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ABSTRACT

In developing a direct injection gasoline engine, the in-cylinder fuel air mixing is key to good performance and emissions. High speed visualization in an optically accessible single cylinder engine for direct injection gasoline engine applications is an effective tool to reveal the fuel spray pattern effect on mixture formation The fuel injectors in this study employ the unique multi-hole turbulence nozzles in a PFI-like (Port Fuel Injection) fuel system architecture specifically developed as a Low Pressure Direct Injection (LPDI) fuel injection system. In this study, three injector sprays with a narrow 40° spray angle, a 60° spray angle with 5° offset angle, and a wide 80° spray angle with 10° offset angle were evaluated. Image processing algorithms were developed to analyze the nature of in-cylinder fuel-air mixing and the extent of fuel spray impingement on the cylinder wall. Test data reveal that for a given cylinder head, piston configuration and intake air port flow characteristics, injector spray pattern plays a dominating role in how the fuel-air mixture is formed. If an appropriate injector spray pattern is chosen, the in-cylinder fuel mixing can be enhanced by minimizing fuel impingement on cylinder wall, piston top, and intake valves, thus producing a more homogeneous fuel-air mixture prior to the ignition. Engine designers can select a specific spray pattern to improve the fuel-air mixture optimized for specific parameters such as engine head, piston, valve configuration, intake air flow characteristics, fuel injection strategy, injector mounting and operating conditions.

INTRODUCTION

Developing a high efficiency and clean combustion engine with enhanced combustion performance, improved fuel economy and reduced engine emissions has been a principal goal of vehicle manufacturers and component suppliers. Among many enablers advancing the current internal combustion engine technology to achieve such goals is the development of direct injection (DI) gasoline engines. As fuel is injected directly into the engine cylinder, this engine technology offers great flexibility to control the fuel injection strategy with respect to various engine operation modes. In particular, the fuel-air mixture preparation in the combustion chamber has also been identified as one of the key factors that greatly influence the combustion characteristics of the engine performance [1]. Hence, optimizing the fuel mixture homogeneity is a key engine design parameter.

Recently, there has been a resurgent effort by various vehicle manufacturers and suppliers to develop and manufacture a second generation DI gasoline engines which overcome the challenges of the first generation production engines [2]. The main focus has been shifted to the stoichiometric homogeneous-charge engines which are designed particularly for the North American automobile markets [3]. For homogeneous combustion mode, the injection timing occurs during the intake stroke and a homogeneous fuel-air mixture is generally formed. The mixture is therefore maintained at stoichiometric condition in the cylinder prior to the ignition event. A homogeneous mixture can generally be formed by creating a spray with moderate to wide cone angle, well-atomized drops, and an appropriate level of spray tip penetration for optimizing the fuel-air mixing. In addition, the fuel injection timing window has to be precisely controlled in order to minimize any cylinder wall and/or piston wetting while maintaining the charge homogeneity in the cylinder.

When developing combustion systems for DI gasoline engines, it is important to achieve optimal fuel-air mixture for ignition. Depending upon the combustion chamber configuration and the engine operating modes, the fuel mixture strategy may require different levels of control over key spray characteristics including spray pattern, cone angle, penetration, and drop size. If the injectors can be designed to offer spray tailoring flexibility, engine designers may utilize the injectors to deliver the specific flow and spray requirements without major compromises and limitations when running the engine at its optimized configuration.

High speed imaging has evolved as a primary optical diagnostic technique for investigating the characteristics of ultra-fast motion events. The short time duration between frames and high image quality with good image resolution make high speed imaging an ideal optical tool to study the highly transient fuel spray characteristics applicable in an engine configuration. Earlier research by Hamady et al. [4] studied the fuel spray characteristics from various injector nozzles using a high speed imaging system. Using a similar technique, Kawajiri et al. [5] were able to investigate the interaction between spray and air motion in a cylindrical vessel with swirling intake gas motion similar to that in an engine. In addition, high speed imaging visualization from consecutive cycles was also applied to study fuel distribution, ignition, and combustion characteristics [6-8] under realistic engine speed and load configurations. More recently, Hung et al. [9] combined high speed imaging with time-resolved laser diffraction to characterize the transient nature of the gasoline pulsing sprays under atmospheric condition. Transient characteristics such as drop sizing, intra-pulse and pulse-to-pulse interactions throughout and in between consecutive injection cycles were readily resolved.

Hardware improvement has also been a key contributor to the major advances in imaging technology over the past decade. The technology of high speed camera systems has also shifted from a traditional 16 mm film camera such as a rotating prism cine camera at a maximum rate of around 10,000 frames per second to advanced CMOS or CCD based sensors with digital image format at a frame rate exceeding 100,000 frames per second with kilo pixel resolution. Visible and ultra-violet laser light sources have also been advanced to operate at repetition rates as high as 50 kHz. These powerful illumination sources work very well with the intensified or regular cameras, enabling visible imaging or planar laser induced fluorescence (PLIF) imaging on fuel spray to be performed. Timing devices have also been improved significantly such that the timing of the injection events can be reliably synchronized with the camera and light source. In addition, sophisticated algorithms and innovative analysis techniques have also been developed. Large quantities of digitized images can be stored and analyzed efficiently, thus allowing both qualitative and quantitative information to be extracted. For example, mechanisms of fuel film formation on the cylinder wall have been identified as a potential source of smoke and hydrocarbon emissions from DI gasoline engines. Various studies have been performed [10, 11] to focus on investigating the mechanisms of the fuel film formation. In particular, Drake and Fansler [12] used high speed imaging at a rate of 4,500 frames per second to visualize the fuel film deposition on piston top from two different types of fuel injectors. Using a RIM (Refractive Index Matching) technique, they were able to process the images and quantify the fuel film area, thickness and volume deposited on piston top as a function of engine crank angle. They concluded that the spray structure could be a dominant factor in affecting the amount of film deposited on piston for stratified charge combustion during late injection.

