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Abstract

In this paper, a combustion analysis of a compression ignition (CI) engine operating with biodiesel blends has been experimentally investigated to examine the effects of biodiesel on the combustion process and the characteristics of the engine noise produced by the combustion process. The experiment with different loads and speeds was conducted on a four-cylinder, four-stroke, direct injection and turbocharged diesel engine fuelled with biodiesel (B50 and B100) and normal pure diesel. The signals of vibration, acoustic and the in-cylinder pressure were measured for analysing the engine combustion. In order to examine these signals, a coherent power spectrum analysis method is used to investigate the engine noise signals for locating the frequency band which is closely related to the combustion process. Subsequently, the Wigner-Ville distribution is employed to analyse the energy distribution of engine noise in the time-frequency domain. A band-pass filter based on fractional Fourier transform is developed to extract the main combustion induced noise from the engine noise for comparison. The interim results show that the sound energy level of combustion noise of engine fuelled by biodiesel and its blends is higher than that of engine fuelled by diesel. This is also identical to the variation of in-cylinder pressure. Hence, acoustic signals of the CI engine provide useful features for remote engine condition monitoring and supply fuel monitoring.

Key words: Vibro-acoustics; CI engine; Combustion; Biodiesel; Condition monitoring

1. Introduction

The demand for using petroleum-based fuel has gradually been increased in numerous areas in recent times. However, the resources of this fuel are non-renewable and the remaining global ones are sufficient to meet demand up to 2030 (1). Therefore, a demand to develop alternative fuels which are cheaper and environmentally acceptable (2) has been considered to reduce the dependency on fossil fuel due to the limited resources. It pointed out that biodiesel is one of the most promising renewable, alternative and environmentally fuels (2, 3). Several studies (2-7) have been conducted to research engine performances, combustion and emission characteristics of diesel engines fuelled with biodiesel in comparison with those of petroleum-based diesel.

However, the measurement methods used in most of these researches are inconvenient or costly in real application. For instance, a pressure sensor on the cylinder head of a test engine for in-cylinder pressure measurement has to be fixed. The measurement of the gaseous emissions is based on a costly special emission analyser which includes many gas sensors. Hence, airborne acoustic or vibration measurement is widely used for engine condition monitoring and combustion diagnosis [(8-11)], since airborne acoustic is not only easy to install in real application but also measurement with more comprehensive information in a remote way. Some acoustic-based monitoring researches have been proposed for engine during recent decades. Fujimoto [(14)] investigated the effect of the oil film on the piston slap induced noise. His study found that the oil film formed between the piston skirt and the cylinder reduced the clearance and acted as a damper, thus reducing the piston slap force to some extent. Periede et al. [(10)] carried out similar investigation and concluded that the piston slap noise was also proportional to the cylinder bore dimension. Based on analysing the complex engine noises using coherent power spectrum analysis, Shu and Liang [(9)] concluded that the noise of low-frequency band and high-frequency band are the main resources of machinery noise and combustion noise, respectively. Most of the engine noise is produced by the combustion process, but additional noise [(13)] such as the injection, inlet and exhaust noise due to the injection of fuel and the intake and exhaust valves are all make up a fraction of the overall noise.

In this paper, the combustion noise level and characteristics are investigated according to the in-cylinder pressure variation during the combustion process. The paper is organised as follows, the CI engine combustion process is presented in section 2. Section 3 introduces the experimental test rig and testing procedure. The analysis results and discussion are detailed in section 4. Finally, the conclusions are summarized in section 5.

2. CI engine combustion process

The combustion of engine is a complexity process due to the combustion mechanism. The main parameters used for analysing the characteristics of the combustion process are cylinder pressure, ignition time delay, and heat release rate (HRR) [(14)]. All these parameters are based on the variations of cylinder pressure. Hence, the combustion parameters can be calculated based on the cylinder pressure data. The other important combustion parameters such as combustion duration and intensity can be estimated from the heat release rate variation over an engine cycle. In addition, the HRR can be used for identifying the start of combustion, indicating the ignition delay for different fuels, showing the fraction of fuel burned in the premixed mode and differences in combustion rates of fuels [(15)]. The HRR can be computed from a simplified approach which was derived from the first law of thermodynamics [(16)] as expressed in Equation (1).

