Abstract

Premixed diesel combustion offers the potential of high thermal efficiency and low emissions, however, because the rapid rate of pressure rise and short combustion durations are often associated with low temperature combustion processes, noise is also an issue. The reduction of combustion noise is a technical matter that needs separate attention. Engine noise research has been conducted experimentally with a premixed diesel engine and techniques for engine noise simulation have been developed. The engine employed in the research here is a supercharged, single cylinder DI diesel research engine with a high pressure common rail fuel injection system. In the experiments, the engine was operated at 1600 rpm and 2000 rpm, the engine noise was sampled by two microphones, and the sampled engine noise was averaged and analyzed by an FFT sound analyzer. The structural attenuation 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. With the heat release history approximated by the Wiebe function, the pressure history could be calculated from the fitted curves of the heat release, and the simulated combustion noise was calculated from the pressure history and structural attenuation.

Two simulations were conducted for this paper. Since combustion noise arises in a trade-off relation with thermal efficiency in general, the best heat release shape for low engine noise and high thermal efficiency was first investigated. The simulation parameters of the heat release shape are the crank angle at 50% burn (CA50), the maximum rate of heat release, the combustion duration, and the initial rise in the high temperature heat release. Based on the heat release data of the engine tests, initial conditions were set and the simulation was conducted with these initial conditions. The simulation results show that the CA50 and degree of constant volume are related to the thermal efficiency and the initial rise in the high temperature heat release is related to the engine noise. For the conditions studied in this research, the thermal efficiency is maximum when the CA50 of the heat release occurs at 4.0 °CA ATDC, and the optimum engine operation for lower engine noise with higher thermal efficiency can be achieved by a mild initial rise in the high temperature heat release (the parameter-M=2.0 in the Wiebe function) and a high degree of constant volume combustion.

The effects of the connecting rod/crank radius ratio (L/R ratio, Figure 10) on combustion noise and thermal efficiency were also investigated. The L/R ratio was changed from 2.5 to 8 in 0.5 steps, and the heat balance and combustion noise were calculated. The simulation results show that improvements in the thermal efficiency can be achieved with decreases in the L/R ratio, caused by the cooling loss reduction, but that L/R ratio changes do not result in a reduction in combustion noise. Further, the heat release shape for the optimum thermal efficiency and combustion noise was calculated from the points of the L/R ratio, maximum rate of heat release, and combustion duration.
The results of the pre-mixed diesel engine operation and the engine test data were compared with the simulation data. The parameters for this comparison were the intake oxygen content and the fuel injection timing. The engine test data shows that the parameter-M in the Wiebe function increases with delays in the CA50 position, and that there is a strong linear relation between the CA50 and the Wiebe parameter-M.

**Introduction**

For next generation vehicles, the regulations of NOx and PM emissions are tighter and higher thermal efficiencies are required. The PCCI diesel engine has the potential of low NOx emissions combined with diesel-like high engine efficiency. However, the ignitability of diesel fuel is sufficiently high, so various mechanical attachments, such as EGR and superchargers, are used to achieve the PCCI operation. As the premixed duration becomes longer, the pre-mixed air-fuel mixture ignites in one burst in the cylinder and this leads the PCCI combustion noise.

Reducing the maximum rate of pressure rise is known as an empirical method to reduce engine noise. Several papers have reported on combustion noise analysis of diesel engine combustion. Kojima reported the relation between the combustion noise and the burning rate \[1\], and clarified the characteristics of combustion noise generation \[2, 3\]. The ringing intensity control reported by Sjoberg \[4\] and Johansson \[5\], and the relation between combustion noise and pressure history was reported by Scarpati \[6\] and Ghandhi \[7\]. The tonal research by Torii \[8\] separated the combustion noise and mechanical noise. Multiple injection strategies were applied to achieve noise reductions by Badami \[9\] and Fuyuto \[10\].

