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Combustion Characteristics of a Dual Fuel Diesel Engine with Natural Gas (Lower limit of Cetane Number for Ignition of the Fuel)

Yasufumi Yoshimoto
Niigata Inst. of Technology

Eiji Kinoshita
Kagoshima Univ.

Shanbu Luge and Takatoshi Ohmura
Niigata Inst. of Technology

ABSTRACT

Dual fuel diesel engines using compressed natural gas (CNG) are an attractive low polluting application, because natural gas is a clean low CO_2 emitting fuel with superior resource availability. In dual fuel diesel engines with natural gas as the main fuel the natural gas is supplied from the intake pipe and the pre-mixture formed in the cylinder is spontaneously ignited by an injected spray of ordinary gas oil. Dual fuel engines of this type have the advantages that only limited engine modifications are needed and that low calorie gas fuels such as biogas can be used. To clarify the influence of the cetane number (C.N.) of the ignition fuel on the ignition performance, combustion characteristics, and emissions of the dual fuel operation, the present study used standard ignition fuels prepared by n-hexadecane and heptamethylnonane which define the ignitability of diesel combustion. The experiments focused on determining the lower C.N. limit of the ignition fuel and used standard ignition fuels of different C.N., 30 to 55, in five C.N increments. It was found that at high loads the dual fuel operation needs ignition fuels with C.N. higher than 45, while normal diesel operation is possible with C.N. 35 to 40 fuels. It was confirmed that for ordinary gas oil and for fuels with C.N. higher than 45, there is a strong negative correlation between the coefficient of variance of IMEP and the brake thermal efficiency.

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INTRODUCTION

Dual fuel diesel engines using natural gas are an attractive low polluting option for diesel engines, because natural gas is a clean low CO_2 emitting fuel with superior resource availability. The present study supplies compressed natural gas (CNG) as the main fuel from the intake-pipe and the pre-mixture formed in the cylinder is spontaneously ignited by the injection of ordinary gas oil or other standard ignition fuels. Dual fuel engines of this type have the advantages that only limited modifications of presently used engines are needed for cleaner operation and that it is possible to use low calorie gas fuels such as pyrolysis gas produced from wood- and waste-biomass [9][12], as well as that at high load conditions a relatively high thermal efficiency can be obtained. Studies using this type of dual fuel operation have been reported with conventional diesel engines [1],[2],[3],[4],[5],[6],[7],[8],[9],[10],[11],[12],[13],[14],[15],[16],[17],[18],[19],[20], and it is considered that the above mentioned features would suit cogeneration systems well [5][8].

In a previous study [19], one of the authors used fatty acid methyl esters, methyl oleate and methyl palmitate, major components of biodiesel, as the ignition fuel with a small single cylinder DI diesel engine. The results showed that when the CNG supply was less than 75% of the heat energy available, the brake thermal efficiency was similar to that of ordinary diesel operation at high loads (BMEP=0.67MPa). Further, when the CNG supply was above 75%, ignition became very unstable and the brake thermal efficiency decreased significantly as well as HC emissions increased sharply. The reason for this was considered to be the appearance of misfiring, and a following study [20] aimed to clarify the influence of the cetane number (C.N.) of the ignition fuel on the ignition performance, combustion characteristics, and emissions of the dual fuel operation. The experiments used standard ignition fuels prepared by n-hexadecane and heptamethylnonane which define the ignitability of diesel combustion, and focused on the effects of fuels with better ignitability than ordinary gas oil such as fuels with higher cetane numbers, 70 and 100. The results showed that while the CNG ratio where misfiring occurs decreased somewhat with increasing C.N., the combustion stability was little improved.

To determine the lower C.N. limit in dual fuel operation, the present study used standard ignition fuels with fuels of different C.N., 30 to 55, in five C.N increments, and the influence of the ignition fuels with lower C.N. was examined in detail. It was found that at high loads the dual fuel operation needs ignition fuels with C.N. higher than 45, while normal diesel operation is possible with C.N. 35 to 40 fuels. It was confirmed that for ordinary gas oil and for fuels with C.N. higher than 45, there is a strong negative correlation between the coefficient of variance of IMEP and the brake thermal efficiency. Details of the results are described in the following.

EXPERIMENTAL APPARATUS AND METHODS

Fuel

The experiments used CNG (compressed natural gas) as the main fuel premixed with fresh intake air and with standard fuels for the ignition: six different ignition fuels with C.N. 30 to 55 in five C.N. increments were prepared by mixing n-hexadecane and heptamethylnonane based on Eq. (1) which defines the compression ignition characteristics, and ordinary JIS #2 gas oil was used as the reference fuel. Tables 1 and 2 show the particulars of the test gas and the tested ignition fuels.

$$C.N. = nHD(vol.\%) + 0.15 \times HMN(vol.\%) \tag{1}$$

Table 1. Properties of the used gas

Test main fuel			CNG
Composition	CH_4	vol.%	89.0
	C_2H_6	vol.%	6.4
	C_3H_8	vol.%	3.7
	C_4H_{10}	vol.%	0.9
Mean molecular weight		g/mol	18.36
Density		$kg/m^3{}_N$	0.819
Net calorific value		$MJ/m^3{}_N$	40.37
Net calorific value		MJ/kg	49.30
Stoichiometric air-fuel ratio			17.05

