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Novel Lubrication Condition Estimation Method by Sliding Ultrasonic Vibration Generated at Piston–Cylinder Liner

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In this paper, a convenient and rapid method is proposed for estimating the lubrication condition and characteristics at the piston–cylinder liner, where sliding motion similar to that like in a reciprocating engine occurs. In general, condition monitoring and diagnosis of the lubricant analyze density, type, form, and the chemical properties, including those of the wear particles. These conventional analysis methods are practical to use, have high reliability, and can often extract extensive information on the wear phenomenon. However, it takes a long time to analyze and assess the results. The proposed method is used to create a three-dimensional graph for the high-frequency vibration (termed sliding ultrasonic vibration) that results from direct contact of the sliding parts of the piston–cylinder liner. The theoretical oil film thickness and degree of contamination are obtained from the sampled lubricant. As a result, more efficient condition monitoring and diagnosis of the lubrication can be performed by applying the proposed method than with conventional methods.

Keywords: Reciprocating Machinery, Condition Monitoring and Diagnosis, Sliding Ultrasonic Vibration, Lubricant, Degree of Contamination, Oil Film Thickness.

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1. Introduction

In the condition monitoring and diagnosis of machinery, many of the state variables [1] generated by its machinery are typically used. In particular, vibration and acoustic signals [2] generated by dynamic machinery are the most common state variables, and the lubrication, pressure, temperature, and current [3,4], which are also obtained from its machinery, are well known as effective state variables. In choosing these state variables, it is necessary to consider the sensitivity of the characteristics to the phenomenon, measurement environment, cost, and convenience of the respective transducers.

In the condition monitoring and diagnosis of the lubricant [5], the number of abrasion particles per unit volume (including the diagnosed lubricant), its configuration, the size of the abrasion particles, and the chemical properties of the diagnosed lubricant can generally be estimated by analyzing the actual extracted lubricant. The above lubricant analysis method is very practical for achieving high reliability and obtaining much information about the lubricant contamination. However, the analysis method requires significant skills of the analyst and a large amount of time to obtain accurate analysis and estimation results. Therefore, a more rapid and convenient method that can obtain information about the lubricant contamination is currently desirable. In this paper, it is proposed that the novel lubrication condition estimation method [6,7] can rapidly and conveniently estimate the degree of lubricant contamination and the lubrication condition (oil film thickness) in the sliding portion of the piston–cylinder liner in the reciprocating machinery.

Studies about the condition monitoring and diagnosis [8,9,10,11] of reciprocating machinery using audio frequency vibration and acoustic analysis generated from driven diesel engines have been conducted. The purpose of these studies was to detect faults of the diagnosed diesel engines; in contrast, the purpose of our study is to monitor the acting stress that is the original cause of the faults. The concept of this study is based on the idea of proactive maintenance [12], which removes the causes of the faults. From the viewpoint of preventive maintenance, the monitoring and diagnosis of the lubricant contamination and lubrication condition are very important because the original cause of the faults in reciprocating machinery is related to the lubrication at the sliding portion of the piston–cylinder liner. Consequently, the quantification of the high-frequency vibration (sliding ultrasonic vibration) generated by direct contact in the sliding portion of the piston–cylinder liner has been carried out in this study. In a prior study, an example with a sliding bearing [13] as the target has related the wave motion generated from the sliding portion of the mechanical elements to its lubrication; however, a lubricant contamination and lubrication condition monitoring method associated with reciprocating machinery based on the amount of sliding ultrasonic vibration cannot be found in any prior studies.

The proposed method is characterized by a three-dimensional curve that experimentally consists of an amount of sliding ultrasonic vibration in the frequency domain and the theoretical oil film thickness at that time; furthermore, the number of abrasion particles per unit volume is determined by analyzing the actual extracted lubricant. The lubricant contamination and lubrication condition can be rapidity and conveniently estimated from the amount of sliding ultrasonic vibration based on the proposed three-dimensional curves. High-efficiency condition monitoring and diagnosis of the lubrication based on the sliding motion can be expected using the present method.

