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ABSTRACT

With the increasing public interest in energy supply and the environment, attention has focused on the development of ecological and efficient combustion technologies. One of these technologies could be the use of natural gas as supplement fuel for diesel fuel in DI diesel engines. The great availability at attractive prices and the clean nature of combustion are the most important advantages of natural gas compared to conventional diesel fuel. In the present work are given theoretical and experimental results for the combustion mechanism of natural gas in a compression ignition environment, with special emphasis on the combined heat release rate of natural gas and diesel fuel, the duration of combustion and the ignition delay period. Results are also provided for the formation history of pollutants inside the combustion chamber of a DI diesel engine operating in dual fuel mode (with natural gas fuelling). The model used is a two – zone phenomenological one describing the combustion mechanisms of natural gas and diesel fuel. Natural gas is ignited from the diesel fuel and the existence of a flame front is considered to describe its combustion rate. The experimental investigation was conducted on a single cylinder DI diesel engine properly modified to operate under dual fuel conditions using natural gas as supplementary fuel. The experimental results seem to be in good agreement with the theoretical ones, obtained from the simulation model. Comparing the results under normal diesel and dual fuel operation a serious effect of the presence of natural gas on exhaust emissions and main combustion characteristics is observed. As far as the exhaust emissions are concerned, the presence of gaseous fuel affects positively (reduction) the values of NO and Soot. On the contrary dual fuel operation has a negative effect on CO emissions. Concerning the combustion analysis of dual fuel operation, heat release rate is affected seriously by the presence of natural gas in the combustion chamber. Compared to normal diesel operation, the increase of natural gas quantity results to a decrease of the percentage of heat released during premixed combustion period while ignition delay increases. Combustion duration is higher under dual fuel operation at low load, but with the increase of load the difference is decreased and it becomes even lower compared to normal diesel operation at high load.

INTRODUCTION

The dual fuel diesel engine with pilot ignition is a conventional diesel in which the main amount of energy release during the combustion process comes from the combustion of gaseous fuel which is inducted into the engine along with the air intake. The ignition source is a small (pilot) amount of diesel fuel injected late in the compression stroke inside the cylinder (Dual Fuel Operation) [1*]. A suitable gaseous fuel for dual mode operation is natural gas which due to its relatively high auto-ignition temperature allows to maintain the compression ratio (CR) of conventional diesel engines. This kind of engine is referred to as Pilot-Ignited Natural Gas Diesel (P-I.NG&D.) engines [1-4]. In recent years P-I.NG&D. engines have received considerable attention as a result of their advantage in the field of pollutant emissions and increased availability of natural gas at attractive prices compared to diesel fuel [3-5]. A main advantage of P-I.NG&D. engine compared to conventional diesel one is the simultaneous reduction of NO and particulates emissions. During the last years, extensive experimental investigations along with computer simulations have been conducted in P-I.NG&D. engines [1-4,6-11].In the present contribution a two-zone model has been used to describe the combustion process of a P-I.NG&D. engine. The current model is an improved version of a previous one presented in the past [6-8].

*Numbers in brackets designate references at the end of the paper

The model predicts apart from cylinder pressure trace and heat release the concentration of soot, nitric oxide and carbon monoxide pollutant emissions. The main purpose of the model is to describe and understand the combustion process taking place in P-I.NG&D. engines. Theoretical results are provided at various engine loads at constant engine speed.

Parallel to the theoretical investigation, an experimental one has been conducted on a single cylinder high speed direct injection diesel test engine modified to operate under dual fuel mode [2,6]. The engine was supplied with natural gas from a low-pressure distribution network. The data obtained from the experimental investigation have been used to validate the model and understand the complex combustion nature under dual fuel operation. From the comparison of theoretical and experimental findings it is revealed that the simulation model developed predicts adequately engine performance and pollutant emissions trend with engine load under dual fuel operation. From both theoretical and experimental findings, important information is derived revealing the effect of the presence of gaseous fuel in the cylinder charge on the combustion mechanism. This is accomplished through the comparison of the cylinder pressure traces, ignition delay period, intensity of the heat release and combustion duration under normal and dual fuel operation. Furthermore, concerning pollutant emissions (NO, CO and Soot), it is revealed the effect of gaseous fuel on their formation by comparing values to those under normal diesel operation. This information is extremely valuable if one wishes to apply this technology on new or existing DI diesel engine.

BRIEF DESCRIPTION OF DUAL FUEL COMBUSTION PROCESS

As previously mentioned, in P-I.NG&D. engines, gaseous fuel is inducted inside the cylinder along with the air intake while small amount of diesel fuel is used for ignition [1,9-11]. At the end of intake stroke the cylinder charge is a homogeneous mixture of air and gaseous fuel. The cylinder charge during compression is treated as a single zone (unburned zone) in which a uniformity in space of pressure, temperature and composition is considered. During the compression stroke the cylinder charge is compressed to high pressure and temperature as TDC is approached. Prior to 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 its value of the volume change. The boundary of this conical jet defines the second zone entitled "burning zone" (Figure 1) [6-8]. According to the model inside the burning zone takes place combustion process and for this reason the main constituents of the zone are combustion products, unburned evaporated diesel fuel, unburned gaseous fuel and air which 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 initiates after the auto-ignition of the evaporated diesel fuel. The time interval between the start of diesel 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 approach TDC. Simultaneous, the mass of evaporated diesel fuel increases forming a combustible mixture with the one entrained inside the burning zone. Thus, after 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 negligible thickness based on experimental data from which a flame thickness under actual engine conditions of approximately 0.2 mm is reported [12]. Thus, the outer boundary of the burning zone is defined by the flame front whose penetration inside the unburned zone is controlled by the flame speed calculated using 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 (Figure 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 [13] 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 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.

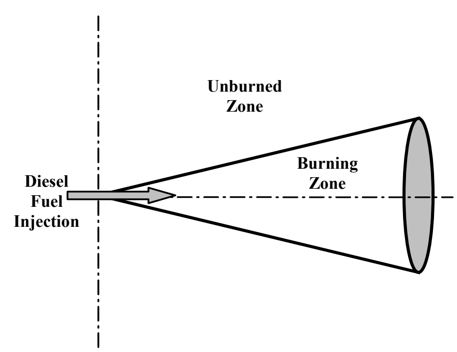

Figure 1. Definition of the Burning Zone before Initiation of Combustion

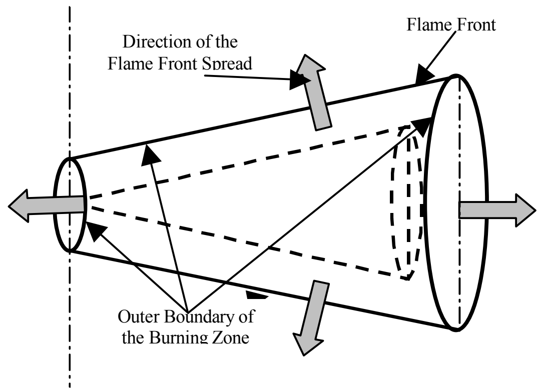

Figure 2. Definition of the Burning Zone after Initiation of Combustion

THE MATHEMATICAL TREATMENT OF THE MODEL

As mentioned, the model developed by the authors [6-8], has been improved in the present work to take into account details concerning the mixing and combustion process of natural gas. It is a relatively simple model, which is used to describe the operation of existing diesel engines where the main part of the liquid fuel is replaced by natural gas. Its purpose is to predict apart from engine performance and concentration of pollutants, the main factors affecting the combustion process (ignition delay period, the intensity of the heat release mechanism, the combustion duration etc.) when the engine runs under dual fuel operation. The model is based on the fundamental laws of mass, momentum and energy conservation applied for each zone separately and for the entire of cylinder charge. Furthermore, for the development of the model presented in the present work have been made some assumptions which had been made in the earlier version of the model [6-8]. These assumptions have as follows :

