The Impact of Total Equivalence Ratio on Environmental Behavior of a Natural Gas Dual Fuel Diesel Engine

Roussos G. Papagiannakis and Dimitrios T. Hountalas

***Abstract* —**Towards the reduction of exhaust emissions from diesel engines, automotive engineers have proposed various solutions, one of which is the use of natural gas as a supplement for the commercial liquid diesel fuel (Dual Fuel Operation-DFO). In a dual fuel compression ignition engine, auto-ignition of the injected diesel fuel provides ignition centers for turbulent flame propagation throughout the lean homogeneous gaseous fuel-air mixture. Engine performance, for any quantity of diesel fuel improves with the increased admission of natural gas. This improvement appears to be dependent on the total equivalence ratio. In the present work, experimental results are provided concerning the effect of the total equivalence ratio on performance characteristics and environmental behavior of an existing compression ignition engine modified to operate under diesel-natural gas DFO operating mode at various combinations of load and engine speed. By comparing the results, an important effect of the total equivalence ratio on some critical engine performance parameters (i.e. cylinder pressure and total heat release traces, ignition delay period, duration of combustion, brake efficiency, NO, CO, unburnt HC and soot emissions) is observed. The main objective of this comparative assessment is to record and to comparatively evaluate the relative impact each one of the total equivalence ratio, brake torque and engine speed, on engine performance characteristics and emitted pollutants. Furthermore, the present investigation deals with the determining of optimum combinations between the aforementioned parameters referred before since at high brake torque and high engine speed, the simultaneous increase of the total equivalence ratio may lead in undesirable results about engine performance characteristics. The conclusions of the specific investigation will be extremely valuable for the application of this technology on existing DI diesel engines.

***Keywords* —** dual fuel engine; natural gas; equivalence ratio.

I. Introduction

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 [1-8]. Natural gas being an environmentally friendly fuel has emerged as an alternative fuel to potentially replace conventional diesel fuel in some applications to meet the stringent emission regulations while maintaining the performance of the diesel counterpart. Dual fuel engines are generally more efficient than conventional spark-ignited natural gas engines due to the higher compression ratio and lean operation. Another reason for the interest in the investigation of dual fuel engines is due to the increasingly stringent regulation of NOx and particulate emissions (soot). Dual fuel engines have been employed in some mobile sources such as fleet vehicles, heavy-duty trucks, buses, railway locomotives, and marine vessels and in construction and agricultural field applications. However, application in on-highway trucks has not proved very promising due to the low number of refueling stations available and poor light load performance. Dual fuel engines, as the name implies, operate on two fuels, primarily gaseous fuel which is typically natural gas and a secondary liquid fuel like diesel. Most of the energy comes from the primary fuel, which has a high self-ignition temperature. The liquid fuel, which has a relatively low self-ignition temperature, initiates the combustion process in the cylinder through timed injection. Unlike that of spark ignited natural gas engines, dual fuel engines have multiple ignition sources, which results in more complete combustion of charge compared to a single spark in the spark ignited natural gas engine. Due to the high compression ratios of the compression ignition engines, dual fuel mode of engine operation achieves high fuel conversion efficiencies operating on leaner air-fuel mixtures. Dual fuel operation has the potential to provide operational characteristics that are comparable and perhaps superior to those of the conventional diesel or spark ignition engines. The oxides of nitrogen known as NOx and soot are reduced considerably in dual fuel operation. This superior performance may be achieved only if effective measures can be ensured both for the avoidance of knock at high loads and

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incomplete combustion of gaseous fuel at light loads [1-8].

A number of experimental and theoretical investigations concerning the dual fuel diesel – natural gas operating mode have been reported in the international literature [7-30]. Various researchers [7-30] have published extensive theoretical and experimental investigations concerning the combustion processes occurring inside the combustion chamber of a 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 [31-35].

A primary objective of the present work is to examine the effect of the total equivalence ratio under various combinations of brake torque and engine speeds, primarily from the viewpoint of some critical engine performance characteristics 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 diesel engine properly modified to operate under dual fuel operating mode. The engine is supplied with natural gas from the public 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 brake torque, 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 total equivalence ratio 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. 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 total equivalence ratio increases. Furthermore, under dual fuel operating mode, the effect of total equivalence ratio in combination with engine operating conditions (i.e. brake torque 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 normal diesel operation.

II. Experimental Facilities and Test Cases Examined

A. *Experimental Installation*

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 with a bowl in piston combustion chamber. A schematic layout of the test installation used is shown in Figure 1. The engine has a bore of 85.73 mm, a stroke of 82.55 mm and a connecting rod length of 180 mm. The compression ratio is 17.6 and the normal operation speed range is between 1000 and 3000 rpm. A high-pressure fuel – pump, having an 6.5-mm diameter plunger is connected to a three-hole injector nozzle (each hole having diameter of 0.23 mm) which is located in the middle of the combustion head. The injector nozzle opening pressure is 180 bar. The engine is coupled to a Heenan & Froude hydraulic dynamometer. Other technical data of the engine are given in Table 1.

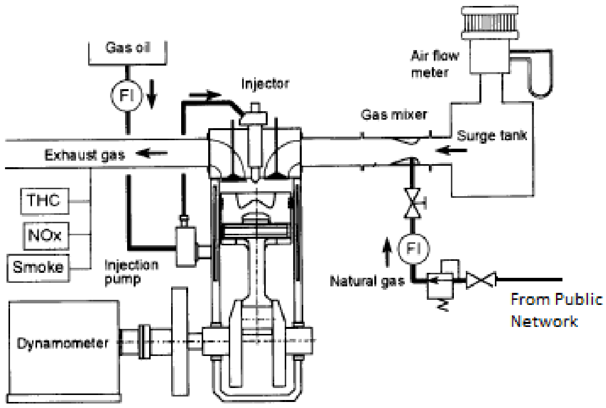

Fig. 1 Schematic layout of the test installation

Table 1. Basic data for Lister LV1 test engine

Type of Engine	Single Cylinder, 4-Stroke,DI
Cylinder Dead Volume	28.03cm^3
Inlet Valve Opening	15^{o}CA before TDC
Inlet Valve Closure	41^{o}CA after BDC
Exhaust Valve Opening	41^{o}CA before BDC
Exhaust Valve Closure	15^{o}CA after TDC
Inlet Valve Diameter	34.5mm
Exhaust Valve Diameter	31.5mm
Static Injection Timing	26^{o}CA before TDC

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 [31-35]. 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.