2. Reciprocating Slide-type Friction Testing Machine and Analytical Method

Fig.1 shows a schematic view of the applied reciprocating slide-type friction testing machine (hereinafter referred to as the testing machine), the circulation flow of the lubricant, and the measurement system. Furthermore, Figs. 2, 3, and 4 show the piston, cylinder liner, and piston ring of the actual reciprocating engine used in this study, respectively. The sliding portion (hereinafter referred to as the piston part) of the testing machine made in imitation of the actual engine consists of the piston, the cylinder liner, and the piston ring. The specifications of the actual reciprocating engine used are shown in Table 1. Fig. 5 shows a schematic view of the sliding portion of the testing machine, which can replicate the reciprocating motion of the cylinder liner by mounting it on a bed driven by a stepping motor. The migration distance and speed of the piston part can be controlled using a C++ program, and the vertical load acting on the piston can be controlled using weights placed on the testing machine. An arbitrary temperature of the lubricant at the piston part is set by an oil bath and a thermocouple installed in the piston, and its lubricant is sent to the sliding portion using a pump. The three kinds of lubricants tested in this experiment are additive-free turbine oil 46 (ISO viscosity grade 46, hereinafter referred to as additive46), additive-free mineral oil 22 (ISO viscosity grade 22, hereinafter referred to as additive22), and additive-free mineral oil 10 (ISO viscosity grade 10, hereinafter referred to as additive10).

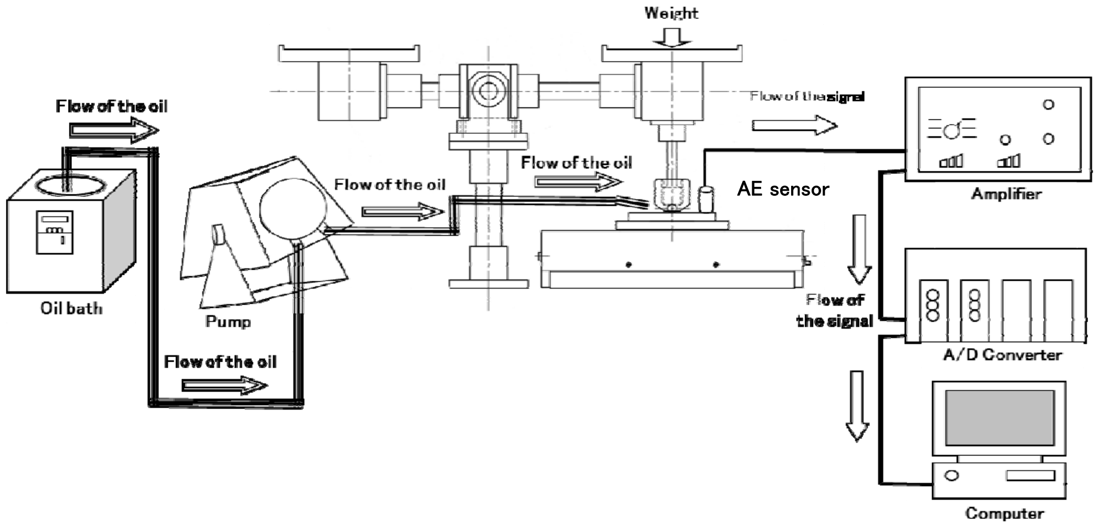

Fig. 1 Reciprocating slide-type friction testing machine and its analytical system

The oil film thickness at the piston part is controlled by varying the viscosity and temperature of the respective lubricants. The generated sliding ultrasonic vibration is measured by an acoustic emission (AE) sensor (Physical Acoustics Corp. S9220) and a high-frequency hydrophone (Bruel & Kjaer type 8103) located in the piston part. However, the hydrophone is 30 [mm] from the sliding part. The sampling frequency of the obtained time series data is set to 500 [kHz], and the cut-off frequency is 200 [kHz]. The measurement time per point is 0.5 [s], and five points of data are measured per condition. The obtained five points of data are translated to frequency domain data by fast Fourier transform (FFT). The stroke of the reciprocating motion is set to 20 [mm], and the reciprocating sliding speed of the piston part is set to 150 [rpm] by the analytical system using a C++ program. New lubricants filtered by filter paper with a pore diameter of 0.8 [μm] are used in this experiment. The applied lubricants that passed through the sliding part during the experiment are sampled in bottles for each of the experimental conditions, and later, the contamination degree coefficient that expresses the contamination level of the sampled lubricants is estimated. Table 2 shows the basic specifications of the applied lubricants and their temperatures; the lubricant sampling time for each of the temperature conditions is set to 28.75 [min] because the number of reciprocating motions on the piston part is equalized at each temperature, and the amount of sampled lubricant at each temperature ranged from 1 to 1.7 liters.

