Spray and Combustion Visualization in an Optical HSDI Diesel Engine Fuelled with Biodiesel and Diesel using Multiple Injection Strategy

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Abstract
An optically accessible single-cylinder high speed direct-injection (HSDI) Diesel engine was used to study the spray and combustion processes using biodiesel fuel with multiple injection strategies. Influences of injection timings on liquid fuel evolution and combustion characteristics were studied under similar loads. In-cylinder pressure was measured for heat release analysis. High-speed combustion video was also captured for all the studied cases using the same frame rate. NOx emissions were measured in the exhaust pipe. Different combustion modes including conventional diesel combustion and low-temperature combustion were confirmed from the heat release rates and the combustion images. Soot luminosity was found consistently lower for bio-diesel fuel than the European diesel fuel showing less soot emissions for all the cases. However, for NOx emissions, under conventional combustion cases, it was found that biodiesel fuel leads to increased NOx emissions. But for low-temperature combustion modes, NOx emissions were lower for bio-diesel fuel than the diesel fuel. Simultaneous reduction of NOx and soot was achieved for advanced low-temperature combustion mode.

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### Introduction

Compared to gasoline engines, Direct-Injection (DI) diesel engines offer higher thermal efficiency. Because of worldwide environmental concerns, the emission regulations have become more and more stringent. Exhaust emissions like oxides of nitrogen (NOx) and Particulate Matter (PM) must be reduced for diesel engines to meet future emission standards. New techniques or combustion concepts should be developed to help solve these problems.

Multiple injection strategies have been reported for simultaneous reduction of NOx and PM in large-bore DI diesel engines [1-3] and small-bore HSDI diesel engines [4-6]. Results by Nehmer and Reitz [1] showed that pulsed injection might provide a method to reduce PM and allow for reduction of NOx from controlled pressure rise. The effectiveness of double, triple, and rate shaped injection strategies to simultaneously reduce NOx and PM was also evaluated [2]. Numerical simulations were carried out to explore the mechanism of soot and NOx reduction for multiple injection strategies [3]. The multiple injection strategy has a similar effect to the retarded single injection on NOx reduction. Reduced soot emissions are due to the fact that the soot producing rich region is not replenished when the injection pulse is terminated and restarted. Zhang [4] investigated the effect of a pilot injection on NOx, soot emissions, and combustion noise in a small diesel engine. Smoke emission was seen relevant to the pilot flame and reducing the pilot flame at the main injection starting timing can reduce smoke emissions. By optimizing the EGR rate, pilot timing and quantity, main timing, and dwell between the main and pilot injections, simultaneous reduction of NOx and PM was obtained in an HSDI diesel engine [5]. Another study on pilot injection was done by Tanaka et al. [6]. It was shown that simultaneous reduction of combustion noise and emissions is possible by decreasing the influence of the pilot burned gas through minimizing the fuel quantity and advancing the pilot injection timing.

Combustion concepts like Homogeneous Charge Compression Ignition (HCCI) combustion have been shown to be effective for NOx and PM reduction. Onishi et al. [7] discovered this combustion concept, called Active Thermo-Atmosphere Combustion (ATAC). They showed that ATAC is a new combustion mode different from combustion in traditional Spark-Ignition (SI) and Compression-Ignition (CI) engines. Another research group also found a similar phenomenon [8]. The studies in four-stroke SI-engines were done by Najt and Foster [9] and Thring [10]. The HCCI combustion in diesel engines was reported much later. Due to the low volatility of diesel fuel, it is difficult to form a homogeneous charge. Because of the flexibility of the multiple injection strategies in controlling the mixing and combustion processes, they have also been employed in DI diesel engines for HCCI-like combustion modes.