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1}\left(\gamma P \frac{dV}{d\theta} + V \frac{dP}{d\theta}\right) \tag{1}$$

where, $dQ/d\theta$ is the heat release rate across the system boundary into the system, P is the in-cylinder gas pressure, V is the in-cylinder volume, γ is the ratio of specific heat whose ranges is from 1.3 to 1.5 for heat release analysis, θ is the crank angle. Moreover, $P(dV/d\theta)$ is the rate of work transfer done by the system due to system boundary displacement [(17)].

Figure 1 presents the cylinder pressure and HRR of a diesel engine. It indicates that the combustion process can be divided into three distinguishable stages. The first stage is the premixed period where the rate of burning is very high and the combustion time is short (for only a few crank angle degrees) as well as the cylinder pressure rises rapidly. The second stage is the main heat release period corresponding to a period of gradually decreasing HRR and lasting about 30 crank degrees, namely mixing controlled period. The third stage is the late combustion period which corresponds to the tail of heat release diagram in which a small but distinguishable HRR throughout much of the expansion stroke.

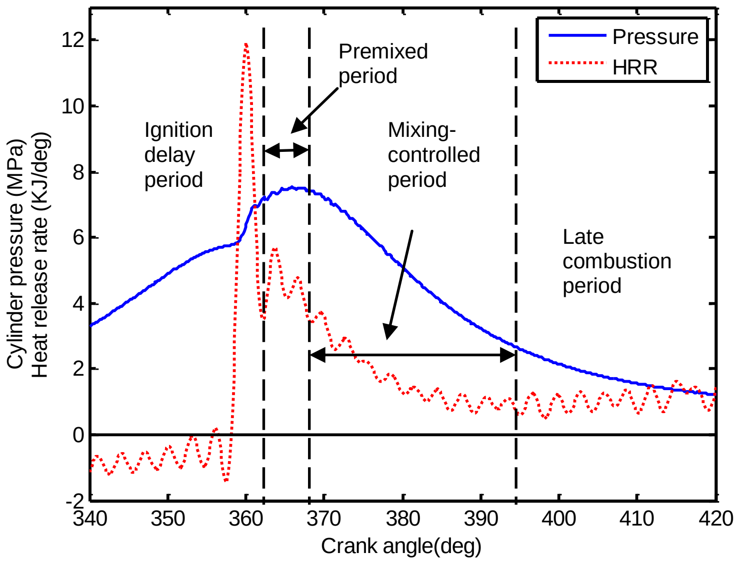

Figure 1. Cylinder pressure and heat release rate of diesel engine

The vibration and acoustic signals of test engine are related to the combustion process. Combustion noise is a complex noise whose level and sound quality strongly depend on the fuel combustion and is one of the main engine noise sources [(18)]. It occurs towards the end of the compression stroke and subsequent expansion stroke. The rapid pressure changes due to the combustion process results in vibration transmitted through engine structures, forms a part of the airborne noise, and contributes to the entire engine noise level. Therefore, the pressure variation in the engine cylinder plays an important role in the analysis of the combustion characteristics, combustion noise of any fuel, and the sound quality related to the combustion parameters [(18)]. On the other hand, the combustion process and engine fuels could be monitored through analysing the sound quality of engine combustion noise.

3. Experimental test rig and testing Procedure

The experiment was conducted on a test rig which consists of diesel engine, charge amplifier for vibration signal, filter and amplifier for sound signal, ADC, load, and PC for saving data and analysis. The diesel engine is a four-cylinder, four-stroke, turbocharged, water-cooled, and direct injection. The load to the engine was provided by a 200 kW AC dynamometer with 4 quadrant regenerative drive with monitoring and absorbing capability for both steady and transient conditions. The schematic diagram of test rig and engine specifications is shown in Figure 2 and Table 1, respectively.

