Applied Thermal Engineering 102 (2016) 1001–1010

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

NO_X reduction and efficiency improvements by means of the Double Fuel HCCI combustion of natural gas–gasoline mixtures

Emiliano Pipitone *, Giuseppe Genchi

Università degli Studi di Palermo, Viale delle Scienze, Edificio 8, 90128 Palermo, Italy

ARTICLE INFO

Article history:
Received 13 November 2015
Accepted 9 April 2016
Available online 13 April 2016

Keywords:
HCCI
Internal combustion engine
Natural gas
Gasoline
Spark ignition

ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) and Double Fuel (DF) combustion represent two innovative processes sharing a strong potential for pollutant emissions and fuel consumption reduction. HCCI regards the auto-ignition of a homogeneous premixed charge of air and fuel, featuring very low NO_X emissions and good efficiency. Double Fuel (DF) instead indicates the simultaneous combustion of gasoline and natural gas (or gasoline and LPG), premixed with air by the port injection of both fuels within same engine cycle. Since fuel mixtures enhances the HCCI performances widening the range of possible operating conditions, the authors tested the HCCI combustion process using natural gas–gasoline mixtures on a CFR engine, endowed with two injection systems for the accurate control of both fuel mixture composition and overall air-to-fuels ratio. Stable and knock-free DF-HCCI combustions were obtained by combining different intake temperatures and compression ratios, in a range of engine load between 20% and 54% of the maximum load obtained with normal Spark Ignition (SI) operation. Noticeable increment of engine efficiency (+23%) and a strong reduction of pollutant emissions were found in comparison to SI operation, above all in terms of NO_x, which was reduced by 2 order of magnitude.

1. Introduction

Energy efficiency as well as pollutant emissions reduction are the main research and development purposes on modern internal combustion engines design. As a consequence, researchers and manufactures are continuously impelled to develop innovative and affordable solutions. Nowadays engine downsizing [1,2] and super charging [2–4] as well as cutting-edge valvetrain control systems [5–7] are widely spread in the automotive market, thanks to their prerogative of higher efficiency. For similar reasons various hybrid engine systems [8,9] recently have gained a good commercial success. Moreover, in the last years, the increasing cost of conventional fuels promoted the use of alternative fuels as well as fuel mixtures. Gaseous fuels, such as Natural Gas (NG) and Liquefied Petroleum Gas (LPG), represent today a concrete alternative to conventional fuels for road vehicles propulsion and stationary engines, since these are characterized by a relatively low cost and lower environmental impact. For these reasons in the last 20 years, gaseous fuels [10,11] as well as liquid fuel mixtures [12] have been deeply studied with the aim to experience their compatibility and properties as alternative fuels for spark ignition engines.

Many researchers carried out studies on the use of methane or hydrogen in spark ignition engines [13], and on the use of mixtures of gaseous fuels, such as natural gas, with particular attention to efficiency improvement, pollutant emissions and on the effects of the variation of its chemical composition [14]. Moreover, in recent years, the ethanol–gasoline blends have been extensively studied [12] and are nowadays used in the automotive field.

The use of fuel mixtures has been also studied by the authors in previous works [15,16], where the simultaneous combustion of a homogeneous mixture of gasoline and gaseous fuel (NG or LPG) in stoichiometric proportion with air has been experimentally tested, with several proportion between gasoline and gas, thus exploiting the good qualities of both fuels to obtain cleaner and more efficient combustions. The addition of the gaseous fuel to gasoline strongly increased the knock resistance [17,18], thus allowing to run the engine with stoichiometric mixtures even at full load: with respect to pure gasoline operation a considerable increment in engine efficiency was observed (+26%), together with HC and CO reduction in the order of 90%, without noticeable power losses (−4%). Bi-fuel engines, nowadays widely spread, are already endowed of a double injection system and can hence easily take advantage of this third combustion mode, which is referred to as *Double Fuel* (DF) combustion and is quite different from the well-known Dual Fuel combustion, in which, instead, the auto-ignition

* Corresponding author.
E-mail address: emiliano.pipitone@unipa.it (E. Pipitone).

http://dx.doi.org/10.1016/j.applthermaleng.2016.04.045
1359-4311/

Nomenclature

Abbreviations and symbols

ATDC	After Top Dead Centre
BTDC	Before Top Dead Centre
CAD	Crank Angle Degree
CFR	Cooperative Fuel Research
CI	Compression Ignition
CO	Carbon Oxide
COV	Coefficient of Variation
CR	Compression Ratio
DME	Dimethyl Ether
EGR	Exhaust Gas Recirculation
G_{Air}	inlet air mass flow
G_{NG}	natural gas mass flow
$G_{Gasoline}$	gasoline mass flow
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing
ITE	Indicated Thermal Efficiency
LPG	Liquefied Petroleum Gas
LPP	Location of Pressure Peak
MON	Motor Octane Number
$m_{Gasoline}$	injected mass of gasoline
m_{NG}	injected mass of natural gas
NO	Nitric Oxide
NG	Natural Gas
RCCI	Reactivity Controlled Compression Ignition
RPM	Revolutions Per Minute
TDC	Top Dead Centre
THC	Total Hydrocarbons
T_{IN}	inlet air–fuels mixture temperature
SI	Spark Ignition
VCR	Variable Compression Ratio
x_{NG}	natural gas mass fraction
α	air to fuel ratio
α_{ST}	stoichiometric air to fuel ratio
λ	air-excess factor = α/α_{ST}

of a small quantity of one of the two fuels acts as igniter to start the flame propagation combustion of the second fuel. The simultaneous combustion of gas and gasoline has been investigated also by other researchers [19–22], both in naturally aspirated and supercharged SI engines.

In light of their results, the authors decided to exploit NG–gasoline mixtures to experience a combustion process that combine the DF and the Homogeneous Charge Compression Ignition (HCCI) combustion, a particular combustion process [23,24] in which a homogeneous mixture of air and fuel is auto-ignited through compression. HCCI combustion has the potential to significantly reduce NO_X and particulate emissions, while achieving high thermal efficiency and allowing the use of a variety of fuels. The HCCI combustion process is able to combine the best features of a Spark Ignition (SI) engine and Compression Ignition (CI) engine. Similar to a SI engine, fuel and air are homogeneously mixed, thus preventing fuel-rich diffusion combustion and strongly reducing the particulate matter emissions, usually associated with conventional Compression Ignition (CI) combustion. Moreover, the auto-ignition of a homogenously premixed charge allows the use of large excess of comburent air, thus eliminating the high-temperature flames of conventional SI engine combustion and reducing Nitrogen Oxides (NO_X) emissions respect to conventional CI and SI combustion. Furthermore, similarly to CI engines, HCCI combustion power output is ordinary controlled by unthrottled operation, acting on the air-to-fuel ratio, thus reaching higher thermal efficiency compared to SI engines in part load condition.

However, respect to conventional engines that have a direct control on the combustion phase, through the spark timing in SI engines or through the fuel injection timing in CI engines, HCCI engines achieve auto-ignition of the homogeneous mixture around the Top Dead Centre (TDC), thus presenting some difficulties in the control of both combustion phase and rate of heat release, in a wide range of operative conditions. In particular, at idle, in-cylinder thermodynamic conditions are not sufficient to cause auto-ignition while at high loads the rate of heat release results excessive and entails heavy knocking conditions. These considerations, together with transient and cold start matters, are the main HCCI technical challenges that are currently object of many researches. Various systems have been proposed: supercharging [25] and Exhaust Gas Recirculation (EGR) [26] are widely experienced as well as variable valve timing [27] and Variable Compression Ratio (VCR) [28]. Many studies focuses on the use of different fuels (gas [29] and liquid [30]), fuel blends involving gasoline [31,32] or natural gas [33,34]. In many cases some of the above mentioned systems are simultaneously used or experienced to improve the operating range of the HCCI combustion. A comprehensive review of the studies carried out on both HCCI and RCCI (Reactivity Controlled Compression Ignition, a variant of HCCI) is available in [35]. As can be noticed, despite the great number of researches carried out on HCCI performed by means of several pure and mixed fuels, there is no evidence of any HCCI study involving the use of natural gas–gasoline mixtures. The combined use of these two widely spread fuels offers the possibility to vary the auto-ignition properties of the mixture by simply changing the mixture composition, thus achieving strong knock resistance by the use of high natural gas concentration, or low knock resistance with a greater presence of gasoline. This has the potential to solve the problems related to poor thermodynamic conditions and to the engine load control. Modulating the fuel mixture composition could allow hence to control the overall resistance to auto-ignition and hence help to properly adapt the reactivity of the fuel mixture to the in-cylinder thermodynamic conditions. Moreover, this particular fuel mixture has strong potentialities for HCCI application, since both fuels are commercially available in common filling stations of many countries. All these considerations induced the authors to carry out a proper experimental study on the Double Fuel HCCI.

