

An experimental study on spray and combustion characteristics based on fuel temperature of direct injection bio-ethanol-gasoline blending fuel[†]

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Abstract

This study investigated the spray and combustion characteristics of a direct injection spark ignition type system based on the changes in the temperature of the blended fuel (with bio-ethanol and gasoline). The test was performed in a chamber with a constant volume. The diameter and width of the chamber were 86 mm and 39 mm, respectively. The bio-ethanol test fuel was blended at volume ratios of 0 %, 10 %, 20 % and 100 %. The temperature of the fuel was set as -7 , 25 and 35 °C. The fuel injection pressure and ambient pressure were set as 4.5 and 0.5 MPa, respectively. The shape and characteristics of the spray were investigated through a spray experiment. The increase in the fuel temperature changed its density and viscosity; this in turn increased spray penetration and spray area and increased the bio-ethanol blending ratio. The combustion visualization and experimental analysis indicated that the decrease in the fuel temperature and the increase in the bio-ethanol blending ratio led to the high viscosity and low heating value. This resulted in an increase in the ignition delay and a decrease in the rate of heat release. It is necessary to adjust the spray strategy and ignition timing to adopt bio-ethanol blended fuel as an alternative fuel.

Keywords: Fuel-temperature; Bio-ethanol; DI (Direct injection); Constant volume chamber (CVC); Ignition delay; Spray; Combustion; Emission

1. Introduction

Several studies have focused on the development of vehicles with high fuel efficiency in order to reduce carbon dioxide emissions. The global issues of limited energy resources and the impact of greenhouse gases on the atmospheric environment are some reasons driving the development of high fuel efficiency vehicles. Typical examples of high fuel efficiency vehicles include clean diesel vehicles with a high thermal efficiency and vehicles with hybrid technology. As the era of energy diversification begins, the use of traditional fossil fuels is reduced by the utilization of alternative forms of energy and bio-mass [1, 2]. Bio-ethanol is proposed as an alternative to gasoline. Bio-ethanol is produced by the fermentation of plants such as sugar cane, corn, and sweet potato produces bio-ethanol. Bio-ethanol could be used in a conventional Port fuel injection (PFI) engine. The latent heat of vaporization of bio-ethanol in PFI engines is used to cool the air-fuel mixture and results in the increased density of the air-fuel mixture and the oxygenate characteristics. This results in an improvement in the combustion characteristics and the blend

of bio-ethanol and gasoline at a fixed ratio reduces harmful emissions. Because of the improvement in the combustion characteristics through the increased density of the air-fuel mixture and oxygenate characteristics obtained by cooling the air-fuel mixture by the latent heat of vaporization of bio-ethanol when such a mixture is used in a conventional Port fuel injection (PFI) engine. However, there is a condition restriction on the application of that the homogeneous air-fuel mixture. The mixture must be formed within a short period of time when it is applied to Gasoline direct injection (GDI) technology because of the high viscosity and latent heat of vaporization of bio-ethanol. The formation of the non-homogeneous air-fuel mixture directly affects the engine power and results in the reduction of engine torque produced. This and the leads to the creation of exhaust gas [3-6]. As reported by Crane et al. 80 % of the harmful emissions from a vehicle are produced in the cold-start and warm-up driving conditions. Hence, studies to reduce the harmful emissions from such driving conditions are in progress [7]. In particular, there is a significant amount of interest in reducing the quantity of harmful emissions from the cold-start at low external temperatures in winter. Therefore, this study focused on the identification of the spray and combustion characteristics of the bio-ethanol-gasoline fuel mixture at extremely low temperatures and room temperature. The

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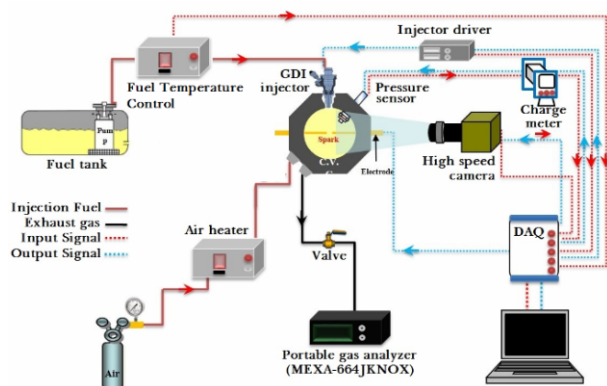


