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Research Paper

Effects of combustion duration characteristic on the brake thermal efficiency and NOx emission of a turbocharged diesel engine fueled with diesel-LNG dual-fuel

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HIGHLIGHTS

- The study was conducted on a diesel-LNG dual-fuel engine at low and medium loads.
- Diesel injection timing greatly affects the centroid angle of combustion duration (α).
- α is correlated with the brake thermal efficiency and NOx emission.
- Keeping α in 1–2 °CA ATDC may obtain optimal thermal efficiency and NOx emission.

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ABSTRACT

The diesel-LNG (liquefied natural gas) dual-fuel combustion mode was conducted on a high-pressure common-rail six-cylinder diesel engine to find an assistant parameter to assess the brake thermal efficiency (η_e) and nitrogen oxides (NOx) emission of the diesel-LNG dual-fuel engine. The results show that the diesel injection timing has a prominent impact on the centroid angle of combustion duration (α) which is closely related to η_e and NOx emission. At low and medium loads, when α is near to top dead center (TDC) and is after TDC, the η_e and NOx emission are higher. Nevertheless, when α is before TDC, the result of NOx emission is opposite. Therefore, for optimal η_e and NOx emission at low and medium loads, it would be the best way altering diesel injection timing to retard α to ATDC and ensuring α in 1–2 °CA ATDC.

1. Introduction

Diesel engines, a sort of conventional internal combustion engines, are facing with two kinds of challenges: the crisis of petroleum and the increasingly stringent emission regulations. Consequently, the attention of researchers is dramatically drawn to the technology of dual-fuel (DF) engines. Compared to conventional diesel engines, DF engines have more economical and environmental benefits [1–4].

In recent years, alcohol is the most common alternative fuel used in vehicle engine, and many researchers used it to substitute diesel in DF engines [5–7]. Prashant et al. [6] investigated the effect of ethanol addition on the performance of DF engines. The results showed that with increased ethanol content, the peak pressure rising rate and peak in-cylinder pressure increased. Additionally, Gómez et al. [7] found that ethanol addition apparently decreased the soot emission. For DF engines, natural gas (NG) is also a perfect alternative fuel and presents promising potential owing to its clean property of combustion and rich reserves. So, a large number of numerical and experimental studies respecting the utilization of natural gas on DF engines have been conducted [8–15]. Zhang et al. [16] studied the effect of diesel injection pressure on combustion process in a heavy-duty DF engine. The results showed that approximately increasing diesel injection pressure ameliorated the combustion process and improved η_e. Li et al. [17] found that when the diesel injection pressure was increased, the maximum in-cylinder pressure increased and the combustion noise deteriorated. Nithyanandan et al. [18] investigated the effect of

Abbreviations: DF, dual-fuel; CA, crank angle; *CCR*, co-combustion ratio (%); *HRR*, heat release rate (kJ/°CA); α, centroid angle of combustion duration (°CA); η_e, brake thermal efficiency (%).

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energy-based substitution rate on soot characteristic in a diesel-NG dual-fuel engine. Their results showed that as the energy-based substitution rate increased, the soot emission declined and the proportion of large particle size increased. Papagiannakis et al. [19] found that a low NOx emission of DF, compared with conventional diesel engine, could be achieved with the increasing diesel fuel supplementary ratio. Zhang and Song [20] also found that when the co-combustion ratio (*CCR*) was increased, the brake specific fuel consumption, HC emission and CO emission increased, while the smoke emission decreased, and was maintained a low level under low and medium loads.

Except for altering diesel injection pressure and *CCR*, Cameretti et al. [21] investigated the effects of diesel injection timing on performance of a DF engine, the results showed that at medium engine loads, the start of combustion advanced with increased diesel injection timing. Yang et al. [22] found that at low loads, as the diesel injection timing is advanced, the quantity and mass of particle dramatically declined with a louder combustion noise. Xu et al. [23] studied the effects of diesel pre-injection timing on combustion and emissions of a DF engine. Their study found that closely advanced pre-injection timing resulted to higher in-cylinder pressure, the improvement of combustion and increase in NOx emission, while the early pre-injection timing resulted to the opposed variation trend. Compared to *CCR*, altering diesel injection timing is more effective to control combustion models and get higher η_e [24]. In addition, there were other researchers investigating the effect of NG port injection timing and found that at low and medium loads, approximately retarding NG port injection timing improved η_e, while the NOx emission increased [25,26]. Especially at medium loads, the NOx emission was rather high. In order to resolve the issue, the exhaust gas recirculation (EGR) was applied in DF engines. Abdelaal and Hegab [27] found that the application of EGR significantly reduced the NOx emission with increased η_e at medium loads.

