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TIME SERIES ANALYSIS OF CYLINDER COVER VIBRATION AND DIESEL ENGINE FAULT DIAGNOSIS

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Abstract

The working conditions of diesel engine components can be detected by monitoring the cylinder cover vibration signals. In order to analyze these vibration signals, both time series analysis and Fourier spectrum method are adopted. The characteristics of cylinder cover vibration signals excited by valve impacts and combustion forces are discussed, and some criteria suitable for diesel engine vibration monitoring and fault diagnosis are proposed. The experimental investigations were carried out on a medium speed four-stroke 6-cylinder marine diesel engine. The cylinder cover surface vibration signals are measured and analyzed for diagnosing the working conditions of valve clearances and engine cylinder loads. The results show that it is viable and effective to detect the variations in valve clearance and engine cylinder load from cylinder cover vibration signals. Furthermore, this investigation shows that autoregressive spectrum is smoother than Fourier spectrum, with more accurate frequency locating, so it is more suitable to diesel engine condition monitoring and fault diagnosis.

INTRODUCTION

Timely detection of incipient malfunctions of diesel engines is desirable for avoiding costly unanticipated shutdown as well as preventing severe damage to the engine. The engine cylinder cover vibration is the result of the structure response of the engine to its internal exciting forces. The engine malfunctions, which affect these forces, also affect the vibration signals of the cylinder cover. This paper conducts an investigation on diesel engine condition monitoring using cylinder cover vibration signals. The aim of this investigation was to demonstrate the feasibility and effectiveness of using engine cylinder cover vibration signals for detection of the working conditions of valve clearances and the engine cylinder loads. In order to analysis these vibration signals, both time series analysis and Fourier spectrum method are adopted. The characteristics of cylinder cover vibration signals excited by valve impacts and combustion forces are

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discussed, and some criteria suitable for diesel engine vibration monitoring and fault diagnosis are proposed.

The experimental investigations are carried out on a medium speed four-stroke 6-cylinder marine diesel engine. The cylinder cover surface vibration signals are measured and analyzed, and the relationships between the character parameters of cylinder cover surface vibration and the working conditions of valves and engine cylinder loads are presented. The analytic and experimental investigations indicate that the variations in valve clearance and engine load can be distinguished by use of cylinder cover vibration signals. The results show that it is viable and effective to detect the variations in valve clearance and engine cylinder load from cylinder cover vibration signals.

TIME SERIES AND AUTOREGRESSIVE SPECTRUM ANALYSIS

Time series analysis is one of the modern spectrum estimation methods. In this method, the parametric model, which is equivalent to the original system, is established. In contrast to Fourier technique, this is a parametric approach, the system's parameters describing the signal, and thus also its PSD (Power Spectral Density) [1], [2]. From the data model, the transfer function of the system and the PSD of the signal can be estimated.

The autoregressive-moving average data model (ARMA model) may be considered to produce data $\{x_t\}$ when excited by a white noise sequence $\{a_t\}$, as follows:

$$x_t - \sum_{n=1}^{N} \phi_n x_{t-n} = a_t - \sum_{m=1}^{M} \theta_m a_{t-m} \tag{1}$$

$$\{a_t\} \sim NID(0, \sigma^2)$$

Where Φ_n and θ_m are autoregressive coefficients and moving average coefficients respectively. The model is denoted as ARMA(N,M). By introducing the backward shift operator B, which is defined as $Bx_t = x_{t-1}$, then we rewrite Equation (1) as

$$(1 - \sum_{n=1}^{N} \phi_n B^n) x_t = (1 - \sum_{m=1}^{M} \theta_m B^m) a_t \tag{2}$$

It can be simplified as

$$\Phi(B)\, x_t = \theta(B)\, a_t \tag{3}$$

$$x_t = \frac{\theta(B)}{\phi(B)} a_t \tag{4}$$

Where $\varphi(B) = 1 - \varphi_1 B - \varphi_2 B^2 - \cdots - \varphi_n B^N$

$\theta(B) = 1 - \theta_1 B - \theta_2 B^2 - \cdots - \theta_M B^M$

If all the θ_m (m=1,2,3 …M) equal to zero, while φ_n (n=1,2,3 …N) are not, then Equation (4) becomes:

$$x_t - \sum_{n=1}^{N} \varphi_n x_{t-n} = a_t \tag{5}$$

Equation (5) denotes the autoregressive model (AR model).

