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A study on the spray and engine combustion characteristics of diesel–dimethyl ether fuel blends

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Abstract

The purpose of this study was to compare the spray characteristics, the combustion characteristics and the emissions (nitrogen oxides, carbon monoxide, hydrocarbon and smoke) of typical fuels (100% diesel and 100% dimethyl ether) and diesel–dimethyl ether fuel blends in a constant-volume chamber and a single-cylinder direct-injection diesel engine. The spray characteristics were investigated by varying the ambient pressure and the fuel injection pressure using a common-rail fuel injection system with various fuel mixture ratios. The spray characteristic research parameters were the spray shape, the penetration length and the spray angle at the seven-hole injector. Common types of injector were used (Bosch). Two types of fuel blended by mass fraction were employed. Typical fuels (100% diesel and 100% dimethyl ether) and fuel blends with diesel:dimethyl ether mixture ratios of 95:5 and 90:10 were used. The injection pressure was fixed at 70 MPa while the ambient pressure was varied (0 MPa, 2.5 MPa and 5 MPa). The combustion experiments were conducted in a single-cylinder engine equipped with a common rail. The injection pressure was 700 bar at 1200 r/min. The amount of injected fuel was adjusted to obtain a fixed input calorific value of 972.2 J/cycle in order to make a comparison between the fuel types. The results showed that the injection quantity was greater with diesel fuel but not for dimethyl ether fuel and for the fuel blend with the higher mixing ratio. The spray penetration length increased with increasing ambient pressure for diesel but decreased for dimethyl ether with a larger spray angle. The ignition delay and the heat release rate decreased with increasing dimethyl ether which led to a higher indicated mean effective pressure and higher thermal efficiency because there was less negative work in the expansion. The total hydrocarbon emissions and the carbon monoxide emissions decreased with increasing dimethyl ether, but the nitrogen oxide emissions generated were greater owing to the increased combustion temperature.

Keywords

Diesel–dimethyl ether fuel blends, emission characteristics, spray characteristics, reactivity

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Introduction

Engine exhaust emissions, which include carbon monoxide (CO), hydrocarbon (HC) and carbon dioxide (CO_2) emissions, are a continuing concern from an environmental perspective. Compression ignition engines, which are subject to strict regulation of CO_2 emissions and fuel economy, are considered for use in environmentally friendly cars. However, the air–fuel mixture, which exists as a heterogeneous mixture in the combustion process, occurs in local areas of high equivalence ratio. As a result, harmful particulate matter (PM) emissions are generated. In order to meet the strict regulations regarding compression engines, several research institutes have investigated new combustion technologies, after-treatment devices and alternative fuels.

There are some alternative fuels, including natural gas and liquefied petroleum gas (LPG) fuel, which drastically reduce the PM emissions. However, these can only be used in spark ignition engines because of their low cetane number, since the thermal efficiencies of spark ignition engines are lower than those of compression ignition engines.

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Another alternative fuel is dimethyl ether (DME), which can be used in compression ignition engines owing to its high cetane number and low ignition temperature. Moreover, the high cetane number which results from the low ignition temperature and almost instantaneous vaporization can result in a shorter total ignition period. The low boiling point of DME fuel leads to quick evaporation when a liquid-phase DME spray is injected into the engine cylinder and, as a result, can lead to better mixing with the air in the engine cylinder. Smokeless combustion can be expected for DME because of its high oxygen content and the absence of C–C bonds in its molecular structure. Despite the extremely low exhaust emission performance of DME fuel, relatively higher NO_x emissions have been reported by many researchers.[1–3] On the other hand, DME fuel requires a higher injected mass to obtain the same amount of energy as diesel fuel because of its lower density and combustion enthalpy. Furthermore, current diesel fuel injection systems are not suitable for DME fuel owing to leakage problems because of its low viscosity. Moreover, the lubricity of DME fuel is lower than that of diesel fuel, which leads to wear problems. Therefore, a study on the development of a suitable mixed-fuel injection system is required.[4,5]

Investigations have been conducted on the effect of dual-fuel engines. There are many additive fuels for diesel such as LPG, compressed natural gas and DME. Most of these blends maintain the performance of a conventional engine, but the production of nitrogen oxides (NO_x) depends on the circumstances. In order to compensate for this, an exhaust after-treatment system is required in addition to modifications to the injection system. Control of the combustion could also be difficult because of the differences in the cetane and the octane numbers. Recently, fuel blends containing DME have begun to be studied in an attempt to solve these problems associated with dual-fuel engines. Komatsu et al.[6] studied combustion characteristics within the engine using a DME–methane fuel blend. Additionally, Bielaczyc et al.[7] and Delgado and Paz[8] reported the engine performance and the emission characteristics using an ethanol–gasoline fuel blend. Furthermore, Sivalakshmi and Balusamy[9] studied the performance and characteristics of the exhaust gas in a diesel engine using a biodiesel–DME fuel blend.

Ikeda et al.[10] investigated a diesel–DME fuel blend containing 40% DME using an injection pressure of 170 bar. As a result, when the injection timing was retarded, the NO_x emission levels of the fuel blend and the diesel fuel were similar, and the smoke emissions decreased while the HC emissions increased. However, when the brake mean effective pressure (BMEP) was increased, the HC production levels of the fuel blend and the diesel fuel were similar, the NO_x emissions decreased and the smoke emissions increased. In this paper, there is no comparison of the DME fuel and diesel fuel.

The previous study employed a mechanical fuel supply system rather than a common-rail system. However, the density and the elastic modulus of DME, which is a highly compressible fuel, change with the pressure and the temperature. As a result, it is difficult to obtain the required amount of fuel injection as well as to control the amount of fuel injected under appropriate operating conditions. The fuel injection pump requires a high capacity and measures the pressure and temperature changes, as required by an electronic rather than a mechanical fuel supply system and, for this reason, is suitable for the common rail-system.[11]

Wang et al.[12] investigated diesel–DME fuel blends with mixtures containing 10 mass %, 20 mass % and 30 mass % DME at an injection pressure of 180 bar. As a result, for high-load conditions, fuel blends have stronger effects on the smoke emissions while, at low BMEPs, the fuel blends have only slight effects on the smoke reduction. With the addition of DME, the NO_x emissions decrease slightly, while the CO emissions and the unburned HC emissions increase for most operating conditions.

