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Experimental investigation on the performance and emissions characteristics of ethanol/diesel dual-fuel combustion

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A B S T R A C T

The diesel compression ignition (CI) engine has higher durability and thermal efficiency than the gasoline spark ignition (SI) engine. However, the high levels of nitrogen oxides (NOx) and particulate matter (PM) emissions are major problems of diesel engines due to the auto-ignition of heterogeneous mixtures. The dual-fuel combustion concept could be solution to environmental concerns. Dual-fuel combustion can be implemented by substituting some of the diesel fuel with high-volatility fuels, such as gasoline and natural gas. The premixed mixture condition can be improved to diminish localized rich and stoichiometric regions. Notably, ethanol has greater potential to reduce PM emissions because it is highly volatile and readily oxidized.

For these reasons, in this research, the effects of varying the ethanol substitution ratio on engine performance and emissions under the dual-fuel combustion condition were experimentally investigated under various load conditions. The test engine was a heavy-duty single-cylinder diesel engine with two direct injectors. Engine speed was fixed at 1000 rpm and the load condition was varied for an indicated mean effective pressure (IMEP) ranging from 0.2 to 0.8 MPa. The ratio of ethanol to the total input energy was controlled from zero to nearly 50% of the input energy. The NOx and PM emissions decreased with increasing ethanol substitution and the mean size of the PM emissions decreased. For the mid-load condition (IMEP 0.6 MPa), the substitution was increased to 63%, but for low and high loads, higher ethanol fractions could not be used because of insufficient ignition energy at low loads and sharp increment of the in-cylinder pressure under high loads.

1. Introduction

The diesel engine is widely used in the transportation and stationary power plant sectors because of its high thermal efficiency and durability. Because the diesel engine is based on compression ignition (CI) combustion from auto-ignition of high cetane-number fuel, that is, diesel fuel, a higher compression ratio can be used than with a gasoline spark ignition (SI) engine. Additionally, while almost all gasoline engines operate under the stoichiometric condition using a three-way catalyst (TWC) to achieve combustion stabilization (including flame propagation speed), diesel engines can run under the lean mixture condition and thereby achieve high thermal efficiency and low carbon dioxide (CO_2) emissions.

Although the diesel engine has excellent thermal efficiency, there is an environmental problem related to the high levels of nitrogen oxides (NOx) and particulate matter (PM) emissions [1]. Basically, the engine-out NOx emissions from a diesel engine are lower than those of a gasoline engine because of the lean mixture combustion [2]. However, a diesel engine cannot use a TWC because of this lean operating condition. For this reason, expensive after-treatment approaches such as selective catalytic reduction (SCR) or lean-NOx trap (LNT) have been applied to current diesel engines to reduce their NOx emissions. Additionally, the high PM emissions from the locally rich mixture combustion zone of the diesel engine can be mitigated by a diesel particulate filter (DPF). These add-ons increase the manufacturing cost of diesel engines. The problem is not restricted to light-duty diesel engines for passenger vehicles, but also extends to heavy-duty diesel engines for trucks and diesel engines for power generation.

Hence, the substitution of diesel fuel by other fuels is a good approach to resolve the environmental issues. Dual-fuel combustion, which can be implemented by using two different fuels, is a representative method. It is difficult to replace all of the diesel fuel with secondary fuels because most secondary fuels do not auto ignite as readily as diesel fuel [3,4]. Thus, replacing only some of the diesel fuel with secondary fuels is a general approach for a dual-fuel engine. In such an engine, controlling two fuels independently using two different fuel-injection equipment (FIE) systems is recommended. In this way, an independent injection strategy can optimize the engine performance

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under various engine operating conditions. For the secondary fuels, low-reactivity and high-volatility fuels such as gasoline, natural gas, and alcohol are commonly used to improve the premixed mixture condition [5–9]. Among these, ethanol is an oxidized fuel, which provides an opportunity to enhance oxidation of the PM emissions [10–13].

Rakopoulos et al. [10] examined the effects of ethanol/diesel fuel blends on the emissions and performance of a heavy-duty diesel engine under the dual-fuel combustion mode. They used a single FIE and ethanol was blended with diesel at 5% and 10% based on the total amount of energy before the experiments. The results showed that the blends reduced PM and carbon monoxide (CO) emissions. However, the fuel conversion efficiency suffered. Since two fuels were already blended before supplying into the cylinder, combustion characteristic of blended fuel combustion was similar with that of neat diesel combustion in the heat release rate (HRR). In addition to that, the small amount of ethanol substitution rate did not influence on the combustion, significantly. On the other hand, Sarjovaara et al. [11] performed dual-fuel combustion experiments using two separate FIE systems with E85 and diesel fuels under various load conditions. They found reduced NOx emissions with increasing E85 ratio, but the combustion efficiency worsened; there was no consistent trend in the thermal efficiency because the combustion phase was not optimized. Padala et al. [12] also conducted dual-fuel combustion experiments with ethanol and diesel fuel for various ratios. There was a clear "NOx versus PM trade-off" relationship with increasing ethanol content. Additionally, a small amount of ethanol substitution enhanced the combustion efficiency relative to that of neat diesel combustion, while ethanol fraction as 50% made combustion deterioration. Although combustion efficiency was introduced, there was no more details in the energy budget of ethanol/diesel dual-fuel combustion including heat and exhaust losses. Other studies have confirmed the reduction of PM emissions with increasing ethanol substitution, but reported many different trends in NOx emissions, combustion behavior, and thermal efficiency. Also, there were rarely described for the energy fractions of ethanol/diesel dual-fuel combustion, which is important to understand the major cause of energy loss.

