Flash Boiling Spray of Diesel Fuel Mixed with Ethane and Its Effects on Premixed Diesel Combustion

Y. Kobashi[1]*, S. Hirako[2], A. Matsumoto[2], K. Naganuma[2]

[1] Division of Energy and Environmental Systems, Graduate School of Engineering, Hokkaido University, Japan

[2] Department of Mechanical Engineering, Kanazawa Institute of Technology, Japan

Abstract

Mixtures of gaseous and liquid fuels have the potential to induce flash boiling under in-cylinder conditions close to the top dead center, something that can be expected to be suitable for a premixed diesel combustion that demands a way to enhance lean mixture formation. The present study mixes ethane, which is a natural gas component, into diesel fuel as ethane has a high vapor pressure that facilitates flash boiling and as ethane has a low cetane number that prolongs the ignition delay. This paper investigates spray characteristics of a diesel fuel – ethane mixture, and the engine performance and exhaust emissions are evaluated in a single cylinder engine. The test results show that the flash boiling enhances the lean mixture formation, and realizes combustion with low soot and low NOx without EGR (exhaust gas recirculation) while also improving thermal efficiency. An additional test was performed to identify the effects of the flash boiling and the ignition delay, and the results show that the long ignition delay plays the

* Corresponding author at: Kita 13, Nishi 8, Kita-ku, Sapporo, Hokkaido, Japan
E-mail address: kobashi@eng.hokudai.ac.jp (Y. Kobashi)

most important role in the thermal efficiency and emission changes and that the flash boiling facilitates the formation of leaner mixtures that lower the overall combustion rate and achieves very low NO_x emissions.

Keywords: Flash Boiling, Premixed Diesel Combustion, Ethane, Mixed Fuel, Emissions

Nomenclature:

$dp/d\theta_{max}$	Maximum pressure rise rate [MPa/deg.CA]
$dQ/d\theta$	apparent heat release rate [J/deg.CA]
ΔP_{sat}	difference between the pressure on bubble point curve and the in-cylinder pressure [MPa]
p_{inj}	fuel injection pressure [MPa]
Q_{1st}: Q_{2nd}	ratio of first and second injection quantities
V_{et}	volume fraction of ethane in the mixed fuels [vol.%]
z	distance from injection nozzle [mm]

Greek symbol:

η_i	indicated thermal efficiency [%]
ρ_a	ambient density [kg/m^3]
θ_{inj}	signal incident timing of fuel injection (single injection case) [deg.CA ATDC]
$\theta_{inj.1st}$	signal incident timing of first fuel injection [deg.CA ATDC]
$\theta_{inj.2nd}$	signal incident timing of second fuel injection [deg.CA ATDC]
τ_{ign}	ignition delay defined as the period from the injection signal incidence until the start of the main combustion

1. Introduction

In modern diesel engines employing premixed diesel combustion, high-pressure fuel injection with advanced timings would promote atomization and facilitate the leaner mixture formation, and exhaust gas recirculation (EGR) has been used to lengthen ignition delays and to ensure the formation of the premixed mixture in diesel combustion [1-2]. Despite these advantages, earlier timings of fuel injection with high pressures increase the unburned emissions due to the fuel reaching the squish area in the combustion chamber [3], and high EGR rates make it difficult to control the intake oxygen concentration in transient operation due to the slowness of the response.

Gasoline-fueled compression ignition engines have attracted increasing attention [4-6], since the low reactivity and high volatility of gasoline are suitable qualities characteristics to enhance lean mixture formation. Manente et al. demonstrated 57% indicated thermal efficiency in a single cylinder heavy duty with the gasoline-fueled compression ignition [7]. In this strategy, however, the ignition timing control for a wide range of operating conditions is still challenging as commonly-available gasoline is too resistant to auto-ignition. To make the operation

easier while retaining the advantage of the lean mixture formation, won et al. proposed the use of blends of gasoline and diesel fuel [8], since it is preferable to have fuels with a better reactivity than those of gasoline, and it is practical to blend those commercially-available fuels. Researchers at University of Birmingham applied the blends of gasoline and diesel fuel as well, and the mixing fractions were optimized to improve the premixed diesel combustion [9-10].

Prior to this attracting attention, researchers at Doshisha University proposed that the use of mixed fuels consisting of a high volatility fuel with low reactivity (such as gasoline and gaseous fuels) and a high reactivity fuel with low volatility (such as diesel fuel) would make it possible to obtain better characteristics of evaporation and control of the ignition timing [11]. In addition to the above benefits, they suggested that the high volatility fuel induces flash boiling when the fuel is discharged into a combustion chamber with lower pressure than its saturation pressure [12]. Following Senda, Wada et al. [13-14] applied mixed fuels consisting of *i*-pentane (C_5H_{12}) and *n*-tridecane ($C_{13}H_{28}$) to a premixed charge compression ignition engine with very early injection. Anitescu et al. visualized the flash boiling sprays of the blends of gasoline and diesel fuel, injected into ambient air [15]. However, there have been only a few studies addressing a flashing spray injected into engine-relevant conditions near top dead center.

As one of them, the previous paper by the authors introduced a flashing spray in the premixed diesel combustion with late injections [16]. Gaseous fuel, ethane was used to achieve the flash boiling even near top dead center, and it was mixed into diesel fuel to ensure successful compression ignition. Another study reported the results in the literature [12], where liquid petroleum gas (LPG) was mixed into *n*-tridecane ($C_{13}H_{28}$), and the mixed fuel was introduced into the conventional diesel combustion with late injections while heating the fuel to ensure the flash boiling conditions. Both the studies reported the improvements of the efficiency and the emissions with the flashing sprays. However, the characteristics of the flash boiling spray injected into an engine-relevant high ambient density conditions were not examined, and the mechanism that improved the combustion was not figured out.

In the present paper, the spray characteristics of the diesel - ethane mixed fuel were investigated at a variety of mixing fractions. Engine tests were performed while introducing the flashing spray into the premixed diesel combustion, and data analysis was carried out to clarify the combustion improvement mechanism and to isolate the effects of the flash boiling and the ignition delay on the engine performance and the exhaust emissions.

