

Low Temperature Premixed Diesel Combustion with Blends of Ordinary Diesel Fuel and Normal Heptane

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ABSTRACT

Premixed diesel combustion blending high volatility fuels into diesel fuel were investigated in a modern diesel engine. First, various fractions of normal heptane and diesel fuel were examined to determine the influence of the blending of a highly ignitable and volatile fuel into diesel fuel. The indicated thermal efficiency improves almost linearly with increasing normal heptane fraction, particularly at advanced injection timings when the fuel is not injected directly into the piston cavity. This improvement is mainly due to decreases in the other losses, ϕ_{other} which are calculated with the following equation based on the energy balance.

$$\phi_{other} = \eta_u - \eta_i - \phi_{ex}$$

η_u : The combustion efficiency calculated from the exhaust gas compositions

η_i : The indicated thermal efficiency

ϕ_{ex} : The exhaust loss calculated from the enthalpy difference between intake and exhaust gas

The decreases in the other losses with normal heptane blends are due to a reduction in the unburned fuel which does not reach the gas analyzer.

Next, a blend of 50 vol% iso-octane and 50 vol% diesel fuel was examined to determine the influence of blending a low ignitability and high volatility fuel similar to gasoline into diesel fuel. With the iso-octane blend the range of premixed diesel combustion can be expanded to higher intake oxygen concentrations due to a longer premixing duration with increasing ignition delay, and smokeless combustion is possible at low intake oxygen concentrations. The unburned fuel arising from adhesion of fuel spray on the cylinder wall can be avoided with the high volatility fuel, resulting in higher efficiency.

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INTRODUCTION

Low temperature premixed diesel combustion with longer ignition delays due to low intake oxygen concentrations with large quantities of exhaust gas recirculation (EGR) can establish simultaneous reductions in the nitrogen oxide (NO_x) and particulate emissions in current diesel engines.^{1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11} Further, compared with conventional diesel combustion, premixed diesel combustion offers the potential to improve the thermal efficiency with reductions in the cooling loss due to lower combustion temperatures and increases in the degree of constant volume heat release due to shorter combustion durations. However, premixed diesel combustion occasionally causes intense heat release events, resulting in larger combustion noise, and the combustion characteristics strongly depend on the fuel properties.^{6, 7, 8, 9, 10} With low volatility fuels including

diesel fuel, advanced fuel injection to maintain the premixing duration causes increases in unburned fuel amounts due to the impingement of the fuel sprays on the cylinder wall or residual fuel in the squish area and piston crevices, causing the deteriorations in thermal efficiency. The authors have reported that the thermal efficiency improved significantly with high volatility fuels such as normal heptane due to suppression of the unburned fuel amounts in the premixed diesel combustion.¹¹

The purpose of this study is to determine the influence of blending high volatility fuels including normal heptane and iso-octane into diesel fuel on the characteristics of premixed diesel combustion. With the blends of normal heptane the influence of the blending of high volatility fuels with ignitabilities similar to diesel fuel are investigated, and the influence of blending a low ignitability and high volatility fuel like gasoline is evaluated with an iso-octane blend.

EXPERIMENTAL SETUP

The experiments were conducted on a single cylinder, boosted, four-stroke cycle, direct injection diesel engine with common rail and low-pressure loop cooled EGR, with the specifications of the test engine shown in [Table 1](#).

[Table 2](#) shows the experimental conditions. The intake gas was boosted at 175 kPa (absolute) with a supercharger driven by an electric motor, and the intake gas temperature was maintained at 40°C with an intercooler and heater attached after the supercharger. The cooled EGR was realized with the exhaust gas passed through a diesel particulate filter (DPF) and an EGR cooler. The recirculated exhaust gas was mixed with the intake air before the supercharger with the low-pressure loop. The exhaust gas pressure was maintained with a valve at 175 kPa (absolute), the same as the boost pressure, modifying the turbo-charging. The engine speed was everywhere set at 2000 rpm, the fuel injection pressure at 160 MPa, and the indicated mean effective pressure (IMEP) at 0.8 MPa.

