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Comparative Evaluation of the Effect of Partial Substitution of Diesel Fuel by Natural Gas on Performance and Emissions of a Fumigated Dual Fuel Diesel Engine

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ABSTRACT

Reduction of exhaust emissions is a major research task in diesel engine development in view of increasing concerns regarding environmental protection and stringent exhaust gas regulations. Simultaneous reduction of NOx emissions and particulate matter is quite difficult due to the soot/NOx 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. 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. sulphur compounds). The present contribution is mainly concerned, with an experimental investigation of the characteristics of dual fuel operation when liquid diesel is partially replaced with natural gas under ambient intake temperature in a DI diesel engine. In the present work the results from this investigation concerning data are given for performance and exhaust emissions. Furthermore, through this experimental work the effect of liquid fuel percentage replacement by natural gas i.e. various values of gaseous fuel / total fuel mass ratios are examined. 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; performance; emissions

1. 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 [1-6]. Natural gas is fumi-

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gated into the intake air and premixed with the air during the induction stroke. 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 [3-6]. The specific type of dual fuel engine is referred to as "Fumigated Natural Gas Diesel Engine".For this kind of engine, 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, while the desired engine power output (i.e. engine load) is controlled by changing the amount of the gaseous fuel (natural gas) [3-6].

Various researchers [1-26] 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.

A primary objective of the present work is to examine the main characteristics of the dual fuel combustion under various engine operating conditions, 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 diesel engine properly modified to operate under dual fuel operating mode. From the analysis of the experimental measurements, important information is derived revealing the effect of engine load in combination with the natural gas concentration both on the combustion mechanism occurring inside the combustion chamber and on the formation of pollutant emissions (NO, CO, HC and Soot). 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 so that the engine operation becomes friendlier to the environment, without deteriorating its performance compared to that of conventional operation.

2. Experimental section

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 [22-26].

Other technical data of the engine are given in Table 1. In Fig.1 an analytical schematic layout of the test installation used is presented. As shown, the adjustment of gaseous fuel supply is accomplished through a control valve located after the flow-meter. Afterwards, the gaseous fuel flows towards the intake of the engine and is mixed with the intake air. The main instrumentation of the test installation used, is:

Table 1. Basic data of Lister LV1 diesel engine

Type	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

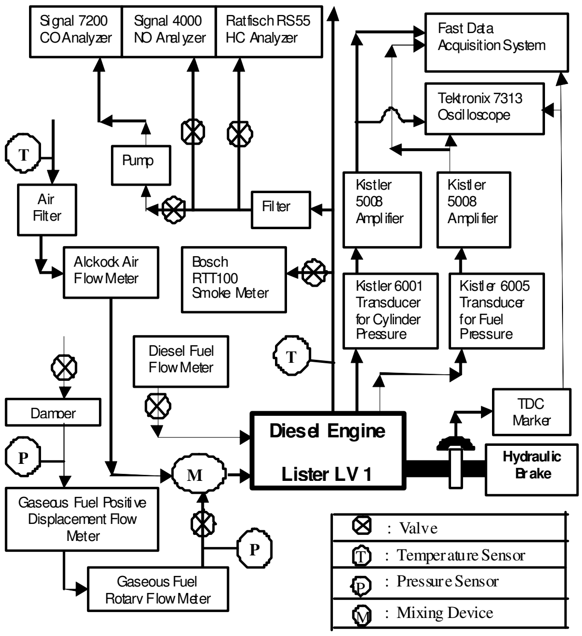

Fig.1: Schematic layout of the test installation.

- An Alckock (viscous type) air flow meter.
- A rotary displacement gaseous fuel flow meter.
- Tank and flow meter for the liquid diesel fuel.
- Temperature sensors for the exhaust gas, inlet air and gaseous fuel.
- T.D.C. marker (magnetic pick-up).
- Kistler piezoelectric transducer for the main chamber coupled to Kistler charge amplifier and connected to a high speed sampling system.
- Kistler piezoelectric transducer for measuring the fuel – line pressure just before the injector.
- A fast data-acquisition and recording system was used to record the pressure diagrams obtained by the piezoelectric transducers. At each engine operating point (i.e. load and natural gas concentration) 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.
- A Bosch RTT100 smoke meter is used to measure smoke levels in the exhaust gas. NO emissions are measured using a chemilumescent analyzer (type: Signal 4000) while HC emissions are measured with a flame-ionization detector (type: Ratfisch RS55). The last two devices were fitted with thermostatically controlled heated lines. CO is measured with a nondispersive infra-red analyzer (type: Signal 7200).

