Thermal Efficiency and Environmental Performance of a Natural Gas – Diesel Compression Ignition Engine – An Experimental Approach.

R.G. Papagiannakis[a]

[a]*Thermodynamic & Propulsion Systems Section, Department of Aeronautical Sciences, Hellenic Air Force Academy, Dekelia Air Force Base, 1010 Dekelia Attikis, Greece*

Abstract. Reduction of exhaust emissions is a major research task in diesel engine development in view of increasing concern regarding environmental protection and stringent exhaust gas regulations. Simultaneous reduction of NO_x emissions and particulate matter is quite difficult due to the soot/NO_x trade-off and is often accompanied by fuel consumption penalties. Towards this aim, automotive engineers have proposed various solutions, one of which is the use of alternative gaseous fuels as a supplement for the commercial liquid diesel fuel. This type of engine, which operates fuelled simultaneously with conventional diesel oil and gaseous fuel, is called “dual fuel” diesel engine. The majority of the dual fuel diesel engines operate on the fumigation principle where the liquid diesel fuel is partially replaced by gaseous one fumigated into the intake air. One of the gaseous fuels used commonly in compression ignition engine is natural gas, which has a relatively high auto- ignition temperature and moreover is an economical and clean burning fuel. The high auto – ignition temperature of natural gas is a serious advantage against other gaseous fuels since the compression ratio of most conventional DI diesel engines can be maintained. Moreover the combustion of natural gas produces practically no particulates since natural gas contains less dissolved impurities (e.g. sulfur compounds). In the present work, experimental results are provided concerning the combustion of natural gas in a compression ignition environment. The experimental investigation has been conducted on a single cylinder, naturally aspirated; high speed direct injection diesel engine properly modified to operate under dual fuel mode. By comparing the results, an important effect of engine operating point (i.e. load and engine speed) in combination with the presence of natural gas on the exhaust emissions and main combustion characteristics is observed. Through the experimental results provided here, it is observed that the main dual fuel combustion characteristics (i.e cylinder pressure, heat released rate, duration of combustion, ignition delay, total brake specific energy consumption, maximum combustion pressure) are affected remarkably so by the liquid fuel percentage replacement by natural gas as by the engine operating conditions (i.e. load and engine speed). As far as the exhaust emissions are concerned, it is revealed that the effect of the dual fuel operation on NO, CO and HC emissions is affected by the engine operating point (i.e. load and engine speed). Thus, the main objective of this comparative assessment is to elaborate the relative impact of each one of the above mentioned parameters on engine performance characteristics and exhaust emissions. Furthermore, an endeavor is made to determine the optimum combinations of these engine operational parameters. The conclusions of this study may be proven to be considerably valuable for the application of this technology on existing DI diesel engines.

Keywords: Dual Fuel Combustion; Natural Gas Fumigation; Supplement Ratio; Performance; Emissions.
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INTRODUCTION

The worldwide energy consumption is constantly increasing and it will certainly increase during at least the 21st century. Nowadays, around 80% of the world primary energy demand is satisfied by fossil fuels [1,2,3,4]. Awareness of limitations of fossil fuels reserves and the fact that burning of fossil fuels has a major contribution to the greenhouse gases emission has lead to a growing interest in the use of alternative fuels, particularly for the operation of internal combustion engines that are also the main “energy consumer”. A promising solution is the use of natural gas as a supplement for the conventional diesel fuel (dual fuel natural gas diesel engines), owing to its inherent clean nature of combustion combined with the high availability at attractive prices [1-4,5,6]. Thus, many conventional compression ignition engines can also operate on dual fuel principle (i.e. natural gas and diesel fuel). For the majority of the compression ignition engines, natural gas is most usually inducted with the air during the induction stroke. They are mixed and compressed as in a conventional diesel engine. The mixture does not auto- ignite due to the high auto-ignition temperature of methane which is the main constituent of the natural gas. At the same point near the top dead centre (TDC), an amount of the liquid diesel fuel is injected through the conventional diesel fuel injection system. Diesel fuel auto-ignites and creates ignition sources for the surrounding gaseous fuel mixture [5,6]. The specific type of dual fuel engine is referred to as “Fumigated Natural Gas Diesel (FNGD) Engine”. Natural gas is fumigated into the intake air and premixed with the air during the induction stroke. Under dual fuel operating mode, the amount of the gaseous fuel fumigated into the intake air replaces an equal amount of the inducted combustion air since at constant engine speed the total amount of the inducted mixture has to be kept constant. Furthermore, in fumigated dual fuel operating systems, the desired engine power output is controlled by changing the amount of the gaseous fuel (natural gas). Most current dual fuel engines are made to operate either on dual fuel principle with diesel ignition, or simply as conventional diesel engines [5,6].

A number of experimental and theoretical investigations concerning the dual fuel diesel – natural gas operating mode have been reported in the international literature [5-22]. Various researchers [5-22] have published extensive theoretical and experimental investigations concerning the combustion processes occurring inside the combustion chamber of a fumigated dual fuel diesel-natural gas compression ignition engine. During the last years, the present research group has reported experimental investigations along with computer simulations conducted on such kind of engines [23-27].

A primary objective of the present work is to examine the main characteristics of the dual fuel combustion under various combinations of load and engine speeds, primarily from the viewpoint of engine performance and exhaust emissions, where the liquid diesel fuel is partially replaced by natural gas at various percentages. For this purpose, an extended experimental investigation has been conducted on an existing single cylinder, naturally aspirated, high speed, direct injection (HSDI) diesel engine properly modified to operate under dual fuel operating mode. The engine is supplied with natural gas from the local low pressure distribution network after making the appropriate modifications. During the experimental investigation, pressure measurements are taken from the engine combustion chamber and the liquid fuel injection system using a high speed sampling device, while pollutants are measured at the engine exhaust. Moreover, measurements are taken of engine power, liquid and gaseous fuel consumption, exhaust gas temperature, and intake air mass flow rate.

From the analysis of the experimental measurements, important information is derived revealing the effect of engine operating conditions (i.e. load and engine speed) in combination with the natural gas concentration on the combustion mechanism occurring inside the combustion chamber, by estimating the ignition delay period, the duration of combustion and the intensity of the heat release mechanism as a function of the natural gas concentration in the cylinder charge. Moreover, under dual fuel operating mode, from the examination of both the experimental cylinder pressure and its derived heat release rate diagrams, for each engine operating condition (i.e. load and engine speed), important information is derived concerning the possibility of the appearance of knocking phenomena as the mass of supplementary fuel increases. Furthermore, under dual fuel operating mode and at various liquid fuel percentage replacement ratios by natural gas, the effect of engine operating conditions (i.e. load and engine speed) on the formation of pollutant emissions (NO, CO, HC and Soot) is revealed, by comparing the related values to the corresponding ones obtained under normal diesel fuel operation. The information derived from the present investigation is extremely valuable if one wishes to apply dual fuelling on an existing high speed direct injection diesel engine. It will be accomplished through the estimation of the proper combination of the engine operating condition and the ratio of diesel-natural gas fuel consumption so that the engine operation becomes friendlier to the environment, without deteriorating its performance compared to that of the conventional diesel operation.

EXPERIMENTAL SECTION

Experimental Installation

Facilities to monitor and control engine variables were installed on a single-cylinder test-bed Lister LV1 experimental engine. This is a four-stroke, naturally aspirated, air-cooled engine with a “bowl-in-piston” combustion chamber having a bore of 85.73 mm, a stroke of 82.55 mm and a rod length to crank radius of 3.6. The compression ratio is 18:1 and the nominal speed range is between 1000 and 3000 rpm. The liquid fuel is injected inside the cylinder through a three-hole injector nozzle (hole diameter of 250 μm), located near the combustion chamber center with an opening pressure of 180 bar. The engine is coupled to a Heenan & Froude hydraulic dynamometer [23-27].

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 volume air flow rate is measured. It must be stated here that 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 cylinder chamber. 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 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 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 intake of the engine and is mixed with the intake air.

The main measuring instruments were: an Alcock (viscous type) air flow meter; tanks and flow meters for fuel; temperature sensors for the exhaust gas, inlet air, lubricating oil and cooling water; a TDC marker (magnetic pick-up); an rpm indicator and a Kistler piezoelectric transducer for the combustion chamber pressure [23-27]. Another similar piezoelectric transducer was fitted to the high-pressure fuel pipe (from pump to injector) near the injector. A fast data-acquisition and recording system was used to record the pressure diagrams obtained by the piezoelectric transducers.

Exhaust gas analyzers were used to measure smoke, nitrogen oxide (NO), total unburned hydrocarbons (HC) (equivalent propane) and carbon monoxide (CO) at the tailpipe. A Bosch RTT100 smoke meter was used to measure smoke levels in the exhaust gases, NO emissions were measured with a Signal chemiluminescent analyzer and the HC emissions were measured with a Signal flame-ionization detector. The last two devices were fitted with thermostatically controlled heated lines. Finally, CO was measured with a Signal non-dispersive infrared analyzer.

At each operating point (i.e. load and engine speed) forty cycles were acquired on a time basis. For the estimation of the mean indicator diagram averaging took place over the indicator diagrams of 40 consecutive cycles. Each one of the measured indicator diagram is converted to a crank angle basis one using the engine speed which is measured every half revolution of the crankshaft. Thus, for each engine speed the actual sampling rate in degree crank angle is determined from the precise engine speed which is estimated from the TDC signal and the desired crank angle resolution.

Description of Test Fuels

The main properties of the liquid diesel fuel used are given in Table 1. 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 1. As observed, methane is the main constituent of the natural gas resulting in a relatively high octane number, which makes it suitable for engines of high compression ratio. Furthermore, its C/H ratio is low resulting in a significant reduction of the specific CO_2 emissions.

TABLE 1. Basic Characteristics of the fuels used.