2. Experimental Setup and Conditions

2.1 Fuels tested

The fuels mainly tested here are mixtures of ethane and JIS No.2 diesel fuel in a variety of mixing fractions. The properties of the tested fuels are detailed in Table 1. The JIS No.2 Diesel fuel has a high boiling point and high reactivity, and ethane has a low boiling point and the octane number of 108. A pressure-temperature diagram of the diesel fuel – ethane mixed fuel, calculated with SUPERTRAPP (NIST), is shown in Fig.1. Diesel fuel consists of multiple components, and it was replaced by *n*-tridecane ($C_{13}H_{28}$) in the calculations. With increasing ethane fraction, V_{et} shifts the saturated liquid curve to higher pressures, and it can be expected that the flash boiling will take place at the in-cylinder pressure with a high fraction of ethane. The present study employed the difference between the pressure on the saturated liquid curve and the ambient pressure, termed ΔPsat, to express the degree of the flash boiling. A part of these experiments use a mixed fuel consisting of *i*-octane and Shellsol (MC421) to distinguish the effects of the flash boiling and the ignition delay as Shellsol is mainly composed of large isoparaffins with high distillation temperatures and low cetane numbers, and a mixed fuel of this kind would not induce flash boiling in the engine cylinder. The mixed-in fraction of Shellsol was determined to match the ignition timing with the mixed fuel of ethane and diesel fuel.

[Table 1]

[Fig.1]

2.2 Fuel supply system

Figure 2 shows a diagram of the fuel supply system. To maintain ethane in the liquid phase during introduction into the fuel supply system, a bladder accumulator (at ② in Fig.2) pressurized with nitrogen was used and the pressure in the flow path of ethane was maintained above the vapor pressure. When the flow path was filled with ethane, the ethane was introduced into the mixing tank (at ① in Fig.2) where diesel fuel was already accumulated, and the mixture of ethane and diesel fuel was agitated at 600 rpm for 30 minutes. Then the mixed fuel was transferred to the double-acting cylinder (③) facilitated by the intensifier (④) to be compressed to the injection pressure of 40 MPa.

[Fig.2]

2.3 Spray visualization in a constant volume vessel

The spray was visualized in a constant volume vessel with shadowgraph photography. The vessel equips with two quartz glass windows lying in the same optical

path line. The diameter and thickness of the quartz glasses are 120 mm and 45 mm, respectively, and 90 mm in diameter is available as a field of view. A fuel injector with a single hole nozzle (0.1 mm in diameter) was attached at the top of the vessel.

The arrangement for the shadowgraph photography is schematically shown in Fig.3. A xenon lamp was used as the light source with a plano-convex lens (100 mm diameter, 1000 mm focal length) which collimates the light, and the collimated light passing through the vessel was focused on a lens of a high-speed video camera (SA1.1, Photron) with the second plano-convex lens (100 mm diameter, 1000 mm focal length). The spray images were taken with the recording rate of 30000 frames per second (fps), and the light intensity was adjusted with an ND filter. The spatial resolution for this setup was 0.072 μm/pixel.

The analysis began while subtracting a background image without the injection from spray images, in order to eliminate the background noise. An intensity threshold level which was used to separate the spray region from the background ambient gas was carefully determined with the as-recorded spray images of V_{et} = 60 % as this fuel was most volatile, and it was most difficult to separate the spray region. Pixels with intensity levels below the threshold were defined as the spray region, and the spay angles were measured. The spray tip penetration was measured with the ensemble-averaged images.

The experimental conditions are detailed in Table 2. The vessel was filled with the 0.90 MPa CO_2 at the room temperature to achieve the engine-relevant ambient density of 16.1 kg/m^3. The fuel was injected from a single hole 0.10 mm diameter nozzle. The pressure in the vessel was lower than the in-cylinder pressure, and ΔP_{sat} in the spray experiment was positive when the ethane fraction, V_{et} was above 30 vol.%. All the ΔPsat detailed in Table 2 were calculated assuming the fuel temperature of 300 K in Fig.1.

[Figure 3]

[Table 2]

2.4 Engine test

The specifications of the single cylinder diesel engine used in the present study are detailed in Table 3. The single cylinder diesel engine equips with a toroidal type combustion chamber. The bore and stroke are 83 and 85 mm, the displacement volume is 460 cc, and the compression ratio is 19.3. The fuel injection system introduced in Fig.2 was used, and a 10 hole nozzle with the nominal hole diameters of 0.101 mm was employed. The experimental conditions are detailed in Table 4. The engine rotation speed was maintained constant at 1000 rpm. The mixed-in fraction was varied as an

experimental parameter while adjusting the fuel flow rate to maintain the constant heat supply of 400 J/cycle. As the primary purpose of the present paper is to investigation the relation between the spray characteristics of the diesel fuel – ethane mixed fuel and the characteristics of the premixed diesel combustion, the engine tests were performed at the constant rotation speed and engine load. The variations of these experimental parameters were not tested but should be examined in future work.

Table 5 provides the accuracy and the detection limit of measurements. The soot concentration was measured with a smoke meter (AVL, 415S). As the accuracy is unknown, the detectable limit is detailed in Table 5. The exhaust gas concentrations of CO, CO_2 and NO_x were measured with an FT-IR exhaust gas analyzer (HORIBA, MEXA-6000FT), and THC emissions were measured with an FID analyzer (HORIBA, MEXA-1170HFID). The accuracy of all the gas concentrations was within ±1% FSO. The in-cylinder pressures were measured with a pressure transducer (Kistler, 6125C) which has accuracy better than ±0.4% of full scale output (FSO), while a charge amplifier (Kistler, 5010B) was used to convert the charge signal from the pressure transducer into an output voltage. Unlike the experiments where repetition of measurements is possible, so that an error analysis is performed, as in Ref. [17] dealing with a hybrid cooling-drying system, it is difficult for engine tests to repeat many tests

at each operating point due to the many data points involved, and in engine tests, cycle-by-cycle variations in the combustion processes may cause a significant error and deteriorate the accuracy of measurements. In the present study, to mitigate the effects of the cycle-by-cycle variations, the data obtained in the experimental conditions where the coefficients of variance of indicated mean effective pressure (IMEP) were less than 3% were used for the evaluation, except for some data in Fig.10. The profiles of the in-cylinder pressure were recorded for 120 cycles, and the averaged pressure was employed to calculate the apparent heat release rate. The exhaust gas concentrations were measured for 30 seconds, and the averaged values were adopted.

