

A PREDICTIVE MODEL FOR THE COMBUSTION PROCESS IN DUAL FUEL ENGINES AT PART LOADS USING A QUASI DIMENSIONAL MULTI ZONE MODEL AND DETAILED CHEMICAL KINETICS MECHANISM

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Abstract This work is carried out to investigate combustion characteristics of a dual fuel (diesel - gas) engine at part loads, using a quasi - dimensional multi zone combustion model (MZCM) for the combustion of diesel fuel and a single zone model with detailed chemical kinetics for the combustion of natural gas fuel. Chemical kinetic mechanisms consist of 184 reactions with 50 species. 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 and heat release rate (H.R.R). The dual - fuel engines, inevitably suffer from lower thermal efficiency, higher carbon monoxide and unburned hydrocarbon at part local conditions. Therefore this paper is an attempt to investigate the combustion phenomena at part loads and using methods such as injection timing advance, increasing pilot fuel quantity, and intake air throttling to improve the mentioned problems. It was found that the advancing of diesel injection timing gave little improvement in combustion parameters but other proposed methods promoted better combustion. Predicted values show good agreement with corresponding experimental values over a whole range of engine operating conditions. Implications will be discussed in details.

Key Words Multi Zone Combustion Model, Chemical Kinetics, Natural Gas, Dual Fuel Engines

چکیده کار حاضر به منظور بررسی خواص احتراقی یک موتور دوگانه سوز در بار جزئی با استفاده از یک مدل احتراقی چندمنطقه ای شبه بعدی برای احتراق سوخت دیزل و یک مدل احتراقی تک منطقه ای با مکانیزم سینتیک شیمیایی مفصل برای احتراق سوخت گاز طبیعی انجام شده است. مکانیزم سینتیک شیمیایی شامل ۱۸۴ واکنش و ۵۰ گونه شیمیایی می باشد. این مدل احتراقی قادر به پیشگویی توسعه فرایند احتراق با زمان و پارامترهای عملکردی مهم مرتبط از جمله فشار، دما و آهنگ گرمای آزاد شده است. موتورهای دوگانه سوز در بارهای جزئی به ناچار متحمل بازده حرارتی پایین و منوکسید کربن و سوخت نسوخته با مقدار زیاد می شوند. از اینرو این مقاله به بررسی پدیده احتراق در بارهای جزئی می پردازد و در این ارتباط از روشهایی نظیر آوانس زمان پاشش سوخت دیزل، افزایش مقدار سوخت دیزل و خفانش هوای ورودی برای بهبود مشکلات ذکر شده در بارهای جزئی استفاده شده است. نتایج نشان می دهند که آوانس زمان پاشش سوخت دیزل تاثیر اندکی در بهبود پارامترهای احتراقی داشته و دو روش دیگر، فرایند احتراق را بهتر بهبود می دهند. مقادیر پیش بینی شده توسط مدل در توافق خوبی با کارهای تجربی پیشین در تمامی شرایط عملکردی موتور می باشند. 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 emissions without

an 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 diesel and spark ignition engine operations. As the gaseous fuel is mixed with the intake air in the

intake manifold, the mixture formation is greatly modified and then undergoes a multi - point ignition due to the compression ignition and combustion of 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 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.

A suitable computer - based analytical model can provide an adequate means for describing details of the combustion process in engines and help to reduce prohibitive development time and cost usually involved in the conversion of diesel engines to dual fuel operation. However, the developments of comprehensive simulation models for dual fuel engine operations have so far been very limited due to the complex combustion processes involved. In dual fuel engines, zero dimensional models were the main 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 to near full load. The model could not be applied to predict neither exhaust emissions nor the operation at light loads when lean mixtures are employed. This is due mainly 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 simulates a combustion process by using a multi - zone combustion model for diesel pilot jet combustion and a conventional S.I. combustion model for simulating of combustion of premixed gas/air mixture. 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 which perhaps are due to weakness of employed emission sub models.

Abd Alla et al. [7,8] developed a quasi-two zone model, which was used for prediction of the combustion processes in dual fuel engines and some of their performance features. Their model emphasizes the effects of chemical kinetic activity of the premixed gaseous fuel on the combustion performance, while the role of pilot fuel in the ignition and heat release processes are considered by using two superposed Wiebe's 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.

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

Karim and Liu [14] developed a multi - zone thermodynamic model which was able to describe the combustion processes 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 to the end of the expansion process. The associated formation and concentrations of exhaust emissions are also established. The model not only can predict the onset of the knock but also attends to the more demanding case of predicting the part load engine performance with the associated partial oxidation reactions and the production of exhaust 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 by 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 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 empirical dual fuel engine simulation has been carried out using the Wiebe model for different speeds and air fuel ratios.

Pirouzpanah and Jafarmadar [16] 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. The chemical kinetic mechanism consisted of 325 reactions with 53 species (GRI3). The heat release rate of pilot fuel in this model was considered by two Wiebe functions. Predicted values of performance parameters for dual fuel operation show good agreement with the corresponding experimental data.

On the other hand, multi - dimensional models also applied to the simulation of the combustion process in dual fuel engines [17-21]. This type of model requires a significant amount of computer power even for the prediction of turbulent flow with a simplistic description of the complex chemical reaction which takes place during combustion. Accordingly, most of the combustion models used are either the zero dimensional or quasi-dimensional types.

In recent years, chemical kinetic modeling has become an important powerful tool for the analysis of combustion systems [1,22]. Such computer models have contributed to the understanding and solution of longstanding practical combustion problems in dual fuel engines, including performance or pre - ignition reactions, emission, part load performance and knocking [20].

The present contribution describes a quasi - dimensional thermodynamic model that was developed to describe the combustion processes of dual fuel engines. This model consists of a quasi - dimensional multi zone combustion model (MZCM) for the combustion of diesel fuel and a single zone model with detailed chemical kinetics for the combustion of natural gas fuel. A chemical

kinetic mechanism consists of 184 reactions with 50 species. 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 and heat release rate. The dual fuel engines, at part loads, inevitably suffer from lower thermal efficiency and higher carbon monoxide and unburned hydrocarbons. Therefore, this paper is an attempt to investigate the combustion phenomena at part loads and using methods such as advancing pilot fuel injection timing, increasing pilot fuel quantity, and intake air throttling to improve the mentioned problems.

