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ABSTRACT

During the last years a great deal of effort has been made for the reduction of pollutant emissions from direct injection Diesel Engines. Towards these efforts engineers have proposed various solutions, one of which is the use of gaseous fuels as a supplement for liquid diesel fuel. These engines are referred to as dual combustion engines i.e. they use conventional diesel fuel and gaseous fuel as well. The ignition of the gaseous fuel is accomplished through the liquid fuel, which is auto-ignited in the same way as in common diesel engines. One of the fuels used is natural gas, which has a relatively high auto-ignition temperature. This is extremely important since the CR of most conventional diesel engines can be maintained. In these engines the released energy is produced partially from the combustion of natural gas and from the combustion of liquid diesel fuel. The aim for the usage of dual-fuel combustion systems is mainly to reduce the particulate emissions (soot) by replacing diesel fuel with natural gas partially or entirely. For this reason in the present work are given preliminary results of a theoretical investigation using a simulation model developed for dual fuel engines. The model is a two-zone combustion one, taking into account, on a zonal basis, details of diesel fuel spray formation and the mixing with the surrounding gas, which is a mixture of air and natural gas. The main difference from a conventional diesel engine combustion system is that the natural gas is already mixed and ready for combustion. The natural gas burning initiates after the ignition of the diesel fuel and its rate depends on the rate of entrainment of surrounding gas inside the fuel jet formed. A soot model has been used to estimate the formation of soot while a detailed equilibrium model has been used to determine the concentration of chemical species. For nitric oxide the extended Zeldovich mechanism is used. The model is applied on a single cylinder test engine located at the author's laboratory at various operating conditions of the engine. The amount of liquid fuel supplemented by natural gas has been varied and its affect on engine performance and emissions has been examined. From these preliminary results it is revealed a serious effect on the heat release rate inside the engine cylinder and a reduction of particulate emissions when compared to experimental data obtained from the engine using diesel fuel only.

INTRODUCTION

The use of alternative gaseous fuels in engines for the production of power has been increasing worldwide. This has been prompted by the cleaner nature of their combustion compared to conventional liquid fuels as well as their relative increased availability at attractive prices. Diesel engines can be made to operate on gaseous fuels efficiently [1-4][1]*. These engines usually have the gaseous fuel mixed with air at the start of the compression stroke. This mixture does not auto-ignite due to its higher self ignition temperature. The resulting mixture after the compression stroke is ignited through the ignition of the amount of diesel fuel injected. Diesel fuel can auto-ignite readily creating ignition sources for the surrounding gaseous fuel mixture. Most current dual fuel engines are made to operate either on gaseous fuels with diesel ignition or only on liquid fuel injection as normal diesel engines [1-4].

A computer based mathematical model can provide an adequate way for describing details of the complicated mixing, combustion and pollutants emission processes in these engines. The present contribution describes a model that simulates dual-fuel combustion using a two-zone approach to simulate the combustion of premixed gas/air charge.

Its main purpose is to describe the operation of existing diesel engines, where part of the liquid fuel is replaced by gaseous fuel. These engines are usually referred to as fumigated ones and the replacement of liquid fuel with gaseous aims mainly to the reduction of soot emissions.

Furthermore due to the partial replacement of liquid fuel with gaseous one no specific modifications are required and thus the technique can be applied on existing engines.

1. Numbers in brackets designate references at the end of the paper.

The main aim of the present model is to estimate, apart from power and efficiency, the concentration of soot emissions and nitrogen oxides at the engine exhaust (NO), by replacing diesel fuel with natural gas partially at various amounts ranging from 0-40%. For this reason preliminary results from the modeling are provided for various engine operating conditions. To validate the model under normal diesel operations and to compare the findings under dual-fuel operation an extended experimental is conducted on a single cylinder test engine. From the analysis of results obtained it is revealed that the simulation model developed predicts adequately engine performance and pollutants under normal diesel engine operations. Furthermore comparing the findings for dual-fuel operation with the standard diesel one, a serious effect of the gaseous fuel is observed on engine performance and soot, NO emissions. To validate the findings of the dual-fuel modeling an experimental investigation is currently under way and results will be presented in the near future.

GENERAL DESCRIPTION OF THE MODEL

The cylinder charge at the start of compression stroke is a homogeneous mixture of gaseous fuel and air, that has different thermodynamic properties from the air charge in normal diesel engines. We replace a part of liquid fuel with gaseous fuel in order to have the same power output.

For combustion a two-zone model is considered [5]. In each zone there is a uniformity in space of pressure, temperature and composition at each instant of time, neglecting heat transfer between the zones. The first zone consists of air/gaseous fuel (unburned zone) and the second zone consists of combustion products unburned mixture and evaporated liquid fuel (burning zone). These zones are separated by the area of the conical jet, which is formed during the injection of the diesel fuel [6]. Steady state jet theory, including wall impingement, is used to describe the fuel-air mixing process. Ignition of the gaseous fuel, which is already mixed and ready for combustion, occurs as a result of the diesel jet auto-ignition after the ignition delay period of the diesel fuel [4]. Due to the lean nature of the surrounding mixture no flame is considered, i.e. the combustion rate of gaseous fuel depends on its entrainment rate inside the burning zone. Of course this consideration will have to be modified in the case of normal dual-fuel engines where the liquid fuel is a very small percentage of the total and used for the ignition of the main charge.

For the heat transfer calculation between the charge and cylinder wall, the Annand formula is employed [7]. For prediction of soot emissions, the amount of net soot is calculated considering the difference between the rates of soot formation and soot oxidation inside the burning zone [6-14].

Dissociation of combustion products is taken into account by incorporating the Vickland et al [15] method including eleven species. For the formation of nitric oxides the extended Zeldovich chain reaction mechanism is considered [6-10,13-16,19].