Table 1 Basic specifications of the actual engine

Engine name	Ogawa Seiki FT-300
Engine type	OHV formula Horizontally opposed Two-cylinder Four-stroke engine
Cylinder capacity	50[cc]
Bore	33.6[mm]
Stroke	27.5[mm]

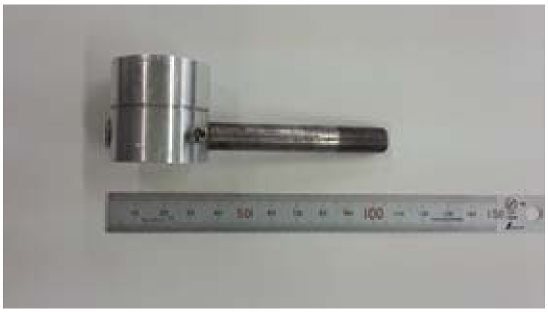

Fig. 2 Piston part

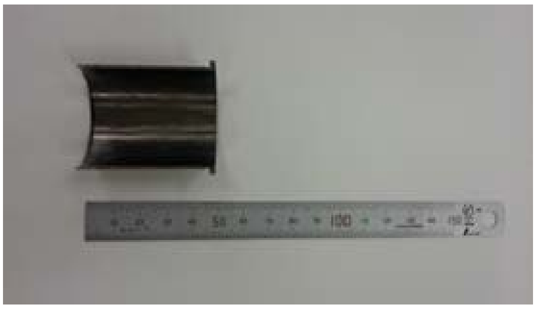

Fig. 3 Cylinder liner part

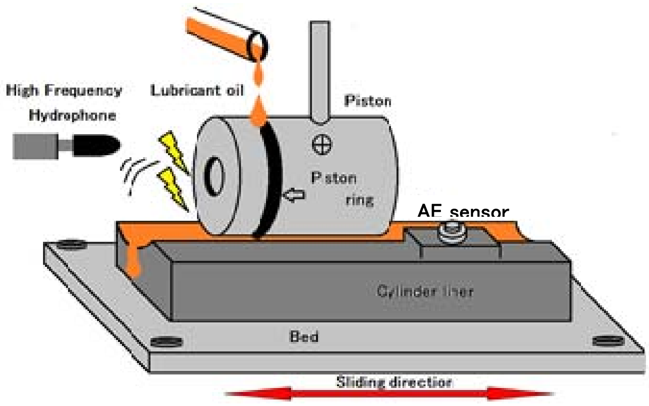

Fig. 5 Schematic view of the sliding portion of the testing machine

Fig. 4 Piston ring parts

Table 2 Basic specifications of the applied lubricants

Lubricant name	Temperature of the piston section[℃]	ISO Viscosity grade [mm/s^2]
Additive-free turbine oil 46	60,90,120,150	46.4
Additive-free machine oil 22	60,90,120,140	21.1
Additive-free machine oil 10	60,90,110,130	9.76

3. Analytical Results and Discussions

Fig. 6 shows the time series data including the sliding ultrasonic vibration for the additive10 case at a temperature of 110 [°C] obtained from one stroke of the reciprocating motion using the AE sensor. In particular, the large amplitude in the time series corresponds to the part, including the frequency components of the sliding ultrasonic vibration. Subsequently, the time series is translated into the frequency domain data shown in Fig. 7 by FFT. The frequency bands that are quantified as sliding ultrasonic vibration are defined as 5 [kHz] to 200 [kHz] in this experiment, and the low-frequency components within 5 [kHz] are cut by a digital low-pass filter because the driving frequency components and the natural frequency components of the testing machine are within 5 [kHz]. Only the frequency components associated with the sliding vibration must be extracted to obtain accurate evaluations.

