In-cylinder Combustion Visualization of a Direct-injection Spark-ignition Engine with Different Operating Conditions and Fuels

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ABSTRACT

A direct-injection and spark-ignition single-cylinder engine with optical access to the cylinder was used for the combustion visualization study. Gasoline and ethanol-gasoline blended fuels were used in this investigation. Experiments were conducted to investigate the effects of fuel injection pressure, injection timing and the number of injections on the in-cylinder combustion process. Two types of direct fuel injectors were 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. Experiments were performed at 1500 rpm engine speed with partial load. In-cylinder pressure signals were recorded for the combustion analyses and synchronized with the high-speed combustion imaging recording. Visualization results show that the flame growth is faster with the increment of fuel injection pressure. The mass fraction burned curves, calculated from the averaged in-cylinder pressure data, show that the burn duration is shorter for the higher fuel injection pressure. It is observed that with the advancement of the injection timing, the flame size increased faster than that of retarded injection timing. Comparing different fuels, it was found that the flame growth is much faster for ethanol-gasoline blended fuels than for gasoline.

1. INTRODUCTION

Improvement in fuel economy and reduction in exhaust emissions are the main goals behind the new developments in internal combustion engines. The concept of the direct-injection spark-ignition (DISI) engine has the potential of achieving such goals. For the DISI technology, the fuel is directly injected into the engine cylinder, which offers great flexibility to control accurately fueling quantity, fuel injection timing, its duration and the number of injections.

Various optical methods have been used by researchers to investigate the in-cylinder combustion process of internal combustion engines. Several studies have been reported on in-cylinder combustion visualization of both spark-ignition and diesel engines. Kozuka et al. (2003) presented high speed Schliren photographs of combustion in a premixed charge gasoline engine. Bladh et al. (2005) studied the flame propagation in a spark-ignition engine using laser-induced fluorescence of cool-flame species. The authors tracked the movement of the flame front by using two Nd:YAG lasers operated at 355 nm for two consecutive measurements within the same engine cycle with adjustable time separation between the pulses. Sauter et al. (2006) used a multi-fibre visualization technique for in-cylinder combustion visualization of a gasoline engine.

In addition, due to continuous increase in the cost of fossil fuels, there is an increasing interest in the use of alternative fuels in order to reduce reliance on conventional fossil fuels (Tongroon and Zhao, 2010). The term ‘alternative fuels’ is commonly used to identify energy sources that are not of petroleum origin. Alternative fuels that are most used at present are natural gas, liquefied petroleum gas, and biofuels. Kowalewicz and Wojtyniak (2005) presented the use of alternative fuels into the fuel market and the results of investigations applied to spark-ignition and compression-ignition engines. Ethanol is one of the most popular biofuels...
that has been tested extensively in internal combustion engines. This can be produced from biomass: potatoes, cereals, beets, sugar cane, wood, brewery waste, and many other agricultural products and food wastes in the process of fermentation. It can also be produced from natural gas and crude oil. Ethanol is not considered toxic; it is soluble in water and is biodegradable. It is more flammable than gasoline (Kowalewicz and Wojtyniak, 2005). Li et al. (2008) investigated controlled auto-ignition combustion with ethanol in a single-cylinder gasoline direct-injection engine. Ethanol has a higher octane number than gasoline and thus the potential for higher efficiency, by increasing compression ratio, exists. However, its lower heating value leads to fewer driving miles per gallon when it is used as a direct replacement in current gasoline engines (Mittal et al., 2008). Also, vehicle modifications are required to operate gasoline vehicles on high ethanol content blended fuels due to its poor startability in cold weather and the requirement of ethanol corrosion resistant materials for the fuel system. Therefore, the blend of ethanol (in gasoline) is more widely used in the vehicles that are designed to operate on gasoline fuel. This reduces the consumption of fossil fuels, and due to the high latent heat of evaporation of ethanol, combustion temperature decreases compared to vehicles operating on pure gasoline. This enables higher torque and higher thermal efficiency due to reduced cooling heat loss (Nakata et al., 2006). In the United States, the blend of 10 percent ethanol in gasoline is widely used in many fuel blends. In Brazil, the ethanol percentage is even higher and this is already in use for several years (Brinkman, 1981).

