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Experimental study on improvement of ignitability of methanol fuel

Fuels - Alternative & New Fuels

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ABSTRACT

At present, as countries around the world accelerate their efforts towards decarbonization with the aim of becoming carbon neutral by 2050, Japan's shipping sector is aiming for the commercial operation of 'zero-emission ships' that do not emit greenhouse gases by 2028.

While green methanol is attracting attention as a possible alternative fuel, technical issues such as low ignitability have been pointed out, and it is necessary to aim to accumulate basic research knowledge for early commercialization.

In this study, to compensate for the low ignition quality of methanol, dimethyl ether (DME), which can be produced by the dehydration reaction of methanol, was mixed in several proportions and its ignition and combustion characteristics were investigated in a constant volume vessel. The results showed that the ignition delay of the methanol-DME fuel spray was strongly influenced by the more reactive DME, and that the spray flame was expected to have low soot combustion, as the brightness of the flame was significantly lower than that of the diesel oil used in the comparison.

Furthermore, two diesel combustion systems suitable for methanol were investigated on the same engine.

One of these is a direct in-cylinder injection (DI/MIX) system in which methanol is mixed with biodiesel. The other is a system in which methanol is injected into the intake port for premixed intake and biodiesel is injected directly into the cylinder as ignition oil (DI/PI).
Experimental results show that DI/MIX system has a steeper heat release ratio with higher constant volume degree of combustion than DI/PI system, which indicates higher thermal efficiency and lower CO and THC emission.

1 INTRODUCTION

Efforts are currently underway around the world to reduce greenhouse gas (GHG) emissions from various industries. Ships are no exception, and the GHG reduction strategy adopted at IMO_MEPC80 strengthened the targets for the 2030 and 2040 phases compared to the proposal adopted in 2018.One way to reduce GHG emissions is to use methanol, ammonia or hydrogen as fuel instead of fuel oil.

According to our own projections of the future diffusion rate of alternative fuels, methanol is expected to spread in the market ahead of ammonia, and we believe that the rate of ammonia will eventually exceed that of methanol, but not until after 2040.Furthermore, methanol can be made carbon neutral by using green methanol, which is obtained by using CO2 and hydrogen from renewable energy sources.

This study focuses on methanol fuel, which is liquid at ambient temperature and pressure and relatively easy to handle despite its toxicity, is in rapidly growing demand on ships, especially large container ships. To compensate for its low ignition properties, dimethyl ether (DME), which can be produced by the dehydration reaction of methanol, was mixed in several mixing fractions. The ignition and combustion characteristics were investigated in a constant volume vessel.

Furthermore, methanol and biodiesel (FAME) were compared on the same engine between a direct injection of the fuel mixture into the cylinder (DI/MIX) and a premixed intake with methanol injected into the intake pipe and FAME injected directly into the cylinder as ignition oil (DI/PI).

Methanol-DME and methanol-FAME monofuel blends have the potential to utilise methanol without the need for complex fuel systems such as dedicated injectors, and this study will highlight their usefulness.

2 DIMETHYL ETHER(DME) AND METHANOL BLENDS

2.1 Experimental apparatus and methods

2.1.1 A constant volume vessel

A premixed combustion type constant volume vessel was used in the experiments. An overview of the constant volume vessel and the surrounding system is shown in Figure 1. The combustion chamber of the constant volume vessel has a pancake shape with a diameter of 85 mm and a thickness of 44 mm. After filling the combustion chamber with a premixed gas comprising ethylene, oxygen and nitrogen, high-temperature, high-pressure environment equivalent to a top dead centre condition of a diesel engine was made with flame propagation through spark discharge from a spark plug. The flow rate of the premixed gas was controlled by a mass flow controller.

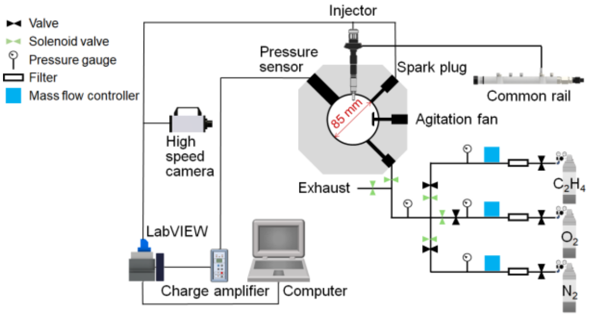


Figure 1. Schematic of experimental system consisting of constant volume vessel

The pressure in the vessel was measured by a piezoelectric pressure sensor (KISTLER, 6054C). The output of the pressure sensor was fed into a charge amplifier (KISTLER, Type 5011), which amplified the output and converted it into a voltage, which was then fed into a CompactRIO (National Instrument).

An example of the pressure history in a constant volume vessel is shown in Figure 2. After spark ignition, the pressure rises due to the heat generated by the flame propagation, reaches a peak and then decreases due to heat transfer to the vessel wall. When the desired pressure was reached, an injection signal was applied to the injector and fuel was injected. A K-type thermocouple with a wire diameter of 100 µm was used to measure the ambient temperature, which was placed in the centre of the combustion chamber and exposed to the ambient gas.

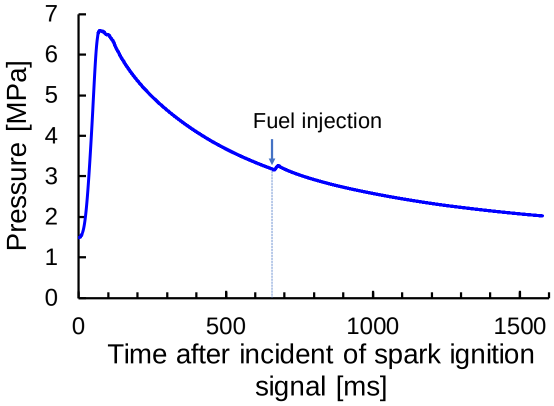


Figure 2. A profile of pressure in a combustion chamber

2.1.2 Fuel specifications and the device for mixing fuels

The cetane number, molecular oxygen content, lower heating value and theoretical air-fuel ratio (AFR) of methanol and dimethyl ether (DME) are shown in Table 1. Table 1 also includes values for JIS No. 2 diesel oil used for comparison. Methanol has a low cetane number of 3, while the cetane number of DME is as high as that of JIS 2 diesel oil. Methanol has a high oxygen content of 50 mass%, which means that its lower heating value and theoretical AFR are lower than those of JIS 2 diesel oil. The same applies to DME with an oxygen content of 34.8 mass%.

