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Emission characteristics of high speed, dual fuel, compression ignition engine operating in a wide range of natural gas/diesel fuel proportions

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ABSTRACT

In the effort to reduce pollutant emissions from diesel engines various solutions have been proposed, one of which is the use of natural gas as supplement to liquid diesel fuel, with these engines referred to as fumigated, dual fuel, compression ignition engines. One of the main purposes of using natural gas in dual fuel (liquid and gaseous one) combustion systems is to reduce particulate emissions and nitrogen oxides. Natural gas is a clean burning fuel; it possesses a relatively high auto-ignition temperature, which is a serious advantage over other gaseous fuels since then the compression ratio of most conventional direct injection (DI) diesel engines can be maintained high. In the present work, an experimental investigation has been conducted to examine the effects of the total air–fuel ratio on the efficiency and pollutant emissions of a high speed, compression ignition engine located at the authors' laboratory, where liquid diesel fuel is partially substituted by natural gas in various proportions, with the natural gas fumigated into the intake air. The experimental results disclose the effect of these parameters on brake thermal efficiency, exhaust gas temperature, nitric oxide, carbon monoxide, unburned hydrocarbons and soot emissions, with the beneficial effect of the presence of natural gas being revealed. Given that the experimental measurements cover a wide range of liquid diesel supplementary ratios without any appearance of knocking phenomena, the belief is strengthened that the findings of the present work can be very valuable if opted to apply this technology on existing DI diesel engines.

1. Introduction

Many internal combustion engines, usually converted from commercial compression or spark ignition engines, have been fueled with various alternative gaseous fuels, such as natural gas, liquefied petroleum gas and less frequently biogases, for use in power generation, transportation and other applications [1–5]. Certainly, one of the main objectives for improving the combustion process of conventional internal combustion engines is to find effective ways to reduce exhaust emissions, without making significant modifications on their mechanical structure [6–9]. Various solutions have been proposed, and among them the use of natural gas as a supplement to the conventional liquid fuel possesses a dominant place, owing to its inherent clean nature of combustion [4,5,10–13]. Natural gas produces practically no particulates since it contains few dissolved impurities (e.g. sulfur compounds). Moreover, natural gas can be used in compression ignition engines (dual fuel diesel–natural gas engines) since the auto-ignition temperature of the gaseous fuel is higher compared to the one of conventional liquid diesel fuel [4,5,10–13].

Dual fuel diesel–natural gas engines feature essentially a homogeneous natural gas–air mixture compressed rapidly below its auto-ignition conditions and ignited by the injection of an amount of liquid diesel fuel around top dead center position. Natural gas is fumigated into the intake air and premixed with it during the induction stroke. At constant engine speed, the fumigated gaseous fuel replaces an equal amount of the inducted combustion air (on a volume basis) since the total amount of the inducted mixture has to be kept constant. Furthermore, under fumigated dual fuel operating mode, the desired engine power output (i.e. brake mean effective pressure) is controlled by changing the amounts of the fuels used. Thus, at a given combination of engine speed and load, the change of the liquid fuel "supplementary ratio" leads to a change of the inhaled combustion air, thus resulting to the alteration of the total relative air–fuel ratio.

A number of experimental and theoretical investigations concerning the dual fuel diesel–natural gas operating mode have been reported in the literature. Various researchers [4,5,10–28] have published extensive theoretical and experimental investigations concerning the combustion processes occurring inside the

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Nomenclature

$\dot{m}$	mass flow rate (kg/s)
X	diesel fuel supplementary ratio (%)
λ	total relative air–fuel ratio

Subscripts

D	diesel
NG	natural gas
st	stoichiometric
tot	total

Abbreviations

AFR	air–fuel ratio (by mass) (kg/kg)
BDC	bottom dead center
bmep	brake mean effective pressure (bar)
BTE	brake thermal efficiency
°CA	degrees crank angle
CO	(specific) carbon monoxide (g/kWh)
COV	coefficient of variance (%)
DI	direct injection
LHV	lower heating value (kJ/kg)
NG	natural gas
NO	(specific) nitric oxide (g/kWh)
rpm	revolutions per minute
TDC	top dead center
HC	(specific) unburned hydrocarbons (g/kWh)

combustion chamber of a dual fuel diesel–natural gas compression ignition engine. Specifically, several investigators have presented theoretical and experimental investigations on the effect of the pilot diesel fuel amount and its injection timing on the performance of a dual fuel engine. Furthermore, other theoretical and experimental investigations have also been conducted on the combustion phenomena occurring inside the combustion chamber of an advanced injection low pilot-ignited natural gas diesel engine fueled with natural gas.

During the last years, the present research group has reported experimental investigations along with computer simulations conducted on such kind of engines [29–35]. In order to examine the effect of the total air–fuel ratio on brake efficiency and exhaust emissions of a dual fuel diesel engine, an extended experimental investigation was conducted on a single cylinder, naturally aspirated, air-cooled, high speed, direct injection diesel test engine, located at the authors' laboratory, properly modified to operate under dual fuel (i.e. diesel and natural gas) mode. The results reported here concern brake thermal efficiency, exhaust gas temperature, and NO, CO, HC and Soot emissions for various engine operating conditions, i.e. loads and speeds. The main objectives of this comparative assessment are to record and evaluate the relative impact of the above mentioned total air–fuel ratio on engine efficiency and emitted pollutants. Furthermore, the present investigation deals with determining the optimum range of variation of the parameter referred to above, since at high engine load conditions the significant change of this specific parameter may lead to undesirable effects on engine performance characteristics.

The information derived from the present investigation can be extremely valuable if one wishes to apply dual fueling on an existing high speed, direct injection diesel engine. It will be accomplished through the estimation of the proper total air–fuel ratio, which is affected by the diesel–natural gas fuels consumption ratio, so that the engine operation becomes friendlier to the environment without deteriorating its performance compared to that of the conventional diesel operation.

2. Experimental procedure

2.1. Experimental facilities

Facilities to monitor and control engine variables are installed on a test bed, single cylinder, naturally aspirated, four-stroke, air-cooled, direct injection, high speed, Lister LV1 diesel engine having a bowl-in-piston combustion chamber. The most important technical data of the test engine used are provided in Table 1. A high pressure fuel pump having a 6.5 mm diameter plunger is connected to a three hole injector nozzle (each hole having a diameter of 0.23 mm), which is located in the middle of the cylinder head. The injector nozzle opening pressure is 180 bar. The engine is coupled to a Heenan and Froude hydraulic dynamometer.

