Trade-off improvements by combining EGR and supercharging in a diesel engine with next generation bio-alcohol blended FAME fuels

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　　Isopentanol and isobutanol were selected as the next generation bio-alcohols in the study, and four kinds of bio-alcohol blended fuels were prepared by blending at a constant mass ratio of 3:7 into the two kinds of FAME fuels. The base fuels used here were methyl laurate (LME) and a blend with equal proportions of methyl palmitate (PME) and LME, termed PLME. The bio-alcohol blended FAME fuels were supplied to a small single cylinder direct injection diesel engine, and the effects on engine performance, combustion characteristics, and exhaust emissions were investigated. The boost pressures were set at two conditions: 100 kPa (without supercharging) and 120 kPa (with supercharging), and the EGR rates were varied from 0 to 28.5%. The results showed that regardless of EGR rate, and with and without supercharging, the trade-off relation between NOx and smoke emissions with all the bio-alcohol blended FAME fuels improved significantly while maintaining good brake thermal efficiency compared with conventional diesel fuel operation. Further, as a result of the ignition delay measurements employing standard fuels, it was found that the estimated cetane numbers of the tested bio-alcohol blended FAME fuels are strongly dependent on the estimated cetane number of the base fuel.

Key words: Diesel engine, Fatty acid methyl ester, Next generation bio-alcohol, EGR, Supercharging,
　　　　　　Trade-off improvement

1. INTRODUCTION

　　To suppress carbon dioxide emissions and prevent depletion of petroleum resources, studies have been conducted on the application of biofuels such as biodiesel and bio-alcohol which are promising renewable energy sources as alternative fuels for diesel engines. Biodiesel is produced from transesterification reactions with vegetable oil and methanol, and it can suppress PM, soot, and smoke emissions in diesel combustion due to its oxygenated fuel characteristic. Biodiesel is treated as having carbon neutral properties, and can reduce CO_2 emissions in the life cycles [1], leading to numerous studies having been conducted [2-7]. Depending on the kind of vegetable oil, the kind and composition of the fatty acid methyl esters (FAME) that are contained in biodiesel differ and these different kinds of FAME and their compositions have been reported to affect the combustion and emission characteristics of diesel engines [3].

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In a previous paper [5], the author used blended fuels with high purity FAME (reagents) and ordinary diesel fuel (JIS No.2 gas oil) in a diesel engine, and investigated the effects of FAME kinds and blending ratios on engine performance, combustion characteristics, and emissions. In that study, single droplet combustion using the tested fuels was also performed, and the combustion behavior was observed with visualization by a shadow graph method. Further, the soot formed during the droplet combustion was collected by adhering on a glass fiber filter, and the mass of the soot was measured. The experiments used eight kinds of FAME from C8:0 (methyl caprylate) to C18:3 (methyl linolenate), and these FAME fuels were blended into diesel fuel with mass ratios of 0 to 100%. The soot formation rate was defined as the mass ratio of the collected soot to the fuel droplet, and the results showed that the soot formation rates of all the tested fuels decreased significantly with increasing FAME blending ratios (with increasing oxygen content in the blends) [5].

For the application of bio-alcohols to diesel engines, bio-ethanol has been adopted for study as a research object from the early stages of this research. When ethanol with a high oxygen content is blended with diesel fuel, significant reductions in PM and smoke emissions can be obtained [8]. Further, the smoke reduction effect of the ethanol blended fuel is more effective when EGR is applied to reduce NOx [9]. However, ethanol has drawbacks such as low cetane number, poor miscibility with diesel fuel, and the hygroscopic characteristics. Butanol has the features of a biofuel and can be produced by the classical ABE (acetone-butanol-ethanol) fermentation method, and many studies have been reported as bio-butanol is superior to bio-ethanol as a diesel fuel replacement [10-15]. When butanol is blended with biodiesel, the oxygen content of the blends increase, and this may be expected to lead to further reductions in PM and smoke emissions [11,13]. Further, the low temperature fluidity which is a drawback of biodiesel, is significantly improved [13,15].

In recent years, research and development of high efficiency microbial fermentation methods have progressed with the production of next generation bio-alcohols using non-food based sources like cellulosic biomass [16,17]. The resulting next generation bio-alcohols contain other components than ethanol, somewhat larger volumes of butanol (C4) and pentanol (C5) with more carbons and are higher alcohols. For these reasons, studies of the application of next generation bio-alcohols as a supplement to diesel fuels have been strongly promoted in recent years [18-36]. The cetane numbers of C4/C5 bio-alcohols are still significantly lower than that of diesel fuel, and to ensure ignitability such bio-alcohols are generally used by blending with conventional diesel fuel or biodiesel.

To obtain simultaneous reductions in NOx and PM emissions from diesel engines, it is well known that combining with EGR and supercharging technology is an effective measure [4]. The concept is that to reduce NOx emissions by decreasing oxygen concentration and flame temperature in local combustion regions with EGR, while supercharging compensate for the output loss caused by increasing EGR rates. The present study investigated the effects of combining cooled EGR and supercharging with fueling by next generation bio-alcohol blended FAME in a DI (direct injection) diesel engine equipped with a jerk type injection system. The next generation bio-alcohols used here were isopentanol (C5) and isobutanol (C4), and these alcohol isomers are produced variously depending on the production

processes. In recent research, research and development to improve the yield of isopentanol and isobutanol with conversion technologies by modern fermentation processes have been progressing [16].

