c^{\;2/3}} \qquad (3)$$

where f is friction factor; N is number of contact asperities; V is the sliding speed; W_c is the contact load between asperities and K_I is the constant depends on material properties and surface topography of seal.

All previous models predict AE signals due to direct asperity contacts. However, based on the tribological behaviour of seals, the mating surfaces may be well separated because of the hydrodynamic effects. Thus it is necessary to calculate frictional energy resulted from the shear effect of the moving lubricant at the interface area. As it is known from fluid mechanics, tangential viscous friction acts in the sliding direction can be expressed as follow:

$$F = \iint_{dA} \tau \, dA \qquad (4)$$

where τ is viscous shear stress of liquid film and A is the sealing interface area (nominal contact area). If Z axis is aligned with the thin direction of the lubrication film, then Based on Newton's viscous flow law shear stress on the fluid element (see Figure 4) in the x or sliding direction is

$$\tau(z) = \mu \frac{\partial v}{\partial z} \tag{5}$$

where v is the velocity of the fluid along the boundary and z is the height in the thin direction of fluid. Based on pioneering work of Lebeck [19], the main component of fluid velocity in sliding direction can be expressed as:

$$v = \frac{1}{2\mu} \frac{\partial p}{\partial x} (z^2 - zh) + V \frac{z}{h} \tag{6}$$

where h is relative film thickness. Considering the fact that for such a small differential element shown in Figure 4, it is logical to assume:

$$\frac{dp}{dx} = 0$$

Substituting Equation (6) into Equation (5) will result in:

$$\tau = \mu \frac{V}{h} \tag{7}$$

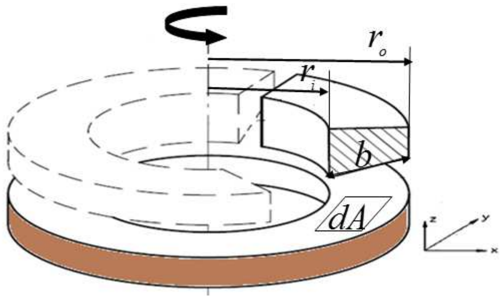


Figure 4 Differential elements in viscous friction model

The differential friction force dF can be expressed as (see Figure 4):

$$dF = \tau\, dA \tag{8}$$

Substituting Equation (7) into Equation (8) yields:

$$dF = \mu \frac{V}{h} dA \tag{9}$$

The acoustic energy release rate (power) at any arbitrary differential element of fluid can be expressed as:

$$\dot{U}_{AE} = \iint V\, dF \tag{10}$$

Substituting dF from Equations (9) into Equation (10) will result in:

$$\dot{U} = \int_{r_i}^{r_i+b} \int_{r_i}^{r_i+b} V \;\mu \frac{V}{h}\, dx\, dy = \mu \frac{V^2 b^2}{h} \tag{11}$$

As long as the asperity height distribution follows normal distribution in face seals, application of AE RMS value to characterize frictional processes is logical. Based on detailed discussion found in the work of Towsyfyan et al. [18], RMS value of the AE signal can be expressed as is:

$$Vrms = \sqrt{\dot{U}_{AE}} \tag{12}$$

Substituting Equation (11) into Equation (12) and defining all constants with K_{II}, RMS voltage of AE signal generated by viscous friction in terms of sliding speed and viscosity can be expressed as:

$$V_{rms} = V\sqrt{K_{II}\,\mu} \tag{13}$$

Thus, the comprehensive mathematical model which can be used for AE condition monitoring of mechanical seals is proposed in Table 2.

Table 2. AE RMS model in different lubrication regimes

Friction regime	AE amplitude (V_{rms})
Hydrodynamic Lubrication(HL)	$V\sqrt{K_{II}\,\mu}$
Mixed Lubrication(ML)	$\sqrt{K_I\, f\, N\, V\, W_c^{2/3}} - V\sqrt{K_{II}\,\mu}$
Boundary Lubrication(BL)	$\sqrt{K_I\, f\, N V\, W_c^{2/3}}$

It should be noted that the friction force in the ML region is the sum of the tangential viscous friction (from HL) and direct asperity contact friction (from BL). However, based on Stribeck curve, the friction goes down by increasing rotational speed under ML regime. This comes from the effect of lubricant on reducing the severity of contact among asperities which results in decreasing friction by progressing Stribeck curve under ML regime. Thus, the term $V\sqrt{K_{II}\,\mu}$ is used with negative values.