The main properties of the liquid diesel fuel used are given in Table 2. The present liquid fuel is the representative of a typical commercial automotive diesel fuel. The composition of the natural gas used, based on data obtained from the supplier at the time of measurements, is given also in Table 2. As observed, methane is the main constituent of the natural gas resulting in a relatively high octane number, which makes it suitable for use in engines with high compression ratio. The stoichiometric air–fuel ratio of the liquid diesel fuel AFR_D^{st} has been determined through the application of the stoichiometric combustion equation, taking into account its (elemental mass) composition. The stoichiometric air–fuel ratio of natural gas AFR_{NG}^{st} is then the sum of the products of the stoichiometric air–fuel ratio of each one of the gaseous fuel hydrocarbon constituent with its mass fraction, which was in turn determined through the precise volumetric composition and the molar masses of the constituent and the fuel. The stoichiometric air–fuel ratio for each hydrocarbon constituent is determined through the application of the stoichiometric combustion equation.

The air drawn-in by the engine is passed first through a filter and then through a viscous type air-flow meter, where the inducted volumetric air-flow rate is measured. There is no need to use an air damper to damp the air pulsations, since the highly viscous element damps the pulsations of the air-flow and produces a smooth average flow towards the engine cylinder. The fuel system consists of both liquid diesel fuel injection and natural gas supply subsystems. The engine can be easily switched over to operate on

Table 1
Basic data of the test engine.

Engine type	Single cylinder (four-stroke) (DI)
Bore	85.73 mm
Stroke	82.55 mm
Connecting rod	180 mm
Compression ratio	17.6: 1
Cylinder dead volume	28.03 cm^3
Normal operation speed	1000–3000 rpm
Inlet valve opening	15 °CA before TDC
Inlet valve closure	41 °CA after BDC
Exhaust valve opening	41 °CA before BDC
Exhaust valve closure	15 °CA after TDC
Inlet valve diameter	34.5 mm
Exhaust valve diameter	31.5 mm
Static injection timing	26 °CA before TDC
Maximum power output	6.7 kW at 3000 rpm
Maximum torque	25 Nm at 2000 rpm

Table 2
Basic characteristics of the fuels used.

Liquid diesel fuel (CEN EN-590)	
Cetane number	52.5
Density	833.7 kg/m^3
LHV	42.74 MJ/kg
Sulfur content	45 mg/kg
Stoichiometric air–fuel ratio	14.5 kg/kg
Natural gas (ISO 6974-6)	
Methane	98% (v/v)
Ethane	0.6% (v/v)
Propane	0.2% (v/v)
Butane	0.2% (v/v)
Pentane	0.1% (v/v)
Nitrogen	0.8% (v/v)
Carbon dioxide	0.1% (v/v)
LHV	48.6 MJ/kg
Stoichiometric air–fuel ratio	17.3 kg/kg

either pure diesel fuel (normal diesel operation) or diesel and natural gas (dual fuel operation) operating modes. The engine is supplied with natural gas obtained from the mains of the local distribution network. The gaseous fuel, before entering the engine cylinder, passes through a small tank (to dampen the pressure fluctuations from the engine intake) and two flow meters (for accuracy), a positive displacement and a rotary flow one. The adjustment of gaseous fuel supply is accomplished through a control valve located after the flow meter. Then, the gaseous fuel flows towards the engine intake and mixes with the intake air.

The main instrumentation used for pollutant emissions measurements consists of a Bosch smokemeter to measure smoke levels in the exhaust gas. Further, NO emissions are measured using a chemiluminescent analyzer, while (total) unburned HC emissions are measured with a flame ionization detector. The last two devices are fitted with thermostatically controlled heated lines. CO is measured with a non-dispersive infrared analyzer. The main characteristics of the instrumentation for measuring the pollutant emissions are given in Table 3.

2.2. Test cases examined

For all test cases examined during the experimental investigation, the air inlet temperature was measured about 23 °C and the air absolute humidity about 50%. Measurements were taken at three engine speeds of 1500, 2000 and 2500 rpm, and four different engine loads corresponding to brake mean effective pressures (bmep) of 1.2, 2.4, 3.7 and 4.9 bar, respectively, under both normal diesel ($x = 0\%$) and dual fuel ($x > 0\%$) operations. According to the experimental procedure, under dual fuel operating mode at a given engine speed sufficient amount of liquid diesel fuel is provided to achieve a percentage of the desired engine power output. The rest of the percentage of the desired engine power output is achieved by using only natural gas, which is fumigated into the air intake.

The diesel fuel supplementary ratio x, which represents the quotient of the mass flow rate of natural gas divided by the total fuel (diesel and natural gas) mass flow rates, is given by the formula:

$$x = \frac{\dot{m}_{\mathrm{NG}}}{\dot{m}_{\mathrm{D}} + \dot{m}_{\mathrm{NG}}} \cdot 100\ (\%) \tag{1}$$

The term $\dot{m}_{\mathrm{D}}$ represents diesel fuel consumption measured by a flow meter appropriate for liquid fuel, while $\dot{m}_{\mathrm{NG}}$ is the gaseous fuel consumption measured by a rotary displacement gaseous fuel flow meter.

Further, the corresponding total relative air–fuel ratio (i.e. taking into account both fuels) is given as:

$$\lambda_{\mathrm{tot}} = \frac{\dot{m}_{\mathrm{air}}}{AFR^{\mathrm{st}}_{\mathrm{NG}} \cdot \dot{m}_{\mathrm{NG}} + \mathrm{AFR}^{\mathrm{st}}_{\mathrm{D}} \cdot \dot{\mathrm{m}}_{\mathrm{D}}} \tag{2}$$

where, $\mathrm{AFR}^{st}_{\mathrm{D}}$ and $\mathrm{AFR}^{\mathrm{st}}_{\mathrm{NG}}$ correspond to the stoichiometric air–fuel ratio (by mass) for the diesel fuel and the natural gas, respectively. The corresponding values are provided in Table 2. The mass flow rate of the inhaled air $\dot{m}_{\mathrm{air}}$ is measured by a viscous type airflow meter.

It is pointed out that at constant engine speed the amount of the gaseous fuel fumigated into the intake air replaces an equal amount (on a volume basis) of the inducted combustion air, since the total amount of the inducted mixture has to be kept constant. Furthermore, the desired engine load (i.e. bmep) is controlled by changing the amounts of the fuels used. Sufficient amount of liquid diesel fuel is provided to achieve a percentage of the desired engine load, while the rest of the percentage of the desired bmep is reached by using only natural gas fumigated into the intake air, thus resulting to the reduction of the inhaled combustion air. Then, at a given engine speed (i.e. the mass flow rate of the inducted mixture air – gaseous fuel is kept constant), the change of the liquid fuel "supplementary ratio" leads to a change of the inhaled combustion air, thus resulting to the alteration of the total relative air–fuel ratio calculated by Eq. (2). In the present work, the maximum percentage of the liquid diesel fuel substituted by natural gas is up to 86%, while the least total relative air–fuel ratio goes down to 1.28 depending on engine operating conditions, i.e. load and speed. Moreover, for the same engine load, the liquid fuel supplementary ratio decreases with the increase of the engine speed, which discloses an increase of the injected diesel fuel. This is probably attributed to the increase of engine mechanical losses.