The oil film thickness h_0 based on Grubin's theory [14] can be calculated using equations (1) to (5) below. The equations of Grubin are based on Elastrohydrodynamic Lubrication Theory (EHL theory). Table 3 lists the necessary constant values for estimating the oil film thickness of each lubricant, specifically, α, μ_0 , ε, and E, which indicate the pressure viscosity coefficient, absolute viscosity of the lubricant, Poisson's ratio, and Young's modulus, respectively; and U, R, L, and W, which indicate the average moving velocity of the piston part, the equivalent radius at the contact site of the piston, the length at the contact site of the piston ring, and the applied load, respectively.

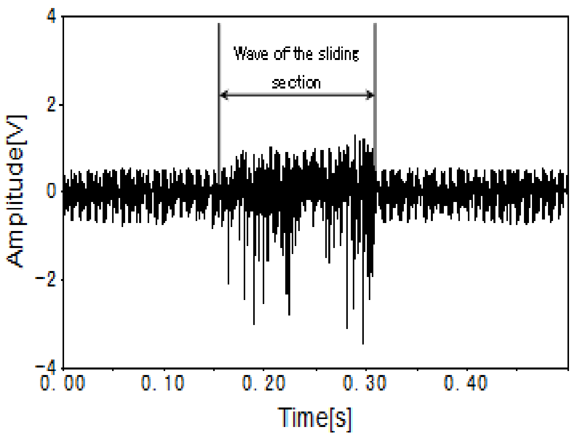

Fig. 6 Obtained time series of data for additive10 at an oil temperature of 110 [°C]

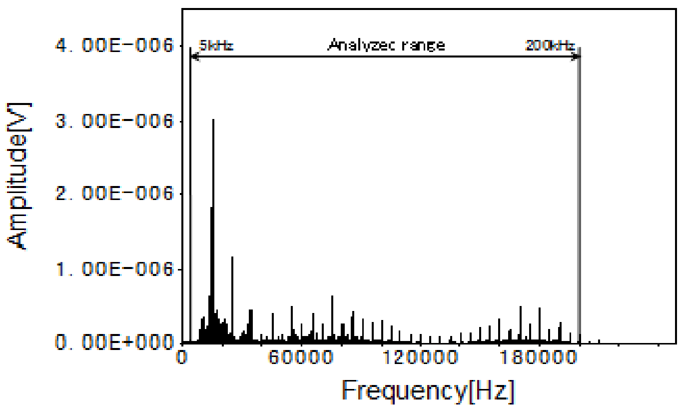

Fig. 7 FFT data of additive 10 at an oil temperature of 110 [°C]

Table 3 Applied constant values used to calculate the oil film thickness

	Oil temperature 60[°C]		
Lubricant oil name	Additive-free 46	Additive-free 22	Additive-free 10
α : Viscosity coefficient pressure [mm^2/N]	0.188	0.159	0.132
μ_0 : Absolute viscosity of lubricant oil [Nsec/mm^2]	1.86E-09	9.63E-10	4.92E-10
ε : Poisson's ratio[—]	0.3		
E: Longitudinal elastic modulus [kgf/mm^2]	21,000		
U: Rotational speed [mm/s]	314		
R: Turning radius [mm]	16.7		
L: Contact length [mm]	1.2		
W: Weight [N]	53.46		

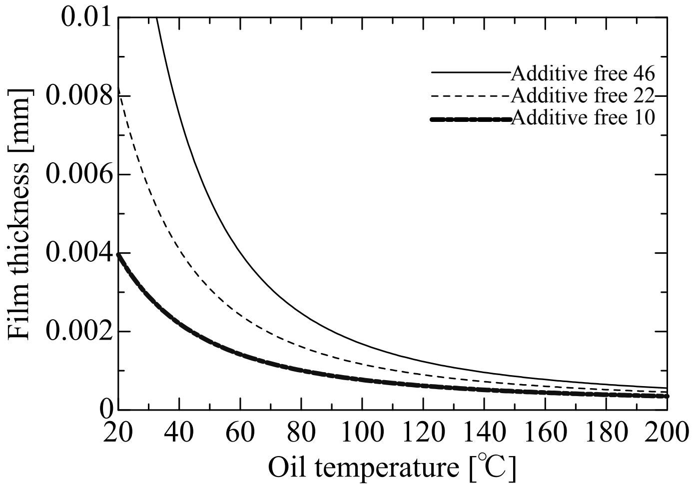

Fig. 8 Oil film thickness of each lubricant

$$h_0 = 1.95R\frac{\left(G\overline{U}\right)^{0.727}}{\overline{W}^{0.091}} \quad (1)$$

