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(3-05) Effect of EGR and Preheating on Natural Gas Combustion Assisted with Gas-Oil in a Diesel Engine

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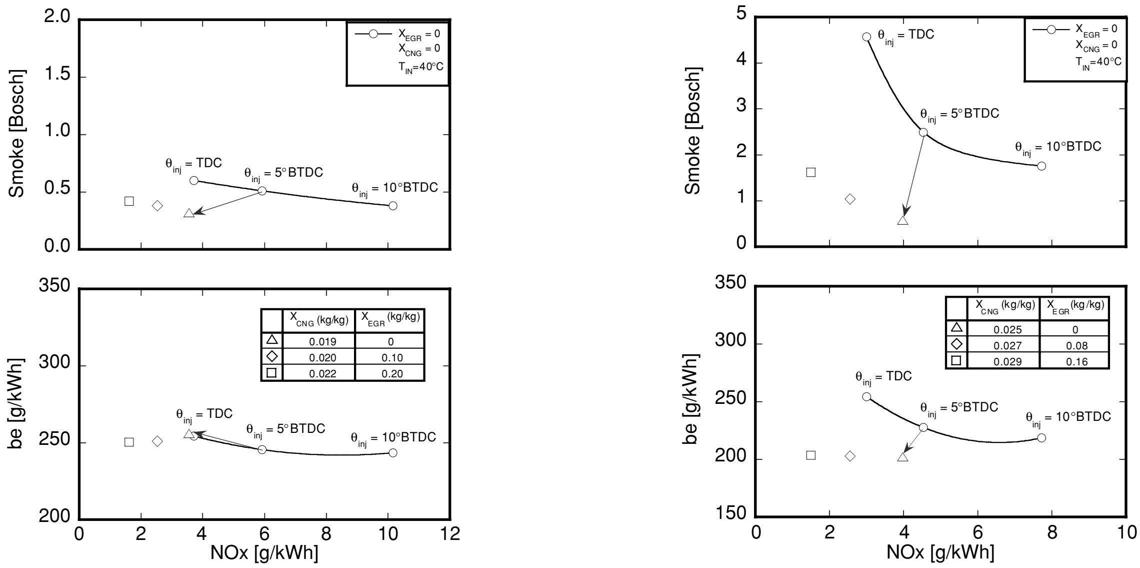
Key Words: Diesel Engine, Natural Gas, EGR, Preheating, Exhaust Emissions

ABSTRACT

In order to reduce NOx and smoke simultaneously and also to improve markedly the trade-off between smoke and NOx without deteriorating fuel consumption, natural gas was charged homogeneously into the intake air and was burned igniting by a small amount of gas oil injection in a four cylinder naturally-aspirated DI diesel engine. Combustion tests were carried out by changing the ratio of the amount of natural gas and the amount of gas oil first, secondarily the intake preheating temperature, and thirdly the EGR rate respectively. Effects of the respective parameter on the ignition and the burning rate of natural gas, exhaust emissions and specific fuel consumption were investigated. It is found that significant improvement of Smoke-NOx trade-off can be obtained without deteriorating fuel consumption by the suitable combination between the natural gas charge rate, the EGR rate and the intake preheating temperature for each engine load condition as shown in Fig.1. Concluding remarks are as follows; (1) A high burning rate of natural gas results in shortening the combustion duration, then, leads to lower fuel consumption. (2) Too high burning rate of natural gas results in increase of NOx, on the other hand, too low burning rate of natural gas results in increases of fuel consumption and THC. (3) Increases in fuel consumption and THC due to incomplete combustion of natural gas at the low load are improved drastically by raising the intake charge temperature up to 120°C, which increases the burning rate of natural gas. (4) Increase in NOx due to high burning rate of natural gas at the high load is improved by lowering the intake charge temperature below 60°C, which suppresses the burning rate of natural gas. (5) High EGR rate shows suppression effect on the burning rate at the high load, however, the burning rate is hardly affected by EGR at the low load. (6) NOx is reduced effectively by EGR without deteriorating fuel consumption, which is due to the water content brought by EGR as well as the decrease in oxygen concentration of the intake charge.



(a) Pme=0.33MPa, T_{IN}=120°C (b) Pme=0.66MPa, T_{IN}=45°C

Fig.1 Improved trade-off of dual fueled natural gas engine in combination with Preheating and EGR (θ_{inj}=5°BTDC)

INTRODUCTION

To meet extremely stringent emission standards in automotive engine field, extensive researches have been carried out to explore various ways to reduce NOx and particulate emissions from diesel engines. In various methods for reducing both exhaust NOx and smoke in diesel engines, utilizations of natural gas (Kusaka, et al.[1], Goto, et al.[2], Ishida, et al.[3]), dimethyl ether (Kajitani, S., et al.[4]), methanol (Ishida, et al.[5]) and gasoline/ gas-oil blend (Ishida, et al.[6]) have been examined for low emission vehicles. These fuels form the pre-mixture easily because they have a common feature of much lower evaporation temperature than gas oil, then, smoke can be reduced markedly. In these fuels, natural gas is expected most to be the alternative fuel from the viewpoint of its infrastructure, however, it is a key to successfully control both ignition and the burning rate of natural gas having a high ignition temperature.