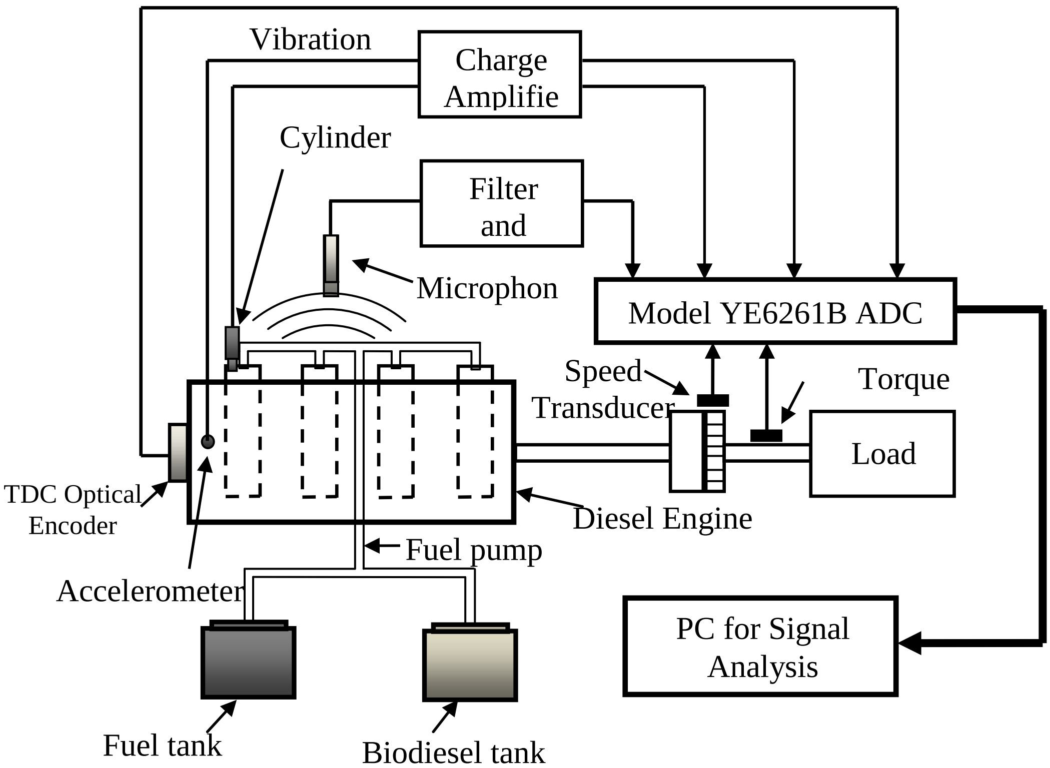

Figure 2. Schematic diagram of test rig

Table 1. Specifications of the test engine

Type of engine	Turbocharged diesel engine
Number of cylinders	4
Bore	103mm
Stroke	132mm
Compression ratio	18.3
Injection system	Direct injection
Displacement	4.399 litre
Cooling system	Water
Maximum power	74.2 kw @ 2200 rpm

In the experiment, the test engine was tested with different fuels and operated at different speeds and loads. Concretely, engine was fuelled with rapeseed oil, its blends involved B50 and B100, and pure diesel at the constant speeds of 900rpm, 1100rpm and 1300rpm. B50 is a mixture of 50% rapeseed oil and 50% diesel whilst B100 is 100% rapeseed oil. The loads were varied from 0 to 420Nm with an interval of 105Nm at each constant speed. The details of the operating conditions are listed in Table 2.

Table 2. Operating conditions

Fuel	Speed (rpm)	Load (Nm)
Pure Diesel	900 - 1300	105 – 420
B50	900 - 1300	105 – 420
B100	900 - 1300	105 – 420

4. Results and Discussion

4.1 Cylinder pressure

Cylinder pressure versus crank angle over the compression and expansion strokes of the engine running cycle can be used to obtain quantitative information on the progress of combustion [19]. Figure 3 shows the cylinder pressures versus crank angle for different fuels (Diesel, B50, and B100), engine loads (105Nm, 210Nm, 315Nm and 420Nm) and at the constant engine speed of 1100rpm. It can be seen that the peak cylinder pressure was higher for rapeseed oil biodiesel at all tests. This is due to the high oxygen content of biodiesel which contributes to the combustion process [5]. The fuels achieve complete combustion resulting in a higher in-cylinder pressure. Moreover, the higher viscosity of biodiesel can enhance fuel spray penetration and thus improving air-fuel mixing [17]. Conversely, a higher viscosity of biodiesel can also lead to bad fuel injection atomization. The peak cylinder pressure of B50 is slightly higher than that of B100, especially in high load. This is possibly because of the higher viscosity of B100. The viscosity of the biodiesel is increased with the increase of biodiesel percentage in the blends. However, the higher viscosity leads to the decrease of combustion efficiency due to the bad fuel injection atomization [2].