The liquid fuel used is dodecane ($C_{12}H_{26}$) representing widely the commercial diesel fuel and the gas fuel used is a mixture of methane (CH_4) and propane (C_3H_8) in proportion (90%-10%) in (v/v), which is a typical representative of natural gas fuel.

MATHEMATICAL TREATMENT

CONSERVATION OF ENERGY – As already stated inside the combustion chamber there are considered two zones, a burning one consisting of air, evaporated fuel, gaseous fuel plus burned products and an unburned one consisting of air and gaseous fuel. Each zone posses its own temperature and composition. The pressure is the same for the both zones. The first low equation for each zone may be written as [6,9]

$$dU_{b,u} = dQ_{b,u} - P \times dV_{b,u} + \sum (dm_{bu} \times h_{bu}) + dm_{fb} \times h_f \tag{1}$$

where "$dm_{fb}h_f$" is the total enthalpy addition to the burning zone from the liquid diesel fuel and "$\Sigma dm_{bu}h_{bu}$" is the total enthalpy addition to the burning zone from the unburned zone due the entrainment rate of unburned mixture. The internal energies and enthalpies in the above equation considered are total, so that the heat of combustion is taken into account implicitly. The temperature, pressure and volume of each zone is obtained from the integrating of a set of three ordinary first order differential equations with unknowns "T_b", "T_u" and "P". These equations have been derived after some mathematical elaboration from the first law and perfect gas state equations for both zones and the volume balance. The system of the three equations has as follows

$$dT_u = \left(dQ_u - R_u T_u dm_u + V_u dP + h_u dm_u - u_u dm_u \right) / m_u C_{pu} \tag{2}$$

$$dT_b = \left\{ \begin{array}{l} dQ_b - R_b T_b dm_b - m_b T_b dR_b + V_b dP + h_u dm_{ub} \\ + h_{fev} dm_{fev} - u_b dm_b - m_b \left[\sum_{k=\text{species}} dy_k u_k \right]_b \end{array} \right\} / m_b C_{pb} \tag{3}$$

$$dP = \left\{ \begin{array}{l} \frac{R_u}{C_{pu}}\left[dQ_u - R_u T_u dm_u + h_u dm_u - u_u dm_u\right] + \\ R_u T_u dm_u + R_b T_b dm_b + m_b T_b dR_b - PdV + \\ \frac{R_b}{C_{pb}}[dQ_b - R_b T_b dm_b - m_b T_b dR_b + h_u dm_{ub} + \\ h_{fev} dm_{fev} - u_b dm_b - m_b \left(\sum_{k=species} dy_k u_k \right)_b] \end{array} \right\} / \left(V_{cyl} - \frac{R_u}{C_{pu}} V_u - \frac{R_b}{C_{pb}} V_b \right) \tag{4}$$

The values of "T_b", "T_u" and "P" must satisfy the continuity equation and volume balance as well as, that

$$\begin{aligned} m_u + m_b &= m_{total} \\ V_u + V_b &= V_{cyl} \end{aligned} \tag{5}$$

HEAT TRANSFER MODEL – At each time step, we calculate the total area of the cylinder as

$$A_{tot} = A_{piston} + A_{liner} + A_{cyl.Head} \tag{6}$$

Once the area is known and employing the Annand expression [14], we have

$$dQ = A_{tot} \times \left[a \times \lambda \times \frac{Re^b}{D} \times \left(T_g - T_w\right) + c \times \left(T_g^4 - T_w^4\right) \right] \tag{7}$$

where "α,b,c" are constants and "λ" is the thermal conductivity. The only problem existing is the application of the above equation during the combustion stroke, since the burning zone is not entirely in contact with the cylinder wall surfaces. [8] For this reason a bulk average temperature of both zones is used as follows

$$T_g = \frac{\sum_{i=b,u} m_i \times C_{vi} \times T_i}{\sum_{i=b,u} m_i \times C_{vi}} \tag{8}$$

The thermal conductivity and dynamic viscosity are calculated using this bulk average temperature. The total heat exchange rate is then distributed among the two zones according to their mass, temperature and specific heat capacity as follows

$$dQ_{b,u} = dQ \times \frac{m_i \times C_{vi} \times T_i}{\sum_{i=b,u} m_i \times C_{vi} \times T_i} \tag{9}$$

ESTIMATION OF INJECTION RATE – The volume flow rate of diesel fuel and its injection rate are calculated using the following equations :

$$\begin{aligned} \dot{q}_f &= C_{Dinj} \times \frac{\pi \times d_{noz}^2}{4} \sqrt{\frac{2 \times \Delta P_{noz}}{\rho_f}} \\ \dot{m}_f &= \rho_f \times \dot{q}_f \end{aligned} \tag{10}$$

where "ΔP_{noz}" is the pressure difference across the injector hole as

$$\Delta P_{noz} = P_{fl} - P_{cyl} \tag{11}$$

where "ρ_f" is the liquid fuel density, "d_{noz}" is the diameter of the nozzle and "C_{Dinj}" is the nozzle discharge coefficient. The fuel line pressure "P_{fl}" is an input to the program together with the dynamic injection timing. The total mass of injected liquid fuel per cylinder cycle is determined from the integration of EQ. (10) [16].

MASS ENTRAINMENT RATE – After initiation of fuel injection the burning zone begins to form and penetrates into the combustion chamber. The present model, while based on simplified relations, represents the above mechanism with satisfying accuracy. For jet penetration the proposal of Hiroyasu et al [13] has been adopted.