2. Experimental setup

The experimental test were carried out using a Cooperative Fuel Research (CFR) engine [36] manufactured by Dresser Waukesha (engine specifications are reported in Table 1). The use of this engine is currently prescribed by ASTM for fuel octane rating and, thanks to its robust construction, it also allows to experience various combustion process such as HCCI, even in knocking

Table 1
CFR engine specifications [36].

Manufacturer	Dresser Waukesha
Model	F1/F2 Octane
Compression ratio	4.5–16
Bore	82.6 mm
Stroke	114.3 mm
Connecting rod length	254.0 mm
Displacement	611.2 cm^3

condition. The CFR engine is a four-stroke two valve stationary single-cylinder spark-ignition engine. The particular engine arrangement allows to vary quickly and accurately the Compression Ratio (CR) from 4.5 to 16 by moving the engine head (fixed to the cylinder sleeve) with respect to the piston. The combustion chamber is of discoid type and its basic configuration does not change with the compression ratio.

The standard CFR engine is connected to an electric synchronous motor that maintains constant the revolution speed (900 RPM or 600 RPM), both in fired and in motored condition. Two electric heaters were connected to two independent PID control systems Omega CN4116 with the aim to maintain both inlet air temperature (T_{AIR}) and intake air/fuel mixture temperature (T_{IN}) at the required values during the tests.

As regards fuel supplying, a standard CFR engine features an original carburettor system which does not allow the use of gaseous fuels. The authors hence endowed the CFR engine with two independent injection systems (Figs. 1 and 2) and a Venturi air flow metre in order to realize each desired NG–gasoline mixture and to accurately control the overall air–fuels ratio (a more detailed description is given in [17,18]).

Gaseous fuel mass flow was measured by means of a Bronkhorst mini CORI-FLOW® Coriolis effect mass flow metre while gasoline mass flow was deduced on the basis of the imposed injection time by means of a precise injector flow chart previously experimentally determined [17,18].

A computer (n.1 in Fig. 1) equipped with a National Instruments DAQCard 6062E was used to manage the two injection systems and perform data acquisition, by means of an expressly designed software developed by the authors in LabVIEW environment.

The in-cylinder pressure, measured by means of a Kistler AG piezoelectric pressure sensor placed on the combustion chamber, and all the relevant quantities (manifold absolute pressure as well as air and fuels mass flows) were acquired by means of the above mentioned DAQCard 6062E using as trigger and scan clock the pulses generated by a 360 pulses per revolution incremental optical encoder connected to the engine crankshaft.

The knock occurrence was monitored analysing the in-cylinder pressure sensor. To this purpose a second personal computer (n.2 in Fig. 1) was used to process the in-cylinder pressure signal, acquired by means of a second National Instruments DAQCard 6062E, with a scan rate of 200 kHz.

The test bench was also endowed with an exhaust gas analyser (Motorscan 8020) used to measure the concentration of CO, CO_2, O_2, Total Hydrocarbons (THC) and Nitric Oxide (NO) in the engine exhaust emissions.

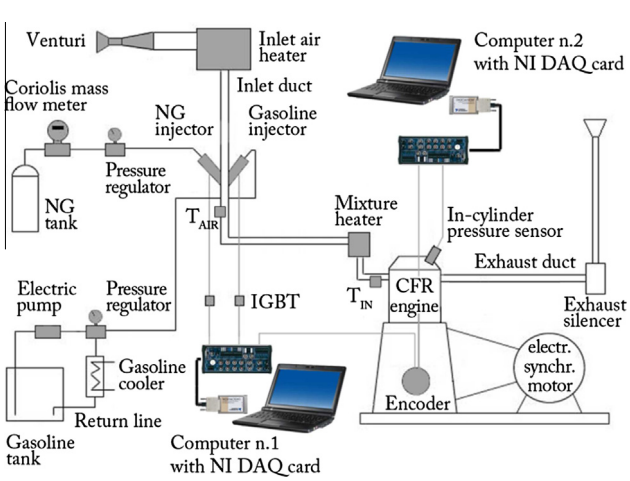

Fig. 1. Experimental system layout.

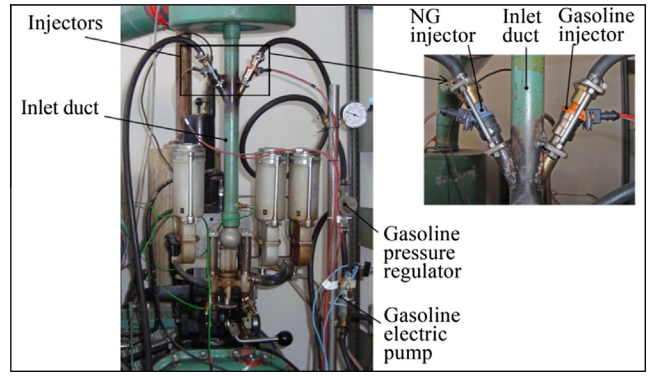

Fig. 2. Fuel supply systems: NG injector and gasoline injector placed in the inlet duct.

3. Test methods

In this experimental study, the auto-ignition of NG–gasoline mixtures were experimentally tested under different proportions between the two fuels, in order to define the range of the DF-HCCI operating conditions without knocking, and to evaluate the corresponding performances in terms of indicated efficiency and pollutant emissions. It must be remarked that in the test performed, no external EGR has been employed to decrease knock tendency; moreover, due to a substantial valves overlap, the engine used for the test does not benefit from internal EGR. All this helps to understand the reason why no HCCI combustion could be performed by using only gasoline or only natural gas, without causing heavy knocking or combustion instability. The combined use of the two fuels, instead, allowed to perform stable and safe HCCI combustions, with very encouraging results, as shown further on.

The HCCI tests were carried out running the CFR engine at 900 RPM, employing three different compression ratios (14, 15, 16) and three intake temperatures T_{IN} (135, 150, 165 °C), with a constant coolant temperature of 100 °C. By means of preliminary evaluations, it was observed that these parameters allow to experience a wide range of different Double Fuel HCCI operative conditions on the CFR engine.

If m_{NG} and $m_{Gasoline}$ denote the injected mass of NG and gasoline respectively within the same engine cycle, the proportion between the two fuels may be represented by the percentage NG mass fraction x_{NG}:

$$x_{NG} = \frac{m_{NG}}{m_{NG} + m_{Gasoline}} \cdot 100 \tag{1}$$

As resumed in Table 2, in the test performed the NG mass fraction was varied from 20% to 80% (or to the maximum value allowing stable HCCI operation without knocking occurrence). For each NG mass fraction, the engine load was varied (from the stability limit to the incipient knocking) acting on the overall air-to-fuel ratio α, usually expressed in terms of air-excess factor λ, defined as:

$$\lambda = \frac{\alpha}{\alpha_{ST}} = \frac{G_{Air}}{16.9 \cdot G_{NG} + 14.7 \cdot G_{Gasoline}} \tag{2}$$

Table 2
Double Fuel HCCI test conditions.

Engine speed	900 RPM
Inlet air temperature T_{AIR}	40 °C
Intake Air/fuel temp. T_{IN}	135, 150, 165 °C
Engine coolant temperature	100 °C
Compression ratio (CR)	14, 15, 16
NG mass fraction	from 20% to 80%
Manifold absolute pressure	1.0 bar

being G_{Air}, G_{NG} and $G_{Gasoline}$ the mass flow rates of air, NG and gasoline, while 14.7 and 16.9 are the stoichiometric air-to-fuel ratios of gasoline and natural gas respectively.

For each test condition, 50 consecutive complete in-cylinder pressure cycles were acquired together with Manifold Absolute Pressure (MAP), inlet air mass flow, NG and gasoline mass flows. These data allowed the calculation of the engine mean pressure cycle, which was used to compute both Indicated Mean Effective Pressure (IMEP) and Indicated Thermal Efficiency (ITE).