The objective of this paper is to investigate how spray pattern would affect the fuel mixture preparation in a DI gasoline engine under realistic speed and load conditions. Several key parameters including the injector spray pattern, injection timing, and fuel pressure are evaluated. The observations are based upon the fuel distribution in the combustion chamber as well as fuel impingement on the cylinder wall as a function of crank angle degree. Further, imaging analysis techniques are presented to reveal cylinder wall impingement. The results will be used to correlate the engine combustion and emissions performance in the subsequent single cylinder dynamometer combustion testing, which will be reported in a future study.

LPDI FUEL SYSTEM AND MULTI-HOLE FUEL INJECTORS

The multi-hole fuel injectors are developed as part of a patented Low Pressure Direct Injection (LPDI) fuel delivery system for DI gasoline engines [13, 14]. The LPDI fuel delivery architecture is very similar to the current port fuel injection (PFI) fuel system in production, i.e., no engine driven pump is required. This LPDI fuel system is designed for an engine to run in stoichiometric homogeneous-charge combustion mode and to achieve improved combustion, better fuel economy, and reduced emissions. It consists of a high-efficiency in-tank positive displacement pump and motor module, fuel injectors and fuel rail with an integral pulse damping feature and controls. The in-tank fuel pump delivers a nominal fuel pressure of 2 MPa to the fuel injectors connected to a common-rail through a chassis fuel line. An injector driver integrated into a powertrain control module (PCM) controls the fuel injection, timing and duration. A pressure sensor mounted on a fuel rail provides fuel pressure feedback to the PCM and the fuel pump controller to regulate fuel flow and fuel rail pressure using a closed-loop control.

One of the key components in this fuel system is the multi-hole high turbulence fuel injectors that inject fuel in relatively well-atomized drops directly into the cylinder at much lower pressures than competing high pressure DI fuel systems. It has been shown in a previous study [15] that multi-hole fuel injectors offer the spray pattern tailoring flexibility over other existing fuel injectors using either the swirl or slit nozzles. This is because the hole pattern, hole orientation, internal flow cavity, and number of holes on a multi-hole nozzle can all be precisely designed to control individual spray plumes and the overall spray distribution. This injector utilizes a novel high-turbulence multi-hole nozzle to produce a soft spray at 2 MPa fuel system pressure with relatively well-atomized drops. Since this pressure is significantly lower than that of other existing high pressure gasoline DI systems, it enables an attractive cost effective solution for DI fuel system implementation by eliminating the expensive high-pressure fuel pump and related parts.

The fuel spray characteristics of current production intent injectors used in this study are produced by an eight-hole nozzle configuration. The internal nozzle geometry and geometrical parameters have been designed to offer different spray characteristics. Prior to running these injectors in the optical engine, injector spray tests were performed to evaluate the spray geometry, drop sizing, and injector dynamic flow characteristics in a test bench. Whenever possible, the spray measurement and characterization were carried out according to the test setup, procedure and reporting guidelines based on the SAE Gasoline Fuel Injection Standards Committee recommendations [16]. Images of spray pattern formation were recorded using a spray imaging system. Direct illumination on the spray was provided by a strobe light located at a slight angle from the direction of the camera. Spray geometrical parameters such as spray angle, offset angle, and spray trip penetration were extracted from the Mie-scattered spray images. The spray drop sizing was performed using Phase Doppler Interferometry. Statistical drop diameters and volume flux were measured and reported. In addition to the spray characteristics, an automated injector flow stand was used to measure the dynamic flow rate of the injector as a function of the injection pulse width. The static flow rate was extracted from the dynamic flow curve. The test fuel used in the spray characterization tests was n-Heptane and the fuel pressure was set at 2 MPa. The injection pulse width for the spray characterizations was set to 1.5 ms.

Figures 1 to 3 show the spray images of three spray patterns produced with various multi-hole nozzle configurations. These images were recorded at the same time delay of 1.5 ms after the start of the injection (SOI) pulse. Each spray is shown as an ensemble average image by averaging ten individual images. The image size in these figures is approximately 55 mm wide by 65 mm high. The baseline narrow spray with a spray angle 40° and 0° offset angle (denoted as 40/0) is depicted in Figure 1. As expected, this spray exhibits a strong symmetry along the injector axis. Figure 2 shows a wider spray with the spray angle of 60° and a 5° offset angle (denoted as 60/5). The offset angle of this spray is achieved by designing the valve seat and nozzle configurations without distorting the basic symmetry of the spray. Similar to the narrow spray, the spray pattern is also symmetric with respect to the injector axis, tilted by the designed offset angle of 5°. The front end of the spray is relatively uniform across the spray tip. The overall spray angle is wider because each inclined spray plume from an individual hole is increased proportionally from that of the baseline nozzle. As a result, the spray is hollower in the inner core region. Using the similar design principle, a spray with a much larger spray angle of 80° and a 10° offset angle (denoted as 80/10) can be produced, as shown in Figure 3. It is worth mentioning that even though these sprays are generated using the same basic eight-hole nozzle configurations, the fact that different spray patterns can be achieved due to different internal geometrical configurations in the nozzles demonstrates the capability to tailor the spray pattern by using the multi-hole nozzles.

