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# Combustion performance of dual-injection using n-butanol direct-injection and gasoline port fuel-injection in a SI engine

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In this study, combustion performance of dual-injection using n-butanol direct-injection (DI) and gasoline port fuel-injection (PFI) was evaluated. Dual-injection with various mass fraction of gasoline PFI and n-butanol DI were examined in a single cylinder SI engine operating at 1500 r/min, wide-open-throttle and stoichiometric air-fuel ratio ($\lambda = 1$). Indicated mean effective pressure (IMEP), knock behaviors, cylinder pressure and fuel consumption performance of dual-injection were compared with those of gasoline single-injection. At maximum brake torque (MBT) spark timings, dual-injections can produce higher engine IMEP when compared with gasoline PFI single-injection. Due to the increased engine IMEPs, dual-injections of 80% gasoline PFI-20% n-butanol DI (G80B20) and 50% gasoline PFI-50% n-butanol DI (G50B50) exhibited higher knock propensity and heavier knock intensity. When the mass fraction ratio of n-butanol DI reached to 80%, in-cylinder cooling effects of n-butanol vaporization dominated and led to a decrease of knock occurrence. Through in-cylinder pressure measurement, relatively high maximum combustion pressure with earlier crank angle were observed under dual-injection modes. Increase of the mass fraction of n-butanol DI, dual-injections resulted in higher fuel consumption rates. Nevertheless, comparisons of the indicated specific energy consumption rates of each injection mode indicated that n-butanol gasoline dual-injection had superior fuel conversion efficiencies.


## 1. Introduction

Internal combustion engines contribute significantly to the rising global $CO_2$ emissions and the depletion in supply of fossil fuels. There is an increasing need to revolutionize the energy supply chain in order to reduce the petroleum-based fuels consumption and alleviate the potentially damaging effect of global warming [1]. Biofuels offer a promising option as alternative fuels for the internal combustion engines given their high energy density, renewability and low lifecycle emissions of $CO_2$ [2]. Renewable biofuels can be produced from lignocellulosic biomass feedstock, such as wood, non-edible plants and non-edible residues of food. Increasing use of biofuels in the internal combustion engines provides a great potential to improve the sustainability in transportation. In Europe, it is demanded that all European Union member states must conform to a 10% minimum target on the use of biofuels or other renewable fuels in transportation by 2020 [3]. In the US, tax incentives have been provided to promote the use of bioethanol in gasoline [4]. For large energy consuming developing countries, such as China [5], India and Brazil [6], promoting biofuels will help to improve regional energy security and reduce dependency on imported oil. Therefore, for the automotive sector, it is obliged to design compatible systems with these alternative biofuels and to optimize their use not only in neat form but also in blends with traditional fuels.

Nowadays, when using biofuels in internal combustion engines, most of them are blended with petroleum-based fuels as additives, like ethanol-gasoline blends [7]. Normally, biofuel-gasoline blends are externally mixed with a specified blending ratio, and are injected either into intake port or into in-cylinder chamber. However, the fixed blending ratio and the single-injection strategy precludes the possibility of altering biofuel-gasoline mixture instantly according to the engine requirement. For some fuels, like methanol, has phase separation issues when blending with gasoline given its very small water tolerance [8]. Therefore, it is necessary to develop new injection systems to offer greater flexible and stable biofuel-gasoline blends for internal combustion engines.

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Despite the increased complexity and cost, combination of port fuel-injection (PFI) and direct-injection (DI) offers a possible approach to meet the aforementioned demands. When PFI and DI injection are used simultaneously (dual-injection), biofuel-gasoline blending ratio can be changed immediately by separately injecting different quantities of biofuels and gasoline into engines, thus introducing fuel flexibility and avoiding phase separation problems. Using dual-injection strategy, gasoline can be injected with PFI system, while the biofuels with higher latent heat of vaporization can be injected through DI system. It is beneficial to reduce in-cylinder temperature and combustion temperature due to the more significant cooling effect of biofuels. Consequently, lower in-cylinder charge temperature increases volumetric efficiency [9], decreases NOx emissions and reduces knock propensity [10]. Hence, dual-injection has potential to be one of the promising techniques for better use of biofuels in internal combustion engines in the future.

Typically, the lower heating value of alcohols enlarges as the carbon atom numbers increase and the oxygen contents decrease [11]. n-Butanol is a 4-carbon alcohol, doubling the carbon number and containing 34.2% more energy density when compared with that of ethanol. Main fuel properties of n-butanol and popular gasoline alternative fuel, methanol and ethanol are outlined in Table 1. Relatively high lower heating value of n-butanol helps to reduce fuel consumption and to obtain better mileages. However, a noticeable drawback of n-butanol is the relatively low RON and MON compared with those of methanol and ethanol. Abnormal combustion of knock is more likely to occur when using n-butanol as gasoline substitutes or additives in SI engines.