3. Uncertainty analysis of the experimental data

For each engine operating mode, i.e. normal diesel and dual fuel operation, two sets of measurements are taken. At each engine operating point three measurements were taken and, thus, the values reported for all measured parameters are the mean ones from six different measurements. This makes it possible to estimate the repeatability of measured data and the relevant measuring error. To estimate the accuracy of the measurements, the coefficient of variance (COV) for each measured parameter is determined. This represents the standard deviation of each magnitude as a percentage of its mean value. The value of COV for each measured parameter is presented in Table 4, from where it is apparent that the measurements are quite repeatable, especially concerning engine performance characteristics.

Table 3
Basic instrumentation for pollutant emissions measurements.

Pollutant	Instrument	Type	Operating principle
Soot	Bosch	RTT100	Optical
NO	Signal	4000 Series	Chemiluminescence
HC	Ratfisch	RS55	Flame Ionization
CO	Signal	7200 Series	Non-dispersive infrared

Table 4
Coefficient of variance (COV) for the measured quantities.

Measured quantity	COV (%)
Exhaust gas temperature	3.1
Nitric oxide	3.5
Carbon monoxide	2.9
Unburned hydrocarbons	3.2
Soot	3.9

4. Results and discussion

In this section, experimental results are provided concerning the effect of the total air–fuel ratio on engine efficiency and pollutant emissions of the high speed, dual fuel, compression ignition engine, operating in a wide range of natural gas/diesel fuel proportions. By total it is meant that both the natural gas and diesel fuel masses are taken into account. It is reiterated here that, for the same engine operating point (i.e. load and engine speed), the increase of the liquid fuel supplementary ratio is accompanied by a decrease of the total relative air–fuel ratio, since the increased mass flow rate of the gaseous fuel fumigated into the intake air results to a considerable decrease of the inhaled combustion air.

All figures to follow, with the exemption of Fig. 2 that shows several heat release rate diagrams, provide in a bar chart arrangement each performance or emission parameter as a function of the liquid fuel supplementary ratio *x* and the total relative air–fuel ratio λ at various loads (bmep of 1.2, 2.4, 3.7 and 4.9 bar), for the engine speeds of 1500, 2000 and 2500 rpm (from now on, for clarity, λ_{tot} is referred to simply as λ). In the subsections below, the effect of dual fuel operation on the following parameters is examined: brake thermal efficiency, exhaust gas temperature (in °C), nitric oxide emissions, carbon monoxide emissions, unburned hydrocarbons emissions and soot density (in mg per liter of exhaust gases), consecutively. The nitric oxide, carbon monoxide and unburned hydrocarbons emissions are specific emission values, expressed in g/kWh.

4.1. Brake thermal efficiency and heat release rate diagrams

Fig. 1a–c illustrates the variation of the experimental values of the brake thermal efficiency (BTE). For each test case examined, the brake thermal efficiency is calculated by multiplying the reciprocal value of the total brake specific energy consumption by 3.6. The total brake specific energy consumption, expressed in MJ/kWh, is computed from the measured brake power output, the measured mass flow rates of both diesel and natural gas and their lower heating values. Observing this figure, it is revealed that the natural gas–diesel dual fuel operation affects the engine brake thermal efficiency since it has influence on the history of combustion in the cylinder charge.

In order to support in more detail the conclusions regarding the effect of the examined parameters on the brake thermal efficiency and pollutants, total heat release rate curves corresponding to various combinations of total relative air–fuel ratios and engine operating points (i.e. engine load and speed) are provided in Fig. 2a–d; here, 180 °CA (degrees crank angle) correspond to "hot" TDC position. Examining Fig. 2a–d, it is revealed that the presence of natural gas in the cylinder charge affects the combustion process as detailed in the following three paragraphs.

It is observed that the initiation of combustion under dual fuel operating mode starts later compared to the respective one under normal diesel operation. This is due to the fact that under dual fuel operation the cylinder charge (i.e. gaseous fuel – air mixture) has higher overall specific heat capacity compared to the respective one (i.e. air) under normal diesel operating mode. This has an increasing effect on the ignition delay period of the liquid diesel fuel. This becomes more evident at high liquid fuel supplementary ratios.

During the initial stages of the premixed controlled combustion phase it is revealed that the total heat release rate decreases with the increase of the gaseous fuel concentration in the cylinder charge. This is due to the lower amount of diesel fuel burned during the specific combustion phase and also to the fact that the com-

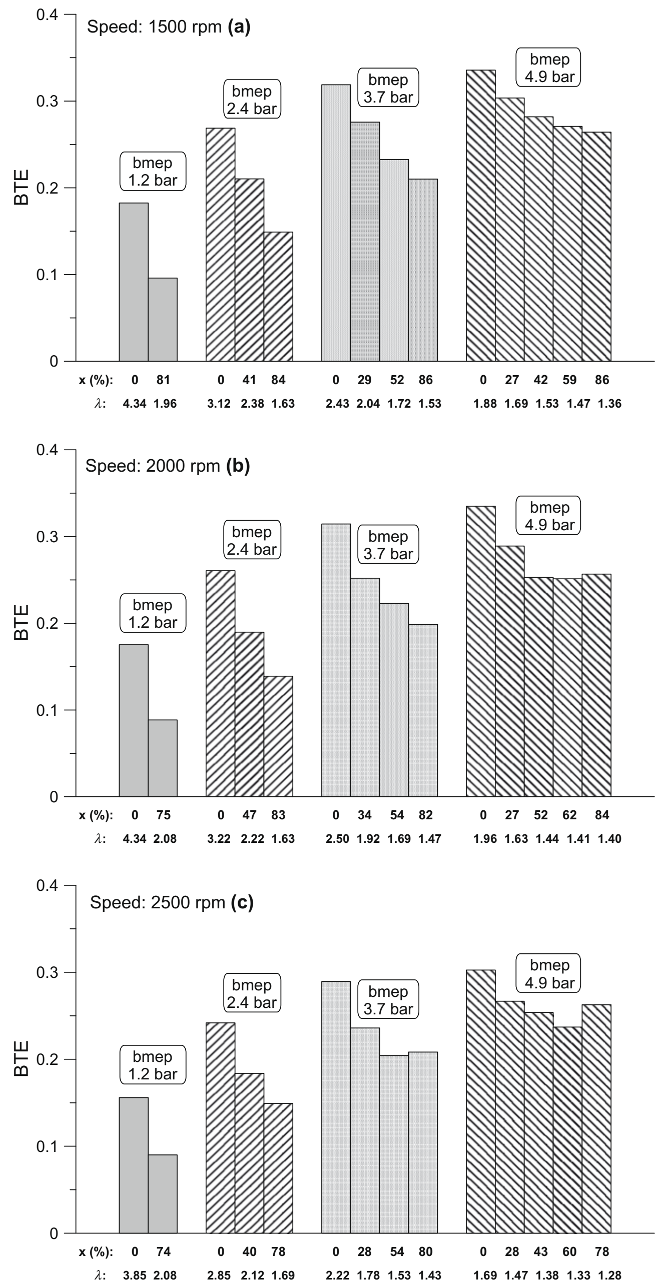

Fig. 1. Bar chart diagram of the brake thermal efficiency (BTE) at various loads (bmep) as function of the supplementary ratio '*x*' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

bustion of the gaseous fuel has not yet progressed enough, since the cylinder charge conditions (i.e. cylinder charge temperature, gaseous fuel concentration etc.) do not favor the existence of the flame front.

