

Experimental Evaluation of Compression Ratio Influence on the Performance of a Dual-Fuel Methane-Diesel Light-Duty Engine

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ABSTRACT

The paper reports an experimental study on the effect of compression ratio variation on the performance and pollutant emissions of a single-cylinder light-duty research diesel engine operating in DF mode.

The architecture of the combustion system as well as the injection system represents the state-of-the-art of the automotive diesel technology. Two pistons with different bowl volume were selected for the experimental campaign, corresponding to two CR values: 16.5 and 14.5. The designs of the piston bowls were carefully performed with the 3D simulation in order to maintain the same air flow structure at the piston top dead center, thus keeping the same in-cylinder flow characteristics versus CR.

The engine tests choice was performed to be representative of actual working conditions of an automotive light-duty diesel engine. Moreover, the test methodology was designed in order to carry out emission and fuel consumption (FC) estimation on New European Driving Cycle (NEDC) test procedure. A proper engine Dual-Fuel calibration was set-up respecting prefixed limits in terms of in-cylinder peak firing pressure, cylinder pressure rise, IMEP cycle-to-cycle variation, and gaseous emissions.

The results evidence a significant impact of CR on the THC level (mainly CH_4), as well as on the combustion efficiency. Benefits on combustion noise and full load performance in DF mode can be attained reducing the CR. The results also show a great potential on the CO_2 reduction beyond a NG substitution rate of 50%. Moreover, the study indicates that such limit could be overcome with specific combustion system design as well as a proper engine calibration.

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INTRODUCTION

The more and more stringent emission legislation, the instability of the fuel price and the uncertainties on fossil fuel reserves, push researchers to pursue high efficiency and low emission technologies in the field of internal combustion engines (ICE). The use of alternative fuels, together with the adoption of alternative combustion concepts are possible routes to reach these objectives. In the case of compression ignition (CI) engines, the combination of alternative combustion concepts as PCCI, HCCI, RCCI and gaseous fuels as natural gas (NG) lead to the dual-fuel (DF) combustion concept, that has received increased attention in recent years [1, 2, 3, 4, 5, 6, 7].

In DF CI engines, the gaseous fuel is premixed with air in the intake manifold and subsequently compressed as in a conventional diesel engine. A certain amount of high-cetane liquid fuel is then injected at

the end of the compression stroke, providing the energy to ignite the mixture and initiate the combustion. Within this framework, due to the high octane number (ON) of NG, DF combustion concept can be considered as a sort of reactivity controlled combustion (RCCI) combining direct injection (DI) of a high-reactivity fuel, with a premixed low reactivity fuel. In this way, combustion phasing and duration can be, to some extent, controlled through the reactivity gradient [1, 2]. By the way, it is well assessed that the DF combustion concept has the benefits to reduce NOx and particulate matter (PM) emissions [8].

The high ON of NG makes the fuel suitable for CI engines which usually operate with relatively high compression ratio (CR).

Furthermore, NG is mainly constituted by methane (CH_4) and has a favorable H/C ratio in terms of CO_2 reduction compared to

conventional fuels. However, at lower loads the low reactivity premixed fuel tends to generate over lean NG-air regions inside the combustion chamber with subsequent higher CO and THC emissions [1, 7]. Moreover, DF systems based on port fuel injection of the gaseous fuel enable a relatively simple conversion of existing engines, maintaining the full diesel capability in case of NG lack [9].

Various investigations have also been conducted to explore the parametric effects on the combustion and emissions performance. They report that through the air/fuel ratio and injection parameterization the emission characteristics of a DF engine at low load conditions can be significantly improved, especially in terms of THC [1, 7]. The effects of the premixed fuel supply method combined with variable in-cylinder charge bulk motion have also been investigated, reporting that port fuel NG can probably induce some stratification capable to influence the flame intensity and propagation and then the initial combustion duration. Accordingly, DF combustion is significantly influenced by the premixed mixture and its distribution inside the cylinder [10].

