

Noise and Emissions Reduction by Second Injection in Diesel PCCI Combustion with Split Injection

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ABSTRACT

An author's previous studies addressed a combustion system which reduces emissions, noise, and fuel consumption by using PCCI with the split injection of fuel. This concept relies on the premixed combustion of the first injected fuel and accelerated oxidation by the second injected fuel. Although this combustion system requires the optimization of the timing of the second injection, the details of how noise and emissions are reduced have not been elucidated.

In this paper, the authors explain the mechanism whereby emissions and noise are reduced by the second injection.

In-cylinder visualizations and numerical simulations both showed an increase in smoke and CO as the second injection timing was advanced, as induced by the inhibited oxidation of the rich flame. When the second injection timing is excessively retarded, the amount of soot forming around the near-nozzle increased. The second fuel injection at the optimum timing can mix with the air in the inner-region of the cavity, such that no soot is formed in the near-nozzle region.

Combustion noise spectra analysis revealed that noise canceling occurs between two peaks in the pressure rise rate at the optimum timing for the second injection, which reduces the overall combustion noise by lowering the maximum frequency component of the noise spectrum. The period of this frequency is twice the interval between the two peaks of the pressure rise rate. We named this noise reduction technique 'noise-cancelling spike combustion', as it relies on the interference of a "spike" in the pressure increase with the preceding peak.

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INTRODUCTION

The widespread use of diesel engines, which have a high thermal efficiency, has been assumed to be one means whereby CO₂ emissions can be reduced and so suppress global warming. At the same time, however, stricter emissions regulations, in particular related to nitrogen oxide (NO_x), have been scheduled in many regions around the world. In general, diesel engines without any emissions aftertreatment system emit more NO_x and soot than gasoline engines with three-way catalytic converters. Nowadays, in the developed countries, it is common for diesel engines to be fitted with an oxidation catalytic converter and/or a diesel particulate filter. To conform with the regulations to be implemented in the future, the addition of a NO_x aftertreatment system, such as a "lean NO_x trap" (LNT) or "selective catalytic reduction" (SCR), would further increase costs. Therefore, many researchers have been

investigating the concept of "premixed charge compression ignition" (PCCI) as a means of reducing NO_x and soot emissions [1, 2, 3].

In the co-author's previous research [4], PCCI combustion with a single injection presented two challenges, as follows;

1. By advancing the injection timing well ahead of top dead center (TDC), the amounts of unburned hydrocarbons (HC) and carbon monoxide (CO), as well as the fuel consumption, increased, although the amount of NO_x and the combustion noise fell.
2. By adjusting the injection timing such that it is closer to TDC, the rate of heat release increases steeply along with the amount of NO_x emissions and combustion noise, although the HC emissions and fuel consumption decrease.

To resolve these issues incurred by single-injection PCCI, the authors have used split injection, controlling the timing of each injection and the amount of fuel that is injected, devising a combustion concept capable of greatly reducing emissions and combustion noise across the wide range of the new European driving cycle. This combustion concept (see Figure 1) was named “diesel staggered premixed ignition with accelerated oxidation (D-SPIA)”. This has two features; premixed combustion of the first injected fuel, and accelerated oxidation of the HC/CO by the combustion of the second injected fuel. Timing the injection around TDC reduces the fuel consumption and splitting the injection into two reduces the maximum heat release rate and thus reduces the combustion noise.

To realize this concept, the control of the second injection timing (interval between the first and second injections) and the amount of fuel that is injected are significant factors. Emission measurements and the second injection timing, as reported in a previous paper, are shown in Figure 2. Low emissions and low-noise combustion were achieved by optimizing the second injection timing. Although the amounts of smoke, CO and combustion noise rapidly increase as a result of advancing or retarding this timing relative to the optimum timing of the second injection, the reason for this has not been explained.

In this paper, the authors revealed the reason for the increase or decrease in the amount of smoke and CO depending on the timing of the second injection. To achieve this, they used both in-cylinder visualizations and numerical simulations. The behavior of the rich flame, soot formation, and soot oxidation were analyzed by using in-cylinder images and a two-color method. The histories of the quantitative equivalence ratio, temperature, and CO emissions were analyzed by using numerical simulations. In addition, combustion noise spectra analysis revealed how the combustion noise is reduced by focusing on the interaction between two peaks in the rate of increase of the combustion pressure.