Recent studies conducted with C4/C5 next generation bio-alcohols [18-36] can be categorized into several groups: for the kind of alcohol used, the studies have been conducted with only n-butanol [27,32,33,35], with only isobutanol [30], with only n-pentanol [19,20,29,31,34,36], with n-butanol and isobutanol [21], with n-butanol and n-pentanol [23,24,25], with isobutanol and n-pentanol [22,26,28], and with all four butanol isomers [18]. For the base fuel, studies have been carried out with diesel fuel based alcohols [18,21,22,26-28,30,33,34,36], with diesel fuel and biodiesel blend based alcohols [19,23,24,29], with neat biodiesel based alcohols [31,35], with diesel fuel and neat vegetable oil blend based alcohols [25], and without any base fuel, using neat bio-alcohols, [20,32]. For the blending ratio of bio-alcohol, lower ratio blends than 20 vol.% [23-25,29,33,35], with up to 35 vol.% [19,21,26,27,31], up to 40 vol.% [18,22,30,36], up to 45 vol.% [28], a higher blending ratio of 50 vol.% using n-pentanol [34], and operation with neat n-pentanol [20] as well as with neat n-butanol [32] have been reported. For injection systems, common-rail type fuel injection were used in the studies [18-20,27,32,33,34,36], and the application of EGR [21,22,28,30,36], supercharging [19,20], and combining EGR and supercharging [18,27,32,34] have also been reported.

An overview of published papers combining EGR and supercharging will be detailed next. Zheng et al. [18] conducted experiments fueled by four kinds of butanol isomer blended diesel fuel (20 and 40 vol. %) using a modified single cylinder engine with a common-rail type injection system. A wide EGR rate range from 0 to 65% was set with a constant boost pressure of 200 kPa using a compressor, and EGR was simulated by pure CO_2. They examined the effects of butanol isomer blended diesel fuel on the indicated thermal efficiency, combustion characteristics, and emission characteristics (soot, NOx, and THC) under normal combustion and low temperature combustion. The results showed that the ignition delay becomes longer in the order of isobutanol/diesel > 2-butanol/diesel > n-butanol/diesel > tert-butanol/diesel, while the soot emissions tend to decrease in the reverse order of the ignition delays. That study did not vary the boost pressure and the influence of supercharging on the engine characteristics is not clear.

Cheng et al. [27] have suggested that partial premixed low temperature combustion is effective to obtain a simultaneous reduction in NOx and smoke emissions while maintaining a high thermal efficiency. To examine the influence of the different reactivities and oxygen content of fuels on diesel combustion, they used 10 and 30 vol. % n-butanol blended diesel fuel at a moderate EGR rate (about 26% EGR, intake oxygen concentration 17.6%). The experiments were conducted over a wide range of injection timings (-33 to 5 °CA.ATDC), fuel injection pressures (100, 120, 140 MPa), and engine loads (BMEP of 0.43, 0.86, and 1.12 MPa). The results showed that the 30% n-butanol blended diesel fuel can provide significant reductions in smoke emissions without deterioration in fuel consumption at appropriate injection timings, indicating that the PPCI combustion regions can be expanded. The experiments used a four cylinder light-duty diesel engine with a common-rail type injection system, an air compressor, a turbocharger, and a cooled EGR device, but the study does not mention the boost

pressure conditions.

Han et al. [32] modified an originally four-cylinder engine with a common-rail type injection system to a single cylinder engine, and performed experiments at boost pressures of 1.5-2 bar (gauge pressure) using a compressor at cooled EGR rates of 0-55%. That study established that the compression ignition of the (partially) premixed n-butanol and air mixture is capable of producing diesel-like engine efficiency and significant reductions in NOx and smoke emissions. However, the combustion became rapid with increasing engine load and the engine output was limited below 50% of the baseline capability. In the next stage, the experiments were conducted with full use of a split injection strategy, and here the rapid combustion was effectively suppressed. The study finally realized high load operation similar to diesel fuel operation using neat n-butanol, a very impressive research achievement.

Pan et al. [34] used a blended diesel fuel with 25% or 50% n-pentanol (P25 and P50) using a four-cylinder turbocharged diesel engine with a common-rail type injection system at a constant 10% EGR rate condition. That study reported a single injection experiment with a constant injection pressure of 120 MPa, and here the n-pentanol blended diesel fuels decreased soot emissions, while the BSFC, HC, and NOx increased. It was concluded that the P50 fuel gives a 1.86% decrease in brake thermal efficiency, and a significant reduction in soot (77.15%), a result achieved without engine modification. In this study, the spray angle and spray penetration were also measured with a constant volume vessel at conditions of 120 MPa fuel injection pressure, ambient pressure 5 MPa, and ambient temperature 673K. It showed that the atomization characteristics with the n-pentanol blended diesel fuel are better than those of diesel fuel alone.

As described above, there are a few reports combining EGR and supercharging in diesel engines and that employ next generation bio-alcohols, and the effects on the trade-off relation between NOx and smoke emissions using a jerk type injection system are not fully known. Studies with the next generation bio-alcohol isopentanol have been reported: for example, there is research on the engine performance and combustion mechanism of HCCI with a gasoline based engine [37], kinetic research of detailed reactions in the HCCI combustion process [38], and measurements and modeling of the laminar flame speeds of isopentanol/air mixtures [39]. But findings with diesel combustion are very limited. In a literature survey by the author here, no research other than a previous paper [40] by the present author was located. Details of these results will be described in the following.

2. EXPERIMENTAL APPARATUS AND METHODS

2.1 Test fuels

For the blend of bio-alcohols in this study, methyl laurate (LME, purity＞96%) and equal proportions of methyl palmitate (PME, purity＞95%) and LME, termed PLME, were used as the base fuel. The LME is a major component of coconut oil biodiesel and PME is about half of the component in palm oil biodiesel. Both LME and PME are methyl esters of saturated fatty acids having twelve and sixteen carbon atoms respectively, and have better ignitability than ordinary

diesel fuel. The study used isopentanol and isobutanol as the next generation bio-alcohols, and the blending ratio of the bio-alcohol to base fuel was a constant 3:7 (mass ratio). The blended fuel with LME and isopentanol, or with LME and isobutanol is expressed as LiP30 or LiB30. Similarly, PLiP30 or PLiB30 are blended fuels with PLME and isopentanol or with PLME and isobutanol, respectively. The experiments were conducted using seven kinds of fuels: neat LME, neat PLME, LiP30, LiB30, PLiP30, PLiB30, and ordinary diesel fuel (JIS No. 2 gas oil) as a reference. In addition, n-HD (n-hexadecane) and HMN (heptamethylnonane), which are standard fuels for defining cetane numbers, were used to estimate the cetane number of the bio-alcohol blended FAME fuels.

