

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

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ABSTRACT

Dual fuel diesel engines using compressed natural gas (CNG) are an attractive low polluting application, because natural gas is a clean low CO₂ 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 CO₂ 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.

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

Table 1. Properties of the used gas

Test main fuel		CNG
Composition	CH ₄ vol. %	89.0
	C ₂ H ₆ vol. %	6.4
	C ₃ H ₈ vol. %	3.7
	C ₄ H 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°C] (g/cm ³)	0.816	0.773 ^(a)	0.785 ^(b)
Viscosity [30°C] (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°C ^(b) 25°C ^(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

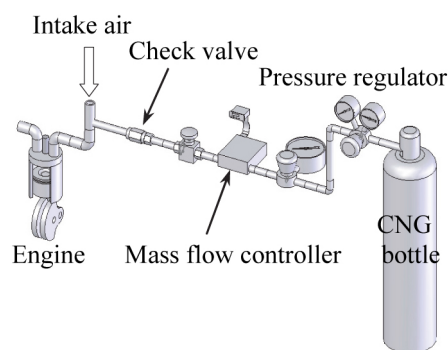


Figure 1. Experimental setup

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):

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

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

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 $Q_g/Q_t \approx 77\%$. 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 $Q_g/Q_t \leq 77\%$ 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 $41\% \leq Q_g/Q_t \leq 77\%$ 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 $Q_g/Q_t=0\%$ ($\phi_g=0$) conditions, ordinary diesel operation, and Figure 4(b) shows dual fuel operation at $Q_g/Q_t=59-63\%$ ($\phi_g=0.41-0.42$). 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.