Effect of EGR and Preheating on Natural Gas Combustion Assisted with Gas-Oil in a Diesel Engine*

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In order to reduce NO_x and smoke simultaneously and also to improve markedly the trade-off between smoke and NO_x 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 clarified experimentally. It is found that significant improvement of smoke-NO_x trade-off can be obtained without deteriorating fuel consumption by the suitable combination between the natural gas charge rate, the intake preheating temperature and the EGR rate for each engine load condition.

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

1. 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 easily form the pre-mixture 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 an 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 between NOx and smoke markedly in a diesel engine. In the present experiment, the homogeneously charged natural gas was ignited by injecting the amount of gas oil having a good ignitability but a high evaporation temperature. Various combustion tests on 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 was increased first while the amount of gas oil decreased to keep the engine output equal. 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 was found that the suitable combination for each engine load between the natural gas charge rate, the EGR rate and the intake preheating resulted in a marked improvement in the trade-off between smoke and NOx and between fuel consumption and NOx.

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Nomenclatures

be : Brake specific fuel consumption [g/kWh]
$Lift$: Needle valve lift [mm]
NOx : Brake specific nitrogen oxides emission [g/kWh]
P : In-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

2. Experimental Apparatus

The test engine was a four-cylinder high-speed naturally aspirated direct injection diesel engine for automobiles, which was the type 4JB1-2 manufactured by ISUZU Motors Ltd.; 93 mm bore, 102 mm stroke and a compression ratio of 18.2. The special VE-type fuel injection pump was used for adjusting the fuel injection timing arbitrarily. 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.

In the dual-fuel engine system shown in Fig.1, 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, the compressed natural gas "CNG" was stored in the vessel at high pressure of 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.1. 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 from 0 to 0.20.

The test engine was operated under two kinds of constant brake mean effective pressure at 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, the CNG charge rate was increased first, then, the intake temperature and the EGR rate was changed under the constant CNG charge rate of 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, while the gas oil injection rate was adjusted to obtain the respective brake mean effective pressure. In both low and high loads, the weight flow ratio between natural gas and gas oil was approximately equal to 80/20 in maximum. 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 the 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

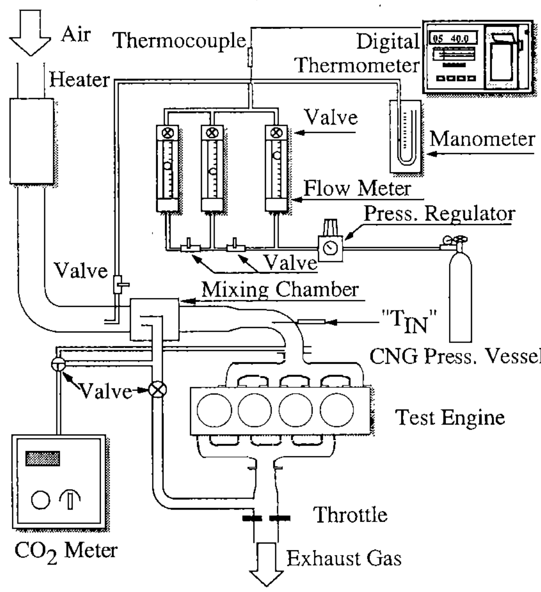


Fig. 1 Natural gas supply system and EGR system

Table 1 Composition of tested natural gas (13A)

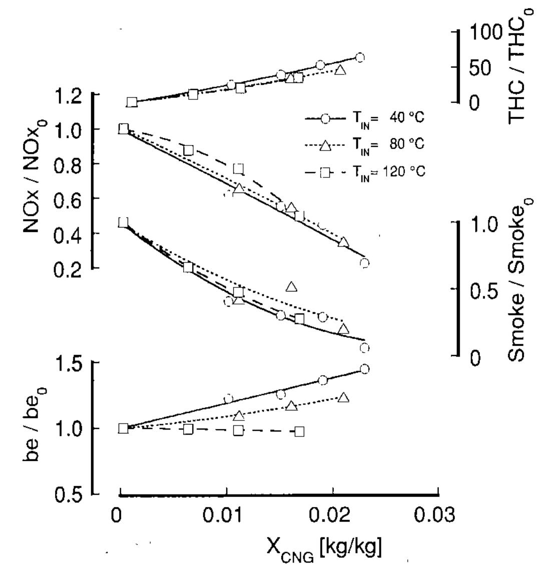
CH_4	C_2H_6	C_3H_8	C_4H_{10}	C_5H_{12}	H_2, N_2, CO_2
87.65	7.22	1.65	3.30	0.05	0.13 (%)

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.