In our previous paper \[11\], the combustion noise parameters were investigated by statistical analysis of engine test data. The maximum rate of pressure rise is the parameter most strongly related to combustion noise, but the maximum rate of heat release (\(RHR_{\text{max}}\)), combustion duration, and CA50 also affect combustion noise. For example, the combustion noise data tested under one maximum rate of pressure rise condition can be correlated with the \(RHR_{\text{max}}\) and the combustion duration, as in the following equation 1. This equation is valid for our test engine (Table 1) at the specific engine speed of 1600 rpm. (Because there is a clear relation between the \(RHR_{\text{max}}\) and CA50, the parameters of the \(RHR_{\text{max}}\) and combustion durations are chosen as the parameters.)

\[
CN = -0.544(CD) + 0.0275(RHR_{\text{max}}) + 85.526
\]

\(CN\): Combustion noise \([J]\)

\(CD\): combustion duration \((CA10-CA90) [^\circ\text{CA}]\)

\(RHR_{\text{max}}\): Maximum rate of heat release \([J/^\circ\text{CA}]\)

Further, an engine noise simulation technique, which predicts the combustion noise from heat release data, was developed and noise characteristics have been researched \[12\]. Simulation of this kind can evaluate virtual combustion which cannot occur in engine tests. For example, the effects of combustion timing (CA50) on the combustion noise and thermal efficiency were investigated. The simulation results with the CA50 varied from −12.6 to 15.0 \(^\circ\text{CA ATDC}\) without changing the heat release shape are shown in Figure 1, and the indicated thermal efficiency and combustion noise are shown in Figure 2.

Figure 2 suggests that engine operation with a high degree of constant volume (the indicated thermal efficiency divided by the thermal efficiency of constant volume combustion at TDC) can attain high thermal efficiencies but that high combustion noise is induced. This may be termed “the sound of efficiency”. The combustion noise analysis was conducted from the aspects of the maximum rate of heat release, combustion period, and the initial rise of the high temperature heat release. It was found that the combustion noise is strongly related to these combustion parameters.

Based on the previous research results, two further simulations were conducted in this paper. Since the combustion noise is in a trade-off relation with thermal efficiency in general, the best heat release shape for low engine noise and high thermal efficiency was first investigated. Then, the effects of the connecting rod/crank radius ratio (L/R ratio) on the combustion noise and thermal efficiency were investigated from a mechanical standpoint, because the piston motion, surface area to volume ratio, and volumetric rate of change at the TDC can be changed by the L/R ratio, a parameter which is related to the maximum rate of pressure rise.

Further, the pre-mixed diesel engine was operated and the engine test data was compared with the simulation data.
Figure 2. Indicated thermal efficiency and combustion noise versus combustion timing (CA50)

Experimental

Engine Bench Set Up

A schematic outline of the research engine bench is shown in Figure 3. The engine employed in the experiments was a supercharged, single cylinder DI diesel research engine (PCCI diesel engine), with a high pressure common rail fuel injection system with injection pressure up to 180 MPa possible. 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 (54 cetane number). The intake air was measured by an orifice flow meter, mixed with low pressure cooled EGR (exhaust gas recirculation) gas boosted by a supercharger, and 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 at 45 cycles of crank angle resolved pressure data and averaged over 45 cycles. For all test conditions in this paper, the water temperature and engine oil temperature were kept at 80 °C.

Table 1. Test engine specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
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</tr>
<tr>
<td>Bore x Stroke</td>
<td>85 x 96.9 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>550 cm³</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>18.3 : 1</td>
</tr>
<tr>
<td>Fuel injection system</td>
<td>Common rail</td>
</tr>
<tr>
<td>Nozzle hole</td>
<td>ø 0.125 x 7</td>
</tr>
<tr>
<td>Spray angle</td>
<td>156°</td>
</tr>
<tr>
<td>Connecting rod/crank radius ratio</td>
<td>L/R=3.1</td>
</tr>
</tbody>
</table>

Figure 3. Outline of the engine test set-up

Noise Measurements and Calculations

Measurements of Engine Noise

The engine and combustion noise were evaluated with the noise measurement arrangement shown schematically in Figure 4. Microphones were set at two different sampling positions at the cylinder head top height (one is 1 m from the front of the engine and the other is 1 m away to the left of the engine). The sampled noise was averaged and analyzed by an FFT sound analyzer (Onosokki LA-1410) and the 1/3 octave band filter was used for the analysis of the frequency characteristics.