The experimental results of the DF-HCCI tests were compared, in terms of IMEP, ITE and pollutant emissions, with the output of the CFR engine in ordinary spark ignition mode, either with pure gasoline or with neat NG (Table 3). In the SI mode, the engine load was obviously regulated by throttling inlet air flow, thus obtaining MAP values from 1.0 to 0.6 bar, the latter being the minimum value corresponding to acceptable running stability of the CFR engine. For both pure fuels, the SI mode was carried out with stoichiometric air-to-fuel ratio up to full load, with the aim to maintain a high indicated efficiency for the whole load range. Preliminary tests revealed that to avoid inadmissible knocking phenomena, the maximum allowed compression ratio in spark ignition mode was 6 for the use of gasoline and 7 for NG. The use of the rich mixtures systematically employed in common SI engines at full load (i.e. $\lambda \approx 0.85$) would have certainly allowed higher compression ratios, but the resulting fuel consumption increase would have excessively penalized the SI engine performance with respect to HCCI. With same attention to efficiency, the spark advance was chosen as the minimum value for best efficiency, obtained by setting the Location of in-cylinder Pressure Peak (LPP) as near as possible to 15 Crank Angle Degrees (CAD) After Top Dead Centre (ATDC) without causing knocking phenomena: this procedure, well known in SI engine calibration and control, allows to quickly determine the optimal combustion phase in spark ignition engines, apart from the particular fuel employed, the engine wear, the different ambient conditions, etc. [37].

Dealing with IMEP evaluation, a crucial factor is represented by the determination of engine Top Dead Centre (TDC) position, which should be carried out within the accuracy of 0.1 CAD [38]: a capacitive sensor Kistler 2629B has been employed by the authors for a correct evaluation of the TDC position for each compression ratio adopted in the test.

The entire experimental study was carried out using a single sample of commercial gasoline (whose main characteristics are reported in Table 4) and a single sample of NG, whose properties and composition are reported in Table 5.

It is worth to mention that the NG used for the tests, with methane content of 85.79 vol.%, well represents a mean situation among the different NG compositions that can be encountered all over the world, whose methane content, as reported in [14,42,43], is usually comprised between 75 and 95 vol.%

4. Experimental results and discussion

The experimental study carried out by the authors aimed to test the opportunity to employ NG–gasoline mixtures to perform HCCI combustion for different engine loads without knocking occurrence, and to assess the connected advantages in terms of both engine efficiency and pollutant emissions. The experimental results here exposed hence define the range of existence of the stable and knock free HCCI combustion obtained in Double Fuel mode, as function of compression ratio (CR), intake temperature T_{IN} and mixture composition (i.e. natural gas mass fraction). In particular, both CR and T_{IN} were employed as control parameters for auto-ignition. For fixed values of CR, T_{IN} and NG mass fraction, a certain engine load variation was also achieved by varying the overall air-to-fuel ratio λ. As example, Fig. 3 reports the measured IMEP as function of fuel mixture composition for CR = 16 and T_{IN} = 150 °C.

In particular, the upper boundary curve in Fig. 3 delimits the maximum knock free engine load, while the lower curve instead

Table 3
Test conditions during spark ignition tests.

Engine speed	900 RPM
Inlet air temperature T_{AIR}	30 °C
Intake Air/fuel temp. T_{IN}	52 °C
Engine coolant temperature	100 °C
Compression ratio (CR)	6 for pure gasoline 7 for pure NG
Air-to fuel ratio (α)	Stoichiometric (16.9 for NG, 14.7 for gasoline)
MAP	From 0.6 to 1.0 bar
Spark advance	Minimum value for best efficiency

Table 4
Properties of gasoline used in the tests.

Liquid phase density at 15 °C	730 kg/m3
Equivalent H/C ratio [39]	1.85
Molar mass (assumed) [12,40]	110 g/mole
Stoichiometric ratio	14.7
Lower heating value [41]	43.4 MJ/kg
Motor Octane number [36,17]	84.1

Table 5
Composition and properties of the natural gas used.

Components	vol.%
Methane – CH_4	85.79
Ethane – C_2H_6	7.86
Propane – C_3H_8	1.61
N-butane – C_4H_{10}	0.19
Isobutane – C_4H_{10}	0.28
Butylene – C_4H_8	0.05
Isopentane – C_5H_{12}	0.06
N-pentane – C_5H_{12}	0.06
Carbon dioxide – CO_2	1.04
Nitrogen – N_2	2.96
Helium – He	0.09
Properties	
Reactive Hydrogen/Carbon Ratio	3.76
Stoichiometric air/fuel mass ratio	16.9
Lower heating value [44]	46.67 MJ/kg
Motor Octane Number [36,17]	122.1

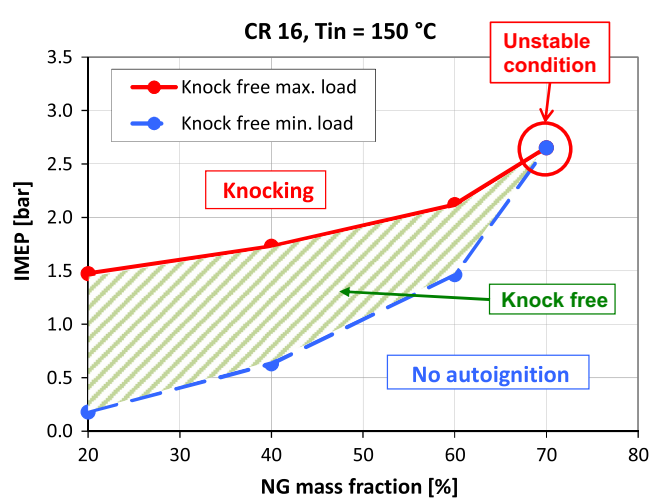

Fig. 3. Range of DF-HCCI operative conditions with T_{IN} = 150 °C and CR = 16: IMEP as function of the NG mass fraction.

represents the minimum achievable engine load in HCCI mode, beneath which no stable auto-ignition processes could be obtained. The two boundary curves have been determined by monitoring the main engine running parameters (maximum in-cylinder pressure and IMEP) for more than 1000 consecutive cycles, in order to exclude any possible slight self-extinguishing auto-ignition condition or self-enforcement knocking phenomena.

Similarly to the other test conditions with different CR and T_{IN}, the engine load resulted proportional to the NG mass fraction (Fig. 3). As the NG mass fraction increases the knock free area progressively reduces; for a NG mass fraction of 70%, the HCCI combustion process resulted unstable since every small variation on the engine control parameters could cause knock occurrence as well as bring to self-extinguishing condition. Considering the strict relation between engine load and air-to-fuel ratio, the same experimental results can be observed, although with opposite slope, in Fig. 4 in terms of air-excess factor λ as function of the NG mass fraction.

The analysis of all experimental data shows that as compression ratio and T_{IN} increase, the amplitude of the HCCI combustion range increases in terms of NG mass fraction.

As shown in Fig. 5, the maximum range of engine load was registered with a CR of 16 and a T_{IN} of 165 °C. In these test conditions it was possible to run the engine with NG–gasoline mixtures comprised between 20% and 80% gaseous fuel mass fraction, which gave as result IMEP values from 1.22 to 2.58 bar. For all the CR used, the IMEP showed a positive trend as function of NG-mass fraction and for each mixture composition the higher maximum knock-free IMEP has been always registered for the lower CR: e.g. with 40% of NG mass fraction, maximum IMEP varied from 1.45 bar of CR 16 to 2.62 bar of CR 14.

Similar considerations can be made for the lower intake temperatures which entailed a reduction of the maximum gaseous fuel content, as shown in Fig. 6 for a T_{IN} of 150 °C and in Fig. 7 for a T_{IN} of 135 °C. The lower boundary condition was found with a CR of 14, in which HCCI combustion can be achieved only using a T_{IN} of 165 °C, with a NG mass fraction ranging from 20% to 50%. For the lower intake temperatures the CR of 14 was not sufficient to achieve HCCI combustion. For fixed CR and mixture composition, a T_{IN} reduction entailed an increase of the engine load: e.g. with CR 16 and 40% of NG mass fraction, maximum IMEP varied from 2.0 bar with a T_{IN} of 165 °C to 3.0 bar with T_{IN} 135 °C.

