ENHANCEMENT OF COMBUSTION PROCESS IN DUAL FUEL ENGINES AT PART LOADS BY USING SUITABLE TECHNIQUES

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Abstract It is a well known fact that, dual fuel engines at part loads, inevitably suffer lower thermal efficiency, higher carbon monoxide and unburned fuel emission. The work in this paper is to investigate combustion characteristics of a dual fuel (diesel-gas) engine at part loads, using a single zone combustion model with detailed chemical kinetics for combustion of natural gas fuel. The authors developed software in which the pilot fuel is considered as a subsidiary zone, and a heat source which is derived from two superposed Wiebe's combustion functions is to account for its contribution to ignite gaseous fuel and the remainder of total released energy. Chemical kinetics mechanism consists of 112 reactions with 34 species. This quasi-two zone combustion model is able to establish the development of the combustion process with time, and the associated important operating parameters such as; pressure, temperature, heat release rate (H.R.R) and species concentration. Therefore this work is an attempt to investigate the combustion phenomenon at part loads, using techniques such as increasing the quantity of pilot fuel and exhaust gas recirculation (EGR) to solve the above mentioned problems. By employing these techniques, it is found that, both these methods have positive effect on the performance and emission parameters, except for NO_x of dual fuel engines at part loads. Predicted values show good agreement with corresponding experimental values in a special engine with operating condition of (1/4 load, 1400 rpm). Implications will be discussed in details.

Keywords Quasi-Two Zone Combustion Model, Chemical Kinetics, Natural Gas, Dual Fuel Engines, Exhaust Gas Recirculation (EGR)

چکیده موتورهای دوگانه سوز در بارهای جزئی به ناچار متحمل بازده حرارتی پایین و مونوکسید کربن و سوخت نسوخته با مقدار زیاد می شوند. کار حاضر به منظور بررسی خواص احتراقی یک موتور دوگانه سوز در بار جزئی با استفاده از یک مدل احتراقی تک منطقه ای با مکانیزم سینتیک شیمیایی مفصل برای احتراق سوخت گاز طبیعی انجام شده است. در نرم افزار توسعه یافته، سوخت آتش زا به صورت یک منطق کمکی در نظر گرفته می‌شود و یک منبع گرمایی که از توابع احتراقی وایب مشتق می‌گردد به منظور در نظر گرفتن سهم آن در اشتعال سوخت گازی و بقیه انرژی آزاد شده کل می‌باشد. مکانیزم سینتیک شیمیایی شامل ۱۱۲ واکنش و ۳۴ گونه شیمیایی می‌باشد. این مدل احتراقی شبه دومنطقه ای قادر به پیشگوئی توسعه فرایند احتراق با زمان و پارامترهای عملکردی مهم مرتبط از جمله فشار، دما، آهنگ گرمای آزاد شده و غلظت گونه‌ها است. از این رو این مقاله به بررسی پدیده احتراق در بارهای جزئی می پردازد و در این ارتباط از روش هایی نظیر افزایش مقدار سوخت آتش زا و بازخورانی گازهای خروجی برای بهبود مشکلات ذکر شده در بارهای جزئی استفاده شده است. نتایج نشان می‌دهند که هردو روش مذکور، تاثیر مثبتی بر پارامترهای عملکردی و آلایندگی به غیر از آلاینده NO_x در شرایط بار جزئی موتورهای دوگانه سوز می‌گذارند. مقادیر پیش بینی شده توسط مدل در توافق خوبی با نتایج تجربی در شرایط عملکردی خاص موتور (۲۵ درصد بار کامل و ۱۴۰۰ rpm) می‌باشند. مفاهیم با جزئیات بیشتر در مقاله بحث خواهد شد.

1. INTRODUCTION

The compression ignition engine of the dual fuel type has been employed in a wide range of applications, to utilize various gaseous fuel resources and minimize exhaust gas emissions

without excessive increase in cost, from that of conventional diesel engines [1]. However, the combustion process in a dual fuel engine tends to display a complex combination of features, both compression and spark ignition engine operations. As the gaseous fuel mixes with the intake air in the manifold, the mixture formation is modified greatly, and then inside the cylinder, the mixture undergoes a multi-point ignition due to the combustion of a pilot diesel fuel spray. Then flame propagation occurs through the premixed natural gas mixture. Thus, dual fuel operation with natural gas fuel can yield a high thermal efficiency almost comparable to an engine operating on diesel fuel at higher loads. However, engine performance and emissions suffer at low loads when operating in dual fuel mode [1-4]. The main reason for this poor light load performance is due to very lean mixtures [2-4]. The lean mixtures are hard to ignite and slow to burn.

Pirouzpanah, et al [3] conducted an experimental study to determine performance and emission characteristics of an automotive direct injection dual-fuelled diesel engine. Cooled EGR was used to resolve the poor light load performance of the engine. Results show that the application of EGR can considerably reduce CO and UHC emissions.

A suitable computer based analytical modeling can provide an adequate means for describing details of a complex combustion process in dual fuel engines, also help to reduce prohibitive development time and cost usually involved in the conversion of diesel engines to dual fuel operation. However, the development of comprehensive simulation models for dual fuel engine operation has been so far very limited due to the complex combustion processes involved. In the dual fuel engines, zero dimensional models were accepted approach for modeling and studying of combustion phenomena and related performance parameters. For example, Karim, et al [2,5] developed a quasi-two zone model, which was used to predict the auto ignition and knock characteristics and overall engine performance of dual fuel engines near full load. The model could not be applied to predict neither exhaust emissions nor the operation at light load when lean mixtures are employed. This is mainly due to the absence of measurement for predicting spatial variations of in-cylinder temperature and composition.

Pirouzpanah, et al [6] developed a model for dual fuel engines at full load conditions, which simulates combustion process by using a multi-zone combustion model for diesel pilot fuel combustion and a conventional S.I. combustion model for modeling combustion of premixed gas/air charge. Also, in this model, for prediction of formation and oxidation rates of pollutants, relevant conventional kinetically controlled mechanisms and mass balances were used. In this model, predicted performance parameters had good agreement with experimental data but there were some discrepancies between the predicted and experimental data for pollutants.