Among the wide literature on DF light duty engines, there is a lack of information about how the combustion performance and emissions characteristics are influenced by the CR variation in the case of DF light duty (LD) diesel engine. Available information are limited to the CR effects on knocking limits and combustion noise for an indirect injection (pre-chamber) engine and in a CR range not applied in current powertrains [11]. Moreover, the information in this regard is generally bounded at few operating points, without the evaluation on the DF performance in a wide area of the engine working map. By the way, since the development of CI ICE pushes the CR towards even lower values, with the aim of reducing NOx and soot emissions, the evaluation of the CR effect assumes a greater importance.

On the basis of the above reasons, and taking into account the current trends in modern automotive diesel engines, it appears interesting to evaluate how the DF combustion is affected by the CR variation [12, 13, 14]. This study has been carried out employing a single-cylinder research engines, having the same combustion system architecture of a Euro5 engines and equipped with a NG/diesel DF fuelling system.

Normally, in DF system a single diesel injection is considered as pilot source for NG combustion. Nevertheless, the modern common rail systems allow the use of multiple injections. Furthermore, the most of current automotive engines are calibrated with a pilot plus main injection strategy in order to control the engine noise and the combustion stability. This aspect was also considered in the present study, performing experiments in DF mode with both a single pilot injection and the conventional Euro 5 injection calibration composed by a pilot plus main diesel injection.

The activities were focused on characterizing the effects of the premixed methane on the global combustion and engine-out emission performances considering practical limits on comfort and combustion

stability parameters as in-cylinder pressure gradient, cycle to cycle variation, etc.. Moreover, by means of a specific test methodology, an estimation of the global emissions, performances and CO₂ on the NEDC was also performed.

The experimental apparatus, the test methodology and the results are described in detail in the following sections.

EXPERIMENTAL SET-UP AND TESTING METHODOLOGY

Single Cylinder Engine

The experimental activities were conducted on a prototype single cylinder CI research engine which has a combustion system design derived from a Euro5 (Eu5) compliant four cylinder engine, representative of the current light duty (LD) diesel technology. The cylinder head is derived from production series but has been modified to run in single-cylinder mode. Auxiliary systems for boost, cooling, lubrication, etc., are not directly coupled to the engine, thus permitting the whole flexibility for the control of these parameters without influencing the load conditions. The customized electronic control unit allows modifying all the engine control parameters. The boosted air is provided and controlled by an external air compressor. This approach offers a great flexibility on the operating conditions. As a result, the combustion characteristics are comparable to those of the reference engine making the outputs transferable to the real multi-cylinder engine.

The engine is equipped with a DF system able to run the engine with liquid and gaseous fuels in the intake manifold [13]. The port fuel injection (PFI) typical for DF configurations is realized through a dedicated PFI injector placed in the not-swirled runner just up-stream the intake valve. The PFI injector, in the case NG injector, is fed by a separate NG low pressure line operating at a pressure of about 5barG.

The diesel and NG FC are measured by means of a gravimetric balance (AVL 715S) and thermal (Brooks SLA 5860) mass flow meters respectively. The engine layout is depicted in [Figure 1](#), while [Table 1](#) describes the main engine characteristics.

[Table 1. Main engine geometrical characteristics.](#)

Displaced volume	477 cc
Stroke	90 mm
Bore	82 mm
Compression ratio	16.5:1; 14.5:1
Number of Valves	4
Diesel Injection System	Common rail (max injection pressure=1800 bar)
Diesel Injector	Solenoid 7 holes microsac
PFI Injector	Multihole - Natural gas