Yu et al.[13,14] investigated diesel–DME fuel blends which contained 25 mass %, 50 mass % and 75 mass % DME according to the intake air temperature, the injection pressure and the BMEP. The difference from other studies was the injection port of the fuel. As a result, the spray angle became larger and the spray penetration length became shorter with increasing DME mass fraction. With increasing DME, the viscosity of the fuel blend decreased, and a slight increase in the vapour pressure of the fuel system was observed. The 50% DME fuel blend is optimal for solving the problems of pure diesel or liquefied DME fuels for homogeneous charge compression ignition (HCCI) engines. An increase in the intake air temperature leads to a very narrow load range because of knocking-like limitations. This suggests that CO_2 is a good way to control the ignition timing and to extend the operable range for HCCI engines.

The objectives of this study are to form oxygenated blends by adding DME to diesel fuel and then to investigate the characteristics of the spray in a constant-volume combustion chamber together with the characteristics of the performance emissions in a single-cylinder compression ignition engine fuelled with different proportions in the fuel blends.

Experimental apparatus

Spray visualization and DME injection system

Figure 1 shows a schematic diagram of the experimental apparatus. The left-hand part of Figure 1 is the fuel injection system. In this experiment, a high fuel injection pressure was maintained in the common-rail system.

The low viscosity and the poor lubricity are caused by fuel leakage and wear in the fuel supply system.

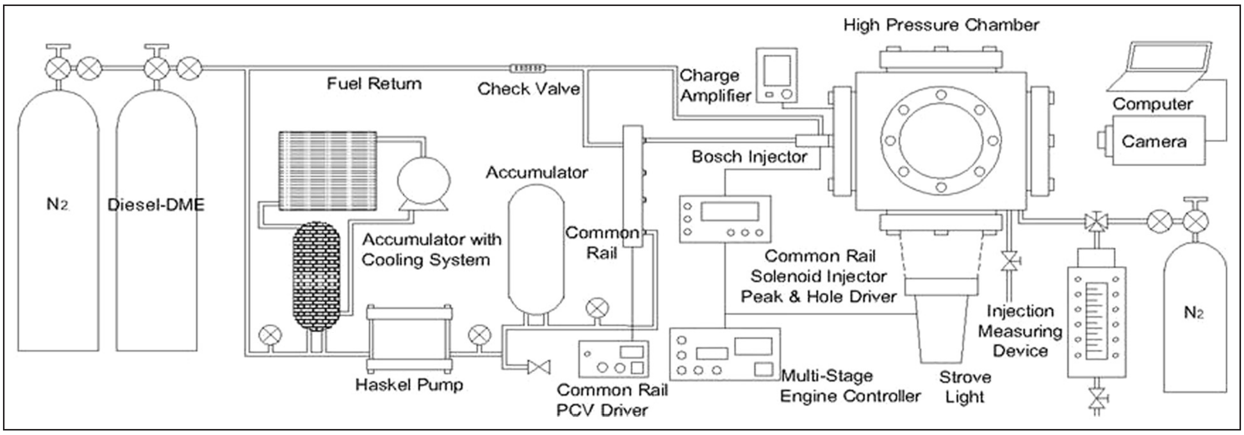

Figure 1. Schematic diagram of the experimental apparatus.
N_2: nitrogen; DME: dimethyl ether; PCV: passenger-carrying vehicle.

Thus, an air compressor (Haskel pump) was used to fix this disadvantage, and 2 vol % biodiesel was added to improve the lubricity of DME. DME exists in the gaseous state at atmospheric pressure owing to its low evaporation point; therefore, a pressure of at least 0.8 MPa should be applied in order to maintain a liquid state in the tank.

DME, which is pressurized at greater than 0.8 MPa using nitrogen pressurization, is supplied to the accumulator. DME cannot be stored in the tank as a liquid because of the difference between the internal and external temperatures in the tank. Therefore, the accumulator wall was wrapped with copper tube to form a circulation system in which the cooled refrigerant was cycled by a compressor, and the internal DME was kept in a liquid state to prevent evaporation.

After pressurization to around 70 MPa, liquid DME was stored via an air compressor in an accumulator installed before the common rail. Fuel was injected at a constant pressure using the common-rail system controlled by a passenger-carrying vehicle drive.

A Bosch solenoid-type seven-hole injector was used. The injection time and the number of injections were controlled by a multi-stage injection engine controller. Fuel, which returned to the common rail from the injector, was again returned to the low-pressure pump and was then circulated.

The right-hand part of Figure 1 shows the spray visualization system in which the constant-volume combustion chamber inside was pressurized by nitrogen. A strobe light was used as a light source, and it was operated by the multi-stage injection engine controller. The internal pressure of the chamber was measured via an installed pressure sensor. The measured pressure signal was amplified with a signal amplifier (Kistler 5015). Further, a camera, which was used to record the spray shape, was operated with a connected computer.

Injection quantity measurement device. On the bottom right of Figure 1, the measurement device used to measure the fuel quantity is included in the schematic diagram. The fuel-quantity-measuring device was designed to endure pressures over 0.8 MPa, which is the pressure required to prevent evaporation of DME.

An engraved scale optical window, which can be used to confirm the injection quantity, was installed in the front part of the measurement system. After completion of the injection quantity measurements, the internal liquid fuel was discharged by opening the drain valve in the bottom of the measuring device.

Experimental apparatus for the engine. Figure 2 is a schematic diagram of the engine facility. The engine in this study is a single-cylinder compression ignition engine with an installed common-rail system. The calculated compression ratio of the engine and the total displacement volume are 19.5:1 and 498 cm^3 respectively. Details of the engine specifications are listed in Table 1. There is no engine control unit in the single-cylinder engine, and therefore the engine controller system must be configured. The crank angle (CA) is signalled from the rotary encoder (Autonics E40S), which is referred to as the position of the crankshaft transferred to the injector driver (TEMS TDA-3300) via the interface box (Zenobalti Co. ZB-100) and the engine controller

Table 1. Engine specifications.

