

Optical Investigation of Dual-fuel CNG/Diesel Combustion Strategies to Reduce CO₂ Emissions

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ABSTRACT

Dual-fuel combustion strategies combining a premixed charge of natural gas and a pilot injection of diesel fuel offer the potential to reduce CO₂ emissions as a result of the high Hydrogen/Carbon (H/C) ratio of methane gas. Moreover, the high octane number of methane means that dual-fuel combustion strategies can be employed on compression ignition engines without the need to vary the engine compression ratio, thereby significantly reducing the cost of engine hardware modifications. The aim of this investigation is to explore the fundamental combustion phenomena occurring when methane is ignited with a pilot injection of diesel fuel. Experiments were performed on a single-cylinder optical research engine which is typical of modern, light-duty diesel engines. A high-speed digital camera recorded time-resolved combustion luminosity and an intensified CCD camera was used for single-cycle OH*chemiluminescence imaging. Experiments were performed for a wide range of equivalence ratios of the premixed charge. At low equivalence ratios, optical engine results revealed that combustion of the premixed charge of methane gas was dominated by spray entrainment and mixture stratification of diesel fuel. At higher equivalence ratios particularly close to stoichiometry, time-resolved natural luminosity images revealed significant modifications in combustion behavior indicating some evidence of flame propagation. Corresponding rates of heat release support the optical measurements in terms of revealing a significant impact on combustion following an increase of equivalence ratio. PLIF-tracer experiments were also performed in order to investigate the influence of in-cylinder fuel distribution on dual-fuel ignition.

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INTRODUCTION

With the progressive introduction of increasingly stringent world-wide vehicle exhaust gas emissions regulations and the need for fuel diversification, it is expected that the use of natural gas in the transportation sector will increase. In this context, dual-fuel combustion strategies are considered as a potentially promising solution for improving the overall efficiency of internal combustion engines. This concept is based on the use of a diesel pilot injection in order to initiate combustion within an overall homogeneous, premixed charge of natural gas. Low CO₂ emissions can be achieved as a result of employing a conventional compression ignition engine with a high compression ratio, in addition to the advantages of using methane as the primary fuel source which has a high knock resistance and low carbon content. Indeed, methane is the fossil fuel with the lowest carbon content, and offers significant reduction of CO₂ emissions compared to diesel fuel. Theoretical gains of up to 26% are achievable, and these values can be further improved if methane comes from a renewable source such as biogas. Furthermore, the

widespread commercial availability of methane at competitive prices in addition to fiscal incentives, especially in North America [1], further enhance the attractiveness of methane as a fuel for the transport industry.

Previous studies reported in the available literature reveal that the development of dual-fuel engines using natural gas were mostly focused on heavy-duty compression-ignition engines with typical diesel substitution ratios around ~60% [2]. Port-fuel gas injection system can easily be implemented in existing engines, so dual-fuel combustion strategies are particularly attractive for heavy-duty engine applications since they can be achieved without the need for major engine hardware modifications. This not only limits engine development costs, but also means that the engine can revert to conventional diesel operation (e.g. in the event of an empty reservoir of methane fuel). Recent work proposed by the authors in [3,4] showed that dual-fuel operation with natural gas can also offer major benefits for light-duty engines, both in terms of engine efficiency and high diesel substitution ratios (95 to 99%).

In parallel to these investigations, recent studies have also been performed in order to evaluate the potential of the in-cylinder blending of fuels with different reactivity. Researchers at the University of Wisconsin-ERC have performed a detailed study of Reactivity Controlled Compression Ignition (RCCI) strategies which enable increased control of HCCI-like operation by tailoring the in-cylinder fuel chemistry. Their work, based upon optical and all-metal engine data in addition to 3D CFD simulations was performed on light and heavy duty engine applications and included a variety of premixed fuels such as iso-octane [5], gasoline [6,7,8] and ethanol [9,10].