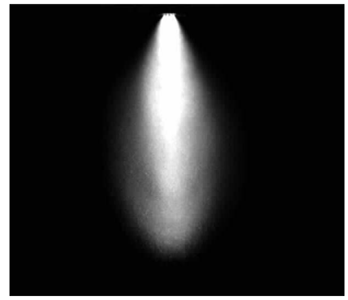

Figure 1. Spray with 40° spray angle / 0° offset angle

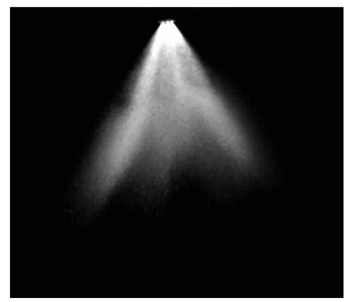

Figure 2. Spray with 60° spray angle / 5° offset angle

Figure 3. Spray with 80° spray angle / 10° offset angle

Figure 4 shows the axial penetrations of three different injector sprays. The penetration length was determined as the axial location of the spray tip from the injector axis. As expected, the 40/0 spray penetrates longer than the other two sprays with larger spray angles. Both 60/5 and 80/10 sprays have very similar penetration characteristics. Figure 5 shows the volume flux and Figure 6 depicts the sauter mean diameter (SMD) distribution of three injector sprays. The measurements were achieved along a line scan perpendicular to the injector axis at 30 mm below the injector tip. Each scan was made at a 2 mm spatial step. The narrow spray shows a Gaussian-like flux distribution with the peak flux along the centerline of the injector axis while the other two wider sprays depict a similar dual peak distribution near the edges of the spray with less flux distribution (hollow) along the injector axis. The peaks on the line scan also show the locations of the plumes which somewhat correspond to the spray angle. The peak-to-peak distance of the volume flux is largest for the 80/10 spray. The SMD values of the 40/0 spray range from 28 to 44 microns; whereas for the 60/5 and 80/10 sprays, the SMD values are between 20 and 42 microns. To convert the radial scan point-wise SMD measurements into a single, line-of-sight value, the SMD was re-processed by weighing the measurement at each location with its corresponding flux density and then normalized. Calculated SMD values for all three sprays are quite similar: that of the 40/0 is about 33.1 microns, whereas the SMD for the 60/5 and 80/10 are 33.4 and 31.8 microns, respectively.

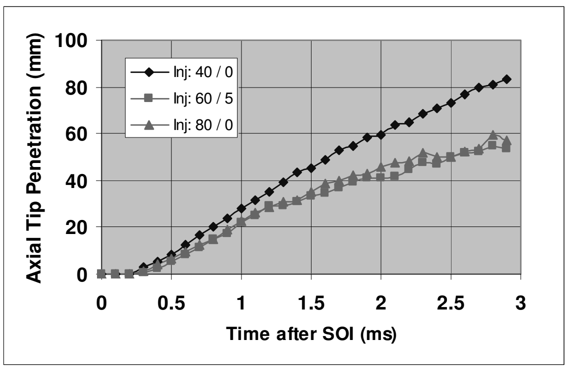

Figure 4. Axial spray tip penetration

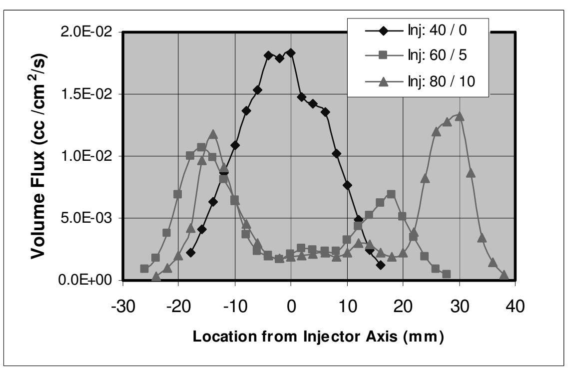

Figure 5. Volume flux distribution

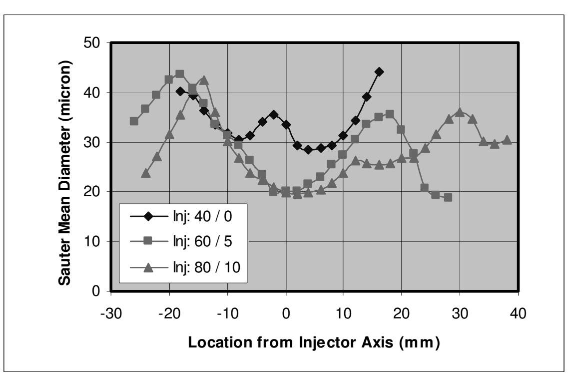

Figure 6. Sauter mean diameter distribution

EXPERIMENTAL SETUP AND TEST CONDITION

SINGLE CYLINDER OPTICAL ENGINE AND FLOW VISUALIZATION DIAGNOSTICS

The 5.4 liter V8 engine rig studied is shown in Figure 7. A production intake air manifold is used and the cylinder head has been modified to accept the low pressure direct injector. Three of the four cylinders on this half cylinder head bank have been deactivated by grinding off their lobes on the camshaft. The head is mounted on top of a single cylinder crank case that has been re-stroked to 105.7 mm to match the crankshaft geometry of the engine as well as to utilize the original connecting rod. Located between the cylinder head and the reciprocating assembly are the quartz cylinder and piston with a quartz insert. Both the cylinder and piston are designed to provide optical access to the inside of the engine cylinder while retaining the original cylinder bore of 90.2 mm. The engine is held at speed by a 15 hp AC motor with a variable speed drive. An optical shaft angle encoder is used to determine crankshaft orientation for the fuel injection control which is accomplished through the use of a timing controller. Both the high speed camera and the fuel injector are triggered from the output signal of the controller.