Abd Alla, et al [7,8] developed a quasi-two zone model, which was used for prediction of the combustion processes in an I.D.I dual fuel engines and some of their performance features. Their model emphasizes the effects of chemical kinetics activity of the premixed gaseous fuel on the combustion process, while the role of pilot fuel in the ignition and heat release processes is considered using two superposed Wiebe's combustion functions. This combustion model is able to establish the development of the combustion process with time and the associated important operating parameters, such as pressure, temperature, rates of energy release and composition. Also they investigated the effects of some methods such as injection timing advance, increasing pilot fuel quantity, exhaust gas recirculation (EGR) to improve poor light load performance.

Karim, et al [9-11] examined the contributions of residual gases to the combustion process analytically, through modeling the autoignition of a homogeneous fuel air mixture, such as methane or propane, in a motored engine. They classified different cases of EGR namely into thermal, kinetic and diluting cases. Also they showed that positive effects of EGR can be moderated by the diluting effects of some inert products in EGR gases.

Karim, et al [12] developed a multi-zone thermodynamic model which was able to describe the combustion processes of dual fuel engines and predict different aspects of their performance. The consequences of the interaction between the gaseous and diesel fuels and the

resulting modification to the combustion processes were considered. A detailed chemical kinetics scheme is employed to describe oxidation of the gaseous fuel starting at compression and ending at the expansion process. The associated formation and concentrations of exhaust gas pollutants are also established. The model not only predicts knock onset but also predicts the more demanding case of low load engine performance with the associated partial oxidation reactions, and the production of exhaust gas emissions. The results demonstrated any measures which increase the size of the combustion regions of the cylinder charge, such as increasing the concentration of the gaseous fuel, or employing a large pilot fuel quantity, can significantly reduce the concentration of the unconverted gaseous fuel and carbon monoxide in the exhaust gases.

Mansour, et al [13] developed a computer program to model the combustion processes in dual fuel engines. A detailed chemical reaction mechanism of natural gas and NO_x were used to predict the main combustion characteristics. This semi-empirical dual fuel engine simulation work has been carried out using the two Wiebe's function model for different speeds and air fuel ratios.

Pirouzpanah, et al [14] developed a quasi-dimensional combustion model to predict the combustion of direct injection dual fuel diesel engines by a detailed chemical kinetic model for gaseous fuel combustion. Chemical kinetic mechanism consisted of 325 reactions with 53 species (GRI3). Heat release rate of pilot fuel with this model was considered by two Wiebe's function. Predicted values of performance parameters for dual fuel operation show good agreement with corresponding experimental data.

Pirouzpanah, et al [15] tried to enhance combustion process in dual fuel engines at part load using EGR. The results show that, with employing hot EGR, taken from very close to the exhaust valve, the amount of EGR which is required to overcome the problems of dual fuel engines at part load conditions is very small. They have shown that, by using this technique, installation of heat exchanger for heating intake charge is no longer required.

Daisho et al [16] conducted an experimental work to determine combustion and exhaust emission characteristics with the objective of improving thermal efficiency and reducing exhaust emissions. The methods tested to achieve these objectives are diesel fuel injection timing advance, intake throttling and EGR. Results of this work show that at part load, diesel injection timing gave little improvement in thermal efficiency and increased NO_x. Intake throttling promoted better combustion and shortened its duration with a consequent improvement in efficiency at higher natural gas fractions. Hot EGR raised thermal efficiency, reduced smoke levels and maintained low NO_x levels.

Abd Alla, et al [17] organized an experimental work to rectify the above mentioned problems of dual fuel engines at part loads conditions by increasing pilot fuel quantity. Through experimental investigations, it can be seen that, the low thermal efficiency and excess emissions at light loads can be improved significantly by increasing pilot fuel quantity.

On the other hand, multi-dimensional models also applied in simulation of combustion process in dual fuel engines [18-22]. These models require a significant amount of computer power even for prediction of turbulent flow with a simplistic description of the complex chemical reactions which are taking place during combustion. Accordingly, in general, most combustion models being used for this purpose are either the zero dimensional or quasi-dimensional types.

In recent years, chemical kinetic modeling has become an important and powerful tool for analysis of complex combustion process in dual fuel engines [1,23]. Such computer models have contributed better understanding and solution of longstanding practical combustion problems in dual fuel engines, including performance, pre-ignition reactions, pollutants emission, part load performance and knocking [21].

The present contribution describes a quasi-two zone thermodynamic model that was developed to describe the combustion process of dual fuel engines. This model consisted of a single zone with detailed chemical kinetics for combustion of natural gas fuel and a subsidiary zone for combustion of pilot fuel to account for its contribution to the ignition of gaseous fuel air mixture. Chemical kinetics mechanism consisted of 112 reactions with 34 species. This so called

"combined" combustion model is able to establish the development of a combustion process with time and the associated important operating parameters, such as pressure, temperature, and heat release rate and species concentration. As mentioned before, the dual fuel engines at part load inevitably suffer from lower thermal efficiency and higher emission of carbon monoxide and unburned fuel. Therefore, this work is an attempt to investigate the nature of poor and complex combustion phenomenon at part loads and some techniques such as increasing pilot fuel quantity and hot EGR are used to solve the above mentioned problems which will be discussed in details in the following sections.

2. DESCRIPTION OF THE PRESENT MODEL

In the present model, the gaseous fuel air mixture in the cylinder is treated as the main zone and subjected to changes in pressure and temperature with time due to piston motion, preignition combustion reactions which may produce some intermediate species such as radicals, carbon monoxide and formaldehyde. These can have profound effects on the subsequent combustion processes of the dual fuel engine. In addition, the presence of the pilot fuel considered as a heat source which is deriving from two superposed Wiebe's combustion functions to account for its contribution to ignition of gaseous fuel, remaining fuel energy and relevant heat release rate.

During the stages of compression, combustion and expansion, full chemical reaction kinetics of gaseous fuel air charge were employed to predict in detail the changes in composition and associated properties. The 112 elementary reactions mechanism has been used and the relevant equations are solved numerically in the present study.