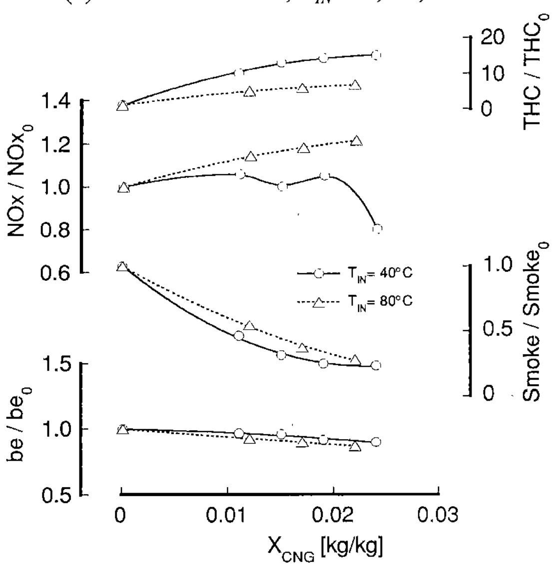
3. Results and Discussion

3.1 Effect of CNG Charge on Combustion

Figures 2(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 X_{CNG}= 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. On the other hand, NOx is increased at the high load case with the high intake temperature of 80°C 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 resulting in 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

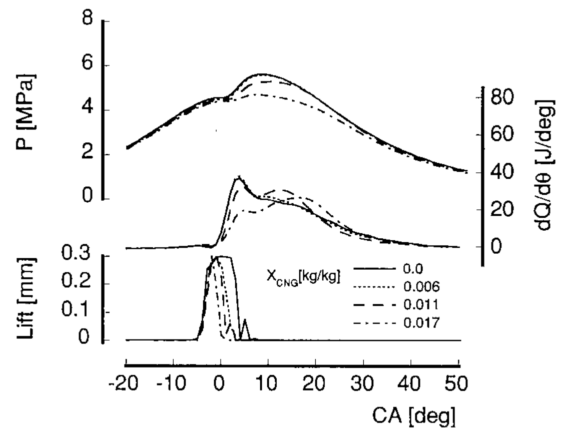


(a) Pme=0.33 MPa, T_{IN}=40, 80, 120°C

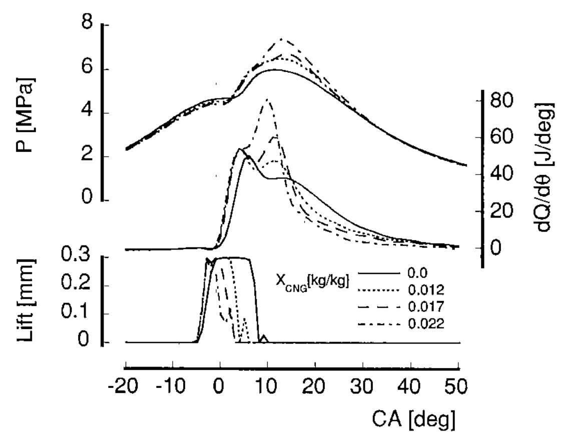


(b) Pme=0.66 MPa, T_{IN}=40, 80°C

Fig. 2 Change in fuel consumption and exhaust emissions due to CNG rate (θ_{inj}=5°BTDC, X_{EGR}=0)



(a) Pme=0.33 MPa, T_{IN}=120°C



(b) Pme=0.66 MPa, T_{IN}=80°C

Fig. 3 Change in combustion history due to CNG rate (θ_{inj}=5°BTDC, X_{EGR}=0) decreases with the CNG rate slightly even in the case of low intake temperature.