The main characteristics of the instrumentation used to measure pollutant emission values are given in Table 2.

Table 2. Instrumentation for pollutant emissions measurements

Pollutant	Instrumentation	Type	Operating Principal
Soot	Bosch	RTT100	Optical
NO	Signal	4000 Series	Chemilumescent
HC	Ratfisch	RS55	Flame Ionization Detector
CO	Signal	7200 Series	Non-Dispersive Infrared-Detector

Under normal diesel (NDO) or dual fuel (DFO) operating mode, at each combination of 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. The main properties of the liquid diesel fuel used are given in Table 3.

Table 3. 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)*

B. Test Cases Examined

Measurements have been taken at three different engine loads corresponding to 9.4 Nm, 14 Nm and 18.6 Nm torque, and three engine speeds of 1500, 2000 and 2500 rpm under both normal diesel (NDO) and dual fuel (DFO) operating modes. Under dual fuel operation and for each combination of load and engine speed, measurements have been taken for various mass flow rates of diesel and natural gas. In Figure 2 is given the variation of the total fuel equivalence ratio as a function of the percentage of diesel fuel supplement from natural gas for various combinations of brake torque and engine speed. 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, an attempt was made, in the present investigation, 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. 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.

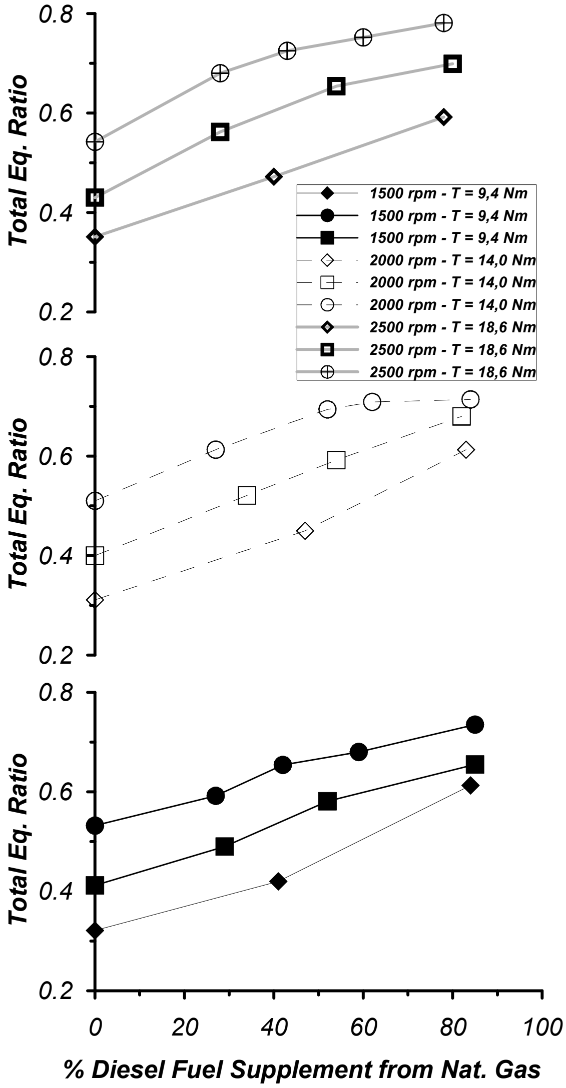

Fig. 2 Total fuel equivalence ratio as a function of the percentage of diesel fuel supplement from natural gas for various combinations of brake torque and engine speed

Furthermore, the desired engine load (i.e. brake torque) is controlled by changing the amounts of the fuels used. Specifically, an amount of liquid diesel fuel is provided to achieve a percentage of the desired brake torque, while the rest of the percentage of the desired brake torque is reached by using only natural gas fumigated into the intake air resulting to the reduction of the inhaled combustion air. Thus, 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, resulting to the alteration of the total fuel equivalence ratio. Taking into account all the aforementioned characteristics, for each one of the examined operating point, the total fuel equivalence ratio (φ_{tot}) is given by the formula:

$$\varphi = \frac{\dot{m}_{NG} \cdot AFR_{st}^{NG} + \dot{m}_{D} \cdot AFR_{st}^{D}}{\dot{m}_{IA}} \qquad (1)$$

where (AFR_{st}^{D}), (AFR_{st}^{NG}) correspond to the stoichiometric air to fuel ratio (by mass) for diesel and natural gas, respectively, ($\dot{m}_{D}$) and ($\dot{m}_{NG}$) represent diesel fuel and natural gas consumptions respectively and ($\dot{m}_{IA}$) represent the mass of the inducted air.

C. Estimation of the Total Combustion Rate

For each operating condition, forty engine pressure cycles are recorded and from these the mean cylinder pressure trace is obtained. Using the measured cylinder pressure diagrams and the TDC pickup signal the ignition delay, combustion duration and the total rate of heat release is estimated. Combustion duration and intensity are estimated from the total heat release rate, which is important information for the combustion mechanism in diesel engines. The heat release rate diagram also provides us with valuable information for the initial stage of combustion where most of the NO is formed. The net heat release rate is determined by applying the first thermodynamic law as follows [36-38]:

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

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

$$\frac{dQ_{gross}}{d\phi} = \frac{dQ_{net}}{d\phi} - \frac{dQ_{w}}{d\phi} \qquad (3)$$

where the heat loss rate ($dQ_w / d\phi$) (negative from gas to walls) is obtained by using the heat transfer model of Annand [39] which requires the cylinder gas temperature (T_g). For this reason, it is assumed that the cylinder content behaves as a perfect gas and the mean cylinder gas temperature is obtained using the ideal gas low where the cylinder pressure is obtained from the mean cylinder pressure trace, the specific gas constant is calculated from the mean gas composition estimated from the initially trapped mass and the fuel burned up to the current engine crank angle and the trapped mass at inlet valve closure is estimated from an open cycle simulation of the engine [36-37], using the measured mass flow rate of air and gaseous fuel. Using the previous methodology, we obtain a good estimation 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 [36-38].