Fig. 1. Schematic illustration of the experimental apparatus.

characteristics were investigated by considering the seasonal conditions of bio-ethanol-gasoline fuel mixtures of different blending ratios inside a Constant volume chamber (CVC) with a direct injection system.

2. Experimental apparatus and method

2.1 Experimental apparatus

A CVC with a diameter and width of 86 mm and 39 mm, respectively, was constructed to investigate the spray and combustion phenomena. An observation window made of reinforced glass with a diameter and thickness of 120 mm and 25 mm, respectively, was installed on the front side of the chamber. A high-speed digital camera was mounted for photographing the spray and combustion characteristics. Peripheral devices, such as of the CVC an intake valve, an exhaust valve, a piezo type pressure sensor, and a spark plug were installed at the center of the CVC. A high-pressure fuel supply system comprised of a gear pump and an anti-explosive motor was constructed to supply high-pressure fuel to the injector. The high-pressure fuel was passed through a fuel temperature controller and was supplied to the injector after the required cooling or heating to reach the target temperature was performed. The injector that was mounted on the CVC wall was a guided type 6-hole injector used in a 2.0 L Theta II GDI engine. The fuel was sprayed into the CVC by using an injector driver. The pressure regulator and the temperature controller were connected to maintain the ambient conditions inside the chamber. A data acquisition board, DAQ NI-6251 board from National Instruments using the LabVIEW program was connected to a personal computer. The data acquisition board was used to control the experimental apparatus and collect data. An emission analyzer (using Horiba MEXA-554 JKNOX) analyzed the exhaust gas after the combustion experiment was terminated. A schematic illustration of the experimental apparatus is shown in Fig. 1.

An injection quantity measurement apparatus is shown in Fig. 2. The apparatus was connected to the ambient pressure chamber and the injector, and was installed in a combined form. It was measured using the CAS Inc. balanced precision.

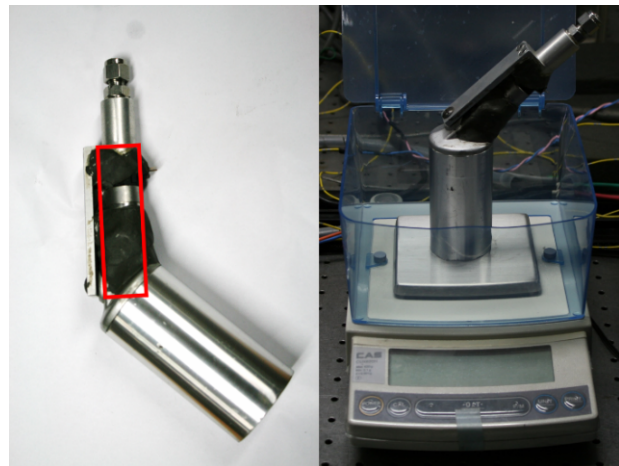


Fig. 2. Injection quantity measurement apparatus.

2.2 Experimental method

The temperatures were set as $-7\text{ }^{\circ}\text{C}$, $25\text{ }^{\circ}\text{C}$ and $35\text{ }^{\circ}\text{C}$, reflecting the annual lowest, average, and highest temperatures in Korea, respectively, to identify the spray and combustion characteristics according to the fuel temperature of the bio-ethanol-gasoline blend fuel. The fuel used in the experiment was defined by the volume ratio of bio-ethanol and was called E0, E10, E20 and E100 according to the blending ratio. The spray characteristics experiment was performed with respect to the bio-ethanol blend rate and fuel temperature for 4 ms. For combustion characteristics experiment, an injection quantity measuring instrument was developed because the density characteristics of the fuel changed according to the fuel temperature and blending ratio. The injection quantity measuring instrument was connected to the high pressure chamber and injector. The injection quantity was measured by the difference in the mass of the injected fuel in the measuring instrument following the spraying. Fuel was supplied according to the injection quantity measurement data to obtain the stoichiometric air-fuel ratio. The fuel injection pressure was set as 4.5 MPa and the ambient pressure was set as 0.5 MPa. The ignition timing was set as 1.5 ms after the end of the injection. This time excluded the fuel injection time, and measured the time during the compression stroke under the middle speed/middle load conditions in the 2.0 L GDI engine.