- The mixture inside each zone after initiation of injection is assumed to be homogeneous. For each zone there is uniformity of temperature and composition. Furthermore, the model assumes that the pressure is uniform inside the combustion chamber.
- The wall jet theory of Glauert [14] is used to describe the fuel jet after the wall impingement.
- Heat transfer between the zones is neglected.
- Inside the burning zone dissociation of combustion products is considered for by incorporating the Vickland et al. method including eleven species [15].
- The liquid fuel used is normal dodecane, which represents adequately 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 in the present model are given in Table 1.
- The temperatures of the unburned (T_u) and burning zone (T_b) and the cylinder pressure (P) are obtained from a set of three ordinary first order differential equations derived after some mathematical manipulation from the first thermodynamic law and the perfect gas state equation [6-8]. The first thermodynamic law and perfect gas state equations applied on each zone have as follows:

$$dU_{u,b} = dQ_{u,b} - PdV_{u,b} + \sum\left(dm_{u,b}h_{u,b}\right) + \underbrace{dm_{dies}^{prep}h_{dies}}_{\text{Enthalpy Addition to the burning zone from the liquid diesel fuel}} \quad (1)$$

$$P \cdot V_{u,b} = m_{u,b} \cdot R_{u,b} \cdot T_{u,b} \quad (2)$$

Table 1. Basic Characteristics of Diesel Fuel and Natural Gas

Basic Characteristics of Liquid Diesel Fuel		
Cetane Number	:	52,5 (-)
Density :	:	833,7 (kg/m^3)
Heating Value	:	42,74 (MJ/kg)
Sulfur Content	:	45 (mg/kg)
Basic Characteristics of 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)

- The heat exchange rate for each zone (dQu,b) is calculated by employing the well-known Annand formula [6-8,16], while the calculation of the rate of diesel fuel injection described in [6-8], depends on its density, the area of the injector hole and the velocity of injection. It must be stated here that the pressure across the injector has been obtained from the measured values of fuel line and cylinder pressure traces [6-8,18,19].
- As mentioned, the initiation of combustion inside the burning zone takes place after the ignition of the injected pilot fuel [1,6-8,9-11]. In the current work, the ignition delay period of the pilot fuel is a function of the burning zone temperature, of the cylinder pressure and the equivalence ratio of the fuel vapor – air mixture inside the burning zone [6-8, 18-20].
- The combustion of pilot injected diesel fuel is assumed to take place in two subsequent stages (premixed and diffusion combustion periods. Thus, the semi-empirical combustion model of Whitehouse – Way is used to describe the preparation and combustion rates of the pilot liquid fuel [6-8,18,19,22-24].

- The NO formation is assumed to be a non-equilibrium process controlled by chemical kinetics [12,17-18,23-26]. Thus, for the estimation of NO formation the extended Zeldovich mechanism with minor modifications has been adopted in the current work [6-8].
- 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 semi-empirical mathematical formula has been used to describe the soot formation mechanism [19,22-26] taking into account details about the cylinder pressure, the temperature of the burning zone and the concentrations of the oxygen and the unburned liquid fuel inside the burning zone [6-8].

MODEL MODIFICATIONS - IMPROVEMENTS

Below, is given an analysis of the modifications–improvements respecting to the definition of the burning zone, the calculation of the laminar flame speed and the definition of the gaseous fuel combustion rate. Furthermore, details on the simulation model which is used here for the first time, for the calculation of carbon monoxide concentration are presented.

ENTRAINMENT RATE OF UNBURNED MIXTURE - As mentioned the quantity of gaseous fuel-air mixture inside the burning zone is controlled by the spread of the zone towards the unburned one. The mass entrainment rate into the burning zone is calculated from the volume change rate of the zone (dV_b^{tot}), as follows [6-8]:

$$dm_b = \rho_u \cdot dV_b^{tot} \tag{3}$$

Prior to combustion initiation, the rate of volume change of the burning zone due to its penetration inside the combustion chamber is calculated from:

$$dV_b^{tot} = dV_b = d\left[\frac{\pi}{3} \cdot \tan^2 \theta \cdot S^3(t)\right] \tag{4}$$

where (θ) is the jet cone angle estimated from the correlation [19]:

$$\theta = \left(\frac{d_{inj}^2 \cdot \rho_u \cdot \Delta P}{\mu_u}\right)^{0,25} \tag{5}$$

The penetration length of the burning zone S is estimated using the correlation proposed by Hiroyasu [13].

After combustion initiation (Figure 2) the two zones are separated by a thin flame front, which covers the outer area of the burning zone and propagates into the unburned zone having a direction perpendicular to the outer surface of the conical zone. According to this assumption, the rate of the volume change of the burning zone depends on the penetration of the zone $dV_{b,1}$ and on the spread of the flame front towards the unburned zone $dV_{b,2}$. Thus, the total rate of the volume change is estimated from the formula:

$$dV_b^{tot} = dV_{b,1} + dV_{b,2} \tag{6}$$

After wall impingement the burning zone follows a path parallel to the cylinder walls. Thus, to determine the zone history after impingement, the wall jet theory of Glauert is applied [14].

If combustion initiates before wall impingement (Figure 3), the volume change of the zone due to its penetration $dV_{b,1}$ is given by EQ(4) where the penetration length S is the maximum of the value estimated from the jet velocity u_{jet} and the turbulence flame speed u_t. The volume change due to the existence of the flame front $dV_{b,2}$ is given by the formula :

$$dV_{b,2} = d\left[\frac{\pi}{3} \cdot S(t) \cdot \left(R_1^2 + R_0 R_1 + R_0^2\right)\right] \tag{7}$$

where : $Ypsos = S(t) \cdot \tan\left(\frac{\theta}{2}\right), \qquad R_1 = u_t dt + Ypsos$

and t is the time interval from the initiation of diesel fuel injection.

After wall impingement (Figure 4) the variation of fuel jet volume $dV_{b,1}$ is given by:

$$dV_{b,1} = d\left[\frac{2\pi}{3} \cdot \frac{\delta_0}{r_0} \cdot r^3(t_w)\right] \tag{8}$$

where the values of δ_0, r_0 and r are calculated from the Glauert wall jet theory [14] where t_w is the wall time interval. The variation of the flame front volume $dV_{b,2}$ is estimated from the formula:

$$\begin{aligned} dV_{b,2} = {} & d\left[\frac{\pi}{3} \cdot Y(t_w) \cdot \left(R_1^2 + R_0 R_1 + R_0^2\right)\right] + \\ & d\left[\frac{\pi}{3} \cdot X^2(t_w) \cdot \left(2Z_1(t_w) + Z_0(t_{w0})\right)\right] \end{aligned} \tag{9}$$

where: $Y(t_w) = \frac{D}{2} - \delta_0 - (Z_0 + u_t dt \cos(\frac{\theta}{2}))$,