Calculations of Combustion Noise

The model used to analyze the engine noise is outlined in Figure 5. Here, the combustion noise was calculated and defined by the coherence method [13]. The engine noise value consists of the combustion noise level (CNL) and the mechanical noise level (MNL) as shown in equation 2, and there is a relation between the cylinder pressure level and combustion noise level. 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 combustion noise level is calculated from the in-cylinder pressure level and the transfer characteristics H by equation 3.
Figure 5. Engine noise generation model

\[ SPL = CNL + MNL \]  \hspace{1cm} (2)

\[ CNL = CPL \times H \]  \hspace{1cm} (3)

\[ SA = CNL - CPL \]  \hspace{1cm} (4)

\[ CNL = SA + CPL \]  \hspace{1cm} (4')

**SPL**: Sound pressure level [dBA]

**CNL**: Combustion noise level [dBA]

**MNL**: Mechanical noise level [dBA]

**H**: Transfer characteristic

**SA**: Structure attenuation [dBA]

**CPL**: Cylinder pressure level [dB]

The pressure changes produce vibrations, transfer or attenuate in the cylinder block, and are released from the surface of the engine as combustion noise. The frequency characteristics are specific to the engine used. The structure attenuation (SA) can be calculated from the combustion noise level and cylinder pressure level by equation 4 and calculated under the several conditions, it was averaged to make it more representative. Figure 6 shows the frequency characteristics of the structure attenuation. Once the structure attenuation (SA) was calculated, the combustion noise level (CNL) in engine tests and simulations are calculated from the cylinder pressure level (CPL) and the structure attenuation (SA) by equation 4'.

Simulation methods

Methods of Combustion Noise Simulation

The heat release histories were approximated by Wiebe functions, the simulated combustion noise was then calculated from the fitted curves of the heat release and the coherence transfer function, so the effects of the heat release history on the combustion noise and indicated thermal efficiency could be investigated. Figure 8 shows the calculation method for the combustion noise simulation. The heat release history was synthesized by applying the Wiebe function in equation 5 two times (Step 1). The in-cylinder pressure history was calculated from the heat release history by the Runge-Kutta numerical method in equation 6 (Step 2), 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 structural attenuation (Step 3).

\[ \frac{dQ}{d\theta} = 6.9 \cdot \frac{Q_{\text{total}}}{\theta Z} \cdot (M + 1) \left( \frac{\theta}{\theta Z} \right)^M \exp \left\{ -6.9 \cdot \left( \frac{\theta}{\theta Z} \right)^{M+1} \right\} \]  \hspace{1cm} (5)

\[ Q_{\text{total}} \]: Total heat release [J]

\[ \theta Z \]: Combustion duration [°CA]

\[ M \]: Parameter M [-]

\[ \theta \]: Crank angle [°CA]

\[ \frac{dQ}{d\theta} = \frac{\kappa}{\kappa - 1} \cdot P \cdot \frac{dV}{d\theta} + \frac{1}{\kappa - 1} \cdot V \cdot \frac{dP}{d\theta} \]  \hspace{1cm} (6)

\[ Q \]: Heat release [J]

\[ \kappa \]: Ratio of specific heats [-]

\[ P \]: Pressure [Pa]

\[ V \]: Volume [m³]

\[ \theta \]: Crank angle [°CA]
Comparison of Combustion Noise Obtained in the Experiments and Simulations

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 9, and the combustion noise data of the experiments and simulation at 100-8000 Hz frequency are plotted in Figure 10. The frequency characteristics of the simulation were very similar to the experiments, and it verifies that the combustion noise could be evaluated by the simulation.

Table 2. Initial conditions for the simulations

<table>
<thead>
<tr>
<th>Engine speed</th>
<th>rpm</th>
<th>1600</th>
</tr>
</thead>
<tbody>
<tr>
<td>IMEP</td>
<td>MPa</td>
<td>0.4</td>
</tr>
<tr>
<td>Intake pressure</td>
<td>kPa (abs)</td>
<td>140</td>
</tr>
<tr>
<td>Intake oxygen content</td>
<td>vol%</td>
<td>13.0</td>
</tr>
<tr>
<td>Total heat release of HTHR</td>
<td>J</td>
<td>425</td>
</tr>
<tr>
<td>The maximum rate of heat release</td>
<td>J/°CA</td>
<td>80</td>
</tr>
<tr>
<td>Parameter M</td>
<td>-</td>
<td>0.25 to 7.00</td>
</tr>
<tr>
<td>Combustion duration</td>
<td>°CA</td>
<td>5.2 to 6.8</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA ATDC</td>
<td>-3 to 7</td>
</tr>
<tr>
<td>Connecting rod/crank radius ratio</td>
<td>-</td>
<td>2.5 to 8.0</td>
</tr>
</tbody>
</table>