$$\overline{U} = \frac{\mu_0 U}{2E'R} \quad (2)$$

$$G = \alpha E' \tag{3}$$

$$\overline{W} = \frac{W}{E' L R} \tag{4}$$

$$\frac{1}{E'} = \frac{1 - \varepsilon^2}{E} \tag{5}$$

Fig. 8 shows the relationship between the oil film thicknesses based on Grubin's theory and the temperature of the applied lubricant. As shown, the oil film thickness tends to suddenly decrease as the temperature increases; this is because the viscosity of the lubricant decreases as its temperature increases. Fig. 9 shows the relationship between the contamination degree of each lubricant obtained by the experiment and the oil film thickness at that time. The applied approximation method is least squares method and applied approximation functions are exponential functions. The reason why I used the exponential functions for fitting curve, the correlation coefficients between obtained data and fitting curve using exponential function are the highest values. The obtained auto-correlation coefficient between the experimental values and the approximation curve is 0.54. It is shown that the oil film thickness tends to suddenly decrease as the corresponding degree of contamination increases.

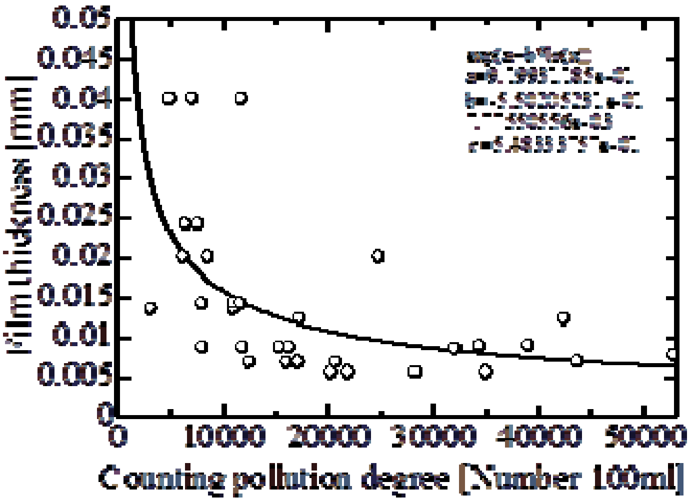

Fig. 9 Oil film thickness and contamination degree determined by the AE sensor

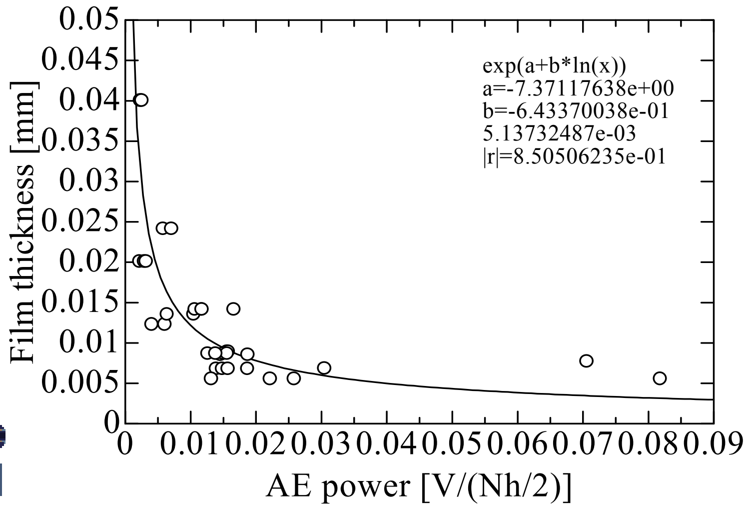

Fig. 10 Oil film thickness and AE power

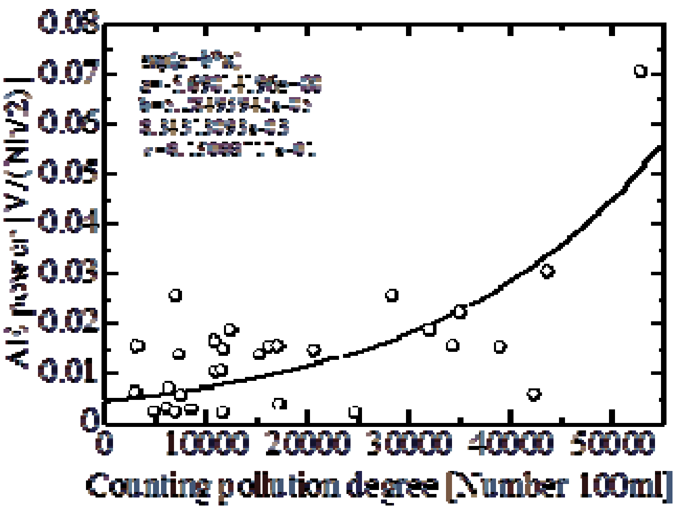

Fig. 11 AE power and contamination degree

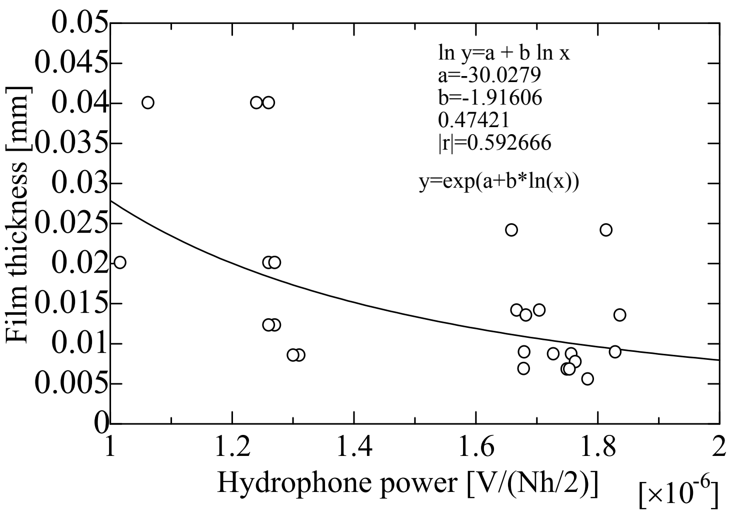

Fig. 12 Oil film thickness and hydrophone power