Inside the jet we assume uniformity of density, temperature and composition. The resulting fuel jet is considered to have the shape of a cone. The volume of unburned mixture entrained by the jet is derived from its volume change. The jet is considered to be steady and consisting of a homogenous gas mixture and its penetration length is given as [10,13]

$$S = 2.95 \times \left(\frac{\Delta P_{noz}}{\rho_u} \right)^{0.25} \times \sqrt{d_{inj} \times t} \tag{12}$$

From the previous equation we obtain, after derivation with respect to time "t", the jet velocity as

$$U = 1.475 \times \left(\frac{\Delta P_{noz}}{\rho_u} \right)^{0.25} \times \sqrt{d_{inj} \times t} \tag{13}$$

If "S_2" and "S_1" represent the jet tip position at times "t_2=t_1+dt" and "t_1" respectively, then the mass entrainment rate before impingement is given by

$$dm = \rho_u \times \frac{\pi}{3} \times \tan^2 \theta \times \left(S_2^3 - S_1^3\right) \tag{14}$$

where "θ" is the jet cone angle given by

$$\tan^2 \theta = \frac{0.75 \times 0.001 \times \rho_u}{\rho_o} \tag{15}$$

In most models the assumption is made that the burning zone after impingement follows a path parallel to the cylinders walls. In the present work the existing well tested wall jet theory of "Glauert" [17] is used to determine the burning zone history after wall impingement upon the cylinder walls. At this point the unburned mixture is entrained only from the wall jet front.

The entrainment rate into the wall jet is estimated from its volume change as follows,

$$dm = \rho_u \times \frac{\pi}{3} \times R_c^2 \times U \times \tan^2\theta \times \frac{\left(t_{w2}^{1.5} - t_{w1}^{1.5}\right)}{4.5 \times t_o^{0.5}} \tag{16}$$

where "t_o" is the transient time for the change of the free jet into a wall jet, "R_c" is the cylinder radius and "t_w" is the wall travel time.

IGNITION DELAY MODEL – Before the injected liquid fuel ignites, it undergoes a delay period. The ignition delay period is estimated from the following correlation [18]

$$S_{pr} = \int_0^t \frac{1}{a_{del} \times P^{-0.757} \times \exp\left(\frac{5500}{T_b}\right)} dt \tag{17}$$

Ignition initiates when the previous integral becomes equal to one, the integration having started from the initiation of injection. At this point we must state that due to its lower cetane number, auto-ignition of the gaseous fuel is avoided, since in the opposite case serious problems would arise [11]. Thus the liquid fuel is the ignition source for the gaseous fuel.

BURNING MODEL FOR THE LIQUID FUEL – The semi-empirical model of Whitehouse-Way is used for calculating the rate of combustion of Diesel fuel. The preparation rate of injected fuel is given by [5]

$$dm_{fpr} = k \times m_{fi}^{1-x} \times m_{fupr}^{x} \times P_{O_2}^{m} \tag{18}$$

where "m_{fi}" is the total amount of fuel injected up to the considered time, "m_{fupr}" is the total unprepared fuel, "P_{O2}" is the partial pressure of available oxygen and "k,x,m" are constants.

The burning rate of the diesel fuel is effected by an Arrhenious type equation. This burning rate is given by

$$dm_{fb} = k' \times P^{0.757} \times \exp\left(\frac{-5500}{T_b}\right) \times \min\left(m_{fav}, m_{aav} \times \phi_{st}\right) \tag{19}$$

where "m_{fav}" is the total amount of unburned fuel, "m_{aav}" is the total amount of unburned air and "φ_{st}" is the stoichiometric equivalence ratio of the liquid fuel.

At the beginning of the combustion period, combustion is controlled by the reaction rate. In a short time, after the prepared fuel during the delay period is consumed, combustion is controlled by the preparation rate.

ESTIMATION OF GASEOUS FUEL BURNING RATE – As already stated in the present work, we replace a part of liquid fuel with gaseous fuel in order to have the same power output. The working media at the start of the compression stroke is a mixture of air and gaseous fuel. Thus the temperature level of the mixture during the compression stroke is very important for establishing whether auto-ignition takes place. Experiments in combustion bombs have shown that auto-ignition of gaseous fuel under diesel-like conditions requires higher temperatures from those achieved during the compression stroke [1,2,3], for this reason auto-ignition is avoided.

During the combustion stroke the mass of the gaseous fuel, entrained inside the burning zone, is proportional the entrainment rate of unburned mass. The combustion rate of gaseous fuel is controlled by the entrainment rate and the local conditions inside the burning zone. The reaction rate is determined using an Arrhenious type equation as follows, [10]

$$\underset{i=CH_4,C_3H_8}{dm_{g_i fb}} = \kappa'' \times \left(\frac{m_{g_i fav}}{m_b}\right)^{z_1} \times \left(\frac{m_{O_2 av}}{m_b}\right)^{z_2} \times \exp\left(\frac{-E_{g_i f}}{R_m \times T_b}\right) \tag{20}$$

where "m_{gifav}" is the available mass of gaseous fuel, "m_{O2av}" is the available mass of oxygen in the burning zone, "E_{gif}" is the activation energy of gaseous fuel and "A, z_1, z_2" are constants. The total rate of heat release is obviously the sum of the two rates, the liquid fuel heat release and the gaseous one.

CHEMICAL SPECIES FORMULATION – Combustion products are defined by dissociation considerations. For the C-H-O system the complete chemical equilibrium scheme proposed by Vickland et al. (1962) is used [15]. For the combustion zone, given its volume, temperature, mass of fuel burned and mass of air entrained, the concentration of each one of the eleven species can be calculated by solving a system of 11 equations containing: 4 atom balance equations (one for each element C,H,O,N) and 7 equilibrium equations. Apart from soot, at any instant of time, the following gas species are considered to be present inside each burned region, all in chemical equilibrium,