Table 2. Particulars of the tested ignition fuels

		Gas oil	nHD	HMN
Density [15℃]	(g/cm^3)	0.816	0.773[(a)]	0.785[(b)]
Viscosity [30℃]	(mm^2/s)	2.82	—	—
Net calorific value[(c)]	(MJ/kg)	42.87	43.62	43.55
Cetane number		60.2[(d)]	100	15
Stoichiometric air-fuel ratio		14.6	14.9	14.90
Carbon	(mass %)	86.1	84.9	84.9
Hydrogen	(mass %)	13.8	15.1	15.1

(a) 20℃ (b) 25℃ (c) measured by the authors (d) cetane index

Table 3. Engine specifications

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

Engine Setup

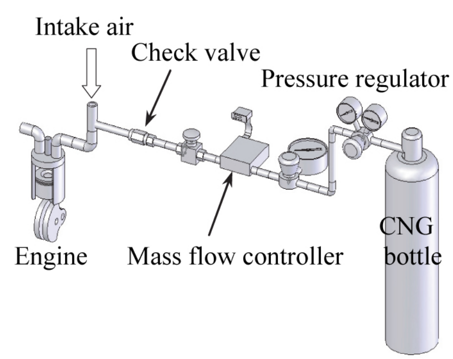


Figure 1. Experimental setup

Figure 1 shows an outline of the experimental apparatus, consisting of a test engine and a CNG supply unit. The tested engine is a 4 stroke, naturally aspirated, water-cooled single cylinder direct injection diesel engine equipped with a toroidal type combustion chamber and the principal specifications of the engine and its fuel injection system are shown in Table 3. The fuel injection system is set to the standard gas oil specifications. As shown in Figure 1, the CNG supply unit is composed of a CNG bottle, a pressure regulator, a mass flow controller, and a check valve. The CNG was fed directly into the intake-pipe, vertical to the air flow. The CNG gas inlet port was 130 mm upstream from the intake-pipe joint of the engine. It was considered that the charge motion with dual fuel operation is similar to that of ordinary diesel operation as the influence of the CNG mixing on the charge motion is small and as the CNG mixing position is well away from the cylinder. The ambient air temperatures and pressure were 25-27°C and 101kPa, respectively.

Measuring Apparatus and Procedure

The combustion pressure was measured with a strain gauge 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 three signals were input into a personal computer, 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, Eq. (2), the degree of constant volume of combustion, η_{glh} is determined from Eq. (3):

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

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

The ignition delay was determined from the crank angle interval between the start of the needle lift and pressure rise due to the combustion. The stability of combustion was evaluated with the cycle to cycle variations in IMEP. The combustion pressure was recorded through about 160 cycles and the distribution of IMEP was determined from 50 continuous cycles. The combustion fluctuation rate, $COV\,(P_i)$ was defined as the standard deviation in the distribution of IMEP divided by the mean value of IMEP.

The NOx emissions were measured using a CLD analyzer, the HC (total hydrocarbons) was measured as ppm methane using a FID analyzer, and the smoke density was measured with an opacity-meter. During the experiments, the engine was operated with 85±2°C cooling water and 70±2°C lubricating oil at a constant engine speed of 1900rpm which corresponds to the maximum brake torque conditions.

With the dual fuel operation, the brake thermal efficiency decreases significantly with decreasing engine load [1][3][5][6][13][18], and the experiments were conducted at a constant BMEP=0.66MPa; the flow rate of the CNG was from 0 to 17L/min (0°C, 1atm). The proportion of CNG supplied, Q_g/Q_t, was defined as the ratio of the heat energy of the supplied CNG Q_g, to the total heat energy available in the cylinder Q_t. When the CNG flow rate is 17L/min, the equivalence ratio of the premixed mixture, ϕ_g, is 0.60, and the proportion of CNG, Q_g/Q_t, is 80%.

EXPERIMENTAL RESULTS AND DISCUSSION

Influence on Engine Performance, Combustion and Emission Characteristics

Figure 2 shows the changes in brake thermal efficiency η_e, the equivalence ratio of the premixed mixture ϕ_g, the equivalence ratio of the total in-cylinder charge ϕ_t, and the exhaust gas temperature as a function of the proportion of CNG supplied, Q_g/Q_t, with the standard ignition fuels (C.N. =30 to 55) and ordinary gas oil. Figure 3 shows the emission characteristics vs. Q_g/Q_t. In the figures, Q_g/Q_t=0% stands for ordinary diesel operation fueled by the ignition fuel only. As shown in Figures 2 and 3, the brake thermal efficiencies decreased remarkably and HC emissions increased sharply at $Q_g/Q_t \fallingdotseq$ 77%. As described later, this is considered to be caused by the appearance of misfiring and this result is similar to previous studies [18],[19],[20]. This section mainly discusses the $Q_g/Q_t \leqq$ 77% region.

