REMARKABLE IMPROVEMENT OF NOx-PM TRADE-OFF IN A DIESEL ENGINE BY MEANS OF BIO-ETHANOL AND EGR

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ABSTRACT

In order to realize a premixed compression ignition (PCI) engine, the effects of bioethanol-gas oil blends and exhaust gas recirculation (EGR) on PM-NOx trade-off have been investigated focusing on ignition delay, premixed combustion, diffusion combustion, smoke, NOx and thermal efficiency. The present experiment was done by increasing the ethanol blend ratio and ethanol and by increasing the EGR ratio in a single cylinder direct injection diesel engine. It is found that a remarkable improvement in NOx-PM trade-off can be achieved by promoting the premixing based on the ethanol blend fuel having low evaporation temperature, large latent heat and low cetane number as well, in addition, based on a marked elongation of ignition delay due to the low cetane number fuel and the low oxygen intake charge. As a result, very low levels of NOx and PM, which satisfies the 2009 emission standards imposed on heavy duty diesel engines in Japan, were achieved without deterioration of brake thermal efficiency in the PCI engine fuelled with the 50% ethanol blend diesel fuel and the high EGR ratio. It is noticed that smoke can be reduced even by increasing the EGR ratio under the highly premixed condition.

KEYWORDS: PCI engine, Bio-ethanol, EGR, PM-NOx trade-off

1. INTRODUCTION

Recent researches have shown that a HCCI engine or a PCI engine will be a promising way to accomplish almost zero emissions in NOx and PM. In order to achieve the severe emission standards imposed on diesel engines such as the post new-long term emission standards for 2009 in Japan, the EURO VI standards for 2012 and the US standards for 2010, it is necessary not only to depend upon the aftertreatment system installed with NOx catalyst and PM catalyst but also to suppress formations of NOx and PM in the spray combustion process by forming the HCCI or PCI condition.

A dual fuel diesel engine, a kind of HCCI engine, fuelled with natural gas (NG) and ignited by a small amount of gas oil was investigated to reduce both NOx and particulate emissions simultaneously by Kusaka, et al.[1], Ishida, et al.[2], Tagai, et al. [3] and Saito, et al.[4] previously, and also studied by Carlucci, A. P., et al.[5] and Papagiannakis, R. G., et al.[6] recently. A similar dual fuel engine fuelled with methanol and assisted by a small amount of gas oil was tested by the authors [7-9]. As a result, a significant improvement in the trade-off between NOx and PM was achieved without deteriorating the thermal efficiency by means of the premixed charge under the conditions of a minimum amount of gas oil for stable ignition and EGR as well. A direct injection diesel engine fuelled with dimethyl ether (DME) was also investigated for low emission diesel engines by Kajitani, S. et al.[10, 11] and Ohmura, T., et al.[12], however, the HCCI engine fuelled with DME alone showed some problems associated with the ignition timing control, the large amount of unburned hydrocarbon emission, and the limited load operation range due to misfire and knock. Then, HCCI engines fuelled with composite fuel of DME and natural gas were investigated by Chen, et al.[13, 14], Konno, et al.[15], Iida, et al.[16] and Ishida, et al.[17-20], and the effects of intake temperature, compression ratio and EGR ratio on the low temperature reaction (LTR) and the high temperature reaction (HTR), misfire limit, knock limit, exhaust emissions and thermal efficiency were clarified. A similar HCCI combustion fuelled with DME and methanol was investigated by Ishida et al.[21], a HCCI combustion fuelled with a vaporized diesel fuel was examined by Ganesh, et al.[22] and a partial HCCI combustion based on early direct injection of diesel fuel was studied by García, et al.[23]. Another HCCI engine fuelled with hydrogen and carbon monoxide reformed from DME was investigated by Shudo, et al.[24, 25] and the one fuelled with the reformed gas from biodiesel and bioethanol was investigated by Tsolakis et al.[26] respectively. It was shown that both natural gas and methanol have a strong effect on suppression of the LTR of DME, and the HTR is retarded properly to near TDC, resulting in a high degree of constant volume combustion. The effect of EGR was focused on improvement in the brake thermal efficiency of a HCCI engine in the authors' studies[18, 20], and it was noticed that the increment of 4% in brake thermal efficiency was achieved by the high EGR ratio of 0.5 compared with the case without EGR. The increase in brake thermal efficiency due to EGR was based on improvement of combustion efficiency mainly resulting from a significant reduction in the unburned hydrocarbon emission. Another focus was put on the achievable highest load in a HCCI engine. Consequently, it was found that the highest load could be achieved under the condition with the minimum DME amount for stable ignition, however, the achieved highest load was limited up to about 0.45 MPa in any studies.

As mentioned in the above, HCCI engines showed an important problem on the limited load range due to misfire and knock, then, a premixed compression ignition (PCI) engine was thought to be better to expand the engine load range. In recent studies on the PCI engine operated with a low cetane number fuel, adopting a high EGR ratio, Ogawa et al.[27] and Li et al.[28, 29] showed experimentally that smoke emission could be decreased drastically by large quantities of cooled EGR and low cetane number fuels, and smoke was dependent strongly upon a long enough ignition delay, in other words, the premixing time between the end of fuel injection and the onset of ignition. It is clear that this low emission could be achieved by the low temperature combustion based on the ultra-high EGR ratio. Ishida et al.[30] showed the effectiveness of gasoline-diesel fuel blends in a diesel engine experimentally, and smoke was reduced markedly by increasing the gasoline blend ratio because ignition delay was elongated due to the gasoline properties with low cetane number and low evaporation temperature.

Similarly, ethanol as a carbon neutral fuel should be noticed as another fuel with low cetane number and low evaporation temperature. Combustion tests on bio-ethanol was done by Kamio et al.[31] in HCCI-SI hybrid combustion, and the effect of water on bio-ethanol HCCI combustion was investigated by Megaritis et al.[32] and Mack et al.[33]. On the other hand, Rakopoulos et al.[34] has done combustion analysis in a diesel engine fuelled with 15% ethanol-diesel fuel blends. The objective of the present study is to show a feasibility of a PCI engine for achieving extremely low PM and NOx by adopting bioethanol-gas oil blends in combination with EGR. The gas oil blended with bio-ethanol was burned in a diesel engine, and the effects of the ethanol blend ratio and the EGR ratio on ignition delay, premixed combustion, diffusion combustion, smoke density, concentrations of NOx, unburned hydrocarbon and carbon monoxide, and the brake thermal efficiency were investigated in detail.

