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Effect of ethanol fraction on the combustion and emission characteristics of a dimethyl ether-ethanol dual-fuel reactivity controlled compression ignition engine

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HIGHLIGHTS

- DME-ethanol dual-fuel combustion (DFC) strategy is introduced.
- RCCI combustion of DME and ethanol can be simultaneously reduced NO_x and soot.
- HC and CO in DME-ethanol DFC are lower than those in diesel or biodiesel DFC.

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ABSTRACT

The purpose of this study was to investigate the effect of the ethanol fraction on the combustion and exhaust emissions characteristics of dimethyl ether (DME)-ethanol dual-fuel reactivity controlled compression ignition (RCCI) engine. In this study, a modified single-cylinder diesel engine was used. The main parameters of this study were the in-cylinder injection timing of DME and the ethanol fraction. The ethanol fraction was found to have a more obvious effect on the indicated mean effective pressure (IMEP) for advanced in-cylinder injection timings than around the top dead center (TDC) conditions. For the same ignition timing, the ethanol fraction had little influence on the IMEP. Increasing the ethanol fraction induced an increase in combustion duration and a decrease in premixed combustion duration (CA10–CA50) around the TDC injection condition. The effect of ethanol on P_{max} was insignificant for CA50. The application of the DME-ethanol dual-fuel combustion strategy caused a significant reduction of $ISNO_x$ without deterioration of ISsoot. In addition, a high ethanol fraction led to a low $ISNO_x$ for the same premixed combustion duration. The ISHC and ISCO emissions increased slightly with increasing ethanol fraction for DME-ethanol dual-fuel combustion. However, the emissions from DME-ethanol combustion were lower than those obtained previously with biodiesel-ethanol and diesel-ethanol dual-fuel combustion.

1. Introduction

Thermal efficiency and fuel economy of compression ignition-type diesel engines is higher than that of spark ignition-type gasoline engines because of the high compression ratio, lean combustion, and low pumping loss [1,2]. However, diesel combustion forms more nitrogen oxides (NO_x) and particulate matter (PM), including soot, owing to its unique combustion characteristics [3–5], contributing to environmental pollution. In addition, exhaust emission and fuel economy regulations for diesel engine vehicles have been progressively tightened owing to increasing concerns about reduction of green-house gases including carbon dioxide (CO_2). Hence, many researchers have tried to reduce exhaust emissions, such as NO_x, PM, hydrocarbons (HCs), and carbon monoxide (CO), using advanced combustion technologies [6–8] and after-treatment systems [9,10]. In fact, a simultaneous reduction of NO_x and PM emissions from compression ignition diesel combustion is very difficult because the two have a trade-off relationship with each other [11]. PM emissions are generally produced by inhomogeneous and locally fuel rich regions, while NO_x is formed in the high temperature region around the stoichiometric air-fuel regions. Recently, many advanced combustion technologies have been introduced for the simultaneous reduction of NO_x

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Nomenclature

ATDC	after top dead center (degree)	HC	hydrocarbon
BTDC	before top dead center (degree)	IMEP	indicated mean effective pressure (bar)
CA	crank angle (degree)	IS-	indicated specific-
CA10	crank angle at 10% of the cumulative heat release	NO_x	nitrogen oxides
CA50	crank angle at 50% of the cumulative heat release	P_{inj}	injection pressure (bar)
CA90	crank angle at 90% of the cumulative heat release	ROPR	rate of pressure rising (bar/degree)
CO	carbon monoxide	P_{max}	maximum combustion pressure (bar)
DME	dimethyl ether	τ_{DME}	DME in-cylinder injection timing (CA degree)

and PM. One of these technologies is premixed low temperature combustion (LTC), which simultaneously reduces both NO_x and PM through reduction of the in-cylinder combustion temperature. The low combustion temperature inhibits the formation of NO owing to its formation reaction having high activation energy [12]. In addition, a long ignition delay provides a sufficient mixture formation period for mixing between air and fuel. Therefore, the fuel rich region in the combustion chamber decreases, and soot formation is inhibited. The advantages of LTC are simultaneous reduction of NO_x and PM, improvement of thermal efficiency due to the short combustion duration, and low heat transfer loss. However, the load limitation resulting from a high pressure rise rate (PRR) and heat release rate (HRR), as well as the difficulty of achieving direct control of combustion phasing, are disadvantages of LTC. Representative combustion strategies for LTC include homogenous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI). Although the HCCI strategy is a thermodynamically attractive technology, controlling the ignition timing and heat release characteristics is difficult [13]. PCCI combustion is a combination of HCCI and conventional direct injection compression ignition, and can be realized through early injection during the initial stages of the compression stroke to provide adequate time for fuel and air to mix, before ignition. The major drawback of the HCCI and PCCI combustion strategies is the high levels of HC and CO emissions. The concept of reactivity controlled compression ignition (RCCI) has been suggested to solve the problems encountered in HCCI and PCCI, such as the difficulties in controlling the ignition timing and combustion phasing, and the high levels of HC and CO emissions. In RCCI, a high-octane fuel, such as gasoline, ethanol, or biogas, is injected into the intake port, while a fuel with high cetane number, such as diesel, biodiesel, or dimethyl ether (DME), is directly injected into the combustion chamber. In RCCI combustion, combustion phasing and ignition timing can be controlled by varying injection timing and adjusting the mixing ratio of the high cetane and high-octane fuel [14].

