Chapter 9
Advances of the Natural Gas/Diesel RCCI Concept Application for Light-Duty Engines: Comprehensive Analysis of the Influence of the Design and Calibration Parameters on Performance and Emissions

Giacomo Belgiorno, Gabriele Di Blasio and Carlo Beatrice

Abstract The increasing energy demand together with the severe emission legislation of the transportation sector requires effective solutions for automotive propulsion systems. The transportation sector is responsible for 23% of the total CO_2 in the European Union. Advanced combustion concepts combined with alternative fuels, if properly tuned, have the potential to increase the efficiencies and lower emissions, compared to the conventional diesel or spark-ignition combustion. This is particularly valid when natural gas (a low reactivity fuel with low auto-ignition characteristics) and fossil diesel (the high reactivity one with high auto-ignition characteristics) are fuelled together in a compression ignition engine approaching the reactivity controlled compression ignition (RCCI) combustion. However, the use of these fuels in an advanced compression ignition engine increases the degrees of freedom for the tuning of the whole system. In order to carry out robust calibrations for efficiency maximization and pollutant minimization, in the whole engine-operating area, a detailed sensitivity analysis to the functional parameters versus fuel consumption and pollutant emissions is a paramount. Based on the experiences carried out at the laboratory of Istituto Motori of CNR, the chapter analyses the correlation between the main design and operating parameters versus the engine performance, providing a scale of degree of influence of the selected parameters as useful information for the engine calibration engineers.

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9.1 Combustion Concepts in Compression Ignition Engines

9.1.1 Conventional Diesel Combustion

Due to its higher thermal conversion efficiency compared to spark-ignition (SI) engines, the compression ignition (CI) engine commonly called diesel engine (from its inventor Rudolf Diesel) is the most diffused engines for medium and heavy-duty applications like commercial vehicles, rail and marine applications. The diesel fuel is injected directly into the cylinder at the end of the compression stroke and then ignited due to the high temperature and pressure reached by the air–fuel mixture near to the top dead centre (TDC). High fuel injection pressure is used to have a desired fuel spray penetration into the combustion chamber with high velocity and good atomization levels. Figure 9.1 shows the typical heat release rate (HRR) trace from the conventional diesel combustion (CDC); the first phase is characterized by the short period called ignition delay (ID) that represents the period between the start of combustion (SoC) and the start of injection (SoI); after the ID period, a spontaneous auto-ignition process starts due to the high temperature and pressure of air–fuel mixture, and this phase is called premixed combustion characterized by the high heat-release gradient. After the premixed combustion and during the injection process, the HRR is controlled by the rate at which mixture becomes available for burning (mixing-controlled combustion phase). Finally, the heat release continues at a lower rate well into the expansion stroke (late combustion phase) (Heywood 1988).

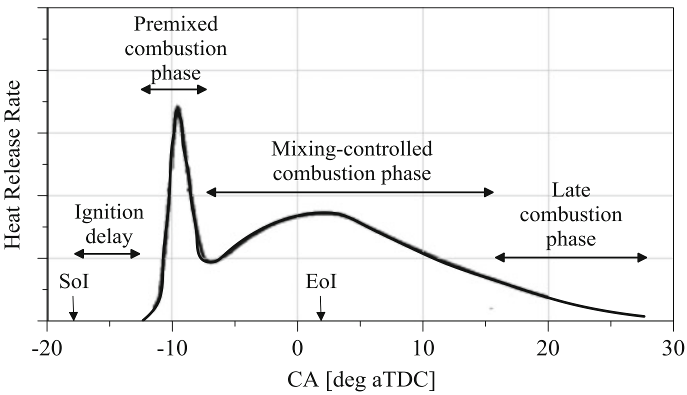


Fig. 9.1 Typical heat release rate identifying different conventional diesel combustion phases

The CDC is characterized by several advantages compared to the SI engine. Higher combustion efficiency, with a higher brake thermal efficiency due to the higher compression ratio, adopted generally around 15:1–18:1 instead of 9:1–10:1 for SI engines. No risk of pre-ignition or knock phenomena is observed due to the different diesel fuel characteristics compared to the gasoline.

One of the disadvantages of the compression ignition (CI) engines is the complex fuel injection system that has a high impact on the complexity of the control strategy and on the cost of engines. While regarding the engine-out emissions, the CI engines have a significant problem regarding the NO_x and soot emissions trade-off (Dec 1997). This is explained by the local fuel-rich zones and high local combustion temperatures that give rise to soot and NO_x, respectively. Fuel properties are also important for the likelihood of soot production. However, Kufferath et al. present an interesting result obtained in a diesel engine for passenger cars in real driving condition, they have demonstrated that combining high level of EGR, thermal management strategy and aftertreatment systems (SCR and DPF) permits to reduce drastically the nitrogen oxide emissions under real driving conditions without penalizing the CO_2 emissions as well (Kufferath et al. 2018).

