

Combustion Noise Reduction with High Thermal Efficiency by the Control of Multiple Fuel Injections in Premixed Diesel Engines

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ABSTRACT

Premixed diesel combustion is effective for high thermal efficiency and reductions of NO_x and PM emissions, but a reduction of combustion noise is necessary for medium-high load engine operation. The control of the fuel injection has become more accurate because of the technical progress of the common rail fuel injection system, and the target heat release shape, calculated by computation, can be achieved by control of EGR, boosting, fuel injection timing, and injection quantity of multiple fuel injections.

In this paper, the reduction of premixed diesel combustion noise maintaining high thermal efficiency has been investigated by the control of injection timings and heating values of multiple fuel injections. There are two aspects of the combustion noise reduction by multiple fuel injections. One is the reduction of the maximum rate of pressure rise in each combustion cycle, and the other is noise reduction effects by the noise cancelling spike (NCS) combustion.

The research was conducted with both engine simulations and experiments. In combustion noise simulations, the heat release history of multiple injections was approximated by Wiebe functions and the simulated combustion noise was calculated from the fitted curve of the heat release and the coherence transfer function. The structural attenuation (SA) of the test engine was calculated from the power spectrum of the FFT analysis of the in-cylinder pressure wave data and the cross power spectrum of the sound pressure of the engine noise by the coherence method, then the combustion noise (CNL) can be calculated from the structural attenuation and cylinder pressure level (CPL) in the simulation, as shown in [equation 4](#). The simulation results were confirmed by the engine tests.

First, the combustion noise reduction by two stage fuel injection was investigated. The maximum rate of pressure rise changes depending on the combustion occurring separately in the compression and expansion strokes. One heat release was set at TDC and the second before or after the TDC. In the late two stage combustion as shown in [Figure 12 \(b\)](#), the combustion noise reduction was most effectively achieved when the heating value of Q_{2nd} is higher than that of Q_{1st} , however in the early two stage combustion in [Figure 12 \(a\)](#), the Q_{1st} heat release occurs during the compression stroke and the combustion noise reduction by the early two stage NCS combustion is more effective than the combustion noise reduction by the late two stage NCS combustion.

Three stage combustion simulations were also investigated at 0.6MPa IMEP and 2000 rpm. The optimum heat release shape for low combustion noise and high indicated thermal efficiency was calculated and the role of each part of the heat release in the three stage combustion is discussed. The simulation predicted 87.1 dBA of combustion noise and 50.3 % of indicated thermal efficiency.

Finally, the effects of multiple fuel injections on the degree of constant volume and combustion noise are analyzed and discussed.

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INTRODUCTION

Much research has been on the reduction of diesel engine combustion noise by engine tests. Kojima reported the relation between combustion noise and burning rates [1] and explained the characteristics of combustion noise generation [2, 3]. Sjoberg [4] and Johansson [5] reported the ringing intensity control and the relation between combustion noise and pressure history was reported by Scarpatti [6] and Ghandi [7].

The maximum rate of pressure rise is a main cause of the combustion noise, and Shibata and Shibaike investigated other factors in the combustion noise generation [8]. There, eighteen engine tests were conducted at one maximum rate of pressure rise condition (1.0 MPa/CA), and it was statistically determined that the maximum rate of heat release (RHR_{max}) and the crank angle of 50 % burn (CA50) are the second and third parameters in the combustion noise. The combustion noise reduction was achieved by EGR and boosting to control the RHR_{max} and CA50 maintaining the maximum pressure rise

rate at 1.0 MPa/CA. Okubo developed a combustion noise simulation method by coherence analysis [9]. This simulation technique was improved for the premixed diesel engine with the Wiebe function [10, 11], making the combustion noise prediction for multiple fuel injections possible.

Fuyuto reported the phenomenon that combustion noise generated in the second combustion could assist in reducing the combustion noise of the first fuel injection, termed “Noise Cancelling Spike Combustion (NCS combustion)” [12]. The NCS effects were investigated by engine tests and simulations, and an optimum heat release shape for low engine noise and high thermal efficiency was suggested [13]. Multiple fuel injection is effective to lower the combustion noise because of the reduction in the maximum rate of pressure rise in each separate combustion and NCS combustion effects.

