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An assessment of the biodiesel low-temperature combustion engine under transient cycles using single-cylinder engine experiment and cycle simulation

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ABSTRACT

An operational strategy was developed to implement LTC (low-temperature combustion) with 50% biodiesel blended fuel (B50), named B-LTC, and evaluated under a combination of a single-cylinder engine experiment and a cycle simulation. The fuel consumption, regulated emissions, and exhaust gas temperature maps were constructed from experiments in a single-cylinder diesel engine at the speed range between 1000 and 1600 rev/min. A dataset of pumping and friction of a 6-cylinder diesel engine was employed to construct a 6000-cm^3 B-LTC engine from the SCRE (single-cylinder research engine) experimental results. The engine maps of the virtual 6-cylinder B-LTC engine were then input into a zero-dimensional (0-D) model for the transient-cycle simulation. The cycle simulation was performed under the two representative transient cycles, namely the WHTC (worldwide harmonized transient cycle) and the NRTC (non-road transient cycle). The WHTC simulation estimated the engine-out $CSNO_x$ (cycle-specific NO_x) of 0.94 g/kWh, which was lowered to 0.30 g/kWh by the SCR (selective catalytic reduction), while CSFC (cycle-specific fuel consumption) and cycle-specific soot (CSsoot) were 310 g/kWh and 0.01 g/kWh, respectively. The NRTC simulation results also showed that the engine-out CSsoot emission was 0.01 g/kWh. The urea-dosing SCR model reduced $CSNO_x$ from 0.99 g/kWh to 0.25 g/kWh. The CSFC was 274 g/kWh in the NRTC simulation.

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1. Introduction

The use of biodiesel-blended diesel fuels has been considered as an effective approach to remove soot emission under the LTC (low-temperature combustion) regime [1–6]. The soot reduction allows the B-LTC (biodiesel-attaining LTC) operation to achieve simultaneous reduction of the nitric oxides (NO_x) and the soot under achievable EGR (exhaust gas recirculation) levels [4]. The lower EGR level required for the simultaneous NO_x and soot reduction helps the B-LTC region expand to higher load conditions [1]. Although NO_x increases in proportion to the biodiesel blend rate [6–10], the increase can be alleviated by an adaptation of the MK (modulated kinetics)-like LTC strategy, which employs late injection timings, high injection pressure, and high swirl flow [11–14].

However, the use of biodiesel-blended fuels leads to a fuel penalty due to the lower energy density. Abuhabaya et al. [15] observed that the maximum engine torque was decreased by 5% in the 20% biodiesel blended fuel (B20) at all engine speeds. Therefore, the biodiesel blend rate should be minimized in spite of the definite advantage in soot reduction. Lee et al. [5] concluded that 60% was the minimum biodiesel blend rate that achieved both NO_x and soot emissions under the acceptable levels at a gross $IMEP_g$ (indicated mean effective pressure) of 900 kPa. It can be inferred that the required blend rate for the simultaneous NO_x and soot removal is reduced at lower load conditions because the soot emission must be lower.

Significant efforts in the optimization of the combustion chamber have been undertaken for soot removal as well. Among the many piston designs, the chamfered bowl, also known as the two-step bowl piston, has shown an outstanding performance in soot reduction. Kim et al. [16] showed that the chamfered-bowl piston exhibited substantial soot reduction in a medium load operation. Yoo et al. [17] performed both computational and experimental investigations on the in-cylinder geometry optimization for a 2400-cm^3, 4-cylinder diesel engine. Their study concluded that the chamfered bowl with a 142-degree included

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angle nozzle was able to meet the soot emission standard in the Tier-4 Final regulation without a DPF (diesel particulate filter). Some studies showed that the chamfered piston exhibited not only lower soot and NO_X [18,19] but also improved fuel economy by 3% [19]. Note that the present study employed a biodiesel-blended fuel, which is a significant factor in the optimization of the in-cylinder geometry [20].

Despite the aforementioned promising results, the performance of the B-LTC concept has not been investigated at wide ranges of speed and load conditions. In particular, the performance under transient-cycle modes has rarely been investigated due to the cost and instrumentation complexity. Most LTC studies investigated the potentials of the combustion concept at a few common operating conditions, such as a 25% load at a medium speed [4]. Zhu et al. [2] performed engine experiments only at an engine speed and IMEP of 1500 rev/min and 0.7 MPa, respectively. The testing conditions for the particulate matter emission comparison were 1500 rev/min and 600 kPa of IMEP in Su et al. [3]. Even the investigation for the load limit of the biodiesel LTC strategy was conducted at a single engine speed of 1500 rev/min [1].