(1) H_2O	2) H_2	(3) OH	(4) H
(5) N_2	(6) NO	(7) N	(8) CO_2
(9) CO	(10) O_2	(11) O	

where the species are referred to by the number in parenthesis, in front of their name, and expressed by means of "kmole" fraction "X". The equilibrium concentrations of the eleven species can be described by the following seven equilibrium reactions ($B=X_1/X_2$):

$$
\begin{aligned}
&\frac{1}{2}H_2 \leftrightarrow H, K_{p1} = \sqrt{P}X_4 \big/ \sqrt{X_2} \\
&\frac{1}{2}O_2 \leftrightarrow O, K_{p2} = \sqrt{P}X_{11} \big/ \sqrt{X_{10}} \\
&\frac{1}{2}N_2 \leftrightarrow N, K_{p3} = \sqrt{P}X_7 \big/ \sqrt{X_5} \\
&2H_2O \leftrightarrow 2H_2 + O_2, K_{P4} = PX_{10} \big/ B^2 \\
&H_2O \leftrightarrow OH + \frac{1}{2}H_2, K_{P5} = \sqrt{P}X_3 \big/ \left(B\sqrt{X_2}\right) \\
&CO_2 + H_2 \leftrightarrow H_2O + CO, K_{P6} = BX_9 / X_8 \\
&H_2O + \frac{1}{2}N_2 \leftrightarrow H_2 + NO, K_{P7} = \sqrt{P}X_6 \big/ \left(B\sqrt{X_5}\right)
\end{aligned}
\tag{21}
$$

where "K_p" are the reactions equilibrium constants.

The solution of the above equations is obtained by incorporating the method proposed by Vickland et al (1962), with some slight modifications to ensure fast conversion.

NITRIC OXIDE CHEMICAL KINETICS – Since the formation of nitric oxides is a kinetically controlled mechanism, the extended Zeldovich chain reaction mechanism is used. The three following reactions are considered

$$
\begin{aligned}
&N + NO \leftrightarrow N_2 + O, k_{1f} = 1.6 \times 10^{10} \\
&N + O_2 \leftrightarrow NO + O, k_{2f} = 6.4 \times 10^{6} \times T \times e^{(-3125/T\ )} \\
&N + OH \leftrightarrow NO + H, k_{3f} = 4.2 \times 10^{10}
\end{aligned}
\tag{22}
$$

After some transformations the kmoles of NO are expressed as follows:

$$
\frac{1}{V} \times \frac{d\left[(NO)V\right]}{dt} = \frac{2 \times \left(1 - \beta^2\right) \times R_1}{\left(1 + \beta \times \dfrac{R_1}{R_2 + R_3}\right)}
\tag{23}
$$

where (NO) denotes concentration, β = $(NO)/(NO)_e$ and subscript "e" denotes equilibrium. "V" is the burned gas volume and R_I (i=1,2,3) is the one way equilibrium rate, for the "i" reaction, defined as follows:

$$
\begin{aligned}
&R_1 = k_{1f} \times (NO)_e \times (NO)_e \\
&R_2 = k_{2f} \times (N)_e \times (O_2)_e \\
&R_3 = k_{3f} \times (N)_e \times (OH)_e
\end{aligned}
\tag{24}
$$

and "k_{if}" is the forward reaction rate constant for the "i" reaction.

SOOT FORMATION MODEL – The modeling of soot formation is a very difficult task in common internal combustion engine simulation models. For this reason researchers usually use semi-empirical models which have been derived as correlation from the analysis of experimental data. In the present work a semi-empirical model, that has been widely tested, is used to predict the rate of soot formation. [6,8-10,13]

The soot formation and oxidation rates are given respectively by,

$$
dm_{sf} = A_f \times m_{fav} \times P^{0.5} \times \exp\left(\frac{-E_{sf}}{R_m T_b}\right)
\tag{25}
$$

$$
dm_{sb} = A_b \times m_s \times \frac{P_{O_2}}{P} \times P^{1.8} \times \exp\left(\frac{-E_{sb}}{R_m T_b}\right)
\tag{26}
$$

where "m_s" is the net soot formed, "P_{O2}" is the partial pressure of oxygen and "A_f" and "A_b" are constants.

The net soot formation rate is then obtained from the expression:

$$
dm_s = dm_{sf} - dm_{sb}
\tag{27}
$$

EXPERIMENTAL FACILITIES AND PROCEDURE

To calibrate the present model an experimental investigation has been conducted on a single cylinder, Lister LV1, direct injection diesel engine located at the authors' laboratory. The results of this investigation are used to calibrate and evaluate the model under normal diesel fuel operation. These results are also used as basis to evaluate the findings when using gaseous fuel as a supplement for liquid fuel. The technical data of the engine are given in Table 1. This is a naturally aspirated, air-cooled, four-stroke engine, with a bowl-in-piston combustion chamber. The normal speed range is 1000-3000 rpm. A three hole injector nozzle (each hole having a diameter of 0.24mm) is located in the middle of the combustion chamber head. The injector nozzle opening pressure is 190 bar. The main parts of the test installation used are:

- Lister LV1 diesel engine.
- Heenan & Froude hydraulic dynamometer.
- Tank and flow meter for diesel fuel.
- TDC marker (magnetic pick-up) and rpm indicator.
- K-type thermocouples for measuring the temperatures of the exhaust gas and engine oil.
- 'Kistler' 6001 miniature piezoelectric transducer for measuring the pressure in the cylinder, flush mounted to the cylinder head and carefully calibrated.
- 'Kistler' 7063 piezoelectric transducer for measuring fuel-line pressure before the injector.
- 'Kistler' 5007 charge amplifiers.

Table 1. Engine basic design data, Lister LV1-Diesel high speed engine

Bore	85.73mm
Stroke	82.55mm
Connecting Rod Length	148.59mm
Compression Ratio	18
Cylinder Dead Volume	28.03cm^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	34.5mm
Exhaust Valve Diameter	31.5mm
Static Injection Timing	28°CA before TDC

The output signals from the magnetic pick-up and piezoelectric transducers are connected to the input of a 'KEITHLEY' DAS-1801ST A/D board, installed on a IBM compatible Pentium II PC.

The exhaust gas analysis system consists of a group of analyzers for measuring all the main gaseous pollutants together with soot concentration. The nitric oxide concentration (NO) was measured by a standard "Signal" chemiluminiscent analyzer having a NO_2 to NO converter switch and fitted with a heated line, while the soot concentration was measured using a Bosch RTT100 analyzer.