Figure 9. Heat release data in experiment and simulation

Figure 10. Plot of the combustion noise of the experiment and simulation

Initial Conditions for the Simulations

Based on the engine test data, the initial conditions for the simulations, listed in Table 2, were defined. The engine speed was 1600 rpm, the boosted pressure was 140 kPa (abs), and the intake oxygen content by EGR was 13.0 vol%. The heat release shape of the high temperature heat release (HTHR) was changed by the parameter-M in the Wiebe function, as shown in Figure 11, while constant heat release was assumed for the low temperature heat release. The total heating value of each HTHR condition in Figure 11 is 425 J, and the peak value of the rate of heat release was maintained at 80 J/°CA, adjusted by the combustion duration. The parameter-M was varied from 0.25 to 7 at the various CA50s, which was changed from −3 °CA ATDC to 7 °CA ATDC. The combustion noise and indicated thermal efficiency were calculated within the ranges of parameter M and CA50. The effect of the low temperature heat release (LTHR) on engine noise is not discussed here, but is taken up in the appendix.

Table 2. Initial conditions for the simulations

<table>
<thead>
<tr>
<th>Engine speed</th>
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<tr>
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<td>5.2 to 6.8</td>
</tr>
<tr>
<td>CA50</td>
<td>°CA ATDC</td>
<td>-3 to 7</td>
</tr>
<tr>
<td>Connecting rod/crank radius ratio</td>
<td>-</td>
<td>2.5 to 8.0</td>
</tr>
</tbody>
</table>

Figure 11. Relation between parameter M and the heat release shape
Method of Indicated Thermal Efficiency Calculations

The cylinder gas composition at each crank angle is calculated for the simulated heat release history. Thus the pressure, mean temperature, specific heats, specific heat ratios, and generated work could be calculated throughout the expansion stroke. The cooling loss is calculated by the Hohenberg equation and the cylinder temperature (the Woschni equation was examined, but the Hohenberg equation was found to be better suited to the engine test data here). The exhaust heat loss is calculated from the enthalpy difference between the intake and exhaust gases. The input energy is the total of the integral of the heat release rate (425 J) and the cooling loss. This allowed the indicated thermal efficiency to be calculated.

Connecting Rod/Crank Radius Ratio

Figure 12. Volume change rate and surface area to volume ratio versus connecting rod/crank radius ratio

The rate of change of volume and the surface area to volume ratio will be most critical for efficiency and noise around TDC, and the effects of the connecting rod/crank radius ratio on the indicated thermal efficiency and combustion noise were investigated. During the combustion duration, the absolute value of the rate of change becomes smaller and the surface area to volume ratio becomes larger with increases in the connecting rod/crank radius ratio (L/R ratio), as shown in Figure 12. The L/R ratio was used as a parameter, and the effects of the rate of change in volume and the surface area to volume ratio on the indicated thermal efficiency and combustion noise were investigated. The following assumptions were made in these simulations.

1. When the L/R ratio is changed, the crank radius changes, but the displacement (550 cm$^3$), compression ratio (16.3:1), and length of connecting rod (150 mm) were constant.
2. The structural attenuation is the same for all L/R ratios

Results and Discussion

The Effects of CA50 and Parameter-M in the Wiebe Function on the Indicated Thermal Efficiency and Combustion Noise

Figure 13 shows the effects of CA50 and parameter-M in the Wiebe function on the indicated thermal efficiency and combustion noise. As shown in Table 1, the L/R ratio of the test engine was 3.1. The indicated thermal efficiency increases with the increase in CA50, and is at the maximum when the CA50 is at 4 °CA ATDC. The location of CA50 relative to TDC can be interpreted as the high degree of constant volume combustion. Regardless of the parameter-M value, the thermal efficiency is almost everywhere similar at the same CA50 position. The simulation here suggests that in the case of CA50=4 °CA ATDC, the combustion noise can at most be reduced by 2.2 dBA with the increase in parameter-M. Here, the combustion process was artificially constrained to be the same combustion duration at every parameter-M value in the simulation. As discussed later, the parameter-M and combustion period changed simultaneously and considerable combustion noise reduction could be observed in the simulations and engine tests.