Overall, optical methods have been useful tools for experimental investigations of combustion in internal combustion engines. To the best of our knowledge, visualization studies evaluating the effects of fuel injection pressure and injection timing on direct-injection spark-ignition combustion are not available. Therefore, an experimental study is performed to investigate the effects of fuel injection pressure and injection timing (including split timings) on the combustion process inside a direct-injection spark-ignition engine. Different fuels, i.e. gasoline and ethanol-gasoline blended fuels, are used and compared in this work. Experiments are performed at 1500 rpm engine speed with partial load with variation in fuel injection pressure, injection timing and the number of injections. The paper is organized as follows. Section 2 provides details of the experimental setup. The experimental results with discussions are presented in Section 3. Finally, conclusions are drawn in Section 4.

2. EXPERIMENTAL SETUP

The engine used in this study is a four-valve (two intake and two exhaust valves) 0.4 liter single-cylinder spark-ignition engine. It has a bore diameter of 83 mm and a stroke length of 73.9 mm. A flat-top optical piston is used to provide optical access to the cylinder. This provided a compression ratio of 9.75:1. A view of the combustion chamber geometry showing intake and exhaust valves, direct-injection fuel injector, spark plug and the pressure transducer is depicted in Figure 1 (Mittal et al., 2011a). It should be noted that in this paper, zero crank angle degree corresponds to the top dead center (TDC) of the compression, and therefore, –180 crank angle degrees (CAD) corresponds to the bottom dead center (BDC) of the intake, i.e. 180° BTDC (before top dead center). Experiments are performed at 1500 rpm engine speed with partial load condition (0.45 bar MAP). This partial load condition was selected to operate the engine within the stress limits of the optical flat-top piston. Mittal et al. (2011b) also used this partial load condition to investigate the effects of charge motion control on fuel spray development and combustion using the same engine. Different fuel injection timings are considered, i.e. 240°, 210° and 180° BTDC. Two types of multi-hole fuel injectors are used; (i) low-pressure direct-injection (LPDI) 9-hole injector with the fuel pressure of 3 MPa and (ii) high-pressure direct-injection (HPDI) 7-hole injector with fuel pressures of 5 and 10 MPa. In addition, the effects of split (or dual) injection are also studied and compared to the corresponding cases of single injection by maintaining the same relative air-to-fuel ratio (λ), which is the inverse of fuel-to-air equivalence ratio (φ). With the split injection, the second injection was 90 CAD apart from the first injection and the two injection pulse widths were equal. Three different fuels, namely, gasoline, E50 and E85, were used in order to compare the 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). Table 1 summarizes the key parameters studied in this work.

<table>
<thead>
<tr>
<th>Table 1. Test parameters for combustion visualization</th>
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<tr>
<td><strong>Parameter</strong></td>
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<tr>
<td>Fuel</td>
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<tr>
<td>Fuel injection pressure</td>
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<td>Engine speed</td>
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<td>Engine load</td>
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<td>Injection timing</td>
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The combustion images are captured with a Photron APX-RS non-intensified high-speed CMOS camera system. The camera was set to operate at 10 kHz, which provided an image size of 512×512 pixels every 100 microseconds. At this frame rate with 1500 rpm engine speed, each frame corresponds to 0.9 CAD. For each test condition, the engine was first motored to reach the desired speed, i.e. 1500 rpm. Once the engine was stabilized, a signal from the Opal-RT prototype engine controller was sent to the fuel injector to trigger the start of injection at a specific crank angle position as well as to trigger the camera at the spark timing crank angle position to start recording the specified number of images in consecutive cycles. The fuel injection duration at each test point is selected to achieve the desired relative air-to-fuel ratio. For each imaging test, forty consecutive cycles are recorded to visualize the combustion process with 200 frames from each cycle. Due to optical engine limitations, fuel supply was cut off as soon as the camera recorded the specified number of cycles. In-cylinder pressure was recorded with one degree of crank angle resolution which was synchronized with the imaging signal. A Kistler piezoelectric pressure transducer was used with the measurement range varying from 0 to 250 bar. The averaged in-cylinder pressure data was then used to evaluate engine performance. Mass fraction burned (MFB) and burn durations are determined using the well-known Rassweiler-Withrow method (Rassweiler and Withrow, 1938). A linear model for the polytropic index during the combustion process is used to evaluate the pressure change due to the volume change (Mittal et al., 2009a).