The experimental conditions are shown in Table 1, and the physical properties of the bio-ethanol and gasoline are shown in Table 2. Table 3 presents the injection durations for each experimental condition.

A high-speed digital camera set at 4000 fps was used for photographing the spray and combustion. The shooting resolution was 256×256 pixels, and it was set to acquire the specification of the high-speed camera and the quantitative visualization data. Additionally, the rate of heat release and the mass fraction burned according to the piezo-type pressure sensor were used to perform a combustion analysis.

The shadow-graphy method was employed for spray visu-

Table 1. Experimental conditions.

	Spray	Combustion
Diameter × width [mm]	86 × 39	
Displacement [cm ³]	228	
Test fuel	E0, E10, E20, E100	
Fuel temperature [°C]	-7, 25, 35	
Injection pressure [MPa]	4.5	
Ambient pressure [MPa]	0.5	
CVC wall temperature [°C]	60	
Ignition position		Center of CVC
Ignition timing		EOI + 1.5 ms

Table 2. Fuel properties of gasoline and bio-ethanol.

	Gasoline	Bio-ethanol	
Chemical formula	C _n H _{1.87n}	C ₂ H ₅ OH	
Molecular weight [kg/kmol]	114.15	46.07	
Oxygen [%wt]	0	35	
Octane number	86-94	98-100	
Density [kg/m ³ @ 20°C]	732	792	
Viscosity	@ -70°C [cp]	2.92	12
	@ 37.8°C [cp]	0.71	1.2
Vapor pressure @ 37.8°C kPa	61.0	15.5	
Latent heat of vaporization [kJ/kg]	420	854	
Auto-ignition temperature [°C]	257	423	
LHV [MJ/kg]	43.47	26.87	
A/F ratio	14.7	9.00	

Table 3. Fuel injection durations [ms].

	Fuel					
	E0	E10	E20	E100		
Fuel temperature [°C]	-7	6.82	7.06	7.32	10.37	
	25	7.09	7.34	7.60	10.75	
	35	7.18	7.43	7.69	10.88	

alization, which was performed by using a continuous light source, LED and a high speed camera. The images from the injection experiment were used to identify the injection characteristics for the spray penetration and spray area. The spray penetration was defined as the distance from the injector nozzle at the upper chamber to the tail end of the spray. The spray area was defined as the area of the droplet distribution at the end of the injection.

The high-speed camera performed the combustion visualization. The pressure sensor measured the pressure changes inside the chamber due to the combustion. During the combustion process, 10000 pressure data points were obtained per second. The energy formula and the rate of pressure increase for the burned gases were used to calculate the rate of heat release. The Eq. (1) Weibe function was used to calculate the mass fraction burned. The exhaust gas was sampled and analyzed after the combustion.

$$x_b = 1 - \exp \left[-a \left(\frac{t - t_s}{\Delta t} \right)^{m+1} \right] \quad (1)$$

where x_b denotes the design variables, t is the time, t_s is the start of combustion, Δt is the total combustion duration ($x_b = 0$ to $x_b = 1$), and a and m are adjustable parameter. Varying a and m changes the shape of the curve significantly. Actual mass fraction burned curves have been fitted with $a = 5$ and $m = 2$.

