

Active Injection Control for Enabling Clean Combustion in Ethanol-Diesel Dual-Fuel Mode

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ABSTRACT

In this work, an active injection control strategy is developed for enabling clean and efficient combustion on an ethanol-diesel dual-fuel engine. The essence of this active injection control is the minimization of the diffusion burning and resultant emissions associated with the diesel injection while maintaining controllability over the ignition and combustion processes. A stand-alone injection bench is employed to characterize the rate of injection for the diesel injection events, and a regression model is established to describe the injection timings and injector delays. A new combustion control parameter is proposed to characterize the extent of diffusion burning on a cycle-to-cycle basis by comparing the modelled rate of diesel injection with the rate of heat release in real time. The test results show that the proposed parameter, compared with the traditional ignition delay, better correlates to the enabling of low NO_x and low smoke combustion. Particularly in the ethanol-diesel dual-fuel combustion, this new parameter can be controlled by actively adjusting the fuel ratio for each engine cycle. An active injection control strategy is therefore implemented to simultaneously control the engine load, combustion phasing, and the extent of diffusion burning via cycle-to-cycle adjustments on the fuelling rates, fuel ratio, and the diesel injection timing respectively. The control is validated by engine tests, and efficient combustion is achieved up to 19.5 bar IMEP with low NO_x and low smoke emissions.

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INTRODUCTION

The low temperature combustion (LTC) is increasingly recognized as a viable clean combustion strategy for diesel engines towards meeting the ever stringent emission regulations [1, 2, 3]. However, due to the low volatility and high reactivity of the majority of conventional diesel fuels, efficient LTC operation is usually limited to low engine loads [4-5]. The enabling of diesel LTC at higher engine loads generally requires a lowered compression ratio and/or a postponed combustion phasing, which can substantially reduce the engine efficiency [6-7].

The LTC pathway that circumvents the formation of nitrogen oxides (NO_x) and soot is illustrated in Figure 1. The primary factors are the local flame temperature and the local equivalence ratio. Theoretically, the local flame temperature needs to remain below a threshold (e.g. 1800 K) to suppress the NO_x generation, while the local equivalence ratio needs to be fuel-lean to avoid the soot formation [8-9]. In practice, the enabling of LTC requires a lean, diluted, and highly homogeneous cylinder charge to be prepared prior to the combustion events. For LTC enabling with conventional diesel fuels, advanced air and fuel handling is usually necessary to counteract the high reactivity of the diesel fuel so that extra time is gained for better fuel-air mixing prior to the ignition and combustion events [10-11]. While the exhaust gas recirculation (EGR) and intake boost are the primary techniques for engine air management, the fuel injection plays a pivotal role in diesel LTC enabling, and innovative injection

control strategies have been developed to achieve clean combustion in diesel engines [12, 13, 14, 15]. In general, these injection strategies either advance or postpone the injection events away from the conventional injection timing window and create a separation between the injection and combustion events to allow a sufficient time period for fuel-air mixing. In such engine operations, the ignition process of the (partially) premixed cylinder charge is largely decoupled from the injection events, and the combustion rate essentially depends on the kinetics of the chemical reactions [16].

The combustion mode of reactivity controlled compression ignition (RCCI) that uses two fuels with contrasting reactivity for in-cylinder fuel blending adds an extra leverage for modulating the reactivity of the cylinder charge. Extensive RCCI research has been carried out recently with encouraging improvements in engine efficiency and emissions [17, 18, 19, 20, 21]. Yu et al. [18] compared the engine emissions and efficiency for dual-fuel application with the fuel blends on a single-cylinder diesel engine. The results showed that comparable engine efficiency to that of the conventional diesel combustion was achieved in both the dual-fuel and the fuel blend cases while an increase of gasoline substantially improved the fuel-air mixing and resulted in 90% reduction in smoke emissions. Ma et al. [19] studied the diesel injection strategies for gasoline-diesel dual-fuel combustion to optimize the efficiency, emissions, and maximum pressure. Results from these tests indicated that the strategy using an early single-shot

diesel injection along with high gasoline ratios offered ultralow NO_x and smoke emissions and high thermal efficiency. An additional late diesel injection was able to raise the engine load (up to 13.9 bar IMEP) but penalties in high soot emissions were observed. In order to achieve RCCI operation at higher engine loads (e.g. 16~23 bar IMEP), a late timing for the intake valve closing (IVC) and a resultant lowered effective compression ratio are often used in the literature to gain better control of the combustion phasing and thus to avoid excessive pressure rise rates.

