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Towards improvement of natural gas-diesel dual fuel mode: An experimental investigation on performance and exhaust emissions

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ABSTRACT

The use of natural gas in compression ignition engines as supplement to liquid diesel in a dual fuel combustion mode is a promising technique. In this study, the effect of DF (dual fuel) operating mode on combustion characteristics, engine performances and pollutants emissions of an existing diesel engine using natural gas as primary fuel and neat diesel as pilot fuel, has been examined. At moderate and relatively high loads, the results show very interesting behavior of dual fuel operating mode in comparison to conventional diesel, both for engine performance and emissions. It showed a simultaneous reduction of soot and NO_x species over a large engine operating area. Moreover, it showed the possibility to obtain lower BSFC (brake specific fuel consumption) than conventional diesel engine. However, this mode presents some deficits at low loads, especially concerning unburned hydrocarbons and carbon monoxide emissions. Understanding those deficiencies is a key of such engines improvement. Some suggestions for new measures towards DF mode improvement are deduced.

1. Introduction

NG (Natural gas) is one of the most promising available fuels for internal combustion engines. Few alternative fuels offer the distinct and attractive advantages of natural gas [1]. Recently, environmental and economical concerns have motivated many governments to consider natural gas as fuel for passenger vehicles as well as stationary engines [2]. A promising technique for its use in internal combustion engines is the dual fuel concept [3].

In dual fuel engines, the gaseous fuel is inducted along with the intake air and is compressed like in a conventional diesel engine. The air-gaseous fuel mixture does not autoignite due to its high autoignition temperature limit [4]. A small amount of neat diesel or biofuel [5,6] is injected near the end of the compression stroke to ignite the gaseous mixture. Diesel fuel autoignites and creates ignition sources for the surrounding air–gaseous fuel mixture. The pilot liquid fuel, which is injected by the conventional diesel injection system, contributes only with a small fraction in the engine power output [7].

The dual fuel engine has been employed in various applications with different gaseous fuels due to their cleaner combustion compared to conventional liquid fuels [8,9]. However, natural gas seems to be an excellent candidate because of its worldwide usage. It has a high octane number, and therefore, it is suitable for engines with relatively high compression ratios. It mixes uniformly with air, resulting in efficient combustion and a substantial depletion of some emissions in the exhaust gas [3]. Moreover, it is possible to apply this technology on existing diesel engines with minor modifications. The potential benefits of using natural gas in diesel engines are both economical and environmental.

However, to be more attractive, some aspects must be improved for best performance and less emissions [10]. One of the main problems with dual fuel operating mode is that, at low load, the engine efficiency decreases compared to conventional diesel. The unburned hydrocarbons and carbon monoxide emissions are also higher in dual fuel mode [4,11,12].

Some experimental and theoretical investigations concerning the dual fuel operating mode using natural gas have been reported in specialized literature [4,13,14]. The effect of some engine parameters, such as injection timing [15,16], pilot diesel fuel amount [17], gaseous fuel – air mixing system [18], air inlet preheating and EGR (Exhaust Gas Recirculation) [19] on the engine performance were also examined. It was found that improvements in engine

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performance and reduction in some emissions can be achieved by optimization of some parameters such as advancing the injection timing or increasing the amount of pilot fuel. EGR with intake heating can also be a promising solution for improving engine efficiency and reducing CO, THC (total hydrocarbon) and NO_x emissions.

The present study aims to examine the effect of dual fuel mode on combustion characteristics, engine performance and exhaust emissions. Experimental investigations are used to achieve this work and the obtained results aim first, to identify the engine operating conditions which are able to provide mechanical and environmental benefits within this mode.

A conventional DI (direct injection) diesel engine has been properly modified to operate in dual fuel mode. The engine is first operating in conventional mode with neat diesel fuel, then in dual fuel mode in order to achieve a real comparison. The obtained results are compared with previous studies to better understand such operating mode (dual fuel) and related problems, so that to make conclusions and recommendations which can aid for improving the natural gas dual fuel operating mode.

2. Experimental setup and experimental procedure

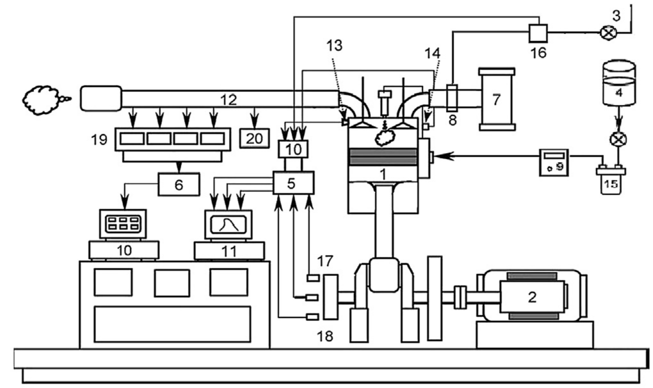
2.1. Engine test cell

A single cylinder air cooled Lister Petter (TS1) diesel engine with output power of 4.5 kW at 1500 rpm is used to carry out engine tests. The basic data of this engine are given in Table 1. The experimental set up scheme is shown in Fig. 1. The engine is connected to an automatic controlled eddy current dynamometer. An orifice meter connected to a large tank is attached to the engine for air flow measurements. The diesel fuel flow rate is measured with a Coriolis type flow meter. Chromel alumel thermocouple in conjunction with a slow speed digital data acquisition system is used for measuring the exhaust gas temperature.

To carry out dual fuel engine experiments, the engine was conveniently modified. It is supplied with natural gas from the local distribution network. Natural gas flow rate is measured using a thermal flow meter (Bronkhorst F112AC-HB-55V). Adjustment of gaseous fuel supply is accomplished through a control valve. Then, the gaseous fuel is mixed with the air in the engine intake manifold. The composition of natural gas used in the present tests is given in Table 3.

2.2. Combustion data acquisition

A rapid digital data acquisition system (AVL – Indiwin) in conjunction with two AVL piezoelectric transducers is used to get the in-cylinder and diesel fuel injection pressures. The data from 100 consecutive cycles at an increment of 0.1 crank angle, are recorded and averaged, with a specific AVL software, to obtain combustion parameters [11,20,21]. An optical shaft position encoder AVL 364C is used to determine the angular position and the engine rotational speed.



1. Test engine
2. Dynamometer
3. Gaseous fuel supply
4. Gasoil tank
5. A/D card for pressure
6. A/D card for Analyzer
7. Air tank
8. Air-gas Mixer
9. Diesel flow meter
10. Charge Amplifier
11. Fast data acquisition system
12. Slow data acquisition system
13. Cylinder pressure sensor
14. Injection pressure sensor
15. Liquid fuel filter
16. Gaseous fuel flow meter
17. TDC encoder
18. Speed sensor
19. Exhaust gas analyzer
20. Smoke meter

Fig. 1. Experimental setup scheme.

2.3. Emission instrumentation

Unburned HC (hydrocarbon) species are captured using the FID technique from sampling exhaust gas. The NO and NO_x concentrations in the exhaust gas are measured with the chemiluminescence technique using a TOPAZE 32M analyzer. Carbon monoxide (CO) and carbon dioxide (CO_2) concentrations are measured by infrared radiation absorption using a MIR 2M analyzer. Oxygen (O_2) concentration is given by a paramagnetic sensor. Smoke levels are observed continuously using the smoke meter (TEOM 1105).

Table 1
Engine specifications.