3. Results and consideration

3.1 Spray visualization results

The spray shape of the bio-ethanol-gasoline blended fuel was photographed at the various temperatures, and the results are shown in Fig. 3. The spray development processes were arrayed in 10 ms intervals for each blended fuel under the fuel temperature conditions. The results indicated that the width at the tail end of the spray was narrower when the temperature

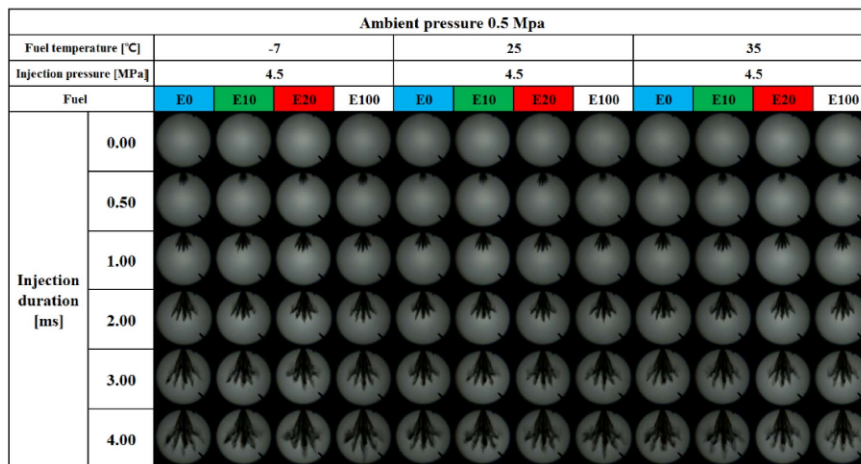


Fig. 3. Spray visualization for various fuel temperatures and bio-ethanol blending ratios.

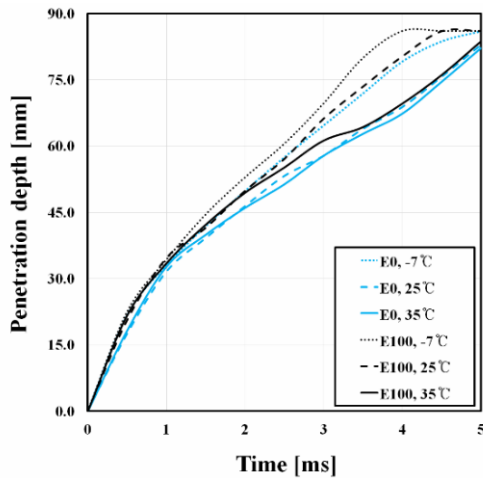


Fig. 4. Spray penetration at various temperatures of the E0 and E100 fuel.

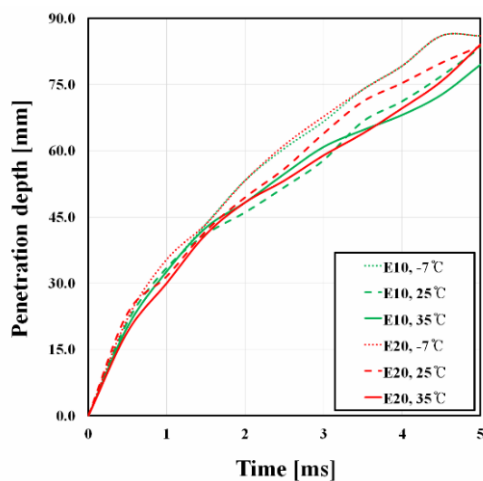


Fig. 5. Spray penetration at various temperatures of the E10 and E20 fuel.

was lower. The spray penetration increased when the bioethanol blending ratio increased.

The results for the spray penetration and spray area are shown in Figs. 4-6. The spray penetration and spray area increased in each blended fuel when the fuel temperature decreased. This was because the kinetic energy of the injected fuel was large owing to the increase in the fuel droplet size caused by the increased physical viscosity characteristics and the fuel gasification when the temperature decreased. The spray penetration and spray area increased as the bioethanol blending ratio increased. This was because the viscosity and latent heat of vaporization were higher than those of gasoline. Conversely, the difference in the spray penetration was smaller at 25 °C and 35 °C, which were above room temperature when compared with that at -7 °C. This was because of the similar viscosity between the bio-ethanol and gasoline when the viscosity was reduced rapidly at room temperature [8-11].