As shown in Fig. 7, the highest HCCI engine load, corresponding to an IMEP of 3.28 bar, has been achieved running the engine at the lower T_{IN} (135 °C) with a CR of 15. This result is reasonably due to

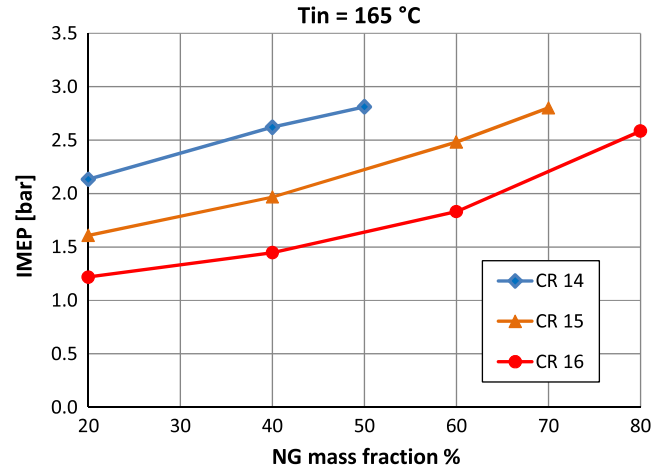

Fig. 5. Maximum HCCI engine load as function of the NG mass fraction at T_{IN} of 165 °C.

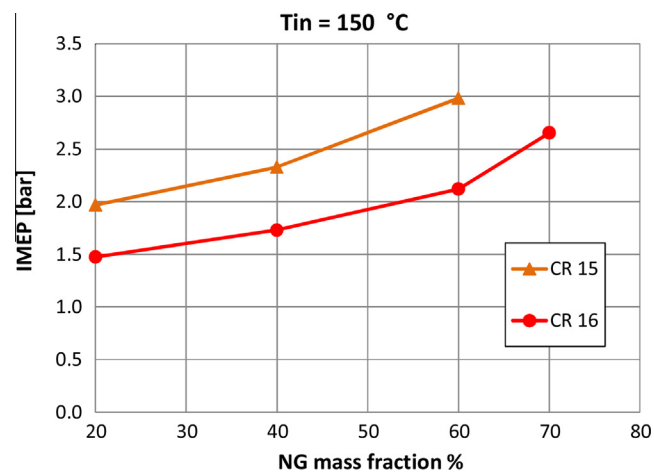

Fig. 6. Maximum HCCI engine load as function of the NG mass fraction at a T_{IN} of 150 °C.

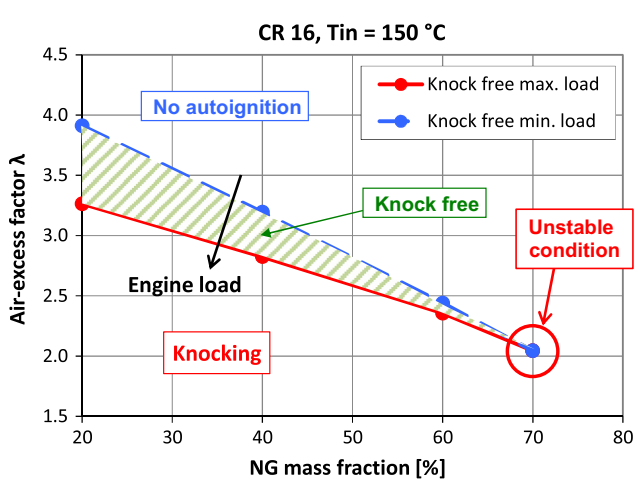

Fig. 4. Range of HCCI operative conditions: air-excess factor λ as function of the NG mass fraction.

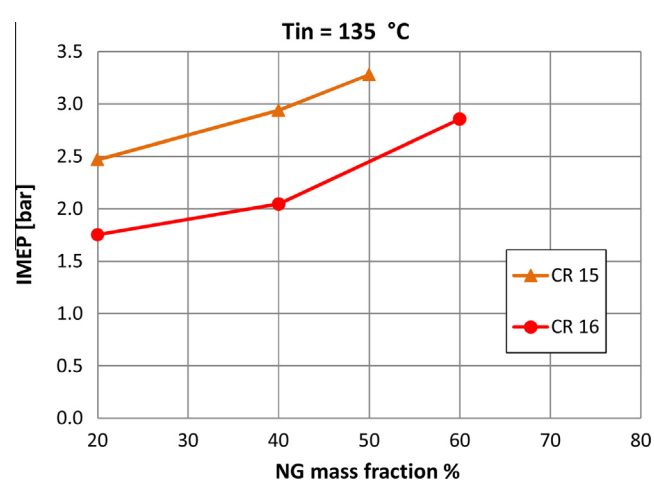

Fig. 7. Maximum HCCI engine load as function of the NG mass fraction at a T_{IN} of 135 °C.

the lighter thermodynamic conditions (corresponding to the lower T_{IN} and CR) which in turn, during the combustion process, produced lower rates of heat release with respect to the other operative conditions.

All the available experimental data have been collected in the diagram of Fig. 8, which represents the knock free DF-HCCI combustion zone as function of IMEP and NG concentration, delimited by the upper and lower bound curves, respectively (CR = 15, T_{IN} = 135 °C) and (CR = 16, T_{IN} = 165 °C). The two black arrows indicate the increasing direction of the two control parameters: intake temperature T_{IN} and compression ratio CR.

The experimental results show that, once fixed both CR and T_{IN}, the maximum knock-free engine load is proportional to the NG mass fraction. In other words, in all the tested operative conditions, the increase of the knock-free engine load always required the increase of the mixture NG content. This can be explained considering that an higher engine load entails higher in-cylinder pressure and temperature, which, in turn, require a stronger auto-ignition resistance to avoid knocking phenomena; this can be obtained by increasing the concentration of natural gas in the mixture, as demonstrated by the authors in a previous experimental work [17]: the overall knock resistance of a NG–gasoline mixture can be expressed as a polynomial function of the NG mass fraction or, alternatively, as function of the H/C ratio of the mixture.

The whole map of DF-HCCI tested conditions is also illustrated in Fig. 9 in terms of air-excess factor λ as function of IMEP. Also in this diagram the knock free DF-HCCI combustion zone is delimited by the same two boundary curves of Fig. 8. Fig. 9 clearly shows that, similarly to Compression Ignition (CI) engines, the load increase is obtained reducing the overall air-to-fuel ratio. The values of λ corresponding to a knock free DF-HCCI combustion vary from 3.20 to 1.75, and are quite higher than those of an ordinary SI engine, usually comprised between 0.8 and 1.1.

The analysis of the experimental results show that the indicated efficiency of the DF-HCCI combustion is proportional to the engine load. As summarised in Fig. 10, all the tested conditions show a very similar trend and could be represented by a single curve; the higher indicated efficiency (0.293) has been registered for the higher IMEP of 3.28 bar, with CR = 15 and T_{IN} = 135 °C.

For a better analysis of the DF-HCCI combustion results, a comparison was made with the results obtained by the same engine operated in the ordinary SI mode, both with pure natural gas and pure gasoline: Fig. 11 reports the indicated engine efficiency values measured with both combustion modes as function of engine load in terms of IMEP.

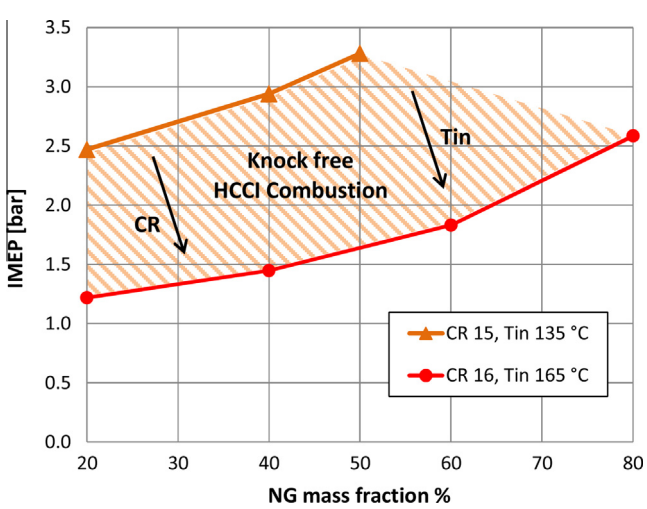

Fig. 8. Range of HCCI operative conditions: IMEP as function of the NG mass fraction.