The frequency characteristics of the combustion noise at CA50=4 °CA ATDC are shown in Figure 14. With the increase in parameter-M, the combustion noise above 1000 Hz is reduced, but the peak noise at 1000 Hz (point A in Figure 14) does not change. This suggests that if the peak of the power spectrum can be reduced, a considerable reduction in combustion noise may be expected.

In the engine operation here, the parameter-M changes from 0.5 to 2.0, and the heat balance of the thermal efficiency, cooling loss, and exhaust loss were calculated under the M=1.0 condition. The result is shown in Figure 15. The cooling loss is lower than that in the conventional diesel engine operation because the lower IMEP and high EGR operation are assumed as in Table 2. The cooling loss decreases, with the increase in the CA50 position. The thermal
efficiency was the maximum at CA50=4 °CA ATDC, but a too large delay of CA50 causes a lowering of the thermal efficiency and enhances the exhaust loss.

The effects of connecting rod/crank radius ratio and Parameter-M in the Wiebe function on the indicated thermal efficiency and combustion noise

To evaluate the effects of the connecting rod/crank radius ratio and parameter-M in the Wiebe function, the CA50 was set at 4 °CA ATDC, where the indicated thermal efficiency was the maximum in this simulation. Figure 16 shows the effects of connecting rod/crank radius ratio (L/R ratio) and parameter-M in the Wiebe function on the indicated thermal efficiency and combustion noise. The indicated thermal efficiency was the maximum with the 2.5 L/R ratio, and the indicated thermal efficiency decreased with increases in the L/R ratio. As shown in Figure 12, the rate of change in the volume increased with decreases in the L/R ratio. This leads to a 0.59% decrease in the degree of constant volume, but the cooling loss also decreased by 5.9%, as shown in Figure 17, because the surface area to volume ratio becomes smaller when the L/R ratio is small, as shown in Figure 12. The effect of the cooling loss on the thermal efficiency is stronger than that of the degree of constant volume, and the indicated thermal efficiency increased with decreases in the L/R ratio. The effect of the L/R ratio on the overall combustion noise was very small. The overall combustion noise can be reduced with a higher parameter-M, which expresses a milder initial rise of high temperature heat release.

The heat balances of the M=1.0 condition is shown in Figure 18. The parameter is the L/R ratio and the CA50 position is constant at 4 °CA ATDC. The cooling loss decreases, and the exhaust loss and indicated thermal efficiency increase with the decrease in the L/R ratio. The indicated thermal efficiency was the maximum at the L/R ratio=2.5 (51.0%) in the simulation.
**Optimum CA50 Position for the Best Thermal Efficiency versus the Connecting Rod/Crank Radius Ratio**

Figure 19 shows the effects of CA50 and parameter-M in the Wiebe function on the indicated thermal efficiency and combustion noise under the L/R ratio=6.0 condition (short stroke condition). Comparing with Figure 13, the indicated thermal efficiency range of Figure 19 is lower than that of Figure 13, and the CA50 at the maximum thermal efficiency in Figure 19 was more delayed. In Figure 16, the thermal efficiency was compared for the CA50=4 °CA ATDC position, however the optimum CA50 position changes depending on the L/R ratio. To know the optimum CA50 position, the thermal efficiency and combustion noise must be calculated specifically at the optimum CA50 position for various L/R ratios. Figure 20 shows the L/R ratio versus the CA50 for the best indicated thermal efficiency, and Figure 21 shows the best indicated thermal efficiency and combustion noise versus the L/R ratio.