D. 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 4. Considering these values, it is revealed that the measurements are quite repeatable, especially concerning engine performance. As far as pollutant emissions are concerned, for the gaseous ones the COV is less than 3.5% while for soot it is less than 3.9%. To achieve this, the instruments used were calibrated (using reference gas samples for NO, CO and HC) before each measurement. The measurement accuracy and repeatability is adequate to derive sound conclusions concerning the effect of dual fuel combustion on engine performance and emissions.

Table 4. Coefficient of Variance for the Measured Quantities

Measured Quantity	COV (%)
Maximum Combustion Pressure	1.2
Brake Specific Fuel Consumption	0.9
Nitric Oxide	3.5
Carbon Monoxide	2.9
Unburned Hydrocarbons	3.2
Soot	3.9

III. Results and Discussion

As already mentioned, the main scope of the present work is to examine the effect of total equivalence ratio in combination with engine load and engine speed, on performance and exhaust emissions of a naturally aspirated, direct injection, compression ignition engine operating under normal diesel (NDO) and dual fuel (DFO) operating modes, under ambient intake temperature. According to international bibliography [1-8], at each combination of brake torque and engine speed the percentage of liquid diesel fuel replacement by natural gas is one of the most interesting parameter that affects seriously the total equivalence ratio, which is a critical parameter that affects seriously the formation mechanism of some basic exhaust emissions (i.e. NO, CO, Soot etc.). Thus, in the present work and for all engine operating points (brake torque and engine speed) examined, experimental results are provided for normal diesel operation (NDO) and also for dual fuel (DFO) operating modes for various total equivalence ratios. Furthermore, it must be stated here that the gaseous pollutant emissions were not adjusted for the intake air humidity level since during the experimental procedure the absolute humidity level was almost the same.

A. Cylinder Pressure and Total Heat Release Rate Data

Figures 3-8 provide the experimental cylinder pressure and total heat release traces for various values of total equivalence ratio, at low (i.e. 9,4 Nm brake torque) and high (i.e. 18,6 Nm brake torque) loads for 1500, 2000 and 2500 rpm engine speeds. For each engine operating point (i.e. combination of brake torque and engine speed) results are provided for normal diesel operation indicated by NDO characterization and also results are provided for two characteristic test cases of dual fuel operation indicated by DFO characterization. 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. Thus, under DFO operating mode and for all engine operating points (i.e. brake torque and engine speed) examined, the increase of the total equivalence ratio affects slightly the value of the cylinder pressure compared to the respective one observed under NDO one. 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 natural gas during the premixed controlled combustion phase. Moreover, for all engine operating points (i.e. brake torque and engine speed) examined, 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 while the peak of the cylinder pressure occurs later compared to the respective value observed with lower values of total equivalence ratio. 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.

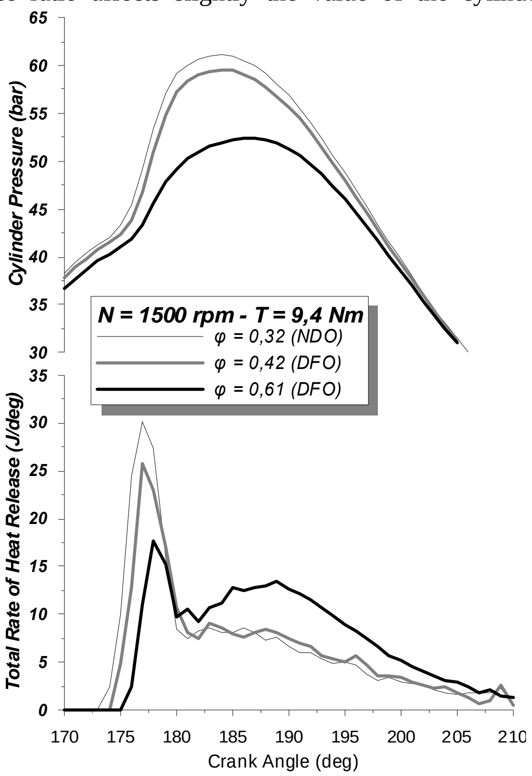

Fig. 3 Experimental pressure and total heat release traces under NDO and DFO modes for 1500 rpm engine speed at 9,4 Nm torque.

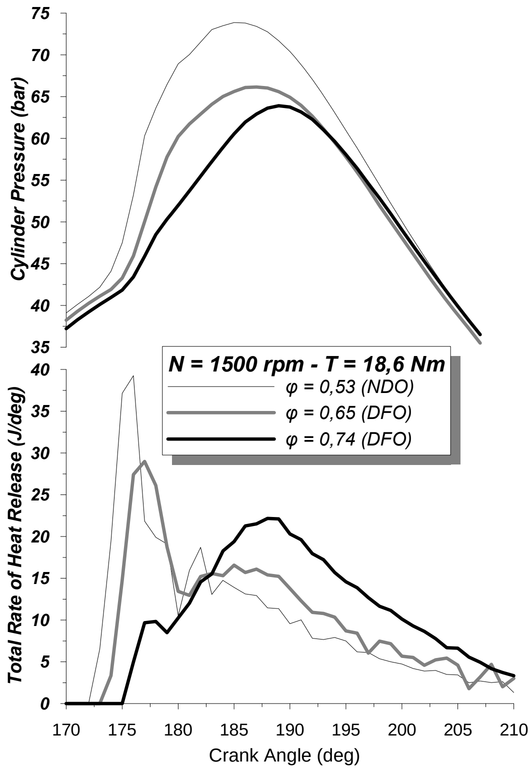

Fig. 4 Experimental pressure and total heat release traces under NDO and DFO modes for 1500 rpm engine speed at 18,6 Nm torque.

