

# Diesel Engine Combustion Noise Reduction by the Control of Timings and Heating Values in Two Stage High Temperature Heat Releases

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## ABSTRACT

Reductions in combustion noise are necessary in high load diesel engine operation and multiple fuel injections can achieve this with the resulting reductions in the maximum rate of pressure rise. In 2014, Dr. Fuyuto reported the phenomenon that the combustion noise produced in the first combustion can be reduced by the combustion noise of the second fuel injection, and this has been named "Noise Cancelling Spike Combustion (NCS combustion)".

To investigate more details of NCS combustion, the effects of timings and heating values of the first and second heat releases on the reduction of overall combustion noise are investigated in this paper.

The engine employed in the research here is a supercharged, single cylinder DI diesel engine with a high pressure common rail fuel injection system. The engine was operated at 2000 rpm and 0.8MPa IMEP, and 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 heat release history, and the simulated combustion noise was calculated from the pressure history and structural attenuation. The test parameter variables are the combustion duration of crank angles of 10% burns in the first and second high temperature heat releases ( $CA10_{1-2}$ ), the duration of the peaks of the maximum rate of pressure rise ( $dP/d\theta_{1-2}$ ), and the heating values of the first and second fuel injections ( $Q_{1st}$  and  $Q_{2nd}$ ). The effects of these parameter variables on the combustion noise were investigated. Simulations showed that the combustion noise was the minimum when the  $dP/d\theta_{1-2}$  was 7.4 °CA, and that 4.1 dBA of combustion noise could be reduced by the NCS combustion, and this was verified by the engine tests. In the simulations, with the  $CA10_{1-2} = 8$  °CA and  $Q_{1st} \cdot Q_{2nd} = 350J \cdot 450J$ , the overall combustion noise was the minimum, and further, the optimum heat release shape for the higher thermal efficiency and lower combustion noise are suggested in the paper.

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## INTRODUCTION

Reducing the maximum rate of pressure rise is known as an empirical method to reduce the combustion noise of premixed diesel engines. Several papers have reported on the combustion noise analysis of diesel engine combustion. Kojima reported the relation between combustion noise and burning rate [1], and explained 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] distinguished the combustion noise and mechanical noise. In a previous paper [9], combustion noise parameter variables were investigated by statistical analysis of engine test data. The maximum rate of pressure rise is the parameter variable most strongly related to combustion noise, but the maximum rate of heat release ( $RHR_{max}$ ), 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  $RHR_{max}$  and the combustion duration, as in the following equation 1. This equation is valid for the test engine used here (Table 1) at the specific engine speed of 1600 rpm. (This was determined as there is a well-established relation between the  $RHR_{max}$  and  $CA50$ , the parameter variables of the  $RHR_{max}$  and the combustion durations were chosen as the variables.)

$$CN = -0.544(CD) + 0.0275(RHR_{max}) + 85.526 \quad (1)$$

CN: Combustion noise [dBA]

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

$RHR_{max}$ : Maximum rate of heat release [J/°CA]

Further, an engine noise simulation technique, which predicts the combustion noise from heat release data, has been developed and noise characteristics have been investigated [10, 11] from evaluations of the heat release shape and mechanical parameters. Figure 1 shows plots of the combustion noise versus thermal efficiency by the simulations, and suggests that much combustion noise is induced with high thermal efficiency in premixed engine operation.

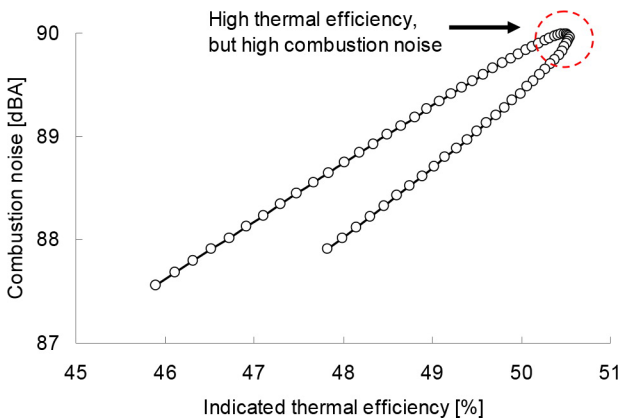


Figure 1. Plot of the indicated thermal efficiency versus combustion noise (Simulation)

To improve on the trade-off relation between combustion noise and thermal efficiency, multiple fuel injections are effective because of the reduction in the maximum rate of pressure rise [12]. Dr. Fuyuto [13] reported the phenomenon that combustion noise produced in the second combustion could assist in reducing the combustion noise of the first fuel injection, and termed this “Noise Cancelling Spike Combustion (NCS combustion)”.