![Figure 19](image1.png)

**Figure 19. The effects of CA50 and Parameter-M on the indicated thermal efficiency and combustion noise (L/R=6.0)**

![Figure 20](image2.png)

**Figure 20. The connecting rod/crank radius ratio versus CA50 under the maximum thermal efficiency conditions**

Since the optimum CA50 for the best indicated thermal efficiency was delayed (Figure 20) and the surface area to volume ratio also increased with increases in the L/R ratio (as shown in Figure 12), the optimum thermal efficiency decreased in Figure 21, but the change in combustion noise was very small.

![Figure 21](image3.png)

**Figure 21. The connecting rod/crank radius ratio versus combustion noise under the maximum thermal efficiency conditions (the optimum CA50s are different depending on the L/R ratio, as shown in Figure 20)**

**Optimum Heat Release Shape and Connecting Rod/Crank Radius Ratio for the Indicated Thermal Efficiency and Combustion Noise**

From the test data in the above simulations, the optimum heat release shape and the L/R ratio were investigated under the test conditions in Table 2. Figure 22 shows the optimum heat release shape with the conditions set here. This figure suggests that the following are the optimum conditions for higher indicated thermal efficiencies and lower engine noise.

1. The L/R ratio low
2. The CA50 at 4 °CA ATDC*
3. A mild initial rise of high temperature heat release
4. A high degree of constant volume

![Figure 22](image4.png)

**Figure 22. The optimum heat release shape for high thermal efficiency and low engine noise calculated by the simulation (the initial conditions are shown in Table 2) (M=2.00)**

Considering engine design and actual heat release shapes, 2.5 was selected as the L/R ratio and 2.0 for the parameter-M. The indicated thermal efficiency and overall engine noise were 51.0% and 86.8 dBA respectively. When the maximum rate of heat release in HTHR increases in Figure 22, the overall engine noise worsens, and in the case that the maximum rate of heat release in HTHR decreases in Figure 22, the engine noise decreases and the thermal efficiency worsens. (* Since the degree of constant volume is affected by the combustion duration, the optimum CA50 changes depending on the engine load.)
Figure 22 was calculated by the initial conditions in Table 2, the maximum rate of heat release was constant and the change in combustion duration was only 1.6 °CA (5.2-6.8 °CA). Here, the effect of the combustion duration and maximum rate of heat release on the combustion noise and indicated thermal efficiency were investigated by simulation. The following RHR$_{\text{max}}$/CD ratio was introduced.

\[
\frac{\text{RHR}_{\text{max}}}{\text{CD}} = \frac{\text{Maximum rate of heat release}}{\text{Combustion duration}}
\]  

(5)

RHR$_{\text{max}}$: Maximum rate of heat release [J]

CD: Combustion duration (CA10-CA90) [°CA]

The simulation was conducted with the 6 data sets (A-F) and the initial conditions are shown in Table 3. The heating value of the high temperature heat release, CA50, parameter-M, and L/R ratio were kept at 425 J, 4 °CA ATDC, 2.0, and 2.5 respectively. The combustion duration in equation 5 changed from 1.8 °CA (data set A) to 22.2 °CA (data set F), the maximum rate of heat release changed from 236.9 J (data set A) to 19.0 J (data set F), and the RHR$_{\text{max}}$/CD ratio changed from 131.8 (data set A) to 0.9 (data set F). The engine speed, IMEP, intake pressure and intake oxygen content are 1600 rpm, 0.4 MPa, 140kPa, and 13.0%, which are the same as in Table 2.

### Table 3. Simulation conditions for the calculation of optimum heat release shape (parameter: RHR$_{\text{max}}$/CD ratio)

<table>
<thead>
<tr>
<th>Parameters</th>
<th>unit</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Combustion duration</td>
<td>°CA</td>
<td>1.8</td>
<td>5.2</td>
<td>7</td>
<td>8.8</td>
<td>13.4</td>
<td>22.2</td>
</tr>
<tr>
<td>Maximum rate of heat release</td>
<td>J/°CA</td>
<td>236.9</td>
<td>79.3</td>
<td>59.4</td>
<td>47.6</td>
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<td>Heating value of HTHR</td>
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<td>CA50</td>
<td>°CA ATDC</td>
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<td>L/R ratio</td>
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</tbody>
</table>