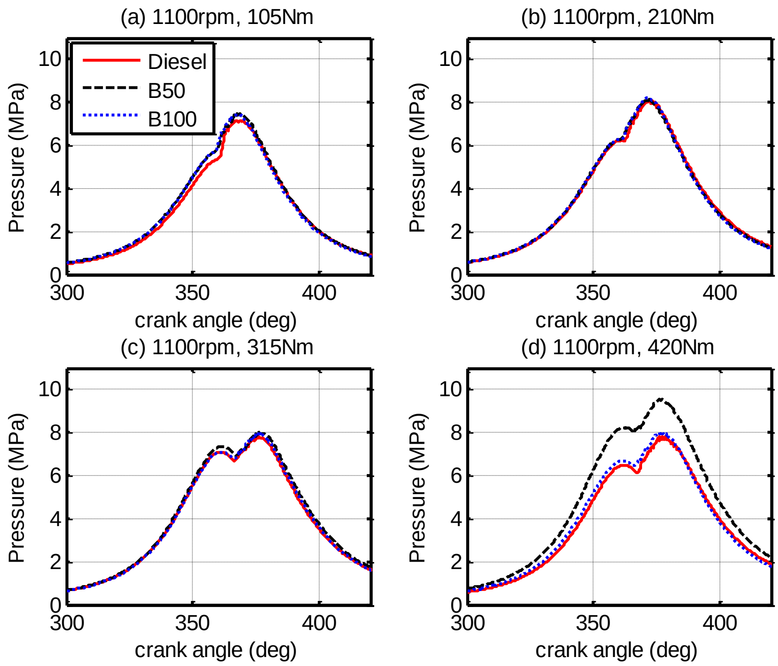

Figure 3. Cylinder pressure at speed of 1100rpm and under different loads

4.2 Analysis of vibration and noise signals

4.2.1 Coherent power spectrum analysis

For analysing the correlation of the engine noise, engine vibration, and combustion process, the coherent power spectrum (CPS) analysis is employed to recognise the noise sources that are related to the combustion process. The input of CPS can be defined as the product of coherent function and input power spectrum [9],

$$G_{yx}(f) = \gamma_{xy}^2 G_{xx}(f) \tag{2}$$

and

$$\gamma_{xy}^2 = \frac{|G_{xy}(f)|^2}{G_{xx}(f)G_{yy}(f)} \tag{3}$$

where x can be considered as the input signal, and y denotes the output signal of measurement system. G_{xx} and G_{yy} are the self-power spectrum of x and y, respectively. G_{xy} is the cross power spectrum between the input signal x and output y. γ_{xy}^2 represents the coherent function of input signal x and output signal y.

In this study, the signal which is measured by acceleration sensor located at the cylinder head of the engine is used as the input signal of the CPS analysis while the output signal is the signal measured by microphone. Figure 4 shows the CPS analysis results. It indicates that the vibration of the cylinder makes significant contribution to the engine noise at the frequency band of 2 kHz - 3 kHz. Furthermore, the noise at the frequency around 5 kHz is also related to the vibration of cylinder head. Figure 4 also indicates that the amplitude of the CPS increases together with the increasing of the engine loads and speeds, which lead to the increase of engine noise accordingly.

