

Experimental Investigation of Diesel-Ethanol Premixed Pilot-Assisted Combustion (PPAC) in a High Compression Ratio Engine

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ABSTRACT

In this work, empirical investigations of the diesel-ethanol Premixed Pilot-Assisted Combustion (PPAC) are carried out on a high compression ratio (18.2:1) single-cylinder diesel engine. The tests focus on determining the minimum ethanol fraction for ultra-low NO_x & soot emissions, effect of single-pilot vs. twin-pilot strategies on emissions and ignition controllability, reducing the EGR requirements, enabling clean combustion across the load range and achieving high efficiency full-load operation. The results show that both low NO_x and almost zero soot emissions can be achieved but at the expense of higher unburned hydrocarbons. Compared to a single-pilot injection, a twin-pilot strategy reduces the soot emissions significantly and also lowers the NO_x emissions, thereby reducing the requirements for EGR. The near-TDC pilot provides excellent control over the combustion phasing, further reducing the need of a higher EGR quantity for phasing control. The minimum ethanol-to-diesel quantity ratio can be quantified by the 'Heat Release Profile Distribution' parameter that identifies the transition from the diesel pilot dominated heterogeneous combustion to predominantly premixed, homogenous ethanol combustion and allows the two fuel quantities to be optimized for emissions and efficiency. Test results with NO_x-soot emission compliance are presented for the full load range of the test engine using the single-pilot PPAC.

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INTRODUCTION

Low temperature combustion (LTC) strategies have the potential to significantly reduce the oxides of nitrogen (NO_x) and particulate matter (PM) from modern diesel engines [1,2,3,4]. Through precise control over the mixture formation process and air path system management, the clean combustion process can be optimized to minimize the energy efficiency penalty typically incurred as a result of the low combustion temperatures [5, 6]. Since the start of the new millennium, intensive research has been reported on a number of LTC strategies to achieve clean combustion; the most widely reported ones being the modulated kinetics (MK) combustion (smokeless rich combustion) [7], homogeneous charge compression ignition (HCCI) combustion [3, 8] and recently, the dual-fuel combustion of fuels with contrasting properties [1, 4, 9].

LTC strategies fundamentally target a long ignition delay (up to several milliseconds) by separating the fuel injection from the combustion event in order to prepare a well-mixed, lean/EGR diluted fuel-air charge [10]. In the MK combustion, this is realized with a near-TDC single fuel injection along with an aggressive application of EGR (50 to 70%) to provide sufficient mixing time before the onset of ignition while still retaining control over the combustion phasing through injection timing adjustment [7, 11]. While NO_x emissions are typically reduced at moderate EGR levels, achieving ultra-low soot emissions require very high EGR levels that typically results in a narrow engine operating regime, where minor changes in

EGR can result in large spikes in the soot emissions (EGR decrease) or significantly deteriorated combustion efficiency and unstable combustion (EGR increase). The ability to maintain a sufficiently long ignition delay greatly diminishes as the engine load level (fuelling amount) or the engine speed increases. Although, the implementation of MK combustion in production vehicles was done as early as 2004, the proposed MK II combustion that extended the strategy to higher loads and higher speeds was never commercially applied.

In contrast, HCCI combustion commonly utilizes multiple incylinder fuel injections at very early injection timings (up to -100°ATDC) to prepare a near-homogeneous cylinder charge. Whereas the long mixing time results in very low soot emissions, high EGR flowrates (50 to 60%) are still required to reduce NO_x to ultra-low levels in high compression ratio engines. The decoupling of the auto-ignition timing from the injection timing typically results in a pre-TDC combustion phasing (0 to -12°ATDC) and therefore requires complex management and control of the intake charge temperature and EGR rate for combustion phasing adjustment [3, 12]. The advanced phasing also results in very high pressure rise rates and peak firing pressures that limit the attainable engine load levels. As such, no implementation of HCCI combustion has been successfully commercialized.

