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Effect of intake oxygen concentration on diesel–*n*-butanol blending combustion: An experimental and numerical study at low engine load

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ABSTRACT

The blends of *n*-butanol and diesel are easy to achieve the premixed low-temperature combustion (PLTC) without larger exhaust gas recirculation (EGR) rate to maintain combustion stability and reduce emissions. Engine experiment and numerical modeling coupled with a new reduced mechanism were used to investigate both combustion and emission characteristics of D100 (pure diesel) and B30 (30% *n*-butanol and 70% diesel) under different intake oxygen concentrations (IOCs: 21%, 19%, 17%, and 15%). With the decrease in IOC, the maximum heat release rate (HRR) of D100 increased, while the maximum HRR of B30 first increased and then decreased. NOx emissions of two fuels differed slightly and both significantly decreased. Soot emission of D100 increased, while that of B30 first increased and then decreased. Moreover, under the same IOC, soot emission of B30 was less than that of D100. The chemical kinetic analysis shows that the addition of *n*-butanol consumed a certain amount of OH, slowed down the oxidation of *n*-heptane and toluene, and finally prolonged the ignition delay period. Additionally, decreasing in IOC suppressed the low-temperature O_2-addition reactions of *n*-heptane and *n*-butanol to retard the reaction processes, prolonging the ignition delay. The numerical results show that as the decrease in IOC, the combustion rate decreased, the timings taken for the complete consumption of reactants and the formation of important intermediate products (CH_2O, CO, H_2O_2, and OH) as well as soot precursors (C_2H_2, A_1, and A_4) were significantly delayed. Besides, the maximum mass fractions of CH_2O, CO, and H_2O_2 increased, while the maximum OH mass fraction decreased. The formation of soot precursors exhibited similar trends, and reached the maximum mass fraction at an IOC of 17%. Specifically, the increase in soot emission of B30 was caused by the decrease in oxidation, while the decrease was mainly induced by the decrease in formation.

1. Introduction

The mass application of internal combustion engines (ICEs) leads to environmental pollution, resulting in the implementation of emission regulations in most countries. Additionally, it is very necessary to improve the engine's fuel economy to overcome the pressure of rising oil prices induced by energy shortage. Compared to spark ignition (SI) engines, compression ignition (CI) engines have higher compression ratios and shorter combustion durations, resulting in high combustion efficiencies and fuel-conversion rates. Therefore, CI engines have great potential for improvement. And the growing popularity of CI engines can also be attributed to its reliability. However, unlike the premixed combustion mode of SI engines, the combustion in a CI engine is mainly subjected to the diffusion combustion mode and significantly depends on the mixture condition of fuel and air. Thus local fuel-rich and high-temperature zones are easily produced in a CI engine, causing massive emissions of soot and NOx. To satisfy increasingly strict emission regulations, several novel combustion modes based on the low-temperature combustion (LTC) have been proposed and extensively investigated [1–3].

It is well-known that a trade-off exists between soot and NOx emission in a traditional CI engine. The LTC strategy can suppress the soot and NOx emissions and simultaneously decrease the fuel consumption [4–6]. An LTC model was performed under a lean gas mixture and low-temperature environment, effectively avoiding the production of soot and NOx. Additionally, a longer premixed phase may accelerate the heat release in combustion, and the dissipation in heat transfer also decreases with the decrease in in-cylinder temperature, both improving the thermal efficiency. Exhaust gas recirculation (EGR) is an effective technological method to achieve LTC in a cylinder [7–10]. Using EGR,

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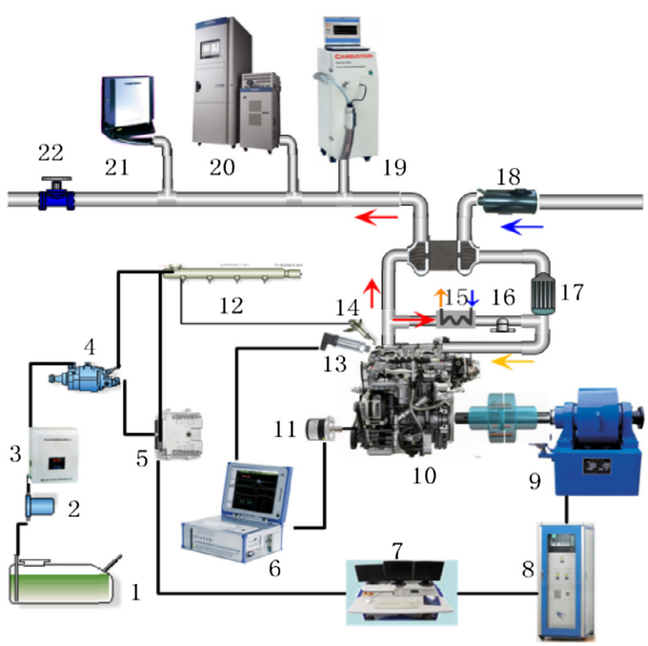
Nomenclature

ICE	internal combustion engine
CI	compression ignition
SI	spark ignition
LTC	low temperature combustion
PLTC	premixed low temperature combustion
HCCI	homogeneous charge compression ignition
DICI	direct injection compression ignition
EGR	exhaust gas recirculation
NOx	nitric oxides
NO	nitric monoxide
N_2	nitrogen
CO_2	carbon dioxide
CO	carbon monoxide
THC	total hydrocarbon
UHC	unburned hydrocarbon
A_1	benzene
A_2	naphthalene
A_3	phenanthrene
A_4	pyrene
PAH	polycyclic aromatic hydrocarbon
D100	pure diesel
B30	30% *n*-butanol + 70% diesel by volume
IOC	intake oxygen concentration
CN	cetane number
CA10	crank angle at 10% accumulated heat release
CA50	crank angle at 50% accumulated heat release
CA90	crank angle at 90% accumulated heat release
VGT	variable-geometry turbocharger
ECU	engine control unit
BMEP	brake mean effective pressure
IMEP	indicated mean effective pressure
HRR	heat release rate
PRR	pressure rise rate
IVC	intake valve closing
EVO	exhaust valve opening
SOI	start of injection
ATDC	after top dead center
CA	crank angle
CFD	computational fluid dynamic
HACA	hydrogen abstraction acetylene addition

the imported exhaust gas does not participate in the combustion, but increases the specific heat capacity of charge, thus reducing the in-cylinder maximum temperature. Moreover, the ignition delay period of the combustible mixture is prolonged, i.e., sufficient period is available for mixing, and therefore the formation of fuel-rich zones can be reduced. Nevertheless, at a high EGR rate, NOx emission can be significantly reduced, but in-cylinder combustion deteriorates, causing a lot of unburned hydrocarbon (UHC) and CO emissions, while the subsequent oxidation of soot is affected. To solve this problem, LTC-related studies mainly focused on the optimization of combustion chamber, fluid characteristics, and fuel injection strategy [11–15]. Recently, the biofuel blending provided the guarantee for the reliability



1: Diesel fuel tank.	2: Fuel filter.	3: Fuel consumption monitor.
4: High pressure fuel pump.	5: ECU.	6: Data acquisition.
7: ECU controller.	8: Dyamometer controller.	9: Eddy-current dynamometer.
10: Diesel engine.	11: Crank angle sensor.	12: Common-rail.
13: Pressure sensor.	14: Direct injector.	15: Heat exchanger.
16: EGR valve.	17: Heat exchanger.	18: Air filter.
19: Cambustion DMS500.	20: Horiba MEXA7100DEGR.	21: AVL 415S smoke meter.
22: Back pressure valve.		

Fig. 1. Schematic diagram of engine setup.