This brief literature review shows that the generic term of “dual-fuel” engines can in fact refer to rather different operating strategies, and combustion regimes are likely to dramatically vary depending on the substitution ratio, equivalence ratio and/or fuel type. Based on this observation, the aim of this study is to perform an optical engine investigation of dual-fuel combustion using methane as the premixed fuel. The aim is to attempt to identify and characterize the combustion regimes occurring in dual-fuel operation. In particular, it is still unknown whether dual-fuel combustion is characterized by auto-ignition (or sequential auto-ignition), flame propagation, diffusion or a combination of these combustion modes. It is vital to gain an improved understanding of such aspects in order to aid the further development and optimization of dual-fuel engine combustion strategies, as well as to support CFD modeling efforts. Experiments reported in this study are performed on a light-duty optical engine, but fundamental combustion mechanisms that will be highlighted are likely to be valid for a wide range of engine sizes.

EXPERIMENTAL SETUP

Optical Engine

Experiments were performed on a light-duty single-cylinder research optical engine, which is based on the production series PSA DW10 4-cylinder engine. Main specifications of the optical engine are summarized in [Table 1](#).

Table 1. Optical engine specifications

Displaced volume	499 cc
Bore	85 mm
Stroke	88 mm
Connecting Rod	145 mm
Compression ratio	15.6:1
Swirl ratio	1.1

[Figure 1](#) presents a schematic of the engine with its optical access. As shown in this figure, a Bowditch piston is used to allow optical access from below with a 45° mirror. The optical windows providing access to the combustion chamber can be cleaned rapidly, since the engine is equipped with a pneumatic piston that enables lifting of the cylinder head.

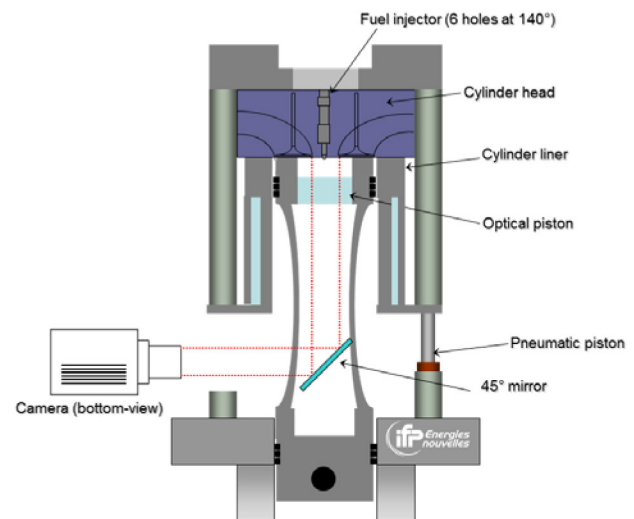


Figure 1. Schematic of the IFPEN single-cylinder optical diesel engine

Piston and Injector Specifications

As illustrated in [Figure 2](#), two pistons with comparable geometry were used in this study:

Piston A was used for the majority of the engine experiments. This piston, which is made of Titanium, was equipped with a flat optical window in the base of the piston bowl made from Sapphire.

Piston B was used exclusively for Planar Laser Induced Fluorescence (PLIF) experiments. In addition to the flat optical window in the base of the piston bowl, optical access was also provided by four quartz windows housed within the piston bowl wall: two wide windows for the laser sheet (width = 38 mm) and two smaller windows which were not used for the current study.

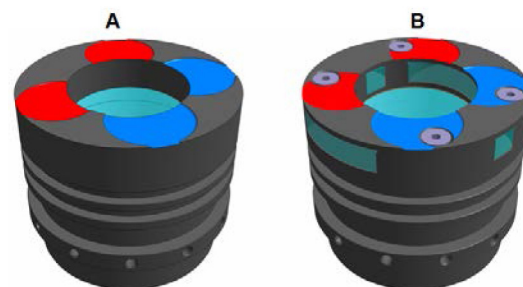


Figure 2. 3D views of the two optical pistons used for imaging experiments

The two pistons were designed to provide the same geometrical compression ratio of 15.6:1. They have the same bowl height (14 mm), but the bowl diameter for piston A is slightly larger (48 mm) than for piston B (46 mm) in order to compensate for the crevice volumes due to additional screws and optical windows. Both pistons have identical piston top-land crevice volumes (1.5 cm³) and squish height: 1 mm between the piston crown and the firedeck.