$X(t_w) = r(t_w)$ and $Z_1 = r\frac{\delta_0}{r_0} + u_t dt$

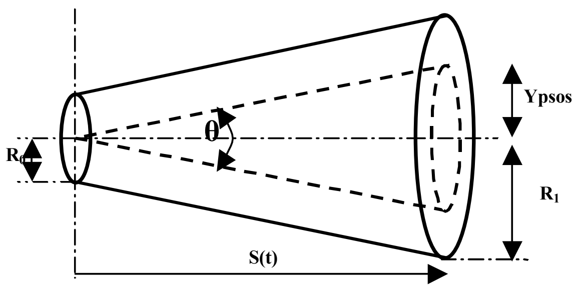

Figure 3. Burning Zone Definition before Impingement

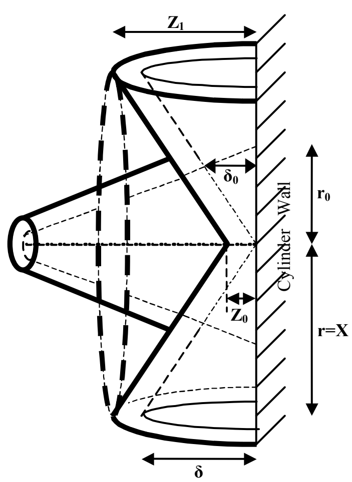

Figure 4. Burning Zone Definition after Impingement

MECHANISM OF FLAME PROPAGATION - The laminar burning velocity is defined as the relative velocity, with which the unburned gas moves inside the flame front and is transformed to products [1,4,8,10]. It is an important intrinsic property of a combustible mixture. The laminar burning velocity developed inside a combustion chamber depends mainly on the equivalence ratio, the temperature of the unburned gas and the pressure [12]. Since methane is the main constituent of natural gas the laminar burning velocity in the present model is obtained by applying a correlation proposed by G.A. Karim [21], which simulates adequately the burning velocity of Methane – Air mixtures. The mathematical formula has as follows:

$$u_l = A + (F_1 \cdot F_2) \cdot \left(F_3 + F_4\left(\varphi_{u,eq} - 1.036\right) + F_5\left(\varphi_{u,eq} - 1.036\right)^2\right) \tag{10}$$

where A, F_1, F_2, F_3, F_4 and F_5 are correlations given in [21] that take into account the gaseous fuel equivalence ratio of unburned zone $\varphi_{u,eq}$, the cylinder pressure and the temperature of the unburned zone. The main advantage of the proposed correlation is that it can predict adequately the laminar burning velocity for non-stoichimetric region.

To allow for the effect of turbulence a correction factor $f_{turb.}$ is used [8]. Its value depends on engine velocity and mixture condition during the combustion phase [8]. Thus the turbulence burning velocity used in the present model is estimated from the following formula:

$$u_t = f_{turb.} \cdot u_l \tag{11}$$

DEFINITION OF GASEOUS FUEL COMBUSTION RATE - The gaseous fuel combustion rate is given by the formula:

$$dm_{comb.}^{N.G.} = dm_{fl}^{N.G.} + dm_R^{N.G.} \tag{12}$$

where $dm_{fl}^{N.G.}$ is the combustion rate caused by the spread of the flame front while $dm_R^{N.G.}$ is the combustion rate of the gaseous fuel entrained inside the burning zone due to the formation and penetration of the fuel jet.

After initiation of combustion, the combustion rate $dm_R^{N.G.}$ is the sum of an Arrhenius type reaction rates of the gaseous fuel constituents, i, as follows [8,18]:

$$dm_R^{N.G.} = \sum_i \left\{K_i \cdot (x_i)^{a_i} \cdot \left(x_{O_2}\right)^{b_i} \cdot \exp\left(-\frac{E_i}{T_b}\right)\right\} \tag{13}$$

where E_i is the activation energy of each constituent of the gaseous fuel and K_i, a_i, b_i are constants.

As already mentioned the gaseous fuel entrained through the flame front is transformed into products. Consequently the combustion rate of the gaseous fuel caused from the spread of the flame front $dm_{fl}^{N.G.}$ is given by the formula:

$$dm_{fl}^{N.G.} = \frac{\rho_u}{MB_u} \cdot \sum_i \left(MB_i \cdot y_{i,u} \cdot dV_b^{tot}\right) \tag{14}$$

Taking into account the combustion rates of the two fuels used, the total heat release rate is the sum of the liquid fuel heat release rate and the gaseous one, as follows:

$$H.R.R. = dm_{comb.}^{dies.} \cdot H_u^{dies.} + dm_{comb.}^{N.G.} \cdot H_u^{N.G.} \tag{15}$$

CARBON MONOXIDE MODEL - According to [8,18], carbon monoxide formed by the combustion process is oxidized to carbon dioxide at a rate that is relatively slow compared to the carbon monoxide formation rate. The two kinetically controlled reactions with their relative forward reaction rate constants have as follows [18]:

$$CO + HO \leftrightarrow CO_2 + H,\ \ k_1 = 6{,}76 \cdot 10^{10} \cdot \exp(\frac{T_b}{1102})$$

$$CO_2 + O \leftrightarrow CO + O_2,\quad k_2 = 2{,}51 \cdot 10^{12} \cdot \exp(\frac{-22041}{T_b})$$

The net rate of carbon monoxide formation is estimated from the differential equation:

$$\frac{d[CO]}{dt} = (R_1 + R_2) - \frac{(R_1 + R_2) \cdot [CO]}{[CO]_e} \tag{16}$$

where the one way equilibrium rates (Ri, i=1,2) are defined as follows:

$$\begin{aligned} R_1 &= k_1 \cdot [CO]_e \cdot [HO]_e \\ R_2 &= k_2 \cdot [CO_2]_e \cdot [O]_e \end{aligned} \tag{17}$$

DESCRIPTION OF THE EXPERIMENTAL FACILITIES AND TEST CASES EXAMINED

EXPERIMENTAL SETUP - In Figure 5 is given a schematic layout of the experimental setup used in the present work. The experiments were carried out on a single cylinder, naturally aspirated, four strokes air-cooled Lister LV-1 diesel engine. Information about the engine is given in Table 2. The engine is coupled to a Heenan&Froude hydraylic dynamometer. Diesel fuel is injected using a “Bryce-Berger” high-pressure fuel pump having a 6.5mm diameter plunger. The fuel pump is connected to a three-hole injector nozzle having an opening pressure of 180 bar, which is located in the center of the combustion chamber. The intake air mass flow rate is measured using a viscous type flow meter while the natural gas one is estimated using a rotary displacement gaseous fuel flow meter. Intake, exhaust and gaseous fuel temperatures are measured using Type-K thermocouples while pressures of the cylinder charge and the liquid fuel line, are measured using piezoelectric transducers and charge amplifiers. The Top Dead Center (TDC) position is estimated by using a TDC marker (magnetic pick-up). A fast data acquisition and recording system based on a PC is used to record measured data. As far as pollutant emissions are concerned, nitric oxides are measured using a chemilumescent analyzer, for the estimation of carbon monoxide concentration a nondispersive infrared analyzer is used while the unburned hydrocarbons are measured using a flame ionization detector. The smoke levels in the exhaust gas are measured using a Bosch smoke meter. The engine is supplied with natural gas obtained from the low-pressure distribution network and the adjustment of engine load under dual fuel operation is accomplished through a control valve located after the gaseous fuel flow meter.