NOMENCLATURE

ATDC	After top dead center
be	Brake specific fuel consumption [MJ/kWh]
BTDC	Before top dead center
CA	Crank angle
CO	Carbon monoxide emission [ppm] or [g/kWh]
CA50	50% burnt point [deg.CA ATDC]
CO_2	CO_2 concentration [%]
$dQ/d\theta$	Net heat release rate [J/degree]
EGR	Exhaust gas recirculation
EtOH30	Blend fuel with 68vol% gas oil, 29vol% ethanol and 3vol% octanol
EtOH50	Blend fuel with 48vol% gas oil, 48vol% ethanol and 4vol% octanol
HCCI	Homogeneous charge compression ignition
HTR	High temperature reaction
Hu	Lower heating value of fuel [J/mg]
Lift	Needle valve lift [mm]
LTR	Low temperature reaction
m_d	Mass rate of diffusion combustion [mg/cycle]
NOx	Nitrogen oxides emission [ppm] or [g/kWh]
p_{me}	Brake mean effective pressure [MPa]
P	In-cylinder pressure [MPa]
PCI	Premixed compression ignition
PM	Particulate matter [mg/cycle] or [g/kWh]
Q_{total}	Total cumulative heat release [J/cycle]
Q_d	Cumulative heat release of diffusion combustion [J/cycle]
Q_p	Cumulative heat release of premixed combustion [J/cycle] ($=Q_{total}-Q_d$)
Smoke	Smoke density [Bosch]
THC	Total unburned hydrocarbon emission [ppm] or [g/kWh]
T	Temperature [°C]
X_{EGR}	EGR ratio (approximated by CO_{2IN}/CO_{2Exh})
θ	Crank angle [degree]
θ_{inj}	Injection timing [deg.CA BTDC]
ε	Compression ratio
Subscripts	
0	Without EGR
e, Exh	Exhaust gas
IN	Intake charge

2. EXPERIMENTAL APPARATUS AND MEASUREMENTS

The test engine was a single cylinder high-speed naturally aspirated direct injection diesel engine, the type NFD 170-(E) manufactured by YANMAR Co., Ltd. Figure 1(a) shows the section of test engine with a horizontal cylinder axis. Piston is a conventional one with toroidal cavity. The bore is 102 mm, the stroke is 105 mm, and the compression ratio is ε=17.8. Fuel injection pump is the conventional type Bosch PFR-1AW with 9 mm plunger dia. Principal particulars of the test engine is shown in Table 1. Figure 1(b) shows the combustion test system. The suction air pressure at the intake manifold was always adjusted at the standard atmosphere pressure of 0.1013 MPa by using the electric blower. Intake charge temperature was kept constant at T_{IN}=40±0.5 °C by using electric heater installed upstream of mixing chamber and EGR cooler was used in the case with EGR, that is, the cooled EGR was adopted. EGR gas was charged into the mixing chamber located at 1,800 mm upstream of the intake manifold. Air flow rate was measured by using entrance nozzle installed in the plenum tank located upstream of surge tank. In order to calculate the fuel flow rate, a weight decrement of fuel tank was measured in the specified time interval of ten minutes by the electronic weighing instrument. The combustion tests were carried out at a constant engine speed of 1,200±5 rpm, 50% of the rated speed. Engine load was controlled by the electrical dynamometer, measuring shaft torque and rotational speed, and it was changed from 20% load (p_{me}=0.13MPa) to 80% load (0.51MPa), which correspond to equivalence ratios of 0.19 and 0.53 in the cases without EGR. The fuel was injected into the combustion chamber directly at a constant injection timing of 5±0.2 deg.CA BTDC, however at the final experiment, it was advanced to 14±0.2 deg.CA BTDC to achieve better NOx-PM trade-off. The test conditions are summarized in Table 2.

Test fuels are low sulfur diesel fuel (gas oil JIS #2 in Japan) having cetane number of 55 and ethanol with that of about 8. Properties of tested fuels are summarized in Table 3. The blend fuel named EtOH30 consists of 68 vol% gas oil, 29 vol% ethanol with 3vol% octanol as a surface-active agent, and EtOH50 consists of 48% gas oil, 48% ethanol and 4% octanol. The solubility between gas oil and ethanol was examined at three kinds of fuel temperature as shown in Fig.2, and the minimum amount of octanol for a stable blend fuel without phase separation was determined experimentally for various ethanol blend ratios. The lower heating value of ethanol 26.8 MJ/kg is fairly smaller compared to that of gas oil 42.9 MJ/kg. Ethanol has lower cetane number, lower evaporation temperature and larger latent heat compared with gas oil, and the cetane number of the blend fuel shown in Table 3 was estimated from the volume ratio between ethanol and gas oil.

The exhaust gas temperature was measured by means of the E-type thermocouple at the exhaust manifold, and temperatures of fuel, air, cooling water and intake charge were measured by the T-type thermocouple. CO_2 concentration was measured at both intake and exhaust manifolds respectively to calculate the EGR ratio approximately. In the exhaust gas analysis, exhaust gas temperature Te, concentrations of carbon monoxide CO, total unburned hydrocarbon THC and nitrogen oxides NOx, and smoke density were measured by means of respective sensor. Table 4 shows summary of equipment, detection principle, and accuracy of measurements. The time-history of in-cylinder pressure and the needle valve lift were measured by using the piezoelectric pressure transducer and gap sensor simultaneously and the outputs were sampled every one-fourth degree in crank angle by means of the 4 channel combustion analyzer CB-467 manufactured by Ono Sokki Co. Ltd. The time-history of combustion pressure was the ensemble average sampled over continuous 350 engine cycles. The data were transmitted to the personal computer and recorded on hard disks.

3. RESULTS AND DISCUSSION

3.1 Effects of Ethanol Blend Ratio and EGR Ratio on Combustion Time-History and Engine Performance

Figures 3(a), (b) and (c) show changes in time-history of combustion due to ethanol blend, EGR and engine load as well respectively. The abscissa denotes the crank angle θ deg.CA, and the ordinates denote the measured in-cylinder pressure P, the rate of net heat release dQ/dθ and the needle valve lift respectively. In the experiment, the injection timing was set at 5±0.2 deg.CA BTDC as shown in the lift curves, and the intake charge temperature was kept constant at 40 °C even in the case with EGR by means of the cooled EGR system. As the ethanol blend ratio increases and also the EGR ratio increases, as shown in Figs.3(a) and (c), the ignition timing is retarded and the ignition delay becomes larger markedly especially in the case of EtOH50, on the other hand, the end of combustion was not retarded. And the premixed combustion increases and the diffusion combustion decreases remarkably. As the engine load increases in the case of EtOH50, the heat release rates in both premixed combustion and diffusion combustion increase but ignition delay is almost unchanged as shown in Fig.3(b). It should be noticed that, in the case of EtOH50 having low cetane number and low evaporation temperature, ignition occurs after the end of fuel injection at low loads such as p_{me}=0.13 and 0.26 MPa.