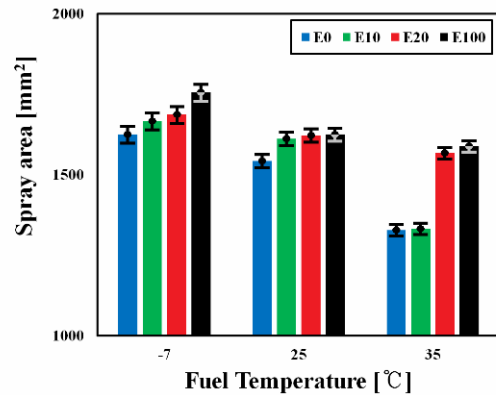


Fig. 6. Spray area at various fuel temperatures and bio-ethanol blending ratios.

3.2 Combustion visualization and analysis

The flame produced by the spark plug in the air-fuel mixture that formed after the end of the injection was photographed, as shown in Fig. 7. The combustion characteristics of the ignition delay time (such as the area of flame kernel creation and flame development) were checked by arraying the visualization results in 30 ms by 5 ms intervals. Later, they were arrayed in 20 ms intervals to check the combustion process according to the propagation. The results indicated that the flame development was slower when the fuel temperature was lower. According to the spray visualization results, the fuel viscosity and vaporization increased at a low fuel temperature resulting in the wall-wetting effect inside the combustion chamber. This reduced the homogeneity of the air-fuel mixture inside the combustion chamber and the fuel quantity in the air-fuel mixture, leading to the slower flame development at lower fuel temperatures. The ignition delay time increased when the bio-ethanol blending ratio increased under the same temperature conditions. This was because of the high viscosity and latent heat of vaporization of the bio-ethanol that resulted in the non-homogeneous air-fuel mixture and the cooling effect of the air-fuel mixture [12].

A cool flame was observed in the initial stage of the flame development in each experimental condition. This cool flame is known as the glowing of OH- or aldehyde series created during the ignition delay time at a low combustion temperature. The cool flame glow increased when the fuel temperature was low or when the bio-ethanol blending ratio increased. This indicated that the ignition delay time was longer when the cool flame was created, owing to the reduced combustion characteristics formed by the non-homogeneous air-fuel mixture [13].

As shown in the combustion analysis results, multiple small flames in the flame-development stage were considered, based on the experimental conditions in the combustion visualization results.

Figs. 8 and 9 illustrate the results for the combustion pressure and rate of heat release. Table 3 shows the results for the