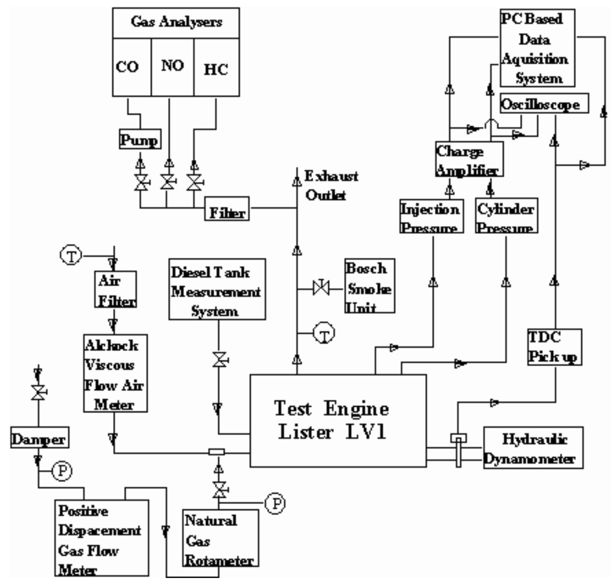

Figure 5. Schematic Layout of the Test Installation

Table 2. Engine Basic Data of Lister LV 1 diesel Engine

Type	Single Cylinder, 4-Stroke,DI
Bore	85.73mm
Stroke	82.55mm
Connecting Rod Length	148.59mm
Compression Ratio	18
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

TEST CASES EXAMINED - The static injection timing is kept constant at 26 ^{0}CA BTDC while the air inlet temperature is 23 ^{0}C for all cases examined. Measurements are taken at four different engine loads corresponding to 20%, 40%, 60% and 80% of full load relative to diesel operation, at 2000 rpm engine speed under both, normal diesel (only diesel fuel) and dual fuel operation. Under dual fuel operation an effort has been made to keep the pilot amount of diesel fuel constant, while the power output of the engine is adjusted through the amount of gaseous fuel. The measurement procedure under dual fuel operation has as follows: At a given constant engine speed pilot diesel fuel is injected to cover approximately the mechanical losses of the engine. Then, keeping constant the flow rate of liquid diesel fuel, the power output is further increased using only gaseous fuel. This procedure is followed until the desired power output is obtained. The corresponding mass flow rates of diesel fuel and natural gas are shown in Table 3 for all cases examined under normal diesel and dual fuel operation.

Table 3. Test Cases Examined at 2000 rpm Engine Speed for Various Engine Loads

2000 rpm Engine Speed			
	Dual Fuel Operation		Normal Diesel Operation
Load (%)	Diesel Fuel Consumption (kg/h)	Natural Gas Consumption (kg/h)	Diesel Fuel Consumption (kg/h)
20	0.200	0.634	0.463
40	0.196	0.855	0.630
60	0.201	0.920	0.786
80	0.198	0.973	0.982

As revealed the amount of pilot diesel fuel, under dual fuel operation varies from 0.196 – 0.201 kg/h at 2000 rpm and can be considered as constant. During the investigation, under normal diesel and dual fuel operation the following measurements are taken at each measuring point:

- Cylinder Pressure Diagrams
- Power Output
- Exhaust Gas Temperature
- Diesel and Gaseous Fuels Consumptions
- Intake Air Mass Flow Rate
- Exhaust Gas Emissions
- Exhaust Smoke Opacity

RESULTS AND DISCUSSION

To verify the ability of the model to predict apart from overall engine performance the main characteristics of dual fuel combustion process, a comparison between experimental and calculated pressure and total heat release traces is given in Figures 6 – 9 for various loads at 2000 rpm engine speed. It must be stated here that the calculated results corresponds to the normal injection timing (NIT) of the engine. In the same Figures are given the measured cylinder pressure and heat release traces under normal diesel operation for the respective engine operating conditions.

Comparing the experimental and the respective theoretical results showing in Figures 6-9, it is observed that the agreement in all cases examined is good revealing the ability of the current model to predict adequately engine performance and the total burning rate for diesel engines operating under dual fuel conditions. At this point it must be stated that predictions for cylinder pressure and total heat release rate provided in Figures 6-9 are considerably improved compared to an earlier version of the present model [7-8]. The model's ability to predict engine performance and total burning rate is encouraging taking into account the complicated combustion process of natural gas in a compression ignition environment. The experimental heat release rates under normal diesel and dual fuel operation shown in Figures 6-9 were obtained from the analysis of the corresponding experimental cylinder pressure traces using a diagnostic code [27].

Observing the experimental cylinder pressure traces under normal diesel and dual fuel operation at each test case examined (Figures 6-9), it is obvious that the presence of natural gas in the cylinder charge affects cylinder pressure. Specifically, the pressure under dual fuel operation diverges from the respective values under normal diesel operation for the same engine conditions. 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 compression stroke is the result of the higher specific heat capacity of the natural gas – air mixture compared to the one of air for normal diesel operation. After initiation of combustion the rate of cylinder pressure rise under normal diesel operation is higher compared to the respective one under dual fuel operation. As shown later on, it is the result of the higher combustion rate of diesel fuel during the ignition delay period. Furthermore, for each load examined, it is observed that the value of maximum cylinder pressure observing under dual fuel operation is lower and occurs later compared to the respective value under normal diesel operation. This is encouraging, since obviously no danger exists for the engine structure because of cylinder pressure if we decide to apply this technology on conventional diesel engines.

Examining the total heat release curves shown in Figures 6–9, it is revealed that the presence of natural gas in the cylinder charge affects seriously the combustion process. It is observed that the initiation of combustion under dual fuel operation starts later compared to the respective one under normal diesel operation while the difference becomes more evident at high engine load. During the initial stage of combustion under dual fuel operation an interesting behavior of the total burning rate is observed. It is revealed that the initial peak in the heat release curve observed under dual fuel operation is lower and appears later compared to the respective one under normal diesel operation.

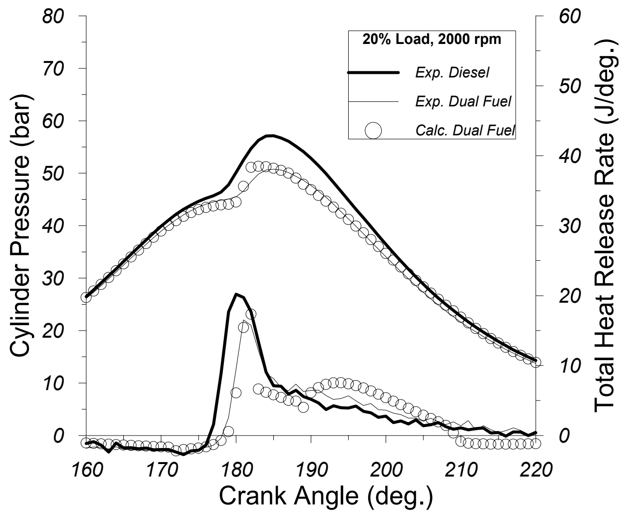

Figure 6. Theoretical and Experimental Pressure and Total Heat Release Traces under Dual Fuel Operation and experimental one under normal diesel operation at 20% Load and 2000 engine speed.

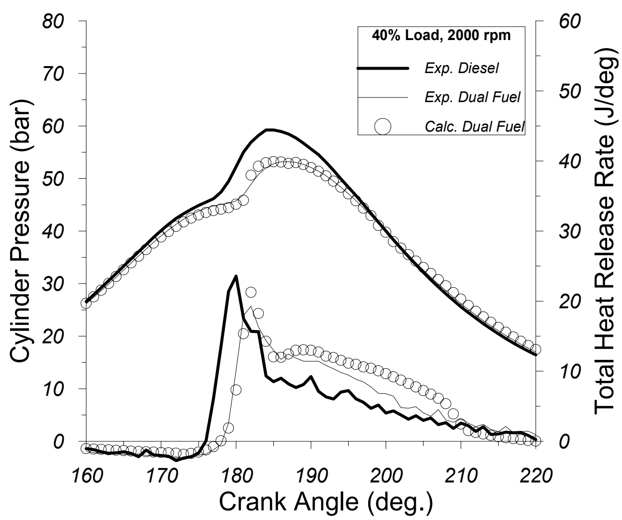

Figure 7. Theoretical and Experimental Pressure and Heat Release Traces under Dual Fuel Operation and experimental one under normal diesel operation at 40% Load and 2000 engine speed.

