

Theoretical Investigation of Combustion Process in Dual Fuel Engines at Part Load Considering the Effect of Exhaust Gas Recirculation

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Abstract

The dual fuel engines at part loads inevitably suffer from lower thermal efficiency and higher carbon monoxide and unburned fuel emission. This work is carried out 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 deriving from two superposed Wiebe's combustion functions to account for its contribution to ignition of gaseous fuel and the rest of total released energy. Chemical kinetics mechanism is consisted 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. Also, this model is able to calculate accumulated heat release, ignition delay and combustion duration of gaseous fuel air mixture. Therefore this paper is an attempt to investigate the combustion phenomenon at part loads and using hot exhaust gas recirculation (EGR) to improve the above mentioned drawbacks and problems. By employing this technique, it is found that, low percentages of EGR, considering its thermal and radical effects, have the positive effect on performance and emission parameters of dual fuel engines at part loads. Predicted values show good agreement with corresponding experimental values in a special engine operating condition (1/4 load, 1400 rpm). Implications will be discussed in details.

Keywords: Quasi-Two Zone Combustion Model, Chemical Kinetics, Natural Gas, Dual Fuel Engines, EGR

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 of both compression and spark ignition engine operations. As the gaseous fuel is mixed 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 the same 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 and Khoshbakhti [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 the complex combustion process in dual fuel engines and 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 and 5] developed a quasi-two zone model, which was used to predict the autoignition 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 measures for predicting spatial variations of the temperature and composition within the cylinder.

Pirouzpanah and Kashani [6] developed a model for dual fuel engines at full load conditions, which it simulates combustion process by using a multi-zone combustion model for diesel pilot fuel combustion and a conventional S.I. combustion model for modeling of 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 discrepancies between the predicted pollutants and experimental data.

Abd Alla et al [7 and 8] developed a quasi-two zone model, which was used for prediction of the combustion process in an I.D.I dual fuel engine. 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 by 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 re-

lease 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.

Hountalas and Papagiannakis [9-13] developed a two zone model for dual fuel engines, which takes into account, on a zonal basis, details of diesel fuel spray formation and its mixing with mixture of air and natural gas. They have investigated the effect of some parameters such as various natural gas percentages and total equivalence ratios on the performance and emission parameters over the whole range of engine operating conditions.

Karim and Liu [14] developed a multi-zone thermodynamic model which was able to describe the combustion process of dual fuel engines and predict 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 kinetic scheme is employed to describe the oxidation of the gaseous fuel right from start of compression stroke to the end of the expansion process. The associated formation and concentrations of exhaust gases are also established. The model not only can predict the onset of knock but also attend to the more demanding case of predicting the low load engine performance with the associated partial oxidation reactions and the production of exhaust gas emissions. Results of this model demonstrated that any measures that tend to 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 reduce markedly the concentration of the unconverted gaseous fuel and carbon monoxide in the exhaust gases.

Mansour et al [15] developed a computer program to model the combustion process in dual fuel engines. A detailed chemical reaction mechanism of natural gas and were used to predict the main combustion characteristics. This semi-empirical dual fuel engine model uses the Wiebe's function combustion model for different speeds and air fuel ratios.

Pirouzpanah and Jafarmadar [16] developed a quasi-dimensional combustion model to predict the combustion process of direct injection dual fuel diesel engines by using a detailed chemical kinetic model for gaseous fuel combustion. Chemical kinetic mechanism was consisted of 325 reactions with 53 species (GRI3). Heat release rate of pilot fuel at this model was considered by two Wiebe's functions. Predicted values of the performance parameters for dual fuel operation show good agreement with corresponding experimental data.

On the other hand, multi-dimensional models were also applied in simulation of combustion process in dual fuel engines [17-21]. This type of models requires a significant amount of computer power even for prediction of turbulent flow with a simplistic description of the complex chemical reaction which takes place during combustion. Accordingly, in general, most of the 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 powerful tool for analysis of combustion systems [1 and 22]. Such computer models have contributed for better understanding and solution of longstanding practical combustion problems in dual fuel engines, including pre-ignition re-

actions, emission, part load performance and knocking [20].

The present contribution describes a quasi-two zone thermo-dynamic model that is developed to describe the combustion processes of dual fuel engines. This model was consisted of a single zone model with detailed chemical kinetics for combustion of natural gas fuel and a subsidiary zone for combustion of pilot fuel to account for its contribution for ignition of gaseous fuel air mixture. Chemical kinetics mechanism was consisted of 112 reactions with 34 species. This so called "combined" 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 and species concentration. Also, this model is able to calculate accumulated heat release, ignition delay and combustion duration of gaseous fuel. As mentioned before, the dual fuel engines at part loads 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 hot EGR is used to solve the above mentioned problems.

