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Effect of Premixed Ethanol Ratio Based on the Same Heating Value on the Atomization of Diesel Fuel Injected in the Cylinder

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Abstract: The objective of this study was to investigate the effect of a premixed ethanol ratio based on the same total heating value in a cylinder on the equivalence ratio distributions and the injected fuel droplet behavior in the cylinder of an RCCI engine. The spray simulation was conducted in two parts. First, we carried out spray validation simulations to determine the spray-influenced factor of the test injector. Next, engine simulations were performed with the spray-influenced factor obtained from the spray validation simulations to investigate the effect of the premixed ethanol ratio based on the same total heating value in a cylinder on the injected fuel atomization and the equivalence ratio distributions. The introduced total heating value was fixed at 595 J based on the lower heating value of diesel, 14 mg. The heating value of the premixed ethanol ratio varied from 0% to 40% based on the same total heating value in the cylinder in steps of 10%. It was revealed that when the premixed ethanol ratio based on the same total heating value in the cylinder was increased, the spray tip penetration value was reduced after 4 deg of diesel was injected because of the short injection duration and the small amount of diesel fuel used. The SMD value was also increased up to 32.58% with an increasing premixed ethanol ratio because of the low kinetic energy of the injected fuel, the short injection duration, the slow evaporation of the injected fuel and the low cylinder temperature.

Keywords: fuel combustion; energy conversion/system; energy systems analysis; mixed alternative fuel

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1. Introduction

A compression ignition (CI) engine is a device that generates power by injecting fuel into compressed air and using the characteristics of the auto-ignition of the injected fuel. It uses a relatively high compression ratio and auto-ignition characteristics and has the advantage of having stronger torque and higher efficiency than a spark ignition (SI) engine. However, unlike SI engines, which have a homogeneous air–fuel mixture, CI engines have a relatively rich mixture of injected fuel but the disadvantages of emitting a large amount of NO_X due to a high combustion temperature as well as soot due to incomplete combustion. Recently, the low-temperature combustion (LTC) method has been frequently applied to reduce exhaust emissions from CI engines. The high combustion temperature is formed by the rich air–fuel mixture in the CI engine, so to implement the LTC method, premixed charge compression ignition (PCCI), homogeneous charge compression ignition (HCCI), and reactivity-controlled compression ignition (RCCI) are used to suppress the rich air–fuel mixture region. Rohani et al. [1] studied the effect of the EGR ratio and the different injection start times using the PCCI method. They found that the PCCI method emits less exhaust emissions due to various auto-ignition points in the air–fuel mixture in a whole cylinder. Pandey et al. [2] suggested that the PCCI method can secure more time for air and fuel mixing than the general CI method; however, it still does not meet various exhaust emission regulations. Thus, Cha et al. [3] and Min et al. [4] progressed their studies further to simultaneously reduce exhaust emissions by using simulated EGR under the PCCI method. They found that specific start times of energizing can reduce exhaust emissions simultaneously. However, the IMEP value also decreased because the injected fuel that was not burned flowed into the crevice volume. Another method for low-temperature reactions is the homogeneous charge compression ignition (HCCI) method because it induces a low-temperature reaction using a homogeneous mixture in the cylinder [5]. Although the HCCI method reduces exhaust emissions, it is difficult to control the exact ignition timing under high-load operation [6,7].

To solve the issue of controlling the exact ignition timing in the HCCI method, the RCCI method has been studied by many researchers. Generally, the RCCI method uses two fuels for the homogeneous air–fuel mixture and the control of ignition timing. First, to form the homogeneous air–fuel mixture, high-octane fuel is injected into the intake port and mixed homogeneously with air. The homogeneous air–fuel mixture enters the cylinder when the intake valve opens, so the air in the cylinder becomes as charged as the homogeneous air–fuel mixture. Additionally, to ignite the homogeneously distributed high-octane fuel, a high-cetane fuel is then used by injecting it directly into the cylinder [8–10]. Thus, the start of combustion in the cylinder can be easily controlled using this method compared to the HCCI method due to the auto-ignition characteristics of the injected high-cetane fuel. However, Singh et al. [11] suggested that substances produced by incomplete combustion, such as carbon monoxide (CO) and hydrocarbon (HC), are significantly discharged by the auto-ignition of the directly injected high-cetane fuel into the cylinder. Jo et al. [12] carried out experimental research aiming to reduce the exhaust emissions from incomplete combustion under different premixed ethanol ratios and energizing timings. The total heating value of ethanol and diesel was constant at 595 J, and the premixed ethanol ratio varied from 0% to 40%. As a result, even though the peak cylinder pressure was decreased with an increasing premixed ethanol ratio, the IMEP (indicated mean effective pressure) had very similar values because the ignition delay was longer due to the decrease in the cylinder temperature as a result of the increased ethanol ratio. In addition, the NO_X emissions were reduced due to the low combustion temperature and low intake air temperature. In their study, the amount of diesel fuel injected for the ignition of the homogeneously distributed ethanol in the cylinder changed according to the ethanol ratio. So, to determine the exact ignition timing needed to reduce the exhaust emissions, analysis of injected diesel fuel droplets is important because the intake air temperature decreases due to the latent heat of the vaporization of the ethanol injected into the premixing chamber. However, in an experimental study, it is difficult to analyze and compare the injected fuel droplet behavior because the amount of injected diesel fuel that is evaporated is difficult to confirm in visualization experiments of fuel droplets, so the exact evaporation rate is difficult to measure.

