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Effect of biodiesel and ethanol on load limits of high-efficiency premixed low-temperature combustion in a diesel engine

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HIGHLIGHTS

- ▶ The effect of biodiesel and ethanol on low temperature combustion (LTC) load limits are investigated.
- ▶ LTC is smoke limited at high load for diesel and biodiesel fuels.
- ▶ Ultra low smoke emission is possible with biodiesel–ethanol fuel.
- ▶ The high load limits of biodiesel–ethanol LTC is caused by a drop of combustion efficiency rather than smoke.
- ▶ High-efficiency LTC load limits can be extended to 0.35–0.82 MPa IMEP with biodiesel–ethanol.

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ABSTRACT

The fuel effect of diesel, biodiesel and biodiesel–ethanol on load limits of high-efficiency premixed low temperature combustion (LTC), defined by nitrogen oxides (NO_x) ⩽ 1 g/kg-fuel, smoke ⩽ 0.5 filter smoke number (FSN) and combustion efficiency ⩾ 96%, is investigated using a diesel engine in this study. Low load operation range is limited by a drop in combustion efficiency for all three test fuels. High load operation range is limited by a sharp increase in smoke with diesel fuel. Biodiesel produces lower smoke at high load but still exceed the limit of 0.5 FSN. Unlike diesel and biodiesel, smoke is no longer the limiting factor for high load operation with biodiesel–ethanol fuel. A blend of 20% ethanol in biodiesel results in ultra-low smoke emission (maximum FSN less than 0.25), which is thought to be caused by the joint effects of better fuel air mixing and higher fuel oxygen fraction.

Since the high load operating limit of biodiesel–ethanol is caused by a drop in combustion efficiency and not smoke as for the other fuels, biodiesel–ethanol was used to further extend the LTC load limits at a higher boosting pressure. When intake pressure is increased from 120 to 150 kPa, biodiesel–ethanol still demonstrates ultra-low smoke, and its high-efficiency LTC operating range is extended from 0.40–0.65 to 0.35–0.82 MPa indicated mean effective pressure (IMEP).

1. Introduction

The diesel engine is an attractive vehicle power source due to its high fuel efficiency, but conventional diesel engine combustion operates in a heterogeneous and high temperature range where both nitrogen oxides (NO_x) and particular matter (PM) are produced. Conventional diesel combustion typically displays a trade-off between these pollutants where NO_x and PM formation are primarily affected by in-cylinder combustion temperature and local equivalence ratio. If combustion is shifted to significantly lower temperatures, NO_x and PM can be lowered simultaneously [1,2]. This combustion mode is called low temperature combustion (LTC).

Cooled exhaust gas recirculation (EGR) is a common strategy to achieve LTC by reducing combustion temperature and prolonging ignition delay. Reduced combustion temperature suppresses NO_x and PM formation. Prolonged ignition delay allows additional time for fuel and air mixing, which is beneficial for reducing fuel rich zones and suppresses soot formation. High EGR rate (more than 55%) is normally required to simultaneously reduce NO_x and PM emission to very low levels, which is usually accompanied by

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Nomenclature

LTC	low temperature combustion
EGR	exhaust gas recirculation
NO_x	$NO + NO_2$
FSN	filter smoke number
IMEP	indicated mean effective pressure
PM	particular matter
HC	hydrocarbon
CO	carbon monoxide
ULSD	ultra-low sulfur diesel
B–E	biodiesel–ethanol
BTDC	before top dead center
CA	crank angle
ATDC	after top dead center
SOI	start of injection
EOI	end of injection
MFB	mass fraction burned

deterioration of combustion efficiency and increased levels of incomplete combustion products [3,4]. Another challenge for LTC is high load operation where smoke emissions and combustion noise exceed acceptable levels. High smoke is a result of increased combustion temperature and reduced in-cylinder oxygen [5]. High combustion noise is caused by LTC being more premixed, which leads to rapid heat release. Thus, extending LTC operation to higher load remains a challenge [6].

Several strategies to solve the problems at high load have been investigated and reported recently. Intake boost can improve combustion efficiency and extend high load limit [7–9]. Low compression ratio can prolong ignition delay, reduce smoke and increase maximum load but with an accompanying drop in thermal efficiency [10]. Multiple fuel injections can reduce peak heat release rate and improve fuel air mixing, resulting in reduced noise and smoke [11].