The engine was equipped with a common-rail injector placed at the center of the cylinder head in vertical position. Injector specifications are given in Table 2, and Figure 3 shows the position of the fuel sprays within the piston bowl at TDC.

Table 2. Injector specifications

Injector system	Common rail
Nozzle	VCO
Static flow rate	220 ml / 30s / 100 bar
Spray angle	140°
Number of holes	6
Hole diameter	131 μm

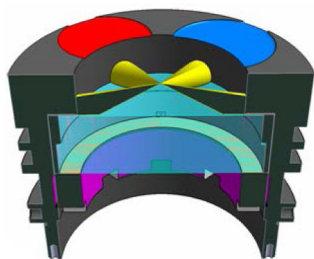


Figure 3. Position of fuel sprays in the piston bowl at TDC

Fuels

Pure methane was mostly used as the premixed fuel. Methane was supplied from a high pressure bottle, and flow rate was controlled with a dedicated flow meter which was specifically calibrated for methane gas.

A limited number of experiments was also performed with iso-octane as premixed fuel. The aim of these specific experiments is to investigate the influence of a higher fuel reactivity for the premixed fuel in dual-fuel combustion mechanisms.

A diesel surrogate fuel referred to hereafter as "GMA" was used in all the experiments. GMA fuel is a mixture of 36% iso-octane and 64% n-Decane by vol. This fuel blend has an overall cetane number of approximately 53, which is very close to the cetane number of commercial European diesel fuel. The use of a surrogate fuel was necessary in order to perform PLIF measurements and thereby avoid undesired fluorescence of aromatic and naphthenic compounds, which are present in the majority of commercial fuels. The use of GMA, which does not contain aromatic compounds, also limited fouling of the optical windows since in-cylinder soot formation was significantly reduced.

Test Cell

Figure 4 presents a schematic of the test cell used in this study.

As illustrated in this figure, the intake flow consisting of air and N_2 was controlled with sonic flow nozzles by adjusting the upstream pressure. A heater allowed precise control of the intake temperature (typically $\pm 1^\circ\text{C}$) and a large plenum was used to dampen pressure waves in the intake line.

Direct injection (DI) of diesel surrogate fuel was performed with the centrally mounted injector described in Table 2. As explained in previous section, some experiments were performed with iso-octane as premixed fuel using a Port Fuel Injector (PFI) located 1 meter upstream of the intake plenum. In order to avoid condensation of the injected fuel, the intake pipe between the PFI injector and the engine was heated.

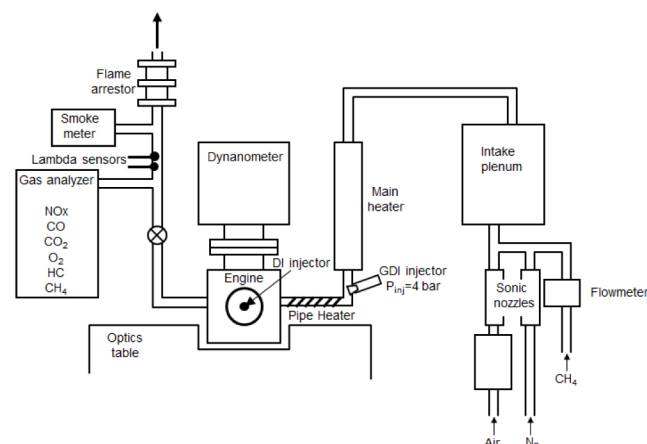


Figure 4. Schematic of the engine test cell

For each experiment, engine operating conditions (i.e. engine speed, pressures, temperatures...) and pollutant emissions were recorded with the commercial Morphee software, while synchronization of the injection settings, cameras and laser triggers were controlled with an in-house code. Calculation of the equivalence ratio was computed offline with the respective measurements of air mass and fuel flows. Air and methane mass flows were computed on-line using the Morphee software, but fuel flows from the DI and PFI injectors were computed offline from calibration curves. Cylinder pressure traces were measured with a cooled 6043A60 KISTLER pressure sensor and recorded every 0.5°CA . Recorded cylinder pressure traces were then analyzed with an in-house code to compute indicated mean effective pressure (IMEP) and apparent rate of heat release (dQa).