The target of this study is to reduce NOx and smoke simultaneously and also to improve the trade-off relationship between NOx and smoke markedly in a diesel engine dual-fueled with natural gas. In the present experiment, the homogeneously charged natural gas was ignited by injecting a small amount of gas oil having a good ignitability but a high evaporation temperature. Combustion tests of the dual-fueled natural gas engine were carried out using a four cylindered naturally aspirated DI diesel engine under both low and high engine load conditions. The charge rate of natural gas in the intake air was changed first relative to the amount of gas oil injected directly into the cylinder. Secondarily, effects of the intake preheating temperature and the rate of exhaust gas recirculation “EGR” on ignition and the burning rate of natural gas were examined each independently. It will be shown that, in order to obtain marked improvement in the trade-off relationships between NOx and smoke and between NOx and fuel consumption, a suitable combination between the natural gas charge rate, the EGR rate and the intake preheating is required for each engine load condition.

EXPERIMENTAL APPARATUS

The test engine was a four-cylinder high-speed naturally aspirated direct injection diesel engine for automobiles, which was the type 4JB1 manufactured by ISUZU Motors Ltd.; 93 mm bore, 102 mm stroke, compression ratio of 18.2 and a maximum output of 64.7 kW(88 PS)/3,600 rpm. The special VE-type fuel injection pump was used to adjust the fuel injection timing arbitrarily and easily. The tested injector was the conventional multi-hole nozzle having four holes of 0.28 mm diameter, and the opening pressure of needle valve was set at 18.5 MPa. Furthermore, the port water injection system was built up on the intake manifold using gasoline injectors with the injection pressure of about 0.5 MPa, and water was injected into each suction port of a four cylinder engine (Ishida, et al.[7]).

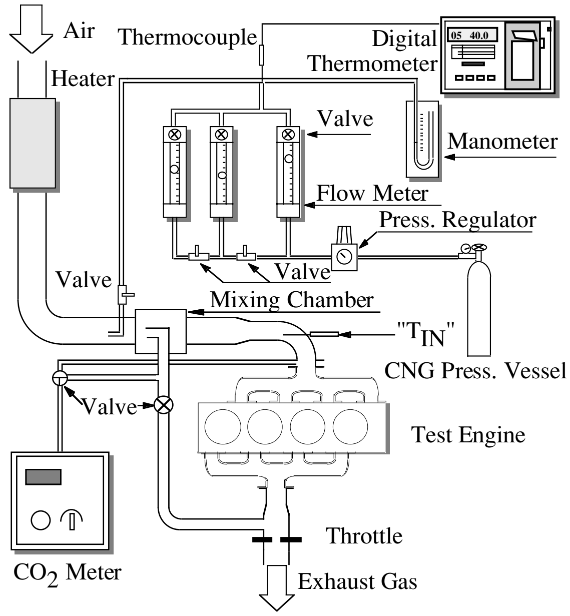


Fig.2 Natural gas supply system and EGR system

Table 1 Composition of tested natural gas (13A)

CH_4	C_2H_6	C_3H_8	C_4H_{10}	C_5H_{12}	H_2, N_2, CO_2
87.65	7.22	1.65	3.30	0.05	0.13 (%)

In the dual-fuel engine system shown in Fig.2, a gas mixer was installed in the intake system for obtaining a complete mixing of natural gas and/or the EGR gas with the intake fresh air, and the natural gas pre-mixture was ignited by diesel sprays injected directly into the cylinder. In this system, natural gas was stored in the vessel as the compressed natural gas “CNG” at the high pressure of about 25 MPa. CO_2 concentration was measured to calculate the EGR rate at the intake manifold and at the EGR return channel respectively as shown in Fig.2 The natural gas charge rate “X_{CNG} [kg/kg]” defined by the intake fresh dry air was varied from 0 to 0.032 which corresponds to the equivalence ratio “ϕ_{CNG}=16.86 X_{CNG}” from 0 to 0.54. The EGR rate “X_{EGR} [kg/kg]” defined by the total intake charge was varied between 0 and 0.20.

The test engine was operated at two kinds of constant brake mean effective pressure condition under a constant speed of 1,700 ± 5 rpm; Pme=0.33 MPa in the low load case and 0.66 MPa in the high load case. In the combustion tests of intake temperature variation and the EGR rate variation, the flow rate of natural gas was kept constant at 5,500 [liter/h] or G_{CNG}=2.37 [kg/h] in the low load case, and 8,300 [liter/h] or G_{CNG}=3.55 [kg/h] in the high load case respectively so as to obtain nearly equal weight flow ratio of 80/20 between natural gas and gas oil in both loads, while the gas oil injection rate was adjusted to obtain the constant brake mean effective pressure. The intake charge temperature “T_{IN}” was measured at the engine inlet, and it was adjusted to the specified temperature ± 0.5°C by using the electric heater even in the case with EGR. The suction air pressure at the engine inlet was also adjusted and fixed at the standard atmospheric pressure of 0.1013 MPa by using the motor-driven blower.

Table 1 shows the composition of tested natural gas. The natural gas is the urban gas fuel named "13A", and gas oil used for igniting natural gas is a ordinary one having a cetane index of 57. The net calorific values of these fuels are 49.12 and 42.91 [MJ/kg] respectively. Furthermore, fuel consumption "be [g/kWh]" shown in the following figures is the reduced one based on the net calorific value of gas oil. The time histories of combustion pressure, fuel nozzle pressure and the needle valve lift were measured using the respective sensors, and those outputs were sampled every one-fourth degree of crank angle simultaneously by means of the four-channel combustion analyzer "CB-467" manufactured by Ono Sokki Co. Ltd. The measured time histories of the experimental results are the ensemble average sampled over continuous 350 engine cycles. Those data were transmitted to the personal computer and recorded on floppy disks.

