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Theoretical study of the effects of pilot fuel quantity and its injection timing on the performance and emissions of a dual fuel diesel engine

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Abstract

Various solutions have been proposed for improving the combustion process of conventional diesel engines and reducing the exhaust emissions without making serious modifications on the engine, one of which is the use of natural gas as a supplement for the conventional diesel fuel, the so called dual fuel natural gas diesel engines. The most common type of these is referred to as the pilot ignited natural gas diesel engine (PINGDE). Here, the primary fuel is natural gas that controls the engine power output, while the pilot diesel fuel injected near the end of the compression stroke auto-ignites and creates ignition sources for the surrounding gaseous fuel mixture to be burned. Previous research studies have shown that the main disadvantage of this dual fuel combustion is its negative impact on engine efficiency compared to the normal diesel operation, while carbon monoxide emissions are also increased. The pilot diesel fuel quantity and injection advance influence significantly the combustion mechanism. Then, in order to examine the effect of these two parameters on the performance and emissions, a comprehensive two-zone phenomenological model is employed and applied on a high-speed, pilot ignited, natural gas diesel engine located at the authors' laboratory. According to the results, the simultaneously increase of the pilot fuel quantity accompanied with an increase of its injection timing results to an improvement of the engine efficiency (increase) and of the emitted CO emissions (decrease) while it has a negative effect (increase) of NO emissions.

Keywords: Dual fuel diesel engine; Natural gas; Pilot ignition; Injection timing; Combustion; Emissions

1. Introduction

One of the main objectives for improving the combustion process of conventional internal combustion engines is to find effective ways to reduce exhaust emissions, without making serious modifications on their mechanical structure. Various solutions have been proposed, and among them the use of gaseous fuels posses a dominant place [1–4]. A good choice 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.

Many compression-ignition engines operate on the dual fuel principle. For these engines the primary fuel is normally a gaseous one, which is most usually inducted with the air during the induction stroke. One of the gaseous fuels used very commonly is natural gas, since its high auto-ignition temperature supports its use in conventional diesel engines. At the same point near top dead center, during the compression stroke, a small amount of diesel fuel is injected through the conventional diesel fuel system to act as a source of ignition for the compressed gaseous fuel–air mixture. The quantity of the pilot diesel fuel per injection is

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Nomenclature

m'_{mix}	mass flow rate of the air–gaseous fuel mixture at inlet valve closure (kg/h)
m'_{NG}	natural gas consumption rate (kg/h)
m'_{D}	diesel fuel consumption rate (kg/h)

Greek

λ	total air excess ratio

Abbreviations

AFR	air to fuel ratio (by mass)
B	before
BDC	bottom dead center
BSFC	brake specific fuel consumption
°CA	degrees of crank angle
CO	carbon monoxide
D	diesel
DI	direct injection
deg	degrees of crank angle
NG	natural gas
NPFQ	normal pilot fuel quantity
NIT	normal injection timing (°CA before TDC)
NO	nitric oxide
rpm	revolutions per minute
ppm	parts per million (by volume)
TDC	top dead center

usually fixed for a given engine; at full load, its amount represents less than 10% of the total fuel amount to the engine. In most dual fuel operating systems the engine power output is controlled by changing only the amount of the primary gaseous fuel added to the air during the induction stroke. Thus, at constant engine speed, the change of the amount of the gaseous fuel results to a change of the inducted combustion air since the total amount of the inducted mixture is kept constant. Most current dual fuel engines are made to operate either on the dual fuel principle with diesel ignition (pilot ignited natural gas diesel engine), or only on diesel fuel injection as normal diesel engines do [5].

Many research studies [5–10] have reported that, under dual fuel operation, the poor utilization of the gaseous fuel observed at low and intermediate loads results in poor engine performance (drop in engine efficiency) and in higher concentrations of carbon monoxide emissions compared to the respective values observed under normal diesel operation. At high load, the improvement of gaseous fuel utilization leads to a relevant improvement of both engine performance and carbon monoxide emissions, but their values continue to be inferior to the respective values observed under normal diesel operation.

According to many research works [5–16], injection timing and pilot diesel fuel quantity are some of the most important variables controlling the performance and exhaust emissions of a pilot ignited natural gas diesel engine. During the last years, the present research group has reported experimental investigations along with computer simulations conducted on such kind of engines [6,10–13]. In the present work, a theoretical investigation is conducted concerning the effect of both pilot fuel quantity and injection advance on the performance and emissions of a single cylinder, high speed, direct injection diesel engine modified to operate under dual fuel mode. The theoretical results have been obtained using a two-zone, phenomenological combustion model. The model has been improved and modified accordingly in order to describe and understand better the combustion process that takes place in pilot ignited natural gas diesel engines; some of the model results have been presented in the past [10–13]. As basis for the validation of the model, experimental results are used concerning the engine behavior under dual fuel mode at fixed pilot fuel quantity and normal injection advance, for various engine loads at constant engine speed.

From the theoretical findings, important information is derived revealing the effect of both parameters on engine performance and pollutant emissions. This is accomplished through the comparison of the calculated maximum cylinder pressure and total brake specific fuel consumption for various combinations of pilot fuel quantity and injection timings, at various loads for the same engine speed. Furthermore, the effect of both parameters is revealed on the formation of pollutant emissions, by comparing the related values to the corresponding ones obtained under normal pilot fuel quantity and normal injection timing. This information is extremely valuable, if one wishes to determine the proper combination of both variables to improve the engine behavior at specific operating conditions (speed and load) under dual fuel mode.

2. Outline of the model

In the present work only an outline of the model is given, on the one hand because of the lack of space and on the other because this is based on an existing model that has been presented in detail in previous publications by the authors [10–13]. As stated above, the main purpose of this work is the theoretical study, which is to be conducted via the simulation model, concerning the effect of pilot diesel fuel quantity and injection timing on both engine performance and emissions.