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 chemical reactions mechanism has been used and the relevant equations are solved numerically in the present study.

The performance of a typical dual fuel diesel 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, combusted and subsequently expanded towards the exhaust stage. During these stages, 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 the energy release, 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:

$$\tau = 4.3 \times 10^{-3} P^{-2.5} \phi^{-1.04} \exp\left(\frac{5000}{T}\right) \quad (1)$$

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 behavior of combustion process, commonly identified as premixed combustion and diffusive combustion, regardless of engine operating conditions. To fit adequately the rate of combustion curve for this two stages behaviour, two superposed Wiebe's combustion functions were combined [23]. The employed two Wiebe's functions have the following form:

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

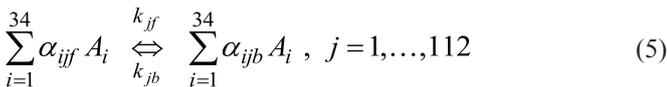
4.2. Gaseous fuel

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:



For the j th reaction, the forward rate constant can be expressed by the following Arrhenius expression:

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

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

The reaction rates for the j th reaction in the forward and backward directions can be expressed respectively in terms of

the concentration as follows:

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

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

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

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

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:

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

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

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

\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 [24]:

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

Thus, the energy equation (11) and the simultaneous set of equations of (10), one for each species, result in a simultaneous set of non-linear first order differential equations which were solved by the authors developed software. This software employs DVODE (Double precision variable ordinary differential equation) for solving stiff and non-stiff ODEs corresponding to the net rate of production of each species and energy equation, respectively. Also it is necessary to mention that the CPU time for solving these simultaneous set of differential equations is less than twenty minutes.

It is known that the main combustion of gaseous fuel in dual fuel engines occurs after ignition of diesel pilot fuel which can be considered as ignition sources of gaseous fuel air mixture. Therefore, it can be assumed that the trend of combustion of gaseous fuel air mixture is similar to the combustion of spark ignition engines. Hence the following definitions are used to characterize the energy release aspects of combustion of gaseous fuel air mixture [25]:

• **Flame development angle or ignition delay period ($\Delta\theta_d$):**

The crank angle interval between the ignition of pilot fuel and the time when a small but significant fraction of the cylinder mass has burned or fuel chemical energy has been released. In this work, this fraction is considered 10 percent.

• **Rapid burning angle ($\Delta\theta_b$):** The crank angle interval required to burn the bulk of the gaseous fuel and air charge. It is defined as the interval between the end of the flame development stage and the end of the flame propagation process in which mass fraction burned or energy release fraction is considered 90 percent.

• **Overall burning angle ($\Delta\theta_o$):** The duration of the overall burning process. It is the sum of $\Delta\theta_b$ and $\Delta\theta_d$.

5. Results and discussion

5.1. Baseline dual fuel engine

Fig. 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 predicted values of cylinder pressure (Fig. 1) show good agreement with those obtained from experimental work [26].

Fig. 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 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 loads, the total energy release rates diagram of the dual fuel engine is very similar to that observed in pure diesel engine operation [14]. It is necessary to know that 0 degree position in all of the figures is related to TDC.

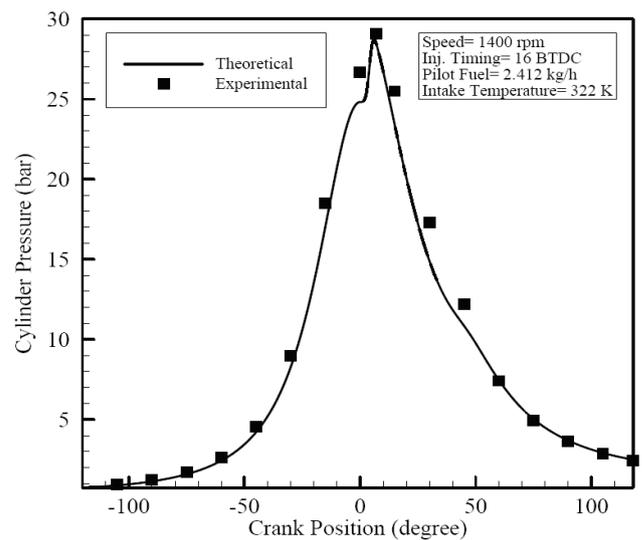


Fig. 1. Comparisons of calculated and experimental [25] cylinder pressure of dual fuel engine. (1/4 full load, energy ratio: 20% pilot diesel, 80% natural gas)

