

Experimental Investigation on CNG-Diesel Combustion Modes under Highly Diluted Conditions on a Light Duty Diesel Engine with Focus on Injection Strategy

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ABSTRACT

In the last decades, emission legislation on pollutant emissions generated by road transportation sector has become the main driving force for internal combustion engine development. Approximately 20% of worldwide emissions of carbon dioxide from fuel combustion come from the transportation sector, and road vehicles contribute up to 80% of those emissions [1].

Light-duty methane gas engines are usually spark-ignited due to similar combustion characteristics for methane gas and gasoline. Since spark ignition requires a low compression ratio to avoid knock problems, gas engines have lower efficiency than diesel engines. A combustion concept that has been successfully applied on large stationary engines and to some extent on heavy-duty engines is dual-fuel combustion, where a compression-ignited diesel pilot injection is used to ignite a homogeneous charge of methane gas and air. This dual-fuel combustion concept is well established for large stationary engines and exists as an after-market solution for heavy-duty engines but does not exist at all for light-duty engines. This concept offers a high degree of flexibility for the engine operation because dual fuel combustion does not require heavy modifications of the original diesel engine architecture so diesel operation could remain unaltered.

This paper presents an initial study of how combustion characteristics in a multi-cylinder dual fuel methane-diesel light duty engine are altered by the injection control strategy adopted on different high substitution ratio operating points under highly diluted conditions (unthrottled). The measurements have been performed under steady state conditions but the impact of injection strategy on transient operation is discussed and analyzed based on emissions and brake thermal efficiency. Results show that high substitution ratios are difficult to operate at low loads regardless the operation mode selected. RCCI or PPCI combustion could be adopted to promote more stable and robust combustion, although those modes require lower substitution ratios to be achieved. This investigation does not aim for an optimization of the injection parameters in dual fuel combustion mode, but it aims to point out and understand the most relevant characteristics and behaviors of a light duty dual fuel CNG-Diesel engine operating under high substitution ratios.

CITATION: Garcia, P. and Tunestal, P., "Experimental Investigation on CNG-Diesel Combustion Modes under Highly Diluted Conditions on a Light Duty Diesel Engine with Focus on Injection Strategy," *SAE Int. J. Engines* 8(5):2015, doi:10.4271/2015-24-2439.

INTRODUCTION

Compressed natural gas (CNG) is a promising fuel to be considered for the future in the light duty engine market. In the short term, low price and low CO₂ emissions per energy content are two of its main attractive characteristics. In the long term, CNG is available almost worldwide and it can be produced from bio-sources [2]. Due to its high H/C ratio compared to conventional fuels, theoretical CO₂ savings up to 26% can be expected, even more if biogas is used.

In most cases, CNG light duty applications are implemented in spark ignition engines, where a fairly simple gas injection system is added and the engine can still be operated with gasoline or CNG independently. However, dual fuel diesel-CNG engines are operated in a different way. CNG is usually port injected (although some direct injection applications exist [3]) and the diesel operation of the engine

remains unaltered. Therefore, the engine admits a mixture of air and CNG which is later ignited by a pilot injection of diesel. That event triggers the main combustion and controls combustion phasing. In this paper, it is referred as conventional dual fuel combustion (CDF).

This concept has been applied to some extent on heavy duty diesel engines. The main limitations that have been found on this kind of heavy duty dual fuel engines are related to methane oxidation in the aftertreatment system and preignition at high load.

The concept of substitution ratio (SR) in this paper is defined as:

$$SR (\%) = \frac{\dot{m}_{CNG} \cdot LHV_{CNG}}{\dot{m}_{CNG} \cdot LHV_{CNG} + \dot{m}_{diesel} \cdot LHV_{diesel}} \times 100$$

Many investigations have been published regarding those two drawbacks on heavy duty engines [4, 5, 6, 7]. Methane oxidation is a key limiting factor for the operating range achievable for this concept, as methane oxidation catalysts require a temperature around 400°C to reach acceptable efficiency [8]. This temperature is sometimes hardly reached for lean mixture at low load and some suggest that a new generation of oxidation catalyst will be required to extend the operating lean limit of these types of engines in the future. Methane is considered around 25 times more powerful as a greenhouse gas than carbon dioxide [9], so that introduces a strong limitation on the exhaust emission from this type of engines. However, nowadays this kind of combustion concept is not considered in any emissions legislation, so there are no fixed emission levels to be fulfilled. Usually, methane emissions levels allowed for spark ignition engines are used when assessing CNG combustion in diesel engines.