Table 1 shows the particulars of the tested fuels. The cetane number of isopentanol is not known, but from the ignition delay behavior in engine experiments [40], it may be estimated that the value would be between 17 (cetane number of n-butanol [25]) and 20 (cetane number of n-pentanol [25]). The cetane numbers of the bio-alcohols used here are much lower than that of diesel fuel, but they have low boiling point characteristics and that can be expected to improve the evaporation characteristics by blending bio-alcohols into the FAME. Isopentanol and isobutanol are oxygenated fuels and the oxygen content in the fuel molecule structure is 18.2% and 21.6%, respectively.

Table 1 Properties of the tested fuels

Test fuel	Diesel	PME	LME	PLME	iso-pentanol	iso-butanol	LiP30	LiB30	PLiP30	PLiB30
Density15 ℃] (kg/m³)	836	867[b]	873	870[c]	813	807	853	851	851[c]	848[c]
Viscosity30 ℃] (mm²/s)	3.89	—	2.90	3.92	—	2.63[e]	2.67	2.58	3.32	3.23
Cetane number	—	74.3[b]	61.2[b]	—	—	<15[f]	—	—	—	—
Cetane index	57.1	—	37.0	47.0	—	—	36.1	37.4	43.1	44.3
Lower heating value(MJ/kg)	42.95	37.12	35.35	36.11[a]	34.62	32.61	35.13[a]	34.53[a]	35.66[a]	35.06[a]
B.P. at 90 % distillation (℃)	336	—	261	316	131[d]	108[d]	261	261	314	314
Carbon[a] (mass %)	86.1	75.5	72.8	74.2	68.1	64.8	71.4	70.1	72.4	71.4
Hydrogen[a] (mass %)	13.8	12.7	12.2	12.5	13.7	13.6	12.7	12.6	12.9	12.8
Oxygen[a] (mass %)	—	11.8	14.9	13.4	18.2	21.6	15.9	16.9	14.8	15.9
Stoichiometric air-fuel ratio[a]	14.6	12.5	11.9	12.2	11.7	11.2	11.8	11.7	12.1	11.9

[a] estimated
[b] from Ref. [41]
[c] estimated from the measured value at 30 ˚C
[d] boiling point
[e] from Ref. [26] at 40 ˚C
[f] from Ref. [26]

2.2 Engine setup

Fig. 1 shows an outline of the experimental apparatus, consisting of a test engine, a device for cooling the EGR, and a supercharging system. The test engine is a small single cylinder, water cooled, four stroke direct injection diesel engine with a jerk-type injection system. Table 2 shows the specifications of the engine and fuel injection system. The experiments used a Roots blower type

supercharger driven by an inverter controlled motor, making it possible to control the boost pressure independent of the engine operating variables. The boost pressures were set at two conditions: 100 kPa (N/A, naturally aspirated operation, without supercharging) and 120 kPa (S/C, supercharged operation). The back pressure in the exhaust pipe was set to a constant 110 kPa during the supercharged operation.
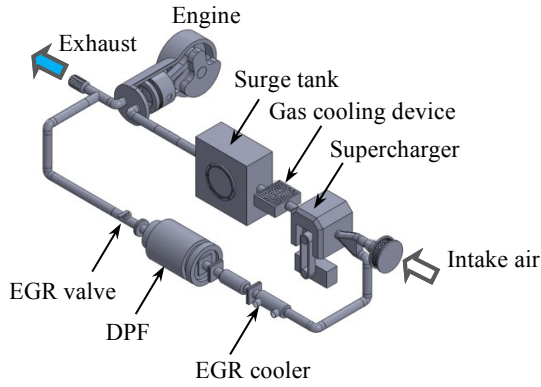
Fig. 1 Engine setup

Table 2 Engine specifications

Engine model	4 stroke, Horizontal, Water cooled
Bore	82 mm
Stroke	78 mm
Stroke volume	411cc (Single cylinder)
Compression ratio	18
Combustion chamber	DI (Toroidal type)
Rated output	5.1 kW/2400 rpm (BMEP=0.62 MPa)
Injection pump	Bosch PFR (Plunger 7 mm)
Injection nozzle	DLLA 150 (4-ϕ0.2)
Opening pressure	21.7 MPa
Injection timing	Fixed (19 $^\circ$CA.BTDC)

A previous study [42], where the boost pressures were varied from 100 to 140 kPa at a constant high load condition, suggested that improvements in diesel combustion were small when the boost pressure was increased above 120 kPa. The EGR rate was adjusted by an EGR valve upstream of the diesel particulate filter (DPF). The experiments employed low pressure loop cooled EGR and the recirculated gas was introduced into an intake pipe upstream of the supercharger. The EGR rate was determined from Eq. (1), and the intake air quantity was measured by a laminar flow meter attached upstream of the supercharger. The gas temperatures in the intake pipe were controlled at 30-34˚C. During the experiments, the engine was operated at a constant engine speed of 1900 rpm which corresponds to the maximum brake torque conditions. The engine load was set at a constant high load condition (BMEP=0.67 MPa) where the trade-off relation between NOx and smoke emissions assumes critical values.

$$EGR \% = \frac{(CO_2)_{intake}-(CO_2)_{ambient}}{(CO_2)_{exhaust}-(CO_2)_{ambient}} \times 100 \tag{1}$$

2.3 Measuring apparatus and procedure

The combustion pressure was measured with a piezoelectric type pressure pick-up and the crank angle was detected by a rotary encoder; the needle lift of the nozzle was monitored by a Hall-effect element. The values of the three signals were recorded digitally, and the rate of heat release and the degree of constant volume of combustion η_{glh} were determined from the average pressure of 50 cycles. Assuming energy equilibrium of the gas in the cylinder as in Eq. (2), the degree of constant volume of combustion, η_{glh}, is determined from Eq. (3):

$$dQ = dQ_E - dQ_C = \frac{1}{\kappa - 1}\left(\kappa P dV + V dP\right) \tag{2}$$

$$\eta_{glh} = \frac{1}{Q}\int \frac{1 - \dfrac{1}{\varepsilon_\theta^{\kappa-1}}}{1 - \dfrac{1}{\varepsilon^{\kappa-1}}} \frac{dQ}{d\theta} d\theta \tag{3}$$

The ignition delay was determined from the crank angle interval between the start of the needle lift and the pressure rise due to the combustion. The NOx emissions were measured using a CLD analyzer, the HC was measured as ppm methane using an FID analyzer, the CO and CO_2 were measured using an NDIR analyzer, and the smoke density was measured with an opacimeter.

