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Ignition control for liquid dual-fuel combustion in compression ignition engines

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ABSTRACT

This paper investigates the characteristics of combustion and emissions for various combinations of liquid fuels under dual-fuel clean combustion modes. The effects of fuel properties on the ignition control are examined along with the optimization of combustion control strategies. With diesel direct injection (DI) as an effective ignition control, the increased use of premixed ethanol in the auto-ignition resistant background can significantly improve the trade-off between the nitrogen oxides (NOx) and the soot emissions. Because of the relatively low reactivity of ethanol fuel, a single-shot diesel micro-pilot injection shows a limited ability to ignite the very lean ethanol mixture at medium loads. When using gasoline to form a relatively higher reactivity premixed background, lower compression ratios and exhaust gas recirculation (EGR) are necessary to avoid the premature auto-ignition that can cause high pressure rise rate and increased NOx and soot emissions. At a compression ratio of 14, ultra-low levels of NOx and soot emissions are achieved simultaneously for diesel-gasoline dual-fuel combustion, with the ignition and combustion phasing being responsively controlled by DI timing. DI butanol is examined as a means of ignition control for dual-fuel combustion. Owing to the relatively lower fuel reactivity, the compression ignition of butanol is difficult and sensitive to the thermal load of the engine and the EGR rate applied. With double-shot butanol DI as the ignition enabler and the primary power producer, butanol-ethanol dual-fuel combustion is realized at an indicated mean effective pressure (IMEP) of 13 bar.

1. Introduction

Low temperature combustion (LTC) is recognized as a viable strategy to tackle the challenge of simultaneously suppressing the NOx and soot emissions in compression ignition engines [1–4]. The LTC requires a more premixed cylinder charge to reduce the soot emissions. Simultaneous reductions in both NOx emissions and soot emissions are achievable, only if the premixed charge is sufficiently lean and diluted. A cylinder charge with suppressed reactivity is preferable in terms of reducing the combustion temperature thus the NOx emissions, as well as alleviating the pressure rise rate if suitable combustion phasing is available. A directly controllable ignition process requires higher reactivity regions to be distributed locally and scheduled temporally within a desired crank angle (CA) window, which is normally engaged by the direct injection (DI) of fuel into the engine cylinder [5,6]. Diesel LTC can be enabled under considerably high EGR rates that prolong the ignition delay for mixing enhancement [7–9]. A high level of intake boost is required to compensate for the oxygen amount. However, the use of a high intake boost risks excessive cylinder pressures that, under high engine loads, can exceed the limit of engine hardware strength [10]. Moreover, the homogeneity required for soot reduction is difficult to achieve as the engine load and the fueling rate increase [11]. The substitution of a portion of the DI fuel, by using port fuel injection (PFI) of a volatile fuel, can alleviate the challenge of in-cylinder mixing for a large amount of fuel. The DI fuel with reduced quantity is necessarily delivered for deterministic ignition control.

Recently, the LTC has been examined through in-cylinder blending of two fuels with different reactivity [12–16]. The dual-fuel combustion uses DI to set off a pilot flame for burning a premixed background fuel mixture that is prepared by PFI. The DI fuel normally has adequate reactivity for deterministic auto-ignition under a high compression ratio (CR), while the background fuel is normally volatile for premixing and with lower reactivity for producing a controllable combustion speed. The ignition process is initiated in the high reactivity regions associated with the direct fuel injection. Owing to the lower reactivity in the background, the combustion process is controllable, in the manner of sequentially

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Nomenclature

Definitions Unit

α	premixed fuel ratio –
CA5	crank angle of 5% mass fraction burnt [°CA]
CA10	crank angle of 10% mass fraction burnt [°CA]
CA50	crank angle of 50% mass fraction burnt [°CA]
CA95	crank angle of 95% mass fraction burnt [°CA]
CD	combustion duration [°CA]
dp/dθ	cylinder pressure rising rate [bar/°CA]
ID	ignition delay [°CA]
IMEP	indicated mean effective pressure [bar]
int. O_2	intake oxygen concentration [%]
LHV_{DI}	lower heating value of DI fuel J/kg
LHV_{PFI}	lower heating value of premixed fuel J/kg
$\dot{m}_{DI}$	fueling rate of DI fuel kg/cycle
$\dot{m}_{PFI}$	fueling rate of premixed fuel kg/cycle
p_inj	pressure of injection [bar]
p_int	pressure of intake [bar]
p_max	peak cylinder pressure [bar]