Hashizume et al. [11] proposed a low soot solution, called MULtiple stage Diesel Combustion (MULDIC) for higher load operating conditions. Although a sooting luminous flame was observed, this luminous flame disappeared quickly, and most of the soot was oxidized rapidly. Smoke and NOx were seen reduced. A multipulse HCCI combustion study was done by Su et al. [12]. They used multiple short injection pulses for early injection or followed by a main injection near Top-Dead-Center (TDC). For very early injection, great increase in Total Hydrocarbon (THC) emissions was seen. A study by Hasegawa and Yanagihara [13] employed two injections called UNiform BUlky combustion System (UNIBUS). The first injection was used to form a pre-mixture. The second injection was used as an ignition trigger. The ignition of the premixed gas could be controlled by the second injection when the early injection maintained a low temperature reaction. Luminous flame was not observed and many ignition points appeared in the combustion chamber. A dual mode operation was used in a Narrow-Angle Direct-Injection (NADITM) concept [14]. The engine is operated in HCCI combustion under partial loads and in conventional diesel combustion at full-load conditions. In order to avoid fuel-liner impingement, a narrow-angle injector was employed. HCCI combustion in a small-bore HSDI diesel engine was investigated by using early multipulse short injection pulses during the compression stroke [15]. In order to decrease the fuel-liner impingement, they used a narrow-angle injector. Results showed a dramatic reduction of NOx and smoke, while HC and CO substantially increased. Another two-stage diesel fuel injection PCCI combustion was carried out by Kook and Bae [16]. A large fuel fraction was injected very early before TDC. A second injection with a small amount of fuel was injected near the compression TDC to ignite all the air-fuel mixtures. Results showed that a narrow-angle injector is favorable for very early injection.

Agricultural fat and oils, in raw or chemically modified forms, have the potential to supplant a significant proportion of petroleum-based fuels. Bio-diesel is of particular interest to the automobile industry and other areas in energy and environment because it significantly reduces particulate matter (PM), hydrocarbon (HC) and carbon monoxide (CO) emissions. The engine testing from three different engines, a Cummins N-14 engine, a Cummins B5.9 engine, and a DDC Series 50 engine showed average reductions of 84.4% in HC, 40.5% in CO, and 38.0% in PM emissions [17].

In addition to its benefits to Environmental Protection Agency (EPA) regulated exhaust emissions of PM, HC and CO, bio-diesel contributes less to global warming than fossil fuels due to its closed carbon cycle.
There is almost no net increase of carbon dioxide (CO2) emission from bio-diesel combustion. Bio-diesel is also the only alternative fuel that has passed the EPA-required Tier I and Tier II Health Effects testing requirements of the Clean Air Act Amendments of 1990. Moreover, bio-diesel is particularly attractive because it is a renewable fuel that can be replenished through the growth of plants or production of livestock, and it has the potential to supplant a fraction of petroleum-based fuels.

One of the main factors impacting the use of bio-diesel is its NOx emission. Bio-diesel has been criticized for its up to 15% higher brake specific NOx emissions comparing to diesel fuels. In the last ten years, numerous studies and measurements of NOx emissions from diesel engines fueled with bio-diesel have been published [17-25]. However, most of the reports were focused on conventional diffusion combustion of bio-diesel. Low temperature combustion has its unique advantage to inhibit the formation of NOx during the combustion process. In addition, compared to petroleum diesel fuel, the cetane number of bio-diesel is generally higher, which results in easier auto-ignition and is beneficial to the low temperature combustion process. The low temperature combustion of bio-diesels has not been reported in the literature, it is of practical interests to investigate the combustion characteristics of bio-diesels in a small bore HSDI diesel engine and compare with the low sulfur European diesel.

Combustion visualization gives a qualitative feel for the effects of differing injection strategies. Imaging of the natural flame luminosity from the combustion event through the use of an optical engine has been a technique that has garnered widespread use [26-32]. These works identified ignition locations, flame temperatures, evidence of flame wall interaction and late cycle events such as soot oxidation.

