International Journal of Architectural, Civil and Construction Sciences
ISSN: 2415-1734
Vol:13, No:2, 2019

The Lubrication Regimes Recognition of a Pressure-Fed Journal Bearing by Time and Frequency Domain Analysis of Acoustic Emission Signals

S. Hosseini, M. Ahmadi Najafabadi, M. Akhlaghi

Abstract**—**The health of the journal bearings is very important in preventing unforeseen breakdowns in rotary machines, and poor lubrication is one of the most important factors for producing the bearing failures. Hydrodynamic lubrication (HL), mixed lubrication (ML), and boundary lubrication (BL) are three regimes of a journal bearing lubrication. This paper uses acoustic emission (AE) measurement technique to correlate features of the AE signals to the three lubrication regimes. The transitions from HL to ML based on operating factors such as rotating speed, load, inlet oil pressure by time domain and time-frequency domain signal analysis techniques are detected, and then metal-to-metal contacts between sliding surfaces of the journal and bearing are identified. It is found that there is a significant difference between theoretical and experimental operating values that are obtained for defining the lubrication regions.

Keywords**—**Acoustic emission technique, pressure fed journal bearing, time and frequency signal analysis, metal-to-metal contact.

I. Introduction

JOURNAL bearings have an important role in various types of rotary machines in different industries and the main benefit of them is the lack of contact between sliding surfaces of the journal and the bearing inner surface and thus, they are long life bearing. But, in severe working conditions of the journal bearings such as low rotating speed and high load applications, the oil film between sliding surfaces may not completely separate them and so metal-to-metal contact occurs. Poor lubrication is the main cause of the journal bearings failures and these kinds of failures generally result when the oil film thickness is too low to prevent the contact of micro asperities of the sliding surfaces. HL, ML, and BL are the three different lubrication regimes that journal bearings can operate in any of them. Thickness of the oil film defines the lubrication situation. When a journal bearing operates under the BL, the sliding surfaces of the bearing and the shaft are practically in direct contact and friction is at its highest level. Lower friction levels occur with the ML, where the sliding surfaces are partially separated by the lubricant and the HL, where the sliding surfaces are completely separated by the lubricant [1]. The Stribeck curve and the design charts of Raimondi and Boyd are useful to determine lubrication situation in the journal bearings. Fig. 1 shows the Stribeck curve [2] that is the variation of friction in relation to duty parameter in the three lubrication regimes. The duty parameter is combination of rotating speed with oil viscosity per apply load. In accordance with the Stribeck curve, by reducing the rotating speed while the other parameters are constant, the eccentricity of the bearing increases and the minimum thickness of the oil film decreases.

Raimondi and Boyd [3] obtained numerical solutions of Reynolds' equation for journal bearings that were presented in the form of charts and tables. The performance parameters of the journal bearings such as minimum oil film thickness and friction coefficient can be obtained from these charts. Real-time monitoring of the lubrication condition of the journal bearings is recognized as a difficult problem. AE is an advanced nondestructive testing technique for real-time condition monitoring and early fault detection of rotating machinery [4] that accompanies almost all physical phenomena in solids and their surfaces [5]. AE is described as propagation of mechanical elastic waves in a structure that produced by an AE source.

In HL regime, the source of AE can be attributed to the friction within the fluid and the friction reaction of the fluid on the surfaces [6]. This means that, there will always be a thin film of oil between the two surfaces in order to prevent to contact asperities of two surfaces. In working conditions of the journal bearings because of some reasons such as sudden force, vibration, oil starvation, non-eccentricity, and excessive height of the surfaces asperities, there is a rupture probability in the oil layer and consequently metal-to-metal contact. Thus, at each revolution of the shaft, excessive surface asperities of the shaft and the bearing that enter into the zone of the minimum oil film, are subject to stochastic pulse loading due to the accidental formation and rupture of micro contacts. Metallic contact is more probable by deteriorating the lubrication conditions and reducing the thickness of the oil layer.

AE signals can be characterized by various features in a broad frequency range from 20 kHz to several MHz's such as amplitude, energy, and count rate that each of them contains some information about the understudied case. Yoon et al. [7] applied AE monitoring to detect incipient failures of the journal bearings caused by several types of abnormal working

Dr. Sadegh Hosseini is with the Department of Mechanical Engineering, Faculty of Engineering, Islamic Azad University (IAU), Varamin, Iran (corresponding author, phone: +989125255970; e-mail: s.hoseini.s@aut.ac.ir).
Dr. Mehdi Ahmadi Najafabadi is with Department of Mechanical Engineering, Amirkabir University of Technology, Tehran, Iran (e-mail: ahmadin@aut.ac.ir).
Prof. Mehdi Akhlaghi was with the Department of Mechanical Engineering, Amirkabir University of Technology, Tehran, Iran (e-mail: mehdiakhlaghi@hotmail.com).