RESULTS AND DISCUSSION

Effect of CNG Charge on Combustion

Figures 3(a) and (b) and Figs.4(a) and (b) show changes due to the CNG charge rate in time histories of the in-cylinder pressure "P [MPa]", the apparent heat release rate "dQ/dθ [J/deg]" and the needle valve lift "Lift [mm]" which indicates the injection duration of gas oil. Figs.3(a) and (b) are the cases of two different intake temperature in the low load, and Figs.4(a) and (b) are those in the high load.

In the low load cases with the intake temperature of 80 and 120°C, as the CNG rate increases, the maximum heat release rate in the initial combustion stage decreases significantly in spite of the increase in ignition delay. This might be based on that the amount of gas oil injected becomes smaller as the CNG rate increases, and the lean mixture of natural gas burns very slowly. As a result, the maximum combustion pressure decreases markedly, which results in a marked reduction in NOx. The second peak of the heat release curve is based on combustion of natural gas, and it becomes higher in the high intake temperature case compared with the first peak at a large CNG rate. Furthermore, as mentioned in the preceding chapter, the brake mean effective pressure was kept

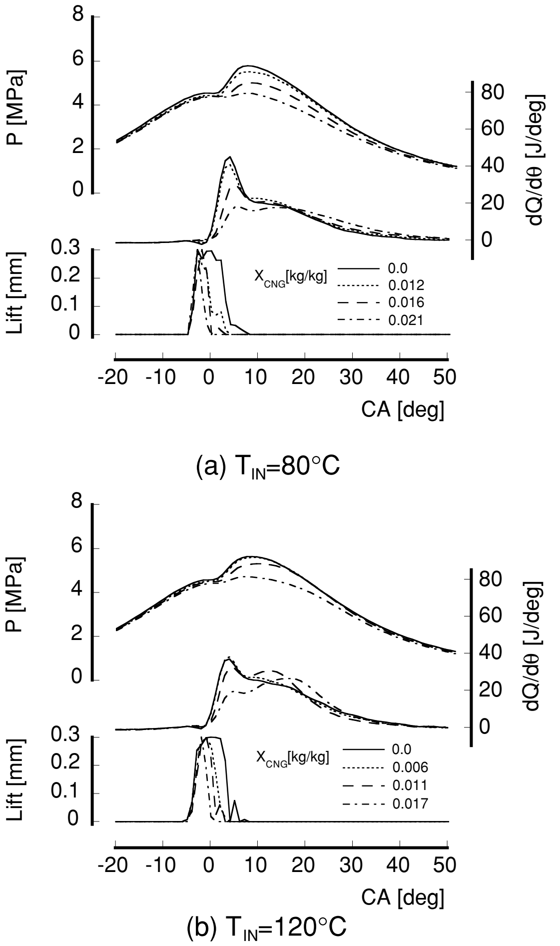


Fig.3 Change in combustion history due to CNG rate (Pme=0.33 MPa, θ_{inj}=5°BTDC, X_{EGR}=0)

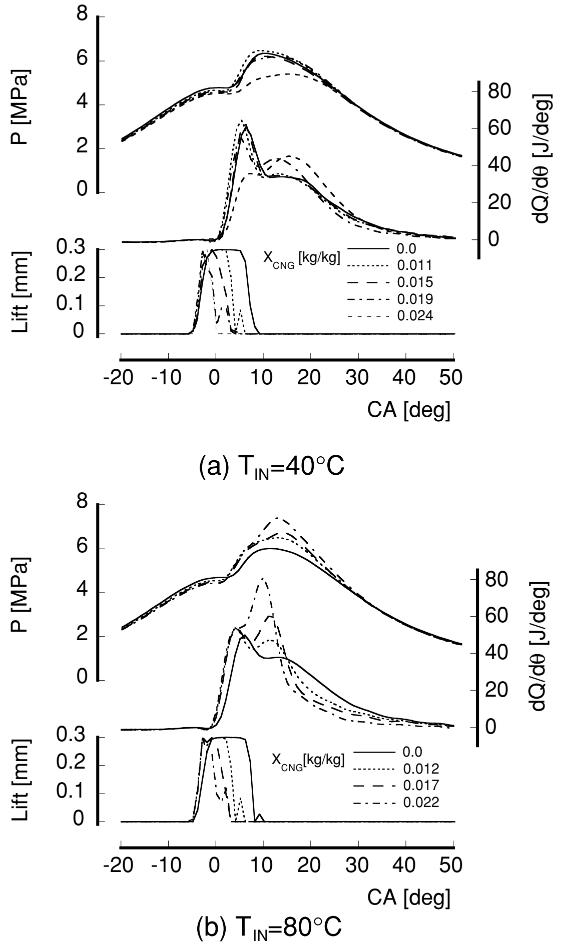
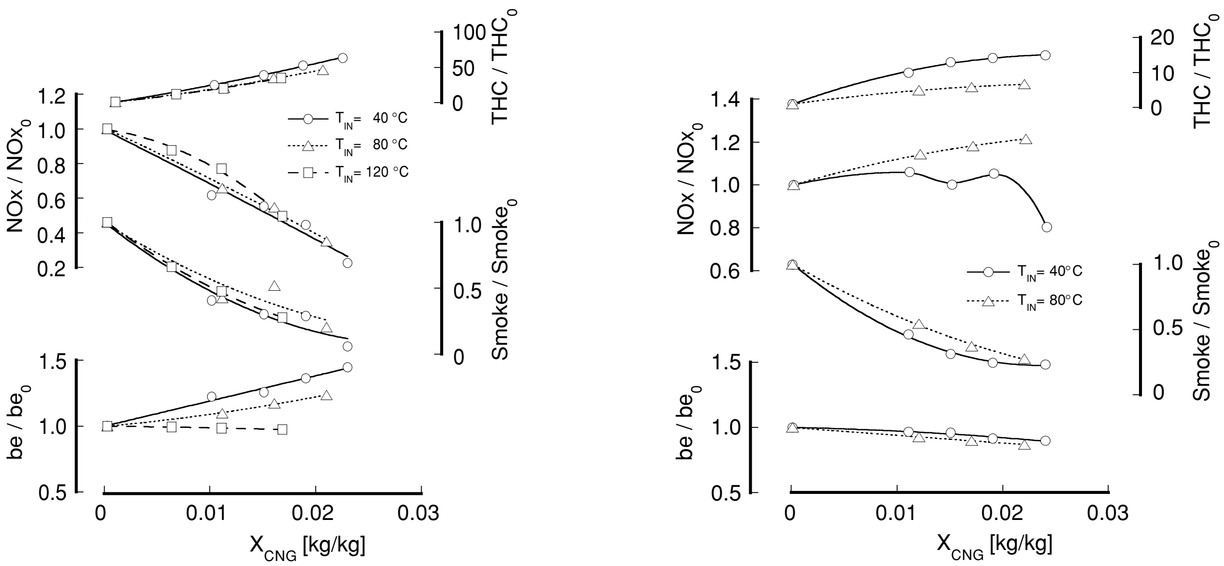