In the current work, the effects of different fuels on combustion processes in an optical engine with realistic piston geometry using different injection timings will be presented.

**Optical Engine and Facility**

The optical engine was built using a single cylinder DIATA research engine supplied by Ford Motor Company. Key aspects of the DIATA engine are listed in Table 1. The design is based on the drop-liner design employed at Sandia National Labs in Livermore, CA. Optical access to the combustion chamber is attained from the side through a window just below the head, or from below through the fused silica piston top, which is attached to a Bowditch-type piston extension as shown in Fig. 1. The optical engine design maintains the geometry of the ports and combustion chamber of the original engine. A complete description of the optical engine used in this study can be found in a previous publication [33].

A Bosch common-rail electronic injection system was used on the research engine, capable of injection pressures up to 1350 bar. A Valve-Covered-Orifice (VCO) injector with six 0.124 mm holes placed symmetrically in the nozzle tip was used. The spray injection angle is 150 degree.

<table>
<thead>
<tr>
<th>Bore</th>
<th>70 mm</th>
</tr>
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<tbody>
<tr>
<td>Stroke</td>
<td>78 mm</td>
</tr>
<tr>
<td>Displacement/Cylinder</td>
<td>300 cc</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>19.5:1</td>
</tr>
<tr>
<td>Swirl Ratio</td>
<td>2.5</td>
</tr>
<tr>
<td>Valves/Cylinder</td>
<td>4</td>
</tr>
<tr>
<td>Intake Valve Diameter</td>
<td>24 mm</td>
</tr>
<tr>
<td>Exhaust Valve Diameter</td>
<td>21 mm</td>
</tr>
<tr>
<td>Maximum Valve Lift</td>
<td>7.30/7.67 mm (Intake/Exhaust)</td>
</tr>
<tr>
<td>Intake Valve Opening</td>
<td>13 CAD ATDC (at 1 mm valve lift)</td>
</tr>
<tr>
<td>Intake Valve Closing</td>
<td>20 CAD ABDC (at 1 mm valve lift)</td>
</tr>
<tr>
<td>Exhaust Valve Opening</td>
<td>33 CAD BBDC (at 1 mm valve lift)</td>
</tr>
<tr>
<td>Exhaust Valve Closing</td>
<td>18 CAD BTDC (at 1 mm valve lift)</td>
</tr>
</tbody>
</table>

Table 1. Engine specifications of the single cylinder DIATA research engine.

A 12-bit high-speed video camera, Phantom v7.3, built by Vision Research, Inc. was used to obtain the combustion videos. Due to the extensive optical access provided by the optical DIATA engine, 3-D like combustion imaging is applicable by adding two mirrors beside the optical engine [34-36]. Combustion images were obtained using the high-speed video camera by setting the operating frame rate at 12000 frames per second with the resolution at 512X256 to capture the images from the bottom and side. For all of the cases, the exposure time was 2 us. The same lens f-stop was used; therefore perceived intensities are directly comparable.

National Instruments LabView version 6.0 is used as the data acquisition and timing software for the engine. An optical shaft encoder with 0.25 crank angle resolution is used to provide the time basis on which all data acquisition timing systems are operated.


Experimental Operating Conditions

Different combustion modes with multiple-injection strategies were studied for pure diesel and biodiesel fuels. The injection strategy included a small first injection with an early pre-TDC timing and a large main injection at or after TDC. The first injection fuel quantity was fixed at 1.5 mm$^3$ for both fuels. The injection timing was changed to achieve different combustion modes by controlling the ambient environment during the main injection event. The injection pressure was maintained at 800 bar. The first injection timing changed from –40 CAD to –20 CAD ATDC by a step of 10 CAD. The main injection timings were chosen to be at TDC and 10 CAD ATDC. Because there was no ignition for first injection at –40 CAD ATDC followed by main injection at 10 CAD ATDC, this case was not included in the current operation conditions. There are five conditions for each fuel. Therefore, ten conditions for the two fuels. The operating conditions are tabulated in Table 2. Some selected physical properties of the two fuels are listed in Table 3.