[Table 3]

[Table 4]

[Table 5]

3. Results and Discussion

3.1 Spray Characteristics of the Diesel Fuel – Ethane Mixed Fuels

Figure 4 shows the spray shape near the nozzle for different volume fractions of ethane, V_{et}. The sprays with positive ΔP_{sat} which may possibly induce flash boiling, disperse in the radial direction directly at the nozzle exit (top of figure) as the fuel flow

containing bubbles formed in the nozzle expands immediately after the discharge. The temporal changes of the whole spray images are shown in Fig.5, clearly showing that the spray width increased with V_{et}. The flash boiling phenomenon enhances the liquid atomization and vaporization and the formation of droplets, promoting the dispersion further in the downstream direction, due to the turbulent motion.

[Fig.4]

[Fig.5]

The spray cone angles and the spray angles are shown in Fig.6. The spray cone angle was defined based on the spray width at z = 5 mm with the shadowgraph images enlarging the region close to the nozzle, and the whole spray images were used to define the spray angle based on the maximum spray width from z = 25 mm to 38 mm. Using 10 sets of the frames, the averaged angles as well as the extent of the fluctuation were calculated. By considering the spatial resolution 0.072 μm/pixel, the resolution of the spray cone angle measured at z = 5 mm is 0.80 degree/pixel, and that of the spray angle measured at z = 32 mm is 0.13 degree/pixel. Taking into account the resolution and the extent of the data fluctuation, both these angles clearly increase with increasing V_{et}. The spray cone angle is more sensitive to V_{et} than the spray angle, as the initial spray dispersion is strongly affected by the flow during the passage of the nozzle (by the

bubbles growth due to the phase-change) and the spray dispersion further downstream is affected by the turbulent motion of the surrounding gas.

In Fig.7, the relationships between the ΔP_{sat} and the spray angle are plotted against the various ambient densities, ρ_a. The open symbols are the data from the previous study using a heated mixed fuel of *n*-tridecane and *i*-pentane (Wada et al., 2007) and the solid symbols are the data obtained in this study. It is difficult to compare the data from the present and the previous study directly due to the differences in the definition of the spray angle, while it is possible to compare the gradient against the ΔP_{sat}. The previous study showed that the effect of the flash boiling on the spray dispersion was less pronounced at the high ambient density conditions, while the present data showed that the spray angle increased evenly with increasing ΔP_{sat} at all the ambient densities here, likely as the diesel fuel – ethane mixed fuel has the higher ΔP_{sat}. Suggesting that a more pronounced effect of flash boiling can be expected by using the diesel fuel – ethane mixed fuel in the engine test, which will be reported below.

[Fig.6]

[Fig.7]

A plot of the spray tip penetration with the diesel fuel – ethane mixed fuel is

shown in Fig.8. The initial spray tip penetrations are largely proportional to the time from the start of injection also with the flash boiling, likely as the dense region in the center of the spray is maintained. At around 0.15 ms after the start of injection, the spray tip penetration is proportional to the square-root of the time, implying that the exchange of momentum between the spray and the surrounding gas has become active. At this point, the part of the fuels with a large ethane fraction lose momentum faster due to the flash boiling effect.

[Fig.8]

3.2 Effects of the Ethane Fraction on Engine Performance and Exhaust Gas Emissions

Figure 9 shows the profiles of the in-cylinder pressure, p and the apparent heat release rate, $dQ/d\theta$ at the time of the fuel injection signal, θ_{inj} at -4 deg.CA ATDC. Considering that the delay in arrival of the injection signal from the actual injection is approximately 4 deg.CA, the diesel fuel auto-ignited immediately after the start of the actual injection. With increasing ethane fraction, V_{et}, the ignition delay increased, and the low temperature heat release (LTHR) which is typically observed in premixed combustion appears prior to the main combustion of V_{et} = 50 vol.%. At this θ_{inj}, it was difficult to establish the combustion with the coefficient of variance of indicated mean

effective pressure (IMEP) less than 5% with V_{et} = 60 and 65 vol.% due to the long ignition delay.

[Fig.9]

Figure 10 shows the variations of the indicated thermal efficiency, η_i and the maximum pressure rise rate, $dp/d\theta_{max}$ with V_{et} and θ_{inj}. The indicated thermal efficiency decreased with increasing V_{et} from 0 to 50 vol.% as the longer ignition delay retarded the combustion and decreased the degree of constant volume. The early injection timing was expected to advance the combustion phase and improve the degree of constant volume, however it deteriorated the indicated thermal efficiency due to the increased cooling loss, and additionally the advanced injection was limited by the very large increase in $dp/d\theta_{max}$. With V_{et} at 60 and 65 vol.%, a better indicated thermal efficiency was obtained with the earlier injection timing around -25 deg.CA ATDC. The longer ignition delays and the flash boiling with these fuel mixtures enabled the formation of leaner mixture combustion that lowered the flame temperature. However, further advances in the injection deteriorated the η_i due to the formation of an excessively lean mixture while retarding the injection deteriorated the η_i, due to the excessively late combustion phase.

[Fig.10]

To further identify the advantages of the diesel fuel - ethane mixed fuels more specifically, the best (optimum) injection timings, those which resulted in the best indicated thermal efficiencies were selected, and the profiles of the optimum apparent heat release rates, $dQ/d\theta$ are shown in Fig.11. The optimum combustion phase with V_{et} = 0, 30 and 50 vol.% was later than that of the V_{et} of 60 and 65 vol.% possibly as the flame temperatures of the V_{et} = 0, 30, and 50 vol.% may be higher and the later combustion resulted in a reduced cooling loss. The leaner mixture of V_{et} = 60 and 65 vol.% formed by the longer ignition delays and the flash boiling may lower the flame temperatures, so that advancing the combustion phase and avoiding the cooling loss made the increase in the degree of constant volume possible.