Direct contact between the surface of the cylinder liner and the piston ring or between both sides through the medium of the metal abrasion particles in the lubricants often occurs by decreasing the oil film thickness. Furthermore, the number of metal abrasion particles in each lubricant increases by increasing the previous amount of direct contact. Fig. 10 shows the relationship between the oil film thickness and the AE power generated at the piston part, where the horizontal axis indicates the AE output per stroke of the piston. The units [V], [N], and [h] represent the voltage, revolutions per hour of the piston, and sampling time of each lubricant, respectively. It is known that the oil film thickness tends to suddenly decrease as the corresponding AE power increases. Direct contact at the sliding portion often occurs due to the thin oil film thickness, and as a result of the aforementioned reason, the increase in AE power is generally able to be well understood. Furthermore, the obtained autocorrelation coefficient between the AE power and the oil film thickness is 0.85. It is concluded that the oil film thickness can be conveniently estimated from the generated AE power based on the sliding ultrasonic vibration. Fig. 11 shows the relationship between the measured contamination degree and the AE power for each applied lubricant. It is shown that the contamination degree tends to suddenly increase as the corresponding AE power increases. This is obviously because of the increasing direct contact at the piston part. The autocorrelation coefficient between the AE power and the contamination degree is about 0.81. As a result of the above reasons, it is concluded that the contamination degree can be conveniently estimated from the generated AE power based on the sliding ultrasonic vibration at the piston part.

