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Yousefi, Amin; Birouk, Madjid; Guo, Hongsheng

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An experimental and numerical study of natural gas/diesel dual-fuel engine at low load: Effect of diesel fuel injection timing

Amin Yousefi†
Department of Mechanical Engineering, University of Manitoba,
Winnipeg, Manitoba, Canada R3T 5V6
Madjid Birouk*
Department of Mechanical Engineering, University of Manitoba,
Winnipeg, Manitoba, Canada R3T 5V6
Hongsheng Guo+
Energy, Mining and Environment Portfolio, National Research Council Canada,
1200 Montreal Road, Ottawa, Ontario, Canada K1A 0R6

Abstract

One of the most promising alternative combustion strategies is natural gas/diesel dual-fuel combustion. It consists of preparing a premixed mixture of a gaseous fuel and air, whose ignition is triggered by the injection of a small amount of more ignitable fuel, usually diesel fuel. However, this combustion mode still suffers from low thermal efficiency and high level of unburned methane and CO emissions at low load conditions. The present paper reports the results of an experimental and numerical study on the effect of diesel injection timings (10 to 50 °BTDC) on the combustion performance and emissions of dual-fuel combustion at 25% engine load. Analysis of OH spatial distribution shows that, at very advanced diesel injection timings, the non-reactive mixture zones are much lower in OH concentration than other injection timings during the last stages of combustion, indicating a more predominant premixed combustion mode. At retarded diesel injection timings, the consumption of premixed fuel in the outer part of the charge is likely to be a significant challenge for dual-fuel combustion engine at low load conditions. However, with advancing the diesel injection timing, the OH radical becomes more uniform throughout the combustion chamber which confirms that high temperature combustion reactions can occur in the central part of the charge. NO_x, unburned methane, and CO emissions are reduced while at the same time the highest indicated thermal efficiency is achieved at very advanced diesel injection timings of 46 and 50 °BTDC.

1. Introduction

Diesel engine has been widely used in transportation and power station industry, due mainly to its higher reliability and superior fuel conversion efficiency. However, due to locally rich air-fuel mixture regions and non-uniform temperature distribution in the combustion chamber, it is very difficult to reduce simultaneously NO_x and soot emissions for diesel engine. One of the most promising alternative combustion strategies is dual-fuel combustion which consists of the preparation of a premixed fuel and intake air mixture, whose ignition is triggered by the injection of a small amount of a more ignitable fuel, usually diesel fuel. A typical dual-fuel combustion combines port fuel injection of a low reactivity fuel to create a well-mixed charge of the premixed fuel and air mixture, and the direct injection of high reactivity fuel (i.e., diesel) as an ignition source. Because of its higher ignition temperature, natural gas is a suitable candidate for low reactivity fuel of dual-fuel combustion. However, some technical issues are still unresolved when CI engine operates under natural gas/diesel dual-fuel mode at low load conditions [1]. Compared to diesel fuel engine, natural gas/diesel dual-fuel engine is known to experience unstable combustion performance, low thermal efficiency, and high levels of unburned methane and CO emissions at low load conditions. This is because that, at low loads and with small quantities of pilot diesel fuel, flame propagation front does not reach portions of the charge situated far away from the pilot ignition nuclei. As a consequence, after an initial fast oxidation of the injected pilot diesel fuel, the rate of combustion slows down leading to incomplete combustion, which in turn results in misfiring or partial burning and hence high level of unburned methane and CO emissions at the engine exhaust [2,3]. For dual-fuel combustion engine, diesel injection timing affects the ignition delay because the in-cylinder charge temperature and pressure change significantly close to the TDC. Advancing diesel injection timing usually increases the ignition delay because the in-cylinder mean

*Professor, corresponding author, majdid.birouk@umanitoba.ca
+*PhD, Senior Research Officer*
†PhD candidate

charge temperature and pressure are lower. However, retarding or advancing diesel injection timing beyond certain limits may result in poor combustion efficiency.