Fig.4 Change in combustion history due to CNG rate (Pme=0.66 MPa, θ_{inj}=5°BTDC, X_{EGR}=0)

constant in the present experiment by adjusting the gas oil injection rate in each case when the CNG rate was changed stepwise. However, the indicated mean effective pressure seems to be decreased when the CNG rate is increased, as seen in the pressure history curves measured at the No.1 cylinder under the low load condition shown in Figs.3(a) and (b). It might be based on the increased deviation of combustion between four cylinders in the cases of large CNG rates.

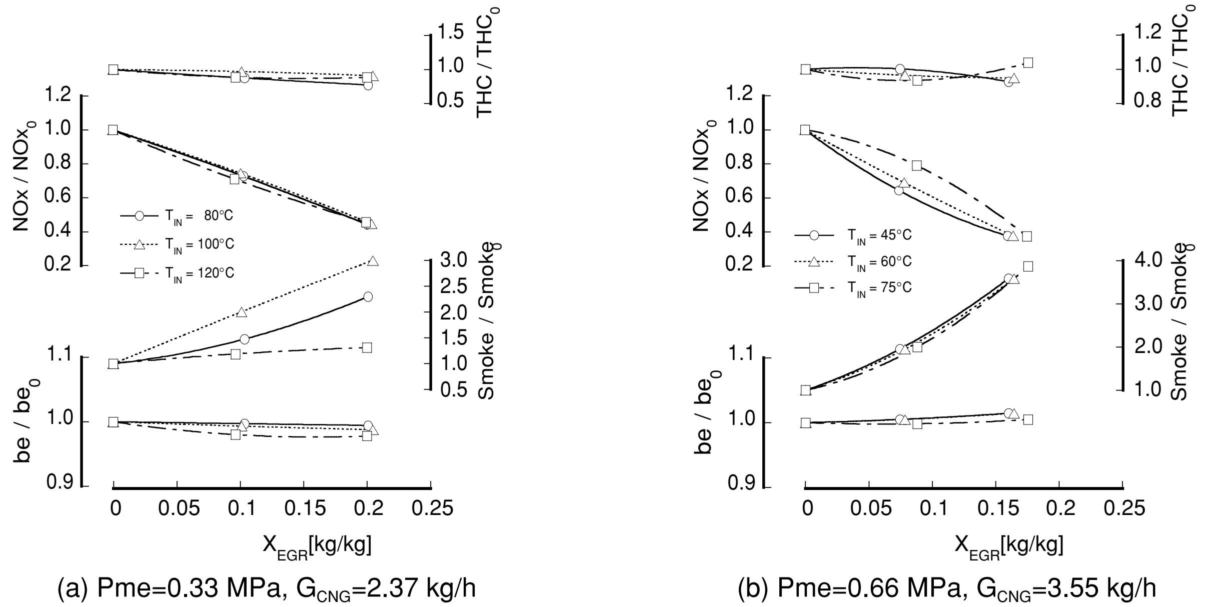
In the high load cases with the intake temperature of 40 and 80°C, ignition delay seems to decrease as the CNG rate increases especially in the high intake temperature case shown in Fig.4(b). The heat release rate in the initial combustion is almost unchanged due to the CNG rate in both intake temperature cases except for the case of X_{CNG}=0.024 and T_{IN}=40°C alone. As seen in Figs.4(a) and (b), the natural gas is ignited after the first peak of the initial combustion, and the natural gas burns much faster in the high load in comparison with the low load judging from the second peak of the heat release rate curve, which results in a marked increases in the maximum combustion pressure and combustion temperature. Obviously the burning rate is higher under the higher intake temperature condition, thus, the higher burning rate results in shortening the combustion duration and leads to lower fuel consumption.