The pressure-temperature diagrams for methanol, DME and a fuel containing a 0.60 mole fraction of DME mixed with methanol (X_{DME} = 0.6) are shown in Figure 3. This figure is the result of a calculation by NIST's REFPROP[1], which can estimate the thermodynamic properties of DME. DME is a gas at room temperature and atmospheric pressure, but can be liquefied at room temperature by pressurising it below 1 MPa. Furthermore, as shown by Senda et al. [2] for hydrocarbon mixtures, a gas-liquid two-phase region bounded by a bubble point curve and a dew point curve is formed on the P-T diagram when DME is mixed with methanol. This reduces the saturated vapour pressure of DME and enables the liquid phase to be maintained at lower pressure.

Table 1. Properties of the fuel tested

Fuel	MeOH	DME	JIS No.2 Diesel fuel
Cetane number	3	55 – 60	57*
Oxygen content [mass%]	50.0	34.8	0
Lower heating value [MJ/kg]	19.9	28.8	42.9
Theoretical air/fuel ratio [-]	6.4	9.0	14.6

*cetane index

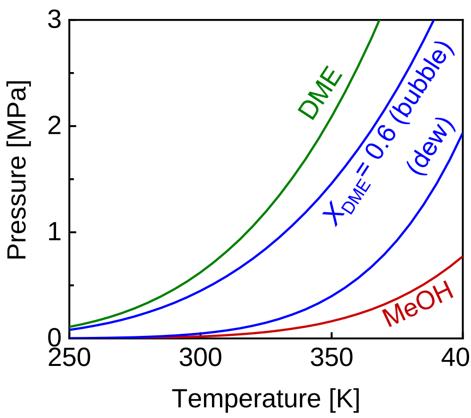


Figure 3. Pressure-temperature diagram of methanol, DME and methanol-DME mixing fuel (molar fraction of DME X_{DME} = 0.6)

Considering these saturated vapour pressure characteristics, the method used in this study is to pressurise DME and mix it with methanol in a liquefied state. The system for mixing fuels and for fuel injection is shown in Figure 4. The mixing section enclosed by the blue line in the diagram has a mixing tank with a 200 cm^3 internal volume, into which an arbitrary amount of methanol was injected in advance according to the mixing ratio and sealed. The mixing tank was then pressurised to above the saturated vapour pressure of DME by nitrogen gas. After setting up these conditions, DME was led from the cylinder to the mixing tank while the pressure in the pipes and mixing tank was gradually reduced. An electromagnetic stirrer was installed inside the mixing tank to stir the methanol/DME mixture and then guide it to the injection section. In the injection section, the fuel pressure was boosted by a fuel pump and stored in the common rail. The fuel injection pressure was defined as the value indicated by a pressure gauge installed in the common rail. A piezoelectric injector (DENSO, G3P) was used for fuel injection.

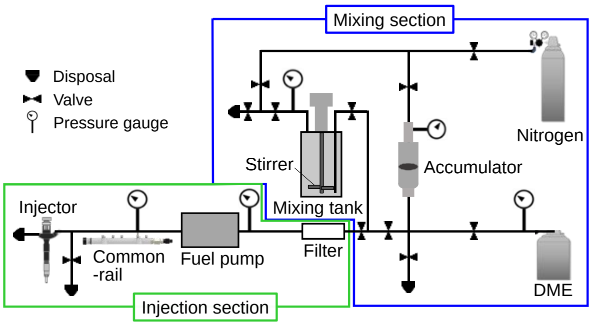


Figure 4. Schematic of the device for mixing methanol and DME and for fuel injection

2.1.3 Optical system

To visualize the spray and its flame, the Schlieren imaging method was used in this study. The optical system is shown in Figure 5. A CW laser with a wavelength of 532 nm was used as the light source. The laser beam was magnified to about the diameter of the combustion chamber by a biconvex lens and collimated by a plano‐convex lens. The collimated light passed through the constant volume vessel and was then focused by passing through the single convex lens again. The concentrated light was then directed to a high-speed camera (Photron, SA-Z) after adjusting the light intensity with an ND filter. A pinhole was placed in front of the camera to remove the light flux components refracted by the atomising flame, thus acquiring the Schlieren image. The acquisition speed was 80 kfps.

In Schlieren photography, the density gradient in the light path is photographed as a black shadow, and the area reflecting the light is also photographed as a black shadow. Thus, in a spray, the density differences between the liquid phase, which was continuous from the jet, and between the spray and the ambient gas were captured as black shadows, and in a flame, the density differences between the hot gas in the flame and the ambient gas were captured as black shadows.

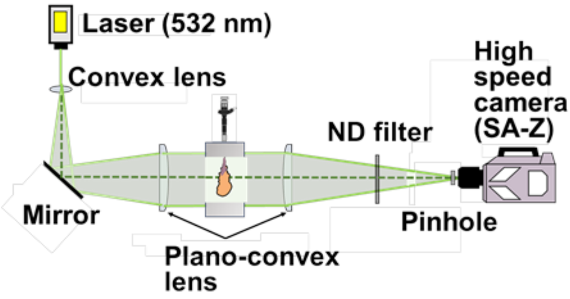


Figure 5. Optical setup for schlieren photography

2.1.4 Experimental conditions

The experimental conditions are shown in Table 2. The diameter of the fuel injection nozzle was 0.125 mm. The injection pressure was 40 MPa. The ambient temperature was defined by the measured value of a K-type thermocouple with a wire diameter of 100μm. The ambient temperature was varied from 1040 K to 790 K. The global atmosphere density averaged over the entire combustion chamber was 14 kg/m^3.

Table 2. Experimental conditions for the constant volume vessel test

Nozzle hole diameter [mm] (number of holes)	0.125 (single)
Injection pressure [MPa]	40
Ambient temperature [K]	1040, 950, 870, 840, 790
Global ambient density [kg/m^3]	14.0

2.1.5 Experimental results and discussion

The Schlieren images of spray combustion for each fuel at an ambient temperature (T_a) of 950 K are shown in Figure 6.