In this paper, two topics are investigated. One is a further investigation of the combustion noise reduction by the dual fuel injections. Here heat release occurs at TDC and the effects of a heat release before TDC (compression stroke) or after the TDC (expansion stroke) on NCS effects were investigated

The other topic is combustion noise reduction by three fuel injections near TDC. For the dual fuel injections, there is generally amplification of the frequency of the combustion noise and the NCS effect is small. The third fuel injection is effective in lowering of the combustion noise of the amplified frequency and the optimum heat release shape of the three stage combustion was investigated by simulations and engine tests. Finally, the effects of multiple fuel injections on the degree of constant volume and combustion noise are analyzed and discussed.

EXPERIMENTAL

Engine Bench Set up

A schematic outline of the research engine bench is shown in Figure 1. The engine employed in the experiments was a supercharged, single cylinder DI diesel research engine, with a high pressure common rail fuel injection system for injection pressures up to 180 MPa. The specifications of the diesel engine are given in Table 1. The fuel used in the experiments was the commercially available #2 Japanese diesel fuel classified by JIS K2204 specifications (a cetane number of 54). The intake air was measured by an orifice flow meter and mixed with low pressure cooled EGR gas boosted by a supercharger, before being supplied to the engine. The intake manifold was equipped with an electric heater to maintain the intake air temperature at 25 °C. The engine was equipped with a pressure transducer (Kisler 6125A) and pressure data was transmitted to a PC as 45 cycles of crank angle resolved pressure data averaged over the 45 cycles. For all test conditions in this paper, the water and engine oil temperatures were maintained at 80 °C.

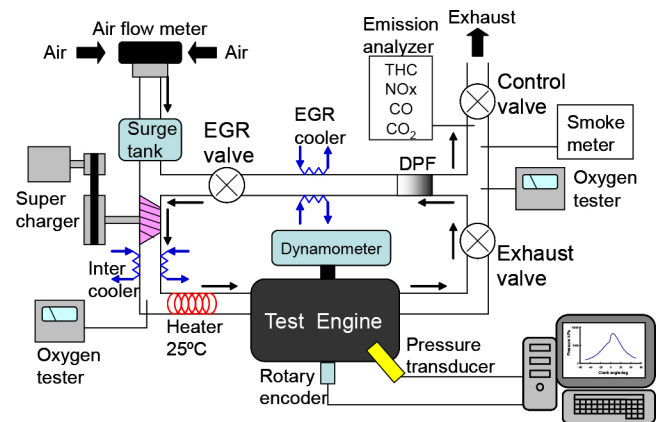


Figure 1. Outline of the engine test set-up

Table 1. Test engine specifications

Number of cylinders	1
Bore x Stroke	$\phi 85 \times 96.9$ mm
Displacement	550 cm ³
Compression ratio	16.3 : 1
Fuel injection system	Common rail
Nozzle hole	$\phi 0.125 \times 7$
Spray angle	156°
Connecting rod/crank radius ratio	L/R=3.1

Noise Measurements and Calculations

Measurements of the Engine Noise

The engine and combustion noise were evaluated with the noise measurement arrangement shown schematically in Figure 2. The supercharger in the right side is surrounded by the acoustic absorption panels and materials, and microphones were set at two sampling positions at the cylinder head top height (one 1 m away from the front of the engine and the other 1 m away to the left of the engine) and engine noise was collected from the left side and front of engines, avoiding the sampling of the transmission noise in the rear side of the engine. The sampled noise was averaged and analyzed by an FFT sound analyzers (Onosokki LA-1410 and DS-3000) and a 1/3 octave band filter was used for the analysis of the frequency characteristics.