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$$x_t = a_t - \sum_{m}^{M} \theta_m a_{t-m} \tag{6}$$

Equation (6) denotes the moving average model (MA model).

Both ARMA models and MA models can be approached and represented by high order AR models [3]. And most mechanical systems can also be simplified as AR models. The AR models are more convenient and more widely used in practice. In this paper, the AR models are used to analyze the vibration characteristics of engine cylinder cover. The model orders can be determined by F–test, or AIC (Akaike Information Criterion), or BIC criterion [4].

The ARMA, MA, or AR spectrum is a kind of parametric spectrum based on the time series model. The power spectrum density (PSD) of vibration signals can be obtained from the parameters of time series models, which is given as:

$$S_x(f) = \sigma^2 \left| \frac{\theta(e^{-i2\pi fT})}{\theta(e^{-i2\pi fT})} \right|^2 \tag{7}$$

Where T is the sampling interval, and $i = \sqrt{-1}$.

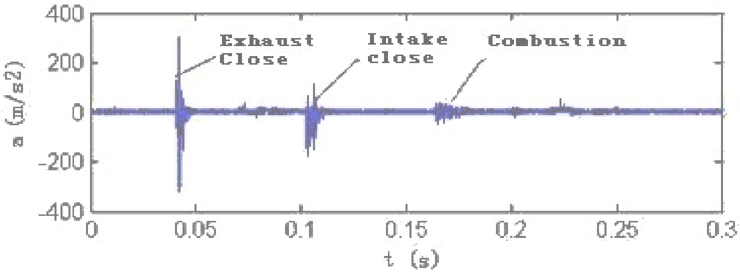
EXPERIMENTAL PROCEDURE AND SCHEME

The engine used in this study is a 6-cylinder, four-stroke, turbo-charged, water-cooled diesel engine. The cylinder bore is 260 mm and the piston stroke is 340 mm, giving an output of 370 kW at 400 r/min. The firing order of the engine is 1-5-3-6-2-4. The valve timing of the engine is: intake valve open at 73° crank angle before top dead center, intake valve close at 37° crank angle after bottom dead center, exhaust valve open at 45° crank angle before bottom dead center, exhaust valve close at 65° crank angle after top dead center.

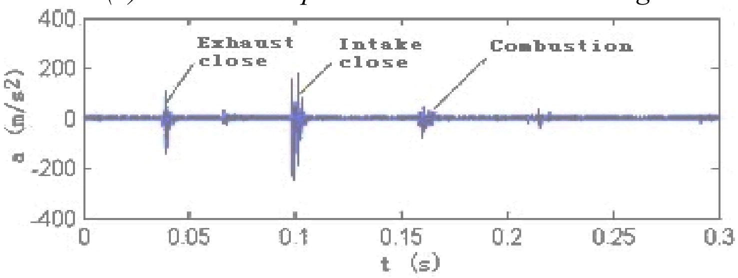
In order to investigate the relationships between the engine surface vibration signals and the working conditions of the engine, the vertical vibration acceleration signals of cylinder cover and the transverse vibration of cylinder block were measured under different engine working conditions. The sampling frequency used in signal processing is 14 kHz, the sampling length is 16383 data points (1.17 s). Then the measured signals were analyzed using the MATLAB software.

CHARACTERISTICS OF ENGINE CYLINDER COVER VIBRATION

While in running condition, there are lots of exciting sources inside the engine and each one contributes some kind of surface vibration with different styles and different quantities. The principal exciting sources of the engine cylinder cover are: the impacts of intake and exhaust valves, the combustion forces (cylinder pressure), and the influences between the cylinders. The vibration response time history (acceleration) of cylinder cover excited by valve impacts and combustion forces are shown in Figure 1.



(a) Vibration response at exhaust valve casing



(b) Vibration response at intake valve

Figure 1 Vibration response of valve impacts and combustion forces.