Fig. 12 represents the relationship between the oil film thickness and the hydrophone power in the case of using the high-frequency hydrophone as a sensor for measuring the sliding ultrasonic vibration. It is shown that the oil film thickness tends to gently decrease as the corresponding hydrophone power increases. The obtained autocorrelation coefficient between the obtained experimental values and the exponential approximation curve is 0.59. The autocorrelation coefficient of the above relation is not as good as the value of 0.85 obtained using the AE sensor. Therefore, it is concluded that the convenient estimation of the oil film thickness is more difficult using the hydrophone than using the AE sensor because the sensitivity of the high-frequency hydrophone is inferior to that of the AE sensor, and the signal-to-noise ratio of the hydrophone is worse than that of the non-contact type sensor for the target facility. Fig. 13 shows the relationship between the contamination degree and the hydrophone power. It is shown that the hydrophone power tends to slightly increase as the corresponding contamination degree increases. The obtained autocorrelation coefficient between the obtained experimental values and the exponential approximation curve is about 0.05. In addition, it is known that according to the inspection of this figure, the deviation between the obtained approximation curve and the experimental values is large. Like in the case of the oil film thickness and the hydrophone power, the sensitivity and signal-to-noise ratio of the hydrophone are inferior to those of the AE sensor. Due to the above reasons, it is concluded that when the hydrophone is used, the precision estimation of the contamination degree that expresses the contamination level of the lubricant is more difficult than in the case using the AE sensor.

Fig. 14 shows the three-dimensional curve that is constructed using the obtained AE power, the measured contamination degree, and the oil film thickness associated with generating the sliding ultrasonic vibration. Each

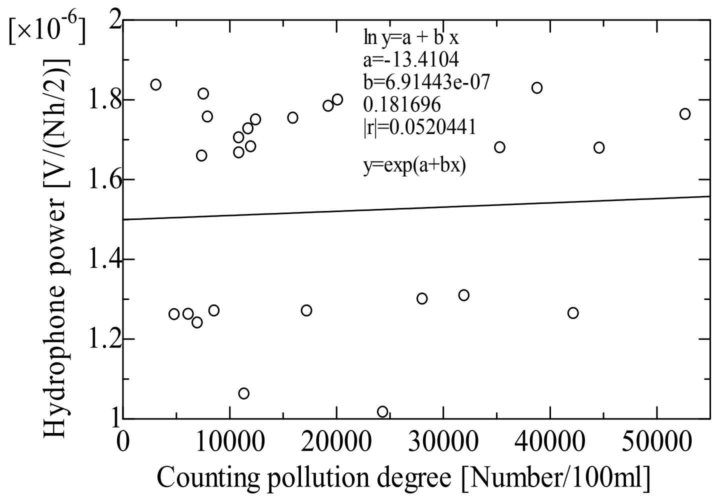

Fig. 13 Hydrophone power and its contamination degree

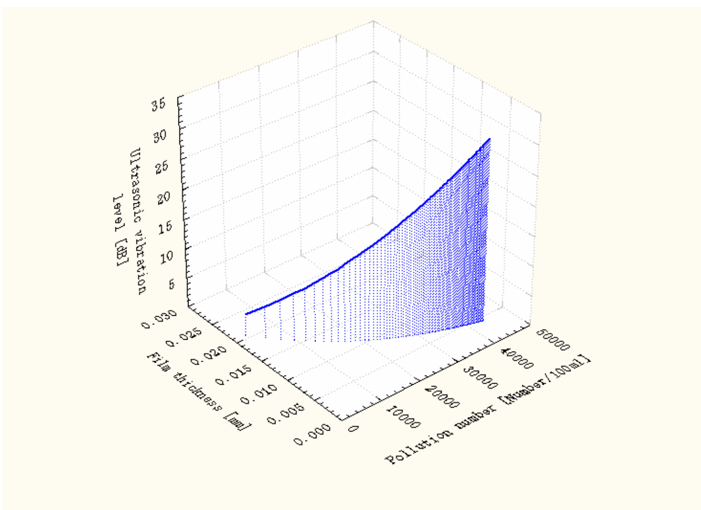

Fig. 14 Three-dimensional curve associated with the obtained ultrasonic vibration level, its oil film thickness, and its measured contamination degree

of the AE power values are represented by converting it into a level value in order to consider the practical use, and the level value is defined as $20\log(S/S_0)$. Here, the reference value S_0 indicates the AE power when the lubricant temperature is 40 [°C]. The approximate three-dimensional curve is acquired by the least-square method. The oil film thickness and measured contamination degree associated with the lubricants at the piston–cylinder liner can be estimated by obtaining the AE power generated by the sliding ultrasonic vibration based on the above three-dimensional curve. Accordingly, the lubricant contamination and lubrication condition monitoring and diagnosis associated with many reciprocating machines can be conveniently implemented using this proposed method.