The second main limitation of dual fuel CNG-diesel engines is the possibility of preignition appearance at high loads. The root of this event has been studied by many researchers and it still remains unclear [10, 11, 12, 13]. Combustion chamber and piston geometry, lubricating oil, diesel injector dribble, etc. have been suggested as possible causes for this phenomenon. Some authors have suggested that standard diesel operation should be adopted at high loads in order to avoid this problem [14], although others have suggested not so radical solutions, for example variable valve timing (variable compression ratio, Miller cycle) [15].

Overall lean operation is the most challenging area for research on this type of combustion. The excess of air usually implies poor flame propagation for high substitution ratios. For that reason, maximum substitution ratio is reduced in this area, meaning that larger amounts of diesel are used. Extreme throttling is sometimes necessary and which when combined with a non-optimized combustion chamber for premixed combustion leads to low combustion efficiency and poor combustion stability. Some researchers [6, 16, 17] have proposed the use of multi-injection strategies or alternative low temperature combustion (LTC) modes at low load in an attempt to overcome this limitation. The idea is to have diesel fuel dispersed in the premixed air-CNG mixture, in order to increase reactivity of the lean charge, with main combustion happening via diesel autoignition or via a later diesel pilot injection.

Low temperature combustion strategies were developed mainly to achieve simultaneous reduction of nitrogen oxides (NO_x) and soot in diesel engines. Ignition timing is driven mainly by the chemical kinetics of the charge. Different concepts have been presented during the last decades. Homogeneous Charge Compression Ignition (HCCI) was probably the first LTC mode introduced. Homogeneous mixture of fuel and air is a requirement and, among others, combustion control is a limiting factor. Fuel is usually port injected. Partially Premixed Compression Ignition (PPCI) is achieved with direct injection systems, where advanced injection strategies lead to a certain degree of premixing of fuel and air, although not as much as with HCCI conditions. Finally, Reactivity Controlled Compression Ignition (RCCI) relies on the mixture of two different fuels, one high cetane number fuel (usually diesel and direct injected) and one high octane number fuel (gasoline, natural gas, etc. usually port injected).

The reactivity of the mixture is controlled by the ratio of the fuels. Heat release in LTC modes is usually divided in two different stages. The first one, so called “Low temperature heat release” (LTHR) is usually linked to pre-reactions of the fuel mixture and depends on the type of fuel used. The second one, so called “High temperature heat release” (HTHR) is normally the main energy release, so it is linked to combustion noise and maximum load [18].

There are fewer investigations of dual fuel combustion in light duty engines than on heavy duty engines. The reason behind this is probably related to the use of two different fuels and poor combustion efficiency at low loads, where passenger cars are usually operated, apart from the ones previously introduced. In the latest years, legislation on maximum CO_2 emitted per kilometer has resulted in a special interest in natural gas fueled vehicles. This paper focuses on how dual fuel CNG-diesel combustion is affected by the injection strategy adopted, trying to point out the main limitations at low loads and how those are affected by the diesel injection characteristics. Not all injection parameters have been studied or optimized because the goal is to globally understand how the engine should be operated and identify the weaknesses and strengths at different operating points.

