

Effect of Swirl Ratio and Wall Temperature on Pre-Injection Chemiluminescence During Starting of an Optical Diesel Engine

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ABSTRACT

Fuel wall impingement commonly occurs in small-bore diesel engines. Particularly during engine starting, when wall temperatures are low, the evaporation rate of fuel film remaining from previous cycles plays a significant role in the autoignition process that is not fully understood. Pre-injection chemiluminescence (PIC), resulting from low-temperature oxidation of evaporating fuel film and residual gases, was measured over 3200 μsec intervals at the end of the compression strokes, but prior to fuel injection during a series of starting sequences in an optical diesel engine. These experiments were conducted to determine the effect of this parameter on combustion phasing and were conducted at initial engine temperatures of 30, 40, 50 and 60 °C, at swirl ratios of 2.0 and 4.5 at 1000 RPM. PIC was determined to increase and be highly correlated with combustion phasing during initial cycles of the starting sequence. This suggests that under starting conditions, combustion phasing is determined by wall film evaporation rates and residual concentrations rather than the fuel injection event. As surface temperatures rise further, wall film evaporation accelerates, and increasing wall film fractions evaporate prior to the end of combustion. As evaporated fuel film is consumed by combustion, PIC decreases. The injection event begins to control the combustion process only when a significant portion of the delivered fuel is evaporated within the combustion period. The sensitivity of combustion phasing to PIC is observed to increase at reduced temperatures, indicating the greater influence of wall film evaporation rates on starting under reduced temperature conditions. Comparison to combustion phasing sensitivity predicted by kinetic models is consistent with the autoignition characteristics of lean

mixtures. Combustion phasing was found to be more sensitive to PIC at higher swirl levels under all temperatures.

INTRODUCTION

Control of the combustion process under the conditions encountered during the starting of a diesel engine remains a difficult challenge. Such conditions are often not conducive to autoignition, and they must be overcome while maintaining performance and emissions requirements. There is a need for a further understanding of the fundamental phenomena governing cold-starting, so that control of this process may be optimized in next-generation engines.

The cold-starting of a diesel engine presents many difficulties. Heat losses are increased since typical surface temperatures are below values found during steady operation. Lubricating oil viscosities are high and battery discharge currents low, resulting in lower cranking speeds. Together with the enhanced blowby from the greater clearances that exist between piston-ring assemblies and cylinder surfaces, charge temperatures are reduced, diminishing reaction rates.

Another degree of variability is added by the presence of residual gases of varying composition. Comprising unburned fuel evaporated from wetted wall surfaces and both complete and partial combustion products remaining in the cylinder from previous, misfiring or partially firing cycles, this composition has been shown to influence the kinetics, autoignition and combustion phasing of the subsequent engine cycle [2,5-9].

The lower injection pressures that may be produced at cranking speeds result in poor spray atomization and mixture preparation. Fuel wall impingement is a particular concern during the starting of small bore engines, where reduced surface temperatures result in unpredictable, uncontrolled and reduced evaporation rates. The effects of parameters including swirl, coolant temperature and bowl geometry on deposited fuel mass during diesel engine starting have been measured experimentally by Tsunemoto *et al* [10]. The percentage of fuel mass adhered to the wall was found to *increase* at lower engine speeds, lower fueling rates, lower temperatures, lower compression ratios and lower swirl rates, provided the combustion bowl to piston diameter ratio is not excessively low. Fuel injection timing where spray was propagated through a burning flame was found to *reduce* the percentage of fuel deposited on the wall.

Various parameters affecting wall film evaporation rates from a single component fuel have been examined by detailed modeling by Oliveira and Hochgreb [11]. These results indicate a complex process including both flash evaporation of fuel film, and its condensation over the course of the expansion and compression strokes of an engine cycle. Increasing wall temperatures and liquid boiling point temperatures, and decreasing heats of vaporization were found to increase evaporation rates. A critical fuel film thickness, or penetration depth, was identified, below which fuel evaporation rates sharply increase [11]. French *et al* have measured fuel film evaporation rates by shadowgraph and schlieren techniques [12], Montorsi *et al* have recorded temperature and fuel film thickness histories resulting from a wall-impinging single and split diesel spray on a temperature-controlled wall in a spray chamber [13], and Kashdan has experimentally observed fuel film lifetimes in an engine by a tracer LIF technique [14]. In the latter work, by application of a naphthalene tracer to a n-dodecane/n-tetradecane fuel in an optical compression ignition engine, the more volatile naphthalene was found to persist as a fuel film well into the expansion stroke. A film of the less volatile, n-tetradecane, was observed to persist on the combustion bowl surfaces into the exhaust stroke and evidence of flash boiling during piston expansion observed [14].