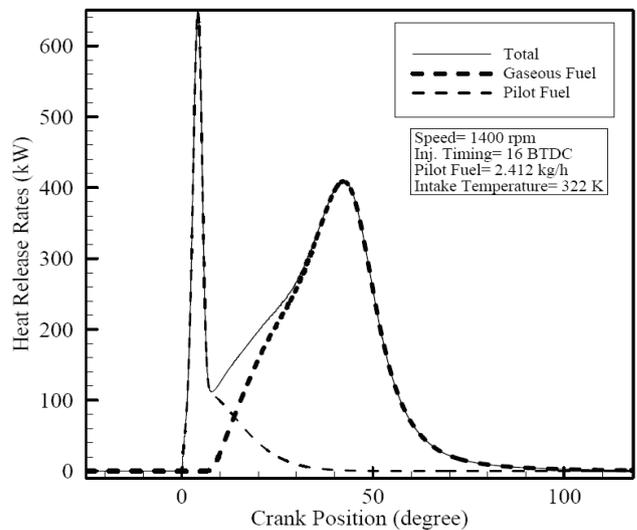


Fig. 2. Variations of the calculated heat release rates of both pilot and gaseous fuels with crank position (1/4 full load, energy ratio: 20% pilot diesel, 80% natural gas)

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

Make and model	Fuels
Mercedes-Benz OM-355	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 @ 2200 rpm
Capacity: 11.58 (liter)	Max. Torque: 824 N.m @ 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

Load	Species (Values)	UHC (ppm)	CO (%)	NO _x (ppm)	O ₂ (%)
1/4 Full load	Experimental	626	0.311	157	13
	Theoretical	562	0.387	0	13.04

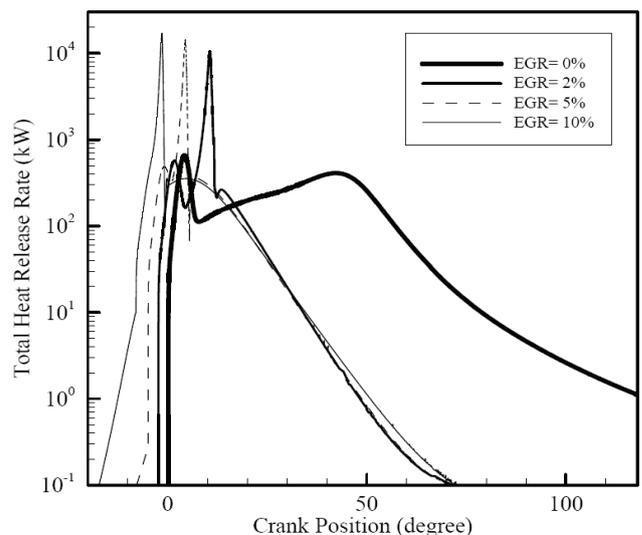


Fig. 3. Variations of the calculated total heat release rate with crank position for different values of EGR.

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 it 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. Therefore the main purpose of this work is to investigate the combustion phenomena at part loads and using EGR to solve the mentioned problems. The low load which is selected for this study is a 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 EGR

In general, EGR is a useful method for reducing NO_x formed in the cylinder due to lowered combustion temperature resulting from the increased inert gas in the cylinder charge (common or traditional effect). As this type of EGR has negative effects on other pollutants and performance parameters, hence it is classified as negative effects of EGR category. But, in dual fuel engines at part load conditions, EGR has sound and positive effects on performance and emission parameters. These improvements are maybe originated from firstly getting better mixture quality of gaseous fuel air charge, secondly using active radicals for enhancement of combustion of gaseous fuel air mixture (Radical effect) and thirdly increasing intake charge temperature (thermal effect). Hence salient features of EGR to be shown are negative and positive effects as mentioned above. To quantify the amount of EGR, the EGR percentage is defined by:

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

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 the composition of cylinder charge at the exhaust valve opening condition by running the authors developed software for baseline dual fuel engine.

Fig. 3 shows logarithmically variations of the calculated heat release rate with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that with increasing percentage of EGR, heat release rate is increased and the combustion process is shifted towards TDC. 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 which show themselves at EGR percentages higher than 2. Also, with increasing EGR, ignition delay is short-

ened and the share of premixed combustion of pilot fuel is reduced. In other words, the share of diffusion combustion of pilot fuel is increased and subsequently the heat release rate of gaseous fuel air mixture extends towards the end of expansion process.

Fig. 4 indicates variations of the calculated accumulated heat release of gaseous fuel air mixture with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that with increasing percentage of EGR, accumulated heat release is increased and the combustion process of gaseous fuel air mixture is shifted towards TDC. The main reason for this trend is perhaps due to overcoming of positive effects of EGR on negative effects of EGR.

