Optical Study of Gasoline Substitution Ratio and Diesel Injection Strategy Effects on Dual-fuel Combustion

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1. Abstract

Ever growing population and increased vehicles have resulted in higher atmospheric concentration of the greenhouse gases, such as carbon dioxide and methane, thus increasing our planet's average temperature leading to irreversible climate changes, which has led to increasingly demanding and more strict legislations on pollutant emission and CO_2, as well as fuel economy targets for the automotive industry. As a result, a great deal of efforts and resources has been spent on the research and development of high efficiency and low emission engines for automotive applications in the attempt to reduce GHG emissions and levels of nitrogen oxides and soot emissions, which affect the air quality.

This research has developed strategies to investigate the combustion characteristics, engine performance, and exhaust emission of diesel-gasoline dual-fuel operation in a Ricardo Hydra single cylinder optical engine running at 1200 rpm, equipped with a high pressure common rail injection system for diesel fuel delivery, and a port fuel injection system, designed and manufactured by the author, for gasoline fuel delivery, in order to allow for dual-fuel operations. In-cylinder pressure measurement is used for calculating all engine parameters, heat release rate, and efficiency. In addition to the thermodynamic analysis of the combustion parameters, high speed imaging of spray and combustion

chemiluminescence was used for the optical analysis of the effect of the above-mentioned parameters on autoignition and combustion processes.

Effects of different substitution ratios and diesel injection strategies at low engine loads were studied when the total fuel energy was kept constant. The three main substitution ratios used in this study include 45%, 60%, and 75%, which also indicates the amount of fuel energy from port-injected gasoline, where the rest is provided by the direct injection of diesel.

Depending on the testing conditions, such as injection strategy and intake conditions, some dual-fuel operations were able to deliver high efficiency and improved emissions compared to that of a pure diesel engine operation, with the diesel-gasoline operation offering more consistency in improved thermal efficiency. The optical analysis of the combustion illustrates the main difference in the flame propagation, distribution and quality for each substitution percentage, as well as the condition under examination. It was observed that combustions with higher concentration of diesel fuel having more diffusion like combustion, especially with diesel injection timings closer to the top dead centre, where there is less time for the two fuel and air to properly mix before combustion occurs, resulting in higher temperature and levels of NOx due to the pockets of high diesel concentrations within the combustion chamber, whereas higher concentration of gasoline, especially at earlier diesel injection timings, resulted in more homogenous fuel mixture, and thus lower combustion temperatures. In other words, When the gasoline substitution ratio is lower, optimized SOI is advanced further, so that richer diesel mixture needs longer ignition delay to have proper combustion timing, and combustion is milder and PHRR is slightly lower due to less local diesel rich mixture area by means of earlier injection timing, and in terms of emissions, lower gasoline substitution ratio, decreases NOx with more homogeneous diesel mixture, and same can be said for THC. Performing the thermodynamics testing with an all metal piston alongside the optical testing allowed for the confirmation of these outcomes.

This study not only delivers an insight to the benefits of dual-fuel engine operation, it also represents the benefits of optical engines in providing better understanding of engine operation and ways of improving it.

2. Acknowledgment

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3. Introduction

In recent years, concern over exhaust gas CO_2 emissions from motor vehicles has been increasing, and to meet the current emission regulations for internal combustion (IC) engines, it has become necessary to develop a thorough understanding of the in-cylinder processes of fuel combustion to simultaneously achieve lower emissions and higher efficiency. To realize this, many studies have been taking place, using innovative in-cylinder optical analysis and laser imaging diagnostic techniques, which have provided a significantly enhanced understanding of the internal combustion. In the meantime, dual-fuel combustion techniques are another well regarded technology for improving IC engines emission output and efficiency. In a compression ignition (CI) dual-fuel engine, the primary fuel is mixed relatively homogeneously with the air during the intake stroke, then the air fuel mixture is ignited by injecting a small amount of diesel fuel, known as the pilot fuel, as the piston approaches the top of the compression stroke, where this diesel pilot fuel, quickly induces autoignition reactions and ignites due to the heat of compression, just as it would in a regular CI diesel engine. The combustion of the diesel pilot then ignites the air fuel mixture in the rest of the combustion chamber (1).