The diagram shows the boundaries of the ignition time determined by the definitions described below as red dashed lines. The position of the first confirmed ignition in the image is marked with a yellow arrow. For the time increments shown in the figure, the ignition timing of DME is slightly delayed compared to JIS 2 diesel fuel, but radial expansion due to ignition occurs upstream for both fuel sprays. X_{DME} = 0.6 has a slight delay in ignition and the white dashed line pointing to the unburned fuel spray extends further downstream; X_{DME} = 0.4, with an even lower DME fraction, has a more prolonged ignition delay and the white dashed line also extends further downstream of the spray, which is even more pronounced for methanol alone. However, for methanol the unburnt fuel vapour extends up to about 65 mm from the nozzle, while ignition occurs at about 35 mm from the nozzle and then rapidly extends the combustion zone downstream of the spray. The reason why ignition does not always occur at the spray tip has not yet been clarified, but fuels with a high oxygen content, such as methanol and DME, may have a low equivalent ratio downstream of the spray, making them unsuitable for ignition.

In Figure 6, only the image of JIS No. 2 diesel oil shows luminous flame from soot; JIS 2 diesel oil has the shortest ignition delay and combustion proceeds in an rich mixture after ignition, which is thought to have resulted in combustion accompanied by soot formation; DME also has an ignition delay close to that of JIS 2 diesel oil and ignition occurs before spray develops, but a luminous flame from soot is not observed. The reason why no bright flame from soot is observed is that DME contains 34.8 mass% oxygen in its molecules. Fuels with a high oxygen content do not produce soot because of the low air-fuel ratio and the lower equivalent ratio inside the spray. Therefore, no luminous flames are observed in methanol-DME blends and methanol with a higher oxygen content than DME and a longer ignition delay.

The rate of heat release (ROHR) for each fuel at an ambient temperature (T_a) of 950 K are shown in Figure 7. The ROHR of a methanol-DME mixture with a DME mole fraction of 0.2 (X_{DME} = 0.2), which is not shown in Figure. 6, has been added to Figure 7. When DME is blended with methanol, the ignition delay is shortened by increasing the blending ratio and the ROHR of premixed combustion, which occurs in the early stages of combustion, is reduced. The reduction in the ROHR of premixed combustion is due to a decrease in the amount of unburnt fuel present from the ignition position to the spray tip, as shown in Figure. 6. As a result, at X_{DME} = 0.6, where the ignition delay is reduced to about DME, the rate of heat release from 1.0 ms after the start of injection shows a typical ROHR profile of the diffusive combustion.

For a wide range of ambient temperatures (T_a), including 950 K, the ignition delay period of each fuel was systematically investigated. Under low-temperature ambient conditions, a low ROHR due to low-temperature oxidation reactions may be measured before the onset of high-temperature oxidation reactions. Therefore, in the present study,

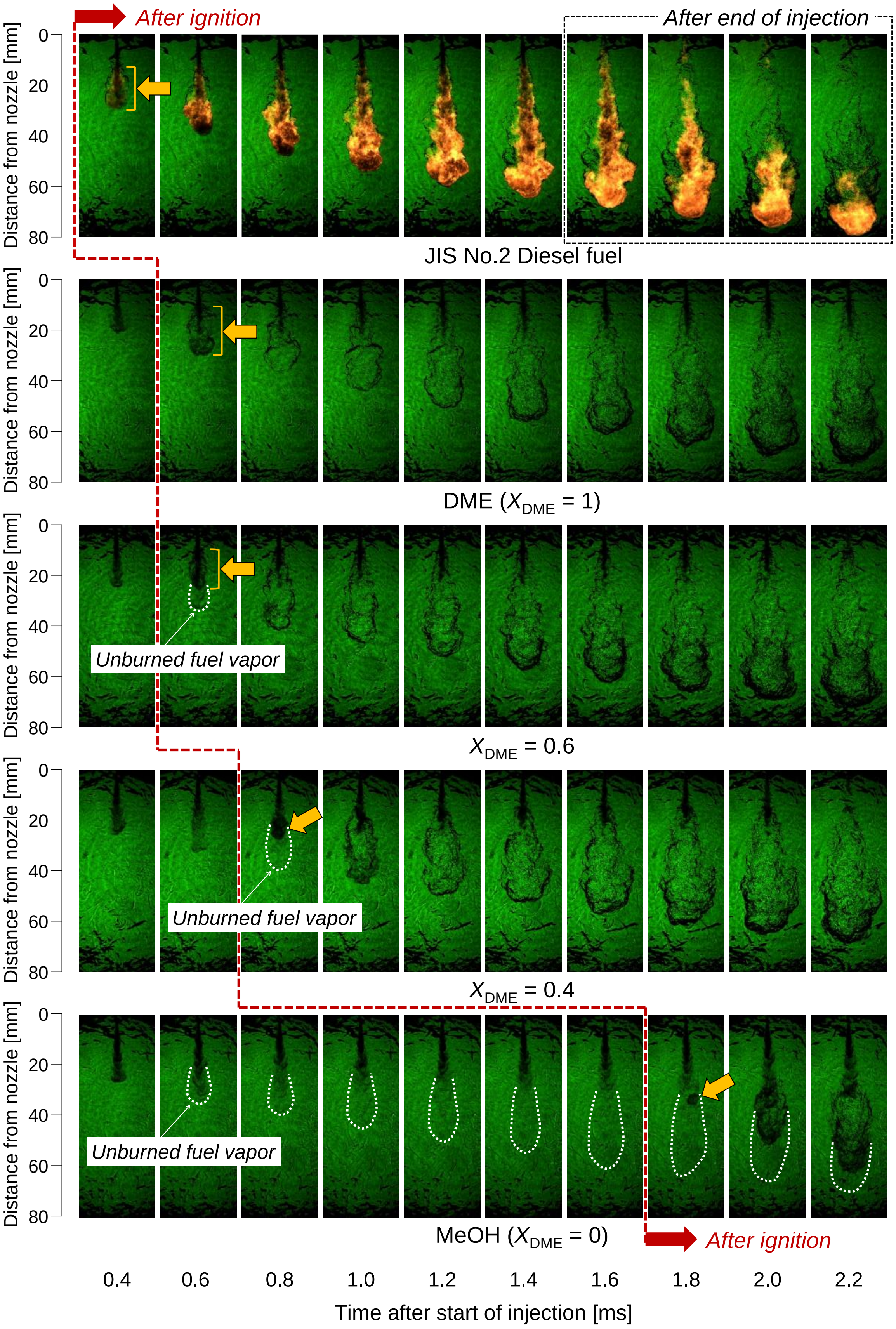


Figure 6. Schlieren images of spray combustion processes with the fuels tested (T_a = 950 K)