Constructor	LISTER-PETER (TS1)
Engine type	4 strokes, compression ignition, diesel direct injection (DI)
Number of cylinders	Single cylinder
Bore × stroke	95.5 × 88.94 mm
Volumetric capacity	630 cm^3
Compression ratio	18
Injection	13° CA before TDC
Injection pressure	250 bars
Power output	4.5 kW at 1500 rpm
IVO	36° CA before TDC
IVC	69° CA after BDC
EVO	76° CA before BDC
EVC	32° CA after TDC

Table 2
Accuracy of measuring instruments and uncertainty of computed parameters.

Parameter	Measuring instruments	Accuracy
Torque	Effort sensor	±0.1 Nm
Engine speed	AVL 364C	±3 rpm
Injection timing	AVL 364C	±0.05° CA
Intake air flow rate	Differential pressure transmitter (LPX5841)	±1% of measured value
Fuel flow rate	Coriolis type mass flow meter (RHM015)	±0.5% of measured value
In cylinder pressure	Piezo-electric (AVL QH32D)	±2 bars
Injection pressure	Piezo-electric (AVL QH33D)	±2 bars
Intake air temperature	Differential pressure transmitter (LPX5841)	±1.6 K
Exhaust gas temperature	K type thermocouple	±1.6 K
Ambient air temperature	HD 2012 TC/150	±0.2 K
Relative humidity	HD 2012 TC/150	±2%
$CO/CO_2/O_2$	MIR 2M	±2% Full scale
NO_x	TOPAZE 32M	±2% Full scale
THC	GRAPHITE 52M	±2% Full scale
Soot emissions	TEOM 1105	±30 ng/s
Computed parameters		Uncertainty
Brake power	–	±1.9%
BSFC	–	±2%
Fuel air Eq. ratio	–	±1.1%

Table 3
Natural gas composition and heating value.

Component	v/v (%)
Methane (CH_4)	90.4 ± 1.7
Ethane (C_2H_6)	6.7 ± 1.6
Propane (C_3H_8)	1.8 ± 0.8
Butane (C_4H_{10})	0.7 ± 0.5
Pentane (C_5H_{12})	0.05 ± 0.03
Inerts ($N_2 + CO_2$)	0.35 ± 0.3
Net heating value (MJ/kg): 49.5 ± 0.2	

2.4. Experimental procedure

Experiments are initially carried out on the conventional diesel engine using neat diesel fuel. The injection timing is set at 13° before TDC (top dead center) for all experiments. The engine is stabilized for each measurement. All thermodynamic parameters (pressures, temperatures, flow rates) of air and fuel flows and mechanical parameters (rotational speed, torque) are measured to get engine performance. Exhaust gas analyzers are calibrated before measurements. Observations are performed for soot, NO_x, HC, CO and CO_2 to analyze the emission characteristics.

With dual fuel operating mode at rated engine speed, pilot diesel fuel is injected to cover approximately 10% of the maximum power output. Then, keeping constant the flow rate of liquid diesel fuel, the engine power output is increased by augmenting the natural gas flow rate. This procedure is followed until the desired power output is obtained. In order to estimate the NG contribution to power output in dual fuel mode, a mass participation rate (Z) is commonly adopted [7]. This parameter can be determined as:

$$Z = \frac{\dot{m}_{NG}}{\dot{m}_D + \dot{m}_{NG}} \cdot 100 \tag{1}$$

where $\dot{m}_D$ (kg/s) is the pilot diesel fuel mass flow rate and $\dot{m}_{NG}$ (kg/s) the natural gas mass flow rate in dual fuel operating mode.

In both operating modes, measurements are carried out for four engine speeds: 1500, 1800, 2000 and 2200 rpm and four applied loads corresponding to 20%, 40%, 60% and 80% of full load for each engine speed. Compared measurements were carried out in almost same ambient conditions. Indeed, first, experiments are done in hours when there is no important gradient of temperature. Moreover, measurements on dual fuel and diesel mode used as baseline are carried out in short time; so that ambient conditions are practically constant.

2.5. Measured data uncertainty analysis

All measurements of physical quantities have some degree of uncertainty, owing to different sources, namely the employed instrumentation, its calibration, observation accuracy and the methodology of experimentation [11]. Therefore, to confirm the repeatability of the experimentation, uncertainty analysis is necessary.

The computed parameters uncertainty is determined according to the principle of root-mean square method [22], on the basis of the corresponding engine components and emission analysis instrumentation accuracy, as declared by their respective manufacturers, as follows:

$$e_R = \left[\left(\frac{\partial f}{\partial x_1}\ e_1\right)^2 + \left(\frac{\partial f}{\partial x_2}\ e_2\right)^2 + \cdots + \left(\frac{\partial f}{\partial x_n}\ e_n\right)^2\right]^{\frac{1}{2}} \tag{2}$$

where e_R is the uncertainty in the computed result R, f is a given function of the computed result. $x_1, x_2, \ldots, x_n$ are the independent measured variables (i.e., $R = f(x_1, x_2, \ldots, x_n)$), $e_1, e_2, \ldots, e_n$ are the corresponding uncertainty values of the independent measured variables.

Table 2 summarizes the accuracy of the measurements along with uncertainty of the computed parameters, based on the instruments' specifications and experimental error analysis.

To increase the reliability of the measured data, all the instruments used are tested and calibrated before experiments. Special emphasis is given to the exhaust gas emissions measurements. All gas analyzers are purged after each measurement, and then calibrated before the next measurement. Moreover, the experiments have been conducted such that ten measurements of each parameter (except for in-cylinder pressure, as detailed in Section 2.2) have been recorded; for each operating point. The quantities reported for all measured parameters, which are then used for further computations, are the arithmetic mean values of the ten measurements [23].

3. Experimental heat release analysis

Heat release analysis is a valuable and largely adopted tool for combustion study in engines. It includes calculations which yield how much heat would have to be added to the cylinder contents in order to produce the observed variations in measured cylinder pressure. Using the average measured pressure diagram and the TDC signal, the rate of heat release is estimated [24].

Considering the space trapped in the cylinder when the valves are closed as a control volume, the net heat release rate is determined by applying the first thermodynamic law using the following expression [25]:

$$\delta Q_{net} = \delta Q_{gross} - \delta Q_w = dU + \delta W \tag{3}$$

where δQ_{gross} is the heat release by combustion, δW is mechanical work done by the system, δQ_w is heat lost to combustion chamber walls and δQ_{net} is the net heat release which is the difference between the gross heat release and the heat lost to the walls.

Considering the cylinder content as an ideal gas (such that $c_p/c_v = \gamma$ and $c_p - c_v = R$), the Equation (3) can finally be modified as:

$$\frac{dQ_{net}}{d\theta} = \frac{dQ_{gross}}{d\theta} - \frac{dQ_w}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{4}$$

where θ is the CA (crank angle), p is the in-cylinder pressure at a given crank angle, V is the cylinder volume at that point, and γ is the specific heat ratio (c_p/c_v).

This type of heat release model is referred to in the literature as zero-dimensional model. The following assumptions were used in this determination: the mixture inside the combustion chamber behaves as an ideal gas, the gaseous contents within the combustion chamber are homogeneous, and the system shall be considered a closed system with constant mass throughout the cycle (neglecting varying mass due to crevice flow, blow-by, and fuel injection) [26]. Therefore, it should be noted that the calculation of the net heat release rate is only valid between the IVC (inlet valve closure) and the EVO (exhaust valve opening) as a constant mass is considered [24].

The value of γ varies with the variation of the gas temperature inside the cylinder, [23]. It ranges from 1.3 to 1.35 for diesel heat release analysis. However, the appropriate values of γ for the most accurate heat release analysis are not well defined [27]. In the heat release analysis, the specific heat ratio, γ, being a function of temperature has an influence on the magnitude of Q_{net} and $dQ_{net}/d\theta$. However, according to Ma et al. [28], the effect of changes in γ, within a typical range, on the phasing of Q_{net} and $dQ_{net}/d\theta$ is negligible. Hence as suggested by Mustafi et al. [24], a constant value (1.35) of specific heat ratio γ, is retained.