condition such as hard particles in the lubrication layer, insufficient lubrication, and metallic contact. Their results showed that AE is an effective tool to detect the incipient failure in journal bearings. Mba [8] detected scratches of a size about 100 µm on a roller bearing using AE method and concluded that this method can monitor the rate of degradation on the bearings. Miettinen and Andersson [9] used the AE method for monitoring the lubrication condition in grease-lubricated rolling bearing. The aim of their work was to clarify how the contaminants in the grease affect the generated AE. Their results showed that even the lowest contaminant particle concentrations were sufficient to induce a clear increase in the AE count level in the test bearing and the hardness of the contaminants were identified by the AE characteristics. Jamaludin et al. [10] studied high frequency stress waves for assessing the lubrication condition within a low-speed rotating bearing. They demonstrated the ability of this technique to assess the amount of grease within the bearing housing, which significantly depended on the lubrication condition in the test bearing. Thus, the properly and poorly lubricated bearings were recognized. Mirhadizadeh et al. [6] investigated the influence of operating variables (speed, load, etc.) of a journal bearing on the generation of AE signals. They proposed in a properly maintained HL regime, the principal source of the AE was the friction in the shearing of the lubricant in addition to the splashing and motion of the oil within the bearing and the power losses of the bearing were directly correlated with the AE levels. In another study, Mirhadizadeh and Mba [11] focused on correlating AE activity to the operating condition on a hydrodynamic bearing. Their results showed that for several particular speed conditions, applied loads had a negligible influence on the level of AE amplitude. However, AE levels significantly were changed by variation of the rotational speeds even though the actual predicted change in the minimum film thickness with increasing speed was less than 1% due to the test conditions. Couturier and Mba [12] monitored bearings under variable speed and load conditions by AE technology. Thus, AE activity was related to the theoretically predicted specific lubricant film thickness under an elastohydrodynamic lubrication regime. Douglas et al. [13] used AE method to provide information pertaining to the interaction between piston rings and cylinder liners, such as asperity contact, lubricant flow and/or blowby, in a range of diesel engines. Analysis of the AE signals in mentioned work generally involved comparisons of the AE energy within windowed sections of the engine cycle. The influences of various factors such as engine speed, load, and lubricant conditions on the average Root Mean Square (RMS) of AE energy were considered. They suggested AE monitoring could be an effective tool for the in situ evaluation of component condition, wear rates, and lubricant performance by further development. Niknam et al. [14] distinguished between lubricated and dry roller bearings under similar working conditions by AE signal parameters. Various levels of speeds and radial loads were implemented for the tests and in each test, several time domain AE parameters were calculated. The lubrication situation of the test bearings were distinguished by the AE parameters. Towsyfyan et al. [15] investigated the AE features of self-aligning journal bearings under different rotational speed, radial load, and lubrication condition. Time domain and frequency domain signal analysis techniques were used and it was concluded that the AE energy levels in high frequency range were higher for the higher radial load and speed condition. For different lubricant cases, AE energy was became high when the viscosity was lower. The research showed that AE can be used to detect lubricant degradation in the journal bearings. Cai et al. [16] examined freight car's journal bearing inner ring looseness by wavelet packet analyzing of the AE signals and a combination of the AE features was used to detect the faults. They extracted the ration between high frequency's internal and total energy as the characteristic parameter that could recognize the seeded faults.

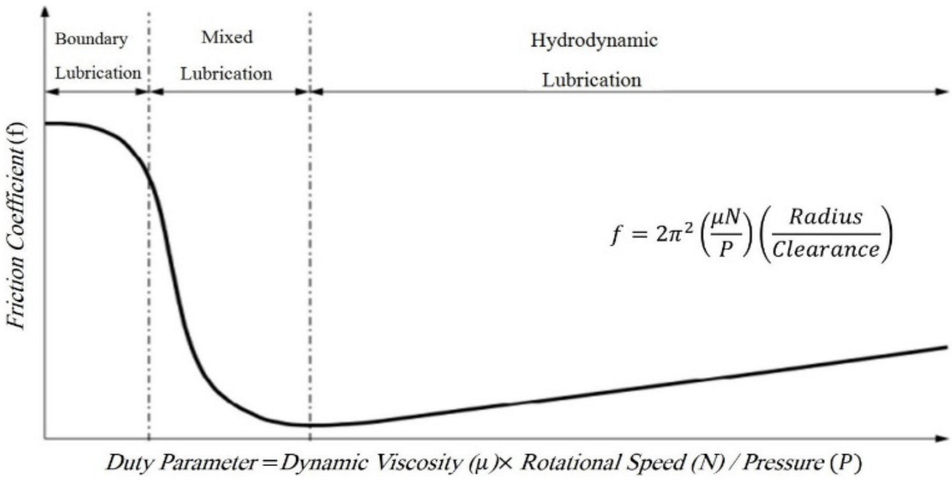

Fig. 1 The Stribeck curve for a sliding bearing [2]

In addition to the parameters mentioned to define the lubrication regimes in a journal bearing by Stribeck curve and Raimondi and Boyd charts, additional factors such as lubricant quality and contaminants, vibration, misalignment, roughness of the sliding surfaces, start/stop system should be considered when assessing the lubrication condition of the journal bearings in critical assets such as large turbines. Due to the large number of variables and factors that have some effects on the lubrication conditions of the journal bearings, achieving theoretical relations to determine the lubrication regimes based on various operating parameters is difficult. The objective of the present research work is to define lubrication regime transitions in a pressure-fed journal bearing in various working conditions using AE method based on time domain and time-frequency domain signal analysis techniques.