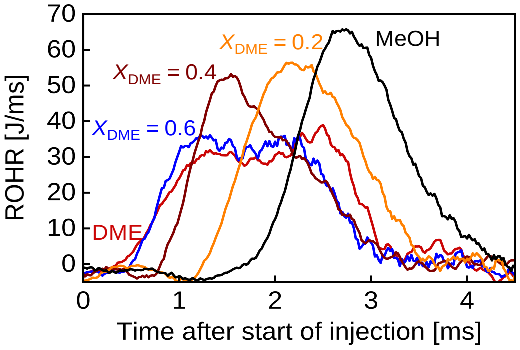


Figure 7. Change in the profiles of the rate of heat release (ROHR) with the mole fraction of DME XMDE (Ta = 950 K).

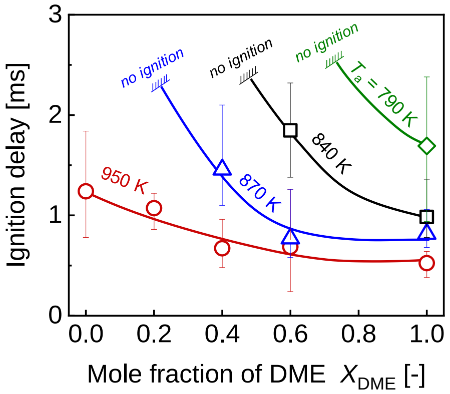


Figure 8. Effect of the mole fraction of DME X_{DME} on ignition delays at various ambient temperatures

ignition was defined with a threshold value based on the derivative of the ROHR. For the present experimental results, ignition was defined as the timing at which the derivative of the ROHR exceeded 28 J/ms^2, which made it possible to treat the onset of the high-temperature oxidation reaction as ignition, not including the low-temperature oxidation reaction period. The ignition delay period was defined as the time between the start of fuel injection and ignition.

Figure 8 shows the variation of the ignition delay period with respect to the mole fraction of DME, varying the ambient temperature from 950 K to 790K. At T_a = 950 K, methanol with a low cetane number also ignites. At this ambient temperature, there is no linear relationship between the ignition delay with respect to the mole fraction, and the ignition delay period is relatively close to that of DME when the DME mixture fraction is above 0.4. In contrast, at T_a = 870 K, methanol and X_{DME} = 0.2 did not lead to ignition. On the other hand, the DME mixing ratio of 0.4 or higher leads to ignition, and the ignition delay period for X_{DME} = 0.6 is comparable to that of DME. However, the difference between the ignition delay periods of X_{DME} = 0.6 and DME increases with decreasing ambient temperature, and at T_a = 790 K even X_{DME} = 0.6 does not lead to ignition. It should be noted, however, that this ambient temperature was measured using a K-type thermocouple with a wire diameter of 100 μm. It should also be noted that the relationship between ambient temperature and ignition delay varies with ambient density.

As shown above, at the measured ambient temperature of 870 K, the ignition delay period of the methanol-DME blends with X_{DME} = 0.6 was comparable to that of DME. The combustion characteristics of the methanol-DME blends with X_{DME} = 0.6 are then compared with those of diesel oil. In particular, the methanol-DME blends are fuels with a high oxygen content. As oxygen-containing fuels are known to improve the afterburn of diesel spray combustion [3], attention was focused on X_{DME} = 0.6 and the ROHR of diesel fuel.

Figure 9 compares the ROHR of JIS No. 2 diesel oil and X_{DME} = 0.6 at an ambient temperature (T_a) 1040 K, which is even higher than the ambient temperature (T_a) 950 K shown previously, where the difference in ignition delay period between fuels is small and the spray diffusion combustion characteristics are comparable. The peak value of the heat generation rate is lower for X_{DME} = 0.6 than for JIS No.2 diesel fuel due to its lower heating value. In addition, the injection period is prolonged in order to supply the same amount of heat. On the other hand, JIS No.2 diesel fuel has a shorter ignition delay and its combustion period is longer than X_{DME} = 0.6, even though the heat generation rate starts up earlier. The fuel injection rate per unit time during the quasi-steady injection period for JIS No.2 diesel fuel and X_{DME} = 0.6 is 2.18 mg/ms and 2.01 mg/ms respectively. Taking into account the lower heating value, the heat delivery rates per unit time for JIS No.2 diesel fuel and X_{DME} = 0.6 are 93.3 J/ms and 54.5 J/ms, respectively. In order to ignore this difference in the heat supply rate per unit time, the ROHR in Figure. 9 were made dimensionless by the heat supply rates of the respective fuels. The results are shown in Figure 10: X_{DME} = 0.6 has a higher ROHR relative to the heat supply rate than JIS No. 2 diesel fuel. JIS No.2 diesel fuel has been found to exhibit a typical ROHR profile of diffusive combustion with luminous flame emission, while the ROHR continues to increase until the end of injection. On the other hand, the ROHR for X_{DME} = 0.6 remains approximately constant from 1.0 ms after the start of fuel injection. Furthermore, the afterburn after the end of fuel injection is significantly shorter for X_{DME} = 0.6. This indicates that X_{DME} = 0.6 increases the heat generation rate during the diffusive combustion period and shortens the afterburn due to the presence of oxygen in the fuel molecules, which leads to dilution of the spray interior.