[Fig.11]

Figure 12 shows the changes in ΔP_{sat} and the ignition delays, τ_{ign}, with the V_{et} at the best injection timings, where τ_{ign} was defined as the period from the injection signal arrival until the start of the main combustion. The τ_{ign} was maintained nearly constant up to when the ratio of the less reactive ethane was high (note that 50 vol.% is equal to 73 mol.%) as also reported previously [11], but a further increase of V_{et} above 60 vol.% increased τ_{ign} strongly, because of the low content of the more reactive diesel fuel component. Due to the high vapor pressure as well as the early injection

accompanied by the long ignition delay, ΔP_{sat} with V_{et} = 60 and 65 vol.% was positive, and a large spray dispersion would be as could be expected.

[Fig.12]

The changes in the maximum pressure rise rate, $dp/d\theta_{max}$ and the exhaust emissions with V_{et} are shown in Fig.13, and the changes in the indicated thermal efficiency, η_i and its related factors are shown in Fig.14. The cooling loss, ϕ_{cool} is calculated as:

$$\phi_{\mathrm{cool}} = \eta_{\mathrm{comb}} - \eta_{\mathrm{i}}/\eta_{\mathrm{th}}\eta_{\mathrm{glh}} \qquad (1)$$

where η_{comb} is the combustion efficiency, η_{th} is the thermal efficiency of theoretical Otto-cycle, and η_{glh} is the degree of constant volume burning [18].

From V_{et} = 0 to 50 vol.%, the best η_i decreased, and $dp/d\theta_{max}$ increased. It appears that with increasing V_{et}, the air-fuel mixture became leaner and closer to stoichiometric, increasing the flame temperature and causing the simultaneous ignition at multiple locations. The further increase in the V_{et} up to 60 and 65 % increased the best η_i and decreased the $dp/d\theta_{max}$. Here, it would appear that the leaner mixtures formed by the flash boiling and the long ignition delay decreased the flame temperature and reduced the cooling loss while lowering the overall combustion rate even with the earlier combustion. The above discussion of the flame temperature was supported by the

changes in the cooling loss, ϕ_{cool}, where the cooling loss increased from V_{et} = 0 to 50 vol.%, and decreased from V_{et} = 50 to 65 vol.%.

NO_x increased from V_{et} = 0 to 50 vol.%, and decreased from V_{et} = 50 to 65 vol.%. No significant soot was detected in the present experiments, and the soot decreased with increasing V_{et}. As the result, combustion without soot and without NO_x was accomplished with V_{et} = 65 vol.%. One drawback here was the increases of THC and CO and the decrease of the combustion efficiency, η_{comb} as the formation of the leaner mixture and the decrease in the flame temperature tend to increase the unburned emissions due to the quenching. This combustion inefficiency had a negative effect on the thermal efficiency, but the reduction in the cooling loss overcame this negative change as well as the slight increase in the degree of constant volume burning, η_{glh}.

[Fig.13]

[Fig.14]

3.3 Isolation of the Effects of Ignition Delay and Flash Boiling

To isolate the effects of the ignition delay and the flash boiling, the mixed fuel of Shellsol (MC421) and *i*-octane was tested, adjusting the amount of *i*-octane to match the ignition delay of the diesel fuel – ethane mixed fuel (V_{et} = 60 vol.%). The profiles of the

in-cylinder pressure, p and the apparent heat release rate, $dQ/d\theta$ are shown in Fig.15. There was a discrepancy in the timings of the LTHR, and the onset of the high temperature heat release of the Shellsol – *i*-octane mixed fuel was earlier with 0.5 deg.CA compared to that of the diesel fuel – ethane mixed fuel. The difference may cause no significant effect on this comparison. The rate of heat release of the diesel fuel – ethane mixed fuel was lower than the Shellsol – *i*-octane mixed fuel even with the same ignition delay, likely as the leaner mixture could reduce the overall combustion rate in combination with the improvement in $dp/d\theta_{max}$ of 0.3 MPa/deg.CA.

[Fig.15]

The indicated thermal efficiency, η_i, the combustion efficiency, η_{comb}, the cooling loss, ϕ_{cool}, and NO_x emissions are plotted for the two mixed fuels in Fig.15. There were no significant differences in the η_i, the η_{comb}, and the ϕ_{cool}, and the η_i of the Shellsol – *i*-octane mixed fuel exceeded 40%. It was evident that the indicated thermal efficiency strongly depends on the ignition delay. It was not possible to determine the effect on the soot emission as no soot was detected in this condition. From the fact that the diesel fuel – ethane mixed fuel resulted in lower NO_x emissions, it appears that the air-fuel mixture of the diesel fuel – ethane mixed fuel was leaner and its flame temperature was lower.

As the result, it may be concluded that the ignition delay plays the most important role in the efficiency and in the emission characteristics, and that the flash boiling enhances the leaner air-fuel mixture formation that lowers the overall combustion rate and the NO_x emissions further.

[Fig.16]

4. Conclusions

The present study introduced mixed fuels of diesel fuel and ethane in premixed diesel combustion, aiming to utilize its flash boiling and ignition delay characteristics. The spray characteristics of the mixed fuels were investigated in a constant volume vessel, and the engine performance and the emissions were evaluated at the IMEP around 0.35 MPa in a single cylinder engine. The conclusions may be summarized as follows:

1. The high pressure on the bubble point curve of the diesel-fuel – ethane mixed fuel induces the flash boiling even in the in-cylinder condition close to the top dead center, expands the spray width right at the nozzle exit and disperses the spray in the downstream region even at the high ambient density.
2. The ethane fraction more than 60 vol.% is needed to gain the ignition delay which

allows the early injection accompanied by the flash boiling and achieved the low temperature combustion conditions.

3. The long ignition delay of the diesel fuel – ethane mixed fuel plays a major role in improving the thermal efficiency, and the flash boiling makes the air – fuel mixture more homogeneous, reducing NO_x emissions.
4. The major drawback of the diesel fuel – ethane mixed fuel is the complex fuel supply system. This may be suitable for stationary engines.