3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Influence of supercharging and EGR on engine performance and emission characteristics with bio-alcohol blended FAME fuels

Fig. 2 shows the changes in brake thermal efficiency, η_e and exhaust gas temperature as a function of the EGR rate with ordinary diesel fuel, neat LME, and the bio-alcohol blended LME fuels. Isopentanol and isobutanol were selected as the next generation bio-alcohols, and a constant blend ratio with alcohol vs. LME of 3:7 (mass) was employed. Here, the boost pressure was changed: 100 kPa (N/A) or 120 kPa (S/C) with the supercharger. The boost pressure 100 kPa is the naturally aspirated operation in the present experiments, and referred to as "without supercharging". Fig. 3 shows the changes in smoke and CO emissions in these cases.
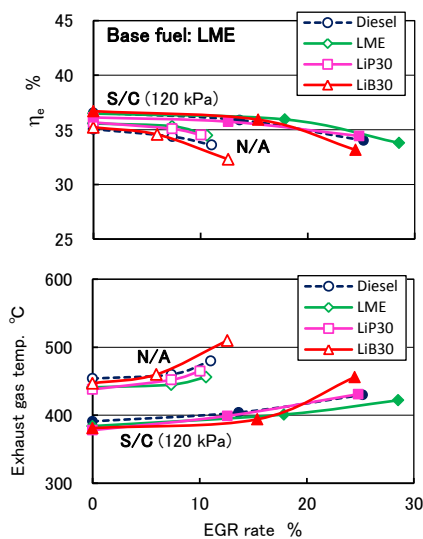
Fig. 2 Results when combining EGR and supercharging on brake thermal efficiency and exhaust gas temperature with the LME based bio-alcohol blends
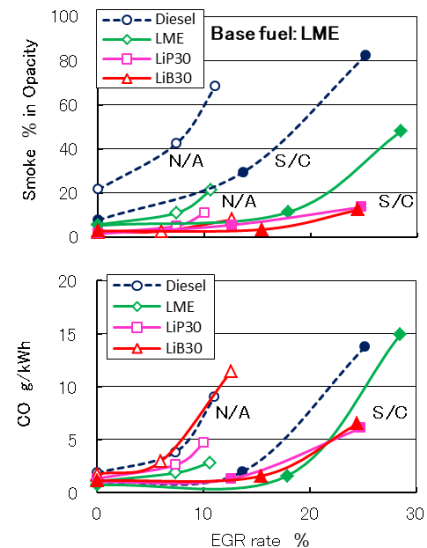
Fig. 3 Results when combining EGR and supercharging on smoke and CO emissions with the LME based bio-alcohol blends

As shown in the top panel in Fig. 2, the brake thermal efficiency η_e, with all the tested fuels increased under the supercharged operation over the tested EGR region. The relations among the brake thermal efficiency η_e, the theoretical thermal efficiency η_{th}, the degree of constant volume of combustion η_{glh}, the combustion efficiency η_u, the rate of cooling loss ϕ_w, and the mechanical efficiency η_m is:

$$\eta_e = \eta_{th} \cdot \eta_{glh} \cdot \eta_u (1 - \phi_w) \eta_m \tag{4}$$

As shown in Fig. 3, the smoke and CO emissions under the supercharged operation decreased significantly, compared to the naturally aspirated operation. Therefore, one reason for the brake thermal efficiency improvement with supercharging can be ascribed to the increased combustion efficiency η_u, due to an increase in the in-cylinder air density. As shown in the bottom panel of Fig. 2, the exhaust gas temperatures under the supercharged operation showed substantial decreases over the tested EGR regions. The reason is considered to be that the temperature rise of the in-cylinder charge is suppressed because of the increasing heat capacity due to supercharging. Here, when LiB30 (blended fuel with LME and isobutanol) was used, the exhaust gas temperatures rose remarkably at the high EGR rate condition even when supercharging was employed. The reason for this will be described in the next section.

Now, the influence of the EGR rate with and without supercharging will be discussed. When the EGR rate was increased to about 10% with the naturally aspirated operation, the smoke emissions with diesel fuel showed significant increases (top panel in Fig. 3), and this EGR rate condition cannot be employed. When supercharging was employed with the bio-alcohol blended LME fuels, the EGR rate could be increased to nearly 20% without large increases in smoke. In this case, the brake thermal efficiency, η_e with supercharging showed that the decreasing trend with increasing EGR rate was suppressed below that of the naturally aspirated operation as shown in the top panel of Fig. 2. Both with and without supercharging, the smoke emissions with neat LME and bio-alcohols blended LMEs decreased substantially over the tested EGR region, as compared with diesel fuel (top panel of Fig. 3). Also, the bio-alcohol blended LME showed lower smoke emissions than those of the neat LME. The reason is considered to be that biofuels are oxygenated, and the additionally available oxygen compensates for local oxygen shortages that occur when EGR is applied.

Fig. 4 shows the changes in smoke emissions as a function of the oxygen content in the fuel for all the tested fuels. It is clear that regardless of supercharging, the smoke emissions decrease linearly with increasing oxygen content in the fuel. These results are in good agreement with the trends of the soot formation rates obtained in the single droplet combustion using different blends with diesel fuel and neat FAME [5]. Therefore, it may be concluded that when different biofuels are employed in diesel engines, the smoke emissions would mainly be controlled by the oxygen content in the fuel.