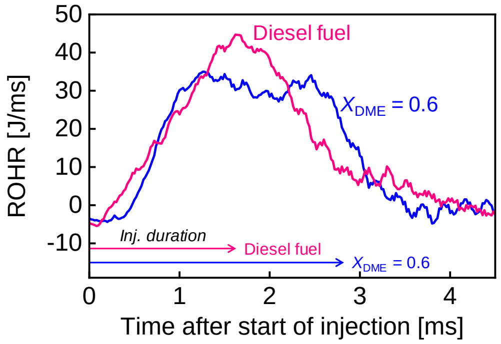


Figure 9. Comparison of the profiles of the rate of heat release between JIS No 2 diesel fuel and methanol-DME blend (X_{DME} = 0.6) (T_a = 1040 K)

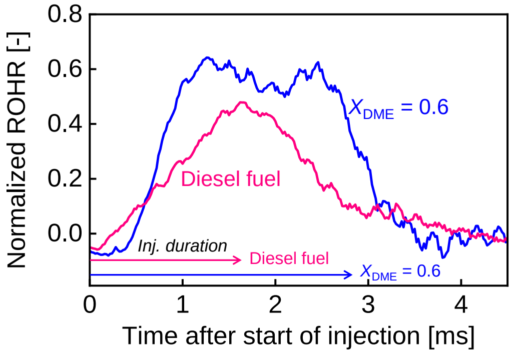


Figure 10. Comparison of the profiles of the rate of heat release normalised by the fuel energy supply rate between JIS No. 2 diesel fuel and methanol-DME blend (X_{DME} =0.6) (T_a = 1040 K)

3 BIODIESEL AND METHANOL BLENDS

3.1 Experimental apparatus and methods

3.1.1 Experimental device

In this study, a comparison was made between a fuel mixture of biodiesel (FAME) and methanol injected directly into the cylinder (DI/MIX) and a system in which methanol is injected into the intake pipe for premixed intake and FAME is injected directly into the cylinder as ignition oil (DI/PI). Figure 11 shows a diagram of the experimental setup used in this test. A water-cooled, four-stroke, single-cylinder, direct-injection diesel engine was used as the test engine. A common rail diesel fuel injection system was used for the direct in-cylinder injection of the FAME-methanol mixture. On the other hand, a multi-hole fuel injection valve for gasoline engines was used for methanol injection into the intake pipe.

In-cylinder pressure was detected using a piezoelectric pressure transducer (Kistler 6052), the signal was captured by a data recorder (Yokogawa Electric DL750) via a charge amplifier (Kistler 5011) and combustion characteristics such as heat generation rate were analyzed using the engine combustion pressure analysis package (Yokogawa Electric diesel Ver. 4.12)

The concentration of total unburnt hydrocarbons (THC) in the exhaust gas was measured by a heated FID analyzer (HORIBA MEXA-1170HFID), carbon monoxide (CO) by a non-dispersive infrared spectrometer (Best Instruments BCC-611AS) and nitrogen oxide (NOx) by a chemiluminescent analyzer (Yanaco ECL-88AO), respectively.

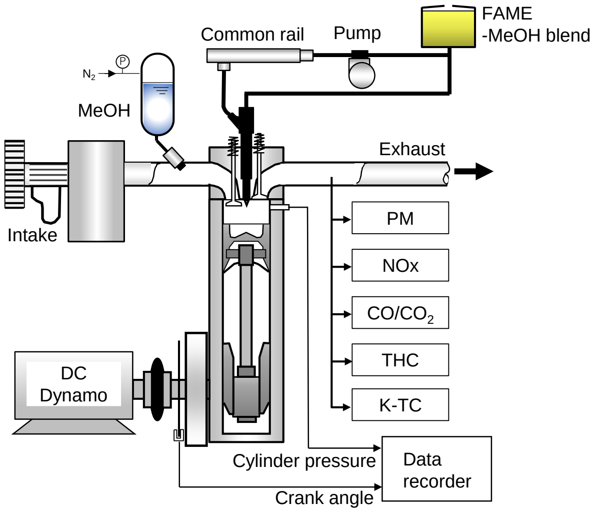


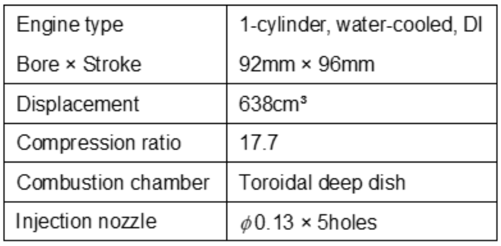
Figure 11. Layout of experimental apparatus

Exhaust particulate PM was measured using a micro dilution tunnel system (F-Techno MIT-2000), whereby exhaust air with a 10-fold dilution was aired over a Teflon-coated filter (Pallflex TX40HI20-WW, 70 mm diameter, 60 mm effective diameter) to collect the particulate matter and weigh it. The exhaust temperature at 350 mm downstream from the exhaust port was measured using a K-type thermocouple.

Table 3 and 4 shows the experimental conditions for DI/MIX and DI/PI: the in-cylinder injection pressure was kept low at 40 MPa for DI/MIX, taking into account the reduced lubricity of the fuel due to the methanol mixture, and the same injection pressure was used for DI/PI. The start of direct in-cylinder injection was adjusted so that the ignition timing was -6 ± 1 deg ATDC.

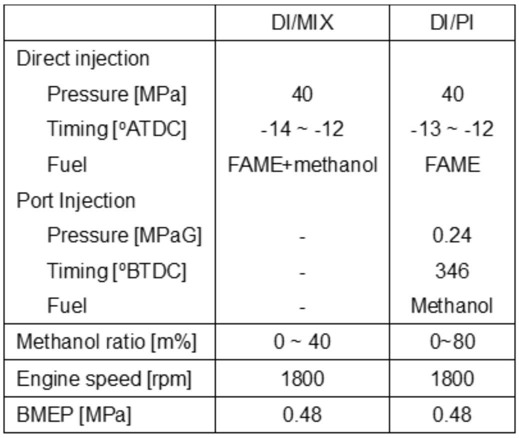
The mass fraction of methanol in the total fuel consumed was varied between 0 and 40 m% for DI/MIX and between 0 and 80 m% for DI/PI. The upper limit of the methanol percentage in DI/MIX was limited to 40 m% due to the limitations of the injection system. The combustion and exhaust characteristics of the two combustion systems were measured at a constant engine speed of 1800 rpm, coolant temperature of 80 °C and break mean effective pressure BMEP = 0.48 MPa.