The model used is a phenomenological two-zone one, examining the closed part of the engine cycle. The cylinder charge during the compression phase is treated as a single zone (unburned zone), with assumed uniformity in space of pressure, temperature and composition. During the compression phase, the cylinder charge is compressed to a high pressure and temperature as TDC is approached. Prior to reaching TDC, a small amount of diesel fuel is injected into the combustion chamber, which atomizes and evaporates forming a conical jet penetrating inside the unburned zone. As the jet penetrates, homogeneous mixture is entrained into the jet and mixes with the evaporated diesel fuel. The quantity of the mixture entrained inside the conical jet is estimated from the value of its volume change. The boundary of this jet defines the second zone, the ‘‘burning zone’’ as shown in Fig. 1 [10–13].

According to the model, inside the burning zone takes place the process of combustion, and for this reason the main constituents of the zone are combustion products, unburned evaporated diesel fuel, unburned gaseous fuel and air that has not yet participated in combustion. Furthermore, the model assumes that there is uniformity in space of pressure, temperature and composition inside the burning zone. The ignition of the charge inside the burning zone commences after the auto-ignition of the evaporated diesel fuel. The time interval between the start of diesel fuel injection and initiation of combustion defines the ignition delay period. During this period, the temperature of the burning zone and the pressure of the cylinder charge increase significantly as the piston approaches TDC. Simultaneously, the mass of evaporated diesel fuel increases forming a combustible mixture with the one entrained inside the burning zone. Thus, after the initiation of combustion, two zones exist (unburned and burning one) separated by a flame front, which is assumed to have the shape of a cone covering the outer area of the jet.

The model assumes that the flame front has a negligible thickness, based on experimental data that report a flame thickness under actual engine conditions of approximately 0.2 mm [17]. Thus, the outer boundary of the burning zone is defined by the flame front, the penetration of which,

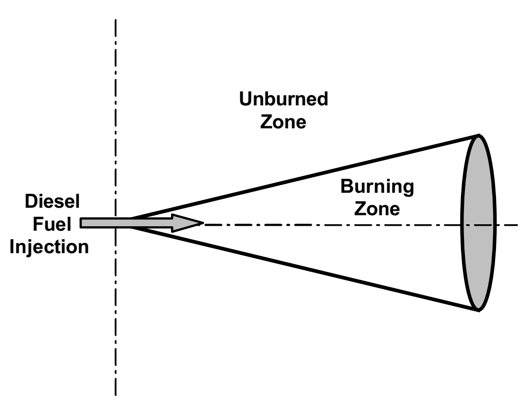

Fig. 1. Definition of the burning zone before the initiation of combustion.

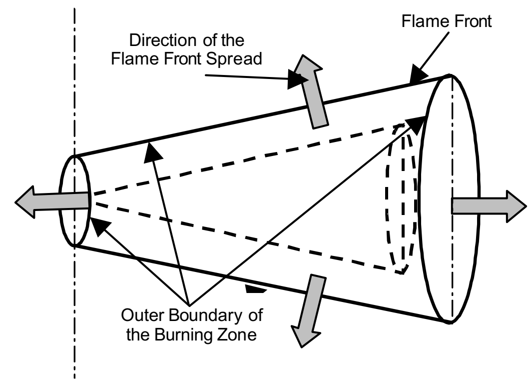

Fig. 2. Definition of the burning zone after the initiation of combustion.

inside the unburned zone, is controlled by the flame speed computed via a consideration of a flame propagation mechanism. The flame front spreads inside the unburned zone in a direction perpendicular to the outer surface of the conical burning zone, as shown in Fig. 2.

The entrainment rate of gaseous fuel inside the burning zone results from the variation of the volume of the burning zone, estimated from the jet penetration theory of Hiroyasu et al. [18] and from the propagation of the flame surrounding the jet. The gaseous fuel mixture entrained inside the burning zone, due to the flame propagation, is transformed immediately into products. Therefore, a portion of the heat released from the gaseous fuel depends actually on the flame propagation speed. The remaining portion depends on the amount of gaseous fuel entrained inside the burning zone from the start of injection, due to the formation and penetration of the fuel jet.

The combustion of this gaseous fuel is described by using an Arrhenius type premixed reaction rate. Consequently, the heat release rate of gaseous fuel is the sum of the two rates described above, while the total rate of heat release is defined as the sum of the diesel and gaseous fuel ones. The model is based on the fundamental laws of mass, momentum and energy conservation applied for each zone separately and for the whole cylinder charge. The mixture inside each zone after the initiation of injection is assumed to be homogeneous. For each zone, uniformity of temperature and composition exists. Furthermore, the model assumes that the pressure is uniform inside the combustion chamber. The wall jet theory of Glauert [19] is used to describe the fuel jet history after wall impingement. Heat transfer between the zones is neglected. Inside the burning zone, dissociation of combustion products is taken care for by incorporating the Vickland et al. method [20] with eleven species considered.

The liquid fuel used is normal dodecane, which represents adequately the commercial diesel fuel. The natural gas used in the current model is a mixture of methane and other hydrocarbons. The main characteristics of the fuels used are given in Table 1. The temperatures of the unburned and burning zones and the cylinder pressure are obtained from a set of three ordinary first order differ-

Table 1
Basic characteristics of diesel fuel and natural gas

Liquid diesel fuel	
Cetane number	52.5
Density	833.7 kg/m^3
Net heating value	42.74 MJ/kg
Sulfur content	45 mg/kg
Natural gas	
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)
Net heating value	48.6 MJ/kg

ential equations, derived after mathematical manipulation of the first law of thermodynamics and the perfect gas state equation [10–13,21].