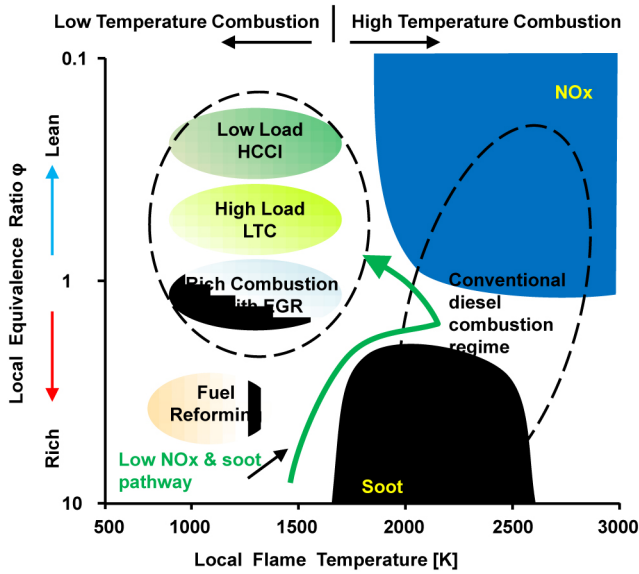


Figure 1. LTC Pathway for simultaneously low NO_x and soot emissions

In terms of the fuelling strategy, the RCCI dual-fuel application operates on a low reactivity fuel (e.g. gasoline or ethanol) introduced at the intake port to create a well-mixed cylinder charge and a high reactivity fuel (e.g. diesel) directly injected into the combustion chamber. The double-shot fuelling strategy is the most commonly adopted for the diesel injection, with the early pilot to raise the reactivity in the squish regions and the second injection to ignite the cylinder charge [22, 23, 24, 25]. It is noted that these diesel injections in RCCI (e.g. pilot@-60° aTDC and main@-30° aTDC) are usually commanded much earlier than those in conventional diesel combustion. The early injection timings raise concerns about the fuel impingement on the cylinder wall and the top land of the piston, and thus low injection pressures are usually applied for a shortened the spray penetration [26]. Moreover, although studies have been performed to investigate the impacts of diesel injection timings on the combustion characteristics and exhaust emissions [27], the criterion for selecting optimal injection timings has not been fully developed so far.

When the early double-injection strategy is adopted, the ignition timing and combustion phasing largely depend on the coordination of the controls over diesel SOI, EGR rates, and IVC timings, as well as engine operating conditions e.g. the fuel ratio, intake temperature and intake pressure [27]. The RCCI strategy improves the ignition control over the homogeneous charge compression ignition (HCCI) by organizing the reactivity distribution across the cylinder charge, despite their similarity in adopting early diesel injections for long ignition delays. For high load operation, RCCI usually requires

assistance from EGR to attain an optimized combustion phasing (e.g. 6~12° aTDC) while adjusting the ratio between the two diesel pilot injections. Although the fuelling control is deemed as the primary measure to modulate the cylinder charge reactivity for RCCI operation, the use of EGR can also effectively affects the overall reactivity, and the combustion sensitivity to EGR variations can incur demanding control challenges [26, 28].

The other widely applied injection strategy for dual-fuel engines uses a single-shot diesel injection near TDC (-15~5° aTDC) that serves as a reliable ignition source to initiate the combustion of the premixed cylinder charge formed by the port injected fuel, intake air, and recirculated exhaust gas [29, 30, 31, 32]. The combustion of the near-TDC injected diesel presents similar characteristics as those of the conventional diesel combustion, and the diesel injection scheduling predominantly determines the ignition timing and combustion phasing. In terms of exhaust emissions, high levels of NO_x are usually observed when the EGR rate is insufficient. Nevertheless, with high fractions of the port injected fuel (e.g. 80% ethanol and 20% diesel), the NO_x emissions can be suppressed via EGR application while minor smoke penalties are observed, thereby substantially improving the NO_x-smoke trade-off for diesel engines [29-30].

For the dual-fuel engine operation, the impacts of adding the secondary fuel to diesel combustion are typically investigated through fuel ratio sweeps. In general, a higher fraction of the port injected fuel (e.g. gasoline and ethanol) leads to improved homogeneity of the cylinder charge and a greater extent of the premixed burning, with penalties in combustion efficiency [21-22, 30-31]. Higher engine efficiency has been observed with the increasing fraction of port-injected fuel due to reduced heat transfer and near-constant volume combustion, as explained in [18, 24, 25, 26]. However, in other studies, slightly reduced engine efficiency has been reported when a high fraction of the port-injected fuel is applied [29-30]. A common consensus from the literatures is that the addition of a secondary fuel (e.g. gasoline and ethanol) at the intake port is beneficial for reducing the NO_x and smoke emissions while producing diesel-like engine efficiency. The selection of an optimal fuel fraction depends on the desired engine performance such as the exhaust emissions, engine efficiency, and combustion controllability, as well as engine operating conditions such as the availability of intake boost and EGR.