9.1.2 Dual Fuel Reactivity Controlled Compression Ignition (RCCI)

Commonly, the dual fuel concept adopted the high reactive direct-injected diesel, while different low reactive fuels are used for the port injection (i.e., natural gas, ethanol and gasoline) (Di Blasio et al. 2013, 2015; Benajes et al. 2015). A dual fuel RCCI combustion process involves two different fuels with different reactivity (auto-ignition resistance). The low reactive fuel is injected via port fuel injector fitted in the intake manifold and the fuel is premixed with air during the intake stroke, while the high reactive fuel is direct injected adopting the common diesel fuel injection system.

The application of the dual fuel combustion concept consists of three main phases. A homogeneous charge of air and low reactive fuel is compressed while near the TDC a high reactive fuel is direct injected and mixed with the air–gas mixture during the ignition delay. After the ignition delay, the high reactive fuel is ignited by the high pressure and temperature from the compression, which in turn ignites the main fuel (low reactive fuel). Finally, the ensuing combustion occurs mainly through the flame propagation of the high auto-ignition fuel–air mixture (Königsson 2014; Karim 2015). Figure 9.2 shows an example of the in-cylinder pressure and HRR traces of the conventional diesel and dual fuel combustions. Both cases suppose the double direct diesel injection strategy. A substitution ratio (Sr) equal to 50% for the DF combustion, defined as (see Eq. 9.1) the ratio between the methane fuel mass and the total fuel mass (diesel plus methane), is adopted. The DF case shows a higher low-temperature heat release compared to the CDC.

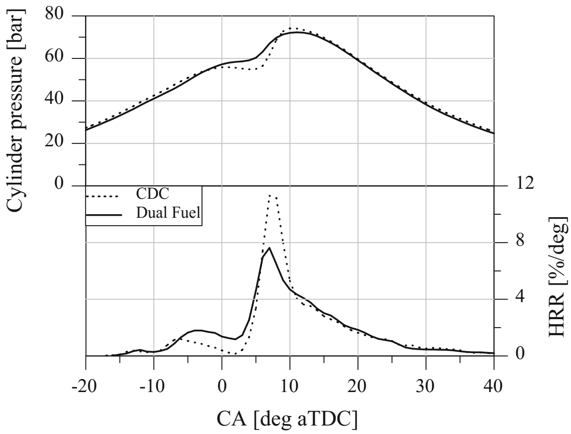


Fig. 9.2 In-cylinder pressure and heat release rate (HRR) for CDC and dual fuel combustion (Sr = 50%) at 2000 rpm and 7.0 bar of IMEP

This is due to the involvement of the premixed methane during the pilot diesel combustion. As a consequence, the max peak of the premixed combustion phase lowers and the combustion speed increases leading to advantages in terms of NO_x and soot formation.

In recent years, alternative combustion concepts for dual fuel engines have been investigated. More attention to the diesel-ignited gas engine since the engine is well suited for reactivity controlled compression ignition (RCCI) combustion (Kokjohn et al. 2011). In this concept, the high reactive fuel is injected early during the compression stroke compared to the conventional dual fuel, to promote mixing within the low reactive fuel and control the combustion characteristics via the reactivity of the mixture (Valladolid et al. 2017). In contrast to the conventional dual fuel, the diesel is injected early in the compression phase with either multiple injections or one single injection. RCCI has proven to be effective in achieving high fuel efficiency and low pollutant emissions (Splitter et al. 2011). Function of the fuel reactivity, Curran et al. (2014) defined the RCCI like a combustion process between the partially premixed combustion (PPC) (Belgiorno et al. 2018a) and direct injection compression ignition (DICI) combustion. RCCI uses two fuels with varying proportions while the PPC runs efficiently on any fuel by adjusting fuel injection and other engine parameters.

One of the advantages of the dual fuel combustion engines is the application of the concept in conventional diesel engines thus without substantial hardware modification. Moreover, adopting methane as alternative fuel permits a significant CO_2 emission reduction (Besch et al. 2015) with a maximum theoretical reduction of 25%, related to the chemical characteristics, supposing the same engine efficiency. However, it has to be considered that one of the main problems of the DF combustion is the CH_4 slip, especially at low load condition. It has to be taken into account that the methane greenhouse gas potential (GHP) is about 25% more than that of CO_2. For this reason, a dedicated development of combustion calibration and aftertreatment systems is necessary to reduce the CO_2 equivalent emission to promote the development of DF combustion engines.