The heat release rate calculated in this study was the NHRR (net heat release rate), which can be used as an indicator of actual heat release for small engines [27]. For each operating point examined, the net HRR (heat released rate) is calculated using the mean cylinder pressure trace, for which data from 100 consecutive cycles, are recorded and averaged.

The mean cylinder gas temperature is obtained using the following expression:

$$T_g = \frac{p\,V}{m\,R} \tag{5}$$

where, the specific gas constant R is calculated from the mean gas composition estimated from the initially trapped mass and the amount of fuel burned to the current crank angle. The trapped mass at inlet valve closure is estimated from an open cycle simulation using the measured mass flow rate of air and gaseous fuel [25].

4. Results and discussions

In this section, experimental results concerning the effect of dual fuel operating mode on engine performance and pollutant emissions are presented and discussed.

4.1. Effect of dual fuel operating mode on heat released and in-cylinder pressure

Results are provided for two engine speeds: 1500 rpm and 2000 rpm. For each one, four applied loads corresponding to 20%, 40%, 60% and 80% of full load are considered (Figs. 2 and 3). Total (gaseous and liquid fuel) net heat released and in-cylinder pressure are used to investigate the effect of dual fuel operating mode on the combustion characteristics.

At low engine loads, in-cylinder pressure corresponding to dual fuel operating mode is slightly lower comparatively to the conventional diesel engine case. The lower cylinder pressures observed under dual fuel operation during the compression stroke are the consequence of the higher specific heat capacity of the NG-air mixtures. At first period of combustion, a similar trend is observed due to the slower combustion rate of the gaseous fuel compared to neat diesel fuel, and because of a later ignition [6,29]. However, at high engine speeds, the in-cylinder mixture is well prepared and the engine is warmer, resulting on faster flame speed. The difference is less significant (Fig. 3-b1, b2). Concerning the total heat released rate corresponding to dual fuel operating mode, it is slightly higher at low loads, compared to the case of conventional diesel during the final period of combustion (Figs. 2 and 3-a1, a2), revealing late combustion of the gaseous fuel. Thus, the effect on the in-cylinder pressure is small since it occurs in the expansion stroke (Figs. 2 and 3-b1, b2).

At high engine loads (Figs. 2 and 3), unlike the results presented in the reference [7], the present work shows that the in-cylinder pressure with dual fuel operating mode becomes higher than the conventional diesel engine case notably at the rapid premixed combustion period. This is the consequence of a higher heat released under dual fuel mode compared to conventional diesel at that period; revealing an improvement in the gaseous fuel combustion. This is also confirmed by the BSFC trend as shown below. This observation is more evident at high engine speed, where the in-cylinder mixture is well prepared and the engine is warmer as mentioned before. However, during the compression stroke and the initial periods of combustion, the cylinder pressure is slight lower for dual fuel mode.

Concerning the in-cylinder pressure peak, it is noticed from Fig. 4, that the maximum in-cylinder pressure of conventional diesel is higher at low loads, than the dual-fuel mode. This tendency is reversed and the difference becomes increasingly significant at higher loads. Hence, the in-cylinder pressure peak becomes higher for dual-fuel mode compared to conventional diesel as a consequence of the above-mentioned improvement of the gaseous fuel combustion. Selim [30] also affirms that the maximum pressure is higher for dual fuel mode than the conventional diesel case.

4.2. Effect of dual fuel mode on the total brake specific fuel consumption

The evolution of the total BSFC according to load has been investigated for several engine speeds both for conventional diesel and dual fuel mode respectively. Fig. 5 shows the results concerning four engine speeds, 1500, 1800, 2000 and 2200 rpm, respectively. The total BSFC is directly calculated from the engine brake power output and the fuel mass flow rate. Thus, no consideration of the difference in LHV (lower heating value) between natural gas and neat diesel is taken into account.

At low engine loads, the total BSFC for dual fuel mode is higher than the conventional one. This observation is confirmed by the results reported by several authors [7,15]. It is especially remarkable at engine speeds of 1500 rpm and 2000 rpm. This reveals a poor utilization of the gaseous fuel, which is confirmed by the THC emissions (Fig. 10), and CO emissions (Fig. 12). It should be noted that when we consider the fact that for dual fuel engines, the load is increased by increasing the gaseous fuel flow rate; it appears that the THC emissions at low load, are relatively high even thought the absolute values are low.

This poor utilization of the gaseous fuel is due essentially to the combination of low temperature (at low loads) and very poor air-natural gas mixture inside the combustion chamber (Fig. 6), resulting in a bad and slow combustion rate of the gaseous fuel.

However, at high and moderate loads (over 50%), the results show that the total BSFC is lower for all the tested engine speeds. At those loads, the improvement of the gaseous fuel utilization due to higher temperatures and richer mixtures, leads to a relevant improvement of the total BSFC with dual fuel mode. Moreover, since the heating value of natural gas is higher than the neat diesel one, the total BSFC is lower for dual fuel mode for this range of loads (i.e., moderate to high loads).

4.3. Effect of dual fuel mode on soot emissions

The evolution of soot emissions according to engine load, were measured for 1500, 1800, 2000 and 2200 rpm engine speeds both for the two engine modes. Fig. 7 presents results for two engine speeds (1500 and 2000 rpm) since the behaviour of soot emissions is quite similar. The results show clearly that dual fuel mode is a very efficient technique to reduce soot emissions especially at high loads where they are important for conventional diesel case.

In fact, for all the examined cases, soot emissions of dual fuel mode are considerably lower compared to those corresponding of conventional diesel particularly for high loads. While, with neat diesel fuel, increasing load has a big influence on soot emissions, they are negligible for dual fuel mode at all loads. Accordingly, the use of natural gas allows a drastic reduction of soot emissions. This is to be expected since natural gas, where methane represents the main component, has very small tendency to produce soot [2].