Figure 4 shows changes in exhaust emissions and fuel consumption due to engine load, where the parameters are the fuel and the EGR ratio. The engine load “p_{me}” was increased by increasing the injection amount of blend fuel at each EGR ratio. The brake specific fuel consumption “be[g/kWh]” in usual was reduced to “be[MJ/kWh]” by compensating the lower heating value of each fuel, and they show almost equal fuel consumption between three cases, however, the one of EtOH50 is a little higher in the low loads and a little lower in the high loads compared with the one of gas oil alone. The higher fuel consumption in the low load results from increases in CO emission and THC emission due to too much lean burn condition with the equivalence ratio of 0.20-0.29. On the other hand, the lower fuel consumption in the high load results from a little earlier end of combustion as shown in Fig.3(a). NOx increases a little by ethanol blend because the premixed combustion increases due to longer ignition delay and lower evaporation temperature, and it is reduced markedly by EGR as shown in Fig.4. Smoke density in the case of EtOH50 is reduced remarkably over the load operation range, in addition, it is decreased further by EGR. Especially in the low loads, smoke density becomes almost zero due to promotion of the premixing resulting from late ignition after the end of fuel injection[27] as shown in Fig.3(b).

Figure 5 shows the NOx reduction rate due to EGR and Fig.6 shows smoke change rate due to EGR. It is reasonable that NOx decreases remarkably as the EGR ratio increases, and about 90% reduction in NOx was achieved by the EGR ratio of 0.5. It should be noticed that, only in the case of EtOH50, smoke was reduced by increasing the EGR ratio. The reason why it was reduced by EGR will be clearly shown in the latter section of this paper.

3.2 Elongation of Ignition Delay due to Ethanol and EGR

Definitions of ignition timing, ignition delay, the premixed combustion and the approximated diffusion combustion are shown in the heat release rate curve calculated from the experimental pressure history. In Fig.7, the ignition timing was defined as the zero-cross point of the dotted line tangential to the heat release rate curve in the initial premixed combustion stage. The ignition delay was defined as the crank angle duration between the start of injection and the ignition point. The heat release rate curve during the diffusion combustion period was approximated by the Wiebe function[35] between the second peak of the heat release curve and the end of combustion. In the present analysis, it was assumed that the diffusion combustion begins at the maximum heat release timing in the premixed combustion, which is a similar procedure in the two zone model analysis by the authors[36, 37]. The cumulative heat release rate of diffusion combustion Q_d was calculated by integrating the Wiebe function, and the premixed combustion quantity Q_p was calculated by subtracting Q_d from the total cumulative heat release Q_{total}. Figure 8 shows an uncertainty of diffusion combustion index "m" of Wiebe function in relation to diffusion combustion quantity Q_d. The tendency that the index m decreases as Q_d increases is quite reasonable although showing some inevitable scattering.

Figure 9 shows variation of ignition delay due to the EGR ratio, where the parameters are the fuel and the engine load. The ignition delay is dependent basically upon the fuel cetane number; the cetane number of the tested fuels varies from 55 of gas oil to 41 of the fuel EtOH30 and 32 of the fuel EtOH50. In the ethanol blend fuels, ignition delay increases mainly based on the low cetane number and secondarily based on the large latent heat and the low evaporation temperature as well. The lower the cetane number, the longer the ignition delay is, and it also increases more by increasing the EGR ratio. However, variation of ignition delay due to engine load is relatively small except for the cases with a high EGR ratio near misfire limit. Figure 10 shows the correlation between the ignition delay and the oxygen concentration in the intake charge mainly dominated by the EGR ratio. The plotted data are the same ones shown in Fig.9, and they are separated into three groups clearly depending on the fuel cetane number. The rate of increase in ignition delay due to decrease in oxygen concentration is much larger in the lower cetane number fuel, in other words, the so-called dilution effect[32] in the intake charge due to EGR appears markedly on the low cetane number fuel. As a result, the premixing can be promoted more in the lower cetane number fuel under a high EGR condition, of course, in the fuel with lower evaporation temperature.

Figures 11(a) and (b) show the relationship between the premixed combustion quantity Q_p and the amount of heat input, in which Q_p consists of combustion of gas oil and/or ethanol; (a) shows comparison of Q_p between the three fuels, and (b) shows the relation between Q_p and the amount of injected ethanol heat. As shown in Fig.11(a), the premixed combustion Q_p based on gas oil alone is almost unchanged by the heat input, or, the engine load. On the other hand, as shown in Fig.11(b), Q_p based on ethanol combustion increases almost linearly with the amount of injected ethanol heat. In other words, the premixing can be promoted significantly by increasing the ethanol blend ratio. Figure 12 shows the relation between Q_p/Q_{total} and NOx, where the parameters are the fuel and the engine load, and the data with different EGR ratio are indicated by the same symbol mark. As the EGR ratio increases, Q_{total} varies little at the specified load but Q_p becomes large. In this case, NOx decreases drastically while Q_p/Q_{total} increases. As shown in Fig.12, Q_p of the low cetane number fuel EtOH50 becomes about twice compared with that of gas oil, however, the increase in NOx is small. It is clear that NOx is not always dependent on the premixed combustion in the present experiment. Generally, NOx increases as the premixed combustion increases because the larger premixed combustion results in the higher combustion pressure, then, the higher combustion temperature. However, as the ethanol blend ratio increases, PCI combustion approaches to HCCI combustion due to promotion of the premixing, then, the high temperature region in the spray combustion becomes small, resulting in suppression of NOx formation rate.

3.3 Relationship between Diffusion Combustion and PM

Figure 13 shows change in the cumulative heat release Q_d during diffusion combustion due to ignition delay, where the parameter is the engine load p_{me}. The data in the figure include the cases with different ethanol blend ratio and the cases with different EGR ratio. Q_d, which seems to be a main factor of smoke emission, decreases almost linearly with increase in ignition delay at any engine load while Q_d is larger at the higher load because of the larger amount of injected fuel.

The mass rate of diffusion combustion "m_d" was calculated by the following equation;

$$m_d=Q_d/Hu \qquad (1)$$

where Hu is the lower heating value of the fuel.

Assuming that all ethanol burns in the premixed combustion stage, and composition in gas oil having higher evaporation temperature burns in the diffusion combustion process because ethanol has fairly low evaporation temperature compared with gas oil, the lower heating value of gas oil was adopted for Hu in Eq.(1). Figure 14 shows a correlation between the mass rate of diffusion combustion m_d and the injection quantity of gas oil. All Q_d data shown in Fig.13 are plotted again in Fig.14. It is clear that m_d correlates well with the injection quantity of gas oil, in other words, m_d is strongly dependent upon injection quantity of gas oil and it is about 70% of the injected gas oil.