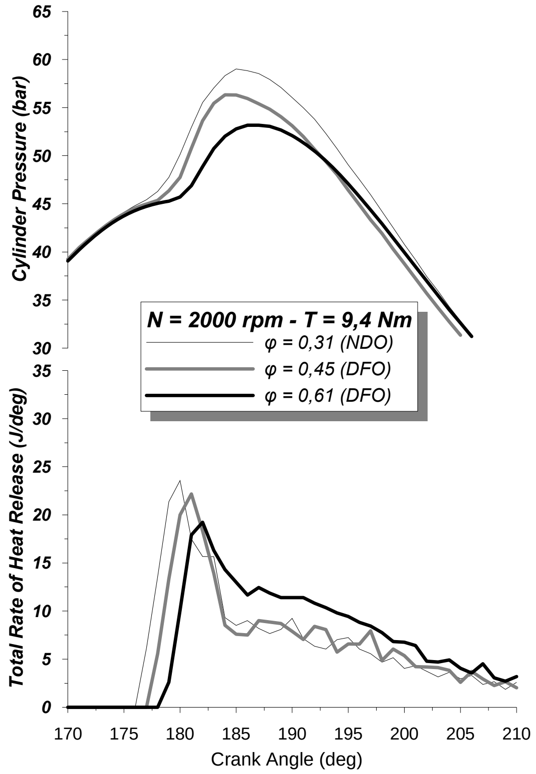

Fig. 5 Experimental pressure and total heat release traces under NDO and DFO modes for 2000 rpm engine speed at 9,4 Nm torque.

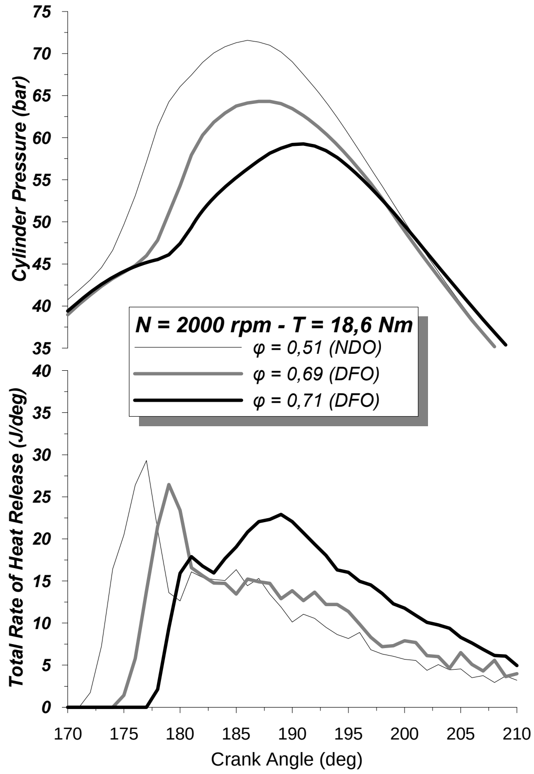

Fig. 6 Experimental pressure and total heat release traces under NDO and DFO modes for 2000 rpm engine speed at 18,6 Nm torque.

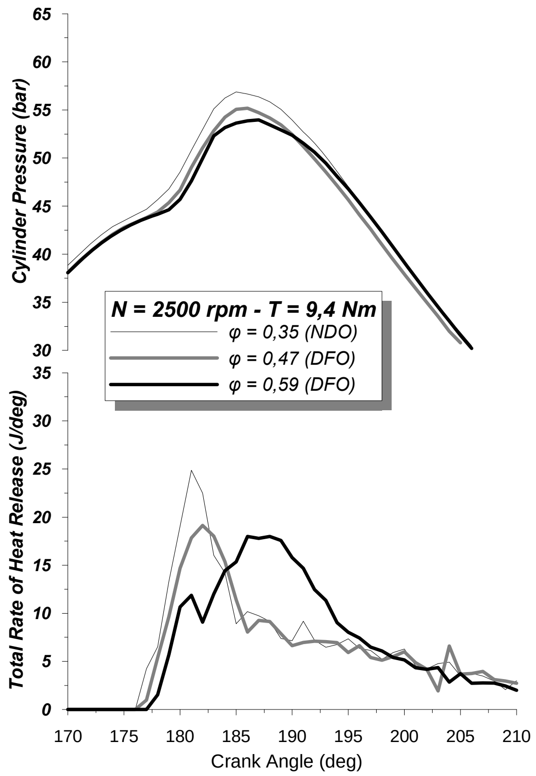

Fig. 7 Experimental pressure and total heat release traces under NDO and DFO modes for 2500 rpm engine speed at 9,4 Nm torque.

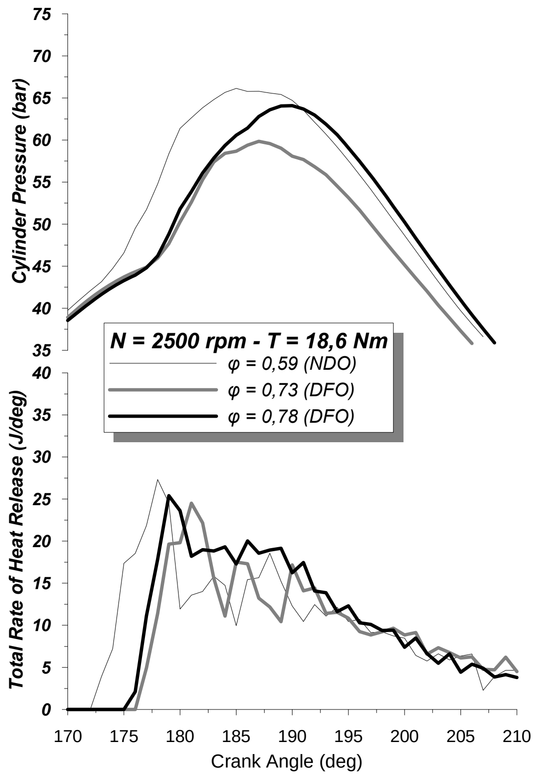

Fig. 8 Experimental pressure and total heat release traces under NDO and DFO modes for 2500 rpm engine speed at 18,6 Nm torque.