Engine type	Water-cooled, 4 cycle diesel
Number of cylinders	1
Bore	83 mm
Stroke	92 mm
Displacement	498 cm^3
Number of valves	2 intake, 2 exhaust
Compression ratio	19.5
Valve timing, intake	8° CA BTDC; 44° CA ABDC
Valve timing, exhaust	54.6° CA BTDC; 7.4° CA ATDC
Fuel system	Common rail

CA: crank angle; BTDC: before top dead centre; ABDC: after bottom dead centre; ATDC: after top dead centre.

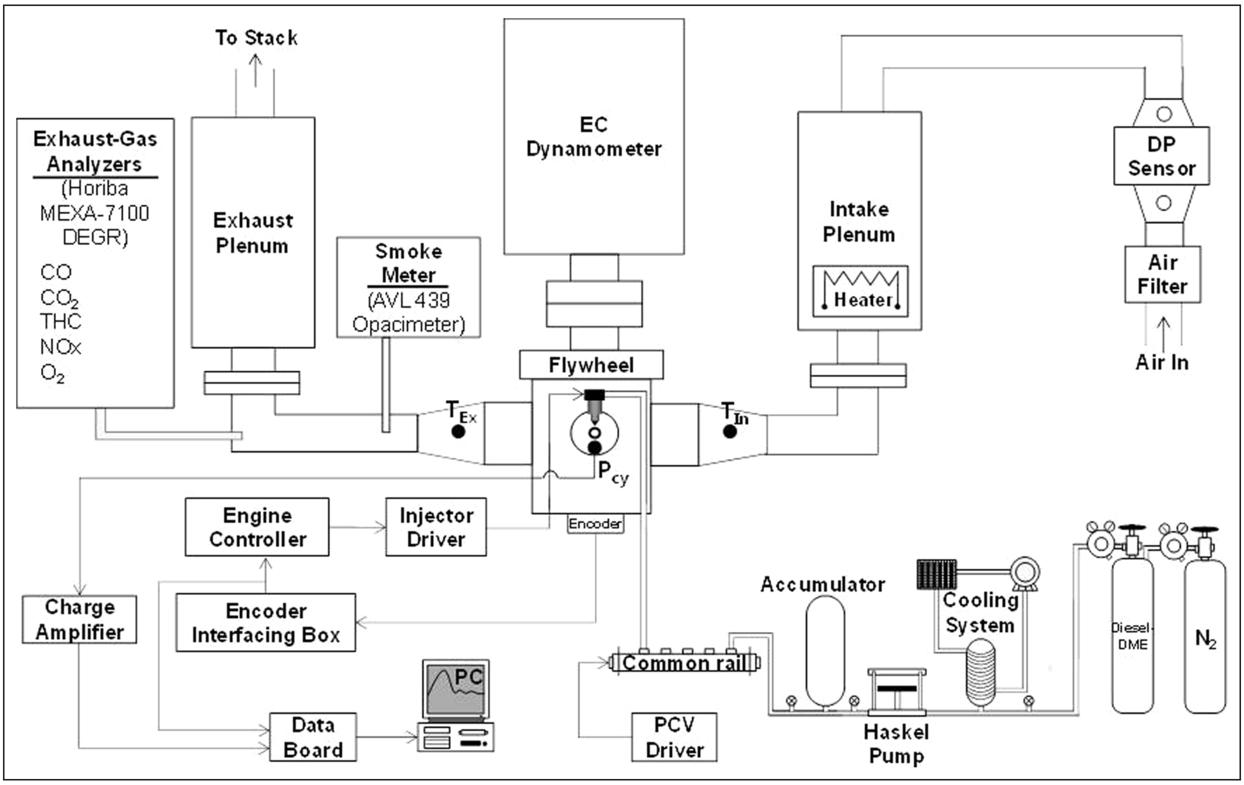

Figure 2. Schematic diagram of the engine facility.
CO: carbon monoxide; CO_2: carbon dioxide; THC: total hydrocarbon: NOx: nitrogen oxides; O_2: oxygen; PC: personal computer; EC: eddy current; PCV: passenger-carrying vehicle; DP: differential pressure; DME: dimethyl ether.

(Zenobalti ZB-8035). This is a means of adjusting the injection times and the injection duration. The pressure in the common rail can be controlled by using the driver control valve (Zenobalti ZB-1100). A 150 kW eddy current dynamometer was used to control the engine speed.

The CO, HC and NO_x emissions were measured using an exhaust gas analysis device (HORIBA MEXA-7100 DEGR), and the smoke emissions were analysed with a smoke meter (AVL 415) to indicate the quantities of the emissions. Each average displacement volume of the exhaust gas flow per minute was recorded during steady-state engine operation. In order to measure an internal cylinder pressure, a piezoelectric-type pressure sensor (Kistler 6056A) was inserted into the hole of the cylinder head, and the pressure data were gathered per 0.1° CA. One average cycle was calculated from the whole data set of 50 cycles.

Test fuel

Ultra-low-sulphur diesel and 98% pure DME (2 vol % biodiesel was added to improve the lubricity) was used to study the spray characteristics of the diesel fuel, the DME fuel and the fuel blends. The fuel properties of diesel and DME are listed in Table 2.

Based on previous research[15] regarding the blending of diesel and DME, phase separation does not occur above 273.83 K and 96.1 kPa. Diesel and DME were blended at room temperature and a high pressure, i.e. 293.83 K and 1000 kPa.

Table 3 shows the compositions of the test fuels. The composition ratio of the test fuel was determined using

Table 2. Properties of the fuels.

Property	Value for the following	
	Diesel	DME
Chemical structure	$C_nH_{1.8n}$	CH_3OCH_3
Cetane number	40–50	$\gg 55$
Vapour pressure (20 °C)	—	0.51 MPa
Autoignition temperature	250 °C	235 °C
Low heating value	42.7 MJ/kg	28.8 MJ/kg
Boiling point (1 atm)	180–370 °C	–25.1 °C
Liquid density	831 kg/m^3	667 kg/m^3
Oxygen content	0%	34.8%

DME: dimethyl ether.

Table 3. Fuel blending ratios of the test fuel.

	Blending ratio (wt %) for the following			
	ULSD	DiDm 95	DiDm 90	DME
Diesel	100	95	90	0
DME	0	5	10	100

ULSD: ultra-low sulphur diesel; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100; DME: dimethyl ether.