The main properties of the fuels (i.e. liquid diesel fuel and gaseous natural gas) used are given in Table 2.

Measurements are 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 operation (100% diesel fuel) and dual fuel operation (natural gas and diesel fuel). In the present work, due to the lack of space, results are presented only for the 1500 rpm engine speed case. Under dual fuel operation part of the liquid fuel is replaced by gaseous so as to maintain the power output of the engine the same as the normal diesel operation at the specific operating point (load, speed). The procedure is as follows: At a given engine speed after having reached a certain power output with diesel fuel only, its flow rate is kept constant and the engine power output is further increased using gaseous fuel until the desired value is reached. Details for the experimental test cases examined are given in Table 3.

Table 2. 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)

Table 3. Test Cases Examined.

bmep (bar)	1500 RPM					2000 RPM					2500 RPM				
	Brake Power (kW)	**m_D (kg/h)**	**m_{NG} (kg/h)**	**λ (-)**	**x (%)**	**Brake Power (kW)**	**m_D (kg/h)**	**m_{NG} (kg/h**	**λ (-)**	**x (%)**	**Brake Power (kW)**	**m_D (kg/h)**	**m_{NG} (kg/h**	**λ (-)**	**x (%)**
2.45	1.47	0,46	0	3.12	0	1.95	0,61	0	3.22	0	2.42	0,86	0	2.85	0
2.45	1.47	0,34	0,23	2.38	41	1.95	0,46	0,39	2.22	47	2.42	0,63	0,42	2.12	40
2.45	1.47	0.12	0.62	1.63	84	1.95	0.18	0.85	1.63	83	2.42	0.25	0.97	1.69	78
3.69	2.20	0,58	0	2.43	0	2.93	0,79	0	2.5	0	3.64	1,06	0	2.22	0
3.69	2.20	0,46	0,19	2.04	29	2.93	0,61	0,33	1.92	34	3.64	0,86	0,36	1.78	28
3.69	2.20	0,34	0,39	1.72	52	2.93	0,46	0,56	1.69	54	3.64	0,63	0,76	1.53	54
3.69	2.20	0.11	0.68	1.53	86	2.93	0.18	0.92	1.47	82	3.64	0.25	1.07	1.53	80
4.92	2.94	0,74	0	1.88	0	3.90	0,98	0	1.96	0	4.87	1,36	0	1.69	0
4.92	2.94	0,58	0,21	1.69	27	3.90	0,79	0,30	1.63	27	4.87	1,06	0,42	1.47	28
4.92	2.94	0,46	0,35	1.53	42	3.90	0,61	0,63	1.44	52	4.87	0,86	0,66	1.38	43
4.92	2.94	0,34	0,50	1.47	59	3.90	0,46	0,76	1.41	62	4.87	0,63	0,97	1.28	60
4.92	2.94	0.12	0.76	1.36	86	3.90	0.18	0.97	1.4	84	4.87	0.25	1.10	1.42	78

The mass flow rate of Natural Gas divided by the total fuel (Diesel and Natural Gas) mass flow rates represents the "supplement ratio" (x), i.e.

$$x=\left\{\dot{m}_{NG}\big/\left(\dot{m}_D+\dot{m}_{NG}\right)\right\}\cdot 100\ (\%) \qquad (1)$$

For each engine operating point (i.e. load and engine speed), the total burning rate is estimated by using the mean cylinder indicator diagram. The estimated heat release rate is the total one due to the combustion of both the liquid fuel and the gaseous one [22-26]. 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 [28-29]. For each engine operating mode, 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.