The heat exchange rate for each zone is calculated by employing the well-known Annand formula [10–13,22], while the calculation of the rate of diesel fuel injection, as described in Refs. [10–12], depends on its density, the area of the injector hole and the velocity of injection. It should be stated here that the pressures across the injector have been obtained from the measured values of fuel line and cylinder pressure traces [10–13,23,24].

As mentioned before, the initiation of combustion inside the burning zone takes place after the ignition of the injected pilot fuel [5,10–16]. In the current work, the ignition delay period of the pilot fuel is a function of the burning zone temperature, the cylinder pressure and the equivalence ratio of the fuel vapor–air mixture inside the burning zone [10–13,23–25] while the combustion rate of the gaseous fuel is calculated by employing the formula described in [26]. The combustion of the pilot injected diesel fuel is assumed to take place in two subsequent stages, i.e. premixed and diffusion combustion. Thus, the semi-empirical combustion model of Whitehouse–Way is used to describe the preparation and combustion rates of the pilot liquid fuel [10–13,23,24,27–29].

The NO formation is assumed to be a non-equilibrium process controlled by chemical kinetics [17,21,23,28–32]. Then, for the estimation of NO formation the extended Zeldovich mechanism, with minor modifications, has been adopted in the current work [10–13]. Further, it is assumed that both the CO formation and oxidation mechanisms are kinetically controlled. Thus, the respective rates are modeled in a similar way as the respective ones of NO, using the kinetics of the main formation and oxidation reactions, combining carbon monoxide, carbon dioxide and hydroxyl [13,23].

Many researches use semi-empirical models, derived as correlations from the analysis of experimental data, to describe the soot formation mechanism. In the present work, a semi-empirical mathematical formula has been used to describe the soot formation mechanism [24,27–32], by taking into account details concerning the cylinder pressure, the temperature of the burning zone, and the concentrations of the oxygen and the unburned liquid fuel inside the burning zone [10–13].

3. Test cases examined

To validate the present model, an experimental investigation has been conducted on a single cylinder, naturally aspirated, direct injection, Lister LV1, diesel engine modified to operate with pilot diesel fuel injection under dual fuel mode [6]. The engine, located at the authors' laboratory, is supplied with natural gas obtained from the city low-pressure distribution network, and the adjustment of the engine load under dual fuel operation is accomplished through a control valve located after the gaseous fuel flow meter. Afterwards, the gaseous fuel flows towards the engine intake and is mixed with the intake air. The data obtained from the experimental investigation have been used to validate the model and understand the complex nature of combustion under dual fuel operating mode. More information about the engine used and its dynamometer is given in Table 2. A schematic layout of the test installation used is shown in Fig. 3.

During the experimental investigation, measurements were taken at four different engine loads corresponding to 20%, 40%, 60% and 80% of full load relative to diesel operation, at an engine speed of 2000 rpm, under both normal diesel (i.e. engine runs only with diesel fuel) and dual fuel operation (i.e. simultaneously combustion of both natural gas and diesel fuel) [6,10,11].

Under dual fuel operation, an effort has been made to keep the pilot amount of diesel fuel constant (normal pilot fuel quantity – NPFQ), while the power output of the engine is adjusted through the amount of natural gas. The measurement procedure under dual fuel operation has as follows: At a given constant engine speed, pilot amount of diesel fuel is injected in order to cover approximately the mechanical losses of the engine. Then, keeping constant the flow rate of liquid diesel fuel, the power out-

Table 2
Basic data of Lister LV1 diesel engine and its dynamometer

Engine type	Single cylinder, 4-stroke, DI
Bore	85.73 mm
Stroke	82.55 mm
Connecting rod length	148.59 mm
Compression ratio	18:1
Cylinder dead volume	28.03 cm^3
Inlet valve opening	15 °CA before TDC
Inlet valve closure	41 °CA after BDC
Exhaust valve opening	41 °CA before BDC
Exhaust valve closure	15 °CA after TDC
Inlet valve diameter	34.5 mm
Exhaust valve diameter	31.5 mm
Type of fuel pump	Bryce-Berger, 180 (bar)
Static injection timing	26 °CA before TDC
Dynamometer	Heenan & Froude, hydraulic

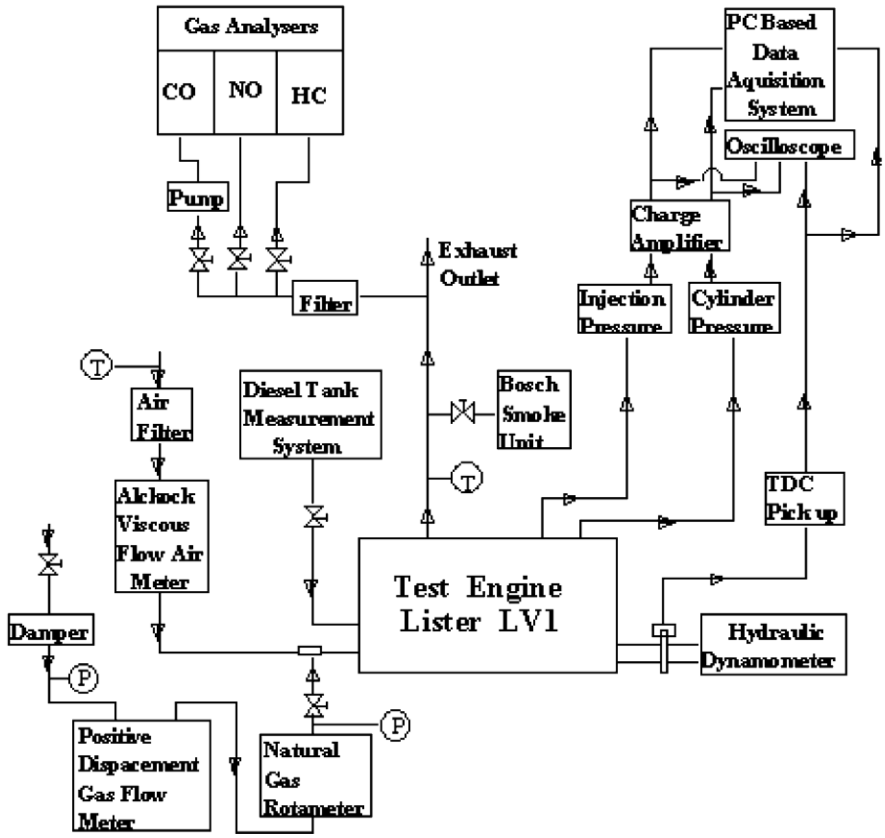

Fig. 3. Schematic layout of the test installation.