Table 2. Summary of the selected operating conditions.

<table>
<thead>
<tr>
<th>Fuel Type</th>
<th>Case Number</th>
<th>Injection Pressure [bar]</th>
<th>First Injection SOI [CAD ATDC]</th>
<th>First Injection Fuel Quantity [mm$^3$]</th>
<th>Main SOI [CAD ATDC]</th>
<th>Main Duration [ms]</th>
<th>IMEP [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>B0</td>
<td>1</td>
<td>800</td>
<td>-40</td>
<td>1.5</td>
<td>10</td>
<td>120</td>
<td>3.98</td>
</tr>
<tr>
<td>B0</td>
<td>2</td>
<td>800</td>
<td>-30</td>
<td>1.5</td>
<td>10</td>
<td>116</td>
<td>3.98</td>
</tr>
<tr>
<td>B0</td>
<td>3</td>
<td>800</td>
<td>-20</td>
<td>1.5</td>
<td>0</td>
<td>120</td>
<td>3.99</td>
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<tr>
<td>B0</td>
<td>4</td>
<td>800</td>
<td>-30</td>
<td>1.5</td>
<td>10</td>
<td>123</td>
<td>3.98</td>
</tr>
<tr>
<td>B0</td>
<td>5</td>
<td>800</td>
<td>-20</td>
<td>1.5</td>
<td>0</td>
<td>113</td>
<td>4.00</td>
</tr>
<tr>
<td>B100</td>
<td>1</td>
<td>800</td>
<td>-40</td>
<td>1.5</td>
<td>0</td>
<td>138</td>
<td>4.00</td>
</tr>
<tr>
<td>B100</td>
<td>2</td>
<td>800</td>
<td>-30</td>
<td>1.5</td>
<td>10</td>
<td>129</td>
<td>4.08</td>
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<tr>
<td>B100</td>
<td>3</td>
<td>800</td>
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<td>1.5</td>
<td>0</td>
<td>133</td>
<td>4.07</td>
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<td>1.5</td>
<td>10</td>
<td>134</td>
<td>3.93</td>
</tr>
<tr>
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<td>5</td>
<td>800</td>
<td>-20</td>
<td>1.5</td>
<td>10</td>
<td>122</td>
<td>4.02</td>
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Table 3. Summary of the selected properties of the two fuels.

<table>
<thead>
<tr>
<th>Fuel Type</th>
<th>Property</th>
<th>B0</th>
<th>European Low Sulfur Diesel Fuel</th>
<th>Soybean Biodiesel Fuel</th>
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<tr>
<td></td>
<td>Specific Gravity</td>
<td>0.837</td>
<td>0.877</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Sulfur (ppm)</td>
<td>196</td>
<td>~0</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Flash Point (°F)</td>
<td>130.4</td>
<td>&gt;201</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Boiling Point (°F)</td>
<td>368.3 (BP)</td>
<td>518.0 (50%)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Viscosity (cps)</td>
<td>3.2 (@40°C)</td>
<td>71 (@25°C)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Cetane Number</td>
<td>54.0</td>
<td>59.9 [ref. 100]</td>
<td></td>
</tr>
</tbody>
</table>