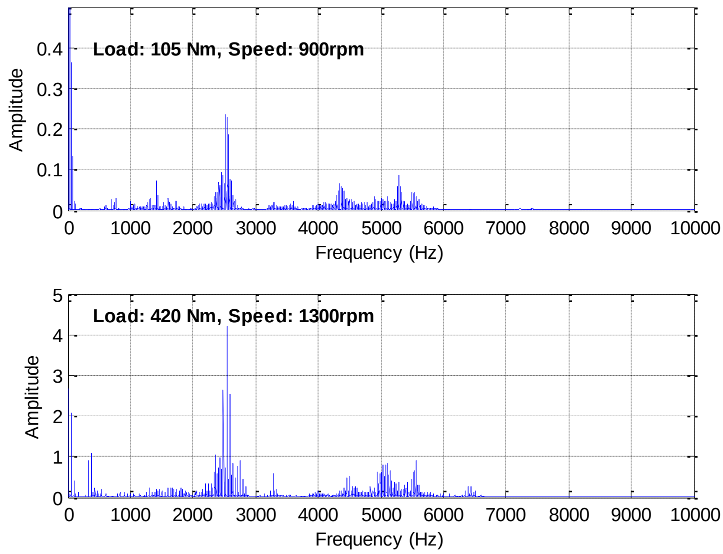

Figure 4. Coherent power spectrum analysis of engine noise

4.2.2 Combustion noise distribution

Wigner-Ville distribution (WVD) which is a simple and effective time-frequency analysis method gives the instantaneous power of the analysed signal at the time t and the frequency ω. The continuous WVD of a general signal $x(t)$ can be defined by Equation (4) [20].

$$W_f(t,\omega) = \int_{-\infty}^{+\infty} x(t+\tau/2)\, x(t-\tau/2) e^{-j\omega\tau} d\tau \tag{4}$$

where $x(t)$ is the real-valued time signal. t and ω are the time and frequency indices, respectively. τ is the redius extending from time t. However, from the Equation (4), the instantaneous power relies on the nature of the signal far away from the time t due to the integral requires an infinite signal length [21]. Hence, the localisation characteristics of the WVD are reduced to some extent. In application, a time window $w(t)$ is often added to the analysed signal before the transformation being carried out.

$$x_w(t) = x(t)w(t) \tag{5}$$

The length of the added window in Equation (5) is much shorter than that of the analytical signal. The window can be able to slide along the time axis so that its centre is always located at time t. This short-timed window capably makes the transformation of capturing the transients with a better resolution. The WVD of windowed signal as shown in equation (5) [22] can be expressed as follow:

$$W_{f_w}(t,\omega) = \frac{1}{2\pi} \int_{-\infty}^{+\infty} W_f(t,\eta)\, W_w(0, \omega-\eta) d\eta \tag{6}$$

where W_f and W_w are the WVD of the analytical signal and the window function, respectively. In fact, the WVD of a signal according to equation (6) is a low-pass smoothed version of the original WVD in frequency domain [20].

Figure 5 shows the WVD of the noise of engine using different fuels at the loads of 105Nm and 420Nm. The length of the window for the WVD is determined based on one complete combustion cycle (720 degree). It can be seen that the main energy of the engine noise is located in the low frequency band which is below 1 kHz at low engine load of 105Nm as shown in Figure 5 (a), (c) and (e). The energy level at the high frequency band (2 kHz-3 kHz) also increases together with the increase of the biodiesel percentage. At the high engine load of 420Nm according to Figure 5 (b), (d) and (f), the main energy of the engine noise is located at the frequency of around 1 kHz and 2 kHz. Especially, the WVD of engine noise at the frequency band of 2 kHz to 3 kHz gives clear indications of noise energy for different fuels. Moreover, considering the cylinder pressures shown in Figure 3, the higher cylinder pressure produces the higher noise energy level at the frequency band of 2 kHz-3 kHz.

According to the analysis of the relationship of engine vibration and engine noise in section 4.2.1, the vibration produced by the combustion process creates the considerable contribution to the engine noise at the frequency band of 2 kHz - 3 kHz. Therefore, it seems that the engine noise at the frequency band of 2 kHz - 3 kHz can be used for monitoring the combustion process and the engine supply fuels.