Liquid Diesel Fuel (CEN EN-590)	
Cetane Number	52.5 (-)
Density :	833.7 (kg/m3)
LHV	42.74 (MJ/kg)
Sulfur Content	45 (mg/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)

Experimental Procedure

For all test cases examined during the experimental investigation, the static injection timing was kept constant at 26 degrees crank angle before top dead center (BTDC), the air inlet temperature was measured about 23 ^oC and the air absolute humidity was measured about 50%. Measurements have been taken at three different engine loads corresponding to 2.45 bar, 3.69 bar and 4.92 bar brake mean effective pressure, and three engine speeds of 1500, 2000 and 2500 rpm under both normal diesel and dual fuel operation. Under dual fuel operation and for each operating point (i.e. load and engine speed), measurements have been taken for various liquid fuel percentage replacement ratios by natural gas (i.e. supplement ratio). In the present work the "supplement ratio" (x) represents the quotient of the mass flow rate of natural gas divided by the total fuel (diesel and natural gas) mass flow rate, and is given by the formula:

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

It must be stated here that under dual fuel operating mode, the amount of the gaseous fuel fumigated into the intake air replaces an equal amount of the inducted combustion air since for specific engine speed the total amount of the inducted mixture has to be kept constant. Furthermore, in the present investigation an attempt was made to conduct all experiments without significant fluctuations in air inlet temperature and lubricating oil temperature as a method to prevent possible discrepancies in engine operation during the tests and mainly, to avoid variations in engine loading.

Under dual fuel operating mode the experimental procedure has as follows: At a given constant engine speed, enough amount of liquid diesel fuel is provided to achieve a percentage of the desired engine power output. The rest percentage of the desired engine power output is reached by using only natural gas, which is fumigated into the air intake. Thus, for example, at 2000 rpm engine speed when the engine runs under dual fuel operating mode with supplement ratio $x = 27\%$ at 4.92 bar brake mean effective pressure (i.e. corresponds to 80% of full engine load), a percentage of the desired engine power output (~75%) is obtained using only diesel fuel while the remaining (~25%) up to the desired engine load is obtained from the gaseous fuel.

ESTIMATION OF THE TOTAL BURNING RATE

For each engine operating point (i.e. load and engine speed), the total burning rate is estimated by using the mean cylinder indicator diagram. Combustion duration and intensity are estimated from the TDC pickup signal combined with the calculated heat release rate, which is a most valuable source of information for the combustion mechanism in diesel engines [28,31]. The net heat release rate is determined by applying the first thermodynamic law as follows [28,31]:

$$\frac{dQ_{net}}{d\varphi} = \frac{C_V}{R}\left(P\frac{dV}{d\varphi} + V\frac{dP}{d\varphi} - \frac{PV}{m}\frac{dm}{d\varphi} \right) + P\frac{dV}{d\varphi} \tag{2}$$

During this analysis, the following assumptions are made:

- The cylinder charge is considered as an ideal gas.
- The composition of the working gas is considered to be variable and is estimated from the trapped mass (air and gaseous fuel) at inlet valve closure and the amount of fuel burned up to the current engine crank angle
- The distribution of thermodynamic properties inside the combustion chamber is considered to be uniform
- No dissociation present in combustion products

- The cylinder is considered as a closed system, so that no variation of cylinder mass due to blow-by or gas leakage through the inlet/exhaust valves is considered.

The gross heat release rate, which provides the actual rate of energy release, is then obtained by

$$\frac{dQ_{gross}}{d\varphi} = \frac{dQ_{net}}{d\varphi} - \frac{dQ_w}{d\varphi} \tag{3}$$

where the heat loss rate ($dQ_w / d\varphi$) (negative from gas to walls) is obtained from Eq. (4) using the heat transfer model of Annand [29],

$$\frac{dQ_w}{dt} = A\left(\alpha_c \frac{\lambda}{D} \mathrm{Re}^b \left(T_w - T_g\right) + c_r \left(T_w^4 - T_g^4\right)\right) \tag{4}$$

where (α_c, b and c_r) are constants, (A) is the total area of the cylinder and (λ) is the thermal conductivity of the working medium. In Eq. (4), it is required to have an estimate of the cylinder gas temperature (T_g). For this reason, it is assumed that the cylinder content behaves as a perfect gas as already mentioned. The mean cylinder gas temperature is obtained using the following expression:

$$T_g = \frac{P \cdot V}{m \cdot R} \tag{5}$$

where the cylinder pressure (P) is obtained from the mean cylinder indicator trace. The specific gas constant (R) is calculated from the mean gas composition estimated from the initially trapped mass and the fuel burned up to the current engine crank angle. The trapped mass at inlet valve closure is estimated from an open cycle simulation of the engine, using the measured mass flow rate of air and gaseous fuel. Using the previous methodology, we obtain a good estimate of the actual rate of heat release inside the combustion chamber, since the net heat release rate does not account for the energy loss due to heat exchange with the cylinder walls. The heat transfer model coefficients are calibrated using an iterative procedure, so that the cumulative gross heat release rate calculated is equal to the total energy released from the combustion of the measured gaseous and liquid fuel mass. This provides us with a good estimate of the rate of fuel consumption (combustion) inside the engine cylinder. Under dual fuel operation, the estimated heat release rate is the total one due to the combustion of both the liquid fuel and the gaseous one [32].

UNCERTAINTY ANALYSIS OF THE EXPERIMENTAL DATA

For each engine operating mode, i.e. normal diesel and dual fuel operation, two sets of measurements have been 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 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 COV for each measured parameter is presented in Table 2. Considering these values, it is shown that the measurements are quite repeatable, especially concerning engine performance characteristics.

TABLE 2. Coefficient of Variance for the Measured Quantities

Measured Quantity	**COV (%)**
Maximum Combustion Pressure	1.2
Brake Specific Energy Consumption	0.9
Exhaust Gas Temperature	3.1
Nitric Oxide	3.5
Carbon Monoxide	2.9
Unburned Hydrocarbons	3.2
Soot	3.9

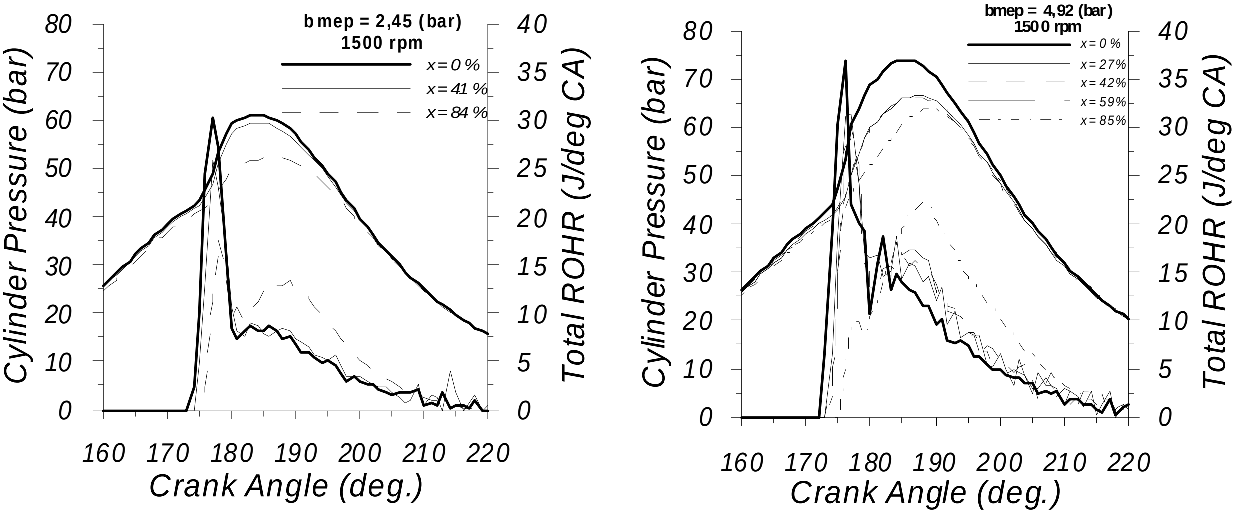
RESULTS AND DISCUSSION

Cylinder Pressure and Total Heat Release Rate Data

Figures 1-3 provide the experimental pressure and total heat release traces for 1500, 2000 and 2500 rpm engine speed, at 2.45 and 4.92 bar brake mean effective pressure, under normal diesel ($X = 0\%$) and dual fuel natural gas-diesel ($X \neq 0\%$) operating modes. Observing these figures, it is obvious that the presence of natural gas in the cylinder charge affects both cylinder pressure and the total burning rate. Specifically, at low load (i.e. 2.45 bar brake mean effective pressure) and for all engine speeds examined, the gaseous fuel affects only slightly the value of the cylinder pressure compared to the one under normal diesel operation. The difference becomes more evident during the last stage of compression and during the initial stage of the combustion process. The difference observed during the last stages of the compression stroke is the result of the higher specific heat capacity of the natural gas – air mixture compared to that of the air for normal diesel operation while the difference observed during the initial stage of the combustion process is the result of the poor combustion of the gaseous fuel during the premixed controlled combustion phase. Moreover, at low load, the increase of engine speed leads to an increase of the cylinder pressure traces observed under dual fuel operating modes which tend to converge to the respective one observed under normal diesel operating mode. This is due to the fact that the increase of engine speed leads to a warmer engine and to an increase of the turbulence inside the combustion chamber. These factors affect positively (enhance) the flame speed contributing thus to the improvement of the gaseous fuel combustion quality (lower ignition delay and faster flame speed).

