Liquid Spray Penetration Under HCCI Conditions

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Abstract

Homogeneous charge compression ignition (HCCI) offers the potential for very low Nitrous Oxide and Particulate matter emission as well as increased thermal efficiency compared to conventional diesel engines. The ability to control HCCI combustion utilizing a cost effective and commercially practical system is of utmost importance. Injection strategy is an important control component in HCCI operation. Injection timing, injection pressure and nozzle configuration all effect homogeneity of the mixture and thus the NO_x and HC emissions. Higher fuel pressure will lead to higher spray velocity, greater penetration and therefore a greater chance of wall impingement. On the other hand, higher pressures lead to faster mixing, so similar mixing can be achieved with later injection timing and high spray pressure. In this experimental study, a high-speed camera was used to visualize the spray development and measure the spray penetration for diesel fuel sprays injected under HCCI conditions. The fuel sprays were injected into a constant-volume pressure vessel with optical access. The pressures in the vessel were in the range that would be encountered during a very early injection in a diesel engine that would allow the fuel and air enough time to form a homogeneous mixture before the piston approaches TDC and ignition conditions are met. The liquid penetration was measured for different fuel injectors for varying ambient pressures.

Introduction

Homogeneous charge compression ignition (HCCI) offers the potential for very low Nitrous Oxide and Particulate matter emission as well as increased thermal efficiency. The ability to control HCCI combustion utilizing a cost effective and commercially practical system is of utmost importance. Injection strategy is the most important control component in HCCI operation. Injection timing, injection pressure and nozzle configuration all effect homogeneity of the mixture and thus the NO_x and HC emissions. The aim of researchers is to find a combination of spray, fuel pressure, and injection timing that will yield both low NO_x and HC. Higher fuel pressure will lead to higher spray velocity, greater penetration and therefore a greater chance of wall impingement. On the other hand, higher pressures lead to faster mixing, so similar mixing can be achieved with later injection timing and high spray pressure.

Materials and Methods

A prototype injector was installed on a pressure vessel with optical access. Several different injectors were utilized with differing number of holes and orifice diameter. The high-pressure portion of the fuel injection system consists of a production belt driven pump, a large electric motor to drive the pump, a common rail, and connections. The motor runs as a constant speed of 1750 rpm and has an output of 10 horsepower. A 480-volt outlet provides the 15 Amps of electricity the motor requires. The fuel pump is a production unit off a John Deere Tractor. It is capable of providing up to 300 MPa of pressure, although only 70 MPa is required to properly operate the injectors. The low-pressure part of the fuel system contains a 5-gallon fuel tank, fuel pump, pressure regulator, and fuel filter. The fuel tank has one outlet line and two return lines, one from the pressure regulator and the other from the high-pressure pump. It has been noticed that air pressure within the pressure vessel can work its way into the fuel tank and pressurize it. It was found that depressurizing the pressure vessel every few runs would keep the pressure in the fuel tank at a reasonable level of less than 10 psi. The auxiliary pump initially supplies the fuel to the high-pressure pump at 7 psi. The pump is powered by a 12-Volt, 16.7-amp power supply. Once a liquid column has reached the high-pressure pump the low-pressure pump no longer needs to operate.

The pressure vessel used is cylindrically shaped with 1.25-inch thick steel walls, and the outside was painted green, shown in Figure 1. The project required that a John Deere production cylinder head be installed onto the vessel. An aluminum adapter plate was machined, and the adapter plate bolts directly into the vessel. A different head could easily be attached to the vessel by machining another one of these adapters. A cylinder liner was provided by

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the sponsor, which slid into the pressure vessel. The liner wall functions as a reference for the radial propagation of the spray.. Using the production cylinder head and liner means the spray flow will be very similar to that of the inside of the actual engine. The vessel is pressurized from the building's central air supply. The compressed building air has had time to cool, which allows the temperature in the vessel to stay low compared to air compressed by a pump. A thermocouple is used to monitor the temperature inside the pressure vessel.