3. RESULTS AND DISCUSSION

Combustion visualization and in-cylinder pressure analyses inside a direct-injection spark-ignition engine are presented. The viewable area through the optical piston is about 51% of the piston top area. Experiments are performed at 1500 rpm engine speed with partial load condition (0.45 bar MAP) as reported earlier. It is worth notifying that the fuel injection quantity is adjusted to maintain the same relative air-to-fuel ratio, i.e. $\lambda=1$, for each test point discussed in this paper. The characteristics displayed in the combustion images, such as the flame sizes, shapes and appearance, provide useful insight into what happens over the combustion period (Aleiferis et al., 2008). It should be pointed out here that the images presented are a two-dimensional representation of the three-dimensional flame development inside the engine cylinder. Also, one should notice that each combustion image shown in this paper is a reduced form of its original image size of 512×512 pixels to 420×420 pixels for improved visibility by eliminating the dark area band of pixels outside the cylinder. The physical size of a pixel is about 0.22 mm.

### 3.1. Effect of Fuel Injection Pressure on Combustion Visualization and In-cylinder Pressure Analyses

Figure 2 shows the stoichiometric combustion images of gasoline at 25.2°, 28.8°, 31.5° and 34.2° after spark timing (AST) with low-pressure direct-injection injector at 3 MPa (upper) and high-pressure direct-injection injector at 5 (middle) and 10 (lower) MPa. The injection current durations are 1.94 ms for LPDI injector at 3 MPa, and 2.35 and 2.06 ms for HPDI injector at 5 and 10 MPa, respectively. It is to be noticed that LPDI injector used in experiments has higher flow rate compared to HPDI injector. The images are enhanced so that the early flame development and its growth are clearly visible to the reader for comparison purposes. Each image shown is a gray scale representation of its RGB image for improved visibility and print quality. In each case the start of injection is at 210° BTDC and the spark timing (ST) is at 35° BTDC based on MBT. The MBT timing at each test point is determined based on the maximum value of the mean IMEP for the spark sweep. The engine is operated at 1500 rpm with partial load condition. Mittal et al. (2011a) showed less overall impingement on in-cylinder surfaces in the same engine at the injection timing of 210° BTDC. Note that the intake valves are located towards the upper half of the images. It is evident from the images that the flame growth is much slower with LPDI injector at 3 MPa compared to HPDI injector with injection pressures of 5 and 10 MPa. It is to be noticed that the early flame development starts close to the in-cylinder central location. This is mainly due to the spark plug location close to the center of the combustion chamber. It is expected that there will be some cycle-to-cycle variations in flame development. Note that some bright rich spots are also visible when fuel is injected at 10 MPa. These hot spots may be occurring due to droplet burning (Aleiferis et al., 2008). One reason for this might be the faster spray tip penetration at this higher injection pressure, and hence, more expected liquid phase gasoline impingement on the piston top. Aleiferis et al. (2008) also found these hot spots with DI injector in their work. The authors discussed the fact that gasoline is particularly
susceptible to these hot spots. However, note that most multi-component fuels exhibit a tendency toward this characteristic.

Figure 2. Flame images of gasoline ($\lambda = 1$, $ST = 35^\circ BTDC$) using LPDI injector at 3 MPa (upper) and HPDI injector at 5 (middle) and 10 MPa (lower)

Figure 3 shows the averaged in-cylinder pressure for gasoline at stoichiometric conditions using LPDI injector at 3 MPa and HPDI injector at 5 and 10 MPa. It can be observed that the peak in-cylinder pressure increases with the increase in injection pressure. It is a well-established fact that higher injection pressure results in better atomization and better mixing. Therefore, higher injection pressure leads to faster
combustion with higher peak in-cylinder pressure. It is noticed that the crank angle at which the peak in-cylinder pressure occurs is 3 CAD earlier for HPDI injector at 10 MPa than that of LPDI injector at 3 MPa. The peak in-cylinder pressures are 12.8, 13.2 and 14.8 bar at 20°, 19° and 17° ATDC for LPDI injector at 3 MPa and HPDI injector at 5 and 10 MPa, respectively.