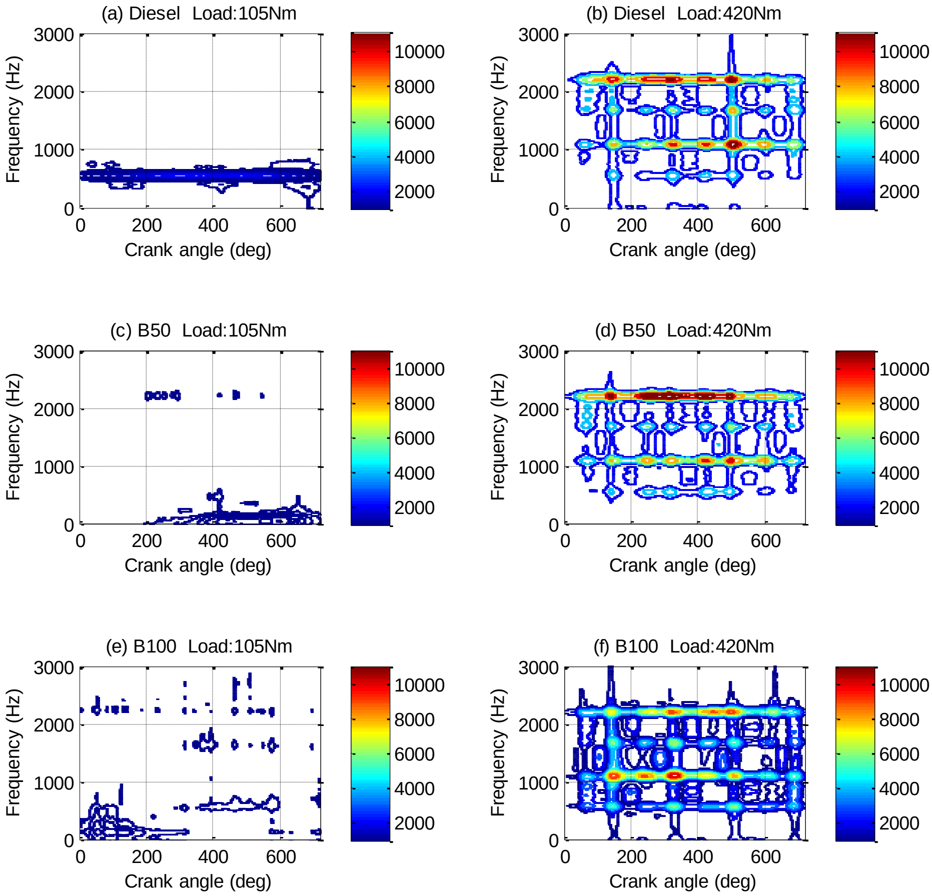

Figure 5. WVD of engine noise at loads of 105Nm and 420Nm

4.2.3 Combustion noise extraction

For extraction the combustion noise, a filter based on fractional Fourier transform (FrFT) is designed to process the engine noise. The FrFT is a time-frequency distribution extended from the classical Fourier transform (FT). FrFT is a technique which rotates the conventional time-frequency domains to a specified angle α which allows the designer to have a better flexibility to extract the useful signals or decrease noisy signals [23]. The α^{th} order FrFT of a signal $x(t)$ can be defined as [24]

$$X_p(u) = \int_{-\infty}^{+\infty} x(t) K_p\,(t,u)dt \tag{7}$$

$$\alpha = p\frac{\pi}{2} \tag{8}$$

where the kernel function $K_p(t,u)$ can be expressed as

$$K_p(t,u)=\begin{cases}\sqrt{\frac{1-jcot\alpha}{2}}e^{j(\frac{1}{2}u^2cot\alpha+\frac{1}{2}t^2cot\alpha-utcsc\alpha)}, & \alpha\neq n\pi\\ \delta(t-u), & \alpha=2n\pi\\ \delta(t+u), & \alpha=(2n+1)\pi\end{cases} \quad (9)$$

It is found that the FrFT of the signal $x(t)$ at the order $\alpha = 0$ is the input signal itself $x(t)$; at the order $\alpha = \pi/2$ corresponds to the FT of the signal $x(t)$; and at the order $\alpha = \pi$ is the inverse of the signal $x(-t)$.

The filtering process based on FrFT is described as follows.

1. Calculate the FrFT of the analytical signal according to the variation of the order α. The range of p is chosen from 0.5 to 1.5. The energy distribution of the analytical signal on the fractional plane can be obtained as shown in Figure 6 (a).
2. Search the peak point (p_0, u_0) on the fractional plane. Then, calculate the FrFT of the analytical signal at the order of $\alpha_0 = p_0\frac{\pi}{2}$.
3. Filter the analytical signal in the fractional domain using a band-pass filter. The centre frequency of the filter is based on the localization of the peak point u_0 on the fractional plane. The energy distribution of the filtered signal on the fractional plane is shown in Figure 6 (b).
4. Calculate the FrFT of the filtered signal at the order of $a_0 = -p_0\frac{\pi}{2}$ to obtain the extracted signal in the time domain.