EXPERIMENTAL SETUP

Test Engine

The tests have been carried out on a 4-cylinder Volvo D4 2 liter diesel engine, modified for dual fuel operation. The original turbochargers have been removed and replaced by a new exhaust manifold and backpressure valve, in order to have full control over intake and exhaust pressures. The CNG injection system consists of 4 Keihin 73cc CNG injectors, which inject the CNG directly in the intake ports via an adaptor plate placed between those ports and the original intake manifold. [Figure 1](#) shows the intake manifold and the adaptor plate used. Gas pressure is fixed to 5 bar absolute. CNG is only injected while the intake valve is open in order to minimize cylinder-to-cylinder variations. The engine has no aftertreatment system. [Table 1](#) contains most relevant engine specifications.



Figure 1. Adaptor plate for CNG injectors attached to intake manifold. Each injector supplies CNG to two different ports (one injector per cylinder)

The engine is controlled via a LabVIEW based program developed at the Division of Combustion Engines at Lund University. High resolution data (every 0.2 CAD) is sampled via a NI PXI-6221 multifunction data acquisition card. Slow signals are logged via a NI PXI-6221 multifunction data

acquisition card while temperatures are measured with a NI PXI-4353 card. Diesel direct injectors, high pressure diesel pump and diesel common rail dump valve are controlled using NI 9751 Direct Injector Driver Module kits. CNG injectors are controlled via an “in-house built” peak and hold driver.

Table 1. Engine specifications

Engine type	Volvo D4
Displacement [cm ³]	1969
N° of Cylinders	4
Compression ratio	15.8
Bore x Stroke [mm]	82 x 93.2
Valves	4
Diesel fuel system	Denso G4S (direct injection)
CNG fuel system	Keihin CNG 73cc (port injection)
Max Power [diesel, kW]	133
Max Torque [diesel, Nm]	400

Fuel Data

Two different fuels have been used in this study. Main specifications are shown in Table 2.

Table 2. Fuel specifications

	CNG	Diesel
Type [mol-% for CNG composition]	CH ₄ 89,4% C ₂ H ₆ 5,87% C ₃ H ₈ 2,29% C ₄ H ₁₀ 0,92% CO ₂ 0,92% Others 0,62%	MK1
Lower Heating Value [MJ/kg]	48,7	45,7
H/C ratio	3,70	2,33
Molecular Weight [g/mol]	18,4	170
AFR _{STOICH}	16,9	15,2

Global lambda calculation should be specified here, because in dual fuel engines it can be obtained using different methods. In this investigation, based on mass flows of each fuel, an “equivalent” fuel is calculated, in terms of composition (basically carbon and hydrogen atoms per molecule). Concentration of oxygen (wet) is measured in the exhaust pipe and then, considering the previously equivalent fuel composition calculated, global lambda is estimated as [19]:

$$\lambda_{Global} = \frac{\left(a + \frac{b}{4} - \frac{c}{2}\right) + \left(\frac{b}{4} + \frac{c}{2}\right) \times X_{O_2}}{\left(a + \frac{b}{4} - \frac{c}{2}\right) - 4.773 \times \left(a + \frac{b}{4} - \frac{c}{2}\right) \times X_{O_2}} \quad (2)$$

where a, b and c are carbon, hydrogen and oxygen atoms per molecule of “equivalent” fuel respectively and X_{O₂} corresponds to the oxygen concentration measured in the exhaust. On the other hand,

local lambda values for CNG and diesel (lambda-CNG and lambda-diesel) are calculated based on their mass flows, air mass flow and stoichiometric air-fuel ratio for each fuel:

$$\lambda_{Local} = \frac{\dot{m}_{air} / \dot{m}_{fuel}}{AFR_{STOICH_{fuel}}} \quad (3)$$

Measurement Equipment

In-cylinder pressure has been acquired using four AVL GH14P pressures sensors, using the existing standard glow plug bores. In-cylinder pressures and torque are measured every 0.2 CAD and 150 cycles were saved and averaged for every operating point under steady state conditions. Exhaust emissions (TUHC, CO, CO₂, O₂, NO_x and CO₂ intake) were measured by a Horiba MEXA-9200F analyzer. Brake values were calculated using a torque sensor HBM T40. All temperatures are measured using K-type thermocouples and system pressures are captured using Keller slow pressure sensors. Air and CNG flows are measured using Bronkhorst F-106CI and F106AI respectively (direct thermal mass flow measurement). Diesel flow was measured using a Sartorius MSE balance. Scale weight was logged for more than 10 minutes and then a linear regression model was fitted to that data to calculate its slope, i.e. diesel fuel consumption.