At high load (i.e. 4.92 bar brake mean effective pressure), as the amount of the replaced liquid diesel fuel increases, the rate of cylinder pressure rise during the initial stage of the combustion process becomes lower compared to the respective one under normal diesel operation while the peak of the cylinder pressure occurs later compared to the respective value observed under normal diesel operation. It is the result of both the later initiation of combustion and the lower premixed controlled combustion rate of diesel fuel due to its smaller amount. At low supplement ratios it becomes more evident at low engine speed, while the increase of engine speed results in a slightly increase of the rate of the cylinder pressure rise especially during the initial stages of combustion. At high supplement ratios, the increase of engine speed results in a considerable improvement of the gaseous fuel combustion quality (faster flame speed) which affects positively the rate of the cylinder pressure rise during the combustion process resulting thus to a considerable increase of the maximum cylinder pressure, which may converge to the respective one under normal diesel operation.

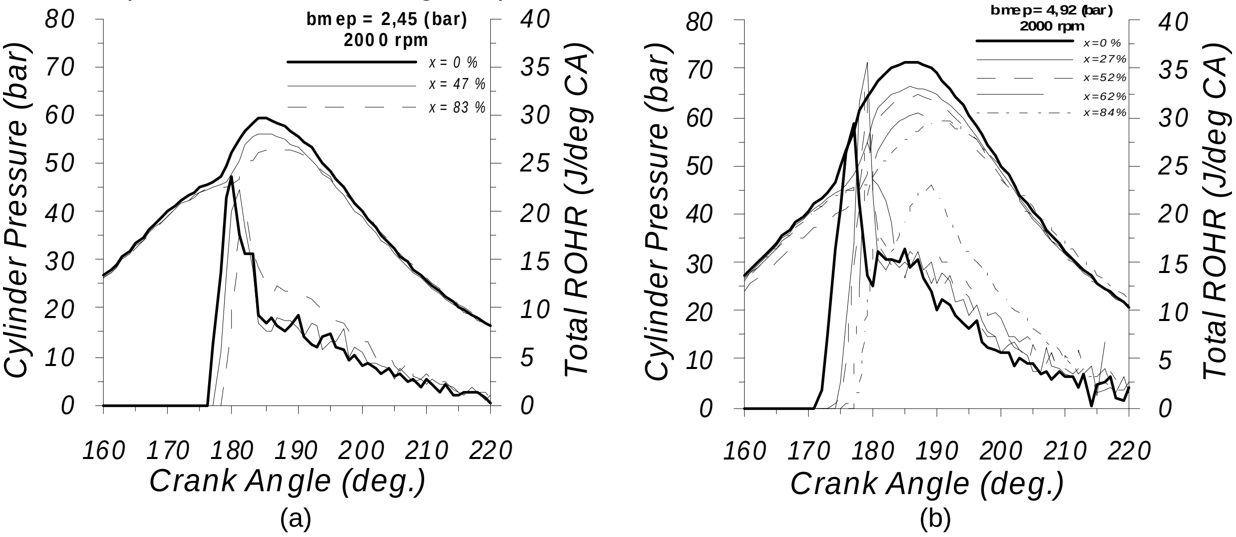
As far as the total heat release rate curves are concerned, it is revealed that the presence of natural gas in the cylinder charge affects the combustion process. The initiation of combustion observed under dual fuel operating modes starts later compared to the respective one under



(a) (b)

FIGURE 1. Experimental pressure and heat release traces under normal diesel and dual fuel operating modes for 1500 rpm engine speed (a) at 2.45 bar and (b) at 4.92 bar brake mean effective pressure.

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. For all engine operating points examined, the total burning rate observed during the initial stages of combustion under dual fuel operating modes 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 combustion 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. The difference becomes more evident at part load and for all engine speed examined.



(a) (b)

FIGURE 2. Experimental pressure and heat release traces under normal diesel and dual fuel operating modes for 2000 rpm engine speed (a) at 2.45 bar and (b) at 4.92 bar brake mean effective pressure.

As far as the second phase of combustion process is concerned, it is revealed that the total burning rate during the diffused controlled combustion phase is considerable higher compared to the respective one observed under normal diesel operation. This is the result of the considerable improvement of the gaseous fuel combustion quality that is caused by the improvement of the cylinder charge conditions (i.e. gaseous fuel concentration, cylinder gas temperature etc.), which contributes significantly to the existence and the fast spread of the flame front surrounding the burning zone. The effect is stronger again at low engine speed, revealing the longest duration of the gaseous fuel combustion. 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.

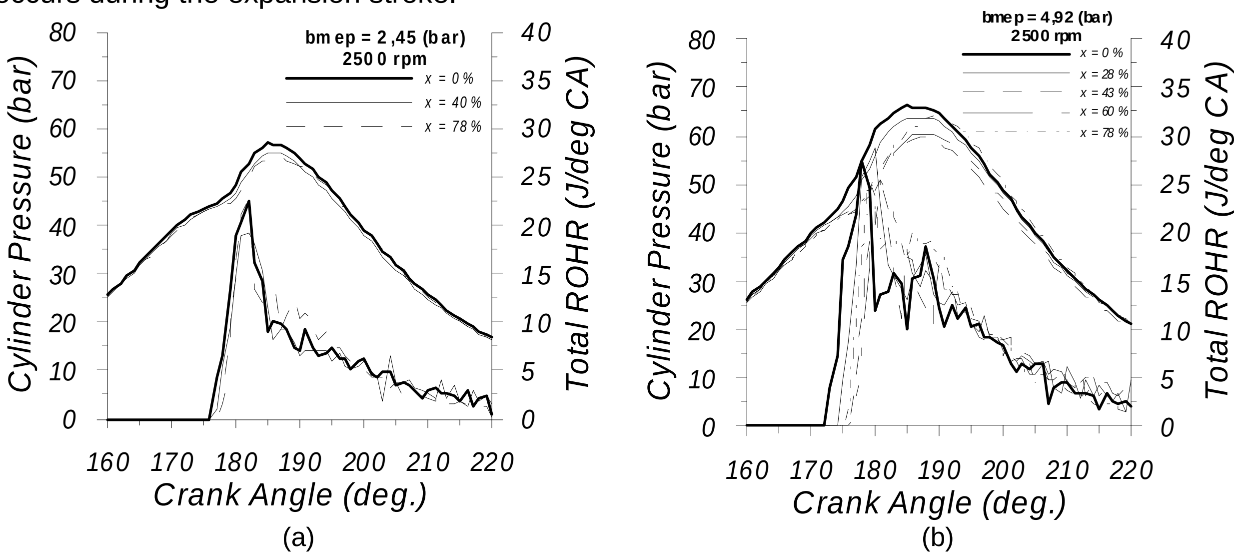


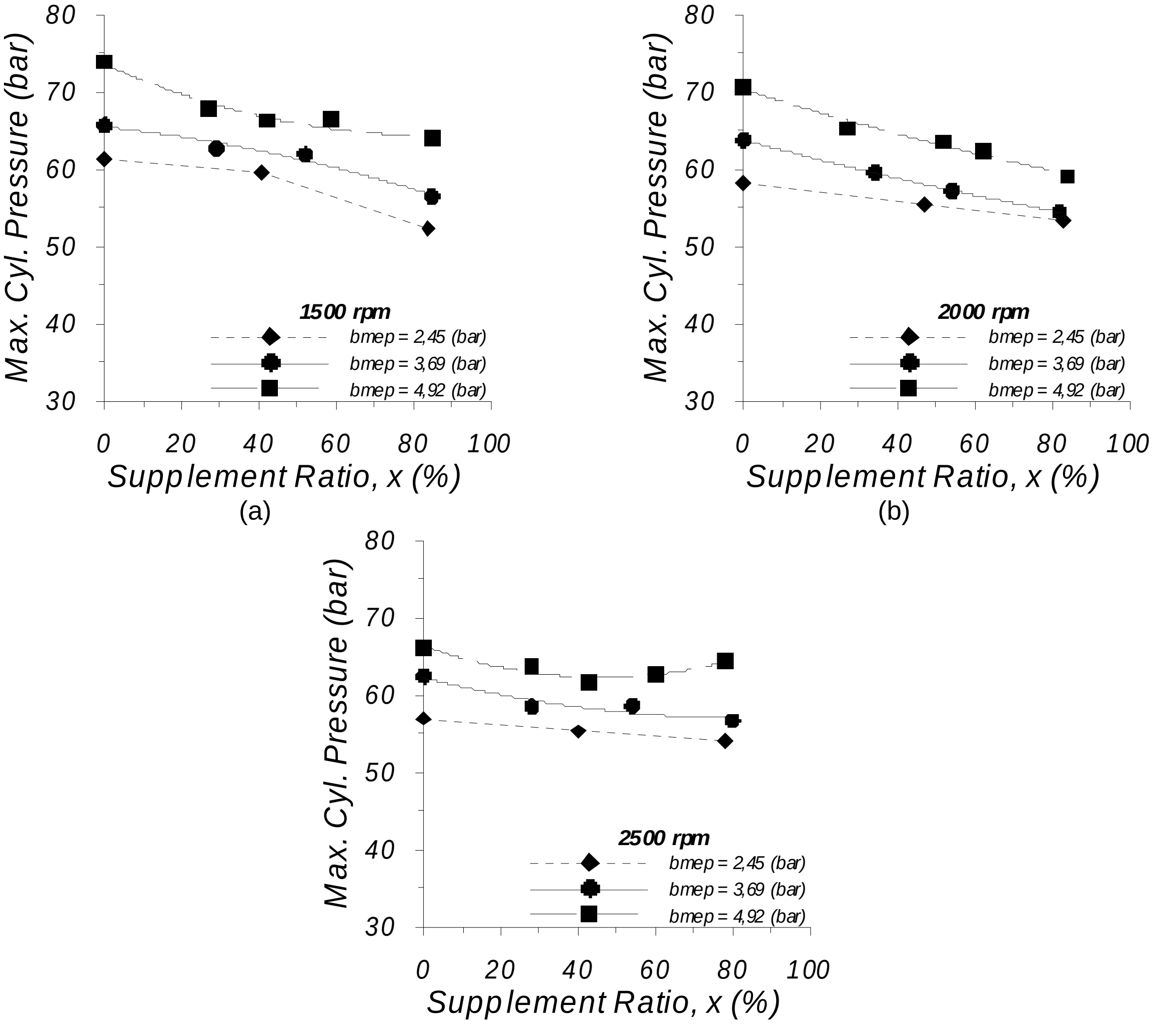
FIGURE 3. Experimental pressure and heat release traces under normal diesel and dual fuel operating modes for 2500 rpm engine speed (a) at 2.45 bar and (b) at 4.92 bar brake mean effective pressure.