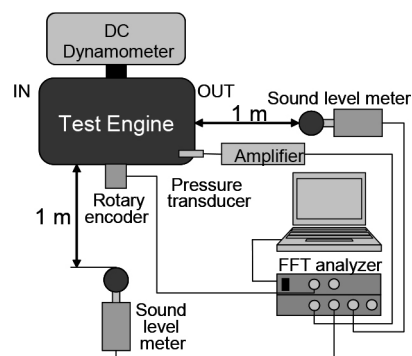


Figure 2. Engine noise measurement setup

Calculations of Combustion Noise [9]

The model used to analyze the engine noise is outlined in Figure 3. The engine noise value is comprised of the combustion noise level by the coherence method (CCNL) and the mechanical noise level (MNL) as shown in equation 1 below, assuming that the cylinder pressure and combustion noise levels are closely related. The transfer characteristic H was calculated from the power spectrum by the FFT analysis of the in-cylinder pressure waveform and the cross spectrum of the sound pressure of the engine noise level and pressure waveform. With this, the coherence combustion noise level (CCNL) is calculated from the in-cylinder pressure level and the transfer characteristics H by equation 2.

The pressure changes produce vibrations, transfer, and attenuation in the cylinder block, before it is released from the surface of the engine as combustion noise. The frequency characteristics are specific to the engine used. The engine structure attenuation (SA) can be calculated from the coherence combustion noise level and cylinder pressure level by equation 3, it was calculated under several conditions and averaged to provide a representative structure attenuation curve. Figure 4 shows the frequency characteristics of the structure attenuation (SA), and the combustion noise level (CNL) in the engine tests and simulations are calculated from the cylinder pressure level (CPL) and the structure attenuation (SA) by equation 4. (The direct noise measurements in Figure 2 are only conducted to determine the engine transfer function H . Once the structure attenuation is obtained, the combustion noise is calculated by equation 4.)

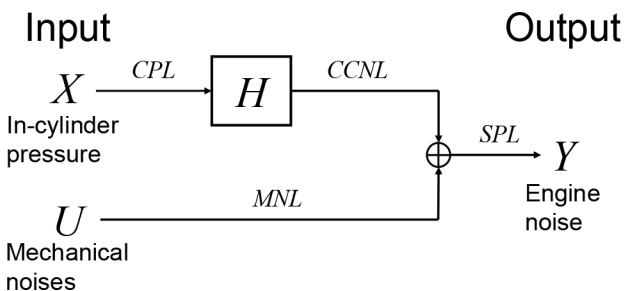


Figure 3. Engine noise generation model

$$SPL = CCNL + MNL \quad (1)$$

$$CCNL = CPL * H \quad (2)$$

$$SA = CCNL - CPL \quad (3)$$

$$CNL = SA + CPL \quad (4)$$

SPL : Sound pressure level [dBA]

$CCNL$: Combustion noise level by cohlernt method [dBA]

MNL : Mechanical noise level [dBA]

H : Transfercharacteristic

SA : Structure attenuation [dBA]

CNL : Combustion noise level [dBA]

CPL : Cylinder pressure level [dB]

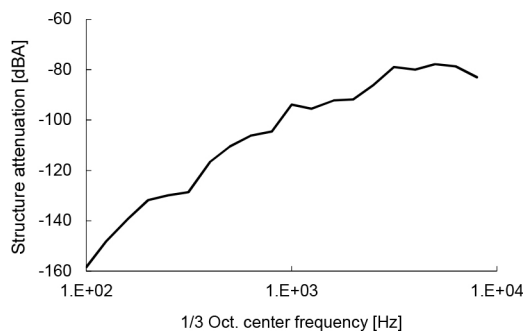


Figure 4. Structural attenuation of the test engine

Figure 5 shows the equal loudness curve of 60 phon pure sound for the human ear. For example, 60 dB of a 1 kHz pure sound (Point 1) is equivalent to 87 dB of 63 Hz (Point 2). This curve shows that the sensitivity of the human ear changes with frequency. Typically the sensitivity of the human ear is low below approximately 600Hz and above 3000 Hz, and is highest in the 1000-3000 Hz range. The sensitivity of the human ear changes depending on the frequency range. Therefore, the “A” weighted sound pressure level, the dBA, was used to evaluate the perceived loudness of the overall combustion noise.

