ABSTRACT

An experimental study is performed to investigate the fuel impingement on cylinder walls and piston top inside a direct-injection spark-ignition engine with optical access to the cylinder. Three different fuels, namely, E85, E50 and gasoline are used in this work. E85 represents a blend of 85 percent ethanol and 15 percent gasoline by volume. Experiments are performed at different load conditions with the engine speeds of 1500 and 2000 rpm. Two types of fuel injectors are used; (i) High-pressure production injector with fuel pressures of 5 and 10 MPa, and (ii) Low-pressure production-intent injector with fuel pressure of 3 MPa. In addition, the effects of split injection are also presented and compared with the similar cases of single injection by maintaining the same amount of fuel for the stoichiometric condition. Novel image processing algorithms are developed to analyze the fuel impingement quantitatively on cylinder walls and piston top inside the engine cylinder. Qualitative details of spray tip penetration are also presented to reveal the effects of ethanol fuels compared to that of gasoline. It is found that the split injection is an effective way to reduce the overall fuel impingement on in-cylinder surfaces. No significant difference is observed on fuel spray pattern when gasoline is compared with E50 and E85. However, spray tip penetration is slightly higher with gasoline than that of ethanol fuels. Results also show that the wall impingement is higher with gasoline compared to ethanol fuels.

INTRODUCTION

Improvement in fuel efficiency and reduction in exhaust emissions are the main goals behind the new developments in internal combustion (IC) engines. The concept of direct-injection spark-ignition (DISI) engine has the potential to achieve such goals. In this technology, fuel is directly injected into the engine cylinder, which offers great flexibility to control the fuel injection timing, its duration and the number of injections. Note that the fuel-air mixture preparation in the combustion chamber is one of the key factors that influences the in-cylinder combustion characteristics and hence the engine performance (Hung et al., 2007). Therefore, optimizing the fuel-air mixture homogeneity is an important parameter for the engine designers. In general, a homogeneous fuel-air mixture is achieved by injecting the fuel during the intake stroke. However, due to in-cylinder injection and higher injection pressures, the fuel impingement levels on in-cylinder surfaces in DISI engines are typically higher than those in port-fuel injection (PFI) engines (Pereira et al., 2007). This results in an increase in the levels of un-burned hydrocarbons and smoke emissions, which reduces the potential fuel economy benefits associated with the direct-injection engines. Therefore, it is important to control the fuel injection timing precisely in order to minimize the fuel impingement on in-cylinder surfaces.

Several studies have been reported on fuel spray pattern visualization and its influence on mixture formation inside the cylinder of direct-injection systems. Grimaldi et al. (2000)
studied the spray characteristics inside a gasoline direct-injection (GDI) system with high pressure modulation. The authors used a laser sheet technique for spray visualization, where the radiation of Nd-Yag pulsed laser was scattered by the spray droplets lying on the lighted plane and collected by a CCD camera. The spray images were analyzed in terms of spray penetration and global shape. Hung et al. (2003) used a Mie scattering technique for spray pattern visualization inside a direct-injection spark-ignition engine. The authors used the presence probability image (PPI) technique (Grimaldi et al., 2000) to quantify the pulse-to-pulse variability of the macroscopic fuel spray characteristics. Kawajiri et al. (2002) investigated the spray behavior in a constant-volume cylindrical vessel, in which a swirling gas motion similar to that in a DI engine cylinder was generated. The experimental results were also compared with the numerical simulation results by the authors. They found that the main part of the hollow-cone spray was influenced strongly by the air motion, although the influence of the air motion on the central part was weak. Hung et al. (2007) used high speed imaging to visualize the spray pattern in a single-cylinder direct-injection gasoline engine. Three spray patterns, i.e. a narrow 40° spray angle, a 60° spray angle with 5° offset angle, and a wide 80° spray angle with 10° offset angle, were studied by the authors. They concluded that for a given cylinder head, piston configuration and intake port flow characteristics, injector spray pattern plays a dominating role in how the fuel-air mixture is formed. It was suggested that if an appropriate injector spray pattern is chosen, the in-cylinder fuel-air mixing can be enhanced by minimizing the fuel impingement on in-cylinder surfaces, thus producing a more homogeneous fuel-air mixture prior to ignition. Aleiferis et al. (2008) studied the spray development of gasoline, iso-octane and ethanol in a spark-ignition engine. The authors found that the spray characteristics of fuels differ between hot and cold engine operation to a large extent. Few differences were noticed in the spray development among all fuels at an engine head temperature of 20° C. However, the spray cone angle was reduced considerably at an engine head temperature of 80° C. Gasoline and E85 115 (a blend of 85 percent ethanol and 15 percent iso-octane) appeared to demonstrate a classical spray collapse at this engine head temperature of 80° C. Iso-octane was found to be the least sensitive to the elevation of engine head temperature in terms of spray collapse.