Thus, the dual fuel combustion concept would permit the combination of the benefits of the high diesel engine thermodynamic efficiency with the use of alternative fuels (lower CO_2). Compared to the conventional combustion, the DF exhibits same stable combustion (e.g., in terms of the cycle-to-cycle variation), and combustion noise and phasing controls thank to the flexibility of the diesel fuel injection strategies (Di Blasio et al. 2017).

As widely reported in literature, the main DF disadvantages are related to the lower combustion efficiency (typically in the range of 85–95%) due to the low reactive premixed fuel trapped in the crevice volumes and not involved by the flame front (Di Blasio et al. 2013, 2015). Additional issues derive from the non-optimized combustion chamber for DF RCCI operation as in this case study. While the DF technology is almost consolidated for the heavy-duty and marine engines (e.g., Volvo and Wartsila engines), possible engine technological and control advances still represent a challenge for the improvement of the DF concept for light-duty engines application.

9.2 Analysis of the Main Design and Calibration Parameters of Methane–Diesel Dual Fuel Combustion

In this chapter, some suggestions about the combustion chamber design and calibration parameters that most affect the combustion characteristics, emissions and efficiencies at low load conditions are reported. The dual fuel research activity was performed at Istituto Motori of the National Research Council of Italy. A single cylinder engine setup, equipped with a Euro 5 light duty combustion architecture, was utilized for the experimental campaign.

To make the results representative of the reference multi-cylinder engine (MCE), the boundary conditions or the control variables of the auxiliary systems (boost, EGR, oil cooling, water cooling, fuel cooling) are set as those of the reference MCE. The main engine geometrical characteristics are reported below in Table 9.1.

Table 9.1 Single cylinder engine geometrical characteristics

Displaced volume	477.5 cc
Stroke	90 mm
Bore	82 mm
Compression ratio	16.5, 15.5, 14.5
Number of valves	4
Diesel injection system	Common rail
Diesel injector	Solenoid 7 holes microsac
PFI injector	Multihole

The indicated pressures are measured by means of a Kistler 6125B flush mounted piezo-quartz transducer, fitted inside the head glow plug hole and acquired with a resolution of 0.1 CAD. The average pressure signal, for the calculation of the indicated mean effective pressure (IMEP) and the apparent heat release (HR) and HRR, is averaged over 128 consecutive cycles. The heat release analysis is based on the first law of thermodynamics as reported by Heywood (1988). The engine-out gaseous emissions in terms of THC, MHC, NO_x, CO, CO_2 and O_2 are measured using an integrated emissions test bench. Two dedicated flame ionization detectors (FIDs) have been used to measure the total unburned hydrocarbons (THC) and methane unburned hydrocarbons (MHC) engine-out emissions.

The commercial EN590 compliant diesel fuel and pure methane (CH_4) were used as direct and port-injected fuel, respectively. The fuel characteristics are listed in Table 9.2. Since natural gas is constituted by a mix of hydrocarbons, the composition and consequently its specifics are variable and dependent on the source of supply. For this reason, CH_4 was used because of its fixed physical–chemical characteristics.

Common dual fuel combustion adopts a single diesel injection strategy; however, a double diesel injection strategy has demonstrated a reduction of the heat transfer loss (before the TDC) and then an improvement in terms of gross-indicated efficiency (up to 1.5% units) with a reduction of the peak pressure rise rate (PRR_{max}) (of about 60%), THC and NO_x emissions levels (of about 10 and 30%, respectively) compared to the single injection strategy (Di Blasio et al. 2017; Belgiorno et al. 2018b). For these reasons, all the results reported in this chapter refer to the adoption of the double diesel injection strategy for both CDC and DF combustions.

The dual fuel combustion is characterized by the high amount of natural gas not involved by the flame front that produces high emission level of MHC and then low net-indicated efficiency compared to the CDC (Fig. 9.3), especially at low load (below 7 bar of IMEP), due to not optimal combustion process (long combustion duration, high combustion and heat transfer losses), similar trend was observed in literature (Di Blasio et al. 2015; Belgiorno et al. 2018b).

As widely recognized, the main penalty of the dual fuel concept application is the CH_4 emissions slip, especially at low load conditions. For this reason, the chapter mainly focuses on how the engine calibration and design parameters impact on its reduction by preserving or improving the global engine efficiency. During the

Table 9.2 Fuel properties

Feature	Diesel fuel (EN590)	Methane
Density (kg/m^3) STP	840	0.788
LHV (MJ/kg)	42.95	49.5 ± 0.2
A/F_{stoich} (–)	14.7	17.2
H/C (–)	~1.86	4