As shown in the top graph of Figure 2, the brake thermal efficiencies, η_e with gas oil maintained values similar to those of ordinary diesel operation with CNG. For the fuels with C.N. 45, 50, and 55, the η_e were slightly lower than in ordinary diesel operation in the 41% $\leqq Q_g/Q_t \leqq$ 77% region, but these fuels still maintain good brake thermal efficiencies. Summarizing, the ignition fuels with relatively higher C.N. and with the CNG supply in the optimum regions, resulted in good heat energy conversion efficiencies similar to those of the ordinary diesel operation. The present study was carried out at a limited range of high load conditions, but the results show clearly advantageous characteristic, also as there are few reports where better heat energy conversion efficiencies could be obtained in dual fuel operation [18],[19],[20]. However, the η_e decreases considerably with the C.N. 40 ignition fuel at the CNG supply ratio, Q_g/Q_t beyond 21%, this decrease is stronger with the C.N. 35 ignition fuel. With the C.N. 30 ignition fuel, the η_e is clearly lower than in ordinary diesel operation. In summary, it was found that at high loads the dual fuel operation needs ignition fuels with C.N. higher than 45, the conditions where ordinary diesel operation is possible is with C.N. values around 40 to 35.

Figure 4 shows the results of the combustion analysis of the dual fuel operation with gas oil and standard fuels with C.N. 40 to 55 as the ignition fuels. Here Figure 4(a) shows the Q_g/Q_t=0% (ϕ_g=0) conditions, ordinary diesel operation, and Figure 4(b) shows dual fuel operation at Q_g/Q_t=59-63% (ϕ_g=0.41-0.42). Both Figures 4(a) and 4(b) show that the ignition timings delay with decreasing C.N. of the ignition fuel. Compared with Figure 4(a) which was operated without CNG, Figure 4(b) shows that with CNG the ignition timings delayed considerably and the peak values of the premixed combustion increased with decreasing C.N. of the ignition fuel.

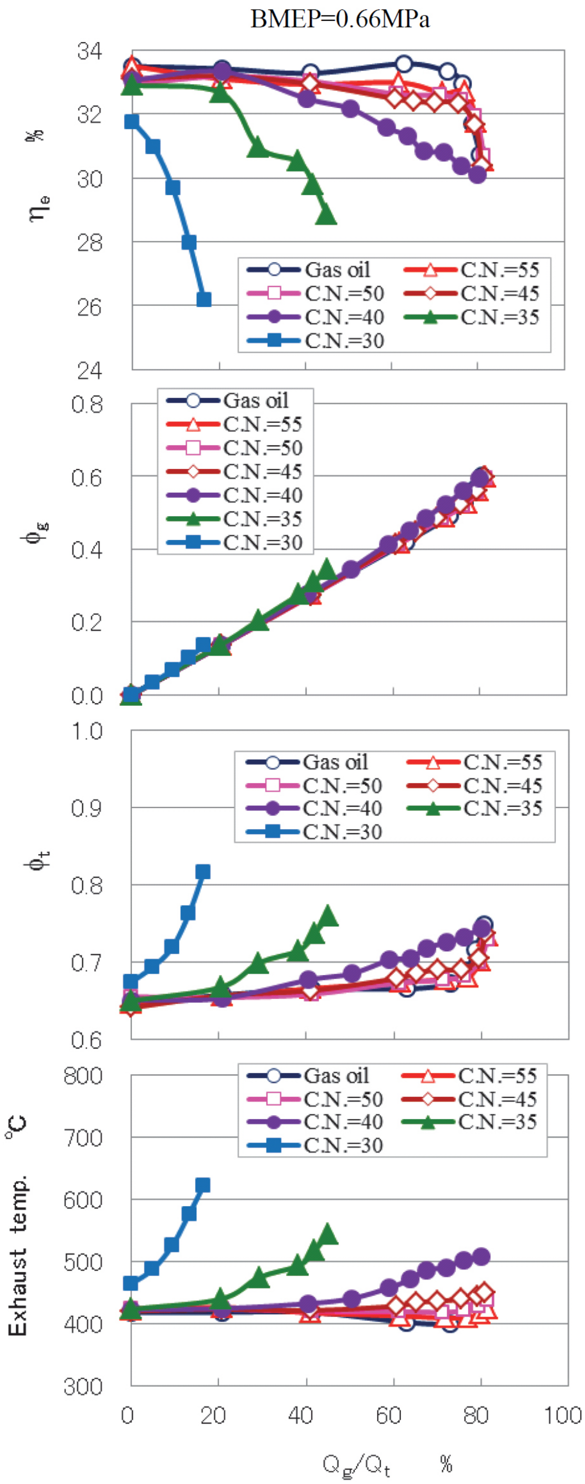


Figure 2. Engine performance with fuels of various cetane numbers at different CNG supply ratios, Q_g/Q_t