The objective of this paper is to investigate the fuel impingement on cylinder walls and piston top inside a direct-injection spark-ignition engine. Novel image processing algorithms are developed to analyze the fuel impingement on in-cylinder surfaces. Gasoline and ethanol-gasoline blended fuels are compared in this work. Experiments are performed at different engine speeds and load conditions with the variation in fuel injection timing, fuel pressure and the number of injections. In the following sections, a detail of experimental setup is first outlined, followed by the procedural details of fuel impingement analysis on in-cylinder surfaces. Results of various experiments are then presented. Finally, concluding remarks are summarized from this work.

**EXPERIMENTAL SETUP**

The engine used in the present work is a four-valve, two intake and two exhaust valves, 0.4 liter single-cylinder spark-ignition engine although not fired. Quartz cylinder is used to provide the optical access to the cylinder. Engine specifications are listed in Table 1 with the bore diameter of 83 mm and the stroke length of 73.9 mm. It should be noted that in this paper 0° crank angle corresponds to the top dead center (TDC) of the compression, and therefore −180 and 180 crank angle degrees (CADs) correspond to the bottom dead center (BDC) of the intake, i.e. 180° BTDC (before top dead center) and power strokes, i.e. 180° ATDC (after top dead center) respectively. Figure 1 shows the experimental rig with the high-speed camera and laser in place for spray visualization. In order to be able to fit with multiple configurations of the DI injectors and ignition spark plugs and coils, the cylinder head was designed to be able to use both the low-pressure and high-pressure direct-injection injectors. To study the ionization combustion feedback, either 8 mm or 14 mm spark plugs may be fitted to the single cylinder head, however not used in this study. The head also accommodates a pressure transducer to record the in-cylinder pressure data. A view of the combustion chamber geometry showing intake and exhaust valves, direct-injector, spark plug and the pressure transducer is illustrated in Figure 2. The custom-designed piston was used, which allowed the geometrical compression ratio (knock-limited) from a baseline of 9.75:1 to increase up to 13.5:1. The increased compression ratio was largely made possible by a re-design of the piston bowl with high mid section, shallow bowl contour, and four cavities matching the valve profiles.