Wall fuel film formation processes have been investigated by Stanton and Canton [15] by application of the continuity and momentum equations in a KIVA-II code simulating referenced experiments. Droplet stick, rebound, spread and splash impingement regimes were considered where the wall temperature is below that of the fuel boiling point and pure adhesion temperature. Swirl was also found to influence the spread of the fuel film and wall vortices were found to redirect spray droplets in directions away from the wall [15]. Thermal and evaporative processes were subsequently included by the same authors in a later work [16]. Fuel film thicknesses of a n-dodecane and number 2 diesel fuel (DF2) were calculated, which were found to vary in time and with load at wall temperature of 400 °C. For the 150

mm bore engine configuration at full load, the fuel-wetted fraction of the piston crown area is reported to exceed 60% [16]. The methodology of the previous model has been further utilized by Nakama *et al* to investigate the effects of fuel temperature on wall film thickness in a gasoline-fueled direct injection engine [17]. The experimentally-confirmed results indicate that the late-cycle wall film mass may be decreased from near 80 % to 35% by increasing fuel temperature from 300 to 423 °K [17].

While liquid film development has been studied extensively, the effects of film evaporation on engine performance, particularly during starting, remain largely unknown. This work reports experimental results gathered from a series of engine starting sequences over a range of initial engine temperatures and two swirl ratios. During each starting sequence, ultraviolet chemiluminescent emission was measured at the end of the compression stroke prior to the fuel injection event. The relationship of this quantity, believed to be strongly linked to wall film evaporation, was analyzed and related to the combustion phasing of the subsequent engine cycle during the starting sequences. The effects of initial engine temperature and swirl ratio on this relationship are examined.

EXPERIMENTAL SETUP

ENGINE – The engine used in this investigation is a single-cylinder, direct-injection AVL Model 5402 optical-access research engine with a bore of 85 mm, stroke 90 mm and resulting displacement of 510 cm³. The geometric compression ratio has been measured to be 15:1. The intake ports are equipped with a swirl control valve, allowing the control of swirl ratios between 2.0 and 4.5. This engine is equipped with a Bosch CR CP1 common-rail fuel injection system rated to 1350 bar injection pressures. Engine and injection system specifications are shown in Tables A1 and A2 of the Appendix.

Optical access is provided via a classic Bowditch extended piston in which is mounted a 20 mm-thick fused-silica (quartz) window, as shown in Figure 1. The placement of a mirror inside the piston assembly at 45° relative to the cylinder axis provides a view of the entire 40 mm-diameter combustion bowl. The combustion bowl is of rectangular cross-section, and the flat surface geometry of the combustion chamber is interrupted only by valve pockets placed in the piston bowl rim and valve seat chamfers in the cylinder head. Oil-less lubrication is provided by slotted graphite rings and the sealing by uninterrupted bronze-teflon rings. Compressed air directed at the underside of the piston window provides cooling of this assembly during operation.