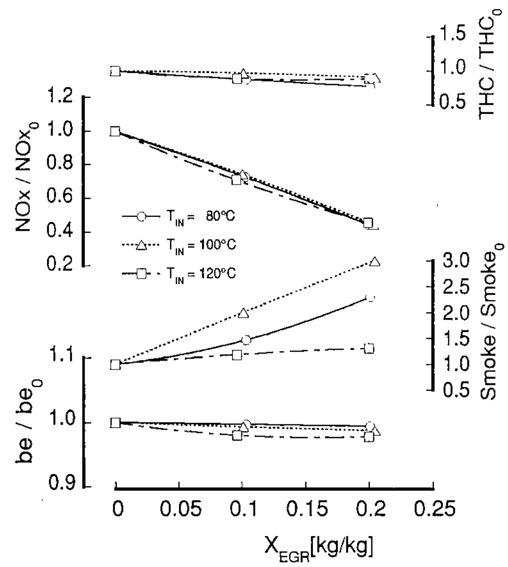
Figures 3(a) and (b) show examples of time-history change due to the CNG charge rate in the in-cylinder pressure "*P*" measured at No.1 cylinder, the apparent heat release rate "$dQ/d\theta$" and the needle valve lift "*Lift*" which indicates the injection duration of gas oil.

In the low load case with the intake temperature of 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, then, the marked reduction in NOx results. The second peak of the heat release curve is based on combustion of natural gas, and it becomes higher than the first peak at the large CNG rate in the high intake temperature case of 120°C. Furthermore, the indicated mean effective pressure seems to be decreased when the CNG rate is increased although the brake mean effective pressure is kept constant in the present experiment by adjusting the gas oil injection rate in each case. It is mainly due to the increased deviation of combustion between four cylinders in the cases of large CNG rates.

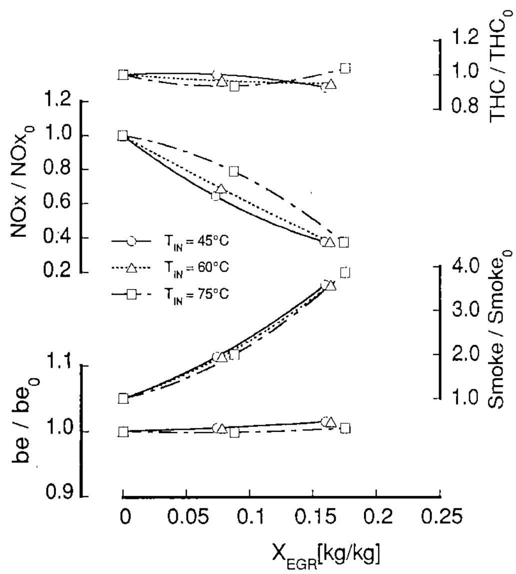
In the high load case with the intake temperature of 80°C, as the CNG rate increases, ignition delay tends to decrease slightly and the heat release rate in the initial combustion is almost unchanged. It is clearly seen that the natural gas is ignited after ignition of gas oil, and the natural gas burns much faster in the high load compared with the low load case comparing the second peak of the heat release rate curve between Figs.3(a) and (b). The higher burning rate results in a marked increases in the maximum combustion pressure and NOx, on the other hand, it results in shortening the combustion duration which leads to lower fuel consumption.

3.2 Effect of EGR on Combustion

Figures 4(a) and (b) show the effect of EGR on exhaust emissions and fuel consumption under the constant CNG charge condition; 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. 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%. The reduction rate of NOx due to EGR is larger in the high load than the low load. This is mainly due to the fact that the water