The modern dual-fuel combustion strategy employs two fuels of contrasting reactivity to manage the cylinder charge preparation and the ignition control under lean, EGR diluted conditions. A low-reactivity, volatile fuel (gasoline and ethanol being the most commonly reported ones) is used to prepare a highly-homogeneous, bulk cylinder charge. The high volatility permits the port injection of the fuel (simpler injection system) while the low-reactivity ensures that auto-ignition does not occur under the diesel-like high compression ratios. The reactivity of the cylinder charge is then modified by introducing a high-reactivity 'pilot' fuel, typically by means of a diesel direct injection (DI) to manage the auto-ignition timing.

Two variations of the dual-fuel combustion exist, depending on the injection timing of the high-reactivity fuel. In Reactivity Controlled Compression Ignition (RCCI) combustion, the cylinder charge reactivity is modified through two early diesel pilot injections (1st pilot: -80 to -50°ATDC; 2nd pilot: -45 to -30°ATDC) to provide adequate control over the combustion initiation and phasing [4, 13]. The mixing time for the last pilot injection is long enough such that the soot emissions are ultra-low. Previous results indicate that moderate to high engine loads are achievable with RCCI combustion while retaining or exceeding diesel-like thermal efficiencies [4]. However, a relatively complex combustion phasing control through pilot timing management 'on-the-fly' is required to enable transient operation.

The other implementation of dual-fuel combustion is the Premixed Pilot-Assisted Combustion (PPAC) that employs a *near-TDC* pilot injection timing to provide diesel-like direct control over the combustion initiation and phasing [9, 14]. While up to 2 diesel pilot injections may be used to achieve the necessary fuel blending for reactivity modification, the timing of the last pilot is always close to TDC. This greatly simplifies the combustion control requirements and provides a linear correlation of combustion phasing with the pilot injection timing that also enables easier handling of transient operation. Ultra low emissions of NO_x and soot are obtained at low-to-mid loads; however, at higher engine loads and speeds, the relatively shorter mixing time for the near-TDC pilot may adversely affect the soot emissions. Therefore, the pilot injection quantity optimization is critical to minimize the impact on the soot emissions.

The implementation of both the MK and HCCI combustion strategies is further hindered by the requirements of flowing large amounts of EGR at elevated intake boost pressures. First, the simultaneous availability of high boost pressure and high EGR flow rates at low loads typical for these combustion modes is usually not possible due to the constraints of the turbocharging systems (energy availability in the exhaust is limited at such engine operations). Second, the effectiveness of EGR in terms of the intake charge dilution is a function of the engine load and intake pressure, and therefore requires a precise control on the intake charge dilution for maintaining stable operation close to the combustion stability limits [15]. Last, the disproportionate response times of the air-path and combustion systems compound the control problem during transient events such as variations in the engine load demand. The current practice is to momentarily close the EGR valve during tip-in or tip-out events since the EGR valve opening for steady-state operation is mostly based on look-up tables calibrated during the development stage, with a few

model-based corrections. Therefore, the implementation of these combustion modes over the speed/load range is difficult without the ability to track and enable the use of EGR during transients [5].

The pathway to a clean-burning, high efficiency engine should optimize both the combustion features and the fuel properties as shown in Figure 1. It can be observed that the PPAC strategy combines the benefits of both the HCCI and MK combustion and thereby, overcomes a number of challenges of MK and HCCI combustion. The long mixing time, together with the low reactivity of the bulk fuel usually minimizes or even eliminates the EGR requirements for *postponing the onset of ignition or to control the combustion phasing*. Since soot emissions are very low, only low-to-moderate amounts of EGR are required to achieve ultra-low NO_x emissions. The load range of dual-fuel combustion is much wider than that for the MK and HCCI combustion, which minimizes the need to carryout *mode-switching* on-the-fly to enable clean combustion across the engine operating map.

Recent LTC research has focused on renewable, oxygenated fuels such as ethanol and butanol for the low-reactivity fuels [16,17,18]. Ethanol fuel has gathered particular interest because of its high volatility that allows it to be injected into the intake port to form a highly premixed cylinder charge and results in significant lowering of the compression-end temperature and pressure (charge cooling effect due to its heat of vaporization). Unlike gasoline (octane number: 87~94) which tends to auto-ignite as the load increases, the low reactivity of ethanol (research octane number: 110~115) allows for its effective use in high compression ratio diesel engines.