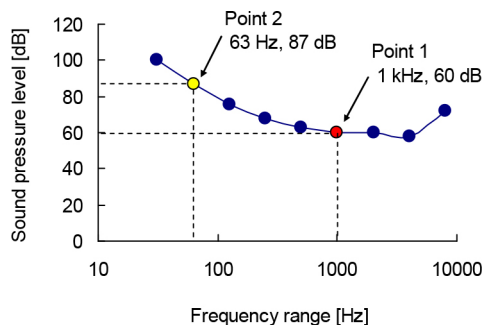


Figure 5. Equal loudness curve of a pure sound at 60 phon for the human ear

Simulation Methods

Wiebe Curve Fitting to Three Stage Combustion

The heat release histories were approximated by the Wiebe function, the simulated combustion noise was then calculated from the fitted curves of the heat release and the coherence transfer function, as a result the effects of the heat release history on the combustion noise and indicated thermal efficiency were investigated. An example of the heat release shape in three stage injection is shown in Figure 6. The heat release history was synthesized by introducing the Wiebe function in equation 5 four times: one time for the low temperature heat release (LTHR) and one each for the first, second, and third high temperature heat releases (HTHRs). In each of these heat release calculations, the Wiebe functions were fitted to the heat release history of three stage combustion in the engine test and the parameters M and θ_z in Table 2 were obtained and used in the further simulations.

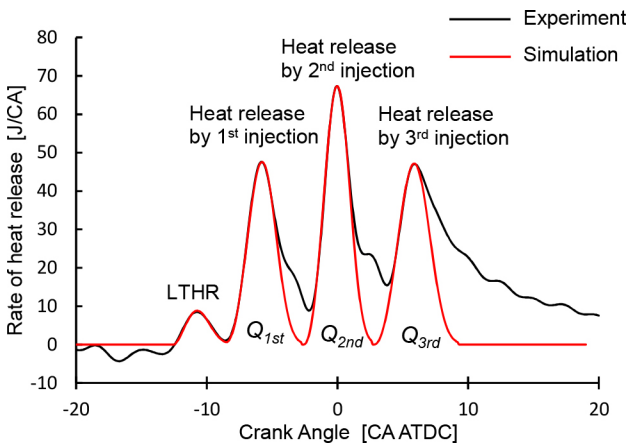


Figure 6. Approximation by the Wiebe function for the three stage combustion

Table 2. The values of parameters M and θ_z in the Wiebe function for Q_{1st} , Q_{2nd} and Q_{3rd}

	LTHR	Q_{1st}	Q_{2nd}	Q_{3rd}
M	1.5	2.15	2.2	2.1
θ_z	4.7	5.9	5.3	6.5

The in-cylinder pressure history was calculated from the heat release history by the Runge-Kutta numerical method in equation 6, and the kappa is calculated every 0.20 degree of crank angle from the in-cylinder temperature and the cylinder gas composition. The overall combustion noise and frequency characteristics of the combustion noise were calculated from the frequency characteristics of the in-cylinder pressure by Fourier transformation and the structure attenuation.

$$\frac{dQ}{d\theta} = 6.9 \cdot \frac{Q_{total}}{\theta_z} \cdot (M+1) \cdot \left(\frac{\theta}{\theta_z}\right)^M \cdot \exp\left\{-6.9 \cdot \left(\frac{\theta}{\theta_z}\right)^{M+1}\right\} \quad (5)$$

Q_{total} : Total heat release [J]

θ_z : Combustion duration [°CA]

M : Parameter M [-]

θ : Crank angle [°CA]

$$\frac{dQ}{d\theta} = \frac{\kappa}{\kappa-1} \cdot P \cdot \frac{dV}{d\theta} + \frac{1}{\kappa-1} \cdot V \cdot \frac{dP}{d\theta} \quad (6)$$

Q : Heat release [J]

κ : Ratio of specific heats [-]

P : Pressure [Pa]

V : Volume [m³]

θ : Crank angle [°CA]

To evaluate the accuracy of the simulation data, the combustion data obtained by engine test experiments and simulations were compared. The heat history of the engine data and the approximate history of the heat release by the Wiebe function are plotted together in Figure 7, and the combustion noise data of the experiments and simulation at 100-8000 Hz are plotted in Figure 8. The frequency characteristics of the simulation were very similar to the experiments, and this verifies that the combustion noise could be evaluated by the simulation.