The performance of a typical dual fuel engine with known dimensions, compression ratio and valve timing is simulated by the described model. The fresh charge of the intake mixture (NG+air) is assumed to be initially homogenous and fully mixed during the suction stroke. The mixture, following intake valve closure, is then compressed, burned and subsequently expands towards the exhaust stroke. During these processes, chemical reaction kinetics were used to predict in detail the changes in the concentration and properties of the mixture (NG+air) in the cylinder and subsequent changes of the energy release rates with time. When the pilot diesel fuel is injected into the combustion chamber, its contribution to energy release, the composition of the cylinder charge, temperature and pressure are taken into account. Thus, a quasi-two zone combustion model is employed where the main zone is the gaseous fuel air mixture with its detailed chemical kinetic reaction activity. At a later stage, following pilot fuel ignition, an overlapping secondary and smaller subsidiary zone due to the combustion of the pilot fuel is also considered. The two zones are assumed, for simplicity, to be interactive primarily thermally with no direct chemical interaction between the two types of fuels.

3. ASSUMPTIONS

1. This model assumes that the pilot fuel acts as a heat source and provides thermal energy for ignition of the gaseous fuel air charge. Once ignition begins, the energy released by the pilot fuel is incorporated in the model, raising the pressure and mean temperature of the whole cylinder charge beyond those levels due to piston motion and the chemical reactions of the gaseous fuel air charge.
2. It is assumed throughout that, there is no temperature, pressure and concentration gradients within the cylinder charge.
3. Gas leakage from the cylinder, once the valves are closed, was assumed to be negligible.
4. Active radicals which are introduced to the engine cylinder by EGR, have no effect on the chemical kinetics mechanism but they combine with existing free radicals which were originated from gaseous fuel ignition and therefore, they can have profound effect on the chain branching.
5. All components of the mixtures were considered to behave as ideal gas.

4. MATHEMATICAL TREATMENT

4.1. Pilot Fuel

4.1.1. Ignition delay of the pilot fuel The pilot fuel undergoes a definite ignition delay period based on engine operating conditions. A large number of correlations based on experimental and/or theoretical investigations are available to calculate the ignition delay period. In this work following formulation [7] is used:

$$\tau = 4.3\times10^{-3}P^{-2.5}\phi^{-1.04}\exp(5000/T) \quad (1)$$

Where the ϕ is defined as the ratio of the mass of the stoichiometric amount of air required for the combustion of both of the gaseous and the pilot diesel fuels to the mass of the actual amount of air drawn in:

$$\phi = (14.75\dot{m}_p + 15.22\dot{m}_{CNG})/\dot{m}_a \quad (2)$$

Ignition starts as soon as the following condition is satisfied:

$$\int_0^t dt/\tau \geq 1.0 \quad (3)$$

Also, for calculation of this period, a subroutine has been developed in which chemical kinetic and energy equations (Equations 5-12 which will be discussed in the next sections) are introduced for considering different effects of parameters such as thermodynamics and physical properties, contribution of preignition energy release, heat transfer and EGR on temperature and pressure of ignition delay definition.

4.1.2. Pilot fuel heat release model Diesel engines, generally exhibit two stages of combustion behavior process, commonly identified as premixed combustion and diffusive combustion, regardless of engine operating conditions. To fit the rate of combustion curve adequately for these two stages, two superposed Wiebe's combustion functions were combined [24]. The employed two Wiebe's functions have the following form:

$$dQ/d\theta = 6.9\frac{Q_p}{\theta_p}(M_p+1)\left(\frac{\theta-\theta_0}{\theta_p}\right)^{M_p}\exp\left(-6.9\left(\frac{\theta-\theta_0}{\theta_p}\right)^{M_p+1}\right)$$
$$+6.9\frac{Q_d}{\theta_d}(M_d+1)\left(\frac{\theta-\theta_0}{\theta_d}\right)^{M_d}\exp\left(-6.9\left(\frac{\theta-\theta_0}{\theta_d}\right)^{M_d+1}\right) \quad (4)$$

4.2. Gaseous Fuel A single zone analytical model that incorporates detailed chemical kinetic mechanism was employed to investigate the gaseous fuel combustion characteristics. The scheme consisted of 112 chemical reaction steps and the following 34 chemical species:

C_2H, C_2H_2, C_2H_3, C_2H_4, C_2H_5, C_2H_6, C_3H_8, nC_3H_7, iC_3H_7, C_3H_6, CH_4, CH, CH_2, CH_3, CH_2CO, CH_2O, CH_3O, CHO, CH_3CHO, CH_3CO, CO, CO_2, H_2, H, HO_2, H_2O_2, H_2O, O, O_2, OH, N_2, N_2O, N, NO.

With employment of the detailed chemical reaction kinetics in the model, all changes in the concentration and properties of the mixture and the consequent changes of energy release rate could be established with the time during the compression, combustion and expansion stages. The energy and species equations for the homogenous reacting system become:

For this reaction scheme, the intermediate reaction steps may be represented as following:

$$\sum_{i=1}^{34}\alpha_{ijf}A_i \underset{k_{jb}}{\overset{k_{jf}}{\Leftrightarrow}} \sum_{i=1}^{34}\alpha_{ijb}A_i \quad j=1..112 \quad (5)$$

For the jth reaction, the forward rate constant k_{jf} can be expressed by the following Arrhenius expression:

$$k_{jf} = A_{jf}T^B\exp\left(\frac{-E_{jf}}{TR}\right) \quad j=1..112 \quad (6)$$

While backward rate constant k_{jb}, which is of similar form to that of k_{jf}.