[Table 3](#) shows the properties of the tested fuels. First, several fractions of normal heptane and ordinary diesel fuel (JIS No.1) blended to give 0 vol%, 25 vol%, 50 vol%, 75 vol%, and 100 vol% of normal heptane fractions in the blended fuels (the XX vol% normal heptane blend fuel in the following) were examined to evaluate the influence of blending high volatility fuels with ignitabilities similar to diesel fuel. As shown in panel A of [Table 2](#), the study used three fuel injection timings set at 10°CA, 25°CA, and 40°CA before top dead center (BTDC) and at one intake oxygen concentration which was set at 11% with the cooled EGR. [Figure 1](#) shows the positions of the injected fuel sprays and piston at several fuel injection timings. When fuel is injected at 40°CA BTDC, almost all of the fuel sprays remain in the squish area and impinge on the cylinder wall. At 25°CA BTDC, the fuel sprays impinge on the lip of the piston cavity and a portion of the spray remains in the squish area. At 10°CA BTDC, almost all of the fuel sprays remain in the piston cavity.

Next, a blend of 50 vol% iso-octane and 50 vol% diesel fuel (the 50 vol% iso-octane blend fuel in the following) was examined to evaluate the influence of blending a low ignitability and high volatility fuel similar to gasoline into the diesel fuel. As shown in panel B of [Table 2](#), the intake oxygen concentrations were set between 9% and 15% with the cooled EGR. The 50% heat release crank angle (CA50) was maintained at the optimum timing (4°CA after top dead center (ATDC)) for the indicated thermal efficiency changing the fuel injection timing. Lubricity improver (R655) was added to the normal heptane and iso-octane in a volume fraction of 300 ppm.

In this study, the other losses, ϕ_{other} are calculated with the following equation:

$$\phi_{other} = \eta_u - \eta_i - \phi_{ex}$$

η_u : The combustion efficiency calculated from the exhaust gas compositions

η_i : The indicated thermal efficiency calculated from the fuel consumption and the in-cylinder pressure

ϕ_{ex} : The exhaust loss calculated from the enthalpy difference between intake and exhaust gas

The other losses, ϕ_{other} are almost entirely comprised by the cooling losses, but a portion is unburned fuel losses which do not appear in the exhaust gas analyzer data due to fuel entering the crank case or passing through the sample port of the gas analyzer.

The exhaust gas was analyzed with an automotive exhaust gas analyzer (MEXA-1600: HORIBA) including NDIR (non-dispersive infrared absorption) for CO and CO₂, CLD (chemical luminescence detector) for NO_x, and HFID (heated flame ionization detector) for THC. The oxygen concentrations in the intake and exhaust gases were measured with a portable oxygen tester (POT-101: SHIMAZU), and the smoke concentration was measured with a Bosch-type smoke meter.

[Table 1. Specifications of the test engine](#)

Engine type	DI single cylinder
Bore and stroke	φ 85 × 96.9 mm
Displacement	550 cm ³
Compression ratio	16.3
Nozzle hole	φ 0.125 × 7 - 156°

[Table 2. Experimental conditions](#)

Combustion mode	Premixed diesel combustion
Engine speed	2000 rpm
IMEP	0.8 MPa
Fuel injection pressure	160 MPa
Boost pressure (abs.)	175 kPa
Intake gas temperatures	40°C
Coolant temperatures	80°C
Lubricant temperatures	80°C
A. Influence of high ignitability and volatility fuel fraction	
Oxygen concentration	11%
Fuel injection timings	10°, 25°, 40°CA BTDC
n-Heptane fraction	0 vol% – 100 vol%
B. Influence of low ignitability and high volatility fuel	
Oxygen concentration	9% or misfire – 15%
Fuel injection timings	Varied for CA50
CA50	4°CA ATDC

Table 3. Properties of the tested fuels

Property	JIS 1 diesel	Normal heptane	iso-Octane
Density [g/cm^3]	0.834	0.689	0.692
10% Distillation temperature [$^{\circ}\text{C}$]	217	98.4 ^a	99.2 ^a
50% Distillation temperature [$^{\circ}\text{C}$]	281		
90% Distillation temperature [$^{\circ}\text{C}$]	342		
Carbon [wt%]	86.33	83.90	84.04
Hydrogen [wt%]	13.7	16.10	15.95
Lower heating value [MJ/kg]	42.9	44.56	44.4
Octane number (RON) ^b	—	0	100
Cetane number	54.2	—	—