Figure 15 shows the relationship between the particulate matter "PM" in the exhaust gas and the diffusion combustion mass "m_d", where the parameter is the EGR ratio "X_{EGR}". Figure 16 shows the relationship between PM and the diffusion combustion mass parameter "$m_d/(1-X_{EGR})$". PM is dependent on both factors of m_d and X_{EGR} as shown in Fig.15, however, it is dependent on the unique factor of $m_d/(1-X_{EGR})$ as shown in Fig.16. The factor of $(1-X_{EGR})$ represents the fraction of fresh intake air in the intake charge, then, $m_d/(1-X_{EGR})$ means the diffusion combustion mass per unit fresh air, in other words, a kind of fuel/air ratio like. If the EGR ratio is constant, PM increases as m_d increases, and also PM increases as the EGR ratio increases if m_d is constant. It cannot be simply determined whether the PM decreases or increases because variation of m_d is dependent strongly upon ignition delay. In order to reduce PM at the time when the EGR ratio increases, the value of $m_d/(1-X_{EGR})$ should become smaller, for instance, as shown in Fig.16 by two solid circles on the correlation line. This condition is written by the following equation;

$$m_d/(1-X_{EGR}) < m_{d0}/(1-0) \quad (m_d=m_{d0} \text{ at } X_{EGR}=0)$$

then,

$$m_d/m_{d0} < (1-X_{EGR}) \qquad (2)$$

Figure 17 shows change due to the EGR ratio in the normalized diffusion combustion mass m_d/m_{d0}. The right hand side of Eq.(2) corresponds to the thick solid line with its gradient of (-1). If the value of m_d/m_{d0} locates in the region below the thick solid line, the PM can be reduced by EGR. The experimental data shown in Fig.17 are identical to the ones shown in Fig.6. Only in the case of 50% ethanol blend fuel EtOH50, PM could be reduced by increasing the EGR ratio, and NOx, of course, could be reduced simultaneously by EGR.

3.4 Improved Results of PM-NOx Trade-off

Figure 18 shows the relation between the 50% burnt point CA50 and the EGR ratio in the case of the 50% ethanol blend fuel EtOH50. The CA50 was defined as the crank angle at which the cumulative heat release becomes 50%. At the fuel injection timing of 5 deg.CA BTDC, the EGR ratio was limited up to 0.2 due to misfire as shown in Fig.11, on the other hand, the EGR ratio for the misfire limit was 0.38 at p_{me}=0.51 MPa at the fuel injection timing of 14 deg.CA BTDC. The figure indicates that misfire occurs if the CA50 reaches between 10 and 15 deg.CA ATDC, which is seen not only in the present case but also in the HCCI engine[12]. In order to achieve less NOx, the EGR ratio should be increased more. If the fuel injection timing is advanced from 5 to 14 deg.CA BTDC, a high EGR ratio is allowable for the misfire limit.

Figure 19 shows the PM-NOx trade-off lines in the cases of gas oil and EtOH50. The ordinate denotes the specific particulate matter PM g/kWh and the abscissa denotes the specific emission of NOx g/kWh. As shown in Fig.19, the original trade-off based on gas oil, showing high levels of PM and NOx, is drastically improved by means of the 50% ethanol blend fuel EtOH50. By adopting the allowable highest EGR ratio in the case of EtOH50 having low cetane number and low evaporation temperature, both NOx and PM was reduced simultaneously below the post new-long term emission standards for 2009 in Japan. Very low levels of 0.4 g/kWh in NOx and 0.006 g/kWh in PM could be achieved at the fuel injection timing of 14 deg.CA BTDC without deterioration of thermal efficiency, which are really achieved by promotion of the premixing.

CONCLUSION

The goal of the present study is to show one of the approaches for achieving low NOx and low PM combustion in a diesel engine by utilizing bio-ethanol and adopting EGR. The gas oil was blended with bio-ethanol for realizing a PCI engine in the present experiment. The concluding remarks obtained here are as follows;

(1) Ignition delay increases with increase in the ethanol blend ratio, and also increases more by increasing the EGR ratio, resulting in promotion of the premixing.

(2) A marked increase in ignition delay due to ethanol blending is based on low cetane number, low evaporation temperature and large latent heat of ethanol.

(3) Increase in ignition delay due to EGR is caused by the so-called dilution effect with the low oxygen charge.

(4) Increase in the ethanol blend ratio promotes the premixing, and results in a decrease in the diffusion combustion quantity.

(5) It is found that PM is a function of $m_d/(1-X_{EGR})$ alone; where m_d is the mass rate of diffusion combustion and X_{EGR} is the EGR ratio. Furthermore m_d is dominated by the injection quantity of gas oil.

(6) It is noticed that, in the case of the 50% ethanol blend fuel, PM could be reduced by increasing the EGR ratio due to a remarkable increase in ignition delay.

(7) Very low levels of NOx and PM could be achieved without deterioration of the thermal efficiency by promotion of the premixing.

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Table 1 Principal particulars of test engine

Model type	NFD-170(E), Single cylinder, Water-cooled, Four-stroke	
Bore and stroke	102x105 [mm]	
Compression ratio	17.8	
Maximum power and speed	12.5 [KW]/2,400 [rpm]	
Valve timing	Opening	Closing
Intake	8 [° BTDC]	40 [° ABDC]
Exhaust	50 [° BBDC]	12 [° ATDC]
Fuel pump	Bosch PFR-1AW with 9 mm plunger	
Injection nozzle	4 holes with 0.29 dia. [mm]	
Opening pressure	19.6 [MPa]	

Table 2 Combustion test conditions

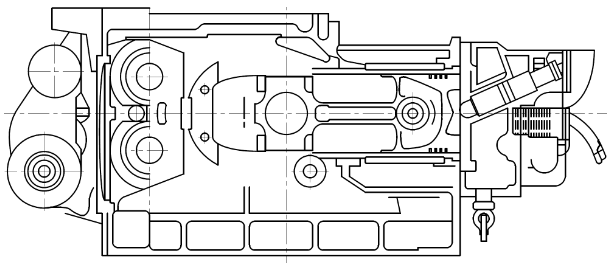
Engine speed	1,200 rpm
Suction air pressure	0.1013 MPa
Intake air temperature	40 °C
Injection timing	5, 14 deg.CA BTDC
Mean effective pressure; p_{me}	0.13(20%)-0.51(80%) MPa
EGR ratio; X_{EGR}	0 - 0.5 or misfire limit

Table 3 Properties of test fuels

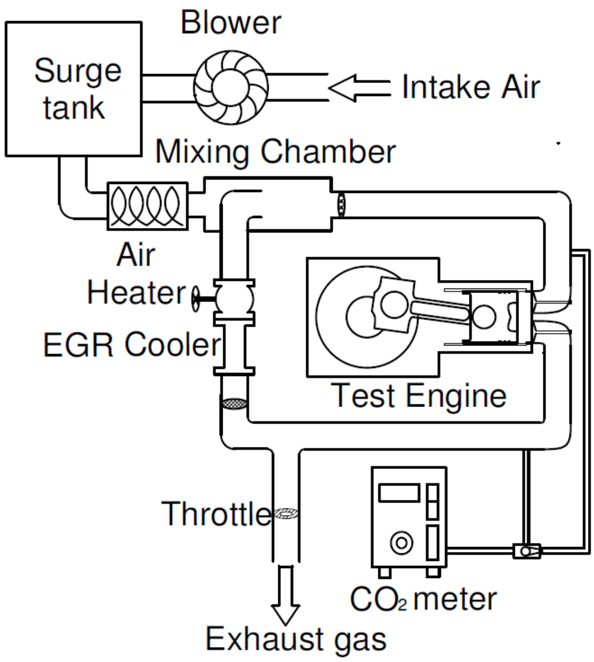
	Gas oil	EtOH30	EtOH50	Ethanol
Gas oil [vol%]	100	68	48	0
Ethanol [vol%]	0	29	48	100
Octanol [vol%]	0	3	4	0
Lower heating value [MJ/kg]	42.9	38.3	35.2	26.8
Latent heat [MJ/kg]	0.25	0.43	0.55	0.86
Boiling point [°C]	190-350	-	-	78.3
Oxygen [wt%]	0	10.1	16.8	34.8
Cetane No.	55	41	32	8