Therefore, this study was conducted to investigate the effect of the premixed ethanol ratio, based on the same total heating value in a cylinder, on the equivalence ratio distributions and the injected fuel droplet behavior in the cylinder. At the same time, the results for the premixed ethanol ratio were compared and analyzed in terms of spray tip penetrations, spray evolutions, and the equivalence ratio distributions in the cylinder. This work carries on from a previous study on the investigation of the simultaneous reduction in exhaust emissions and the improvement in combustion performance according to the premixed ethanol ratio based on the same total heating value in the cylinder in an RCCI engine. The behavior of fuel droplets injected into the cylinder is changed because the density and pressure in the cylinder are changed by the premixed ethanol ratio rather than by the general air in the cylinder, so this study is considered essential. Additionally, this work provides data for the optimization of conditions to improve combustion performance and reduce the exhaust emissions from RCCI engines.

2. Experimental and Numerical Descriptions

2.1. Experimental Descriptions

The schematic of the experimental setup for securing reliability in the validation of the numerical analysis is shown in Figure 1. The experiment setup consisted of a test chamber, an Ar-ion laser, a common-rail injection system, a high-pressure chamber, and an intensified charge-coupled device (ICCD) camera. In the common-rail injection system, the mini sac type test injector has a 0.168 mm hole diameter with 5 holes and 154 deg of inclined spray angle. The injection pressure and the injection mass were constant at 100 MPa and 14 mg. The ambient pressure in the chamber changed from 2.5 MPa to 3.5 MPa to simulate the pressure change in the cylinder because it changed according to the crank angle. The experiment was conducted to take images of the spray evolutions. The intervals of spray evolutions were set to 0.4 ms from 0 ms to 1.2 ms and then it was compared with the spray simulation results such as the spray cone angle and the spray tip penetration according to each time of spray evolution. The spray tip penetration was measured by image post-processing. The obtained images by the experiment were changed to black and white and the changed images were controlled by the threshold to obtain clear images. The threshold values were adjusted to leave the liquid fuel.

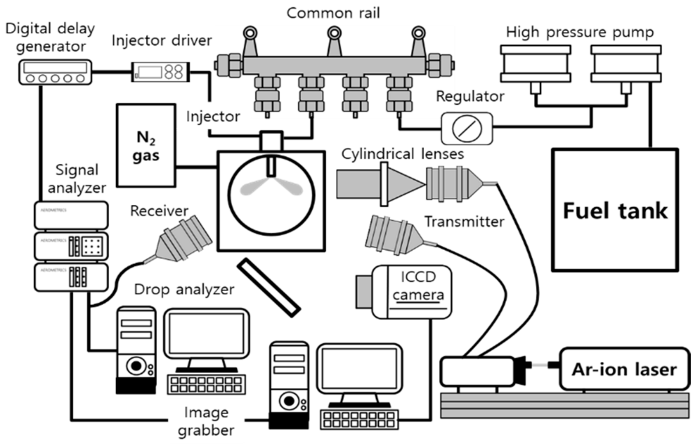


Figure 1. Schematic of an experimental setup.

2.2. Numerical Descriptions

The spray simulation was conducted by dividing it into two parts. First, spray validation simulations were performed to find the spray-influenced factor of the test injector. Next, the engine simulations were performed with the spray-influenced factor obtained from spray validation simulations to investigate the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the injected fuel atomization and the equivalence ratio distributions.

Through the spray validation simulation, the spray-influenced factors were able to be obtained, and these were applied in the engine simulation. In spray-influenced factors, there is a break-up time, inclined spray angle, and so on. So, to validate the spray models, a cylindrical geometry mesh was generated and applied in the spray validation simulation, which is the same shape as the test chamber used in the experiment and it is 140 mm in diameter and 50 mm in height, and the size of the mesh is a hexahedron of 1 mm $\times$ 1 mm $\times$ 1 mm (width $\times$ length $\times$ height). The size of each mesh was generated by referring to the previous study [4] and the program manual [13]. In addition, to express the physical phenomena such as turbulence, break-up, evaporation, and wall interaction in the numerical analysis, sub-models were employed by the k-zeta-f model, KH-RT model, Dukowicz model, and Mundo–Tropea–Sommerfeld model, respectively. Among the sub-models, the KH-RT model was employed for the break-up model. The KH mechanism is used in high relative velocity and high ambient density and the RT mechanism is applied in rapid deceleration of the droplets, causing the growth of surface waves at the droplet stagnation point [13,14]. Furthermore, the KH mechanism is based on the WAVE model equation as follows (1)~(4).

$$R_a = C_1 \Lambda \tag{1}$$

$$\tau_a = \frac{3.7 C_2 R}{\Lambda \cdot \Omega} \tag{2}$$

$$\Lambda = 9.02 \cdot r \frac{\left(1 + 0.45 \cdot Oh^{0.5}\right)\left(1 + 0.4 \cdot T^{0.7}\right)}{\left(1 + 0.87 \cdot We_g^{1.67}\right)^{0.6}} \tag{3}$$

$$\Omega = \left(\frac{\rho_d r^3}{\sigma}\right)^{-0.5} \frac{0.34 + 0.38 \cdot We_g^{1.5}}{(1 + Oh)\left(1 + 1.4 \cdot T^{0.6}\right)} \tag{4}$$

Here, *Oh* and *We* denote the Ohnesorge number and Weber number, respectively.