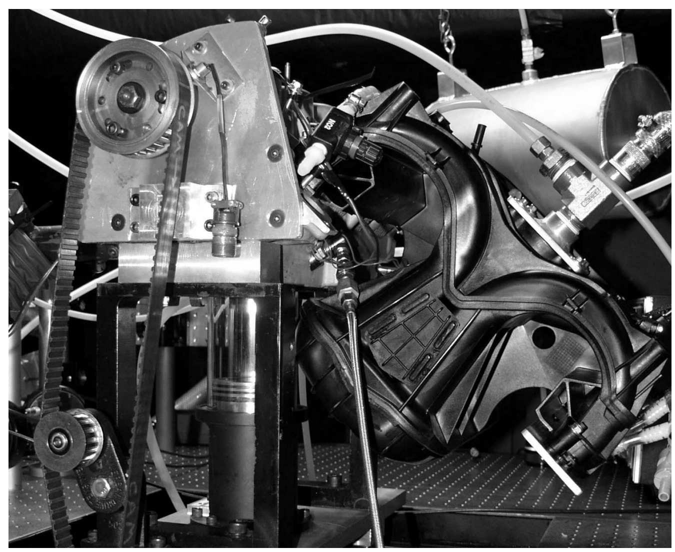

Figure 7. View of the single cylinder optical engine

Figure 8 shows the geometry of the single cylinder head and port configuration. This cylinder head has two intake valves and one exhaust valve. The spark plug is located near the center of the roof. Figure 9 shows the middle section view of the cylinder head. This view shows both the orientation and mounting locations for the injector and the spark plug. The outlines of the three injector spray orientation are also depicted in the same figure. The injector was side-mounted onto the cylinder head at an angle of 35° from the horizontal axis. The mounting location was selected such that there was no interference between the fuel injector and the existing coolant passage surrounding the cylinder head.

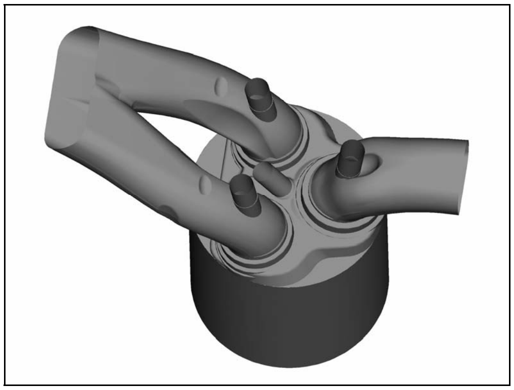

Figure 8. Single cylinder head configuration with intake and exhaust ports

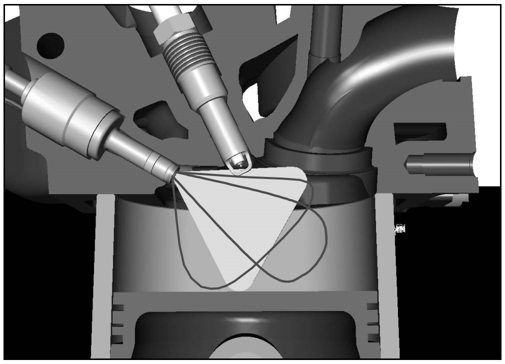

Figure 9. Placement of spark plug and fuel injector (with spray configurations outlined)

EXPERIMENTAL PROCEDURE

All fuel spray imaging tests were performed with the engine motored only. A Mie scattering technique was used to visualize the liquid phase of the fuel dispersion inside the combustion chamber through the quartz cylinder liner wall as well as the quartz piston insert. The fuel spray was imaged with a non-intensified high speed digital video camera. The camera was set to operate at 10 kHz which provides an image resolution of 512 by 512 pixels to cover a spatial imaging area of approximately 123 by 123 mm. A high repetition rate pulsed copper vapor laser, synchronized with the high speed camera and the fuel injection timing logic, was used to illuminate the liquid fuel dispersion. A fiber optics cable was used to direct the laser pulse through the quartz piston insert into the cylinder, as shown in Figure 10. This arrangement maximized the illumination quality inside the cylinder and minimized much of the secondary scattering from the internal reflection of the quartz wall. The 20 Watt laser provided the high intensity short pulse duration (about 25 ns) for visualization. This equates to an energy level of approximately 2 mJ per laser pulse.

For this series of optical engine tests, a laboratory type fuel supply system was used which consisted of a fuel bladder, pressure regulator, and compressed nitrogen bottle. Premium grade gasoline was used as test fuel and it was delivered to the fuel injector at which the injection pressure was regulated to either 2 or 3 MPa, which was dictated by different fuel flow and fuel charge mixing requirements at various engine load conditions. Two engine speed and load points were selected for this study: a part load, 1500 RPM point at a manifold air pressure (MAP) of 45.5 kPa absolute, and a full load, 2500 RPM at wide open throttle (WOT). For each test condition, the engine was first motored to reach the desired RPM. Once the engine was stabilized, a signal from the controller was sent out to the fuel injector to trigger the start of injection (SOI) at a specific crank angle of piston location. The same signal was also used to trigger the high speed camera to start recording the image sequence. Based on the previous test data of fuel flow and lambda calculation on this firing engine, the fuel injection duration was adjusted at each load condition to achieve a stoichiometric air fuel ratio. For each imaging test, 300 consecutive frames from each injection cycle were recorded to visualize the fuel dispersion during the intake and compression strokes. Five injection cycles were normally filmed to allow for a quick assessment of cycle-to-cycle variation. The optical chamber was cleaned periodically to ensure that there was no debris remained in the chamber to affect the image background quality. The test matrix in Table 1 summaries the key parameters studied in this investigation. In addition, an in-depth in-cylinder flow field investigation on this engine head and cylinder configuration was made using the Molecular Tagging Velocimetry (MTV) as previously demonstrated by Schock et al. [17]. This investigation will be reported in a future study.