The reaction rates for the jth reaction in the forward and backward directions can be expressed respectively in terms of the concentration as follows:

$$R_{jf} = k_{jf}\prod_{i=1}^{34}(\rho x_i)^{\alpha_{ijf}} \quad j=1..112 \quad (7)$$

$$R_{jb} = k_{jb} \prod_{i=1}^{34} (\rho x_i)^{\alpha_{ijb}} \quad j = 1..112 \tag{8}$$

The piston displacement is a known function of time, hence the density-time is also known in terms of the mass content of the cylinder:

$$\rho(\theta) = \frac{M}{V(\theta)} \tag{9}$$

The net rate of production of each species will be a function of the rate of all the reaction steps involved that are proceeding simultaneously and are given by a relation of the form:

$$-\rho \frac{dx_i}{dt} = \sum_{j=1}^{112} (\alpha_{ijf} - \alpha_{ijb})(R_{jf} - R_{jb}) \quad i = 1..34 \tag{10}$$

The energy equation for the homogeneous reaction system assumed to be an ideal gas and becomes:

$$\sum_{i=1}^{34} x_i \overline{R}T \times \frac{V'(t)}{V(t)} + \sum_{i=1}^{34} \left\{ \left(\int_{T_0}^{T} C_{vi}\, dT + \Delta U_{fi} \right) \frac{dx_i}{dt} + x_i C_{vi} \frac{dT}{dt} \right\} = \dot{Q}_p + \dot{Q}_{Loss} \tag{11}$$

$\dot{Q}_p$, is the heat release rate of pilot fuel which is derived from two superposed Wiebe's combustion functions. $\dot{Q}_{Loss}$, is heat loss rates of the natural gas air mixture which is considered directly and it is included by the Annand heat transfer correlation with the following form [25]:

$$d\dot{Q}_{Loss} = A_s \left\{ \frac{aK}{D} Re^b (T - T_w) + c(T^4 - T_w{}^4) \right\} \tag{12}$$

Thus, the energy Equation 11 and the simultaneous set of equations of (10), one for each species, and result in a simultaneous set of non-linear first order differential equations which were solved by the authors developed software. Also it is necessary to mention that the CPU time for solving these simultaneous set of differential equations is less than twenty minutes.

5. RESULTS AND DISCUSSION

5.1. Baseline Dual Fuel Engine Figure 1 shows typical variations of the calculated cylinder pressure with crank position when a small quantity of liquid diesel fuel was maintained to provide pilot ignition in a DI six cylinder engine which its specifications are shown in Table 1. The corresponding experimental values [26] are also shown. The predicted values show good agreement with the corresponding experimental results.

Figure 2 represents the corresponding variations of the heat release rates with crank position at part load calculated by the described model for the considered dual fuel diesel engine. It can be seen that the total heat release rate of the cylinder charge occurred mainly in two stages. In the first stage, the energy is released by combustion of small quantity of pilot fuel which is associated with high rates. The second stage of the combustion occurs due to combustion of gaseous fuel air charge, which is associated with lower rates of the exothermic combustion reactions result in lower energy release rates immediately following pilot ignition. Also it can be seen that at part load, the total energy release diagram of a dual fuel engine is very similar to that observed in pure diesel engine operation.

Table 2 shows a comparison between experimental and theoretical values of major species of OM-355 dual fuel engine at part load condition (1/4 full load, energy ratio: 20 % pilot diesel, 80 % natural gas). It can be seen that all of the theoretical values are in good agreement with the experimental amounts except for NO_x specie. This discrepancy may be related to the combustion of the pilot fuel which has not been considered in this model. Furthermore, the weakness of the chemical kinetics mechanism for prediction of NO_x concentration in the very lean mixtures of natural gas and air at part load conditions is evident. Also it can be seen that the concentrations of CO and UHC are higher at part load in comparison with the full load experimental values. Therefore the main purpose of this work is an attempt to investigate the combustion phenomena at part loads and some methods such as increasing pilot fuel quantity and EGR are used to solve the mentioned problems. For implementing the above mentioned techniques

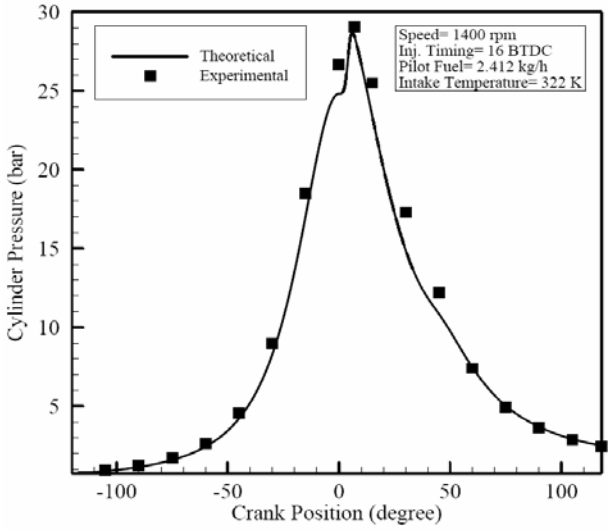

Figure 1. Comparisons of calculated and experimental [25] cylinder pressure of dual fuel engine. (1/4 full load, energy ratio: 20 % pilot diesel, 80 % natural gas).

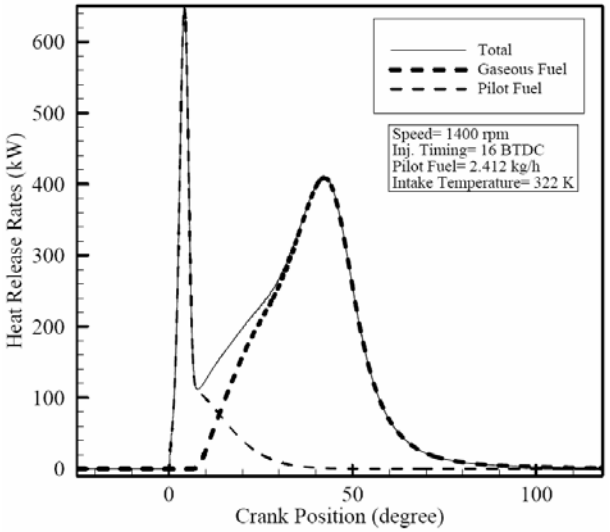

Figure 2. Variations of the calculated heat release rates of both pilot and gaseous fuels with crank position (1/4 full load, energy ratio: 20 % pilot diesel, 80 % natural gas).

TABLE 1. General Specifications of OM-355 Dual Fuel Engine.