In this paper, the simulations of combustion noise and thermal efficiency in the NCS combustion were conducted with evaluations of the heat release timings and shapes, and engine tests substantiated the findings of the simulations. The test parameter variables are the combustion duration of crank angles of 10% burns in the first and second high temperature heat releases ( $CA10_{1-2}$ ), the duration of the peaks of the maximum rate of pressure rise ( $dP/d\theta_{1-2}$ ), and the heating values of the first and second fuel injections ( $Q_{1st}$  and  $Q_{2nd}$ ), and with the results an optimum heat release shape for a higher thermal efficiency and lower combustion noise is suggested.

## EXPERIMENTAL

### Engine Bench Set up

A schematic outline of the research engine bench is shown in Figure 2. 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.

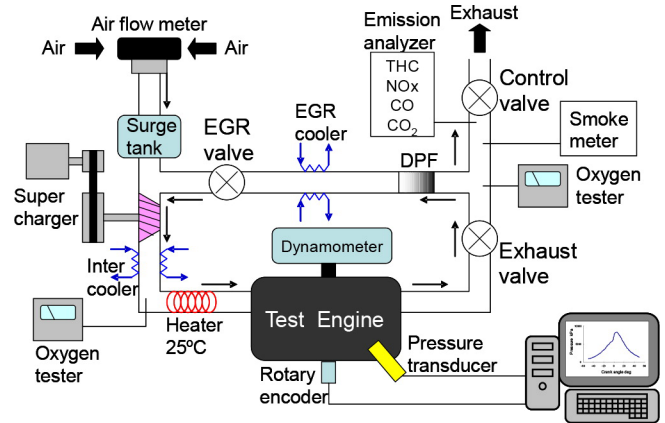


Figure 2. Outline of the engine test set-up

Table 1. Test engine specifications

|                                   |                     |
|-----------------------------------|---------------------|
| Number of cylinders               | 1                   |
| Bore x Stroke                     | φ85 x 96.9 mm       |
| Displacement                      | 550 cm <sup>3</sup> |
| Compression ratio                 | 16.3 : 1            |
| Fuel injection system             | Common rail         |
| Nozzle hole                       | φ0.125 x 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 3. Microphones were set at two different sampling positions at the cylinder head top height (one is 1 m away 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 a 1/3 octave band filter was used for the analysis of the frequency characteristics.

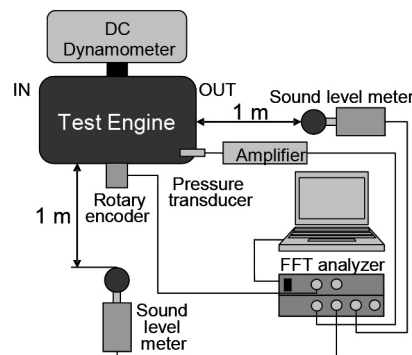


Figure 3. Engine noise measurement set-up

## Calculations of Combustion Noise [14]

The model used to analyze the engine noise is outlined in Figure 4. 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 2, and the cylinder pressure and combustion noise levels 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 3.

The pressure changes produce vibrations, transfer, and attenuate in the cylinder block, before release 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 coherence combustion noise level and cylinder pressure level by equation 4, it was calculated under several conditions and averaged to provide a representative structure attenuation curve. Figure 5 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 5. (The direct noise measurements in Figure 3 are only conducted to determine the engine transfer function  $H$ . Once the structure attenuation is obtained, the combustion noise is calculated by equation 5.)