Results and Discussion

4.1 In-cylinder Pressure and Heat Release Rate

The pressure traces for the ten cases are shown in Fig. 2 grouped by the two fuels. The in-cylinder pressure shows quite different features for different injection timings. However, the fuel effects on the in-cylinder pressure are not so obvious. For different injection strategies, the in-cylinder pressures are quite similar and closely follow the motoring pressure before -20 CAD ATDC. Then with heat release from the first injection, the pressure starts to deviate from the motoring curves. Some pressure drop due to fuel evaporation can be seen for the cases with first injection timing at –20 CAD ATDC. The heat release from the first injection at –40 CAD ATDC is small leading to a small increase in the TDC pressure compared with the motoring pressure. The cases with an injection timing at –30 CAD ATDC have higher TDC pressure than those with first injection timing of –40 CAD ATDC due to more heat release from the first injection. This is due to that the air-fuel mixture is too lean to combust completely for the first injection of Case 1. But for Cases 3 and 5 with a first injection timing at –20 CAD ATDC, a significantly higher heat release is seen than Cases 1, 2, and 4. This results in higher ambient temperature and pressure for Cases 3 and 5. Due to the difference in the heat release of the first injection, the ignition delay of the main injection is greatly different. At the same main injection timing, an early first injection leads to longer ignition delay and more rapid pressure rise. By delaying the main injection timing, the ignition delay is elongated for Cases 4 and 5 compared with Cases 2 and 3. A lower pressure rise rate is seen for the retarded main injection cases. A lower pressure rise rate results in less combustion noise for quieter engine operation. The fuel effects on the in-cylinder pressure are hardly seen only based on the in-cylinder pressure. No obvious difference is found for the two fuels. The ignition delay and peak pressures are quite similar for the same injection strategy.

Figure 2: In-cylinder pressure for the two fuels with multiple-injection strategy

The computed heat release rate curves for these conditions are shown in Fig. 3. Injection strategy greatly influences the heat release pattern and the combustion mode. For Case 1 with an early first injection timing at –40 CAD ATDC, there is only a small amount of heat from the first injection. The ambient temperature is lower for Case 1, which leads to a longer ignition delay for the main injection. As a consequence, a premixed combustion mode is observed with a much higher heat
release rate peak. By retarding the first injection timing, more heat is released from the first injection and the ambient temperature is higher. The ignition delay becomes shorter with a later first injection timing. Relatively high heat release rate peaks are seen for Cases 3 and 5. When the ignition delay becomes shorter, the combustion mode becomes more diffusion-combustion dominated. This diffusion combustion mode is observed for Case 3 with a small portion of premixed combustion and a large portion of mixing controlled combustion. The heat release rate peak is much smaller than that of Case 1. The combustion mode of Case 2 is between Case 1 and Case 3. For Case 5, there is more premixed combustion compared with Case 3, but the heat release rate has a long tail indicating slow diffusion combustion after the premixed combustion phase. With a retarded main injection timing for Case 4, the ignition delay is greatly elongated. The heat release rate shows a single peak premixed combustion mode. Again, the fuel effects on heat release rate curves are hardly seen from the current results.

Figure 3: Heat release rate for the two fuels with multiple-injection strategy

4.2 Combustion Images

The combustion processes for the ten cases were visualized using the high-speed video camera. The camera was triggered by the pulse from the TDC shaft encoder. This frame rate is corresponding to an interval of 0.75 CAD between two sequential images at an engine speed of 1500 rpm. From the combustion images, there is no flame seen for the first injection of Case 1. For Case 2, there are some weak flame points with very low flame intensity. Thus, only the flame of the first injection for Case 3 is presented in this section followed by the main injection flame images for all five injection strategies with diesel and biodiesel fuels.