The most unique feature of this pressure vessel is the slanted window attachment (see Figure 2). The experiment requires that the spray pattern be imaged from the bottom up. The light that illuminates the inside of the vessel and the light reflected off the spray pattern both enter and exit the vessel along the same path and through a single window at the end of the pressure vessel (there was originally a window at the opposite end of the pressure vessel, but that is where the engine head and fuel injector are located in the current setup). A mirror placed between the light source and the window reflects the image toward the camera. The interface for the light between the outside and inside of the vessel is a glass window. A significant proportion of the light directed to the glass will be reflected off it. A window placed at the end of a cylindrical pressure vessel will reflect the light back toward the mirror and into the camera. The view must be straight from the bottom up, so the mirror's position is fixed. In addition the window into the vessel had to be large enough to tolerate the mirror obstructing some of the incoming light. The proportion of the area blocked by the mirror needed to be much smaller than the light-entering area of the window. The solution was to use a large window tilted at a 38.25° angle that allowed an ample aperture and reflected the light out away from the imaging components. The window is designed to withstand a pressure differential of 165 psi and is the pressure-limiting component of this vessel. The window is designed to operate at temperatures below 500 °F or about 530 K.

The data acquisition system consists of an ultra high-speed camera and ultra high-intensity flashlamp. The camera has the ability to take 16 images at specified intervals. The setup provides distortion free, high-contrast images at a number of different focal lengths. The ability to determine if wall impingement occurred can be discerned from the images. The camera is a La Vision Ultra Speed Star 16. F-mount camera lenses can be attached to the camera. Lenses of 60, 85 and 105 mm focal length were used. Each 16 pictures has a resolution of 640 x 512 pixels.. Because the camera is controlled electronically, its speed is limited by the time it takes to transfer images, which is 1 μs, and the maximum framing rate is 1 MHz. The shortest possible exposure time is 0.5 μs. In these experiments the frame rate was varied as needed between 50 kHz and 1 kHz. The exposure length was also varied between 10 and 80 μs, depending on whether a beam splitter or mirror was used to reflect the image into the camera. The flash lamp in use during these experiments is a high-intensity xenon discharge lamp that uses 200 J of energy during operation, with a flash duration of up to 8 ms.

Timing is very important for this experiment because the spray will traverse the cylinder in as little as 3 ms. Delay from the flash lamp to the time when the camera starts is set to 2000 μs so the lamp has time light up. The firing delay from the number 0 injector to the number 3 injector is 1.5 crank angle degrees, it is noted that the time of delay shortens as simulated engine speed is increased. At 1200 rpm the total delay is roughly 75 ms. The camera control trigger delay is about 52 ms at 1200 rpm, the 23 ms discrepancy is caused by an electronic lag as the signal works through the system. The start of injection can vary by up to 3 ms from exposure to exposure.

A fuel with similar properties to diesel, but with a higher flash point was important to prevent any heat release accidents. The fuel also has a lower aromatic hydrocarbon content which will reduce any long-term exposure side effects. The fuel used is VISCOR 1487AW/2 Calibrating Fluid, which meets SAE J-967 and ISO 4113 specifications for Diesel fuel simulation. The Viscosity of the fluid is 0.250 cSt and the specific gravity is 0.821.

Results and Discussion

Single row 6 and 16 hole fuel injectors were compared to a prototype two-row 20 hole HCCI injector. The pressure in the vessel was varied to determine the effectiveness of the injectors to avoid wall wetting at Bottom-Dead Center (BDC) until roughly negative 100 degrees after top dead center (ATDC). The time between the exposures was varied to capture fine start of injection details as well as the stagnation of spray propagation. The average velocity was interpolated from the displacement of the spray front between frames.

A stark difference was observed between the 6 and 20-hole injector. The difference in the appearance of the spray pattern's projection confirms that if avoiding wall wetting and creating a much more homogeneous mixture was the objective, the 20-hole design accomplishes it. A comparison between the spray propagation of the two injectors is made in Figure 4. Six frames or, 430 μs have elapsed since the beginning of injection. The exposure was 30 μs and the field of view was the same for both images at 74 by 59 mm. The amount of fuel injected is 25.1 mg of fuel per stroke. The spray pattern of the HCCI injector (outlined in red) propagates roughly 5 more mm before stagnating. The spray pattern from the conventional 6-hole injector is shown to impinge on the wall in Figure 4, a wideangle image of 105 by 84 mm. The left frame was taken 0.75 ms after SOI, the right frame was taken at 1.25 ms after SOI, with an exposure duration of 30 μs.

