Vibration Monitoring for Wear Condition of Cylinder Liner and Piston Ring in Marine Diesel Engine

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Abstract

The cylinder liner and piston rings of present large marine diesel engines are in severe lubricating condition with the trend of higher engine output. Sometimes their abnormal wear due to scuffing appears in the earlier stage of the voyage despite manufacturer's effort in the structural and metallurgical modification. In such situation many researchers are studying the method of the lubricating condition monitoring from viewpoint of Condition Based Maintenance (CBM).

The purpose of this study is to establish the technique for wear condition monitoring of cylinder liner and piston rings based on liner vibration signals. This paper describes the results of preliminary test carried out in shop trial of a 2-stroke cycle engine, and examines the effectiveness of the monitoring method by the analysis of vibration data measured in the period of running-in of the engine.

1. Introduction

Recent large marine diesel engines are operated in the condition of increased maximum pressure and longer piston stroke. As a consequense of these trends, the problem of abnormal wear of cylinder liner and piston rings (hereafter cited as liner/rings) is increasingly raised, and for early detection of the abnormal wear not a few studies have been done on the lubricating condition monitoring.

Among the monitoring methods, measurement of oil-film thickness or oil sampling through a hole made in liner wall[1] has high sensitivity in the detection of abnormal condition. However these methods are not so popular because of the expensiveness of the equipment and the difficulties in employment for ships in service.

On the other hand there is an attempt to evaluate the wear condition of liner/rings by vibration measurements[2]. Vibration monitoring as a method of condition monitoring is usually applied to general rotational machines such as steam turbines, turbo compressors, pumps and so on, but application to reciprocating machines such as diesel engines is rare. This is because vibrations of diesel engines are induced by several kinds of exciting forces and the relationship between vibration responses and abnormal conditions is not known in detail.

While the goal of this study is to establish the technique for wear condition monitoring using vibration data, at the beginning it is to be examined whether vibration signal is effective or not as a signal for diagnosis. Because vibration signals include several kinds of signals irrelevant to friction, rubbing vibrations induced by the friction of liner/rings are to be distinguished from other vibrations and the availability of vibration measurements must be estimated.

This paper describes the results of preliminary test which was carried out in shop trial of a 2-stroke cycle crosshead engine (uni-flow scavenging, rated output: 9,156kW, rated speed: 114 rpm). Based on the vibration data measured in the period of running-in of the engine, the effectiveness of the method is examined.

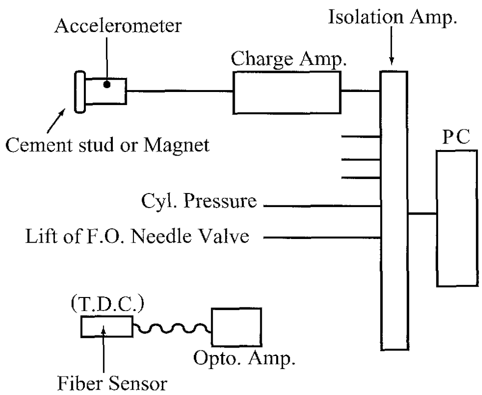

Fig.1 Outline of measuring system

2. OUTLINE OF VIBRATION MONITORING

This monitoring system is mainly aimed at the measurement of high frequency vibrations which are induced by rubbing of liner/rings. In preliminary test, however, cylinder pressure and lift of F.O. needle valve were additionally measured to clarify the features of vibrations in combustion process. Fig.1 shows the outline of the system including all measured items. Vibration signals were measured by accelerometers and charge amplifiers containing low-pass filter, and output signals from the charge amplifiers were acquired by PC incorporating A/D conversion board via isolation amplifiers.

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Representative locations of vibration measurements are shown in Fig.2. In preliminary test, measurements of vibration were made at several locations on one cylinder unit. This is because the vibrations are induced by the exciting force such as combustion and exhaust valve closing as well as the friction of liner/rings, and measurements at other location than cylinder liner are necessary to specify the vibration sources. While most of the accelerometers were mounted on magnet base attached to cylinder, for the measurement of cylinder liner vibration, additional accelerometers were directly fitted to the exposed liner surface using adhesive cement to observe the difference in responses due to contact resonance.