Regardless of the kind of fuel, the CO emissions tend to increase with increasing EGR rate, specifically in the naturally aspirated operation with diesel fuel and with LiB30, as shown in the bottom panel in Fig. 3. The reason for the substantial increases in CO emissions with ordinary diesel fuel

without supercharging is considered to be that the oxygen concentration in local combustion regions decreased due to the EGR. For the LiB30 fuel, as will be further described in the next section, the ignition delay becomes substantially longer as the EGR rate increases, and it is considered that this causes an excessively lean mixture, resulting in an expansion of the incomplete combustion regions. Overall, regardless of the kind of fuel, significant reductions in CO emissions are obtained when applying supercharging.
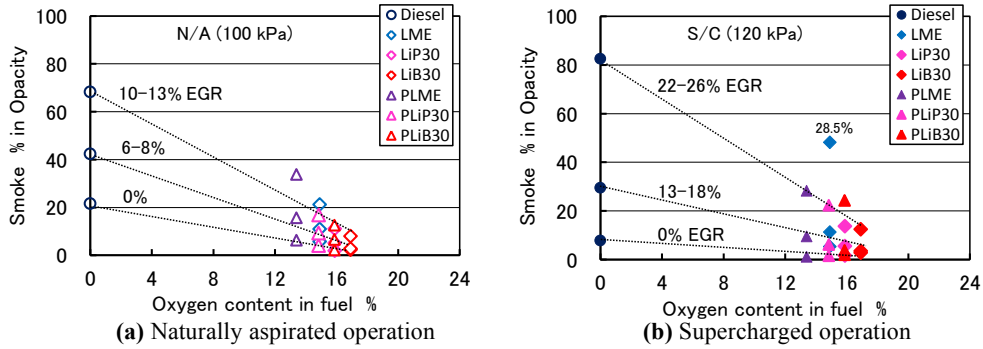
Fig. 4 Smoke emissions as a function of the oxygen content in the tested bio-alcohol blended fuels under N/A and S/C operation

Fig. 5 shows the changes in NOx and HC emissions as a function of the EGR rate with diesel fuel, neat LME, and bio-alcohol blended LME fuels. Regardless of supercharging, the NOx emissions decreased substantially with increasing EGR rates. The reason is considered to be that the oxygen concentration and flame temperature in local combustion regions decreased due to the EGR. As shown in the bottom panel of Fig. 5, the HC emissions were low in all the tested conditions.
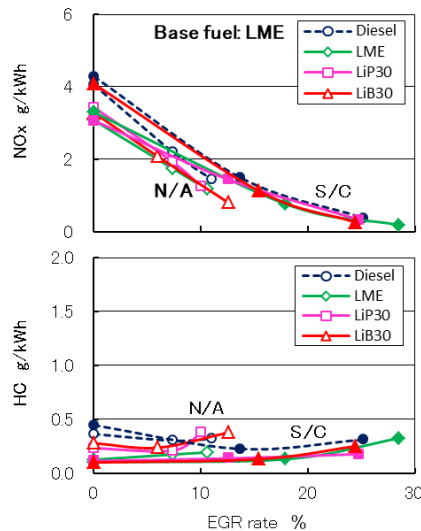
Fig. 5 Results when combining EGR and supercharging on NOx and HC emissions with the LME based bio-alcohol blends

3.2 Effects of supercharging and EGR on combustion characteristics for bio-alcohol blended FAME fuels

Fig. 6 shows the influence of supercharging and EGR on indicator diagrams and the heat release rate histories with the LME based bio-alcohol blends. Figures 6(a) and 6(b) show the influence of EGR in the naturally aspirated operation (without supercharging) for the tested fuels. Also, Figures 6(c) and 6(d) show the differences in indicator diagrams and heat release rates with and without EGR and with supercharging. Here, Fig. 6(b) is with 10 to 13% EGR without supercharging, and Fig. 6(d) is the supercharged operation with 25 to 28.5% EGR. Comparing Figures 6(a) and 6(b), with EGR in the naturally aspirated operation the ignition timings of both C5/C4 bio-alcohols blended LME fuels are much retarded, and the peak values of the heat release rates in the premixed combustion phase are increased. With EGR, the oxygen concentration in the cylinder charge entrained into the spray flux decreases. This would allow the conclusion that fuel-air mixing until ignition is possible becomes delayed resulting in the retardation of the ignition timing. The comparison of Figures 6(a) and 6(c) clearly showed that the ignition timings for LiB30 with supercharging advance remarkably. The main reason for this is that the increase in-cylinder pressure due to supercharging promotes the chemical reactions in the ignition. As an another view, Lee et al. [4] has described that the main factor in the improvement of ignition characteristics due to supercharging is that the amount of air entrained into the spray flux increases resulting in mixture formation promoting.

(a) 0% EGR, N/A   **(b)** 10～13% EGR, N/A   **(c)** 0% EGR, S/C   **(d)** 25～28.5% EGR, S/C
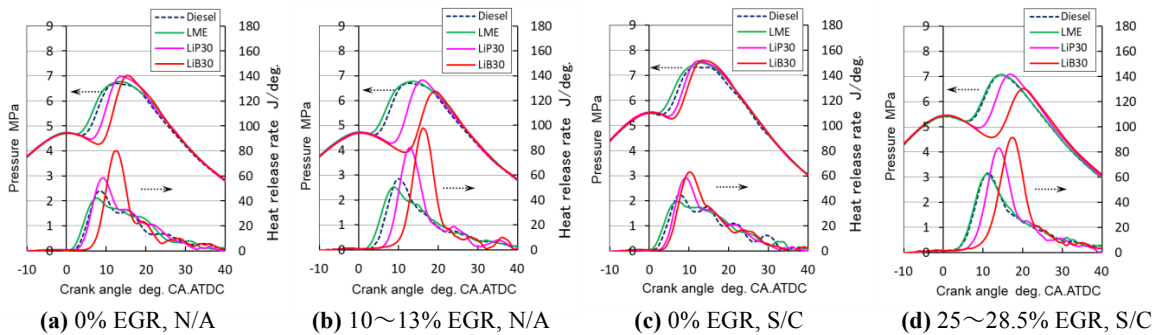
Fig. 6 Results when combining EGR and supercharging on indicator diagrams and heat release rates with the LME based bio-alcohol blends