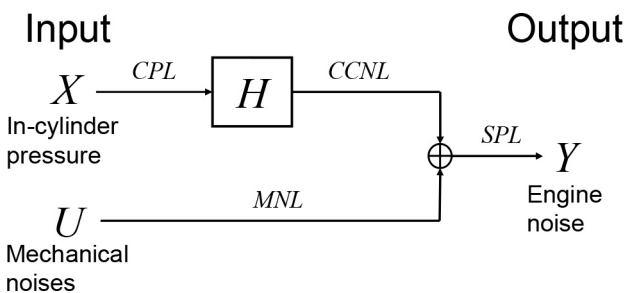


Figure 4. Engine noise generation model

$$SPL = CCNL + MNL \quad (2)$$

$$CCNL = CPL * H \quad (3)$$

$$SA = CCNL - CPL \quad (4)$$

$$CNL = SA + CPL \quad (5)$$

$SPL$  : Sound pressure level [dBA]

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

$MNL$  : Mechanical noise level [dBA]

$H$  : Transfer characteristic

$SA$  : Structure attenuation [dBA]

$CNL$  : Combustion noise level [dBA]

$CPL$  : Cylinder pressure level [dB]

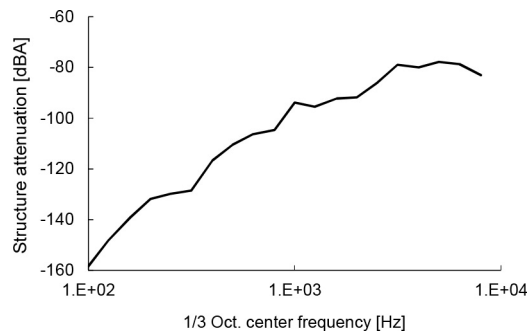


Figure 5. Structural attenuation of the test engine

Figure 6 shows the equal loudness curve of pure sound at 60 phon 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 500Hz and above 8 kHz, and is highest in the 1-4 kHz 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.

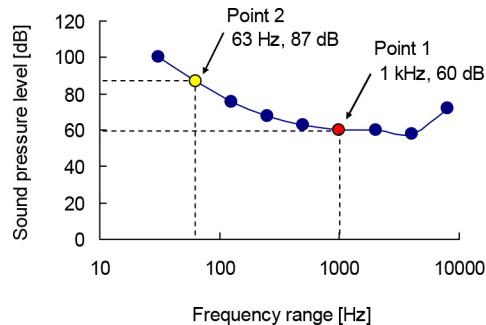


Figure 6. Equal loudness curve of a pure sound at 60 phon for the human ear

## Simulation Methods

### Methods of Combustion Noise Simulation and the Combustion Noise in the First and Second Injections

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 could be further investigated. An example of the heat release shape in the two stage injection is shown in Figure 7. The heat release history was synthesized by applying the Wiebe function in equation 5 three times: one time for the low temperature heat release (LTHR) and one each for the first and second high temperature heat releases (HTHR). In each of these heat release calculations, the parameter  $M$  in equation 5 was arbitrarily set

to 2.0, because the shape of each heat release becomes symmetrical at  $M=2.0$ . (The effects of parameter  $M$  on the combustion noise and heat release is analyzed in the reference paper of 10.)

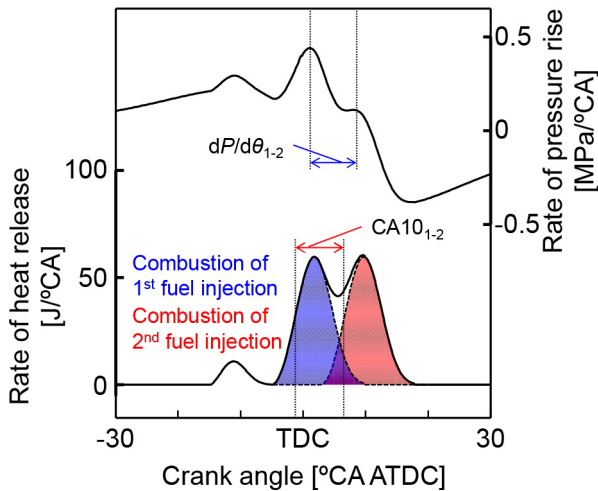


Figure 7. Definitions of crank angle durations of CA10s ( $CA10_{1,2}$ ) and the peaks of the maximum rate of pressure rise ( $dP/d\theta_{1,2}$ ) of the first and second fuel injections

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 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 structural 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]

$\theta_z$  : Combustion duration [°CA]

$M$  : Parameter  $M$  [-]

$\theta$  : Crank angle [°CA]

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

$Q$  : Heat release [J]

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

$P$  : Pressure [Pa]

$V$  : Volume [m<sup>3</sup>]

$\theta$  : 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 (the  $M$  is not 2.0) are plotted together in Figure 8, and the combustion noise data of the experiments and simulation at 100-8000 Hz frequency are plotted in Figure 9. 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.