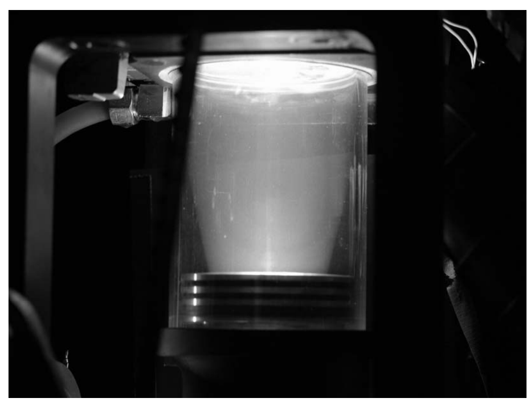

Figure 10. Illumination of the cylinder through the quartz piston insert

Parameter	Description
Fuel Injector Type	40 / 0 (spray angle / offset angle) 60 / 5 80 / 10
Fuel Injection Pressure	2 MPa 3 MPa
Injection Timing	330° CA BTDC (Intake Stroke) 300° CA BTDC (Intake Stroke) 270° CA BTDC (Intake Stroke)
Engine Load	Part Load: 1500 RPM / 45.5 kPa MAP Full Load: 2500 RPM / WOT
Injection Pulse Width	Part Load: 1 – 2 ms (nominal) Full Load: 4 – 7 ms (nominal)

Table 1 Test parameters for the optical study

RESULTS AND DISCUSSIONS

SPRAY PATTERN EFFECT ON IN-CYLINDER FUEL MIXTURE FORMATION

Figure 11 shows the comparison of three injector sprays on fuel mixture distribution in the combustion chamber at the engine part load condition of 1500 RPM and a MAP pressure of 45.5 kPa absolute. If the single cylinder engine was combusting, this MAP pressure would correspond to about 3.3 Bar IMEP (Indicated Mean Effective Pressure). Previous test data on this single cylinder engine of 3.3 Bar IMEP at 1500 RPM corresponded to a 2.62 Bar BMEP (Brake Mean Effective Pressure), which is normally referred to as the World Wide Mapping Point (WWMP). The injection pressure was regulated to a baseline level of 2 MPa. The SOI was set at 300° crank angle (CA) before top dead center (BTDC). With the adjusted cam phasing timing, the intake valve was lifted to about 8 mm at this SOI. The injection pulse width (duration) was set to correspond to lambda one (stoichiometric) condition. These images were recorded within an injection cycle at different crank degrees. It is worth mentioning that the injector driver has a 1 ms pre-charge delay, and so it corresponded to the delay of either 9 CAD/ms (at 1500 RPM) or 15 CAD/ms (at 2500 RPM) before the fuel spray was observed at the top of the cylinder. The first image of the sequence was shown at 277.5° BTDC, where the initial portion of the spray entering the cylinder was found to be about the same for all three sprays. The intake air did not have much effect on the beginning of the spray. The narrow spray of 40/0 showed a slightly stronger axial penetration along the injector axis into the cylinder. At 255° BTDC, the fuel charge started to show some noticeable differences in the fuel distribution. The 40/0 spray penetrated more directly across the cylinder towards the liner wall while the 60/5 and 80/10 sprays were moving more towards the central region of the cylinder. They produced very minimal fuel impingement on the opposite side of the liner wall. It also shows that at this SOI timing, the leading portion of the sprays impinged slightly on the piston top. However, as the cycle progressed to 210° BTDC, the fuel distributions among all three sprays were quite similar.

The fuel distribution at 2500 RPM with full load WOT is displayed in Figure 12. The SOI was again set at 300° BTDC. In this case, the fuel injection pressure was increased to 3 MPa. At full load, the images show that both the spray angle and offset angle were critical factors to affect how fuel was dispersed. In addition, the effect of intake air on fuel mixing was more dominant when the engine was running at full load condition. When the intake valves opened, the intake air diverted the spray slightly towards the direction of moving piston. The initial spray dispersion at an early CA of 262.5° BTDC seemed to be quite similar for all three sprays. However, as the cycle progressed to 225° BTDC, the liquid fuel of the 40/0 spray moved directly towards the opposite side of the cylinder wall. The 60/5 spray penetrated more towards the central region of the cylinder and less on the cylinder wall. The spray was tilted more towards the piston and it created a slightly better fuel dispersion within the cylinder. The impinging location of fuel on the cylinder wall was further away from the top of the cylinder along the stroke than the previous 40/0 spray. It is believed that the enhanced dispersion is due to two factors: Firstly, a wider spray angle improves the fuel dispersion; and secondly, the additional spray offset angle of 5° from the injector axis moves the spray even more towards the piston direction. As expected, the 80/10 spray directed the fuel dispersion even more towards the central region of the cylinder without any noticeable fuel impingement on the cylinder wall. At 197.5° BTDC, the fuel impingement of the 40/0 spray was very pronounced. At 150° BTDC, both intake valves were almost fully closed. The fuel dispersion of the 40/0 spray was quite localized in the upper half of the cylinder closer to the exhaust valve. The 60/5 spray improved the fuel distribution slightly, but the 80/10 spray seemed to provide the best fuel air mixing in the cylinder.

The images, particularly at high load point, confirm that the spray pattern has a dominant effect on how fuel is dispersed inside the cylinder. The narrow spray of 40/0 created a significantly reduced core dispersion of fuel droplets in the central region of the cylinder. A narrower spray usually has a higher axial spray penetration. Since the spray penetration was along its injector axis which was mounted at an angle of 35° inclined from horizontal, the tip of the spray penetrated directly across the cylinder along the injector mounting axis and impinged on the opposite side of the cylinder wall at a location somewhere near the middle of the cylinder. This impingement location was found to be closely related to the geometrical mounting of the injector. A wider angle of fuel spray not only produced a more homogeneous fuel-air mixture by dispersing the mixture formation more in the upper to central region of the cylinder, it also reduced the penetration along its injector axis. In addition, since the spray was designed to bend towards the piston by either 5° or 10°, the spray was able to propagate more directly towards the piston. However, for all three sprays, it was also noticed that there was a lack of fuel distribution in the upper half of the cylinder closer to the intake valves. It is believed that the lack of fuel dispersion was partially due to the constraint of the injector mounting orientation.