The *RT* model is described by Ω (the fastest-growing frequency) and the *K* (wave number). Its equations are as follows (5) and (6).

$$\tau_t = C_5 \frac{1}{\Omega_t} \tag{5}$$

$$\Lambda_t = C_4 \frac{\Pi}{K_t} \tag{6}$$

where τ_t and Λ_t are defined as the break-up time constant and the wave length and it has equations as follows (7) and (8), respectively.

$$\Omega_t = \sqrt{\frac{2}{3\sqrt{3\sigma}} \frac{g_t |\rho_d - \rho_c|^{1.5}}{\rho_d + \rho_c}} \tag{7}$$

$$K_t = \sqrt{\frac{g_t |\rho_d - \rho_c|^{1.5}}{3\sigma}} \tag{8}$$

To investigate the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the equivalence ratio distributions and the injected fuel droplet behavior, the engine simulation was conducted by using engine mesh as shown in Figure 2. The size of the engine mesh is also a hexahedron of 1 mm × 1 mm × 1 mm (width × length × height) as the test chamber. The specifications of the test engine and test injector were listed in Table 1. The introduced total heating value was fixed at 595 J based on the lower heating value of diesel 14 mg. The heating value of the premixed ethanol ratio varied from 0% to 40% based on the same total heating value in-cylinder in steps of 10%. It was assumed that all the ethanol fuel flowing into the cylinder through the premixing chamber was evaporated and introduced in a vaporized state.

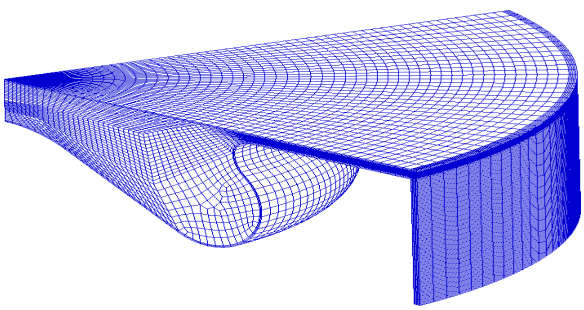


Figure 2. Geometry of test engine mesh.

Table 1. Detailed specifications of test engine and test injector.

	Item	Specification
Engine	Engine type	Single cylinder
	Bore/Stroke	83 mm/92 mm
	Displacement	498 cc
	Compression ratio	17.7
Injector	Nozzle	5 hole mini-sac type
	Hole diameter	0.168 mm
	Inclined spray angle	154 deg

Since the lower heating value of ethanol is 26.8 MJ/kg, the mass of ethanol was increased by 2.22 mg (59.5 J) in order to increase the ethanol ratio by 10% under the same total heating value. Since the lower heating value of diesel is 42.5 MJ/kg, the introduced total heating value into the cylinder excluding the occupied heating value by ethanol was adjusted to decrease the diesel injection mass by 1.4 mg (59.5 J), and the detailed properties of the test fuels are listed in Table 2 [15].

Table 2. Properties of test fuels [15].

Item	Diesel	Ethanol
LHV: lower heating value [MJ/kg]	42.5	26.8
Latent heat of evaporation [kJ/kg]	250	846
Density @ 20 °C [kg/m^3]	838.2	789.4
Carbon content [% mass]	86.7	52.14
Hydrogen content [% mass]	12.71	13.13
Sulfur content [% mass]	0.041	-
Oxygen content [% mass]	-	34.73
Flash point [°C]	67	13
Kinematic viscosity @ 40 °C [mm^2/s]	2.8271	1.056
Typical formula	$C_{14.09}H_{24.78}$	C_2H_6O
Cetane number	42.6	8.5
Lubricity, HFRR @60 °C [μm]	Max. 520	Max. 605

To express the ethanol flowing from the pre-mixing chamber, the initial air composition in the cylinder was adjusted by inducing a change in mass fraction, as listed in Table 3. In the simulation, Diesel-D_1 ($C_{13}H_{23}$) fuel was applied by referring the library to a commercial program [13]. Furthermore, ethanol fuel was used for the homogeneous air–fuel mixture in the cylinder. Since it is an oxygen-containing fuel, it helps combustion and at the same time has a higher heating value than methanol compared to the same mass. The premixed ethanol mass is determined according to the lower heating value of ethanol; the mass fractions of ethanol in the cylinder were adjusted according to the premixed ethanol mass. From these results, the mass fractions of O_2 and N_2 were determined by the ratio of 1:3.31, which are atmospheric conditions. As described above, the diesel mass was changed from 8.4 mg to 14 mg, and the ethanol mass also varied from 0 mg to 8.88 mg according to the premixed ethanol ratio based on the same total heating value in-cylinder. So, the injection data according to the various diesel injection mass were measured using Bosch's suggestion [16] as shown in Figure 3. The start of energizing timing and the injection pressure of diesel fuel were fixed to BTDC 12 deg and 100 MPa, respectively. The detailed numerical analysis conditions for engine simulation are listed in Table 4.