2. DESCRIPTION OF THE PRESENT MODEL AND ASSUMPTIONS

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, pre-ignition combustion reactions which may produce some intermediate species such as radicals, carbon monoxide and formaldehyde. These can have profound effects on subsequent combustion processes of the engine. In addition, the presence of the pilot fuel is considered as a heat source which is deriving from quasi - dimensional multi - zone model to account for its contribution to ignition and remaining fuel energy and remaining energy release. Also it is assumed that, in the modeling of pilot jet combustion by MZCM, there is uniform pressure in all zones, but there exists zonal temperature distribution within the pilot jet.

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 184 elementary reaction mechanisms have been used in the present study which was taken from Ref. 2 and 15.

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 (CNG+Air) is assumed to be initially homogenous and mixed during the suction stroke. The mixture,

following intake valve closure, is then compressed, combusted and subsequently expanded to the exhaust stage. During these stages, reaction kinetics were used to predict, in detail, the changes in the concentration and properties of the mixture (CNG+Air) in the cylinder and subsequent changes of the energy release rates with time. When the pilot diesel fuel injected into the chamber, its energy release and its contribution to the composition of the cylinder charge, including its temperature and pressure are taken into account. Thus, this quasi – multi - zone combustion model consists of a main zone, involving the gaseous fuel air mixture with its detailed chemical kinetic reaction activity and the multi - zone regions due to the combustion of the pilot fuel as shown in Figure 1.

In this model it is assumed for simplicity that, the interaction between the two types of fuel to be primarily thermal with no direct chemical interaction between them.

This model assumes that the pilot fuel acts as a heat source that provides thermal energy for the ignition of the gaseous fuel air charge and also the species evaluated from MZCM can be considered. The ignition delay period of the pilot fuel is calculated by MZCM and the effect of gaseous fuel on this period is assumed to be negligible. 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 reactions of the gaseous fuel air charge. It was assumed throughout that, there were no temperature, pressure and concentration gradients within the cylinder charge. Gas leakage from the cylinder, once the valves closed, was assumed to be negligible. All components of the mixtures were considered to behave as an ideal gas. For calculation of pressure and temperature, average pressure and mass averaged temperature, which are derived from the two models, are considered as pressure and temperature of dual fuel engine.

3. MATHEMATICAL TREATMENT

3.1. Pilot Fuel For modeling of pilot jet combustion, the multi - zone combustion model

[6,24] is modified in the researchers laboratory and used in the present work. Different zones in pilot fuel combustion are shown in Figure 2. It is assumed that the quantity of air that is utilized for combustion of pilot jet fuel and gaseous fuels are proportional to the mass ratio of those fuels that are supplied to the engine simultaneously.

The following equations are sufficient to

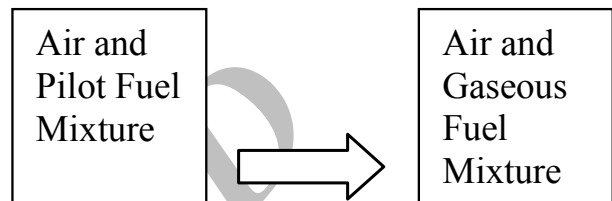


Figure 1. The scheme of combustion model.

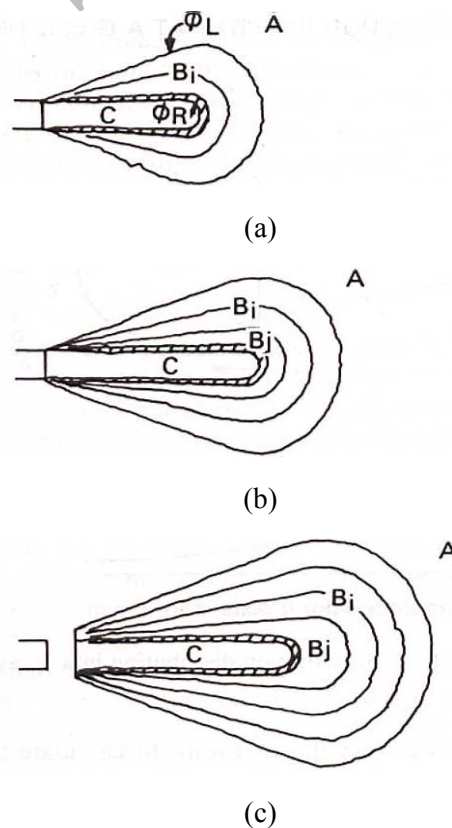


Figure 2. Evolution of combustion zones at pilot fuel combustion: (a) incipient ignition in 'n' zones, (b) continuous entrainment and mixture preparation and (c) end of injection continuous mixing [24].

calculate the cylinder pressure and temperature and composition of each zone. Conservation of Mass:

$$\frac{dM_c}{d\theta} = \frac{dM_f}{d\theta} + \frac{dM_{ec}}{d\theta} - \frac{dM_{mp}}{d\theta} \quad (1)$$

$$\sum \frac{dM_{ebi}}{d\theta} + \frac{dM_{ec}}{d\theta} = -\frac{dM_a}{d\theta} \quad (2)$$

$$\frac{dM_{bi}}{d\theta} = \frac{dM_{ebi}}{d\theta} \quad i = 1, 2, 3, \dots, (j-1) \quad (3)$$

$$\frac{dM_{bj}}{d\theta} = \frac{dM_{ebj}}{d\theta} + \frac{dM_{mp}}{d\theta} \quad (4)$$