Figures 5(a) and (b) show the effect of the CNG charge rate on exhaust emissions and fuel consumption, where the parameter is the intake temperature. Each ordinate is normalized by the value measured under the zero CNG condition. In both cases of low and high loads, smoke is reduced remarkably by about 80% when the



(a) Pme=0.33 MPa (T_{IN}=40, 80, 120°C) (b) Pme=0.66 MPa (T_{IN}=40, 80°C)

Fig.5 Change in fuel consumption and exhaust emissions due to CNG rate (θ_{inj}=5°BTDC, X_{EGR}=0)



(a) Pme=0.33 MPa, G_{CNG}=2.37 kg/h
θ_{inj}=5°BTDC (T_{IN}=80, 100, 120°C)

(b) Pme=0.66 MPa, G_{CNG}=3.55 kg/h
θ_{inj}=5°BTDC (T_{IN}=45, 60, 75°C)

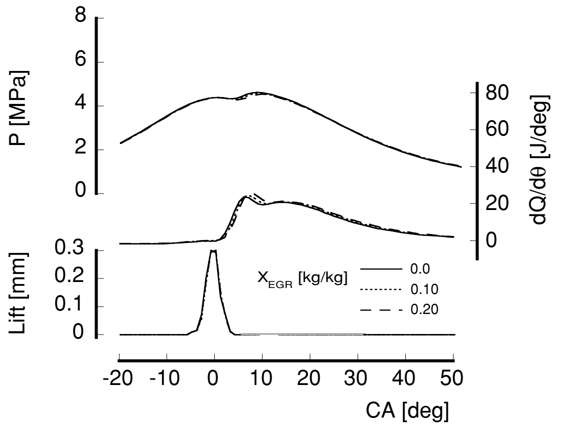
Fig.6 Change in fuel consumption and exhaust emissions due to EGR under constant CNG charge condition

CNG rate is 0.023 [kg/kg] (ϕ_{CNG}=0.39) . NOx is decreased markedly by about 80% at the low load because of the marked decrease in combustion pressure resulting in a decrease in combustion temperature. NOx is increased, on the other hand, at the high load with high intake temperature because combustion pressure increases due to the high burning rate of natural gas. With respect to fuel consumption, it increases markedly with the CNG rate in the low load case with the low intake temperature of 40°C, which is mainly due to an incomplete combustion of natural gas as seen by the large increase in the unburned hydrocarbon emission. However, by raising the intake temperature up to 120°C, the increase in fuel consumption due to CNG is suppressed even in the case of low load. This indicates that fuel consumption can be improved by the increase of intake temperature as suggested by Kusaka, et al.[1]. On the other hand, at the high load, fuel consumption decreases with the CNG rate slightly even in the case of low intake temperature.

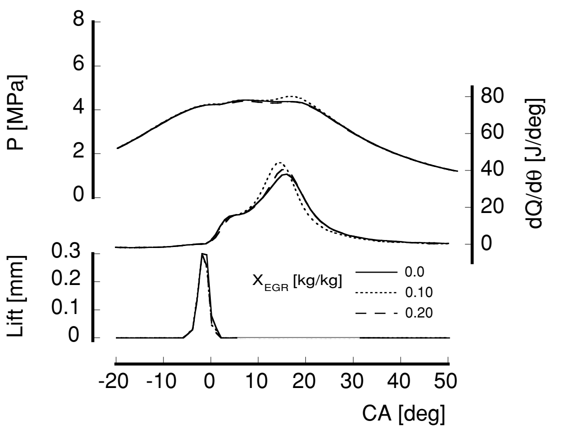
Effect of EGR on Combustion

Figures 6(a) and (b) show the effect of the EGR rate on exhaust emissions and fuel consumption under the constant CNG charge condition; G_{CNG}=2.37 [kg/h] in the low load which corresponds to ϕ_{CNG}=0.30~0.37 and G_{CNG}=3.55 [kg/h] in the high load which corresponds to ϕ_{CNG}=0.42~0.54 respectively. The parameter is the intake charge temperature. NOx is significantly reduced almost linearly with an increase in the EGR rate, and it is almost independent on the intake temperature. NOx is reduced by 50% in the low load when the EGR rate is 20%, and it is 60% in the high load only by the EGR rate of 16%, that is, the reduction rate of NOx due to EGR is larger in the high load than the low load. This is due to the fact that the water vapor concentration in the exhaust gas is about twice in the high load compared with the low load if the EGR rate is equal in both loads. The large reduction in NOx is, judging from the Zel'dovich mechanism on NO formation, based on the water content included in the EGR gas at first, and secondarily the decrease of oxygen content in the intake charge with EGR. Furthermore, it was shown by the authors[3] that the NOx reduction rate due to EGR is about twice of the one due to the port water injection[7].