The combustion images of the first injection of Case 3 for the two fuels are shown in Fig. 4. The flame images are greatly different for different fuels. Ignition occurs at about –7.75 CAD ATDC for B0. Referring to the heat release rate in Fig. 3, the heat release rate has passed its peak when the early flames occur. This shows that the first injection combustion is an HCCI-like combustion with most of the heat released in a flameless combustion process for B0. Only some pieces of small weak flame pockets are observed in the bowl region. At about –4.00 CAD ATDC, the early flame becomes much weaker and starts to disappear. Relatively strong flame is seen for B100 with a more distributed structure. There are some local luminous flames corresponding to the spray jets in the bowl region. For the two fuels, the early flame occurs at a timing close or after the heat release rate peak timing, which implies an HCCI-like or PCCI-like combustion mode for the first injections of the four fuel blends. At a timing close to TDC, the combustion flame for the first injection are almost burnt out. These observations are ascribed to the low volatility of the bio-diesel fuel. As discussed in the previous section, the mixing time of the first injection for Case 3 is very short. Under this short mixing time, the effect of low volatility becomes more obvious for the bio-diesel fuel. The fuel vapor diffuses and penetrates to a shorter distance than the pure diesel fuel. The low volatility of the bio-diesel makes the fuel vapor more confined in the region around the liquid jets in a richer condition. Therefore, strong flame luminosity is observed for bio-diesel fuel.
Figure 4: Combustion images of the first injection for Case 3 at different crank angles for the two fuels with multiple-injection strategy.

The combustion images of the main injection for Case 1 are depicted in Fig. 5. Ignition occurs at about 7.25 CAD ATDC for both fuels. The early flame is quite weak and located around the spray tip near the bowl wall. Some early flames are seen in the squish region, which implies that there is some fuel vapor in the squish region due to the spray bowl lip impingement. The flame development after ignition makes the weak flame fill in the squish region. The flame in squish region has lower flame luminosity with little soot formation. At 11.00 CAD ATDC, little flame is seen in the squish region for B0. Some is left for B100 in the squish region. This shows a slightly stronger fuel impingement on the bowl lip occurring for the B100 than the diesel fuel. But for this main injection timing, most of the fuel is confined in the bowl region. Strong luminous flame is seen in the bowl region for both fuels at 11.00 CAD ATDC. However, a careful observation shows a slightly lower flame intensity for B100 than B0. This can be related to the soot formation in the combustion flame with higher soot luminosity indicating higher soot formation. The flame structure is similar for both fuels. For late cycle crank angles, the flame is most confined in the bowl region and burnt out due to soot oxidation. At 35 CAD ATDC, the flame is almost burnt out for both fuels with very low flame intensity. By comparing the late cycle combustion images, a great difference in the soot oxidation process can be observed. The soot oxidation process for B100 is obviously higher than that of B0. For the bio-diesel fuel, although the fuel has a lower volatility, due to its higher oxygen content than the pure diesel fuel, the soot formation rate is lower and the soot oxidation rate is higher. Both factors result in great reduction in soot emissions for the bio-diesel fuel compared with the pure petro-diesel fuel.

Figure 5: Combustion images of the main injections for Case 1 of the two fuels.

Compared with the combustion images of Case 1, different features are observed in Fig. 6 for Case 2. Ignition occurs at about 5.00 CAD ATDC for B100 with B0 slightly earlier at 4.25 CAD ATDC. Ignition locations are similar in the vicinity of the spray tip near the bowl wall or over the bowl lip. Flame develops further into the bowl and the squish region due to the flame impingement on the bowl lip from the push of the injection momentum. The fact that liquid spray is injected into early luminous flames shows the diffusion flame
combustion feature. Near the end of fuel injection at 8.00 CAD ATDC, strong flame is observed in the bowl near the bowl wall and in the squish region. Without the jet momentum after the end of injection, flame develops upstream to the central bowl region. Under the strong swirl motion in the combustion chamber, a donut shape flame is formed in the piston bowl at 11.00 CAD ATDC. The overall flame luminosity is relatively close for both cases with B0 slightly higher with more soot formation. After the flame luminosity passes its peak, the soot oxidation process becomes dominant and it makes the soot concentration start to drop. Again, a higher soot oxidation rate is observed for B100 than B0. Due to a higher soot concentration, the oxidation process takes longer than Case 1 and some late cycle flames come out of the bowl going to the region above the piston.