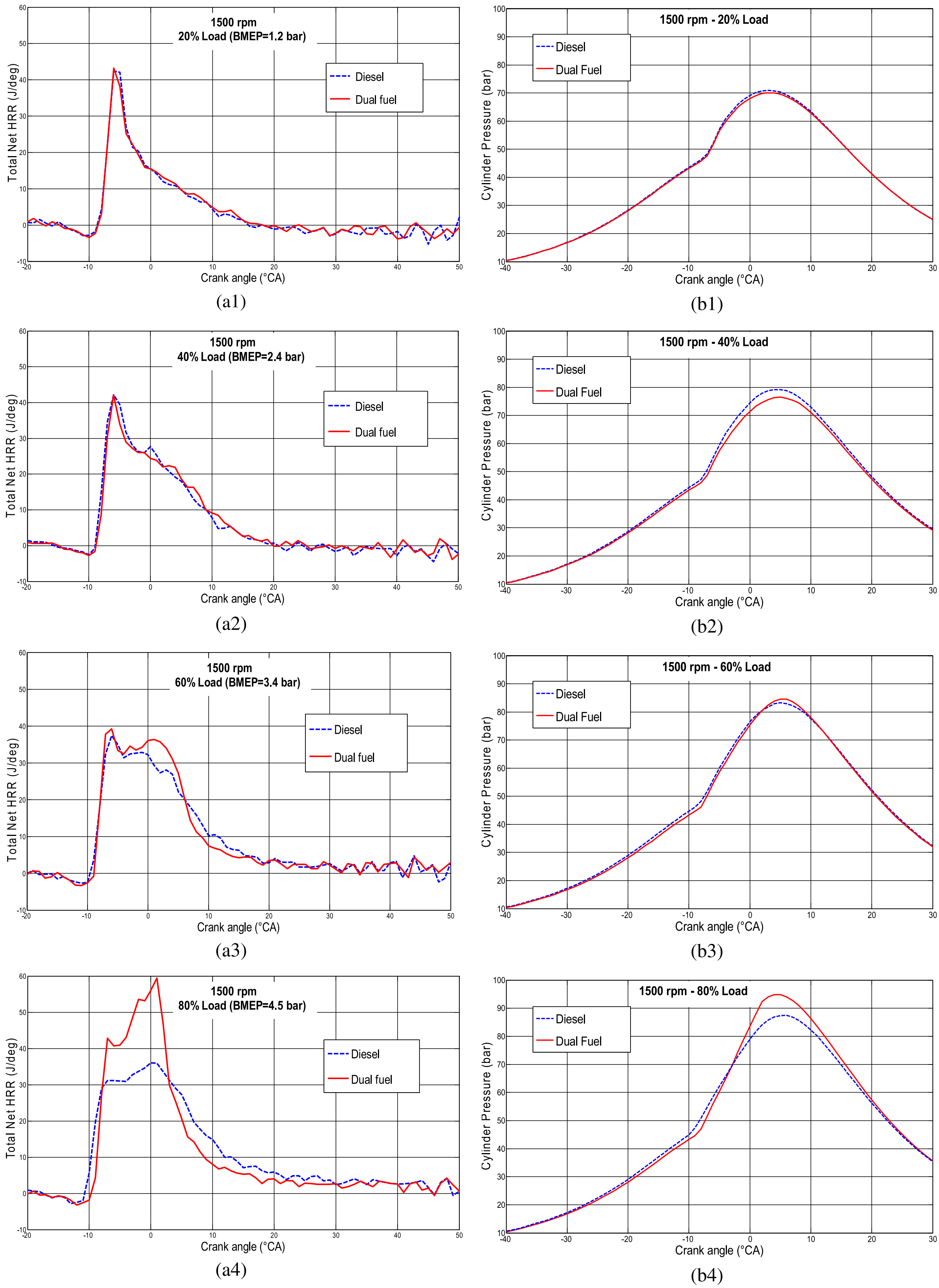


Fig. 2. Total heat released rate and in-cylinder pressure at 1500 rpm engine speed.

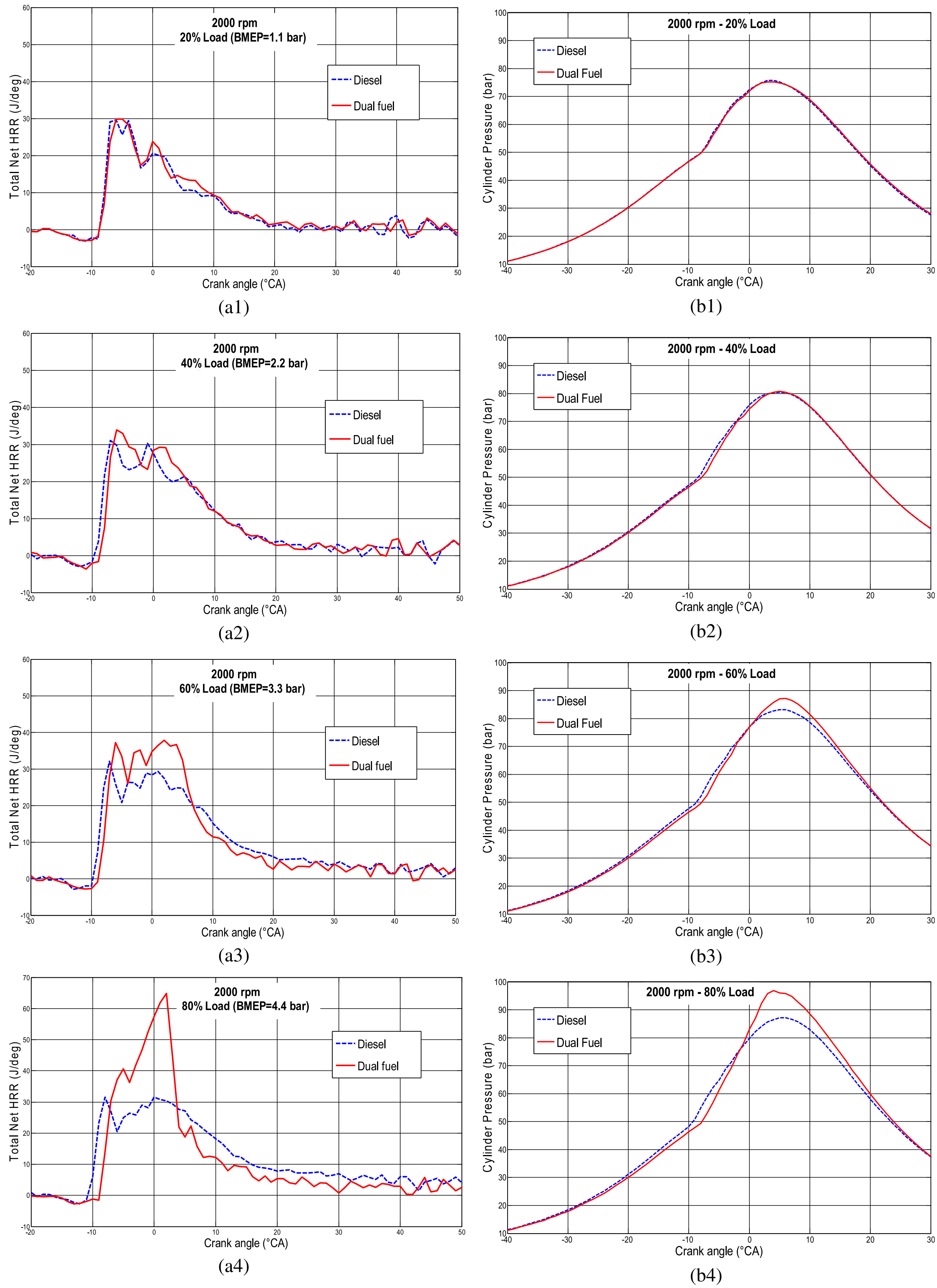


Fig. 3. Total heat release rates and in-cylinder pressure at 2000 rpm engine speed.

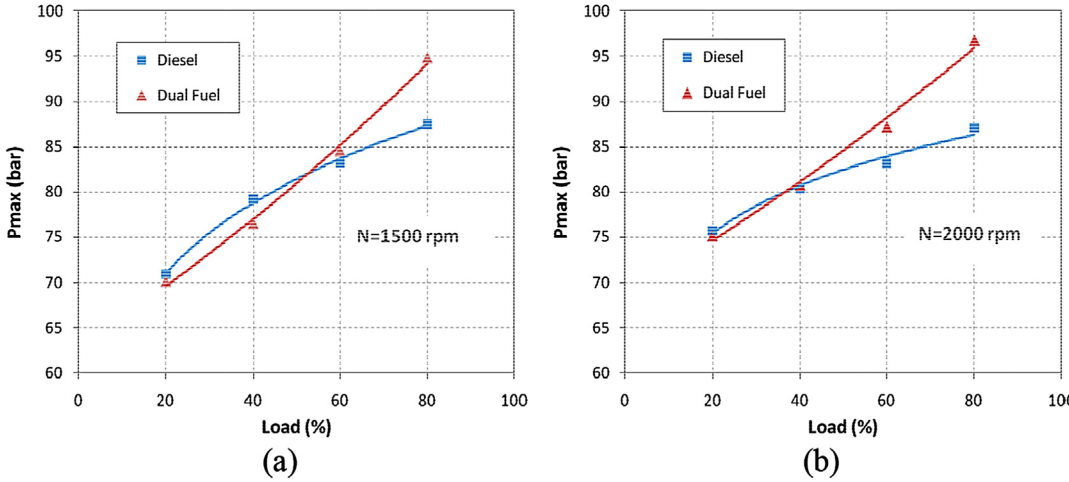


Fig. 4. Comparison of the maximum in-cylinder pressure for various loads at (a) $N = 1500$ rpm (b) $N = 2000$ rpm.

