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Combustion characteristics of a gasoline engine with independent intake port injection and direct injection systems for *n*-butanol and gasoline

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ABSTRACT

N-butanol, as a sustainable biofuel, is usually used as a blend with gasoline in spark ignition engines. In this study, the combustion characteristics were investigated on a four-cylinder spark ignition gasoline engine with independent port fuel injection and direct injection systems for *n*-butanol and gasoline in different operating conditions. The results show that in the case of port fuel injection of *n*-butanol with direct injection gasoline at a given total energy released in a cycle, indicated mean effective pressure is slightly affected by spark timing at stoichiometry while it changes much more with delayed spark timing in lean burn conditions and is much higher in lean burn conditions compared to stoichiometry at given spark timings. With the increase of *n*-butanol percentage in a fixed total energy released in a cycle at given spark timings, ignition timing advances, combustion duration shortens, indicated mean effective pressure and indicated thermal efficiency increase. For the cases of port fuel injection of *n*-butanol with direction injection gasoline and port fuel injection of gasoline with direction injection *n*-butanol at a fixed total energy released in a cycle, their indicated mean effective pressures are close. But their combustion processes are dependent on fuel injection approaches.

1. Introduction

The improvement in vehicle/engine fuel efficiency and the use of alternative fuels in internal combustion engines are the effective measures for environmental protection and emission reduction. There are a variety of alternative fuels for gasoline engines. Recently, there is a growing interest in sustainable fuels, due to their lower lifecycle emissions of carbon dioxide (CO_2). Biofuels such as ethanol produced from sugar plants and corn can be used in gasoline engines. But the biofuels produced from edible crops and vegetables tend to raise food prices. Therefore, the biofuels produced from lignocellulosic materials, such as wood, vegetable waste and non-edible plants, are more favorable for the well-to-wheel CO_2 balance without negative impact on food supply.

N-butanol can be produced in a similar process to ethanol. It can be used as an alternative fuel for diesel engines. Armas et al. [1] found that the blend with 16% *n*-butanol and 84% diesel by volume can decrease smoke opacity and particle concentration during warm engine start when the coolant and lube oil temperatures are less than 90 °C. However, it causes combustion instabilities at cold start when the coolant and lube oil temperatures are about 17 °C. In a six-cylinder, turbocharged and after-cooled heavy duty direct injection engine fueled with diesel and its blends with 8% and 16% *n*-butanol by volume, respectively, Rakopoulos et al. [2] found that with the increase of *n*-butanol in the *n*-butanol-diesel blend, smoke, carbon monoxide (CO) and nitrogen oxide (NOx) emissions decrease. However, hydrocarbon (HC) emissions increase with increased *n*-butanol content in the blends. Similar change trends of smoke, CO, HC and NOx emissions with *n*-butanol in the blends were also found in a single-cylinder, naturally-aspirated, high-speed direct injection engine when diesel and its blends with 8%, 16% and 24% *n*-butanol by volume were used [3]. At constant specific NOx emissions, the addition of *n*-butanol to diesel can significantly improve soot and CO emissions without a serious impact on break specific fuel consumption [4]. Through port fuel injection of *n*-butanol and direct injection peanut biodiesel in a single cylinder diesel engine, Soloiu et al. [5] found that premixed charge compression ignition coupled with low temperature combustion can decrease in-cylinder pressure by 25% and delay peak pressure in the power stroke at 800 rpm and 0.1–0.3 MPa indicated mean effective pressure (IMEP) compared to the case of burning diesel. In the meantime, the reduction in soot is about 98%, and NOx emissions are decreased by 98% at 1 IMEP and 77% at 3 IMEP through controlling the combustion phases. But HC and CO emissions are very high. At 0.1–0.5 MPa

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IMEP and 1400 rpm, it was found [6] that acetaldehyde emissions increase with decreased IMEP and increased *n*-butanol fraction in the total energy released in a cycle, due to port fuel injection of *n*-butanol. In addition, the use of *n*-butanol and biodiesel in low temperature combustion regimes increases formaldehyde emissions at 0.1–0.3 MPa IMEP and 800 rpm [7].