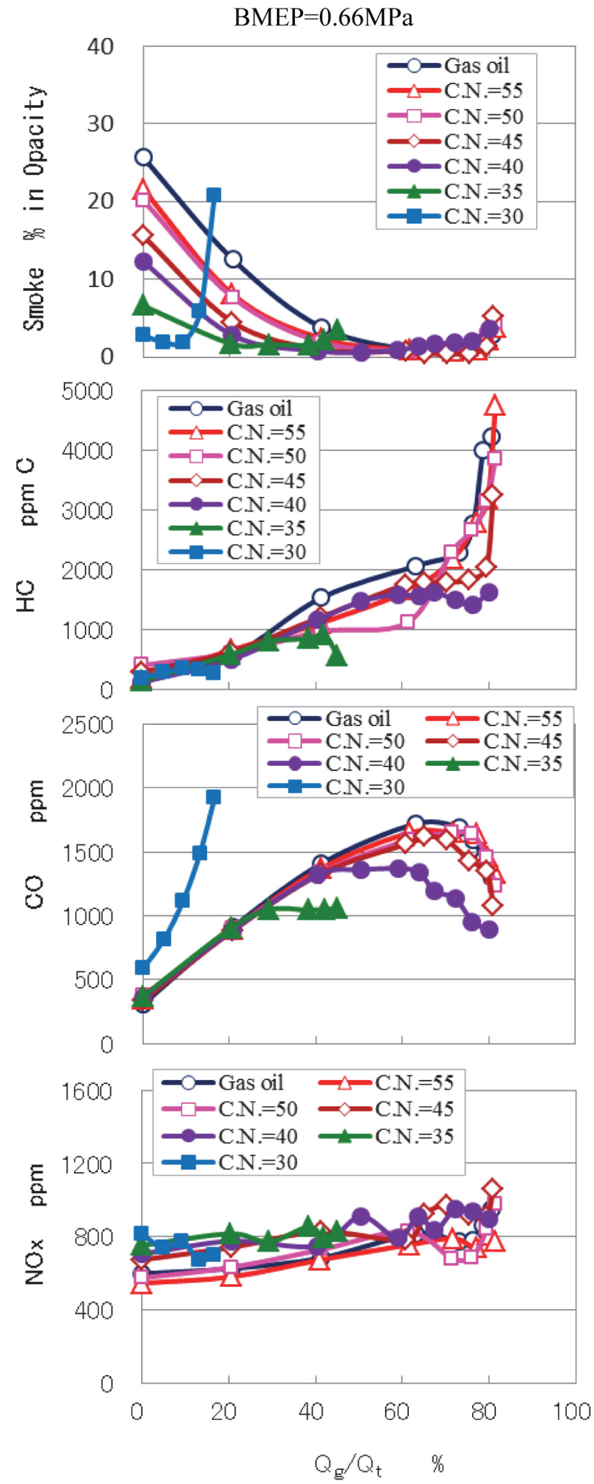
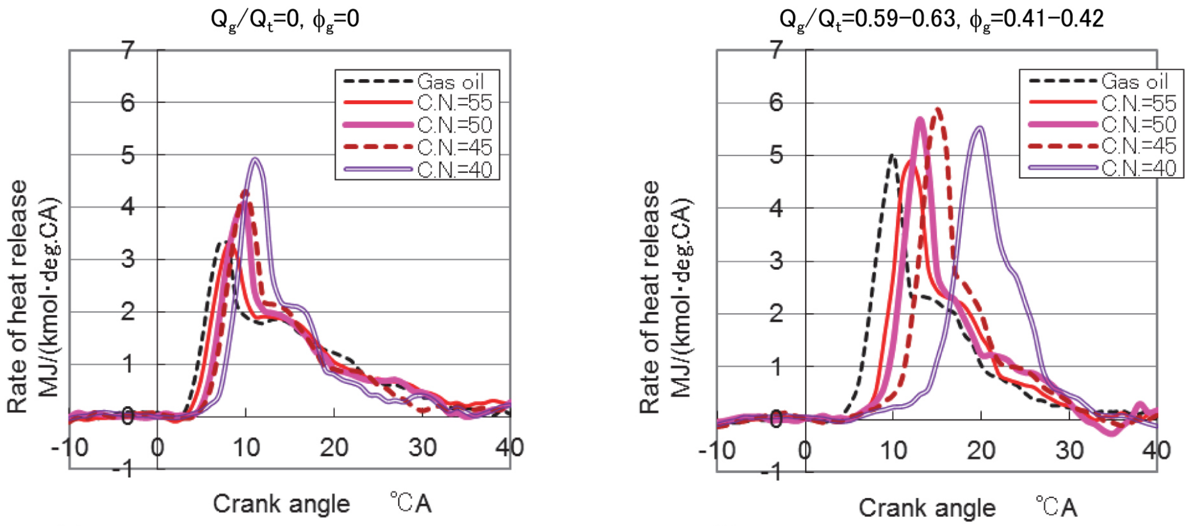


Figure 3. Exhaust emissions with fuels of various cetane numbers at different CNG supply ratios, Q_g/Q_t



(*a*) (*b*)

Figure 4. Heat release rates with gas oil and fuels of various cetane numbers (BMEP=0.66MPa)