Conservation of Energy:

$$\begin{aligned} \frac{d(M_c U_c)}{d\theta} = & -P \frac{dV_c}{d\theta} + \frac{dQ_c}{d\theta} + \frac{dM_f}{d\theta} H_f \\ & + \frac{dM_{ec}}{d\theta} H_a - \frac{dM_{mp}}{d\theta} H_{mp} \end{aligned} \quad (5)$$

$$\frac{d(M_a U_a)}{d\theta} = -P \frac{dV_a}{d\theta} + \frac{dQ_a}{d\theta} - \frac{dM_a}{d\theta} H_a \quad (6)$$

$$\frac{d(M_{bi} U_{bi})}{d\theta} = -P \frac{dV_{bi}}{d\theta} + \frac{dQ_{bi}}{d\theta} + \frac{dM_{ebi}}{d\theta} H_a$$

$$i = 1, 2, 3, \dots, (j-1) \quad (7)$$

$$\begin{aligned} \frac{d(M_{bj} U_{bj})}{d\theta} = & -P \frac{dV_{bj}}{d\theta} + \frac{dQ_{bj}}{d\theta} + \frac{dM_{mp}}{d\theta} H_{mp} \\ & - \frac{dM_{ebj}}{d\theta} H_a \end{aligned} \quad (8)$$

Equation of State:

$$P_i V_i = m R_i T_i \quad (9)$$

Volume Constraint:

$$\sum V_{bi} + V_a + V_c = V_{ch}(\theta) \quad (10)$$

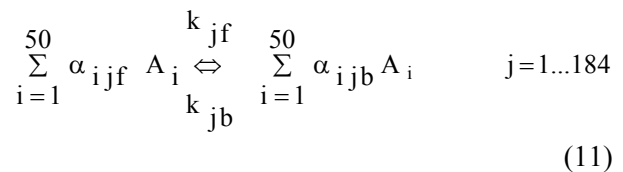
From the above equations, combustion parameters of pilot fuel at part loads such as H.R.R can be calculated.

3.2. Gaseous Fuel A single zone analytical model that incorporates detailed chemical kinetic mechanisms was employed to investigate the gaseous fuel combustion characteristics. The scheme consisted of 184 reactions steps and the following 50 chemical species:

C₂H, C₂H₂, C₂H₃, C₂H₄, C₂H₅, C₂H₆, C₃H₈, nC₃H₇, iC₃H₇, C₃H₆, CH₄, CH, CH₂, CH₃, CH₂CO, CH₂O, CH₃O, CHO, CH₃CHO, CH₃CO, CO, CO₂, H₂, H, HO₂, H₂O₂, H₂O, O, O₂, OH, N₂, N₂H₂, NNH, NO, HNO, NH₃, NH₂, N₂O, N, NO₂, HCN, HCCO, CN, NH, C₂N₂, HOCN, HNCO, NCO, HCNO, C.

With employment of detailed 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 time during the compression, combustion and expansion stages. The energy and species equation for the homogenous reacting mixture becomes:

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



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

$$k_{jf} = A_{jf} T^B \exp\left(-\frac{E_{jf}}{TR}\right) \quad j=1 \dots 184 \quad (12)$$

While the 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:

$$R_{jf} = k_{jf} \prod_{i=1}^{50} (\rho x_i)^{\alpha_{ijf}} \quad j=1 \dots 184 \quad (13)$$

$$R_{jb} = k_{jb} \prod_{i=1}^{50} (\rho x_i)^{\alpha_{ijb}} \quad j=1\dots 184 \quad (14)$$

The piston displacement is a known function of time [19] and hence the density - time is also known in terms of the mass content of the cylinder:

$$\rho(\theta) = \frac{M}{V(\theta)} \quad (15)$$

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 is given by a relation of the following form:

$$-\rho \frac{dx_i}{dt} = \sum_{j=1}^{184} (\alpha_{ijf} - \alpha_{ijb}) (R_{jf} - R_{jb}) \quad i=1\dots 50 \quad (16)$$

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

$$\sum_{i=1}^{50} X_i \bar{R}T \times \frac{V'(t)}{V(t)} + \sum_{i=1}^{50} \left\{ \left(\int_{T_0}^T C_{Vi} dT + \Delta U_{fi} \right) \frac{dx_i}{dt} + x_i C_{Vi} \frac{dT}{dt} \right\} = \dot{Q}_p + \dot{Q}_{Loss} \quad (17)$$

\dot{Q}_p is the heat release rate of pilot fuel which is derived from multi - zone combustion model. \dot{Q}_{Loss} , is heat loss rates of the natural gas air mixture which is not considered directly and it is included by reducing the relevant quantity of pilot fuel heat release rate.

Thus, the energy Equation 17 and the simultaneous set of Equations of 16, one for each species, result in a set of non - linear first order simultaneous equations which were solved by Chemkin II software.

4. RESULTS AND DISCUSSION

Figure 3 shows typical variations of the calculated cylinder pressure with a crank position when a small quantity of liquid diesel fuel was maintained

to provide pilot ignition in a DI six cylinder engine. The specifications of engine are shown in Table 1. The corresponding experimental values [23] are also shown. The predicted values show good agreement with the corresponding experimental results.

Figure 4 represents the corresponding variations of the heat release rates with the crank position 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 tends to be smooth and it is associated with low rates. The second stage of the combustion occurs due to flame propagation within the gaseous fuel air charge. This is associated with faster rates of the exothermic combustion reactions. This results in much greater energy release rates immediately following pilot ignition and, thus, is the main component of energy release of the dual fuel engine. Because of the high temperature, very fast reaction rates of the gaseous fuel air mixture are produced. Thus, the large quantity of energy release of the gaseous fuel air mixture is achieved in a very short time.

Figure 5 shows typical variations of the calculated cylinder pressure with the crank position at part load condition (1/4 full load) for a DI six cylinder dual fuel engine which its specifications are shown in Table 1. The corresponding experimental values [23] are also shown. The predicted values show good agreement with the corresponding experimental results.

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: 22.8% pilot diesel, 77.2% natural gas). It can be seen that all of the theoretical values are in good agreement with the experimental amounts except for CO and CH₄ species which perhaps originates from a weakness of corresponding employed chemical kinetics reaction mechanism.