Although, there were many studies made on the method of improving η_e and reducing NOx emission, the centroid angle of combustion duration (α) was not proposed as an intuitionistic parameter reflecting combustion statues and used to evaluate η_e and NOx emission at low and medium loads for diesel-NG dual-fuel engine. The purpose of this study is to reveal the relationships between α and η_e as well as NOx emission, and to validate that the centroid angle of combustion duration is a proper parameter to reflect combustion statues. In this study, the data of combustion and emissions were collected and analyzed.

2. Experimental apparatus and methodology

2.1. Test engine system

The test engine adopted in this study was a high-pressure common-rail, six-cylinder diesel engine added with an injection system of liquefied natural gas (LNG). The specifics of the engine are listed in Table 1.

Table 1
Engine specifications.

Items	Parameters
Bore × Stroke	112 mm × 132 mm
Displacement	7.8 L
Compression ratio	17.5:1
Intake valve opening	13.5 °CA before top dead center
Intake valve closing	38.5 °CA after bottom dead center
Exhaust valve opening	56.5 °CA before bottom dead center
Exhaust valve closing	11.5 °CA after top dead center
Diesel injection system	High-pressure common-rail
Rated torque	1080 N·m @ 1400–1600 r/min
Rated power	199 kW @ 2200 r/min

The schematic of test engine system is shown in Fig. 1. An added LNG supply system, including LNG tank, magnetic valve, vaporizer, NG filter, regulator, NG rail and mixer, was used in the test system. The magnetic valve and NG rail were controlled by DF electronic control unit (ECU), which controlled NG injection timing and pulse width. Under DF mode, the DF ECU and the original ECU were simultaneously employed, and the signals of coolant temperature, crankshaft position, camshaft position and the pressure of diesel rail were shared by them. The injection of diesel was controlled by a high-pressure common-rail system controlled by the original ECU. Before LNG was induced into NG rail, it got through a vaporizer warmed by engine coolant, and the pressure of NG was maintained at 0.85 MPa by the regulator. Finally, NG was induced into the mixer mounted about 50 cm upstream from the intake manifold to generate homogeneous air-NG mixture.

Besides, an eddy current dynamometer (CW260, CAMA), produced by Nan Feng Company in China, was applied to control the speed and torque of the test engine for anticipant operating conditions. A crank angle sensor was also used to monitor the position of crank angle (CA). To measure the in-cylinder pressure, a piezoelectric pressure transducer (6052A, Kistler) was mounted on the first cylinder-head. However, for the combustion analyzer (Kibox, Kistler) to analyze the charge signal, the signal from the pressure transducer was so tiny that it was necessary to apply a charge amplifier (5019B, Kistler).

2.2. Test methodology and conditions

In DF engines, homogeneous mixture of NG and air is generated by injecting NG during the intake stroke, and is ignited by directly injected diesel. Thus, the combustion process of DF could be seen as the interplay of diffusive combustion of diesel and flame propagation combustion of NG-air premixed charge. So, when the *CCR* is low, the diffusive combustion of diesel plays a dominated role in the combustion process, resulting in two-peak combustion phenomenon. Nevertheless, a single-peak phenomenon is observed under relatively high *CCR*. Additionally, knock is the main issue of the diesel-LNG dual-fuel engines. So, the experiments were conducted at two different engine loads corresponding to 30% and 60% of full load with the engine speed (n) of 1400 r/min. Diesel injection pressure (P_{in}) was the optimum under its operation condition. All engine test conditions are listed in Table 2.

Besides, *CCR* was kept constant by fixing the injection pulse and injection pressure of two fuel. And, the *CCR* was as high as possible at each condition due to the lower price of NG compared to commercial diesel. Under high *CCR*, the combustion of NG premixed charge ignited by small amounts of diesel is pre-mixture combustion, so that investigating α is meaningful. Furthermore, the adjustments of diesel injection timing depended on the operation of test engine for fear of running roughly and knocking.

Prior to carrying out the experiment, the test engine was warmed until the temperature of coolant and lubricant reached 85 °C. In addition, the calculation of heat release rate (*HRR*) was on the basis of averaged value of 100 consecutive cycle in-cylinder pressures. As to emission, an exhaust gas analyzer (DiGas 4000, AVL) was applied to measure the emission, such as NOx, THC and CO. Furthermore, the consumption of LNG and diesel were measured by volume flow meter and mass flow meter, respectively.

Additionally, the uncertainties of measured parameters affect the experimental reliability. In this study, all apparatuses were calibrated before the experiment and these measured data were recorded more than twice. The maximum CoV (coefficient of variation) of each measured parameter was calculated and was used to