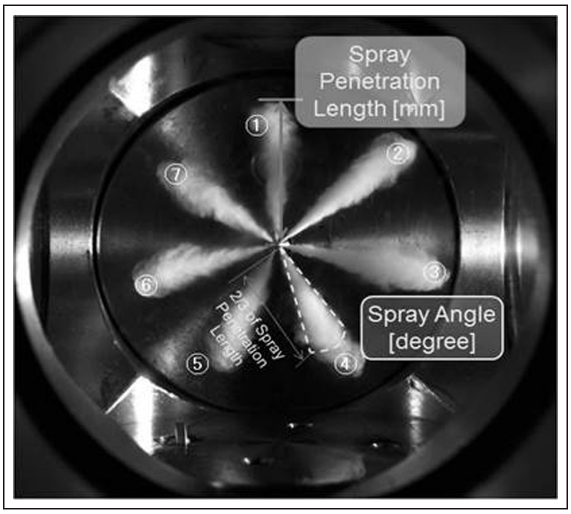

Figure 3. Definitions of the spray penetration length and the spray angle.

$$\mathrm{DiDm}(\%) = \frac{\mathrm{diesel(kg)}}{(\mathrm{diesel} + \mathrm{DME})(\mathrm{kg})} \times 100 \tag{1}$$

Experimental methods

Fuel blending method. Fuel was newly created by blending diesel and DME, which was charged into the 20 kg LPG gas tank. The LPG tank was placed on a digital scale, a new zero point was obtained, and then fuel was charged depending on the weight ratio. First, diesel pressurized to 1 MPa was introduced into the tank using a low-pressure pump.

The chamber's internal pressure was maintained above 0.8 MPa using nitrogen to prevent vaporization of DME. The DME fuel, which was pressurized to 1 MPa by nitrogen, was next charged into the tank. Then, the tank was shaken to mix the liquid DME and diesel in the tank. The fuel blend was used at least 6 h later to ensure that it was sufficiently mixed. For safety, the tank was charged to less than 70% of tank capacity.

Measurement method for the fuel quantity. In this study, the fuel injection pressure was 70 MPa. The fuel quantity was measured using a Bosch solenoid-type commercial injector (hole diameter, 0.134 mm). In all test conditions, the injection time was 1 ms, and the injections were carried out identically for 1000 repetitions.

Method of spray visualization. Fuel was injected when the injection pressure was 70 MPa and the ambient pressure was 0 MPa, 2.5 MPa or 5 MPa. In all conditions, the fuel injection period was 1 ms.

The main measurement factors in this study were the spray penetration length and the spray angle, as shown in Figure 3. The spray penetration length was measured from the injector nozzle tip to the end of the spray droplet, and the spray angle was defined as the angle between two points at the outer boundary at two thirds of the spray penetration length.

Table 4. Operating conditions.

Engine speed	1200 r/min
Injection pressure	700 bar
Injection timing	340–365° CA
Input calories	972.2 J/cycle
Intake pressure	Naturally aspirated
Intake temperature	25 ± 2 °C
Coolant temperature	80 ± 1 °C

Table 5. Injection quantities.

Fuel	Injection quantity (ml/cycle)
Diesel	0.0909
DME	0.0905
DiDm 95	0.0898
DiDm 90	0.0893

DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

Method of engine experimentation. Table 4 shows the operating conditions. The intake air temperature and the coolant temperature were approximately 25 ± 2 °C and 80 ± 1 °C respectively. The injection pressure was 700 bar using a common-rail system at 1200 r/min. The amount of injected fuel was adjusted to obtain a fixed input calorific value of 972.2 J/cycle in order to compare the fuel conditions. The range of injection timings was 340–365° CA.

To quantify the amount of fuel injected in a single injection, all measurements were carried out over 1000 cycles, and the injected fuel was collected. However, for the measurement to be effective on DME fuel, a substantially different fuel measurement system was required because the boiling point of DME fuel is much lower than that of diesel fuel. To measure the injection quantity of DME fuel more accurately, the DME fuel was injected into the pressurized chamber by nitrogen at 40 bar.

Experimental results

Injection quantity measurement results. Table 5 shows each injection quantity for the diesel fuel, the DME fuel and the fuel blends. The overall values were the averages for 1000 separate injections. The largest injection quantity occurred when conventional diesel fuel was injected; the next highest quantity was that of DME.

Table 5 shows that the injection quantity of the diesel–DME fuel blend decreased owing to the addition of DME to diesel. This is because the kinematic viscosity coefficient of DME is lower than that of diesel.

Spray visualization results. Figures 4 to 7 show the spray shape of each fuel (diesel, DiDm 95, DiDm 90 and

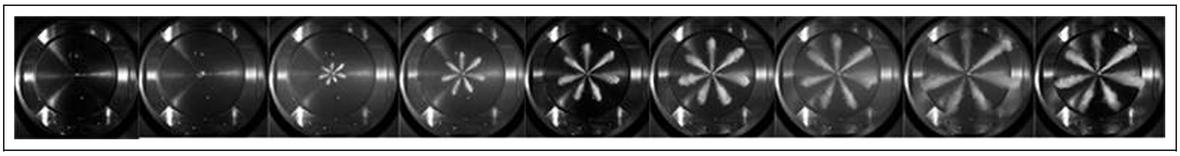

Figure 4. Diesel spray shape (P_{inj} = 70 MPa; P_{amb} = 5 MPa).

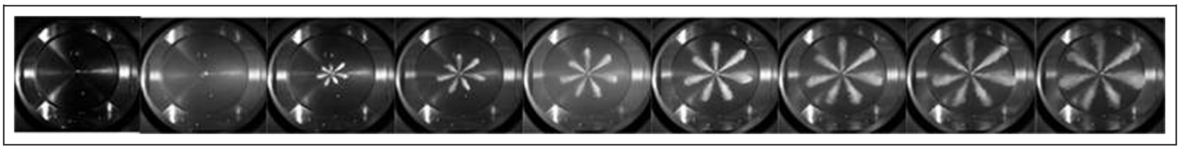

Figure 5. DiDm 95 spray shape (P_{inj} = 70 MPa; P_{amb} = 5 MPa).