The objective of the present study is to assess the B-LTC concept using a dual mode strategy for the LTC and CDC (conventional diesel combustion) modes under the representative transient cycle modes, namely the WHTC (worldwide harmonized transient cycle) and the NRTC (non-road transient cycle). The potential of the concept to meet the soot emission standard without the use of a particulate filter was also investigated. The LTC operation was limited on the basis of the engine intake system capability. Lee et al. [5] showed that the intake condition requirement for NO_X and soot removal was an intake pressure and EGR levels of 200 kPa and 50%, respectively, at 900 kPa $IMEP_g$. Those intake conditions were achievable for a 6-cylinder diesel engine equipped with a LP-EGR (low-pressure EGR) system. Thus, 900 kPa $IMEP_g$ was determined as the highest load condition for the LTC operation. From 900 kPa to 1500 kPa of $IMEP_g$, the conventional diesel combustion mode was employed. The chamfered-bowl piston, which showed a substantial soot reduction [16], was employed in the present study.

2. Engine experiments

2.1. Instrumentation

A single-cylinder version of the 6-cylinder diesel engine was employed for engine experiments. The engine specifications are listed in Table 1. The SCRE (single-cylinder research engine) was coupled with an AC (alternating-current) dynamometer for active loading. The specifications of the AC dynamometer are listed in Table 2. Five cylinders, from cylinder 1 to cylinder 5, were deactivated by through-holes on the pistons, and the rocker arms were removed.

Table 1
Engine specifications.

Parameters	Nominal values	Units
Bore	100	mm
Stroke	125	mm
Displacement volume	982	cm^3
Compression ratio	17.4	–
Swirl ratio	1.5	–
Piston bowl type	Chamfered	–
Fuel injection system	CRDI	–
Max. fuel pressure	160	MPa
Injector nozzle holes	0.160 × 7	mm

Table 2
The specifications of the AC dynamometer.

Parameters	Units	Nominal values
Rated speed	rev/min	1800
Rated torque	Nm	600
Max speed	rev/min	2200
Max power	kW	100

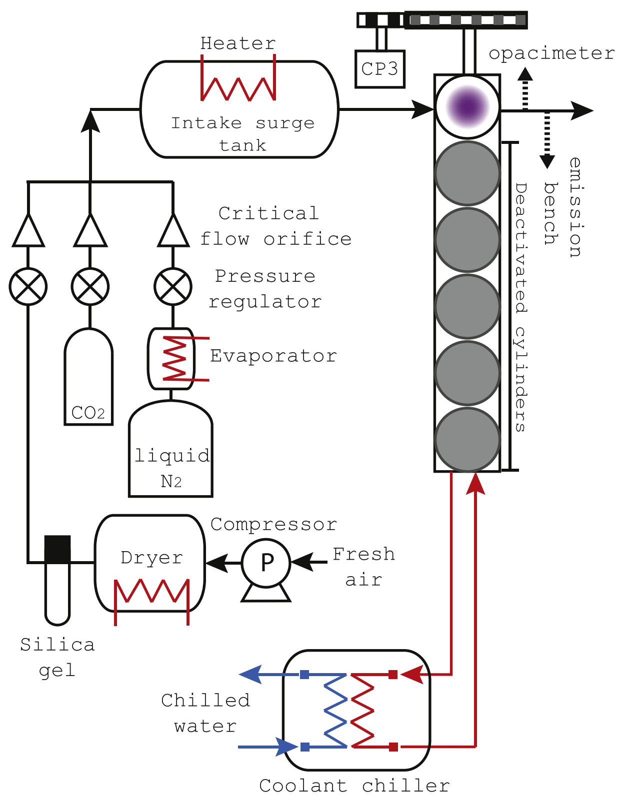

Fig. 1. A schematic diagram of the intake system.

The engine was equipped with the CRDI (common-rail direct-injection) fuel injection system (Bosch, Germany). The CP3 high-pressure fuel pump (Bosch, Germany) was driven by the engine crankshaft, as illustrated in Fig. 1.

Intake and exhaust manifolds were replaced with runners that connected an engine head with surge tanks. The intake surge tank was equipped with a submerged electric heater for intake temperature control. The volumes of the two surge tanks were approximately 100 times larger than the engine displacement

Table 3
The volume ratio of the three gas species in the synthetic intake.

Intake pressure kPa	Intake oxygen %	Air %	N_2 %	CO_2 %
100	13	62.1	31.0	6.9
	15	71.6	23.2	5.2
	17	81.2	15.4	3.4
200	11	52.4	38.9	8.7
	13	61.9	31.1	7.0
	15	71.5	23.3	5.2
	17	81.0	15.5	3.5
250	11	52.4	38.9	8.7
	13	61.9	31.1	7.0
	15	71.4	23.3	5.2
	17	81.0	15.5	3.5

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