Figure 4 shows the mass fraction burned curves calculated from the averaged in-cylinder pressure data shown in Fig. 3 for gasoline at different injection pressures using both LPDI and HPDI injectors. MFB is a measure of the fraction of thermal energy released due to combustion of air-fuel mixture inside an engine cylinder, with respect to the total energy released at the end of combustion during a cycle. This also signifies how fast the chemical energy is released (Mittal et al., 2009).

It can be observed that the burning is faster with the increase in injection pressure. As discussed earlier, this is mainly due to improved mixing with high injection pressures. Note that the 10% burn is at 0 and −2 CAD for LPDI injector at 3 MPa and HPDI injector at 10 MPa, respectively. Similarly, the 90% burn is at 27 and 23 CAD at 3 and 10 MPa of injection pressures, respectively. This shows that the total burn duration decreases with the increase in injection pressure, which is 27 and 25 CAD for LPDI injector at 3 MPa and HPDI injector at 10 MPa, respectively. It is to be noticed that in this paper, combustion duration is defined based on crank angle degrees from 10% to 90% of mass burned.

3.2. Effect of Fuel Injection Timing on Combustion Visualization and In-cylinder Pressure Analyses

Figure 5 shows the stoichiometric combustion images of gasoline at 25.2°, 28.8°, 31.5° and 34.2° after spark timing for different injection timings, i.e. 240°, 210° and 180° BTDC. The engine is operated at 1500 rpm with partial load condition and the spark timing is at 35° BTDC based on MBT. HPDI injector at 5 MPa is used in each case. As reported earlier, the injection quantity is adjusted to maintain the stoichiometric condition for each test point. The injection current durations are 2.35 ms for both 240° and 210° BTDC injection timings and 2.4 ms for 180° BTDC injection timing. Note that the injected fuel quantity is slightly higher at 180° BTDC injection timing compared to injection timings of 240° and 210° BTDC. This is expected due to improved mixing at earlier injection timings. It can be observed from the images that flame growth is faster with the advancement of injection timing. This may be due to the fact that a more homogeneous mixture is achieved with early injection by allowing more mixing time. Note that some bright spots are visible at the injection timings of 240° and 180° BTDC. This may be due to higher peak impingement levels on the piston top at these injection timings than that of 210° BTDC (Mittal et al., 2011).

Figure 6 shows the averaged in-cylinder pressure for gasoline at different injection timings using HPDI injector at 5 MPa. It can be observed that the peak in-cylinder pressure increases with the advancement of injection timing from 210° to 240° BTDC. One reason might be the improved homogeneous mixture with early injection. Note that the faster flame growth is observed with the advancement of injection timing (see Fig 5). However, mixture formation is a very complex process, which may be influenced by several factors, e.g. ambient conditions, intake manifold and intake port geometries, in-cylinder flow, fuel injection pressure etc. The crank angle at which the peak in-cylinder pressure occurs is 4 CAD earlier for the injection timing at 240° BTDC with peak in-cylinder pressure of 14.3 bar compared to 210° BTDC.
with peak in-cylinder pressure of 13.2 bar. Note that there is no significant difference in peak in-cylinder pressure values when fuel is injected at 210° and 180° BTDC. With the retard in injection timing the mean IMEP increases, which is 2.51, 2.59 and 2.66 bar for injection timings at 240°, 210° and 180° BTDC, respectively.
Figure 7. Mass fraction burned for gasoline at stoichiometric conditions for different injection timings using HPDI injector at 5 MPa

![Figure 7](image)

Figure 7 shows the mass fraction burned curves calculated from the averaged in-cylinder pressure data shown in Fig. 6 for gasoline at different injection timings. It can be observed that 10% burn is at $-4$, 0 and $-1$ CAD for injection timings at 240°, 210° and 180° BTDC, respectively. Also, note that 90% burn is at 22, 26 and 27 CAD for the injection timings at 240°, 210° and 180° BTDC, respectively. This shows that 10 to 90% burn is slower with retarded injection timing at 180° BTDC compared to advanced injection timings at 210° BTDC and 240° BTDC. The shorter burn duration at advanced injection timing is due to improved mixing by allowing more mixing time compared to retarded injection timing. However, note that the burn duration is the same with further advancement of the injection timing from 210° BTDC to 240° BTDC.