4.1. Effects of Gaseous Fuel Quantity Figure 6 shows logarithmical variations of the calculated total heat release rate with the crank position for different values of the total equivalence ratios for a

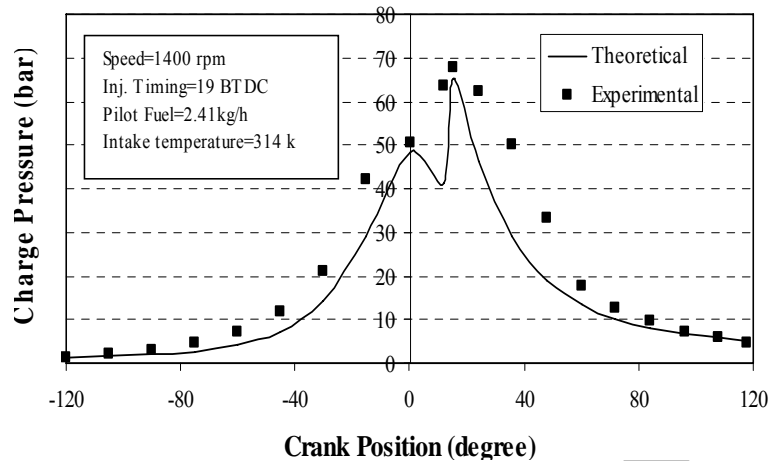


Figure 3. Comparisons of calculated and experimental [23] charge pressure of dual fuel engine. (Full load, Energy ratio: 10% pilot diesel, 90% natural gas).

TABLE 1. General Specifications of OM-355 Engine.

Engine: OM-355 Diesel - Natural gas	Make and model: Mercedes - Benz
Type: Direct injection, naturally aspirated, Heavy duty Vehicle Diesel, four stroke	No of Nozzles/Injector: 4
Cylinders: 6	Nozzle opening pressure: 195 (bar)
In – line - Vertical	Max. Power: 240 (HP) at 2200 (RPM)
Bore*Stroke: 128 (mm)*150 (mm)	Max. Torque: 824N.m at 1400 (RPM)
Capacity: 11.58 (liter)	Engine Speed: 1400 (RPM)
Compression Ratio: 16.1:1	

dual fuel engine fuelled with natural gas while employing a fixed pilot diesel injection quantity. As shown, when employing a gaseous fuel air mixture that is too lean, the calculated heat release rate is low and extends towards the end of expansion process. However, with increasing the concentration of the gaseous fuel in the cylinder charge, the reaction time is correspondingly shortened and the total heat release rate is increased.

Figure 7 illustrates the variation of the calculated

charge pressure with a crank position for different values of equivalence ratios for a dual fuel engine fuelled with natural gas while employing a fixed pilot diesel injection quantity. It can be seen that, by increasing the gaseous fuel quantity, the cylinder pressure and its rates, increase markedly. This is because increasing the gaseous fuel quantity in the mixture enhances reaction rates within the gaseous fuel and results in a larger size of combustion zones around the ignition sources (i.e. liquid droplets).

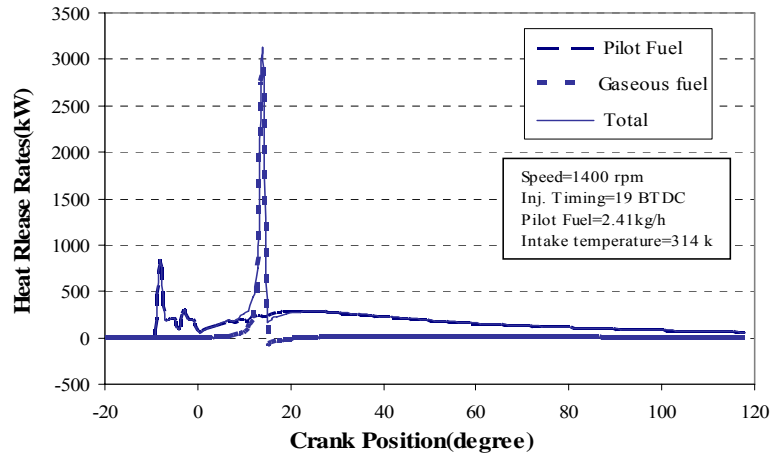


Figure 4. Variations of the calculated heat release rates of both pilot and gaseous fuels with crank position (full load, Energy ratio: 10% pilot diesel, 90% natural gas).

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

Specie (Values)	CH ₄ (ppm)	CO ₂ (%)	CO (%)	NO _x (ppm)	O ₂ (%)
Experimental	626	4.25	0.311	157	13
Theoretical	0	4.16	0.0021	173	12.25

Figure 8 describes the variation of the calculated charge temperature with a crank position for different values of equivalence ratios for a dual fuel engine fuelled with natural gas while employing a fixed pilot diesel injection quantity. The charge temperatures are increased with the increase of gaseous fuel quantity in the cylinder charge and this may result in the variations of the charge temperature with equivalence ratios consistent with trends observed in the energy release process of the mixture.

It is already well known that, a dual fuel engine at part loads inevitably suffers from lower thermal efficiency, higher carbon monoxide, and unburned

fuel. Therefore this paper attempts to investigate the combustion phenomena at part loads and using methods such as advancing injection timing, increasing pilot fuel quantity, and intake air throttling solve the mentioned problems. The low load which is selected for this study is a load with an equivalence ratio equal to 0.3 and engine speed is the maximum torque speed equal to 1400 rpm.