Figure 5 shows the ignition delays, the maximum heat release rates, $(dQ/d\theta)_{max}$, the timings of the $(dQ/d\theta)_{max}$, and the degree of constant volume of combustion η_{glh}, as a function of the CNG supply. These combustion characteristics represent average values which were obtained by analyzing three recorded signals, the crank angle, nozzle needle lift, and cylinder pressure, over 50 continuous cycles. As shown in the top graph of Figure 5, the ignition delay is prolonged with decreases in the C.N. of the ignition fuel as well as with increases in the CNG supply. Specifically, the ignition delays with the C.N. 30 and C.N. 35 fuels show very large increases with increasing CNG supply ratios, Q_g/Q_t. The reason why the ignition delay increases with the dual fuel operation is due to the mixture formation of the sprayed ignition fuel and air being delayed, and the mechanisms may be considered to be explained by the following three points (processes): in the present study, the experiments were carried out at a constant output condition (BMEP=0.66MPa). When the CNG supply ratio increases, the spray momentum decreases because of the decreased injection quantity. Therefore, the mixing speed of the fuel spray and the entrained air decreases and this results in a slowness of mixture formation. The second possible cause is that, as shown in the second graph in Figure 2 (ϕ_g vs. Q_g/Q_t plot), the equivalence ratio of the CNG-air premixed mixture, ϕ_g increases with increasing CNG supply, resulting in much higher equivalence ratios around the spray flux peripheral. The CNG alone has an extremely low cetane number and the above effect would also result in the later formation of a combustible mixture. A further, third, reason may be that the cylinder charge temperature close to the point of the liquid fuel injection decreases due to the higher overall specific heat capacity of the CNG-air mixture, compared to that at ordinary diesel operation [14].

The relation 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 cooling loss ϕ_w, and the mechanical efficiency η_m is:

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

For the lower C.N. fuels such as C.N. 30 or 35, the major combustion process shifts considerably to the expansion side because of the extremely delayed ignition and this trend is accelerated with the increasing CNG supply. As shown in the bottom graph of Figure 5 (η_{glh} vs. Q_g/Q_t plot), the degree of constant volume of combustion, η_{glh} decreases significantly with C.N. 30 and 35. Further, the cooling loss, ϕ_w probably increases because of the much delayed combustion. Consequently, as shown in the top graph of Figure 2 (η_e vs. Q_g/Q_t plot), the above factors result in significant decreases in the brake thermal efficiency.

Figure 6 shows the relation between the exhaust temperature and brake thermal efficiency. All of the data obtained is plotted here, and there is a strong negative correlation between these two parameters. Overall, the data suggests that the increase of the exhaust heat loss with the extremely low C.N. fuels is one of the factors which plays a role in the significant decreases in the brake thermal efficiency. The magnitude of the exhaust heat loss is estimated to increase about 5% when the exhaust temperature increases 100°C.

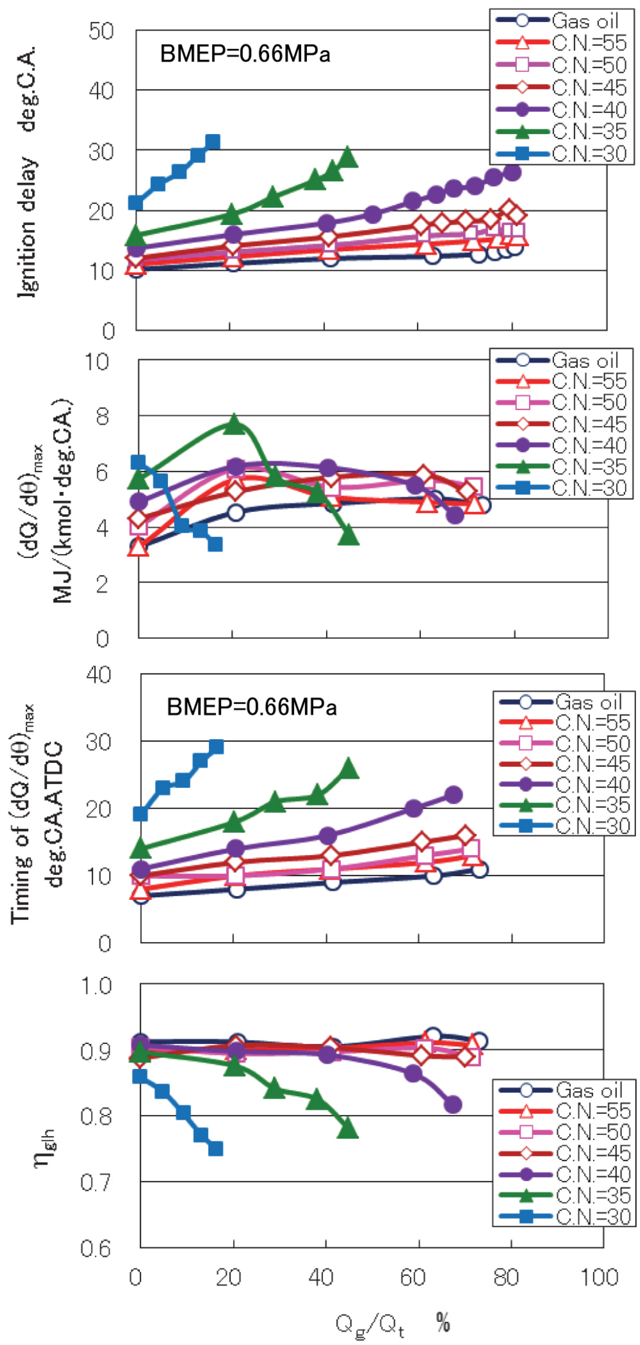


Figure 5. Combustion characteristics with gas oil and fuels of various cetane numbers at different CNG supply ratios, Q_g/Q_t