The simulation results are shown in Figure 23. The abscissa shows the overall combustion noise and the ordinate is the indicated thermal efficiency. With the decrease in the RHR$_{\text{max}}$/CD ratio from 131.8 to 2.4 (data sets A to E), the engine noise decreased maintaining the indicated thermal efficiency, but further reduction of the RHR$_{\text{max}}$/CD ratio (data set F) induces a reduction in thermal efficiency. Figure 24 shows the frequency characteristics of the combustion noise for data sets A to F. Combustion noise above 500 Hz is reduced with the decrease in the RHR$_{\text{max}}$/CD ratio, and this is the cause of the reduction in the overall combustion noise.

In Figure 23, the best plot for the thermal efficiency and overall combustion noise is data set E (RHR$_{\text{max}}$/CD ratio=2.4) but it is difficult to achieve this heat release shape by engine tests. Considering actual combustion conditions, the shape of data set D, the second best and suggested in Figure 25, would be the optimum here.

### Figure 23. Overall combustion noise versus indicated thermal efficiency (Parameter: RHR$_{\text{max}}$/CD ratio shown in Table 3)

### Figure 24. The combustion noise for various RHR$_{\text{max}}$/CD ratios

### Figure 25. Optimum heat release shape for combustion noise and indicated thermal efficiency

**Comparison of the Indicated Thermal Efficiency and Combustion Noise Obtained in the Engine Tests and Simulated Results**

In the simulations, it was possible to change the CA50 without changing the heat release shape like shown in Figure 1, and the effects of CA50 on the indicated thermal efficiency and engine noise were investigated, however, this is not possible in tests with an actual engine, because more than two parameters, for example, combustion duration and the maximum value of heat release, change. So, the effect of one of the parameters on engine noise alone can be investigated in the simulation, but the measured combustion noise in the engine test is the combined value of these
parameters. In this chapter, simple engine tests were conducted, and the detailed noise characteristics were analyzed and discussed based on the simulation data.

The boost pressure by supercharger, intake oxygen content by EGR, and the injection timing are the mechanical parameters that can change the heat release shape of the HTHR. For the comparisons here, the engine was operated under high EGR conditions, 9.5 vol% and 12.0 vol% intake oxygen content, and the injection timing was varied from −25 °CA ATDC to −15 °CA ATDC in 5 °CA increments. The IMEP and engine speed were everywhere at 0.8 MPa and 2000 rpm respectively. The engine tests were conducted under the 6 different conditions, as detailed in Table 4.

Table 4. The engine test conditions with the lowered intake oxygen content by EGR (parameter: injection timing)

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<tbody>
<tr>
<td>1</td>
<td>−25</td>
<td>9.5</td>
<td>175</td>
<td>0.8</td>
<td>2000</td>
</tr>
<tr>
<td>2</td>
<td>−20</td>
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<td></td>
<td></td>
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<tr>
<td>3</td>
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<tr>
<td>4</td>
<td>−20</td>
<td>12</td>
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<td>5</td>
<td>−15</td>
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<td>6</td>
<td>−10</td>
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The heat release data and the curves fitted by the Wiebe function (simulation) were compared, and the effects of the fuel injection timing on the indicated thermal efficiency and engine noise are discussed.

Relation between CA50 and Parameter-M, and the Effects of Intake Oxygen Content on Combustion Noise

The CA50 position of the heat release data in the engine tests versus the parameter-M calculated by the fitted curve of the heat release is shown in Figure 26. There is a clear linear relation between the CA50 and parameter-M: the parameter-M increases (the initial rise of heat release becomes milder) with delays in the CA50 position. The parameter-M could independently be changed in the simulation, however, both the parameter-M and CA50 change simultaneously in the engine tests. The parameter-M of the 9.5vol% intake oxygen content is always larger than that of the 12.0vol% intake oxygen content, showing that the initial rise in the high temperature heat release becomes milder with decreases in the intake oxygen content by EGR. Further, the slope of the linear 9.5 vol% intake oxygen content curve is steeper than the 12.0 vol% intake oxygen content curve (also linear), showing that the milder heat release at the initial stage of the high temperature heat release is more difficult to achieve under the higher intake oxygen content condition of the engine operation.