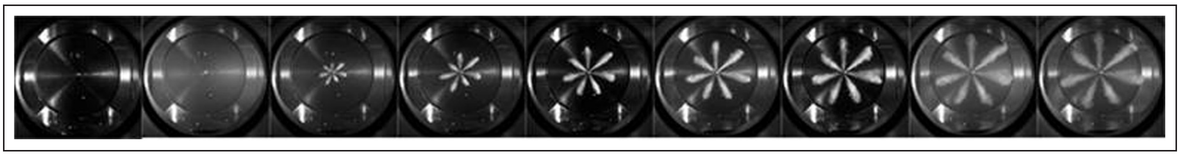

Figure 6. DiDm 90 spray shape (P_{inj} = 70 MPa; P_{amb} = 5 MPa).

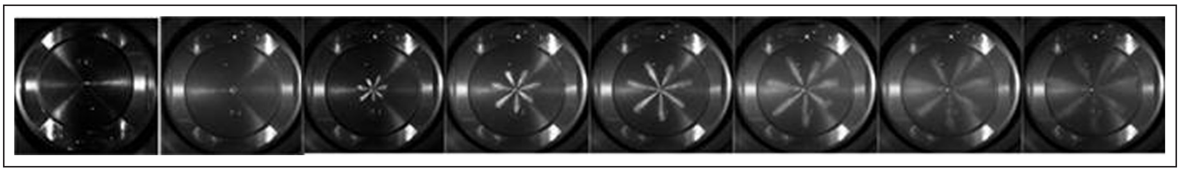

Figure 7. DME spray shape (P_{inj} = 70 MPa; P_{amb} = 5 MPa).

DME respectively) using the same Bosch injector when the injection pressure was 70 MPa and the ambient pressure of the combustion chamber was 5 MPa.

The photographs show the spray growth process at intervals of 0.4 ms until full growth from after the start of injection. Diesel fuel required 3.2 ms to reach the combustion chamber walls as fully developed spray droplets. It can be confirmed that most spray droplets were formed within 2.8 ms before reaching the combustion chamber walls owing to the good DME evaporation characteristics.

The atomization of fuel accelerated as a result of mixing DME fuel with the inside air of the cylinder, thereby improving the combustion characteristics.

Spray penetration length measurement results. The main point in the analysis of the spray penetration length was how quickly the air and the fuel became fully mixed. In other words, the key point was how rapidly the air–fuel mixture proceeded along the spray penetration length. Figure 8 shows that most injected fuel at a high pressure reached the combustion chamber wall in a liquid state within 1.4 ms under non-pressurized conditions in the combustion chamber. However, the behaviour varied depending on the fuel. Diesel fuel had the fastest development speed, while DME fuel showed the slowest

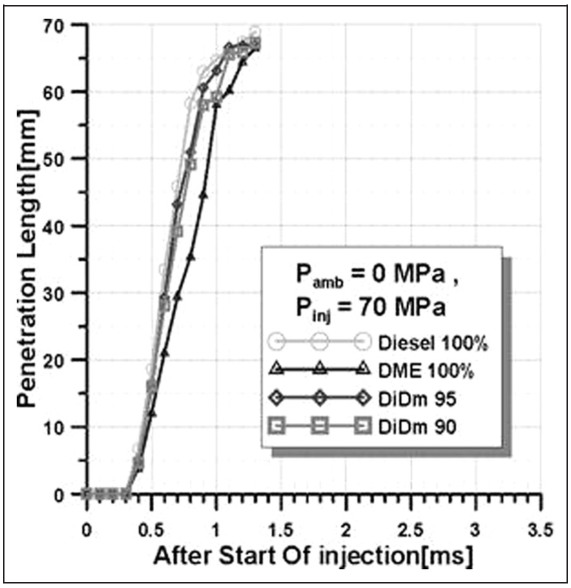

Figure 8. Spray penetration length of fuel at P_{amb} = 0 MPa.
DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

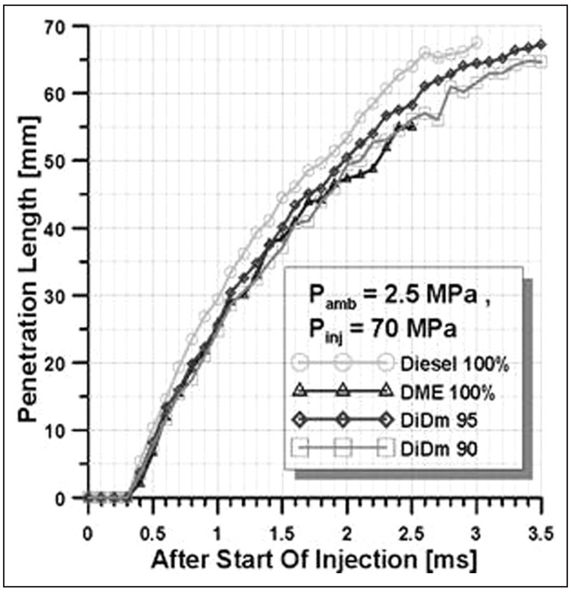

Figure 9. Spray penetration length of fuel at $P_{amb} = 2.5$ MPa. DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

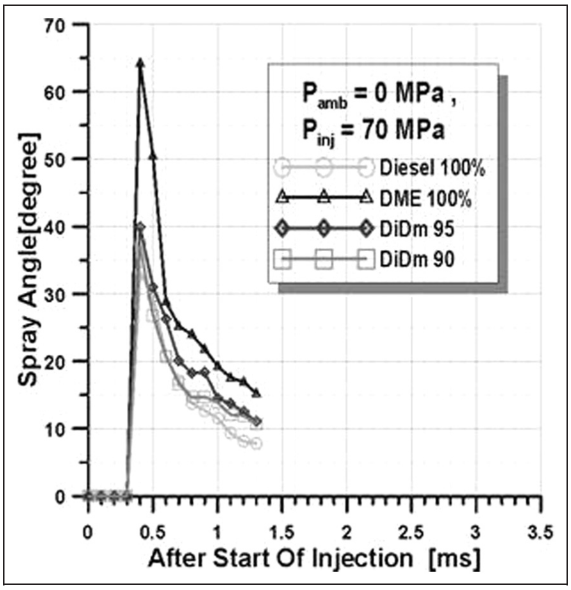

Figure 11. Spray angle of fuel at $P_{amb} = 0$ MPa. DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

