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Combustion performance and emissions of 2-methylfuran diesel blends in a diesel engine

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HIGHLIGHTS

- This is the first comprehensive study of diesel–MF blends in CI engine.
- Ignition delay, combustion duration and emissions were studied.
- Soot emission from diesel–MF blends are much less than pure diesel.
- High MF fraction in fuel blends is not suitable for CI engine.

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ABSTRACT

As an important renewable engine fuel, 2-methylfuran (MF) has drawn intense attention due to the breakthroughs in its production methods. In the present study, combustion characteristics and emissions of a four cylinder direct-injection compression-ignition (DICI) engine fueled with diesel–MF blends (DM) and pure diesel are investigated. The tests were performed at constant speed of 1800 rpm and varying loads from 0.13 to 1.13 MPa brake mean effective pressure (BMEP). Results show that diesel–MF blends have different combustion performance from pure diesel. The combustion phase of fuel blends is retarded with the increase of MF fraction. Diesel–MF blends are characterized with longer ignition delay and shorter combustion duration. Meanwhile, diesel–MF blends show higher brake thermal efficiency (BTE) than pure diesel. However, diesel–MF blends lead to higher NO_x emissions than pure diesel and the NO_x emissions are increased with the increase of MF fraction. The soot emissions from diesel–MF blends are significantly reduced compared to pure diesel. The CO and HC emissions from tested fuels are nearly the same at medium and high engine loads. Pure diesel produces lower CO emissions and higher HC emissions than that of diesel–MF blends at low loads of 0.13–0.38 MPa BMEP.

1. Introduction

With the petroleum storage reduction and global warming, innovations of the energy supply chain are increasingly needed, especially in the transportation industry. Researches [1–4] on the production of fuels derived from biological renewable feedstocks have been greatly intensified over the past decade. Sustained researches have been performed with bio-fuels [5–10], such as bio-ethanol which is the most commonly used in engines due to its renewability and high octane number. However, bio-ethanol has several significant drawbacks in terms of combustion and emission performances, such as low energy density, high latent heat of vaporization and water absorption. Therefore, the search for superior alternatives to bio-ethanol is very important in the area of energy development.

Recently, it is discovered that furan-based fuels such as 2,5-dimethylfuran (DMF) and 2-methylfuran (MF) may be promising alternatives for internal combustion engines because of the breakthrough in their mass production methods, which were reported by the Nature and Science [11,12] respectively. Researchers [13,14] have independently disclosed and further developed a high efficiency approach of converting fructose into MF. First, three oxygen atoms were removed through dehydration from fructose to produce 5-hydroxymenthylfurfural (HMF); second, two oxygen atoms were removed through hydrogenolysis to produce MF. As fructose is abundant and renewable, MF can be a renewable fuel produced by this method. MF with ideal physicochemical properties shown in Table 1 is more attractive than bio-ethanol. The much higher density and lower latent heat of vaporization for MF can improve the mixture formation and cold starting

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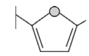
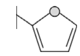
Table 1
Properties of diesel, bioethanol, DMF, MF and gasoline [15–19].

Parameters	Diesel	Bioethanol	DMF	MF	Gasoline
Chemical formula	C_{12}–C_{25}	C_2H_6O			C_4–C_{12}
Research Octane number	20–30	109	101	103	96.8
Motor Octane number	–	90	88	86	85.7
Octane number	–	108	119	–	90–99
Cetane number	52.1	8	9	–	10–15
Oxygen content (%)	0	34.78	16.67	19.51	0
Stoichiometric air/fuel ratio	14.3	8.95	10.79	10.05	14.7
Density at 20 °C (kg/cm^3)	826	790.9	889.7	913.2	744.6
Water solubility (wt%,20 °C)	Negligible	Miscible	Negligible	Negligible	Negligible
Latent heating (kJ/kg) at 25 °C	270–301	919.6	332	358	373
Lower heating value (MJ/kg)	42.5	26.9	33.7	31.2	42.9
Initial boiling point(°C)	180–370	78.4	92	64.7	32.8
Auto-ignition temperature(°C)	180–220	434	286	–	420

performance of engine. The basic water insolubility of MF makes its storage steady. MF also offers an about 34% higher energy density than bio-ethanol, thus reducing the fuel consumption. MF has similar properties as DMF due to their analogous chemical structures and is more competitive than DMF to some extent. Because of lower initial boiling point than DMF, MF evaporates rapidly and improves the mixtures formation in engine cylinder which leading to complete combustion. In addition, its flash point (−22 °C) is lower than DMF (16 °C), which would make the cold start of engine more easily. Practically, engines fueled with MF can operate at higher compression ratio without knock combustion due to the relatively higher RON number, which also makes MF to be a superior alternative fuel for diesel.

