Study on Combustion Noise from a Running Diesel Engine Based on Transient Combustion Noise Generation Model

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ABSTRACT: This study investigates the characteristics of combustion noise generation in a diesel engine under running condition using wavelet transformation and transient combustion-noise-generation model. The results show that combustion noise largely contributed to the total engine noise in the early stage of the expansion stroke. Maximum combustion impact energy had a predominant effect on maximum combustion noise power and therefore on maximum engine noise power for each cycle. The combustion noise power exponentially decayed with time. The duration of combustion noise depends mainly on the maximum combustion noise power, which is controlled by the maximum combustion impact energy and transmission-radiation rate.

KEY WORDS: heat engine, noise. Diesel engine, Combustion noise, Combustion noise generation model, Wavelet transformation [A1]

1. Introduction

Engine noise is mainly categorized into three noises: combustion noise, combustion-induced mechanical noise and mechanical noise. Indeed, diesel engines produce knocking such as impulsive and irregular noises, which are unpleasant (1-3). From a commercial viewpoint, this may be the main drawback of diesel engines since sound quality has become an important value. Consequently, car manufacturers have devoted significant efforts to mitigating engine noise in order to comply with regulations and meet customer demand (3-4). The combustion noise is generated by the combustion impact excited in the combustion chamber. The combustion-induced mechanical noise is generated by impacts of mechanical parts induced by combustion. A typical example is piston slap noise. The mechanical noise is generated by impacts of mechanical parts such as gears and valves unrelated to combustion (5,13). The vibration transmits through the engine structure from each excitation point to the outer surfaces, resulting in noise radiation from them. Combustion noise may be reduced actively by both hardware and settings optimization to control combustion noise characteristics. The use of active actions to control combustion noise usually has a negative impact on combustion performance, drive ability and pollutant emissions. Split injection, which is fuel injection consisting of one pilot and one main injection, is one of the most effective strategies for reducing the maximum rate of rise of in-cylinder pressure and thus for controlling combustion noise in automotive direct-injection diesel engines (6-8). An analysis of in-cylinder pressure, based on an innovative decomposition of the signal and improved spectrofilter is proposed to predict and separate engine combustion and mechanical noise (9-10). Besides the effective strategy of split injection, other parameters are significant for controlling combustion noise, such as diesel-fuel injection timing and pressure, which also affect the fuel burning rate (9,11-12). Miyamoto et al. (12) researched the effect of diesel-fuel injection timing on the maximum rate of in-cylinder pressure rise for a diesel engine with hydrogen addition to the intake air. The result showed that in the case of diesel-fuel injection timing of 4-6 degree ATDC and 10 vol. % hydrogen fraction, the maximum rate of in-cylinder pressure rise was lower than the allowable limit, i.e., 0.5 MPa/deg.

In order to reduce the engine noise effectively, it is necessary to clarify the generation mechanism of vibration and noise in the structure. Haddad et al. (13) estimated the radiation efficiency of several diesel engine components in the reverberation room. Chiatti et al. (14) made the connection between cylinder pressure signal and the rate of pressure rise by analyzing the cylinder pressure signal. Kojima et al. (15-18) analyzed combustion noise with a steady-state noise-generation model, in which the combustion-noise power was proportional to the combustion-impact power. Komori et al. (19) experimentally investigated the transient relationship between combustion impact and noise. It was shown that there was a time delay between combustion-impact power peak and combustion-noise power peak. In order to investigate the transient characteristics of the combustion noise, Miura et al. (20) proposed a transient noise-generation model for the combustion impact in a single-cylinder diesel engine. Some researches have also applied the transient combustion noise-generation model but mostly for single explosion engines excited by a single explosion (20-22). Sugimoto et al. (22) and Sera et al. (22) investigated noise-generation characteristics in a single explosion four-cylinder engine for different noise-radiation parts and different excited cylinders.

The engine noise characteristics of a running engine are different from those of a single explosion engine. A single explosion engine generates only combustion noise, while the noise from a running engine contains combustion noise, combustion-induced mechanical noise and mechanical noise. The main objective of this study is to improve understanding of combustion noise generation in a running...
We applied the transient combustion-noise-generation model to a single-cylinder direct-fuel injection diesel engine operated in different working conditions, considering the accumulation and the decay of vibration energy in the engine structure excited periodically by the combustion impact in the cylinder.