In the past few years, one of the concepts that has been examined by several researchers, is the diesel-gasoline RCCI combustion. Shuaiying Ma et al. (2) investigated the effects of diesel injection strategy on gasoline/diesel dual-fuel combustion, emissions, fuel economy and the operation range with high efficiency and low emissions on a modified single-cylinder diesel engine with port fuel injection of gasoline and direct injection of diesel fuel with rapid in-cylinder fuel blending, employing single and double injection strategies at 1500 rev/min and 50 mg/cycle total equivalent diesel fuelling rate. From their results they reported that this combustion mode had the capability of achieving high efficiency with near zero NOx and soot emissions by using an early single injection of diesel with high gasoline substitution ratio.

In a similar investigation on a heavy duty diesel engine, Jesús Benajes et al. (3) perform an experimental and numerical study to understand mixing and auto-ignition processes in RCCI combustion conditions, using gasoline and diesel as low and high reactivity fuels, respectively, by the means of three parametrical studies with diesel direct injection and gasoline port fuel injection, as well as a detailed analysis in terms of air/fuel mixing process by means of a 1-D spray model. Their results show that as the Diesel/Gasoline fuel ratio is reduced, the ignition delay increases and thus extending the mixing time and the first combustion stage gets lowered while the second one is enhanced, also that the advance of the diesel injection timing extends these effects over the combustion process. Finally, it is stated that compared to conventional neat diesel combustion, a slight reduction in terms of NOx and a very important reduction in terms of soot were achieved with the RCCI combustion.

Furthermore in a numerical investigation by J. Li et al. (4) on the effect of reactivity gradient in an RCCI engine fuelled with gasoline and diesel, the role of fuel reactivity gradient was examined numerically by comparing a dual-fuel mode combustion with other hypothetical cases under one specific load condition, where a chemical reaction mechanism was initially developed aiming at a modelling study on dual-fuel and blend fuel combustion in internal combustion engines fuelled by gasoline/diesel and gasoline/biodiesel. Ignition delays were validated for 100% diesel, 100% gasoline and 100% biodiesel under 102 conditions in total. They further conducted three dimensional CFD calculations under 3 conditions, including pure diesel combustion, and gasoline/diesel combustion, with both single and split injection strategies in the engine. To investigate the fuel reactivity gradient, they compared the gasoline/diesel combustion with a single injection, with other three hypothetical cases, one of which was dual-fuel mode without fuel reactivity gradient, and the other two were the blend fuel mode but with different injection timings. Their result illustrates that the fuel reactivity gradient could retard the ignition timing, reduce heat release rate, and ease peak pressure rise rate, as well as resulting in low levels of NOx and soot emissions from the diesel-gasoline dual-fuel combustion.

Study has shown (5) that in a 2.44 L heavy duty single cylinder compression ignition test engine , with a similar diesel-gasoline fuel delivery system, improving the fuel mixture could result in having increased operational domain for the premixed dual-fuel combustion. This resulted in substantial reduction in NOx and soot emissions, with net indicated efficiency of 50%, at 11bar IMEP. Further study by means of computer simulation revealed that areas with high concentration of diesel tend to ignite first.

Study has also shown (6) that different fuel proportions in dual-fuel partially premixed combustion (PPC) effects efficiency and emission and output, where with port injected gasoline to direct split injected diesel ratio of 89 to 11, 53% net indicated efficiency and considerable reduction in soot and NOx is attainable.

In a study (7) by means of computer simulation of diesel-gasoline dual-fuel combustion in a one cylinder compression ignition diesel engine, where the gasoline substitution ratio was increased from 0% to 85%, combustion pressure as well as temperature were reduced with the increase of gasoline substitution ratio. This was also true for the levels of soot and NOx emission output as a result.

Other, more recent studies (8) (9) have also demonstrated the effects of diesel to gasoline fuel ratio, in a dual-fuel combustion, on efficiency and emission output, and the ability to improve on these under certain operation conditions.

These studies, along with a few others have shown the potential of diesel-gasoline dual-fuel engine combustion in delivering both enhanced performance as well as much improved emission.