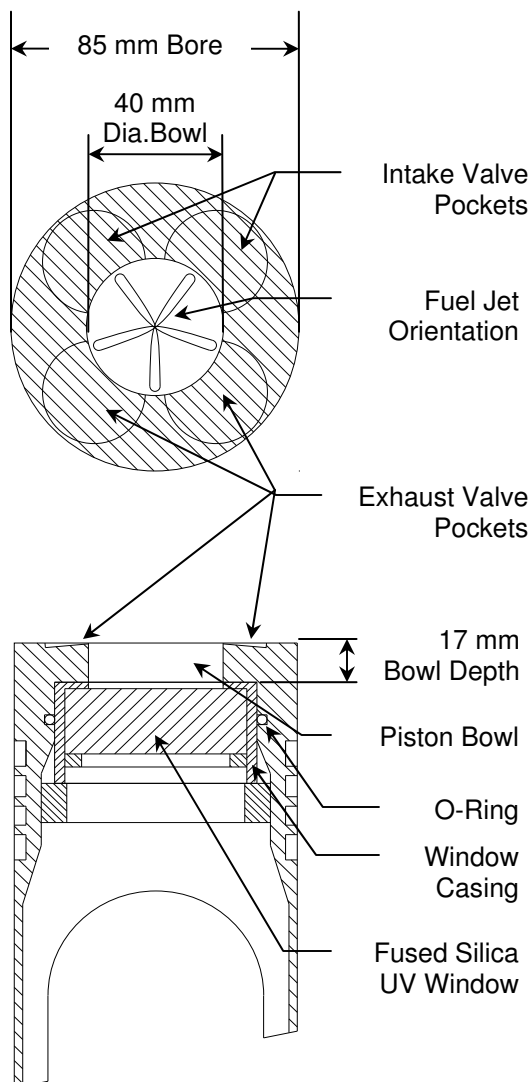


Figure 1. Extended piston and combustion bowl geometry.

ENGINE INSTRUMENTATION – To measure cylinder pressures, a non-cooled piezo-electric pressure transducer (AVL GH 12 D) is mounted through an opening in the cylinder head. A needle lift sensor (MicroEpsilon) is installed in the fuel injector to detect the displacement of the injector needle. An additional pressure transducer measures the pressure in the line between the fuel rail and injector. Sensor signals are recorded by a high-speed data acquisition system triggered by an optical encoder wheel mounted on the engine crankshaft. This instrumentation is used to calculate the Apparent Rate of Heat Release (ARHR) using the analysis provided by Heywood [18], and is shown normalized by the engine displacement. A more detailed description about the engine and instrumentation has been reported earlier [19].

Chemiluminescent images of the combustion chamber were captured through the piston window using an intensified charge-coupled device camera (ICCD), (Princeton Instruments PI-MAX2) optimized for operation in the ultraviolet spectrum. The intensifier has a multi-

alkali photocathode whose nominal sensitivity spans the range from 125 to 900 nm. This range effectively covers the chemiluminescence spectrum of species such as HCHO, OH, CH, C₂ and CO₂ [20] that occurs in the early stages of diesel autoignition [21], as well as the yellow-orange thermal emission of sooty combustion.

A 3-element, ultraviolet lens (Electrophysics) of 78 mm focal length and maximal aperture f/3.8 was used in these experiments. This lens exhibits a great degree of chromatic aberration. To maintain repeatability of the focal plane between experiments, the lens was focused through a 405 nm narrow-band filter, and the filter subsequently removed prior to data collection.

Figure 2 shows the experimental layout with the ICCD aligned with the axis of the extended piston through the 45° mirror. In such a configuration, the camera field of view, as represented by the white circle in Figure 3, covers the 40 mm combustion bowl. Swirl motion is in the clockwise direction.

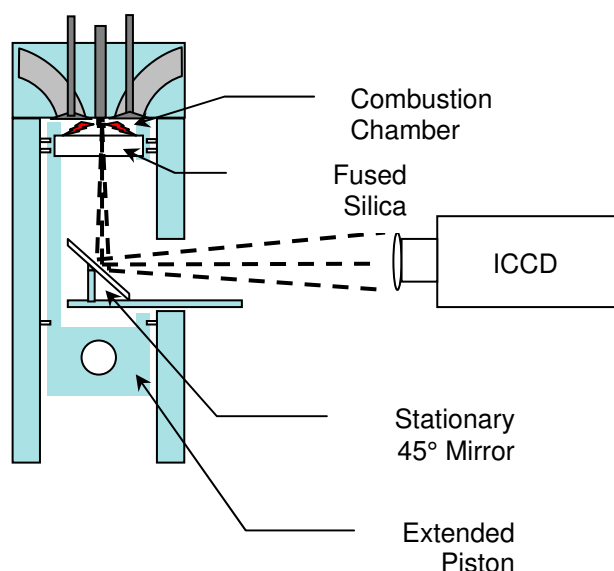


Figure 2. Experimental setup and optical access via Bowditch piston.

ENGINE OPERATING CONDITIONS – The work reported here consists of measurements taken from a series of engine starting sequences with initial cylinder head and block coolant temperatures of 30, 40, 50 and 60 °C at swirl ratios of 2.0 and 4.5. Prior to each starting sequence, the engine was conditioned for two hours by the circulation of coolant and lubricating oil through the respective circuits at the prescribed temperature. At the end of the conditioning period, the engine was motored to a constant steady speed of 1000 RPM, at which time data collection, imaging and fuel injection were begun. The engine speed was held constant by the dynamometer, and fuel injection, imaging and data collection were continued for approximately three

hundred cycles. The experimental engine speed of 1000 RPM was chosen as the minimum value below which resonance of the engine-dynamometer driveshaft is expected.