It results from the lower amount of diesel fuel burnt during the premixed controlled combustion phase and also to the fact that the combustion of gaseous fuel hasn't yet progressed enough since the cylinder charge conditions do not favor the existence of the flame front. Thus the total heat released during the premixed combustion phase is considerably lowered compared to the respective one observed for the same engine load under normal diesel operation (Figure 10). Furthermore, the combustion rate during the premixed controlled combustion phase under normal diesel operation is considerable sharper for all load examined compared to the respective values under dual fuel operation. It must be stated here that as engine load increases under dual fuel operation the duration of premixed controlled combustion phase becomes slightly lower since the combustion of the premixed fuel is faster.

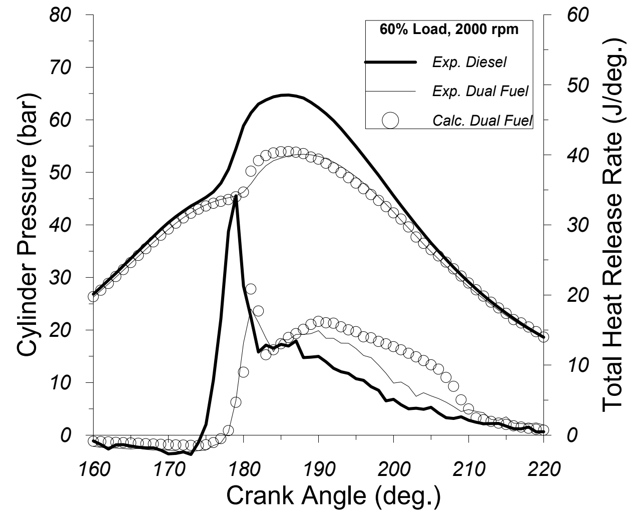

Figure 8. Theoretical and Experimental Pressure and Heat Release Traces under Dual Fuel Operation and experimental one under normal diesel operation at 60% Load and 2000 engine speed.

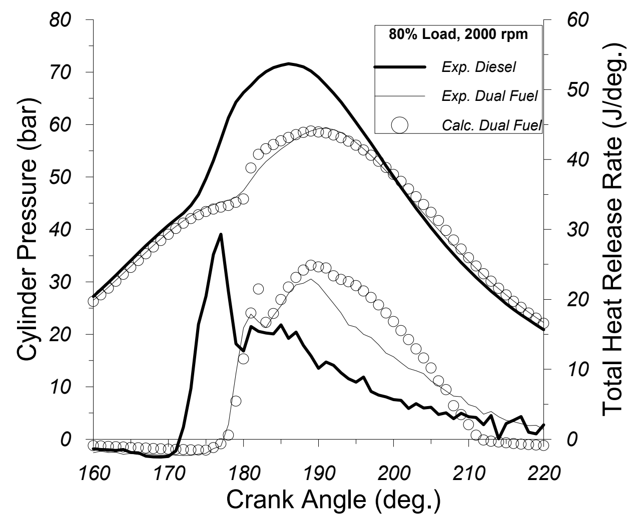

Figure 9. Theoretical and Experimental Pressure and Heat Release Traces under Dual Fuel Operation and experimental one under normal diesel operation at 80% Load and 2000 engine speed.

The lower heat release rate during premixed combustion phase and the higher specific heat capacity of the gaseous mixture under dual fuel operation are the main causes for the lower and delayed appearance of maximum combustion pressure observed under dual fuel operation.

For all cases examined during the second phase of combustion (diffusion phase) it is revealed that the calculated heat release trace under dual fuel mode is slightly overestimated compared to the respective measured one. This is the result of the overestimation of the turbulent flame speed. It is obviously that the part of the model which simulates the combustion mechanism during the diffusion phase needs a further improvement. Comparing the heat release results during this part of combustion under pure diesel and dual fuel modes, it is observed that the total burning rate under dual fuel operation is higher compared to the one under normal diesel operation for all loads examined. The difference is lower at low engine loads due to the inefficient quality of gaseous fuel combustion since the conditions of the cylinder charge don't contribute to the fast spread of the flame front. On the other hand, at high engine load the difference becomes evident revealing that the quality of the gaseous fuel utilization is improved compared to the respective one at low load. This improvement results from the fast spread of the flame front surrounding the burning zone. But, as shown in Figures 6-9, this difference doesn't affect significantly the cylinder pressure concerned to the respective period since we are in the expansion stroke.

In Figure 11 is given the variation of duration of combustion under normal diesel and dual fuel operation with engine load at 2000 rpm engine speed. As recognized the duration of combustion under dual fuel operation is higher compared to the one under normal diesel for all test cases examined. At low load the poor utilization of gaseous fuel leads to a considerable difference while at high load the improvement of gaseous fuel combustion causes the duration of combustion under dual fuel operation to converge to the respective value of normal diesel.

Figure 12 provides the comparison between calculated and experimental values of ignition delay period under dual fuel operation versus engine load at 2000 rpm engine speed. For comparison, in the same figure is given the ignition delay period of pure diesel operation. As observed, under dual fuel operation the model predicts adequately the trend of the ignition delay as engine load changes. At low load the model under-predicts slightly the absolute value which is expected from a two-zone model. As shown for all cases examined ignition delay under normal diesel operation is considerable lower compared to the respective values under dual fuel operation. Furthermore, under dual fuel operation as engine load increases supplying the engine with higher amounts of gaseous fuel, ignition delay follows an opposite trend compared to the one under normal diesel operation since ignition delay increases with load resulting to high differences between the modes i.e. dual fuel and normal diesel operation.

Figure 13 provides the comparison between calculated and experimental values of engine efficiency under dual fuel operation versus engine load at 2000 rpm engine speed. The variation of engine brake efficiency under normal diesel operation is given in the same figure. As observed the model predicts accurately the measured values of engine efficiency. It must be stated here that the measured efficiency is estimated from the measured brake power output and mass flow rates of diesel and natural gas. Observing Figure 13 it is revealed that for all cases examined the efficiency under dual fuel operation is considerable lower compared to the respective one under normal diesel. It is the result of the lower, compared to normal diesel, premixed controlled combustion rate observed under dual fuel operation during the initial stage of combustion.

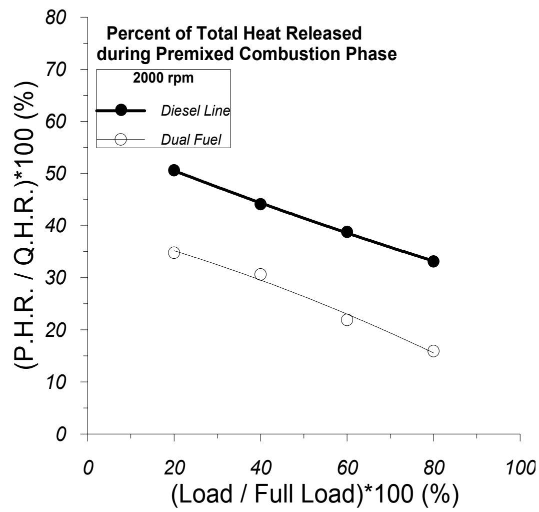

Figure 10. Comparison between the percents of Total Heat Released during Premixed Combustion Phase under Dual Fuel and normal diesel operation versus engine load for 2000 engine speed.