This paper is organized as follows: The CWT is introduced in Section II. Section III describes experimental procedure and the set up that was used for the investigation. Finally, Section IV provides a discussion of the results and benefits of using AE signals features to monitor lubrication regimes of the journal bearings, and draws conclusions.

II. Continuous Wavelet Transform

The time domain signal analysis methods are statistical characteristics of a signal such as mean value, standard deviation, RMS, energy, and count rate. The frequency domain is another explanation of a signal and these features are based on the Fourier transform of a signal such as power spectral density and frequency spectrum. These signal-processing techniques are not suitable for non-transient signal analysis. Recent studies have focused on time-frequency domain techniques such as Wavelet Transform (WT), which are suitable for identifying and analyzing machine failures that are transient in nature [17]. WT uses inner products to assess the similarity between a signal and a wavelet function or a mother wavelet. CWT is a correlation between the signal $x(t)$ and the conjugate $\bar{\psi}(t)$ of the chosen mother wavelet $\psi(t)$ [18]:

$$CWT(a,b) = \frac{1}{\sqrt{|a|}}\int_{-\infty}^{\infty} x(t)\bar{\psi}\left(\frac{(t-b)}{a}\right)dt \quad (1)$$

Magnitude of the CWT coefficients is directly dependent upon the correlation between the signal $x(t)$ and the mother wavelet $\psi(t)$ [19].

III. Experimental Setup

Several pressure-fed journal bearings of the same type but with different clearances were used for the tests under specified working conditions. The bearings were made of phosphor bronze and had radial oil groove. The inner diameter of 35 mm, width of 63 mm, surface roughness (Ra value) of 1.5 µm, axial groove length of 6.3 mm, circumferential groove extent of 5 mm and oil hole diameter of 5 mm were the bearing dimensions and shaft diameter was 35 mm that was made of hardened steel. According to the design guidelines for the journal bearings [20], the sizes of the test bearings were chosen. Fig. 2 shows a schematic of the testing apparatus. At both ends of the shaft, two self-adjustment ball bearings were used to support test set up and plastic strips with a thickness of 10 mm between the shaft and the ball bearings were implemented to reduce noise. A pneumatic system was used to apply loads to the test bearings and a 1.5 kW induction motor, giving an output of 0-2500 rpm that was controlled by a variable speed drive. The bearings operating parameters were controlled throughout the procedure of the tests using different sensors (Table I). A contact tachometer to measure the shaft rotational speed, an oil pressure gauge to measure the pressure of inlet oil, a load cell to measure applied loads and three temperature sensors to measure the temperature of the bearing body, inlet oil, and outlet oil were used. In order to record AE events, AE software AEWin and a data acquisition system (PAC) with a maximum sampling rate of 40 MHz were implemented. Two broadband, resonant-type, single-crystal piezoelectric transducers from Physical Acoustics Corporation (PAC) were used as the AE sensors.

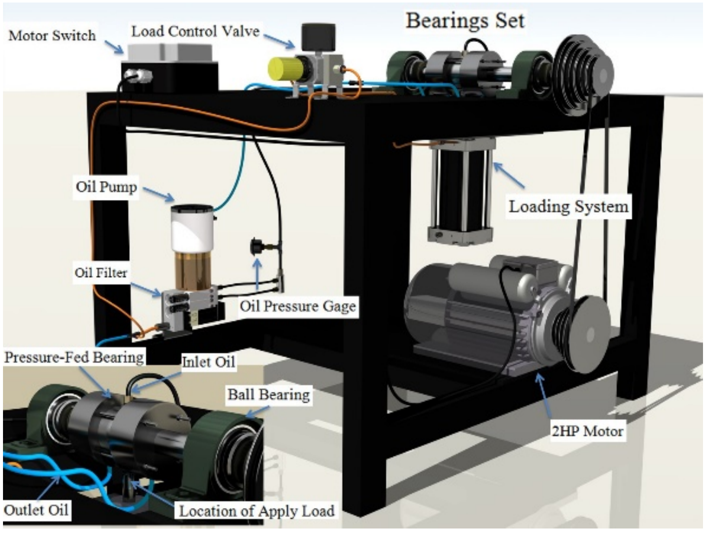

Fig. 2 Schematic representation of the bearing test set

TABLE I
THE SENSORS SPECIFICATIONS USED TO REGISTER OPERATING CONDITIONS OF THE TEST BEARINGS

Type of sensor	Measurement range	Accuracy
Temperature sensor	0–140℃	1℃
Oil pressure gauge	0-25 bar	1%
Tachometer	0-20000 rpm	0.1% + 1 digit
Load cell	0-5 KN	0.5%

In each of the test conditions, the applied loads were constant and due to low fluctuations in temperature of the lubricant during the tests, the oil viscosity was assumed constant. Thus, the three lubrication regimes were distinguished based on the rotational speed variation. The AE data with sample rate of 1 MHz were registered after the outlet oil temperature had reached constant value. AE sensors were connected to 40 dB pre-amplifier and were mounted on the body of the bearing. One of the AE sensors was located on the outer shell of the bearing and the other was located in the bearing housing on the lateral surface of the bearing (Fig. 3). Based on fundamentals of lubrication theory and the Raimondi and Boyd charts, the minimum friction coefficients, the minimum oil film thicknesses, and the friction torques for different operating conditions were obtained (Table II). Then, the theoretical values of duty parameters that were defined the three lubrication regimes were estimated using the obtained performance parameters and the Stribeck curve (Table III).