RESULTS AND DISCUSSION

Table 3 contains information about operating conditions which are common for all the points analyzed in this paper.

Table 3. Common settings for all points analyzed

Diesel Common Rail Pressure [bar]	1000
CNG Common Rail Pressure [bar]	4.75
Intake Manifold Pressure [bar]	1.10
Exhaust Manifold Pressure [bar]	1.35
Inlet Temperature [°C]	27
EGR (%)	0
Throttling (%)	0
Coolant Temperature [°C]	84
Speed [rpm]	1500

Lean Low Load Operation

Dual fuel combustion is well established in stationary applications operating under stoichiometric conditions, where three-way catalyst are used to reduce exhaust emissions. That means that the diesel injection is used in the same way that the spark is used in spark ignition engines. The diesel spray triggers the combustion start. However, vehicle application requires broader operating conditions and therefore stoichiometric operation is not always advisable, especially at low load because excessive throttling at some operating points would be required.

Table 4. Diesel injection specifications used for low load operation under different combustion modes

Mode	Strategy	Timing		Flow		SR
-	-	CAD ATDC		mg/cyl		%
<i>CDF</i>	Single	-14		1.7 to 6.9		50 to 90
<i>PPCI</i>	Double	1 st	-35 to -20	75%	1.7	90
		2 nd	-10	25%		
<i>RCCI</i>	Single	-35		8.9		45

Table 4 contains information about the different injection settings used on the different experiments performed and presented in the following sections.

All tests have been carried out around IMEPg = 6 bar and 1500 rpm. The objective is to analyze how substitution ratio affects the combustion process and to identify which limitations appear at SR > 80%. Specific emissions are based on brake power.

Single Diesel Injection Strategy (CDF)

For this case, a conventional dual fuel diesel pilot strategy has been tested. Diesel injection timing has been fixed to -14 CAD ATDC, which gives an acceptable combustion phasing for SR higher than 70%. The goal is not to optimize the combustion but analyze how the different amounts of diesel and CNG affect the heat release process under highly diluted conditions. For all the operating conditions selected, the amount of energy supplied (diesel + CNG) has been kept constant. Global lambda value is around 2.5 for all cases.

In terms of exhaust emissions, the typical trade-off between TUHC and NO_x for dual fuel engines is observed at these operating conditions (Figure 2). For high substitution ratios, hydrocarbon emissions increase due to flame quenching and crevices, while the lower diesel amounts and a global lean mixture promote lower NO_x levels. In all conditions it seems that diesel is the main contributor to NO_x emissions (lean air-CNG mixture), formed in stoichiometric zones around the diesel spray, so higher substitution ratios lead into really low NO_x levels.

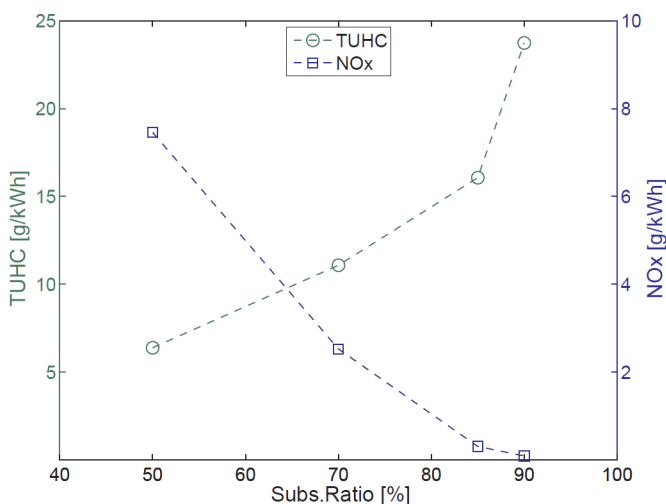


Figure 2. Trade-off between TUHC and NO_x emissions for different substitution ratios at 6 bar IMEPg, 1500 rpm and $\lambda_{\text{GLOBAL}}=2.5$ (diesel single injection @ -14 CAD ATDC)

