

In-Cylinder Particulate Matter and Spray Imaging of Ethanol/Gasoline Blends in a Direct Injection Spark Ignition Engine

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ABSTRACT

A single cylinder Direct Injection Spark Ignition (DISI) engine with optical access was used to investigate the effects of ethanol/gasoline blends on in-cylinder formation of particulate matter (PM) and fuel spray characteristics. Indolene was used as a baseline fuel and two blends of 50% and 85% ethanol (by volume, balance indolene) were investigated. Time resolved thermal radiation (incandescence/natural luminosity) of soot particles and fuel spray characteristics were recorded using a high speed camera. The images were analyzed to quantify soot formation in units of relative image intensity as a function of important engine operating conditions, including ethanol concentration in the fuel, fuel injection timing (250, 300 and 320 °bTDC), and coolant temperature (25 °C and 90 °C). Spatially-integrated incandescence was used as a metric to quantify the level of in-cylinder PM formed at the different operating conditions. The experiments were conducted at stoichiometric conditions, fixed engine speed of 1500 RPM, a load condition of approximately 5.5 bar IMEP_n, with a fixed intake manifold absolute pressure of 76 kPa. Significant reduction in in-cylinder soot formation was observed with the higher ethanol content in the fuel, regardless of fuel injection timing. Fuel impingement was documented in the spray imaging, and fuel impingement was a large factor affecting the level of PM emissions. Retarded fuel injection timing reduced the PM formed in-cylinder for the two fuel blends and the baseline gasoline. Higher coolant temperatures reduced liquid fuel on piston and cylinder wall surfaces, and therefore also reduced in-cylinder PM formation. The in-cylinder crank-angle-resolved data presented in this study are the first of their kind to document in-cylinder PM formation of ethanol fuel blends. The results provide insight on how ethanol fuel blends can be used to reduce DISI PM formation at varying levels of ethanol concentration in the fuel.

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INTRODUCTION

Fuel economy and CO₂ emissions advantages make direct injection (DI) engines attractive for passenger car and light duty truck fleets. Direct injection of the fuel into the combustion chamber decreases the charge temperature and thus increases the volumetric efficiency of the engine and reduces the knock potential at higher compression ratios [1]. DI engines have higher thermal efficiency and higher power output than the port fuel injected gasoline counterparts. DI also improves the transient response of the engine; however, DI spark ignited engines can have higher particulate matter (PM) emissions compared to gasoline port fuel injection or diesel engines equipped with diesel particulate filters [2], [3].

Important factors affecting soot formation are the local equivalence ratio, and temperature, and chemical structure and thermo-physical properties of the fuel [4]. Methods to reduce PM emissions from DISI engines are important, and ethanol fuel blends may be a means to reduce PM emissions, while maintaining DI engine performance.

Ethanol can be produced from renewable resources, and ethanol may provide some CO₂ emissions benefit depending on the fuel processing and feed stocks used. Hill et al. [5] quantified and monetized the life cycle climate change and health effects of greenhouse gas and PM emissions from gasoline and advanced bio-ethanol. For each billion ethanol-equivalent gallons of fuel produced and combusted in the U.S., the combined climate-change and health costs are 469

million U.S. dollars for gasoline, but only 123-208 million U.S. dollars for cellulosic ethanol depending on feed stock (prairie biomass, miscanthus, corn stover, or switchgrass).

The addition of alcohols to diesel fuel can considerably decrease the PM emissions of diesel engines [6], [7], but the effects of ethanol blends in direct injection spark ignited (DISI) engines are less well understood and some results are contradictory. Chen et al. [8] concluded that an increase in ethanol addition leads to an increase in PM both in particle number and mass, but a decrease in PM emissions has been demonstrated by Francqueville [9] and Ericsson et al. [2] over the entire driving cycle for higher (E85) content of ethanol in the fuel. Storey et al. [10] showed decreased PM emissions for lower content of ethanol in the fuel (E10-E20), while He et al. [11], [12] concluded that low ethanol content in fuel exhibits almost the same particle emissions as gasoline.

There are significant differences in important thermophysical properties of ethanol and gasoline that may be the cause of the differences observed in these studies. Ethanol has a significantly higher kinematic viscosity and enthalpy of vaporization compared to gasoline. The higher viscosity results in lower Reynolds numbers and thus smaller turbulence intensity affecting the spray breakup. The high enthalpy of vaporization leads to higher localized charge cooling with ethanol. Higher surface tension of ethanol results in lower Weber numbers, which slow the breakup process [13]. The combination of these properties can affect fuel spray penetration, break up and mixing; however, in the study by Park et al. [13], the authors concluded the axial spray tip penetration and droplet size of E85 fuel blends were similar to gasoline.