The series of experiments conducted for this preliminary study, involved as already mentioned only the diesel fuel (no gas was added) in a variety of engine operating conditions. The following discussion refers to the results for an engine speed of 1500 rpm and four different loads, namely 20%, 40%, 60% and 80% of full engine load. Currently efforts are given to conduct experiments using gaseous fuel as a supplement for liquid fuel taking into account the observations obtained from the modeling. These results will be used to evaluate the findings of the newly developed dual-fuel model.

RESULTS AND DISCUSSION

Figures 1 to 4 present the comparison between theoretical and experimental pressure and heat release traces in the case of the 100% Diesel fuel, for 1500 rpm engine speed and four different loads namely 20, 40, 60 and 80% of full load respectively. It can be observed that the agreement in all cases is good, a fact that is promising for the utilization of the model to predict engine performance for dual fuel engines. It must be stated that the values of the two-zone model's constants are held the same for the entire range of operating parameters variation and for all fuel mixtures considered in the present study. Results for the experimental heat release rate curves in the above figures were obtained from the analysis of the corresponding experimental cylinder pressure traces using a diagnostic code [20].

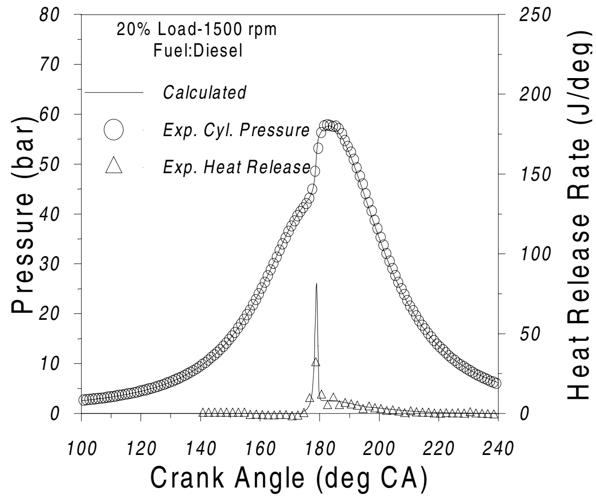

Figure 1. Comparison between experimental and calculated pressures and heat release traces at 1500 rpm engine speed and 20% load under 100% diesel fuel operation.

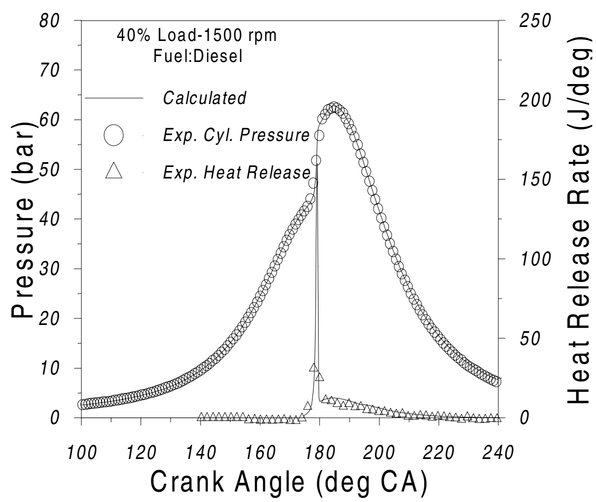

Figure 2. Comparison between experimental and calculated pressures and heat release traces at 1500 rpm engine speed and 40% load under 100% diesel fuel operation.

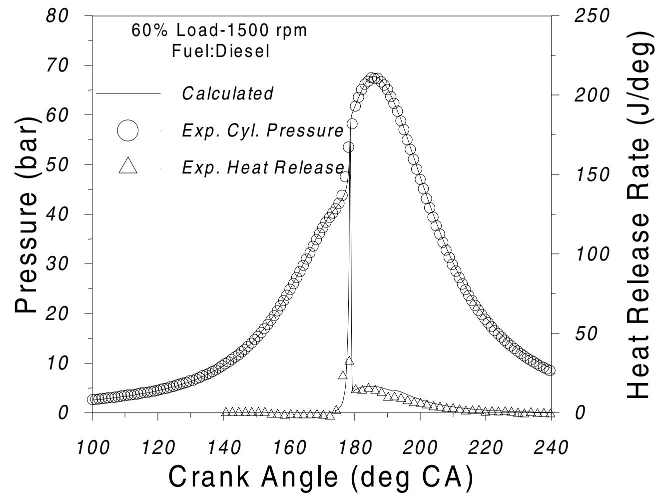

Figure 3. Comparison between experimental and calculated pressures and heat release traces at 1500 rpm engine speed and 60% load under 100% diesel fuel operation.

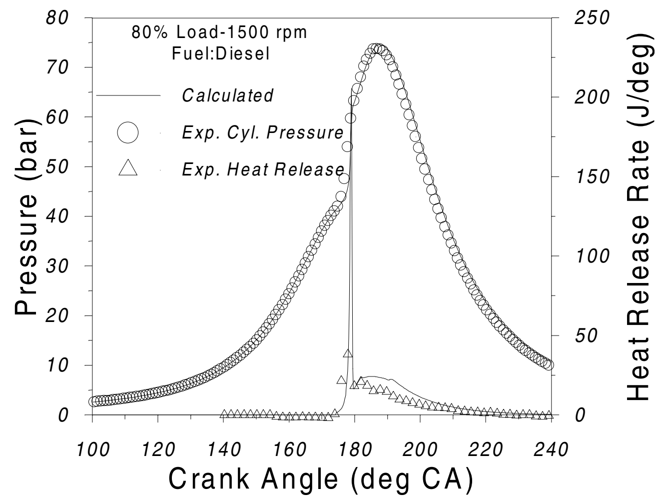

Figure 4. Comparison between experimental and calculated pressures and heat release traces at 1500 rpm engine speed and 80% load under 100% diesel fuel operation.