put is further increased by using only gaseous fuel. This procedure is followed until the desired power output is obtained.

As it is known [5], the main disadvantage of the dual fuel operation is the negative impact of dual fuel combustion on engine efficiency and CO emissions. For this reason, in the present work, it was attempted, by using the current model, to investigate the use of increased pilot fuel quantity combined with advanced injection timing in order to compensate for the negative effect of dual fuel combustion on engine efficiency and CO. Thus, at each engine load, the pilot amount of diesel fuel was increased by fifty (150% NPFQ) and seventy five per cent (175% NPFQ), respectively, relative to the normal pilot fuel quantity (NPFQ).

To calculate the natural gas consumption, for the 150% and 175% of the normal pilot fuel quantities, the following expression was used:

$$\lambda = \frac{\dot{m}_{\mathrm{mix}} - \dot{m}_{\mathrm{NG,i}}}{(\mathrm{AFRD_{st}} \cdot \dot{m}_{\mathrm{D,i}} + \mathrm{AFRNG_{st}} \cdot \dot{m}_{\mathrm{NG,i}})} \tag{1}$$

with $i = \mathrm{NPFQ}$, 150% NPFQ, 175% NPFQ, and where AFRDst and AFRNGst are the stoichiometric air to fuel ratio (by mass) for the diesel and the natural gas fuels, respectively. In the present theoretical investigation, the mass flow rate of the air–gaseous fuel mixture at the inlet valve closure ($\dot{m}'_{\mathrm{mix}}$) and the total air excess ratio (λ) are considered to be constant at each engine load condition.

Table 3
Test cases examined at 2000 rpm engine speed for various engine loads

Load (%)	λ	m'_{D} (kg/h)	m'_{NG} (kg/h)	NIT deg BTDC
Normal pilot fuel quantity (NPFQ)				
20	2.08	0.200	0.630	16
40	1.63	0.196	0.860	16
60	1.47	0.201	0.920	15
80	1.38	0.198	0.970	14
150% of NPFQ				
20	2.08	0.300	0.545	16
40	1.63	0.294	0.780	16
60	1.47	0.301	0.842	15
80	1.38	0.297	0.904	14
175% of NPFQ				
20	2.08	0.350	0.499	16
40	1.63	0.343	0.740	16
60	1.47	0.352	0.801	15
80	1.38	0.346	0.868	14

Thus, for each amount of pilot fuel quantity, the consumption of natural gas is calculated from the above relation. Moreover, at each load and pilot fuel quantity, the injection timing was increased (before TDC) by 4°, 6° and 8° crank angle BNIT, respectively, compared to the normal injection timing (NIT).

Table 3 shows, for each test case examined the diesel fuel and natural gas mass flow rates, and the normal injection timing obtained from the experimental investigation.

4. Results and discussion

To verify the model's ability to predict the engine performance and concentration of the most important pollutant emissions (NO, CO, soot), an extended theoretical and experimental investigation has been conducted in the past by the authors. It should be stated here, that the results obtained from that investigation corresponded to the engine operation under dual fuel mode with normal pilot fuel quantity (NPFQ) and normal injection timing (NIT), with the results presented previously [6,10–13]. As revealed, there was a good agreement between calculated and experimental pressure and total heat release traces, and furthermore the trend of pollutant formation mechanisms with engine load was predicted with reasonable accuracy. Furthermore, to confirm the truthfulness of the model predictions concerning the effect of each parameter (pilot fuel quantity and its injection advance) on maximum cylinder pressure, brake specific fuel consumption and pollutant emissions (NO, CO and soot), a comparison has been made between the results obtained from the model application and experimental ones came from the international bibliography [5,10,33–38]. Comparing the theoretical and experimental results, it is revealed that, despite the difference observed in absolute values that is expected for a two zone phenomenological model, it manages to predict with a good coincidence the trend of performance and pollutant emissions variation with the change of both pilot fuel amount and its injection timing. This is then promising for the use of that model to examine the effect of both parameters (pilot fuel quantity and injection advance) on engine performance and pollutant emissions.

The main disadvantages of the dual fuel combustion are its negative impact (decrease) on engine efficiency and carbon monoxide emissions, with respect to these values under normal diesel operation (i.e. engine runs with 100% diesel fuel). The pilot fuel quantity accompanied with its injection timing are two of the most important variables, that have a controlling influence on the performance and pollutant emissions of an engine operating under dual fuel mode.