3. Test Systems and Procedures

The driving power of test rig 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 3000 rpm. A John Crane Type 1648 MP pusher cartridge seal and a stainless steel tube formed a pressurized chamber. Commercial silicon carbide (stationary ring) and antimony carbon (rotating ring) were used as seal face material. An auxiliary circulating system, was connected with the chamber to pressurize the water and take away the heat generated by the friction of mechanical seal.

Two WD S/N FQ36 AE sensors (sensor1 in radial direction and sensor 2 in axial direction) with an operating frequency range from 100 kHz to 1MHz were employed to obtain the AE signals, allowing high frequency events due to frictional process to be monitored. In this paper, only the results of sensor 1 are presented. The signal from AE sensor is amplified and acquired by a 2MHz high speed data acquisition system with 16 bit resolution.

The aim of this experimental study is to demonstrate a correlation between the AE activities with the duty parameters that refer to different regimes. It means that the seal need to be tested under a wide range of duty parameters. To achieve this, the rig was operated under 4 incremental loads of 5 bar, 6 bar, 7 bar and 8 bar at consecutive speed increments from 300 to 2100 rpm with a step of 300 rpm. According this load condition, the closing force can be calculated based on Equation (2). These operating conditions can achieve a minimum duty parameter of 4.9×10^{-9} at which it is more convinced that the seal is under the mixed lubrication or boundary conditions according to Table 1. In the meantime, the maximum duty parameter of 4.9×10^{-8} indicate that the seal may operates under hydrodynamic regimes. Thus, it allows AE signals to be examined in wide lubrication conditions and henceforth develops effective AE based diagnostics. The tests were performed at a temperature around 31.5°C by circulating cooling water over the system.

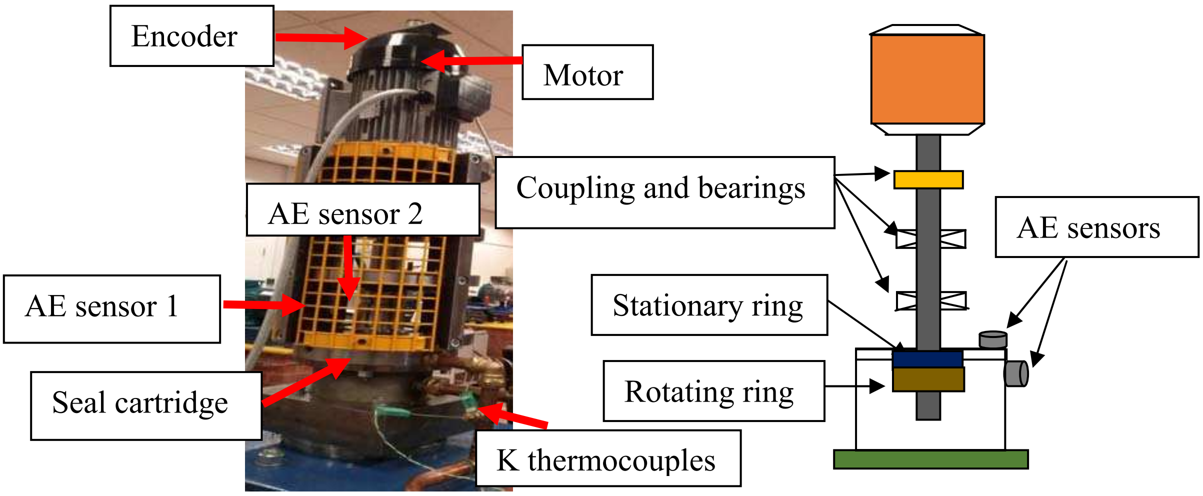


Figure 5 Layout of the test rig (left) and schematic illustration (right)

4. Results and Discussion

4.1 AE signal characteristics in the time domain

To gain understanding of AE responses to friction processes, Figure 6 presents typical AE signals under different speed and pressures settings. Figure 6 (a) and (b) shows that AE amplitudes decrease with speed increase, indicating that asperity reactions are less with increasing in speed and hence showing that the lubrication condition may moves from ML to HL. As the speed increases continuously, shown in Figure 6 (c)–(e), the amplitudes become larger gradually. This indicates more effect of viscous friction and hence shows that the lubrication is in HL regimes. Therefore, this speed dependency allows these two different regimes to be indicated.
Moreover, the comparison of AE responses between low and high operating pressures shows that a clearer difference can be seen at the low speed whereas it is insignificant at the high speeds. This indicates that more asperity interaction due to load happen at the low speed, which is consistent with that predicated using Equation (3). On the other hand at high speeds the fully developed HL helps to reduce or eliminate the effect of asperity interactions. Therefore, it has been demonstrated further that the AE responses will be dominated by the viscous friction i.e. speed but not the load as shown in Equation (13).
In addition, the AE signals from ML to HL regime contains less burst responses due to decreasing asperity contact in large peaks.