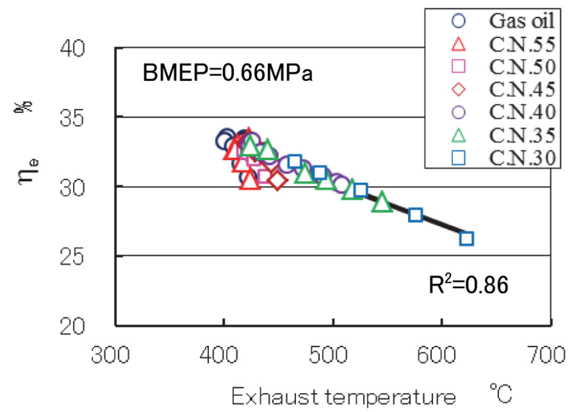


Figure 6. Plot of exhaust temperature vs. brake thermal efficiency

As shown in Figure 5, for the relatively higher cetane number fuels (C.N.≧45) and gas oil, $(dQ/d\theta)_{max}$ increases with increasing Q_g/Q_t while the timings of the $(dQ/d\theta)_{max}$ delays. The η_{glh} maintains relatively high values allowing the conclusion that, when an ignition fuel with suitable C.N. is used, the premixed combustion is promoted with the dual fuel operation. From Figure 4(b), for the fuels with C.N.≧45 and gas oil, the quantity of heat released in the premixed combustion increases and the timing of the completion of the combustion are similar. It may be considered that despite of the longer ignition delay, the changes in the heat release diagrams resulted in maintaining high η_{glh} values as shown in the bottom graph of Figure 5. As described above, the dual fuel operation resulted in longer ignition delays and promotion of premixed combustion. It may be considered that the amount of heat release during premixed combustion increases because the formation of combustible mixture is promoted due to the longer ignition delays and that the combustion of the CNG-air pre-mixture occurs simultaneously due to the presence of the CNG-air pre-mixture entrained in the ignition fuel spray. As a result, local gas temperatures rise and this resulted in a gradual increase in NOx concentrations (bottom graph of Figure 3) with increasing CNG supply. It may be considered that the consequent combustion coexists with diffusion combustion of the sprayed ignition fuel as well as flame propagation combustion in the CNG-air pre-mixture.

With the lower cetane number fuels (C.N.≦40), the maximum heat release rates, $(dQ/d\theta)_{max}$ decrease where the CNG supply ratio reaches a lower threshold value, and this decrease in $(dQ/d\theta)_{max}$ plays a larger role with decreasing cetane numbers of the ignition fuel. In these cases, the timings of the $(dQ/d\theta)_{max}$ is extremely delayed and the η_{glh} decreased significantly resulting in the remarkable decrease in the brake thermal efficiency.

As shown the top graph of Figure 3, the smoke densities with all the ignition fuels except for C.N. 30 show remarkable reductions until a CNG supply of $Q_g/Q_t ≒ 60\%$. Also, the smoke reductions were promoted with decreasing cetane number, caused by the promotion of premixed combustion. A further reason for the smoke reductions is considered to be that the soot formation in the fuel rich zone may be suppressed as the injected quantity of the ignition fuel decreases with the increase in CNG supply. Here, the C.N. = 30 fuel showed remarkable increases in the smoke densities at $Q_g/Q_t \geqq 13\%$. As will be described later, this may be considered to be due to in-complete combustion caused by the appearance of misfiring. The HC emissions increase with increasing CNG supply at $Q_g/Q_t \leqq 77\%$. With dual fuel operation, unburned mixture easily enters the quenching regions such as at the cylinder wall and piston top clearance, giving rise to HC emission increases. A further reason for the increase in HC emissions may be that there are parts of the CNG-air mixture that are not entrained into the spray flux of the ignition fuel. As the CNG-air mixture is very lean and outside the flammable range, the flames do not propagate here and this mixture may remain in the cylinder without burning something that would also result in higher HC emissions.

The CO concentration with all the ignition fuels except for the C.N. 30 case increases with increasing CNG supply (Figure 3), but CO tends to decrease or to remain constant when Q_g/Q_t is above a threshold value, controlled by the cetane number of the ignition fuel. The point where the CO concentration starts to decrease is about $Q_g/Q_t \fallingdotseq 65\%$ for the higher cetane number fuels (C.N.$\geqq$45) and gas oil. For conditions below $Q_g/Q_t \fallingdotseq 65\%$ (ϕ_g=0.45), the ignition fuel quantities decrease and the spray combustion regions decrease with increasing CNG supply because of the lower equivalence ratio in the CNG-air premixed mixture ϕ_g (second graph from the top, Figure 2). This suggests that the quenching region being outside of the spray combustion expands, and the CO emissions, in-complete combustion products, increase. For the regions with Q_g/Q_t>65%, the equivalence ratios, ϕ_g, keep increasing with increasing CNG supply. It is considered that the temperature in the regions of flame propagation combustion rise and the result is a decrease in CO concentration with the combustion improvements in these regions. The results for the C.N. 40 and 35 are due to similar reasons, but the change points in the graph (the maximum) would shift to lower Q_g/Q_t values (less CNG supplied), because the injected fuel quantities increase due to the large deterioration in the brake thermal efficiency. The significant increase in the CO emissions with the C.N. 30 fuel is considered to be due to the appearance of in-complete combustion cycles where smoke emissions increases.