2. Experimental apparatus and method

The engine used for this experiment is a naturally aspirated single-cylinder water cooled four stroke cycle diesel engine. Diesel fuel is injected directly into the combustion chamber with common-rail fuel-injection equipment. Table 1 lists the engine specifications. A schematic of the experimental apparatus is illustrated in Fig. 1.

Table 1 Engine specifications

| Engine type                          | Single cylinder four stroke cycle DI diesel |
|--------------------------------------|---------------------------------------------|
| Bore x stroke                        | 85 mm x 97.1 mm                              |
| Displacement                         | 551 cm³                                     |
| Compression ratio                    | 16.7                                        |
| Charged method                       | Naturally aspirated                          |
| Piston cavity shape                  | Re-entrant type                              |
| Diesel fuel injection system         | Common rail                                  |
| Nozzle                               | Minisac hole nozzle                          |
| Hole diameter x number of fuel injector | φ 0.139 mm x 6               |

The sound pressure, in-cylinder pressure and TDC mark were recorded on a personal computer through an A/D converter (CONTEC ATP-32F) with a sampling frequency of 40960 Hz. The analyzed frequency range was from 500 Hz to 10000 Hz and the analyzed frequencies in this study are all center frequencies in the 1/3 octave band. The experimental data were analyzed with 100-cycle average with wavelet transformation and discussed using a transient combustion noise-generation model.

All the calculated terms in this study was calculated using the following process:

Experimental data of sound pressure was transferred through wavelet transformation to get sound pressure level in time and frequency domains.

Experimental data of sound pressure and in-cylinder pressure were first transferred through wavelet transformation to get data in time and frequency domains. After wavelet transformation, the data continued to be transferred through transient combustion noise generation model to get combustion impact power level, combustion impact energy, engine noise power, decay rate and transmission-radiation rate. Wavelet transformation, condition for wavelet transformation, transient noise generation model and all the above mentioned terms will be defined in the following sections.

3. Wavelet transformation

Wavelet transformation technique was employed in this study. Use of simpler Fourier transformation analysis will give the results which have only frequency domain and lack of time domain. Use of wavelet transformation technique will give the results in both frequency domain and time domain, so that we can understand the engine noise not only at every crank angle but also at every frequency.

Wavelet transformation used in this study is continuous wavelet transformation (CWT) with complex mother wavelet.

The continuous wavelet transformation is defined as the sum over all time of the signal multiplied by scaled, shifted versions of the wavelet function \( \psi(t) \).

\[
C(a,b) = \frac{1}{a} \int f(t) \psi \left( \frac{t-b}{a} \right) dt
\]

where:

- \( C(a,b) \): CWT coefficients
- \( f(t) \): Signal to be analyzed
- \( \psi(t) \): Mother wavelet
- \( a \): Scaling factor
- \( b \): Shifting factor

Scaling factor determine the degree to which the wavelet is compressed or stretched. Low scale values compress the wavelet and correlate better with high frequencies. The low scale CWT coefficients represent the fine-scale features in the input signal vector. High scale values stretch the wavelet and correlate better.
with the low frequency content of the signal. The high scale CWT coefficients represent the coarse-scale features in the input signal.

A complex Morlet wavelet is defined as

\[ \psi(t) = \frac{1}{\sqrt{2\pi f_b}} e^{2\pi i f_c t} e^{-t^2} \]  

(2)

depending on two parameters:

- \( f_b \): Bandwidth parameter
- \( f_c \): Wavelet center frequency

Bandwidth parameter \( f_b = 3 \) and wavelet center frequency \( f_c = 1.6 \) are used and Complex Morlet wavelet cmor3-1.6 is shown in Fig. 2.

### 4. Transient combustion noise-generation model

Transient combustion noise generation model is a model which helps us to improve understanding about combustion noise generation mechanism and engine structure transmission-radiation path through which combustion noise, combustion induced mechanical noise and mechanical noise radiated. Fig. 3 shows a schematic of the transient combustion noise generation model.