Maximum Cylinder Pressure

Figures 4 (a, b and c) provide the variation of the maximum cylinder pressure as a function of the supplement ratio for 2.45, 3,69 and 4.92 bar brake mean effective pressure at 1500, 2000 and 2500 rpm engine speed, respectively. Observing these figures it is revealed that under dual fuel operating modes the increase of the supplement ratio, keeping engine load constant, leads to a significant decrease of the maximum cylinder pressure. For the same engine speed the slope each one of the load curves being almost the same for the entire range of the supplement ratios examined. At part load and low engine speed conditions the decrease of the maximum cylinder pressure becomes more evident at low supplement ratios. As engine speed increases the improvement of the gaseous fuel combustion affects positively the maximum cylinder pressure revealing thus to smaller decrease of the maximum cylinder pressure.

At high load the maximum combustion pressure starts to decrease with the increase of gaseous fuel concentration in the cylinder charge. Further increase of the amount of the gaseous fuel leads, beyond a certain value, to an increase of the maximum combustion pressure, which tends to converge to the one under normal diesel operation. The effect becomes more evident at high engine speed.

It should be mentioned that under dual fuel operating modes examined, the lower heat release rate during premixed controlled combustion phase and the higher specific heat capacity of the natural gas – air mixture are the main reasons of the lower and delayed appearance of maximum combustion pressure compared to normal diesel operation. This is encouraging since, apparently, no danger exists for the engine structure associated to cylinder pressure, if the specific technology is to be applied on conventional diesel engines.



(a) (b) (c)

FIGURE 4. Variation of Maximum Cylinder Pressure as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

Exhaust Gas Temperature

The variation of the measured exhaust gas temperature with diesel fuel supplement ratio is given in figures 5 (a, b and c), for various combinations of engine load (i.e. 2.45 bar, 3.69 bar and 4.92 bar brake mean effective pressure) and engine speed (i.e. 1500, 2000 and 2500 rpm). As well recognized [28,30-32], at each engine load the exhaust gas temperature is favored, in general, by the increase of the engine speed. Thus, for the same supplement ratio, keeping constant engine load, the increase of engine speed leads to an increase of the exhaust gas temperature. Moreover, at each engine speed the increase of engine load leads to a considerable increase of the exhaust gas temperature.

Under dual fuel operating mode at low load, the effect of the percentage of liquid fuel replacement on the exhaust gas temperature is almost negligible. Moreover, at low load and for all engine speeds, the exhaust gas temperature measured at extremely high supplement ratios seems to be slightly lower compared to the respective one observed under normal diesel operation.

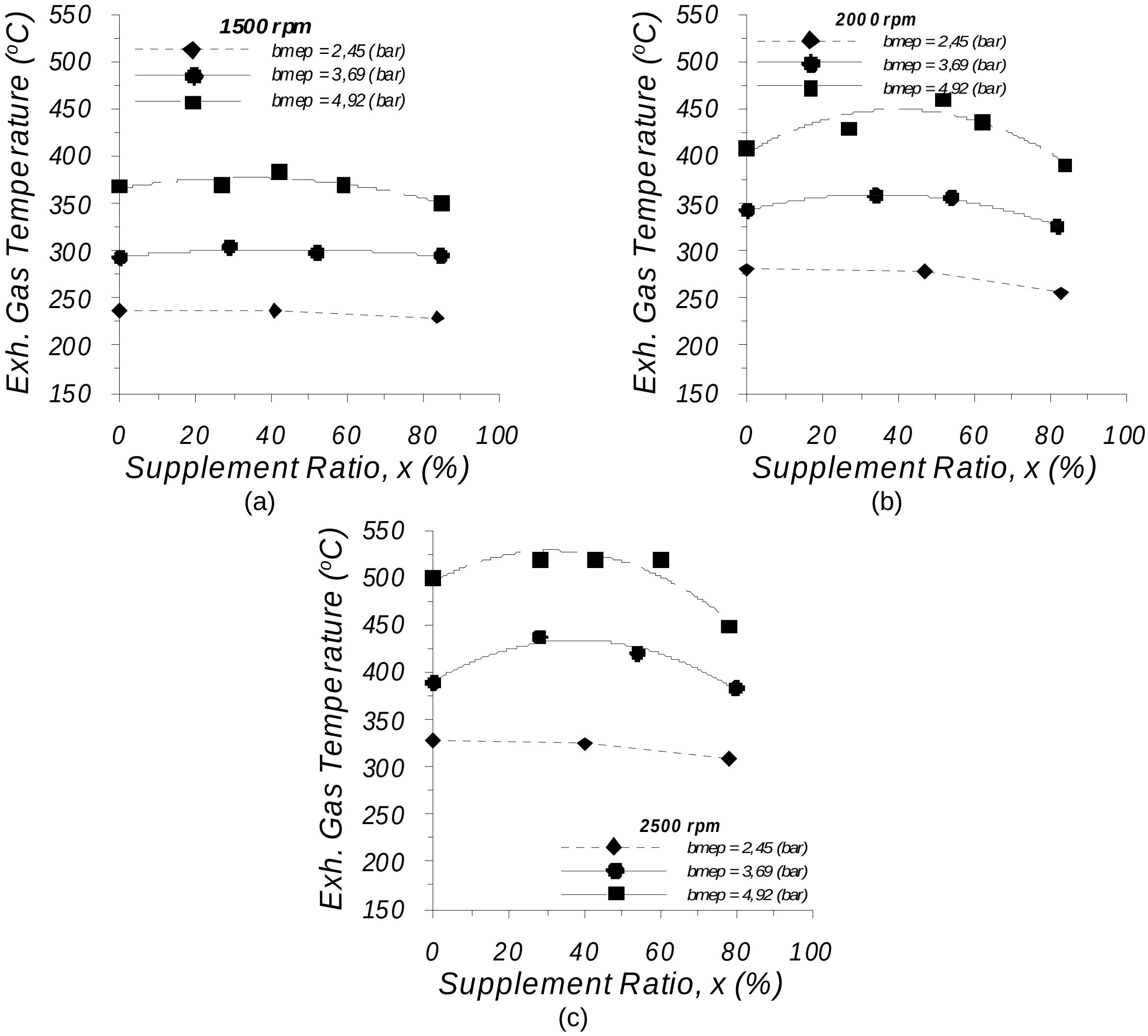
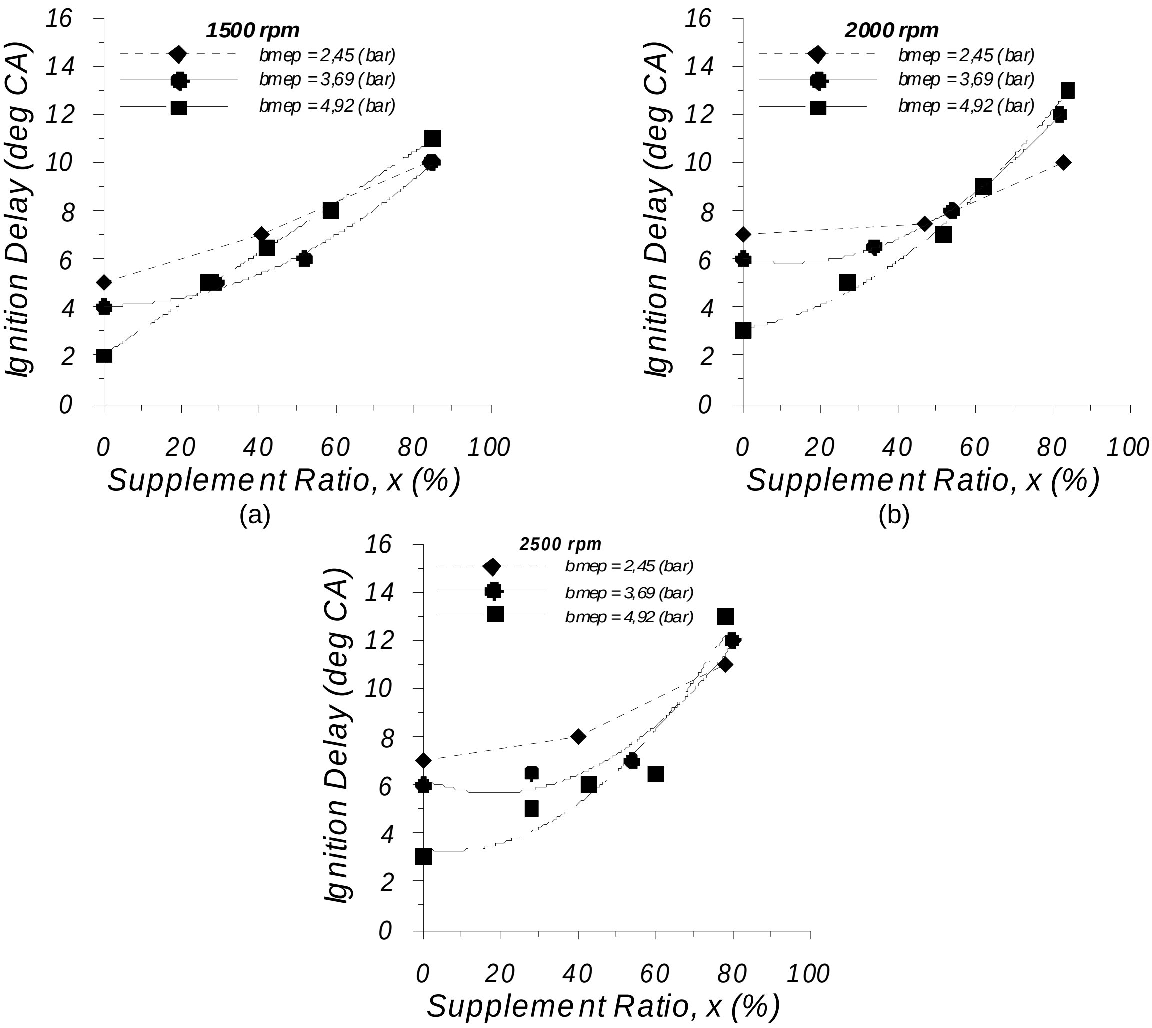


FIGURE 5. Variation of Exhaust Gas Temperature as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

At high engine load, as the percentage of liquid fuel replacement increases, there is an increase of the exhaust gas temperature. Further increasing the supplement ratio beyond a certain limit, leads to a slight decrease of the exhaust gas temperature which may in some cases seems to be lower than the respective one observed under normal diesel operation. In some cases, it follows an incremental way and the specific trend becomes more evident at high engine speed.