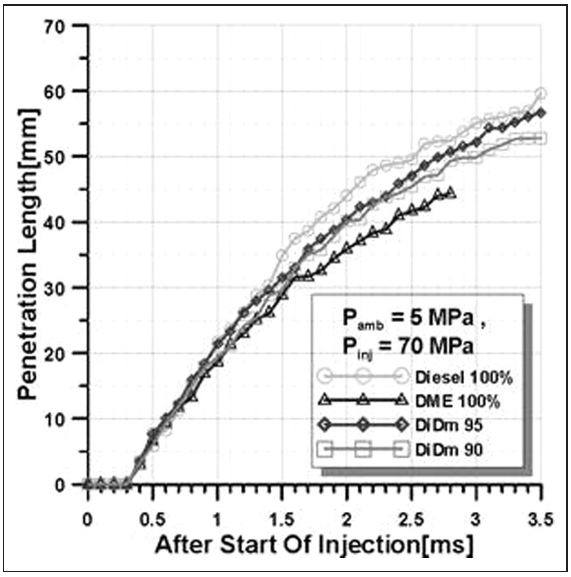

Figure 10. Spray penetration length of fuel at $P_{amb} = 5$ MPa. DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

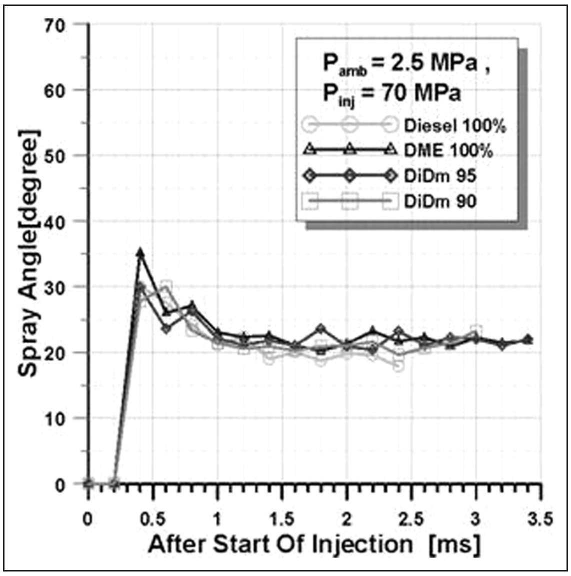

Figure 12. Spray angle of fuel at $P_{amb} = 2.5$ MPa. DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

development speed. In addition, the spray growth speed decreased depending on the DME mixing rate.

In Figures 9 and 10, the spray penetration lengths at pressures of 2.5 MPa and 5 MPa respectively are provided. Diesel fuel showed the fastest droplet development. The droplet development for DME fuel was the slowest, but the spray was fully developed within 2.5 ms and 2.8 ms respectively because of the rapid spray evaporation characteristics. As before, the spray penetration length became shorter with increasing DME mixing rate.

Spray angle measurement results. Figures 11 to 13 show the spray angles for each fuel. The spray angles are affected by the injection pressure, the ambient pressure, the injector nozzle's geometry and the properties of the fuel, among other factors.

The spray angles were analysed according to the fuel when using the same injector under an equivalent injection pressure and various ambient pressures. We confirmed that the spray angle was larger when the ambient pressure was 2.5 MPa or 5 MPa. The spray angle increased owing to the pressurized ambient conditions, which suppress the progress of the spray in the axial direction.

As shown in Figure 11, the spray angle was unstable during the initial spray development at 0.8–1.2 ms, and a stable spray angle was observed after 1.2 ms.

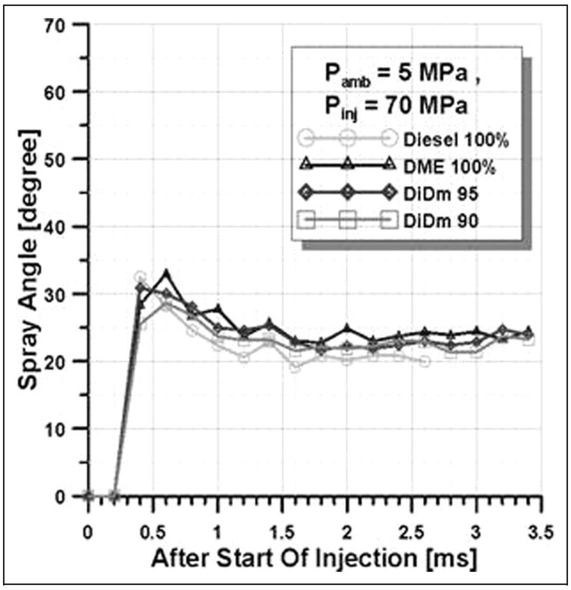

Figure 13. Spray angle of fuel at $P_{amb} = 5$ MPa.
DME: dimethyl ether; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100.

The spray angle for DME fuel was the largest, followed by those for DiDm 90, DiDm 95 and diesel fuel in decreasing order, as shown in Figure 11. The reason that DME produces a larger spray angle is because of the rapid progression of droplet atomization due to the good evaporation characteristics of DME.

Combustion characteristics results. Figure 14 shows the heat release rate and the cylinder pressure trace of the DME fuel, the diesel fuel and the fuel blends at an injection timing of 340° CA. The peak heat release rate with the DME fuel was higher by 10 bar than those of the diesel fuel and the fuel blends. This is due to the rapid vaporization, the high cetane number and the good mixing of DME. The peak cylinder pressures for the fuel blends were similar to that for diesel fuel. The slope of the pressure curve for diesel fuel was the steepest. On the other hand, the slope of the pressure curve for the fuel blends became smoother with increasing DME fuel addition.