Make and Model:Mercedes-Benz OM-355	Fuels: Diesel and Natural Gas
Type: Direct Injection, Naturally Aspirated, Heavy Duty Vehicle Diesel, Four Stroke	No of Nozzles/Injector: 4
Cylinders: 6, In-line-Vertical	Nozzle Opening Pressure: 195 (bar)
Bore * Stroke: 128(mm) * 150(mm)	Max. Power: 240 (HP) at 2200 (RPM)
Capacity: 11.58 (liter)	Max.Torque: 824 N.m at 1400 (RPM)
Compression Ratio: 16.1:1	Pilot Fuel Injection Timing: 16˚CA BTDC

TABLE 2. Comparison of Experimental and Theoretical Values of Major Exhaust Species of OM-355 Dual Fuel Engine at 1400 rpm.

Species (Values)	UHC (ppm)	CO (%)	NO_x (ppm)	O_2 (%)
Experimental	626	0.311	157	13
Theoretical	562	0.387	0	13.04

in the model, following considerations are taken into account:

1. The amount of total fuels energy which is introduced to the engine cylinders is not changed by using the above mentioned methods.

2. To quantify the amount of EGR, the EGR percentage is defined by:

$$EGR(\%) = \left(\frac{\dot{m}_{EGR}}{\dot{m}_a + \dot{m}_{CNG} + \dot{m}_{EGR}} \right) \cdot 100 \qquad (13)$$

Also, for applying EGR to the model, the thermal effect of EGR which can increase the temperature of the charge was considered in addition to species concentrations which were obtained from composition of cylinder charge at the exhaust valve opening condition by running the authors developed software for baseline dual fuel engine.

3. To compare the effects of these techniques on emission parameters, the amount of variables for these methods are defined in which indicated thermal efficiency is considered constant as indicated in Table 3.

The low load which is selected for this study is a ¼ full load with equivalence ratio equal to 0.52 and engine speed is the maximum torque speed equal to 1400 rpm.

5.2. Effects of the Mentioned Methods

Figure 3 shows logarithmically variations of the calculated heat release rate with crank position for different methods. It can be seen that, by using the above mentioned methods, heat release rate is increased and the combustion process is shifted towards TDC meanwhile contribution of EGR is higher than other methods. EGR can promote the combustion process due to increasing total equivalence ratio, intake temperature of the charge and preparing better fuel air mixing ready for combustion. Apart from its thermal effect, EGR tends to improve the preignition reaction rates of the cylinder charge by suitably seeding the intake charge with partial oxidation products that are sources of fruitful active radicals. These positive effects of EGR are moderated by the diluting effects of some products in EGR. Also, with increasing pilot fuel quantity, the volume of the charge that is affected by the combustion of the pilot fuel envelope will increase thus the burned fraction of the gaseous fuel air mixture will be raised.

Figure 4 describes variations of the calculated cylinder pressure with crank position for different methods. As indicated in this figure, the cylinder peak pressure and consequently cylinder pressure during combustion and expansion processes is increased by using each of the above mentioned methods. These trends are consistent with those observed in the heat release rate curves.

Figure 5 indicates variations of the calculated cylinder temperature with crank position for different methods. It can be seen that, cylinder temperature is increased but contribution of EGR is higher than other method. The main reason for this trend is perhaps due to increasing intake charge temperature and using active radicals which exist in EGR gases.

Figure 6 indicates logarithmically variations of the calculated mole fraction of OH radical with

TABLE 3. The amount of Variables for Different Methods.

Methods	Variations	Baseline	Modified
EGR	EGR (%)	0	2
Increasing pilot fuel quantity	Mass flow rate(kg/hr)	2.412	2.6

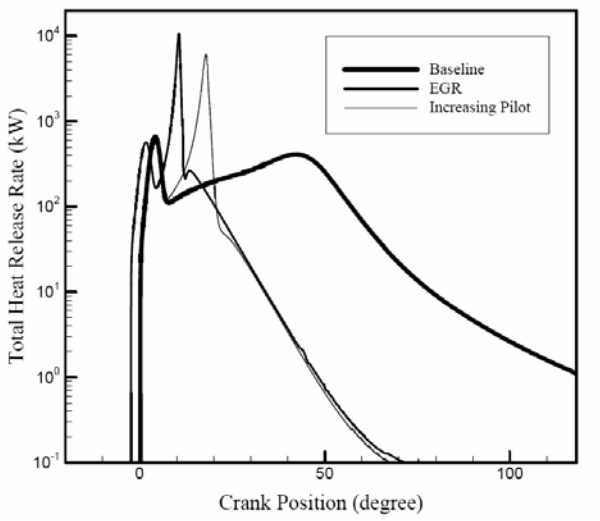

Figure 3. Variations of the calculated total heat release rate with crank position for different methods.

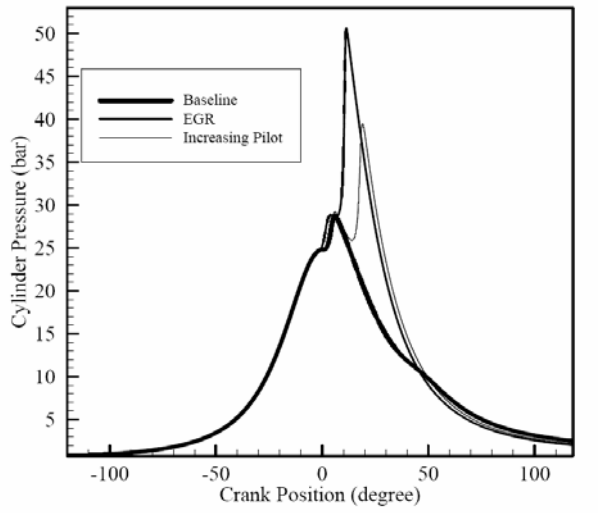

Figure 4. Variations of the calculated cylinder pressure with crank position for different methods.

crank position for different methods. It can be seen that the OH radical produced from the combustion of fuels increases by using each of the above mentioned methods. Sensitivity analysis shows that the OH radical has the greatest effect on ignition delay period through the reaction:

$$CH4 + OH = CH3 + H2O \quad (14)$$

Thus, as indicated in Table 4, increasing the OH radical by using each of the above methods decreases the ignition delay period of the pilot fuel.