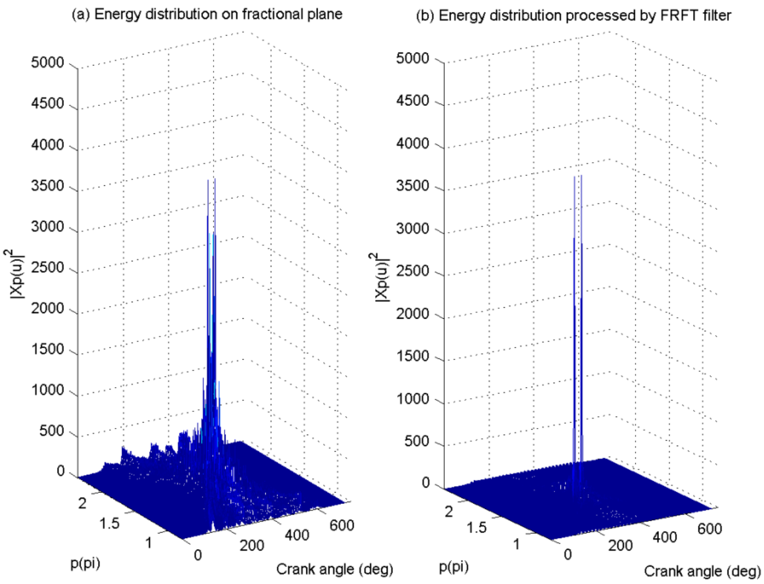

Figure 6. Energy distributions on the fractional plane

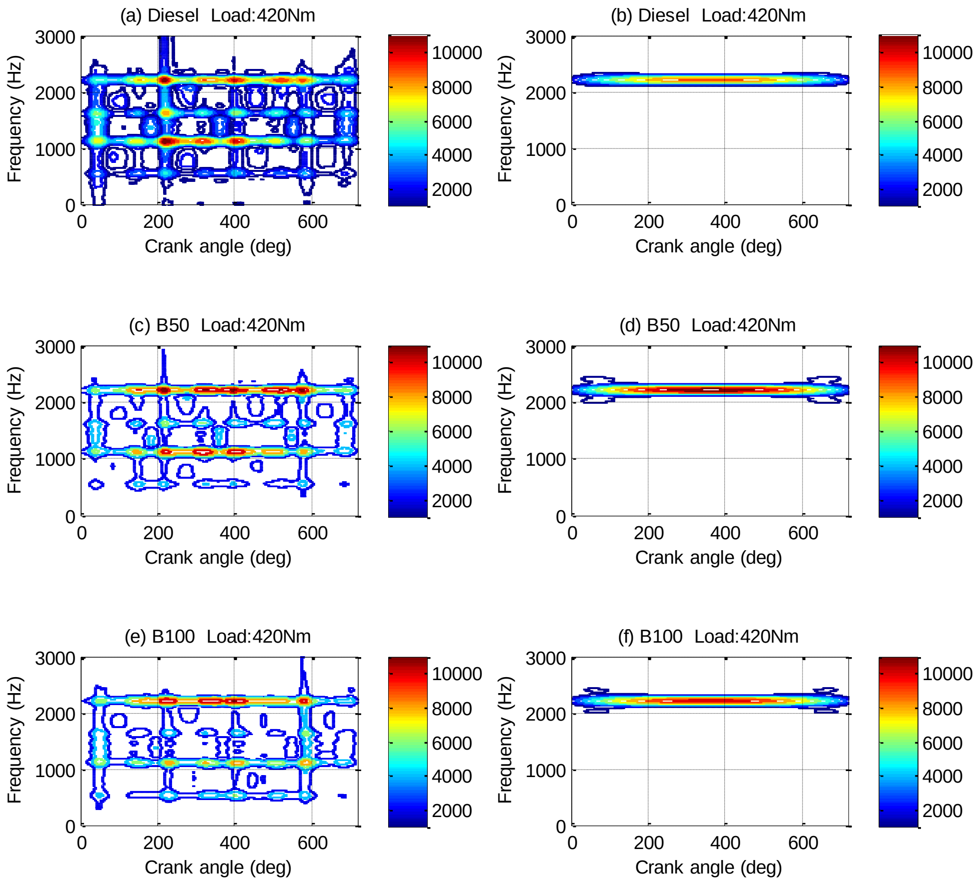

Figure 7. WVD of engine noise processed by FRFT filter

The WVD of the combustion noise which extracts by the FRFT band-pass filter is shown in Figure 7 (b), (d) and (f). For the comparison, Figure 7 (a), (c) and (e) show the WVD of the unfiltered engine noise at the load of 420Nm for different fuels of diesel, B50 and B100. It can be seen that the combustion noise is extracted from the total engine noise and its energy level on the time-frequency plane reflects the differences of fuels.

