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Full Length Article

Experimental and numerical investigations on the cyclic variability of an ethanol/diesel dual-fuel engine

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HIGHLIGHTS

- The cyclic variations of ethanol/diesel dual fuel mode were investigated.
- The cyclic variation of IMEP increased gradually with the increase of ethanol ratio.
- Simulations showed that UHC emission was sensitive to initial in-cylinder temperature.
- The cyclic variations increased with retarded diesel injection timing.

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ABSTRACT

Experiments and simulations were used to explore the cyclic variability of an ethanol/diesel dual-fuel engine. The experiments were conducted on a light-duty diesel engine fueled with port injection of ethanol and direct injection of diesel. The influences of engine load, ethanol proportion and diesel injection timing on the cyclic variability were investigated. In-cylinder pressure traces of 150 consecutive cycles were acquired at each test condition. Consequently, cyclic variations of ignition timing (CA5), combustion phasing (CA50), accumulated heat release (Q_f) and indicated mean effective pressure (IMEP) were used to quantify the cyclic variability. The experimental results showed that the fluctuations of CA5 and CA50 were very small throughout the premixed ratio sweep for both the light and high loads. However, the cyclic variation of IMEP was increased with the increasing ethanol ratio at both load conditions. And the cyclic variation of IMEP was reduced at the high load. The variations of IMEP and Q_f showed similar magnitudes and trends at the test conditions, indicating that the cyclic variation of IMEP was mainly caused by the variation of Q_f. Simulations confirmed that the combustion of premixed ethanol showed larger variations to initial in-cylinder temperature fluctuations at the light load, and consequently resulted in higher cyclic variation of Q_f. At the high load, the increased combustion temperature enhanced the oxidation of ethanol, which contributed to stabilizing the combustion and reducing the cyclic variations. The cyclic variations increased with retarded diesel injection timing, which was mainly due to the reduced combustion temperature.

1. Introduction

Diesel engines have the advantage of high fuel efficiency while accompanied by high soot and NO_x emissions. Over the past few decades, the research of diesel engines has faced severe challenges. In order to decrease the emissions of diesel engines to meet the stringent emission regulations in the future, researchers have proposed many advanced combustion strategies. Most of the advanced combustion strategies can be considered as the Premixed Compression Ignition Low Temperature Combustion (PCI-LTC), such as Homogeneous Charge Compression Ignition (HCCI), and Premixed Charge Compression Ignition (PCCI). Many studies have confirmed that these strategies could simultaneously decrease soot and NO_x emissions while maintaining the high thermal efficiency of diesel engines [1–4]. However, the single-fuel PCI strategy has problems in controlling the combustion phasing and extending to the high load engine conditions. Recent studies proved that the dual-fuel PCI mode could solve these problems, such as Reactivity Controlled Compression Ignition (RCCI) [5], Homogeneous Charge Induced Ignition (HCII) [6], and dual-fuel Highly Premixed Charge Combustion (HPCC) [7].

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Ethanol is usually used as port injected fuel in dual-fuel PCI studies because of its low reactivity. Moreover, ethanol molecules have higher oxygen content and can significantly increase the ignition delay of diesel fuel, which result in reduced soot emission. Studies showed that with ethanol premixed, ultra-low soot and NO_x emissions can be realized with nearly 60% of indicated thermal efficiency in dual-fuel PCI combustion [8–12]. However, the unburned hydrocarbon (UHC) emissions of the ethanol/diesel mode in light load condition were increased due to the enhanced quenching effect near the wall and crevice regions [11,12].

Dual-fuel engines are often observed with high cyclic variations. There are many factors that can result in cyclic variations in the engine operations, such as fluctuations in the in-cylinder gas composition, temperature, and pressure. For the PCI-LTC strategies, the in-cylinder mixture is fuel lean and the combustion temperature is lower than the traditional combustion strategies. The PCI-LTC strategies have the problem of high cyclic variations which has been investigated by many researchers. Shahbakhti and Koch [13] investigated the cyclic variations of HCCI combustion based on a diesel engine. The results confirmed that the cycle to cycle variability of the HCCI engine was strongly affected by the ignition timing. With ignition timing close to top dead center (TDC), HCCI combustion exhibited lower cyclic variations than ignition occurring late after TDC. Saxena and Bedoya [14] proved that the cyclic variations of HCCI combustion were very sensitive to small fluctuations of compressed gas temperature. You-cheng et al. [15] investigated the influences of exhaust gas recirculation (EGR) rate on the cyclic variations of HCCI combustion based on a diesel engine. The results showed that high EGR rate resulted in unstable combustion. The dual-fuel PCI strategy also has the features of fuel-lean mixture and low combustion temperature. Like HCCI engines, the dual-fuel PCI engines have the high cyclic variation issue as well. As the cyclic variations of combustion affect engine emissions, efficiency and noise significantly, the cyclic variability of dual-fuel engine has been studied by many researchers. Jia et al. [16] investigated the cyclic variability of methanol/diesel RCCI combustion on a diesel engine using multi-dimensional simulations. The results showed that in the simulations, the variations of the 50% burn point of RCCI combustion could be well predicted with fluctuations of initial in-cylinder temperature. Wang et al. [17] investigated the cyclic variability of a dual-fuel engine fueled with port injected dimethyl ether (DME) and direct injected diesel. The results showed that the coefficient of variation (COV) of IMEP increased with DME quantity. Wang et al. [18] explored the cyclic variability of a methanol/diesel dual-fuel engine. It was found that the cyclic variations of the methanol/diesel dual-fuel combustion were increased with increasing methanol proportion, especially at the light load conditions.