The present study reports a detailed experimental and numerical investigation of the effect of diesel injection timing on the combustion performance and emissions of natural gas/diesel dual-fuel combustion at low load. A single-cylinder diesel engine is modified to operate in natural gas/diesel dual-fuel mode with natural gas as the primary fuel (75% energy fraction) and diesel as the pilot fuel. In both experiment and simulation, the operating conditions are kept constant and only the pilot diesel injection timing is varied (10-50° BTDC with 4° increment) to examine the combustion performance and emissions of dual-fuel engine under 25% engine load (BMEP= 4.05 bar).

2. Experimental setup

The engine used in this investigation is a modified single-cylinder version of Caterpillar's 3400-series heavy-duty engine. More details about the experimental setup and engine configuration can be found elsewhere [4]. Natural gas was injected into the intake port by a fuel injection manifold. Diesel fuel was directly injected into the cylinder using a prototype common-rail fuel injector system. The start of injection and injection pulse width for both diesel and natural gas were controlled via a National Instruments and LabVIEW-based software. The flow rates of diesel, natural gas, and air were measured by a TRICOR mass flowmeter, a Bronkhorst mass flowmeter, and a turbine mass flowmeter, respectively.

3. Numerical model

Numerical simulations were performed using AVL FIRE software coupled with CHEMKIN solver for flow and chemistry calculations. A reactions mechanism, consisting of 42 species and 168 reactions, developed at Chalmers University [5] was used in the calculation. The Kong-Reitz combustion model was used in the simulation. It assumes that the reaction rate of each species is determined by the kinetic process and the relative magnitude of mixing and reaction, which can be characterized by a local Damköhler number defined as the ratio of flow mixing to kinetic time scale [6]. The break-up process of diesel spray was simulated using WAVE model based on the physical properties of diesel fuel [7]. Dukowicz model was used for the heat-up and evaporation of droplets. It assumes that droplets evaporate in a non-condensable gas environment. Therefore, it uses a two-component system in the gas-phase which consists of vapor and non-condensable gas where each component may be composed of a mixture of different species. The computational mesh was created by FIRE ESE-Diesel platform. Since the diesel injector has six equally spaced nozzle's orifices, a sector mesh of 60° was used to model one spray plume to take advantage of the axial symmetry. The computational domain at the TDC is shown in Fig. 1.

4. Results and discussion

Figure 2 shows the effect of pilot diesel injection timing on the in-cylinder pressure and HRR of natural gas/diesel dual-fuel mode under 25% engine load. It can be seen that there is a perfect match between the measured and calculated in-cylinder pressure and HRR profiles, confirming the ability of the CFD model. It can be observed from Fig. 2 that advancing the injection timing up to 30 °BTDC increases the maximum in-cylinder pressure and also shifts the in-cylinder peak pressure close to the TDC. This is mainly attributed to the variations in the in-cylinder temperature during the diesel injection and before SOC and also to the local equivalence ratio inside the ignition pockets. Figure 3 displays the in-cylinder temperature contours at 5 °CA after diesel injection timing for some selected examined cases. Advancing the diesel injection timing up to 30 °BTDC leads to a reduction in the average in-cylinder temperature. This consequently leads to a prolonged ignition delay and as a result, more premixed mixture is formed during the ignition delay period. This can be confirmed by looking at the spatial and temporal contours of total equivalence ratio before the SOC in Fig. 4. This figure shows that advancing diesel fuel injection timing reduces the fuel concentration gradient in the cylinder and avoids the local fuel rich combustion zones. Therefore, combined with appropriate average in-cylinder temperature, a larger number and a wider space distribution of ignition kernel are produced.

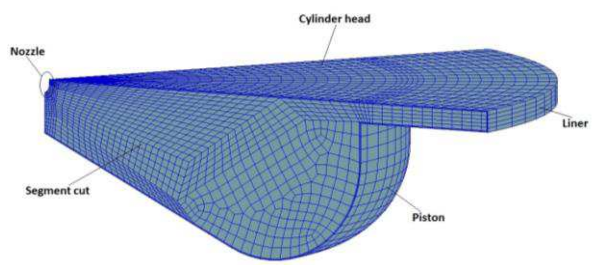


Fig. 1: Computational domain of the combustion chamber at the TDC.