Previous work including that by the authors has demonstrated the potential of the PPAC strategy to meet the challenging task of achieving high energy efficiency together with ultra-low NO_x and soot emissions [9, 19]. However, a number of issues have been identified that need to be addressed to make PPAC a viable strategy for implementation in production engines. These issues include a methodology to select the optimum distribution of fuel quantity between the bulk and pilot fuels, the use of a single-pilot vs twin-pilots based on the engine load and smoke limit, the improvement of the combustion efficiency at light loads as well as load range extension and management to cover the full operating range of the engine.

The first part of this research has been carried out on a single-cylinder research engine and is reported here. In this work, detailed empirical investigations of the diesel-ethanol PPAC are carried out at 1500 rpm (consistent with speeds in the heavy-duty vehicle drive cycle) to determine the optimum boundary conditions of PPAC, including the minimum ethanol fraction for ultra-low NO_x & soot emissions, effect of single pilot vs twin pilot strategies on emissions and ignition controllability, reducing the EGR requirements and achieving high efficiency full-load operation. To quantify the minimum ethanol-to-diesel quantity ratio, a '*Heat Release Profile Distribution*' parameter is proposed and validated for identification of the transition from the diesel pilot-dominated heterogeneous combustion to predominantly premixed, homogenous ethanol combustion and for optimizing the fuel quantity distribution between the two fuel quantities for low emissions and high efficiency. Test results for the single-pilot PPAC with NO_x-soot emission compliance are presented for the full load range of the test engine.

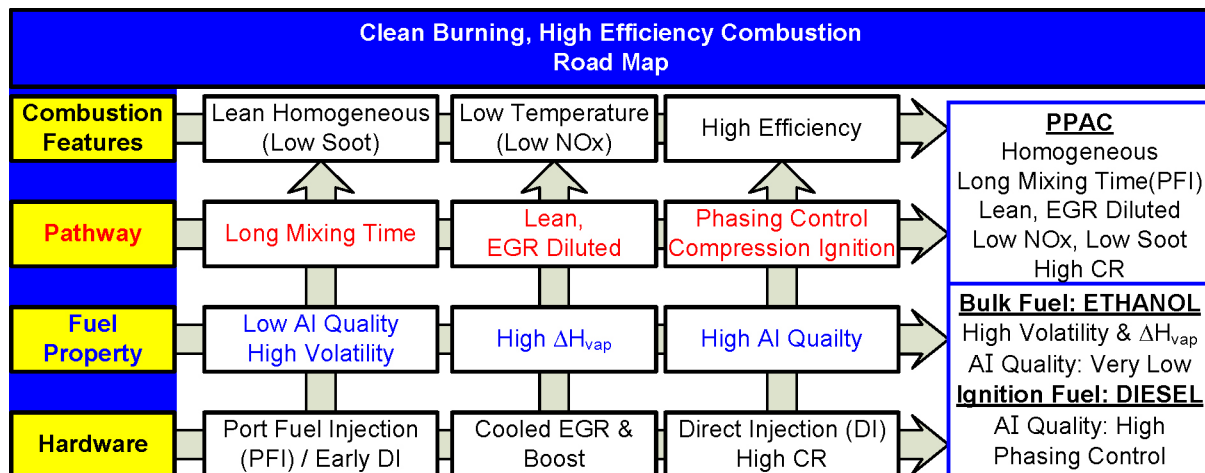


Figure 1. Road Map for Clean Burning, High Efficiency Combustion

EXPERIMENTAL SETUP

The experimental work was conducted on a single-cylinder diesel engine based on a modified 4 cylinder diesel engine and connected to an eddy current dynamometer. The 4-cylinder engine was configured to operate in a 3 cylinders-to-1 cylinder configuration (single-cylinder mode) where 3 cylinders operate in the conventional diesel mode at low load for stable engine operation while the single-cylinder (cylinder # 1) is used for conducting systematic research with independent control of EGR, intake boost pressure, exhaust backpressure and fuel injection timing and quantity [20]. A schematic of the test setup is shown in Figure 2 and the specifications of the test engine are given in Table 1.