<table>
<thead>
<tr>
<th>Specifications</th>
<th>Measure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore (mm)</td>
<td>83</td>
</tr>
<tr>
<td>Stroke (mm)</td>
<td>73.9</td>
</tr>
<tr>
<td>Rod length (mm)</td>
<td>123.2</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>13.5:1</td>
</tr>
<tr>
<td>Intake valve timing</td>
<td>Opens 355 CAD Closes -122 CAD</td>
</tr>
<tr>
<td>Exhaust valve timing</td>
<td>Opens 124 CAD Closes -332 CAD</td>
</tr>
</tbody>
</table>
FUEL INJECTION SYSTEM
When developing combustion systems for DI engines, it is important to achieve optimal fuel-air mixture prior to ignition (Hung et al., 2007). 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. For this engine, side-mounting option for installing the fuel injector is not feasible due to the packaging constraints. In addition, the tilted angles for the inlet and exhaust valves are quite large, at 21.5 degrees and 24.1 degrees, respectively. The dimensions and their physical constraints of the engine cylinder head only allows the injector to be placed on the top side at about 12 degrees along the piston axis in the vicinity of the spark plug (i.e., a centrally-mounted injector configuration). Therefore, two types of multi-hole fuel injectors are considered; (i) Low-pressure direct-injection (LPDI) 9-hole injector with the fuel pressure of 3 MPa (Hung et al., 2007) and (ii) High-pressure direct-injection (HPDI) 7-hole injector with the fuel pressures of 5 and 10 MPa. The advantage of the multi-hole injector is that the hole pattern, hole orientation, internal flow cavity, and number of holes of a nozzle can all be designed to control individual spray plumes and the overall spray pattern, providing enhanced droplet dispersion in the cylinder with reduced fuel impingement on cylinder walls. For this series of optical engine tests, a laboratory type fuel supply system is used which consists of a fuel bladder, pressure regulator and compressed nitrogen bottle.

EXPERIMENTAL PROCEDURE AND OPERATING CONDITIONS
A Mie scattering technique is used to visualize the liquid phase of the fuel dispersion inside the combustion chamber. The fuel spray was imaged with a Photron APX-RS non-intensified high-speed CMOS (complimentary metal-oxide semiconductor) camera and a Nikon 105 mm AF micro lens. The camera was set to operate at 10 kHz, which provided an image size of 512 x 512 pixels. At this frame rate each frame at 1500 and 2000 rpm of engine speeds correspond to 0.9 and 1.2 crank angle degrees, respectively. A high repetition rate pulsed copper vapor laser, synchronized with the high-speed camera and the fuel injection timing logic, is used to
illuminates the liquid fuel dispersion. A fiber optics cable is used to direct the laser pulse inside the engine cylinder through the quartz cylinder. The 20 watt laser provides the high intensity short pulse duration of about 25 ns for visualization purpose. Three different fuels, namely, E85, E50 and gasoline, are used in order to compare ethanol-gasoline blended fuels with gasoline. E85 represents a blend of 85 percent ethanol and 15 percent gasoline by volume. Similarly, E50 represents a blend of 50 percent ethanol and 50 percent gasoline (by volume). Two engine speeds, 1500 and 2000 rpm, with the fuel pressures of 3 MPa (using LPDI injector), 5 and 10 MPa (using HPDI injector) are considered. Different fuel injection timings (270, 240, 210 and 180 ° BTDC) are studied to optimize the injection timing that minimizes the fuel impingement on in-cylinder surfaces. In addition, the effects of split (or dual) injection are also studied and compared with the corresponding cases of single injection by maintaining the same amount of fuel. For each test condition, the engine was first motored to reach the desired rpm, i.e. 1500 or 200 rpm. Once the engine was stabilized, a pulse signal generated by the Cosworth engine controller was sent out to the fuel injector to trigger the start of injection at a specific crank angle position as well as to trigger the camera to start recording the specified number of images in consecutive cycles. The fuel injection duration (or pulse width) at each test point is defined to achieve a stoichiometric air-fuel ratio based on gasoline fuel. Similar pulse duration is considered for the corresponding test point with ethanol-gasoline blended fuels for direct comparison with gasoline. For each imaging test, five injection cycles are recorded to visualize the fuel dispersion with 400 consecutive frames from each cycle. The test matrix in Table 2 summarizes the key parameters studied in this study.