With further retardation of the first injection for Case 3, the ambient temperature during the main injection becomes much higher than that of Case 2, which results in a more diffusion combustion mode as seen in Fig. 7. The combustion process is poor in terms of soot emissions with a significantly high soot concentration formed in the flame. Ignition delay is very short for Case 3, namely about 1.5 CAD after the liquid fuel comes out of the nozzle. Ignition occurs at 2.75 CA ATDC for all fuels. The location is different from Cases 1 and 2 and is around the liquid jet near or behind the spray tip. Early flame develops fast with luminous flames enclosing the liquid spray at about 5.00 CAD ATDC. The liquid penetration is further limited by the high temperature early flames. The flame is pushed downstream by the liquid jet momentum and impinges on the bowl lip. Then flame is split into two parts into the squish region and the piston bowl. At the end of injection, namely at about 8.00 CAD ATDC, strong luminous flames fill in the squish region and the near wall region in the bowl. The jet structure corresponding to the six spray jets is obviously observed. The longer spray-flame overlap greatly inhibits the air-fuel mixing and increases the soot formation during the early combustion stage. The effects of different volatilities for the bio-diesel and diesel fuels are quite limited because of the short liquid penetration. At 11.00 CAD ATDC, highly luminous flame fills out the whole piston bowl. Comparing the combustion images at 17, 29 and 41 CAD ATDC, it is apparent that B100 has a higher soot oxidation rate than diesel fuel. Due to the high soot concentration in the combustion flame, it takes longer to burn out during late cycle crank angles. Late cycle flame comes out of the bowl due to air motion and piston downward movement. The shorter liquid penetration for Case 3 limits the air-fuel mixing region to a smaller region than in Case 2. The available oxygen for Case 3 is less than that of Case 2. Based on these results, the combustion and soot formation processes for the bio-diesel blends are not only determined by the mixing rate, namely fuel volatility, but also greatly influenced by the oxygen availability in the combustion flame. When the mixing rate is limited, the role of the oxygen content becomes dominant.

![Combustion images of the main injections for Case 2 of the two fuels](image-url)
With retarded main injection timing for Case 4, the combustion flame is totally different from Case 2 as seen in Fig. 8. Due to a lower ambient temperature, ignition delay occurs near the end of injection from about 17.00 to 17.75 CAD ATDC. The difference of the early flame in Case 4 from Case 2 is that the early flame develops more slowly with a weak flame luminosity. This lower temperature flame elongates the air-fuel mixing time without soot formation, even though there is some evidence of liquid fuel going into the early flames. Another difference is that the liquid penetration for Case 4 is longer than Case 2 with more oxygen available during the combustion process. The larger mixing region also makes the mixture leaner than Case 2. Thus, a weak and more uniform combustion flame is observed in Case 4 for both fuels. The combustion is similar to PCCI-like or the low-temperature mixing controlled combustion mode. The flame intensity is significantly lower than Cases 1-3 with less soot formation in the flame.

**Figure 7**: Combustion images of the main injections for Case 3 of the two fuels

**Figure 8**: Combustion images of the main injections for Case 4 of the two fuels
Although the main injection timing for Case 5 is greatly retarded compared with Case 3, the ambient air temperature is high enough to generate a diffusion flame combustion mode with a very short ignition delay as shown in Fig. 9. Ignition occurs at about 13.25 CAD ATDC for both fuels around the liquid jets. A longer overlap of liquid spray and flame leads to highly luminous sooting flames. Due to lower ambient pressure and temperature, the liquid penetration is longer than Case 3. Compared with Case 3, a longer liquid penetration makes the flame more distributed in the squish region for Case 5. The flame-bowl lip impingement also pushes more flame into the squish region at later crank angles. Bio-diesel fuel has less flame luminosity showing less soot formation than B0. As the temperature drops to less than 1700 K, the oxygen concentration has little influence on the soot oxidation rate. However, the soot oxidation rate is always proportional to the soot concentration [37]. For Case 5 with a retarded main injection timing, the flame temperature is lower than Case 3, especially for late crank angles. Therefore, the soot oxidation rate is mainly dependent on the soot concentration in the flame. Based on this observation, the soot reduction effects of the bio-diesel fuel will be more pronounced when the combustion occurs close to TDC with a higher flame temperature. For a late main injection strategy, the main contribution for the soot reduction of bio-diesel comes from a lower soot formation process but not the faster soot oxidation process.