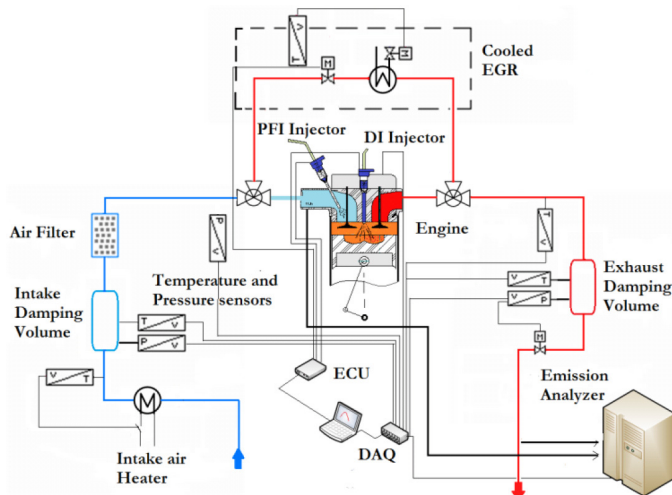


Figure 1. Research engine experimental layout.

The pressure traces were measured by a Kistler piezo quartz transducer fitted inside the combustion chamber. The apparent heat release (HR) and heat release rate (HRR) were real-time calculated by the AVL IndiMicro indicating system based on the pressure signal. The pressure signal was averaged over 128 consecutive cycles. The filter smoke number (FSN) was measured by an AVL 415S smoke-meter and converted in PM emission by means a consolidate correlation implemented in the device processor [15]. Whereas the smoke meter reached its measurement limit (≤ 0.01 FSN), its data were cross checked with the more accurate AVL Microsoot Sensor. The THC (total unburned hydrocarbon) and NO_x emissions were detected by the ABB and Ecophysics devices respectively while the MHC (methane hydrocarbon) and CO-CO₂-O₂ by the Emerson devices. The NMHC (non-methane hydrocarbon) were evaluated through the difference of THC and MHC emissions.

The main characteristics of the fuels used, for direct injection (Commercial European diesel) and for port injection, (methane) are listed in Table 2. For this study, pure methane (CH₄) was preferred to the NG because of its fixed physical-chemical characteristics, whereas NG has a variable composition depending on the source of supply. However, despite NG represents the reference commercial fuel, the use of CH₄ does not invalidate the results of the work.

Table 2. Fuel properties.

Feature	EN590 Diesel	Methane
Density [kg/m ³] STP	840	0.788
Autoignition temperature [°C]	300	595
CN	53	-
ON	-	≥ 120
LHV [MJ/kg]	42.95	49.5±0.2
AFR _{stoich} [-]	14.7	17.2
H/C [-]	~1.86	4

TEST METHODOLOGY

The test procedure was designed in order to characterize and evaluate the performance and emissions of a dual fuel (DF) combustion using methane (CH₄) as premixed fuel setting the compression ratios (CR) at 16.5 and 14.5. Considering the current trends in automotive diesel technology, the two CR were selected as representative of the border values of the most engines present on the market today [14].

As better described later, the methodology chosen for the experimental campaign was not only aimed to a simple parametric analysis of the CR on DF combustion process, but also to carry out quantitative comparison between a diesel and a dual fuel configuration employing practical engine calibration in terms of efficiency, pollutant and CO₂ emissions estimated on NEDC homologation cycle.

The CR was varied changing only the piston bowl design. The designs of the two piston bowls, corresponding to the two CR values, were carefully performed with the 3D simulation. Starting from the design of the production series piston shape (CR 16.5), the new bowl shape was created according to the following guidelines:

- To maintain the same air flow structure at the end of compression stroke in order to assure about the same swirl and turbulence characteristic patterns versus CR;
- To avoid an increase in the squish height and keep constant the internal diameter, in order to increase the k-factor and to keep the same squish flow inside the bowl during compression stroke;
- To keep the same lip profile of the bowl in order to guarantee the same structural robustness at the rated power when the engine operates in diesel mode.

In this way, all the characteristics of air-flow mixing process (spray/wall interaction, spray dispersion, premixed gas distribution etc.) and the oxidation pathway of the combustion products in the modified bowl shapes should be similar to those in the original production piston. Thus, the effects of the CR as geometrical (and thermodynamic) parameter on the engine behavior should be quite isolated.

Figure 2 shows the bowl geometries for both CRs.