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Table 1 Properties of the tested fuels

	Ethane	Diesel fuel	Shellsol MC421	*i*-octane
Carbon number	C2	C10 – 26	C11.2	C8
Lower calorific value [MJ/kg]	47.3	43.1	44.0	44.4
Density [kg/m^3]	340*	820	774	690
Boiling point [ºC]	-89	200 - 350	185 - 199	99
Ignition point [ºC]	515	250	417	411

* at 293 K, 4 MPa (calculate with SUPERTRAPP)

Table 2 Experimental conditions in the constant volume vessel

Volume fraction of ethane, V_{et}	[%]	0	30	50	60
Pressure difference in flash boiling, ΔP_{sat}	[MPa]	-0.9	1.2	2.2	2.5
Injection pressure, p_{inj}	[MPa]	40			
Injection period	[ms]	2			
Nozzle hole diameter	[mm]	0.10			
Ambient pressure, p_a	[MPa]	0.90			
Ambient temperature, T_a	[K]	293			
Ambient density, ρ_a	[kg/m^3]	16.3			
Ambient gas		CO_2			

Table 3 Specifications of the single cylinder diesel engine

Displacement	[cm^3]	460
Bore	[mm]	83
Stroke	[mm]	85
Geometric compression ratio		19.3 : 1
Combustion chamber shape		Toroidal type
Fueling system Configurations of injection nozzle		Direct injection hole dia.: 0.101 mm number of holes: 10

Table 4 Experimental conditions in the single cylinder diesel engine

Engine rotation speed [rpm]	1000
Injection pressure, p_{inj} [MPa]	40
Timing of injection signaltiming, θ_{inj} [deg.CA ATDC]	-30 to 1
Fuel energy supplied [J/cycle]	400
Indicated mean effective pressure [MPa]	around 0.35
Air intake condition	Natural aspiration (No EGR or boost)

Table 5 Accuracy of measurements

Measurements	Accuracy
CO	±50 ppm
CO_2	±0.05 vol.%
NO_x	±10 ppm
THC	±50 ppm
Smoke	0.02 mg/m^3 (detection limit)
In-cylinder pressure	±0.04 MPa

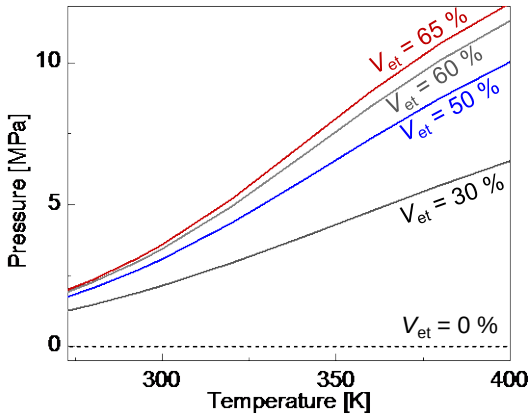

Figure 1 Pressure-temperature diagram of the C_2H_6 – n-$C_{13}H_{28}$ mixed fuels with various mixing fractions of ethane, V_{et}

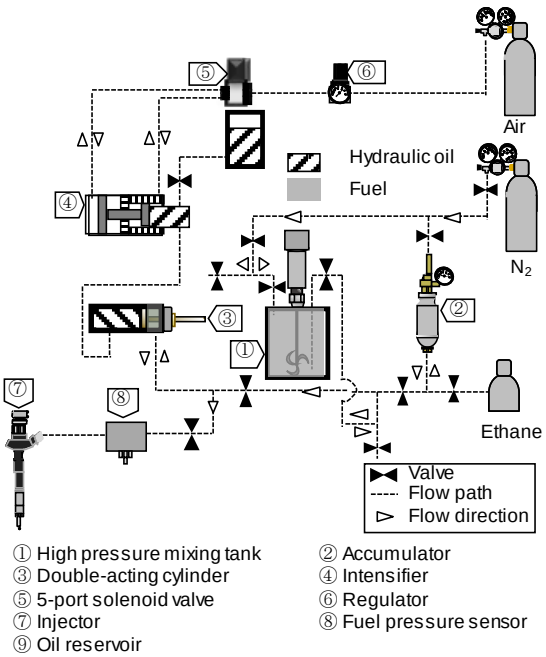

Figure 2 Schematic of the fuel supply system including the fuel mixing and the pressure parts

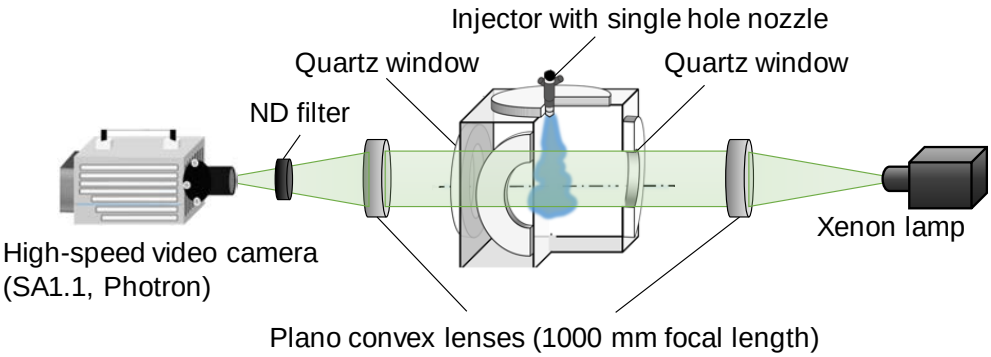

Figure 3　Optical setup for shadowgraph photography

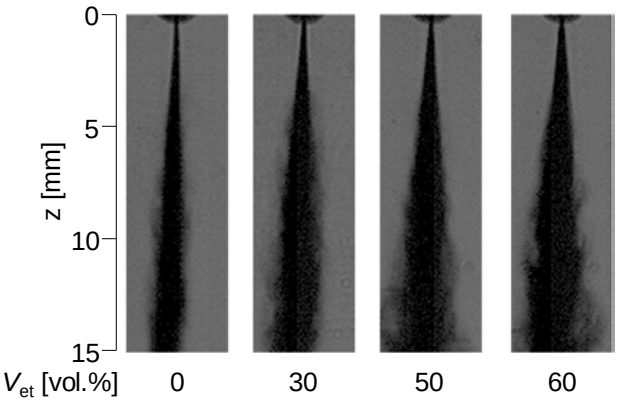

Figure 4　Changes in spray shape near the nozzle exit (top of panels) with ethane fraction, V_{et}