Table 4 Summary of equipment, detection principle, andaccuracy of measurements

Measuring item	**Detection principle**	**Equipment (Maker)**	**Scale range**	**Accuracy**
CO	Constant potential electrolysis	testo350M (TESTO)	0-10000[ppm]	±5%Linearity
NOx	CLD (chemical luminescence detector)	ECL-77A (Yanaco)	0-1000[ppm]	±1%FS
THC	FID (flame ionization detector)	EHF-710H (Yanaco)	0-1000[ppm]	±1%FS
Smoke	Bosch type	DSM-10 (BANZAI)	0-10[Bosch]	±2% Linearity
CO_2	NDIR(non-dispersive infrared)	BIR-200 (BEST SOKKI)	-	±1%FS
In-cylinder pressure	Piezoelectric transducer/ charge amplifier	6125B/5011B (KISTLER)	0-250[bar]	±0.5%FS
Needle valve lift	Gap sensor	PU-03A (AEC)	0-1[mm]	0.3[μm]
Crank angle	Encoder	PP-011A (Ono Sokki)	-	±0.25 deg
Torque	Torque detector/meter	DSTP-10/ DMT-408 (Ono Sokki)	0-100[Nm]	±0.5%FS
Temperature	Thermo-couple	E type/T type (YAMARI)	0-900/0-350[°C]	±1.7/±1.0[°C]
Fuel flow rate	Electronic weighing	EB-32KD (SHIMADZU)	0-32[kg]	±1[g]
Air flow rate	Inlet nozzle pressure drop	ISP-3-50 (SHIBATA)	0-50[mmAq]	±1%FS



(a) Section of test engine



(b) Combustion test system

Fig.1 Test engine and combustion test system

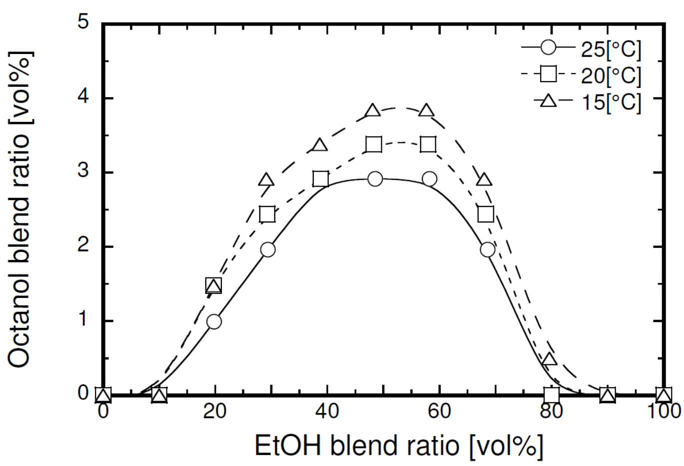
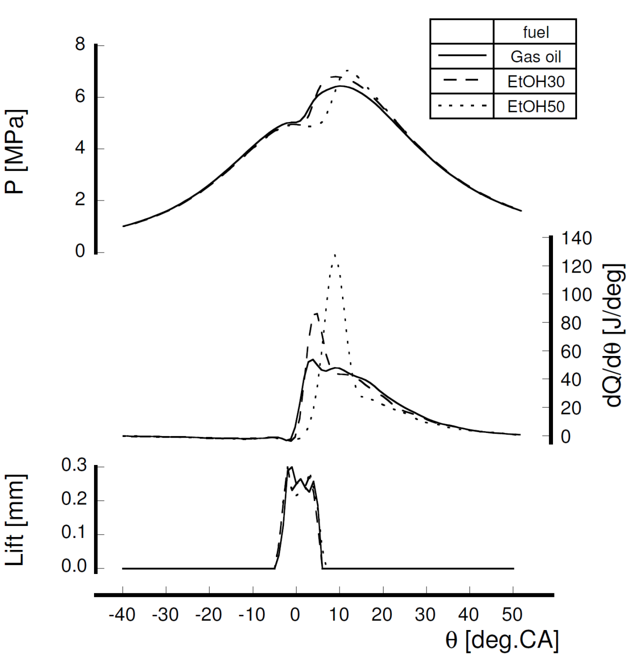
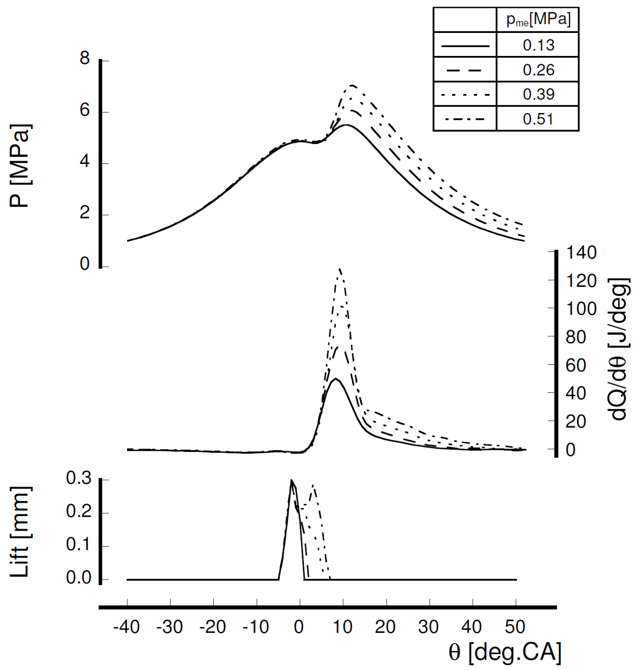