Transient combustion noise generation model applied for one cycle consists of three processes; the accumulation process of vibration energy excited by the combustion impact, decay process of vibration energy in the structure and radiation process of combustion noise from the structure surface. These three processes are defined as follow:

The first process is the accumulation of vibration energy in the engine structure. In this process the combustion impact power \( W_{ci}(f,t) \) is generated in the cylinder, supplies power to the engine structure and accumulates as the vibration energy \( E_{ci}(f,t) \) in the engine structure.

\[
W_{ci}(f,t) = \frac{p_{ci}(f,t)^2}{\rho c} A_{ci}\]

(3)

where \( p_{ci} \) is in-cylinder pressure, \( \rho_c \) is the impedance of medium, and \( A_{ci} \) is the inner surface area of the cylinder. The in-cylinder pressure was analyzed both in time and frequency domains using wavelet transformation.

The accumulation of vibration energy without the decay process is defined as

\[
E_{ci}(f,t) = \eta_v(f) \int_0^t W_{ci}(f,t) dt \]

(4)

where \( \eta_v(f) \) is a vibration-transmission efficiency, which is the ratio of the total energy supplied by the combustion impact generated in the cylinder to the vibration energy near the surface area. This equation can also be expressed as

\[
\frac{dE_{ci}(f,t)}{dt} = \eta_v(f) W_{ci}(f,t) \]

(5)

The decay of vibration energy without the accumulation process is defined as

\[
\frac{d\ln E_{ci}(f,t)}{dt} = -c(f) \]

(6)

where \( c(f) \) is the decay rate, which has the unit of s\(^{-1}\). This equation can also be expressed as

\[
\frac{dE_{ci}(f,t)}{dt} = -c(f) E_{ci}(f,t) \]

(7)

At the same time with the first and the second processes, the third process occurs with the radiation of combustion noise. The radiation of combustion noise is defined as

\[
W_{cn}(f,t) = b(f) E_{ci}(f,t) \]

(8)

where \( b(f) \) is the acoustic conversion coefficient, which has the unit of s\(^{-1}\). The combustion noise power is defined as

\[
W_{cn}(f,t) = \int_{A_{in}} \frac{p_{cn}(f,t)^2}{\rho c} dA \]

(9)

where \( p_{cn} \) is the sound pressure and \( A_{in} \) is the diffusion area of
combustion noise. The sound pressure was analyzed both in time and frequency domains using wavelet transformation.

In actual cases, the accumulation and decay processes occur at the same time. The summation of equations 5 and 7 give the following equation

$$\frac{dE_c}{dt}(f, t) = \eta(f)W_{ci}(f, t) - c(f)E_c(f, t)$$

Equations 8 and 10 give the following equation

$$\frac{dW_{cm}}{dt}(f, t) = b(f)\eta(f)W_{ci}(f, t) - c(f)W_{cm}(f, t)$$

Since the combustion impact occurs in a finite amount of time, after the completion of the combustion impact input, i.e., $W_{ci}(f, t) = 0$, only the decay process occurs and Eq. 11 is transformed to calculate the decay rate $c(f)$ as

$$c(f) = -\frac{d\ln W_{cm}(f, t)}{dt}$$ (12)

The transmission-radiation rate $b(f)\eta(f)$ is obtained through the integration of Eq. 11 from the start of combustion until an arbitrary time as

$$b(f)\eta(f) = \frac{W_{cm}(f, t) - W_{cm}(f, 0) + \int_0^t W_{ci}(f, t) dt}{\int_0^t W_{ci}(f, t) dt}$$

where $W_{cm}(f, 0)$ is 0 for single explosion engines.

This transient combustion noise generation model can be applied to a running engine only if the combustion-induced mechanical noise and mechanical noise are much smaller than the combustion noise.

5. Results and discussion

First, we studied the characteristics of combustion impact generated in the combustion chamber. Fig. 4 shows the combustion impact power levels, which were calculated as $10\log(W/Y/W_0)$ from in-cylinder pressure through wavelet transformation and transient combustion noise generation model with the reference value $W_0 = 10^{12}$ W, for different fuel injection timings, IT, at different engine speeds of 1500 and 1000 rpm and different engine loads. The engine load is expressed by the indicated mean effective pressure, IMEP. As shown in Fig. 4, the combustion impact power of all working conditions had greater value for low frequency range and continued over a longer crank angle in comparison with that for high frequency range. The combustion impact power decreased by delaying the fuel injection timing for all frequency ranges. For the same fuel injection timing, the combustion impact power for 1500 rpm continued for a longer crank angle at all frequency ranges in comparison with that for 1000 rpm. The combustion impact power also decreased by decreasing engine load for all frequency ranges.