On the other hand, smoke increases with the EGR rate at both low and high loads, however, it is noticed that the increase in smoke due to EGR is very small in the low load case with the high intake temperature of 120°C. With respects to fuel consumption and the unburned

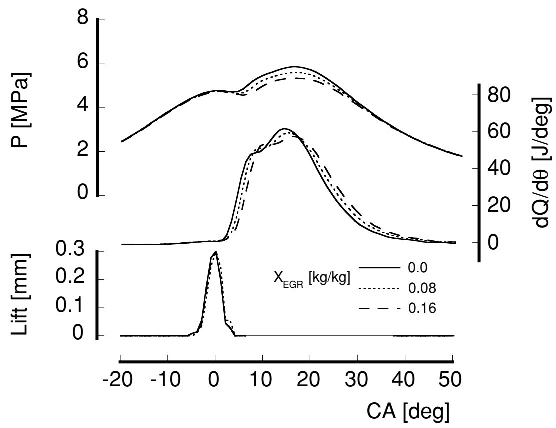


(a) T_{IN}=80°C (X_{EGR}=0, 0.10, 0.20)

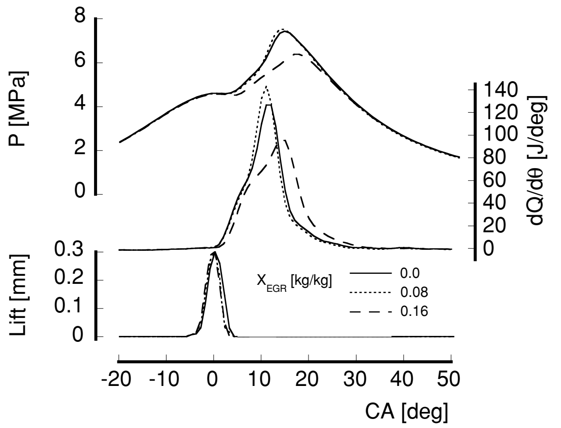


(b) T_{IN}=120°C (X_{EGR}=0, 0.10, 0.20)

Fig.7 Change in combustion history due to EGR under constant CNG charge condition (Pme=0.33MPa, G_{CNG}=2.37 kg/h, θ_{inj}=5°BTDC)

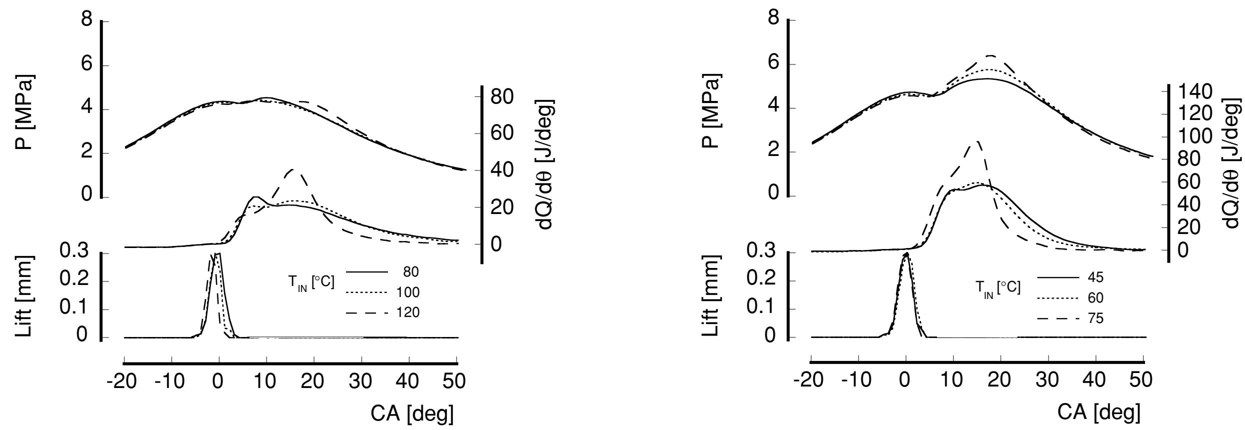


(a) T_{IN}=45°C (X_{EGR}=0, 0.08, 0.16)



(b) T_{IN}=75°C (X_{EGR}=0, 0.08, 0.16)

Fig.8 Change in combustion history due to EGR under constant CNG charge condition (Pme=0.66MPa, G_{CNG}=3.55 kg/h, θ_{inj}=5°BTDC)



(a) Pme=0.33MPa, G_{CNG}=2.37 kg/h, X_{EGR}=0.20
θ_{inj}=5°BTDC (T_{IN}=80, 100, 120°C)

(b) Pme=0.66MPa, G_{CNG}=3.55 kg/h, X_{EGR}=0.16
θ_{inj}=5°BTDC (T_{IN}=45, 60, 75°C)

Fig.9 Change in combustion history due to Preheating under constant CNG charge condition