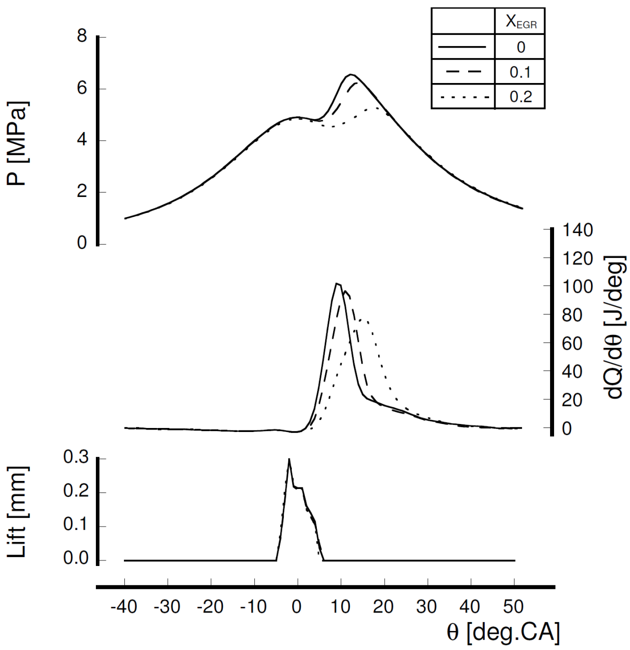
Fig.2 Effect of octanol on solubility between gas oil and ethanol



(a) Effect of ethanol blend ratio (p_{me}=0.51 [MPa])



(b) Effect of engine load (EtOH50)



(c) Effect of EGR ratio (EtOH50, p_{me}=0.39[MPa])

Fig.3 Change in combustion time-history

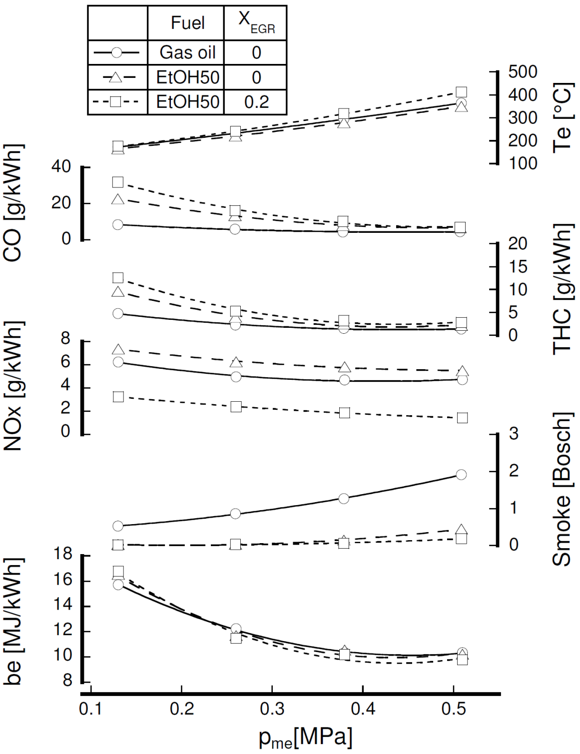


Fig.4 Effects of ethanol and EGR on exhaust emissions and engine performance

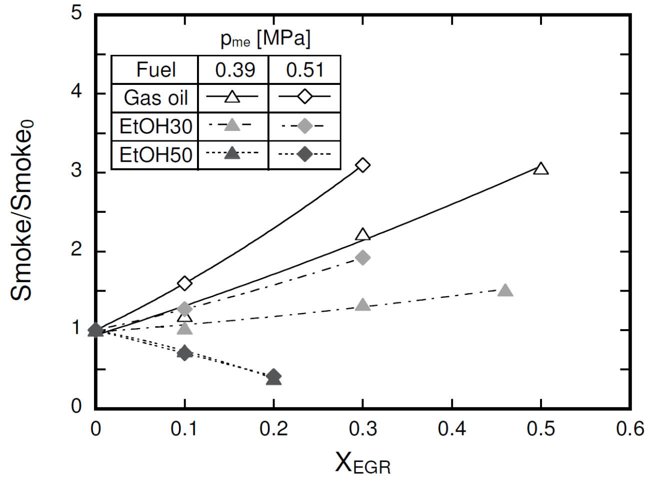


Fig.6 Change in smoke due to EGR

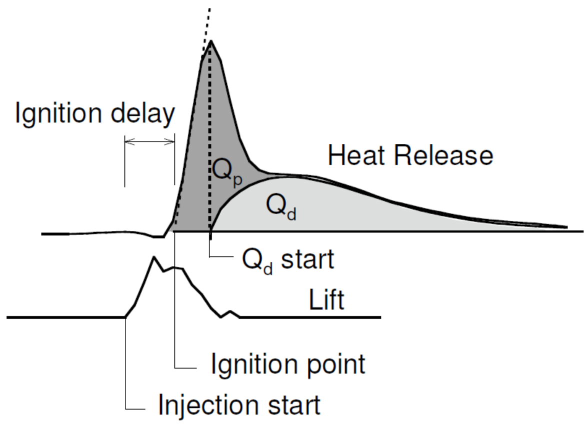


Fig.7 Definitions of ignition delay, premixed combustion Q_p and diffusion combustion Q_d

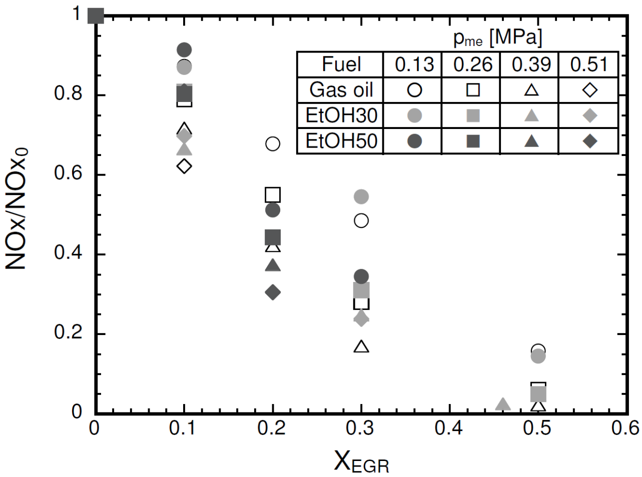


Fig.5 NOx reduction rate due to EGR

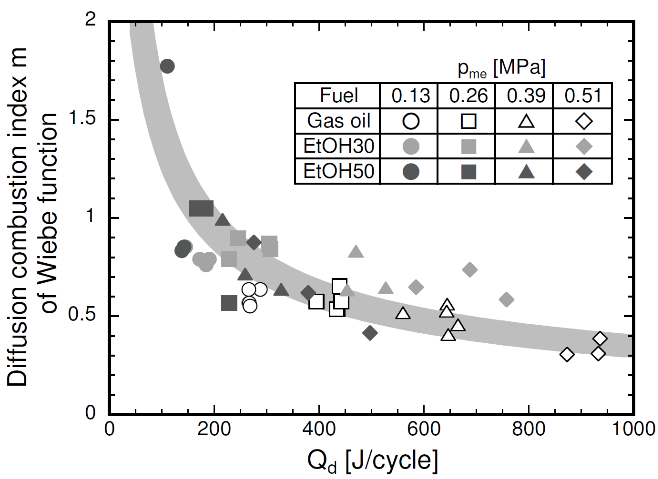


Fig.8 Uncertainty of diffusion combustion index “m” approximated by Wiebe function