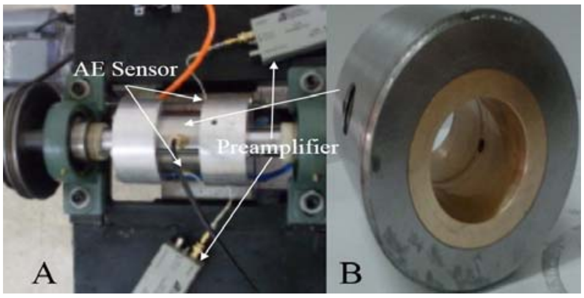

Fig. 3 A- Location of the AE sensors. B- The pressure-fed journal bearing

TABLE II
PERFORMANCE FACTORS OF THE TEST BEARINGS

Inlet Oil Pressure (bar)		Rotational Speed (rpm)		Apply Load (kN)		Oil Temperature (°C)	
Min	Max	Min	Max	Min	Max	Min	Max
0.5	2.5	10	1700	0.78	3.92	48	56
Oil Viscosity (Pa.s)		Sommerfeld Number		Minimum Oil thickness (μm)		Friction factor	
Min	Max	Min	Max	Min	Max	Min	Max
0.0835	0.0540	5.5462e-04	3.6473	0.8	70	8.9985e-04	0.1485

TABLE III
DUTY PARAMETERS FOR THE THREE LUBRICATION REGIMES

Lubrication Regime	Duty parameter ranges	Duty Parameters: Viscosity (μ) in 55 °C (Pa.s)	Speed (N) (rpm)	Pressure (P) (MPa)
Hydrodynamic	$\frac{\mu N}{P} > 4.35$	0.0568	N > 40	0.641
			N > 100	1.282
			N > 200	2.564
			N > 300	3.206
Mixed	$0.87 < \frac{\mu N}{P} < 4.35$	0.0568	N < 40	0.641
			10 < N < 100	1.282
			40 < N < 200	2.564
			40 < N < 300	3.206
Boundary	$\frac{\mu N}{P} < 0.87$	0.0568	-	0.641
			N =10	1.282
			N < 40	2.564
			N < 40	3.206

IV. RESULT AND DISCUSSION

Registered AE signals were processed using time domain features and among them, energy and count were considered as the more sensitive features. Fig. 4 shows AE energy versus time during the tests for the two bearings under three different loads. The rotational speeds were gradually decreased from 1700 rpm to 10 rpm in 16 steps. For each step of the rotational speeds, the RMS of the AE energy were calculated in order to determine whether the energy was changed in the testing process and with red dash lines are shown in the Fig. 4. It was remarkable that in all the tests, the curve of the RMS of AE energy and the Stribeck curve had the same behavior at the three lubrication regimes. Therefore, the friction level of the lubrication regimes was directly proportional to the energy of the AE signals that were generated by various operating conditions of the test bearings. Rotational speeds with the lowest magnitude of AE energy that are defined in Fig. 4 were where the minimum value of the frictional wear between the journal and bearing surfaces occurred. Therefore, the rotational speed with the minimum AE energy is the boundary between the HL and ML regimes. In the HL regime, the loss of the frictional energy, which was due to rotational motion of the oil film, was the main source of the AE generation, and in ML and BL regimes, friction and wear due to the cyclic contact of the excessive asperities between sliding surfaces were the main source of the AE generation. In other words, the AE energy reduction in the HL regime occurred mainly by decreasing the sliding velocity and the increase of AE energy in ML and BL regimes was due to increase of metallic contact which was caused by the oil film thickness reduction. The rotational speeds that were defined the boundary between HL regime and ML regime had higher values rather the values were expected by the theoretical lubrication relations that are brought in Table III. The theoretical value of the duty parameter, which was predicted the boundary between HL regime and ML regime was 4.35. While the minimum experimental value of the duty parameter for the bearing with 0.07 mm clearance was 7, for the bearing with 0.1 mm clearance was 8, and for the bearing under poor lubrication with 0.085 mm clearance it was 34. Significant difference between the experimental values and the theoretical value of the duty parameter were due to the large number of working variables that were caused unpredictable effects on the lubrication regime transitions. In other words, the lack of consideration of all the operating factors had effect on the accuracy of the lubrication models.

