Effects of Natural Gas Percentage on Performance and Emissions of a Natural Gas/Diesel Dual-Fuel Engine

Zhiqin Jia and Ingemar Denbratt

Abstract Due to rising costs of conventional fossil fuels, and increasingly stringent limits on emissions (especially "greenhouse gases"), use of cleaner, cheaper gaseous fuels in internal combustion engines is expected to increase in the future. A popular application is the operation of heavy-duty diesel engines using combinations of compressed natural gas (CNG), supplied with the intake air and diesel injected to initiate combustion. Extensive efforts have already been made in both industry and academia to minimize pollutant emissions and maintain diesel-equivalent performance in this dual-fuel mode, but further knowledge of effects of fundamental parameters is required to optimize the combustion. Thus, this paper presents an experimental investigation of the influence of the CNG to diesel fuel ratio on the performance (effective expansion ratio, pressure and heat release rates) and emissions (HC, CO, NO_x and CO_2) from a CNG/diesel dual-fuel engine operating under varying load conditions but constant engine speed.

1 Introduction

The automotive industry is still highly dependent on conventional fossil fuels globally. However, due to increases in their prices and concern about environmental pollution there is growing interest in the use of alternative fuels. Among these alternative fuels, compressed natural gas (CNG) is a popular choice because of its relatively clean combustion and global availability at attractive prices. Furthermore, with minor modifications heavy-duty diesel engines can operate using mixtures of CNG and diesel. Considerable efforts have been made to assess and improve the performance and emissions of engines operating in such dual-fuel combustion mode recently [1–5], but further optimization is required.

Z. Jia (✉) · I. Denbratt
Chalmers University of Technology, Gothenburg, Sweden
e-mail: zhiqin@chalmers.se

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Table 1 Basic engine specification

Engine type	Volvo D12C
Bore	131 mm
Stroke	150 mm
Connecting rod	260 mm
Compression ratio	17:1
Engine speed	1500 rpm
No. of nozzle orifices	7
Injector nozzle diameter	0.225 mm
SOI electric signal	6 BTDC

In heavy duty dual-fuel operation the CNG is injected into the intake manifold and mixed with air before being inducted into the cylinder. The mixture enters the cylinder during the intake stroke and is compressed during the compression stroke, as in a conventional diesel engine. A certain amount of diesel oil is injected in the cylinder at the end of compression stroke to initiate the gas-air mixture combustion. Compared to CNG, diesel oil has relative high cetane number which can auto-ignite under compression. While CNG has relative high octane number which prevent CNG from auto-igniting during the compression stroke.

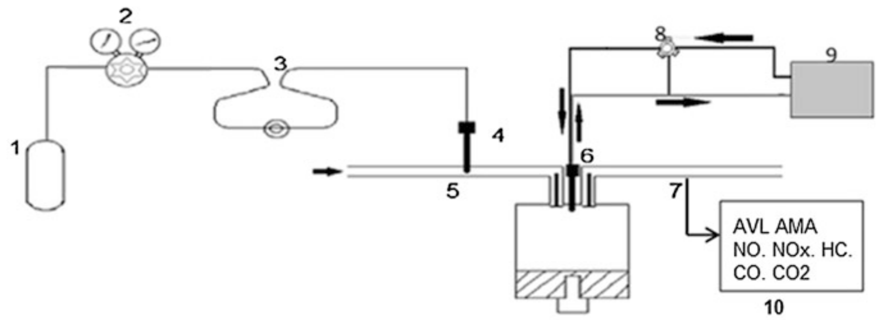
In the presented work we conducted tests with a single-cylinder version of the Volvo D12C diesel engine, with a displaced volume of 2.02 L. Mixtures of CNG and diesel oil with various mass ratios were used to examine their effects on engine performance and emissions.

2 Experimental Setup and Methodology

2.1 The Single-Cylinder Engine

The single-cylinder variant of the Volvo heavy duty D12C engine used in the experimental studies was equipped with a four-valve cylinder head, a centrally positioned diesel injector and a quiescent combustion system. The major engine specifications are listed in Table 1. In addition the engine was equipped with oil, water and air conditioning systems, and a short route EGR system with cooling from the outlet of the engine. The inlet air was supercharged by a screw-compressor, dehydrated in a dryer and temperature-regulated.