Figures 7 to 9 show typically the mole fractions of the active radicals with crank position for different methods. It can be seen that they reach very high values in the early stages of the reaction and therefore it can be concluded that combustion is started with their sufficient concentrations. Meanwhile, in EGR method, these species reach their maximum concentrations earlier than the other method.

Figure 10 describes variations of the calculated mole fraction of H_2O_2 with crank position for different methods. As indicated in this figure, at different methods, the mole fraction of this specie has its maximum value where OH radical is low and vice versa. This is perhaps due to an important role of the following reaction in the chemical kinetics mechanism:

$$OH + OH + M = H_2O_2 + M \quad (15)$$

Figure 11 indicates variations of the calculated mole fraction of CH_2O with crank position for different methods. It can be seen that the amount of this specie at the end of expansion stroke for baseline dual fuel engine is considerable but by using each of the methods, its concentration is zero at earlier part of expansion stroke.

Figure 12 indicates variations of the calculated mole fraction of O_2 with crank position for different methods. As indicated in this figure, by using each of the methods, consumption of this specie is increased.

Figure 13 shows variations of the calculated mole fraction of CH_4 with crank position for different methods. It can be seen that, consumption of CH_4 by EGR method is started earlier than the other method. This trend is consistent with O_2 consumption.

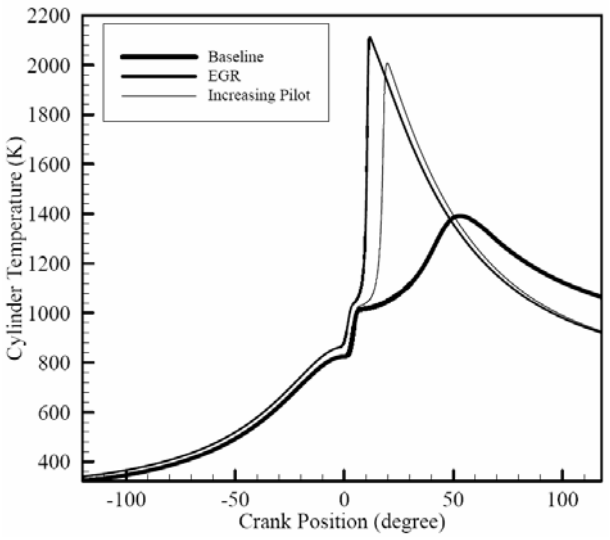

Figure 5. Variations of the calculated cylinder temperature with crank position for different methods.

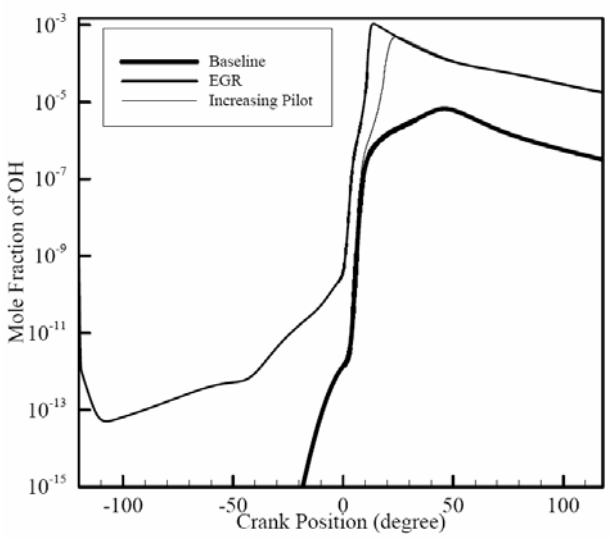

Figure 6. Variations of the calculated mole fraction of OH radical with crank position for different methods.

TABLE 4. Variations of the Ignition Delay Period for Different Methods.

Method	Baseline	EGR	Increasing Pilot
Ignition Delay Period (Degree)	16.2	13.7	16.1

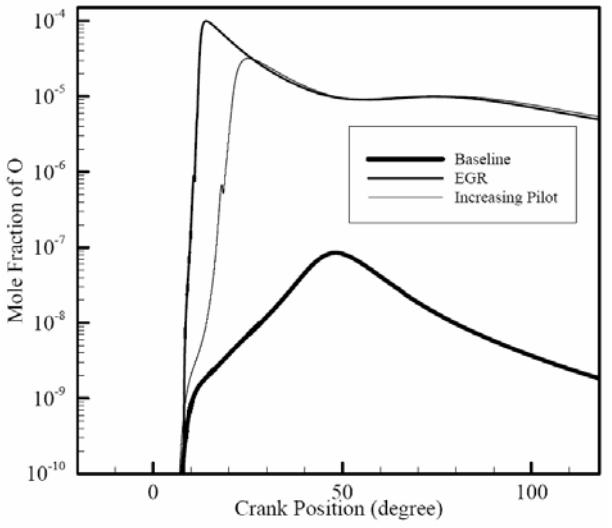

Figure 7. The logarithmical variations of the calculated mole fraction of O radical with crank position for different methods.

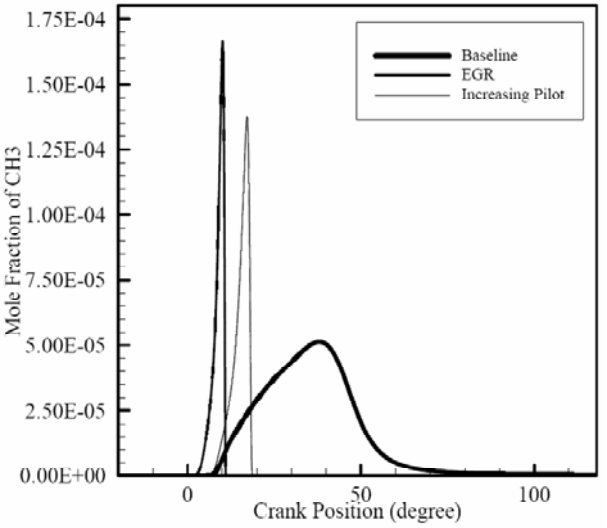

Figure 9. Variations of the calculated mole fraction of CH_3 with crank position for different methods.