Other factors are important in creating the local conditions which lead to high soot formation rates. Temperature affects the spray characteristics of the fuel. Differences in the spray pattern of different fuels have been demonstrated by Serras-Pereira et al. [14] in a motoring engine. However the effects of residual heat from combustion were not identified in the study. The timing of the fuel injection affects the homogeneity of the charge. Earlier SOI can increase fuel impingement on the piston bowl and thereby produce a film of liquid fuel on the piston top which can yield local fuel rich areas and pool fires. Barone et al. [15] concluded that DISI particle morphology was a function of fuel injection timing, with fewer liquid droplets and more single solid sub-25 nm spheres generated by retarded injection.

Experiments to capture time resolved in-cylinder imaging of soot formation and the corresponding fuel spray properties have not been previously conducted for ethanol/gasoline fuel blends. The objective of this work was to understand the fundamental effects of ethanol blends on fuel spray properties and in-cylinder soot formation. An optically accessible single cylinder engine was used to acquire high speed imaging data of fuel spray and PM formation in the engine as a function of the ethanol content in the fuel, the fuel injection timing, and engine coolant temperature. These data provide the first

direct insight into the physical mechanisms controlling PM formation in ethanol fuel blends in DISI engines.

EXPERIMENTAL APPROACH

The study was conducted using one of the single-cylinder, optically-accessible research engine facilities at the University of Michigan. A schematic of the engine set-up is shown in Figure 1. The engine cylinder head hardware was based on a modern 4-valve gasoline direct injection engine geometry. Axial and orthogonal optical access was available through piston window inserts and transparent cylinder liners. For this study, side-view imaging through the transparent cylinder liner was used with a metal piston (with no optical window) that included a bowl geometry that helps fuel preparation at certain conditions. The aluminum cylinder head featured a side-mounted fuel injector with dual overhead camshafts and 4 valves around a centrally mounted spark plug. The engine geometry used a $\phi 89.0$ mm bore and 81.4 mm stroke, yielding 0.51L displacement with nominally 9.4:1 compression ratio. The single cylinder engine design is capable of replicating intake and exhaust valve event timings consistent with a twin-independent variable camshaft timing system, and therefore the valve timing used for all experiments was intake valve opening (IVO) at 11° after top dead center (aTDC) and exhaust valve closing (EVC) at 38° aTDC. A partial-length cylinder liner with coolant jacket was used with a transparent fused silica cylinder liner (height of 25 mm) to enable orthogonal imaging of the combustion chamber.

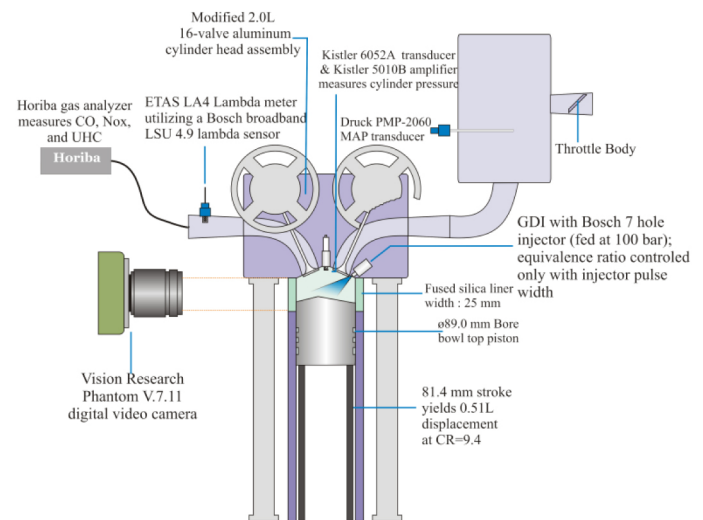


Figure 1. Engine setup schematic

In-cylinder pressure measurements were made using a piezoelectric transducer (Kistler 6052A) and charge amplifier (Kistler 5010B), and intake pressure was monitored using an absolute pressure sensor in a water cooled fitting (Kistler 4045A2) and amplifier (Kistler 4618). Manifold absolute pressure was also measured using an amplified output pressure transducer (GE-Druck PMP-2060). The crank angle

was encoded using a BEI encoder (H20EB-37-F12-SS-360) with 360 signals per revolution. The timing of top dead center was measured with a TDC marker sensor. The signals were used for engine speed control, fuel injection timing, and spark timing. The pressure and crank angle encoder data were acquired using a data acquisition system (National Instruments Compact cDAQ) system operating at 60 kHz. The in-cylinder pressure and the piston phasing data were acquired after the engine was thermally equilibrated and was at a stable speed. The data were acquired for 5 seconds, corresponding to approximately 60 combustion cycles (at 1500 RPM) at each experimental condition.