From the combustion images, it is found that injection strategies greatly influence the combustion mode. Two important factors control the combustion process and the soot generation process. The first one is the mixing process, which dominantly determines the combustion mode transition from premixed combustion to typical diffusion flame combustion. Of course, the mixing process is greatly influenced by different injection parameters. The other one is the oxygen content in the fuel. A higher oxygen content helps in reducing soot formation and increasing soot oxidation. Therefore, the soot exhaust emissions are lower for the bio-diesel fuels.

4.3 NOx Emissions

Because of thermal loading limitation for the optical engine, it is risky to run the optical engine in a continuous firing mode. Instead, the optical engine was operated in a skip-fire mode with an injection pattern of 3 injection cycles followed by 10 motoring cycles. An MEXA-720NOx analyzer was used to measure the diluted NOx concentration. This non-sampling NOx analyzer provides faster response by using a NOx sensor.
installed on the exhaust pipe. The sensor has a response time about 0.7 second. The final NOx emission values were corrected based on the duty-cycle of the operation.

The NOx emission results are quite complicated. For Cases 1 and 2, the NOx emissions are relatively lower than that of Case 3. This shows that the diffusion combustion mode results in higher NOx emissions. By retarding the main injection timing, NOx emissions are reduced. Much lower NOx emissions are found for Case 4 further confirming the PCCI-like combustion mode for this case. Higher NOx emissions are seen for higher bio-diesel content with conventional like combustion mode as Cases 2 and 3. But the NOx emission is lower for B100 under a low temperature combustion mode with large gap between the two injection events as Cases 1, 4, and 5. Based on the above discussion, it is concluded that a multiple-injection strategy can greatly change the emission behavior for different fuels under similar load conditions. Fuel effects can be used to fine-tune the combustion performance. It is also observed that there is no certain trend obtained for the effects of bio-diesel on the NOx emissions. This is quite different from the single-injection strategy cases with conventional combustion, where NOx emissions increase for bio-diesel fuel. Under a certain injection strategy, it is possible to have lower NOx emissions B100 than B0. The soot luminosity is lower for B100. Simultaneous reduction of soot and NOx emissions is possible by using a multiple-injection strategy with a longer dwell time between the two injections.

Conclusions
In this paper, the effects of European low sulfur diesel fuel and bio-diesel fuel on the combustion process were experimentally investigated in a small-bore HSDI diesel engine using multiple injection strategies. Five injection strategies were studied showing the injection timing influences on the combustion modes. Less luminous flame was observed for biodiesel fuel than the European low sulfur diesel fuel indicating lower soot concentration in the combustion flame. Compared with conventional combustion mode, low-temperature combustion modes resulted in lower soot formation. For conventional-like mode, namely case 2 and 3, higher NOx emission was seen for bio-diesel fuel than the European low sulfur diesel fuel. However, for other cases, bio-diesel fuel led to lower NOx emissions. For a certain type of fuel, retarding injection timing resulted in significant reduction in NOx emissions. A multiple-injection strategy can greatly change the emission behavior for different fuels under similar load conditions. Fuel effects can be used to fine-tune the combustion performance. It is also observed that there is no certain trend obtained for the effects of bio-diesel on the NOx emissions. This is quite different from the single-injection strategy cases with conventional combustion, where NOx emissions increase for bio-diesel fuel. Under a certain injection strategy, it is possible to have lower NOx emissions B100 than B0. The soot luminosity is lower for B100. Simultaneous reduction of soot and NOx emissions is possible by using a multiple-injection strategy with a longer dwell time between the two injections.

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