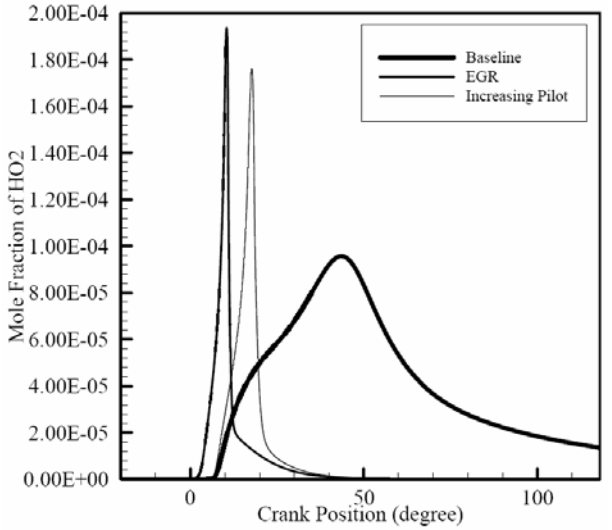

Figure 8. Variations of the calculated mole fraction of HO_2 with crank position for different methods.

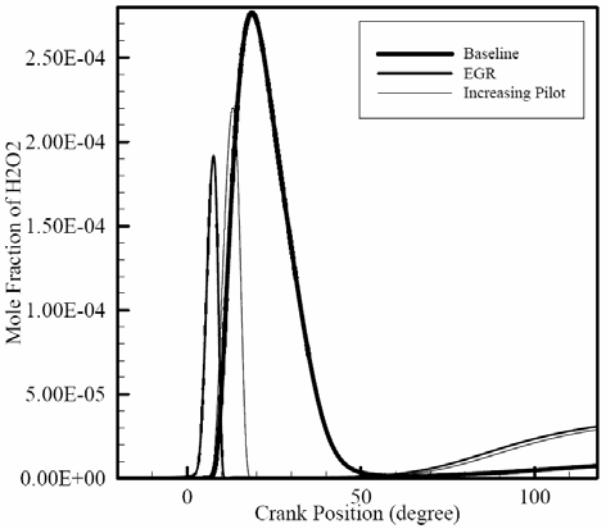

Figure 10. Variations of the calculated mole fraction of H_2O_2 with crank position for different methods.

Figure 14 describes variations of the calculated mole fraction of CO with crank position for different methods. As indicated in this figure, by employing the above improvement methods, CO emission is reduced which is due to better mixture preparation for combustion in the EGR methods and preparing required ignition sources for ignition of gaseous fuel air mixture in the EGR and increasing pilot fuel quantity methods.

Figure 15 indicates variations of the calculated concentration of NO_x with crank position for different methods. It can be seen that, all improvement methods raise NO_x emission relative to baseline dual fuel engine. Apart from the effect of the inert gas brought by EGR, hot recirculated exhaust gas and very active radicals would raise

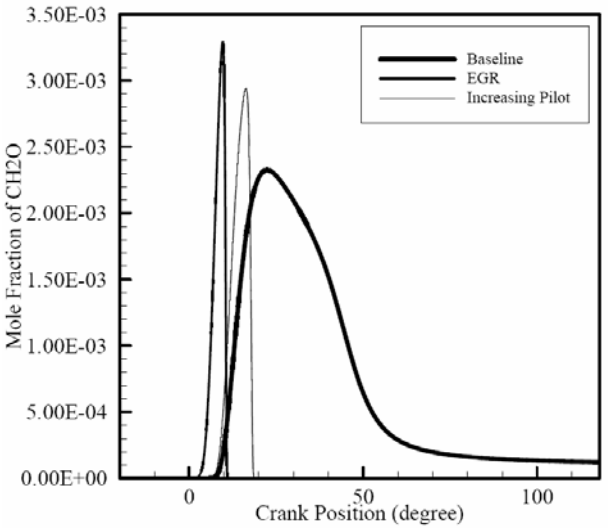

Figure 11. Variations of the calculated mole fraction of CH_2O with crank position for different methods.

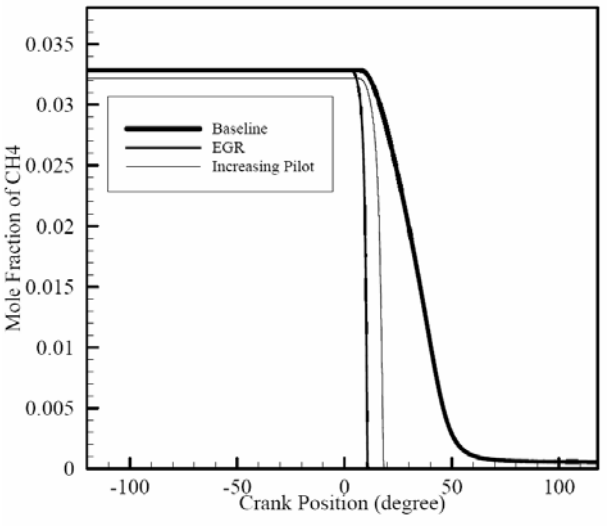

Figure 13. Variations of the calculated mole fraction of CH_4 with crank position for different methods.

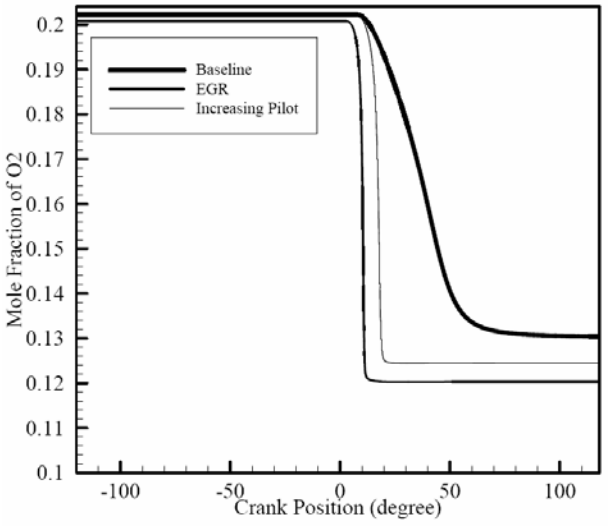

Figure 12. Variations of the calculated mole fraction of O_2 with crank position for different methods.