Table 4. COV 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

3. Results and discussion

3.1. Cylinder pressure and total heat release rate data

Figures 2a-b provide the experimental pressure and total heat release traces for 1500 rpm engine speed, at 2.45 and 4.92 bar brake mean effective pressure, under normal diesel and dual fuel natural gas-diesel 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), 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 re sult of the poor combustion of the ga

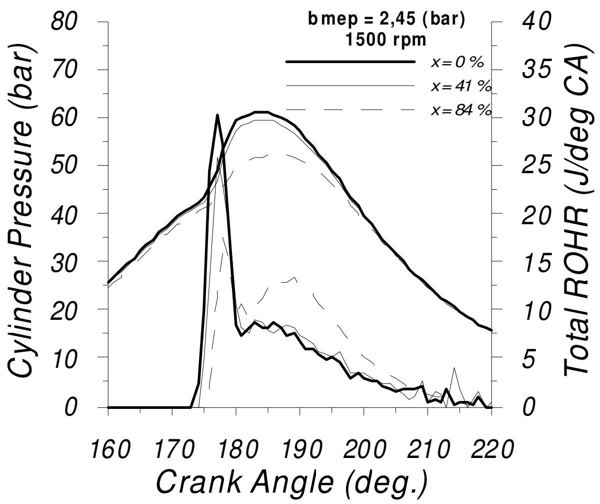

Fig.2a: Experimental pressure and heat release traces under normal diesel and dual fuel operating modes at 2.45 bmep and 1500 rpm engine speed..

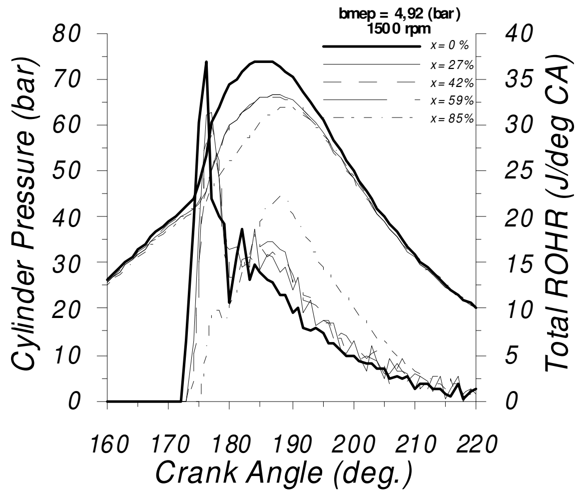

Fig.2b: Experimental pressure and heat release traces under normal diesel and dual fuel operating modes at 4.92 bmep and 1500 rpm engine speed.

seous fuel during the premixed controlled combustion phase. 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

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 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 conditions. 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 considerably 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. 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.

3.2. Maximum cylinder pressure

Fig.3 provides 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 engine speed. Observing this fig. it is revealed that under dual fuel operating mode the increase of the supplement ratio, keeping engine load constant, leads to a significant decrease of the maximum cylinder pressure. The slope each one of the load curves being almost the same for the entire range of the supplement ratios examined.

At part load the decrease of the maximum cylinder pressure becomes more evident at low supplement ratios.

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.

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. Thus, no danger exists for the engine structure associated to cylinder pressure, if the specific technology is to be applied on conventional diesel engines.

3.3. Exhaust gas temperature

The variation of the measured exhaust gas temperature with diesel fuel supplement ratio is given in fig.4. 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, 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. At high engine load, as the percentage of liquid fuel replacement increases, there is a slight 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.

3.4. 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 [28-29]. The variation of ignition delay period as a function of the supplement ratio is given in fig.5.

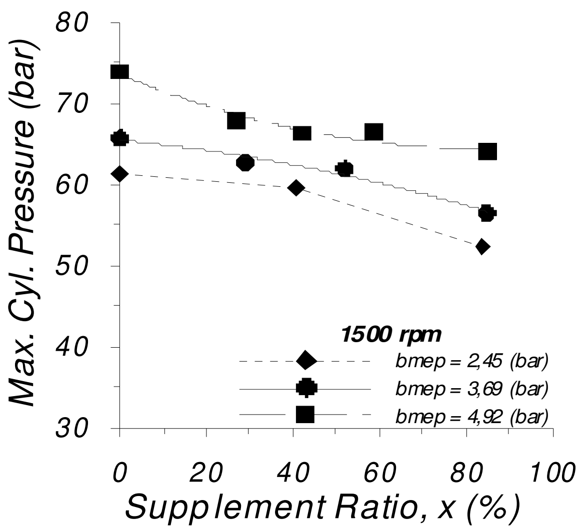

Fig.3 Maximum Cylinder Pressure variation as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