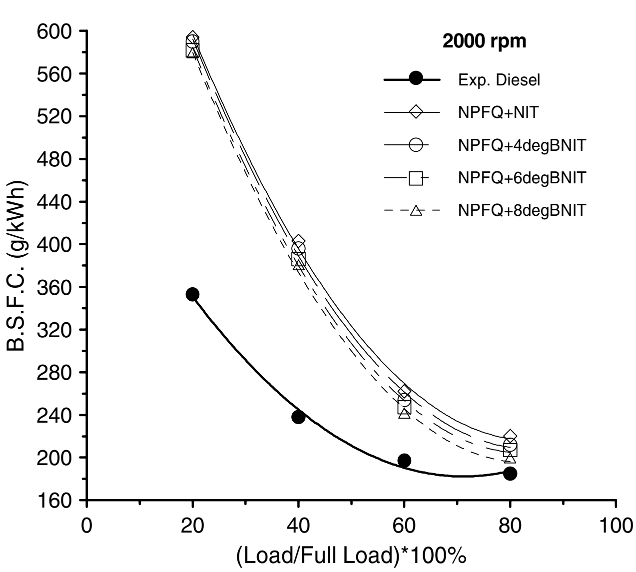

Fig. 4. Brake specific fuel consumption versus engine load for various injection timings at NPFQ.

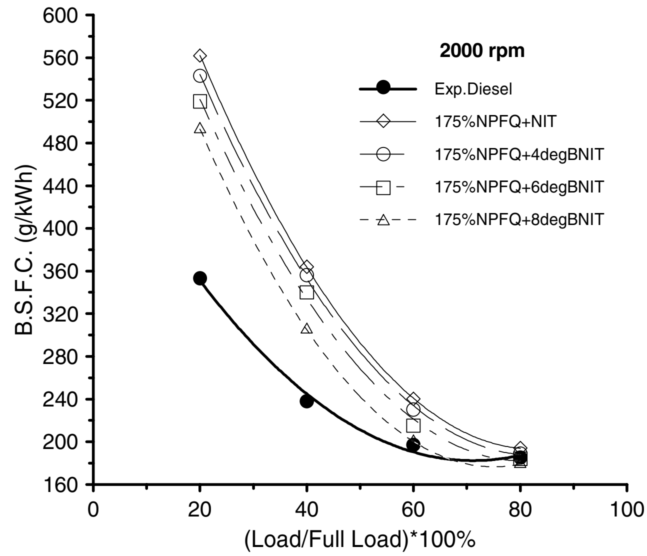

Fig. 5. Brake specific fuel consumption versus engine load for various injection timings at 175% NPFQ.

Figs. 4 and 5 provide the variation of the calculated values of the total brake specific fuel consumption under dual fuel operation as a function of engine load, for various injection timings at NPFQ and 175% of NPFQ operating modes. In the same figure, experimental brake specific fuel consumption values are given under normal diesel operation referring to the normal injection timing. It must be stated here that the total brake specific fuel consumption is estimated from the calculated brake power output and the calculated mass flow rates of diesel and natural gas, for each test case examined.

As shown in Table 1, the heating value of natural gas is higher compared to the one of diesel fuel used. Thus, the absolute values of engine efficiency under dual fuel operation would be even lower if the total brake specific fuel consumption reported in Figs. 4 and 5 were corrected to the heating value of the diesel fuel. But the trend of the engine efficiency variation with the change of both pilot fuel amount and its injection timing is similar to the one of the total brake specific fuel consumption presented in Figs. 4 and 5, revealing thus the effect of both parameters on engine efficiency.

Observing Figs. 4 and 5, it is revealed that the brake specific fuel consumption under dual fuel operation, especially at low and intermediate engine load conditions, is considerably higher compared to the respective one under normal diesel operation. This is the result of the lower premixed controlled combustion rate observed under dual fuel operation during the initial stage of combustion [10]. Examining the theoretical values under dual fuel operation, it is revealed that for the same injection timing the increase of pilot diesel fuel amount leads to an improvement (decrease) of the brake specific fuel consumption compared to the one observed under normal pilot fuel quantity mode. The use of a larger pilot fuel quantity leads to a higher total heat release rate during the premixed controlled combustion phase. It results to an increase of the cylinder charge temperature, which affects positively the combustion rate of the gaseous fuel especially during the second phase of combustion (diffusion phase) since it becomes more efficient. Moreover, for each pilot fuel amount under dual fuel mode, the injection advance leads to a slight improvement (decrease) of the brake specific fuel consumption compared to the one at normal injection timing. It results from the increase of the ignition delay period which leads to a sharper increase of the total heat release curve during the premixed controlled combustion phase a fact that improves the fuel conversion efficiency since the duration of combustion becomes shorter. This improvement is shown to be more sensible at high engine load conditions and high amounts of pilot diesel fuel, where the values of the total brake specific fuel consumption under dual fuel operation tend to converge to the respective ones under normal diesel operation.

Figs. 6 and 7 provide the variation of the calculated values of the maximum combustion pressure under dual fuel operation as a function of engine load, for various injection timings, at NPFQ, 150% of NPFQ and 175% of NPFQ operating modes. In these figure, the respective experimental values under normal diesel operation are given.

As observed, peak cylinder pressure is affected by the quantity of the pilot diesel fuel and by its injection timing as well. For the same injection advance, as the pilot fuel amount increases, keeping the engine load constant, peak cylinder pressure increases significantly. The higher heat release rate during the premixed controlled combustion phase and the lower specific heat capacity of the gaseous mixture, since the increase of pilot diesel quantity leads to lower amount of gaseous fuel, are the main causes for the increase of the maximum cylinder pressure. Furthermore, for the same pilot fuel quantity, the increase of injection advance, keeping the load constant, leads to an increase of the peak cylinder pressure since more of the fuel burns before the top dead center (TDC). At low and intermediate loads, the increase of the maximum cylinder pressure is not so severe to cause problems on the engine structure. On the other hand, at high load, the simultaneous increase of both parameters leads to a significant increase of maximum combustion pressure compared to the respective values for normal pilot fuel quantity and