Another area of interest to researchers in the recent years, is the use of optical engines for the purpose of better understanding the in-cylinder processes such as fuel injection and distribution, combustion quality and characteristics, and the formation of exhaust fumes. Single cylinder optical access engines have been used as they allow for the application of non-intrusive qualitative and quantitative optical diagnostic techniques to gain a detailed insight of all the processes taking place within the combustion chamber from fuel injection, mixing, combustion and the formation of emissions. Nicolas Dronniou et al. (10) explored the fundamental combustion phenomena occurring when methane is ignited with a pilot injection of diesel fuel, on a single-cylinder optical research engine by high-speed imaging of combustion luminosity and single-shot OH*chemiluminescence Imaging. Results showed that combustion of the premixed charge of methane gas was dominated by spray entrainment and mixture stratification of diesel fuel and there were significant modifications in combustion behaviour indicating some evidence of flame propagation when the mixture was near stoichiometric.

Furthermore, Matthew Blessinger, Joshua Stein and Jaal Ghandhi (11), investigated reactivity controlled compression ignition combustion for three fuel combinations of isooctane-diesel, PRF90-diesel, and E85-diesel, at 1200 rpm, 160 kPa absolute intake pressure, and fixed total fuel energy using optimal operating condition for each fuel combination, chosen based on combustion performance from SOI timing and premixed energy fraction sweeps. They found that the heat release duration scaled with the difference in reactivity between the premixed and directly injected fuel, and a small difference gave rise to short heat release duration, similar to that of HCCI combustion. On the other hand, with increase in the difference, the heat release period increases. The high-speed optical data showed that the combustion happened in a staged manner from the high-reactivity zones to low-reactivity zones and the range of ignition timing was found to scale with the difference in the reactivity of the two fuels.

As reported by a recent study (12), there can be significant difference in the optical engine and a metal engine in fuel mixing and combustion, and emissions formation when results were compared between these two types of engines due to different materials on the combustion chamber wall heat transfer characteristics. Thus, it is preferable that both thermodynamic engine experiments and in-cylinder studies are performed in the same engine with minimum change, as it is the case of the present study. The same single cylinder engine was used for the thermodynamic engine testing with a metal piston to identify the most appropriate diesel-gasoline dual-fuel ratio and injection strategies and then adopted for the in-cylinder optical studies by installing an optical window in the piston crown. The combustion, engine performance and emission characteristics of the diesel-gasoline dual-fuel engine operations were studied by means of in-cylinder pressure, fuel consumption and emission measurements with different diesel and gasoline ratios and diesel injection timings. Based on the analysis of such data, additional studies were then performed on the in-cylinder mixture formation, ignition and combustion through high speed imaging of fuel injection and combustion images.

Changzhao Jiang, Xiao Ma and Hongming Xu (13) studied the combustion of DMF and ethanol under dual-injection strategy, where high speed imaging and thermal investigation were carried out to study DMF and gasoline dual-injection on a single cylinder, direct injection spark ignition optical engine, by combining direct injection and port fuel injection simultaneously, resulting in two different fuels to be blended in the cylinder with any ratio. For all their tests, gasoline was injected through PFI and different amounts of DMF or ethanol were injected through DI. For each of the predetermined IMEP, 3bar and 5bar, bio-fuel DI fraction was increased from 0% to 100%. The flame morphology was analysed, and normalized flame area data was then used to study the effect of the dual-injection strategy and fuels, showing that DMF-gasoline dual-injection combustion has higher flame propagation speed and shorter combustion duration than baseline 100% gasoline PFI. The flame luminance of DMF-gasoline is much higher than ethanol-gasoline and pure gasoline, and that flame propagation speed of ethanol increases with the increase of IMEP, while the engine load conditions have less influence on DMF-gasoline dual-injection flame propagation speed.

Within the past decade, new camera and laser technologies have provided the ability for the improvement of high-speed imaging diagnostics for measurements at frame rates corresponding with the time scales of turbulent mixing of the fuel, fuel injection, combustion, and emission formation in internal combustion engines (14).