Table 3 shows Variations of the calculated ignition delay, rapid burning and overall burning periods of gaseous fuel air mixture for different values of EGR. It can be seen that all of these periods are long at baseline condition of dual fuel engine and hence combustion inefficiency can be originated from these long periods. With increasing the EGR up to 2 percent, ignition delay and combustion duration of gaseous fuel air mixture are shortened to reasonable amounts and hence combustion process is promoted. Also, with increasing the EGR to more than 2 percent, combustion duration of gaseous fuel air mixture is increased. Therefore, combustion process is deteriorated relative to the case of EGR percentage is equal to 2.

TABLE 3. Variations of the calculated ignition delay, rapid burning and overall burning periods of gaseous fuel air mixture for different values of EGR.

EGR (%)	0	2	5	10
$\Delta\theta_d$ (°CA)	21.3	11.3	8.23	10.26
$\Delta\theta_b$ (°CA)	33.2	2.7	6.17	6.18
$\Delta\theta_o$ (°CA)	54.5	14	14.4	16.44

Fig. 5 describes variations of the calculated cylinder pressure with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. As indicated in this figure, the charge peak pressure is increased with increasing EGR up to 5 percent and afterward it is reduced with increasing EGR. These trends are consistent with those observed in the heat release rate curves.

Fig. 6 indicates variations of the calculated cylinder temperature with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that with increasing EGR, cylinder temperature is increased but this trend is not remarkable. The main reason for this trend is perhaps due to overcoming of positive effects of EGR on negative effects of EGR.

Fig. 7 shows variation of total work done with crank position for different values of EGR. It can be seen that with increasing the EGR up to 2 percent, the total work done is increased and afterward it reduces. These trends are consistent with those observed in the heat release rate and accumulated heat release curves.

Fig. 8 describes variations of the indicated thermal efficiency for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. As indicated in this figure, indicated thermal efficiency reaches its maximum amount with 2 percent of EGR.

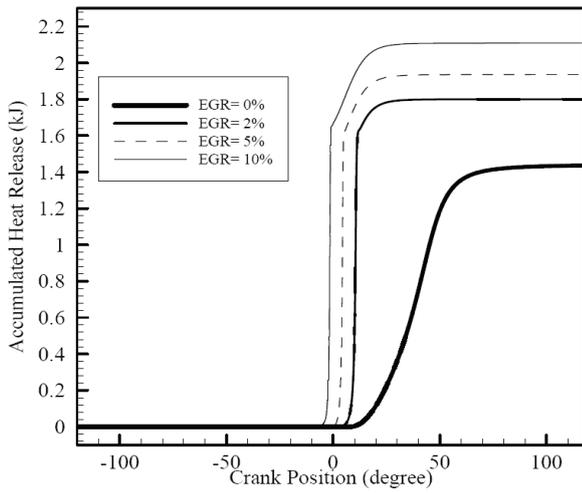


Fig. 4. Variations of the calculated accumulated heat release of gaseous fuel air mixture with crank position for different values of EGR

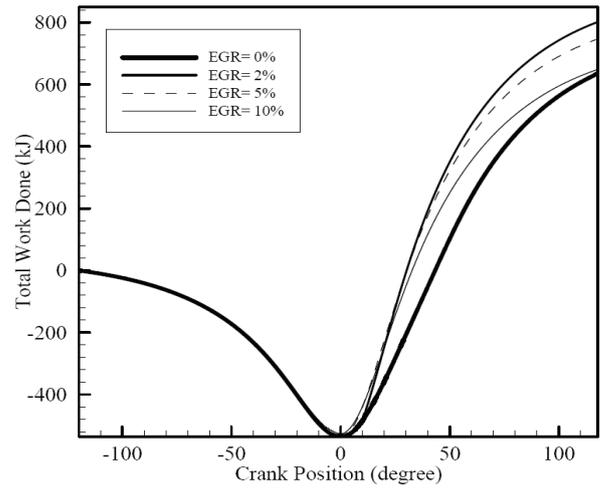


Fig. 7. Variations of the calculated total work done with crank position for different values of EGR.