A schematic layout of the experimental setup is shown in Fig. 1. The engine was modified to enable dual-fuel combustion by adding a gas injection system to the intake manifold. CNG was obtained from a local distribution network. Before entering the engine cylinder, the CNG was passed through a pressure regulator to reduce its pressure from 200 to 14 bar and then through a gas flow meter. Following the injection signal the gaseous fuel was injected into the engine's intake and mixed with the intake air. The gas and diesel fuel control systems were completely



1) CNG Cylinder 2) Pressure regulator 3) Coriolis flow meter 4) Gas injector
5) Intake air 6) Diesel injector 7) Exhaust gas 8) High pressure pump
9) Diesel tank 10) AVL emission equipment

Fig. 1 Schematic layout of the experimental setup

independent from each other. A typical rail-type fuel injection system was used to inject the diesel.

2.2 *Methodology*

The experiments were performed with an engine speed of 1500 rpm, an SOI of 6 CAD BTDC (electrically signaled), an injection pressure of 2000 bar for the diesel oil and at load points of both 25 % (6 bar BMEP) and 50 % (12 bar BMEP). The mass ratio of CNG was varied from 0 to ~90 % under each of the load conditions. The engine's performance was characterized by measuring the in-cylinder pressure, heat release rate, fuel consumption and gas temperature. Engine emissions (NO_x, HC, CO and CO_2) were also recorded. The NG mass ratio was calculated using the following equation:

$$x = \frac{\dot{m}_{NG}}{\dot{m}_{NG} + \dot{m}_{Diesel}} * 100\%$$

Here, $\dot{m}_{NG}$ and $\dot{m}_{Diesel}$ are the mass flow rate of the NG and diesel, respectively.

3 Results and Discussion

As already mentioned, the objective of the study was to investigate effects of the NG to diesel ratio on the performance and emissions of a DI diesel engine operating in dual-fuel mode, at constant engine speed (1500 rpm) and two loads (6 and 12 bar

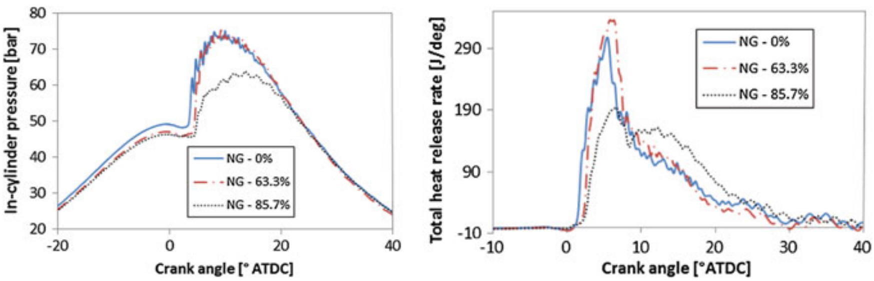


Fig. 2 In-cylinder pressure and total heat release rate traces obtained from tests with pure diesel and diesel mixtures with 63.3 and 87.5 % NG at 1500 rpm and 6 bar BMEP

BMEP). The results, presented in the following sections, do not include measurements of soot emissions, because they were very low (non-detectable in some cases). This is consistent with expectations because soot emissions are not generally problematic in CNG/diesel combustion.

3.1 Cylinder Pressure and Total Heat Release Rate Traces

Figure 2 shows in-cylinder pressure and total heat release rate traces obtained from tests at the light load cases (6 bar BMEP) at 1500 rpm. The traces in the left panel show that the in-cylinder pressure increased much more rapidly after ignition in tests with both pure diesel and 63.3 % NG than with 83.7 % NG (which also resulted in a lower peak pressure). The heat release rate curves in the panel indicate that the combustion with pure diesel is predominantly premixed, with a minor contribution from diffusion combustion. A possible explanation for this is that less fuel is needed at light load, which can be premixed well during the ignition delay time. Therefore, use of 63.3 % NG provides little advantage under these conditions. Furthermore, increasing the NG percentage to 85.7 % results in a slower combustion rate and thus a lower and later pressure peak than pure diesel. The lower burning rate may be due to deterioration of the combustion since less diesel fuel is available to ignite the mixture. The heat release rate trace obtained with 85.7 % NG shows that the combustion duration was longer than with pure diesel, indicating that combustion was less efficient. The traces also show that the ignition delay was longer with both dual-fuel mixtures than with pure diesel.