The effect of increasing the pressure in the vessel should have a profound effect on the fine spray propagating from the 20-hole injector. Early injection will allow more time for mixing but can lead to wall impingement, especially if swirl is introduced. As stated earlier, the particular concern is that spray propagation will result in fuel getting trapped between the outside of the piston bowl and the cylinder head at top dead center. Early injection, while the cylinder pressure is low, can result in the trapping of fuel in this area. The higher cylinder pressures later in the cycle can stall the advance of the spray, retaining it in the bowl area. Here the effect of cylinder pressure was investigated. The injector was fired at low to medium load conditions. The ambient air density was varied from 1.1 to 7.9 kg/m^3. Those pressures correspond to pressures that would be encountered during the first half of the compression stroke. A 75 μs exposure with 175 μs between frames was used. Approximately 3.3 ms elapse from the start of injection to the end of imaging. All conditions besides pressure remained the same.

The data was compared to penetration analysis conducted by Naber et al. (1996). In Naber & Siebers correlation, the penetration length scale x^+ is defined as:

$$x^+ = \frac{d_f \times \sqrt{\tilde{\rho}}}{a \times \tan(\theta/2)} \qquad (1)$$

where d_f is effective diameter of fuel stream exiting the orifice diameter, $\tilde{\rho} = \rho_f / \rho_a$ is the density ratio, a is the tangent of the measured angle to the tangent of the spray dispersion angle and the value used for this report is a = 0. 066. θ is the spray angle. The time scale, t^+, is define as

$$t^+ = \frac{d_f \times \sqrt{\overline{\rho}}}{a \times \tan(\theta/2) \times U_f} \qquad (2)$$

where U_f is the fuel velocity at the orifice exit. The formula used for correlation relates the non-dimensionalized time to the non-dimensionalized penetration distance as:

$$\tilde{t} = \frac{\tilde{s}}{2} + \frac{\tilde{s}}{4}\sqrt{1+16\tilde{s}^2} + \frac{1}{16}\ln\left(4\tilde{s} + \sqrt{1+16\tilde{s}^2}\right) \qquad (3)$$

Other correlations in the literature include those of Dent and Hiroyasu & Arai. It has been shown in the literature that there is not a significant difference between the correlations of Hiroyasu and Naber for typical diesel conditions.

In the experiments used to develop Naber’s correlation, a single-hole injector was used with a research grade D-2 diesel used as the fuel. An injection pressure of 137 MPa, ambient temperature of 451 K, and an orifice diameter of 0.257 mm was used. Under those conditions it appears that an ambient gas density of 124 kg/m^3 would yield greater penetration with their injector, than normal atmospheric density (1.1 kg/m^3) with the 20-hole injector. At 1.25 ms the injector spray would penetrate about 50 mm, while our 20-hole in this experiment would penetrate 31 mm.

In addition late penetration with the 20-hole injector would be much smaller, as the spray starts to stagnate toward 1.5 ms. In the Naber injector the propagation of the spray appears to continue linearly from 1.5 to the end of data at 3.0 ms. Observations of longer duration image series show the pattern of the 20-hole injector come to nearly a stop at an elapsed time of over 3 ms. This behavior may be the result of the greater area of the spray front that the 20-hole injector has. On high load conditions at very low pressures, pressures that would correspond to the piston at less than –90 degrees ATDC, the spray pattern loses its momentum before coming in contact with the wall. By the time the pattern from the 20-hole injector can traverse one quarter of the way, a convention injector will impinge on the wall at these low pressures. The measured spray penetrations in the current experiment are shown in Figure 3.

It was shown that the ambient pressure inside the vessel had a profound effect upon spray propagation. This would be expected from a spray consisting of very small drops, as the diameter of these drops shrink so does the time needed to slow them down. The initial velocity is also slower aiding in their short propagation length. The spray penetration of a prototype 20-hole injector was found to be significantly less than that of a conventional injector. The injector was shown to avoid wall impingement at relatively low ambient air density.