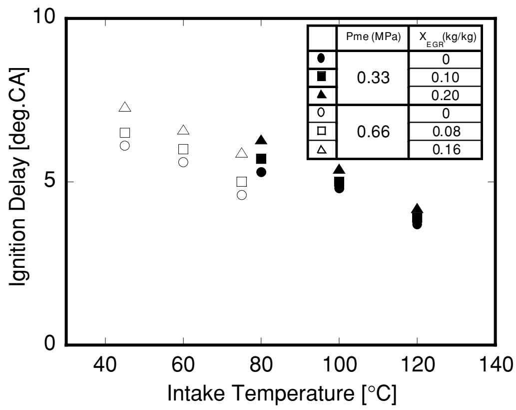


Fig.10 Change in ignition delay due to intake temperature and EGR

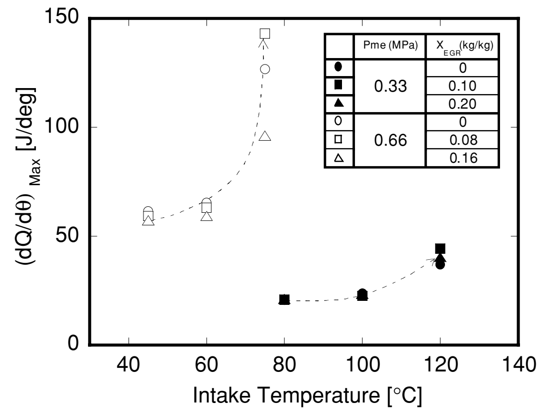


Fig.11 Effect of intake temperature on burning rate of natural gas

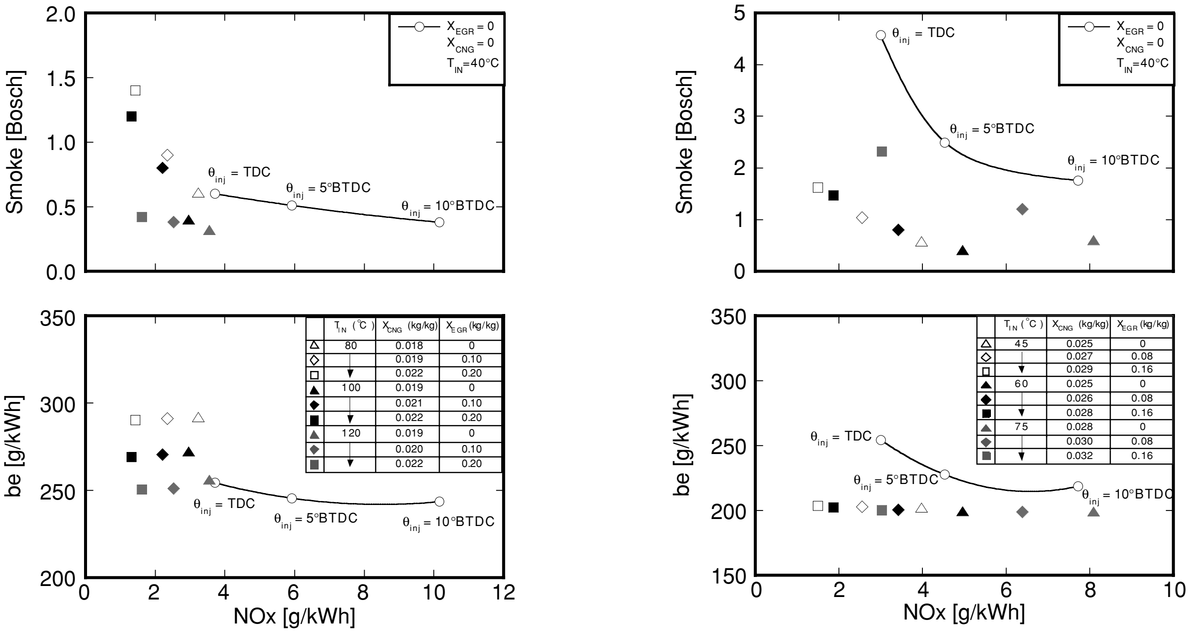
hydrocarbon emission, their variations due to EGR are small at both low and high loads.

Figures 7(a), (b) and Figs.8(a), (b) show changes in time histories of combustion due to EGR under the constant CNG charge condition of G_{CNG}=2.37 [kg/h] (ϕ_{CNG}=0.30~0.37) in the low load and G_{CNG}=3.55 [kg/h] (ϕ_{CNG}=0.42~0.54) in the high load respectively. Fig.(a) shows the case of low intake temperature, Fig.(b) is that of high intake temperature, and the parameter is the EGR rate. In the cases of low intake temperature under both low and high loads, as shown in Fig.7(a) and Fig.8(a), the second peak of the heat release curve becomes slightly lower by EGR, resulting in lower maximum combustion pressure. On the other hand, in the cases of high intake temperature shown in Fig.7(b) and Fig.8(b), the second peak of the heat release curve, which seems to be the maximum burning rate of natural gas, is suppressed by EGR with the rate of about 20%, and it is observed obviously at the high load shown in Fig.8(b). However, it is interesting that the burning rate of natural gas is hastened slightly when the EGR rate is about 10%. Furthermore, ignition delay tends to increase slightly by EGR.

Effect of Intake Preheating on Combustion

Figures 9(a) and (b) show changes in time histories of combustion due to the intake charge temperature under the constant CNG and EGR rates condition; Fig.9(a) shows the case of low load (ϕ_{CNG}=0.37), Fig.9(b) is that of high load (ϕ_{CNG}=0.49~0.54) respectively, and the parameter is the intake temperature. Figures 10 and 11 show changes due to the intake charge temperature in ignition delay and the second peak of the heat release rate curve which represents the maximum burning rate of natural gas, in which the parameter is the EGR rate.