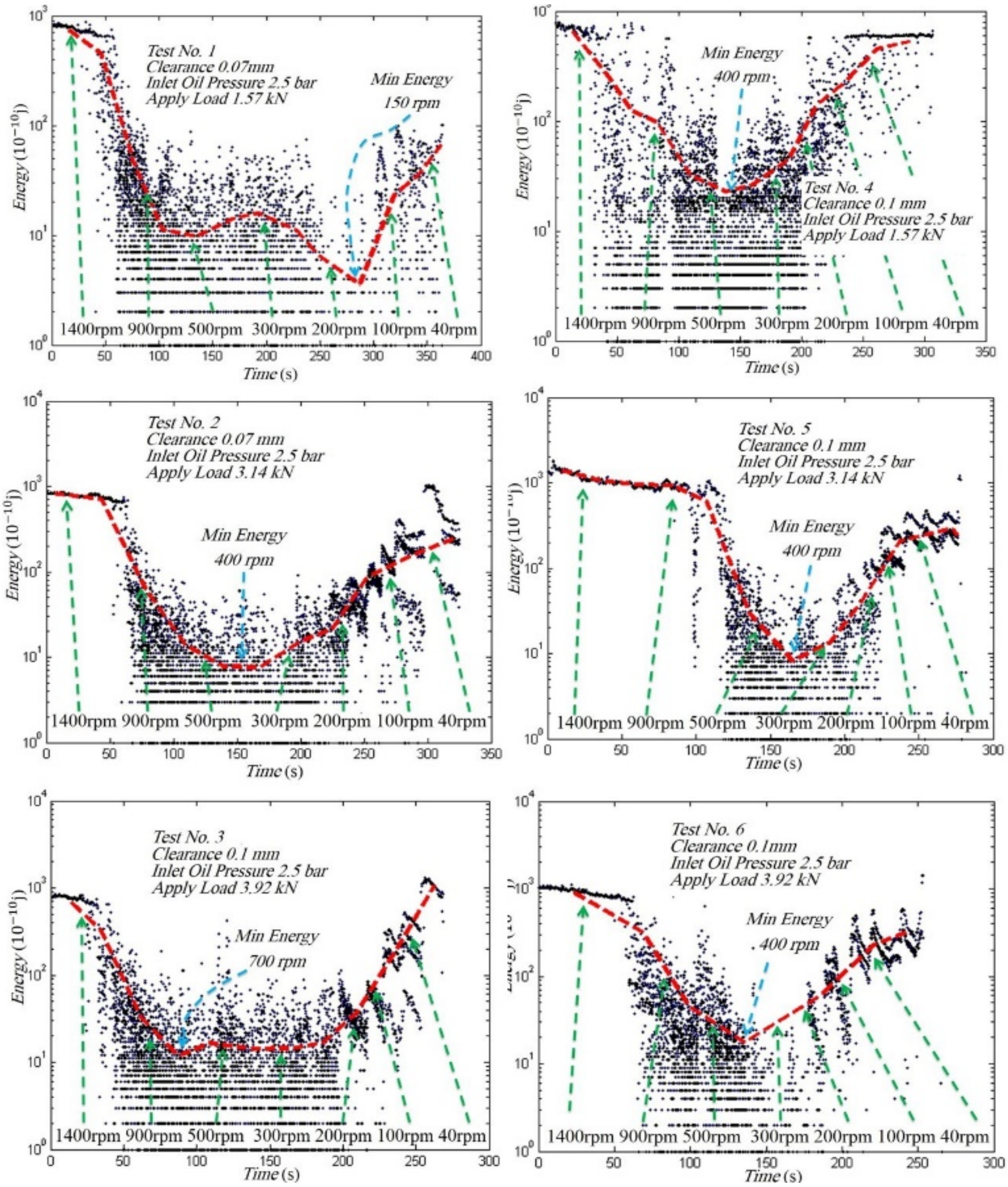

Fig. 4 The AE energy vs. working conditions of the test bearings

As shown in Fig. 4, in order to assess the effect of clearance size of a journal bearing on the lubrication condition, two bearings with minimum and maximum clearance sizes based on design parameters were tested. In the bearing with the minimum clearance (0.07 mm), the apply loads had significant effect on the lubrication regimes transition and by increasing the apply load, the rotational speeds which defined the boundary between HL and ML regimes, were increased. The bearing with the maximum clearance (0.1 mm) had negligible effect on the lubrication regimes and at the almost the same rotational speed, the minimum AE energy was registered. The same test conditions on a bearing with mediocre clearance (0.085 mm) and very low inlet oil pressure were performed (Fig. 5). Due to the low inlet oil pressure, except in the tests that had the low load magnitude (test No. 7), in the high magnitude applied loads, the lubrication regimes were not distinguishable from each other and ML and BL regimes were the dominant lubrication regimes (test No. 8). The clearance size and inlet oil pressure had observable effects on the lubrication conditions, which these factors were not considered in the lubrication theories.

The time domain features presented only some part of the information that was contained in the AE signals. In order to further evaluation of the AE signals, CWT was used and this signal analysis technique showed how the AE signal's energy were distributed over specified range of frequencies and time samples. According to the sample rate of 1 MHz, the frequency range of the AE signal was extended from 50 kHz to 500 kHz, where normal operating vibration of the test bench equipment had no influence in the registered signals. The CWT coefficients clearly depend on the mother wavelet and they will have large magnitudes while there is a good match between a mother wavelet and a signal. The magnitude of CWT coefficient reflects the signal energy intensity in the both time and frequency domains simultaneously.