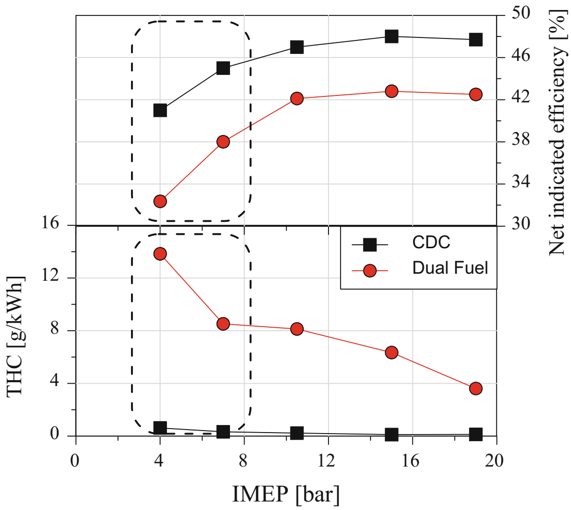


Fig. 9.3 THC and net-indicated efficiency trends function of IMEP for CDC and dual fuel (Sr = 50%) at 2000 rpm

parametric analysis of each of the considered variables, the combustion phasing (CA50) was kept constant and at the reference calibration value for diesel combustion, to have into a certain extent a fair comparison between CDC and Dual Fuel combustion (Di Blasio et al. 2015). The effect of each of these parameters has been investigated.

9.2.1 *Substitution Ratio on MHC and Efficiency*

In order to quantify and qualify the low reactive port-injected fuel, it is useful for defining the substitution ratio (Sr) with methane (CH_4) on a mass basis according to the following equation:

$$Sr[\%] = \frac{m_{CH_4}}{m_{CH_4} + m_{diesel}} \cdot 100 \tag{9.1}$$

where m_{CH_4}, m_{diesel} are the mass flow rates of CH_4, diesel fuel, respectively.

In order to discriminate the single effect of the substitution ratio, the analysis was performed without utilizing EGR and keeping constant the compression ratio (CR=15.5), the combustion phase (MBF50), and the engine load (IMEP). The effects on the MHC and on the gross-indicated efficiency trends, at the more critical low load conditions, are reported. For a more comprehensive analysis of the regulated emissions and efficiencies, it is possibly referring to literature results (Belgiorno et al. 2018b; Papagiannakis et al. 2010; Papagiannakis and Hountalas 2003).

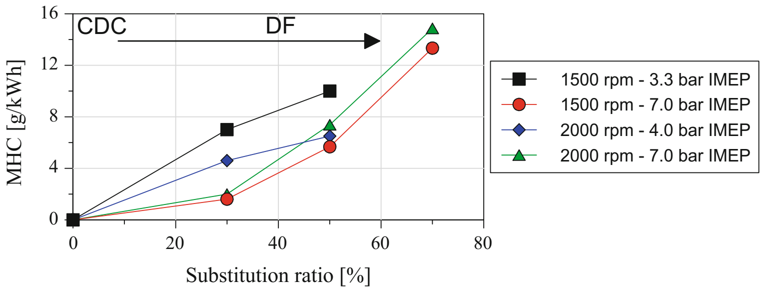


Fig. 9.4 Methane unburned hydrocarbon emissions function of substitution ratio

The CDC is characterized by very low unburned hydrocarbons emission level compared to spark-ignition and dual fuel combustion. In the DF case, most of the total unburned hydrocarbon (THC) is composed by the low reactive methane fuel (MHC), about 90–95% of the THC (Di Blasio et al. 2015; Fraioli et al. 2017), trapped in the squish and crevices volumes (Fraioli et al. 2017; Königsson et al. 2013; Di Blasio et al. 2017b). For this reason, it is fundamental to optimizing the DF combustion by reducing the MHC emission level also because the CH_4 emissions contribute massively to the greenhouse gaseous emissions (GHG) and with a factor of 25 in comparison with CO_2.

Figure 9.4 reports the net-specific MHC emissions function of substitution ratio for different test points. The MHCs drastically increase passing from the CDC to DF; 30% of Sr causes a THC-increase of 3 times. In particular, the MHC trend can be divided into two steps. The first step shows a less than proportional increase when varying the Sr from 0 to 30%; this can be justified by the prevalent diesel fuel able to involve more premixed methane. The second step shows an almost linear variation from 30 to 70% of Sr that is related to the gradual increase of the trapped methane in the crevices at the expense of the diesel fuel decrease. At low load (below 4 bar of IMEP), the maximum Sr adopted was 50% because of high level of MHC up to 5000 ppm was achieved and to minimize the CH_4 slip, a limit of Sr was imposed.

Regarding the gross-indicated efficiency (Fig. 9.5), passing from the CDC to DF, the thermodynamic efficiency decrease up to 15% units. This is explainable by both, the lower combustion rate that increases the heat transfer (HT) loss, and the lower combustion efficiency. For lower Sr values, the HT loss has a higher impact than the combustion loss vice versa for higher Sr values.