Abbreviations

CA	crank angle
CR	compression ratio
CO	carbon monoxide
CO_2	carbon dioxide
DI	direct injection
EGR	exhaust gas recirculation
FPGA	field-programmable gate arrays
HC	hydrocarbon
HCCI	homogenous charge compression ignition
HCLD	heated chemiluminescent detector
HFID	heated flame ionization detector
HRR	heat release rate
LHV	lower heating value
LTC	low temperature combustion
NDIR	non-dispersive infra-red detectors
NOx	nitrogen oxides
O_2	oxygen
PFI	port fuel injection
RCCI	reactivity controlled compression ignition
SOI	start of injection
TDC	top dead center

progressing from the higher reactivity regions to the lower reactivity regions. Dual-fuel operations not only broaden the fuel applicability, but also enhance the potential for high efficiency clean combustion in compression ignition engines. Pilot fuel injections can be scheduled early during the compression stroke to modulate the chemical reactivity for either enabling the ignition or shaping the heat release, e.g. in the reactivity controlled compression ignition (RCCI) mode [6,14,15]. A near top dead center (TDC) pilot injection can improve the control over the start of combustion, and hence the overall combustion phasing [17]. The properties of the fuels and the fueling strategies are critically important for enabling LTC and broadening its operating range. Extensive laboratory investigations on LTC have been performed in the authors' lab in the past years [18–22]. A range of practical and promising future fuels and a variety of fueling strategies are tested to explore the pathway to broaden the fuel applications, especially in high load operations under LTC [18]. In this paper, the authors intend to examine the effects of fuel properties on dual-fuel operation in light of achieving LTC with simultaneous low-NOx and low-soot emissions, with a particular emphasis on the ignition control.

A variety of liquid fuels are selected with considerations of the delivery suitability, and the reactivity for ignition and combustion control. The tested DI pilot fuels include diesel and butanol because of their better lubricity, while gasoline and ethanol are used as premixed fuels. The investigation begins with the diesel-ethanol dual-fuel combustion, a combination with the most contrasting fuel properties among the selected fuels. The ignition and combustion processes are studied regarding the effects of premixed fuel ratio, micro-pilot timing, and DI pressure. The examination then moves to diesel-gasoline dual-fuel combustion that utilizes a background fuel with a higher reactivity. The effect of compression ratio is examined along with the application of EGR, for suppression of the auto-ignition tendency and enabling controllable ignition and combustion process for diesel-gasoline dual-fuel operation. The last section of the investigation demonstrates the feasibility of employing DI butanol, a lower reactivity fuel compared to diesel, to ignite the premixed background fuel. The butanol-ethanol dual-fuel combustion is preliminarily examined with DI timing sweeps.

2. Experimental setups

A schematic diagram of the engine test set-up is shown in Fig. 1. Two test engines were employed for the experimental investigations, with the specifications given in Table 1. A commercial four-cylinder diesel engine was configured to perform single-cylinder research with an eddy current dynamometer. Three cylinders of the engine were operated in the conventional diesel high temperature combustion mode to start and then stabilize the engine. The remaining cylinder was comprehensively instrumented for dual-fuel operation. The intake system and the exhaust system of the research cylinder were separated from those of the other three cylinders. The other test engine is a laboratory single-cylinder research engine (SCTE) coupled with a direct current dynamometer. The air management and the fuel delivery systems were similarly configured for both the engines. The engine intake was supplied with conditioned compressed-air, with pressure being precisely controlled to simulate engine boost. The EGR ratio was adjusted with coordinated control over actuation of the EGR valve and the exhaust back-pressure valve.

PFI was implemented through an in-house built fuel supply and conditioning system, with an injection pressure set at 7 bar gauge. Two PFI injectors were installed in the intake runners of the research cylinder in order to deliver a sufficient amount of fuel for high load operation. The injectors were driven by a driver circuit based on a commercial chip LM1949. The DI fuel delivery was realized by a high-pressure common rail fuel injection system, with closed-loop control over the rail pressure. The DI fuel injectors were driven by EFS IPoD drivers. The fuel injection scheduling for both the DI and the PFI fuels were controlled by the real-time controllers with embedded field-programmable gate arrays (FPGA) from National Instruments. The fuels tested were diesel, butanol, ethanol and gasoline. A comparison of the fuel properties is shown in Table 2. A definition of the premixed fuel ratio (α) for the dual-fuel operation is given in Eq. (1).

$$\alpha = \frac{\dot{m}_{PFI} \times LHV_{PFI}}{\dot{m}_{PFI} \times LHV_{PFI} + \dot{m}_{DI} \times LHV_{DI}} \quad (1)$$

where α, premixed fuel ratio; $\dot{m}_{PFI}$, fueling rate of premixed fuel (kg/cycle); $\dot{m}_{DI}$, fueling rate of DI fuel (kg/cycle); LHV_{PFI}, lower