At high brake torque and high engine speed, as total equivalence ratio tends to the stoichiometric value, there is a considerable increase of the rate of the cylinder pressure rise especially during the initial stages of combustion. The increase of engine speed leads to a warmer engine and to an increase of the turbulence inside the combustion chamber while an increase of the total equivalence ratio favors the existence and the fast spread of the flame front. The aforementioned 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). 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 DFO operating modes 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 NDO 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 natural gas 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. 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 higher compared to the respective one observed under normal diesel operation. This is the result of 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. However, this does not have any significant effect on the cylinder pressure, since the major part of the gaseous fuel diffused controlled combustion occurs during the expansion stroke.

B. *Peak Combustion Pressure*

Figure 9 provides the variation of the duration of combustion as a function of the total equivalence ratio (φ) for 9,4 Nm, 14 Nm and 18,6 Nm brake torque at 1500, 2000 and 2500 rpm engine speed, respectively. It must be stated here that each one of the three circles indicated with NDO includes the peak cylinder pressure values observed under normal diesel operation for each one of the three brake torques examined at 1500, 2000 and 2500 rpm engine speed, respectively. Since the maximum cylinder pressure is a critical parameter concerning the resistance of engine structure, the effect of natural gas concentration on maximum cylinder pressure is a great of interest.

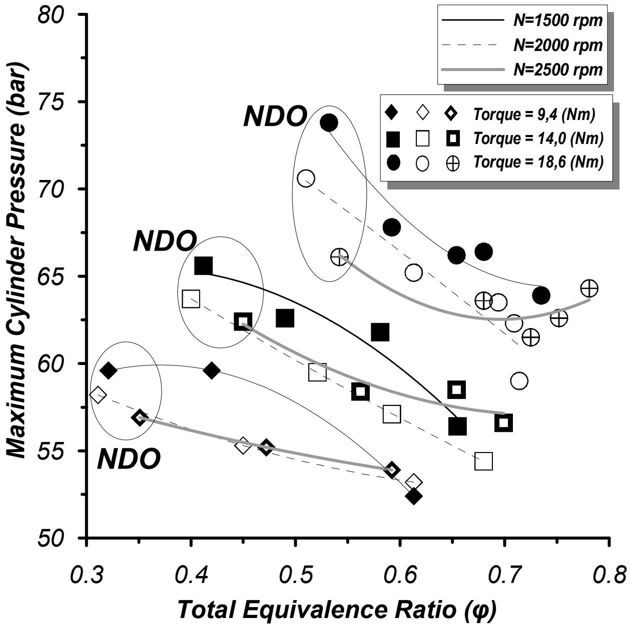

Fig. 9 Maximum Cylinder Pressure vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

Observing this figure it is revealed that under DFO operating mode, the presence of natural gas in the cylinder charge affects positively the maximum cylinder pressure. Specifically, for low and intermediate brake torques and for all engine speeds examined, the increase of the total equivalence ratio, leads to a decrease of the maximum cylinder pressure observed. At high load the maximum cylinder pressure starts to decrease with the increase of total equivalence ratio in the cylinder charge while a further increase of the total equivalence ratio beyond a certain value of $\varphi \geq 0,7$ leads to a slight increase of the maximum combustion pressure. The effect becomes more evident at high engine speed. However, tt must be stated here that for the same engine speed the slope each one of the brake torque curves being almost the same for the entire range of the total fuel equivalence ratios examined. Eventually, 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.

C. *Ignition Delay Period*

The variation of ignition delay period as a function of the total equivalence ratio (φ) for various combinations of brake torque (i.e. 9,4 Nm, 14 Nm and 18,6 Nm) and engine speed (i.e. 1500, 2000 and 2500 rpm) is given in figure 10. 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 [38, 40]. Examining this figure it is shown that under NDO operating mode the increase of the engine load, keeping constant engine speed, results in a slight decrease of the ignition delay period.

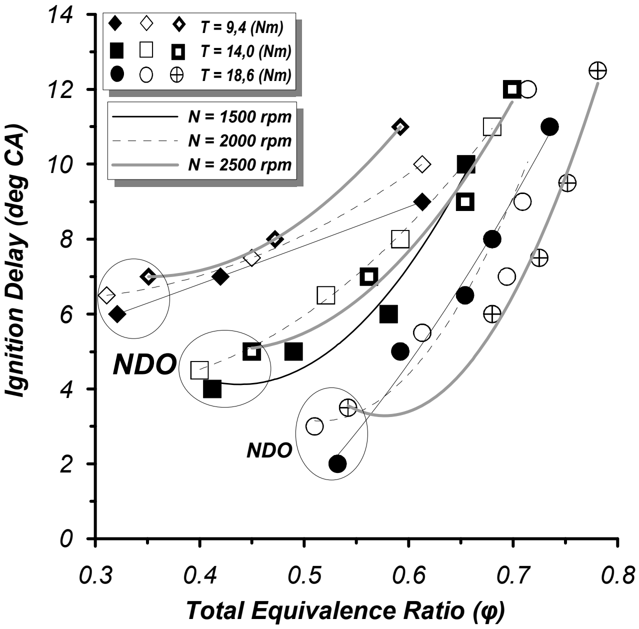

Fig. 10 Ignition Delay Period vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

Under DFO operating mode, the presence of natural gas in the cylinder charge affects the ignition point of the liquid diesel fuel. Thus, for all engine operating points (i.e. brake torque and engine speed) examined, the increase of the natural gas concentration in the cylinder charge resulting in an increase of the total equivalence ratio, leads to an increase of the ignition delay period of the injected liquid 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 operation. 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. The effect is stronger at high brake torque and high engine speed where the rate of ignition delay period rise with total equivalence ratio becomes more evident as compared to the respective one observed at low and intermediate brake torque values. Moreover, for each brake torque, keeping constant the total equivalence ratio the increase of engine speed seems to have a negligible effect on the ignition delay period.