### Table 2. Test parameters for spray visualization

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
<td>E85, E50 and gasoline</td>
</tr>
<tr>
<td>Fuel injection</td>
<td>3 (LPDI injector), 5 and 10 (HPDI injector) MPa</td>
</tr>
<tr>
<td>Engine speed</td>
<td>1500 and 2000 rpm</td>
</tr>
<tr>
<td>Engine load</td>
<td>Full- (WOT) and part- (0.3 bar MAP) load</td>
</tr>
<tr>
<td>Injection timing</td>
<td>270, 240, 210 and 180 ° BTDC</td>
</tr>
<tr>
<td>Number of injection</td>
<td>Single and dual</td>
</tr>
</tbody>
</table>

**IMAGE PROCESSING TECHNIQUE FOR FUEL IMPINGEMENT ANALYSIS**

In direct-injection spark-ignition engines, fuel is directly injected into the engine cylinder and the injection pressure is generally higher than that of port-fuel injection engines. Thus fuel impingement levels on in-cylinder surfaces in DISI engines are typically higher than PFI engines. Therefore, it is important to control the fuel injection timing precisely in order to minimize the fuel impingement on in-cylinder surfaces. Novel image processing algorithms are developed to analyze the spray images for fuel impingement on in-cylinder surfaces, i.e. cylinder walls and piston top, based on crank angle degree. This technique is useful to optimize the fuel injection pressure, injection timing and the number of injections for improved engine performance.

To analyze the fuel impingement on cylinder walls (left and right) and on piston top based on crank angle degree, first the cylinder boundaries are identified in each frame. Note that the boundaries are defined in the first (or any) frame with respect to the point of interest (POI). This is to be done only once for a given experimental setup. The boundaries are then updated based on the new identified location of this point of interest due to piston motion. In general, the point of interest is selected so that the most possible details are captured within the specified block size. The point of interest used in this work is shown in Figure 3 (left) at 240° BTDC and the cylinder boundaries (at 186.9° BTDC) in Figure 4 (left) at 1500 rpm engine speed. A normalized dot product is used to identify the best match for piston motion detection or to locate the new position of POI in the processed frame. A block size of 31×31 pixel matrix (centered at POI) from the first frame is considered as a template. To identify the best match in the processed frame, this template is moved to its previous frame's POI location. Note that it is not required to consider the whole image for the best match when there is prior information about the maximum possible piston displacement between any two consecutive frames. This is to reduce the computational time. Figure 3 shows the location of POI in the first frame (left) with a circular mark and this identified location in the second frame (right) with a square mark (red color) at 1500 rpm engine speed when the start of injection (SOI) is at 240° BTDC. The observed displacement is 2 pixels in the X direction and 0 pixels in the Y direction.

For the analysis of fuel impingement on cylinder walls and piston top, a thin area band of 4 pixels (~ 0.75 mm) inside the cylinder is considered in this work; see Figure 4 (right). Green-channel data of RGB spray image (background subtracted) is considered in the processing due to its broad range of histogram distribution compared to the red- and blue-channels data. Fuel index, which shows the presence of liquid fuel on respective in-cylinder surfaces, is then defined.
based on weighted-mean-normalized intensity. For example, the fuel index at frame $i$ for the left wall impingement is:

$$f_{lw_i} = \frac{\sum_{j=1}^{N_{lw_i}} I_j}{N_{lw_i}} \times \frac{1}{L-1} \times \frac{A_{lw_i}}{A_{lw_i,\text{HRTDC}}} \tag{1}$$

where $i$ is the frame number and $j$ is the processed pixel for left wall impingement. $N_{lw_i}$ represents the total processed pixels for left wall impingement with intensity value, $I$, greater than zero. Each gray scale image considered in this processing is an 8 bit image and therefore, $L = 2^8$, and $A$ represents the total considered area, where,

$$A_{lw_i} = \sum_{j=1}^{N_{lw_i}} 1 \tag{2}$$

and

$$A_{lw_i,\text{HRTDC}} = \sum_{j=1}^{N_{lw_i,\text{HRTDC}}} 1 \tag{3}$$

Similarly, the fuel index (at each frame) for the right wall impingement is:
where each term is self explanatory, as discussed earlier. However, for the piston top impingement note that $A_{pti} = A_{pti (SOI)}$, and hence the fuel index (at each frame) is as follows:

$$fi_{pti} = \left( \frac{\sum_{j} I_{f}}{Np_{pti}} \right) \times \frac{1}{L-1} \times \frac{A_{pti}}{A_{pti (SOI)}}$$

(4)