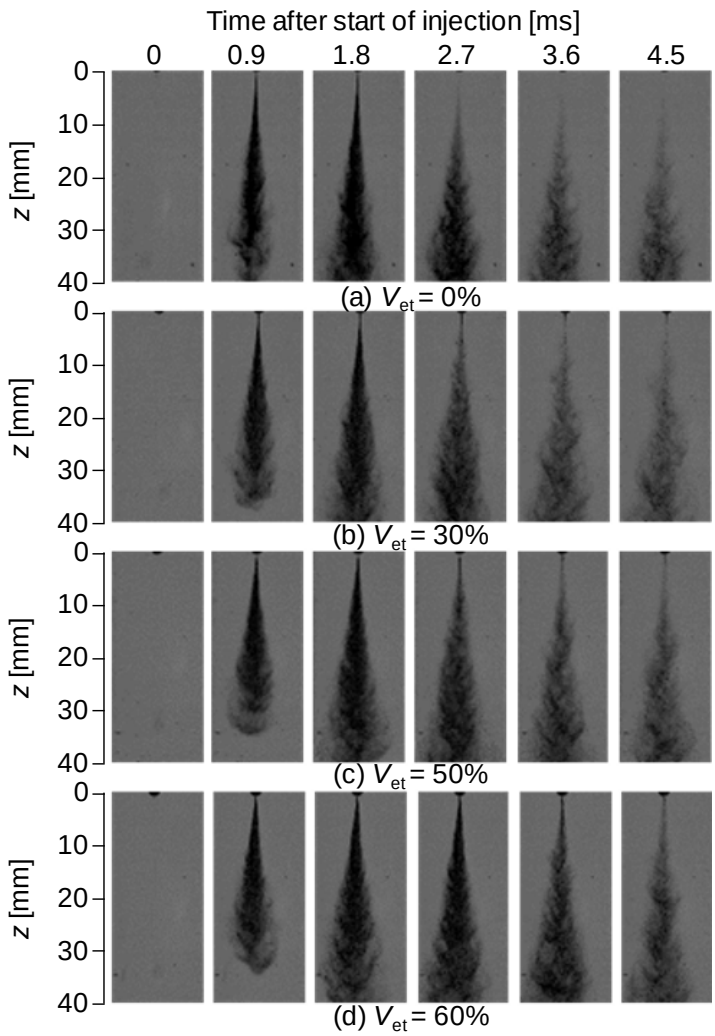

Figure 5 Temporal changes in spray shapes with the ethane fraction, V_{et}

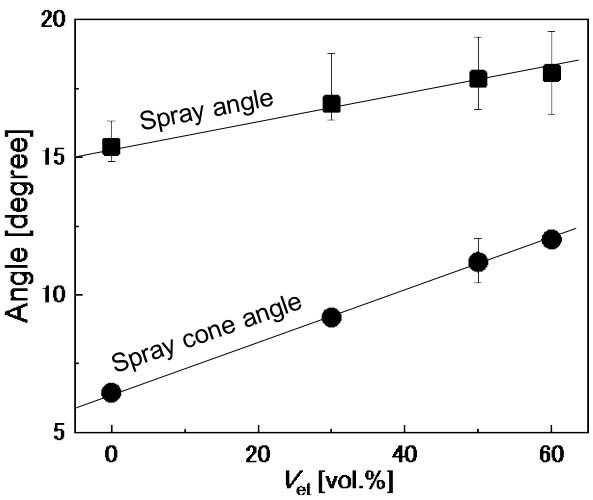

Figure 6 Changes in spray cone angle and spray angle with the ethane fraction, V_{et}

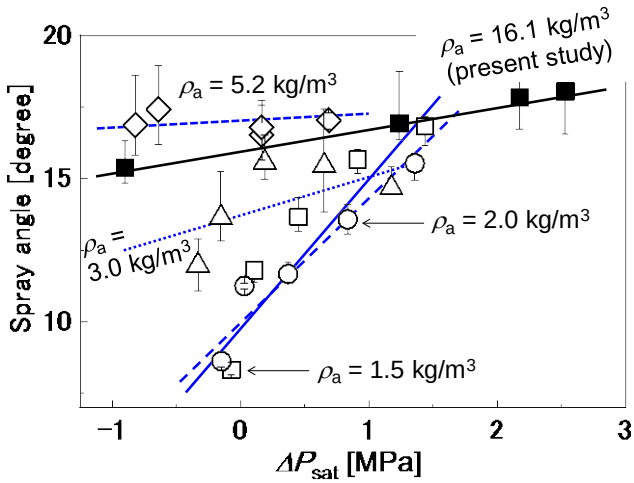

Figure 7 Relation between ΔP_{sat} and the spray angle at various ambient densities, ρ_a
Open symbols: previous study (Wada et al., 2007)
Solid symbol: present study