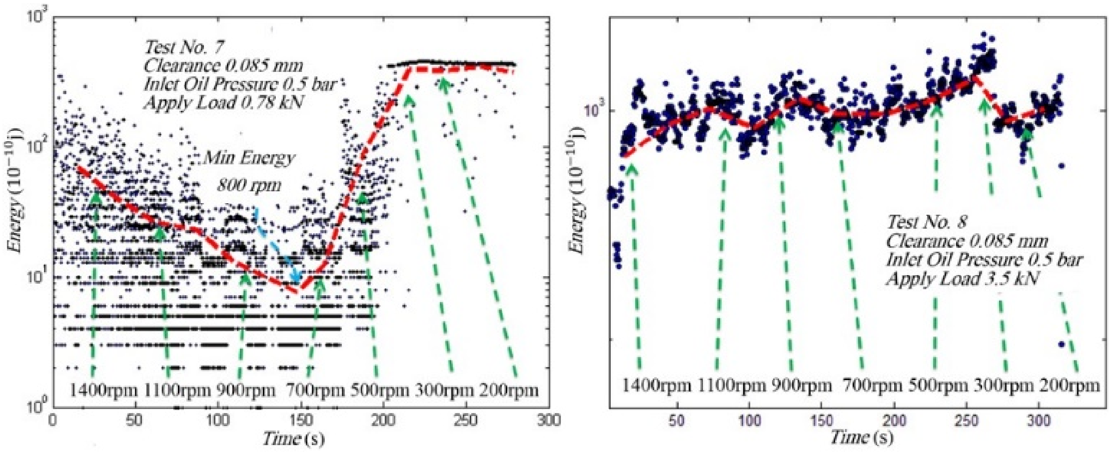

Fig. 5 The AE energy vs. working conditions for the bearing with low inlet oil pressure

The time domain features presented only some part of the information that was contained in the AE signals. In order to further evaluation of the AE signals, CWT was used and this signal analysis technique showed how the AE signal's energy were distributed over specified range of frequencies and time samples. According to the sample rate of 1 MHz, the frequency range of the AE signal was extended from 50 kHz to 500 kHz, where normal operating vibration of the test bench equipment had no influence in the registered signals. The CWT coefficients clearly depend on the mother wavelet and they will have large magnitudes while there is a good match between a mother wavelet and a signal. The magnitude of CWT coefficient reflects the signal energy intensity in the both time and frequency domains simultaneously. For the registered AE signals and among the common mother wavelets, Daubechies 3 wavelet was chosen as the mother wavelet based on maximum matching mechanism [21]. Due to the large number of registered waveforms during the tests, for each of the lubrication regimes, 500 sequential waveforms were selected to calculate the CWT of them. The results of the CWT of the AE waveforms for the bearing with 0.07 mm clearance under mentioned working conditions is represented in Fig. 6. The trend of the CWT coefficients was the same as the results obtained for the RMS of the AE signals and the coefficients magnitude had shown the progress of the Stribeck curve for the three lubrication regimes in the frequency range between approximately 50 kHz and 200 kHz. In addition, in the lubrication regimes that had the possibility of metallic contacts (ML and BL regimes), the increase of the CWT coefficients between 200 kHz and 500 kHz occurred, and they had larger magnitude of coefficients or AE energy intensities in less oil film thicknesses. It can be concluded from the CWT results that metal-to-metal contacts generated AE signals contained in the frequency band from 200 kHz to 500 kHz. The HL regime and BL regime had almost the same level of AE energy in almost all of the tests, while the frequency characteristics of them were clearly different. Due to increase of the metal-to-metal contacts rate between the shaft and the bearing in the BL regime, the CWT coefficients were significantly increased. For the load of 3.92 kN in the similar bearing, CWT on the AE waveform is shown in Fig. 7. As can be seen, the magnitudes of the CWT coefficients for the two tests in the frequency between 50 kHz and 200 kHz were almost equal, whereas these magnitudes in the frequency between 200 kHz and 500 kHz were completely different. The increase of the load was caused more increase in the CWT coefficients values in the frequency range between 200 kHz and 500 kHz, which was due to further and more severe metal-to-metal contacts.

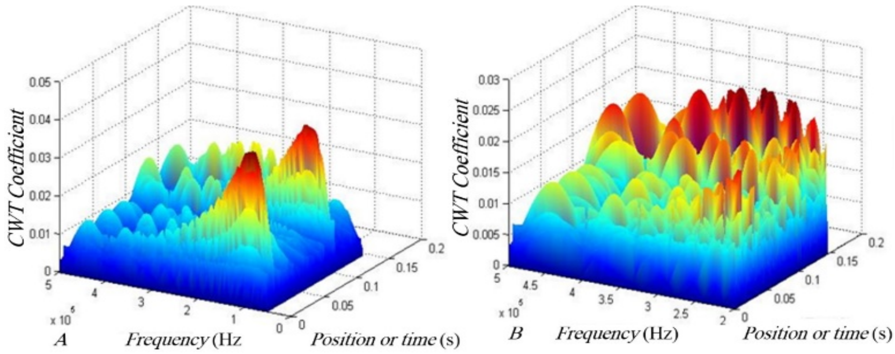

Fig. 6 CWT of the AE signal for the bearing with 1.57 kN applied load. A- frequency between 50 kHz and 500 kHz. B- frequency between 200 kHz and 500 kHz