Vibration Characteristics Induced By Intake and Exhaust Valve Impacts

Valve impact is a kind of broadband exciting force, which mainly affects the vertical vibration of the cylinder cover and has little influence on cylinder block surface vibration [5], [6]. The abnormity of valve clearance is a kind of common malfunction of the valves. Investigation shows that if the valve clearance is larger than the normal clearance, the valve velocity obviously becomes larger while the valve is closing, the shocking force to the cylinder cover also becomes significantly larger, and thus leads to a distinct change of the vibration signal of the cylinder cover. So the vibration signals of cylinder cover while the valve is closing can be used for diagnosing the abnormity of valve clearance.

The time domain response of the cylinder cover surface vibration excited by the valve impacts happens in a very short time period. Experimental investigation shows that the cylinder cover vibration response induced by the valve closing lasts about 10 ms, the relative sampling length used for AR modeling is 140 data points. The computation results show that the data length of 140 points is long enough to obtain proper time series model. The AR spectra (A, B, C, D) and the time histories (a, b, c, d) of cylinder cover vibration signals at different valve clearances are shown in Figure 2.

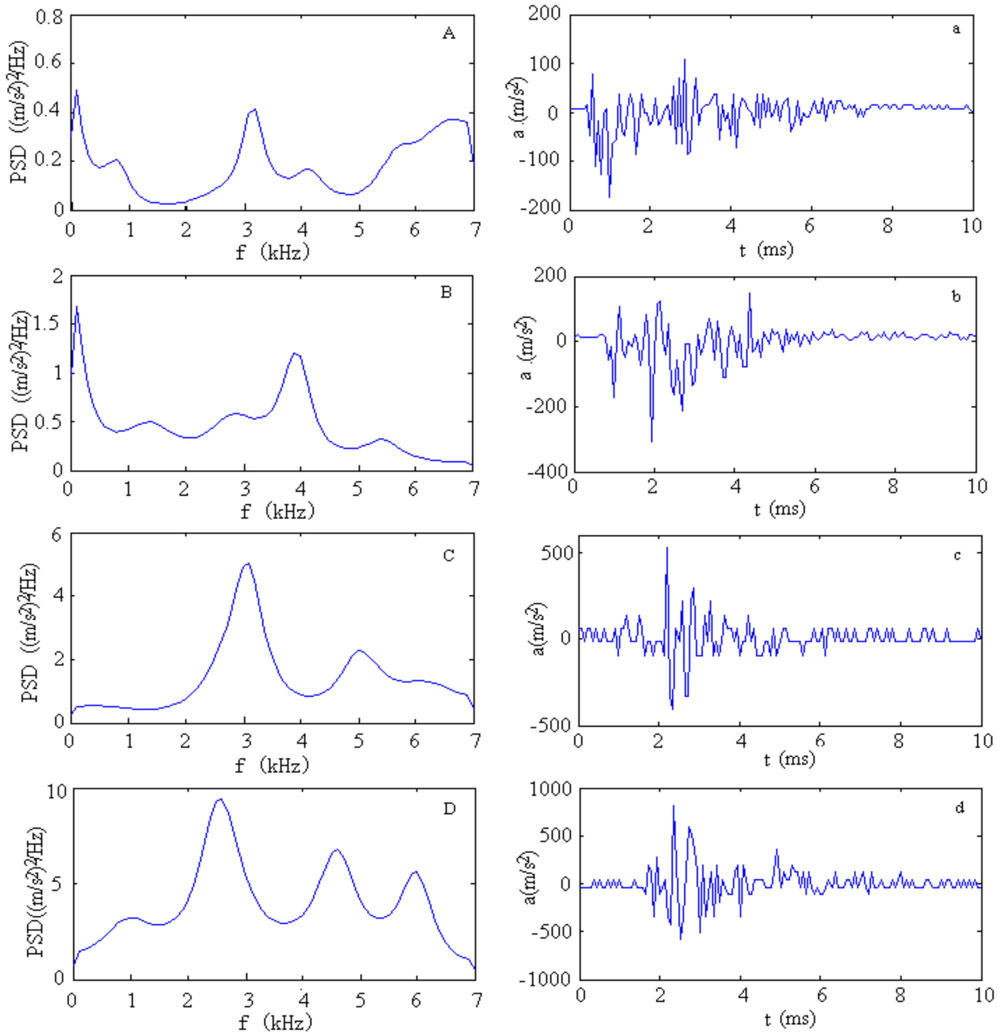


Figure 2 AR spectra (A, B, C, D) and time histories (a, b, c, d) of cylinder cover vibration signals at different valve clearances
(A) Valve clearance 0.2 mm; (B) Valve clearance 0.4 mm;
(C) Valve clearance 0.7 mm; (D) Valve clearance 0.9 mm

As can be seen from Figure 2, the vibration accelerations of cylinder cover increase obviously when the valve clearance becomes larger. Experimental results show that the maximum vibration acceleration is less than 400 m/s^2 when the valve clearance is normal (0.4 mm).