For an existing configuration of injector mounting orientation and cylinder geometry in this engine, a wider spray angle with an offset angle bent towards the piston made the spray less likely to impinge on the cylinder wall. It is likely to reduce any liquid fuel film formation on liner walls which usually leads to a high level of unburned hydrocarbon and other smoke particulates. The fuel impingement on the piston top is strongly dependent upon the SOI timing. The potential fuel impingement on the piston top may also be minimized if the injection timing can be retarded further up to a reasonable level without degrading the fuel mixture quality.

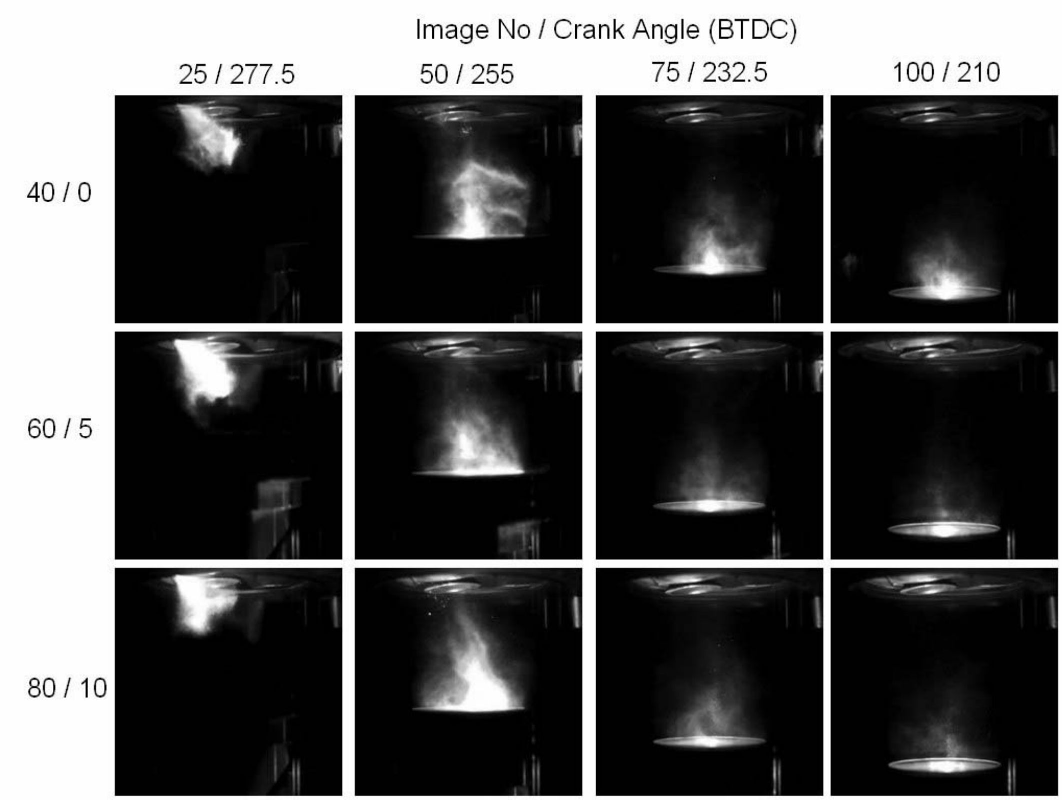

Figure 11. In-cylinder fuel mixture formation at 1500 RPM / 45.5 kPa MAP / 2 MPa fuel pressure / SOI at 300° CA BTDC

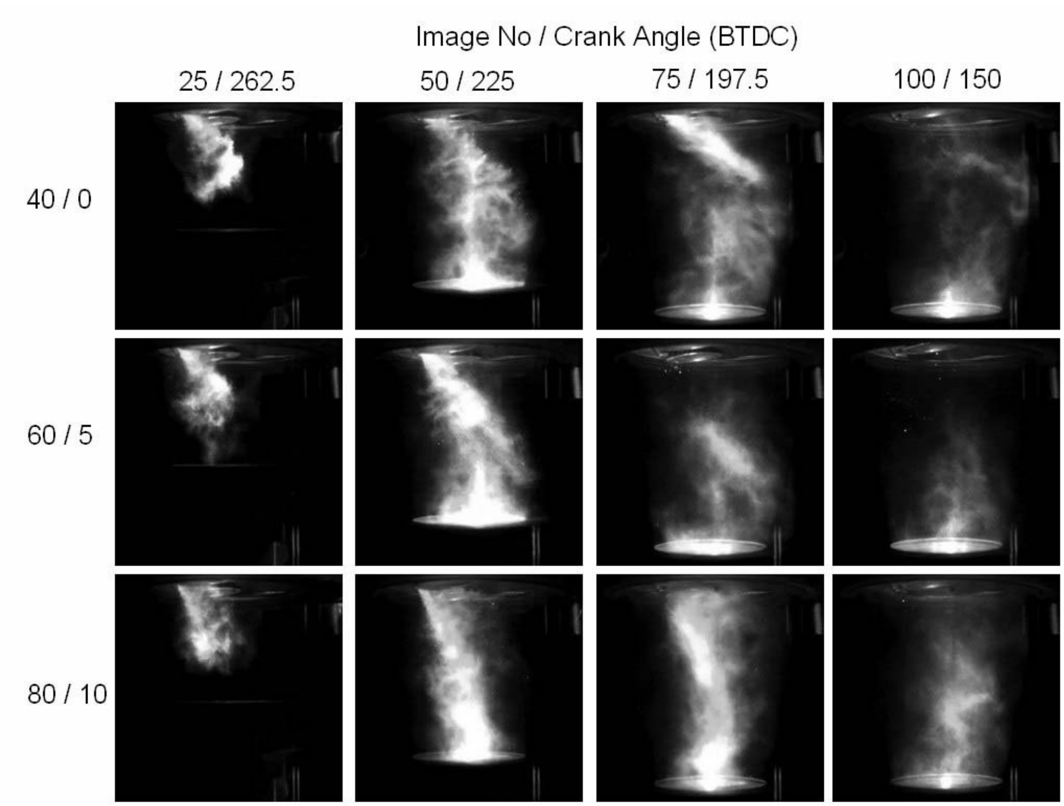

Figure 12. In-cylinder fuel mixture formation at 2500 RPM / WOT / 3 MPa fuel pressure / SOI at 300° CA BTDC

The 80/10 spray produced a better fuel mixture in the cylinder. However, it was also found that due to the close proximity of the injector tip to the intake valves, the larger spray angle also caused a slight amount of fuel spray impingement on the inner side of both intake valves, as shown in Figure 13.