9.2.2 *Effect of the Compression Ratio on MHC and Efficiency*

The combustion chamber geometry is one of the most important design parameters influencing the engine performance and emission. To investigate the effect of the

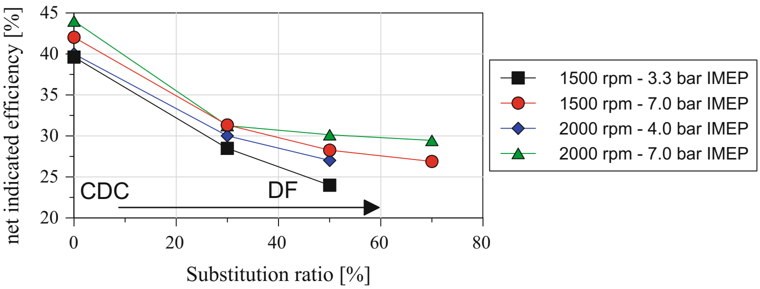
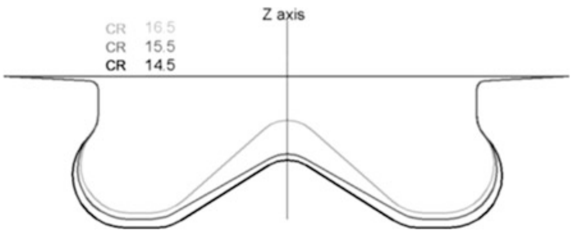


Fig. 9.5 Net-indicated efficiency function of substitution ratio

Fig. 9.6 Bowl geometries for all the compression ratio tested



CR (16.5, 15.5 and 14.5) on the DF combustion, three different piston bowl geometries have been designed. The variation of the CR target was realized by redesigning the bowl volume (Fig. 9.6). In particular, the bowl profile for each CR was defined by means of a 3D CFD simulation using the KIVA-3V code. Starting from the design of the production series piston shape (CR 16.5), the other bowl shapes were generated according to the following guidelines:

- same air flow structure at the end of the compression stroke to assure the same swirl and turbulence characteristics versus CR;
- same squish height and constant internal diameter to increase the k-factor and keep the same squish flow inside the bowl during the compression stroke;
- same bowl lip profile to guarantee the same structural robustness at the rated power.

The following analysis concentrates at low load engine operating conditions (from 3.3 to 7 bar of IMEP) at 1500 and 2000 rpm. A constant Sr and equal to 50% is adopted, employing a conventional double pulse diesel injection strategy (pilot-main) to control the combustion noise and/or the peak pressure rise rate (Di Blasio et al. 2015).

A high sensitivity of the MHC emission with the CR variation is detected independently of the engine speed and load condition in DF mode. In fact, lowering the CR, it is expected that the flame-quenching phenomenon would emphasise the MHC increase. Thus, following a rational thinking, the MHC reduction for smaller CRs is totally ascribable to the reduction of the premixed methane–air charge

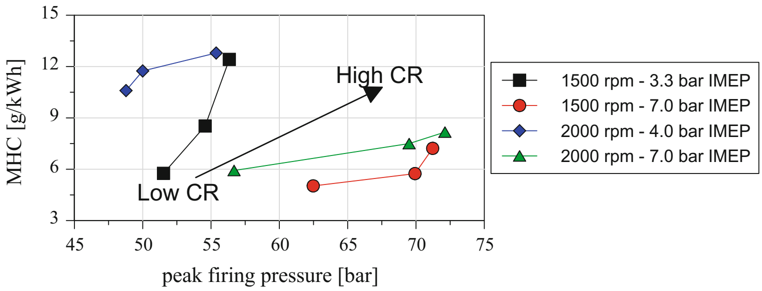


Fig. 9.7 Methane unburned hydrocarbon emissions function of peak-firing pressure, obtained by varying the compression ratio (CR)

trapped into the crevices volumes. In this regard, the crevices volume is assumed to be the annulus volume between the piston top and the first piston ring. Since the CR variation is realised through the bowl enlargement (see test methodology section), the top land crevice volume of the three CR was the same and the variation of the bowl/crevice (b/c) up to 10% units less compared to the high compression ratio value (Belgiorno et al. 2018b) (Fig. 9.7).

As the CR reduces, the maximum in-cylinder pressure drops with a significant impact on the air-methane mixture concentration trapped into the crevice volumes. Starting from the in-cylinder pressure traces and applying the ideal gas law at TDC, reducing the CR, a methane mass reduction of about 5 and 13% at 15.5 and 14.5 compared to the CR 16.5 was calculated. Thus, for reduced CR both the increase of the bowl volume combined with the lower-trapped methane mass in the dead volumes contribute to the reduction of the MHC emissions formation.