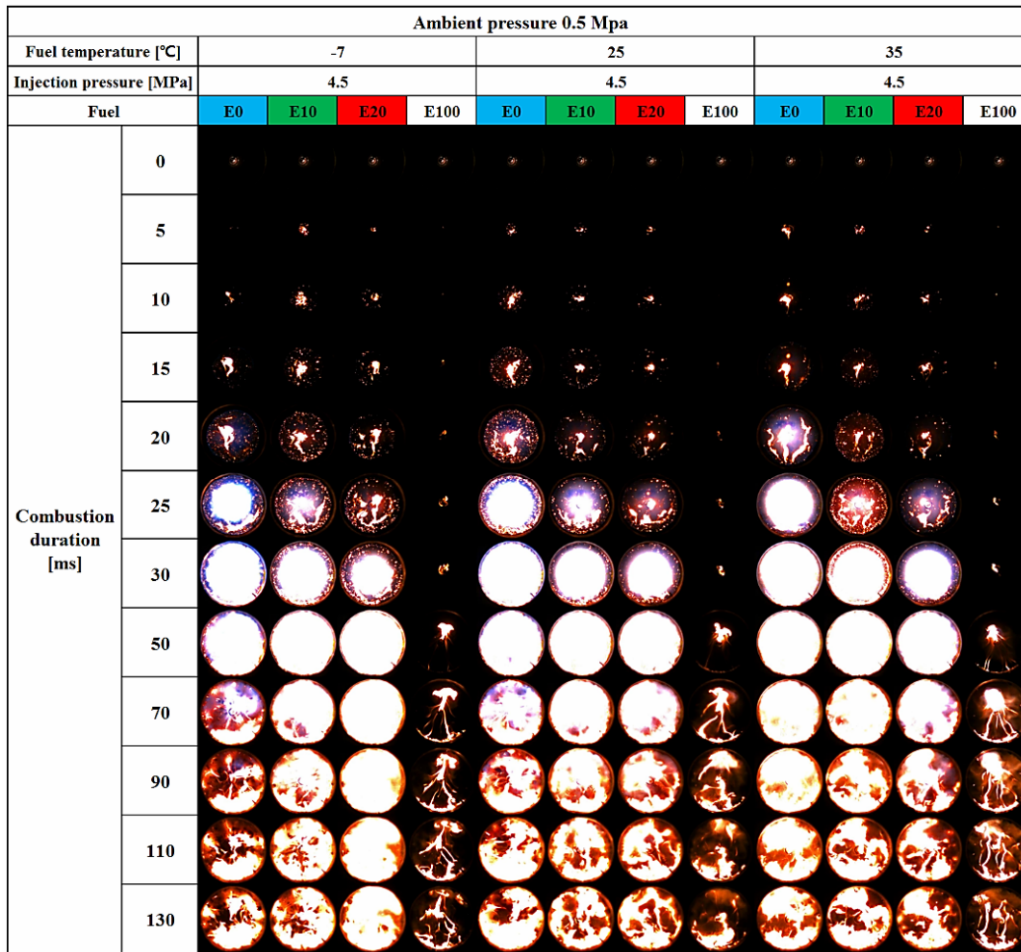


Fig. 7. Combustion visualization for various fuel temperatures and bio-ethanol blending ratios.

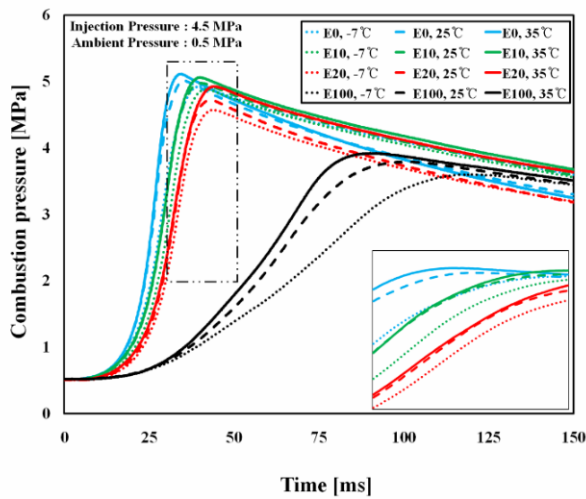


Fig. 8. Combustion pressures for various fuel temperatures and bio-ethanol blending ratios.

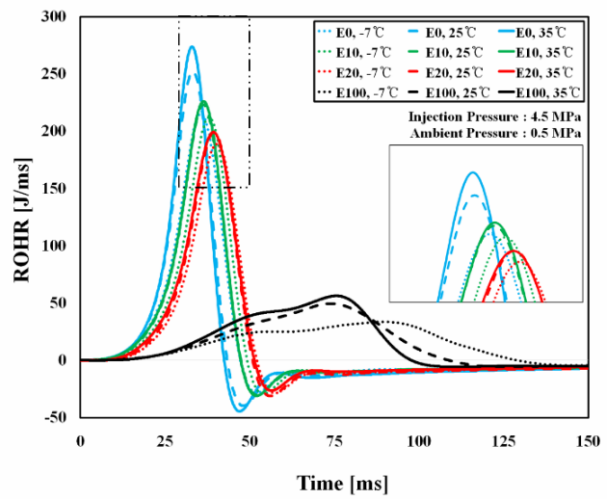


Fig. 9. Rate of heat release for various fuel temperatures and bio-ethanol blending ratios.

mass fraction burned. The combustion speed was low and the combustion peak pressure was the lowest at the low temperature of $-7\text{ }^{\circ}\text{C}$. There were clear differences in the heat-release

rate and mass fraction burned. The flame development period was lengthened during the period with a 10 % mass fraction burned and a rapid combustion time corresponding to 10-90 %.