As far as the diffused controlled combustion phase is concerned, it is revealed that at low load conditions (Fig. 2a and c) the total heat release rate observed under dual fuel operating mode is almost the same with the respective one observed under normal diesel operation, so revealing the poor utilization of the gaseous fuel. At high liquid fuel supplementary ratio, the cylinder charge conditions favor the existence of the flame front, resulting in a slight improvement of the gaseous fuel combustion. At high load, the total heat release rate during the diffused controlled combustion phase increases as the concentration of the gaseous fuel increases, so revealing that the quality of the gaseous fuel combustion has

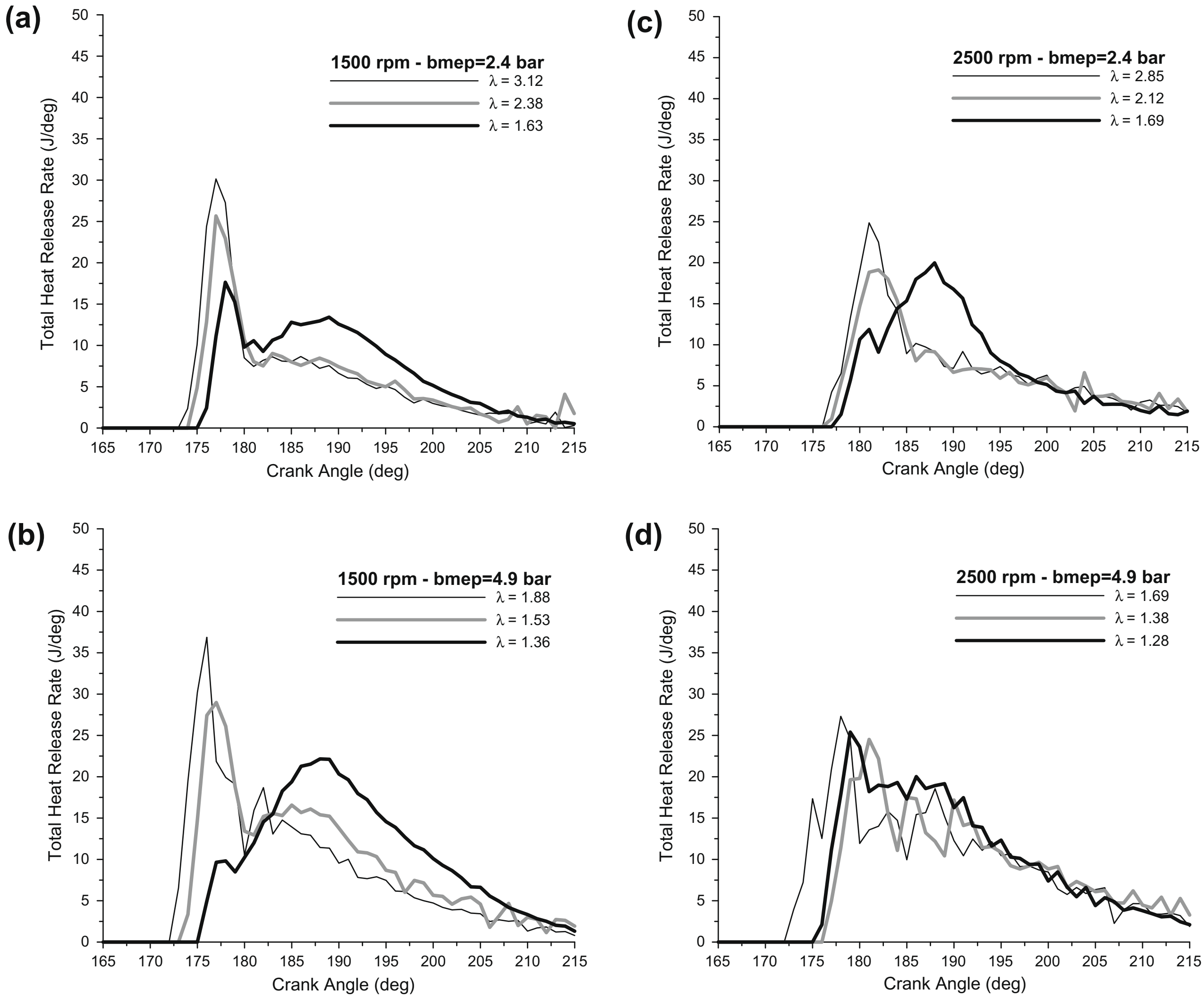

Fig. 2. Total heat release rate diagrams for various total relative air–fuel ratios 'λ', at 1500 rpm engine speed and bmep = 2.4 bar (a), at 1500 rpm engine speed and bmep = 4.9 bar (b), at 2500 rpm engine speed and bmep = 2.4 bar (c), and at 2500 rpm engine speed and bmep = 4.9 bar (d).

been improved compared to the respective one at low load. However, this does not have any significant effect on the cylinder pressure, since the major part of the gaseous fuel combustion occurs during the expansion stroke.

At low and intermediate load conditions and for all engine speeds examined, the decrease of the total relative air–fuel ratio results in a decrease of the brake thermal efficiency. The effect is more pronounced at low load (bmep = 1.2 bar) where the engine efficiency observed under dual fuel operation ($x > 0\%$) is almost half as compared to the one observed under normal diesel operation ($x = 0\%$). This is the outcome of the extremely lower premixed controlled combustion rate observed under dual fuel operation. For low total relative air–fuel ratios ($\lambda = 1.96$ or $\lambda = 2.08$), the extremely low amount of diesel fuel used affects adversely the quality of the diesel fuel spray, resulting in poor diesel fuel preparation, especially during the ignition delay period, which has an adverse effect on the gaseous fuel combustion process. At intermediate loads, the decrease of the total relative air–fuel ratio leads to a milder decrease of BTE as compared to the respective one observed at low load. Specifically, at bmep of 2.4 bar the decrease of BTE is up to 75% while at bmep of 3.7 bar the corresponding decrease is up to 40%. This is the outcome of the improvement of the gaseous fuel combustion quality observed at the specific engine operating conditions (Fig. 2a and c). This improvement may be attributed to higher natural gas burning rates, especially during the premixed controlled combustion phase, since the decrease of the total relative air–fuel ratio affects the ignition delay period of the injected liquid fuel. The higher values of diffusion controlled combustion rates observed under dual fuel operation do not improve engine efficiency, because they occur late into the expansion phase (Fig. 2a and c).