Table 3. Engine specifications



Engine type	1-cylinder, water-cooled, DI
Bore × Stroke	92mm × 96mm
Displacement	638cm^3
Compression ratio	17.7
Combustion chamber	Toroidal deep dish
Injection nozzle	ϕ0.13 × 5holes

Table 4. Experimental condition



	DI/MIX	DI/PI
Direct injection		
Pressure [MPa]	40	40
Timing [^{o}ATDC]	-14 ~ -12	-13 ~ -12
Fuel	FAME+methanol	FAME
Port Injection		
Pressure [MPaG]	-	0.24
Timing [^{o}BTDC]	-	346
Fuel	-	Methanol
Methanol ratio [m%]	0 ~ 40	0~80
Engine speed [rpm]	1800	1800
BMEP [MPa]	0.48	0.48

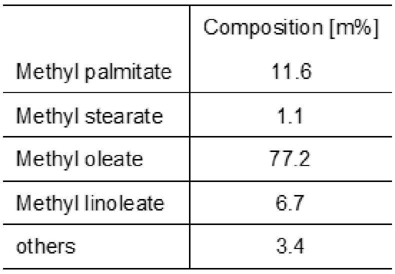
3.1.2 Properties of the fuel tested

In this study, a fuel mixture of methanol (Nacalai Tesque, CAS No. 21914-45, Purity>99.0%) and methyl oleate reagent (Kanto Chemical Co., Ltd., CAS No. 25304-02, Purity>65%) was used as FAME. Three fuel mixtures were prepared and used in the DI/MIX experiments, with methanol mass fractions MR of 0 m%, 20 m% and 40 m%, respectively. The fuel properties of each fuel and methanol are listed in Table 5. Table 6 shows representative values for the fatty acid composition of the test FAMEs.

Table 5. Fuel specifications

MR [m%]	0 (FAME)	20	40	100 (Methanol)
Density [g/cm^3]	0.88	0.86	0.84	0.79
LHV [MJ/kg]	36.6	33.3	29.9	19.9
Carbon [m%]	76.5	68.7	60.9	37.5
Hydrogen [m%]	12.1	12.2	12.3	12.5
Oxygen [m%]	11.4	19.1	26.8	50.0
Theorical Air [kg/kg]	12.4	11.2	10.0	6.4
Cetane number	43*	-	-	3

Table 6. Typical composition of used FAME

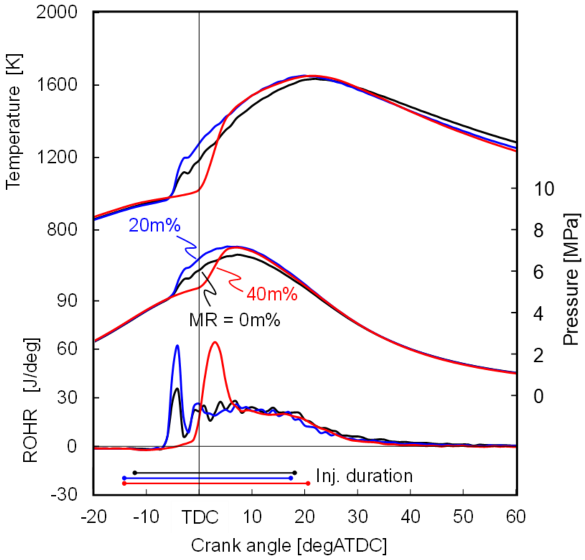


	Composition [m%]
Methyl palmitate	11.6
Methyl stearate	1.1
Methyl oleate	77.2
Methyl linoleate	6.7
others	3.4

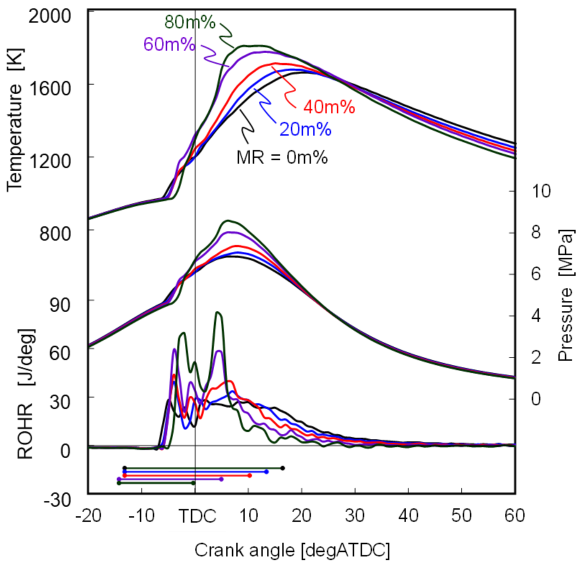
3.2 Experimental results and discussion

Figure 12 shows the crank angle evolution of the in-cylinder pressure, rate of heat release (ROHR) and average in-cylinder gas temperature when the engine is operated at various methyl fuel rates using (a) the DI/MIX system and (b) the DI/PI system. The straight line below the rate of heat release diagram represents the period of fuel injection into the cylinder, which was kept constant at BMEP = 0.48 MPa and the injection timing was set so that the start of heat release was -6 ± 1 deg ATDC.

From Figure 12(a), comparing the onset of heat generation of the fuel in the DI/MIX system, the maximum value of the initial heat release rate peak is higher at MR = 20m% compared to MR = 0m%, and the maximum value increases further at MR = 40m% with a delay of about 7^{o}CA in its onset. The reason for the change from MR=0m% to 20m% can be attributed to the high vaporability of methanol. The 20 m% methanol mixture increased the amount of premixed air formed during the ignition delay period, which may have led to premixed combustion after ignition, resulting in an increase in the initial heat release rate peak. Furthermore, when the methanol fraction was increased up to 40 m%, the onset of heat release was -6 deg ATDC,



(a) DI/MIX.



(b) DI/PI.