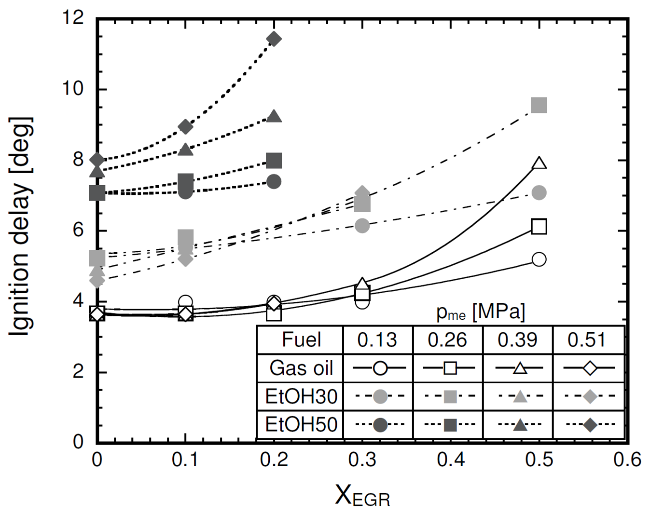


Fig.9 Effects of ethanol and EGR on ignition delay

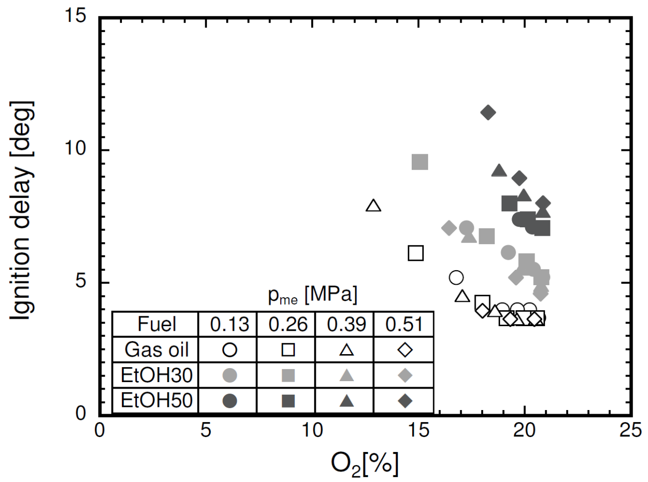
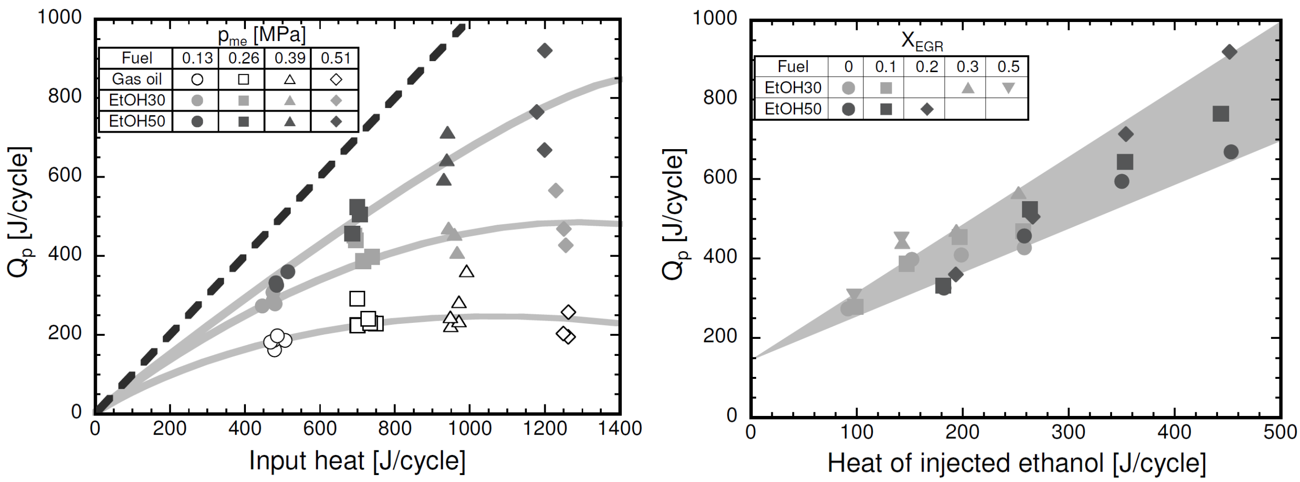


Fig.10 Change in ignition delay due to oxygen concentration in intake charge



(a) Q_p vs total amount of input heat

(b) Q_p vs amount of injected ethanol heat

Fig.11 Relationship between premixed combustion Q_p and amount of heat input

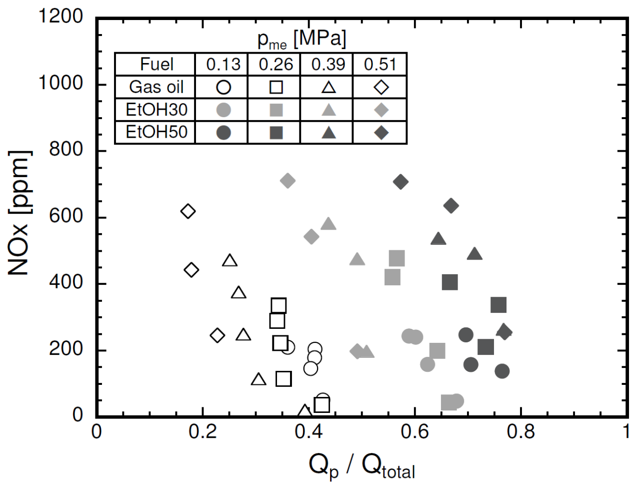


Fig.12 Effect of Q_p/Q_{total} on NOx

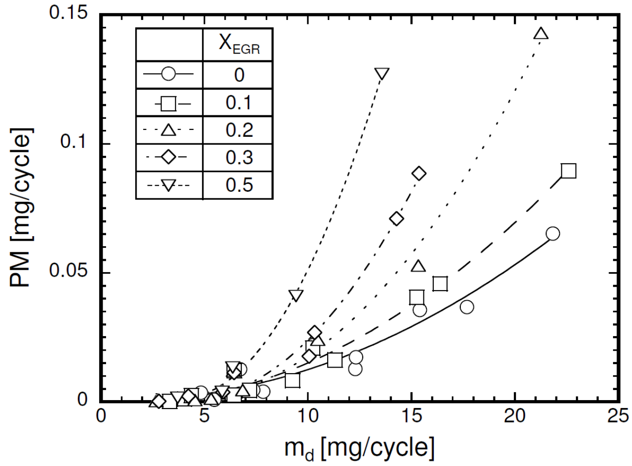


Fig.15 Relationship between PM, diffusion combustion mass m_d and EGR ratio X_{EGR}