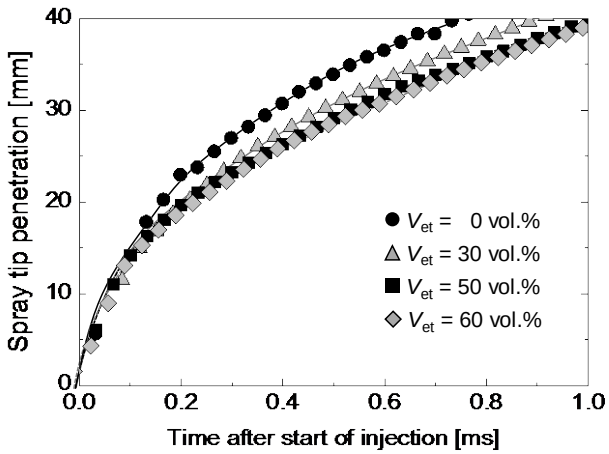

Figure 8 Plot of spray tip penetrations for different ethane fractions, V_{et}

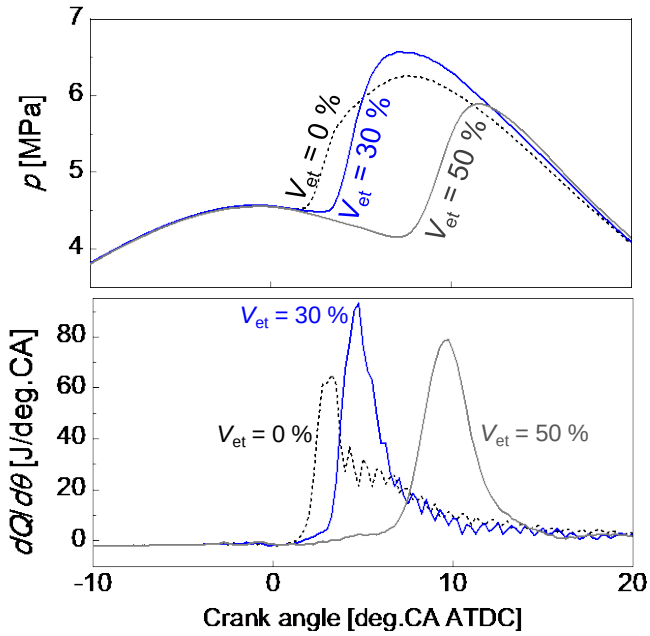

Figure 9 Profiles of in-cylinder pressures, p and apparent heat release rates, $dQ/d\theta$ at the injection timing, θ_{inj} of -4 deg.CA ATDC

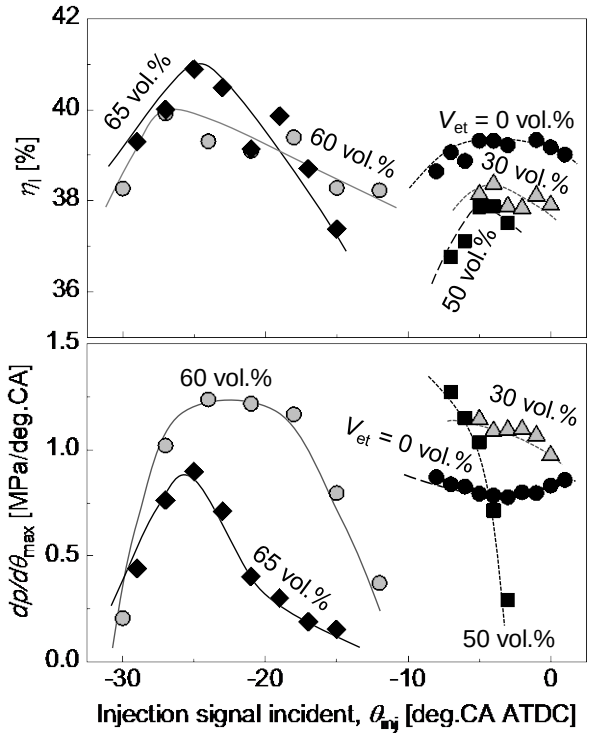

Figure 10 Variations in indicated thermal efficiency, η_i and maximum pressure rise rate, $dp/d\theta_{max}$ with the injection signal incident timing, θ_{inj} and the ethane fraction, V_{et}

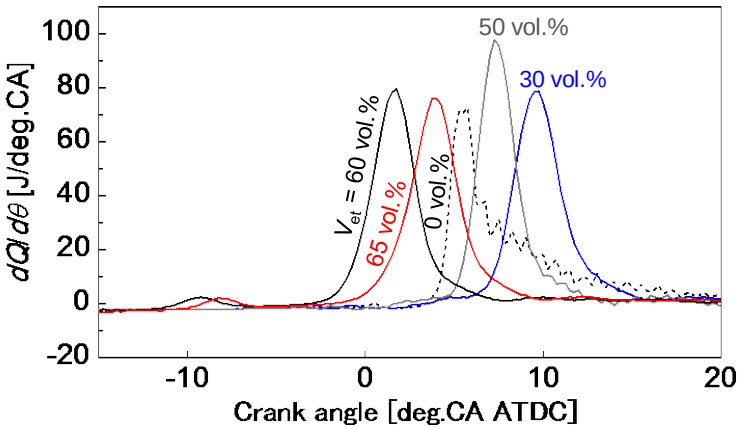

Figure 11 Profiles of the apparent heat release rate, $dQ/d\theta$ at the optimum injection timings

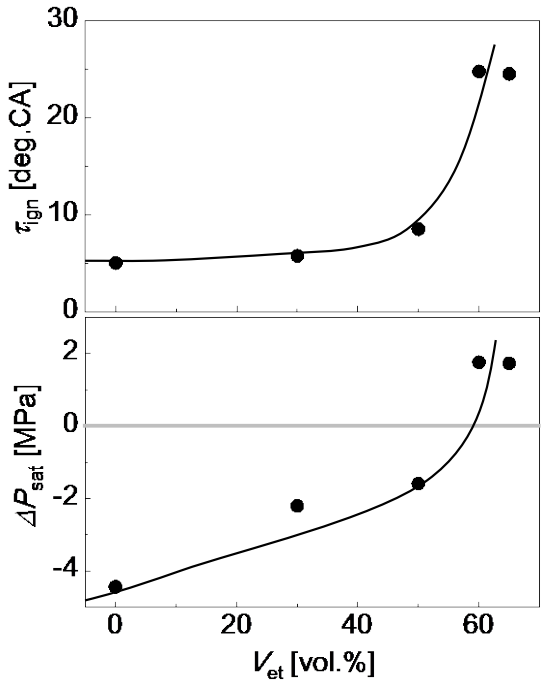

Figure 12 Plots of the ignition delays, τ_{ign} and differences between the in-cylinder pressure and pressure on the bubble point curve, ΔP_{sat} versus ethane fraction, V_{et} at the optimun injection timings