As shown in Fig. 3, at the high load (12 bar BMEP), the in-cylinder pressure after ignition increased more rapidly and there was a higher peak pressure with 69 % NG than with pure diesel, possibly due to greater evaporation and mixing of the pilot diesel spray during the pre-ignition period and higher gas concentrations surrounding the spray [5]. The heat release rate traces in the figure indicate that with both pure diesel and 69 % NG the combustion processes included two-phase combustion (pre-mixed combustion and diffusion-controlled/pre-mixed NG/air

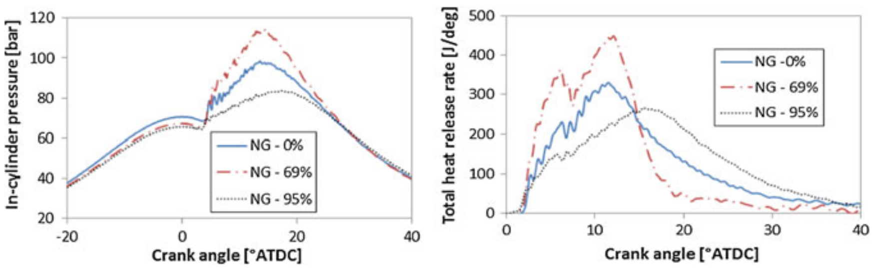


Fig. 3 In-cylinder pressure and total heat release rate traces obtained from tests with pure diesel and diesel mixture with 69 and 95 % NG at 1500 rpm and 12 bar BMEP

mixture combustion). However, the 69 % NG mixture burned more rapidly and resulted in an overall higher heat release due to more fast burning of pre-mixed NG/air during the second phase combustion than diffusion-controlled combustion of pure diesel case. Increasing the load from 6 to 12 bar BMEP raises the temperature in the cylinder, and thus the flame propagation speed. Therefore, dual-fuel combustion with 69 % NG at the higher load was much more efficient than with 63.3 % NG at the lower load. However, with 95 % NG the combustion was less efficient, manifested in a slower heat release rate (Fig. 3), presumably due to the small quantity of pilot diesel. Thus, the pressure increased more slowly and the peak pressure was lower than in tests with both pure diesel and 69 % NG.

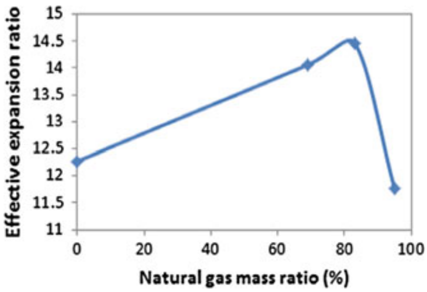
As shown in Fig. 4, at the high load the effective expansion ratio (EER) [6] increased with increases in the NG ratio to ca. 83 %, but declined with further increases to a lower level than in tests with pure diesel.

$$EER = \frac{\int_{SOC}^{EOC} HHR(\theta) \cdot ER(\theta) \cdot d\theta}{\int_{SOC}^{EOC} HHR(\theta) \cdot d\theta}$$

where θ, HHR(θ) and ER(θ) are the crank angle, the instantaneous heat release rate and the instantaneous expansion ratio.

In all the cases discussed above, under both low and high load conditions, the in-cylinder pressure decreased with increases in the NG percentage during the

Fig. 4 EER as function of natural gas mass ratio at 1500 rpm and 12 bar BMEP



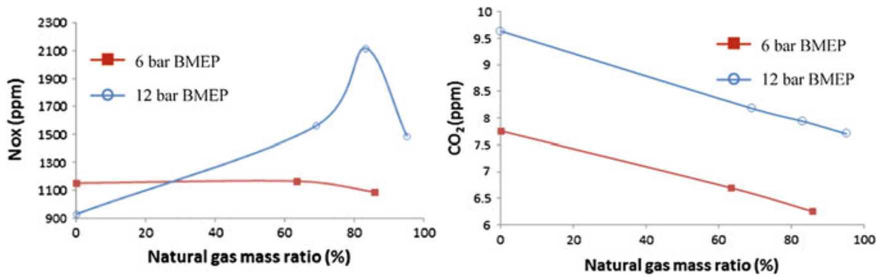


Fig. 5 Nitric oxide and carbon dioxide as function of natural gas mass ratio at 1500 rpm for 6 and 12 bar BMEP

compression stroke, presumably due to reductions of the mixture heat capacity ratio γ with increases in the NG mass ratio.