Figure 2 also shows that the vibration power induced by valve impacts is mainly distributed in the frequency bands of 0 ~2 kHz, 2 ~5 kHz, and 5 ~7 kHz. The vibration power of cylinder cover increases significantly while the clearance becomes large, and more vibration powers are distributed in high frequency band. If the total power of the vibration signal is P, and the power in some frequency span is P_s, then the ratio of power in this frequency span R_P is

$$P_S = \sum_{k=1}^{K} PSD(k) \cdot \Delta f \tag{8}$$

$$R_P = P_S / P \tag{9}$$

Where K is the number of spectrum lines in the frequency span, PSD(k) is the value of power spectral density at number k, and Δf is the frequency resolution. The comparison of vibration power and the ratio of power R_P at different valve clearances are listed in Table 1.

Table 1 Comparison of vibration power and R_P *at different valve clearances*

Valve Clearance	Total Power $(m/s^2)^2$	Band 1 Power 0~2kHz	R_P 1	Band 2 Power 2~5kHz	R_P 2	Band 3 Power 5~7kHz	R_P 3
0.2 mm	11.816	2.339	0.198	4.374	0.370	5.103	0.432
0.4 mm	29.676	8.539	0.288	17.450	0.588	3.687	0.124
0.7 mm	93.485	8.655	0.093	62.541	0.669	22.289	0.238
0.9 mm	266.799	35.493	0.133	166.644	0.625	64.662	0.242

It can be seen from Table 1 that the total vibration power and the vibration power at high frequency increase obviously as the valve clearance becomes large. When the clearance is larger than normal (0.4 mm), more than 85% of the total vibration power is concentrated in the high frequency bands of 2 ~5 kHz and 5 ~7 kHz, and the vibration power in the band of 2 ~5 kHz increases significantly. More than 60% of the total vibration power will be concentrated in the band of 2 ~5 kHz when the valve clearance becomes excessive large. Therefore R_P can be used as a criterion for diagnosing valve malfunctions.

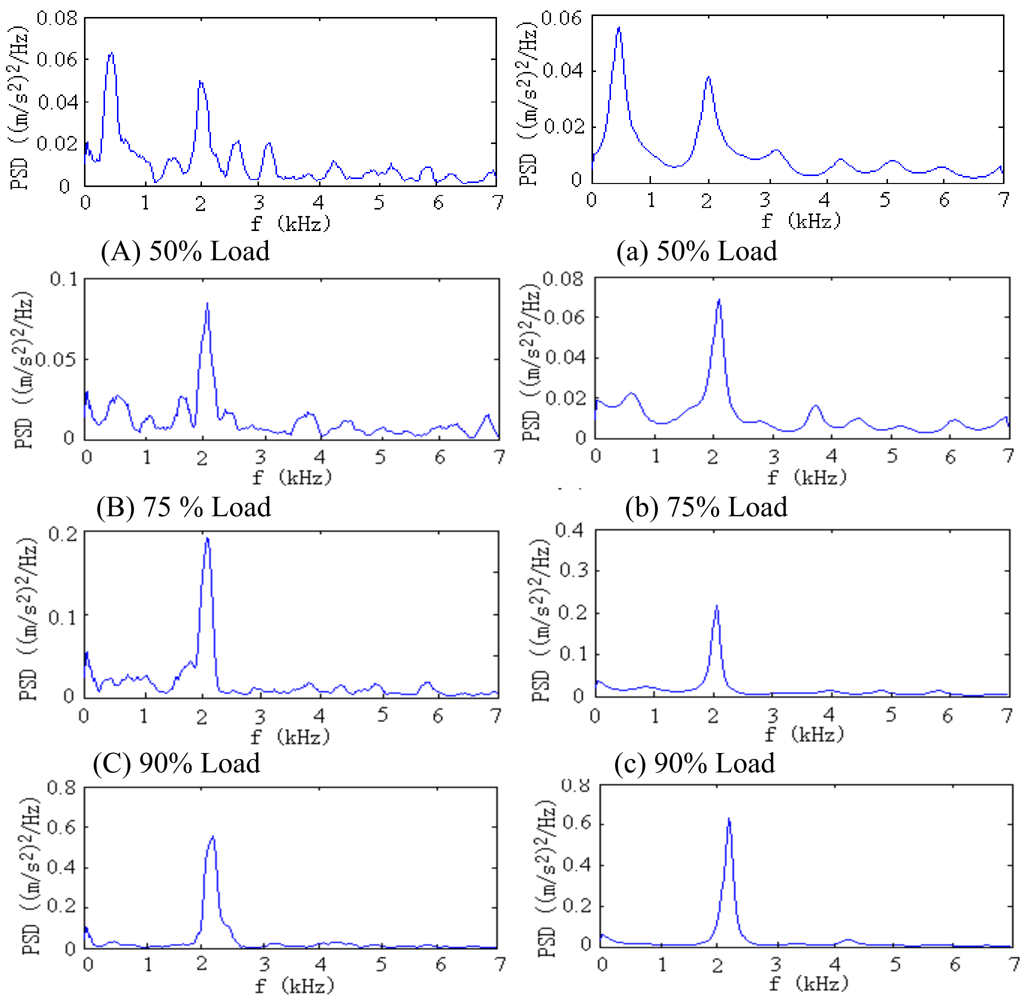
Vibration Characteristics Induced By Combustion Forces