In order to optimize the use of n-butanol in SI engines, numerous researches have been conducted on studying its combustion characteristics, exhaust emissions and engine performance. Szwaja et al. [14] evaluated engine combustion characteristics by examining n-butanol gasoline blends with 0%, 20%, 60% and 100% volumetric n-butanol in a PFI single cylinder SI engine. Results showed that at the same spark timing, the maximum in-cylinder pressures advanced and raised with the increase of n-butanol blending ratios. It indicated that addition of n-butanol led to shorter combustion duration and earlier combustion phasing characterized by crank angle degree of 50% MFB. Similar results were observed in Dernotte's work [15]. 20%–80% volume of n-butanol in gasoline were tested in a PFI engine under condition of 2000 r/min and 2.62 bar break mean effective pressure. Engine combustion stability was improved and ignition delay was reduced by addition of n-butanol in any rate. Wigg et al. [16] investigated the regulated gas emissions of n-butanol in a PFI SI engine and compared them with neat gasoline. Results demonstrated that n-butanol had benefits in reducing CO and NOx emissions, whereas producing higher HC emissions when compared with those of gasoline. Particle number (PN) emissions of a DI SI engine fueling with 10% and 20% volumetric n-butanol gasoline blends under different exhaust gas recirculation (EGR) rates were studied by Zhang et al. [17]. It was found that the maximum PN concentration gradually decreased and the corresponding distribution shifted toward smaller size with increasing n-butanol content. For knocking combustion performance of n-butanol and its blends in DI SI engines, it was recently reported by the authors of this paper [18]. Furthermore, the authors also performed experiments to study the effects of EGR in combination of different compression ratios and intake pressures on knock behavior of n-butanol [19]. However, previous studies of n-butanol in SI engines were tested either with PFI or DI, separately. There are limited information on further optimizing the use of n-butanol in SI engines with dual-injection system.

Dual-injection technology in SI engines has been proposed and explored by many automotive original equipment manufacturers (OEMs). One of the representative works were conducted by Ford Motor Co. with their 'Ecoboost' turbo-charged DI engines [20]. PFI gasoline and DI E85 ethanol-gasoline blend fuel (15% gasoline and 85% ethanol, by volume) was used to evaluate the effects of dual-injection on engine efficiency and knock suppression at high loads. Others like Toyota Motor Co. and Audi AG have also investigated the potential of dual-injection in SI engines [21]. The results demonstrated that its advantages in improving engine output torque, reducing fuel consumption and emissions under certain engine conditions. Cohn et al. [22] further examined hydrous ethanol and gasoline dual-injection in a boosted SI engine. The hardware modifications in their work were modest and the results showed that ethanol-gasoline dual-injection were in favor of cooling the charge and suppressing the knock. Thanks to the flexibility of dual-injection system, Wang et al. [23] focused their studies on comparing the differences of alcohol PFI combining with gasoline DI between gasoline PFI combining with alcohol DI. The tested alcohols included methanol, ethanol and hydrous ethanol. Catapano et al. [24] studied the effects of dual-injection on engine emissions by using gasoline PFI and ethanol DI in a small SI engine. Recently, a novel gasoline alternative biofuel, 2,5-dymethylfuran has also been involved in the test of dual-injection by Xu and his group [25].

Dual-injection has become an attractive strategy for optimization of biofuel gasoline combustion in SI engines. Among the emerging biofuel candidates, n-butanol is very competitive given its high energy density. However, most of the existed researches of

**Table 1**
Fuel Properties of gasoline and alcohols [12,13].

| | Gasoline[a] | Methanol[b] | Ethanol[a] | n-Butanol[a] |
|---|---|---|---|---|
| Molecular formula | $C_2$-$C_{14}$ | $CH_3OH$ | $C_2H_5OH$ | $C_4H_9OH$ |
| Lower heating value (MJ/kg) | 42.9 | 20.08 | 26.83 | 36.01 |
| Research octane number (RON) | 95 | 111 | 108 | 96 |
| Motor octane number (MON) | – | 88.6 | 90 | 78 |
| Stoichiometric air-fuel ratio | 14.46 | 6.43 | 8.94 | 11.12 |
| Energy density of stoichiometric air-fuel mixture (MJ/kg) | 2.77 | 2.7 | 2.69 | 2.96 |
| Heat of vaporization (kJ/kg) | 373 | 1098 | 838 | 584 |
| Density @ 20 °C (kg/m$^3$) | 744.6 | 791.3 | 789.4 | 791.3 |
| H/C ratio | 1.795 | 4 | 3 | 2.5 |
| Gravimetric oxygen content (%) | – | 50 | 34.8 | 21.6 |
| Boiling point (°C) | 25–215 | 64.5 | 78.4 | 117.7 |
| Auto-ignition temperature (°C) | ~300 | 470 | 434 | 385 |
| Viscosity (mm$^2$/s) at 40 °C | 0.4–0.8 | 0.59 | 1.08 | 2.63 |
| Solubility of compound in water at 20 °C (weight %) | negligible | miscible | miscible | 7.7 |

[a] Reference [12].
[b] Reference [13].