In addition to engine parameters, fuel properties also benefit LTC load limits. Low cetane number fuels have longer ignition delay, which can improve fuel air mixing. Reduced soot and higher load LTC operation range have been observed with low cetane number fuels [12,13]. Gasoline has high resistance to auto-ignition and is more volatile than diesel, both of which can reduce smoke emission. Extended LTC high load limit is reported by Han et al. using a blend of diesel and gasoline [14].

Previous studies have shown that smoke is a major limiting factor for LTC at high load with diesel fuel [5,15]. In this study, the potential of biodiesel and biodiesel–ethanol fuels to extend the high load limit of LTC mode is investigated because biodiesel and ethanol are viable alternative fuels that have been shown to reduce smoke from conventional diesel combustion and the low cetane number of ethanol has the potential to improve fuel air mixing [16–18]. Late injection, boosting and moderate EGR are used because these strategies can effectively enhance combustion efficiency and control combustion noise in LTC mode [3–5,7–9,15]. This is the first study to compare the load limits of diesel, biodiesel and biodiesel–ethanol fuels in a high-efficiency premixed low-temperature combustion mode using late injection, boosting and EGR.

2. Experimental method

2.1. Test apparatus

Experiments were carried out on a single cylinder engine based on a General Motors/Isuzu 1.7 L four-cylinder direct-injection compression-ignition diesel engine. Compression ratio was reduced from 19:1 to 15:1 by changing the original piston to one with a larger combustion bowl.

The engine has four valves per cylinder and a centrally placed common rail fuel injector. An EGR valve and EGR cooler were installed to draw cooled exhaust gas to an intake tank where exhaust gas was mixed with intake air. Specifications of the test engine are shown in Table 1.

Cylinder pressure was measured by a water-cooled Kistler 6041A pressure sensor at a resolution of 0.2 crank angle degrees across 200 consecutive cycles for each test point. Heat release was calculated based on a zero-dimensional ideal gas combustion model with Hohenberg's correlation used for heat transfer calculation [19,20].

Gaseous exhaust emissions were measured by a Horiba MEXA-7500DEGR emission bench and reported as fuel-specific emission index with units of g/kg-fuel. Smoke emission was measured by an AVL 415s smoke meter and shown as filter smoke number (FSN). Combustion noise was monitored by an AVL 450s combustion noise meter and reported in decibels.

EGR rate was calculated from the ratio of intake and exhaust carbon dioxide (CO_2) concentrations, as shown in Eq. (1). Combustion efficiency and equivalence ratio are calculated based on emissions measurements from the emission bench. Equivalence ratio is calculated based on carbon and oxygen balances.

$$\text{EGR rate} = (\text{Intake } CO_2 \text{ concentration}/\text{Exhaust } CO_2 \text{ concentration}) \times 100\% \quad (1)$$

2.2. Test fuels and method

Three fuels were tested in this study: ultra-low sulfur diesel (ULSD), neat soybean methyl ester biodiesel, and 80% biodiesel blended with 20% ethanol (biodiesel–ethanol, abbreviated B–E). The Fuel specifications are shown in Table 2. Considering the unmodified design of fuel injection system, fuel ethanol content was limited to 20% [21]. It should be noted that blending ethanol increases the flammability risk compared to biodiesel and diesel. Since E85 (85% ethanol and 15% gasoline) has the same issue and flex fuel vehicles (FFV) that can operate on E85 have been deemed safe, biodiesel–ethanol can also utilize similar engineering features to address this flammability problem.

Throughout this study the engine speed was held constant at 1500 r/min, injection pressure was maintained at 100 MPa and combustion noise was maintained below 90 dB. Engine coolant, oil, and EGR coolant temperatures were set to 85 °C and intake temperature was set to 65 °C. Exhaust pressure was adjusted to be 15 kPa higher than intake pressure to enable sufficient EGR flow.

Table 1
Summary of engine specifications.

Displacement	425 cm^3
Bore	79.0 mm
Stroke	86.0 mm
Connection rod length	160.0 mm
Compression ratio	15:1
Injector nozzle hole number	6
Injector nozzle spray angle	150°