To clarify this, in this study, we investigated the effect of ethanol fuel as the secondary fuel in a dual-fuel engine on the combustion and emission characteristics under four different load conditions, focusing on the NOx and PM emissions. For each load condition, the diesel injection timing was fixed while ethanol-diesel ratio was changed to avoid additional modifications of the injection strategies. The ethanol-diesel dual-fuel combustion needs to be optimized to improve engine performance [14]. That study did not consider the optimization process but only evaluated the effect of ethanol substitution. The ethanol substitution ratio was varied from zero to nearly 50% for each load condition. This was possible because the low heating value (LHV) of ethanol is almost half that of diesel fuel, and ethanol has a high octane number that resists auto-ignition. The use of a very high fraction of ethanol fuel such as for reactivity controlled compression ignition (RCCI) combustion (using a substitution rate of about 70%–90%) was not evaluated and this fuel was regarded only as the secondary energy source in other research [9]. In addition to examining the emission characteristics, the energy fractions were also investigated for all of the cases in terms of combustion analysis. Herein, the advantages and disadvantages of increasing ethanol substitution in dual-fuel combustion are discussed from environmental (emissions) and energy perspectives.

2. Experimental set-up

2.1. Experimental apparatus

A high-speed direct-injection (HSDI) single-cylinder diesel engine with 1.8 L displacement was used for the experiments. The engine head was modified to accept both a solenoid-type diesel direct injector (DI) and a piezo-type gasoline direct injector (GDI) in the same combustion chamber. Most previous research used port fuel injector (PFI)-DI systems for dual-fuel combustion experiments, so that the reactivity gradient in the cylinder was controlled only by diesel injection from the DI system. However, if a low-reactivity fuel, namely, ethanol, was also injected into the cylinder directly, it was possible to implement a higher reactivity gradient. For this reason, the two fuels were injected into the cylinder directly. The ethanol-diesel ratio was calculated according to the input energy. Detailed specifications of the engine are provided in Table 1. A 55-kW DC dynamometer was used to control engine speed and load. A laminar flow meter (model Z50MY15-2; Meriam Instrument Co.) was used to measure the air flow rate and a wide-band lambda meter (model LA4; ETAS Co.) was used to measure the air-to-fuel (AF) ratio. The NOx (detection limit: 5000 ppm; resolution: 1 ppm), total hydrocarbon (THC) (detection limit: 5000 ppm; resolution: 1 ppm), CO (detection limit: 50,000 ppm; resolution: 1 ppm), CO_2, and O_2 concentrations were measured using an exhaust gas analyzer (model AMA i-60; AVL LIST GmbH), and the PM mass concentration was measured using an aerosol monitor (model Dusttrak DRX 8533; TSI Inc.). The number and the number size distribution of the PM emissions were measured using the Fast Mobility Particle Sizer (model FMPS 3091, TSI Inc.). The FMPS is based on an electric aerosol spectrometer and measures particle sizes ranging from 5.6 to 560 nm with a size resolution of 32 channels. It can measure particle size distributions at a frequency of 1 Hz. To analyze the combustion characteristics, a pressure transducer (model 6052C; Kistler Instrument Corp.) and an adapter (model 6542Q27; Kistler) were installed at the cylinder glow plug and set to measure the cylinder pressure at every 1 degree of crank angle by synchronizing to the signal from the encoder using a combustion analyzer (model DE500, Dewetron Inc.). A schematic diagram of the entire experimental set-up is shown in Fig. 1. The specifications of each fuel are listed in Table 2.

Table 1
Engine specifications.

Parameter	Specification
Cycle [stroke]	4
Displacement [L]	1.8
Number of cylinder [–]	1
Bore [mm]	130
Stroke [mm]	140
Compression ratio [–]	17.1
Injection system [–]	Common rail direct injection
Intake system [–]	Natural aspiration
Number of holes of diesel injector [–]	8
Hole diameter of diesel injector [mm]	0.124

2.2. Energy fraction calculations

The indicated thermal efficiency (ITE) was calculated (using Eq. (1)). The combustion, heat transfer, and exhaust losses were calculated (using Eqs. 24).

$$\text{Gross indicated thermal efficiency} = \frac{W_{gross}}{m_{ethanol} \times Q_{LHV\ of\ ethanol} + m_{diesel} \times Q_{LHV\ of\ diesel}} \quad (1)$$

$$\text{Combustion loss} = \frac{m_{THC\ of\ each\ cycle} \times Q_{LHV\ of\ fuel} + m_{CO\ of\ each\ cycle} \times Q_{LHV\ of\ CO}}{m_{ethanol} \times Q_{LHV\ of\ ethanol} + m_{diesel} \times Q_{LHV\ of\ diesel}} \quad (2)$$