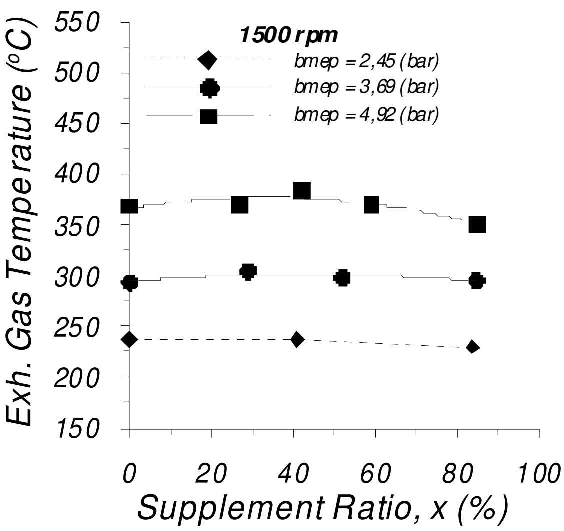

Fig.4: Exhaust Gas Temperature variation as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

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, results in a considerable decrease of the ignition delay period. Under dual fuel operating mode (x≠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 by 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 [6]. 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, 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.

3.5. Duration of combustion

Fig.6 provides 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 respectively.

Observing this figure, it is revealed that at low load conditions, 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.

At high load, 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.

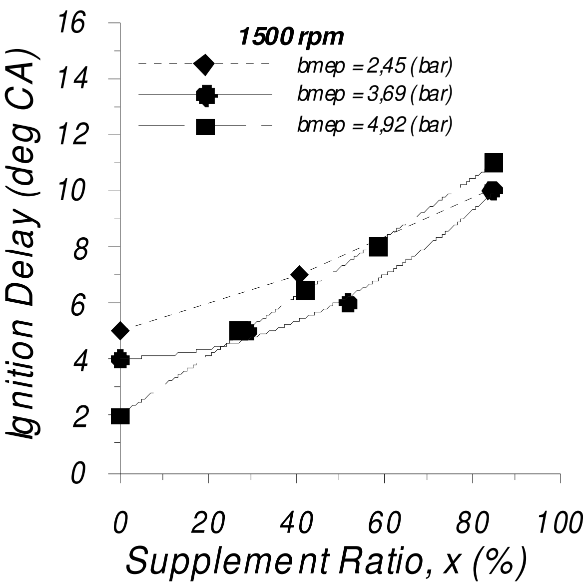

Fig.5: Ignition Delay period as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

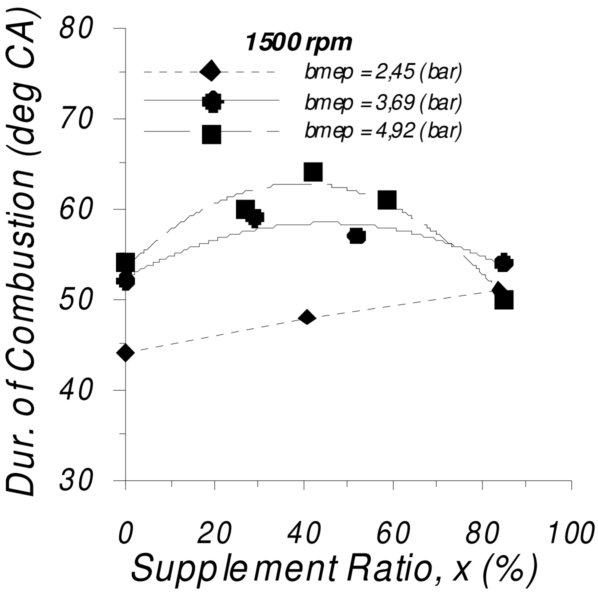

Fig.6: Duration of Combustion as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

3.6. Total brake specific energy consumption

The variation of the total brake specific energy consumption with diesel fuel supplement ratio is given in fig.7.

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 total brake specific energy consumption is affected considerably by the presence of the gaseous fuel in the charge mixture.

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 in an increase of the total brake specific energy consumption. The increase becomes more evident at extremely high supplement ratios, 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, 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 in a slight improvement of the engine efficiency. This is the result of the improvement of the gaseous fuel utilization observed at high load.

3.7. Nitric oxide (NO) emissions

Fig.8 provides 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 rpm engine speed, respectively. As known [28-30], the formation of nitric oxides is favored, in general, by high oxygen concentration and high charge temperature.