Ethanol is often used as port fuel in dual-fuel PCI combustion research. However, few studies on the cyclic variations of ethanol/diesel dual-fuel mode are conducted so far. In the present study, experiments and simulations were used to study the cycle to cycle variations of the ethanol/diesel dual-fuel engine. The cyclic variations of CA5, CA50, Q_f and IMEP were calculated for quantifying the engine cyclic variability.

2. Experimental setup

A single-cylinder, naturally aspirated, light-duty diesel engine was used to perform the experiments. Fig. 1 shows the schematic of the engine layout. The specifications of the diesel engine are given in Table 1. In the present study, the ethanol was delivered with a port fuel injection system and diesel was directly injected into the cylinder. Table 2 shows the properties of ethanol and diesel used in the present study. The pressure of port fuel injection was maintained at 5 bar and the injection timing was −297°CA after top dead center (ATDC). The included spray angle of diesel injector was 154°. As diesel was delivered with a mechanical fuel injection system, the start injection pressure of the diesel injector was 20 MPa and the peak injection pressure was 35 MPa. And three direct injection timings were investigated, including −17°CA, −21°CA and −25°CA ATDC. Table 3 gives the specifications of the port and direct injection systems.

The in-cylinder pressure was acquired with a Kistler cylinder pressure transducer. The TDC was measured by an Optical Encoder. For each test condition, in-cylinder pressure traces of 150 consec-

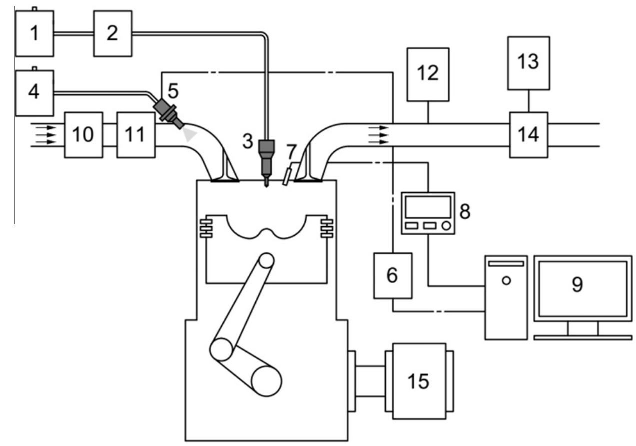

Fig. 1. Engine setup. 1. Diesel tank; 2. Diesel consumption meter; 3. Diesel injector; 4. Port fuel tank; 5. Port fuel injector; 6. Port fuel ECU; 7. Pressure transducer; 8. Charge amplifier; 9. Computer; 10. Intake air flow meter; 11. Intake surge tank; 12. Smoke meter; 13. Exhaust analyzer; 14. Exhaust surge tank; 15. Dynamometer.

Table 1
Engine specifications.

Bore × stroke (mm)	105 × 115
Displacement (L)	0.996
Geometry compression ratio	18.0:1
Piston bowl shape	ω
Intake valve open timing[a] (°CA ATDC)	−391
Intake valve close timing[a] (°CA ATDC)	−108
Exhaust valve open timing[a] (°CA ATDC)	111
Exhaust valve close timing[a] (°CA ATDC)	343

[a] CA: crank angle, 0°CA is the TDC of the compression stroke.

Table 2
Fuel properties of ethanol and diesel [10,19].

	Diesel	Ethanol
Molecular formula	C12–C25	C_2H_5OH
Research octane number (RON)	25	107
Oxygen content (wt.%)	0	34.8
Density (g/ml)	0.84–0.88	0.785
Lower heating value (MJ/kg)	42.5	26.8
Stoichiometric ratio	14.5	9
Enthalpy of vaporization (kJ/kg)	0.27	0.84

Table 3
Specifications of port and direct injection systems.

Port-injector	Steady flow rate @ 5 bar	4.24 ml/s
	Injection pressure	5 bar
Direct-injector	Number of holes	5
	Included spray angle	154°
	Nozzle diameter	0.28 mm
	Injection pressure	20 MPa