In agreement to literature results carried out on SI-PFI engines, the observed trends of MHC reducing the CR depend on three main factors (Belgiorno et al. 2018b; Russ et al. 1995):

- the b/c ratio increment produces a higher methane mass fraction trapped in the piston bowl despite the crevice volume;
- a lower in-cylinder pressure during compression stroke reduces the methane–air mixture packaging into the crevices;
- a higher gas temperature in the late expansion stroke promotes the methane oxidation.

However, the CR has an opposite effect on the cycle conversion efficiency, indeed, a reduction of about 3.0% units is detected on the gross-indicated efficiency, passing from CR 16.5 to 14.5, related to the higher in-cylinder HT loss, lower in-cylinder thermodynamic temperature and then higher combustion duration (Di Blasio et al. 2017; Belgiorno et al. 2018b) (Fig. 9.8).

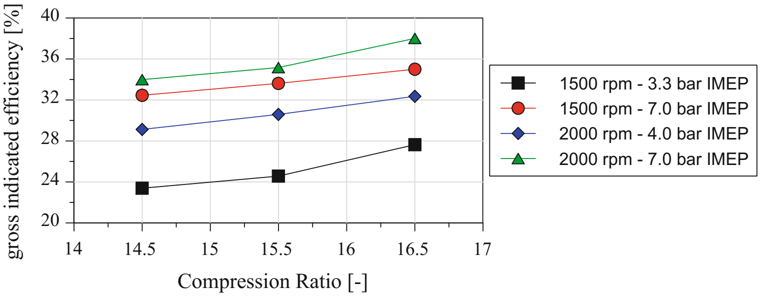


Fig. 9.8 Gross-indicated efficiency function of the compression ratio (CR)

9.2.3 Effect of Exhaust Gas Recirculation on MHC and Efficiency

It is worth investigating the effect of one of the most employed NO_x emission technologies for CI and SI engines, that is, the exhaust gas recirculation (EGR). In particular, the EGR dilutes the in-cylinder mixture reducing the maximum in-cylinder temperatures and then the NO_x formation. Additionally, for SI engines, it can permit, to some extent, the control of the knocking phenomena permitting to increase the specific rated power target. Thus, the EGR effect on efficiency and on MHC emissions is evaluated in DF mode.

The results of the EGR variation effect on MHC and gross-indicated efficiency are presented in Figs. 9.9 and 9.10. As previously explained, only the most critical low speed and load operating conditions have been considered adopting a Sr of 50%. Moreover, to discriminate the EGR effect, also the combustion phasing was kept constant, obviously, the SoIs are varied as a consequence.

Increasing the EGR rate, the MHC reduces (Fig. 9.9) and for values beyond 45%, the relative air–fuel ratio (λ) reduces to 1.07 and near to the stoichiometric value and the specific MHC emission (g/kWh) drops of about 55% compared to the

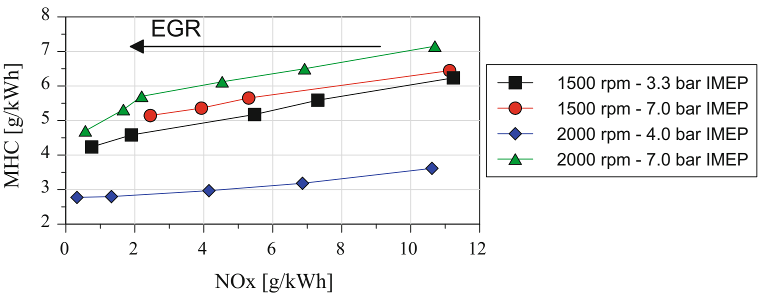


Fig. 9.9 Methane unburned hydrocarbons sensitivity versus NO_x emissions changing the EGR levels

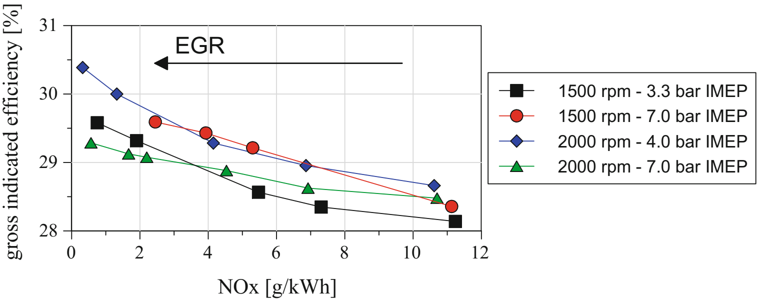


Fig. 9.10 Gross-indicated efficiency sensitivity versus NO_x emissions changing the EGR levels

no EGR case. The difference between the specific and raw MHC emission is a consequence of the reduced exhaust mass flow rate in the presence of EGR and then of lower in-cylinder pressure that reduces drastically the methane mass trapped into the crevices volumes and a higher flame velocity that promotes the methane oxidation (Belgiorno et al. 2018b).