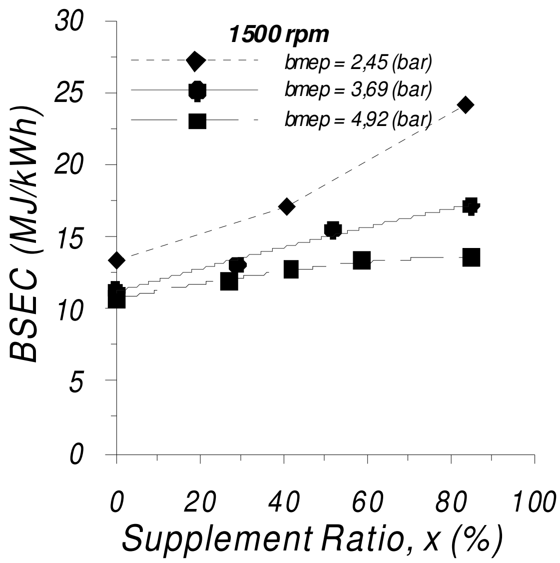

Fig.7: Total Brake Specific Energy Consumption as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

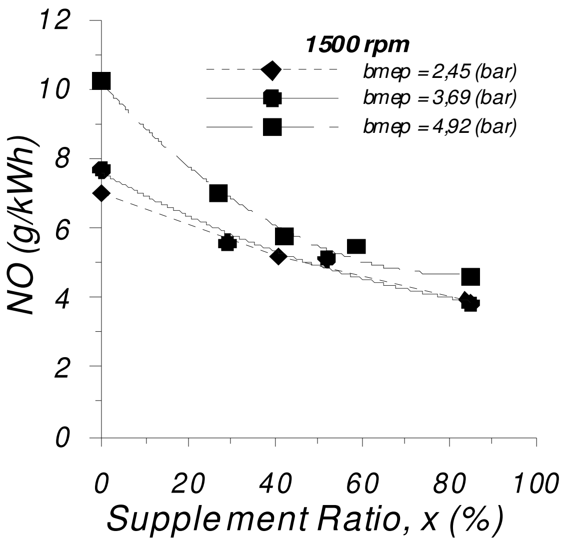

Fig.8: Specific nitric oxide concentration as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

Examining fig.8, 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 is lower compared to the one observed under normal diesel operation. 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. 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.

3.8. Carbon monoxide (CO) emissions

Fig.9 provides 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 rpm engine speed, respectively. As known [28-29], the rate of CO formation is a function of the total 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 fig.9, it is revealed that for the same operating point CO emissions under dual fuel operation are significantly higher compared to the respective one under normal diesel operation. Under normal diesel operation, the fuel-lean (λ>>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, leads to an increase of carbon monoxide concentration and this becomes more evident at 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, while for gaseous

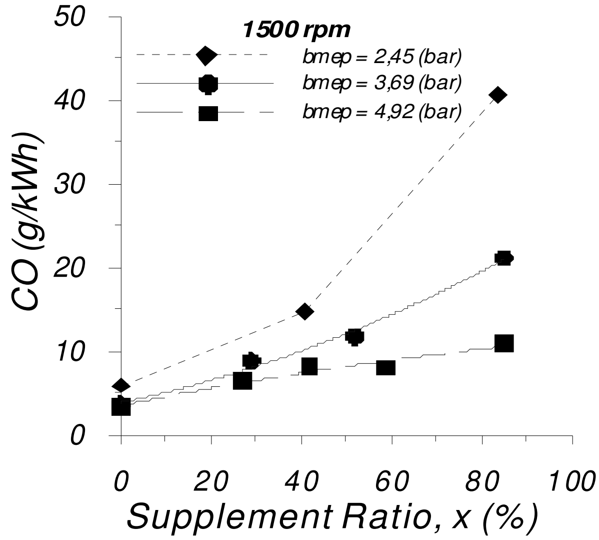

Fig.9: Specific carbon monoxide concentration as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

fuel concentration values beyond a certain value, the emitted CO starts to decrease probably as a result of improvement of the gaseous fuel combustion.