Optical Diagnostics

Combustion Experiments

The majority of experiments were performed on the basis of simultaneous measurements of natural luminosity (NL) and OH^* chemiluminescence. Unless otherwise stated, all combustion experiments were then performed with the setup shown in Figure 5. As illustrated in this figure, a beamsplitting optic was used to ensure that the combustion luminosity signal reached each of the two cameras.

Wavelengths above 400 nm were transmitted to a “Photron” high-speed camera. Images were acquired at 36000 images/sec and an exposure time of 28 μ s ($28 \mu\text{s} = 1/36000 = 0.2^\circ\text{CA}$ at 1200 rpm). A combination of a 500 nm short wave pass (SWP) and a wide band-pass filter BG 39 (370-570 nm) was placed in front of the 85 mm f/1.8 lens. With this combination of optics, light recorded was in the 400-500 nm range, limiting the contribution of soot incandescence on the collected signal.

Wavelengths below 350 nm were reflected by the beam-splitter to an intensified “PIMAX-II” camera equipped with a UV 45 mm f/1.8 lens to record OH*chemiluminescence. A 16 nm wide band-pass filter centered at 312 nm was used to detect chemiluminescence of OH* species and a 358 nm SWP filter was used to limit the contribution of other species (e.g. CH* chemiluminescence, soot incandescence). The exposure time for chemiluminescence imaging was set at 28 μ s.

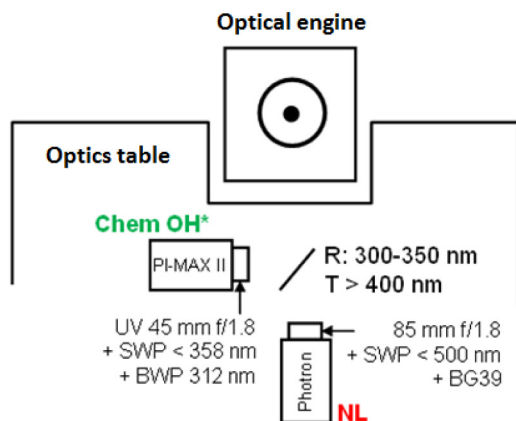


Figure 5. Setup for simultaneous combustion imaging of NL/OH*chemiluminescence

PLIF Experiments

Figure 6 shows the optical setup used for PLIF-tracer experiments. The aim of these experiments was to characterize the temporal and spatial evolution of the in-cylinder fuel distribution of pilot fuel. The PLIF experiments were performed in motored engine conditions using pure nitrogen instead of air. A concentration of 1% naphthalene by mass was added as a molecular tracer to the GMA fuel. PLIF of the naphthalene fuel tracer mixture was excited using the fourth harmonic (i.e. 266 nm wavelength) of a Nd:Yag laser. A set of optics was installed between the laser and the engine to provide a ≈ 1 mm thick horizontal laser sheet that entered the combustion chamber through the sapphire ring and the optical windows housed within the walls of Piston B (see Figure 2).

Figure 7 presents the position of the laser sheet relative to the fuel sprays. As illustrated in this figure, the width of the laser sheet was limited by the optical access in the piston.

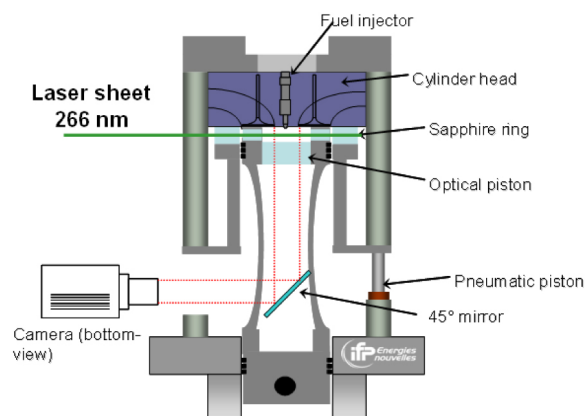


Figure 6. Setup of the optical engine for the PLIF-tracer experiments

A PI-MAX II intensified camera with a UV 45 mm f/1.8 lens was used to detect tracer fluorescence. Fluorescence times are very short, so the camera exposure duration was limited to 50 ns. A 300 nm long wave pass (LWP) blocked elastically scattered 266 nm laser light and a band-pass filter centered at 340 nm was used to isolate naphthalene fluorescence.