D. Duration of Combustion

Figure 11 provides the variation of the duration of combustion as a function of the total equivalence ratio (φ) for 9,4 Nm, 14 Nm and 18,6 Nm brake torque at 1500, 2000 and 2500 rpm engine speed, respectively. Examining this figure, it is observed that at low brake torque and for all engine speeds examined, the increase of the total equivalence ratio caused by the increase of natural gas concentration in the cylinder charge leads to longer durations of combustion as compared to the respective ones observed under NDO operating modes.

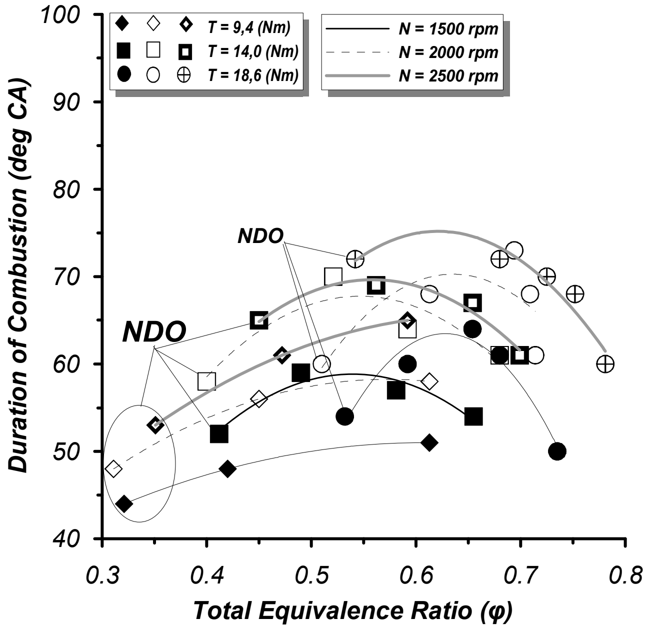

Fig. 11 Duration of Combustion vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

At high brake torque and for all engine speeds examined, the duration of combustion increases with increasing total equivalence ratio and beyond a certain value of (φ_{tot}) it starts to decrease, as a result of the high cylinder charge temperature and the faster combustion rate of the natural gas. The effect becomes more evident for values of $\varphi \geq 0,7$ where the warmer engine in combination with the high natural gas equivalence ratio has a positive effect on the existence and the fast spread of the flame front surrounding the burning zone. Thus at specific operating conditions the increase of total equivalence ratio beyond of a critical limit results to a considerable improvement of the gaseous fuel combustion quality which, in many cases, may lead to shorter duration of combustion compared to the respective one observed under NDO operating mode.

E. Total Brake Specific Energy Consumption

The variation of the total brake specific energy consumption with total equivalence ratio is given in figure 12, for various combinations of brake torque (i.e. 9,4 Nm, 14 Nm and 18,6 Nm) and engine speed (i.e. 1500, 2000 and 2500 rpm). In each one of the three circles indicated with NDO are summarized three values of BSEC corresponding to normal diesel operation (NDO). 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 natural gas in the charge mixture. Examining this figure, it is revealed that for all engine operating points examined the engine efficiency under dual fuel operation (DFO) is lower compared to the respective one under normal diesel (NDO).

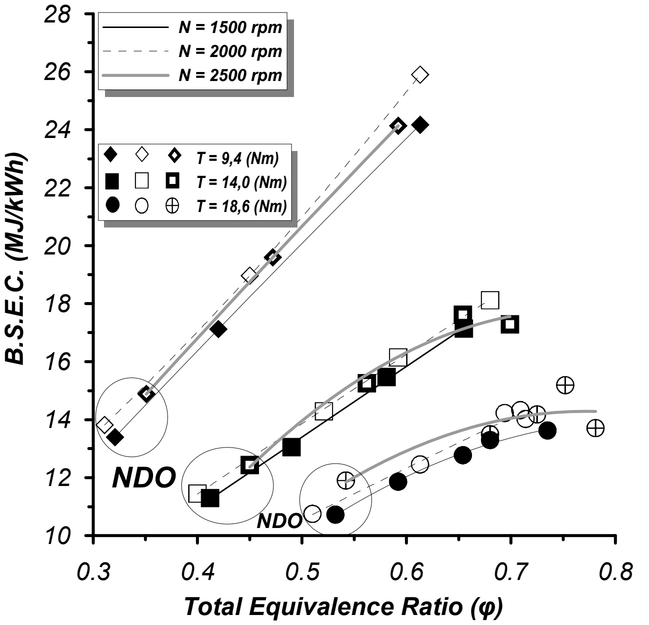

Fig. 12 Brake Specific Energy Consumption vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

At low brake torque and for all engine speed examined, the increase of the total equivalence ratio (φ) results to a considerable increase of the total brake specific energy consumption. At high brake torque and for all engine speed examined, the increment of the total equivalence ratio (φ) leads initially to a slight increase of the total brake specific energy consumption, while a further increase of the total equivalence ratio beyond a critical value ($\varphi \simeq 0{,}7$) results to a slight improvement of the engine efficiency. This is the result of the considerable improvement of the natural gas combustion quality. Under DFO operating mode at each brake torque, keeping constant equivalence ratio, the increase of engine speed seems to have a negligible effect on the engine efficiency.

F. Nitric Oxide (NO) Emissions

Figure 13 provides the variation of the specific nitric oxide concentration as a function of the total equivalence ratio (φ) for 9,4 Nm, 14 Nm and 18,6 Nm brake torque at 1500, 2000 and 2500 rpm engine speed, respectively. As well recognized [38, 40], the formation of nitric oxides is favored, in general, by high oxygen concentration and high charge temperature. Examining this figure, NO emission is affected considerably by the presence of natural gas in the charge mixture. In general, NO concentration observed under DFO mode is lower compared to the one observed under NDO operation at the same engine operating conditions (engine speed, brake torque).

