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An investigation of diesel–ignited propane dual fuel combustion in a heavy-duty diesel engine

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HIGHLIGHTS

- Diesel–propane dual fuel combustion studied on a modern heavy-duty diesel engine (HDDE).
- Limitations to maximum achievable propane substitutions identified over a range of BMEPs (5–20 bar).
- Systematic analysis of combustion heat release, efficiencies, NO_x, smoke, HC, CO emissions.
- One of the first studies of exhaust particle size distributions with diesel–propane combustion in HDDE.
- Fueling strategy and diesel injection timing studies performed at 10 bar BMEP.

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ABSTRACT

This paper presents a detailed experimental analysis of diesel–ignited propane dual fuel combustion on a 12.9-l, six-cylinder, production heavy-duty diesel engine. Gaseous propane was fumigated upstream of the turbocharger air inlet and ignited using direct injection of diesel sprays. Results are presented for brake mean effective pressures (BMEP) from 5 to 20 bar and different percent energy substituted (PES) by propane at a constant engine speed of 1500 rpm. The effect of propane PES on apparent heat release rates, combustion phasing and duration, fuel conversion and combustion efficiencies, and engine-out emissions of oxides of nitrogen (NO_x), smoke, carbon monoxide (CO), and total unburned hydrocarbons (HC) were investigated. Exhaust particle number concentrations and size distributions were also quantified for diesel–ignited propane combustion. With stock engine parameters, the maximum propane PES was limited to 86%, 60%, 33%, and 25% at 5, 10, 15, and 20 bar BMEPs, respectively, either by high maximum pressure rise rates (MPRR) or by excessive HC and CO emissions. With increasing PES, while fuel conversion efficiencies increased slightly at high BMEPs or decreased at low BMEPs, combustion efficiencies uniformly decreased. Also, with increasing PES, NO_x and smoke emissions were generally decreased but these reductions were accompanied by higher HC and CO emissions. Exhaust particle number concentrations decreased with increasing PES at low loads but showed the opposite trends at higher loads. At 10 bar BMEP, by adopting a different fueling strategy, the maximum possible propane PES was extended to 80%. Finally, a limited diesel injection timing study was performed to identify the optimal operating conditions for the best efficiency-emissions-MPRR tradeoffs.

Abbreviations: AHRR, apparent heat release rate (gross); ATDC, after TDC; BDC, bottom dead center; BMEP, brake mean effective pressure; NO_x, brake-specific oxides of nitrogen; CA10-90, crank angle duration between 10% and 90% of cumulative heat release; CA5, crank angle at which 5% of cumulative heat release occurs; CA50, crank angle at which 50% of cumulative heat release occurs; CAD, crank angle degrees; CO, carbon monoxide; dB, decibels; DATDC, degrees after TDC; DBTDC, degrees before TDC; DCAT, Drivven combustion analysis toolkit; D_p, particle diameter; ECU, engine control unit; EEPS, engine exhaust particle sizer; EGR, exhaust gas recirculation; EUP, electronic unit pump; FCE, fuel conversion efficiency (brake); FID, flame ionization detector; FSN, filter smoke number; HC, total unburned hydrocarbons; ID_A, apparent ignition delay; IMEP, indicated mean effective pressure; LFE, laminar flow element; LHV, lower heating value; MPRR, maximum pressure rise rate; PES, percent energy substitution (by propane); PM, particulate matter; SOC, start of combustion; SOI, start of injection (commanded) of pilot fuel; TDC, top dead center; VNT, variable nozzle turbocharger.

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1. Introduction

Despite their high fuel conversion efficiencies, conventional diesel engines suffer from high engine-out particulate matter (PM) and oxides of nitrogen (NO_x) emissions. Current (2010) US EPA emissions standards dictate that heavy duty diesel engines should comply with PM and NO_x limits of 0.013 g/kWh and 0.268 g/kWh, respectively [1]. In addition, intense energy sustainability debates have provided the impetus to investigate alternatives to liquid fossil fuels. In this regard, dual fuel combustion has received renewed interest due to its adaptability for alternative fuels and due to selected performance and emissions benefits compared to conventional diesel combustion [2]. Dual fuel combustion [3–5] is an approach that utilizes a high-cetane (easy-to-autoignite) "pilot" fuel such as diesel [5], biodiesel [6–8], or dimethyl ether [9] to ignite a low-cetane (difficult-to-autoignite) "primary" fuel. While gaseous low-cetane fuels such as natural gas, biogas, hydrogen, etc., have also been considered for dual fuel combustion [10–17], propane is a relatively more attractive option in the United States due to the existing widespread propane distribution network and the ease of storage and transportation of propane in the liquid phase at typical pressures of 100–200 psig (700–1400 kPa). In this regard, some previous studies [18–22] have examined the performance, emissions, and combustion characteristics of diesel–ignited propane (diesel–LPG) dual fuel combustion in both heavy-duty and light-duty diesel engines.