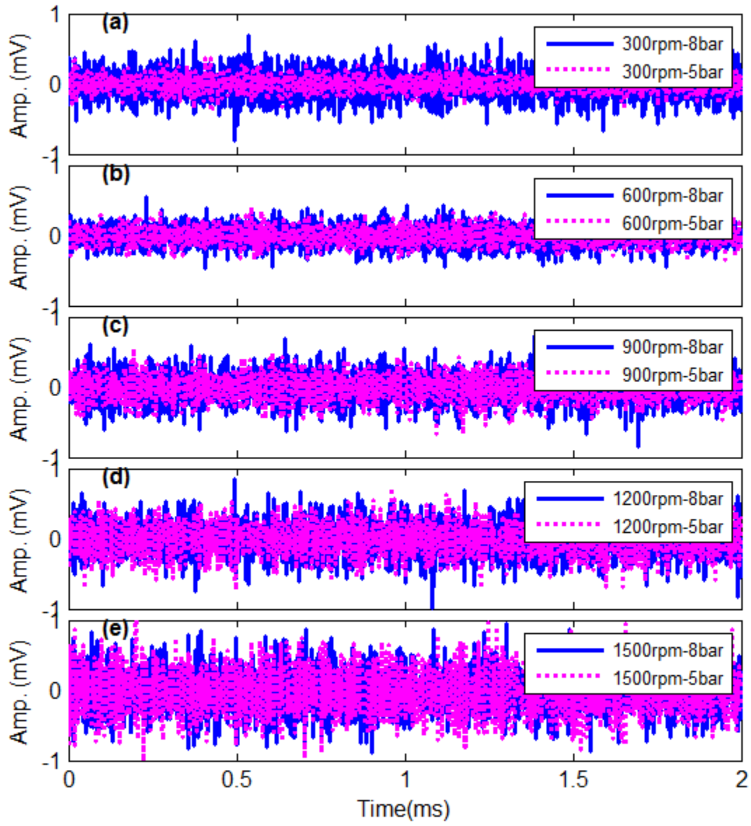


Figure 6 AE signals for different loads and speeds

4.2 AE RMS value

To examine the connection between AE and lubrication conditions, RMS values of the AE signals are calculated and presented versus the duty parameter as shown in Figure 7. For each curve, load is constant and speed increases incrementally from 300 to 2100 rpm with the step size of 300 rpm. As can be seen, the minimum point of Stribeck curve fluctuates in a narrow transition region between 1.00×10^{-8} and 1.5×10^{-8}, however this transition range is not consistent with Table 2. It is the common characteristic that transition occurs at different *G* values as reported in the literature [6, 20].

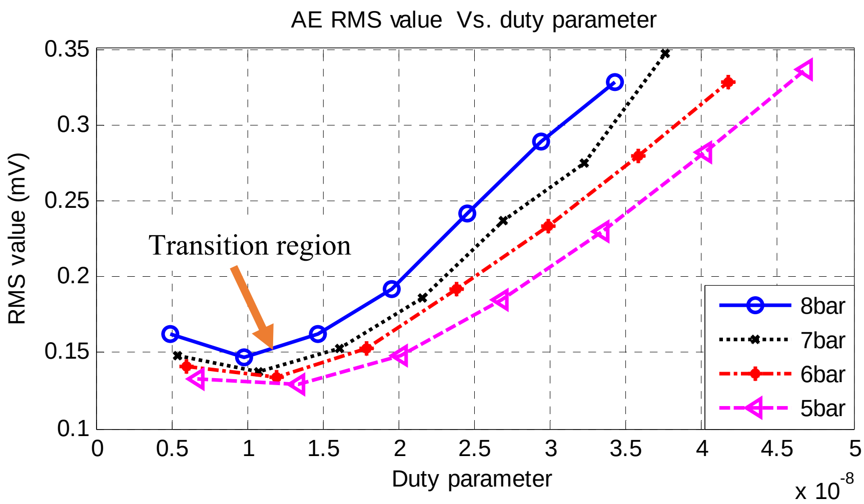


Figure 7 RMS vs. duty parameter

4.3 AE kurtosis value

To investigate the distribution of AE signals, kurtosis values are calculated and presented versus the duty parameter as shown in Figure 8. It shows that in the left hand side of transition region the kurtosis value is relatively high, showing that frictional contact occurs not only in the asperities with normal height distribution, but also in large asperities which their height is significantly larger than average height of asperities (right hand tail of normal distribution). By increasing the speed, the faces become fully separated due to hydrodynamic effect and viscous friction at sealing interface generates more continues AE waves. This is why Kurtosis changes in a narrow range around zero under the HL regime.