4.4. Effect of dual fuel mode on NO_x emissions

Fig. 8 provides the variation of NO_x emissions for conventional diesel and dual fuel mode, according to engine load, at 1500, 1800, 2000 and 2200 rpm engine speed.

At low and moderate loads, NO_x concentration of dual fuel mode is lower in comparison to the conventional diesel one. As widely recognized, the formation of thermal NO_x is mainly favored by two parameters: high oxygen concentration and high charge temperature [31]. For these loads, the charge temperature for both modes are almost equal (Fig. 9a). On the other hand, the higher oxygen concentration of conventional diesel (Fig. 9b) leads to more significant level of NO_x emissions.

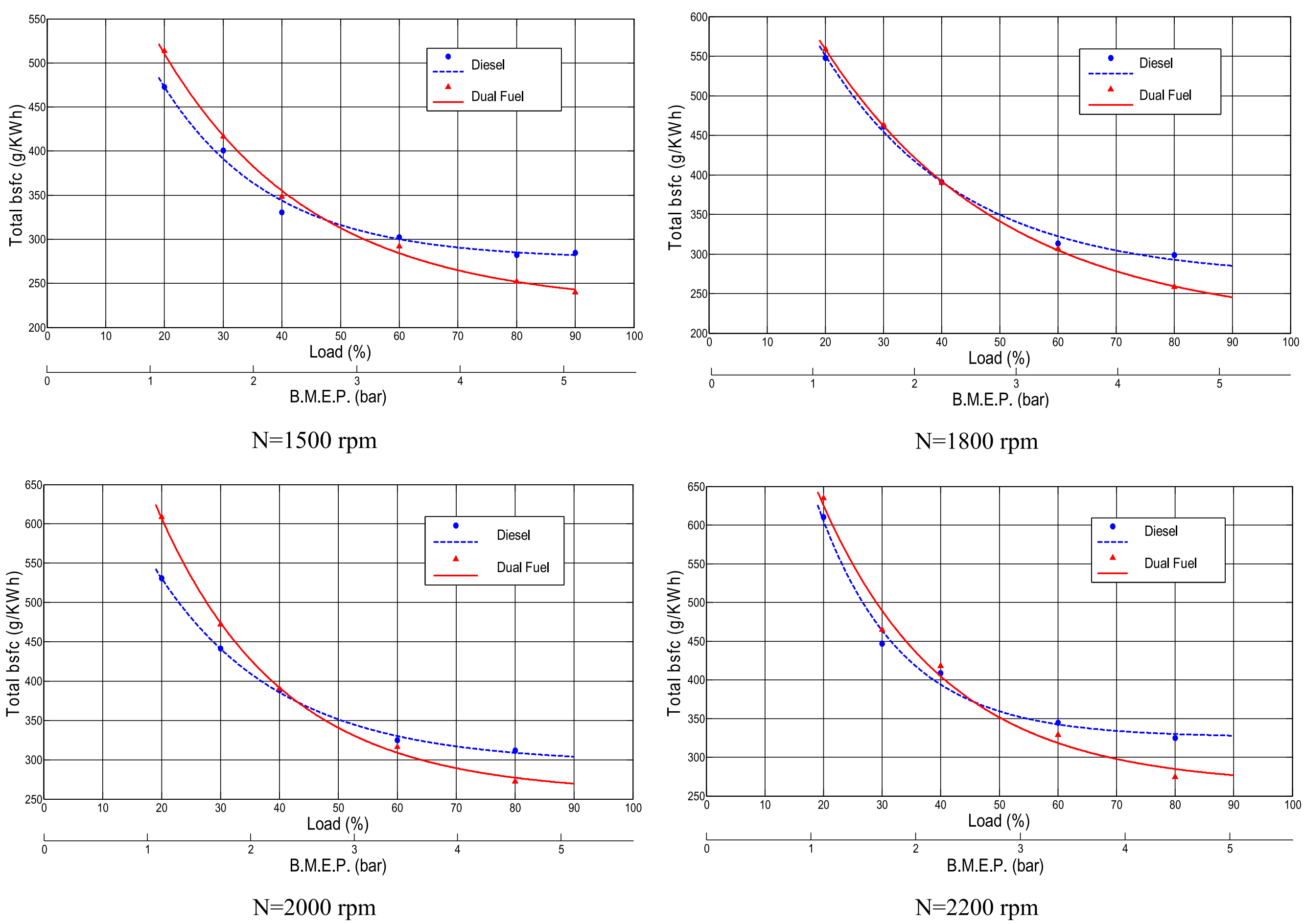


Fig. 5. Total BSFC according to load at different engine speeds.

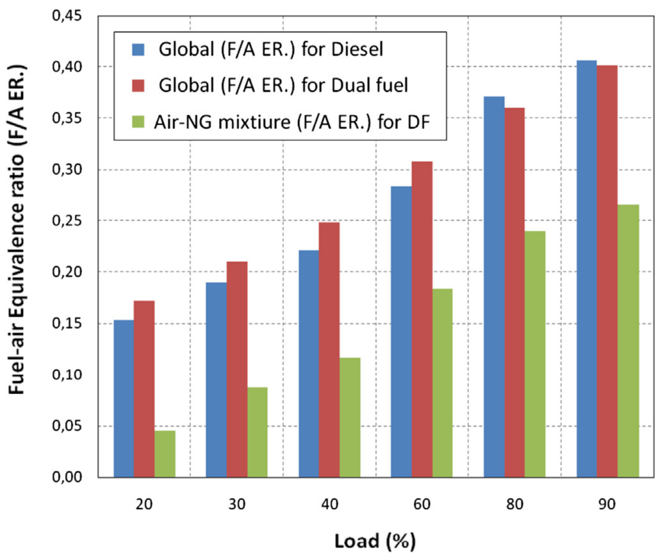


Fig. 6. Fuel air equivalence ratio at various loads for both dual fuel mode and conventional diesel.

However, at higher loads, our results show that the NO_x concentration of dual fuel mode becomes higher than the conventional diesel one at all engine speeds. This is due first, to a higher charge temperature for dual fuel mode. Indeed, Fig. 9a which depicts maximum cylinder gas temperature reveals a higher charge temperature for this mode. This higher temperature is due to a more important heat release in the early premixed combustion stage (Figs. 2 and 3), which is the result of the gaseous fuel combustion improvement at those loads, as explained in Section 4.1. Moreover, Fig. 9b shows that the oxygen concentration in exhaust gas, which can give an idea about the available oxygen in the charge, is higher for dual fuel mode than for conventional diesel at high loads. This higher concentration is particularly the result of a better combustion and thus of a lower consumption (Fig. 5) and less CO emissions (Fig. 12), but also of less quantities of oxygen combined with carbon to form CO_2 (Fig. 13). When the engine speed is increased, this change of trend appears earlier (lower charges) and the difference is more important (Fig. 8).