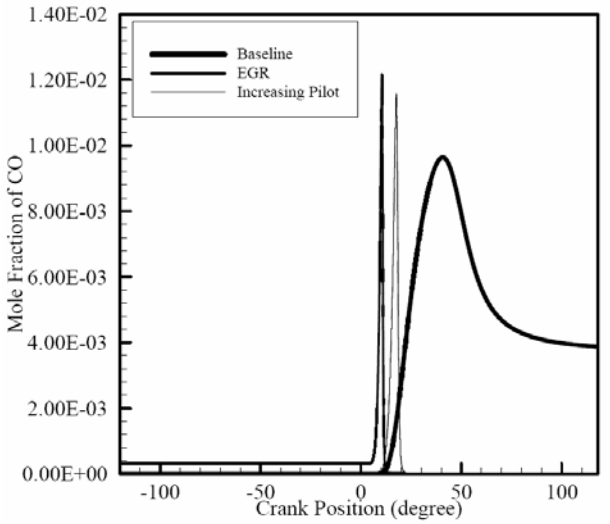

Figure 14. Variations of the calculated mole fraction of CO with crank position for different methods.

the charge temperature; thereby they can enhance the combustion process and raise the NO_x emission. Here, the thermal and chemical effects of EGR may be acting together. Also, preparing required ignition sources for combustion of gaseous fuel air mixture by pilot fuel increasing, would raise the charge temperature, thereby it can raise NO_x emission.

6. CONCLUSIONS

The present quasi-two zone combustion model, combined with the detailed chemical kinetics scheme, can provide a description of the main features of a combustion process in dual fuel engines. The corresponding performance of dual fuel engines can be analyzed by this model for the

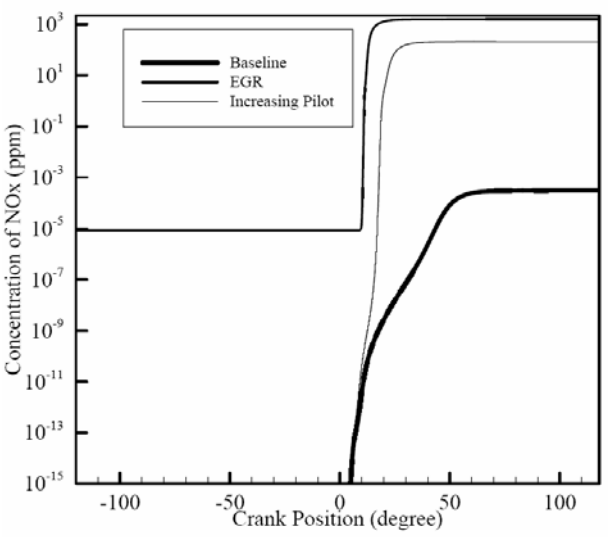

Figure 15. Variations of the calculated concentration of NO_x with crank position for different values of EGR.

whole engine operating conditions. As already known, the dual fuel engines at part loads inevitably suffer lower thermal efficiency and higher emission of carbon monoxide and unburned fuel. Therefore, this work is an attempt to investigate the combustion process of a dual fuel engine at part loads using methods such as EGR and increasing pilot fuel quantity to solve the above mentioned problems. Hence it can be concluded that:

1. By using the above mentioned methods, combustion process is improved and shifts towards TDC, meanwhile contribution of EGR is higher than other methods.
2. By employing EGR method, ignition delay period is reduced too much due to higher amount of intake charge temperature and the presence of very active radicals. But using other method makes this trend less remarkable.
3. Results of this work show that, by employing EGR method, NO_x emission is increased and yet this trend is not so significant using other method.
4. Both of the mentioned methods can decrease other major pollutants emission (UHC and CO) relative to the baseline dual fuel engine which may be due to better mixture preparation for combustion and also preparing required ignition sources for ignition of gaseous fuel air mixture.

7. ACKNOWLEDGEMENTS

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8. NOMENCLATURE

a	Heat transfer constant
A	Pre-exponential factor
A_s	Surface area for heat transfer [m^2]
b	Heat transfer constant
B	The temperature exponent
c	Heat transfer constant
C_V	Specific heat at constant volume $\left[\frac{kJ}{kg\,.\,K}\right]$
D	Cylinder bore [m]
E	Activation energy $\left[\frac{kJ}{kg}\right]$
k	Reaction rate constant
K	Thermal conductivity of gas $\left[\frac{W}{m\,.\,K}\right]$
M	Mass [kg]
M	Third body
M_d	Shape factor of heat release during diffusion combustion
M_p	Shape factor of heat release during premixed combustion
P	Pressure [bar]
Q	Heat loss [J]
Q_d	Quantity of pilot fuel to be consumed during diffusion combustion [J]
Q_p	Quantity of pilot fuel to be consumed during premixed combustion [J]
$\dot{Q}_{Loss}$	Heat loss rate of gaseous fuel [W]
$\dot{Q}_p$	Heat release rate of pilot fuel [W]
$\overline{R}$	Universal gas constant $\left[\frac{kJ}{kg\,.\,K}\right]$
R	Reaction rate
Re	Reynolds number
t	Time [S.]
T	Temperature [K]

$T_\circ$	A reference temperature [K]
T_w	Wall Temperature [K]
U	Internal energy $\left[\frac{kJ}{Kg}\right]$
V	Volume [m^3]
x	Concentration of species $\left[\frac{mol}{Kg}\right]$
α	A stoichiometric coefficient
ρ	Density $\left[\frac{Kg}{m^3}\right]$
θ	Crank angle [deg]
ϕ	Total equivalence ratio
$\theta_\circ$	Crank angle position of start of combustion [deg]
θ_d	Combustion duration of diffusion combustion [deg]
θ_p	Combustion duration of premixed combustion [deg]
τ	Ignition delay period of pilot fuel [deg]

Subscripts

a	Air
air-Th	Air with throttling
air-without-Th	Air without throttling
b	Relating to backward reaction
CNG	Compressed natural gas
EGR	Exhaust gas recirculation
f	Relating to forward reaction
i	Relating to species "i"
j	Relating to reaction step "j"
p	Pilot

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