Figure 12. Crank-angle histories of ROHR, cylinder pressure and average temperature

but the low ignitability of methanol became apparent, and the onset of the main heat release was considered to be delayed. On the other hand, Figure 12(a) shows that the end of heat release for three fuels of DI/MIX remains almost unchanged when the methanol fraction is increased. From the above results, among the test fuels, MR = 40 m% started thermogenesis closest to the top dead center and had the shortest combustion period.

On the other hand, the results for DI/PI in Figure 12(b) show that the first peak of heat release rate is not delayed as in DI/MIX, even when MR is increased to 80 m%. On the other hand, increasing MR shortens the injection period of the FAME and the end of heat release is earlier, indicating that the higher the methanol percentage, the shorter the heat release period. A second peak appears around 5 deg ATDC in the heat release rates for MR = 60 m% and 80 m%. This is the result of compression auto-ignition of the dilute methanol premixture at an end-gas region and is assumed to be similar to the phenomenon observed in natural gas dual-fuel engines [5].

The relationship between the break thermal efficiency BTE, the cycle variability COV of the indicated mean effective pressure IMEP, the exhaust gas temperature EGT and the degree of constant volume of combustion DCV and methanol fraction is shown in Figure 13, comparing DI/MIX and DI/PI.

Figure 13 shows that the thermal efficiency increased with increasing MR in DI/MIX. This can be attributed to the fact that, as shown in Figure 12(a), the main heat release period shifts to just after the top dead center as a result of increasing the methanol ratio, which increases the DCV. Another reason for increased BTE of DI/MIX with increasing MR is slightly decreased exhaust gas temperature which leads to decrease the exhaust losses. In DI/PI, on the other hand, as mentioned earlier, the combustion end was accelerated as MR increased, resulting in an increase in DCV, a decrease in exhaust temperature but a decrease in BTE and a worsening of COV. The reason for the lower BTE in DI/PI is due to the lower combustion efficiency of the methanol premix, as described below.

Figure 14 shows the relationship between the concentrations of NOx, PM, CO and THC in the exhaust gas and the methanol fraction in comparison with DI/MIX and DI/PI. Firstly, with respect to NOx, in DI/MIX, NOx increases once when MR is increased from 0m% to 20m%, but the value is lower than 0m% when MR is increased to 40m%. This can be explained by the in-cylinder gas temperature history around the top dead center in Figure 12(a). Compared to MR = 0 m%, the increased initial heat release at MR = 20 m% results in a higher initial temperature rise rate and higher NOx emissions. However, when the methanol fraction is increased to MR = 40 m%, the onset of the main heat release is delayed until around the top dead center, resulting in a shorter high temperature holding time and thus lower NOx. On the other hand, NOx in DI/PI increased almost monotonically with respect to MR. This is thought to be because the start of the main heat release in DI/PI was not affected by MR, and the cylinder temperature increased as MR increased.

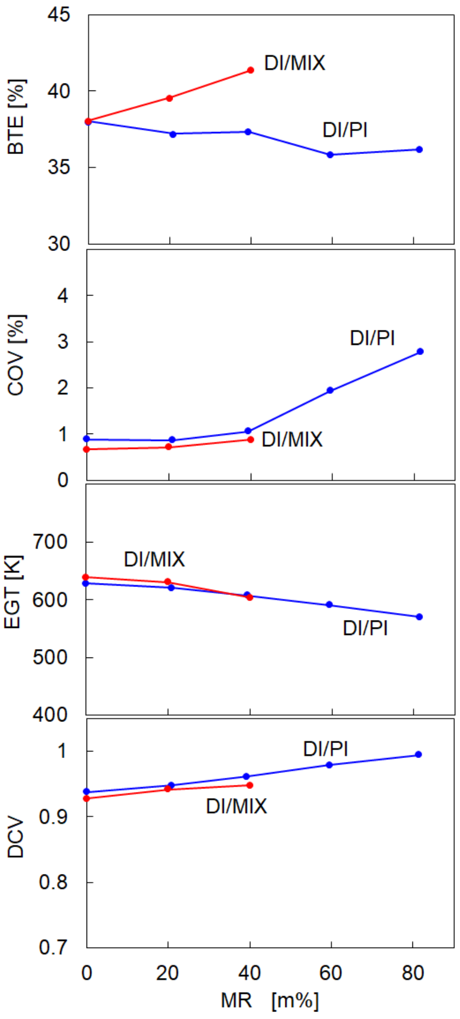


Figure 13. Change in brake thermal efficiency: BTE, coefficient of variance: COV for IMEP, exhaust gas temperature: EGT, and degree of constant volume: DCV with methanol mass fraction: MR

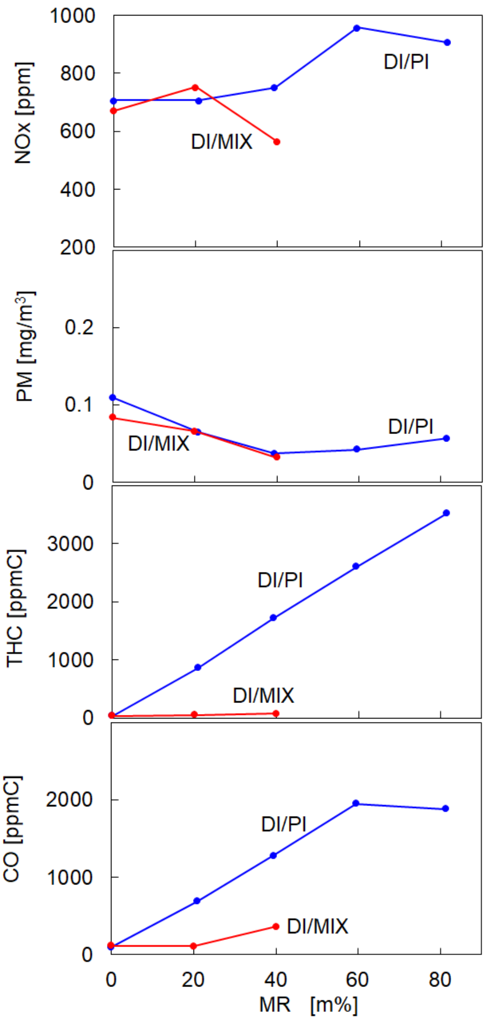


Figure 14. Change in NOx, PM, THC, and CO emissions with MR

The unburned components CO and THC showed a monotonic increase with respect to MR for both combustion methods, but the DI/PI method showed a particularly high rate of increase. Therefore, as shown in Fig. 15, the combustion efficiency of DI/MIX was close to 100% even at MR = 40 m%, whereas that of DI/PI decreased linearly with MR. In order to consider the reasons for such a decrease in the combustion efficiency of DI/PI, the relationship between the equivalence ratio and the methanol fraction is shown in Figure 16.