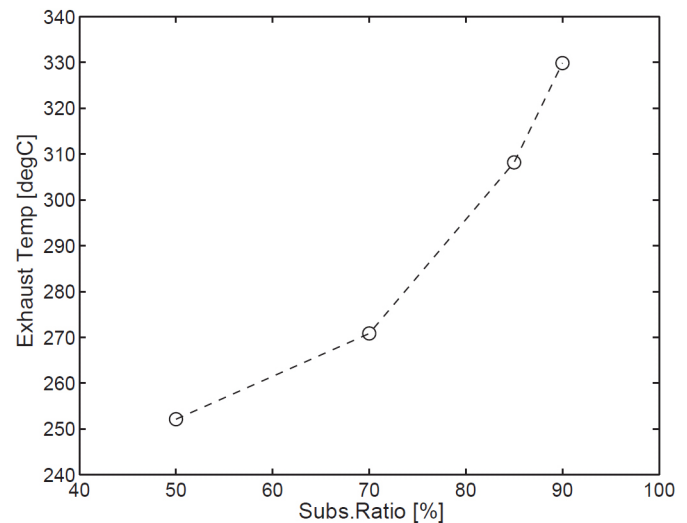


Figure 3. Mean exhaust port temperatures for different substitution ratios at 6 bar IMEPg, 1500 rpm and $\lambda=2.5$ (diesel single injection @ -14 CAD ATDC)

Regarding exhaust temperatures, as can be seen in Figure 3, higher substitution ratios result in higher exhaust temperatures. However, none of the points analyzed reach temperatures above 400°C, which would be the minimum required for high efficiency methane oxidation in the aftertreatment system.

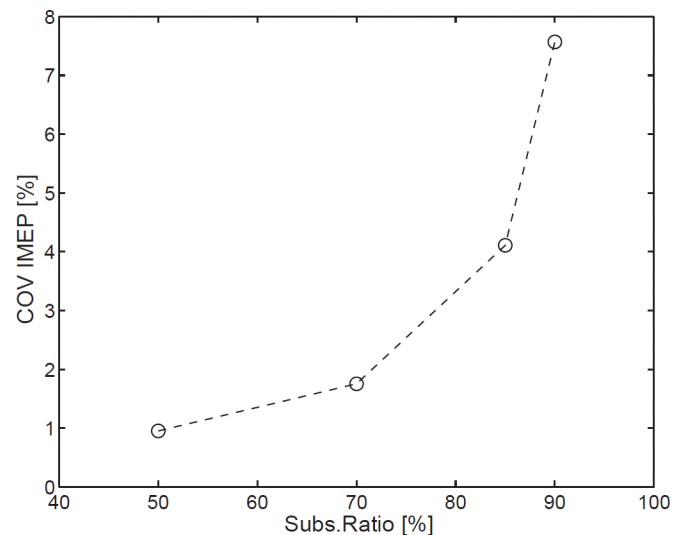


Figure 4. Coefficient of variation of IMEPg for different substitution ratios at 6 bar IMEPg, 1500 rpm and $\lambda_{\text{GLOBAL}}=2.5$ (diesel single injection @ -14 CAD ATDC)

In terms of combustion robustness, it seems that lower substitution ratios promote more stable processes, as it can be observed in Figure 4. Higher substitution ratios imply the use of reduced amounts of diesel fuels, which combined with a lean environment and the low reactivity of methane results in larger cycle-to-cycle variations.

Figure 5 shows overall combustion and brake efficiencies.

Combustion efficiency decreases when higher substitution ratios are used, which clearly affects brake efficiency. In order to analyze the trends found, a closer look at the combustion process is necessary.