Next, we studied the characteristics of engine noise. Figs. 5 to 7 show the sound pressure levels of engine noise from the thrust side and anti-thrust side for different fuel injection timings at 1500 and 1000 rpm and different engine loads at -12 deg. ATDC. The sound pressure level of engine noise in all conditions decreased with the delay of fuel injection timing and decrease in engine load. There were differences in frequency characteristics and sound pressure levels of engine noise from the thrust side and anti-thrust side. This was possibly due to differences in combustion impact transmission, combustion noise radiation, combustion-induced mechanical noise and mechanical noise from both sides of the engine. As the fuel injection timing was delayed, the crank angle for start of engine noise rise were also delayed both for 1500 and 1000 rpm. This is a similar tendency to that shown in Fig. 4. According to the transient combustion noise generation model, the combustion noise attains the maximum level at the end of combustion in each cycle because the combustion impact energy accumulated in the engine structure attains the maximum level at the end of combustion. This fact suggests that the engine noise in the early stage of the expansion stroke in this experiment could be greatly affected by combustion noise. The same results can be obtained under all the experimental conditions. In a comparison of the 1500 and 1000 rpm conditions, there were differences not only in the frequency range but also in the sound pressure level when changing the fuel injection timing and engine load.

Frequency characteristics of A-weighted sound pressure level through FFT analysis are shown in Fig. 8 for different fuel injection timings at 1500 and 1000 rpm and different engine loads at -12 deg. ATDC. Largest contribution frequencies to the overall sound pressure level for all working conditions are 830 Hz, 2100 Hz, 2650 Hz and 3300 Hz as can be seen in Fig. 8. As explained in the previous section, the analyzed frequency range are from 500 Hz to 10000 Hz in this study, but the above four frequencies are chosen as the most important frequencies at which we can make clear the effect of combustion impact on engine noise.

In order to elucidate the effect of combustion impact on engine noise, the relationship between the maximum combustion impact energy and maximum engine noise power was studied. The maximum combustion impact energy in one cycle is defined as the accumulated energy by combustion impact during combustion period $t_b$ which is time from the start of combustion until the end of combustion.

$$E_{ci,max} = \int_0^{t_b} W_{ci} dt$$ (14)

The maximum combustion impact energies are plotted against the fuel injection timing at different frequencies with the engine speed of 1500 rpm in Fig. 9. The analyzed frequencies here are the large contribution frequencies to the overall A-weighted sound pressure level. The available heat produced by diesel-fuel combustion was kept constantly at 1.0 ± 0.05 kJ/cycle even if the fuel injection timing was varied. As shown in Fig. 9, as the injection timing was delayed from -23 deg. ATDC to 2 deg. ATDC, the combustion impact energy first decreased, then increased and decreased again. At higher frequencies of 2650 Hz and 3300 Hz, combustion impact energies at 2 deg. ATDC decreased dramatically due to lower combustion impact power at the same frequencies as shown in Fig. 4. The maximum engine noise power, which is the average value of the maximum engine noise power for 100 cycles from the thrust side and anti-thrust side, is plotted against the fuel injection timing at different
Fig. 4 Combustion impact power level for different fuel injection timings at different engine speeds and loads

(a) Thrust side                                        (b) Anti-thrust side

Fig. 5 Engine noise for thrust side and anti-thrust side for different fuel injection timings at 1500 rpm

(a) Thrust side                                      (b) Anti-thrust side
Fig. 6 Engine noise for thrust side and anti-thrust side for different fuel injection timings at 1000 rpm

Fig. 7 Engine noise for thrust side and anti-thrust side at different engine loads for the injection timing of -12 deg. ATDC at 1500 rpm
Fig. 8 Frequency characteristics of A-weighted sound pressure level for different working conditions.

Fig. 9 Dependencies of maximum combustion impact energy on fuel injection timing at different frequencies at 1500 rpm.

Fig. 10 Dependencies of maximum engine noise power on fuel injection timing at different frequencies at 1500 rpm.
frequencies with the engine speed of 1500 rpm in Fig. 10. A comparison of Figs. 9 and 10 shows that the maximum engine noise power tended to have a close relationship with that of the combustion impact energy for the same frequency in the aspect of dependence on fuel injection timing.