Currently, there are several investigations on the combustion and emission of DMF in internal combustion engine. Xu et al. [20,21] first investigated on the use of DMF as a biofuel in the engine. The experiments highlighted that DMF has shorter combustion duration and higher combustion efficiency than gasoline. Zhang et al. [16] evaluated the combustion and emission of DMF in a diesel engine with low temperature combustion. The results indicated that when the DMF blending ratio was 40%, the trade-off relationship between NO_x and soot disappeared and soot emissions were close to zero. Rothamer and Jennings [19] studied the knocking propensity of DMF-gasoline blends. They found that both DMF and ethanol exhibited significant impacts on improving autoignition resistance of gasoline. Recently, Some remarkable researches [22,23] also discussed the performance of the engine with DMF. Results showed that DMF also has significant reduction in soot emissions. All these studies show that DMF is compatible with the existing combustion systems due to its analogous combustion properties to gasoline. Meanwhile, the studies highlight the competitiveness of DMF, but its high emission level of NO_x is a concern because of its high flame temperature.

However, little interest has been triggered in MF. Matthias et al. first investigated the mixture formation and the combustion performance of MF in a DISI engine [24]. Experimental results reveal that MF has excellent combustion stability compared to conventional fuel, especially in cold conditions. Wang et al. [15] comparatively studied the combustion and emission of MF, DMF, gasoline and ethanol in a DISI engine. They concluded that MF had better combustion characteristics and knock suppression ability than both DMF and gasoline. However, the much higher NO_X emissions for MF remain to be resolved. Furthermore, Haiqiao Wei et al. studied the combustion and emissions of MF-gasoline blends in an spark ignition (SI) engine [18] and highlighted that low addition of MF made these blends more competitive than ethanol–gasoline blends.

Until nowadays, to the authors' knowledge, few researches have been carried out with MF as a diesel additive in a DICI engine. Compression ignition of pure MF is difficult in a DICI engine due to its high octane number. Thus, in this study, different mass fractions of MF were added into diesel and the combustion and emission of different MF ratio fuel blends are examined. Experiments were conducted at a constant engine speed of 1800 rpm and break mean effective pressure (BMEP) from 0.13 to 1.13 MPa in a 4-cylinder, 4-stroke DICI engine. Results were compared with that of pure diesel. The regulated emissions (HC, NO_x and CO) as well as the soot were measured and analyzed.

2. Experimental

2.1. Engine and instrumentation

All experiments were carried out on a modified four-cylinder, 4-stroke, water-cooled, DICI engine coupled with a common rail fuel injection system as illustrated in Fig. 1. The primary specifications of the engine are listed in Table 2. An eddy current (EC) dynamometer was used to maintain the engine at a constant speed of 1800 rpm (±5 rpm) and adjust the torque output. The engine working parameters, such as the injection timing and injected fuel mass, were controlled and monitored with an Electrical Control Module (ECU). The injection timing was constantly fixed at 7.5 crank angle (CA) before top dead center (BTDC) by the ECU manager software.

The in-cylinder pressure was measured using a Kistler 6025C pressure sensor, which was flush fitted with wall of the cylinder head. The signals were delivered to a charge amplifier and then received by a CB-466 combustion analyzer. Pressure data was taken at 0.25 crank angle degree (CAD) intervals for 100 successive cycles and then averaged and lightly smoothed based on a five-data points weighted smoothing. A supernumerary compressor and air conditioning system were used to control the pressure and temperature of intake air. The intake air temperature was stably maintained at 25 °C (±0.5 °C). The coolant temperature of engine was precisely stabilized at 85 °C (±1 °C) by a temperature controller, while the lubricating oil temperature was at 87 °C (±2 °C) along with the increasing load.

The gaseous emissions were measured using an AVL gas analyzer with accuracy of 1 ppm for HC and CO and 0.1% for NO_x. The smoke was measured by a smoke opacimeter (NH-T6) with accuracy of 0.01 m^{-1}. Exhaust samples were pumped from the exhaust port through a long tube to the analyzer.

2.2. Test fuels and experimental procedures

Fuels applied in this study are conventional diesel and MF, whose properties are shown in Table 1. Among them, as the base

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