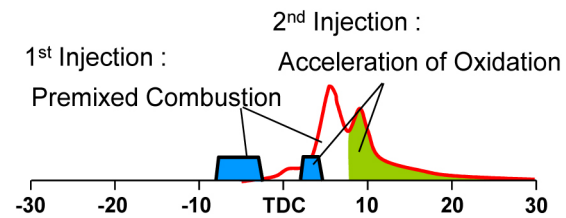


Figure 1. Concept of combustion system

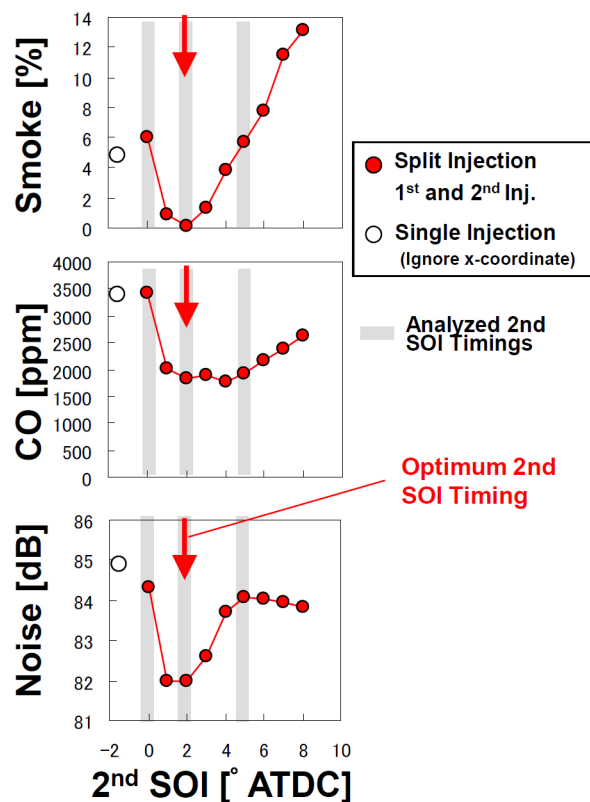


Figure 2. Effects of timing of second injection on smoke and CO emissions, as well as combustion noise. SOI is “start of injection.”

ANALYSIS METHODS

In-Cylinder Visualization

The in-cylinder phenomena occurring between fuel injection and combustion were observed from the bottom of the cylinder by using an optically accessible engine with an elongated piston [5]. The shape of the combustion chamber, with a flat-disk cavity, is shown in Figure 3. The diameter and depth of the cavity were adjusted to approximate the re-entrant piston cavity of a real engine.

A shadowgraph method was used to visualize the fuel spray and ignition process. In addition, a direct-photograph method was used to visualize the luminous flame behavior after ignition. Images were acquired by a high-speed CMOS camera (Photron SA1.1) operating at 20,000 fps. No camera filters were employed for the shadowgraph imaging. The source of the parallel light for the shadowgraph imaging was a halogen lamp. A blue filter was used to allow the transmission of only the blue components of the light from this lamp to make it easy to distinguish between the natural luminosity of the soot and the parallel light of the shadowgraph. Neutral density (ND) filters and the aperture of the lens were used to reduce the exposure for the luminosity imaging. For the two-color method, we used the R-and B-component of each pixel in the direct photograph of the luminous flame. The center wavelength of the R and B filters on the CMOS sensor were 470 nm and 620 nm, respectively.

The intake air was mixed with N_2 gas and CO_2 gas to reproduce the same low- O_2 EGR conditions as in a real engine. Fuel was injected over 20 cycles. After each cycle in which fuel was injected, the optical engine was run without any fuel being injected for three cycles to eliminate the effects of residual gas.

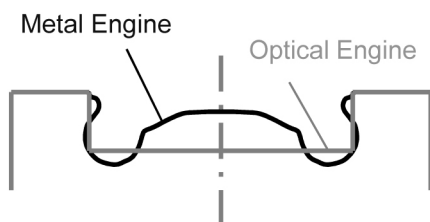


Figure 3. Shapes of combustion chambers

Numerical Simulation

AVL Fire v2009 was used to calculate the in-cylinder spray development and combustion process. This supports the use of a series calculation methodology that links the three-dimensional two-phase nozzle internal flow calculation with the in-cylinder spray/combustion calculation. This improves the accuracy of the spray penetration and mixture formation in the cylinder [6, 7]. First, the internal flow of the nozzle was calculated by taking cavitation into account, after which the calculated profiles at the nozzle exit, including the void fraction and turbulent kinetic energy, were transferred to the in-cylinder spray calculation. The ECFM-3Z combustion model was used

to calculate the CO oxidation in detail [8]. The shape of the combustion chamber for this calculation was the same as that of a real engine. The parameters for the spray model were adjusted so as to be in good agreement with the visualized spray penetration and the mixture formation, while those for the combustion model were adjusted to reproduce the measured heat release rate histories for a metal engine.