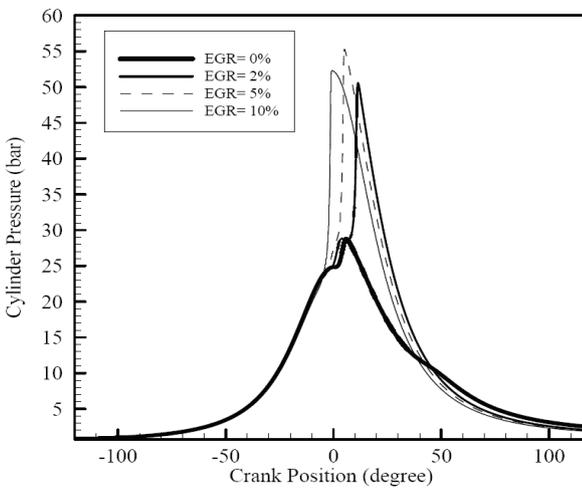


Fig. 5. Variations of the calculated cylinder pressure with crank position for different values of EGR.

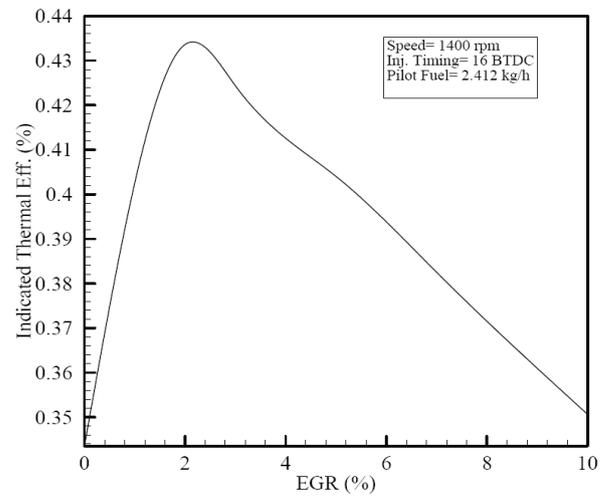


Fig. 8. Variations of the calculated indicated thermal efficiency with different values of EGR.

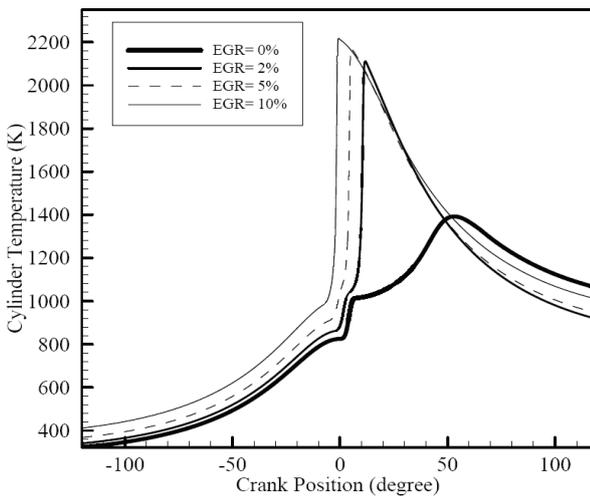


Fig. 6. Variations of the calculated cylinder temperature with crank position for different values of EGR.

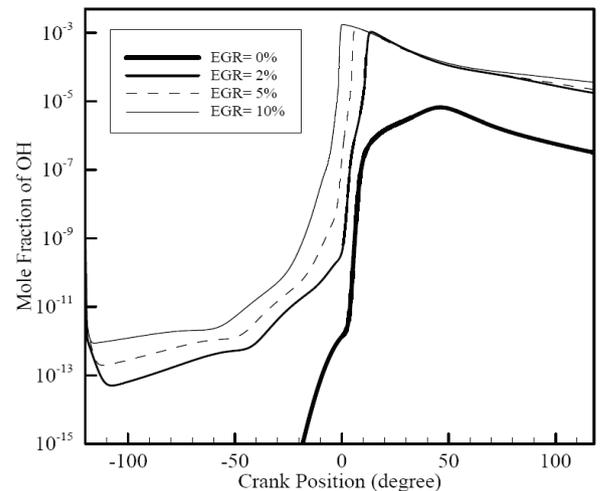
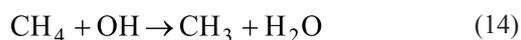


Fig. 9. Variations of the calculated mole fraction of OH radical with crank position for different values of EGR.

Fig. 9 indicates logarithmically variations of the calculated mole fraction of OH radical with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that the OH radical produced from the combustion of fuels increases with the increasing of EGR. Sensitivity analysis shows that the OH radical has the greatest effect on ignition delay period through the reaction:



Thus, as indicated in Fig. 10, increasing the OH radical with increasing EGR, decreases the ignition delay period of the pilot fuel.

Figs. 11 to 13 show typically the mole fractions of the active radicals with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. 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.

Fig. 14 indicates variations of the calculated mole fraction of CH_2O with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that the amount of this specie at the end of expansion stroke at baseline dual fuel engine condition ($\text{EGR} = 0$) is considerable.