Table 4. Combustion peak pressures [bar] and burn durations for 0-10 %, 10-90 % and 0-100 % burned mass fractions [ms] with various fuels and fuel temperatures [°C].

Fuel	Temp.	P_{max}	t_{0-10}	t_{10-90}	t_{max}
E0	-7	49.33	11.1	19.8	57.0
	25	50.11	9.7	17.5	50.1
	35	51.15	9.6	17.1	49.3
E10	-7	48.84	11.5	20.5	59.1
	25	49.78	11.1	19.9	57.2
	35	50.63	11.0	19.8	56.8
E20	-7	45.76	12.2	21.8	62.8
	25	47.23	12.0	21.4	61.7
	35	49.28	12.3	21.9	63.1
E100	-7	36.05	32.9	59.9	169.5
	25	37.95	28.2	50.6	145.5
	35	39.21	24.9	44.7	128.4

There were minor differences in the results of the P-t diagram, rate of heat release, and mass fraction burned between the fuel temperatures of 25 °C and 35 °C. This resulted from the similarity in the physical properties of the fuel between the conditions.

The combustion speed and combustion peak pressure reduced when the bio-ethanol blending ratio increased under the same temperature conditions. This was because of the high viscosity and latent heat of vaporization of the bio-ethanol (as indicated by the spray visualization), which resulted in the non-homogeneous air-fuel mixture and low heat release from the bio-ethanol.

The multiple small flames observed in the flame development stage under each condition were created by the ignition energy at the period of the droplet existence of the sprayed fuel. The flame did not affect the pressure change or heat release when the results for the rate of heat release and the burned mass fractions were matched.

The combustion peak pressures were reduced when the bio-ethanol blending ratio was increased under the same temperature condition. This was because of the high viscosity and latent heat of vaporization of the bio-ethanol (as indicated by the spray visualization), which resulted in the non-homogeneous air-fuel mixture and low heat release from the bio-ethanol. The multiple small flames observed in the flame development stage under each condition were created by the ignition energy at the period of the droplet existence of the sprayed fuel. The flame did not affect the pressure change or heat release when the results for the rate of heat release and mass fraction burned were matched.

3.3 Results for exhaust gas

The results for the exhaust gas according to the changes in the bio-ethanol blending ratio are shown in Figs. 10-13. The CO and HC were created from incomplete combustion in the

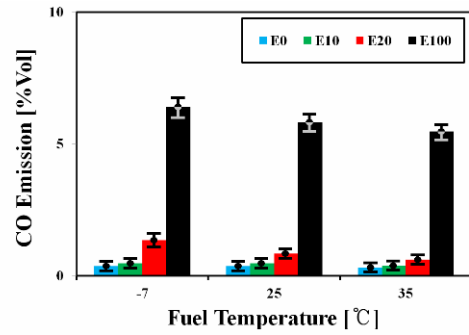


Fig. 10. CO emissions for various fuel temperatures and bio-ethanol blending ratios.

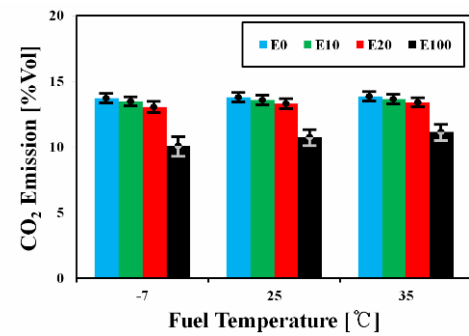


Fig. 11. CO₂ emissions according to various fuel temperatures and bio-ethanol blending ratios.