In this work, the authors intend to develop and implement an active injection strategy that controls the engine load, combustion phasing, and fuel fraction based on real-time feedbacks for ethanol-diesel dual-fuel combustion. This injection strategy applies ethanol port injection and a single-shot diesel injection near TDC. A new combustion control parameter 'δ' is proposed to characterize the diffusion burning of diesel injection, wherein 'δ' is defined as the temporal overlap of injection and combustion events and it is computed by comparing the rate of diesel injection with the rate of heat release for each engine cycle. The correlation between δ and exhaust emissions is examined through a series of engine tests running in both diesel LTC and dual-fuel modes. The control of the fuel ratio is employed in the dual-fuel operation to modulate the

diffusion burning and minimize smoke emissions. The active injection control is validated and applied in engine tests for enabling clean and efficient dual-fuel combustion at high engine loads.

EXPERIMENTAL SETUP

The research is conducted on a single-cylinder test engine re-configured from a 4-cylinder production Ford diesel. The specifications are given in Table 1. Three cylinders of this engine are operated in conventional diesel combustion mode, while the research is conducted on the single cylinder that has completely separated intake and exhaust manifolds to prevent emission cross-contamination. In addition, in order to deliver ethanol, two gasoline-type injectors (to provide sufficient fuel flow rates) are installed at the intake runners of the single research cylinder.

Table 1. Specifications for the single-cylinder test engine.

Engine Type	Single-cylinder
Displacement [cm ³]	499
Bore × Stroke [mm]	86 × 86
Compression ratio [-]	18.2:1
Maximum cylinder pressure [bar]	180
Diesel injection system	Common-rail (1600 bar)
Diesel injector	Solenoid, 6-hole, hole diameter 0.13 mm, umbrella angle 155°
Ethanol fuelling system	Intake port injection (up to 7 bar abs)

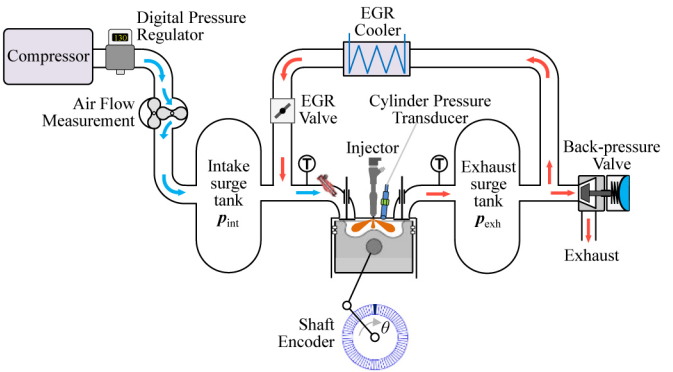


Figure 2. Test engine setup, simulated intake boost control, exhaust backpressure control, and EGR control.

The test engine is integrated into the advanced research platform that offers independent control of the intake boost and EGR, as illustrated in Figure 2. The intake boost is simulated by compressed air supplied from an auxiliary compressor, and the pressure at the intake manifold is controlled by a digital pressure regulator. The air flow rate is measured by an air flow meter (Roots 2M175) installed upstream of the intake surge tank. A pneumatically controlled valve is installed downstream of the exhaust surge tank to control the exhaust backpressure by regulating the exhaust outflow rate. In the EGR loop, tap water is used to cool the EGR and a digitally controlled EGR valve is installed to regulate the flow area of the EGR stream. The final EGR rate is therefore modulated through the control of the EGR valve opening and the differential pressure (different between exhaust backpressure and intake boost) across the EGR valve. During the

tests, the intake temperature is approximately 25°C without EGR, and the intake temperature increases up to 45°C at high EGR rates. The engine is connected to an eddy current dynamometer for engine power and speed management. An optical encoder with 0.1°CA resolution is installed on the engine crank to determine the engine crank angle. An external conditioning unit is employed to circulate the engine coolant and maintain the coolant temperature at 80°C during the engine tests.