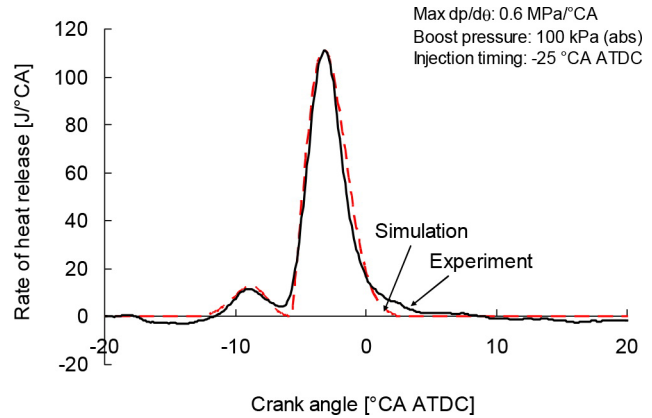


Figure 8. Heat release data in experiment and simulation

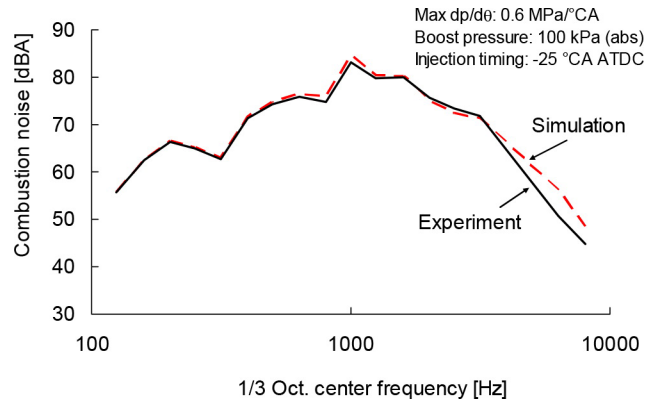


Figure 9. Plot of the combustion noise of the experiment and simulation

In Figure 7, the  $CA10$  values in the first and second HTHRs were calculated and the  $CA10_{1,2}$ , the crank angle duration of the  $CA10$  in the first and second HTHRs was defined as the variable parameter in the simulation, with the second injection crank angle timing was chosen as a parameter variable in the engine tests, because the  $CA10$  of the second fuel injection cannot be determined from the heat release data of the engine tests. Further, the crank angle duration of the peaks of the maximum rate of pressure rise between the first and second fuel injections were defined as  $dP/d\theta_{1,2}$ , and used to calculate the frequency characteristics of noise reduction and amplification in the NCS combustion. Equations 7 and 8 show the amplifying and reduction frequencies. The  $\Delta t$  in the equations is the duration (sec) of  $dP/d\theta_{1,2}$ , and  $\Delta t$  equals to  $(dP/d\theta_{1,2})/12000$  as the engine is operated at 2000 rpm.

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

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

$$\text{Reduction frequency [Hz]: } fn = (2n_C + 1) / 2\Delta t \quad (8)$$

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

### Method of the Indicated Thermal Efficiency Calculations

The cylinder gas composition at each crank angle is calculated for the simulated heat release history. This allows the pressure, mean temperature, specific heats, specific heat ratios, and generated work to be calculated throughout the expansion stroke. The cooling loss is calculated by the Hohenberg equation and the cylinder temperature from IVC timing to EVO timing (the Woschni equation was examined, but the Hohenberg equation was found to agree better with 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 and the cooling loss. Altogether this made it possible to calculate the indicated thermal efficiency.

### Test Conditions for Simulations and Engine Tests

The initial conditions for both the simulations and engine tests were a 12.5% intake oxygen concentration, an engine speed of 2000 rpm and a 175 kPa (abs) boost pressure.

The noise reduction effects by the two stage injection were investigated in Test 1 and confirmed by the engine tests in Test2. Then the optimum heat release shape for high thermal efficiency and low combustion noise was investigated by simulation in Test 3 and proved by the engine tests in Test4.