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

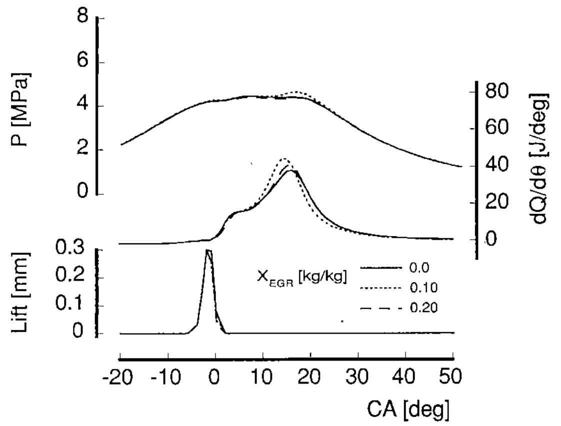


(b) Pme=0.66 MPa, G_{CNG}=3.55 kg/h, θ_{inj}=5°BTDC, T_{IN}=45, 60, 75°C

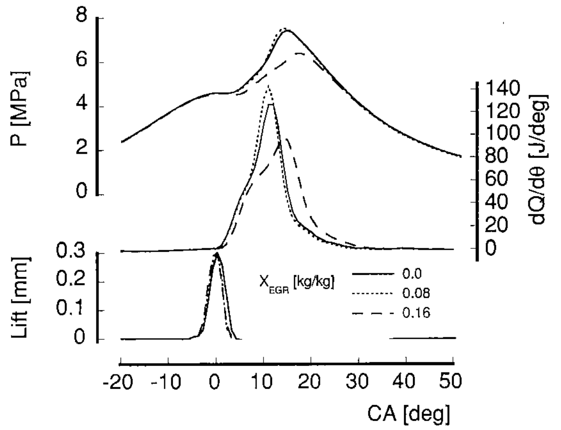
Fig.4 Change in fuel consumption and exhaust emissions due to EGR under constant CNG charge condition

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. 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. It is also seen that fuel consumption and the unburned hydrocarbon emission hardly increase by EGR at both low and high loads.

Figures 5(a) and (b) show changes in time history of combustion due to EGR at the high intake temperature cases under the constant CNG charge rate condition of

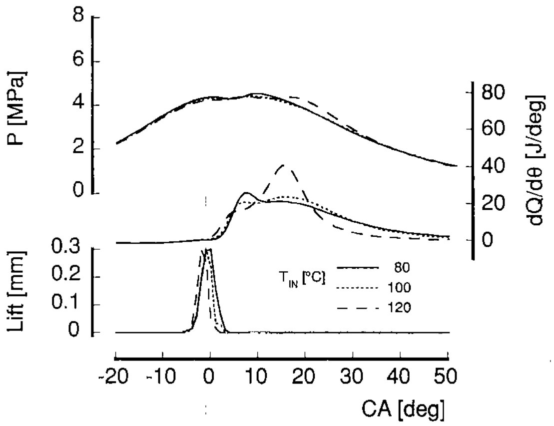


(a) Pme=0.33 MPa, G_{CNG}=2.37 kg/h, θ_{inj}=5°BTDC, T_{IN}=120°C, X_{EGR}=0, 0.10, 0.20

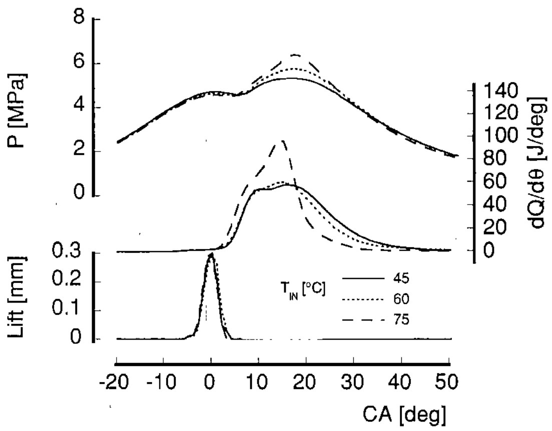


(b) Pme=0.66 MPa, G_{CNG}=3.55 kg/h, θ_{inj}=5°BTDC, T_{IN}=75°C, X_{EGR}=0, 0.08, 0.16

Fig.5 Change in combustion history due to EGR under constant CNG charge condition



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



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

Fig. 6 Change in combustion history due to preheating under constant CNG charge condition

G_{CNG}=2.37 or 3.55 [kg/h] in each load, in which the parameter is the EGR rate. It can be seen in these figures that the second peak value of the heat release curves, which is the maximum burning rate of natural gas, becomes larger once at the small EGR rate, then, it is suppressed by the large EGR rate. It is interesting that the burning rate of natural gas is hastened slightly if the EGR rate is about 10%. Ignition delay increases slightly by EGR in the high load but does not in the low load.