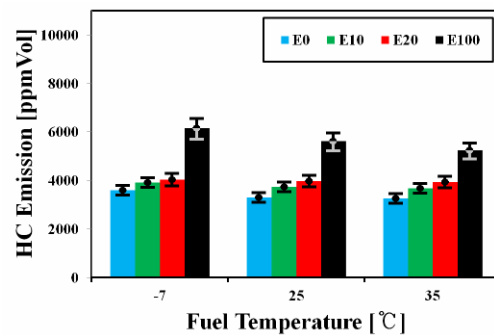


Fig. 12. HC emissions for various fuel temperatures and bio-ethanol blending ratios.

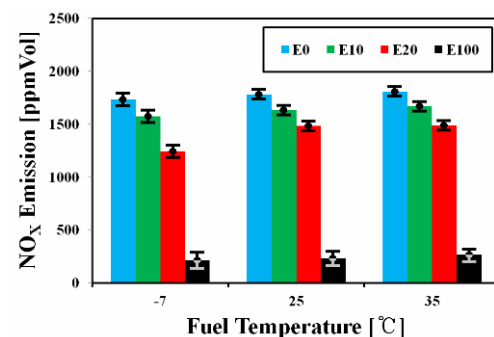


Fig. 13. NO_x emissions for various fuel temperatures and bio-ethanol blending ratios.

locally rich area of the non-homogeneous air-fuel mixture. The emission quantity increased when the bio-ethanol blending ratio increased and the fuel temperature decreased. Consequently, the combustion efficiency was reduced by the increased bio-ethanol blending ratio and decreased fuel temperature in the combustion analysis results.

CO₂ was created from the complete combustion and was affected directly by the homogeneity of the air-fuel mixture. NO_x was mainly formed when the combustion temperature was high and the air was excessive. The results for the emission characteristics of NO_x and CO₂ exhibited the same trend according to the experimental conditions. The emission quantity was maximized at a high fuel temperature.

The increased fuel temperature was an advantageous condition for creating the homogeneous air-fuel mixture. The emission quantities of CO₂ and NO_x reduced when the bioethanol blending ratio increased. The creation of the non-homogeneous air-fuel mixture and the reduction of the H/C ratio of the fuel had a large effect in the case of CO₂. In the case of NO_x, the increase in the bio-ethanol blending ratio formed the non-homogeneous air-fuel mixture and lowered the combustion temperature and heat release, reducing the heat released from the fuel.

4. Conclusions

The following results were obtained from the spray and combustion characteristic experiment according to the fuel temperature changes of the bio-ethanol-gasoline blend fuel in the CVC with direct injection spark ignition.

(1) According to the spray-visualization results, the spray penetration and injection area increased when the bio-ethanol blending ratio increased and the fuel temperature decreased.

(2) The ignition delay time increased as the viscosity and latent heat of vaporization increased at a fuel temperature of 7 °C, but the combustion was small at 25 °C and 35 °C, as the viscosity difference was similar.

(3) The high viscosity and large latent heat in the bio-ethanol indicated an increase in the ignition delay time and the decrease in the heat-release rate when the blending ratio increased.

(4) According to the measurement results for the exhaust gas, the CO and HC emissions increased and the CO₂ and NO_x emissions decreased as the quality of the air-fuel mixture decreased for low fuel temperatures. The CO and HC emission increased and the CO₂ and NO_x emissions decreased when the non-homogeneous air-fuel mixture and low heating value were low and when the bio-ethanol blending ratio was high.

Acknowledgment

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Nomenclature

P_{max}	: Combustion peak pressure
t_{0-10}	: Mass fraction burned for 0-10 %
t_{10-90}	: Mass fraction burned for 10-90 %
t_{max}	: Mass fraction burned for 0-100 %
CVC	: Constant volume chamber
GDI	: Gasoline direct injection
PFI	: Port fuel injection
E0	: Gasoline 100 %
E10	: Gasoline 90 % + bio-ethanol 10 %
E20	: Gasoline 80 % + bio-ethanol 20 %
E100	: Bio-ethanol 100 %
LED	: Light emitting diode
LHV	: Low heating value

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