Ethanol fuel was injected at the back of the intake valve in the intake manifold using a gasoline port injection (PFI) system, suitably calibrated for ethanol fuel delivery. A constant injection timing of 10°ATDC (during intake stroke) was used for the ethanol injection throughout the tests. The diesel fuel was directly injected into the combustion chamber using the engine's common rail solenoid injector-based fuel system. The specifications for the port-fuel injector and the diesel injector are given in Table 2. No modification was made to the stock piston.

The intake manifold pressure and exhaust backpressure were controlled to ± 0.5 kPa with PID controllers while the EGR valve opening was precisely controlled using the controller area network (CAN), implemented through LabVIEW-based systems. The fuel injection timing and quantity as well the common-rail injection pressure controls were implemented with a FPGA-based real-time deterministic system. The engine coolant temperature was precisely controlled to 90°C with an external conditioning system to minimize the discrepancies in the testing results.

The engine indicating system consisted of an AVL GU13P cylinder pressure transducer mounted through the glow-plug access-hole and connected to the combustion analysis system through a Kistler 510B charge amplifier. The pressure data was resolved at 0.1°CA and 200 consecutive combustion cycles were recorded for each data point. The data was processed both online and offline to compute combustion metrics such as the indicated mean effective pressure (IMEP), start of combustion (SOI), net heat release rate, cumulative heat release and the crank angle of 50% heat release (CA50) [21, 22]. In this work, the pumping work is included in all the calculations and the reported values of IMEP and indicated efficiency are on a net basis.

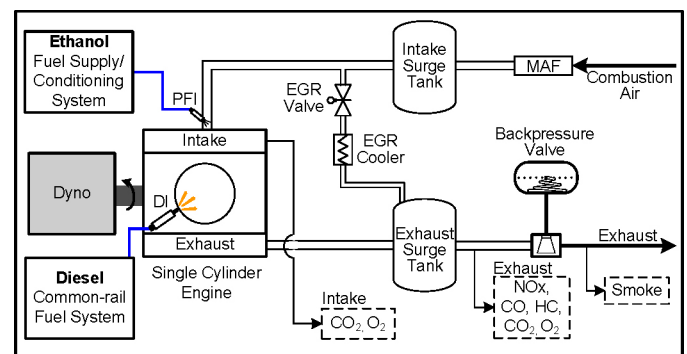


Figure 2. Schematic of the Test Setup

Table 1. Test Engine Specifications

Type	Single Cylinder, 4 stroke DI Diesel
Displacement	499.5 cm ³
Bore x stroke	0.086 m x 0.086 m
Compression ratio	18.2:1
Injection System	Common-rail
Maximum injection pressure	180 MPa
Peak firing pressure	16 MPa (160 bar)

Table 2. Details of the Fuel Injection Systems

	DI	PFI
Type	Solenoid	Solenoid
Number of Holes	6	4
Hole Diameter [μm]	130	-
Included Spray Angle	155°	50°
Injection Pressure [MPa]	60-150	0.4-0.8

The measured exhaust emissions included NO_x, HC, CO, CO₂ and O₂ concentrations while the intake concentrations of CO₂ and O₂ were measured after EGR. The gaseous emissions were averaged for 10 seconds after attaining steady state operation. The particulate matter (PM) in the exhaust was measured using an AVL 415S smoke meter. Hydrocarbon-free, dry combustion air was supplied from a combustion-air conditioning system at a fixed temperature of 24°C and 30% RH. EGR was implemented using a combination of the exhaust backpressure and EGR valve opening. The EGR rate was estimated as the ratio of intake CO₂ concentration to the exhaust CO₂ concentration.

The spectral analysis of the PPAC exhaust was carried out with an MKS MultiGas 2030 Fourier Transform Infrared Spectroscopy (FTIR) continuous gas analyser. The system has a spectral resolution of 0.5-128 cm⁻¹ and performs 1 scan/second @0.5 cm⁻¹. The exhaust sample gas for the FTIR was also supplied from the standard sample conditioning unit (heated sampling line & filter, chiller); therefore, only the light hydrocarbon species have been analysed and reported in this work due to the above mentioned limitation of the exhaust sampling setup. The hydrogen concentration in the exhaust was measured using a V & F H-Sense H₂ gas analyser, based on the electron pulse ionization mass spectrometry principle.