In terms of the condition of A/D conversion, the system is so designed that vibrations have high responses at the frequency range from 5 kHz to 30 kHz taking advantage of the resonance characteristics of accelerometer (shown in Fig.3). The sampling frequency was adjusted to 100 kHz per channel, which is higher than usual, considering the setting of cut-off frequency of low-pass filter. Table 1 shows the specifications of vibration measuring system.

Table 1 Specifications of vibration measuring system

Accelerometer B&K type 4371 Natural frequency: 48 kHz
Charge amplifier B&K NEXUS type 2692A L.P.F. cut-off frequency: 30 kHz L.P.F. attenuation slope: 40 dB/dec H.P.F. cut-off frequency: 1 Hz
A/D conversion board National Instruments type PCI-MIO-16E-1 Channels: 16 SE Max. sampling rate: 1.25 MS/s Resolution: 12 bits

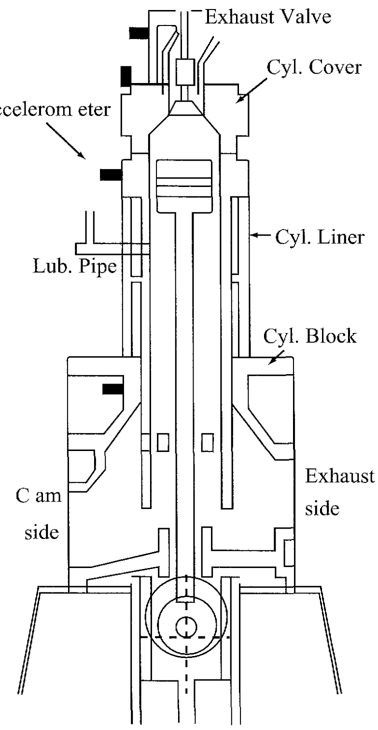

Fig.2 Arrangement of accelerometers

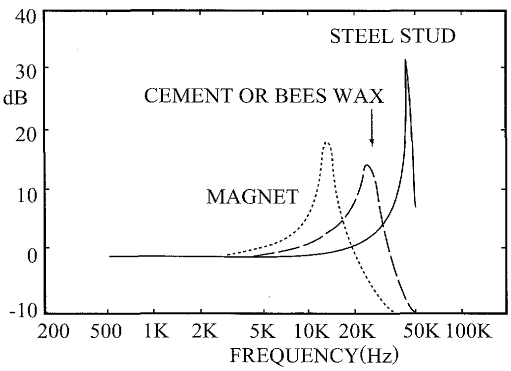

Fig. 3 Frequency characteristics of accelerometer

3. Results of the measurement
3.1 Features of waveform

As a procedure for the test running of engine, in a way of stepwise increase of load, the running for proper fit of liner/rings was initiated and subsequently various kinds of performance test were carried out. The results of measurement at 75% load in the period of running-in are demonstrated in Fig.4.

The signals shown in this figure, in order from top to bottom, represent rotation mark (T.D.C.), lift of F.O. needle valve, cylinder pressure and accelerations of exhaust valve, cylinder cover, cylinder liner and cylinder block. At cylinder cover the acceleration was measured in the vertical direction and at other locations measured in the horizontal direction. And all acceleration signals shown here are the results of measurements in use of magnet base. From Fig.4, vibration sources are summarized as follows.

(1) Closing of exhaust valve and flow of exhaust gas

In the signal measured at exhaust valve, an excessive vibration due to the flow of exhaust gas is observed at the crank angle of approximately 45 degrees (before B.D.C.), and this vibration is detected also in other locations. At the crank angle of approximately 90 degress (after B.D.C.), an impulsive large vibration occurs due to the closing of exhaust valve, that is an impact force on the valve seat. This impulsive vibration is distinguished particularly in the signals measured at cylinder cover and cylinder liner. Naturally, the amplitude of these vibrations is large at the upper position of cylinder unit.