The application of high-speed imaging in IC engines, permits the visualization of sequential and spatial resolution of fuel spray, as well as its combustion. The experimental setup simply requires a source of light, i.e. laser beam, and a device for image capture, i.e. high-speed camera. Essentially, for high speed visualization, the source of light is selected with respect to its brightness, spectral distribution and its repetition rate, as a result of the necessity of a significant amount of light over a short period of time. Likewise, because of their high repetition rate, high-speed cine film cameras are normally used. Herfatmanesh and Lu (15) visualized the fuel injection formation and combustion process, by using a copper vapour laser and a NAC Memrecam FX6000 high speed camera, in a research optical diesel engine. This technique was also widely employed in many studies (9)(16)(17)(18)(19).

Where there are many studies that have been completed with the focus on benefits of dual-fuel operations, as well as some studies in optical analysis of internal combustion by means of recording the combustion chamber events, there is little research done in which both aspects, dual-fuel operation characteristics, as well as optical analysis of these characteristics have been examined simultaneously, and on the same engine, which is what this study has provided.

4. Experimental apparatus and methodology

4.1. Optical engine

All the experimental testing in this study was carried out on a single cylinder Ricardo Hydra Optical engine, with an extended cylinder block, a standard production 4-cylinder diesel engine cylinder head and a common rail fuel injection system, designed to be representative of a typical modern high-speed direct injection (HSDI) diesel engine. The engine specifications are presented in Table 1. The engine is mounted on a Cussons Technology's single cylinder engine test bed consisting of a seismic mass engine mounting, a 30 kW DC dynamometer and engine coolant and oil circuits.

Ricardo Hydra Single Cylinder Optical Engine	
Bore	86 mm
Stroke	86 mm
Swept Volume	499 cm^3
Compression Ratio	16:1
Piston Bowl Diameter/Depth	43.4/11.6 mm, re-entrant bowl with flat bottom
Swirl Ratio	1.4
Engine Speed for Testing	1200 rpm

Table 1 Ricardo Hydra Engine Specifications

Optical access through the extended piston design allows the combustion chamber to be visualised through the axis of the cylinder via a fused silica window mounted in the crown of the piston. A 45-degree mirror was mounted between upper and lower sections for viewing the combustion chamber, Figure 1. In addition, three rectangular cut-outs are machined for fitting with the fused silica windows as the side optical access.