The YAG laser was triggered at 10 Hz, so only one image per engine cycle was collected. For each crank angle of interest, ten images corresponding to ten different engine cycles were acquired.

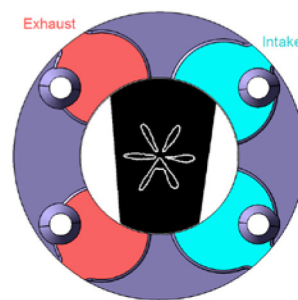


Figure 7. Position of the laser sheet relative to the fuel sprays

Note to the reader: The intensified CCD camera used in the present study could only record one image per cycle at 1200 rpm. Consequently, all single-shot OH*chemiluminescence and PLIF images correspond to different engine cycles. In contrast, NL images were acquired with a high-speed camera that provides time-resolved information within the same engine cycle. In this paper, all NL images are extracted from the “most representative cycle”. This has been determined by applying a method of least squares in which each of the single cycle in-cylinder pressure traces was compared to the ensemble-averaged pressure trace, thus ensuring the selected single-cycle pressure trace is representative of the ensemble-averaged cycle.

RESULTS AND DISCUSSION

This section is dedicated to the experimental results and is divided into major two parts:

- Influence of equivalence ratio is first studied to highlight the influence of this key parameter on dual-fuel combustion.
- PLIF experimental data are then analyzed and provide complimentary information relative to the auto-ignition behavior.

It should be noted that injector dribble phenomena are discussed in the [appendix](#) of this paper to explain the source of some intense bright spots observed in NL images. Although this phenomenon is not specifically linked to dual-fuel combustion, we will show that its consequences are likely to be more significant than in conventional diesel combustion strategies.

Equivalence Ratio Sweep

Dual-fuel combustion strategies that were investigated revealed that the equivalence ratio of the premixed fuel is a very important parameter in terms of its impact on combustion behavior. In order to study the influence of the equivalence ratio of the premixed fuel, the flow of methane was varied while maintaining all the other engine parameters constant. Engine operating conditions for this equivalence ratio sweep are detailed in [Table 3](#). As indicated in this table, DI settings for the pilot fuel were kept constant to provide a quantity of pilot injection equal to 1.2 mg/stroke, which corresponds to an equivalence ratio of approximately 0.05. When increasing the mass flow rate of methane, the pilot quantity was kept constant, but the ratio (i.e. the mass of pilot fuel relative to the total mass of fuel) is decreasing. The DI settings indicated in [Table 3](#) correspond to the electrical start of injection. Injection rate measurements revealed that the real fuel injection occurs between 348.8 and 351.8 °CA. Because experiments were performed with constant intake pressure at 0.7 bar absolute, air flow was reduced when increasing mass flow of premixed methane.

Table 3. Engine operating conditions for the equivalence ratio sweep

	ϕ	ϕ_{premixed}	ϕ_{pilot}	Pilot ratio	Air flow	IMEP
	[-]	[-]	[-]	[%]	[kg/h]	[bar]
$\phi=0.94$	0.94	0.88	0.06	6.5	10.78	5.32
$\phi=0.85$	0.85	0.79	0.06	7.1	10.88	5.01
$\phi=0.75$	0.75	0.69	0.06	7.9	11.10	4.53
$\phi=0.66$	0.66	0.60	0.06	8.8	11.37	4.08
$\phi=0.56$	0.56	0.50	0.06	10.2	11.62	3.39
$\phi=0.44$	0.44	0.39	0.05	12.4	11.98	1.95
$\phi=0.33$	0.33	0.27	0.05	16.6	12.15	0.49
$\phi=0.05$ (DI only)	0.05	0.00	0.05	100.0	12.55	0.52