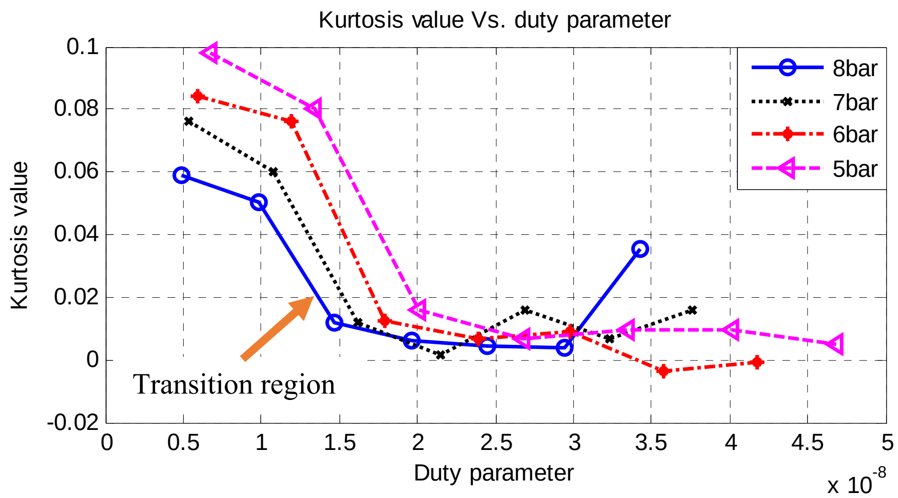


Figure 8 Kurtosis vs. duty parameter

4.4 AE characteristics in the frequency domain

Figure 9 shows the spectra of AE signal when the seal operates under different speed and pressures. Figure 9 (a) and (b) show that amplitudes of AE spectra decrease first with speed increase. Then, as the speed increases continuously, shown in Figure 6 (c)–(e), the amplitudes become larger gradually. This is consistent with the AE signal characteristics in time domain. On the other hand, by increasing the speed, the difference between the amplitude of AE spectra related to low and high load decreases gradually and finally in Figure 9 (e), changing the load has no significant effect on it. This indicates that when hydrodynamic lubrication is well established, the AE responses are more dominated by the viscous friction due to change in speed but not the load as shown in Equation (13). In addition, this effect is clearer for high frequency signals (between 50 kHz to 300 kHz) and changing the load has no significant effect on the amplitude of low frequencies (under 50 kHz). It may prove that the AE signals generated by interaction of asperities are mainly located in high frequency band whereas low frequency events are more affected by viscous friction i.e. speed.

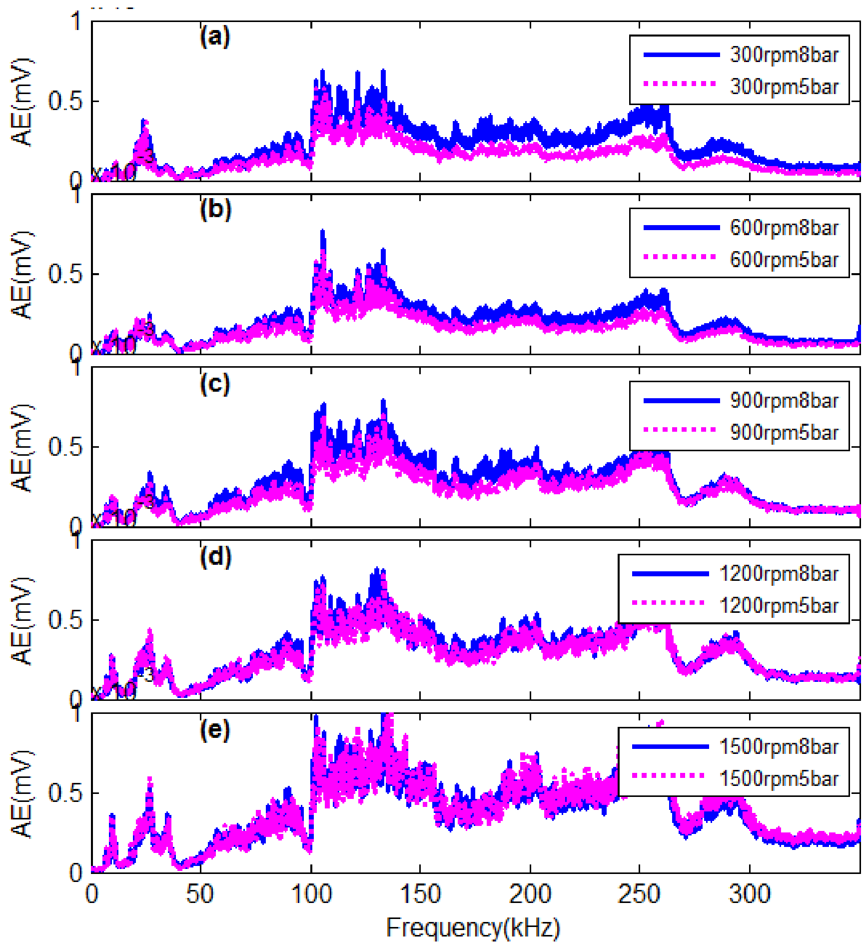
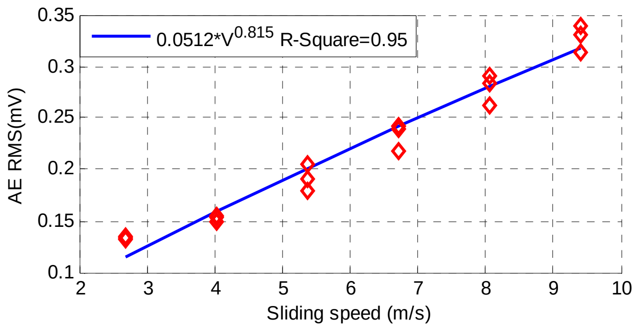