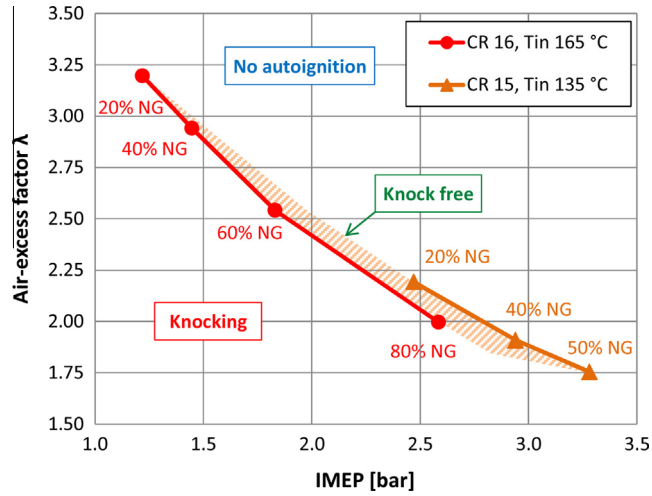

Fig. 9. Range of DF-HCCI operative conditions: air-excess factor λ as function of the IMEP.

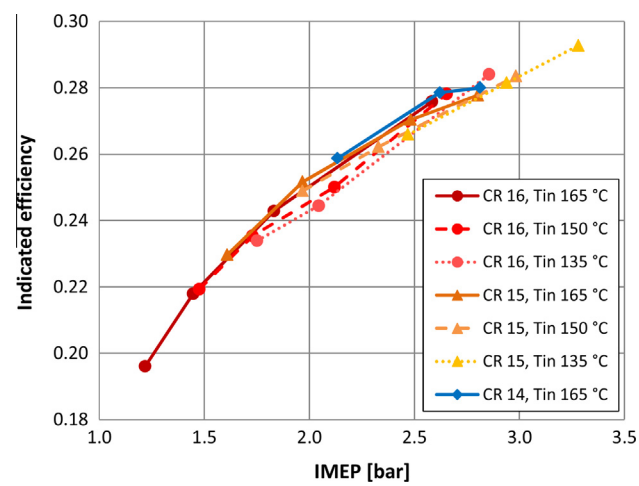

Fig. 10. Indicated efficiency of HCCI combustion as function of the IMEP.

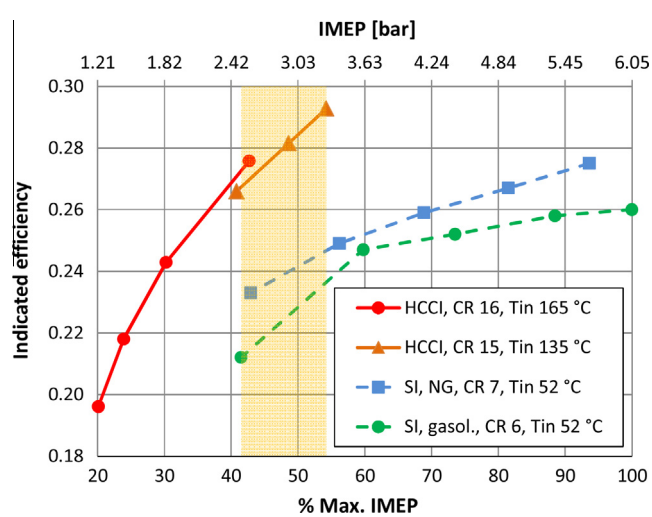

Fig. 11. Indicated efficiency as function of the IMEP.

The two solid lines represent the same boundary curves shown in Fig. 8: one refers to the higher thermodynamic conditions (i.e. CR = 16 and T_{IN} = 165 °C), which also allowed the wider engine load variation in DF-HCCI mode, while the other (i.e. CR = 15 and T_{IN} = 135 °C) refers to the thermodynamic conditions which produced the higher IMEP and the best efficiency. The dashed curves instead refer to the results obtained in the SI mode, using alternatively pure gasoline or pure NG. As mentioned, in the SI mode, the engine load was regulated by means of a traditional throttle valve. The pure natural gas spark ignited combustion shows higher efficiency than the pure gasoline mainly due to the higher compression ratio (7 rather than 6) allowed by the very high knock resistance of NG (as reported in Table 5) and to the better mixing capability with air.

As can be deduced from Fig. 11, the DF-HCCI engine load ranged from 20% to 54% of the maximum engine load obtained with gasoline spark ignition (i.e. 6.05 bar IMEP). Moreover, in the IMEP range between 2.5 and 3.28 bar (underlined by the coloured area in Fig. 11), the DF-HCCI combustion featured significantly higher efficiencies in comparison with conventional SI mode, with increment of about 23% with respect to pure gasoline SI mode. The higher DF-HCCI indicated efficiency has several contribution: the adoption of higher compression ratios, the absence of pumping losses and the substantially shorter combustion duration which characterize HCCI with respect to SI mode. It is worth to note that the SI combustion has been tested in stoichiometric condition, both with gasoline and with natural gas, with the aim to maintain a good efficiency for the comparison with DF-HCCI results, even if, at full load, gasoline spark ignition engines are systematically fuelled with rich mixture, which prevent dangerous knocking phenomena but induce strong efficiency reduction.

In light of the experimental results, the DF-HCCI combustion represents an efficient alternative to the traditional SI combustion, above all in partial load conditions, where throttle regulated SI engines are affected by pumping losses. In HCCI mode, instead, the engine load is regulated, similarly to common compression ignition engines, by varying the air-to-fuel ratio, as shown in Figs. 9 and 4.

As previously mentioned, the combustion phase in the HCCI mode mainly depends on the thermodynamic condition of the air/fuel mixture at IVC (Intake Valve Closure) and on the compression phase. In all the knock-free DF-HCCI conditions tested, the combustion phase always occurred with very low retard with respect to the TDC, as confirmed, for example, by Fig. 12: here the mean in-cylinder pressure trace (evaluated over 50 consecutive pressure cycles) and its standard deviation are shown as function of the crank position, for the DF-HCCI combustion with CR = 15, T_{IN} = 150 °C and 40% of natural gas in the fuel mixture.

The stability of the DF-HCCI combustion performed can also be considered good, being the Coefficient of Variation (COV, i.e. the ratio between standard deviation and mean value) of IMEP about 3%. Considering the whole set of test conditions, the measured IMEP COV ranged from 1.9% to 7.63%, with an overall mean value of 3.62%.

The good stability of the DF-HCCI combustion obtained is also confirmed by the relatively low dispersion of the in-cylinder pressure values: for the case of Fig. 12, a maximum standard deviation of about 4 bar is reached, which roughly corresponds to 13% of the peak in-cylinder pressure of 30.62 bar. With reference to all the tested conditions, the maximum value of the ratio between standard deviation and in-cylinder pressure ranged from 2.76% to 14.54%, with an overall mean value of 8.72%.

The experimental tests carried out with the DF-HCCI combustion revealed also interesting advantages in terms of pollutant emissions. Both Carbon Oxide (CO) and Carbon Dioxide (CO_2) emissions resulted almost halved in comparison with the values measured in SI mode with both gasoline and NG, fundamentally due to the higher air-to-fuel ration. As shown in Fig. 13, the measured CO concentration in the exhaust gases resulted from 0.09 to 0.15 vol.% for the DF-HCCI combustion, while in SI mode the concentration resulted between 0.28 and 0.52 vol.%. As can also be noted from the same Fig. 13, the CO emissions measured with DF-HCCI combustion exhibit a slight reduction with IMEP increment: this has been explained considering that, as already mentioned, in HCCI mode, an engine load increase implies higher combustion temperature which promotes the CO oxidation.

As regards CO_2 emissions, the DF-HCCI combustion revealed a linear trend as function of IMEP (Fig. 14), which is rather obvious, being the HCCI engine load increased by enriching the air–fuels mixture. The values recorded in DF-HCCI mode are comprised between 3.61 and 6.64 vol.%, quite consistent with the air–fuel ratios employed. In SI mode, instead, being the air–fuel adopted constantly maintained to the stoichiometric value, the CO_2 emissions resulted higher and constant with varying engine load: 11.80 vol.% with NG and 15.23 vol.% with gasoline.