Fig. 15 indicates variations of the calculated mole fraction of O_2 with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. As indicated in this figure, with increasing of EGR, the mole fraction of this specie at the beginning of compression stroke is reduced and consumption of this specie is increased.

Fig. 16 shows variations of the calculated mole fraction of CH_4 with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that consumption of CH_4 at higher amounts of EGR is started earlier than lower percentages of EGR.

Fig. 17 describes variations of the calculated mole fraction of CO with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. As indicated in this figure, with increasing EGR, CO emission is reduced which is due to better mixture preparation for combustion and positive effects of active radicals.

Fig. 18 indicates variations of the calculated concentration of NO_x with crank position for different values of EGR for a fixed pilot fuel quantity of the dual fuel engine. It can be seen that with increasing of EGR, NO_x emission is increased but this trend is not so much. Apart from the effect of the inert gas brought by EGR, hot recirculated exhaust gas and very active radicals would raise the charge temperature; thereby they can influence combustion and NO_x emission. Here, the thermal and chemical effects of EGR may acting together.

6. Conclusions

The present quasi-two zone combustion model, combined with the detailed chemical kinetics scheme, can provide a de-

scription of the main features of the combustion process in dual fuel engines. The corresponding performance of dual fuel engines can be analyzed by this model at whole engine operating conditions. As already known, the dual fuel engines at part loads inevitably suffer from 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 EGR to solve the above mentioned drawbacks. Hence it can be concluded that:

1. With employing hot EGR, taken from very close to the exhaust valve or using internal EGR, the amount of EGR that is required to overcome the problems of dual fuel engines at part load conditions is low. The improvements may be originated from getting better mixture quality of gaseous fuel air charge, higher intake temperature and using very active radicals for ignition of gaseous fuel air mixture. Applying this technique, installation of heat exchanger for heating of intake charge is no longer required.

2. Ignition delay period of pilot fuel is reduced due to higher amounts of intake charge temperature and the presence of very active radicals. Also by using higher percentages of EGR, the ignition delay period is reduced too much which may have negative effects on performance of dual fuel engines.

3. Ignition delay and combustion periods of gaseous fuel air mixture are reduced due to higher amounts of intake charge temperature and the presence of very active radicals. Also by using higher percentages of EGR, these periods are increased which can have negative effects on performance of dual fuel engines.

4. The unburned hydrocarbons formed in the cylinder of dual fuel engines is mainly composed of methane which decreases as the EGR increases. This is due to improvement and enhancement of oxidation process in the lean mixtures. Also the presence of very active radicals and higher amounts of intake charge temperature can promote the combustion process and therefore reduce concentration of unburned hydrocarbons emission in exhaust gases.

5. Results of this work show that, at lower percentages of EGR, NO_x emission is increased but this trend is not so remarkable. In general, apart from rather negative effect of EGR, the results also show that, performance and emission of dual fuel engines at part loads are enhanced.

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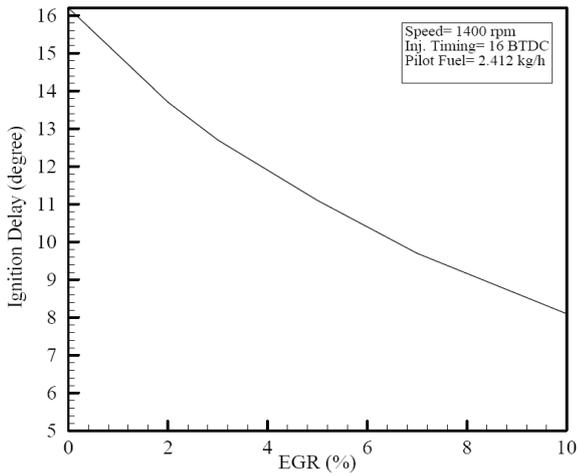


Fig. 10. Variations of the calculated ignition delay period of pilot fuel with different values of EGR.

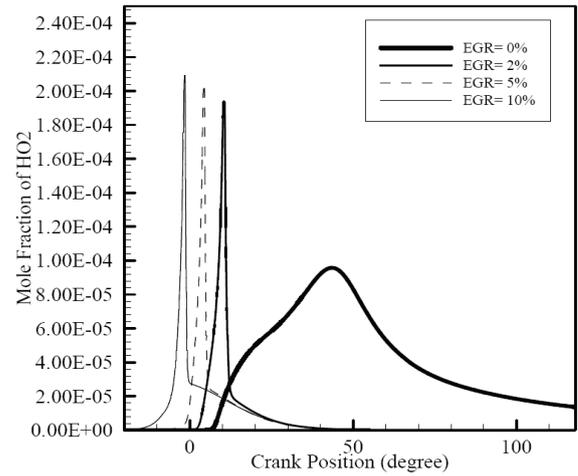


Fig. 13. Variations of the calculated mole fraction of CH3 with crank position for different values of EGR.