4.5. Effect of dual operation on unburned hydrocarbon emissions

The results for unburned hydrocarbon emissions, as a function of engine load, for the two modes are given in Fig. 10 at 1500, 1800, 2000 and 2200 rpm engine speeds. For all the tested engine speeds, the trend of total hydrocarbon (THC) emissions is similar. At any engine load, THC emissions for dual fuel mode are considerably higher in comparison to those corresponding to conventional diesel case. At low load, as mentioned before, the low temperature level and the high air-fuel ratio of gaseous fuel mixture induce a bad and slow combustion. Hence, important quantities of methane do not participate to combustion. However, at lower loads, even if the combustion conditions are not favorable, because gaseous fuel participation is small (Fig. 11), the THC emissions are lower. When the load increases, at the first stage, the emissions still increase since the gaseous fuel participation increases but the combustion quality is not sufficient to lower the THC emissions. For high loads, higher charge temperature level and richer gaseous fuel result in a further improvement in the combustion process, and as a consequence a decrease in the unburned hydrocarbon emissions occurs. Similar trend is provided by Abd Allah et al. [15].

4.6. Effect of dual fuel mode on carbon monoxide emissions

Fig. 12 illustrates the variation of carbon monoxide emissions for the two engines modes, as a function of engine load. Only results for 1500 and 2000 rpm engine speeds are reported.

As known, the rate of CO formation depends on the unburned gaseous fuel availability and the mixture temperature [29]. As shown in Fig. 12, CO emissions are higher for dual fuel mode at low and moderate loads. However, CO concentration for dual fuel mode decreases with the increase of engine load as a result of the improvement of gaseous fuel utilization. On the other hand, for very high loads, because of locally very rich mixtures in conventional diesel, which result in bad combustion, the CO emissions are significantly higher.

4.7. Effect of dual mode on carbon dioxide emission

Carbon dioxide (CO_2) is the most prominent human made Greenhouse gas. However, natural gas, mainly composed by methane, has one of the lowest carbon contents among hydrocarbons, resulting in a potential of CO_2 emission lower than that of neat diesel [32].

The present results confirm that the use of natural gas in a dual fuel engine is an interesting technique to reduce this greenhouse gas especially at high loads. Fig. 13 shows the variation of CO_2 emissions for conventional diesel engine and dual fuel mode, as a

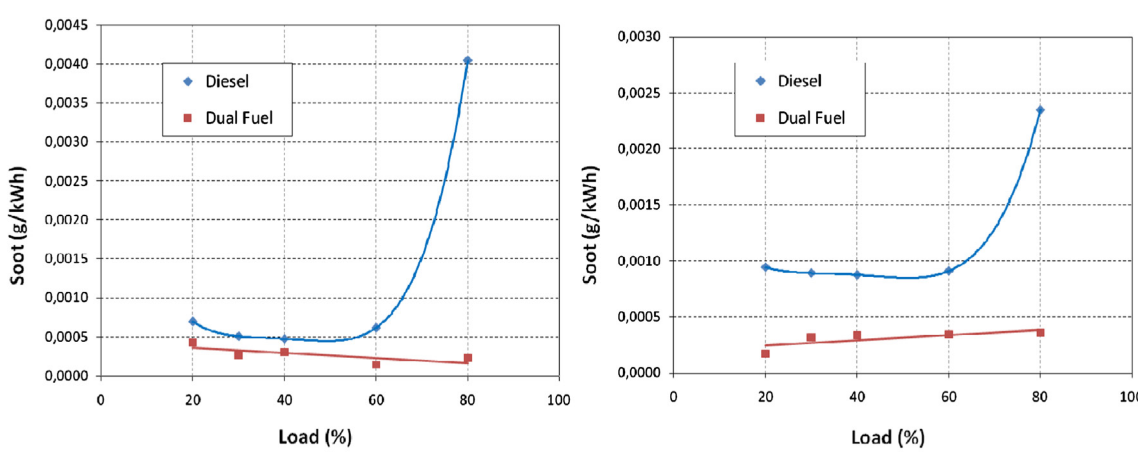
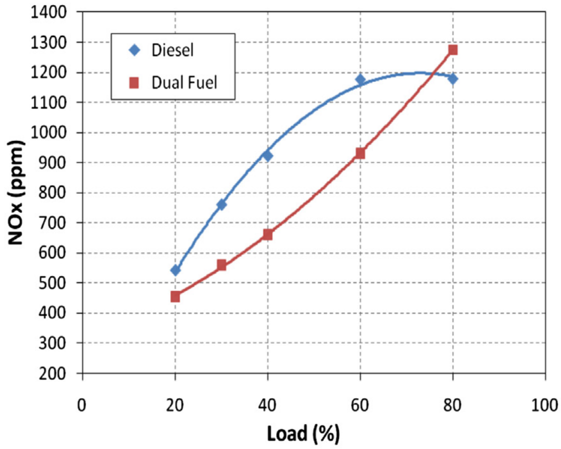
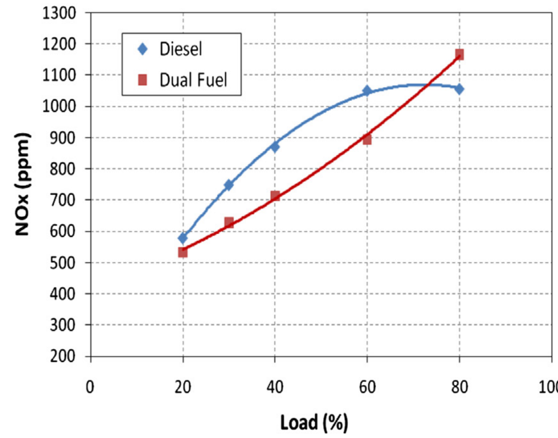


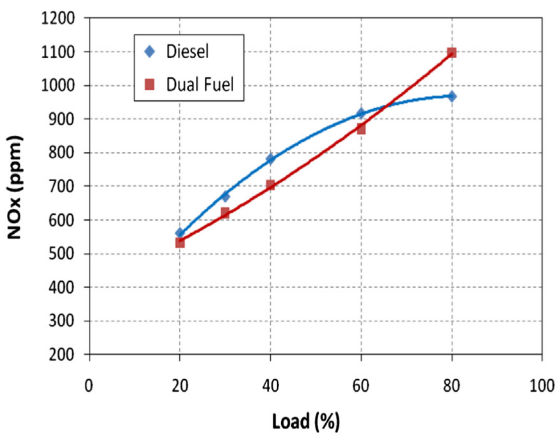
Fig. 7. Soot emissions at various loads for both dual fuel and conventional diesel modes at different engine speeds.



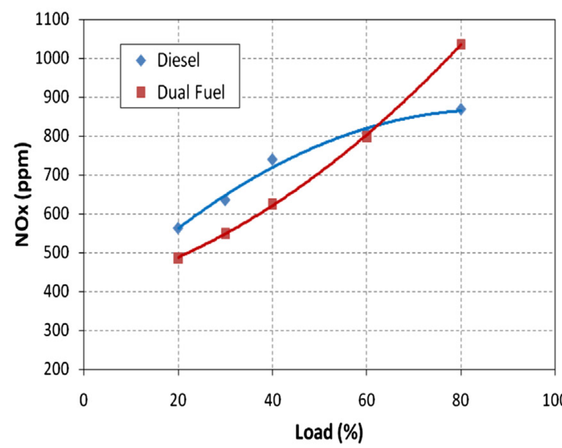
(a) N=1500 rpm



(b) N=1800 rpm



(c) N=2000 rpm



(d) N=2200 rpm

Fig. 8. Variation of NO_x emission according to load at different engine speeds.

function of engine load. Only results for 1500 and 2000 rpm engine speeds are reported as examples.