At high load and low engine speed, the decrease of the total relative air–fuel ratio caused by the increased gaseous fuel percentage inside the cylinder charge has a slight effect on the engine efficiency compared to the respective one observed under normal diesel operation. Thus, the maximum decrease of the brake thermal efficiency observed with the total relative air–fuel ratio is up to 15%. This is the result of the improvement of the gaseous fuel combustion quality. The specific improvement is caused by the amelioration of the cylinder charge conditions, which contributes significantly to the existence and the fast spread of the flame front surrounding the burning zone (Fig. 2b).

At high load and high engine speed, the brake thermal efficiency starts to decrease slightly with the decrease of the total relative air–fuel ratio. A further decrease of the total relative air–fuel ratio, beyond a certain value, leads to a slight increase of the brake thermal efficiency. This is due to the fact that the increase of the engine speed combined with low air–fuel ratio has a positive effect on the existence and spread of the flame front surrounding the burning zone. Specifically, the increase of engine speed leads to a warmer engine and to an increase of the turbulence inside the combustion chamber. These factors have a positive effect (enhance) on the flame speed, thus contributing to the improvement of gaseous fuel combustion quality (lower ignition delay and faster flame speed) (Fig. 2d).

4.2. Exhaust gas temperature

Fig. 3a–c provides the measured values of the exhaust gas temperature. Observing this figure, it is revealed that for each combination of load and total relative air–fuel ratio examined the increase of engine speed results to higher exhaust gas temperatures due to the longer duration of combustion. Moreover, for each engine speed examined the natural gas–diesel dual fuel combustion causes an increase in the exhaust gas temperature.

At low load and for low diesel fuel supplementary ratio, the effect of the natural gas–diesel dual fuel combustion on exhaust gas temperature is almost negligible as compared to the respective one observed under normal diesel operation ($x = 0\%$). At high diesel fuel supplementary ratio the exhaust gas temperature is slightly lower compared to the one observed under normal diesel operation. This is due to the fact that the increased proportion of the gaseous fuel inside the cylinder chamber reduces the air availability inside the combustion chamber, which enhances the combustion quality of the gaseous fuel, thus resulting in a shorter duration of combustion (Fig. 2a and c).

At high load the increase of the liquid fuel supplementary ratio leads to an increase of the exhaust gas temperature. A further increase beyond a certain value leads to a decrease of the exhaust

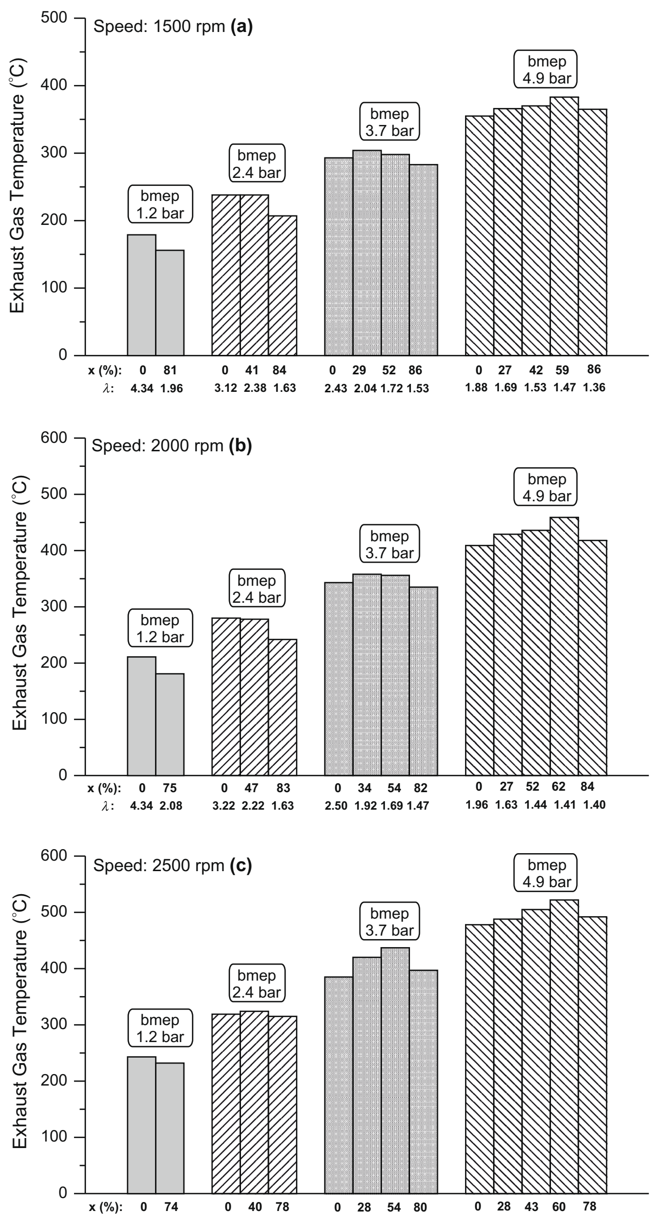

Fig. 3. Bar chart diagram of the exhaust gas temperature at various loads (bmep) as function of the supplementary ratio 'x' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

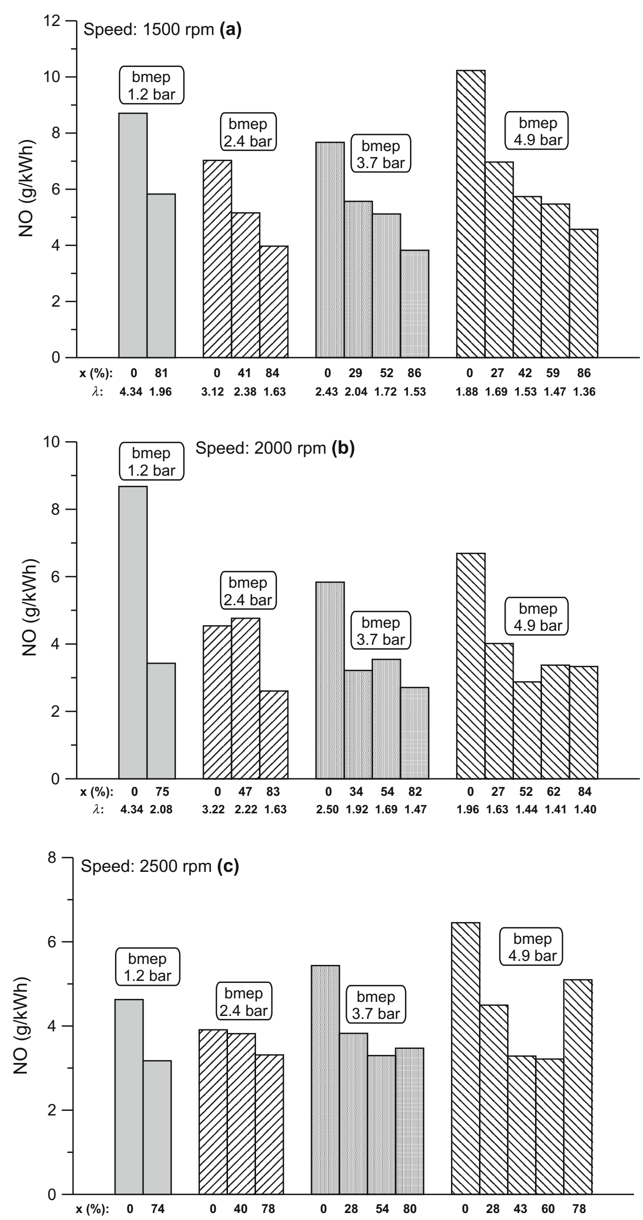

Fig. 4. Bar chart diagram of the emitted specific nitric oxide (NO) at various loads (bmep) as function of the supplementary ratio 'x' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