Table 3. Initial air composition in the cylinder according to the heating value ratio of diesel and ethanol.

Fuel Injection Ratio [Main/Premixed]	Fuel Amount [mg]	Heating Value [J]	O_2 Mass Fraction [-]	N_2 Mass Fraction [-]	Ethanol Mass Fraction [-]
D100/E0	14/0	595/0	0.23200	0.76800	0.00000
D90/E10	12.6/2.2	535.5/59.5	0.23081	0.76407	0.00512
D80/E20	11.2/4.4	476/119	0.22984	0.76084	0.00932
D70/E30	9.8/6.7	416.5/178.5	0.22876	0.75273	0.01396
D60/E40	8.4/8.9	357/238	0.22781	0.75413	0.01806

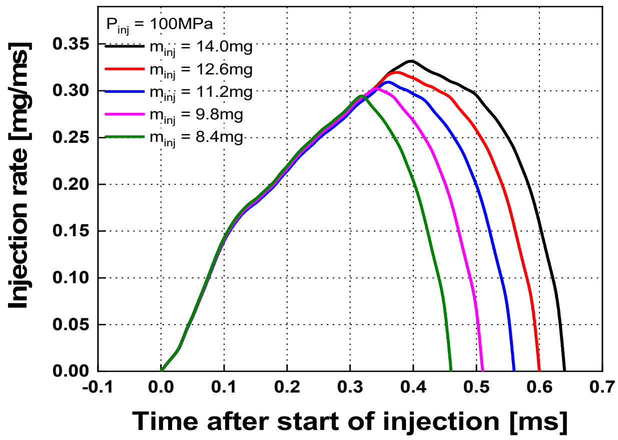


Figure 3. Injection rate according to the injection mass of diesel fuel.

Table 4. Numerical analysis conditions for engine simulation.

Contents	Numerical Analysis
RPM	1800
Injection pressure [MPa]	Diesel: 100, ethanol: 10
Total heating value [J]	595
Start of energizing timing [ATDC deg]	−12
Fuel injection ratio [main fuel/premixed fuel]	D100/E0, D90/E10, D80/E20,D70/E30, D60/E40

3. Results

3.1. Validation of Numerical Models

In this work, the reliability of the applied sub-model in the numerical analysis was secured by comparing the results of spray simulations with experiment results. To secure the exact reliability, the results of spray characteristics were compared according to the change in the ambient pressure because it was changed according to the crank angle. The ambient pressure in the chamber was changed from 2.5 MPa to 3.5 MPa. Figure 4 shows the comparison results of spray evolution according to the ambient pressure. The spray evolution results in the simulation were traced well from the results of the experiment. As mentioned above, if the value of the break-up time constant is small, the injected fuel droplet splits rapidly, and then the spray tip penetration is shortened. In addition, when the ambient pressure is high, the spray cone angle is wider, as shown in Figure 5, because the fuel droplet was broken up well by the high ambient pressure, which increases the drag force.

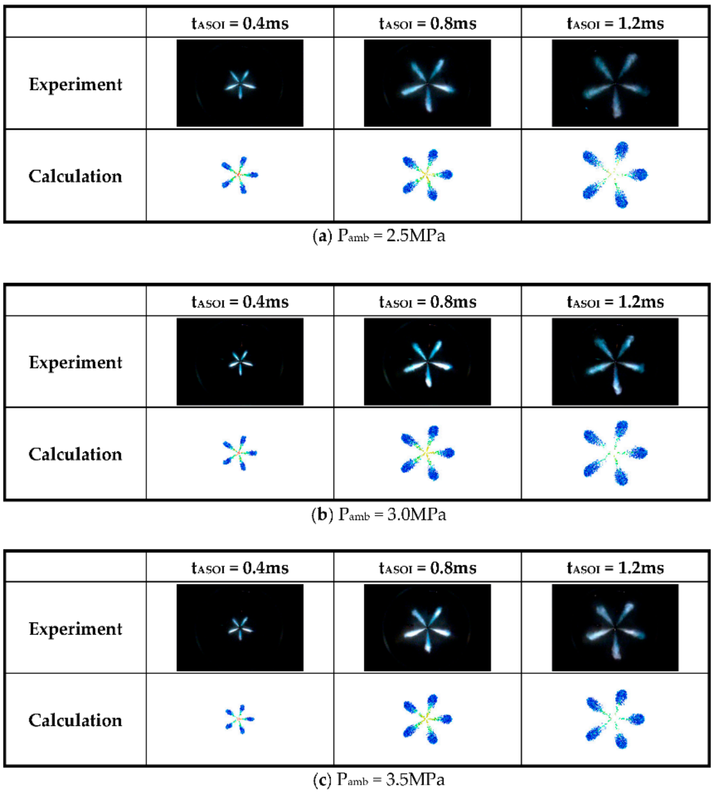
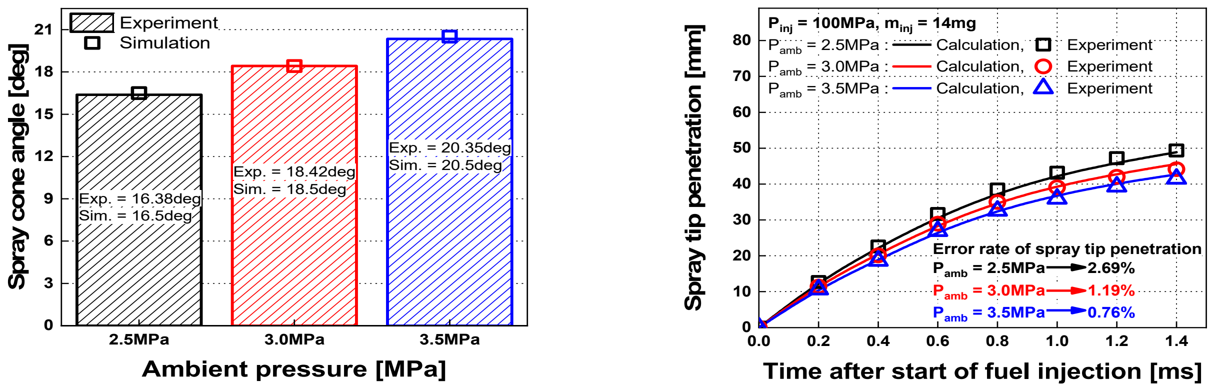