Figure 5 shows the fuel index values evaluated for each frame for the piston top impingement with E85 when fuel injection pressure is 5 MPa and the start of injection is at 180° BTDC. The start of impingement on the piston top is calculated using the second derivative of the fuel index data. Due to the fact that derivative operation is very sensitive to noise, a two-way low pass filter is used (Zhu et al. 2007). Mittal et al. (2009) used this to filter the in-cylinder pressure data in a diesel engine. The upper graph in Figure 5 shows the comparison between the experimental (image processing) and filtered fuel index data. The peak of the second derivative (lower graph of Figure 5) is at 160.2° BTDC and shows the location where fuel impingement starts on the piston top. Due to this and to compare the different cases based on overall fuel index (see Equation 6), the fuel index value of the previous frame (161.1° BTDC in this case) with respect to the frame where impingement starts is subtracted from each frame's fuel index value. This processed data is also shown in the lower graph of Figure 5.

$$fi_{pt} = \int_{SOI}^{\theta} fi_{pti} d\theta$$

(6)

RESULTS AND DISCUSSION
EFFECT OF INJECTION PRESSURE ON FUEL SPRAY DEVELOPMENT

Fuel spray development and its impingement on cylinder walls and piston top inside a direct-injection spark-ignition engine are presented. Figure 6 shows the spray development of E85 with low-pressure direct-injection injector at 3 MPa (left) and high-pressure direct-injection injector at 5 (middle) and 10 (right) MPa. The start of injection is at −240 crank angle degree (or 240° BTDC) for each case and the engine is operated at 1500 rpm with full-load condition. At this engine speed each frame corresponds to 0.9 crank angle degree. Note that the intake valves are located towards the left side of the images. As expected, the spray is first observed when the injection pressure is high, e.g at 233.7° BTDC for 10 MPa. Similarly, it is first observed at 232.8 and 229.2° BTDC for 5 and 3 MPa, respectively. The LPDI injector (at 3 MPa) shows
a hollow cone spray pattern and a strong symmetry along the injector axis. Note that the overall spray angle (with LPDI injector) at 3 MPa is wider when compared to the HPDI injector at 5 and 10 MPa. It is evident from the images that the spray tip penetration is faster with increased injection pressure. Thus piston top impingement starts early with higher injection pressure. Also, noticed is the fuel impingement on the right wall at 209.4° BTDC with LPDI injector. This is not observed at this point (209.4° BTDC) when HPDI injector is used, however, there is a vortex formation towards the lower right half of the image with 10 MPa which eventually ends up with some fuel impingement on the right wall at later crank angle degrees. There is no right wall impingement when fuel is injected at 5 MPa. The spray images clearly show that the left wall impingement is significantly less with LPDI injector at 3 MPa compared to the HPDI injector, which is higher with 10 MPa compared to 5 MPa. The spray images show well-atomized drops with HPDI injector due to high injection pressure at 5 and 10 MPa. The drop size is comparatively larger with LPDI injector at 3 MPa.

**EFFECT OF INJECTION PRESSURE ON FUEL IMPINGEMENT**

Figure 7 shows the effect of fuel injection pressure on left (upper) and right (middle) wall impingement and on piston top (lower graphs) with E85 when engine is operated at 1500 rpm with full-load condition. The start of injection is at 240° BTDC for all the cases. The results confirm the observations of spray images (see the spray development in Figure 6 at 200.4° and 191.4° BTDC) that the left wall impingement is significantly less with LPDI injector at 3 MPa compared to the HPDI injector at 5 and 10 MPa. The impingement starts at 209.4, 204.9 and 193.2° BTDC for 10, 5 and 3 MPa, respectively. However, right wall impingement is significantly high with LPDI injector than that of HPDI injector. It is to be noticed that the piston top impingement starts early with higher injection pressure. However, the peak values and the overall impingement are higher with 3 MPa. The overall fuel impingement index for piston top is 10.5 with HPDI injector (at 5 and 10 MPa) and 12.3 with LPDI injector at 3 MPa. Overall, results show that HPDI injector at 5 MPa reduces the fuel impingement on in-cylinder surfaces and provides well-atomized drops for mixing. It is to be noticed at this point that the fuel index values can be directly compared between different test points for a given experimental setup. However, change in experimental setup may result in overall lower (or higher) values based on optical path and camera position; e.g. see Cronhjort and Wahlin (2004) for different illumination arrangements. Nonetheless, the experimental setup is not changed in this study. Hence the results are directly comparable. It is also expected that the overall fuel index values may be lower if the engine was fired because of relatively faster evaporation of fuel (due to hot conditions) than that of cold conditions of the motored engine.