4.2. Effects of Injection Timing In the dual fuel engines at low loads, the quantities of pilot fuel injected natural gas are reduced. In such cases, ignitability and flame propagation are expected to become poor, thereby resulting in deterioration in

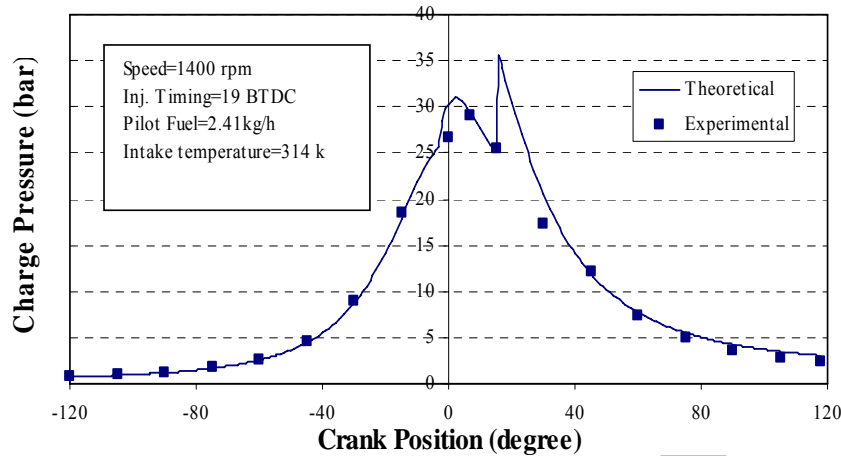


Figure 5. Comparisons of calculated and experimental [23] charge pressure of dual fuel engine. (Part load, Energy ratio: 22.8% pilot diesel, 77.2% natural gas).

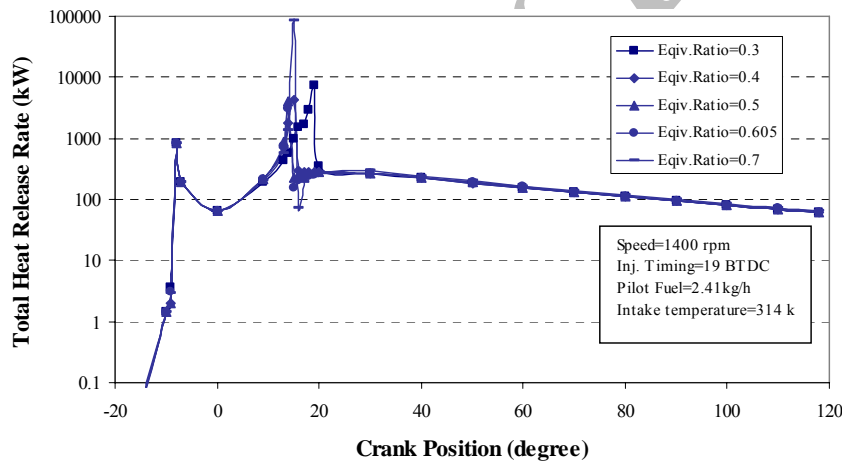


Figure 6. Variations of the calculated total heat release rate with a crank position for different values of equivalence ratio.

the combustion process and also thermal efficiency. So, for this purpose advancing fuel injection timing can be selected to improve the combustion process. Figure 9 shows the effect of diesel fuel injection timing under low load conditions on the rate of heat release. As indicated in this figure, retarding the injection timing deteriorates the combustion process (due to a shifting burning process towards expansion stroke), thus the calculated heat release rate is low and extends towards the end of the expansion process. But with more advancing of the injection timing, better overall combustion process

is achieved which may be due to the increased residence time and activity of the partial oxidation reactions which effectively broadened the lower combustion limit boundary of the overall lean mixtures.

Figure 10 describes the effect of diesel fuel injection timing under low load conditions on the charge pressure. It can be seen that with retarding the injection timing, the trend of the cylinder pressure tends towards the motoring conditions. Also with more advancing of the injection timing, the increasing of the cylinder pressure is not

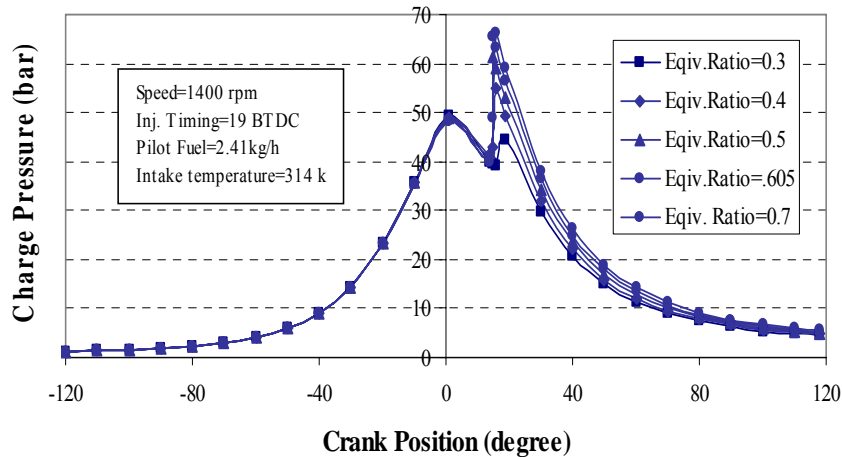


Figure 7. Variations of the calculated charge pressure with a crank position for different values of equivalence ratio.

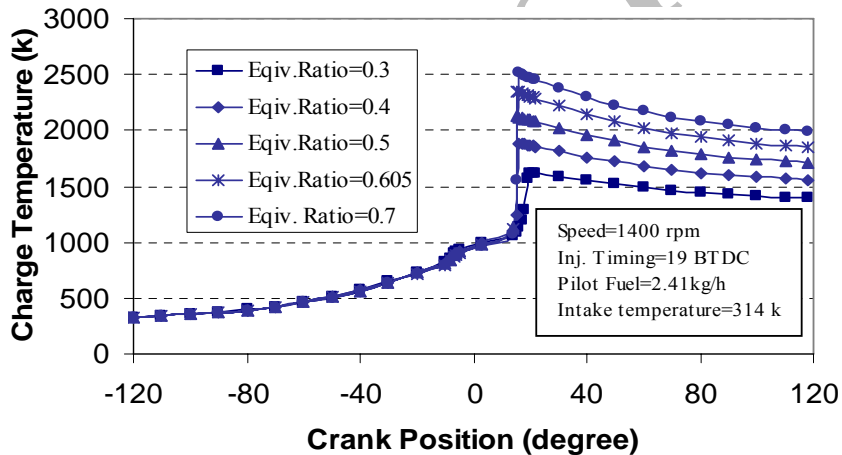


Figure 8. Variations of the calculated charge temperature with crank position for different values of equivalence ratio.

remarkable.