Many researchers have investigated LTC, HCCI, PCCI, and RCCI using various port fuels and various in-cylinder injection fuels in order to achieve low emission and high thermal efficiency combustion. Ma et al. [15] investigated the effect of diesel injection strategies on gasoline-diesel dual-fuel combustion. They found that the lowest ISFC (indicated specific fuel consumption) could be achieved in dual-fuel combustion with very low NO_x and soot emissions by early second injection timing. Gasoline-diesel dual-fuel combustion characteristics in a heavy-duty diesel engine were studied experimentally and numerically by Hanson et al. [16] and Kokjohn et al. [17]. Curran et al. [18] reported a 90% reduction in both NO_x and soot emission, and an improvement in the thermal efficiency of a single-cylinder diesel engine. Although many studies have been conducted on dual-fuel combustion, LTC, and RCCI strategies in diesel engine, additional research is necessary. In particular, it is necessary to study the application of alternative fuels, such as dimethyl ether (DME) and ethanol in RCCI combustion. In this study, the premixed chamber was installed on the intake pipe, and ethanol was injected into the premixed chamber in order to investigate the effect of ethanol port injection. The intake air conditions (oxygen concentration, the intake air temperature, etc.) were changed by the ethanol injection, and it had an influence on the combustion and emission characteristics. In addition, the in-cylinder direct injection timings of DME had an effect on combustion and emission characteristics. The purpose of this study is to investigate the effect of the ethanol fraction and in-cylinder DME injection timings on the combustion and emission characteristics of an RCCI engine system.

2. Experimental apparatus and procedure

In this study, a single-cylinder diesel engine with a DME-ethanol dual-fuel supply system was used. A schematic diagram of the experimental apparatus is shown in Fig. 1. The single-cylinder diesel engine has a displacement of 373.3 cc, a compression ratio of 17.8, and a re-entrant type piston bowl shape. The detailed engine specifications are shown in Table 1. The engine speed was controlled by a DC (direct current) dynamometer (55 kW h). The in-cylinder injection timing and injection quantity of DME were controlled using a timing pulse generator (Blue Planet, TPG-28MP) and an injector driver (TEMS, TDA-3300). In order to analyze the combustion characteristics, such as combustion pressure, heat release rate, and ignition characteristics, the in-cylinder combustion pressure data were measured by the pressure sensor and the piezoelectric transducer (Kistler 6067A80) coupled to a charge amplifier (Kistler 5011B), which amplified and converted the measured charge signal into a voltage signal. The pressure sensor was installed on the cylinder head at the position of the glow plug. The transducer converted the in-cylinder electric signal to its equivalent in-cylinder pressure value.

Because DME is in the vapor phase under atmospheric conditions, liquefied DME was supplied by a high-pressure pump (Haskel HSF-300) after pressurizing it to 0.6 MPa. The pressurized DME was directly injected into the combustion chamber using a 6-hole injector with a hole diameter of 0.128 mm. The fuel return system of the injector was modified to maintain DME in the liquid phase in the fuel return line. To avoid damage to the injection system and improve the lubricity of DME, 1000 ppm of a lubricity improver (Lubrizol) was added to the pure DME. DME injection pressure was fixed at 60 MPa, and the injection timing was varied from before top dead center (BTDC) 25° to top dead center (TDC). Properties of DME and ethanol fuel are summarized in Table 2. Ethanol was used as a premixed fuel in this study. To supply ethanol, the intake system of the single-cylinder diesel engine was modified as shown in Fig. 1 (dotted line). A premixing chamber was added to the intake pipe to allow the intake air and ethanol to mix sufficiently. Ethanol was injected through a 6-hole direct injection injector that was controlled by an injector driver (TEMS, TDA-3300A). The ethanol injection pressure, which was fixed at 3 MPa, was controlled using nitrogen gas and a pressure regulator.