The solid line in Figure 16 is the total equivalence ratio of methanol and FAME combined, while the dashed line is the equivalence ratio of the methanol premixture in the DI/PI system. As shown in Figure 16, the equivalence ratio of the methanol premixture in DI/PI is low at about 0.3 even at MR = 80 m%. On the other hand, under high MR conditions, the spray flame of FAME for the ignition source is smaller, resulting in incomplete flame propagation in the methanol premixture, which is considered to have led to a significant increase in CO and THC. The emission of unburnt gases from the deflagration zone, such as the crevice above the piston top ring, is also considered to be a factor in the increase in THC and CO for DI/PI. On the other hand, the increase in CO and THC in DI/MIX is considered to be suppressed compared to DI/PI, as the combustion in DI/MIX is mainly diffusion combustion, which prevents combustion at low equivalent ratios.

As for particulate PM, it decreased monotonically with respect to MR for both combustion types, as PM is mainly produced from FAME.

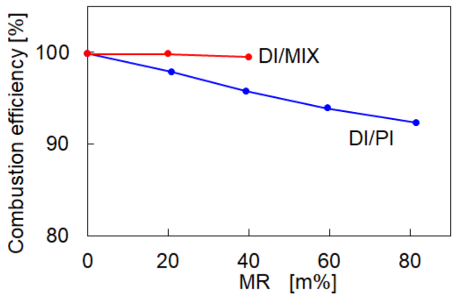


Figure 15. Change in combustion efficiency with MR

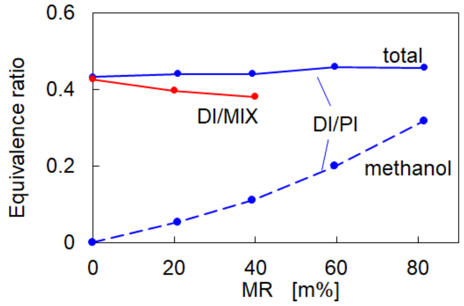


Figure 16. Change in equivalence ratio with MR

4 SUMMARY

4.1 Dimethyl ether (DME) and methanol blends

A liquid-liquid mixture of methanol and DME was investigated for use in diesel engines by improving the ignition properties of methanol in a constant volume vessel. The properties were compared with those of JIS No. 2 diesel oil. The findings are summarized as follows.

1. Mixing methanol with DME shortens the ignition delay. At an ambient temperature of 870 K, a short ignition delay period equivalent to DME is achieved by mixing DME with a mole fraction of 0.6, so that the flame has the appearance of a typical diesel spray combustion with diffusive combustion.

2. Methanol-DME blends have a lower lower heating value and a longer injection period than JIS 2 diesel oil. Nevertheless, the oxygenated nature of the methanol-DME blends leads to a dilution of the spray and a higher heat release rate per unit heat supplied compared to JIS 2 diesel oil. In addition, the combustion period is shortened due to a shorter afterburn.

3. Spray combustion of methanol-DME blends produces very little soot, as they hardly emit a bright flame.

4.2 Biodiesel and methanol blends

A comparison was made between a fuel mixture of biodiesel (FAME) and methanol injected directly into the cylinder (DI/MIX) and a system in which methanol is injected into the intake pipe for premixed intake and FAME is injected directly into the cylinder as ignition oil (DI/PI). The findings are summarised as follows.

1. In the DI/MIX system, increasing the methanol fraction delays the onset of premixed combustion, resulting in a steep heat release history.

2. In the DI/MIX system, increasing the methanol fraction improves the degree of constant volume and reduces exhaust losses, thus increasing the thermal efficiency.

3. In the DI/PI system, the onset of premixed combustion is not delayed by increasing the methanol fraction, but the combustion period is shortened by an earlier end of combustion.

4. In the DI/PI system, CO and THC emissions increase significantly compared to the DI/MIX system when the methanol fraction is increased. This worsens the unburnt losses and reduces the thermal efficiency.

5. These results show that DI/MIX can be expected to have higher thermal efficiency and lower emissions than DI/PI, although the fuel injector restrictions limit the methanol fraction that can be operated.

5 FUTURE PROSPECTS

By blending DME with methanol, the ignition delay was improved, and diffuse combustion was observed with very little soot compared to conventional diesel fuels. It was also found that blending FAME with methanol resulted in higher thermal efficiency and lower emissions compared to DI/PI, although DI/MIX is limited in the methanol fraction that can be operated due to fuel injector restrictions.

Now that the potential of the methanol-DME and methanol-FAME blends has been identified, they will be operated in a single-cylinder test machine to determine their performance.

6 DEFINITIONS, ACRONYMS AND ABBREVIATIONS

AFR	Air-Fuel Ratio
BMEP	Break Mean Effective Pressure
BTE	Break Thermal Efficiency
CO	Carbon monoxide
COV	Cycle variability of the indicated mean effective pressure
DCV	Degree of constant volume of combustion
DI/MIX	System that injects a mixture of FAME and methanol directly into the cylinder
DME	Dimethyl Ether
EGT	Exhaust gas temperature
FAME	Biodiesel
GHG	Greenhouse Gas
MR	The mass fraction of methanol in the total fuel consumed
NOx	Nitrogen oxide
PI/DI	System in which methanol is injected into the intake pipe for premixed intake and FAME is injected directly into the cylinder as ignition oil
PM	Exhaust particulate
ROHR	Rate of Heat Release
T_a	Ambient Temperature
THC	Total unburnt hydrocarbons
X_{DME}	A mole fraction of DME mixed with methanol

7 REFERENCES AND BIBLIOGRAPHY

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