In the graph of the heat release rate, the ignition delay of the DME fuel is shorter than that of the diesel fuel owing to fast vaporization. When the fuel blend is injected into the cylinder, an ignition delay is expected because the fuel blend shortens vaporization with increasing DME ratio. However, the predicted results and the actual results are different, and the ignition delay of the fuel blend was increased more than that of the diesel fuel. The reason for this result is that the diesel fuel and air in the cylinder were blocked by the earlier vaporization of the DME fuel. Moreover, the ascending order of the slope of the heat release rate was diesel fuel, DiDm 95, DiDm 90 and DME fuel, while the heat release rate curve gradient for the diesel fuel was the steepest.

Figure 15 shows the ignition delay, the peak heat release rate, the indicated mean effective pressures (IMEPs) and the thermal efficiencies for the DME fuel, the diesel fuel and the fuel blends at an injection timing of 340° CA. Figure 15(a) and (b) show the ignition delay and the peak heat release rate respectively. As mentioned earlier, the ignition delay for DME fuel is the shortest, and the ignition delay increases with increasing DME fuel mass fraction. On the other hand, the peak heat release rate decreases with increasing

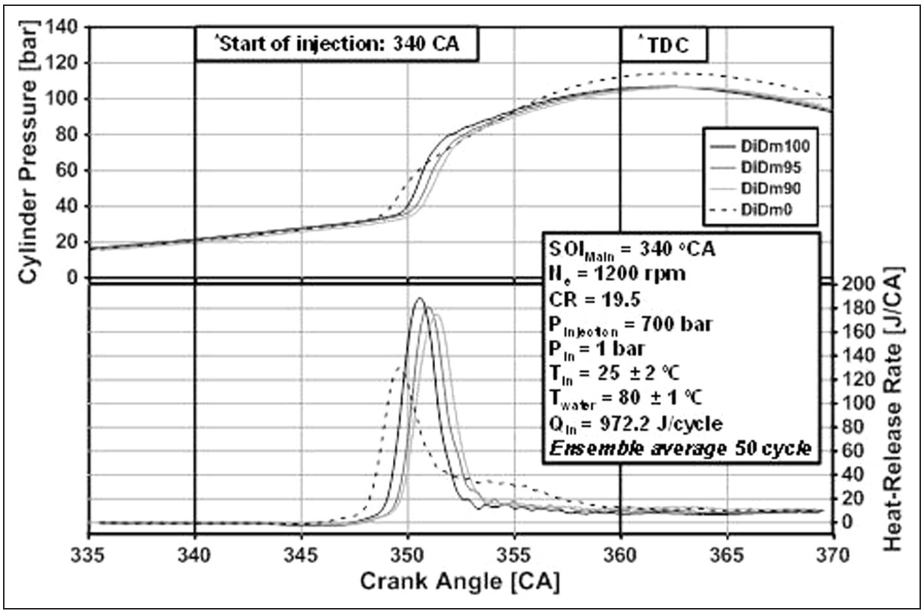

Figure 14. Cylinder pressures and heat release rates for the fuel blends with the SOI_{Main} at 340° CA.
CA: crank angle; TDC: top dead centre; SOI: start of injection; rpm: r/min; CR: compression ratio.

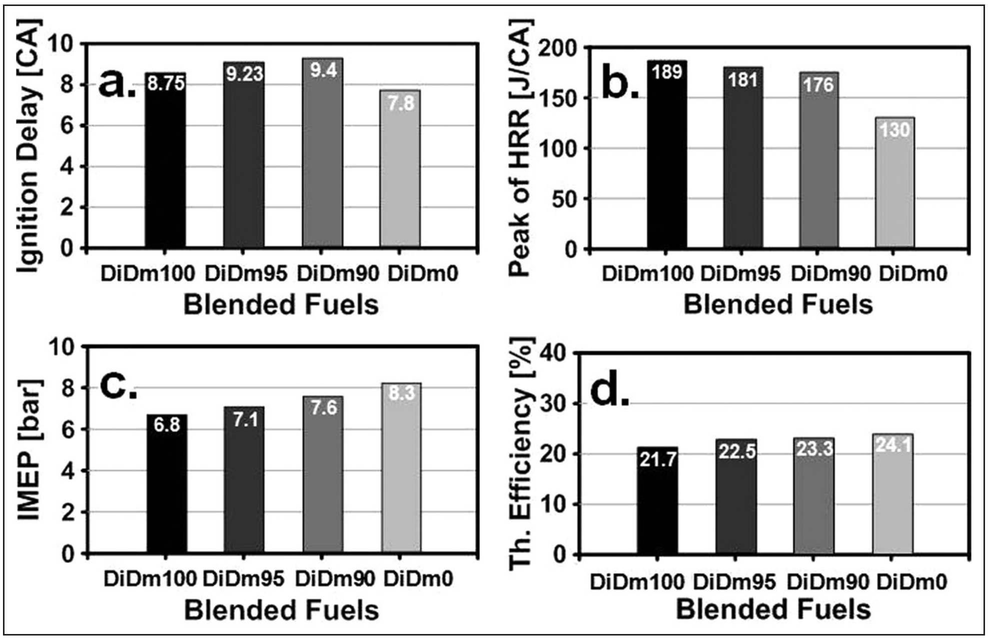

Figure 15. Combustion characteristics including (a) the ignition delay, (b) the peak of the heat release rate, (c) the IMEP and (d) the thermal efficiency for the fuel blends at $SOI_{Main} = 340°$ CA.
CA: crank angle; DiDm: {diesel (kg)/[diesel + DME) (kg)]} $\times$ 100; HRR: heat release rate; IMEP: indicated mean effective pressure; Th.: thermal.

DME fuel mass fraction. Figure 15(c) and (d) show the IMEP and the thermal efficiency results respectively. As briefly discussed above, and as can be observed in Figure 15(c), the IMEPs for the fuel blends were higher than that of the diesel fuel. The IMEPs for DiDm 95 and DiDm 90 are 0.3 bar and 0.8 bar respectively higher than that of the diesel fuel. As the DME ratio in the fuel blend increased, a longer ignition delay was gradually indicated. Since a negative performance in the compression stroke was converted to a positive performance, the IMEP increased. Although the ignition delay for the DME fuel was the shortest, the IMEP was the highest by 8.3 bar because the peak cylinder pressure for the DME fuel influences the IMEP. The thermal efficiency increased in the same order as for the IMEP.