For all experiments	
Pilot settings	SOI _{pilot} = 346 °CA - DOI _{pilot} = 400 µs - m _{pilot} = 1.2 mg/st
Intake conditions	P _{intake} = 0.7 bar abs. - T _{intake} = 40 °C - 0% dilution
Exhaust conditions	P _{exhaust} = 1 bar abs. - T _{exhaust} = 94 to 487 °C depending on ϕ
Other	Injection pressure = 400 bar - Coolant temperature = 70 °C

[Figure 8](#) presents cylinder pressure traces and apparent rates of heat release for experiments listed in [Table 3](#). The reader should note that three sets of curves corresponding to equivalence ratios $\phi=0.05$, $\phi=0.56$ and $\phi=0.94$ have been deliberately highlighted (i.e. thicker lines) in this figure. Experiments for $\phi=0.05$ were performed without any premixed

fuel to provide a reference case corresponding to a pilot fuel injection only. Experiments for $\phi=0.56$ and $\phi=0.94$ will be referred as low and high equivalence ratio cases hereafter.

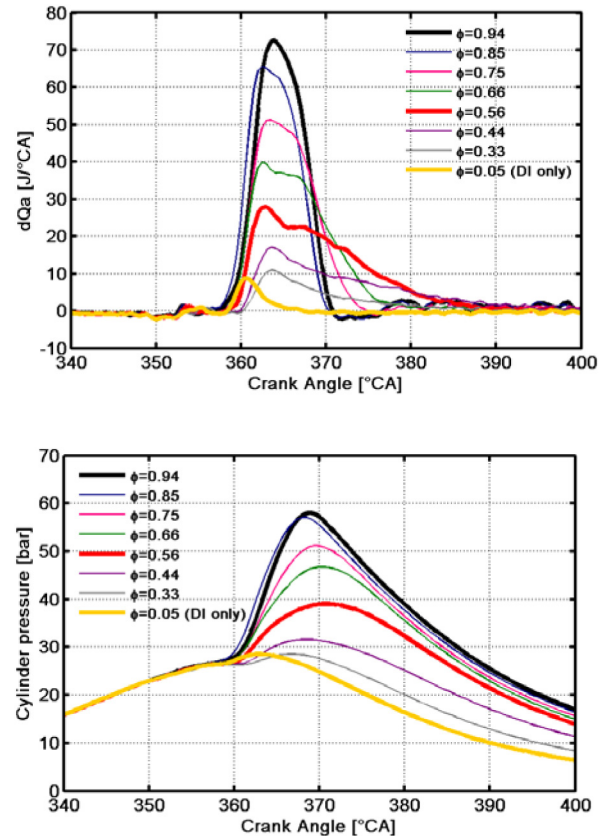


Figure 8. Cylinder pressure traces and apparent heat release rates for an equivalence ratio sweep. Engine operating conditions are reported in [Table 3](#)

Analysis of NL Movies

[Figure 8](#) shows that the shape of the rate of heat release strongly depends on the equivalence ratio of premixed fuel. However, it is interesting to note that the initial rates of heat release (and therefore the combustion speeds during the very early stages of combustion) are comparable regardless of the equivalence ratio. This tends to show that the initial stages of combustion are dictated by auto-ignition of the fuel corresponding to the pilot injection.

At low equivalence ratio ($\phi < 0.6$), the rate of heat release is characterized by a long asymptotic tail resulting in long combustion durations. [Figure 9](#) presents the NL images corresponding to the $\phi=0.56$ condition (thick red curve in [Figure 8](#)). This image sequence shows that combustion initiates near the walls of the piston bowl at 360-361 °CA, and then extends rapidly towards the center of the piston bowl although the very central region within the bowl is marked by a clear absence of NL signal. This first stage of combustion corresponds to the rapid increase of the rate of heat release visible in [Figure 8](#) between 360 and 363 °CA. Following this first stage, the rate of heat release progressively decreases.