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Effect of the diesel injection strategy on the combustion and emissions of propane/diesel dual fuel premixed charge compression ignition engines

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ABSTRACT

In this research, the effects of the propane ratio and diesel direct injection strategy on the emissions and combustion characteristics of a propane-diesel dual fuel PCCI (premixed charge compression ignition) engine were evaluated. Under low speed and low engine load, the propane ratio was varied from 30 to 70% to investigate the effect of the LHV (low heating value) fuel content in the supplied fuel (propane and diesel mixture). The EGR rate was adjusted to suit each propane ratio to ensure combustibility and restricted emissions. Early diesel injection strategies have an excellent effect on the dual fuel combustion in terms of simultaneously reducing NOx (nitrogen oxides) and PM (particulate matter) emissions. This concept is based on PCCI combustion, in which the ignition delay is longer than the diesel injection. Meanwhile, although early single diesel injection has been an effective strategy for reducing NOx and PM emissions simultaneously, it was possible to further reduce NOx emissions using an early split injection strategy with a 30% propane ratio. Additionally, when the propane ratio was 70%, the ignitibility deteriorated due to early single diesel injection, which led to a much leaner air–fuel mixture condition locally prior to auto-ignition, causing unstable combustion. As a result, a multiple injection strategy with earlier main injection and a small diesel post-injection as a triggering source was adopted to stabilize dual fuel PCCI combustion with a high propane ratio. The results emphasized that the diesel injection strategy could be adjusted to suit various propane ratios under dual fuel PCCI combustion to reduce NOx and PM emissions while maintaining thermal efficiency.

1. Introduction

The reduction of emissions from diesel vehicles and the depletion of fossil fuels are two major problems currently confronting engine researchers. In 2014, enforcement began of the EURO-6 regulation, which required a reduction of NOx (nitrogen oxide) emissions to half the level required by previous regulation, while the restrictions on PM (particulate matter) emission levels remained the same. This restriction is challenging for internal combustion engines without emissions after-treatment because there is a trade-off relation between NOx and PM emissions under diesel diffusive combustion [1–3]. Regarding the second issue mentioned above, according to a report from the US EIA (Energy Information Administration), the usage of fossil fuel energy for transportation worldwide will increase by 35% within 20 years [4]. In particular, in the future, the demand for diesel fuel may increase drastically compared to other fossil fuels, such as jet fuel and gasoline. Therefore, a clean diesel combustion system with high thermal efficiency is needed.

For conventional diesel combustion, there are two main combustion modes. The first mode is a premixed combustion phase achieved by the injected fuel prior to the SOC (start of combustion). A mixing controlled combustion phase occurred after the premixed combustion phase due to the heterogeneous air–fuel mixture during the combustion period. Because most of the NOx and PM emissions were formed during the mixing controlled combustion phase, it is important to increase the premixed combustion phase by constructing a well-premixed air–fuel mixture before SOC.

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For this reason, to reduce the engine-out NOx and PM emissions simultaneously, PCCI (premixed charge compression ignition) combustion with direct injection fuel, such as diesel or jet fuel, has been the subject of most of the relevant research to date. Certainly, PCCI combustion has the potential to reduce both NOx and PM emissions more effectively than conventional diesel combustion. In this paper, PCCI combustion is defined as combustion in which the ignition delay is longer than the diesel injection. The ignition delay is calculated as the duration between the SOI (start of injection) and SOC. Earlier diesel injection and the use of a higher EGR (exhaust gas recirculation) rate usually lead to an ignition delay exceeding the injection duration. As a result, the premixed combustion phase increases, while the mixing controlled combustion phase decreases [5–7]. The enhanced air–fuel mixture resulting from the prolonged ignition delay produces low PM emission, and the higher EGR rate suppresses NOx emissions.

However, because of early diesel injection, spray-wall impingement occurred under relatively low in-cylinder pressure conditions, which increased the spray penetration [5,8]. Thus, this wall impingement was problematic in terms of THC (total hydrocarbons), and oil dilution issues occurred. Moreover, the homogeneity of the air–fuel mixture from PCCI combustion was still insufficient due to the low volatility of high-reactivity fuels. In addition, a shorter mixing time compared to that for port fuel injection was unavoidable when PCCI combustion was achieved by direct injection only.

From the above results, it could be concluded that although increasing premixed combustion by PCCI combustion would be suitable for reducing NOx and PM emissions, PCCI combustion by direct injection fuel only has limitations. Therefore, dual fuel combustion was introduced to the PCCI combustion concept. Inagaki et al. first suggested the dual-fuel (premixed iso-octane and direct-injected diesel) PCI (premixed compression ignition) concept [9]. The purpose of this study was to reduce the EGR requirement while maintaining low NOx and PM emissions. In this research, premixed fuel was a main power source, and the high-reactivity fuel, i.e., diesel, enhances the overall in-cylinder reactivity. Because a high-volatility fuel, iso-octane, improved homogeneity, it caused near-zero PM emissions.