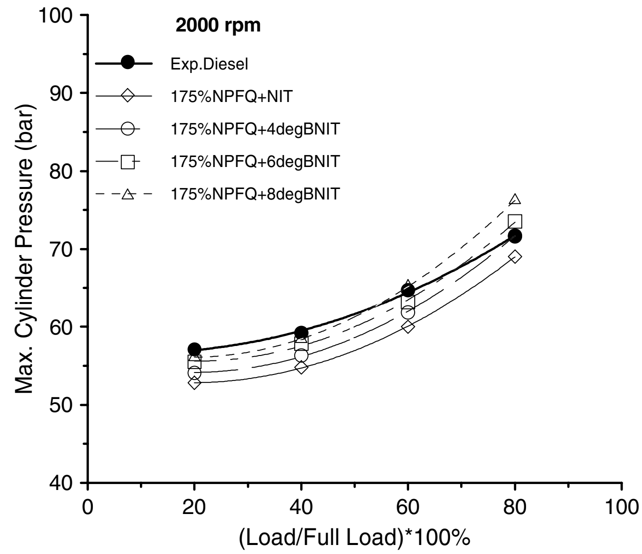

Fig. 7. Maximum cylinder pressure versus engine load for various injection timings at 175% NPFQ.

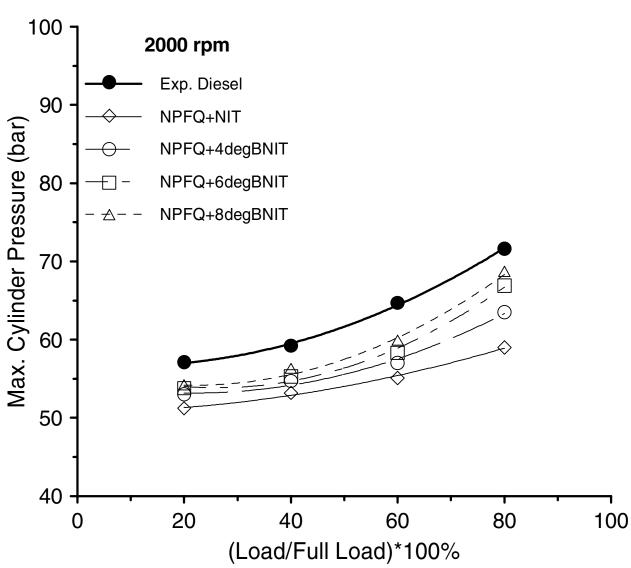

Fig. 6. Maximum cylinder pressure versus engine load for various injection timings at NPFQ.

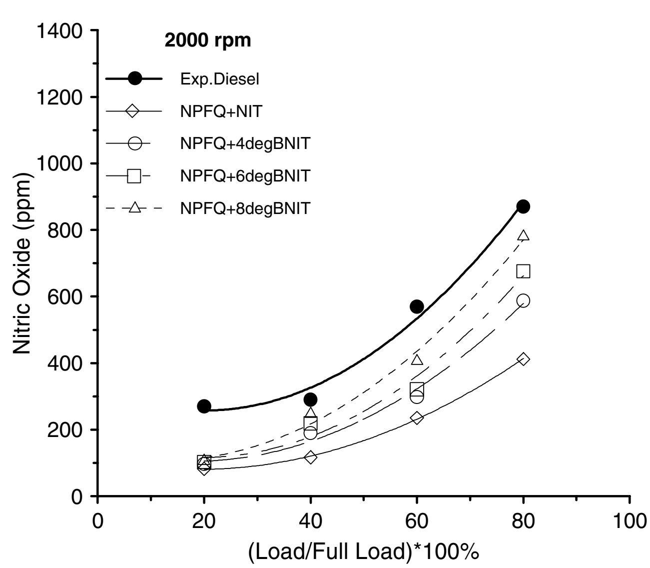

Fig. 8. Nitric oxide emissions versus engine load for various injection timings at NPFQ.

normal injection timing. For extreme values of the pilot fuel quantity and injection advance, it appears that the maximum cylinder pressure tends to be higher compared to the respective one under normal diesel operation, revealing the undesirable effect of the simultaneous increase of both parameters on engine performance as well as on engine structure.

As far as the effect of both parameters on emissions is concerned, Figs. 8–13, provide the variation of the calculated values of NO, CO and soot emissions, respectively, under dual fuel operation versus engine load for various injection timings at NPFQ and 175% of NPFQ operating modes. In the same figures, the measured values for NO, CO and soot under normal diesel operation at the normal injection timing are given. It must be stated here that in Figs. 12 and 13, two *y*-axes are used in order to show clearly the effect of the advanced injection timing on soot formation under dual fuel operation.

Observing Figs. 8 and 9, it is revealed that under dual fuel operation the increase of pilot fuel amount, keeping constant the injection timing, affects as expected NO emissions. The cylinder charge temperature and the local oxygen excess ratio are dominant parameters for NO formation. Specifically, under dual fuel operation, the increase of the pilot fuel quantity, keeping constant the injection timing, results in higher oxygen concentration inside the burning zone accompanied with higher charge temperature, especially during the premixed controlled combustion phase. Thus, the increase of pilot fuel amount leads to an increase of the nitric oxide formation. Moreover, under dual fuel operation, the increase of injection timing at fixed pilot fuel quantity leads to an increase of NO emissions. The increase is more sensitive at high engine load conditions. Advancing the injection timing causes an earlier start of combustion relative to TDC, which leads to an increase of NO formation, since then the charge temperatures inside the combustion chamber increase significantly.