3.3. Effect of Gasoline and Ethanol-gasoline Blends on Combustion Visualization

Figure 8 shows the stoichiometric combustion images of different fuels, namely, gasoline (upper), E50 (middle) and E85 (lower) using HPDI injector at 5 MPa; $\lambda =1$ and injection timing 210° BTDC

![Figure 8](image)
in air over a broad range of equivalence ratio and initial mixture temperature using a constant volume spherical bomb technique. The results showed that at 350 K unburned mixture temperature and 100 kPa pressure, burning velocity data of isooctane-ethanol blends indicate a promotion of isooctane combustion by the addition of ethanol. The maximum burning rates occurred at equivalence ratios between 1.05 and 1.1 independent of fuel. For mixtures leaner than 0.95, isooctane-ethanol blends exhibited faster burning rates than those of neat isooctane and ethanol. It was found that at 350 K unburned mixture temperature and 100 kPa pressure, the laminar burning velocities of the two isooctane-ethanol blends (90% isooctane-10% ethanol and 80% isooctane-20% ethanol) were almost the same at all equivalence ratios considered. However, at increased unburned mixture temperatures, the 80% isooctane-20% ethanol blend exhibited faster burning rates than the 90% isooctane-10% ethanol blend.

3.4. Effect of Single and Split Injections on Combustion Visualization with E85

The effect of single and split (or dual) injections on E85 stoichiometric flame growth is shown in Figure 9 at 25.2°, 28.8°, 31.5° and 34.2° after spark timing with high-pressure direct-injection injector at 5 MPa. The engine is operated at 1500 rpm with partial load condition. The start of injection is at 210° BTDC for each case and the spark timing is at 35° BTDC for gasoline and 24° BTDC for both E50 and E85 based on MBT. Yucesu et al. (2006) studied the variation of MBT spark timing with respect to the compression ratio for gasoline and ethanol-gasoline blended fuels. At a compression ratio of 9:1, the authors found that gasoline has more advanced MBT spark timing than that of ethanol-gasoline blended fuels. However, it is to be noticed that the MBT spark timing is not only related to the compression ratio and type of fuel but also engine speed. It can be observed that the early flame development and its growth are much faster for ethanol-gasoline blended fuels than for gasoline. At 34.2° after spark timing, E85 flame almost propages through the whole cylinder. Gulder (1984) investigated the laminar burning velocities of isooctane, ethanol and isooctane-ethanol blends in air over a broad range of equivalence ratio and initial mixture temperature using a constant volume spherical bomb technique. The results showed that at 350 K unburned mixture temperature and 100 kPa pressure, burning velocity data of isooctane-ethanol blends indicate a promotion of isooctane combustion by the addition of ethanol. The maximum burning rates occurred at equivalence ratios between 1.05 and 1.1 independent of fuel. For mixtures leaner than 0.95, isooctane-ethanol blends exhibited faster burning rates than those of neat isooctane and ethanol. It was found that at 350 K unburned mixture temperature and 100 kPa pressure, the laminar burning velocities of the two isooctane-ethanol blends (90% isooctane-10% ethanol and 80% isooctane-20% ethanol) were almost the same at all equivalence ratios considered. However, at increased unburned mixture temperatures, the 80% isooctane-20% ethanol blend exhibited faster burning rates than the 90% isooctane-10% ethanol blend.

3.5. Effect of Gasoline and Ethanol-gasoline Blended Fuels on In-cylinder Pressure Analyses with Single and Split Injections