Figure 8 shows the variations of sound pressure level (SPL) of the combustion noise under different engine operating condition of different fuels (diesel, B50 and B100), different speeds (900rpm and 1300rpm), and different loads (105Nm 210Nm, 315Nm and 420Nm). It shows that the SPL is increased together with the increase of speeds and loads under all engine fuelled conditions. The SPLs of engine fuelled by biodiesel and its blends (B50 and B100) are slightly higher than those of fuelled by diesel. However, the SPL of engine fuelled by B50 is higher than that of fuelled by B100 under all test engine speeds and loads. This is because of higher peak pressures and heat release rate obtained from the combustion process when the engine is fuelled by B50 according to the combustion parameters analysis. Therefore, it can be concluded that higher pressure produces higher engine vibration and noise.

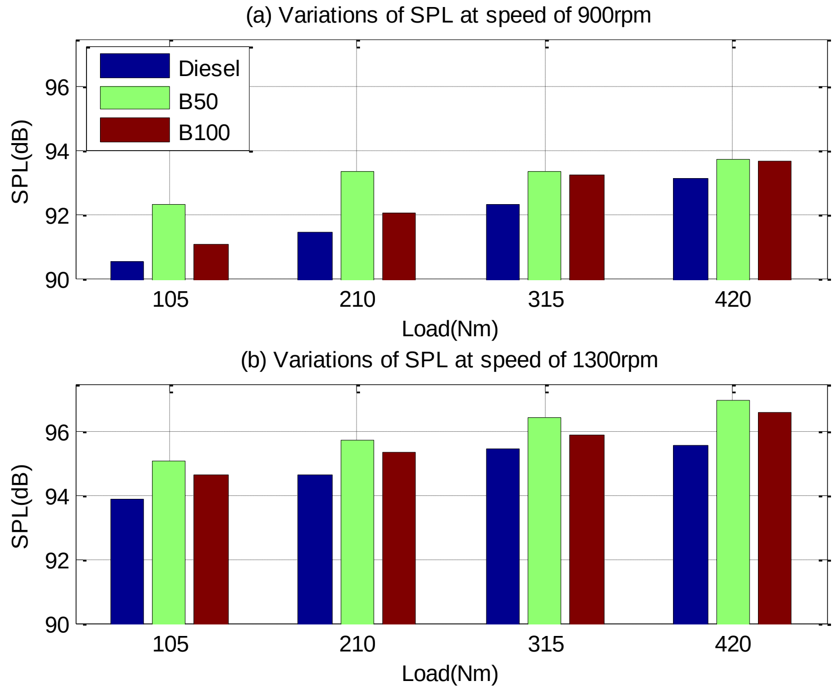

Figure 8. Variations of SPL under different fuels

5. Conclusion

In the this study, an experimental investigation was carried out on the combustion process, vibration, and noise analysis of the CI engine operating with rapeseed biodiesel and its blends under steady state operating conditions. Based on the experimental study, the main results are summarized as follows:

- The peak cylinder pressure of rapeseed oil biodiesel is higher than that of diesel at all tests due to the high contained oxygen of biodiesel which contributes to the combustion process.
- The peak cylinder pressures for both diesel and biodiesel and its blends are increased together with the increase of engine loads and speeds because of that longer ignition delay results in more fuel available for ignition and more energy release during the premixed combustion stage.
- The amplitude of the engine acoustic signals increases according to the increase of engine loads and speeds due to higher engine vibration.
- The SPLs of combustion noise when engine fuelled by biodiesel and its blends are slightly higher than those fuelled by diesel since higher in-cylinder pressure produces higher vibration during the combustion process.

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