Spectral Analysis of Combustion Noise

The combustion noise was determined from the cylinder pressure by using an AVL 450 combustion noise meter with an engine test bench (Figure 4). The frequency spectrum was calculated from the digitally recorded pressure, as follows: First, a pressure trace on a time axis was developed by a Fourier series, and was then converted to a frequency spectrum. The spectrum was first multiplied by a structural frequency filter which simulates acoustic attenuation through an engine block, and was then multiplied by a so-called "A-filter" which applies human acoustic characteristics. These steps generated the combustion noise spectrum that is emitted from the surface of the engine block.

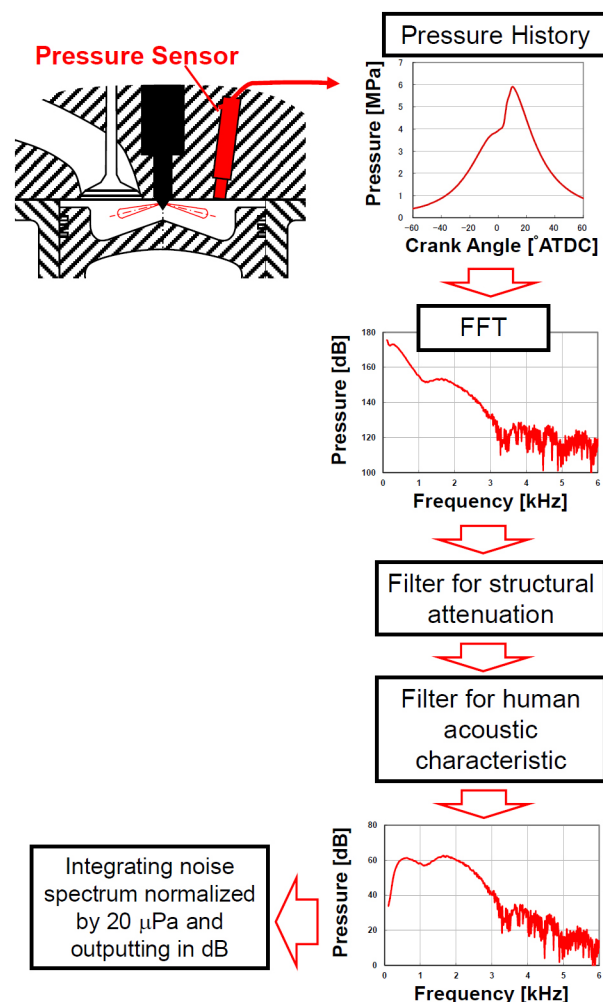


Figure 4. Measuring the combustion noise. An AVL 450 combustion noise meter was used (rather than using a microphone to directly measure the amount of noise being emitted by the engine block).

Engine Specifications and Operating Conditions

Table 1 lists the specifications of the single-cylinder optical engine and four-cylinder real engine used in this study. The bore diameter of both engines was 86 mm. The stroke of the multi-cylinder metal engine was 86 mm and that of the optical engine was 96 mm. The amount of fuel in the first injection was increased for the optical engine because the swept volume of the optical engine is larger than that of the metal engine. Three second injection timings, indicated by the gray lines in Figure 2, were selected; one is the optimum timing at 2° ATDC, and the other two are an advanced timing TDC and retarded timing 5° ATDC, at which both the smoke/CO emissions and combustion noise increase. Figure 5 shows the in-cylinder pressure and heat release rate histories for both the metal engine and the optical engine. The heat release rate history for the metal engine could be very closely reproduced with the optical engine, by maintaining the compression ratio through a reduction in the useless volume of the combustion chamber in the optical engine.