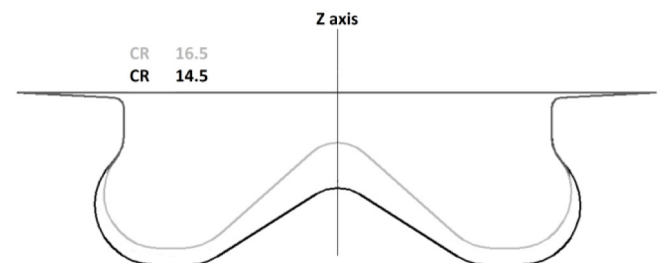


Figure 2. Bowl geometry for both compression ratios.

The air flow patterns for the two CRs at 20 crank angle degrees before the TDC are plotted in Figure 3. These plots are the result of an engine cycle simulation without injection performed with the

KIVA 3V code. Additional details on the CFD simulations can be found in [12]. The main air flow patterns appear very similar, with the main difference located mainly in the central part of the bowl.

Also the swirl ratio values and the moment of inertia versus crank angle, not reported for brevity, were very similar confirming that the two bowl shapes have effect mainly on the thermodynamics of the process.

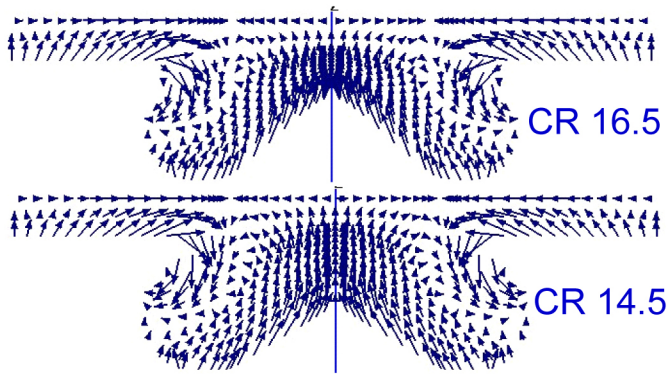


Figure 3. Air flow patterns for the two selected CR values 20 crank angle degrees before the Top Dead Center. The arrows are plotted with the same scale for all CRs.

A nominal Eu5 calibration map was adopted for engine reference calibration. It was directly derived from a production series four-cylinder engine equipped with a DOC + close coupled DPF featuring the Eu5 emission limits [16]. However, the analysis of the present work only takes into account the engine-out emissions.

The first part of this study was targeted at the selection of the diesel fuel injection strategy when the engine operates in DF mode. Usually, the DF configurations employ a single direct injection of diesel fuel to ignite the bulk gas-air charge. On the contrary, current CI diesel engines are generally calibrated with a pilot+main injection strategy, while the most advanced calibrations can employ also multiple injection strategies [17]. Both injection modes (single pilot and pilot+main injection) were preliminary screened in several operating points. As detailed later, the pilot+main injection strategy, in the following indicated as double pilot injection for DF operations, provided a valuable effect on the control of the cycle-to-cycle IMEP coefficient of variation (COV_{IMEP}) and on the pressure rise rate (PRR) in DF combustion mode. As an example, Table 3 reports the COV_{IMEP} and PRR values for two engine operating points at 50% of diesel/ CH_4 substitution ratio on mass basis (defined later) with single and double pilot injection. Similar results on double pulse injection in DF engines were obtained by Aroonsrisopon *et al.* [18].

Table 3. Comparison of COV_{IMEP} and PRR for single pilot and double pilot injection strategy in two engine operating points at a CR of 16.5.

Test point [speed x bmep]	CH ₄ /diesel substitution ratio [%]	Injection strategy	COV_{IMEP} [%]	PRR [bar/deg]
1500x5	50	Double	1.22	2.9
1500x5	50	Single	1.47	3.2
2000x13	50	Double	0.87	4.9
2000x13	50	Single	0.94	6.5

For these reasons, the double injection strategy was preferred to the more conventional single pilot injection and therefore employed in the whole experimental campaign.