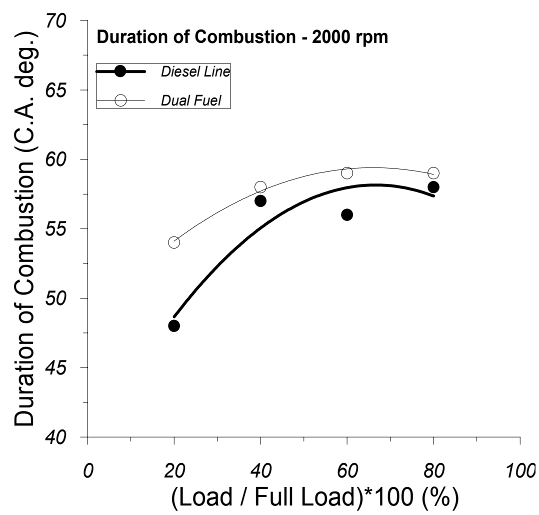

Figure 11. Comparison between the Duration of Combustion under Dual Fuel and normal diesel operation versus engine load for 2000 engine speed.

The higher values of diffusion controlled combustion rates observed under dual fuel operation don't improve engine efficiency since they occur late in the expansion phase. At high load the values of engine efficiency under normal diesel and dual fuel operation seem to converge. This is the result of the considerable improvement of the gaseous fuel utilization observed at high load. In both cases the rather low absolute values are due to the poor mechanical efficiency of the single cylinder test engine.

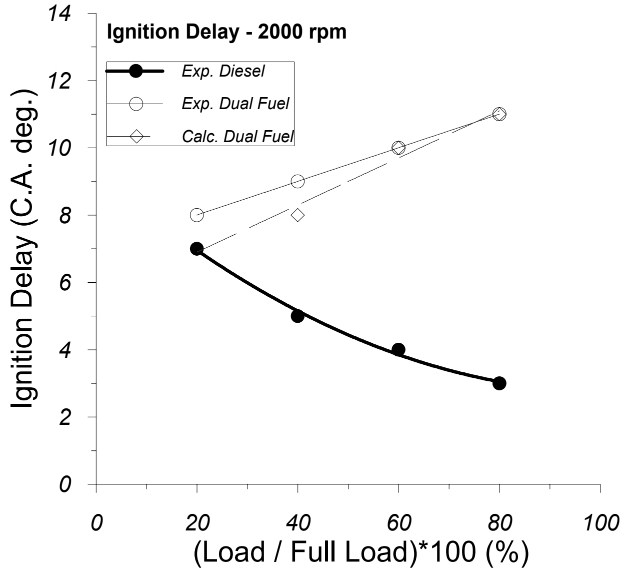

Figure 12. Theoretical and Experimental Ignition Delay Period under Dual Fuel Operation and experimental one under normal diesel operation for 2000 engine speed.

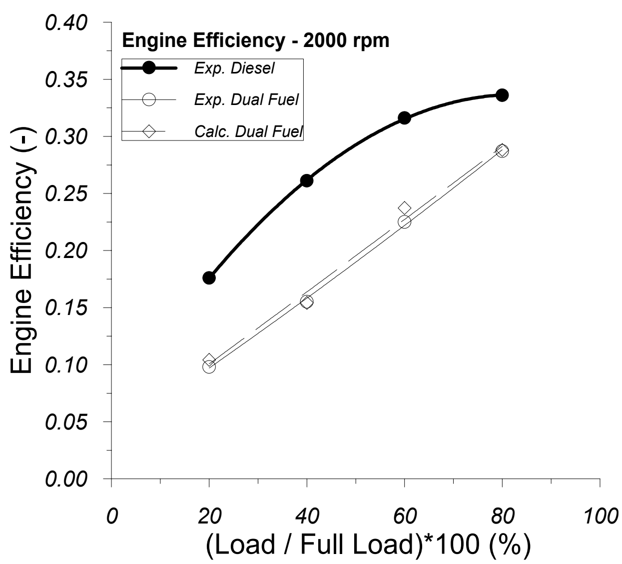

Figure 13. Theoretical and Experimental Engine Efficiency under Dual Fuel Operation and experimental one under normal diesel operation versus engine load for 2000 engine speed.

The comparison between measured and calculated values of brake specific NO emissions under dual fuel operation with engine load is given in Figure 14 for 2000 rpm engine speed. In the same figure is given the brake specific NO emissions under normal diesel operation. Examining the theoretical and the experimental values under dual fuel operation, it is revealed that the model predicts with adequate accuracy the trend of NO concentration with engine load. The theoretical values are slightly lower compared to the experimental ones but it is normal for a two-zone model since it under predicts the burning zone temperature which affects seriously the formation mechanism of the oxides of nitrogen (NO). As revealed from Figure 14, NO concentration under dual fuel operation is considerable lower compared to normal diesel operation. It is the result of the lower concentration of oxygen inside the burning zone due to the presence of the gaseous fuel and the lower premixed controlled combustion rate which results to lower burning temperatures compared to the ones under normal diesel operation.

Figure 15 provides the comparison between calculated and measured values of brake specific carbon monoxide emissions under dual fuel operation as a function of engine load for 2000 rpm engine speed. In the same figure is given the brake specific CO emissions under pure diesel operation. As observed the model predicts adequately the trend following the measured values of carbon monoxide concentration under dual fuel operation. Most encouraging is that the model manages to predict the effect of load on carbon monoxide concentration when the engine runs under dual fuel operating conditions. From Figure 15, it is revealed that carbon monoxide concentration under dual fuel operation is significantly higher compared to the respective one under normal diesel operation even though considerable improvement is observed at high load.

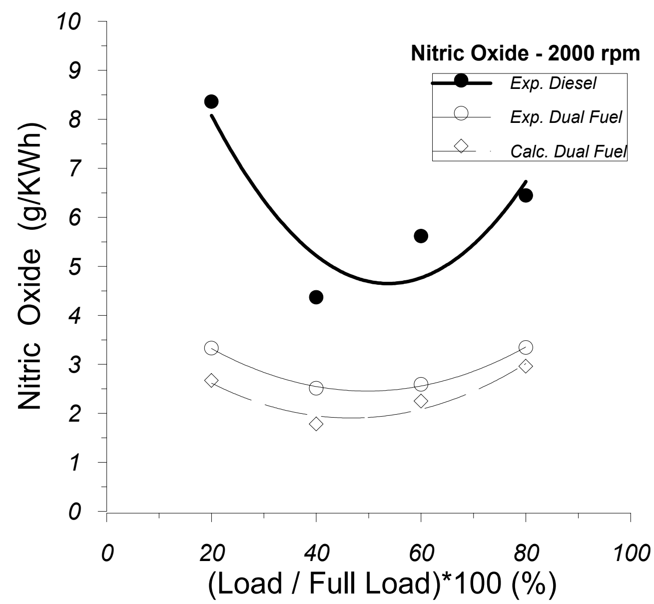

Figure 14. Theoretical and Experimental Nitric Oxides under Dual Fuel Operation and experimental one under normal diesel operation versus engine load for 2000 engine speed

Finally Figure 16 provides the comparison between calculated and measured values of brake specific soot emissions under dual fuel operation versus engine load for 2000 rpm engine speed. In the same figure are given soot values under normal diesel operation at the same engine operating conditions. Observing the experimental and respective theoretical values under dual fuel operation, it is revealed that the current model predicts with adequate accuracy the measured values of soot emissions and their variation with engine load.