gas temperature. This is mainly due to the fact that at high total relative air–fuel ratios the duration of combustion is slightly longer compared to the respective one observed under normal diesel operation. At extremely low total relative air–fuel ratio, the increased gaseous fuel proportion combined with the improved cylinder charge conditions has a positive effect on the existence and spread of the flame front surrounding the burning zone, thus resulting to a considerable shorter duration of combustion as compared to the one observed with higher total relative air–fuel ratios (Fig. 2b and d).

4.3. Nitric oxide

The variation of the specific nitric oxide (NO) emission is provided in Fig. 4a–c. As known [36–38], the NO formation mechanism is predominantly controlled by the cylinder charge temperature and auxiliary by the local oxygen excess availability. Examining this figure, NO emission is affected considerably by the liquid fuel supplementary ratio. For the same combination of engine operating conditions (i.e. engine speed and load), the increase of the liquid fuel supplementary ratio ($x > 0\%$) leads to lower NO concentrations compared to the ones observed under normal diesel operation ($x = 0\%$).

At low and intermediate load conditions and for all engine speeds examined, the increase of the gaseous fuel concentration results to a decrease of the specific NO emissions. The effect is more pronounced at low load where the reduction of the specific NO emission is up to 60%. This is due to the fact that the extremely low charge temperature caused by the poor quality of the natural gas combustion during the premixed controlled combustion phase (Fig. 2a and c) weakens NO formation rate. At intermediate loads the increase of the gaseous fuel concentration leads to a milder decrease of NO emissions compared to the one observed at low load. In this case, the specific trend may be ascribed principally to the increased cylinder charge temperatures as a result of the increased energy released (i.e. heat release rate) inside the combustion chamber, especially during the diffusion controlled combustion phase (Fig. 2a and c). Thus, at bmep of 2.4 bar the maximum decrease of NO emission is up to 45% while at bmep of 3.7 bar the corresponding decrease is up to 30%.

At high load and low engine speed, the increase of liquid fuel supplementary ratio has a positive effect (i.e. reduction) on the specific NO emission as compared to the respective one observed under normal diesel operation. Thus, at specific engine operating conditions, the maximum decrease of NO emission is up to 47%. A possible explanation is, mainly, the lower rate of premixed controlled combustion of the gaseous fuel (Fig. 2b), which leads to lower charge temperature inside the combustion chamber compared to normal diesel operation.

At high load and high engine speed, specific NO emission starts to decrease with the increase of the diesel fuel supplementary ratio. A further increase of x, beyond a certain value, leads to a slight increase of the specific NO emissions. This is due to the fact that the increase of the engine speed leads to a warmer engine and to an increase of the turbulence inside the combustion chamber. These factors have a positive effect (enhance) on the flame speed, thus contributing to the improvement of gaseous fuel combustion quality, especially during the premixed controlled combustion phase (Fig. 2d), thus resulting to an enhancement of the NO formation mechanism.

4.4. Carbon monoxide

The variation of the specific carbon monoxide (CO) emission is provided in Fig. 5a–c. As known [36–38], the rate of CO formation is a function of the air–fuel ratio, the unburned gaseous fuel availability and also the cylinder charge temperature. The aforementioned parameters control also the rate of fuel decomposition and oxidation. Observing this figure, it is revealed that for the same operating point (i.e. engine speed and load) CO emissions under dual fuel operation are significantly higher compared to the respective ones under normal diesel operation. Thus, at each engine operating point (i.e. load and engine speed) the increase of liquid fuel supplementary ratio, which is accompanied with a reduction of the total relative air–fuel ratio, favors the CO formation mechanism.

At low and intermediate load conditions and for all engine speeds examined, the decrease of the total relative air–fuel ratio results to an increase of the specific CO emissions. The effect is more pronounced at low load where the increase of the specific CO emission is up to 600%. This is due to the fact that the extremely

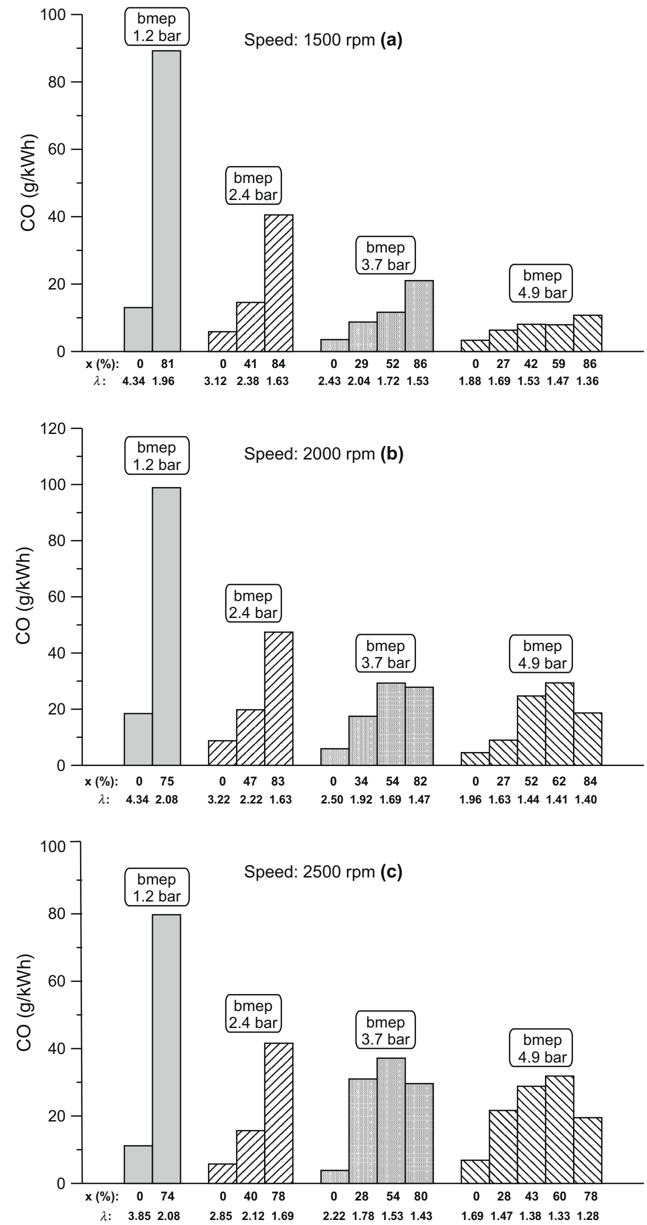

Fig. 5. Bar chart diagram of the emitted specific carbon monoxide (CO) at various loads (bmep) as function of the supplementary ratio 'x' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

poor quality of the gaseous fuel combustion contributes to a sharp increase of the unburned gaseous fuel availability, which combined with the low charge temperature and low oxygen concentration of the cylinder charge enhance the CO formation mechanism.