Figure 4. Validation results of spray evolution characteristics (P_{inj} = 100 MPa).



(**a**) Spray cone angle (t_{ASOI} = 0.8 ms) (**b**) Spray tip penetration

Figure 5. Validation results of spray cone angle and spray tip penetration characteristics (P_{inj} = 100 MPa).

In the comparison between the results of the numerical analysis and the experiment, the error rates of the spray cone angle according to the ambient pressure were within 0.73%, 0.43%, and 0.74%, respectively. The error rates according to the ambient pressure were less than a maximum of 2.69%. From these results, it can be said that the cylinder pressure change due to the crank angle variation can be sufficiently described for the spray cone angle and spray tip penetration.

3.2. The Effect of Ethanol Ratio on the Characteristics of Spray Evolution

The analysis of spray tip penetration is very important because, as reported by Lee et al. [17], the amount of exhaust emissions formation was different according to the collided position of the injected fuel on the piston wall. The injected fuel should be targeted to the point in a cylinder exactly at the desired timing to form a homogeneous air–fuel mixture and ignite. Figure 6 shows the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the visualization of spray evolution characteristics. It was found that when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the collided amount of diesel fuel to the cylinder wall was decreased because the amount of diesel fuel was decreased, as shown in Figure 3. To exactly compare spray tip penetration according to the premixed ethanol ratio based on the same total heating value in-cylinder, Figure 7 shows the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the spray tip penetration characteristics. As shown in Figure 6, the difference in the spray tip penetration value according to the premixed ethanol ratio was difficult to observe until the time after 3 deg of diesel injection started. However, in the case of D60/E40, it was observed that the spray tip penetration value slightly decreased at the time after the start of 4 deg of diesel injection because the diesel injection was complete due to the shorter diesel injection duration compared to the other diesel injection duration. From these results, it was found that, as shown in Figure 8, although the injected fuel kinetic energy was decreased by 20.7% whilst increasing the ethanol ratio based on the same total heating value in-cylinder, the spray tip penetration was not affected the kinetic energy, and it was affected by the injection duration.

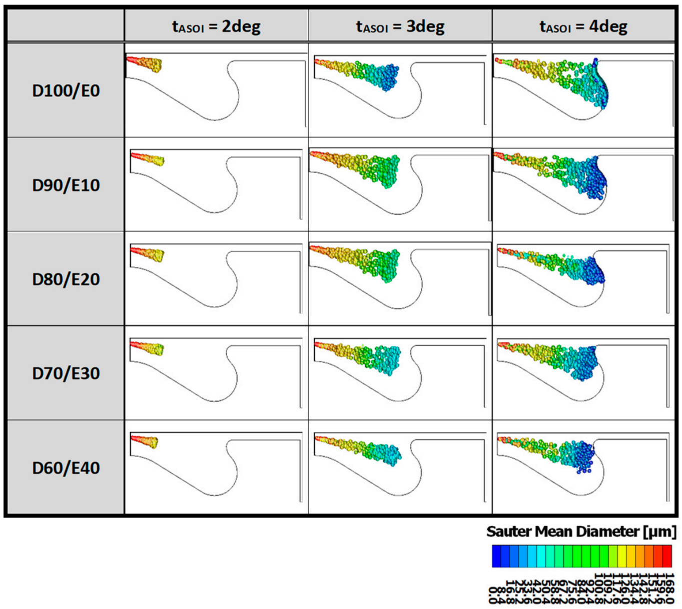


Figure 6. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the visualization of spray evolution characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

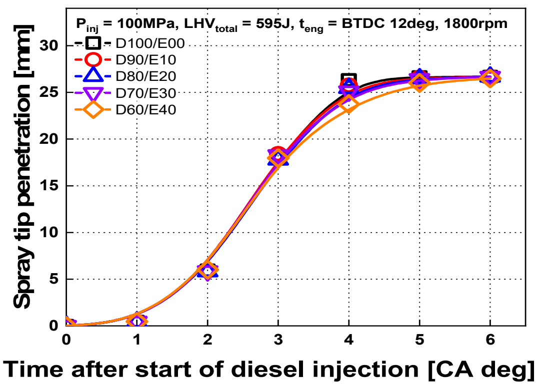


Figure 7. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the spray tip penetration characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