(2) Combustion and opening/closing of F.O. needle valve

The effect of combustion appears as a comparatively large vibration at the upper position of cylinder. It can be seen from careful observation of the waveform measured at cylinder cover that the vibration in combustion process includes the transient vibrations[3] corresponding to the motion of F.O. needle valve, and this transient vibrations are observed also at cylinder liner. These are induced by the impact force which occurs with opening and closing of F.O. needle valve, and are therefore essentially the same phenomenon as mentioned in the case of closing of exhaust valve. It should be noted that the vibrations observed in the period of F.O. injection include above transient vibrations as well as the vibrations due to combustion.

(3) Contact of piston rings with non-continuous parts of cylinder liner surface

There are non-continuous parts such as (upper and lower) cylinder oil grooves and scavenging ports on inner surface of cylinder liner. When piston rings run across the non-cotinuous parts, impulsive vibrations are generated by the contact with the parts. In Fig.4, it is noticeable that two vibrations occur symmetrically before and after T.D.C. at the crank angle corresponding to contact of top-ring with oil grooves. When piston rings run across scavenging ports, in some case relevant vibrations occur and in other case don't; in Fig.4, the vibrations are not apparent. This is supposed to be the phenomenon caused by the positional relationship between scavenging ports and free ends of piston rings.

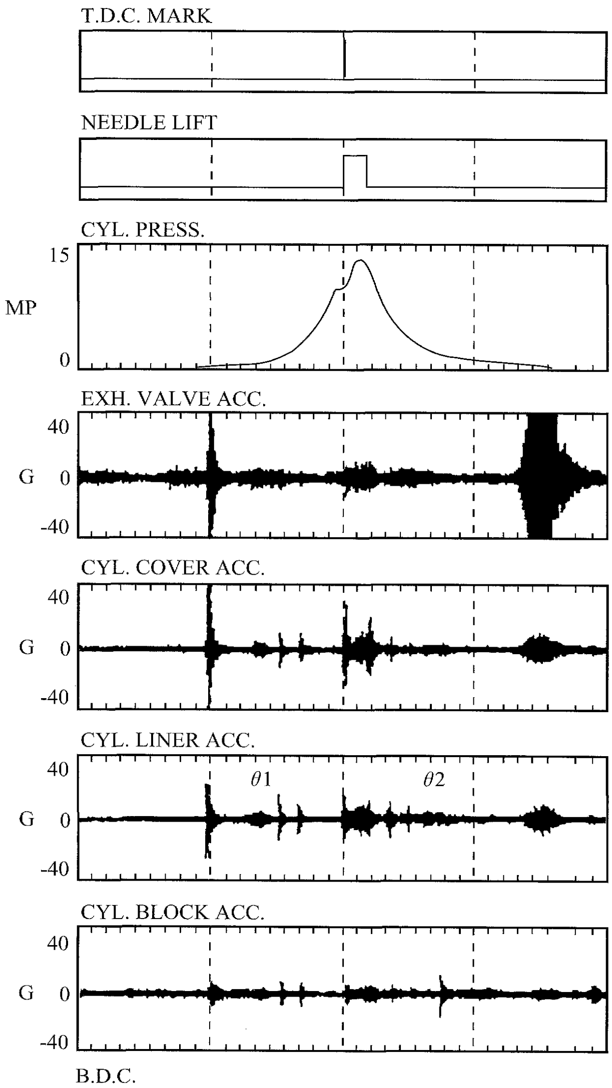

Fig.4 Responses at the location of exhaust valve, cylinder cover, cylinder liner and cylinder block (load:75%)