At intermediate loads the increase of CO emissions with total relative air–fuel reduction is smoother as compared with the respective one observed at low load conditions. This may be attributed mainly to the lower unburned gaseous fuel availability inside the cylinder charge owing to the improvement of quality of the gaseous fuel combustion as compared to the respective one observed at low load conditions. Thus, at bmep = 2.4 bar the maximum increase of CO emission is up to 270% while at bmep = 3.7 bar the corresponding increase is up to 190%.

At high load and low engine speeds the decrease of the total relative air–fuel ratio results to a slight increase of the specific CO emission as compared to the respective one observed under normal diesel operation. Thus, at specific engine operating conditions, the maximum increase of CO emission is up to 70%. Despite the decrease of the total relative air–fuel ratio, the improvement of the gaseous fuel combustion quality, especially during premixed controlled combustion phase, contributes to lower unburned gaseous fuel availability and higher cylinder charge temperature, thus contributing to lower formed CO emissions.

At high engine load and high engine speed, the increase of the gaseous fuel concentration causes a slight increase of the specific CO emissions, while for total relative air–fuel ratios beyond a certain value the emitted CO starts to decrease. A possible explanation is that at higher engine speed the less time available for combustion combined with the considerable improvement of the gaseous fuel combustion process affect adversely the CO formation mechanism.

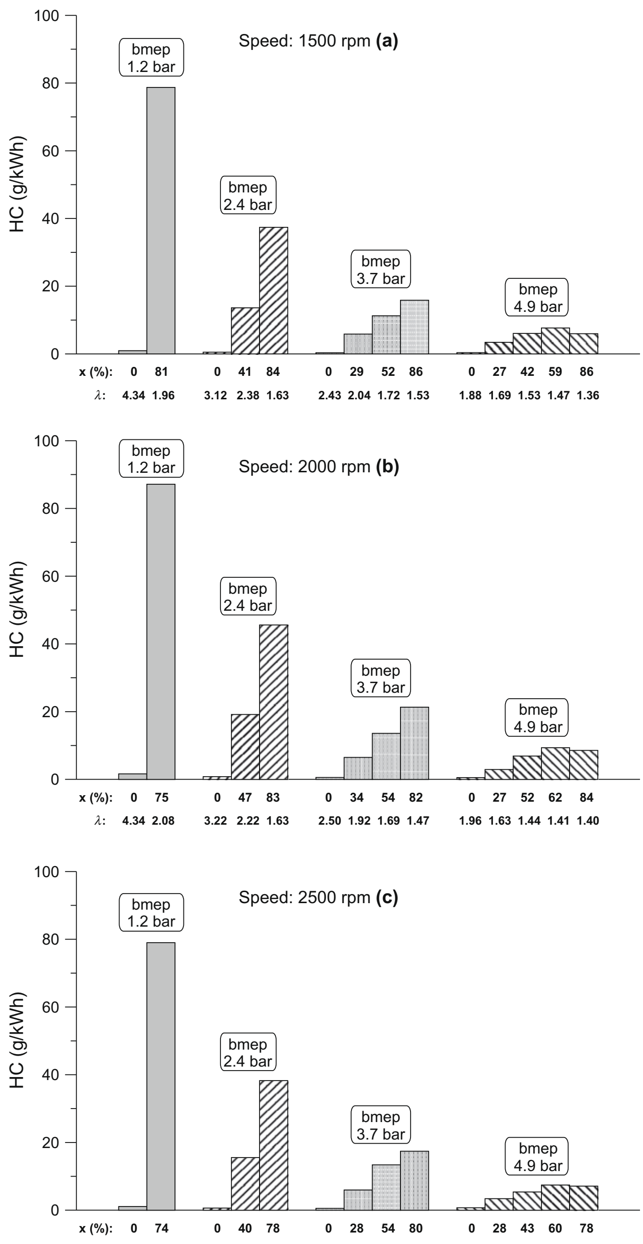

Fig. 6. Bar chart diagram of the emitted specific unburned hydrocarbons (HC) at various loads (bmep) as function of the supplementary ratio 'x' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

4.5. Unburned hydrocarbons

Fig. 6a–c illustrates the variation of the experimental values of the specific (total) unburned hydrocarbons (HC) emissions. As known, the variation of unburned hydrocarbons in the exhaust gases depends on the quality of the combustion process occurring inside the combustion chamber [36–38]. Moreover, under dual fuel operating mode, combustion process is affected considerably by the total relative air–fuel ratio since this specific factor plays a significant role on the flame propagation mechanism. Examining this figure, unburned HC emissions are affected considerably by the presence of the gaseous fuel inside the cylinder charge. At each combination of engine speed and load, the specific unburned HC emissions measured under dual fuel operation mode is higher compared to the respective one observed under normal diesel operation. The difference becomes more intense at low load conditions, thus revealing the considerable amount of gaseous fuel escaping the combustion process.

At intermediate load conditions, as the total relative air–fuel ratio decreases the increase of the unburned HC emissions observed is smoother compared to the respective trend observed at low load conditions. This is due to the improvement of the gaseous fuel combustion process, since the total relative air–fuel ratio tends to stoichiometric values, thus favoring the flame propagation mechanism, which plays also a significant role on the unburned HC oxidation rate.

At high engine load conditions, the decrease of the total relative air–fuel ratio leads to a slight increase of the unburned HC emissions. This is due to the lower amount of natural gas escaping the combustion process, which is caused by the considerable improvement of the gaseous fuel combustion process.

4.6. Soot

The variation of the experimental values of soot (smoke) density is provided in Fig. 7a–c. Examining this figure, it is observed that dual fuel operation is a potential way of reducing soot emissions. Specifically, under dual fuel operating mode and for all engine speeds examined, it is revealed that at low load as the percentage of the liquid diesel fuel substitution increases soot density decreases, since less liquid fuel is injected on a percentage basis and so less soot is formed.