Figure 11 shows the variations of the calculated charge temperature with a crank position for different values of injection timing under low load conditions.

With retarding the injection timing, pilot fuel combustion is delayed, and thus, the temperature of the mixture is not enough to initiate the reactions and propagate the flame in the whole gaseous fuel air mixture and, consequently, incomplete combustion of the gaseous fuel may occur. Also under this condition, the charge temperature in the expansion stroke is high due to

late combustion of the charge. As seen from this figure, more advancing could have allowed longer ignition delay and so larger premixed zones which yield higher combustion temperatures in the combustion process.

4.3. Effects of Pilot Fuel Quantity Figure 12 shows logarithmically the variations of the calculated total heat release rate with a crank position for different amounts of pilot fuel for a dual fuel engine fuelled with natural gas for a fixed total equivalence ratio of 0.3. Increasing the pilot fuel quantity reduces the ignition delay of the pilot

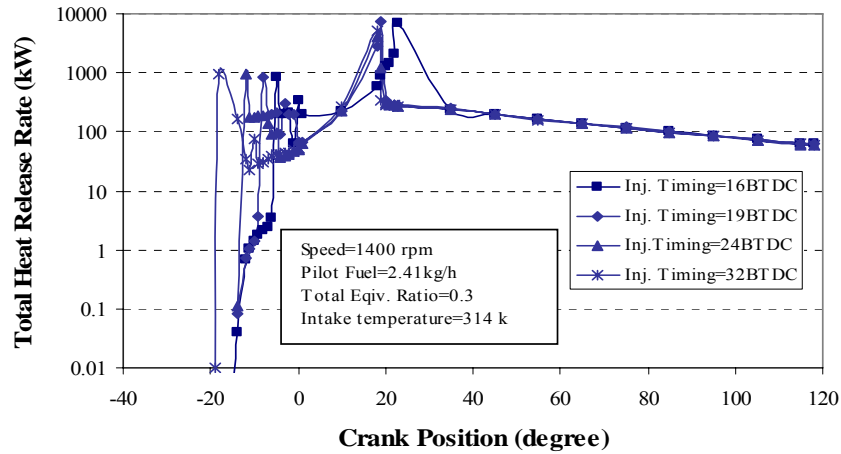


Figure 9. Variations of the calculated heat release rate with crank position for different values of injection timing.

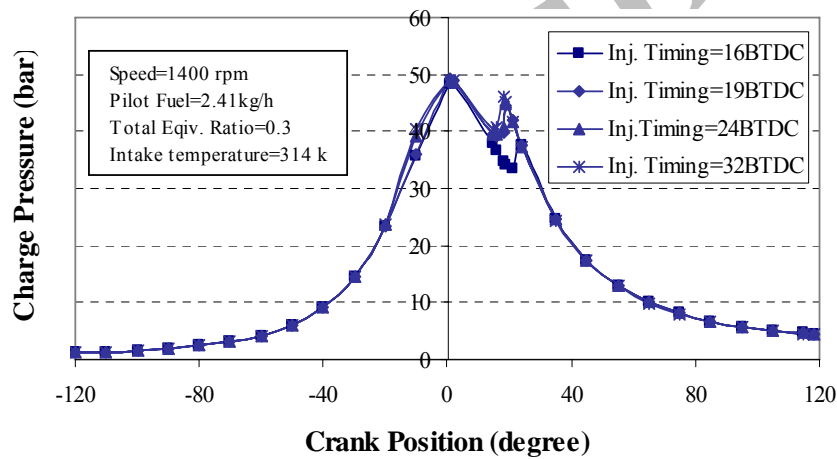


Figure 10. Variations of the calculated charge pressure with crank position for different values of injection timing.

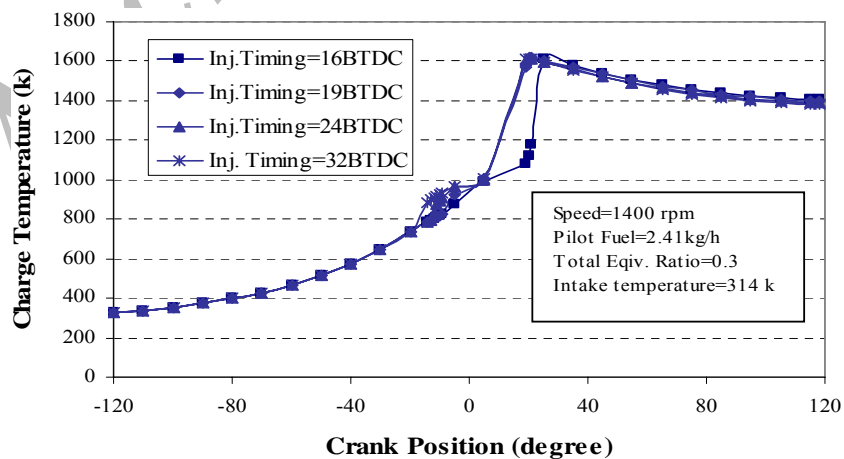


Figure 11. Variations of the calculated charge temperature with crank position for different values of injection timing.

fuel, thus increases the total heat release rate. 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 increasing the burned fraction of the gaseous fuel air mixture. On other hand, by using a small pilot quantity, the flame originating from the pilot ignition sources cannot propagate throughout the whole combustion chamber and thus the amount of heat release rate decreases.

Figure 13 illustrates the variations of the calculated cylinder pressure with a crank position for different amounts of pilot fuel. Here the total equivalence ratio is constant. It can be seen that increasing the pilot fuel quantity increases the charge pressure.