Ignition Delay Period

The ignition delay period is defined as the time interval from the start of injection of the liquid fuel to the initiation of the liquid fuel combustion [28,31,32]. The variation of ignition delay period as a function of the supplement ratio for various combinations of engine load (i.e. 2.45, 3.69 and 4.92 bar brake mean effective pressure) and engine speed (i.e. 1500, 2000 and 2500 rpm) is given in figures 6 (a, b and c). It is well known [28] that under normal diesel operation ignition delay period is affected, in general, by the change of the engine load. Thus, under normal diesel operation (x=0%), the increase of the engine load, keeping constant engine speed, results in a considerable decrease of the ignition delay period.



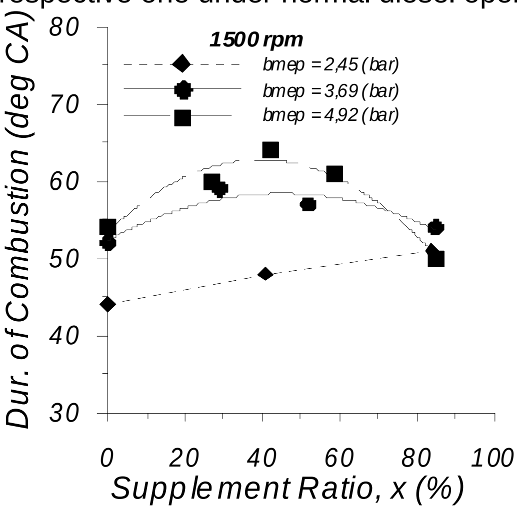
(a) (b) (c)

FIGURE 6. Variation of Ignition Delay period as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

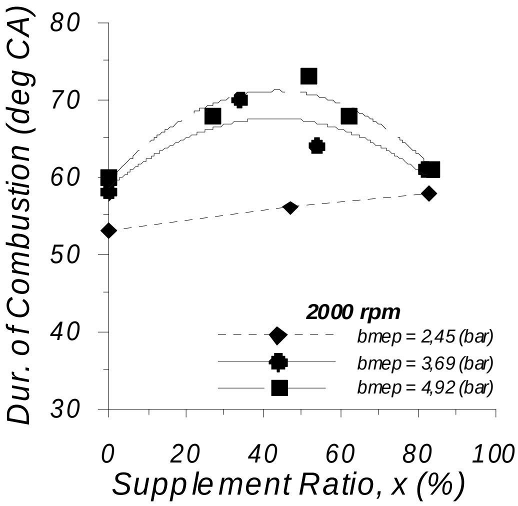
Under dual fuel operating mode ($x \neq 0\%$), the presence of the gaseous fuel affects the ignition point of the liquid diesel fuel. This is due mainly to the reduction of charge temperature close to the point of the liquid fuel injection which is caused to the higher overall specific heat capacity of the gaseous fuel – air mixture as compared with the respective one of the air observed under normal diesel operation5. This has as a result a drop of gas temperature at the start of diesel fuel injection, which obviously affects positively (i.e. increase) the ignition delay period [28]. Thus, for all engine operating points (i.e. load and engine speed), the increase of the supplement ratio leads to an increase of the ignition delay period of the injected liquid fuel. Thus, for the same engine operating point (i.e. load and engine speed), ignition delay period under dual fuel mode is higher compared to the respective one observed under normal diesel operation. At extremely high supplement ratios, the increase of engine load results to a decrease of the ignition delay period which seems to be opposed to the respective trend observed under normal diesel operation. This is explained taking into account the fact that in fumigated natural gas – diesel operating mode, the increase of engine load is accompanied by an increase of the fumigated amount of the natural gas which results in extremely high overall specific heat capacity of the gaseous fuel – air mixture close to the point of the liquid fuel injection.

Duration of Combustion

Figures 7 (a, b and c) provide the variation of the duration of combustion as a function of the supplement ratio for 2.45, 3,69 and 4.92 bar brake mean effective pressure at 1500, 2000 and 2500 rpm engine speed, respectively. Examining these figures, it is observed that for the same supplement ratio the increase of engine speed, keeping constant engine load, results to a slight increase of the duration of combustion. At low load, the increase of the supplement ratio leads to a slight increase of the duration of combustion. Comparing the results between normal diesel and dual fuel operating modes corresponding to low engine load conditions, it is revealed that the duration of combustion under dual fuel operating mode is slightly higher compared to the respective one under normal diesel operation.



(a)



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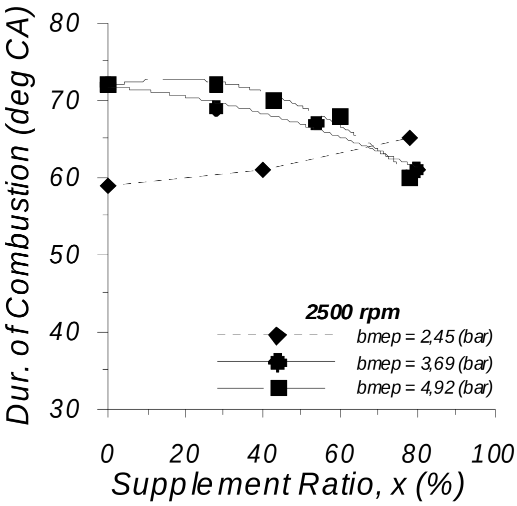


(c)

FIGURE 7. Variation of Duration of Combustion as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

At high load and for low and intermediate engine speed, the duration of combustion increases with increasing natural gas mass ratio and beyond a certain value of the gaseous fuel percentage it starts to decrease, as a result of the high cylinder charge temperature and the faster combustion rate of the gaseous fuel. At extremely high supplement ratios, the duration of combustion tends to become even lower compared to the respective one observed under normal diesel operation. The effect becomes more evident at high engine speed where the warmer engine has a positive effect on the existence and spread of the flame front surrounding the burning zone. Thus, at high load and high engine speed, the increase of supplement ratio results to a considerable improvement of the gaseous fuel combustion quality which leads to a shorter duration of combustion compared to the respective one observed under normal diesel operation.

Total Brake Specific Energy Consumption

The variation of the total brake specific energy consumption with diesel fuel supplement ratio is given in figures 8 (a, b and c), for various combinations of engine load (i.e. 2.45 bar, 3.69 bar and 4.92 bar brake mean effective pressure) and engine speed (i.e. 1500, 2000 and 2500 rpm).