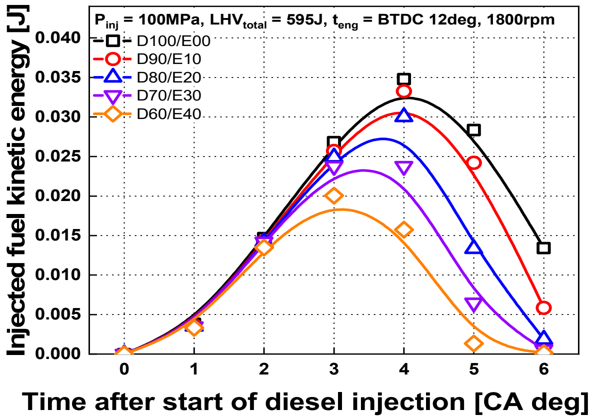


Figure 8. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the injected fuel kinetic energy characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

3.3. The Effect of Ethanol Ratio on the Fuel Behavior Characteristics

The injected diesel fuel for igniting the homogeneously distributed ethanol fuel in the cylinder may form particulate matter due to incomplete combustion by auto-ignition characteristics. The injected fuel droplet that breaks up into smaller sizes can quickly evaporate and mix well with the air in the cylinder, and it induces good combustion performance and reduces the particulate matter.

The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the SMD characteristics was shown in Figure 9. It was observed that when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the SMD value also became larger up to about 32.58%. Generally, it is known that the injected fuel is greatly affected by viscosity (μ) in the injected fuel and Weber number (We) [18]. So, in this study, the density influences Weber number in the cylinder was analyzed as shown in Figure 10. Figure 10 shows the effect of a premixed ethanol ratio based on the same total heating value in-cylinder on the density characteristics.

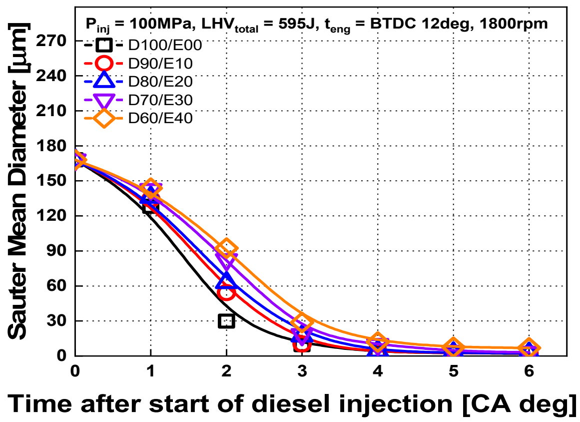
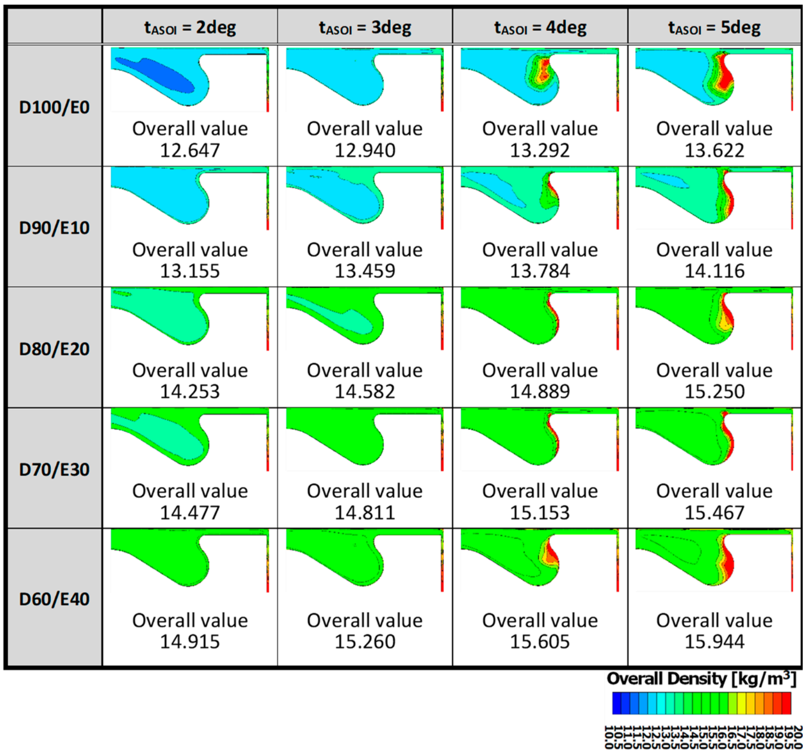
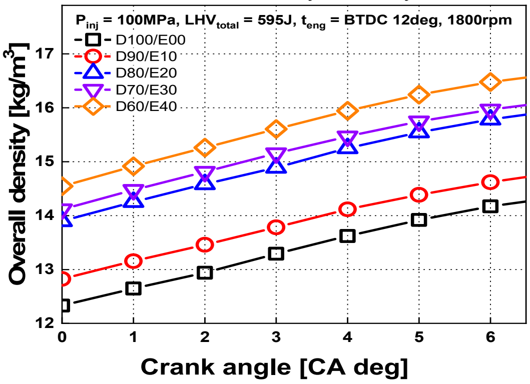


Figure 9. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the Sauter mean diameter characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).