Fig. 7 plots the changes in combustion characteristics, the ignition delay, maximum heat release rate, $(dQ/d\theta)_{max}$, and constant volume of combustion, η_{glh}, as a function of the EGR rate. Fig. 7(a) with the bio-alcohol blended LME fuels under the naturally aspirated operation shows that the ignition delay is remarkably prolonged and that the maximum heat release rate, $(dQ/d\theta)_{max}$ is much higher than with diesel fuel. This tendency becomes stronger as the EGR rate increases, but the values decrease significantly with supercharging. Considering the influence of the tested bio-alcohol kinds, the ignition delays with LiB30 (isobutanol blended LME) are longer than those of LiP30 (isopentanol blended LME), both with and without supercharging. This result is consistent with the general tendency that lower carbon number alcohols have a lower cetane number. With supercharging, the ignition delays of the bio-alcohol blended LME fuels are considerably longer than that of ordinary diesel fuel, but the

constant volume of combustion, η_{glh} show similar values up to around 20% EGR (bottom panel in Fig. 7(a)). This suggests that the tested bio-alcohol blended LME fuels had more rapid combustion speeds after the ignition, and that the premixed combustion phase is promoted. The reason is considered to be due to the bio-alcohol blended LME containing low boiling point components, alcohols, and the mixing of the fuel vapor and air was improved. However, the LiB30 showed a significant decrease in η_{glh} at the maximum EGR rate condition also when supercharging was employed. Therefore, it may be concluded that the significant decrease in the brake thermal efficiency, η_e observed with the LiB30 fuel for supercharged operation at the maximum EGR rate condition (top panel in Fig. 2) was due to the decrease in the η_{glh}. Comparing Figures 7(a) and 7(b), the bio-alcohol blended PLME fuels show significant reductions in ignition delays and lower maximum heat release rates than those of the bio-alcohol blended LME. This reduction in ignition delays arise as methyl palmitate (PME) has a high cetane number of 74.3 [41] and better ignitability than LME.

(a) Base fuel: LME   **(b)** Base fuel: PLME
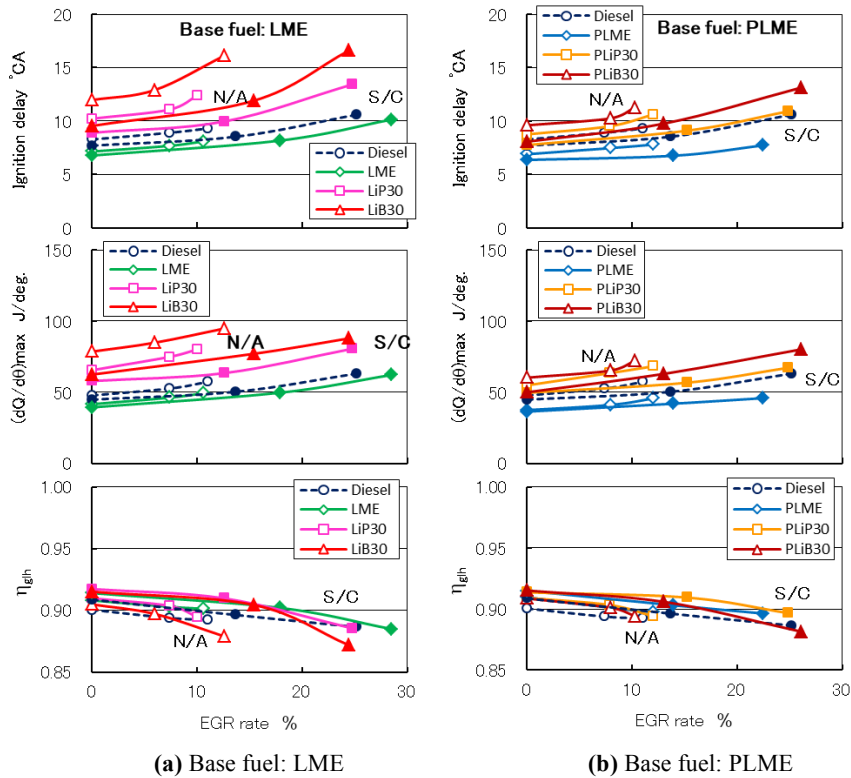
Fig. 7 Results when combining EGR and supercharging on combustion characteristics with the tested bio-alcohol blended fuels

Fig. 8 plots the changes in combustion characteristics, $(Q_E-Q_B)/Q_E$ as a function of the EGR rate with the LME based bio-alcohol blends. Here, Q_E is the heating value of the supplied fuel, Q_B is the total heat release during combustion, and (Q_E-Q_B) represents the heat loss of the in-cylinder charge. Based on this, the value of the ordinate suggests the rate of heat loss during combustion, and allows the conclusion that this factor reflects the magnitude of the rate of cooling loss, ϕ_w in Eq. (4). Regardless of the kind of fuel, the rate of heat loss with supercharging is much lower over the tested EGR region than with the

naturally aspirated operation. Overall, it may be concluded that the decrease in the rate of cooling loss, ϕ_w under the supercharged operation contributes, as one factor, to the increase in the brake thermal efficiency, η_e (top panel in Fig. 2). It may be assumed that the gas temperature drop close to the cylinder wall caused by supercharging results in decreasing the heat loss by this cooling. A further reason may be that significant reductions in smoke emissions caused by supercharging decreases heat transfer by radiation from the soot. Such an effect caused by reduced soot would contribute to the reduction in the rate of cooling loss ϕ_w.
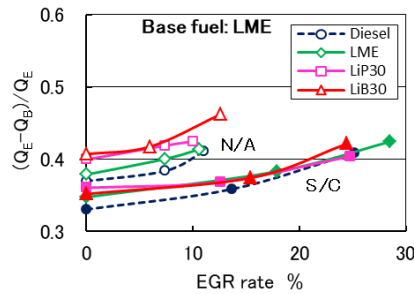