**EFFECT OF INJECTION TIMING ON FUEL IMPINGEMENT**

Figure 8 shows the effect of fuel injection timing at 240, 210 and 180° BTDC on left (upper) and right (middle) wall impingement and on piston top (lower graphs) with E85 when the engine is operated at 1500 rpm with full-load condition. HPDI injector is used in each case with 5 MPa of injection pressure. The results show that left wall impingement increases with the advancement of injection timing. However, right wall impingement is higher when start of injection is at 180° BTDC (or at the BDC of intake) compared to 240 and 210° BTDC. Note that the right wall impingement is almost negligible with SOI at 210° BTDC. The peak impingement on piston top reduces from 240 to 210° BTDC and then again increases from 210 to 180° BTDC. One reason for the higher peak impingement levels at the injection timing of 180° BTDC than that of 240 and 210° BTDC may be due to upward piston motion (i.e. towards the spray) after the injection. Note that at 240 and 210° BTDC injection timings, piston is moving towards the BDC position (i.e. away from the spray) at the time of injection. Overall piston top impingement reduces slightly when SOI is at 180° BTDC compared to the case when SOI is at 240° BTDC. Due to the fact that more homogeneous mixture is achieved when fuel is injected during the intake stroke (by allowing more mixing time), 210° BTDC shows less overall impingement effects on in-cylinder surfaces.

**EFFECT OF NUMBER OF INJECTIONS ON FUEL IMPINGEMENT**

Figure 9 shows the effect of number of injections, i.e. single and split (or dual) injection, on left wall (upper) and piston top (lower) impingement with E85 when engine is operated at 1500 rpm with full-load condition. Note that there is no right wall impingement in both the cases (hence not shown in the plots). The start of injection is at 210° BTDC for both the cases with the injection pressure of 5 MPa. With split injection, the start of second injection is at 120° BTDC (90 CADs apart from the first injection) and the total amount of fuel is same as was used in the single injection. Note that the total amount of fuel (in split injection) is divided equally in both the injections. It is evident that the left wall impingement reduces with the split injection. The overall reduction is about 50% with split injection compared to the single injection. Also, the peak and overall impingement on the piston top is less (about 22%) with split injection compared to the single injection.
Figure 6. Spray development of E85 (SOI 240° BTDC at 1500 rpm engine speed and full-load) with LPDI at 3 MPa (left) and HPDI at 5 (middle) and 10 (right) MPa
Figure 10 shows the effect of engine speed on left (upper) and right (middle) wall impingement and on piston top (lower graphs) with E85 when engine is operated at part-load (30%) condition. Results of two engine speeds (1500 and 2000 rpm) are presented with injection timings at 240, 210 and 180° BTDC. Impingement on in-cylinder surfaces starts at later crank angles at 2000 rpm compared to the lower engine speed of 1500 rpm. Also, it is found that the overall wall impingement reduces at 2000 rpm than that of 1500 rpm. Note that the piston top impingement reduces considerably at engine speed of 2000 rpm than that of 1500 rpm when SOI is at 240° BTDC. However, there is no significant difference in overall piston top impingement at 210 and 180° BTDC with respect to the engine speed.