At high and intermediated load conditions and high total relative air–fuel ratios a slight reduction of soot emission occurs. This is due to the fact that, despite the improvement of the gaseous fuel combustion rate, the concentration of the soot formed is higher than the one under normal diesel operation due to the lower cylinder charge temperatures observed during the premixed controlled combustion phase. At extremely low total relative air–fuel ratios

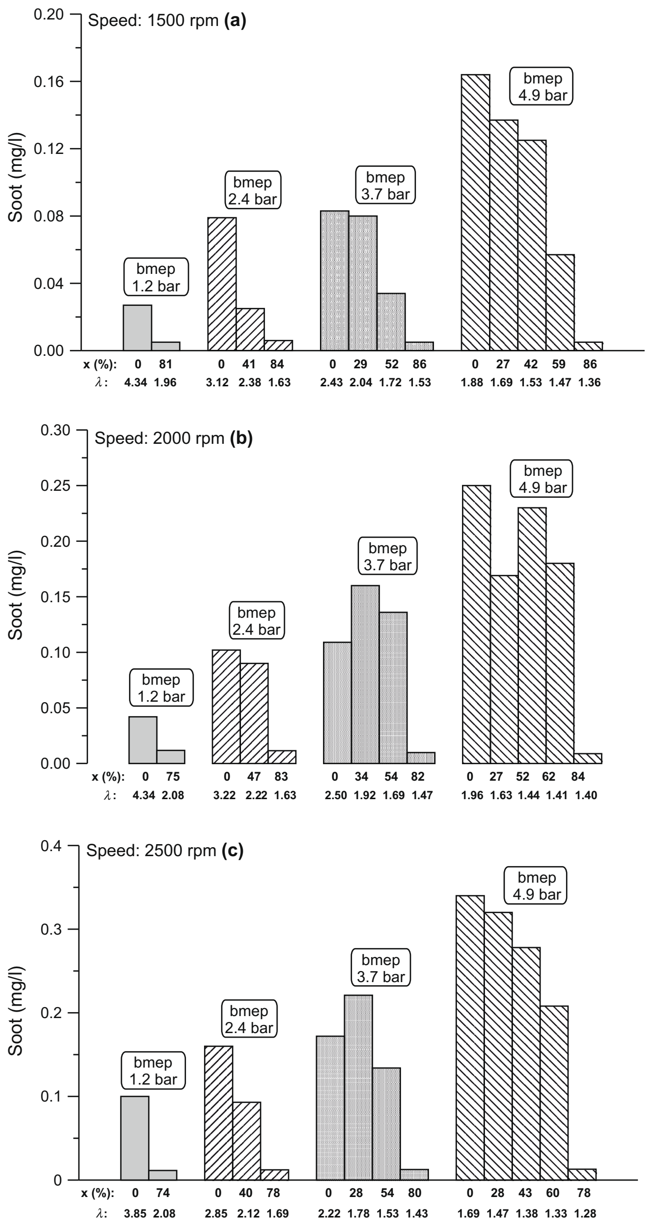

Fig. 7. Bar chart diagram of the emitted soot (smoke) density at various loads (bmep) as function of the supplementary ratio 'x' and corresponding total relative air–fuel ratio 'λ', for the engine speeds of 1500 rpm (a), 2000 rpm (b), and 2500 rpm (c).

the reduction of soot emission becomes more evident compared to the one observed at higher ratios, since the improvement of the gaseous fuel combustion rate results in higher cylinder charge temperatures during the expansion stroke, which, eventually, promotes soot oxidation rate and so contributing to a further decrease of soot concentration. Therefore, at high load and for all engine speeds examined the engine operation under dual fuel mode with extremely low total relative air–fuel ratio leads to a considerable decrease of soot emissions compared to the normal diesel operation.

It is apparent that dual fuel operation mode, using natural gas, is a very effective method to reduce soot emissions at almost all engine operating points. The main reason behind is that natural gas, having methane as the main constituent that is the lower member in the paraffin family, possesses very small tendency to produce soot. Thus, practically, the gaseous fuel produces no soot, while it contributes to the oxidation of the one formed by the combustion of the liquid fuel. In that sense, the increase of ignition delay period observed under dual fuel operation results in a decrease of the heat released during the premixed controlled combustion phase, which, as it is well known [36,37], has a positive (i.e. reduction) effect on soot formation and, at the same time, does not seem to have an adverse (increasing) effect on nitric oxide emissions as it happens in normal diesel engines. Therefore, the use of natural gas has a positive effect on soot emissions (drastic reduction), and at the same time it does not have an adverse effect (increase) on nitric oxide emissions as occurs in normal diesel engines; actually, it seems to have a positive effect by reducing its value, a fact that is very important.

5. Summary and conclusions

An extended experimental investigation has been conducted on a high speed, direct injection, single cylinder, test engine for the purpose of exploring the effect of the total air–fuel ratio on engine efficiency and the emitted pollutants from a compression ignition engine operating under dual fuel mode. The engine has been properly modified to operate under dual fuel mode, without changing its main configuration. Measurements have been taken at various combinations of engine loads and speeds, under both normal diesel and dual fuel operating modes.

From the analysis of the experimental data, it is shown that the decrease of the total relative air–fuel ratio, which is caused by the increase of diesel fuel supplementary ratio, results to a lower brake thermal efficiency compared to the one under normal diesel operation, thus revealing the deterioration of the engine efficiency under diesel–natural gas dual fuel operating mode. The specific deterioration becomes more evident at low and intermediate loads, while at high load and high diesel supplementary ratios the engine efficiency is improved.

Furthermore, for all combinations of engine operating points (i.e. engine speeds and loads), the increase of diesel fuel supplementary ratio results in lower nitric oxide emissions as compared to the respective ones observed under normal diesel operation. At high load and high supplementary ratios, the beneficial effect becomes more evident compared to the one observed at part load conditions.

Dual fuel operation leads to a significant decrease of soot emissions. The positive effect is stronger at high engine load and for relative air–fuel ratios close to stoichiometry.

The increase of diesel fuel supplementary ratio affects adversely (i.e. increase) the concentration of the carbon monoxide emissions. This effect becomes more evident at low and intermediate loads in comparison to the one observed at high load, since at high engine load and low total relative air–fuel ratios a slight decrease of the emitted carbon monoxide concentration is observed. Furthermore, dual fuel operation results also in higher unburned hydrocarbon emissions compared to the ones observed under normal diesel operation. At part load this specific difference is stronger, while at high engine load and low air–fuel ratios a slight decrease is observed.

Taking into account all the findings of the experimental investigation, it is revealed that dual fuel combustion using natural gas as a supplement for liquid diesel fuel is a promising technique for controlling both NO and Soot emissions on existing DI diesel engines, requiring only slight modifications of the engine structure. This is extremely important if one considers the difficulties of controlling both the main pollutants, i.e. NO and Soot, in conventional DI diesel engines. The observed disadvantages concerning brake efficiency, HC and CO can be possibly mitigated by applying modifications on the engine tuning, e.g. injection timing of liquid diesel fuel mainly at part loads.

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