3.2 Nitrogen Oxide Emissions

Nitrogen oxide (NO_x) emissions are influenced by numerous variables, including the combustion temperature, oxygen concentration and combustion duration [7–9]. In our tests NO_x emissions were higher at the low load than at the high load when using pure diesel (Fig. 5, left panel). Furthermore, they slightly decreased with increases in the NG percentage at the low load, possibly due to deterioration of the combustion process and reduction of the charge temperature associated with reduction of the pilot diesel quantity. At high load NOx emissions were higher than at low load with all the tested dual fuel ratios, and increased with increases in the NG percentage. This is presumably because NG-air mixtures are richer at high load, thus flame propagation is faster and the charge temperature higher.

3.3 CO_2 Emissions

CO_2 emissions declined with increases in the NG percentage at both loads, presumably because NG has a lower C/H ratio than diesel. Even including unburned hydrocarbons (UHC, see below), dual-fuel combustion yielded lower CO_2 emissions than pure diesel combustion.

3.4 UHC Emissions

Several processes may contribute to UHC emissions, including inefficient combustion, wall wetting, flame quenching near cold walls and fuel passing through the

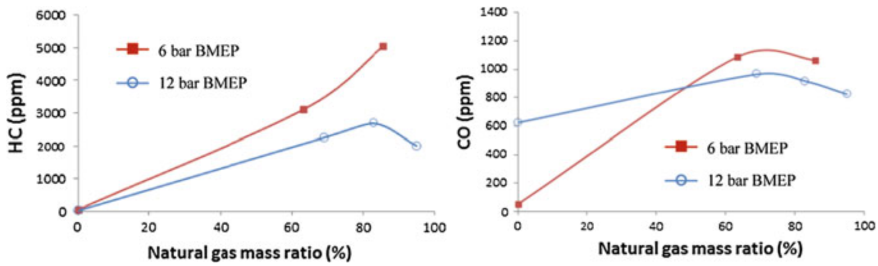


Fig. 6 Unburned hydrocarbon and carbon monoxide as function of the natural gas mass ratio at 1500 rpm for 6 and 12 bar BMEP

engine without burning during valve timing overlaps [7, 8, 10]. In our tests, use of the dual-fuel consistently resulted in higher UHC emission than use of pure diesel, and higher UHC emissions at light load than at high load (Fig. 6, left panel). This can be explained by the leaner mixtures, less efficient and slower combustion, and lower charge temperatures (as detailed above), all of which can contribute to flame extinction (bulk quenching), resulting in some of the fuel mixture reaching the exhaust without burning. At high load UHC emissions increased with increases in the NG percentage up to ca. 83 % NG then declined. However, they were consistently lower at high load than at low load, presumably because of the more efficient combustion, higher charge temperature, faster burning and (hence) more efficient HC oxidization.

3.5 CO Emissions

As shown in Fig. 6 (right panel), CO emissions were lower at light load than at high load when using pure diesel, but higher at low load in the tested dual-fuel cases, presumably due to the leaner mixtures, less efficient combustion, lower charge temperatures and (hence) impairment of CO oxidation. In general, use of dual-fuel generally led to higher CO emissions than use of pure diesel.

4 Conclusions

Our results indicate that when using dual (NG and diesel) fuel in a heavy-duty diesel engine at high load increasing the NG/diesel mass ratio up to a threshold level will increase the peak pressure and reduce the combustion duration. However, further increases will reduce the peak pressure and prolong combustion. We also found that NOx, UHC and CO emissions were higher in dual-fuel combustion mode, although CO_2 emissions decreased with increases in the NG/diesel mass ratio.

At light load use of the dual fuel resulted in a similar peak pressure to pure diesel up to a threshold NG/diesel mass ratio. However, further increases resulted in a reduction in peak pressure and prolonged combustion. NO_x emissions slightly decreased, while UHC and CO emissions increased with increases in the NG/diesel mass ratio. Overall, the results show that careful modulation of the NG/diesel ratio in response to changes in load (and other key parameters) is required to optimize dual-fuel combustion in heavy-duty diesel engines.

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