For all experiments, the fuel was administered in a single injection at an injection pressure of 400 bar, to represent less than typical operating values as may be encountered in the common rail during cranking. The injector command signal was constant at 615 μsec , corresponding to an estimated fuel delivery rate of 11 mg/cycle. The start of injection was fixed and chosen to occur at 10 °bTDC, as a pre-determined optimum that both minimizes the number of misfiring cycles during a starting sequence, and provides a sufficiently long exposure window to capture a pre-injection chemiluminescence signal as explained further below.

The engine is naturally aspirated and inlet pressure during all experiments was atmospheric. The inlet air temperature was maintained at ambient laboratory conditions near 300 °K for all cases examined.

FUEL – Commercial-grade, ultra low-sulfur diesel (ULSD) fuel having the specifications given in Table A3 of the Appendix was used during experiments. The atmospheric pressure distillation curve for this fuel is shown in Figure 4.

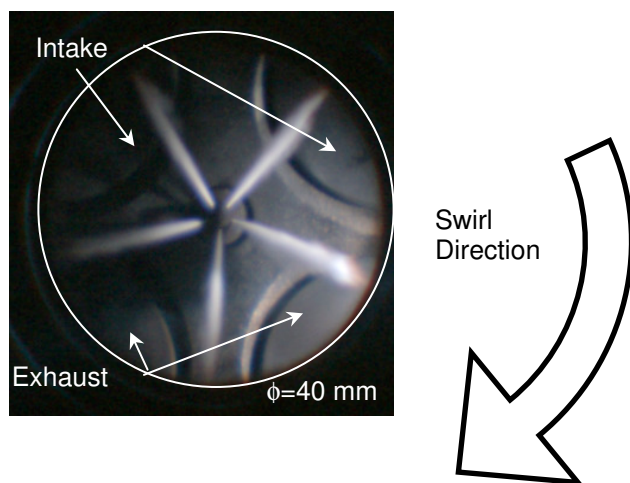


Figure 3. Camera field of view relative to combustion chamber and spray orientation.

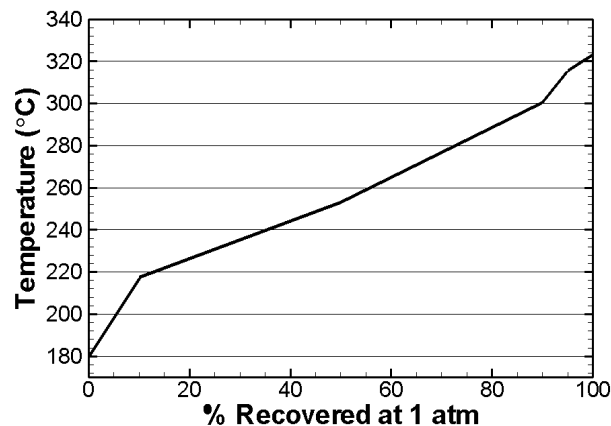


Figure 4. Atmospheric pressure distillation curve for ULSD fuel used during experiments.

PRE-INJECTION CHEMILUMINESCENCE

The measurement of pre-injection chemiluminescence (PIC) has been described earlier [2,22]. During these experiments, unfiltered, broadband, intensified ultraviolet (UV) images are recorded through the transparent piston at the end of the compression stroke, prior to the beginning of fuel injection. The images presented in this work are taken over an exposure period 3200 μsec , or approximately 19 crank-angle degrees (CAD), which are synchronized to begin at 30 °bTDC. Despite the generous exposure period, due to the low intensity of the source light emission it was necessary to use the full gain setting of the CCD intensifier. A single PIC image is obtained from each engine cycle.

Figure 5 illustrates the relative timing of the camera exposure period relative to the cylinder pressure, ARHR, and injector needle lift signal. The data shown is considered typical of a full-firing cycle of the starting sequence, at a coolant temperature of 40 °C and a swirl ratio of 4.5. The crank angle at which 50% of the total apparent heat release for the cycle occurs (CA50) is calculated from the pressure trace and is used as a combustion phasing metric.