### Test 1: Effects of the Two Stage HTHR Combustion on Combustion Noise Reduction (Simulation)

The thermal efficiencies and noise reduction effects of the two stage HTHR combustion were investigated by comparing with the combustion and noise data of single stage fuel injection combustion. The test conditions are shown in Table 2 and the heat release patterns in Figure 10. The heating values of the first and second fuel injections of the two stage HTHR combustion were similar at 400J respectively ( $Q_{1st}:Q_{2nd}=400J:400J$ ), the CA10 of the first heat release was maintained at -1.4 °CA ATDC that was the highest thermal efficiency conditions at 2000 rpm for single stage injection, and the CA10<sub>1-2</sub> was varied from 6.5 to 10 °CA. In the single stage injection, the total heating value of the HTHR is 400 J ( $Q_{1st}:Q_{2nd}=400J:0J$ ) and the CA10 is at -1.4 °CA ATDC. A 50 J heating value of the LTHR is assumed in each case, so the total heating value of the single and two stage HTHR combustion stages are 450 J and 850 J respectively.

Table 2. Test conditions for Test 1 (Simulation)

|                                     |                           | Single stage injection | Two stage injection |
|-------------------------------------|---------------------------|------------------------|---------------------|
| Engine speed                        | rpm                       | 2000                   | 2000                |
| Boost pressure                      | kPa (abs)                 | 175                    | 175                 |
| Intake O <sub>2</sub> concentration | %                         | 12.5                   | 12.5                |
| Heating value                       | LTHR                      | J                      | 50                  |
|                                     | 1 <sup>st</sup> injection | J                      | 400                 |
|                                     | 2 <sup>nd</sup> injection | J                      | 0                   |
|                                     | Total heat release        | J                      | 450                 |
| CA10 of 1 <sup>st</sup> injection   |                           | °CA ATDC               | -1.4                |
| CA10 <sub>1-2</sub>                 |                           | °CA                    | -                   |
|                                     |                           |                        | 6.5 to 10           |

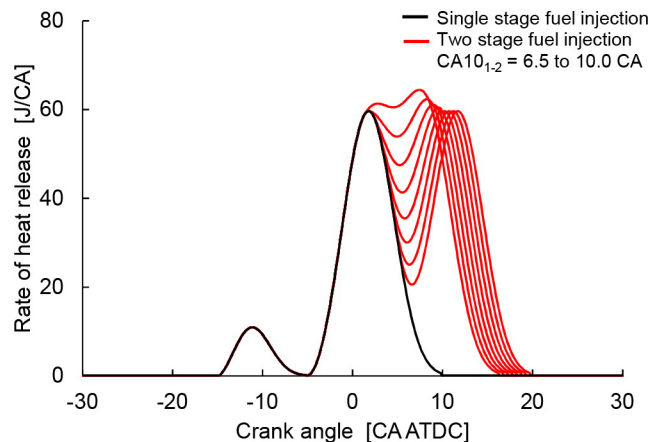


Figure 10. Heat release patterns with Test 1 (Simulation)

### Test 2: Effects of the Two Stage HTHR Combustion on Combustion Noise Reduction (Engine Test)

To evaluate the simulation results in Test 1, engine tests were conducted here, and the thermal efficiency and combustion data with the two stage injections were compared with those of the single stage injection. The test conditions are shown in Table 3. The start of the first injection in the two stage injections was set to -15 °CA ATDC, like in the single stage injection, and the heat release shape of first injection was maintained to be the same heat release shape of single stage injection. Then, the start of the second injection was varied from -2.5 °CA ATDC to 5 °CA ATDC. The IMEPs were 0.4 MPa and 0.8 MPa for the single stage injection and two stage injections respectively. The load of the two stage injection was adjusted to be 0.8 MPa by the second fuel injection.

Table 3. Test conditions for Test 2 (Engine test)

|                                       |           | Single stage injection | Two stage injection |
|---------------------------------------|-----------|------------------------|---------------------|
| Engine speed                          | rpm       | 2000                   | 2000                |
| Boost pressure                        | kPa (abs) | 175                    | 175                 |
| Intake O <sub>2</sub> concentration   | %         | 12.5                   | 12.5                |
| IMEP                                  | MPa       | 0.4                    | 0.8                 |
| SOI of 1 <sup>st</sup> fuel injection | °CA ATDC  | -15.0                  | -15.0               |
| SOI of 2 <sup>nd</sup> fuel injection | °CA ATDC  | -                      | -2.5 to 5           |