Test Fuels

The test fuels are ultralow sulfur diesel (ULSD) and ethanol. The major fuel properties are listed in Table 2. Ethanol has a poor lubricity and a low viscosity for high-pressure direct injection application. The port fuel injection is used for ethanol. Compared with diesel, ethanol offers a superior anti-knocking property (low Cetane number and high Octane number), which helps to reduce the propensity of premature auto-ignition for a premixed fuel-air mixture during the compression process. The diesel injection occurs near TDC and serves as the ignition source to secure reliable ignition and stable combustion. In addition, the high volatility and fuel-borne oxygen of ethanol are deemed beneficial for smoke reduction. The energy density represented by the lower heat value (LHV) of ethanol is however lower than that of diesel, and thus the use of ethanol replacing diesel generally results in an increased fuel consumption for the same energy input into the engine. In this dual-fuel study, the fuel ratio is defined as the energy contribution from ethanol (up to 91%) in relation to the total fuel (diesel and ethanol) energy input.

Table 2. Major properties of test fuels.

Fuel	Diesel	Ethanol
Formula	CH _{1.89}	C ₂ H ₅ OH
Density @ 20°C (kg/m ³)	846	780
Cetane number [-]	46.5	~10
Octane number [-]	~25	~113
Lower heating value [MJ/kg]	43.5	27
Oxygen content by mass [%]	0	34.78
Boiling temperature [°C]	229 – 337 (T ₅ – T ₉₅)	78.3
Lubricity, HFRR WSD [μm]	300@60°C	632@25°C
Kinematic viscosity [cSt]	2.5@40°C	1.52@20°C

Injection Control Setup

The injection control setup is illustrated in Figure 3. A control chassis (NI PXI 1031) embracing a real-time (RT) controller (NI PXI 8110) and two field programmable gate array (FPGA) devices is employed to perform the real-time combustion analysis and to control the fuel injection. These two FPGA devices are Vertex 5 LX85 FPGA (NI PXI 7853R) and Vertex II 3M gate FPGA (NI PXI 7813R), providing 8 analog input channels with independent sampling rate of 750 kHz, and 8 analog output channels with independent update rate of 1 MHz; in addition, the system has 256 configurable digital lines as inputs, outputs, and counters at rates up to 40 MHz. Three direct memory access (DMA) channels are available for ultrahigh-speed data transfer between the RT controller and FPGA device. A high-performance computer is used to host the control software, provide the user

interface, and log the injection control data. Dedicated drivers receive control signals from the control chassis and energize designated actuators for injection control and common-rail pressure control. The cylinder pressure is measured by a piezo-electric pressure transducer (AVL GU13P) and converted to 0~10 Volt analog signal by a charge amplifier (Kistler 5010B, not shown in Figure 3), and it is used as the combustion feedback for the active injection control.

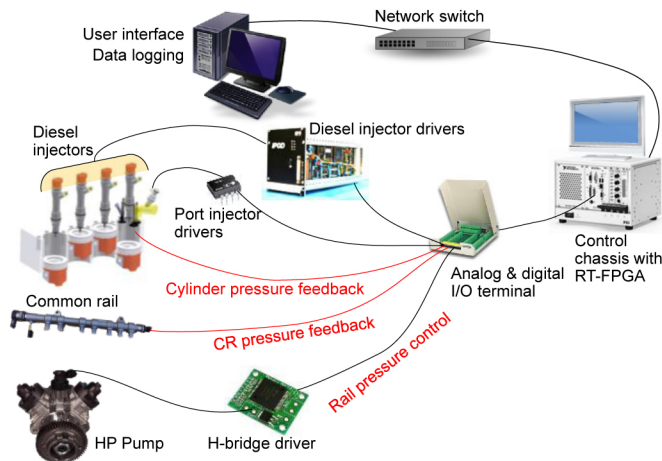


Figure 3. Illustration of injection control hardware setup.

Test Bench for Diesel Injection Characterization

An offline injection bench (EFS 8405 as shown in Figure 4) is employed to characterize the diesel injection events. The test bench includes a complete common-rail injection system and a measurement chamber. The variable parameters are injection duration, the number of injections, injection upstream pressure, and injection downstream pressure (backpressure in the measurement chamber to simulate in-cylinder pressure). The rate of each injection event is measured along with the injection command signal. It is noted that the rate of injection (ROI) essentially characterizes the injection events; however, the direct measurement is usually unavailable on engine applications. As an alternative, the profile of the injector needle lift is commonly used to approximately determine the injection timings (start and end), and efforts have been made to model ROI with an acceptable accuracy [33].