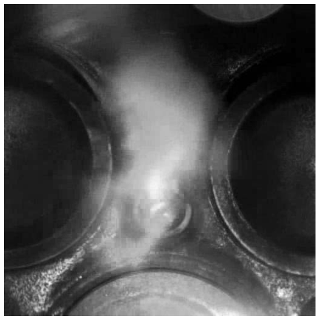

Figure 13. Slight fuel impingement on intake valves due to larger spray angle

FUEL IMPINGEMENT ANALYSIS ON CYLINDER LINER WALL

Based on the distinct features depicted in Figures 11 and 12, it is possible to identify and extract more information on the mixture formation from these images with image processing. Therefore, image processing algorithms have been developed to measure the semi-quantitative information such as the magnitude of fuel spray impingement on cylinder wall and piston top, and fuel-air mixture homogeneity.

For example, to analyze the fuel impingement magnitude on the cylinder wall, a fuel impingement index on the cylinder wall can be defined based upon the illumination intensity of the location (pixel) on the image near the cylinder wall. The methodology of the fuel impingement on the cylinder wall is briefly outlined next. Figure 14 depicts the measurement areas shown as a gray bar along the cylinder wall where the illumination intensity of each pixel in the image is extracted. The size of this area depends on the image orientation and measurement location of interest. For the analysis of fuel impingement on cylinder wall, a thin area band was chosen to be 5 pixels (i^{th}) wide by 300 pixels (j^{th}) long. Then, an average intensity is computed by averaging the pixel intensity across the width (i.e., across the i^{th} direction) of the area at each j^{th} pixel as follows:

$$\bar{I}_{ave,j} = \frac{\sum_{i=1}^{N} I_{i,j}}{N} \tag{1}$$

where $I_{i,j}$ is the intensity of an individual pixel in the measurement area. N is the number of pixels along the width and it is equal to 5 for this analysis.

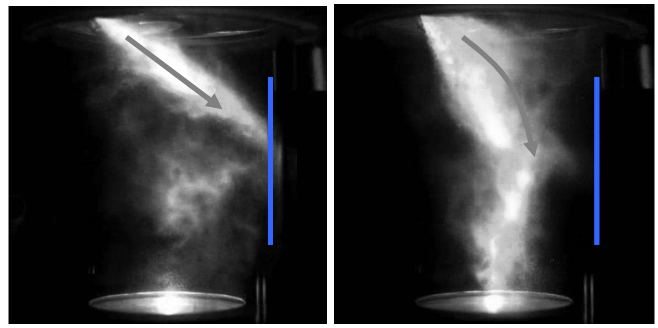

Figure 14. Location of the fuel impingement analysis (illustrated by the measurement region) (Left: 40/0 spray; Right: 80/10 spray)

Figure 15 shows the comparison of the average intensity between the two spray patterns along the measurement line. For both sprays, there was no fuel impingement near the top of the cylinder wall. However, it can be seen that for the 40/0 spray, the average intensity along the cylinder wall started to increase abruptly at about one-third of the stroke distance, and it peaked at about halfway of the cylinder. After the peak, the intensity continued to decrease toward the bottom of the cylinder. Fuel impingement was found to spread more on the lower half of the cylinder wall. Conversely, for the wider spray of 80/10, there was almost no impingement of fuel along the cylinder wall. The average intensity remained very minimal and constant along the entire analysis location.

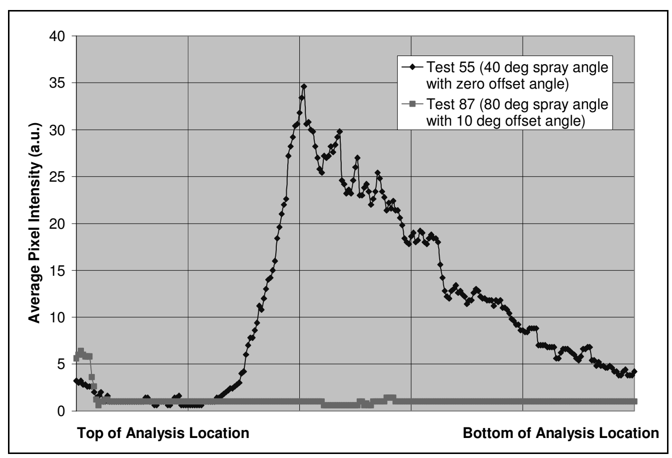

Figure 15. Comparison of ensemble average intensity on wall impingement when piston is at BDC

Since the fuel impingement is strongly transient and it is rapidly changing at different crank angles within an engine cycle, an overall fuel impingement index (FII) at a specific crank angle can also be defined based on the ensemble average intensity over the entire location along j^{th} direction of the measurement domain:

$$FII_{CA} = \frac{\sum_{j=1}^{M} \bar{I}_{ave,j}}{M} \tag{2}$$

where M is the number of pixels along the length of the measurement domain. M is equal to 300 for this analysis.