The most interesting result however concerns the NO_X emissions, which are here represented by their main constituent Nitric Oxide (NO): with respect to the conventional SI mode, the DF-HCCI

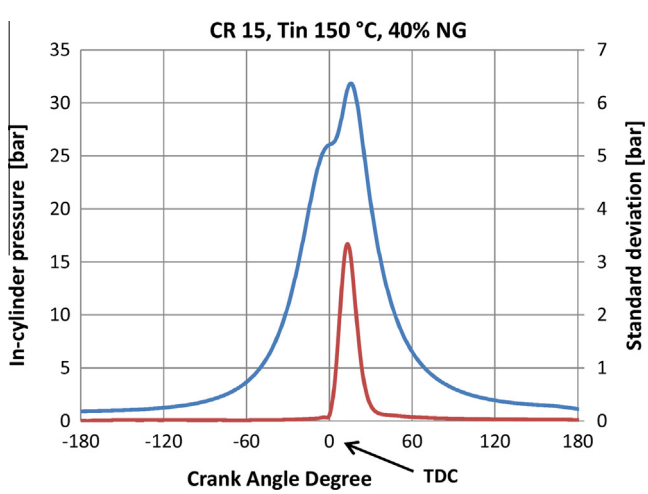

Fig. 12. Mean in-cylinder pressure and its standard deviation for the DF-HCCI combustion with T_{IN} 150 °C, CR 15 and 40% NG mass fraction.

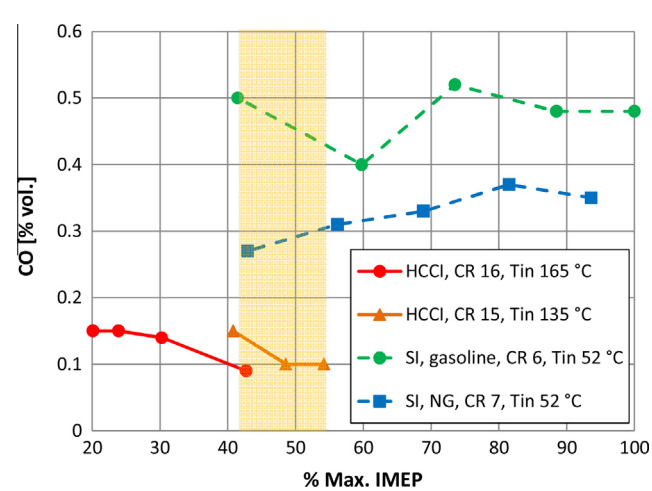

Fig. 13. CO emissions as function of the IMEP.

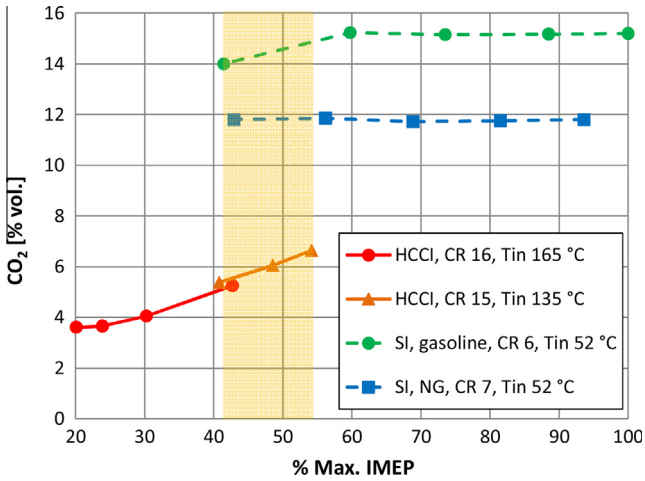

Fig. 14. CO_2 emissions as function of the IMEP.

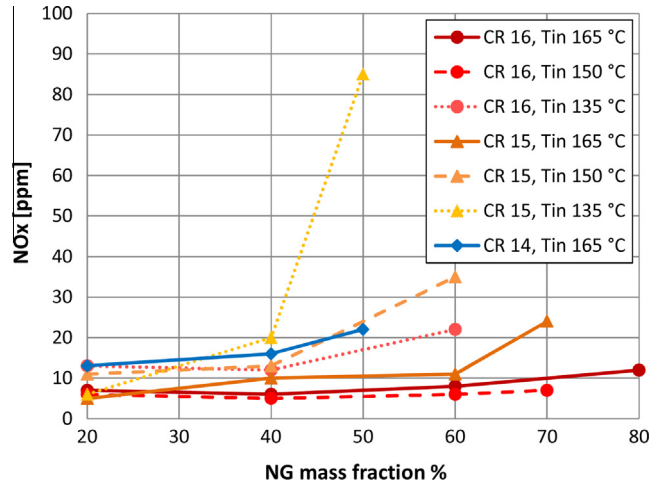

Fig. 16. NO emissions as function of the NG mass fraction measured in all the DF-HCCI test.

combustion revealed the drastic reduction of two order of magnitude, as represented in Fig. 15. As can be observed, in SI mode, the NO emissions resulted between 2000 and 3700 ppm, varying with engine load.

With the DF-HCCI combustion, instead, the NO emissions revealed a mean value of 20 ppm with, reaching the maximum of 85 ppm in correspondence of the maximum engine load experienced with a CR of 15 and T_{IN} 135 °C, as shown in both Figs. 15 and 16.

In particular, similarly to SI combustion, in DF-HCCI mode the NO emissions revealed a slight increase with the engine load (corresponding to the higher NG mass fraction, Fig. 8) due to the corresponding higher temperature conditions of the combustion chamber.

Considering a conventional three-way catalytic converter efficiency of 90%, the lowest NO emissions measured in the SI mode, i.e. 2000 ppm, would be reduced to 200 ppm, which is still well above the mean raw NO concentration of 10 ppm recorded with the DF-HCCI combustion. This negligible emission level results from the good homogeneity of the air/fuel mixture and, above all, from the low combustion temperatures which, due to the lean mixture adopted, characterize the HCCI combustion process.

Nevertheless, these lower combustion temperature entail, as a negative consequence, an incomplete combustion processes which in turn cause higher hydrocarbons emissions if compared to the values measured with conventional SI combustion (Fig. 17).

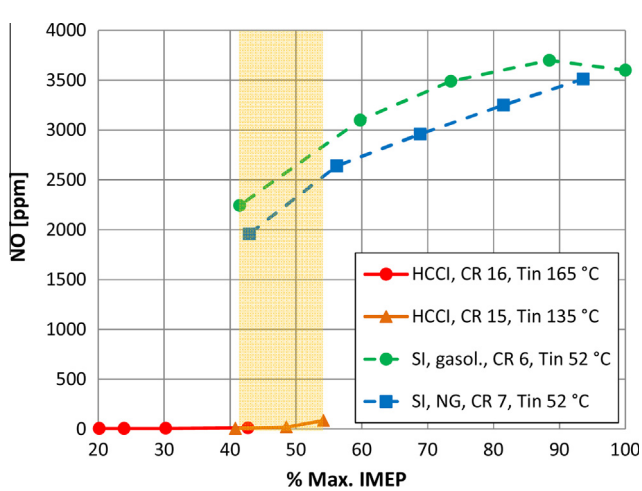

Fig. 15. NO emissions as function of the IMEP.

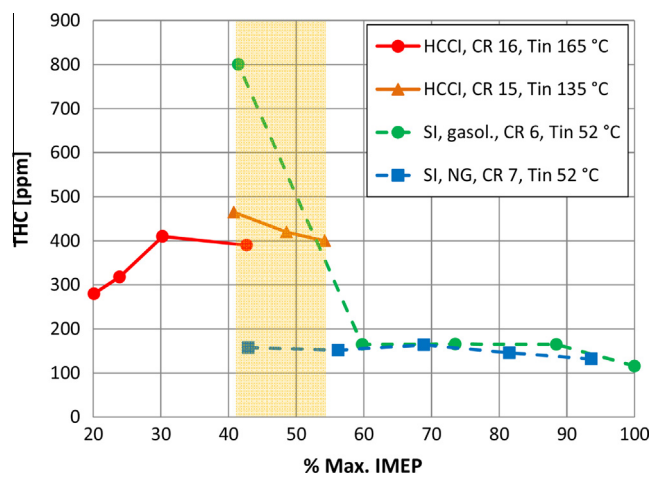

Fig. 17. Total hydrocarbons emissions as function of the IMEP.