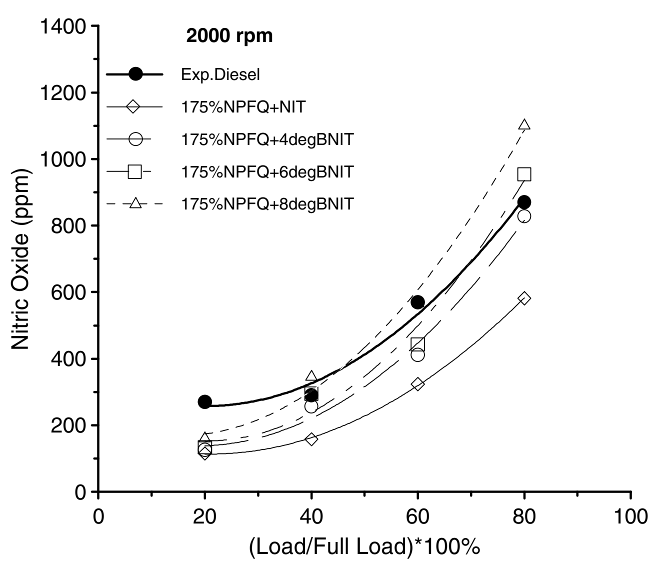

Fig. 9. Nitric oxide emissions versus engine load for various injection timings at 175% NPFQ.

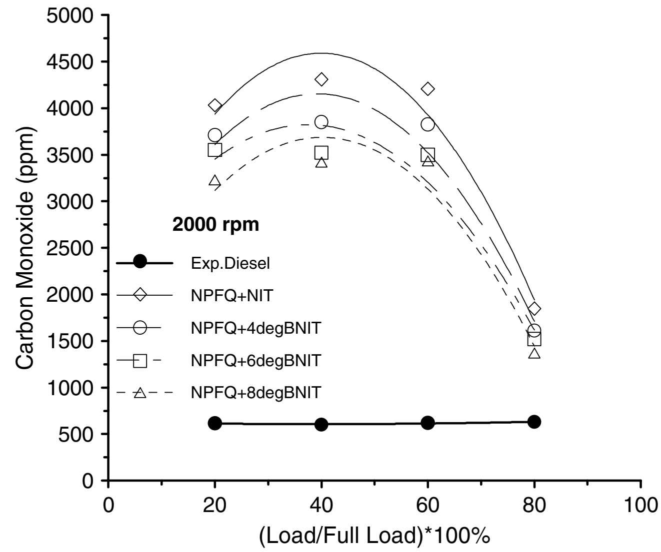

Fig. 10. Carbon monoxide emissions versus engine load for various injection timings, at NPFQ.

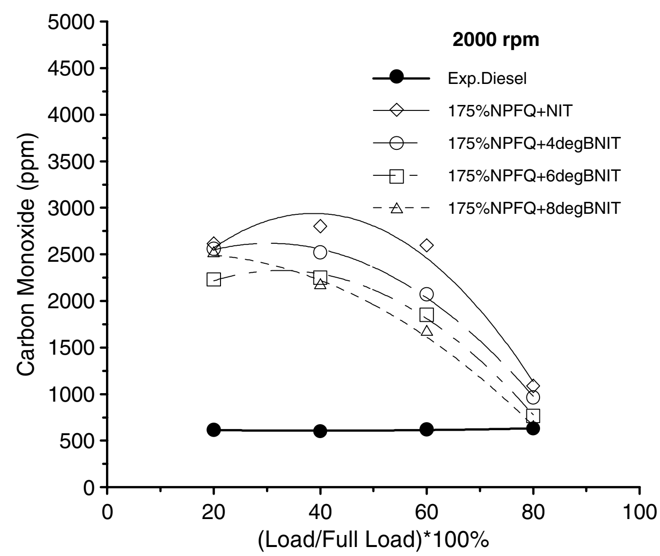

Fig. 11. Carbon monoxide emissions versus engine load for various injection timings, 175% NPFQ.

At high engine load conditions under dual fuel operation, the simultaneous increase of both parameters, compared to the respective normal values, leads to a sharp increase of NO emissions which, for extreme values of both parameters, seem to be higher than the ones under normal diesel operation (negative influence of both parameters).

Examining the values of the carbon monoxide emissions, shown in Figs. 10 and 11, it is revealed that CO emissions increase with increasing engine load and, beyond a certain value of load, they start to decrease as a result of the improvement of the gaseous fuel combustion rate. Observing these figures, it is revealed that the quantity of

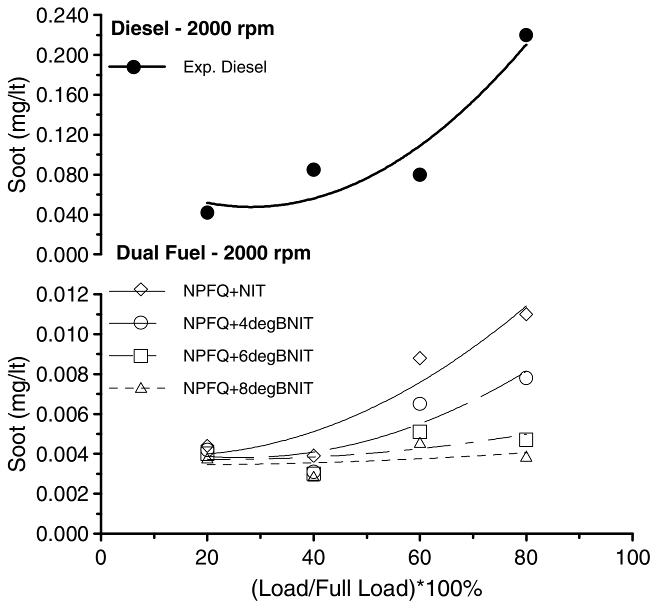

Fig. 12. Soot emissions versus engine load for various injection timings at NPFQ.