(4) friction between cylinder liner and piston rings

The vibrations due to the friction between cylinder liner and piston rings appear especially at cylinder liner in the vicinity of crank angles which maximize piston speed; the angles, shown as $\theta 1$ and $\theta 2$ in Fig.4, equal 66.7 degrees (before and after T.D.C.) in measured engine. Amplitude of the vibrations changes slowly compared with other instantaneous vibrations. In comparison between $\theta 1$ and $\theta 2$, amplitude at $\theta 1$ is generally larger than that at $\theta 2$ though the difference is not apparent in Fig.4. The friction of liner/rings depends upon their lubricating condition in addition to piston speed. Consequently, if piston is moving, rubbing vibrations may appear even in combustion process, which decrease the thickness of oil-film. However, it is difficult in the time domain to distinguish the rubbing vibrations from those induced by combustion.

3.2 Features of spectrum map

When the waveform of measured signals changes transiently in one frame of the analysis, it is a general way to extract the features of the frequency domain using time-frequency analysis. In this study STFT (Short Time Fourier Transform), which is a kind of time-frequency analysis, was employed with following conditions:

- the width of a data frame for FFT is 1024 points (equivalent to approximately 10 msec);
- a quantity of shift of data frame for 2D map is half of frame width;
- hanning window is employed for data window.

And, vibration level (dB re = 10-5 m/sec2) is represented by gradation pattern from white (70 dB) to black (130 dB).

Fig.5 shows the spectrum map of cylinder liner vibrations in running-in of engine (50% load) and in this figure the acceleration signal in use of adhesive cement is demonstrated. As we can see from the spectrum, peculiar curved stripes appears symmetrically with respect to the position of T.D.C. The piston speed of this engine rises to the peak at the crank angles of $\theta 1$ and $\theta 2$ as shown in Fig.6, and the curved stripes of Fig.5 are constituted by a set of curves proportional to the piston speed. At $\theta 1$, for example, fundamental frequency is approximately 1.64 kHz, and many peaks are regularly arranged at the frequencies of integral multiplies of the fundamental frequency.

To examine the geometric changes of curved stripes with engine speed, the relationship between fundamental frequency at $\theta 1$ and engine speeds at loads of 4 cases is shown in Fig.7. As we can see easily, fundamental frequency is directly proprtional to engine speed. It can be therfore understood that vertical intervals of stripes expand with the increase of engine speed.

Generation of stripes is considered to be related to the geometric shape of inner surface of cylinder liner. To progress a capability to keep oil-film thick enough, liner surface of low-speed large marine engine is fabricated into corrugated shape (or wave cut) as shown in Fig.8, and that is the same as measured engine. In this case, from the pitch L of corrugated shape and piston speed V, it follows that the piston ring comes into contact with the swelled part of inner surface at a period of T (= L/V). As shown in Fig.9, if impact forces are applied at a period of T, responses in the frequency domain appear at regular intervals, which are represented by the frequencies of integral multiplies of fundamental frequency 1/T. Fundamental frequencies shown in Fig.5 nearly correspond to the values of 1/T calculated from the piston speed of measured engine, and hence the stripes are considered to be generated by the corrugated shape of liner surface.

In contrast with the above, the spectrum of impulsive vibrations induced by following sources is distributed in the shape of vertical line: closing of exhaust valve, contact of piston rings with non-continuous parts of liner surface, opening and closing of F.O. needle valve. In the case of vibrations induced by combustion and flow of exhaust gas, a spread of the distribution in the direction of time is observed in spectrum map. These spectrum are featured by the continuous distribution in the wide frequency range.