Emission characteristics. Figure 16 shows the CO emissions, the total hydrocarbon (THC) emissions, the NO_x emissions and the smoke emissions for various injection timings. Figure 16(a) and (b) shows the CO results and the THC results respectively. At almost all the injection timings, the CO emissions and the THC emissions for the DME fuel were lower than those for the diesel fuel owing to the low HC concentration in the DME fuel. Furthermore, wall wetting was reduced by the short spray penetration length and fast vaporization. The CO emissions and the THC emissions for the fuel blends were lower than those for the diesel fuel. The CO emissions for DiDm 90 were more similar to those for the DME fuel than to those for DiDm 95.

Figure 16(c) shows the NO_x emissions and, for all the injection timings, the NO_x emissions for the DME fuel were higher than those for the diesel fuel. This is because the combustion temperature is increased by the higher IMEP and power than for the diesel fuel. The NO_x emissions for the fuel blends increased with increasing DME fuel mass fraction.

Figure 16(d) shows the smoke production, which was higher for the DME fuel at all the injection timings because the DME fuel has a high oxygen content, which leads to high oxidation rates of particulates. Therefore, with increasing DME fuel addition, smoke production decreased. The smoke emissions for the fuel blends DiDm 95 and DiDm 90 were reduced by 20% compared with those for the diesel fuel.

Conclusions

In this study, we used diesel fuel, DME fueland fuel blends containing DME fractions of 5 mass % and 10 mass %. The spray characteristics were investigated by varying the ambient pressure with a common-rail system. An experimental investigation was conducted to examine the effects of the fuel blends on the combustion and the emissions characteristics in a single-cylinder direct-injection diesel engine. The results are summarized as follows.

1. The quantity of diesel fuel injected was the highest during the 1 ms injection time, followed by the quantity of DME fuel and then the quantity of higher-DME-mixing-ratio fuel blend.
2. The spray penetration length of the fuel was shorter because of its interference with the development of the spray axis as the ambient pressure increased. The spray penetration length decreased

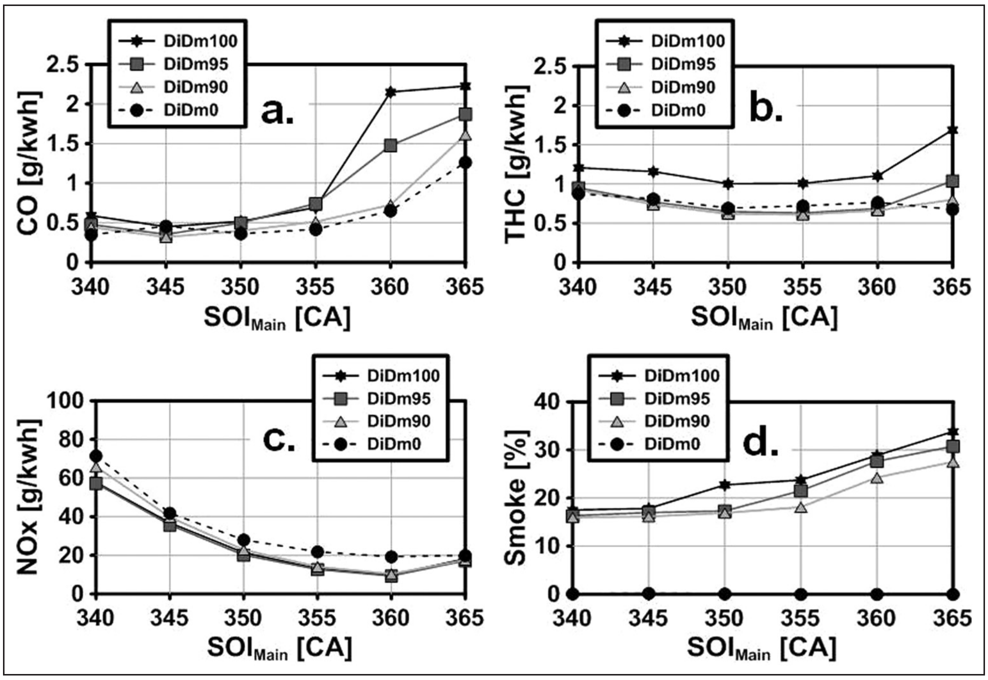

Figure 16. Emission characteristics of (a) CO, (b) THC, (c) NO_x and (d) smoke at various SOI_{Main} values.
CO: carbon monoxide; DiDm: {diesel (kg)/[diesel + DME) (kg)]} × 100; SOI: start of injection; CA: crank angle; THC: total hydrocarbon; NOx: nitrogen oxides.

with increasing DME mixing ratio in the case of the fuel blend.

3. The spray angle for the DME fuel was the largest, followed by those for DiDm 90, DiDm 95 and the diesel fuel in decreasing order. The spray angle increased owing to the increasing DME mass fraction ratio because it is dependent on the good evaporation characteristics of DME.
4. It was found that the ignition delay and the peak heat release rate decrease with increasing DME mass fraction. In addition, the IMEP and the thermal efficiency increased with increasing DME fuel addition. Since a negative performance in the compression stroke is converted to a positive performance, the IMEP also increased.
5. In comparison with the results for the diesel fuel, the THC emissions and the CO emissions decrease with increasing DME mass fraction because of the low HC concentration in the DME fuel and the reduced wall wetting which is due to the short spray penetration length and fast vaporization. With increasing DME addition, the smoke emissions for the DiDm 95 and DiDm fuel blends were reduced by 20% compared with those for the diesel fuel. However, the NO_x emissions increased for the 10 mass % DME fraction; this is because the combustion temperature increased owing to the higher IMEP and power than those for the diesel fuel.

Declaration of conflicting interest

The authors declare that there is no conflict of interest.

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Appendix 1

Notation

P_{amb}	ambient pressure (MPa)
P_{inj}	injection pressure (MPa)

Abbreviations

BMEP	brake mean effective pressure
CA	crank angle
CO	carbon monoxide
CO_2	carbon dioxide
DME	dimethyl ether
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
IMEP	indicated mean effective pressure
LPG	liquefied petroleum gas
NO_x	nitrogen oxides
PM	particulate matter
THC	total hydrocarbon