N-butanol has a number of advantages over ethanol when used in gasoline engines, due to its similar physical properties to gasoline and more easily transportation with gasoline through pipelines because of its lower tendency to separate from the base fuel when contaminated with water [8]. In addition, a greater percentage of butanol in its blend with gasoline than ethanol is possible, due to its closer resemblance in air-fuel ratio to gasoline [9]. Therefore, *n*-butanol can also be used as an alternative fuel for spark ignition (SI) engines and was investigated in different kinds of SI gasoline engines. In a single-cylinder cooperative fuels research engine with *n*-butanol-gasoline blends containing 20%, 60% and 100% *n*-butanol by volume, Szwaja and Naber [10] found that the combustion durations for pure *n*-butanol and the *n*-butanol-gasoline blends with 20% and 60% *n*-butanol by volume in the case of stoichiometric mixture are comparable to those burning gasoline. Moreover, combustion stability at part load is slightly lower for *n*-butanol in comparison with gasoline. In a direct injection SI engine, it was found [11] that the addition of *n*-butanol to gasoline decreases combustion duration and peak particle number concentration. But it degrades anti-knock ability. In a port fuel injection SI gasoline engine, it was found [12] that *n*-butanol-gasoline blends decrease engine specific HC, CO, NOx and particle number emissions compared to gasoline. In a single-cylinder carburetor gasoline engine with the blends containing 3%, 7% and 10% *n*-butanol by volume, it was found [13] that with the increase of *n*-butanol, CO and HC emissions decrease. In the meantime, torque and exhaust temperature decrease at the same speed. In a turbocharged direct injection SI gasoline engine with *n*-butanol-gasoline blends with 15%, 30% and 50% *n*-butanol by volume at stoichiometry, it was found [14] that the addition of *n*-butanol to gasoline increases peak in-cylinder pressure and fastens burning rate. Higher *n*-butanol content in the blend results in slightly higher knock possibility at high loads. In a port fuel injection SI gasoline engine with *n*-butanol-gasoline blends containing 20%, 40%, 60% and 80% *n*-butanol by volume, it was found [15] that *n*-butanol-gasoline blend improves combustion stability at lean burn conditions compared to gasoline. However, HC emissions in the case of the blends with 60% and 80% *n*-butanol are higher than those of gasoline. Lattimore et al. [16] found that the *n*-butanol-gasoline blend with 20% *n*-butanol advances the position of 50% mass fraction burned at 0.85 MPa IMEP compared to gasoline. In addition, the *n*-butanol-gasoline blend can effectively reduce accumulation mode particulate number emissions. In a single-cylinder cooperative fuels research engine with port fuel injection, Pechout et al. [17] found that for the blends with 30% and 50% *n*-butanol by volume, the combustion duration at the early stage is slightly shorter and the main combustion duration is generally comparable to gasoline at stoichiometry. In the meantime, flame propagation is faster with increased butanol content in the blend. However, *n*-butanol can improve thermal efficiency of gasoline engines [18]. The engine power and torque is better when the blend with 35% *n*-butanol by volume was used at optimal spark timing [19] compared to gasoline. In a direct injection four-cylinder engine, Cooney et al. [20] found that at high load, efficiency drop in the case of the blend with 85% *n*-butanol relative to gasoline is caused by lower octane rating of *n*-butanol even though knock combustion does not occur. In an optical direct injection gasoline engine, Aleiferis and Behringer [21] investigated the combustion processes of ethanol, *n*-butanol, isooctane and gasoline and found that at similar flame radii between 8 and 12 mm, the burning velocity in descending order is ethanol > *n*-butanol > isooctane > gasoline. Irimescu et al. [22] found that the laminar flame speed of butanol is higher than that of gasoline. For the blend with 40% *n*-butanol and 60% gasoline by volume, the fuel injection at closed intake valves increases diffusion-controlled flames relative to the case of open intake valves while the fuel injection with opening intake valves can decrease ultrafine carbonaceous particles in the exhaust gases [23].

However, little investigation was done on the gasoline engine fueled with *n*-butanol and gasoline through two independent fuel injection systems. In a spark ignition port fuel injection engine with one injector for gasoline and another for *n*-butanol, Venugopal and Ramesh [24] found that simultaneous injection 50% gasoline and 50% *n*-butanol on the mass basis can decrease HC emissions by 13–15% relative to the case of injecting the blend with 50% *n*-butanol and nitric oxide emissions relative to gasoline. Furthermore, the effect of *n*-butanol on torque and efficiency is dependent on throttle position, injection timing and the percentage of *n*-butanol in the total energy released in a cycle [25].

N-butanol can be used in homogeneous charge compression ignition (HCCI) engines in order to improve the thermal efficiency at low loads as a blend with gasoline. However, the addition of *n*-butanol to gasoline advances the autoignition timing of the HCCI engine [26], due to earlier formation and consumption of ethylene and acetylene at low temperature condition [27]. Moreover, *n*-butanol-gasoline blends autoignite earlier than their ethanol–gasoline counterparts with identical oxygen mass content in the blend at the same intake valve opening/exhaust valve closing timings regardless of engine speeds [28]. However, air dilution can slightly delay autoignition timing [29] and is limited by combustion instabilities [30]. Therefore, dependent on engine loads, the mode switch between spark ignition and HCCI combustion is necessary in the driving cycle when *n*-butanol is used in gasoline engines as an addition.

In fact, *n*-butanol and gasoline can be independently supplied to a spark ignition engine with port fuel injection and direct injection systems. In this case, *n*-butanol can be injected into intake ports or cylinders while gasoline is injected into cylinders or intake ports. Therefore, the amount of *n*-butanol and gasoline to be injected into the cylinders and intake ports can be flexibly controlled independently in the light of engine load and speed. To our knowledge, little research can be found in the literature. Actually, fuel injection approaches can alter the time available for the preparation of air-fuel mixture in the cylinder and hence affect the distribution of different fuels in the cylinder and the subsequent combustion processes after spark discharge. Therefore, there is a coupling relationship between different fuels distributed in the cylinder and combustion characteristics when combustion occurs. Understanding the effect of fuel injection approaches on combustion characteristics of a spark ignition engine in different operating conditions is essential to improve its thermal efficiency.

2. Experimental setup and procedures

A four-cylinder four-stroke gasoline engine with a bore of 82.5 mm, a stroke of 84.2 mm and a compression ratio of 9.6 was used. The amount of fuel to be burned in the cylinder can be electronically controlled independently through port fuel injection and direct injection for changing the energy ratio of port fuel injection/direct injection to the total released in a cycle and relative air-fuel ratio (lambda). The experimental setup is shown in Fig. 1.

The engine was coupled to a water eddy current dynamometer for the measurement of torque. With a linear oxygen sensor mounted in the tailpipe, lambda was measured through an ETAS lambda meter with an accuracy of ±1.5%. The flow rates of