The EGR is increased from 0 to 45% with an increase of the gross efficiency of about 1.0% units. With a high level of EGR, the pilot-ignition delay increases, due to the lower mixture reactivity (lower in-cylinder temperature and less oxygen). This permitting to improve the efficiency by reducing the heat transfer losses during the compression stroke. The higher premixed combustion permits to release more heat closer to the TDC, improving the thermodynamic efficiency.

On this basis of the presented results, it can be drawn that the EGR represents not only an important driver to reduce the NO_x (one order of magnitude less) but also the MHC (up to 55%) while improving the gross-indicated efficiency (up to 1.0% units) compared to the no-EGR strategy, without penalization in terms of soot and combustion noise that are always below the value obtained for CDC.

9.2.4 Effect of Air-to-Fuel Ratio on MHC and Efficiency

Many definitions of the dual fuel combustion can be found in literature, and in general, the DF combustion can be considered as a process in between the flame propagation (SI engine) and diffusive combustion (CI engine) processes (Königsson 2014; Karim 2015; Di Blasio et al. 2017a). For this reason, the in-cylinder air-to-fuel ratio represents a fundamental parameter in the control of the combustion process and emission level. In a compression ignition engine, the air-to-fuel ratio can be varied by means of the injection quantity and thus the load or by means of reducing the intake air mass through the intake throttle valve at constant load. The latter was adopted to investigate the effect of different air-to-fuel ratio impact, keeping constant the Sr equal to 50% and the load, without EGR. The throttle

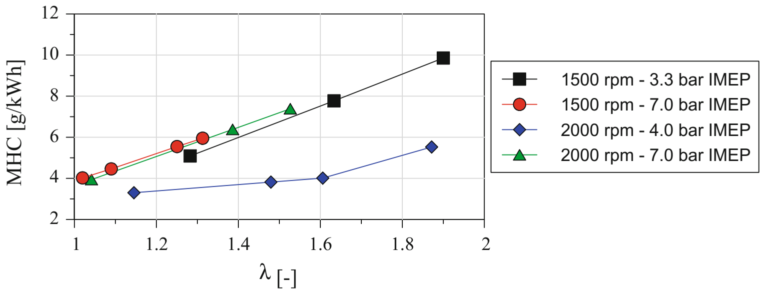


Fig. 9.11 Methane unburned hydrocarbons sensitivity versus λ varying the throttle valve position

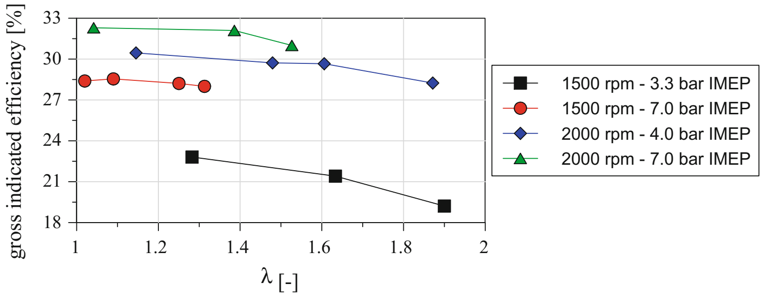


Fig. 9.12 Gross-indicated efficiency sensitivity versus λ varying the throttle valve position

position was varied from the fully opened condition to partialize reaching a lambda value (λ), the inverse of the equivalence ratio, of 1.05.

The analysis shows that near to stoichiometric values the specific MHC emissions converge for each point towards very low level and about 35% less compared to higher lambda values (Fig. 9.11). This is because of the fastening of the combustion process that promotes the methane oxidation process, moreover, the valve throttling lowers the in-cylinder peak pressure reducing the methane mass trapped into the crevice volumes (same consideration reported in CR section).

Looking at the gross-indicated efficiency (Fig. 9.12), reducing the lambda, an improvement of the thermodynamic process is obtained (up to 3%) compared to the fully opened throttle valve, because of the lower combustion losses and lower heat transfer losses (faster combustion). However, higher pumping losses were obtained partializing the throttle valve that at the end penalize the net-indicated efficiency compared to the fully opened throttle valve.

In general, the air-throttling has a beneficial effect on the MHC reduction but at the expense of the emitted smoke and particles, especially when the λ value is at the limit of the diesel operation mode. Thus, since both the DF and the air-throttling act on the λ reduction, the last has to be opportunely calibrated as a function of the substitution ratio and the load. In particular, lower loads that employ lower substitution ratio values require a higher air-throttling, in contrast to higher loads. Concluding, for both speeds, 1500 and 2000 rpm at 7 bar of IMEP and λ in the range 1.3–1.4, the MHC reduces by about 15% without penalties on the net efficiency (Di Blasio et al. 2017; Belgiorno et al. 2018b).