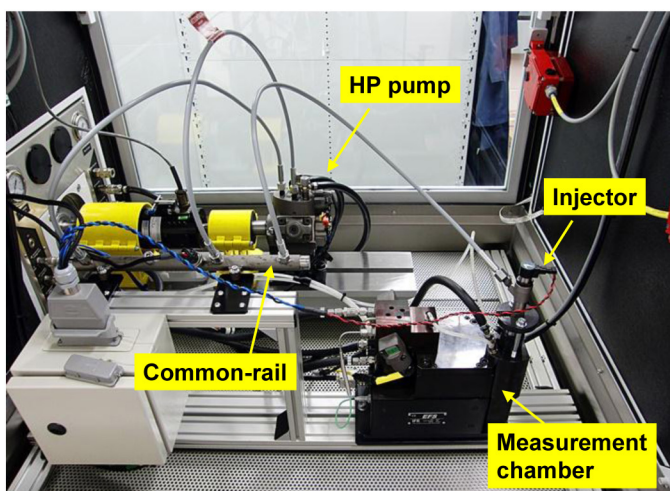


Figure 4. Stand-alone fuel injection bench for diesel injection characterization.

An example of the injection bench results is shown in Figure 5 to demonstrate the differences between the injection command and the actual ROI, along with the corresponding heat release rate of diesel combustion obtained from engine tests using the same injection command. The injection pressure (p_{inj}) is 900 bar and the injection duration is 550 μs . By comparing the injection command and the ROI profile, the tested injector has an opening delay of 230 μs and a closing delay of 610 μs under this particular test conditions. The actual injection duration is 930 μs that is 69% more than the commanded value (550 μs). The long closing delay indicates that a substantial part of the injection process is not captured by examining only the injection command. Compared with the heat release profile, the injection command ends prior to SOC but the actual injection process in fact overlaps with the combustion events in the time domain. By convention, the ignition delay is defined as the time period (or crank angle) between SOC and the commanded start of injection (SOI_{cmd}) / actual start of injection (SOI_{act}), and it is used to assess the extent of premixed burning for diesel combustion. A common consensus is that a longer ignition delay helps to enhance the premixed burning for enabling low NOx and low smoke combustion, however, a quantitative value of the desired ignition delay is rarely reported. In [3] and [10], ignition delay longer than 1ms is suggested to be sufficient for enabling diesel LTC on a single-cylinder research engine. As a matter of fact, a prolonged ignition delay alone, unless being sufficiently long, does not necessarily results in a thoroughly premixed cylinder charge or an elimination of the high temperature diffusion burning. On the contrary, the temporal 'overlap' (as highlighted in Figure 5) between the heat release and actual ROI, to a great extent, represents the diffusion burning duration during which the majority of smoke emissions are considered to be generated. Therefore, compared to the conventional ignition delay, the quantification of this temporal 'overlap' is more relevant to the engine emissions, and the minimization of this 'overlap' (notated as δ) is expected to be effective for smoke emission control.

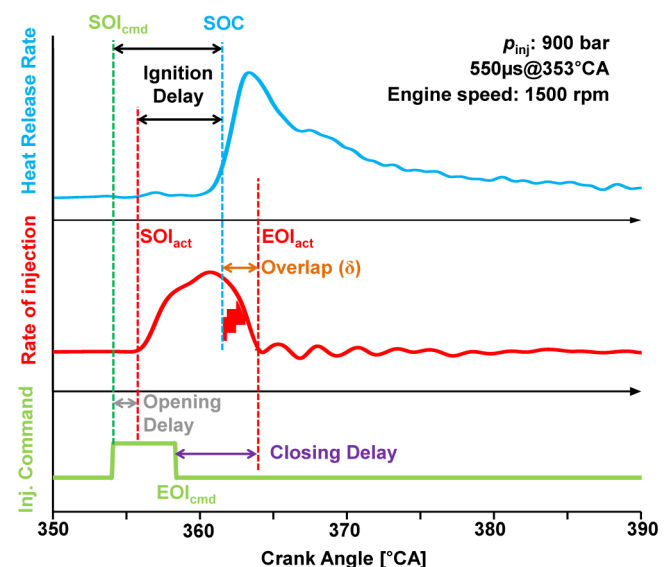


Figure 5. Comparison of injection command, ROI, and heat release rate profiles; definition of SOI_{cmd} , EOI_{cmd} , SOI_{act} , EOI_{act} , SOC, ignition delay and injector opening delay and closing delay.