As shown in Figure 2, the equivalence ratios of the premixed mixture ϕ_g increase linearly with increasing CNG supply rate, while the equivalence ratios of the total in-cylinder charge ϕ_t increase steeply at lower CNG supply rates with the low cetane numbers. The changes in ϕ_t are caused by the increases in the injected quantity of the ignition fuel and correspond to decreases in brake thermal efficiency.

When the CNG supply (Q_g/Q_t) is above 77%, and using ignition fuels with C.N.$\geqq$45 or gas oil, the brake thermal efficiency η_e decreases significantly (Figure 2) while the HC emissions increase sharply (Figure 3). This is considered to be caused by the appearance of misfiring and the next section will discuss this further.

Influence on Cycle to Cycle Variations in Combustion

The ignition became unstable when the CNG supply ratio (Q_g/Q_t) was above 78%, because of the low quantity of fuel injected. The appearance of misfiring cycles was determined by examining a continuous indicator of about 160 cycles for standard ignition fuels with different cetane numbers and CNG supply conditions; further, the coefficient of variance, COV (P_i), was determined by calculating the cycle to cycle variations in the IMEP distribution, and results of both are shown in Figure 7. Here, regardless of ignition fuel (other than C.N. 30) the COV (P_i) are similar to ordinary diesel operation at the $Q_g/Q_t \leqq 77\%$ condition, while the COV (P_i) values increase significantly when the CNG supply is above 78%. Misfiring appears at $Q_g/Q_t \geqq 78\%$ as shown by the sharp increase in COV (P_i) and the incidence increases sharply with increasing CNG supply.

Figure 8 shows the relation between the appearance of misfiring and $COV(P_i)$ for different kinds of fuel. The plot in Figure 8 includes the relatively higher cetane number fuels (C.N.$\geqq$45) and gas oil, and establishes that there is a linear relation between the appearance of misfiring and $COV(P_i)$, with a correlation factor of R=0.93. This would allow the conclusion that the combustion fluctuations are caused by the appearance of misfiring.

The appearance of misfiring cycles results in a sharp increase in HC emissions, as was shown in Figure 3. Also, the misfiring results in increasing equivalence ratios of the total in-cylinder charge ϕ_t (Figure 2) because the injected fuel quantities increase, maintaining the constant engine output. The considerable decrease in the brake thermal efficiency due to the appearance of misfiring can be explained as follows. When the CNG supply rate is excessively high, the ignition timing is considerably delayed and combustion becomes unstable, resulting in misfiring. From Eq. (4), then, the combustion efficiency η_u, the degree of constant volume of combustion η_{glh}, and the mechanical efficiency η_m all decrease, while the cooling loss ϕ_w increases. The net result is a considerable decrease in the brake thermal efficiency.

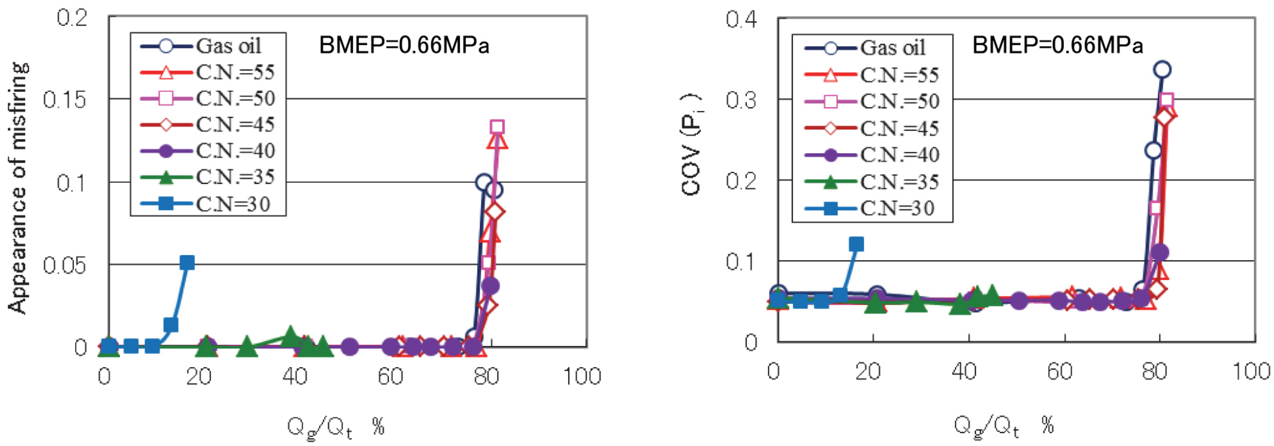


Figure 7. Plot of CNG supply ratios vs. appearance of misfiring and COV(P_i)