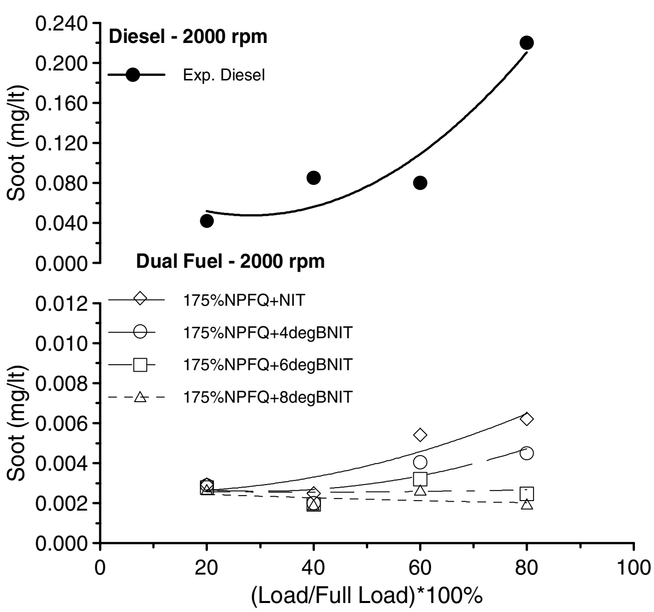

Fig. 13. Soot emissions versus engine load for various injection timings at 175% NPFQ.

the pilot diesel fuel and its injection timing affect seriously the CO concentration levels observed under dual fuel operation. Specifically, for fixed injection timing, the increase of pilot fuel amount, keeping the engine load constant, leads to a decrease of CO emissions since the higher burning rate during the premixed controlled combustion phase and the lower available amount of unburned gaseous fuel inside the burning zone have a negative affect (demote) the CO formation mechanism.

Furthermore, the increase of injection timing, for the same pilot fuel amount, leads to a decrease of CO emissions compared to the respective values for normal injection timing. It results from the increase of CO oxidation mechanism, which is due to the increased time interval during the expansion stroke for which high temperature persists in the cylinder allowed more complete oxidation of CO. The simultaneous increase of both parameters, compared to the respective normal values, leads to a considerable improvement (decrease) of the carbon monoxide emissions observed under dual fuel operating mode.

Observing Figs. 12 and 13, it is revealed that under dual fuel operation the increase of the pilot fuel amount, when injection timing is kept constant, affects positively (decrease) the concentration level of soot emissions.

The increase of the pilot diesel fuel amount contributes to a demotion of soot formation mechanism since the cylinder charge temperature increases and furthermore to a promotion of soot oxidation mechanism due to the increase of the oxygen excess ratio inside the burning zone. The effect is more significant at high engine loads. Furthermore, it appears that the pilot fuel injection advance affects positively (decrease) the concentration of soot emissions, compared to the respective ones observed at normal pilot fuel amount and normal injection timing. This is probably due to the higher cylinder charge temperature which demotes the soot formation mechanism contributing thus to the reduction of soot emissions. For the same engine load, the increased amount of pilot diesel fuel accompanied with an increase in injection timing lead to a significant reduction of the soot emissions compared to the respective values under dual fuel operation with normal values of both parameters. Thus, the difference of the soot emissions observed between normal diesel and dual fuel operating modes becomes higher with the simultaneous increase of both parameters.

5. Conclusions

In the present work, an existing two-zone phenomenological model has been used to examine the effect of the increase of the pilot fuel quantity accompanied with its injection advance, on performance and pollutant emissions of a direct injection, dual fuel, diesel–natural gas engine.

The comparison between normal diesel and dual fuel operation with normal injection timing and normal pilot fuel quantity reveals that the simultaneous use of diesel and natural gas leads to lower peak combustion pressures. Moreover, dual fuel operation leads to higher values of the brake specific fuel consumption compared to the respective ones under pure diesel operation. As far as pollutant emissions are concerned, dual fuel operation affects positively (reduction) nitric oxide and soot emissions, while it causes a considerable increase of carbon monoxide emissions compared to the ones under normal diesel operation. The results of the present work revealed that:

(a) An improvement in the total brake specific fuel consumption is achieved by employing a larger pilot fuel quantity and by advancing its injection timing. At high engine load conditions, the total brake specific fuel consumption observed under dual fuel operation tends to converge with the respective value under normal diesel operation.
(b) The simultaneous increase of pilot fuel amount accompanied with increase of its injection timing results to an increase of the maximum combustion pressure, compared to the respective one observed at normal pilot fuel quantity and normal injection timing. This effect is more evident at high load, where the combination of high injection advance and high pilot fuel amount results in the maximum cylinder pressure, under dual fuel operation, becoming higher compared to the respective one under normal diesel operation. This point to the necessity of determining an optimum combination of both parameters, especially at high engine loads.
(c) As far as emissions are concerned, the negative impacts of dual fuel combustion on CO emissions could be reduced by increasing the pilot fuel quantity and by advancing its injection timing. However, this seems to have a negative effect on NO emissions. This requires a detailed investigation to estimate the proper combination of engine settings to improve its behavior under dual fuel mode, as far as engine efficiency and CO emissions are concerned.

It appears that the use of the advanced injection timing accompanied with an increase of the amount of the pilot diesel fuel can be a potential tool for increasing engine efficiency and reducing CO emissions, when the engine operates under dual fuel mode. On the other hand, by carrying out the adjustments referred to above, part of the benefit obtained by the dual fuel operation is lost, since an increase in maximum cylinder pressure and NO emissions levels are observed. Taking into account the corresponding values under normal diesel operation, it is revealed that, at low and intermediate loads, the adjustment of the pilot fuel quantity and its injection timing are not restricted, as long as they are at high engine load conditions.

At high load, there is a necessity for the determination of an optimum combination of the parameters referred to above, since the simultaneous increase of pilot fuel quantity accompanied with its advancing injection timing results to higher values of maximum cylinder pressure and NO emissions, compared to the respective ones under normal diesel operation.

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