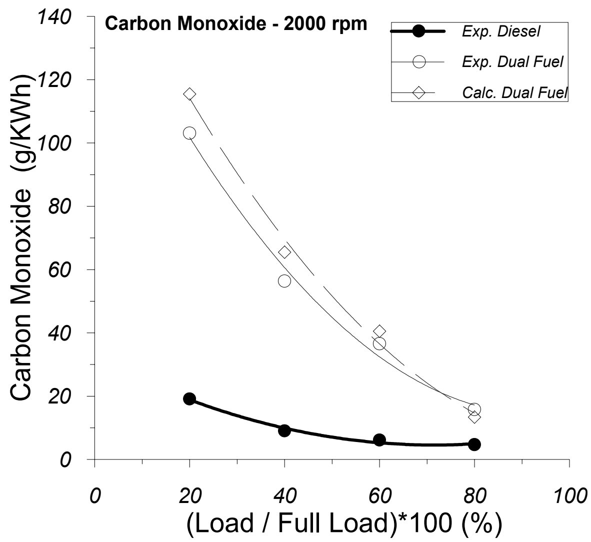

Figure 15. Theoretical and Experimental Carbon Monoxide under Dual Fuel Operation and experimental one under normal diesel operation versus engine load for 2000 engine speed

Carefully observing soot emissions under normal diesel and dual fuel operation, it is revealed that under dual fuel operation the increase of engine load leads to a slight decrease of soot emissions which is the opposite of what takes place under normal diesel operation. Thus, the dual fuel operation could be a potential way of reducing soot emissions. The significant reduction of soot under dual fuel operation is due to the fact that the increase of engine power is accomplished through the increase of gaseous fuel mass flow rate. Thus liquid fuel remains constant and since gaseous fuel does not contribute to soot formation we have extremely low soot values at the engine exhaust.

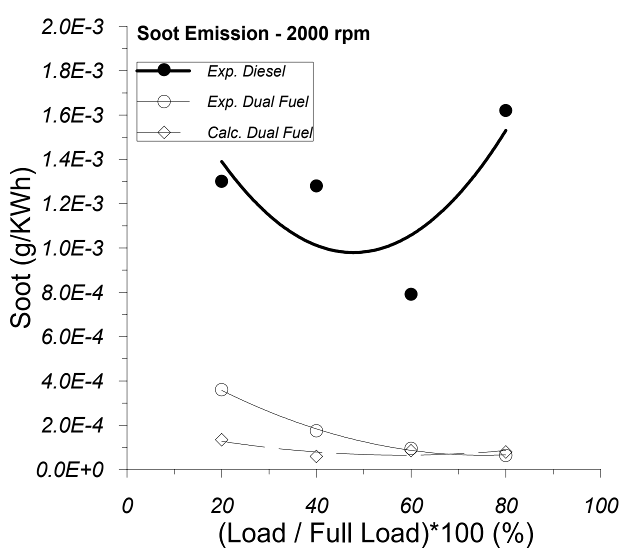

Figure16. Theoretical and Experimental Soot emissions under Dual Fuel Operation and experimental one under normal diesel operation versus engine load for 2000 engine speed

POSSIBILITIES FOR IMPROVING DUAL FUEL OPERATION

As shown in Figures 13 and 15, the main disadvantage of dual fuel operation is the negative impact of dual fuel combustion on engine efficiency and CO emissions. In the present contribution a first attempt was made by using the current model to investigate the use of advanced injection timing to compensate the negative effect of dual fuel combustion on engine efficiency, NO, CO and Soot emissions.

Figure 17 provides the variation of the calculated values of engine efficiency under dual fuel operation as a function of engine load for various injection timings. It must be stated here that at each engine load the injection timing was increased four (4) and eight (8) degrees respectively compared to the normal injection timing (NIT). In the same figure are given experimental engine efficiency values under normal diesel operation referring to the normal injection timing. Examining the theoretical values under dual fuel operation, it is revealed that the injection advance leads to an improvement of engine efficiency compared to the one at normal injection timing. This improvement is shown to be more sensible at high engine load and high values of injection timing, where the values of engine efficiency under normal diesel and dual fuel operation seem to converge. However, observing Figure 17 it is revealed that for all cases examined the efficiency under dual fuel operation is still lower compared to the respective one under normal diesel operation.

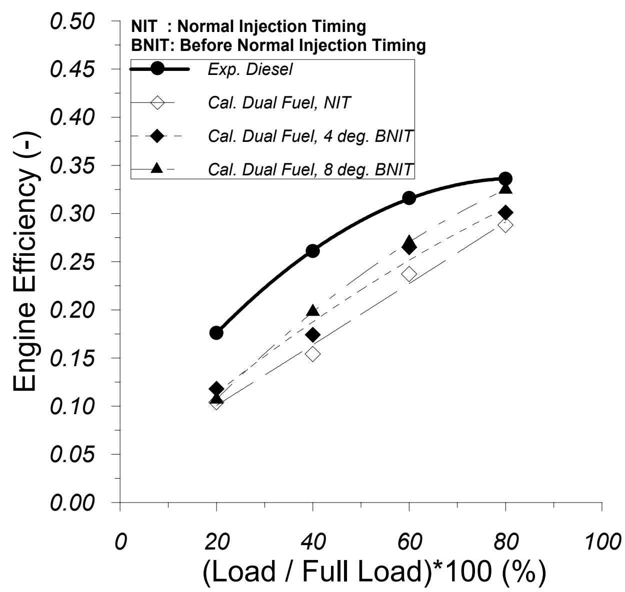

Figure 17. Theoretical Engine Efficiency Variation versus engine load at 2000 rpm engine speed for various values of Injection Timing.

As far as the effect of the advanced injection timing on emissions is concerned, Figures 18, 19 and 20 provide the variation of the calculated values of NO, CO and Soot emissions respectively under dual fuel operation versus engine load for various injection timing. In the same figures are given the measured values for NO, CO and Soot under normal diesel operation at the normal injection timing. It must be stated here that in Figure 20 two y-axis are used in order to be clearer the effect of advanced injection timing on soot formation under dual fuel operation. Observing these Figures, it is revealed that the increase of the injection timing under dual fuel operation affects as expected NO, CO and Soot emissions. Specifically, under dual fuel operation at high engine load conditions the advancement of injection timing leads to a decrease of CO and soot emissions compared to the respective values referring to normal injection timing. In spite of this positive effect, NO emissions are increased but their concentration values are lower compared to the ones under normal diesel operation. At low load the effect of advanced injection timing is relatively low.

Taking into account these results, it appears that the use of the advanced injection timing can be a potential way for increasing engine efficiency and reducing CO emissions. On the other hand part of the benefit obtained by dual fuel operation has been lost. Thus, it is required a more detailed investigation to estimate the proper combination of engine settings to improve engine behavior under dual fuel mode as far as engine efficiency and CO emissions are concerned.