Table 1. Experimental conditions

	Metal Engine & CFD	Optical Engine
Bore x Stroke	86 x 86 mm	86 x 96 mm
Swept Volume	0.5 L	0.55 L
Compression Ratio	15.8	16
Swirl Ratio	2.0	
Nozzle	$\phi 0.09 \times 9$, Piezo	
Engine Speed	1300 rpm	
Injection Pressure	40 MPa	
Injection Amount	10 + 5 mm ³ /stroke.	11.5 + 5 mm ³ /st.
EGR Rate	40	-
O ₂ Concentration	-	14.5 %
Equiv. Ratio (A/F)	0.74 (19.6)	
BMEP	0.37 MPa	-

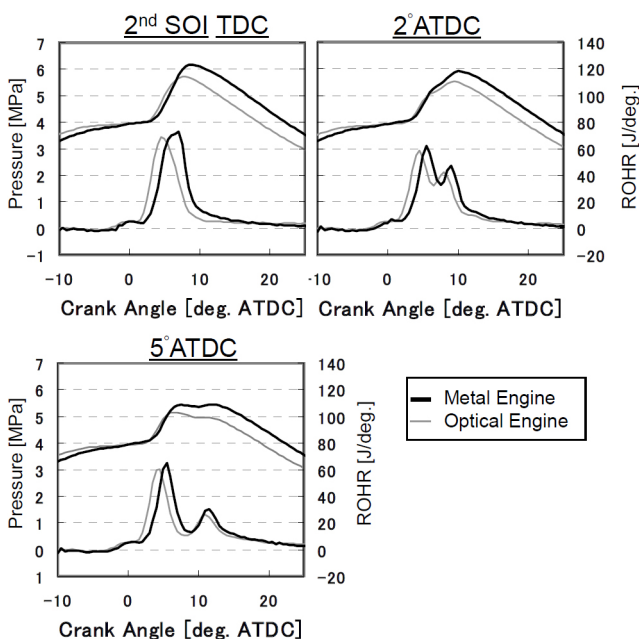


Figure 5. Comparison of in-cylinder pressure and rate of heat release of metal engine and optical engine.

REDUCTION OF SMOKE AND CO EMISSIONS

Increase in Smoke and CO Emissions at Advanced Second Injection Timing

Comparison of Rich Flame and CO Emissions through Numerical Simulation

Figure 2 shows that the smoke and CO emissions increase as the second injection timing is advanced to TDC from the optimum timing of 2° ATDC, thus shortening the interval between the first and second fuel injections. When the interval is set to 0, that is, when there is only a single injection, the smoke and CO emissions were the same as those produced when the second injection timing is TDC (as shown by the white circles in Figure 2). As the smoke and CO emissions increase, an analysis using ϕ -T maps indicated inhibited mixing/oxidation of the rich flame.

Figure 6a shows a ϕ -T map of the CO concentrations as a function of temperature and stoichiometry [9, 10]. Isothermal and isobaric (6 MPa) homogenous reactor calculations were performed by using a detailed chemical reaction model. The chemical reaction was calculated by using n-heptane as the initial fuel. The CO concentration after a residence time of 1 ms relative to the start of the calculation is indicated by the level of gray shading. The soot and NO_x formation zones overlap on the ϕ -T map. The closed curve indicates a typical distribution of in-cylinder gas in the latter part of the combustion period. The dotted rectangle ($T > 1500$ K, $\phi < 1$) indicates the zone in which the CO can be oxidized in a short time [10]. The ϕ -T map shown in Figure 6b plots the local CO mass fraction, ϕ and the temperature at 15° ATDC for each cell of the three-dimensional numerical simulation for a second injection timing of 2° ATDC. The CO mass fraction in each cell is indicated by the level of gray shading. As the CO oxidation proceeds in the CO oxidation zone in the latter part of the combustion period, the CO distribution shown by the ϕ -T map splits into two parts. One part is called the "low-temperature CO" in this paper. In this part, CO oxidation stops due to the temperature being too low. The second part is called the "rich CO." Here, the oxidation of the CO is inhibited by a shortage of oxygen, even while the local temperature is sufficiently high. Figure 6b shows that the majority of the CO in the latter part of the combustion period in this combustion concept is rich CO. Soot, the major component of smoke, exists in the rich-flame region ($\phi > 1$) within the closed curve of Figure 6a, where the amount of oxidation stagnates because of a shortage of oxygen. When the rich flame moves into the CO oxidation zone ($\phi < 1$) and mixes with the surrounding residual oxygen, the oxidation of the CO and soot proceeds.