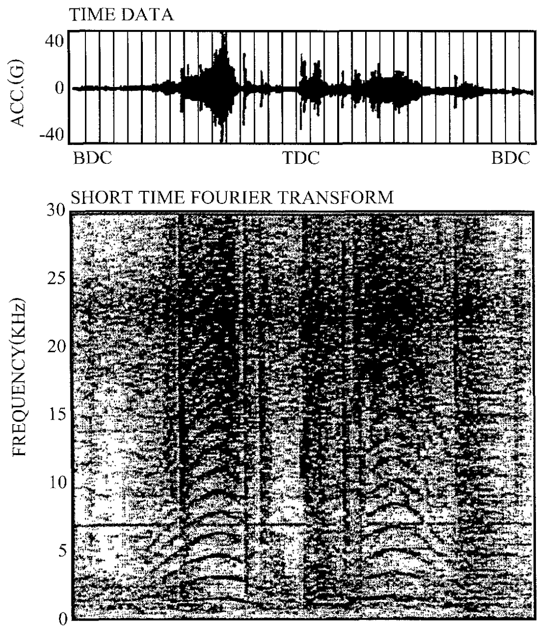

Fig.5 A spectrum map of cylinder liner vibration by STFT (load:50%)

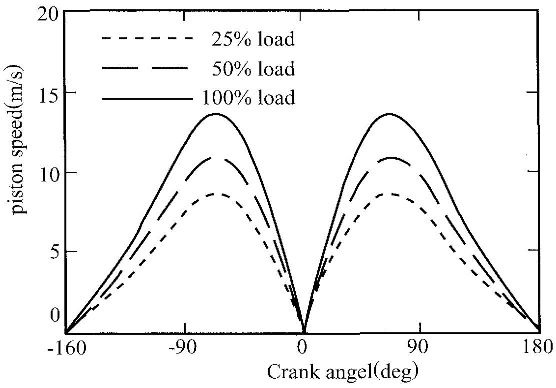

Fig.6 The curves of piston speed

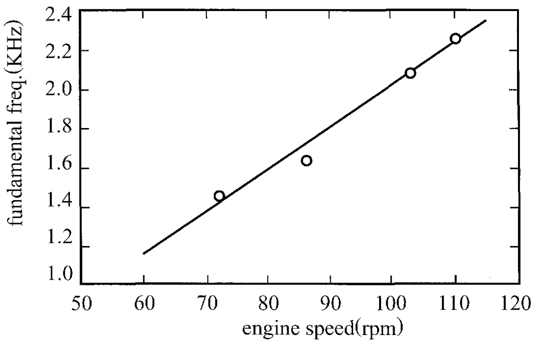

Fig.7 The relation between fundamental frequency and enginespeed at crank angle $\theta 1$

As mentioned above, STFT enables the distinction of vibration sources. From the difference in spectrum distribution, it is possible to distinguish the rubbing vibrations of liner/rings from other vibrations.

In addition, while high responses are conspicuous in frequency range between 20 kHz and 25 kHz at any crank angle in Fig.5, this is caused by the contact resonance of used accelerometer as described later. And, it must be noted that in general the foregoing stripes are caused also by the rubbing vibration in the adjacent cylinder to measured cylinder, though the phenomenon is not apparent in Fig.5.

3.3 Changes of spectrum with time

In order to examine the changes of spectrum with time, FFT analysis with the averaging of eight revolutions was carried out using one frame data (1024 points) at the crank angle of q1. Fig.10 and Fig.11 show the spectrum of cylinder liner vibrations measured at 50% load in running-in and after 20 hours' running respectively, and in both figures examples in use of adhesive cement are demonstrated. For the manner of representation of data, the horizontal axis (frequency axis) with a logarithmic scale is used to compare with the resonance characteristics curves of accelerometers.

In Fig.10, many peaks corresponding to stripes shown in Fig.5 are prominent. However, if the fluctuations are neglected, the response curve is similar to the frequency characteristics of accelerometers in use of adhesive cement shown in Fig.3, and is of the shape on which the contact resonance of approximately 25 kHz has a strong effect. On the other hand, in Fig.11, the significant decrease of responses can be seen at the higher frequency range than 3 kHz, though the existence of contact resonance is confirmed.

Since the principal difference between both figures is running hours of engine, this decrease is supposedly caused by the change of rubbing of liner/rings. It is said that the abnormality such as scuffing of cylinder liner brings the increase of responses at higher frequencies[4]. In this measurement, to the contrary, the decrease of the responses with time was verified taking advantage of running-in of the engine, in which liner/rings are comparatively in severe condition of rubbing.