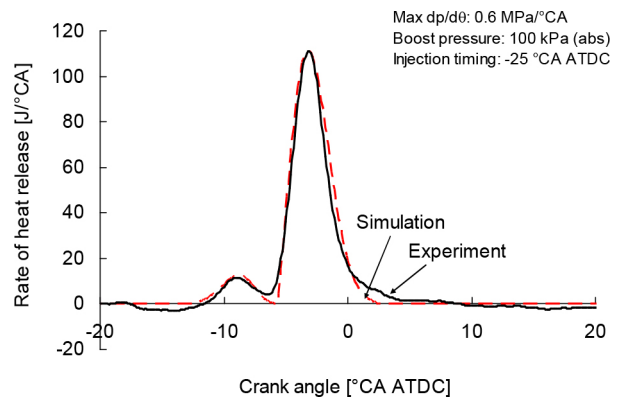


Figure 7. Heat release data in experiment and simulation

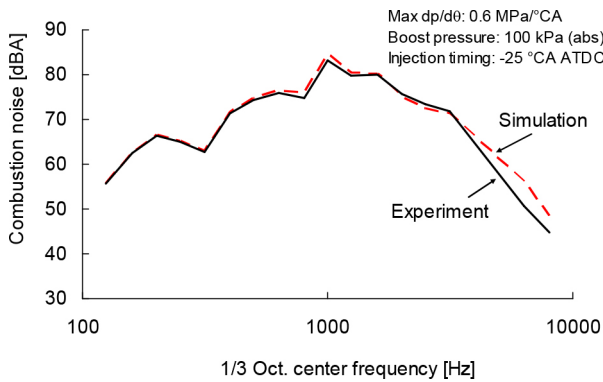


Figure 8. Plot of the combustion noise of the experiment and simulation

Figure 9 shows a rate of heat release and rate of pressure rise in two stage combustion and Figure 10 shows an explanation of amplifying and reduction phenomenon of combustion noise. The time difference in the crank angle at the peaks of the pressure rise rates between the first and second fuel injections is defined as $dP/d\theta_{1-2}$ in Figure 9. Equations 7 and 8 show the amplifying and reduction frequencies between Peaks 1 and 2. The Δt in the equations is the duration (sec) of $dP/d\theta_{1-2}$, and Δt equals $(dP/d\theta_{1-2})/12000$ as the engine is operated at 2000 rpm.

$$\text{Amplifying frequency [Hz]: } fn = n_A / \Delta t$$

(7)

$$(n_A = 1, 2, 3, \dots)$$

$$\text{Reduction frequency [Hz]: } fn = (2n_c + 1) / 2\Delta t$$

(8)