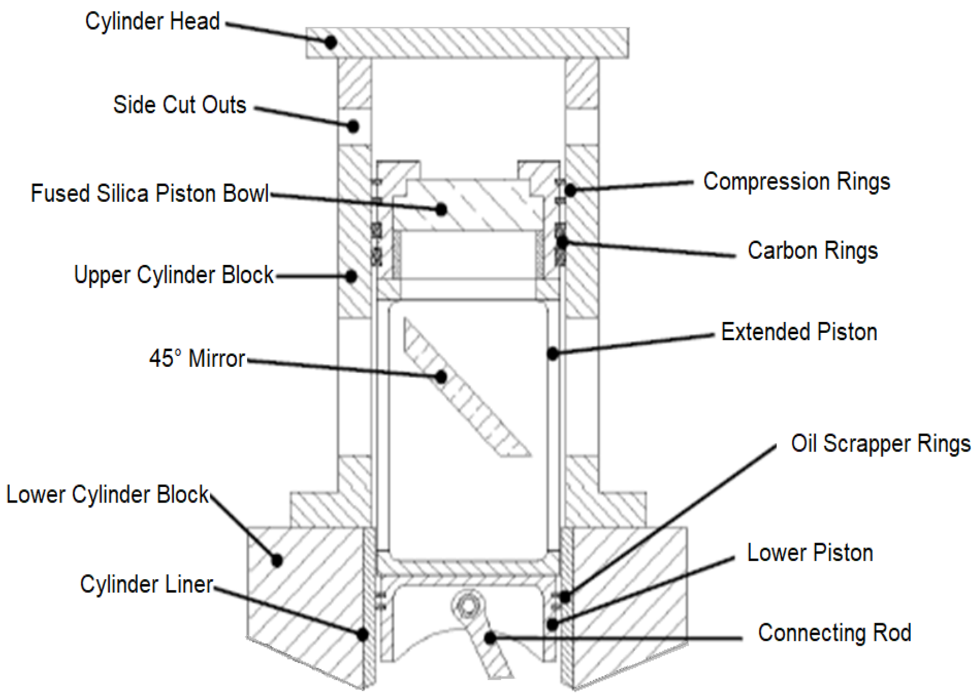

Figure 1 Sectional Schematic View of the Optical Layout

A Delphi multi-hole Valve Covered Orifice (VCO) injector was used and it can provide maximum injection pressure of 1350 bar independent of the engine speed. A pneumatic actuated diaphragm pump was employed to pressurise the diesel fuel to the common rail.

An Eaton M45 supercharger driven by an AC motor rotating at 2600 rpm was employed to provide the compressed air into the intake system. The boost pressure could be manually controlled by adjusting a bypass valve up to a maximum of 0.5 bar gauge pressure. The nitrogen gas delivered from a pressurised bottle was used as simulated exhaust gas recycling (EGR).

4.2. Thermodynamic measurement

A Kistler 6125A piezoelectric pressure transducer, mounted in place of a glow plug in the cylinder head, coupled with a Kistler 5011 charge amplifier, allows for the measurement of the in-cylinder pressure. A National Instruments (NI) data acquisition card and an in-house LabVIEW Program are used for collecting and recording the data. The in-cylinder pressure data was then used to determine the heat release rate, mean effective pressure and ignition delay.

4.3. Optical measurement

High speed video imaging was used to record video images from the diesel fuel spray during the injection phase and the following combustion. The high-speed video camera used for this study was a NAC Memrecam FX 6000, equipped with a high-speed colour CMOS sensor. For the imaging of diesel fuel sprays, a high repetition CU15 copper vapour laser was used as the strobe light and synchronised with the high-speed camera at 10,000 frames per second (fps), which means, at the engine speed of 1200 rpm, one frame equates to 0.72-degree crank angle. The output from the copper vapour laser was coupled to an optical fibre to illuminate the combustion chamber through a 45-degree mirror positioned underneath of glass piston window to visualise the direct diesel fuel spray during the diesel only operations (Figure 2). For combustion imaging, the copper vapour laser was turned off and the high-speed camera was coupled and synchronised with a high-speed DRS Technologies ILS image intensifier with gains varied from 60% to 80% to record weak luminescence of premixed combustion at 6,000 fps, resulting in a resolution of 1.2-degree crank angle between images. The image resolution is dependent on the frame rate, at 10,000 fps, the image resolution was 512×248 pixels, and at 6,000 fps, the image resolution is 512×384 pixels. A Nikon 60 mm f.2.8D was used for the spray imaging and a Nikon 150 mm f.4.5 telescopic lens for operations with the image intensifier. A TDC reference on the captured videos was produced by a LED triggered by reference signal from a shaft encoder.

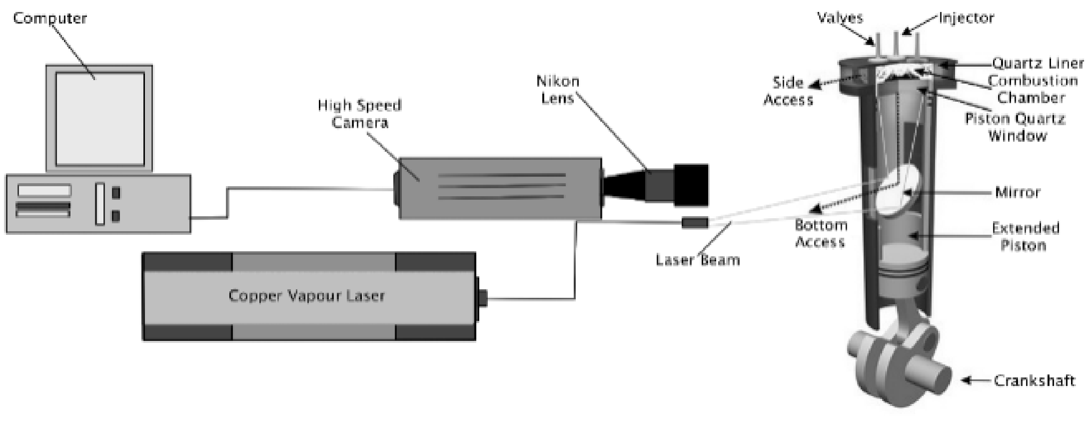

Figure 2 High Speed Camera and Copper Vapor Laser Arrangement (20)