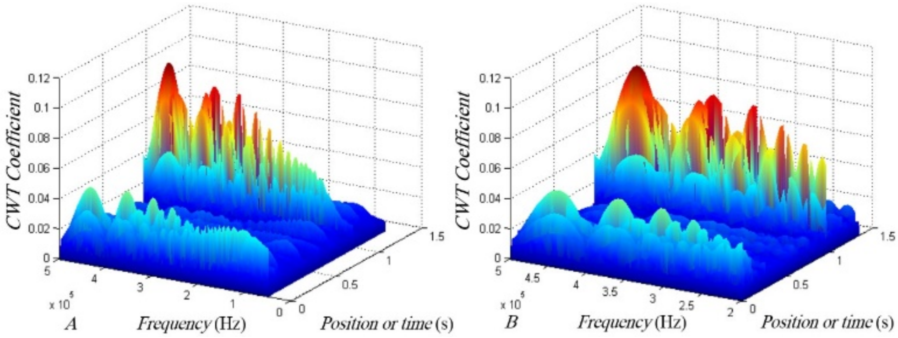

Fig. 7 CWT of the AE signal for the bearing with 3.92 kN applied load. A- frequency between 50 kHz and 500 kHz. B- frequency between 200 kHz and 500 kHz

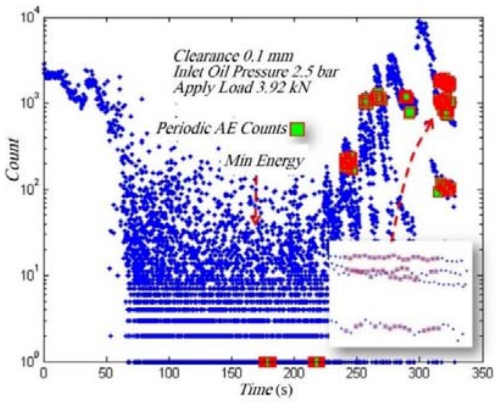

Fig. 8 The AE counts results for the bearing with the inlet oil pressure of 2.5 bar at the various rotational speeds

Every excessive asperity on sliding surfaces between the journal and bearing in region of the minimum oil film position will experience repeated deformation during the rotational motion of the journal over the bearing surface. The repeated deformations at the tips of the excessive asperities or cyclic metal-to-metal contacts in the journal bearings have the ability to produce cyclic AE signals proportional to the number of the shaft rotation. In order to detect these cyclic metallic contacts of the test journal bearings, a MATLAB code was written. Thus, at each rotating speed, clusters of the AE signal counts that had equal magnitude, and also their number was same to the shaft revolutions per second were identified. The results of the two tests are shown in Figs. 8 and 9. As predicted from the previous sections of this investigation, the periodic AE counts signals were not occurred in the HL regime and their number were increased in the BL regime compared to the ML regime. In the Fig. 9 in comparison with the Fig. 8, because of the low inlet oil pressure of the bearing, even at less applied load, the number of periodic AE counts was greater, which indicated that more metal-to-metal contacts were occurred in the same operating conditions.

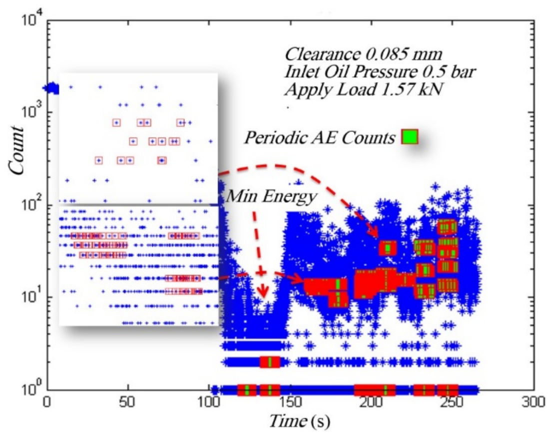

Fig. 9 The AE counts results for the bearing with inlet oil pressure of 0.5 bar at the various rotational speeds

V. Conclusion

The results of the present investigation showed that lubrication condition monitoring in journal bearings is possible by AE method regardless of the design and operating parameters of the bearings. The RMS of AE energy had direct correlation with the friction level of the three lubrication regimes. The experimental values of the working parameters for transition lubrication regimes from HL to ML regime were achieved. Thus, the HL and ML regimes in the test conditions were distinguished from each other. Due to the large number of working factors that had effects on the accuracy of the lubrication theory, there was a significant difference between the experimental values and the theoretical value of the duty parameter, which was defined the boundary between HL and ML regimes. In the bearing with the clearance of 0.07 mm in all the working conditions, the trend similar to the Stribeck curve was observed and both of the apply load and rotational speed had major effect on the lubrication regime situations. In the bearing with the clearance of 0.1 mm, the maximum allowable value for the bearings clearances was selected and the applied loads had negligible effect on the lubrication situation. In the bearing with the clearance of 0.085 mm, very low inlet oil pressure was imposed, and except in the tests with low amounts of the loads at the other loads, lubrication regimes were not distinguishable from each other, and ML or BL regimes were dominant. While, the effects of the clearance size and inlet oil pressure were not considered in the lubrication theories.