Acknowledgements

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Nomenclature

d	orifice diameter
P	pressure
S	spray penetration
t	time
U	injection velocity
ρ	density
θ	spray angle
Subscripts	
g	gas
l	liquid

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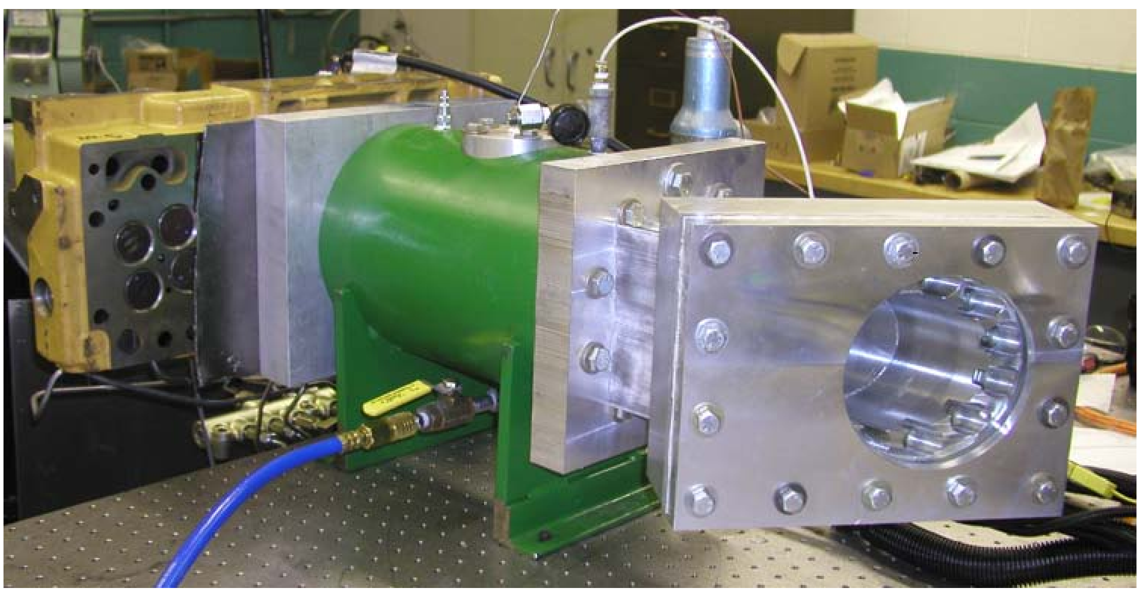

Figure 1: Pressure vessel with engine head and viewing window attached.

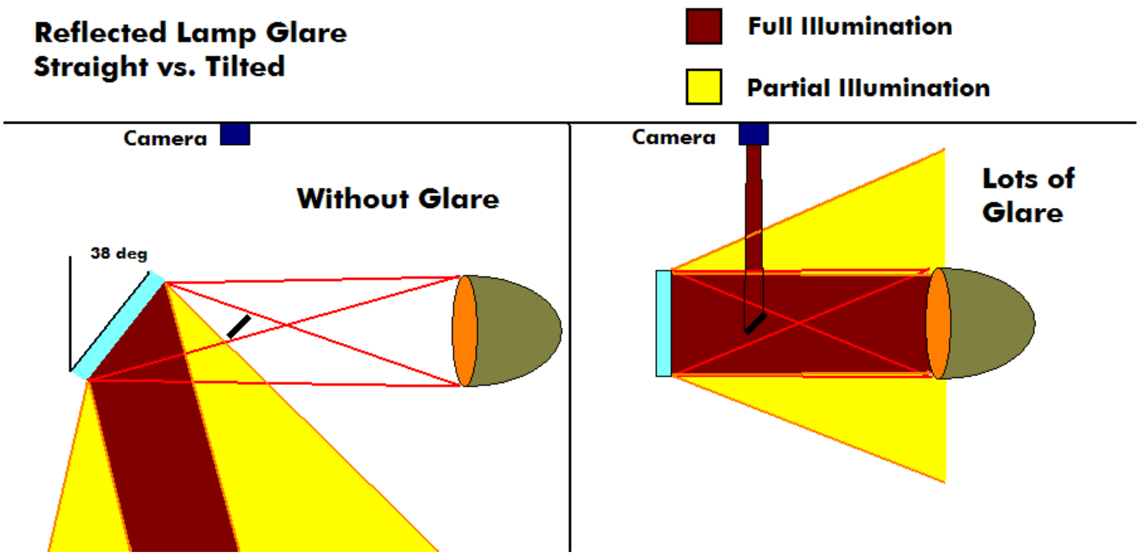

Figure 2: Optical Setup to minimize glare with single window of pressure vessel.