The experiments were conducted at steady-state operating conditions keeping constant the combustion barycenter namely MBF50% (it represents the 50% of the total burned fuel mass) as settled in the reference Eu5 calibration map. The settled reference MBF50% values were assumed to reach the best compromise between the indicated efficiency and the prefixed emission values. Therefore, the DF tests were performed keeping constant all the injection parameters, included the dwell time period between 1st and 2nd injection, and using the nominal values of the reference Eu5 engine calibration. In each tested operating point, the main injection quantity was obviously reduced with the increase of the premixed CH_4 in order to reach the same indicated mean effective pressure (IMEP) of the diesel only mode and the start of the main injection was varied in order to reach the reference MBF50% values.

By the way, the imposed constant MBF50% through the adopted injection strategy is in agreement with the combustion closed loop control methodology recently released for the most recent generation automotive engines [20].

The reference test points for this campaign are reported in Table 4. They were chosen as the most representative operating points of the real engine of equal unit displacement during the execution of the UDC (Urban Driving Cycle) sub-cycle of the NEDC homologation cycle. Indeed, as shown in Figure 4, the selected points are placed in the speed-load plot of the NEDC cycle performed by a vehicle with 1490 kg of inertial mass, equipped with the corresponding four-cylinder engine. In this paper, even if the variables that identify the test points are the speed and the IMEP, to facilitate the interpretation of the correspondent test point of the multi-cylinder engine, to indicate the operating points, the authors always refer to the brake mean effective pressure (BMEP). Refer to Table 4 for the correspondence between BMEP and IMEP. Moreover, in the text, notations like 1500x2 are adopted for the test points.

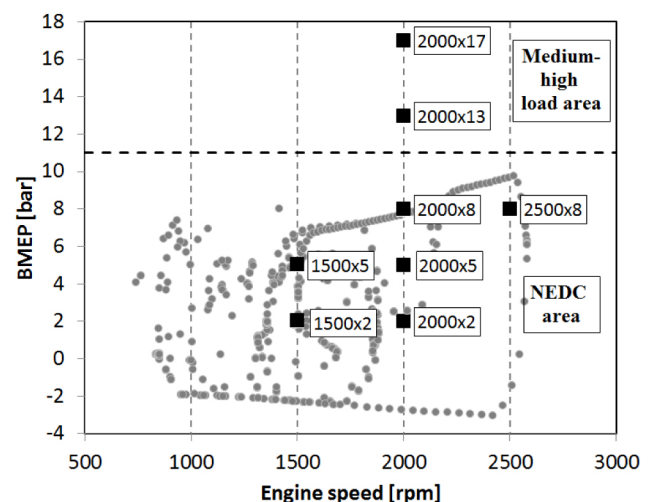


Figure 4. Position of the performed stationary operating points in the speed-load plot of a vehicle with 1490 kg of inertial mass, equipped with the multi-cylinder engine over the UDC (Urban Driving Cycle) cycle.

Additionally, two high load points were included in the test point matrix, 2000 rpm and 13 bar BMEP and 2000 rpm 17 bar BMEP, representing medium speed - high load and medium speed - full load conditions respectively. These conditions are worth being considered also because less explored in literature on DF combustion features. On this way the CR variation effects on the engine behavior at high/full load conditions was also evaluated. In the following, the diesel and dual fuel combustion modes are abbreviated by D and DF respectively.

Table 4. Selected steady-state test points and main operating parameters.

Test point [speed x bmep]	IMEP [bar]	BMEP* [bar]
1500x2	3.3	2
1500x5	7.00	5
2000x2	4.00	2
2000x5	7.00	5
2000x8	10.50	8
2000x13	15	13
2000x17	19	17
2500x8	10.60	8

BMEP* of the real four-cylinder engine of equal unit displacement

Test point [speed x bmep]	EGR [%]	Q_{pil}^{**} [mm ³ /cycle]	Intake Temp. [K]	m_d [kg/h]	P _{rail} [bar]
1500x2	44	1.00	336	0.41	620
1500x5	33	1.50	333	0.78	765
2000x2	38	1.00	338	0.61	714
2000x5	32	1.50	336	0.98	885
2000x8	26	1.00	340	1.40	1248
2000x13	0	1.00	333	1.95	1146
2000x17	0	1.00	323	2.51	1207
2500x8	24	1.00	340	1.59	1240

Q_{pil}^{**} = 1st injection pulse in DF mode

As claimed before, a nominal Eu5 engine calibration was considered as reference when the engine works in diesel mode. However, to perform tests in DF mode several criteria for engine parameter regulation were chosen.