**EFFECT OF ENGINE SPEED ON FUEL IMPINGEMENT**

Figure 10 shows the effect of engine speed on left (upper) and right (middle) wall impingement and on piston top (lower graphs) with E85 when engine is operated at part-load (30%) condition. Results of two engine speeds (1500 and 2000 rpm) are presented with injection timings at 240, 210 and 180° BTDC. Impingement on in-cylinder surfaces starts at later crank angles at 2000 rpm compared to the lower engine speed of 1500 rpm. Also, it is found that the overall wall impingement reduces at 2000 rpm than that of 1500 rpm. Note that the piston top impingement reduces considerably at engine speed of 2000 rpm than that of 1500 rpm when SOI is at 240° BTDC. However, there is no significant difference in overall piston top impingement at 210 and 180° BTDC with respect to the engine speed.
EFFECT OF DIFFERENT FUELS ON SPRAY DEVELOPMENT

Figure 11 shows the spray development of different fuels, namely, E85 (left), E50 (middle) and gasoline (right), with high-pressure direct-injection injector at 5 MPa. The start of injection is at −210 crank angle degree for all the fuels. In each test shown, engine is operated at 1500 rpm with full-load condition. There is no significant difference in spray pattern development and in cone angle when these different fuels are compared. However, it is observed that the spray tip penetration is slightly faster with gasoline than that of ethanol-gasoline blended fuels.
Figure 11. Spray development of E85 (left), E50 (middle) and gasoline (right) at 1500 rpm engine speed and full-load with HPDI at 5 MPa, SOI 210° BTDC
EFFECT OF DIFFERENT FUELS ON FUEL IMPINGEMENT

Figure 12 shows the effect of different fuels, namely, E85, E50 and gasoline, on left (upper) and right (middle) wall impingement and on piston top (lower graphs) with high-pressure direct-injection injector at 5 MPa. The start of injection is at −210 crank angle degree for all the fuels. In each test shown, engine is operated at 1500 rpm with full-load condition. The plots show that the fuel impingement on in-cylinder surfaces starts slightly early with gasoline than ethanol-gasoline blended fuels. Note that the wall impingement is higher with gasoline than E50 and E85. There is no right wall impingement with ethanol-gasoline blended fuels. The piston top impingement is also higher with gasoline than E50, and E85 is in the middle. Overall it can be concluded that gasoline has higher impingement on in-cylinder surfaces that ethanol-gasoline blended fuels.

CONCLUSIONS

An experimental study is performed to investigate the fuel spray development and its impingement on in-cylinder surfaces inside a direct-injection spark-ignition engine. Gasoline and ethanol-gasoline blended fuels are considered with the variation in fuel injection pressure, injection timing and the number of injections. Two types of fuel injectors are used; (i) High-pressure production injector with fuel pressures of 5 and 10 MPa, and (ii) Low-pressure production-intent injector with the fuel pressure of 3 MPa. Novel image processing algorithms are developed to analyze the fuel impingement on cylinder walls and piston top. The technique is useful to optimize the fuel pressure, injection timing and the number of injections to minimize the fuel impingement on in-cylinder surfaces. Results show that split injection is an effective way to reduce the overall fuel impingement on in-cylinder surfaces. No significant difference is observed on spray development when gasoline is compared with E50 and E85, however, the spray tip penetration is slightly higher with gasoline than that of ethanol-gasoline blended fuels. Results also show that the overall impingement is higher with gasoline compared to E50 and E85.

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DEFINITIONS/ABBREVIATIONS

180° BTDC 180 crank angle degrees before TDC of compression

180° ATDC 180 crank angle degrees after TDC of compression

ATDC After top dead center

BDC Bottom dead center

BTDC Before top dead center

CAD Crank angle degree

DI Direct-injection

DISI Direct-injection spark-ignition

GDI Gasoline direct-injection

HPDI High-pressure direct-injection

IC Internal combustion

LPDI Low-pressure direct-injection

MAP Manifold absolute pressure sensor

PFI Port-fuel injection

POI Point of interest

PPI Presence probability image

RPM Revolutions per minute

SOI Start of injection

TDC Top dead center

WOT Wide open throttle