At low loads, since gaseous fuel participation is small (Fig. 11), the difference between CO_2 emissions for conventional diesel and dual fuel mode is not significant. However, for higher loads, due to the increase of engine load under dual fuel mode which is achieved by increasing the amount of natural gas, the difference is more important. In this case, the natural gas contribution to decrease the CO_2 emissions appears more clearly.

4.8. Some suggestions for novel measures towards dual fuel mode improvement

It is clear that dual fuel mode suffers from some deficiencies concerning mainly BSFC, THC and CO emissions at low loads. Our investigation showed clearly and on base of a complete analyze that this poor utilization of the gaseous fuel is due essentially to the combination of low temperature and very poor air-natural gas mixture inside the combustion chamber, resulting in a bad and slow combustion rate of the gaseous fuel. We consider that two approaches can be used for measures towards DF mode improvement:

a .Focus on gaseous fuel combustion promotion: this can be accomplished in three ways.
- Improve air-gaseous fuel mixture: install efficient mixing systems.
- Enhance air-gaseous fuel mixture temperature: preheating the mixture.
- Improve gaseous fuel combustion characteristics, especially lean burn limits and burning velocity: this can be accomplished for instance by Hydrogen enrichment [33].

b. Engine control strategy: a control system can be installed on the engine so that when the engine is operating on DF mode, it switches automatically to diesel mode, under a certain minimum load. Hence, the engine will operate on the DF mode interesting area and avoid the inconvenient domain.

5. Conclusion

In this study, the effect of dual fuel operating mode on combustion characteristics, engine performance and pollutants emissions of an existing diesel engine using natural gas as primary fuel and neat diesel as pilot fuel, has been examined.

At low engine loads, the total BSFC for dual fuel mode is higher than the conventional diesel. The increase in BSFC reveals a poor utilization of the gaseous fuel. This is mainly due to the combination of low temperature and very poor air-natural gas mixture inside the combustion chamber, resulting in a bad and slow combustion of the gaseous fuel.

However, at high and moderate loads, our results show a very interesting behavior of the dual fuel mode compared to conventional diesel. In fact, the total BSFC is lower for all the examined

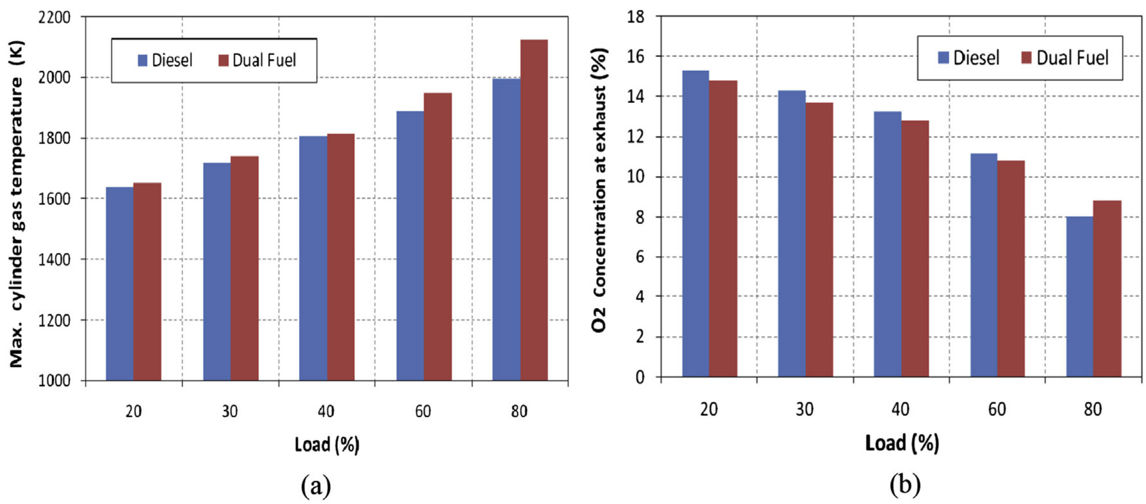


Fig. 9. Maximum cylinder gas temperature (a) and oxygen concentration (b) at various loads (N = 2000 rpm).

engine speeds. At those loads, the enhancement of the gaseous fuel utilization due to higher temperatures and richer mixtures, leads to a relevant improvement in the total BSFC and a higher heat release rate in the premixed-combustion phase for dual fuel mode. Consequently, the in-cylinder pressure peak for dual fuel mode becomes higher than the corresponding one for conventional diesel.

Concerning pollutant emissions, our results show that the use of natural gas for dual fuel mode is a very efficient technique to reduce soot emissions especially at high loads where they are important in

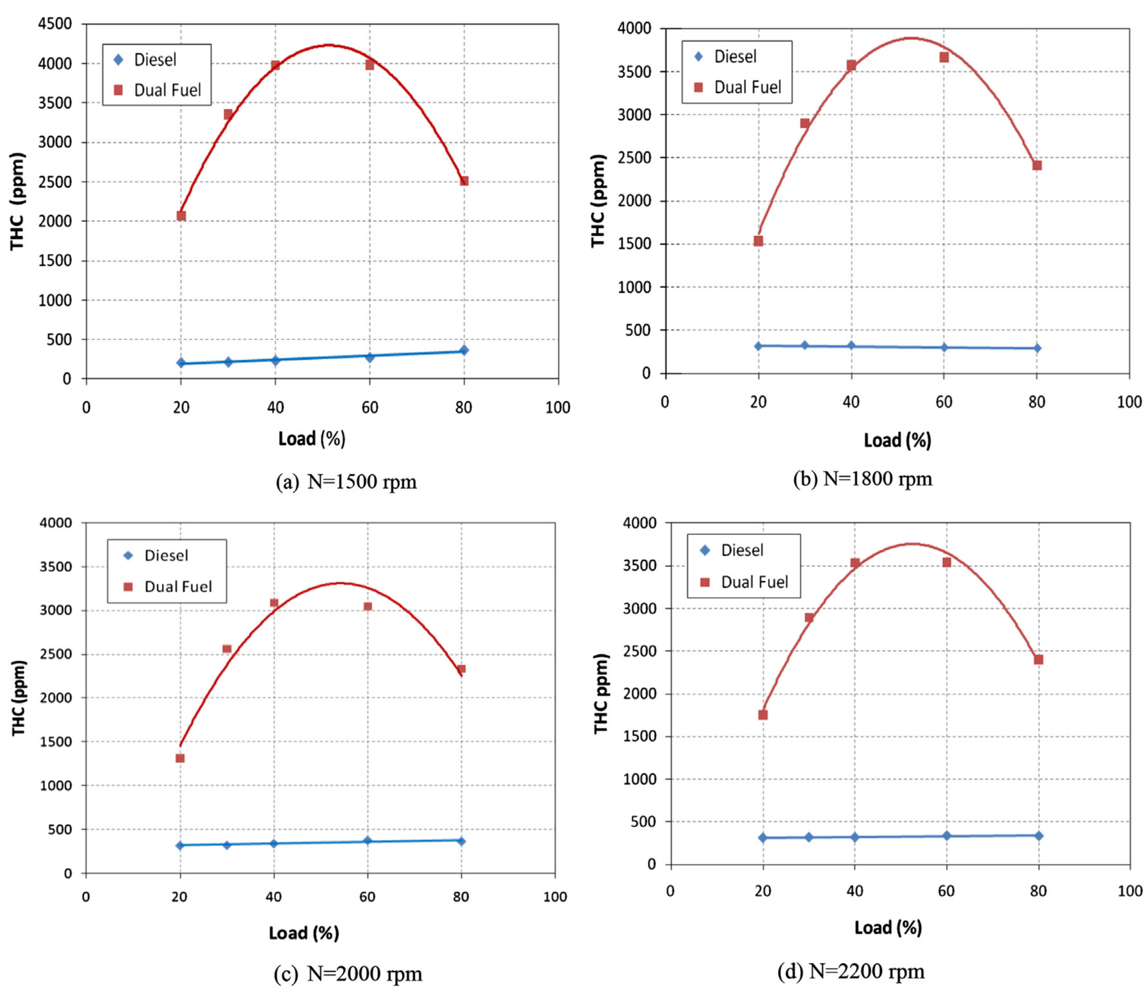


Fig. 10. Unburned hydrocarbon emissions at various loads for both dual fuel and conventional diesel modes for different engine speeds.