Variations in combustion force (or equivalently, cylinder pressure) mean the change of engine loads, which will also result in the change of cylinder cover vibration. The cylinder cover vibration can be used to detect the changes of cylinder loads [7], [8].

According to the engine injection timing, the sections of vibration signal relating to the combustion period are chosen and analysed at different cylinder loads. Both the Fourier spectra (A, B, C, D) and AR spectra (a, b, c, d) of cylinder cover vibration signals at different cylinder loads are shown in Figure 3. The comparison of total vibration powers is listed in Table 2.

Table 2 Comparison of total vibration power

	Total Vibration Power $(m/s^2)^2$			
Load	50% Pe	75% Pe	90% Pe	100% Pe
Fourier Spectrum	2.7504	3.0047	4.4845	9.0800
AR Spectrum	2.5684	2.8049	4.1474	8.3548



(A) 50% Load (a) 50% Load

(B) 75 % Load (b) 75% Load

(C) 90% Load (c) 90% Load

(D) 100% Load (d) 100% Load

Figure 3 PSD of cylinder cover vibration at different cylinder loads

It can be seen from Figure 3 and Table 2 that the shapes of spectra obtained by the two different methods are quite similar, and the obtained total vibration power values are very close. This indicates that the constructed AR models are correct and suitable.

Figure 3 shows that the cylinder cover vibration power induced by combustion forces is mainly concentrated in the band of 1.5 ~2.5 kHz. The vibration power in 1.5 ~ 2.5 kHz band increases apparently as the cylinder load increases.

Comparing AR spectrum with Fourier spectrum, it is obvious that:

(1) The time series method is superior, for some specific situations, to the Fourier technique: it needn't use any window function and no leakage or side lobe in spectra is introduced; the parametric spectra are usually much smoother and have better frequency resolution capabilities than Fourier spectra, especially when the data record is short. So it is more suitable for the analysis of short data records.

(2) A major problem with ARMA model is selecting or determining the proper model orders. Another problem is that the ARMA model is highly sensitive to the

signal-to-noise ratio. Therefore, all the noises associated with measurement should be avoided or minimized.

SUMMARY

Using vibration monitoring and fault diagnosis can improve the reliability and efficiency of diesel engines. The variations in valve clearance and engine load can be detected by use of cylinder cover vibration signals. The maximum vibration acceleration, the maximum value of PSD, and the ratio of power R_P can be used as the criteria for diesel engine vibration monitoring. It is feasible and effective to detect the variations in valve clearance and engine cylinder load from cylinder cover vibration.

The AR spectrum is smoother than Fourier spectrum, with more accurate frequency locating, especially suitable to short data records, so it is more suitable and effective for diesel engine condition monitoring and fault diagnosis.

ACKNOWLEDGMENTS

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