(**a**) Distributions of density in the cylinder.



(**b**) Overall density in the cylinder.

Figure 10. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the density in the cylinder characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

When the premixed ethanol ratio was increased, the density value in the cylinder was also increased by about 3.85% because the intake air temperature was decreased by −4.16%, as shown in Figure 11, due to the latent heat of the evaporation of the ethanol injected into the premixing chamber [12]. Moreover, over time, high-density distributions were observed at the piston bowl wall and the piston rim region because the injected diesel fuel collided with the piston rim region, and it was broken up well by the collision effect. The well-broken small size of diesel droplets was converted to the gas state by evaporation, and this increased the density. Furthermore, it was found that when the time after the start of the diesel injection was 2 deg, although a difference in the injected fuel kinetic energy was observed, as shown in Figure 8, the difference in SMD was large, as shown in Figure 9, because both the injected fuel kinetic energy and the SMD value were obtained as the overall values in the cylinder. In the case of the injected fuel kinetic energy, although the total injected fuel kinetic energy was the same, the injected fuel droplets were well split. However, in the case of SMD, the difference in values appeared to be relatively large because when the ethanol ratio was increased, the diesel fuel injection duration was shorter.

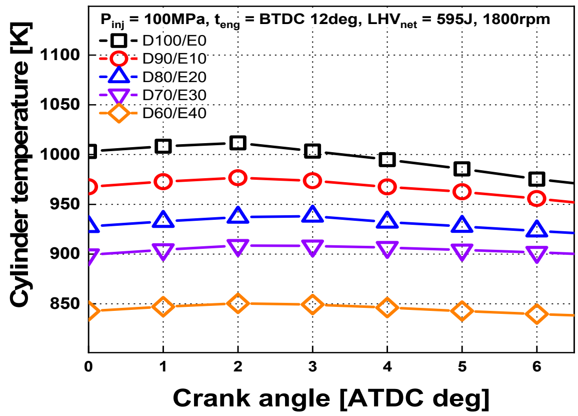


Figure 11. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the cylinder temperature characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

Since the drag force also affects Weber's number, it is compared as shown in Figure 12. Figure 12 shows the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the drag force. The original drag force should be calculated using the local density value; however, it is difficult to obtain the value. So, in this study, the overall density value was applied to calculate the drag force because the density distribution in the cylinder was observed to be almost the same, excluding the crevice volume, as shown in Figure 10a. As shown in Figure 8, although the kinetic energy decreased by 20.7% with an increasing ethanol ratio based on the same total heating value in-cylinder, the drag force showed a similar value due to the increase in the density in the cylinder. However, when the ethanol ratio based on the same total heating value in-cylinder was increased, although the difference in the SMD values was little at the beginning of diesel fuel injection, it decreased after 3 deg, which affects combustion by 32.58%. Thus, it was judged that the drag force was not the affected factor for the SMD.

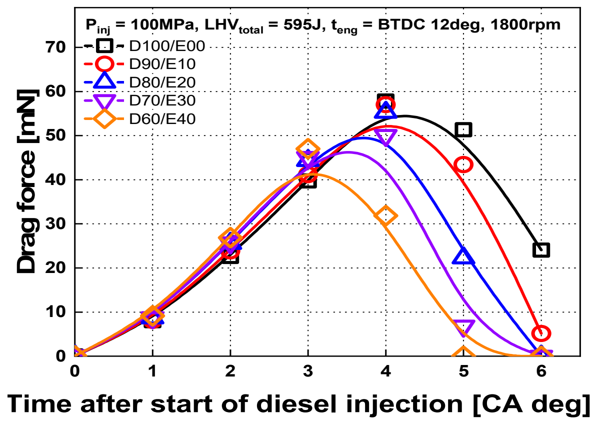


Figure 12. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the drag force in the cylinder characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

Therefore, the evaporation rate, which is another factor affecting SMD, was compared as shown in Figure 13. The injected fuel evaporation rate was determined by the evaporated amount per injected fuel mass at each point in time. It was found that when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the injected fuel slowly evaporated due to the decrease in the intake air temperature by −4.16%.

Consequently, it was found that the kinetic energy and the evaporation of the injected fuel were more affected by the droplet atomization of the injected fuel than the drag force of the injected fuel which affects Weber's number. From these results, it is expected that a low amount of NO_X is generated because a low combustion temperature occurs due to a longer combustion period by the slow evaporating fuel under the conditions of D60/E40.