Consequently, the proportion of premixed combustion is increased. With further advancing the injection timing, the crank angle of the peak pressure is retarded and the maximum in-cylinder pressure is decreased. The ignition delay also increases when advancing the diesel injection timing, and the diesel fuel experiences very long atomization and evaporation processes. As a result, the combustion chamber volume becomes more and more uniform without remarkable fuel stratification during the ignition delay. However, the in-cylinder temperature is not high enough which leads to a retarded SOC and a reduced maximum in-cylinder pressure. For very advanced diesel injection timing, the pilot diesel ignition mode is similar to that encountered in HCCI engine, where late combustion phasing happens in the expansion stroke.

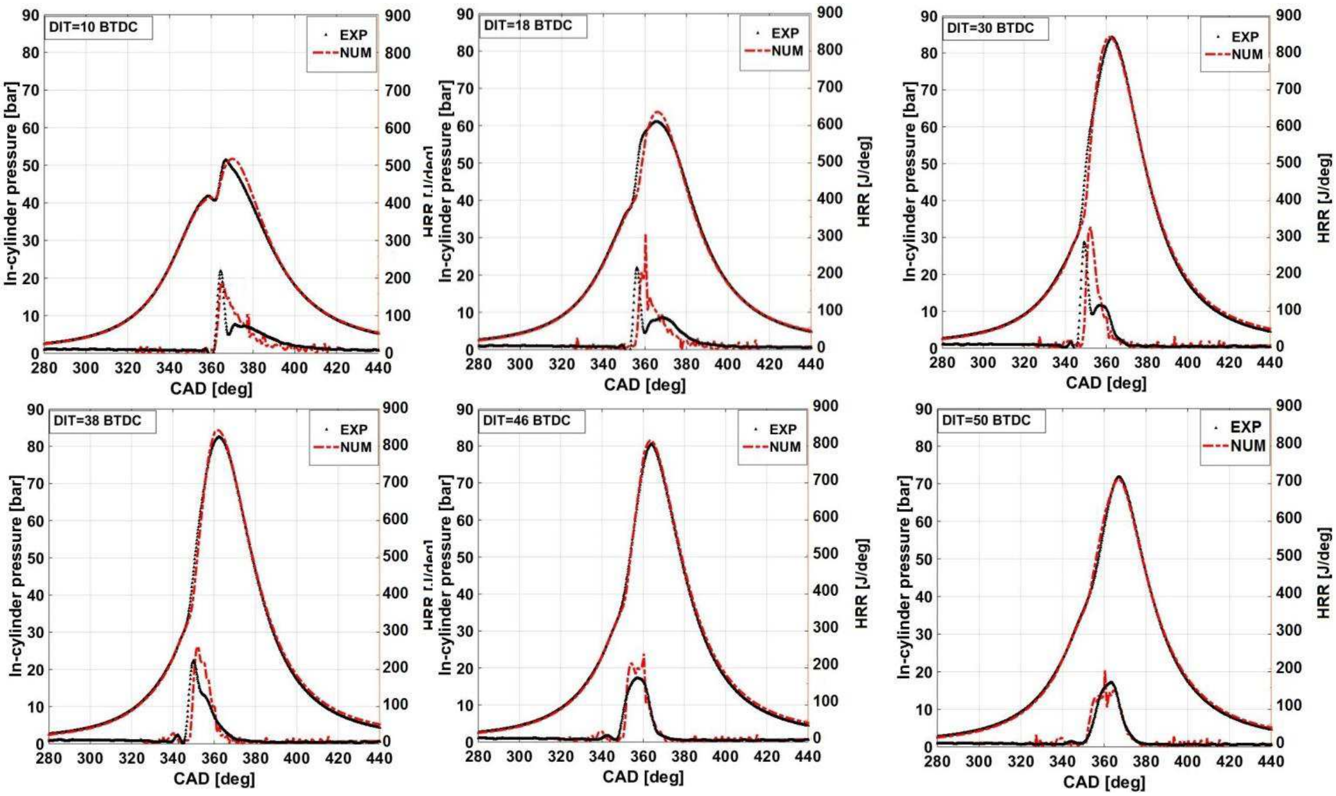


Fig. 2: Experimental and numerical in-cylinder pressure and HRR at different injection timings under 25% engine load.