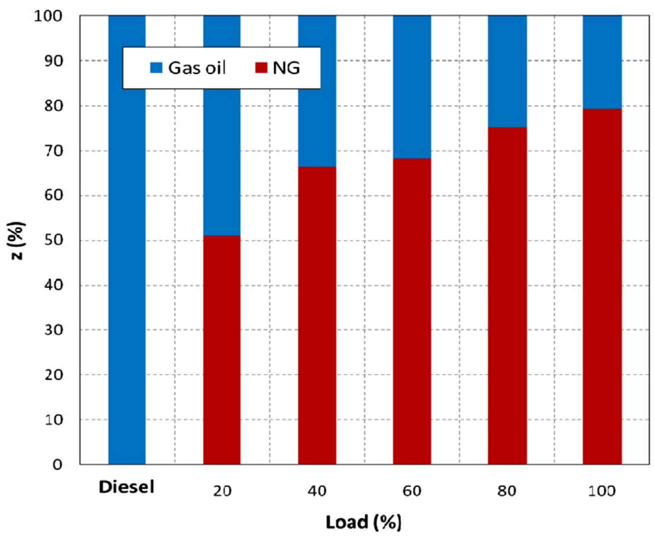


Fig. 11. Variation of the participation rate (Z) as a function of load, $N = 1500$ rpm.

conventional diesel engine. Regarding the nitric oxides concentration (NO_x), a reduction is also observed with dual fuel mode at low and moderate loads. However, for higher loads, the results show that NO_x becomes higher than the corresponding ones of conventional diesel engine. On the other hand, THC emissions are significantly higher for dual fuel mode. CO emissions are also higher at low and moderate loads for dual fuel mode.

Accordingly, dual fuel mode using natural gas is a promising technique for controlling soot and NO_x emissions which are noticeable challenges of diesel engines. Moreover, it showed the possibility to obtain lower BSFC than conventional diesel engine over a large engine operating area. In addition, this technique can be applied on existing DI diesel engines and requires only slight modifications. Nevertheless, the maximum permissible in-cylinder pressure must be verified.

Conversely, some deficiencies concerning BSFC at low engine load, THC and CO emissions are observed. This reveals that measures towards dual fuel mode improvement must focus in gaseous fuel combustion promotion. Some suggestions for new measures towards DF mode improvement are deduced. An idea for an

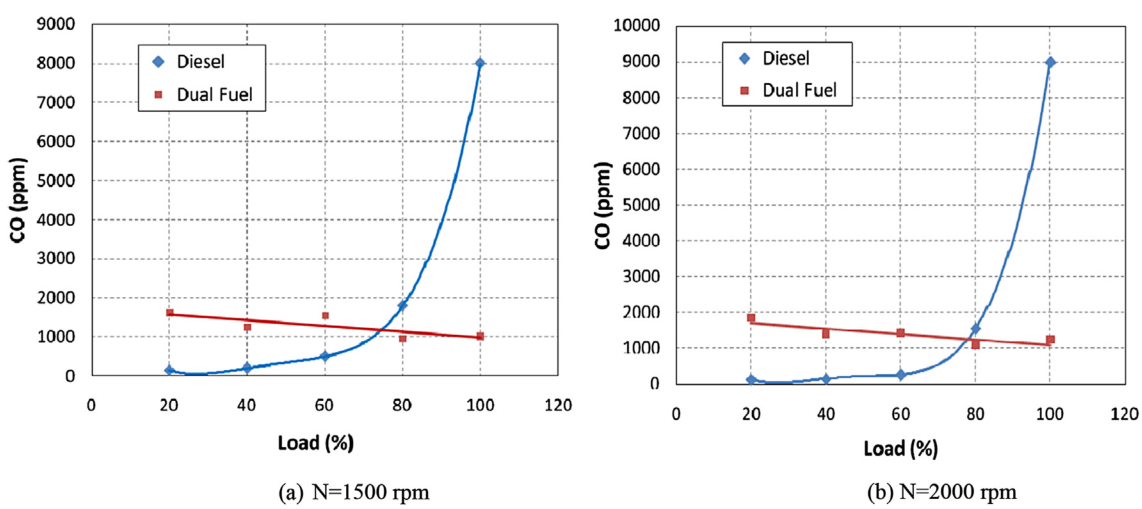


Fig. 12. Variation of the CO emissions according to load, for various engine speeds.

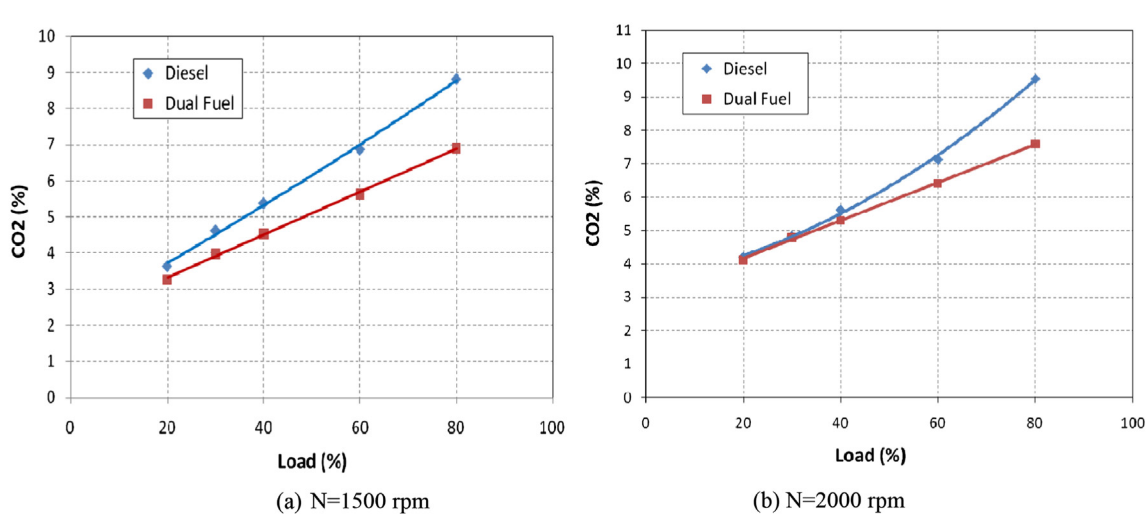


Fig. 13. Variation of the CO_2 emissions according to load, for various engine speeds.

adequate engine control strategy is also proposed. This can contribute to eliminate or at least reduce these deficiencies.

Nomenclature

Notations

$\dot{m}$	mass flow rate [kg/s]
N	engine rotational speed [rpm]
P	instantaneous in-cylinder gas pressure [bar]
Z	natural gas mass participation [%]

Abbreviations

BSFC	brake specific fuel consumption
CA	crank angle
CI	compression ignition
DI	direct injection
EVO	exhaust valve opening
EVC	exhaust valve closing
F/A ER	fuel air equivalence ratio
HRR	heat released rate
IVO	inlet valve opening
IVC	inlet valve closing
LHV	lower heating value
NO_x	nitrogen oxides
TDC	top dead center

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