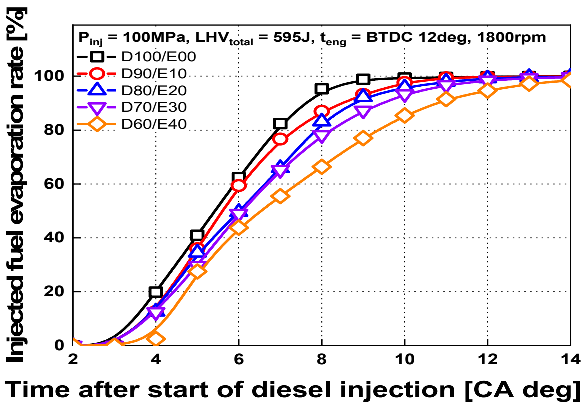


Figure 13. The effect of the premixed ethanol based on the same total heating value in-cylinder on the injected fuel evaporation rate characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

3.4. The Effect of Ethanol Ratio on the Equivalence Ratio Distribution Characteristics

Figure 14 shows the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the equivalence ratio distribution characteristics. At the beginning of the diesel injection (t_{ASOI} = 5 deg), it was observed that when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the overall value of the equivalence ratio increased. The reason is that although the injected fuel slowly evaporated due to the low cylinder temperature, the amount of premixed ethanol also increased, which increased the equivalence ratio. However, at t_{ASOI} = 10 deg, the overall value of the equivalence ratio decreased with an increasing premixed ethanol ratio because many unevaporated diesel droplets remained in the cylinder due to the slowly evaporating injected diesel fuel.

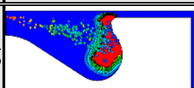
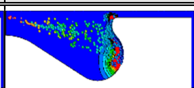
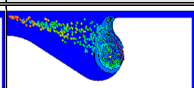
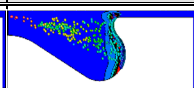
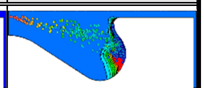
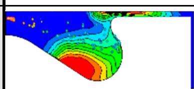
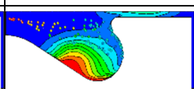
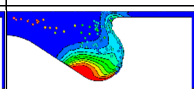
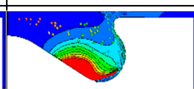
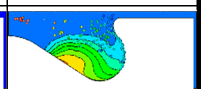
	D100/E0	D90/E10	D80/E20	D70/E30	D60/E40
t_{ASOI} = 5 deg (BTDC 12 deg)	 $\Phi_{overall}$ = 0.1984	 $\Phi_{overall}$ = 0.1367	 $\Phi_{overall}$ = 0.1548	 $\Phi_{overall}$ = 0.1847	 $\Phi_{overall}$ = 0.2323
t_{ASOI} = 10 deg (BTDC 12 deg)	 $\Phi_{overall}$ = 0.4820	 $\Phi_{overall}$ = 0.3324	 $\Phi_{overall}$ = 0.3220	 $\Phi_{overall}$ = 0.3105	 $\Phi_{overall}$ = 0.3081

Figure 14. The effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the equivalence ratio distribution characteristics (P_{inj} = 100 MPa, LHV_{total} = 595 J, t_{eng} = BTDC 12 deg, 1800 rpm).

At t_{ASOI} = 5 deg, a rich equivalence ratio was distributed in the piston bowl because the injected diesel fuel collided with the piston bowl wall, and the collided diesel fuel droplet was atomized by the collision effect so that the smaller diesel fuel droplets evaporated well. Furthermore, at t_{ASOI} = 10 deg, it was observed that a part of the injected fuel was introduced to the squish volume, and the amount of the introduced fuel into the squish volume was decreased with an increasing premixed ethanol ratio based on the same total heating value in-cylinder. In general, as reported by Lee et al. [17], when a small amount of fuel among the total injected fuel is introduced into the piston bowl and a large amount of fuel is introduced into the squish volume, the combustion performance deteriorates because the squish volume has a small amount of air. The small amount of air induces incomplete combustion by the rich equivalence ratio and then a lot of exhaust emissions, such as soot and CO, are generated because it cannot be oxidized by a lean concentration of O_2. In this study, when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the amount of injected diesel fuel decreased, and the amount of diesel fuel introduced into the squish volume was also decreased, as shown in Figure 14. From these results, it is expected that the combustion performance will be promoted by increasing the ethanol ratio based on the same total heating value in-cylinder because the amount of diesel fuel with a high heating value introduced into the squish volume was small.

4. Conclusions

The aim of this study is to investigate the effect of the premixed ethanol ratio based on the same total heating value in-cylinder on the equivalence ratio distributions and injected fuel droplet behavior in the cylinder under the RCCI engine. The conclusions we were able to obtain are as follows:

1. In the case of D60/E40, it was observed that the spray tip penetration value was slightly decreased at the time after the start of the 4 deg of diesel injection because the diesel injection was completed due to the shorter diesel injection duration compared to the other diesel injection amount.
2. When the premixed ethanol ratio based on the same total heating value in-cylinder increased, the SMD value was also more enormous, up to 32.58%, because of the low kinetic energy of the injected fuel by the short injection duration and the slow evaporation of the injected fuel by the low cylinder temperature.
3. At t_{ASOI} = 5 deg, when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the overall value of the equivalence ratio was increased because the amount of premixed ethanol was increased. However, at t_{ASOI} = 10 deg, the overall value of the equivalence ratio was decreased because many unevaporated diesel droplets remained in the cylinder due to the slowly evaporating injected diesel fuel.
4. At t_{ASOI} = 10 deg, it was observed that when the premixed ethanol ratio based on the same total heating value in-cylinder was increased, the amount of injected diesel fuel was decreased, and the amount of diesel fuel introduced into the squish volume was also decreased.
5. It is expected that the combustion performance will be promoted by increasing the ethanol ratio based on the same total heating value of the introduced fuels into the cylinder because the amount of diesel fuel with a high heating value introduced into the squish volume was small.

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