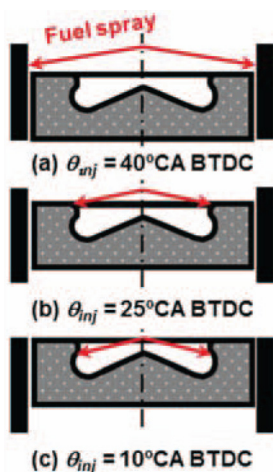
^a Boiling point^b Research octane number

Fig. 1. Positions of the piston and fuel spray at different fuel injection timings for the premixed diesel combustion

EXPERIMENTAL RESULTS AND DISCUSSION

Influence of the Blended Fraction of High Ignitability and Volatility Fuel Into Diesel Fuel

Figure 2 shows the in-cylinder pressures and the rates of heat release (ROHR) with normal heptane, diesel fuel, and the 50 vol% normal heptane blend fuel at the three fuel injection timings, 10°CA, 25°CA, and 40°CA BTDC. For all the fuel injection timings and fuels, the main combustion occurs after the low temperature heat release (LTHR), showing typical premixed diesel combustion. At the injection timings of 25°CA and 40°CA BTDC, the main combustion occurs before TDC and is almost complete at TDC, while at the

injection timing of 10°CA BTDC the main combustion starts after TDC with short combustion durations, showing heat releases suitable to establish high thermal efficiency. The combustion phases of the injection timings at 25°CA and 40°CA BTDC are similar, and the main combustion does not change when advancing the injection timing further than 25°CA BTDC.

Figure 3 shows the premixing duration, θ_{premix} defined as the crank angle from the end of fuel injection to the ignition (crank angle at 10% of the heat release) for various normal heptane fractions under the conditions in Figure 2. The premixing duration decreases linearly and slightly with increasing in the normal heptane fraction and the degree of change with the normal heptane fraction is more significant at the 40°CA BTDC fuel injection than at the later fuel injections. This result suggests that the sensitivity of the ignition delay on the fuel volatility at early injection timings is larger due to the lower in-cylinder temperature.

Figure 4 shows the indicated thermal efficiency, η_i ; the combustion efficiency calculated from the total hydrocarbon (THC) and the carbon monoxide (CO) concentrations in the exhaust gas, η_u ; the degree of constant volume heat release, η_{gh} ; the exhaust loss from the temperature and quantity of exhaust gas, ϕ_{ex} ; the other losses, ϕ_{other} ; the cooling loss with Woschni's equation, $\phi_{Woschni}$; and the quantity ratio of the exhaust and the intake carbon, C_{ex}/C_{in} for various normal heptane fractions under the conditions in Figure 2. The indicated thermal efficiencies at the 40°CA BTDC fuel injection timing are much lower than at the other timings due to the lower combustion efficiency, and the indicated thermal efficiencies improve almost linearly with increases in the normal heptane fraction. At the 25°CA BTDC and 10°CA BTDC fuel injection timings, the degrees of improvement in the indicated thermal efficiencies are smaller than at the 40°CA BTDC fuel injection timing, but the improvement is still almost linear with increasing normal heptane fractions. For all three injection timings the improvement of the indicated thermal efficiency with increasing normal heptane fractions is mainly due to the decrease in the other losses, ϕ_{other} . The other losses are mainly the cooling loss, but a portion is the unburned loss, the portion of the exhaust not appearing in the gas analyzer due to the fuel entering the crank case or escaping through the sample port of the gas analyzer. In the results here the other losses decrease due to reductions in the unburned fuel which does not appear in the exhaust gas analyzer, as the cooling loss with Woschni's equation, $\phi_{Woschni}$ does not change. This is also supported as the carbon quantity ratio of the exhaust and the input, C_{ex}/C_{in} increases with increasing normal heptane ratios and the degree of increase is the largest at the 40°CA BTDC fuel injection timing. Here, some of the carbon quantity ratios, C_{ex}/C_{in} are above 100% due to the inaccuracy in the absolute concentrations of CO_2 or in the exhaust gas volume, the relative accuracies are adequate however.