4. Conclusions

This paper proposes a quantification method for high-frequency vibration called sliding ultrasonic vibration generated by direct contact in the sliding portion of a piston–cylinder liner. Furthermore, the quantified sliding ultrasonic vibration is related to the corresponding contamination degree and oil film thickness based on Grubin's theory. As a result, the oil film thickness and contamination degree associated with the lubricants at the piston–cylinder liner can be conveniently estimated from the level of AE power obtained by the quantified sliding ultrasonic vibration based on the proposed three-dimensional curve associated with the state variables discussed. The lubricant contamination and the lubrication condition monitoring and diagnosis associated with reciprocating machinery can be conveniently implemented using this proposed method.

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References

[1] Iron and Steel Institute of Japan, *Handbook of Condition Diagnosis Technology*, 1st ed., Maruzen Press, pp.1-4, 1986.
[2] R.B.Randall. *Vibration-based Condition Monitoring: Industrial, Aerospace and Automotive Applications,* Wiley Press, pp.3-8, 2011.
[3] HAMID A.TOLIYAT et al., *ELECTRIC MACHINE*, CRC Press, pp.111-153, 2013.
[4] T. Toyota, *How to proceed condition diagnosis of electrical equipment*, Japan Institute of Plant Maintenance, p.13-15, 2005.
[5] T. Toyota, *How to proceed condition diagnosis of mechanical system ~Mainly on oil analyasis~*, Japan Institute of Plant Maintenance,p.32-90, 1998.
[6] Shintaro Shiota, Masahiko Kawabata and Hiromitsu Ohta, Condition Monitoring Technology of Lubrication of a Reciprocating Machinery Based on Generated Sliding Ultrasonic Vibration, *Proceedings of the 66 th JSME meeting, kyushu branch* ,pp.369-370, 2013.
[7] Hiromitsu Ohta, Masahiko Kawabata, Shintaro Shiota and Hidemi Emura, High Precision Method on Lubricant Condition Monitoring Based on Reciprocating Machinery, *Proceedings of the spring meeting of the Society of Plant Engineers Japan*, pp.17-20, 2013.
[8] Ryuichi Kimura, Noboru Nakai, Noboru Kataoka and Hiroshi Mizutani, Real Time Diagnosis System of Diesel Engine by Abnormal Sound Detection, *Journal of the M.E.S.J.*,Vol.28,No.3,pp236-242,1993.
[9] Ryuichi Kimura, Wataru Terashima, Noboru Nakai, Tetsuo Yamada and Shou Takeda, Diagnostic method for 2-Stroke Cycle Diesel Engine by Measurement of Vibration on Cylinder-Jacket - Observation of Change in Normal Vibration Pattern -, *Journal of the M.E.S.J.*,Vol.33,No.10, pp.738-745, 1998.
[10] Senichi Sasaki, Condition Monitoring and Fault Diagnosis for marine engine -About Marine Diesel Engine-, *Proceedings of the ClassNK meeting*, NIPPON KAIJI KYOKAI (ClassNK),pp.1-12, 2000.
[11] Hiromitsu Ohta, Minoru Tuda and Keiji Sasada,Coveninet Condition Monitoring Method for Marine Diesel Engine Based on the Vibration Acceleration of Cylinder Head, *Proceedings of the autumn meeting of the Society of Plant Engineers Japan*, pp.121-125, 2010.
[12] E.C. Fitch, *Proactive Maintenace for Mechanical Systems* Second Edition, FES Inc., (1992).
[13] Kazunori Okamoto, Hiromi Shihara, Tomoya Nagayama and Tadahiko Kurosawa, Optimum Condition Monitoring Technique of Sliding Bearing for Marine Engin, *Proceedings of the ClassNK meeting*, NIPPON KAIJI KYOKAI (ClassNK),pp.71-81, 2006.
[14] Kouichi Watanabe, Elastic Fluid Lubrication Theory (Theory of EHL), NACHI TECHNICAL REPORT, Vol.11D1, NACHI-FUJIKOSHI CORP., 2006.