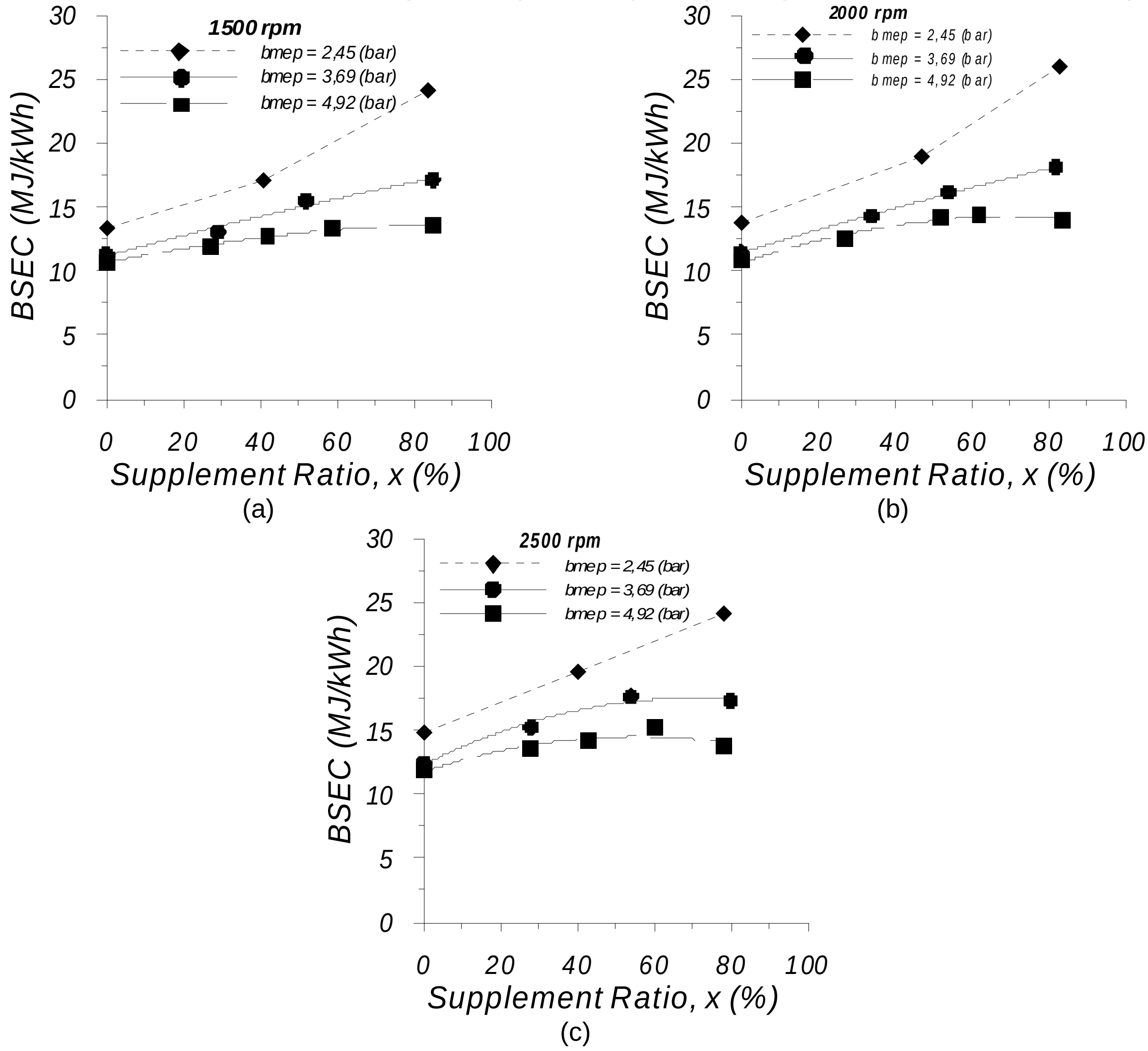


FIGURE 8. Variation of the brake specific energy consumption as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

It should be stated here that the lower heating value of natural gas is higher compared to the one of diesel fuel used, revealing that the total brake specific fuel consumption observed under dual fuel operation would be even higher if it were corrected (reduced) to the heating value of diesel fuel. Thus, in the present contribution, the total brake specific energy consumption has been used instead of the total bake specific fuel consumption. The experimental total brake specific energy consumption is estimated from the measured brake power output, the measured mass flow rates of diesel and natural gas and their lower heating values. Thus, no correction is made to cater for the difference in the lower heating values between natural gas and diesel fuel. As observed the brake specific energy consumption is affected considerably by the presence of the gaseous fuel in the charge mixture. Examining these figures, it is revealed that for all engine operating points examined the engine efficiency under dual fuel operation is lower compared to the respective one under normal diesel.

At low load, the increase of the supplement ratio results to an increase of the total brake specific energy consumption. The increase becomes more evident at extremely high supplement ratios and for low and intermediate engine speeds, where the extremely low amount of diesel fuel used affects negatively the quality of the diesel fuel spray, resulting in poor diesel fuel preparation, especially during the ignition delay period, a fact that has a negative effect on the gaseous fuel ignition process.

At high load and for all engine speed examined, the increment of the gaseous fuel percentage, keeping engine load constant, leads initially to a slight increase of the total brake specific energy consumption, while a further increase of the gaseous fuel supplement ratio results to a slight improvement of the engine efficiency, which tends to converge to the respective one observed under normal diesel operation. This is the result of the improvement of the gaseous fuel utilization observed at high load. The effect becomes more evident at high engine speed, where the dual fuel operation at high supplement ratios becomes more efficient compared to normal diesel operation revealing thus the considerable improvement of the gaseous fuel utilization.

Nitric Oxide (NO) Emissions

Figures 9 (a, b and c) provide the variation of the specific nitric oxide concentration as a function of the supplement ratio for 2.45, 3,69 and 4.92 bar brake mean effective pressure at 1500, 2000 and 2500 rpm engine speed, respectively. As well recognized [28-32], the formation of nitric oxides is favored, in general, by high oxygen concentration and high charge temperature. Examining these figures, NO emission is affected considerably by the presence of gaseous fuel in the charge mixture. In general, NO concentration observed under dual fuel operating mode are lower compared to the one observed under normal diesel operation at the same engine operating conditions (engine speed, load). Specifically, at low load, there is a slight decrease of NO emissions with the increase of the percentage of liquid fuel replacement. The effect becomes more evident at high supplement ratios and low engine speed.

At high engine load, there is a considerable decrease of NO emissions with increased percentages of the liquid fuel replacement until a certain limit where the trend of nitric oxide reduction tends to decrease. A further increase of the supplement ratio leads to an increase of NO concentration which may in some cases be higher than the one observed under normal diesel operation. This becomes more evident at high engine speed and extremely high supplement ratios.