4.4. Exhaust Emission Measurement

The engine exhaust gas emissions of CO, CO_2, O_2, uHC and NOx were measured using a Horiba MEXA-7170DEGR Exhaust Gas Analyser, which comprises of four analysing modules, all controlled by a PC as the main control unit where each of the modules operate with a different measurement principle, and the concentrations of emitted soot was measured by an AVL 415 Smoke Meter, which is equipped with a diaphragm type pump that sucks the exhaust gas into the sampling line and then passed through a paper filter followed by a flow meter; the soot concentration is then measured by the use of a reflectometer, which measures and compares the reflection from the clean and the smoky filter papers.

Measured Variable	Device	Manufacturer	Dynamic Range	Linearity/ Accuracy	Repeatability
CO (low content)	AIA-721A		0-2.5k ppm		
CO (mid-high content)	AIA-722		0-12 vol%		
CO2	AIA-722	Horiba (MEXA 7170 DEGR)	0-20 vol%	≤ ± 1.0% FS or ± 2.0% of readings	Within ± 0.5% of full scale (FS)
NOx	CLA-720MA		0-500 ppm or 0-10k ppm		
O2	MPA-720		0-25 vol%		
Unburnt HC	FIA-725A		0-500 ppm or 0-50k ppm		
Filter Smoke Number	415SE	AVL	0-10 FSN	-	Within ± 0.005 FSN + 3% of reading

Table 2 Emission measurement device specification and accuracy

Table 2 shows the specification and accuracy of the emission measurement device used.

5. Test conditions

The constant and variable engine operating parameters are listed in Table 3 and Table 4 respectively. Diesel was directly injected at 500bar by the common rail fuel injector into the combustion chamber, and gasoline was injected at every TDC, i.e. twice every engine cycle at 0° and 180° CA, in the intake port at 3.0 bar. Engine was operated in low load at 1200 rpm and supplied with a constant input energy of 5.2 kJ/sec, meaning total equivalent ratio is about 0.4 at 150° C of intake temperature and 1 bar of intake pressure, resulting in loads varying from 0.1 bar IMEP to 3.1 bar IMEP. IMEP or other combustion characteristics are calculated from 20 cycles of in-cylinder pressure with in-house code in VBA (Visual Basic for Applications). In this research, apparent heat release rate calculated with a constant heat specific ratio 1.3 is used for analysis, not gross heat release due to difficulty to estimate heat loss accurately. In addition, the experiments were repeated without and with 25% and 40% of EGR. Both single and split injections of diesel were used with a constant dwell angle of 35° CA during the split injections. However, for in-cylinder optical measurements, due to the greater heat loss and leakage with the glass piston crown, both the intake pressure and the intake temperature are raised to 1.1 bar and 150^{o} C respectively, so that the pressure and temperature at TDC remained close to those when engine operating with the metal piston crown. Otherwise, other conditions are kept the same as thermal engine test.

Fixed Testing Parameters	
Parameter	Value
Engine Speed (rpm)	1200
Diesel Rail Pressure (bar)	500
Total Fuel Energy (kJ/sec) Constant	5.2
Target IMEP (bar)	3.0
Intake Pressure (bar)	≈ 1.0 (1.1 for Optical setup)

Table 3 Fixed Testing Parameters

Variable Testing Parameters	
Parameter	Value
Intake Temperature (^{O}C)	100, 150
Substitution Ratio (%)	0 (Diesel only), 45, 60, 75
EGR Rate (%)	0, 25, 40
Diesel Injection Strategy	Single, Split (50/50, 35^{O}CA Dwell)

Table 4 Variable Testing Parameters

This study was conducted as part of a wider study on dual-fuel operations in variety of engines, heavy and light duty, and fuel mixtures, therefore in order to enable comparisons between results of all studies, and also due to the sensitivity of the optical engine and the imaging equipment used in this study limiting the number of tests and increasing time required, only a select number of gasoline substitution ratios, 45%, 60% and 75%, were used.