It is worth to mention, however, that the total hydrocarbons emissions measured in the DF-HCCI combustion mode lie in the range 300–500 ppm and hence can be easily reduced to very low values by means of a common oxidation catalytic converter, especially if the available large excess of air in the exhaust gas is considered.

5. Conclusions

The aim of this work was to experience a particular internal combustion engine process which combines the simultaneous combustion of gasoline and natural gas, called Double Fuel (DF) combustion, with the Homogeneous Charge Compression Ignition (HCCI). No references were found on the HCCI combustion of gasoline–natural gas mixtures, hence the original results here presented, obtained through a wide set of experimental tests carried out by the authors, covers a certain lack of literature and demonstrate that DF-HCCI combustion can be achieved with remarkable advantages in terms of both efficiency and environmental impact.

The range of knock-free running condition in DF-HCCI mode revealed smaller than in conventional spark ignition (SI) mode and strictly depends on the compression ratio (CR) and on the inlet air–fuels mixture temperature T_{IN} (as summarised in Fig. 8). The engine load increase entails higher thermodynamic conditions

which requires an increase of the fuel mixture knock resistance: within certain limits this has been achieved by means of the Double Fuel strategy, increasing the natural gas content of the fuel mixture [17]. For all CR and T_{IN}, the maximum knock free engine load resulted proportional to the natural gas mass fraction.

With the DF-HCCI combustion, the best results, in terms of IMEP and indicated efficiency, have been obtained with a CR of 15 and T_{IN} 135 °C. For lower CR and T_{IN} the auto-ignition process resulted unstable. No HCCI combustion could be obtained by using only gasoline or only natural gas, without causing heavy knocking or unstable running of the engine. The results presented, hence, could be obtained only due to the combined use of gasoline and natural gas.

The DF-HCCI combustion featured noteworthy increments in indicated efficiency (up to +23%) in comparison with conventional SI mode in the shared load range, which was found between 20% and 54% of the maximum SI engine load. The higher efficiency of DF-HCCI is mainly due to the higher compression ratio and to the absence of pumping losses, given the unthrottled operation which characterize the compression ignition engines respect to ordinary spark ignition engines.

The main advantage of the DF-HCCI combustion however has been found in the pollutant emissions, which, with the only exception of total hydrocarbons (Fig. 17), revealed a remarkable reduction with respect to conventional SI mode. However, the large excess of air in the exhaust gas (Fig. 9), should allow a strong reduction of the raw THC by means of a common oxidation catalytic converter.

It is noteworthy that the DF-HCCI combustion allowed to dramatically decrease the NO emissions, reaching a reduction factor of two order of magnitude in comparison with the SI operation, with an average raw value of only 20 ppm (Fig. 15). In light of the constricting environment saving rules and increasing fuel cost, the negligible NO emissions as well as the high indicated engine efficiency, represent the main and most interesting results obtained in this experimental study.

As shown in many other works [23], the limited operating range and some difficulties related to non-steady operation, make HCCI combustion more suitable for stationary or marine engines. Good results have been achieved using HCCI engine in hybrid powertrain systems [45,46].

In light of the experimental results obtained in this study, the Double Fuel operation brings undeniable advantages for the realization of HCCI combustion: in particular, the simultaneous injection of both gasoline and natural gas allows a continuous and accurate control of the fuel mixture knock resistance, which can be adapted to the required HCCI engine load by varying the NG content. For this reason the Double Fuel strategy can be considered, together with EGR or supercharging, an efficient system to enhance HCCI combustion performance in terms of range of operating conditions as well as combustion control in non-steady conditions and during transition phase from other combustion mode (SI or CI).

Acknowledgements

The authors express their acknowledgments to Mr. Beniamino Drago for the technical support provided in the realization of the engine test bench.

References

[1] N. Fraser, H. Blaxill, G. Lumsden, M. Bassett, Challenges for Increased efficiency through gasoline engine downsizing, SAE Int. J. Engines 2 (1) (2009) 991–1008, http://dx.doi.org/10.4271/2009-01-1053.

[2] B. Lecointe, G. Monnier, Downsizing a Gasoline Engine Using Turbocharging with Direct Injection, SAE Technical Paper 2003-01-0542, 2003. http://dx.doi.org/10.4271/2003-01-0542.

[3] D. Petitjean, L. Bernardini, C. Middlemass, S. Shahed, Advanced Gasoline Engine Turbocharging Technology for Fuel Economy Improvements, SAE Technical Paper 2004-01-0988, 2004. http://dx.doi.org/10.4271/2004-01-0988.

[4] T. Lake, J. Stokes, R. Murphy, R. Osborne, et al., Turbocharging Concepts for Downsized DI Gasoline Engines, SAE Technical Paper 2004-01-0036, 2004. http://dx.doi.org/10.4271/2004-01-0036.

[5] Alasdair Cairns, Hua Zhao, Alan Todd, Pavlos Aleiferis, A study of mechanical variable valve operation with gasoline–alcohol fuels in a spark ignition engine, Fuel 106 (2013) 802–813, http://dx.doi.org/10.1016/j.fuel.2012.10.041. ISSN 0016-2361.

[6] S. Verhelst, J. Demuynck, R. Sierens, P. Huyskens, Impact of variable valve timing on power, emissions and backfire of a bi-fuel hydrogen/gasoline engine, Int. J. Hydrogen Energy 35 (9) (2010) 4399–4408, http://dx.doi.org/10.1016/j.ijhydene.2010.02.022. ISSN 0360-3199.

[7] G. Fontana, E. Galloni, Variable valve timing for fuel economy improvement in a small spark-ignition engine, Appl. Energy 86 (1) (2009) 96–105, http://dx.doi.org/10.1016/j.apenergy.2008.04.009. ISSN 0306-2619.

[8] Kamil Çağatay Bayindir, Mehmet Ali Gözüküçük, Ahmet Teke, A comprehensive overview of hybrid electric vehicle: powertrain configurations, powertrain control techniques and electronic control units, Energy Convers. Manage. 52 (2) (2011) 1305–1313, http://dx.doi.org/10.1016/j.enconman.2010.09.028. ISSN 0196-8904.

[9] K.T. Chau, Y.S. Wong, Overview of power management in hybrid electric vehicles, Energy Convers. Manage. 43 (15) (2002) 1953–1968, http://dx.doi.org/10.1016/S0196-8904(01)00148-0. ISSN 0196-8904.

[10] T. Korakianitis, A.M. Namasivayam, R.J. Crookes, Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance and emissions, Prog. Energy Combust. Sci. 37 (1) (2011) 89–112, http://dx.doi.org/10.1016/j.pecs.2010.04.002. ISSN 0360-1285.

[11] H. Albela et al., Performance and emissions of LPG fueled internal combustion engine: a review, Int. J. Sci. Eng. Res. 3 (3) (2012). ISSN 2229-5518.

[12] J.E. Anderson, D.M. DiCicco, J.M. Ginder, U. Kramer, T.G. Leone, H.E. Raney-Pablo, T.J. Wallington, High octane number ethanol–gasoline blends: quantifying the potential benefits in the United States, Fuel 97 (2012), http://dx.doi.org/10.1016/j.fuel.2012.03.017.

[13] Ghazi A. Karim, Hydrogen as a spark ignition engine fuel, Int. J. Hydrogen Energy 28 (2003) 569–577, http://dx.doi.org/10.1016/S0360-3199(02)00150-7.

[14] Hien Ly, Effects of natural gas composition variations on the operation, performance and exhaust emissions of natural gas – powered vehicles, in: NGV 2002 Conference Paper – Effects of Gas Composition, August 2002.

[15] E. Pipitone, S. Beccari, Performance and Emission Improvement of a S.I. Engine Fuelled by LPG/Gasoline Mixtures, SAE Technical Paper 2010-01-0615, SI Combustion and Direct Injection SI Engine Technology, 2010. http://dx.doi.org/10.4271/2010-01-0615.

[16] E. Pipitone, S. Beccari, Performances Improvement of a S.I. CNG Bi-Fuel Engine by Means of Double-Fuel Injection, SAE Paper 2009-24-0058, 2009, ISSN 0148-7191, http://dx.doi.org/10.4271/2009-24-0058.