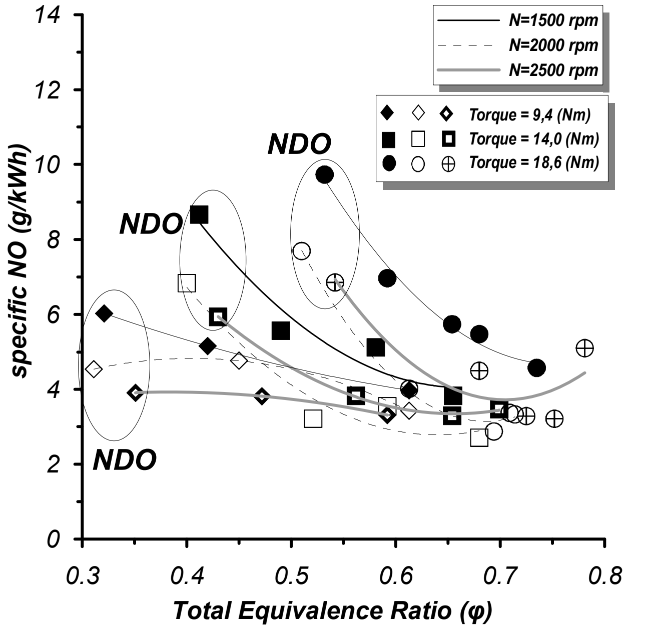

Fig. 13 Specific NO emissions vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

Specifically, at low brake torque and for all engine speed examined, there is a slight decrease of NO emissions with the increase of the total equivalence ratio (φ). At high brake torque, there is a considerable decrease of NO emissions with increased total equivalence ratios until a certain limit where the trend of nitric oxide reduction tends to decrease. A further increase of the total equivalence ratio leads to a slight increase of NO. This becomes more evident at high engine speed and for total equivalence ratios beyond of 0,7.

G. Carbon Monoxide (CO) Emissions

Figure 14 provides the variation of the specific carbon monoxide concentration as function of the total equivalence ratio (φ) for 9,4 Nm, 14 Nm and 18,6 Nm brake torque at 1500, 2000 and 2500 rpm engine speed, respectively. As known [38, 40], 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 this figure, it is revealed that for each engine operating point (i.e. each combination of brake torque and engine speed), CO emissions under DFO operation are significantly higher compared to the respective one under NDO operation mode. Under normal diesel operation and for all engine operating points, the increase of total equivalence ratio leads to lower specific CO emissions. Under DFO operating mode and for the same brake torque, the increase of the total equivalence ratio accompanied with an increase in natural gas concentration in the cylinder charge results in higher CO emissions since the decrease of the total air excess ratio favors the CO formation mechanism.

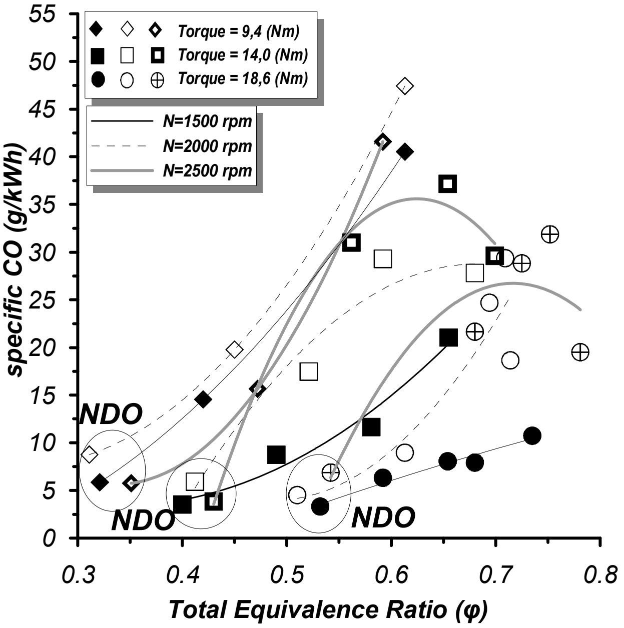

Fig. 14 Specific CO emissions vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

This becomes more evident at low brake torque and for all engine speeds examined where the slow combustion rate of natural gas observed, maintains the cylinder charge temperature at low levels resulting in a reduction of the oxidation process of carbon monoxide. At high brake torque, the increase of the total equivalence ratio causes a more slight increase of CO emissions compared to the one observed at low brake torque, while for total equivalence ratio values beyond a critical value ($\varphi \simeq 0,7$), the emitted CO starts to decrease probably as a result of improvement of natural gas combustion quality. The aforementioned trend becomes more evident at high engine speed. This is the result of a considerable improvement of the gaseous fuel utilization, especially during the diffused controlled combustion phase.

H. Unburned Hydrocarbon (HC) Emissions

The variation of the specific unburned hydrocarbon concentration with total equivalence ratio is given in figure 15, for various combinations of brake torque (i.e. 9,4 Nm, 14 Nm and 18,6 Nm) and engine speed (i.e. 1500, 2000 and 2500 rpm). As known [38, 40], the variation of unburned hydrocarbons in the exhaust gases depends on the quality of the combustion process occurring inside the cylinder chamber. Under DFO operating mode, the combustion process is affected considerably by the total air excess ratio ($\lambda = \left(1/\varphi\right)$) since this specific factor plays a significant role on the flame propagation mechanism.