Unburned hydrocarbon (HC) emissions

The variation of the specific unburned hydrocarbon concentration with diesel fuel supplement ratio is given in fig.10. As known [28-29], 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 fig.10, it is revealed that at each load examined, 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

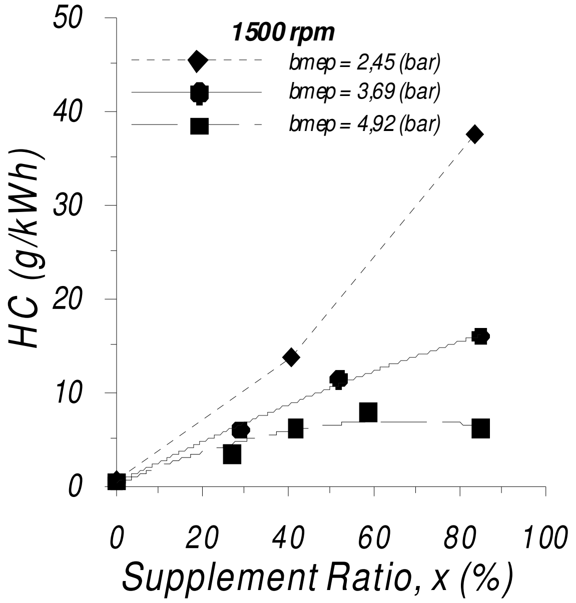

Fig.10: Specific unburned hydrocarbon concentration as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

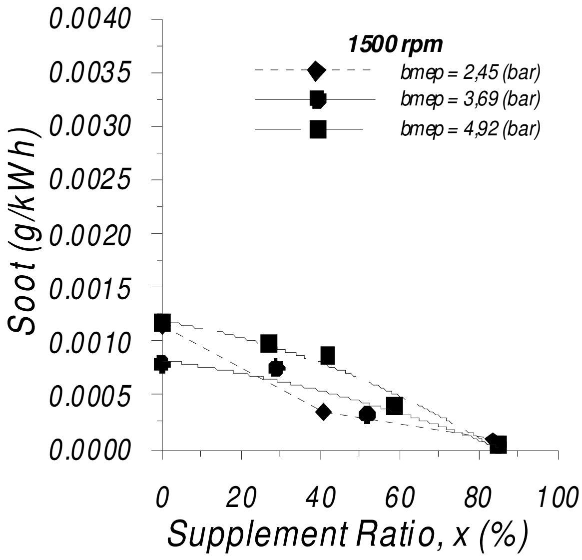

Fig.11: Smoke density as a function of the supplement ratio at various loads (bmep) for 1500 rpm engine speeds..

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 valuesobserved under dual fuel operation are considerably higher compared to normal diesel operation.

3.10. Soot emissions

Fig.11 provides 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 rpm engine speed. Examining this figure it is observed that dual fuel operation is a potential way of reducing soot emissions. Specifically, 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 provide 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.

4. 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. The engine has been properly modified to operate under dual fuel mode without changing its main configuration. From the analysis of experimental data it is revealed that in comparison with normal diesel operation, dual fuel operation results in:

- 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 provides converge to the respective one observed under normal diesel operation.
- A negligible variation of the exhaust gas temperature.
- Higher ignition delay period. Increasing the percentage of liquid fuel replacement increases the ignition delay considerably.
- Lower brake engine efficiency. Specifically, it is revealed that for the same load, 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 decrease of the engine efficiency compared to the one observed at part load.
- Lower specific NO concentration. At high load, the positive effect of the supplement ratio increment on specific NO emissions becomes more evident compared to the one observed at low load conditions.
- A substantial increase of the specific CO emissions. At low load the increase becomes more evident compared to the one observed at high load conditions. 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.

Taking into account all of 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.

5. Nomenclature

$\dot{m}$	mass flow rate, (kg/s)
x	diesel fuel supplementary ratio, (%)
λ	total air-fuel ratio excess ratio, (-)

Subscripts

D	Diesel
NG	Natural Gas

Abbreviations

BSEC	Total brake specific energy consumption, (MJ/kWh)
bmep	Brake mean effective pressure, (bar)
^{0}CA	Degrees of crank angle
CO	(specific) carbon monoxide, (g/kWh)
COV	Coefficient of variance, (%)
DI	Direct injection
HC	(specific) unburned hydrocarbons, (g/kWh)
LHV	Lower heating value, (kJ/kg)

NO (specific) nitric oxide, (g/kWh)

ROHR Total rate of heat release, (J/deg.)

rpm Revolutions per minute

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Biography

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