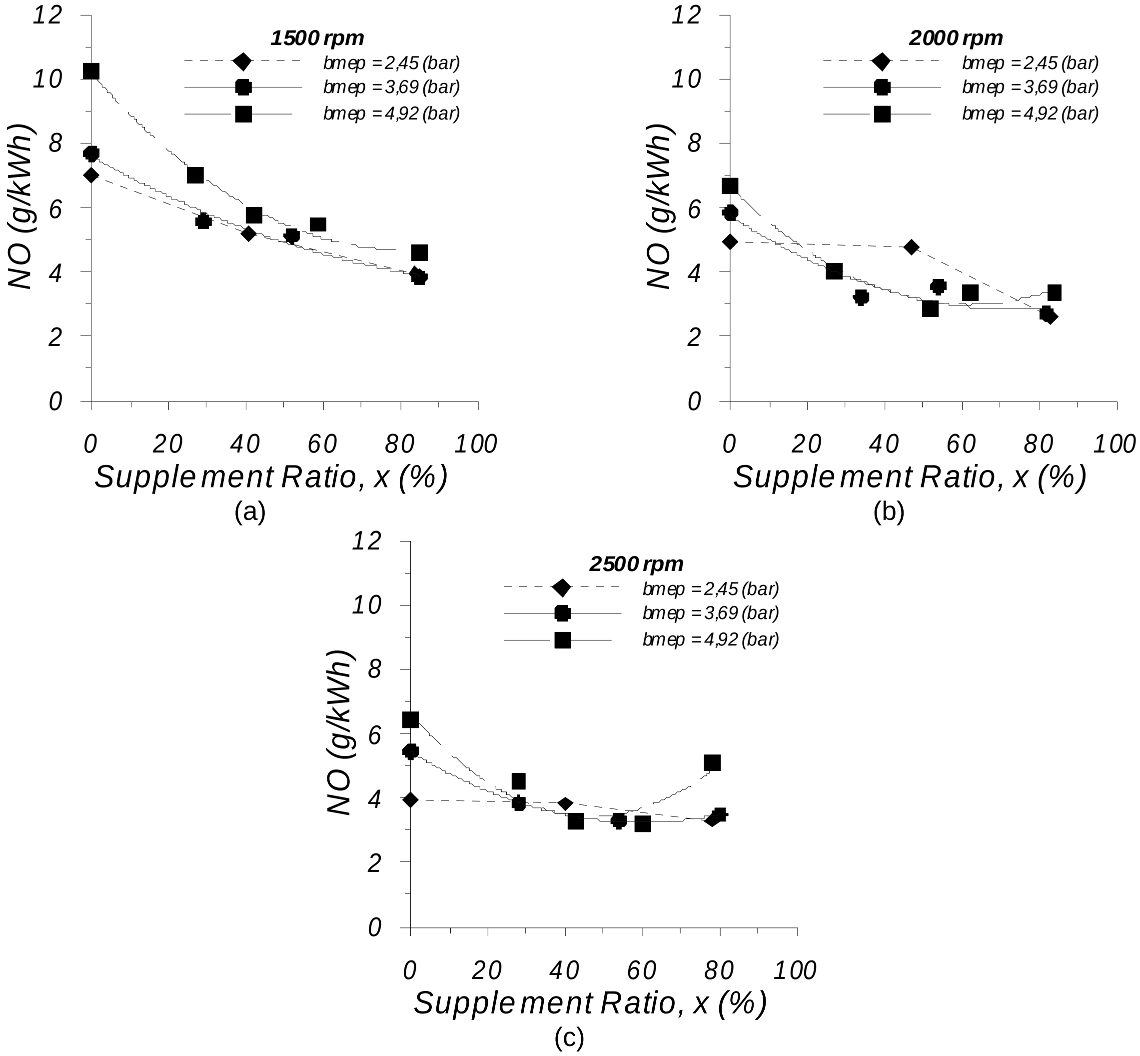


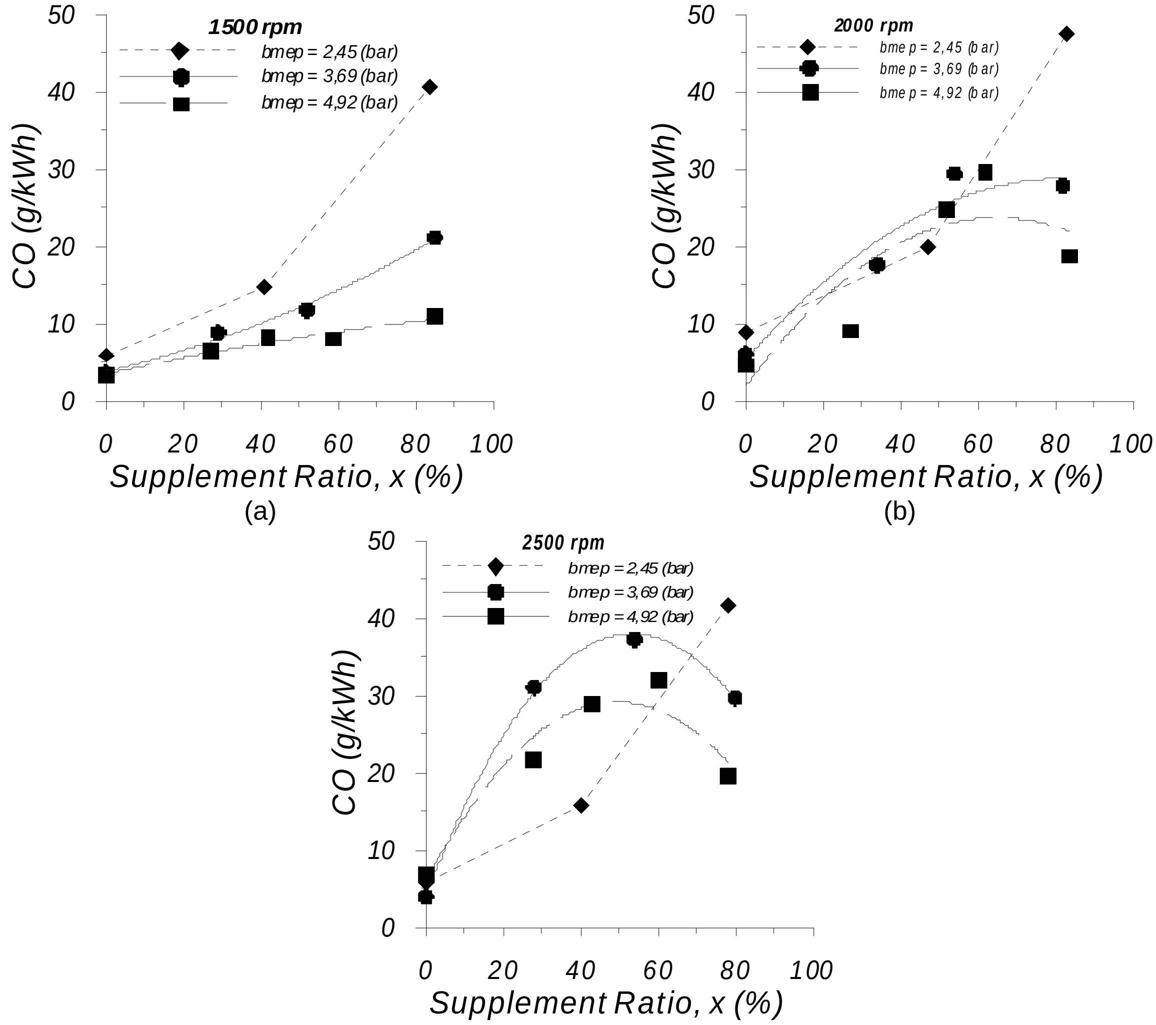
FIGURE 9. Variation of specific NO emissions as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

Carbon Monoxide (CO) Emissions

Figures 10 (a, b and c) provide the variation of the specific carbon monoxide concentration as a function of the supplement ratio for 2.45, 3,69 and 4.92 bar brake mean effective pressure at 1500, 2000 and 2500 rpm engine speed, respectively. As known [28-32], the rate of CO formation is a function of the relative air/fuel ratio, of the unburned gaseous fuel availability and also of the cylinder charge temperature, both of which control the rate of fuel decomposition and oxidation. Observing these figures, 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 one under normal diesel operation. Under normal diesel operation and for all engine operating points, the fuel-lean ($\lambda >> 1$) nature of the mixture leads to extremely low specific CO emissions. Under dual fuel operation and for the same brake mean effective pressure, the increase of the gaseous fuel concentration in the charge results to a decrease of the total air excess ratio favoring thus the CO formation.

At part load, the increase of the gaseous fuel amount, keeping load constant, leads to an increase of carbon monoxide concentration and this becomes more evident at high engine speed and high supplement ratios. This is due to the slow combustion rate of the gaseous fuel, which maintains the cylinder charge temperature at low levels resulting in a reduction of the oxidation process of carbon monoxide.

At high load, the increase of the supplement ratio causes a more intense increase of CO emissions compared to the one observed at low load, especially for intermediate and high engine speed, while for gaseous fuel concentration values beyond a certain value, the emitted CO starts to decrease probably as a result of improvement of the gaseous fuel combustion. At high load and high engine speed, specific CO emission decreases considerably as supplement ratio increases. This is the result of improvement of the gaseous fuel utilization, especially during the diffused controlled combustion phase.



(c)

FIGURE 10. Variation of specific CO emissions as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

Unburned Hydrocarbon (HC) Emissions

The variation of the specific unburned hydrocarbon concentration with diesel fuel supplement ratio is given in figures 11 (a, b and c), for various combinations of engine load (i.e. 2.45 bar, 3.69 bar and 4.92 bar brake mean effective pressure) and engine speed (i.e. 1500, 2000 and 2500 rpm). As known [28-32], the variation of unburned hydrocarbons in the exhaust gases depends on the quality of the combustion process occurring inside the cylinder chamber. Under dual fuel operating mode, the combustion process is affected considerably by the total air excess ratio (λ) since this specific factor plays a significant role on the flame propagation mechanism. Examining these figures, it is observed that at each combination of load and engine speed, the emitted HC concentration measured under dual fuel operating mode is higher compared to the respective one observed under normal diesel operation, while this difference becomes more intense at low load and high supplement ratios.

At high load, as supplement ratio increases the unburned HC emission increases slightly until a certain limit where the concentration of the emitted unburned HC starts to decrease. This is due to the slight improvement of the gaseous fuel combustion process, since the total air excess ratio tends to stoichiometry favoring, thus, the flame propagation mechanism, which also plays a significant role on the unburned HC oxidation rate. But in any case, HC emission values observed under dual fuel operation are considerably higher compared to normal diesel operation. The increase of engine speed from 1500 to 2500 rpm does not seem to have a significant effect on the emitted HC concentration since similar results are observed.

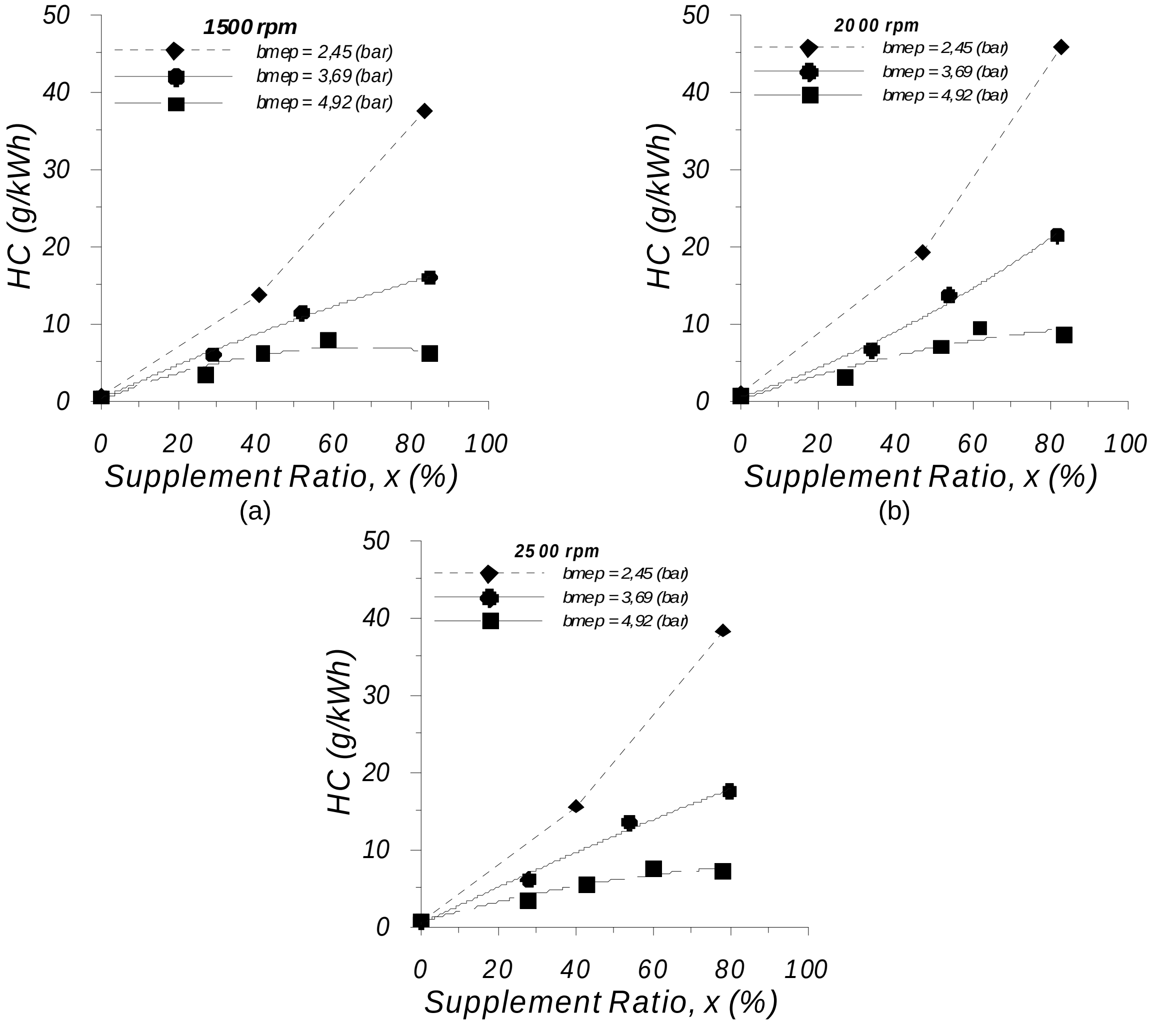


FIGURE 11. Variation of specific HC emissions as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

Soot Emissions

Figures 12 (a, b and c) provide the measured values of smoke density as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressures at 1500, 2000 and 2500 rpm engine speeds, respectively.