Fig. 8 Results when combining EGR and supercharging on the rate of heat loss
with the LME based bio-alcohol blends

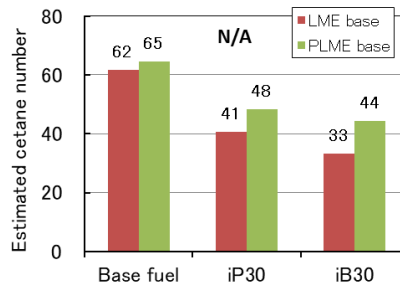## 3.3 Estimation of the cetane numbers of the tested bio-alcohol blended FAME fuels

As shown in Fig. 7, the ignition delays of the bio-alcohol blended PLME were significantly shorter than those of the LME based alcohol blends. To obtain a further and quantitative understanding on the self-ignition characteristics of the bio-alcohol blended FAME fuels, the present study attempted to estimate the cetane number of the tested biofuels. To examine the relationship between ignition delay and cetane number, the further experiments here were conducted using standard fuels that are used to define cetane numbers. Fig. 9 shows the relationship between the EGR rate and ignition delay when standard fuels with the cetane numbers of 70, 55, and 40 were applied to the test engine at a naturally aspirated condition. The unbroken curves in Fig. 9 are approximate curves that represent the ignition delays of each standard fuel as a function of the EGR rate, and the equations for the curves (approximate formulas) are also shown in Fig. 9.
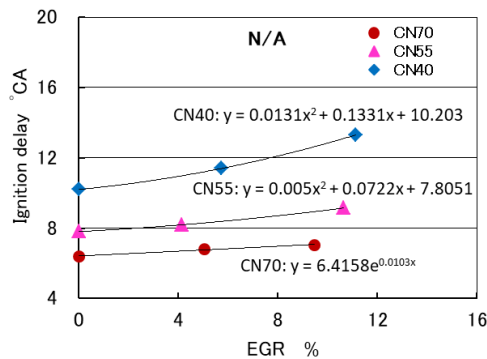
Fig. 9 Plot of EGR rate vs. ignition delay with standard fuels defining cetane number at
the naturally aspirated condition
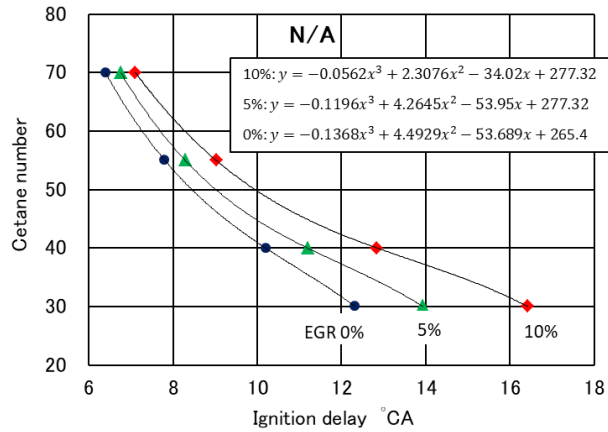
Fig. 10 Plot of ignition delay vs. cetane number with the standard fuels as a function of reference EGR rates at the naturally aspirated condition

Fig. 11 Estimated cetane numbers of the tested bio-alcohol fuel blends using LME and PLME as the base fuels

Regardless of the cetane number of the standard fuels, the ignition delays can be accurately approximated as functions of the EGR rates. With EGR rates of 0, 5, and 10% as references, the ignition delays at the reference EGR rates of each standard fuel were calculated using the approximation formulas shown in Fig. 9 with Fig. 10 showing the results. The unbroken curves in Fig. 10 show approximate curves representing the cetane number of the standard fuels at the three reference EGR rates as a function of the ignition delay. The ignition delays at cetane number 30 were obtained by extrapolation using the approximate formulas shown in Fig. 10. Using the approximate formulas and as shown in Fig. 10, the cetane numbers can be estimated from the ignition delays at the reference EGR rates (0, 5, and 10%) of the fuels being investigation.

Fig. 11 shows the results of the cetane numbers estimated for the six kinds of biofuels used in the present study, using the above procedure. Here, the ignition delays of the tested fuels at the reference EGR rates (0, 5, and 10%) were calculated from the approximate formulas between ignition delays vs. EGR rates, using the ignition delay data as shown of the top panels of Figures 7(a) and 7(b). Graboski et al. [41] reported the cetane number of LME as 61.2, while the present analysis obtained the estimated value of the cetane number as 62. This suggests that the estimation procedure for the cetane number here can be considered adequate. As shown in Fig. 11, there is a small difference in the estimated cetane numbers of LME and PLME, which are the base fuels of the bio-alcohols blends. However, comparing

the estimated cetane numbers of the isopentanol blended FAME and isobutanol blended FAME, it is clear that the cetane number of the alcohol blended fuel strongly depends on the cetane number of the base fuel. In other words, that a small difference in the cetane number of the base fuel is amplified in the alcohol blended fuels.

3.4 Effects of the trade-off improvements with bio-alcohol blended FAME fuels combining with supercharging and EGR

Fig. 12 plots the effects of combining supercharging and cooled EGR for the tested fuels on the trade-off relation for the NOx emissions vs. the brake thermal efficiency, η_e as well as between NOx vs. smoke emissions. Fig. 12(a) is the LME based blends, and Fig. 12(b) shows the case with PLME based blends, respectively. When EGR was employed under the naturally aspirated operation (100 kPa boost pressures, N/A), the NOx emissions decreased significantly for all the tested fuels; but, the smoke emissions, especially with the ordinary diesel fuel, increased substantially, so that the EGR rate was limited to low levels. When the supercharging was employed, the decreasing tendency of η_e was suppressed remarkably even when the engine was operated at high EGR rate conditions, as well as the trade-off relation between NOx and smoke emissions were improved remarkably. Comparing the kinds of tested fuels, the bio-alcohol blended LME fuels showed the better effects in smoke reductions than the bio-alcohol blended PLME fuels. For the brake thermal efficiency η_e, the neat PLME with supercharging tended to be somewhat better than the other tested fuels.