The engine was driven by a hydraulic dynamometer (Micro-Dyn 35) equipped with a control system which regulates the engine speed by compensating between providing load or absorbing net power output. The engine speed was set at 1500 RPM for this study. The engine control module for spark and fuel injection is integrated with the dynamometer control system. Stoichiometry was controlled by varying the fuel injector driver pulse width (PW) duration while setting a constant fuel rail pressure of 100 bar. All experiments in the study were operated at stoichiometric conditions and a load condition of approximately 5.5 bar IMEP_n, with a fixed intake manifold absolute pressure of 76 kPa.

The fuel/air equivalence ratio was measured based on the O₂ in the exhaust using a lambda meter (ETAS LA4) with a broadband lambda sensor (Bosch LSU 4.9). The lambda meter settings were changed for each fuel blend using the appropriate C/O and C/H ratios. The CO, UHC, CO₂, and NO_x emissions in the exhaust were measured using an automotive emissions analyzer (Horiba MEXA-584L).

The baseline fuel was indolene, a reference grade gasoline (EPA Tier II EEE, analyzed at 86.46% by weight carbon, <0.01% oxygen, 13.60% hydrogen; H/C = 1.88; 97.2 RON, 89.0 MON). The indolene was blended with anhydrous ethanol (Sigma Aldrich, ethanol anhydrous, 200 proof with purity $\geq 99.5\%$, H₂O $\leq 0.005\%$) to produce each fuel blend. Each fuel blend was measured and poured into a glass container which was pre-rinsed with ethanol and then dried. The container was shaken vigorously to mix the fuels initially and again before adding to the engine fueling system. Between each blend, the fuel system was purged once using a small amount of the new blend.

The combustion chamber was imaged through the fused silica cylinder liner using a high-speed color digital video camera with a widescreen CMOS array (Vision Research Phantom v7.11, color). A macro 105 mm lens (af micro Nikkor 105mm f2.8 d) was used with the camera to adjust the focal length and f4.0 was selected to optimize the focal depth and light exposure. Two high intensity LED arrays (1 $\times$ 3 array of 3-watt cool white LEDs) were installed at the two sides of the engine (perpendicular to the fuel injection stream) to visualize the spray. In this study, the camera was focused at a plane coinciding with the spark plug ground

electrode. The camera settings were fixed at 1280 $\times$ 552 pixels at 9000 frames per second (corresponding to 1 frame/CAD) with 100 μ s exposure time. Non-filtered emission was recorded via the high-speed color digital camera and time-sequenced with the crank-angle resolved pressure data. The imaging data captured flame propagation (via chemiluminescence), soot formation (via incandescence/thermal emission of the soot particles) and scattering of the LED emission by the fuel spray.

Camera images were time sequenced to a common trigger with in-cylinder pressure data. The imaging data were synchronized with the pressure data with an uncertainty of ~ 0.1 -0.2 ms or 1-2 CAD. The high speed imaging data were acquired for 11000 frames corresponding to 15 consecutive combustion cycles. The images were post-processed to extract temporal and spatial information. Color enhancement algorithms were applied to the images to isolate the soot incandescence and reduce interference from reflections and other stray features. Averages of the imaging data were generated using the data from the 15 combustion cycles.

Figure 2 presents the step-by-step processing of the imaging data. The top panel in the figure shows a frame from the original imaging file corresponding to 420° aTDC. The middle panel presents the results of converting the image to grey scale and after a threshold has been applied to reduce stray reflections and other interfering signals. The bottom panel shows the results of averaging the 15 frames corresponding to 420° aTDC from each of the combustion cycles. A false color scale has been applied to the data in the bottom panel.

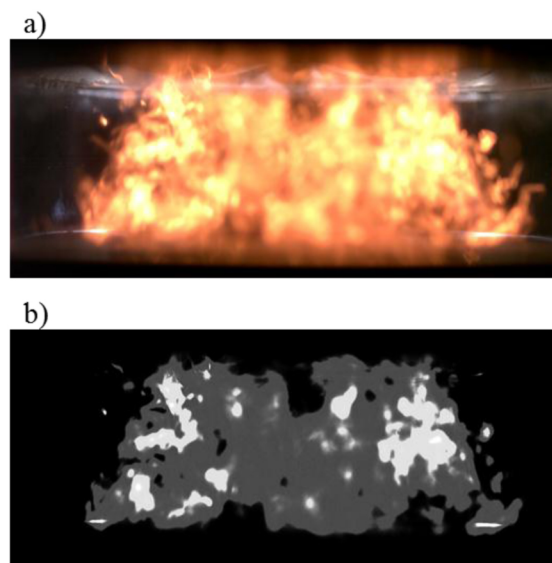


Figure 2. Image processing steps. a) original image, b) background elimination and conversion to grey scale, c) intensity map of all cycles in false scale.