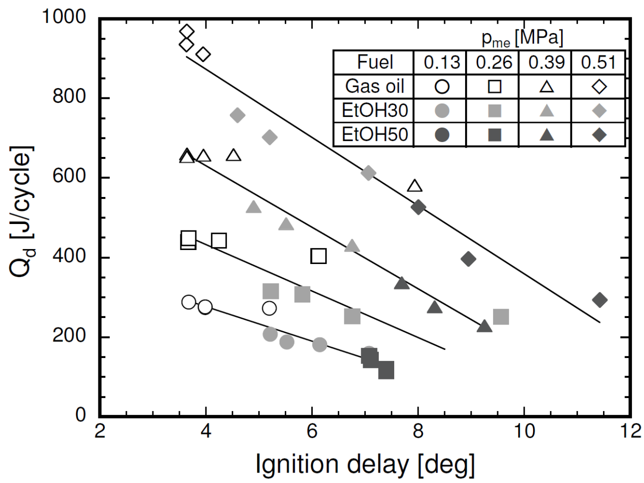


Fig.13 Change in cumulative heat release during diffusion combustion Q_d due to ignition delay

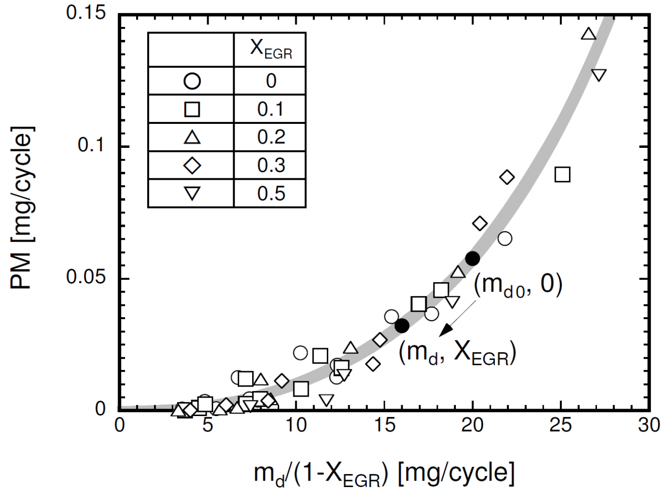


Fig.16 Correlation between PM and $m_d/(1-X_{EGR})$

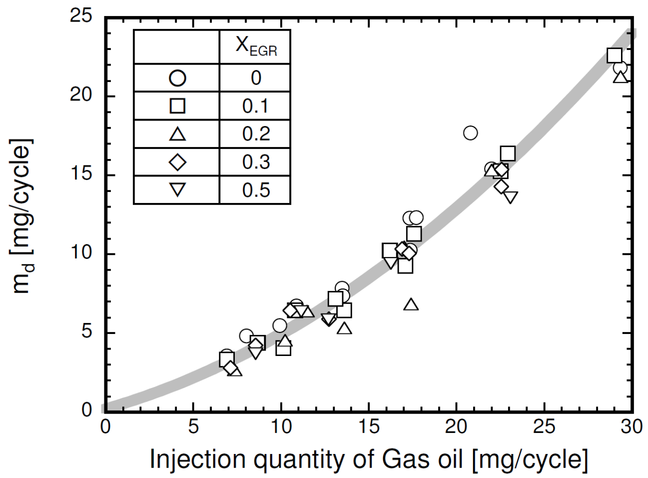


Fig.14 Correlation between diffusion combustion mass and injected amount of gas oil

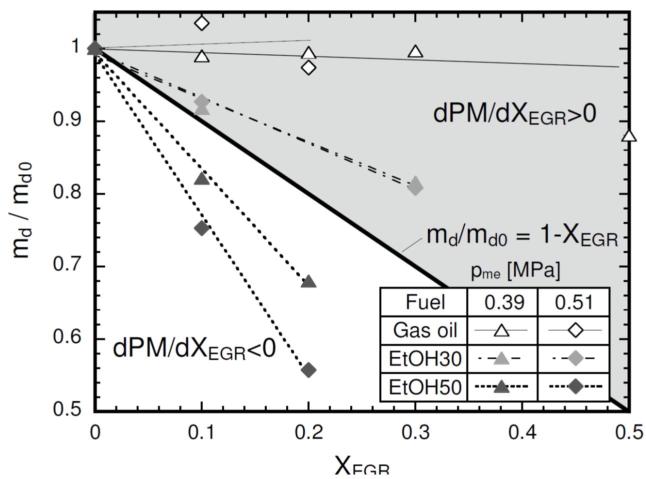


Fig.17 Change in normalized diffusion combustion mass m_d/m_{d0} due to EGR

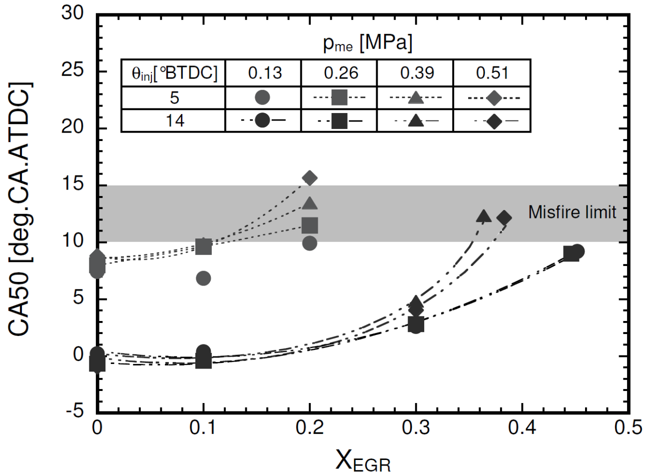


Fig.18 Change in 50% burnt point CA50 due to EGR

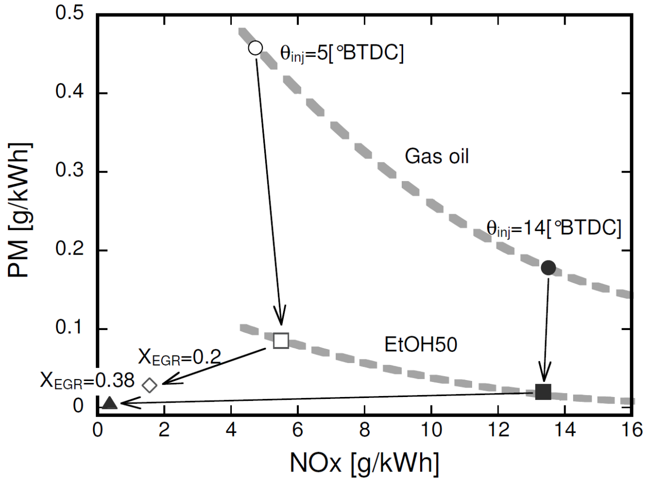


Fig.19 Improvement in PM-NOx trade-off due to ethanol blend and EGR