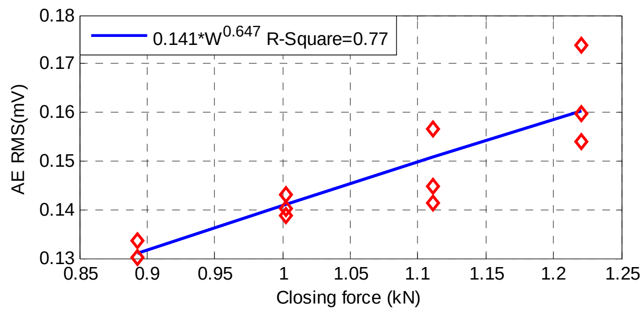
Figure 9 AE spectra under different loads and speeds

4.5 Model verification

Figure 10 (a) shows the relationship between RMS value and the average sliding speed when the rig operated under load 6 bar at six different speeds from 600 to 2100 rpm with the step size of 300 rpm to simulate HL regime, three different tests have been carried out. The average speed is calculated using the average of the outer and inner diameters of sealing interface. Based on Figure 10 (a), the power trend predicted for speed by Equation (13) is achieved approximately and may prove that in the HL regime, RMS value changes linearly with sliding speed. Figure 10 (b) presents the test results to investigate the effect of load on AE RMS value when the seal operated at speed 300 rpm under 4 different load, from 5 bar to 8 bar with the step size of 1bar, to simulate ML regime (three different tests). It is assumed that contact load is linearly proportional to closing force, however in reality contact load is much smaller than closing force. As it is illustrated in Figure 10 (a), the power trend predicted for contact load by Equation (3) is not achieved. This might be because of the influence of contact load on the total number of asperity contact which is not taken into account. Furthermore, it should be noted that the Equations (3) and (13) are derived for pure sliding asperity contact and pure viscous friction respectively, however, under the condition of the tests it can be hardly achieved. It seems that the effect of load on the frictional work [18] and elastic energy released [17] during asperity contact should be more investigated.



(a) RMS versus speed



(b) RMS versus closing force at low speed

Figure 10 Model verification

5. Conclusion

This paper investigates frictional characteristics of mechanical seals using AE measurements. To demonstrate the AE generation mechanism and feasibility of identifying lubrication regimes, an experimental study was carried out based on duty parameter values. The analysis of results produces the following key points:

1. AE can be generated by two main mechanisms i.e. direct asperity interactions and viscous friction of lubricant films.
2. The AE responses correspond to these two types mechanisms with the short duration bursts and continuous waves. In the ML region, the bursts are more dominant due to contact of large asperity peaks. However in the HL regime, the strong shearing occurs in between fluid layers and lead to more contents of continuous AE signals.
3. The combination of RMS and kurtosis values of AE signals gives good indication of ML and HL regimes.
4. Based on the AE model developed in this work, AE RMS value changes linearly with sliding speed in the HL regime, while it is proportional to the cubic root of contact load in ML regime. However, to achieve a solid conclusion, the effect of load during asperity contact on the frictional work and elastic energy released rate should be more investigated in future works.

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