Figure 10 shows the averaged in-cylinder pressure for gasoline, E50 and E85 at stoichiometric conditions using both single and split injections. The peak in-cylinder pressure is higher for gasoline than for ethanol-gasoline blended fuels. It is about 13.2, 10.7 and 11.1 bar for gasoline, E50 and E85, respectively. Abdel-Rahman and Osman (1997) showed that at a compression ratio of 10:1, the E10 fuel blend increases the maximum in-cylinder pressure over that of pure unleaded gasoline (Yucesu et al., 2006). However, the ethanol percentage above 10% results in a decrease of maximum in-cylinder pressure to a value even lower that E0's pressure.
Yucesu et al. (2006) explained that the addition of ethanol to gasoline has two effects on the fuel blend properties: (1) an increase of the octane number, and (2) a decrease in the heating value. These effects have opposite results in terms of engine performance. The first effect dominates up to an ethanol percentage of 10%, after which the second effect starts to take over. The crank angle at which the peak in-cylinder pressure occurs is 7 CAD earlier for gasoline compared to E85 (see Figure 10). Note that the mean IMEP is higher for ethanol-gasoline blended fuels (2.68 and 2.67 bar for E50 and E85, respectively) than that of gasoline (2.59 bar). With split injection, the peak in-cylinder pressure increases for all fuels compared to their corresponding single injection cases; however, the effects are more significant on ethanol-gasoline blends than that of gasoline. The peak in-cylinder pressures with split injection for gasoline, E50 and E85 are 13.7, 11.9 and 13.1 bar, respectively. It is interesting that the crank angle at which the peak in-cylinder pressure occurs is only 1 CAD earlier for gasoline than that of E85 with split injection. With split injection, the mean IMEP for gasoline increases to 2.73 bar compared to 2.59 bar of corresponding single injection case. However, the effect of split injection on mean IMEP is not significant on E50 and E85 (2.70 bar for both) compared to the corresponding single injection cases.

Figure 11 shows the mass fraction burned curves calculated from the averaged in-cylinder pressure data shown in Fig. 10 for different fuels. As shown in the figure, the 10% burn is 5 CAD earlier for gasoline than that of E85 with single injection. Note that the MBT spark timing is advanced for gasoline compared to E50 and E85. The burn duration (10 to 90%) is 26 CAD for gasoline and increases with ethanol content (28 and 27 CAD for E50 and E85, respectively). Li et al. (2008) found that the controlled auto-ignition combustion with gasoline obtained the earlier start of combustion and
higher peak in-cylinder pressure compared to ethanol. It can 
be observed (see Fig. 11) that with split injection the 10% 
burn is faster for all the fuels compared to the corresponding 
single injection cases. This is due to early start of combustion 
process with split injection compared to single injection. 
With split injection, the combustion process starts 2 CAD 
earlier for gasoline and E50, and 4 CAD earlier for E85 than 
that of corresponding single injections. Note that the 10% 
burn (with split injection) is only 1 CAD earlier for gasoline 
than that of E85. The 10 to 90% burn duration for E50 and 
E85 is 28 and 27 CAD, respectively, which is similar to their 
corresponding single injection data. However, burn duration 
increases by 1 CAD for gasoline compared to its 
corresponding single injection case.

4. CONCLUSIONS

A single-cylinder direct-injection spark-ignition engine was 
used for combustion studies of different fuels, namely, 
gasoline, E50 and E85. The engine in use provided optical 
access to the combustion chamber. Experiments were 
conducted to investigate the effects of fuel injection pressure, 
ignition timing and the number of injections on the in-
cylinder combustion process. Two types of direct-injection 
gasoline fuel injectors were 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-cylinder pressure data, synchronized with high-
speed imaging, was recorded for combustion analysis.

Result show that flame growth is faster with the increase in 
fuel injection pressure. Higher fuel injection pressure led to 
faster combustion and higher peak in-cylinder pressure. It 
was also observed that flame growth is faster with the 
advancement of fuel injection timing. In comparing 
combustion events using different fuels, it was found that 
flame growth becomes faster as the ethanol content of 
ethanol-gasoline blended fuels increases. With split injection, 
the flame growth of E85 was further enhanced than that of its 
corresponding single injection case. Higher peak in-cylinder 
pressures were observed for all fuels with split injection 
compared to their corresponding single injection cases. 
However, the effects of split injection on gasoline are less 
significant than on E85.

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DEFINITIONS/ABBREVIATIONS

\( \lambda \) - Relative air-to-fuel ratio

\( \phi \) - Fuel-to-air equivalence ratio

180\(^\circ\) BTDC - 180 crank angle degrees before TDC of compression

25.2\(^\circ\) AST - 25.2 crank angle degrees after spark timing

BDC - Bottom dead center

BTDC - Before top dead center

CAD - Crank angle degree

DI - Direct-injection

DISI - Direct-injection spark-ignition

HPDI - High-pressure direct-injection

IMEP - Indicated mean effective pressure

LPDI - Low-pressure direct-injection

MAP - Manifold absolute pressure

MBT - Maximum Brake Torque

MFB - Mass fraction burned

RPM - Revolutions per minute

ST - Spark timing

TDC - Top dead center