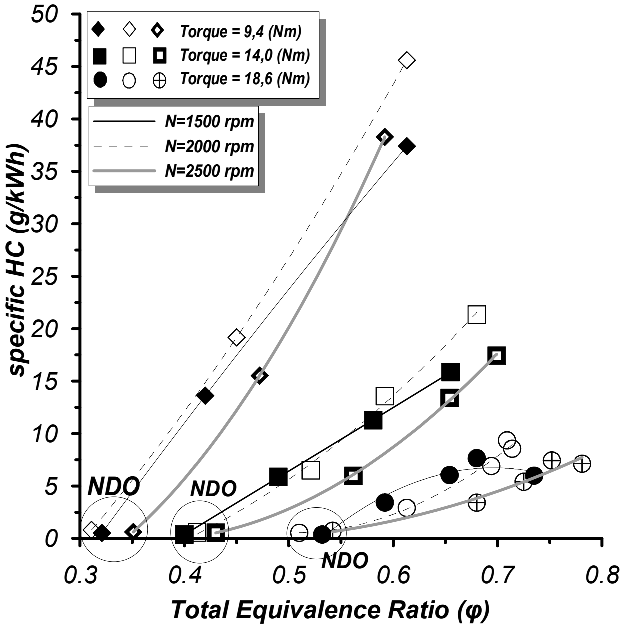

Fig. 15 Specific HC emissions vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

Examining these figures, it is observed that at each combination of brake torque and engine speed, the emitted HC concentration measured under dual fuel operating mode is considerable higher compared to the respective one observed under normal diesel operation. The specific difference becomes more intense at low brake torque. At high load, as total equivalence ratio increases the unburned HC emission increases slightly until a critical limit where the concentration of the emitted unburned HC starts to decrease. This is due to the slight improvement of the natural gas 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. For each combination of brake torque and total equivalence ratio, 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.

I. Soot Emissions

Figure 16 provides the measured values of smoke density function of the total equivalence ratio (φ) for 9,4 Nm, 14 Nm and 18,6 Nm brake torque at 1500, 2000 and 2500 rpm engine speed, respectively. Examining this figure we observe that dual fuel natural gas/diesel operation is a potential way of reducing soot emissions. Specifically, for all engine speed examined it is revealed that at low load as the total equivalence ratio increases, soot concentration decreases. This is due to the fact that, despite the lower quality of the natural gas combustion, the concentration of the soot formed is lower than the respective one observed under normal diesel operation since less liquid fuel is injected on a percentage basis and thus less soot is formed.

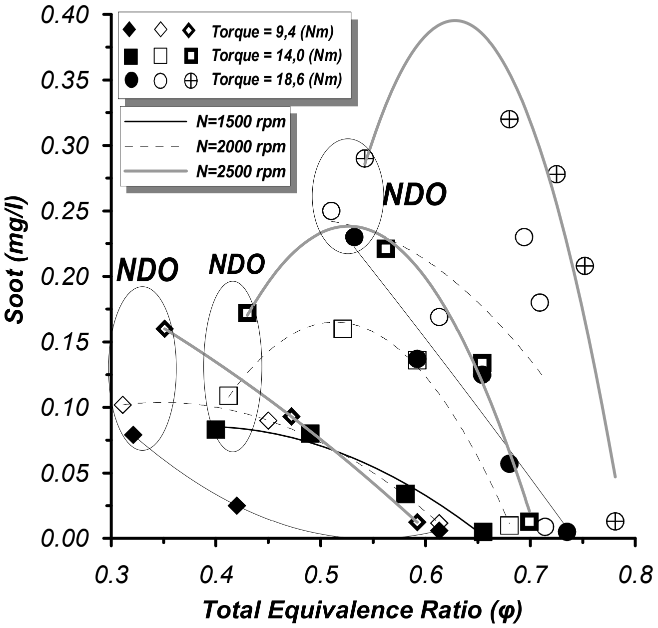

Fig. 16 Soot emissions vs Total Equivalence Ratio under NDO and DFO modes for various combinations of brake torque and engine speeds.

However, at high brake torque and high engine speed, as total equivalence ratio increases soot emissions increase slightly until a critical limit of ($\varphi \simeq 0,7$), where a further increase leads to a considerable decrease of soot emissions. This is due to the fact that, 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 sharp decrease of soot concentration. The reduction of soot with total equivalence ratio for the same value of brake torque 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.

IV. Conclusions

To understand the main engine performance characteristics and the environmental behavior of an existing compression ignition engine modified to operate under dual fuel (natural gas – diesel) operating mode, an extended experimental investigation has been conducted in the past by the authors. 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 brake torque and engine speed for various values of total equivalence ratios. From the analysis of experimental data it is revealed that the increase of the total equivalence ratio caused by an increase of natural gas concentration in the cylinder charge, dual fuel operation results to:

- lower burning rate, especially 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 an existing diesel engine does not seem to affect significantly the engine structure.
- longer duration of combustion. This effect becomes more evident at low brake torque and for all engine speeds. At high brake torque and for all engine speeds examined, the increase of the total equivalence ratio leads initially to a slight longer duration of combustion while a further increase of total equivalence ratio beyond of 0,7 results to considerable decrease of duration of combustion.
- higher ignition delay period. The effect becomes more evident at high total equivalence ratios and high engine speeds.
- higher total brake specific energy consumption. The effect becomes more evident at low brake torque, while at high brake torque the increase of total equivalence ratio leads to a more slight increase of the brake specific energy consumption compared to the one observed at low brake torque. As the total equivalence ratio tends to stoichiometry the engine efficiency is improving slightly.
- lower specific NO concentration. At high brake torque and for all engine speed examined, the positive effect of the total equivalence ratio increment on specific NO emissions becomes more evident compared to the one observed at low brake torque conditions. But, at high brake torque, the increase of total equivalence ratio to values beyond of 0.7, leads to a slight increase of specific NO emissions.
- an increase of the specific CO emissions. At low brake torque the increase becomes more evident compared to the one observed at high brake torque. 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 total equivalence ratio leads to a substantial increase in specific HC emissions. But at high load the specific HC emissions continue to increase with increasing the values of total equivalence ratio and beyond a certain value they start to decrease slightly.
- lower soot concentration. The positive effect is stronger at high values of total equivalence ratio and high brake torque conditions where it is observed a drastic decrease in soot emissions.

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. The observed disadvantages concerning engine efficiency, HC and CO can be possibly mitigated by applying various modifications on the engine tuning, i.e. injection timing of liquid diesel fuel mainly at part loads.

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