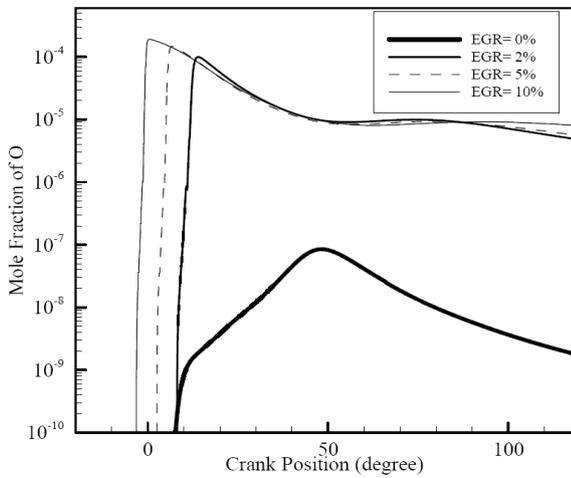


Fig. 11. Variations of the calculated mole fraction of the O radical on a logarithmic scale with crank angle for different values of EGR.

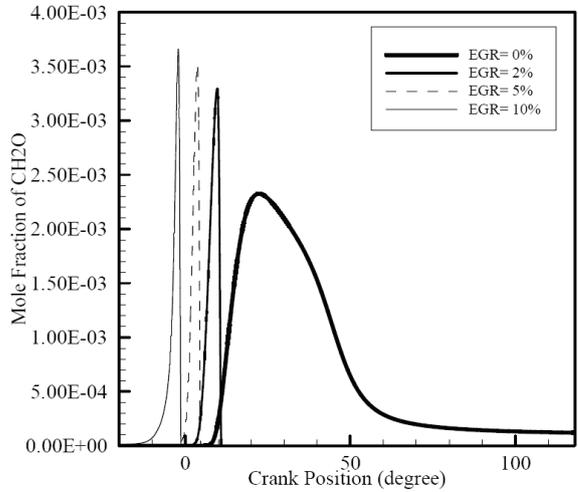


Fig. 14. Variations of the calculated mole fraction of CH2O with crank position for different values of EGR.

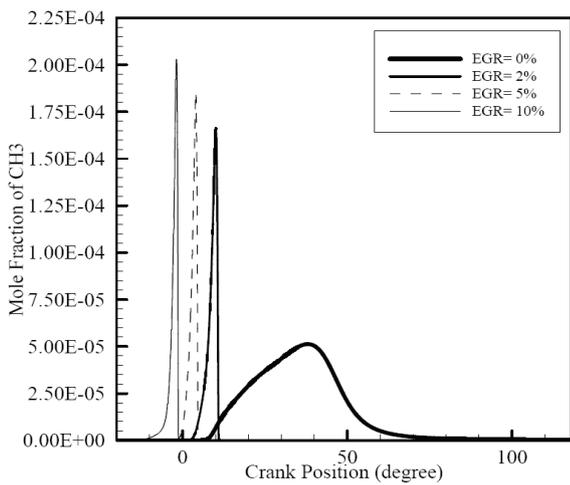


Fig. 12. Variations of the calculated mole fraction of HO2 with crank position for different values of EGR.

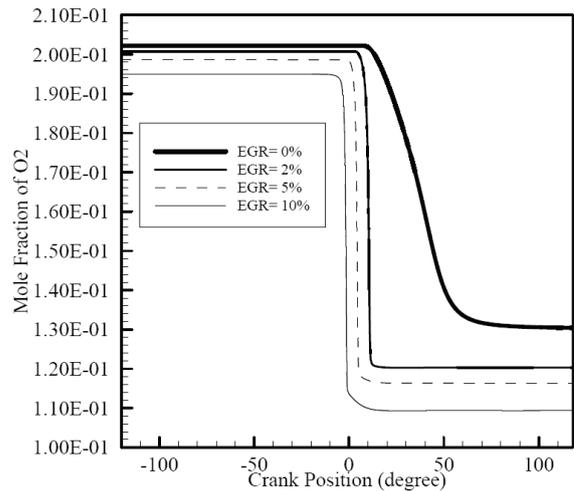


Fig. 15. Variations of the calculated mole fraction of O2 with crank position for different values of EGR.