3.3 Effect of Intake Preheating on Combustion

Figures 6(a) and (b) show changes in time-history of combustion due to the intake temperature under the constant CNG and EGR rates condition; Fig.6(a) shows the case of low load (ϕ_{CNG}=0.37), Fig.6(b) is that of high load (ϕ_{CNG}=0.49~0.54) respectively, and the parameter is the intake temperature. Figure 7(a) shows change in ignition delay due to the intake temperature and the EGR rate, and Fig.7(b) also shows change in ignition delay due to the in-cylinder mean gas temperature at the ignition timing; where the in-cylinder mean gas temperature was calculated from the measured in-cylinder pressure by using the equation of state, and taking into consideration of gas composition change based on CNG charge and EGR. Figure 8 shows change in the second peak value of the heat release rate which represents the maximum burning rate of natural gas.

By raising the intake 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.6 and 8. In the case of low load, the intake temperature must be raised up to about 120°C in order to increase the burning rate of natural gas, in other words, in order to reduce fuel consumption due to increase in the unburned hydrocarbon. In the case of high load, on the other hand, the intake temperature must be lowered

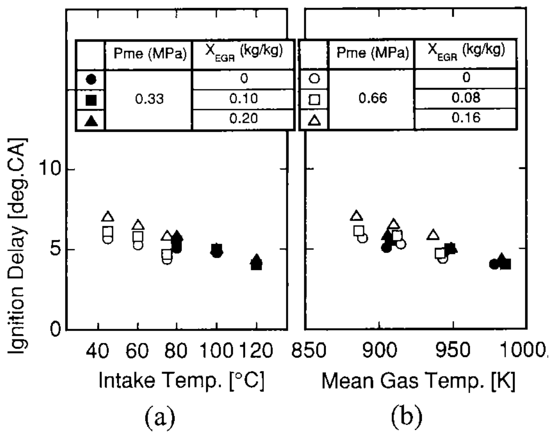


Fig.7 Change in ignition delay due to intake temperature and EGR

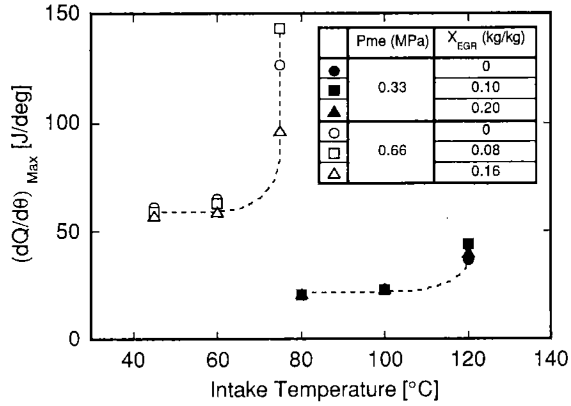
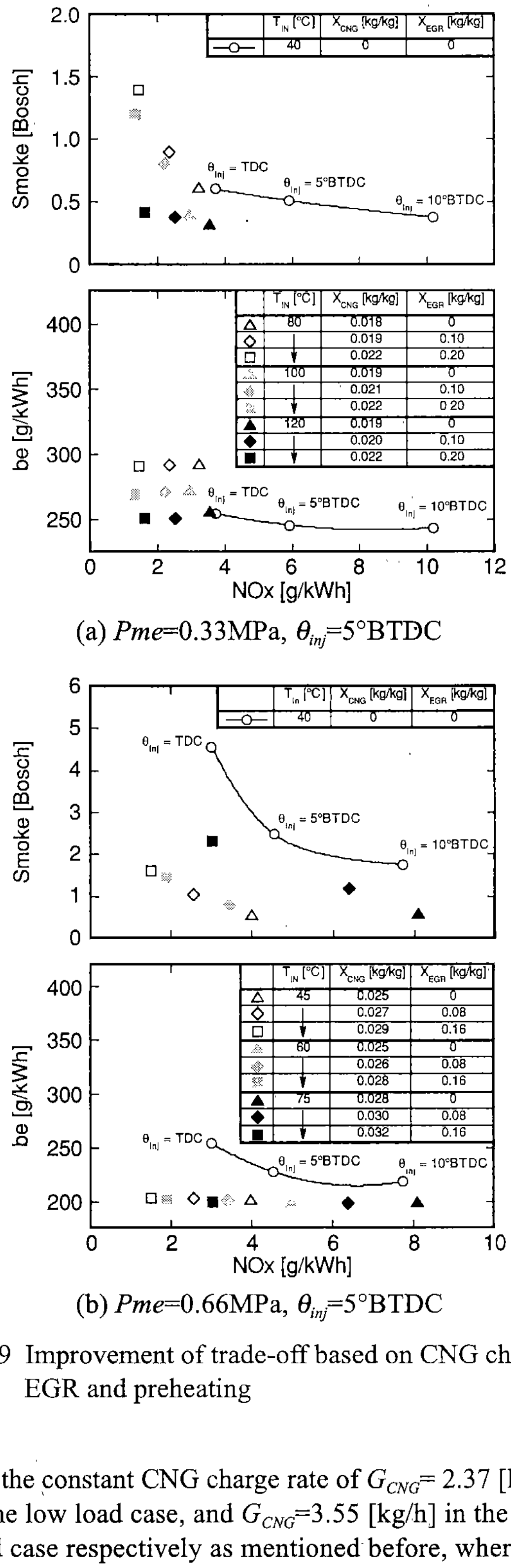


Fig.8 Effect of intake temperature on burning rate of natural gas

below 60°C in order to suppress the burning rate of natural gas for avoiding diesel knock. As shown in Fig.7(a), the ignition delay of gas oil in the natural gas pre-mixture decreases with the intake temperature rise, and increases a little with the EGR rate. These effects of intake temperature and EGR on ignition are similar to the results shown by Nakano, et al.[7]. In Fig.7(a), if the intake temperature is equal, the ignition delay is a little smaller in the high load than the low load due to the difference of CNG charge rate. It seems to be reasonable that the ignition delay is mainly dependent on the in-cylinder mean gas temperature at the ignition timing and it is increased slightly by EGR as shown in Fig.7(b).