The variation of maximum cylinder pressure with load by various percentages of gaseous fuel is given in Figure 5. As shown the maximum combustion pressure is affected by the presence of the gaseous fuel in the corresponding mixtures. Combustion of the diesel-gas fuel mixture results to higher rates of heat release and this in turn results to higher combustion pressures compared to pure diesel fuel under the same conditions. Differences become higher as engine load and percentage of liquid fuel replacement by gaseous fuel increase.

An explanation for the previous, is given by Figure 6, which presents the corresponding heat release rate diagrams for the two extreme mixtures considered at 40 and 80% load.

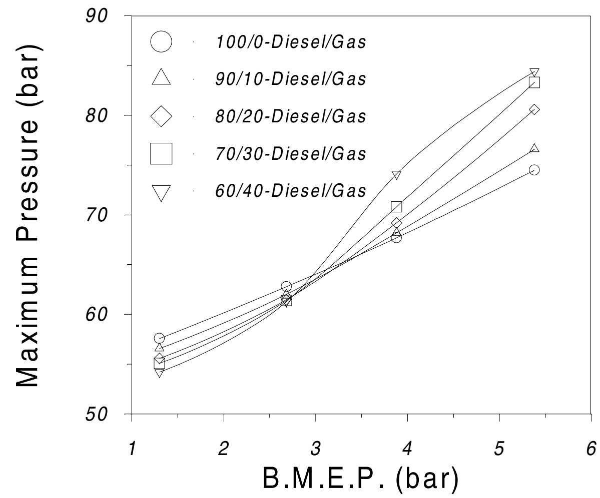

Figure 5. Maximum combustion pressure vs load for various percentages of Diesel/Gaseous fuel.

It is observed that during the premixed-controlled combustion phase of the liquid fuel, a sudden increase in heat release rate occurs, at higher load conditions. This is due to the sharp rate of combustion caused by presence of the gaseous fuel, which is already premixed with air and ready for combustion.

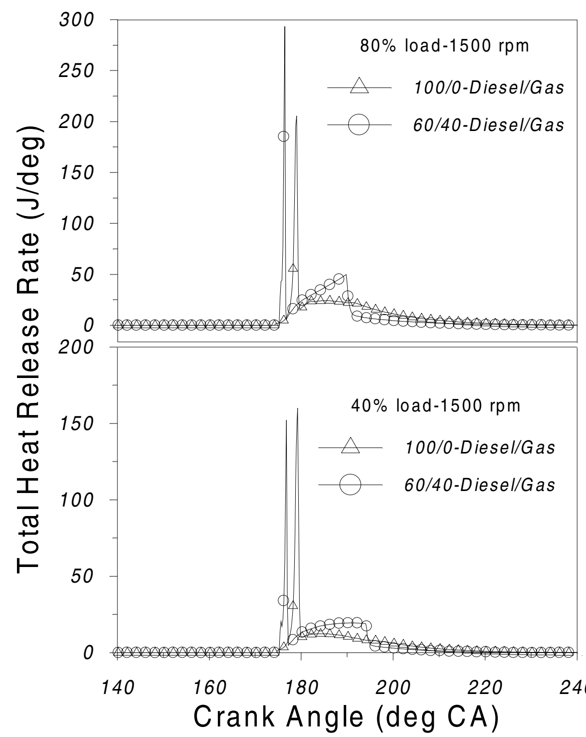

Figure 6. Heat release rate at 1500 rpm and 80%, 40% load for 100/0 and 60/40 Diesel/Gaseous Fuel operation

Figures 7 to 9 present the comparison between theoretical pressure traces in the case of 100%, 90%, 80%, 70%, and 60% Diesel fuel for 1500 rpm and for three different loads. It is observed that under part load conditions the presence of gaseous fuel affect slightly the values of the pressure. This is due to the fact that during the premixed-controlled combustion phase the heat release rate at part load when using a mixture of Diesel-Gas fuel is similar to the one when using 100% diesel fuel. Under high load conditions the increase of the percentage of gaseous fuel replacing the diesel fuel, leads to a sharper rate of heat release during the premixed-controlled phase, as already shown in Figure 6.

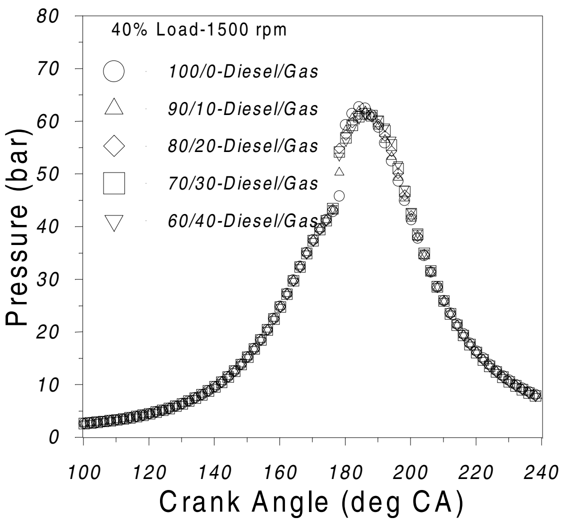

Figure 7. Comparison between calculated Pressure traces for various percentages of Diesel/ Gaseous fuel at 1500 rpm and 40%load.