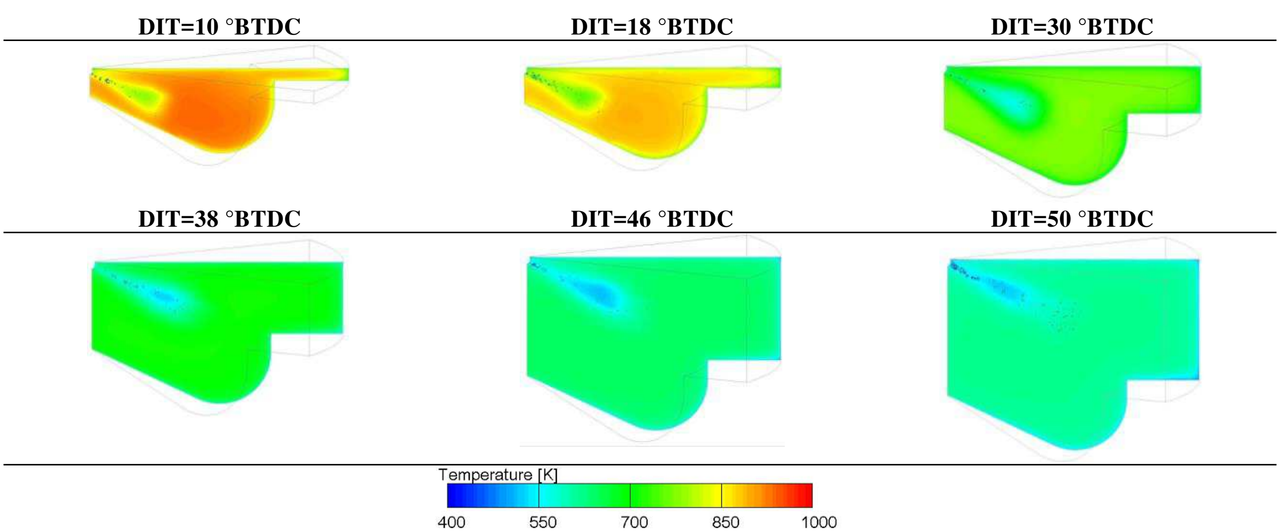


Fig. 3: In-cylinder temperature contours (at 5 °CA ADIT) for various diesel injection timings under 25% engine load.

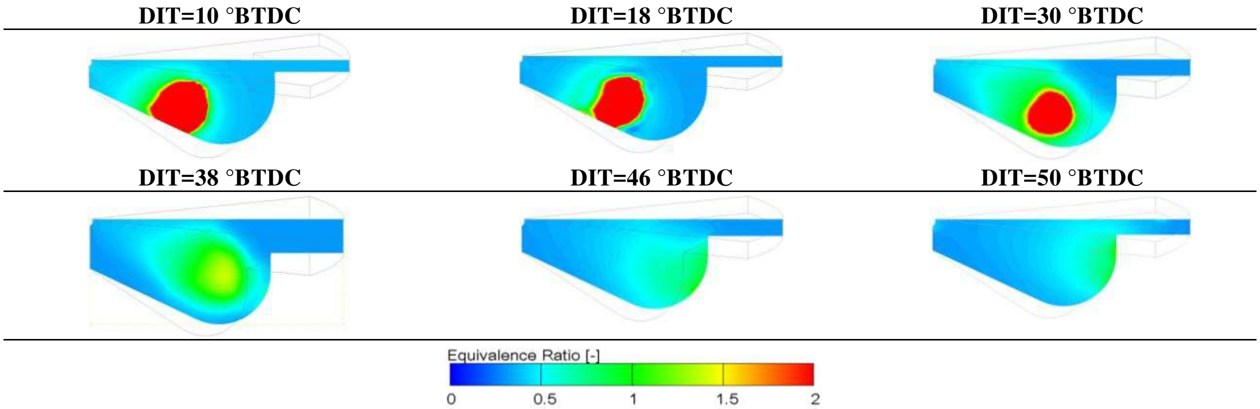


Fig. 4: Spatial contours of total equivalence ratio (at SOC timing) of various diesel injection timings under 25% engine load.