This index can be used to track and analyze the extent of fuel impingement at each crank angle degree over the injection cycle. Figure 16 shows such a plot of crank angle resolved fuel impingement index on the cylinder wall. This figure also reveals several useful facts about the characteristics of such as the sequence and the duration of the fuel impingement. The injection logic pulse started at 270° BTDC when the image sequence was commenced. Taking the injector driver pre-charge delay into account, the fuel spray entered the cylinder at about 246° BTDC. The spray then propagated directly across the cylinder and started to impinge on the cylinder wall at about 210° BTDC. The impingement index started to increase as the piston continued to sweep downwards. For both sprays, even though the peak of impingement was observed to be between 140° and 130° BTDC, the narrow spray with 40/0 resulted in substantially higher cylinder wall impingement than the wider spray of 80/10. Impingement continued to decrease for both sprays as the piston reached about 100° BTDC. Beyond this crank angle, the fuel impingement for both sprays was found to be very minimal.

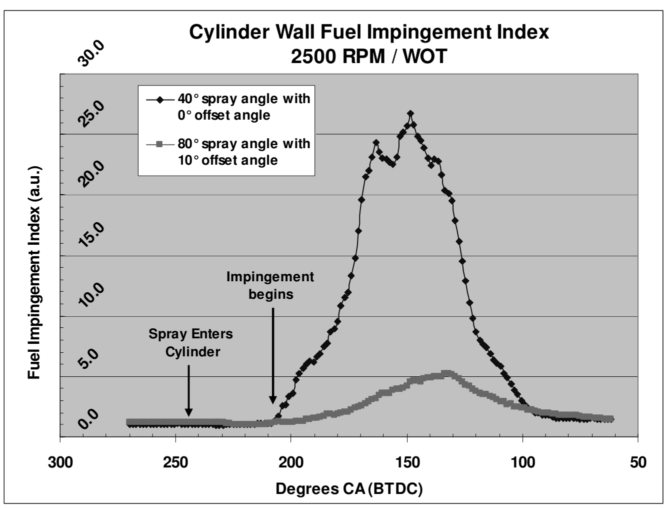

Figure 16. Crank angle resolved fuel impingement index on cylinder liner wall

Similar to any other image analysis methods based on light intensity extracted from the pixels of an image, it is important to realize that this fuel impingement analysis technique mentioned above also requires a consistent light illumination and background in the region of interest in order to minimize any possible inaccuracy or uncertainty. For example, any fuel droplets populated in the bottom of the liner wall or in the shadow of the piston quartz insert may not be accounted for equally due to the uneven light intensity distribution inside the cylinder. In addition, any residual fuel left behind from previous cycles may remain in the cylinder or be deposited on the walls as the cycle progresses. This could potentially over-estimate the quantity and location of the fuel impingement at a particular crank angle. Therefore, a proper background subtraction may be needed to eliminate any contribution of the residual fuel from previous cycles.

Even though this fuel impingement index cannot be used directly to correlate the amount of fuel impinged on the wall, this value indicates the extent as well as the location of the fuel impingement at a specific crank angle within a cycle. Since it is based on the illumination intensity of the pixel, once the images are properly adjusted to correct for any illumination deviation in the imaging setup, it may be useful for comparing other fuel mixing quality between different conditions.

CONCLUSION

High speed imaging was performed to visualize the spray pattern effect on fuel mixture formation as a function of crank angle in a single cylinder engine for direct injection gasoline applications. With the use of the imaging diagnostics to differentiate the fuel mixing characteristics produced by three different spray patterns, it was found that the spray angle, offset angle, and injector mounting orientation had pronounced effects on the fuel mixture preparation. The fuel mixture inside the combustion chamber was affected more by the spray pattern at full load than on part load condition. A narrow spray with 40° spray angle was not able to create a homogeneous fuel mixture in the cylinder. Fuel was found to impinge on the cylinder wall. The location of the impingement was strongly dependent on how the injector was mounted in the cylinder head. However, the widest spray angle of 80° with 10° offset angle produced a better fuel mixture in the cylinder with more homogeneous distribution and less cylinder liner wall impingement. Due to the wide spray angle, a small amount of fuel was also found to impinge on the intake valves. Moreover, for all three spray patterns, there was still a lack of fuel dispersion in the upper part of the region near the intake valves. It is believed that the lack of fuel dispersion was partially due to the constraint of the injector mounting orientation. For this cylinder head configuration, a steeper (more vertical) injector mounting angle may help improve the fuel mixture distribution and overall homogeneity.

Fuel impingement on cylinder liner walls was also investigated by using image processing and analysis algorithms. Using high speed imaging, the transient nature of fuel impingement was resolved as a function of crank angle degree. If a consistent light intensity through the image was ensured, the location and extent of fuel impingement of various spray patterns could be differentiated and compared. Similar image analysis methods may also be applied to evaluate the fuel impingement on the top of the piston. A new injector spray pattern is currently being revised which will not only minimize the fuel impingement on liner wall and intake valves, but enhance the overall fuel distribution. These results will be used to correlate the engine combustion and emission performance in the subsequent single cylinder dynamometer combustion testing.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Center
CA	Crank Angle
CAD	Crank Angle Degree
CMOS	Complementary Metal Oxide Semiconductor
CCD	Charge Coupled Device
DI	Direct Injection
FII_{CA}	Fuel Impingement Index
IMEP	Indicated Mean Effective Pressure
LPDI	Low Pressure Direct Injection
MAP	Manifold Air Pressure
MTV	Molecular Tagging Velocimetry
PCM	Powertrain Control Module
PFI	Port Fuel Injection
PLIF	Planar Laser Induced Fluorescence
RPM	Revolution Per Minute
SOI	Start of Injection
SMD	Sauter Mean Diameter
WOT	Wide Open Throttle
WWMP	World Wide Mapping Point

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