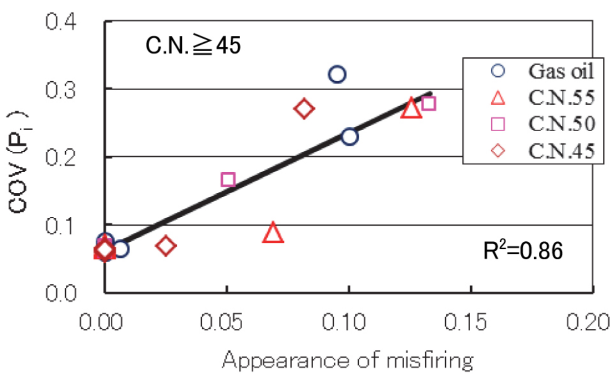


Figure 8. Plot of appearance of misfiring vs. COV(P_i)

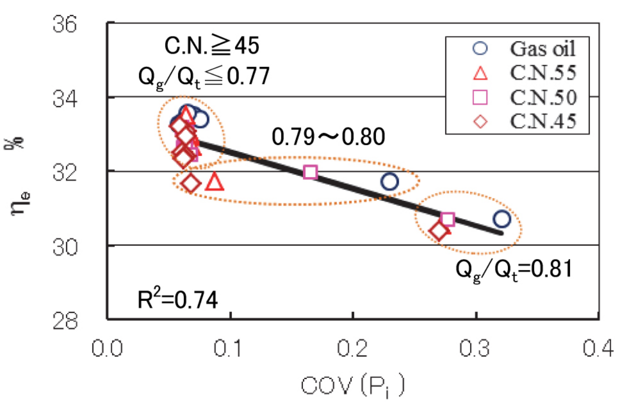


Figure 9. Plot of COV(P_i) vs. brake thermal efficiency

Figure 9 shows the relation between *COV* (P_i) and brake thermal efficiency with C.N. ≧45 fuels and gas oil. Combustion fluctuations increase as the brake thermal efficiency decreases where the CNG supply ratios exceed 78%, and there is a linear relation between the coefficient of variance of IMEP and the brake thermal efficiency. This confirms that when the CNG supply ratio is excessively high, misfiring would occur and the coefficient of variance of the IMEP increases, resulting in the considerable decreases in the brake thermal efficiency.

CONCLUSIONS

Standard fuels with cetane numbers (C.N.) 30 to 55 were prepared and employed as ignition fuels in a dual fuel diesel engine fueled by compressed natural gas (CNG) from the intake-pipe as the main fuel. The influence of the C.N. of the ignition fuel and the CNG supply ratio on the engine performance, combustion, and emission characteristics were examined and compared with gas oil ignition at high loads. The rate of CNG supply was defined as the ratio of the heat energy of the supplied CNG to the total heat energy available in the cylinder. The results of the present study may be summarized as follows:

1. To maintain brake thermal efficiencies similar to ordinary diesel fuel operation, the dual fuel operation needs ignition fuels with C.N. higher than 45, while normal diesel fuel operation is possible with C.N. 40 to 35 fuels.

2. Regardless of the ignition fuel, the ignition timings of the dual fuel operation were delayed with increases in the CNG supply. Compared with the C.N.≧45 standard fuels, C.N.≦40 fuels showed significantly longer ignition delays.

3. The smoke densities with the C.N.≧45 fuels and gas oil showed remarkable decreases with increasing CNG supply. Ignition fuels with lower C.N. (35≦C.N.≦40) showed lower smoke emissions.

4. When the proportion of the CNG supply was above 78%, and with C.N.≧45 fuels and gas oil, the ignition became unstable and the brake thermal efficiencies decreased significantly while the HC emissions increased sharply. The reason may be considered to be due to increases in combustion fluctuations due to the appearance of misfiring.

5. In the dual fuel operation with the C.N.≧45 standard fuels and gas oil, there is a strong negative correlation between the coefficient of variance of IMEP and brake thermal efficiency.

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CONTACT INFORMATION

Yasufumi Yoshimoto
Department of Mechanical and Control Engineering
Niigata Institute of Technology
Phone & Fax: +81(257)22-8116
Yosimoto@mce.niit.ac.jp

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DEFINITIONS/ABBREVIATIONS

CNG - compressed natural gas

C.N. - cetane number

nHD - n-hexadecane

HMN - heptamethylnonane

BMEP - brake mean effective pressure

IMEP - indicated mean effective pressure

θ - crank angle

Q - heat release in cylinder

Q_E - calorific value of fuel

Q_C - cooling heat to cylinder wall

V - gas volume in cylinder

P - gas pressure in cylinder

κ - ratio of specific heats

ε - compression ratio

ε_θ - compression ratio at crank angle

η_{glh} - degree of constant volume of combustion

Q_g/Q_t - CNG supply ratio (Q_g: heat energy of the supplied CNG, Q_t: total heat energy available in the cylinder)

ϕ_g - equivalence ratio of premixed mixture

ϕ_t - equivalence ratio of total in-cylinder charge

η_e - brake thermal efficiency

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

$COV(P_i)$ - coefficient of variance of IMEP (defined as the standard deviation in the distribution of IMEP divided by the mean value of IMEP)