Figure 5 presents the spatial distribution of OH radical mass fraction for different diesel injection timings. It is obvious that OH radical distribution is wider for an injection timing of 30 °BTDC than for other injection timings at the early stage of combustion. However, at a diesel injection timing of 46 °BTDC, the blue non-reactive zones are much narrower than other injection timings during the last stages of combustion, indicating that a more premixed combustion takes place in these cases. Moreover, as the diesel injection timing advances, the highest OH concentration is detected exclusively near the wall region of the piston bowl which corresponds to the fuel rich zones. For example, at a diesel injection timing of 46 °BTDC, the highest OH concentration is located near the piston wall. On the contrary, at a diesel injection timing of 14 °BTDC, the highest OH concentration is closer to the cylinder axis and nozzle tip. These contours reveal that, for late diesel injection timings, the consumption of premixed fuel in the outer part of the charge is likely to be a significant challenge for dual-fuel combustion engine at low engine load. This observation is important, since it could explain why natural gas/diesel dual-fuel combustion mode under low engine load often suffers from high unburned methane emissions. However, with advanced diesel injection timings, the OH radical distribution is more uniform throughout the combustion chamber, which makes the combustion look similar to that in a HCCI engine and, therefore, helps reduce unburned methane emissions.

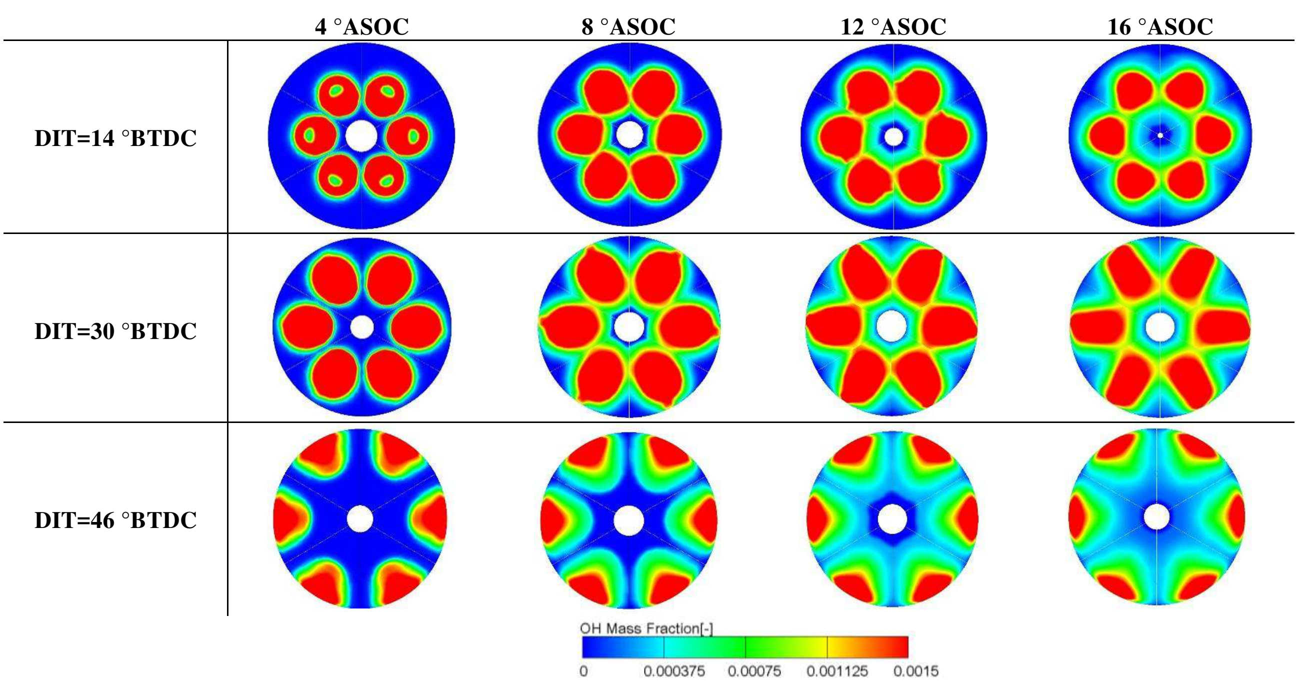


Fig. 5: Spatial contours of OH radical for various diesel injection timings under 25% engine load.

Figure 6 shows the variation of the ITE as a function of diesel fuel injection timing under 25% engine load. In order to support in more detail the results regarding the effect of diesel injection timing on ITE of the dual-fuel engine at low load, it is needed to investigate the heat release corresponding to each examined case (Fig. 2). For diesel injection timing between 10 and 30 °BTDC, the combustion rate of natural gas is very slow, and the fuel utilization is significantly poor. This is because the combustion of natural gas-air mixture is significantly delayed compared to that of diesel-air mixture. The HRR profile exhibits a single obvious peak which is relatively high as the second peak is almost indistinguishable, and thus the HRR