The fuel properties are given in Table 3 [23, 24]. Commercially available laboratory-grade anhydrous ethyl alcohol (ethanol) with a research octane number of 110~115 and research-grade ultra-low sulphur diesel (ULSD) fuel with a cetane number of 46.5 were used for all the tests.

Table 3. Fuel Properties

Fuel	ULSD	Ethanol
Density [15°C, kg/m ³]	846	788
Viscosity [40°C, cSt]	2.5	1.52
Cetane Number [-]	46.5	8 – 11
Research Octane Number [-]	25	110 – 115
Lower Heating Value [MJ/kg]	43.1	27
Oxygen Content [% mass]	0	34.78
Final Boiling Point [1 bar, °C]	340.7	78.3
C-H-O Ratio [-]	C ₁ H _{1.87}	C ₁ H ₃ O _{0.5}
Fuel	ULSD	Ethanol

The quantity of ethanol at any operating point was quantified by the 'Ethanol Fraction' (EF) on an energy input basis and defined as the ratio of the energy contributed by the ethanol quantity to the total energy input from both the fuels.

$$EF = \frac{\dot{m}_{f_{ethanol}} \cdot LHV_{ethanol}}{(\dot{m}_{f_{ethanol}} \cdot LHV_{ethanol}) + (\dot{m}_{f_{diesel}} \cdot LHV_{diesel})} \quad (1)$$

RESULTS AND DISCUSSION

A comprehensive test matrix was setup to investigate and understand the impact of different combustion and fuelling parameters on the emissions and performance of ethanol-diesel PPAC. The parametric analysis was carried out by conducting sweep tests for the ethanol fraction, diesel pilot quantity, EGR and boost pressure. The targeted emission levels were 0.2 g/kWh for NO_x and 0.01 g/kWh for soot across the load range, based on the US EPA 2010 emission standards for heavy-duty on-road compression-ignition engines.

All the testing was carried out at 1500 rpm, consistent with speeds observed in the US EPA 'Not-To-Exceed' and 'European Stationary Cycle' for heavy-duty vehicle drive cycles. The results presented in this work are representative of the trends observed during the experimental work over the complete load range of the engine.

Combustion Characteristics with Ethanol Fraction Variation

The first objective was to determine the optimum boundary conditions of PPAC, including the minimum ethanol fraction for ultra-low NO_x & soot emissions. To this end, an examination of the heat release data revealed that the location of the crank angle of 50% heat release (CA50) within the combustion process correlates well with the changing quantity of ethanol. Based on these analyses, the authors have defined a 'Heat Release Profile Distribution' (HRPD) parameter as follows:

$$HRPD = \frac{\theta_{CA50} - \theta_{SOC}}{\theta_{EOC} - \theta_{SOC}} \quad (2)$$

The term ($\theta_{CA50} - \theta_{SOC}$) represents the time required for accumulation of heat release up to CA50 and normalizing this term by the total combustion duration ($\theta_{EOC} - \theta_{SOC}$) provides a measure of the rate at which heat is released during the early phase of combustion. The values of the heat release parameters are calculated from the online cycle-by-cycle heat release analysis. The *start of combustion* (SOC) is taken as the first zero-crossing after the pilot *start of injection* (SOI) timing and the *end of combustion* (EOC) is taken as the crank angle at which 90% of the total heat has been released.

The results of an EF sweep test are presented in Figure 3. The sweep was started from EF: 0.3 and EGR was increased to 43% to reduce NO_x to almost 0.2 g/kWh. The ethanol fraction was then progressively increased (while reducing the pilot quantity) to maintain the IMEP at 9.8 bar. EGR was held constant and the combustion phasing was maintained at 7°ATDC throughout by adjusting the pilot injection timing to account for the changing ignition delay with increased EF.