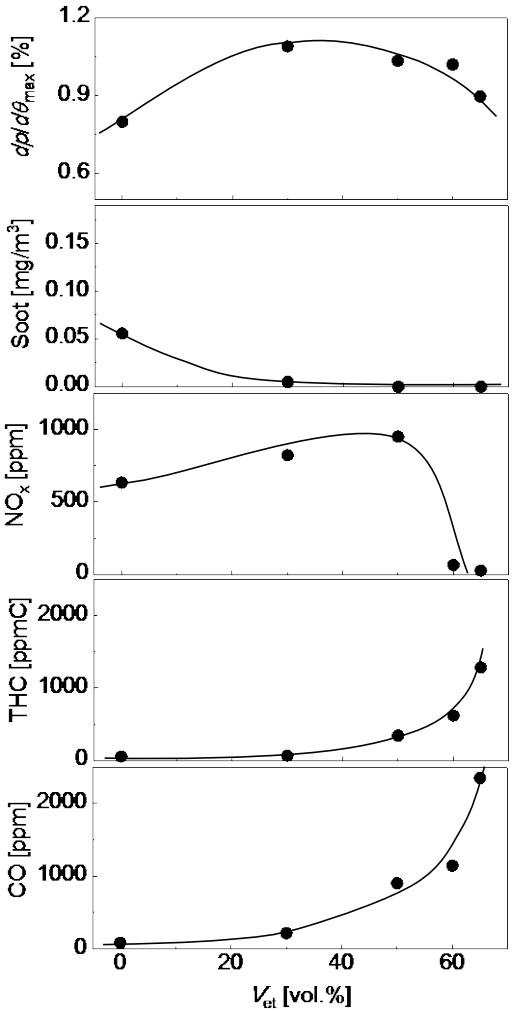

Figure 13 Changes in maximum pressure rise rate, $dp/d\theta_{max}$ and exhaust gas emissions with ethane fraction, V_{et} at the optimum injection timing

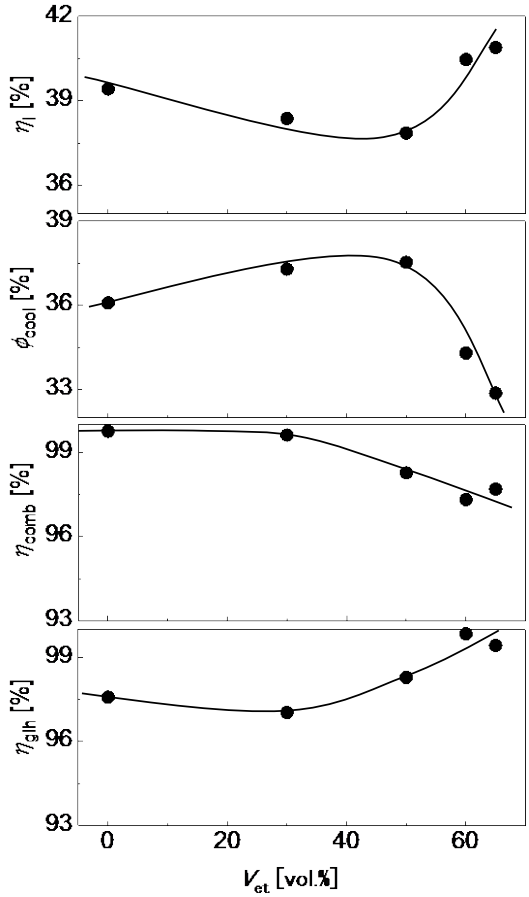

Figure 14 Changes in indicated thermal efficiency, η_i and its related factors, cooling loss, ϕ_{cool}; combustion efficiency, η_{comb}; and degree of constant volume burning, η_{glh} with ethane fraction, V_{et} at the optimum injection timing

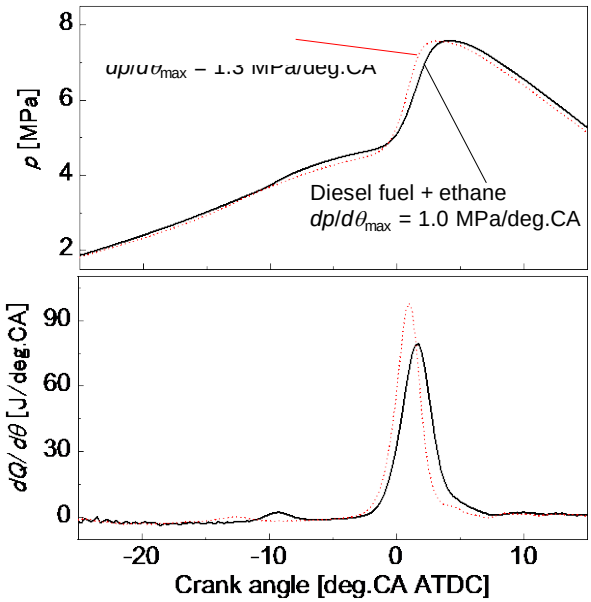

Figure 15 Plot of in-cylinder pressure, p and apparent heat release rate, $dQ/d\theta$ for the mixed fuels of Diesel fuel – ethane and Shellsol – *i*-octane (θ_{inj} = -27 deg.CA ATDC)

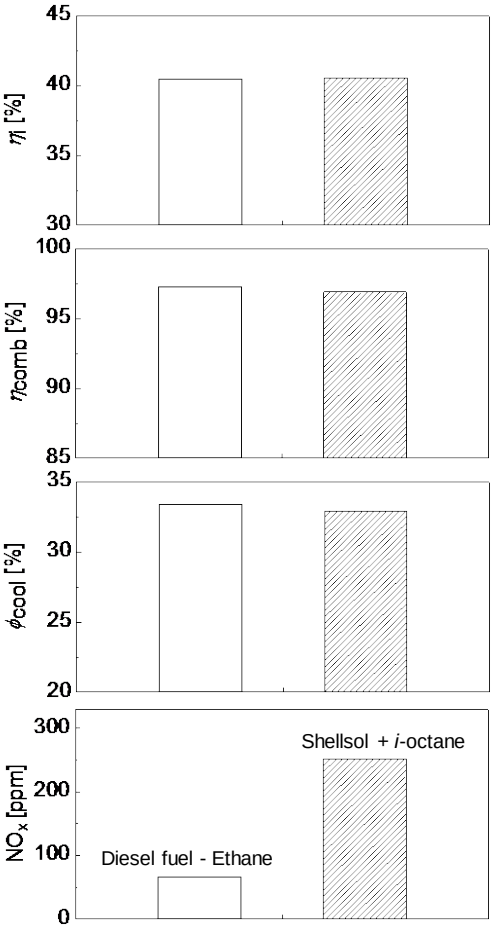

Figure 16 Details of the indicated thermal efficiency, η_i, combustion efficiency, η_{comb}, cooling loss, ϕ_{cool}, and NO_x emissions of the fuel mixtures of Diesel fuel – ethane and Shellsol – *i*-octane (θ_{inj} = -27 deg.CA ATDC)