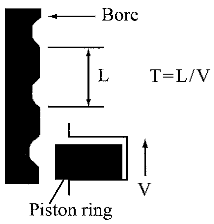

Fig.8 Corrugated surface of cylinder liner

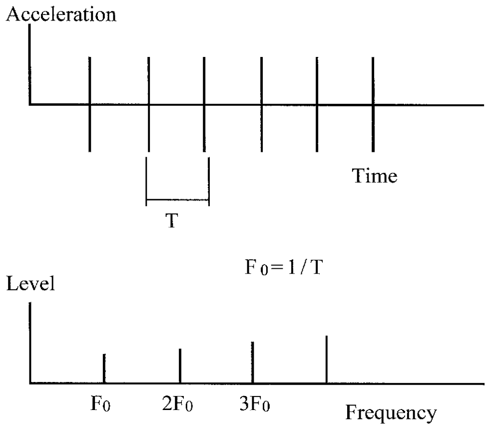

Fig.9 Generation mechanizm of stripes in spectrum map

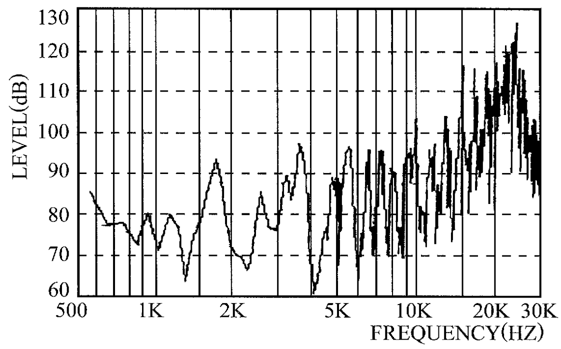

Fig.10 A spectrum of cylinder liner vibrations at $\theta 1$
(load: 50%, running-in)

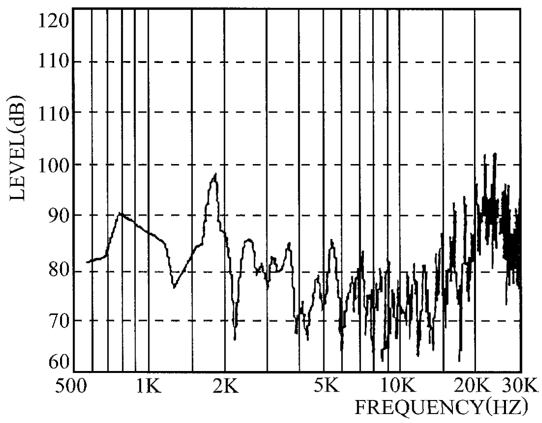

Fig.11 A spectrum of cylinder liner vibrations at $\theta 1$
(load: 50%, 20 hours later)

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Thus, changes of spectrum in lower load, 25% and 50%, were notable, but the changes in higher load than those were not remarkable. This is considered to be related to the way of loading; that is, the engine was initially operated at step-up loads, 25%, 50%, 75% and 90%, and also 20 hours later at the same step-up loads. It is therefore assumed that the rubbing of liner/rings at the first load of 25% and 50% was in severe condition, but subsequently slight rubbing lasted.

4. Conclusion

For the purpose of establishing the monitoring technique for wear condition of liner/rings in large marine engine of 2-stroke cycle, the method using vibration signals of cylinder liner was examined, and for the first step preliminary test was carried out in shop trial of engine. Measurements made it clear that vibration signals include several kinds of vibrations induced by closing of exhaust valve, flow of exhaust gas, combustion, opening/closing of F.O. needle

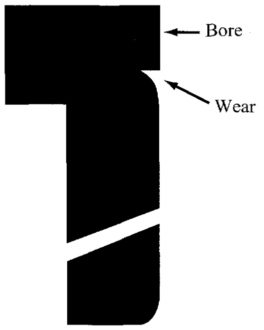

Fig.12 Wear of cylinder liner

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