This concept had an influence on RCCI (reactivity controlled compression ignition) combustion. RCCI combustion could be implemented by supplying low-reactivity fuel, such as gasoline and gaseous fuels, through a PFI (port fuel injector) and high-reactivity fuel, usually diesel fuel, with DI (direct injection) [10]. The ratio of low-reactivity fuel was usually higher than that of high-reactivity fuel. High-reactivity fuel was injected by a split injection strategy. The first injection was performed at near BTDC (before top dead center) 80 CA (crank angle) for squish conditioning and the second one was injected at BTDC 30~45 CA as an ignition source [11]. The overall combustion process was dependent on the reactivity stratification in the cylinder. Because the low-reactivity fuel was well-distributed in-cylinder and the small amount of diesel fuel was used for ignition, this combustion was similar to HCCI (homogeneous charge compression ignition) combustion [10–12]. The only difference between the two combustion concepts, RCCI and HCCI, is whether high-reactivity fuel is supplied. RCCI combustion promises higher thermal efficiency due to a reduction of heat loss from the shortened combustion duration, low combustion temperature and near-zero NOx and PM emissions from homogeneous mixture combustion.

Although RCCI combustion has the advantage of improving thermal efficiency and reducing emissions, RCCI combustion has been mainly used for early diesel injection to enhance the air–fuel mixture condition, similar to HCCI combustion. As a result, combustion phase was decided by the degree of reactivity stratification in the cylinder only, which means SOC could not be controlled by diesel SOI directly. Additionally, the various operating strategies of diesel SOI and EGR rates should be elaborated using different low-reactivity fuel ratios.

Therefore, in this research, different operating strategies for dual fuel combustion for controlling the combustion phase directly by diesel SOI were assessed for various premixed fuel ratios. The premixed fuel ratio (propane ratio in this research) was increased within the range under which stable combustion can be maintained (from 30 to 70%). Stable combustion was assessed by the CoV (coefficient of variation) of the gIMEP (gross indicated mean effective pressure); in this study, the CoV of the gIMEP was not to exceed 5% [13,14]. While the combustion remained stable, the restrictions for NOx and PM emissions were set at under 50 ppm and 3 mg/m^3, respectively [14]. The EGR rate was also varied with the propane ratio to ensure combustion stability and low NOx emission levels. The results of this work emphasized that there was an appropriate diesel injection strategy for each propane ratio to stabilize the dual fuel PCCI combustion and achieve low NOx and PM emissions simultaneously while maintaining the thermal efficiency at same level as in the conventional diesel combustion condition.

2. Experimental procedure

2.1. Experimental apparatus

A HSDI (high-speed direct injection) single-cylinder diesel engine with a 497 cc displacement based on the EURO IV standard was used for these experiments. A piezo injector with a spray pressure of up to 160 MPa was equipped with a common rail system. More detailed specifications of the engine are introduced in Table 1. To control the engine, a 37 kW DC dynamometer was adopted, and the diesel fuel was preserved in the fuel tank during all the experiments. The oil and coolant temperatures were controlled at near 85 °C. Additionally, the diesel fueling temperature was maintained at 40 °C during the experiments. To measure the diesel fuel flow rate, a mass burette type flowmeter (ONO SOKKI, FX-203P) was used. The concentrations of NOx, THC (total hydrocarbon), CO, CO_2 and O_2 were measured using an exhaust gas analyzer (Horiba, MEXA 7100DEGR), and the PM emissions were measured using a smokemeter (AVL, 415S). Thus, the EGR rate based on the volumetric values was calculated from the CO_2 fractions in the exhaust gas and intake gas. To simulate a turbo charger and boosting intake-air, compressed shop-air was used. Then, to control the amount of air and maintain constant flow, sonic orifices and pressure regulators were used. To measure the pressures, an absolute pressure transducer (Kistler, 4045A5) was used, and a relative pressure transducer (Kistler, 6055Bsp) was adopted for in-cylinder pressure. Signals from the pressure transducers were recorded using a scale of one crank angle per 100

Table 1
Engine specifications.

Engine type	Single cylinder (four-stroke) (DI)
Displacement [L]	0.497
Bore [mm]	83
Stroke [mm]	92
Connecting rod [mm]	145.8
Compression ratio	15.5
Number of nozzle holes [-]	7
Spray angle [°]	153
Nozzle diameter [mm]	0.128
Hydraulic flow rate [cc/10 MPa/30 s]	380