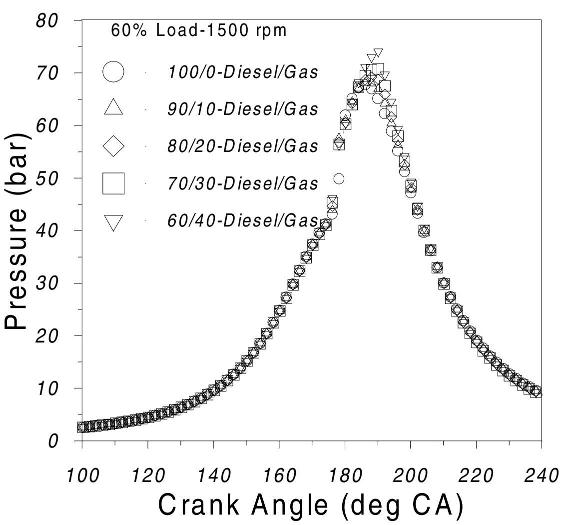

Figure 8. Comparison between calculated Pressure traces for various percentages of Diesel/ Gaseous fuel at 1500 rpm and 60%load.

Figure 10 presents the variation of brake specific fuel consumption with engine load. It is observed that in the case of 100% diesel fuel operation the model predicts accurately the measured "bsfc" values for all loads examined.

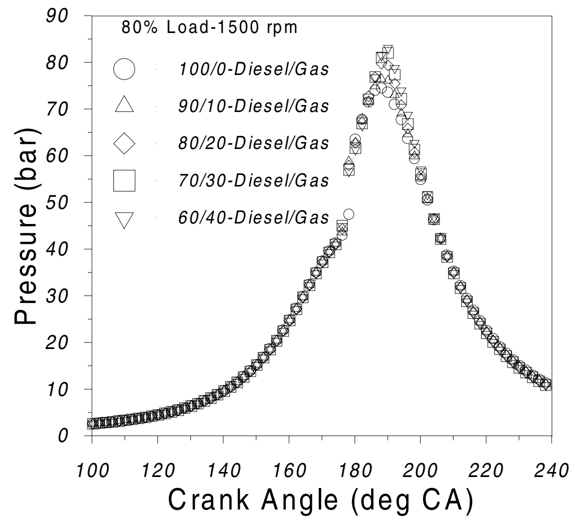

Figure 9. Comparison between calculated Pressure traces for various percentages of Diesel/ Gaseous fuel at 1500 rpm and 80%load.

It is observed that the increment of gas percentage in the fuel results in an analogue increment of engine efficiency, which becomes higher as load increases.

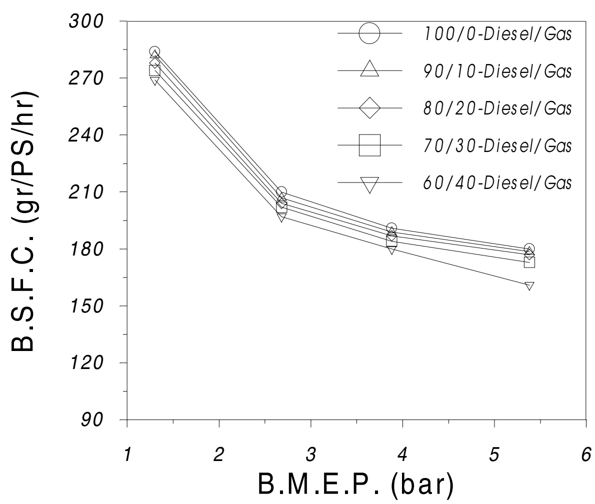

Figure 10. Brake specific fuel consumption at various engine loads for various percentages of Diesel/Gaseous fuel.

The variation of Nitric Oxide concentration with engine load is presented in Figure 11. Comparing the calculated values of NO with the measured ones at 100% diesel fuel operation, it is revealed a good coincidence. The model over-predicts slightly absolute values, which is normal for a two-zone model, but it manages to predict the trend with load. This enables us to use it for NO prediction under dual fuel operation. The formation of Nitric Oxide in the fuel jet is directly related to the local oxygen concentration and the gas temperature. Formation of Nitric Oxide takes place mainly during the premixed and initial part of mixing-controlled combustion, where high local temperatures exist. At part load conditions the "NO" concentrations differ slightly for various percentages of Diesel/Gas mixtures due to the previously mentioned effect on the heat release rate. As load increases nitric oxide concentration increase with the percentage of liquid fuel replaced by the gaseous one.

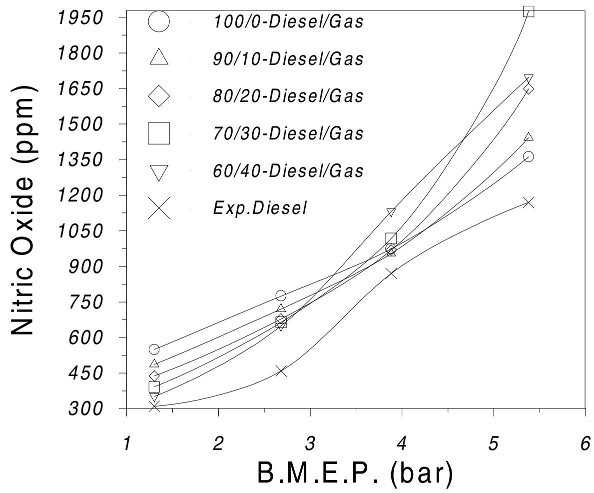

Figure 11. Nitric Oxide emissions at 1500 rpm and various loads for various percentages of Diesel/Gaseous fuel

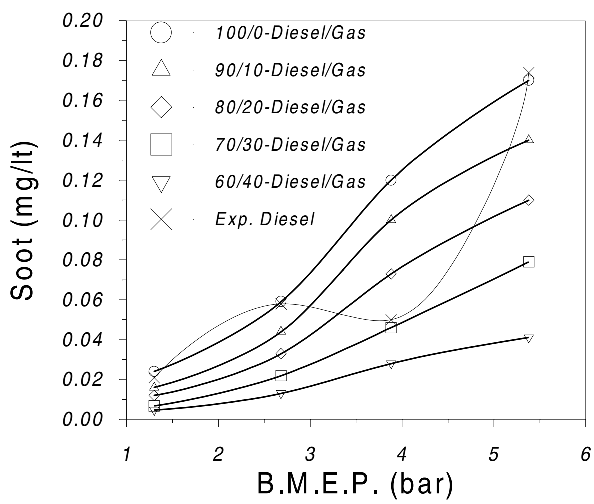

Figure 12. Soot emissions at 1500 rpm and various loads for various percentages of Diesel/ Gaseous fuel