profile is not symmetric. For these cases, the combustion rate of the main combustion stage is too slow, and the combustion efficiency of natural gas is extremely low (second peak of HRR). However, advancing diesel fuel injection timing from 10 to 30 °BTDC increases the second peak of HRR and its crank angle becomes closer to the first peak (diesel premixed combustion). This is due to the fact that advancing diesel fuel injection timing prolongs the ignition delay and thus more premixed natural gas-air and diesel mixture is formed before the SOC. In addition, the prolonged ignition allows the formation of larger number and wider space distribution of ignition kernel. As a result, the combustion efficiency of natural gas and the ITE are improved, as shown in Fig. 6. Dual-fuel combustion with a diesel injection timing between 34 and 42 °BTDC has a faster combustion rate of natural gas-air mixture compared to previous cases and diesel fuel experiences very long atomization and evaporation processes. The SOC timing is retarded and the combustion phasing continues into the expansion stroke. The multipoint premixed combustion dominated by natural gas occurs quickly after the premixed combustion of pilot diesel fuel. For these cases, the premixed combustion of natural gas significantly improves and more heat is released during the premixed combustion stage. This late released heat positively affects the expansion pressure which leads to an increase in ITE, as shown in Fig. 6.

In dual-fuel combustion mode with very advanced diesel injection timing, low temperature chemical kinetics reactions are often observed before the main combustion stage. The premixed ignition of diesel-air-natural gas mixture provides a significant wide ignition source for natural gas, resulting in a very fast combustion rate of natural gas-air mixture. Before the start of the main combustion, a lower peak, which is caused by low temperature chemical kinetics reactions, appears in the HRR profile. There is only one significant peak of the HRR profile during combustion stage which has similar characteristics to that of HCCI combustion mode. For both diesel injection timings of 46 and 50 °BTDC, the HRR profile is almost symmetric and SOC timing is very retarded. Combustion phasing mostly happens during the expansion stroke which increases the expansion pressure and positive engine work. Therefore, the thermal efficiency is the highest for these injection timings (Fig 6).