Figure 14 describes the variations of the calculated charge temperature with a crank position for different amounts of pilot fuel for a dual fuel engine fuelled with natural gas for a fixed total equivalence ratio. As indicated in this figure, the charge temperature decreases with the increasing of the pilot fuel quantity. The reason for this trend originates from the tending and approaching of the combustion of the dual fuel engines towards the combustion of pure diesel engines.

4.4. Effects of Intake Air Throttling To achieve better mixture quality of gaseous fuel air charge at part loads, the reduction of intake air can be used. Throttling of the intake air can be implemented by using a gate valve installed in the intake manifold [25]. The amount of air throttling is defined by:

$$\text{Th}(\%) = \left(1 - \frac{\dot{m}_{\text{air}} - \text{Th}}{\dot{m}_{\text{air}} - \text{without} - \text{Th}} \right) \cdot 100 \quad (18)$$

Figure 15 shows the variations of the calculated heat release rate with a crank position for different values of air throttling used for a fixed pilot fuel quantity of a dual fuel engine. It can be seen that with increasing the intake air throttling, heat release rate is increased and the combustion is shifted to close to TDC. Air throttling can promote better combustion due to increasing total equivalence ratio and preparing better fuel air mixture for combustion.

Figure 16 describes the variations of the

calculated charge pressure with a crank position for different values of intake air throttling for a fixed pilot fuel quantity of a dual fuel engine. As indicated in this figure, the charge pressure is increased with increasing air throttling. Thus, it may result in enhancing the performance of the dual fuel engine with air throttling.

Figure 17 indicates the variations of the calculated charge temperature with a crank position for different values of intake air throttling for a fixed pilot fuel quantity of a dual fuel engine. It can be seen that with increasing air throttling, charge temperature is increased. Thus, it can be concluded that air throttling may increase NO_x emission despite better effects of that on performance parameters.

5. CONCLUSIONS

The present quasi - dimensional model, combined with the detailed chemical kinetic scheme, can provide a description 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 known, the dual fuel engines at part loads inevitably suffer from lower thermal efficiency, higher carbon monoxide and unburned fuel. Therefore this paper is an attempt to investigate the combustion phenomena at part loads and using methods such as advancement of injection timing, increasing pilot fuel quantity, intake air throttling to solve the mentioned problems. It is concluded that:

1. Little improvement in combustion parameters is given by advancing diesel injection timing (Figure 10).
2. Increasing pilot fuel quantity can provide better combustion and this method can reduce NO_x emission by approaching the combustion of dual fuel engines towards pure diesel engine combustion.
3. Air throttling can promote better combustion due to increasing total equivalence ratio and preparing better fuel air mixture for combustion. Also air throttling may increase NO_x emission despite its better effects on the dual fuel engine performance parameters.

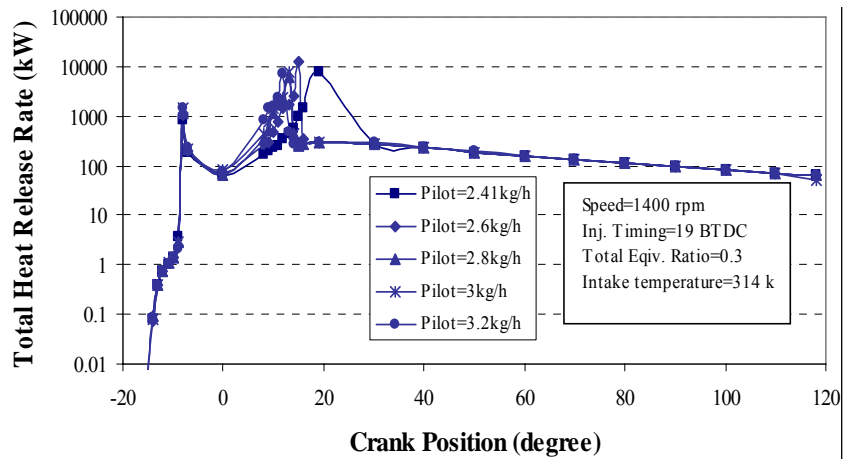


Figure 12. Variations of the calculated heat release rate with crank position for different amounts of pilot fuel.

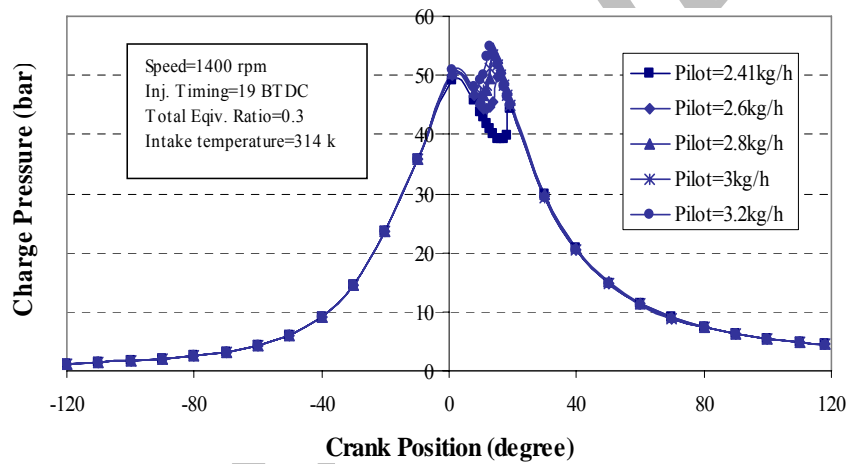


Figure 13. Variations of the calculated charge pressure with crank position for different amounts of pilot fuel.