The overall combustion noise versus the indicated thermal efficiency is shown in Figure 27. There are small differences between the data measured by the engine tests and the simulations, but the trends of both are quite similar (The differences are caused by the heat release shape of the engine test data and the fitted curves, and an example of this for the data set No.2 (Table 4) is shown in Figure 28 (red line and green dotted line)). Figure 27 shows that the combustion noise can be reduced by a delay in the fuel injection timing.
The maximum rate of heat release, combustion duration, CA50, and parameter-M are the important parameters in the reduction of combustion noise, and those parameters are intertwined in complicated ways. The combustion noise simulation in this study is an effective tool for the enlargement of PCCI speed-load operational range.

Conclusions

To achieve higher thermal efficiency and lower combustion noise with pre-mixed diesel engine operation, a simulation of the combustion noise and heat balance were conducted for the heat release shape and the connecting rod/crank radius ratio under the initial conditions of Table 2, determined from the engine test data. The conclusions of this investigation may be summarized as follows:

1. The optimum heat release shape to achieve low engine noise and high thermal efficiency was investigated. The CA50 and degree of constant volume are related to the thermal efficiency and the initial rise in the high temperature heat release is related to the engine noise.
2. With delays in the CA50 position, the cooling loss decreases and the thermal efficiency was the maximum at CA50=4 °CA ATDC, but a too large delay in CA50 causes a lowering of the thermal efficiency and increases the exhaust loss.
3. The thermal efficiency increased with decreases in the connecting rod/crank radius ratio, because of the reduction of the cooling loss, related to the reduction of the surface area to volume ratio.
4. The optimum heat release was calculated for the initial conditions in Table 2. Considerations of the actual engine design and achievable heat release shapes, 2.5 was chosen for the connecting rod crank radius ratio, and the 2.0 was chosen as the parameter-M in the Wiebe function.
5. The effects of the $RHR_{\text{max}}/CD$ ratio (maximum rate of heat release divided by combustion duration) on combustion noise and thermal efficiency were investigated. The data set D ($RHR_{\text{max}}/CD$ ratio=5.4) in Table 3 was optimum for the indicated thermal efficiency (50.9%) and overall combustion noise (76.9 dBA).

The engine tests were conducted and the test results were compared with the simulation data.

6. The parameter-M in the Wiebe function increases with delays in the CA50 position, and there is a strong linear relation between the parameters.
7. The combustion noise of the engine tests reduces 4.0-4.3 dBA, as the maximum peak of the heat release and the combustion period change with the CA50 positions in the engine operation.

References

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Abbreviations
ATDC - After top dead center
CA10 - Crank angle of 10% burn
CA50 - Crank angle of 50% burn
CA90 - Crank angle of 90% burn
CD - Combustion duration (CA10-90)
CNL - Combustion noise level
CPL - Cylinder pressure level
DI - Direct injection
EGR - Exhaust gas recirculation
FFT - Fast Fourier transformation
H - Transfer characteristic
HCCI - Homogeneous charge compression ignition
HTHR - High temperature heat release
IMEP - Indicated mean effective pressure
L - Length of connecting rod
LTHR - Low temperature heat release
L/R ratio - Connecting rod/crank radius ratio
M - Parameter for heat release shape in Wiebe function
MNL - Mechanical noise level
PCCI - Premixed charge compression ignition
R - Crank radius
RHR - Rate of heat release
RHR_{max} - Maximum rate of heat release
RHR_{max}/CD - Maximum rate of heat release divided by combustion duration
SA - Structure attenuation
SPL - Sound pressure level
THE EFFECTS OF LTHR ON COMBUSTION NOISE

The crank angle at 50% burn of LTHR (LTHR CA50) and the heating value of LTHR (LTHR Q) are related to the crank angle of the 50% burn of HTHR (HTHR CA50) and the HTHR shape, as shown in Figure AP-1. The effect of LTHR on the combustion noise in itself is very small, but the LTHR changes the HTHR shape, and the combustion noise changes indirectly. The detailed chemical reactions and physical relations between the LTHR and HTHR are discussed in references [14, 15, 16].

Figure AP-1. HTHR CA50 versus the LTHR CA50 and LTHR heating values [14]