From studying the CWT of the AE signals, the following results were obtained:

- The frequency band from 50 kHz to 200 kHz was related to the lubrication regime transitions.
- Metal-to-metal contact of the two sliding surfaces was associated with wide range of the frequency response between 200 kHz and 500 kHz.

Finally, metal-to-metal contacts in ML and BL regimes by identifying the cyclic and equal AE counts were detected, which hitherto has not been explored by AE method.

References

[1] J. Shigley and C. R. Mischke, Bearings and lubrication: a mechanical designers' workbook, New York: McGraw-Hill, 1990.
[2] E. Shigley, J. Brown and C. Thomas, in Standard handbook of machine design (3rd ed.), New York, McGraw-Hill Professional, 2004, pp. 640-641.
[3] A. A. Raimondi and J. Boyd, "A Solution for the Finite Journal Bearing and its Application to Analysis and Design," Tribology Transactions, vol. 1, no. 1, pp. 194-209, 1958.
[4] H. Yasir, M. Salah, A. Roslan and I. Raja, "Acoustic emission and artificial intelligent methods in condition monitoring of raotating machines- A review," in The national conference for post graduate research, University Malaysia Pahang, 2016.
[5] V. Baranov, E. Kudryavtsev, G. Sarychev and V. Schavelin, "Friction of solids and nature of acoustic emission," in Acoustic emission in friction, Moscow, Tribology and interface engineering series, No. 53, 2007, p. 10.
[6] S. Mirhadizadeh, E. Moncholi and D. Mba, " Influence of operational variables in hydrodynamic bearing on the generation of acoustic emission," Tribology international, vol. 43, pp. 1760-1767, 2010.
[7] D. J. Yoon, O. Y. Kwon, M. H. Chung and K. W. Kim, "Acoustic Emission Monitoring of Incipient in Journal Bearings," The Korean Society for Nondestructive Testing, vol. 14, no. 2, pp. 16-22, 1994.
[8] D. Mba, "The Use of Acoustic Emission for Estimation of Bearing Defect Size," Failure Analysis and Prevention, vol. 8, no. 2, pp. 188-192, 2008.
[9] J. Miettinen and P. Andersson, "Acoustic emission of rolling bearings lubricated with contaminated grease," Tribology International, vol. 33, p. 777–787, 2000.
[10] Jamaludin, Mba and Bannister, "Condition monitoring of slow-speed rolling element bearings using stress waves," Proceeding the IMECHE Part E Journal of Process Mechanical Engineering, vol. 215, no. 27, p. 245–271, 2001.
[11] S. Mirhadizadeh and D. Mba, "Observation of acoustic emission in a hydrodynamic bearings," in Technical Inspection and NDT , Tehran, Iran, 2008.
[12] J. Couturier and D. Mba, "Operational Bearing Parameters and Acoustic Emission Generation," Vib. Acoust , vol. 130, no. 2, 2008.
[13] R. M. Douglas, J. A. Steel and R. L. Rueben, "A study of tribological behavior of piston ring/cylinder interaction in diesel engines using acoustic emission," Tribology International, vol. 39, no. 12, pp. 1634-1642, 2006.
[14] S. Niknam, V. Songmene and Y. H. Joe Au, "The use of acoustic emission information to distinguish between dry and lubricated rolling element bearings in low-speed rotating machines," The International Journal of Advanced Manufacturing Technology, vol. 69, no. 9-12, p. 2679–2689, 2013.
[15] H. Towsyfyan, P. Raharjo, F. Gu and A. Ball, "Characterization of acoustic emissions from journal bearings for fault detection," in NDT 2013, 10th-12th September, Telford, UK, 2013.
[16] C. C. Hai, F. W. Dong, Y. X. Qiao and D. S. Zhen, "The Acoustic Emission Diagnose and Analysis of Journal Bearing‘s Inner Ring Looseness Fault by Truck Wheel," Applied Mechanics and Materials, Vols. 130-134, pp. 54-57, 2011.
[17] A. Djebala, N. Ouelaa and N. Hamzaoui, "Detection of rolling bearing defects using discrete wavelet analysis," Meccanica, vol. 43, no. 2, pp. 339-348, 2008.
[18] W. P. Tse and H. T. W.X. Yang, "Machine fault diagnosis through an effective exact wavelet analysis," Sound and Vibration, vol. 277, pp. 1005-1024, 2004.
[19] Y. Ruqiang and X. Robert, "An efficient approach to machine health diagnosis based on harmonic wavelet packet transform," Robotics and Computer-Integrated Manufacturing, vol. 21, no. 4-5, p. 291–301, 2005.
[20] A. Panthi, J. Balwanshi, A. Chandravanshi and G. Gupta, "Design and Analysis of Hydrodynamic Journal Bearing using Raimondi and Boyd Chart," International Journal Of Core Engineering & Management, vol. 2, no. 3, pp. 109-120, 2015.
[21] S. Hosseini, N. Ahmadi and M. Akhlaghi, "Classification of acoustic emission signals generated from journal bearing at different lubrication conditions based on wavelet analysis in combination with artificial neural network and genetic algorithm," Tribology International, no. 95, pp. 426-434, 2016.