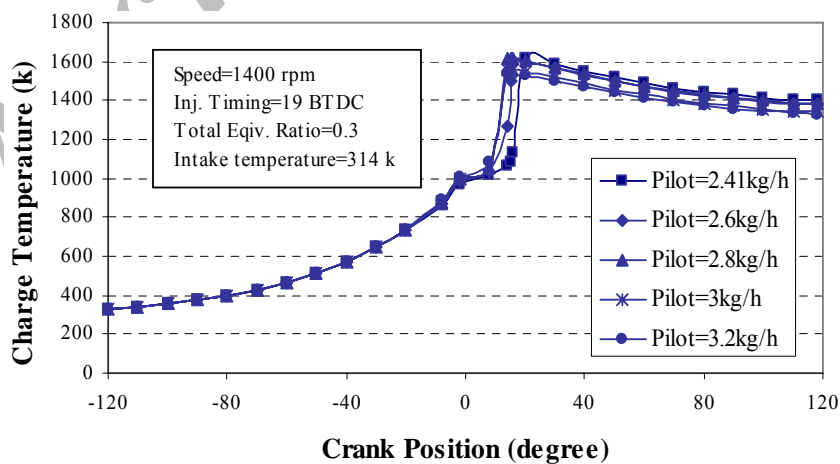


Figure 14. Variations of the calculated charge temperature with crank position for different amounts of pilot fuel.

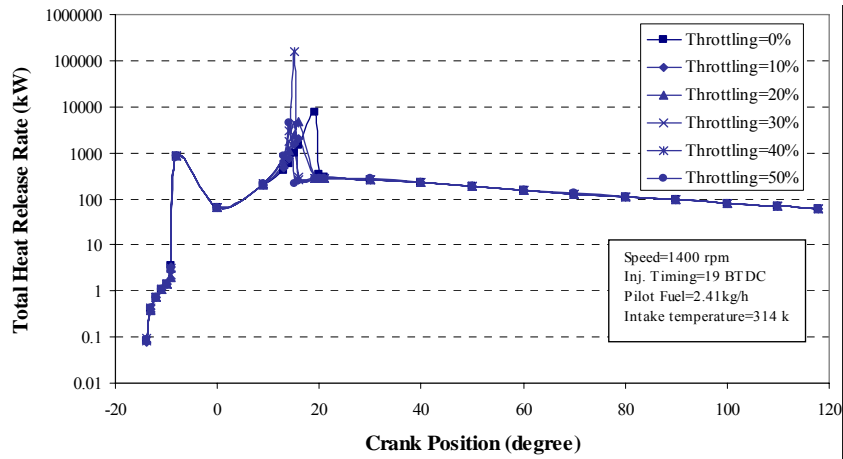


Figure 15. Variations of the calculated heat release rate with crank position for different values of intake air throttling.

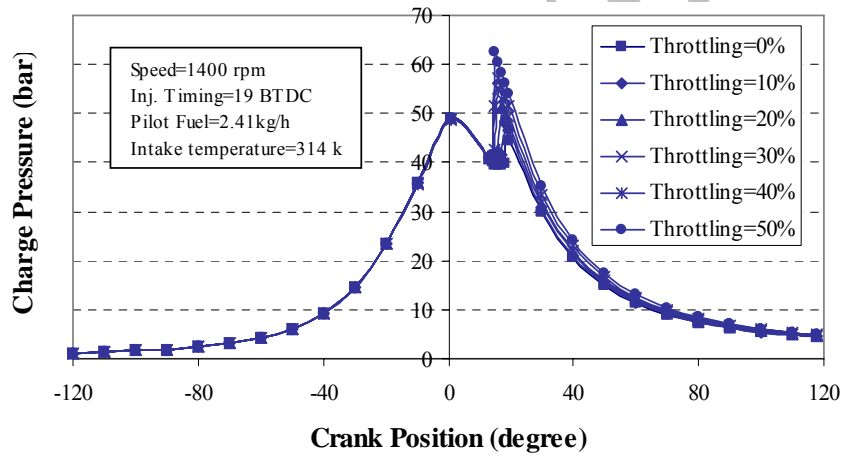


Figure 16. Variations of the calculated charge pressure with crank position for different values of intake air throttling.

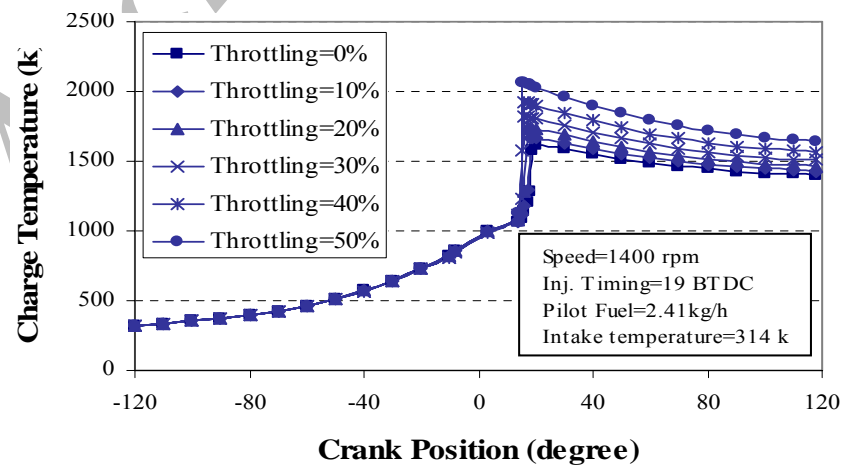


Figure 17. Variations of the calculated charge temperature with crank position for different values of intake air throttling.

6. NOMENCLATURE

M	Mass
Q	Heat loss
T	Temperature
P	Pressure
R	Gas constant
R	Reaction rate
\bar{R}	Universal gas constant
V	Volume
U	Internal energy
H	Enthalpy
A	Pre - exponential factor
B	The temperature exponent
E	Activation energy
C_V	Specific heat at constant volume
α	A stoichiometric coefficient
k	Reaction rate constant
t	Time
ρ	Density
x	Concentration of species
θ	Crank angle
Q_p	Heat release rate of pilot fuel
Q_{Loss}	Heat loss rate of gaseous fuel

Subscripts

i	Relating to species "i"
i	Number of burning zone in pilot fuel
j	Relating to reaction step "j"
j	Preparing zone to pilot fuel
b	Relating to backward reaction
f	Relating to forward reaction
f	Relating pilot fuel
c	Core zone in pilot fuel spray
bj	jth burning zone
e	Entrainment
mp	Mixture prepared
a	Air zone

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