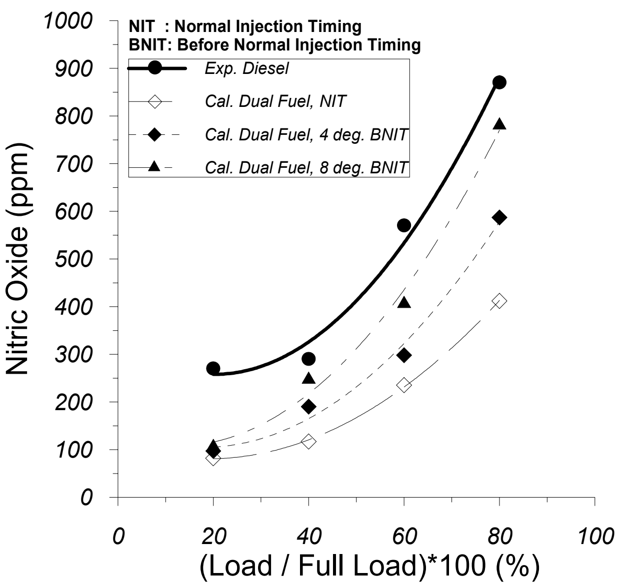

Figure 18. Theoretical NO concentration versus engine load at 2000 rpm engine speed for various values of Injection Timing.

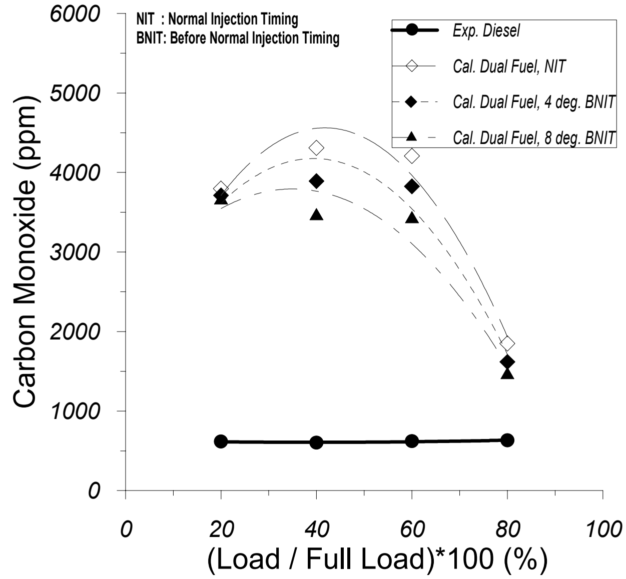

Figure 19. Theoretical Carbon Monoxide concentration versus engine load at 2000 rpm engine speed for various values of Injection Timing.

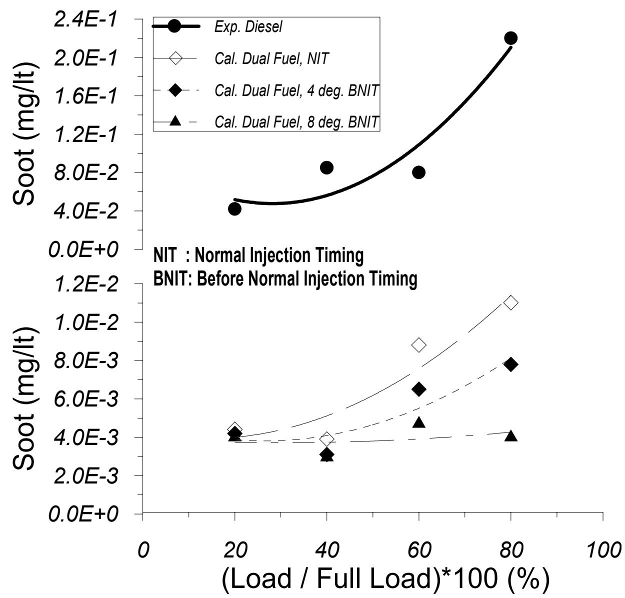

Figure 20. Theoretical Soot Concentration versus engine load at 2000 rpm engine speed for various values of Injection Timing.

CONCLUSIONS

Under the present work an existing two-zone model has been modified to simulate more accurately the operation of dual fuel engines for the prediction of performance and pollutant emissions. Modifications conducted concerned the more detailed description of the various physical phenomena related to the combustion process of an engine operating under dual fuel conditions. Modifications focused on the fundamental description of the simultaneous combustion of natural gas and pilot diesel fuel. For this reason a flame model was developed and employed to describe the combustion of natural gas. According to the philosophy of the current model, combustion of gaseous fuel initiates after the auto-ignition of the prepared diesel fuel and its burning rate depends on the speed with which flame front formed around the burning zone spreads and on the reaction rate of the gaseous fuel entrained inside the burning zone.

To validate the model and understand the combustion mechanism under dual fuel mode an extended experimental investigation was conducted on a high speed DI single cylinder test engine located at the author's laboratory. Measurements have been taken at various loads at constant engine speed under normal diesel and dual fuel operation. For this reason the engine has been properly modified to operate under dual fuel mode without changing its main configuration.

Comparing calculated and measured values under dual fuel operation a good coincidence is observed for both performance and pollutant emissions. Specifically, the model predicts with reasonable accuracy the absolute values but most important it predicts the trends of the combustion and pollutants formation mechanisms.

Concerning the comparison between normal diesel and dual fuel operation, it is revealed that the simultaneous use of diesel and natural gas leads to lower burning rate during the premixed combustion phase which results to lower peak combustion pressure observed later compared to normal diesel operation. Furthermore, dual fuel operation leads to degradation of engine efficiency and increase of ignition delay period compared to pure diesel operation.

As far as emissions dual fuel operation affects positively (reduction) nitric oxides and soot emissions. This is extremely important if we consider the difficulties for controlling both pollutants, NO and Soot, in DI diesel engines. On the other hand the presence of gaseous fuel in the cylinder charge causes a considerable increase of carbon monoxide emissions compared to the ones observed under normal diesel operation. The negative impacts of dual fuel combustion on engine efficiency and CO emissions could be reduced by advancing injection timing but it seems to have a negative effect on NO emissions. This requires a detailed investigation to estimate the proper combination of engine settings to improve engine behavior under dual fuel mode as far as engine efficiency and CO emissions are concerned.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

D	cylinder bore, *m*
E	activation energy, *J/Kmole*
H_u	Lower Heating Value, *J/kg*
h	specific enthalpy, *J/kg*
K_1	constant for liquid fuel reaction rate
m	mass, *kg*
$\dot{m}$	mass flow rate, *kg/s*
MB	Molecular Weight, *kg/Kmole*
N	engine speed, *rpm*
P	pressure, *Pa*
P_{O2}	partial pressure of oxygen, *bar*
Q	heat transfer to walls, *J*
R_m	universal gas constant, *J/KmoleK*
R	gas constant, *J/kgK*
S	penetration length, *m*
t	time, *s*
T	absolute temperature, *K*
U	Total Internal Energy, *J*
u_l	laminar flame velocity, *m/s*
u_t	turbulent flame velocity, *m/s*
u_{jet}	Jet Penetration velocity, *m/s*
V	volume, m^3
x	mass fraction on species
y	kmole fraction on species

Greek

θ	initial jet angle, *rad*
ΔP	pressure difference, *Pa*
λ	thermal conductivity, *W/mK*
μ	dynamic viscosity, *kg/ms*
ρ	density, kg/m^3
φ	crank angle degree
φ_{eq}	equivalence ratio

Subscripts

b	burned zone
comb.	combustion
cyl.	cylinder
D	diesel fuel

del delay

e equilibrium

prep preparation

form formation

fl gaseous fuel combusted due to the spread of the flame front

i index denoting the species of gaseous fuel

inj injected diesel fuel

m bulk average value

N.G. Natural Gas

o initial value

ox oxidation

R combustion of the gaseous fuel entrained into the burning zone

tot total

u unburned zone

w wall

Dimensionless numbers

Re Reynolds

Abbreviations

CR Compression Ratio

CO Carbon Monoxide

DI Direct Injection

NO Nitric oxide

H.R.R. Total Heat Released Rate

ppm parts per million (volume)

TDC Top Dead Centre