3.4 Improvement of Trade-Off

Figures 9(a) and (b) show changes in trade-off relationships between smoke and NOx, and between fuel consumption and 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 operated with gas oil alone, where injection timing of gas oil varied from 10°BTDC to TDC, and the other data were obtained under 5°BTDC,



(a) Pme=0.33MPa, θ_{inj}=5°BTDC

(b) Pme=0.66MPa, θ_{inj}=5°BTDC

Fig.9 Improvement of trade-off based on CNG charge, EGR and preheating

and the constant CNG charge rate of G_{CNG}= 2.37 [kg/h] in the low load case, and G_{CNG}=3.55 [kg/h] 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 the 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.9(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 of 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 required. 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.

The above optimum operating condition may vary from engine to engine, however, the present concept is applicable in most dual-fueled natural gas diesel engines.

4. Conclusions

In order to improve the trade-off between smoke and NOx significantly without deteriorating fuel consumption in a DI diesel engine, homogeneously charged natural gas was burned igniting by a small amount of gas-oil. The effects of the CNG 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 appropriate 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 a short combustion duration, then, a lower fuel consumption. Too high burning rate of natural gas results in increase of NOx, on the other hand, too low burning rate results in increases of THC and fuel consumption.

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

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

(5) Ignition delay of gas oil in the natural gas premixture is mainly dependent on the in-cylinder mean gas temperature at the ignition timing, and is increased slightly by the EGR rate.

(6) A high EGR rate suppresses the burning rate at the high load, however, the burning rate is hardly affected by EGR at the low load.

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