Figure 12 presents the variation of the soot concentration in the engine exhaust with load for various proportions of Diesel/gas mixture. Observing Figure 12 we can see that the current model predicts adequately the experimentally measured soot emissions at 100% diesel fuel operation. This is encouraging and makes the results obtained under dual fuel operation more reliable. A difference is observed at 60% load, which is due to the different behaviour of the engine fuel injection system, as observed from the measured values of fuel injection pressure. As this percentage of gaseous/liquid fuel increase, the soot concentration is decreased since less liquid fuel is injected and thus less soot is formed. Also due to the higher temperatures existing at high percentages of liquid fuel replacement, the soot oxidation rate is higher contributing to a further decrease of emitted soot. The decrease of soot is more intense at high loads.

CONCLUSIONS

Under the present work a new model has been developed to simulate the operation of dual-fuel diesel engines for the prediction of performance and pollutant emissions. This preliminary investigation focuses on existing diesel engines where a part of the liquid fuel is replaced with a gaseous one equivalent to natural gas. The main purpose is to examine the effect of liquid fuel percentage replaced by gaseous fuel on performance and mainly soot and nitric oxide emissions.

For this reason a two-zone model has been developed capable of operating with various percentages of liquid and gaseous fuel. Combustion is initiated by the ignition of the liquid fuel, while the burning rate of gaseous fuel is controlled by its entrainment rate into the burning zone. Due to the lean nature of the surrounding unburned air-gaseous fuel mixture no flame front is considered inside the unburned zone. To validate the model before using it to predict engine performance and pollutants emissions under dual-fuel operation an extended experimental investigation has been conducted on a high speed DI simple cylinder test engine located at our laboratory. Measurements have been taken for both performance and emissions at various operating conditions using diesel fuel only. These are used for model validation and also serve as comparison between normal diesel fuel operation and operation when using various percentages of liquid and gaseous fuel.

Comparing calculated and measured values under normal diesel operation a good coincidence is observed for both performance and pollutant emissions. This enables us to use the model to predict engine behaviour under dual-fuel operation. From the analysis of computational data it is revealed that dual-fuel operation results to higher combustion pressures. These pressures increase with increasing percentages of gaseous/liquid fuel. The effect at high engine loads is more intense. Concerning engine efficiency it is revealed in general that the replacement of liquid fuel with gaseous results to an improvement of engine efficiency. This improvement is more intense at higher engine loads and higher values of the gaseous/liquid fuel ratio.

As far as pollutant emissions are concerned the use of gaseous fuel has a negative effect on NO emissions and a positive on soot. Specifically an increase of NO emissions is observed which is serious at high engine loads and gaseous/liquid fuel ratios. On the other hand soot is seriously decreased when using gaseous fuel as a supplement for liquid fuel. Again this effect is stronger at higher engine loads and gaseous/liquid fuel ratios. But the most important is that when using gaseous fuel we always have a reduction of soot emission regardless of the engine operating conditions.

The results of this preliminary investigation are encouraging and urge us to conduct an experimental investigation to verify the findings. This is currently under progress and results will be given in the near future. Even though it is difficult to generalize the findings of the current preliminary investigation, we believe that they are important since the reduction of soot emissions on existing DI diesel engines is extremely important.

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NOMENCLATURE

α, b, c	= constants
a_{del}	= constant
A	= area, m^2
A_f	= constant in soot formation mechanism
A_b	= constant in soot oxidation mechanism
C_v	= specific heat capacity under constant volume, J/kgK
C_p	= specific heat capacity under constant pressure, J/kgK
C_{Dinj}	= injector hole discharge coefficient
d_{noz}	= njector hole diameter, m
D	= cylinder bore, m
E	= activation energy, J/Kmole
k	= constant for liquid fuel preparation rate
k′	= constant for liquid fuel reaction rate
k″	= constant for gaseous fuel reaction rate
m	= mass, kg, or constant for the liquid fuel preparation rate
$\dot{m}$	= mass flow rate, kg/sec
P	= pressure, Pa
P_{O2}	= partial pressure of oxygen, bar
$\dot{q}$	= volumetric flow rate, m^3/sec

Q = heat transfer to walls, J
R = radius, m
R_m = universal gas constant, J/kmoleK
S = penetration length, m
t = time, sec
T = absolute temperature, K
U = jet velocity, m/sec
u = specific internal energy, J/kg
V = volume, m^3
X = Kmole fraction on species
y = mass fraction on species
x = constant
z_i = constants

Greek

θ = initial jet angle, rad
ΔP = pressure difference, Pa
λ = thermal conductivity, W/mK
μ = dynamic viscosity, kg/ms
ν = kinematic viscosity, m^2/sec
ρ = density, kg/m^3
φ = equivalence ratio

Subscripts

a = air
av = available quantity
b = burned zone
del = delay
ev = evaporated liquid fuel
e = equilibrium
f = fuel
fpr = liquid fuel prepared
fi = liquid fuel injected
fupr = liquid fuel unprepared
fb = fuel burned
g = gaseous fuel or gas mixture
i = index denoting control volume or the kind of gaseous fuel
κ = index denoting the species of the gaseous mixture
o = initial value
tot = total
s = soot
sf = soot formed
sb = soot oxidated
st = stoichiometric
u = unburned zone
w = wall

Dimensionless number

Re = Reynolds

Abbreviations

DI = direct injection
bmep = brake mean effective pressure
bsfc = brake specific fuel consumption
NO = nitric oxide
ppm = parts per million (volume)