$$(n_c = 0, 1, 2, 3, \dots)$$

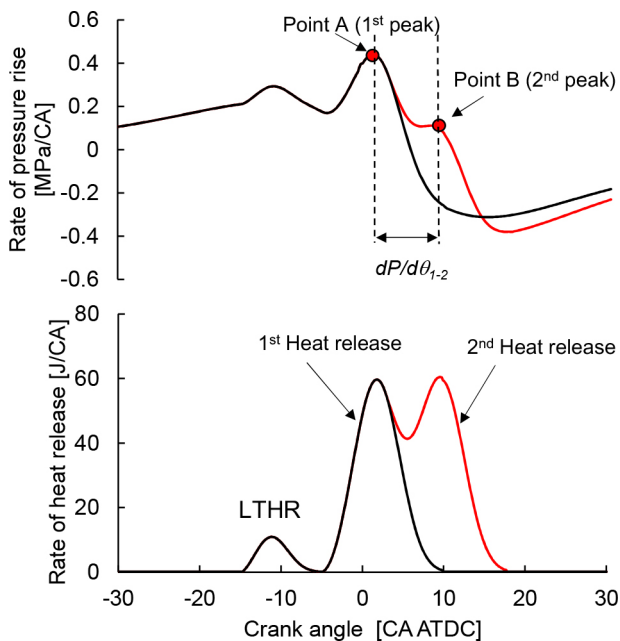


Figure 9. Heat release and pressure rise data in two stage combustion

As shown in equation 4, the combustion noise (CNL) is consisted from structure attenuation (SA) and cylinder pressure level (CPL). The SA is specific and constant for the test engine, and the changes in combustion noise is caused by the CPL. Here, the combustion noise in two stage combustion was explained by the CPL in Figure 10, and the frequencies of noise amplification and reduction by the interferences of two stage combustion noises appear in a cyclic manner. If the noise reduction can be done at the loudest frequency range for the engine, the overall combustion noise can be reduced. This is the basic idea of the two stage NCS combustion.

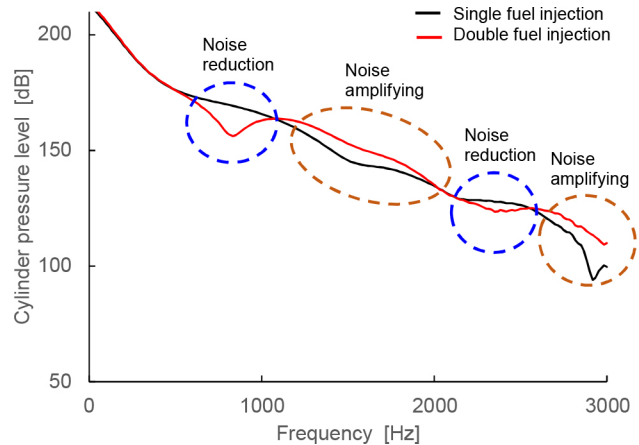


Figure 10. Parameter variables in the simulations

For the three stage combustion in Figure 11, the interferences of combustion noise in Peaks 1-2, Peaks 2-3, and Peaks 1-3 appear, and the combustion noise can be reduced by changing the duration of $dP/d\theta_{1-2}$, $dP/d\theta_{2-3}$, and $dP/d\theta_{1-3}$ and the heating value ratio between Q1, Q2 and Q3. This effect is investigated in Tests 2-4.

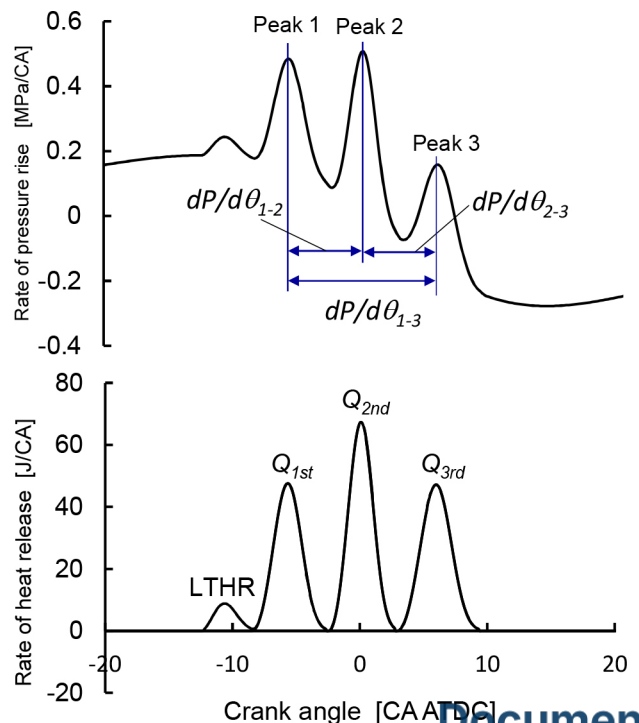


Figure 11. Parameter variables in the simulations