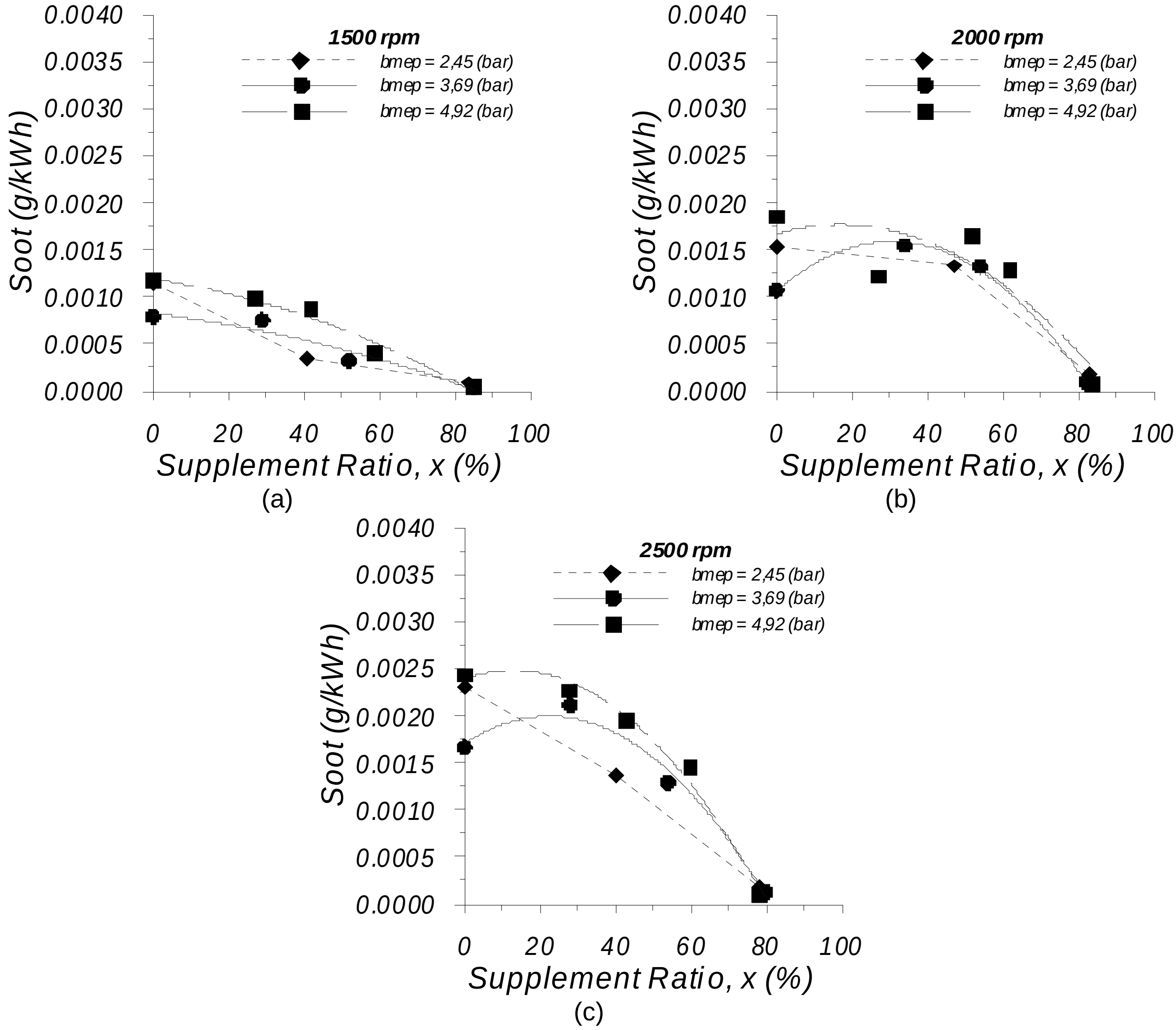


FIGURE 12. Variation of soot density as function of supplement ratio for 2.45, 3.69 and 4.92 bar brake mean effective pressure at (a) 1500 rpm, (b) 2000 rpm and (c) 2500 rpm engine speed.

Examining these figures we observe that dual fuel operation is a potential way of reducing soot emissions. Specifically, for all engine speed examined it is revealed that at low load as the percentage of liquid fuel replacement increases, soot concentration decreases sharply since less liquid fuel is injected on a percentage basis and thus less soot is formed.

At high engine load and low supplement ratios, soot emissions tend to converge to the respective ones under normal diesel operation and in some cases they become even slightly higher. This is due to the fact that, despite the slight 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 temperature observed during the premixed controlled combustion phase. As the percentage of the liquid diesel fuel replacement increases, the reduction of soot emission becomes more evident compared to the one observed at low percentages, since the improvement of the gaseous fuel combustion rate results in higher cylinder charge temperature during the expansion stroke, which, eventually, promotes soot oxidation rate contributing, thus, to a further decrease of soot concentration. The reduction of soot with supplement ratio appears to be steeper at higher engine speeds. In general, dual fuel (natural gas-diesel) operating mode seems to be an efficient way for reducing soot concentration at almost all engine operating conditions. The main reason is probably that natural gas, whose methane is the main constituent, being the lower member in the paraffin family, has very small tendency to produce soot.

CONCLUSIONS

To understand the combustion mechanism under dual fuel operating mode, an extended experimental investigation has been conducted on a high speed, direct injection, single cylinder, test engine located at the author's laboratory. 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 load and engine speed under both normal diesel and dual fuel operating modes. From the analysis of experimental data it is revealed that in comparison with normal diesel operation, dual fuel operation results to:

- lower burning rate during the premixed controlled combustion phase, which results in lower maximum cylinder pressure. This is extremely encouraging, since the use of dual fuel operating mode on a conventional DI diesel engine does not seem to affect significantly the engine structure.
- longer duration of combustion especially at low and intermediate supplement ratios. At high load, duration of combustion increases when increasing the concentration of the gaseous fuel but at high supplement ratios it converges to the respective one observed under normal diesel operation. At high engine speed and high engine load conditions, the increase of the supplement ratio results to shorter duration of combustion to the respective one observed under normal diesel operation.
- a negligible variation of the exhaust gas temperature, especially at low engine speed. But at high engine speed and for all engine load examined, as supplement ratio increases the exhaust gas temperature increases until a certain limit where it starts to decrease. Thus, at extremely high supplement ratios and for the same engine operating point (i.e. load and engine speed), exhaust gas temperature is slightly lower compared to the respective one under normal diesel operation.
- higher ignition delay period. Under dual fuel operation, increasing the percentage of liquid fuel replacement increases the ignition delay considerably. This is due to the reduction of the cylinder charge temperature close to the point of the liquid fuel injection, since under dual fuel operation the cylinder charge has higher overall specific heat capacity compared to the respective one under normal diesel operation.
- higher total brake specific energy consumption. Concerning engine efficiency it is revealed that for the same engine operating point (i.e. load and engine speed), as supplement ratio increases, engine efficiency becomes inferior compared to the respective one observed under normal diesel operation. This effect becomes more evident at part load, while at high load the increase of gaseous fuel supplement percentage leads to a more slight increase of the brake specific energy consumption compared to the one observed at part load, due to the improvement of the gaseous fuel utilization.
- lower specific NO concentration. At high load and for all engine speed examined, the positive effect of the supplement ratio increment on specific NO emissions becomes more evident compared to the one observed at low load conditions. But, at high load and high engine speed, the increase of supplement ratio to extremely high values leads to an increase of specific

NO emissions which may in some cases be higher than the respective one observed under normal diesel operation.

- a substantial increase of the specific CO emissions. At low load the increase becomes more evident compared to the one observed at high load, since at high engine load and high supplement ratios a decrease is observed. But in any test case examined, the specific CO emissions under dual fuel operation are higher compared to the one observed under normal diesel operation.
- an increase of the specific HC emissions. At low load the increase of the gaseous fuel supplement ratio leads to a substantial increase in specific HC emissions. But at high load the specific HC emissions continue to increase with increasing the replacement ratios and beyond a certain value they start to decrease.
- lower soot concentration. The positive effect is stronger at extremely high supplement ratios where it is observed a drastic decrease in soot emissions as compared to normal diesel operation for all engine operating conditions (i.e. load and engine speed) examined.

Taking into account all the above mentioned, it is revealed that dual fuel combustion using natural gas as a supplement for liquid 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 pollutants, NO and Soot, in DI diesel engines. The observed disadvantages concerning engine efficiency, HC and CO can be possibly mitigated by applying modifications on the engine tuning, i.e. injection timing of liquid diesel fuel mainly at part loads. Theoretical results for such an investigation have been published in the past by the present research group [24].

NOMENCLATURE

h	specific enthalpy, J/kg
m	mass, kg
$\dot{m}$	mass flow rate, kg/s
N	engine speed, (rpm)
P	pressure, Pa
Q	heat transfer to walls, J
R	specific gas constant, J/kg K
T	absolute temperature, K
U	total internal energy, J
V	volume, m^3
x	supplement ratio, %

Greek symbols

λ	total air excess ratio or thermal conductivity, W/mK
ρ	density, kg/m^3
φ	crank angle degrees

Subscripts

D	Diesel
NG	Natural Gas
g	gas
w	wall

Dimensionless number

Re	Reynolds

Abbreviations

B	before
BDC	bottom dead center
BSEC	brake specific energy consumption, MJ/kWh
bmep	brake mean effective pressure, bar
CA	degrees of crank angle
COV	coefficient of variance, %
D	diesel
DI	direct injection
deg	degrees of crank angle
FNGD	fumigated natural gas diesel engine
LHV	lower heating value
NDO	normal diesel operation
NG	natural gas
ROHR	rate of heat release
rpm	revolutions per minute
TDC	top dead centre

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