By raising the intake charge temperature, the maximum burning rate of natural gas is increased drastically between 100 and 120°C in the low load case, and it is between 60 and 75°C in the high load case as shown in Figs.9 and 11. In the case of low load, the intake charge temperature has to be raised up to about 120°C in order to hasten the burning rate of natural gas for suppression of increases in the unburned hydrocarbon and fuel consumption based on incomplete combustion of natural gas. In the case of high load, on the other hand, the intake charge temperature has to be lowered below 60°C in order to suppress the burning rate of natural gas



(a) Pme=0.33MPa (θ_{inj}=5°BTDC)

(b) Pme=0.66MPa (θ_{inj}=5°BTDC)

Fig.12 Improvement of trade-off due to EGR and Preheating under constant CNG charge condition

for avoiding diesel knock. As shown in Fig.10, ignition delay decreases with the intake charge temperature rise, and increases slightly with the EGR rate. The effects of intake temperature and EGR on ignition are similar to the results shown by Nakano, et al.[8]. Furthermore, ignition delay is shorter a little in the high load than the low load if the intake temperature is equal.

Improvement of Trade-Off

Figures 12(a) and (b) show changes in trade-off relationships between smoke and NOx, and between fuel consumption k.d NOx due to CNG charge, EGR and intake preheating at the low and high loads respectively. In the figures, the solid line with open circle marks denotes the original trade-off of the test engine operated by gas oil alone, where injection timing is varied from 10°BTDC to TDC, and the other data shown here are obtained under the injection timing of 5°BTDC. In the combustion tests of intake temperature variation and EGR rate variation, the weight flow rate of natural gas relative to that of gas oil was almost kept constant aiming at 80/20; the flow rate of natural gas was about G_{CNG}=2.37 [kg/h] (ϕ_{CNG}=0.30~0.37) in the low load case, and it was about G_{CNG}=3.55 [kg/h] (ϕ_{CNG}=0.42~0.54) in the high load case respectively as mentioned before, where the CNG charge rate “ X_{CNG}” varies slightly depending on the intake temperature and the EGR rate because of a change in the intake air flow rate.

In the low load case, the large reduction in NOx is obtained first by the CNG charge as shown by the difference between the open circle mark and the triangle marks in Fig.12(a), and the further reduction in NOx is obtained by combining with EGR. However, smoke is increased by EGR under the low intake temperature condition, then, the intake preheating up to 120°C is required for the low smoke and low fuel consumption.

In the high load case, smoke is reduced markedly by CNG, however, NOx is not done because of too high burning rate of natural gas. In order to suppress the high burning rate of natural gas, the intake temperature lower than 60°C is necessary. The low enough level of NOx is obtained in combination with EGR and the low intake temperature, however, too high EGR rate leads to deterioration of smoke.

As a result, the trade-off between smoke and NOx is improved without deteriorating fuel consumption by the suitable combination between the CNG charge rate, the intake preheating and the EGR rate in each load. In the low load case, the EGR rate of 20% and the intake temperature of 120°C is suitable as shown by the gray square mark in Fig.12(a), and, in the high load case, the intake temperature below 60°C and the EGR rate of 16% is suitable as shown by the open and solid square marks. Furthermore, these conditions might be suitable only for the present test engine, and they seem to be changed by the injection timing of gas oil and also the CNG charge rate, in other words, the ratio between natural gas and gas oil. However, the present concept is applicable in most dual-fueled natural gas diesel engines.

CONCLUSION

In order to improve significantly the trade-off between smoke and NOx without deteriorating fuel consumption in a DI diesel engine, homogeneously charged natural gas was burned igniting by a small amount of gas-oil injection. The effects of the natural gas charge rate, the intake preheating temperature and the EGR rate on ignition and burning rate of natural gas as well as exhaust emissions and fuel consumption were investigated experimentally. Concluding remarks are as follows.

(1) Significant improvement of Smoke-NOx trade-off was obtained without deteriorating fuel consumption by the suitable combination between the natural gas charge rate, the EGR rate and the intake preheating temperature for each engine load condition.

(2) A high burning rate of natural gas results in shortening the combustion duration and leads to lower fuel consumption.

(3) Too high burning rate of natural gas results in increase of NOx, on the other hand, too low burning rate of natural gas results in increases of fuel consumption and THC.

(4) Increases in fuel consumption and THC due to incomplete combustion of natural gas at the low load are improved drastically by raising the intake charge temperature up to 120°C, which increases the burning rate of natural gas.

(5) Increase in NOx due to the high burning rate of natural gas at the high load is improved by lowering the intake charge temperature below 60°C, which suppresses the burning rate of natural gas.

(6) A high EGR rate shows suppression effect on the burning rate at the high load, however, the burning rate is hardly affected at the low load by EGR.

(7) NOx is reduced effectively by EGR without deteriorating fuel consumption based on the water content brought by EGR and the decrease in oxygen concentration of the intake charge.

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NOMENCLATURE

be	Brake specific fuel consumption [g/kWh]
Lift	Needle valve lift [mm]
NOx	Brake specific nitrogen oxides emission [g/kWh]
P	Cylinder pressure [MPa]
Pme	Brake mean effective pressure [MPa]
$dQ/d\theta$	Heat release rate [J/deg]
Smoke	Smoke density [Bosch]
T_{IN}	Intake charge temperature [°C]
THC	Total unburned hydrocarbon emission [g/kWh]
X_{CNG}	CNG charge rate ($=G_{CNG}/G_{AIR}$) [kg/kg]
X_{EGR}	EGR rate ($=G_{EGR}/G_{INTAKE\text{-}CHARGE}$) [kg/kg]
θ_{inj}	Injection timing of gas oil [° Crank angle]

Subscript

0 Operation condition by gas oil without CNG

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