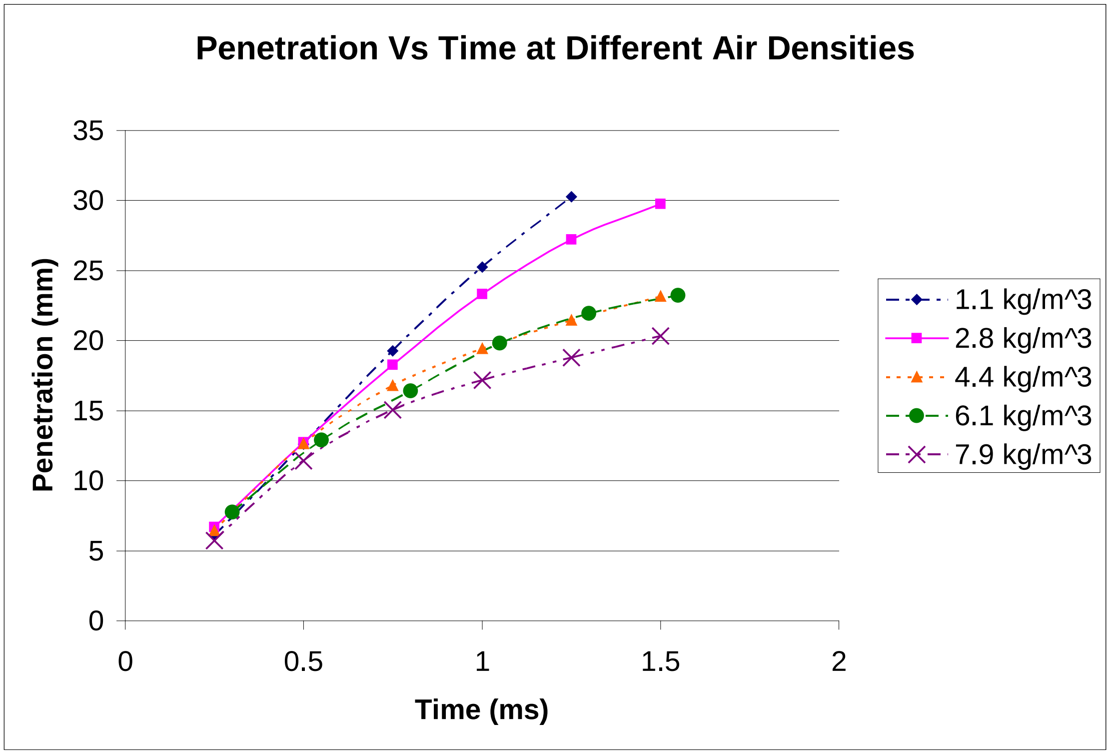

Figure 3: Measured Spray Penetrations for varying ambient densities.

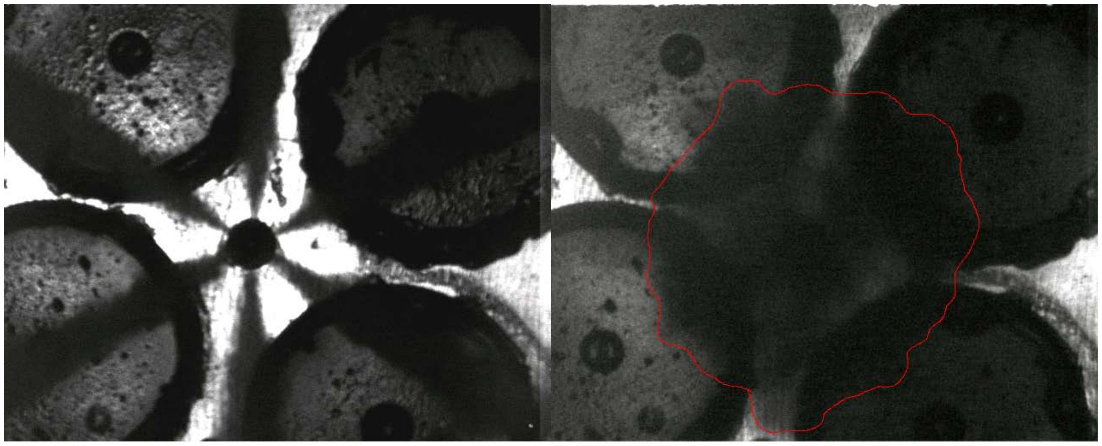

Figure 4: Images of sprays from two different injectors (conventional 6-hole on the left, 20-hole on the right) under comparable conditions.