Table 2 Coefficients of correlation between maximum engine noise power and maximum combustion impact energy for different frequencies

| Frequency [Hz] | Correlation coefficient |
|---------------|------------------------|
| 830           | 0.925                  |
| 2100          | 0.752                  |
| 2650          | 0.972                  |
| 3300          | 0.972                  |

The maximum engine noise power is plotted against the maximum combustion impact energy at different frequencies in Fig.11. The analyzed frequencies here are also the largest contribution frequencies to the overall A-weighted sound pressure level. The correlation coefficient between the maximum engine noise power and the maximum combustion impact energy at different frequencies are listed in Table 2. Generally speaking, linearity correlation requires at least 0.8 of correlation coefficient in statistical sense. For four analyzed frequencies, only at 2100 Hz the correlation coefficient is 0.752, which is a little bit smaller than the required value of 0.8 but for the other three frequencies the correlation coefficients almost reach 1.0. This fact proves that linear correlation between the maximum engine noise power and the maximum combustion impact energy is achievable at all analyzed frequencies independent of fuel injection timing from -12 deg. ATDC to 2 deg. ATDC, engine speed and also engine load. 

In the wavelet transformation, the lower the frequency is, the longer the width of the wavelet is. It means that when low frequency is analyzed, we need more data number to calculate. Moreover, as discussed above, when the decay process is evaluated, the number of data must be increased in the later period from combustion completion, where the combustion noise is no longer predominantly large enough. So in order to reduce errors made by combustion-induced mechanical noise and mechanical noise as well as in order to increase the accuracy of the analysis, 1050 Hz as the lowest frequency is analyzed in the values of the decay rate and transmission-radiation rate shown in Fig. 12. 

As shown in Fig. 12, the decay rate was similar from 1000 Hz to around 5000 Hz and slightly increased with frequency. The decay rate seemed to have the smallest value at 2100 Hz, and vibration energy seemed the most difficult to decrease at this frequency. The transmission-radiation rate tended to increase with frequency from 1000 Hz to around 5000 Hz. This tendency was similar to that for single explosion engines. 

From Fig. 11, because there are linear correlations between maximum engine noise powers and maximum combustion impact energies at all analyzed frequency in the aspect of dependence on different fuel injection timings, engine speeds and also engine loads, as a result combustion noise largely contributed to the total of engine noise in the early stage of the expansion stroke. So, the value of maximum combustion noise power depends on the value of maximum combustion impact energies and the value of transmission-radiation rate as the slope of approximation lines, which varies according to frequency. If the value of maximum combustion impact energy increases, the value of maximum combustion noise power also increases. When the frequency increases, the values of maximum combustion impact energies decrease and make the values of maximum combustion noise power decrease too. But the values of the transmission-radiation rate as the slope of approximation lines shown in Fig. 12 tend to increase with the increase in frequency, so combustion impact energies have higher conversion efficiency to combustion noise power in high frequency.

Combustion noise decrease characteristics depend on maximum combustion noise power and decay rate. Maximum combustion noise power characteristics were explained above. After the combustion is
Fig. 11 Relationships between maximum combustion impact energy and maximum of engine noise power for different working conditions at different frequencies

completed, the decay process becomes the main process and combustion noise decrease characteristics depend on the decay rate. If the decay rate increases, vibration energy will decrease faster, resulting in a faster decrease in combustion noise.

6. Conclusions

We studied noise-generation characteristics in a running single-cylinder diesel engine using a transient combustion noise-generation model for different noise-radiation sides, different fuel injection timings, different engine speeds and different engine...
loads. The main conclusions are as follows:

1. The transient combustion noise-generation model for a single explosion engine can be applied to investigate combustion noise generation from a running engine in the early stage of the expansion stroke, where the combustion noise generated by combustion impact largely contributes to the total engine noise in the frequency range from 500 Hz to around 5000 Hz.

2. The combustion noise peak in one cycle depends mainly on the maximum combustion impact energy and transmission-radiation rate due to the large contribution of combustion noise. The engine noise decrease characteristic depends mainly on the combustion noise peak and decay rate.

3. The engine structure has its own decay and transmission-radiation characteristic of the combustion impact. Transmission-radiation rate increase with frequency. The decay rate is almost independent of frequency.

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