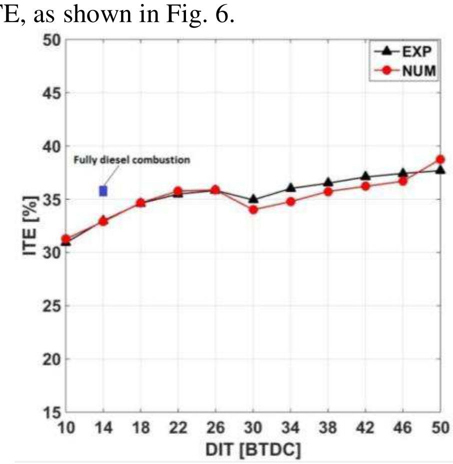
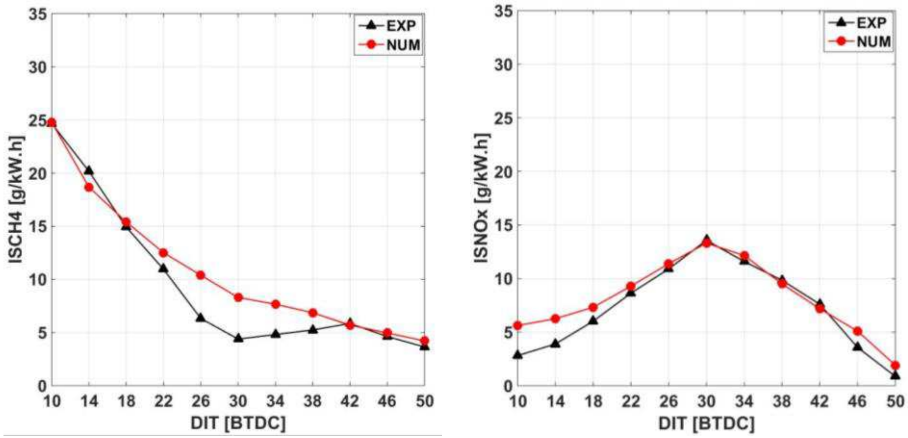


Fig. 6: ITE vs. diesel injection timing for natural gas/diesel dual-fuel mode at 25% engine load.

Figures 7a shows the experimental and numerical trends of the unburned methane emissions versus different diesel injection timings. The experimental results show that methane emissions significantly decrease with advancing diesel injection timing. This trend is well reproduced by the CFD model. However, the model quantitatively over-predicts the unburned methane emissions at diesel fuel injection timing between 26 and 34 °BTDC and also misses the slight increase trend at diesel fuel injection timing between 30 and 38 °BTDC. Both calculated and measured results show that methane emissions are reduced by 6 times when diesel injection timing is advanced from 10 to 50 °BTDC. This is due to the relatively higher combustion rate and greater utilization of premixed natural gas at earlier injection timings.

Figure 7b illustrates the NO_x emissions at different diesel injection timings. It is observed that the experimental $ISNO_x$ trend is well captured numerically. However, the model quantitatively over-predicts NO_x emissions at diesel injection timings of 10 and 14 °BTDC. NO_x emissions consistently increase with advancing diesel injection timing up to 30 °BTDC. This is due to the fact that the local gas temperature becomes higher and more homogeneous as diesel fuel injection timing advances. With further advancing diesel injection timing beyond 30 °BTDC, more combustion reactions take place throughout in the combustion chamber, and the temperature field is more uniform. Consequently, further advancing diesel fuel injection timing up to 50 °BTDC significantly reduces NO_x emissions.



a) ISCH4 b) ISNOx

Fig. 7: CH_4 and NO_x emissions with different injection timings under 25% engine load.

5. Conclusions

The effect of diesel fuel injection timing in the range between 10 and 50 °BTDC on the combustion performance and emissions of natural gas/diesel dual-fuel compression ignition engine at low load has been studied experimentally and numerically. The major findings can be summarized as follows. Analysis of OH spatial distribution for late diesel fuel injection timings showed that the consumption of premixed fuel in the outer part of the charge is likely to be a significant challenge for dual-fuel combustion engine at low engine load. However, with advancing the diesel injection timing, OH radical is more uniform throughout the combustion chamber which confirms that high temperature combustion reactions can occur in the central part of the charge. Very advanced diesel fuel injection timing of 46 and 50 °BTDC simultaneously reduce NO_x, unburned methane, and CO emissions. Moreover, very advanced injection timings can lead to the highest ITE. The results revealed that with advancing diesel injection timing from 10 °BTDC to 50 °BTDC, NO_x, unburned methane, and CO emissions are reduced, respectively, by 65.8%, 83%, and 60% while ITE is increased by 7.5%.

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