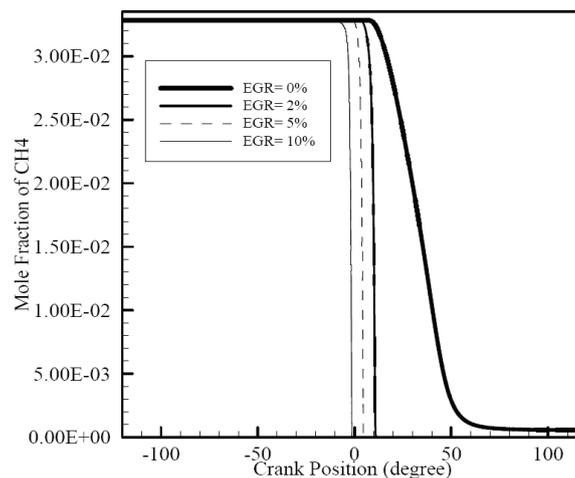


Fig. 16. Variations of the calculated mole fraction of CH4 with crank position for different values of EGR.

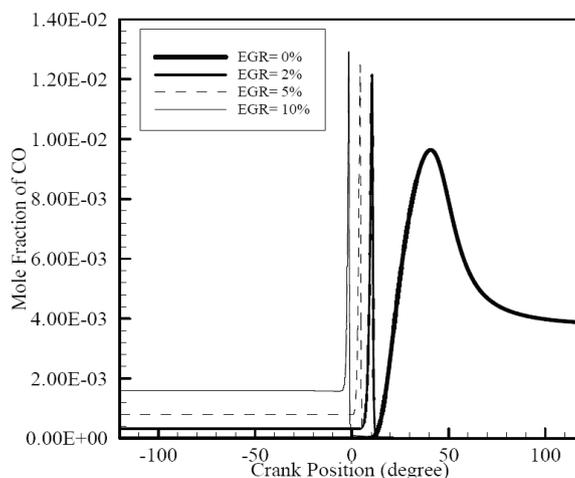


Fig. 17. Variations of the calculated mole fraction of CO with crank position for different values of EGR.

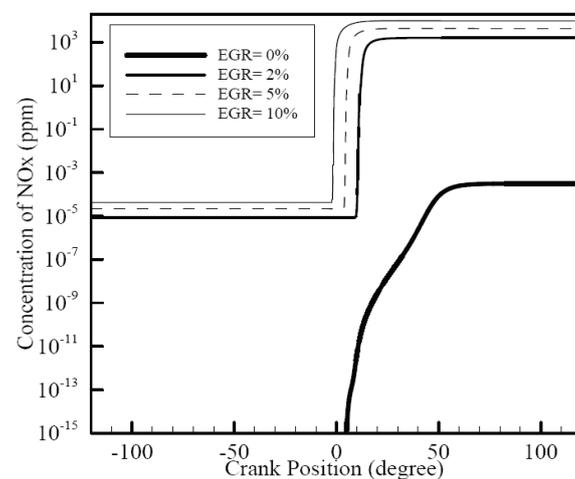


Fig. 18. Variations of the calculated concentration of NOx with crank position for different values of EGR.

NOTATION			
		T_w	Wall Temperature
a	Heat transfer constant	U	Internal energy
A	Pre-exponential factor	V	Volume
A_s	Surface area for heat transfer	x	Concentration of species
b	Heat transfer constant	α	A stoichiometric coefficient
B	The temperature exponent	ρ	Density
c	Heat transfer constant	θ	Crank angle
C_V	Specific heat at constant volume	ϕ	Total equivalence ratio
D	Cylinder bore	θ_c	Crank angle position of start of combustion
E	Activation energy	θ_d	Combustion duration of diffusion combustion
k	Reaction rate constant	θ_p	Combustion duration of premixed combustion
K	Thermal conductivity of gas		
M	Mass	τ	Ignition delay period of pilot fuel
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		
Q	Heat loss		
Q_d	Quantity of pilot fuel to be consumed during diffusion combustion		
Q_p	Quantity of pilot fuel to be consumed during premixed combustion		
\dot{Q}_{Loss}	Heat loss rate of gaseous fuel	Subscripts	
\dot{Q}_p	Heat release rate of pilot fuel	a	Air
\bar{R}	Universal gas constant	b	Relating to backward reaction
R	Reaction rate	CNG	Compressed natural gas
Re	Reynolds number	EGR	Exhaust gas recirculation
t	Time	f	Relating to forward reaction
T	Temperature	i	Relating to species "i"
T_c	A reference temperature	j	Relating to reaction step "j"
		p	pilot

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