Under certain engine operating conditions, dual fuel combustion can provide superior engine performance and lower NO_x and PM emissions compared to straight diesel combustion. In conventional direct injection diesel combustion [23], NO_x is formed in the high-temperature diffusion flame surrounding the diesel jet while PM is formed in fuel-rich premixed regions (equivalence ratios (Φ) from 2 to 4) throughout the cross section of the diesel jet (especially in the head vortex region). In general, both NO_x and PM emissions from conventional DI diesel combustion can be attributed to high-temperature regions with rich-to-stoichiometric equivalence ratios. However, most strategies for NO_x reduction (e.g., exhaust gas recirculation, EGR) result in higher PM emissions and vice versa, leading to the well-known NO_x-PM tradeoff in conventional diesel combustion. By comparison, with dual fuel combustion, NO_x and PM emissions can be simultaneously reduced by increasing the substitution of the low-cetane fuel, which decreases the size of the high-temperature fuel-rich regions. For example, in conventional dual fuel combustion, the low-cetane fuel is inducted with the intake air forming a lean fuel–air mixture and is ignited by the timed injection of the high cetane fuel near TDC [5]. Since a large portion of the fuel energy arises from combustion of the lean fuel–air mixture, there are fewer locally rich areas, which reduce PM formation [5]. Further, as the low-cetane fuel substitution is increased at a constant load, the diesel fueling rate is reduced. Therefore, smaller diesel sprays result in fewer local high-temperature regions, thereby reducing NO_x emissions [5]. While PM mass emissions are reduced as the low-cetane fuel substitution is increased in dual fuel combustion, some studies on premixed charge compression ignition combustion [24,25] indicate that a reduction in PM mass may also be accompanied by an increase in particle number emissions. More recently, Zhou et al. [26] investigated the particle number emissions and particle size distributions in a micro-diesel pilot-ignited natural gas engine and concluded that both pilot injection timing and pilot diesel mass affect the particle number emissions with larger pilot masses leading to higher particle mass concentrations and lower particle number concentrations.

The NO_x and PM benefits possible with dual fuel combustion are partially offset by higher carbon monoxide (CO) and total unburned hydrocarbon (HC) emissions, resulting from partial oxidation and bulk quenching, respectively [5]. In addition, depending on the type of low-cetane fuel used, dual fuel combustion is often constrained by high pressure rise rates and the incidence of knock at high loads [27] and engine combustion instability leading to partial misfire at low loads [28].

Karim [5] described three stages of normal dual fuel combustion: (1) ignition of the high-cetane pilot fuel, (2) ignition of the fuel–air mixture near the pilot fuel spray, and (3) combustion of the remainder of the primary fuel–air mixture by flame propagation. The ignition delay period and the ensuing combustion processes are affected by the percent energy substituted (PES) by the low-cetane primary fuel (i.e., primary fuel concentration in the cylinder charger) as well as the type of primary fuel used [29]. For example, propane substitution has been shown to decrease ignition delay at high engine loads, while methane substitution only increases the ignition delay [30,31]. In addition, Liu and Karim [32] demonstrated that the most important factors affecting the ignition delay period in dual fuel engines include in-cylinder pressure and temperature histories (that are affected by the PES of the gaseous fuel), pre-ignition energy release, heat transfer to the cylinder walls, and residual gas fraction inside the cylinder. While many studies have examined dual fuel combustion on single-cylinder research engines or multi-cylinder light-duty engines (e.g., [18,22]), only a few researchers (e.g., [19]) have reported diesel–ignited propane combustion results from heavy-duty engines. The present work is an attempt to characterize diesel–ignited propane dual fuel combustion, performance, and emissions over a range of engine loads on a modern heavy-duty diesel engine.

2. Objectives

The objectives of this paper are listed below:

1. To investigate the effect of propane substitution on diesel–ignited propane dual fuel combustion on a heavy-duty diesel engine at different loads and the maximum torque speed with stock engine control parameters.
2. To characterize diesel–ignited propane dual fuel combustion based on cylinder pressure and heat release data, fuel conversion and combustion efficiency measurements, and gaseous and particle emissions results.

3. Experimental setup

The experiments were performed on a heavy-duty six-cylinder turbocharged direct-injection diesel engine, whose details are provided in Table 1. As shown in the schematic of the experimental setup (Fig. 1), the engine was coupled to a Froude Hofmann AG500 (500 kW) dynamometer. Two independent controllers were available for the engine: the stock engine control unit (ECU) and a LabVIEW-based, open-architecture Drivven engine controller. Flexible control of all engine parameters was possible using the Drivven engine controller. The stock ECU was used for most of the experiments reported here, except the final set of experiments that were performed to improve propane substitution and to investigate different injection timings at 10 bar brake mean effective pressure (BMEP). A custom-built ECU harness adapter board was used to enable a seamless transition from using the stock ECU to control the engine to the Drivven controller.

3.1. Steady state data acquisition

Steady state data acquisition included measurements of fuel and air mass flow rates, and temperatures and pressures at various