9.2.5 Summary

This chapter reports a parametric analysis of the substitution ratio, EGR, air-to-fuel ratio and compression ratio of a dual fuel combustion in a light-duty engine, at low load (till 7 bar of IMEP), since they are the most critical points for DF operation and they contribute significantly on the emissions and fuel consumption over the whole NEDC and/or WLTC emission homologation cycles for light-duty passenger car vehicles. The conducted investigation permits to evaluate the contribution factor of each analysed parameter the MHC and indicated specific fuel consumption (ISFC). Both the CH_4 slip and fuel consumption reduction are fundamental aspects to be faced with to promote the development of future light-duty combustion engines running in dual fuel concept. The contribution factor of each of the varied parameters on the generic output called X (e.g., MHC or ISFC) is calculated as:

$$\Delta[\%] = \frac{X_{\text{high level}} - X_{\text{low level}}}{X_{\text{low level}}} \cdot 100 \tag{9.2}$$

where the numerator represents the output increment when increasing the factor from a low to the high level (e.g., EGR from 0 to 45%). The denominator represents the output value at the starting level.

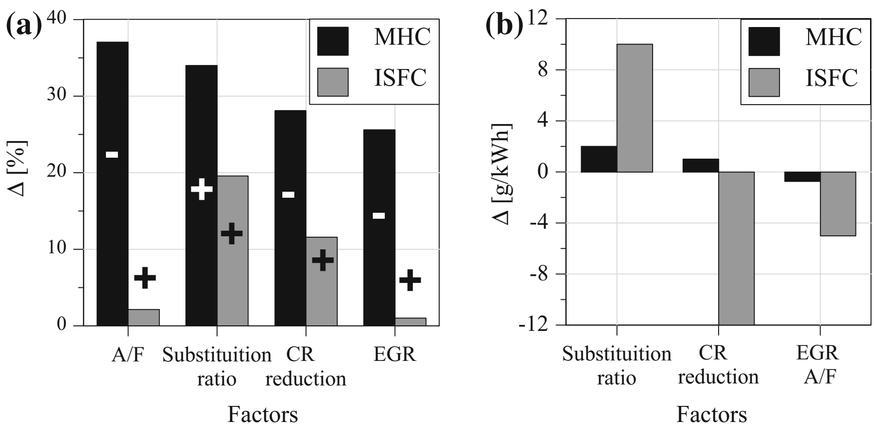


Fig. 9.13 MHC and ISFC relative (**a**) and absolute (**b**) sensitivity versus operating parameters

The contribution factors are reported in Fig. 9.13. The analysis shows a negative impact of the substitution ratio both on MHC and ISFC, and this is mainly related to the non-optimized combustion chamber design for dual fuel operation, especially for what concerns the oxidation of methane in the peripheral and dead regions of the combustion chamber. The specific MHC (g/kWh) increase, as a function of the substitution ratio, is about 2 g/kWh/10%. Additionally, the increase of the substitution ratio leads also to ISFC penalty that is about 10 g/kWh/10%.

The CR reduction to 14.5 reduces the globally indicated efficiency of about 10% with respect to 16.5 notwithstanding the MHC reduction of about 30% due to the combined effect of the bowl/crevice volume ratio increase and peak cylinder pressure decrease. The average MHC reduction per unit of CR reduction is equal to 1.0 g/kWh. While the ISFC increases are about 12.0 g/kWh per unit of CR reduction.

An interesting result can be drawn from Fig. 9.13 in terms of MHC reduction adopting EGR and reducing the A/F without penalties in terms of ISFC. In fact, reducing the A/F, the MHC reduces up to 40% at 1500x3.3 and 2000x7, and more than 20% in the other points. The MHC reduction per units of EGR is about 0.75 g/kWh/10% of EGR while passing from $\lambda = 1.6$ to near stoichiometric conditions an average reduction of about 5 g/kWh is detected for the load range explored.

Optimizing the combustion chamber and the fundamental DF engine operating parameters (EGR, A/F, substitution ratio, etc.), it is possible to achieve very interesting results adopting the dual fuel combustion instead of the CDC concept. Additionally, a development of the combustion chamber and a dedicated aftertreatment can help to improve the efficiency and the CH_4 emissions slip tailpipe to give a gain at this technology. An estimation of the outputs on the New European Driving Cycle (NEDC) shows that adopting only diesel fuel for engine load below 3.3 bar of IMEP, with high level of EGR (from 40 to 45%) while, at medium-high load, adopting a double injection strategy, maximum substitution ratio up to 70% and EGR no more than 35%, it is possible to achieve a reduction of about 6% in terms of CO_2 compared to the CDC under NEDC with an average substation ratio on cycle of about 41% (Belgiorno et al. 2018b).

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