[17] G. Genchi, E. Pipitone, Octane rating of natural gas-gasoline mixtures on CFR engine, SAE Int. J. Fuels Lubr. 7 (3) (2014), http://dx.doi.org/10.4271/2014-01-9081.

[18] E. Pipitone, G. Genchi, Experimental determination of liquefied petroleum gas-gasoline mixtures knock resistance, ASME J. Eng. Gas Turbines Power 136 (12) (2014), http://dx.doi.org/10.1115/1.4027831. 121502–121502-7.

[19] V. Delpech, J. Obiols, D. Soleri, L. Mispreuve, et al., Towards an innovative combination of natural gas and liquid fuel injection in spark ignition engines, SAE Int. J. Fuels Lubr. 3 (2) (2010) 196–209, http://dx.doi.org/10.4271/2010-01-1513.

[20] J. Obiols, D. Soleri, N. Dioc, M. Moreau, Potential of Concomitant Injection of CNG and Gasoline on a 1.6L Gasoline Direct Injection Turbocharged Engine, SAE Technical Paper 2011-01-1995, 2011. http://dx.doi.org/10.4271/2011-01-1995.

[21] M. Momeni Movahed, H. Basirat Tabrizi, M. Mirsalim, Experimental investigation of the concomitant injection of gasoline and CNG in a turbocharged spark ignition engine, Energy Convers. Manage. (2014), http://dx.doi.org/10.1016/j.enconman.2014.01.017.

[22] M. Veiga, R. Mansano, R. Silva, C. Gomes, Injection System for Tri-Fuel Engines with Control of Power by Simultaneous use of CNG and Ethanol or Gasoline, SAE Technical Paper 2010-36-0195, 2010. http://dx.doi.org/10.4271/2010-36-0195.

[23] F. Zhao et al., Homogeneous Charge Compression Ignition (HCCI) Engines: key Research and Development Issues, SAE International, 2003. ISBN of 978-0-7680-1123-4.

[24] Mohammad Izadi Najafabadi, Nuraini Abdul Aziz, Homogeneous charge compression ignition combustion: challenges and proposed solutions, J. Combust. 2013 (2013) 14, http://dx.doi.org/10.1155/2013/783789 Article ID 783789.

[25] J. Hyvönen, G. Haraldsson, B. Johansson, Supercharging HCCI to Extend the Operating Range in a Multi-Cylinder VCR-HCCI Engine, SAE Technical Paper 2003-01-3214, 2003. http://dx.doi.org/10.4271/2003-01-3214.

[26] Tengku Nordayana Akma Tuan Kamaruddin, Mazlan Abdul Wahid, Mohsin Mohd Sies, Review of homogeneous charge compression ignition (HCCI) combustion engines and exhaust gas recirculation (EGR) effects on HCCI, AIP Conference Proceedings, vol. 1440, 2012, pp. 1099–1109, http://dx.doi.org/10.1063/1.4704326.

[27] K. Ebrahimi, C.R. Koch, HCCI combustion timing control with variable valve timing, in: American Control Conference (ACC), 2013, pp. 4429–4434, 17–19 June 2013. http://dx.doi.org/10.1109/ACC.2013.6580522.
[28] D. Gérard, M. Besson, J. Hardy, S. Croguennec, et al., HCCI Combustion on a Diesel VCR Engine, SAE Technical Paper 2008-01-1187, 2008. http://dx.doi.org/10.4271/2008-01-1187.
[29] Kazunobu Kobayashi, Takahiro Sako, Yoshimi Sakaguchi, Satoshi Morimoto, Sumihiro Kanematsu, Kiyoshi Suzuki, Tohru Nakazono, Hiroyuki Ohtsubo, Development of HCCI natural gas engines, J. Natl. Gas Sci. Eng. 3 (5) (2011) 651–656, http://dx.doi.org/10.1016/j.jngse.2011.07.002. ISSN 1875-5100.
[30] Rakesh Kumar Maurya, Avinash Kumar Agarwal, Experimental study of combustion and emission characteristics of ethanol fuelled port injected homogeneous charge compression ignition (HCCI) combustion engine, Appl. Energy 88 (4) (2011) 1169–1180, http://dx.doi.org/10.1016/j.apenergy.2010.09.015. ISSN 0306-2619.
[31] Ali Turkcan, Ahmet Necati Ozsezen, Mustafa Canakci, Experimental investigation of the effects of different injection parameters on a direct injection HCCI engine fueled with alcohol–gasoline fuel blends, Fuel Process. Technol. 126 (2014) 487–496, http://dx.doi.org/10.1016/j.fuproc.2014.05.023. ISSN 0378-3820.
[32] H.-W. Wu, R.-H. Wang, D.-J. Ou, Y.-C. Chen, T.-Y. Chen, Reduction of smoke and nitrogen oxides of a partial HCCI engine using premixed gasoline and ethanol with air, Appl. Energy 88 (2011) 3882–3890, http://dx.doi.org/10.1016/j.apenergy.2011.03.027.
[33] H. El-Din, M. Elkelawy, Z. Yu-Sheng, HCCI Engines Combustion of CNG Fuel with DME and H2 Additives, SAE Technical Paper 2010-01-1473, 2010. http://dx.doi.org/10.4271/2010-01-1473.
[34] M. Fathi, R.K. Saray, M.D. Checkel, The influence of exhaust gas recirculation (EGR) on combustion and emissions of n-heptane/natural gas fueled homogeneous charge compression ignition (HCCI) engines, Appl. Energy 88 (12) (2011) 4719–4724, http://dx.doi.org/10.1016/j.apenergy.2011.06.017.
[35] R.D. Reitz, G. Duraisamy, Review of high efficiency and clean reactivity controlled compression ignition (RCCI) combustion in internal combustion engines, Prog. Energy Combust. 46 (2015) 12–71, http://dx.doi.org/10.1016/j.pecs.2014.05.003.
[36] Standard Test Method for Motor Octane Number of Spark-Ignition Engine Fuel, ASTM International D2700.
[37] E. Pipitone, A comparison between combustion phase indicators for optimal spark timing, ASME J. Eng. Gas Turbines Power 130 (5) (2008), http://dx.doi.org/10.1115/1.2939012.
[38] E. Pipitone, A. Beccari, Determination of TDC in internal combustion engines by a newly developed thermodynamic approach, Appl. Therm. Eng. 30 (14) (2010) 1914–1926, http://dx.doi.org/10.1016/j.applthermaleng.2010.04.012.
[39] Mark E. Myers Jr., Janis Stollstelmer, Andrew M. Wims, Determination of hydrocarbon-type distribution and hydrogen/carbon ratio of gasolines by nuclear magnetic resonance spectrometry, Anal. Chem. 47 (12) (1975).
[40] J.E. Anderson, U. Kramer, S.A. Mueller, T.J. Wallington, Octane numbers of ethanol-and methanol-gasoline blends estimated from molar concentrations, Energy Fuels 24 (12) (2010) 6576–6585, http://dx.doi.org/10.1021/ef101125c.
[41] Bob Boundy, Susan W. Diegel, Lynn Wright, Stacy C. Davis, Biomass Energy Data Book: Edition 4, U.S. Department Of Energy, 2011, ORNL/TM-2011/446. <http://cta.ornl.gov/bedb>.
[42] Byung Hyouk Min, Jin Taek Chung, Ho Young Kim, Simsoo Park, Effects of gas composition on the performance and hydrocarbon emissions for CNG engines, KSME Int. J. 16 (2) (2002) 219–226, http://dx.doi.org/10.1007/BF03185173.
[43] J. Kubesh, S. King, W. Liss, Effect of Gas Composition on Octane Number of Natural Gas Fuels, SAE Technical Paper 922359, 1992. http://dx.doi.org/10.4271/922359.
[44] Standard Practice for Calculating Heat Value, Compressibility Factor, and Relative Density of Gaseous Fuels, ASTM International D3588.
[45] R. Sun, R. Thomas, C. Gray, An HCCI Engine: Power Plant for a Hybrid Vehicle, SAE Technical Paper 2004-01-0933, 2004. http://dx.doi.org/10.4271/2004-01-0933.
[46] B. Lawler, E. Ortiz-Soto, R. Gupta, H. Peng, et al., Hybrid electric vehicle powertrain and control strategy optimization to maximize the synergy with a gasoline HCCI engine, SAE Int. J. Engines 4 (1) (2011) 1115–1126, http://dx.doi.org/10.4271/2011-01-0888.