The first considered parameter was the Exhaust Gas Recirculation (EGR). Assuming the EGR as the main driver for the NOx emission control, for test points employing EGR, it was varied in order to keep constant the engine-out NOx values and equal to those expected by the Eu5 calibration map.

As already reported in the introduction, the three of the main issues related to the DF operating mode in CI engines are the higher PRR, the higher COV_{IMEP} and the greater MHC emissions, especially at low loads. Therefore, DF combustion modes were explored imposing the limits reported in Table 5.

The authors settled the maximum limit of the engine-out MHC emissions to 5000ppm, in order to establish a limit on the combustion efficiency reduction and of the aftertreatment device workload in the assumption of the use of an oxidation catalyst.

Table 5. Constrain values on the engine operating conditions.

Constraints	Maximum values
PRR	80 bar/ms
pfp	160 bar
MHC	5000 ppm
COV _{IMEP}	3 %

The limit of 160 bar of peak firing pressure (pfp) was imposed as thermomechanical structural limit of the system. The limit of 80 bar/ms on PRR was imposed to preserve the current noise vibration harshness (NVH) standards of modern engines. A COV_{IMEP} of 3% was considered as upper limit for an acceptable cycle-to-cycle stability of the engine.

Before every measurement about 2-3 minutes to stabilize the engine outputs were observed.

The percentage of the premixed CH₄ can be quantified on mass basis (r_p) and on energy basis (r_{pe}) according to the following equations:

$$r_p = \frac{m_{CH_4}}{m_{CH_4} + m_d} = \frac{m_{CH_4}}{m_{tot}}; \quad r_{pe} = \frac{m_{CH_4} \cdot h_{CH_4}}{m_{CH_4} \cdot h_{CH_4} + m_d \cdot h_d}$$

Where m_{CH_4} and m_d are the mass flow rates of CH₄ and diesel fuel respectively, while h_{CH_4} and h_d represent the lower heating values (LHV) for methane and diesel fuel respectively. From here forward, the considered premixed ratio r_p is only regarded on mass basis.

The r_p is also used for the calculation of the LHV when using both the premixed (PFI) and the direct injected (DI) fuel:

$$LHV_{DF} = r_p \cdot LHV_{CH_4} + (1 - r_p) \cdot LHV_d$$

The relative air/fuel ratio (λ) is evaluated:

$$\lambda = \frac{m_{air}}{m_{diesel} \cdot AFR_{stoich_{diesel}} + m_{CH_4} \cdot AFR_{stoich_{CH_4}}}$$

The following relations were used for the calculation of the Indicated Specific Fuel Consumption (ISFC), the Indicated Specific Energy Consumption (ISEC), as well as the fuel (η_{fuel}), thermodynamic ($\eta_{thermal}$) and combustion (η_{comb}) efficiencies:

$$ISFC_{total} = \frac{\dot{m}_{tot}(Diesel+CH_4)}{P_{ind}} \quad ISEC = \frac{\dot{m}_{tot} \cdot LHV_{DF}}{P_{ind}}$$

$$\eta_{fuel} = \frac{1}{ISFC \cdot LHV_{DF}} = \eta_{comb} \cdot \eta_{thermal}$$

In the DF case, the ISEC is a useful parameter when comparing the specific consumption of engines employing different fuels.

RESULTS ANALYSIS

The main objective of the following paragraphs is to analyze the CR effect on the global performances and emissions in DF mode. The whole analysis is divided in three parts.