(a) Base fuel: LME   (b) Base fuel: PLME
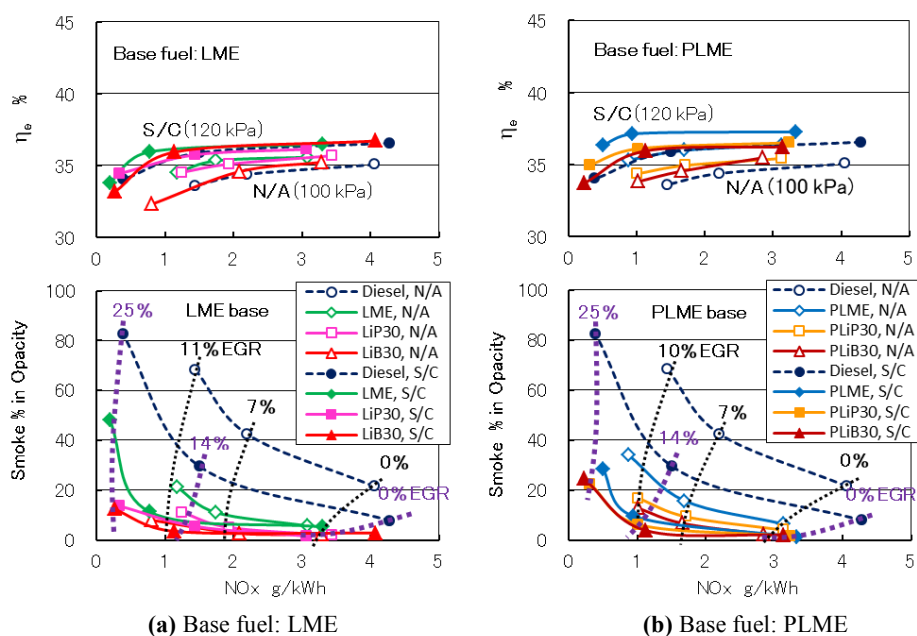
Fig. 12 Trade-off relation between NOx vs. brake thermal efficiency and NOx vs. smoke emissions with the tested bio-alcohol blended fuels

4. CONCLUSIONS

Next generation bio-alcohol blended FAME fuels were supplied to a small single cylinder DI diesel engine equipped with a jerk type injection system. As blending alcohols, methyl laurate (LME), and an equal proportion of methyl palmitate and LME (termed PLME) were used as the base fuels. Four kinds of blended fuels: isopentanol blended LME (termed LiP30), isobutanol blended LME (LiB30), and isopentanol or isobutanol blended PLME (PLiP30 or PLiB30) were prepared with a constant blending ratio of 3:7 (mass). The engine performance, combustion characteristics, and emissions were examined as functions of two boost pressures (100 kPa, N/A, and 120 kPa, S/C) and the EGR rates were varied (0-28.5%). Compared with ordinary diesel fuel, as a reference, the following conclusions were obtained.

(1) All the tested bio-alcohol blended FAME fuels under supercharged operation showed that the trade-off relation between NOx vs. smoke emissions improved substantially over the tested EGR region, maintaining a brake thermal efficiency comparable to ordinary diesel fuel.

(2) Both with and without supercharging, the neat FAME fuels (LME and PLME) reduced smoke emissions significantly compared to diesel fuel operation over the tested EGR region. With bio-alcohol blended FAME fuels further smoke reductions were obtained, as compared with the neat FAME fuels.

(3) The reasons for the brake thermal efficiency improvements with supercharging may be considered to be that the cooling loss reduction due to in-cylinder gas temperature decreases and as combustion efficiency increases due to reductions in smoke and CO emissions.

(4) The isobutanol blended FAME fuels tended to have longer ignition delays than those of the isopentanol blended FAME fuels, but the differences in engine performance and emission characteristics were small when supercharging was employed.

(5) The cetane numbers of the tested bio-alcohol blended FAME fuels were estimated with further experiments employing standard fuels used for defining cetane numbers. The results showed that a small difference in the estimated cetane numbers of the base fuels is amplified by the alcohol blending.

ACKNOWLEDGMENTS

NOMENCLATURE

PM: particulate matter

EGR: exhaust gas recirculation

This system recirculates a part of exhaust gas to the engine combustion chambers. EGR is a very effective technique for reducing NOx emissions in diesel engines due to lowered local oxygen

concentration and flame temperature.

PPCI: partially premixed compression ignition

PCCI: premixed charge compression ignition

HCCI: homogeneous charge compression ignition

N/A: naturally aspirated operation

S/C: supercharged operation

FAME: fatty acid methyl ester

PME: methyl palmitate

LME: methyl laurate

PLME: equal proportions of PME and LME (mass ratio)

PLiP30: 70% PLME and 30% isopentanol blend (mass ratio)

PLiB30: 70% PLME and 30% isobutanol blend (mass ratio)

LiP30: 70% LME and 30% isopentanol blend (mass ratio)

LiB30: 70% LME and 30% isobutanol blend (mass ratio)

θ: crank angle

Q: heat release in cylinder

Q_E: heating value of the supplied fuel

Q_C: cooling due to cylinder wall

Q_B: total in-cylinder heat release

P: in-cylinder gas pressure

V: in-cylinder gas volume

κ: ratio of specific heats

ε: compression ratio

ε_{θ}: compression ratio at crank angle

BMEP: brake mean effective pressure

η_e: brake thermal efficiency

η_u: combustion efficiency

η_{glh}: degree of constant volume of combustion

ϕ_w: rate of cooling loss

$(dQ/d\theta)_{max}$: maximum heat release rate

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