The intake air temperature of 100° C is utilized in order to match the intake temperature used for the optical investigation of diesel only operations from a previous study (9)(21)(22), and it is then increased to 150° C to see the effect of increased intake air temperature, and during optical operations, to partly compensate for the added heat loss due to the optical access points.

The focus of the in-cylinder study in the optical diesel engine is to understand the difference in mixture formation, ignition and combustion with different diesel injection strategies and gasoline substitution ratios. Figure 3 below shows the position of intake and exhaust valve and diesel injection spray angle in image. Dotted lines are injection spray angle.

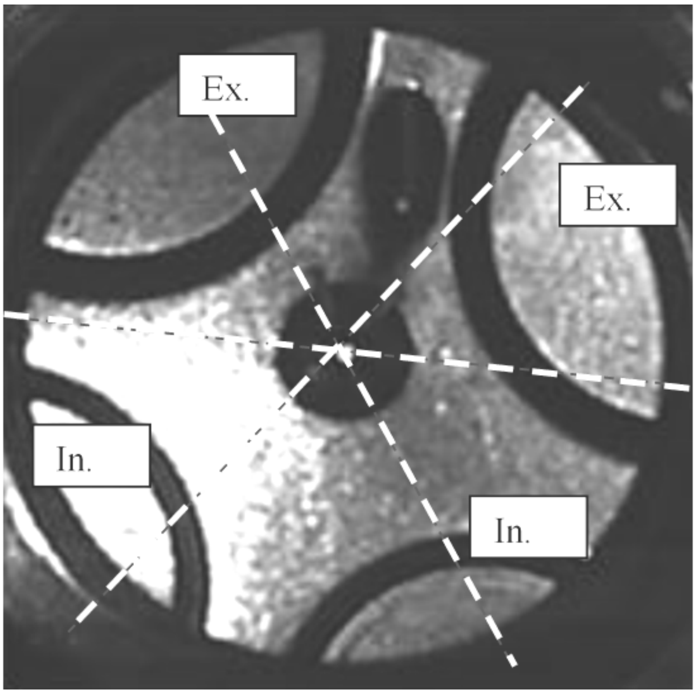

Figure 3 Image of combustion chamber through the glass piston without combustion

6. Results and discussion

6.1. Thermodynamic analysis

6.1.1. Effect of Intake Air Temperature and EGR with 60% Substitution Ratio

Figure 4 shows, from left-top to right-bottom, Combustion Efficiency, Soot, CO, THC, NOx, Indicated Thermal Efficiency, Pmax, CA10-90 (Combustion Duration), CA50 (combustion centre), Ignition Delay (from Injection Signal to CA5), PHRR (Peak Heat Release Rate), PPRR (Peak Pressure Rise Rate), for diesel fuel injection timing.

The data at 150° C of intake temperature without EGR cannot be collected between 10° CA to 45° CA BTDC SOI due to the presence of violent knocking combustion. Also, any injection before 65° CA BTDC at these intake conditions leads to misfire. Misfire also occurs for any injection timing after 5° CA BTDC and before 62° CA BTDC when the intake temperature is at 100°C without EGR. When increasing the EGR to 25% with the 150°C intake temperature, knocking combustion appears between the SOI timings of 35° CA BTDC and 15° CA BTDC, with any SOI timing after 5° CA BTDC or before 56° CA BTDC leading to misfire. By increasing the EGR to 40% for the same intake temperature of 150°C, there is less chance of knocking but misfire happens at a SOI timings before 50° CA BTDC and any SOI timing beyond 5° CA BTDC. Higher intake temperature increases NOx at the same SOI and EGR decreases NOx. Comparing the lowest NOx of each intake condition, NOx at higher intake temperature is lower and without EGR, NOx is lower, because more premixed and leaner mixture of diesel fuel, due to earlier SOI and long ignition delay, decreases NOx levels.

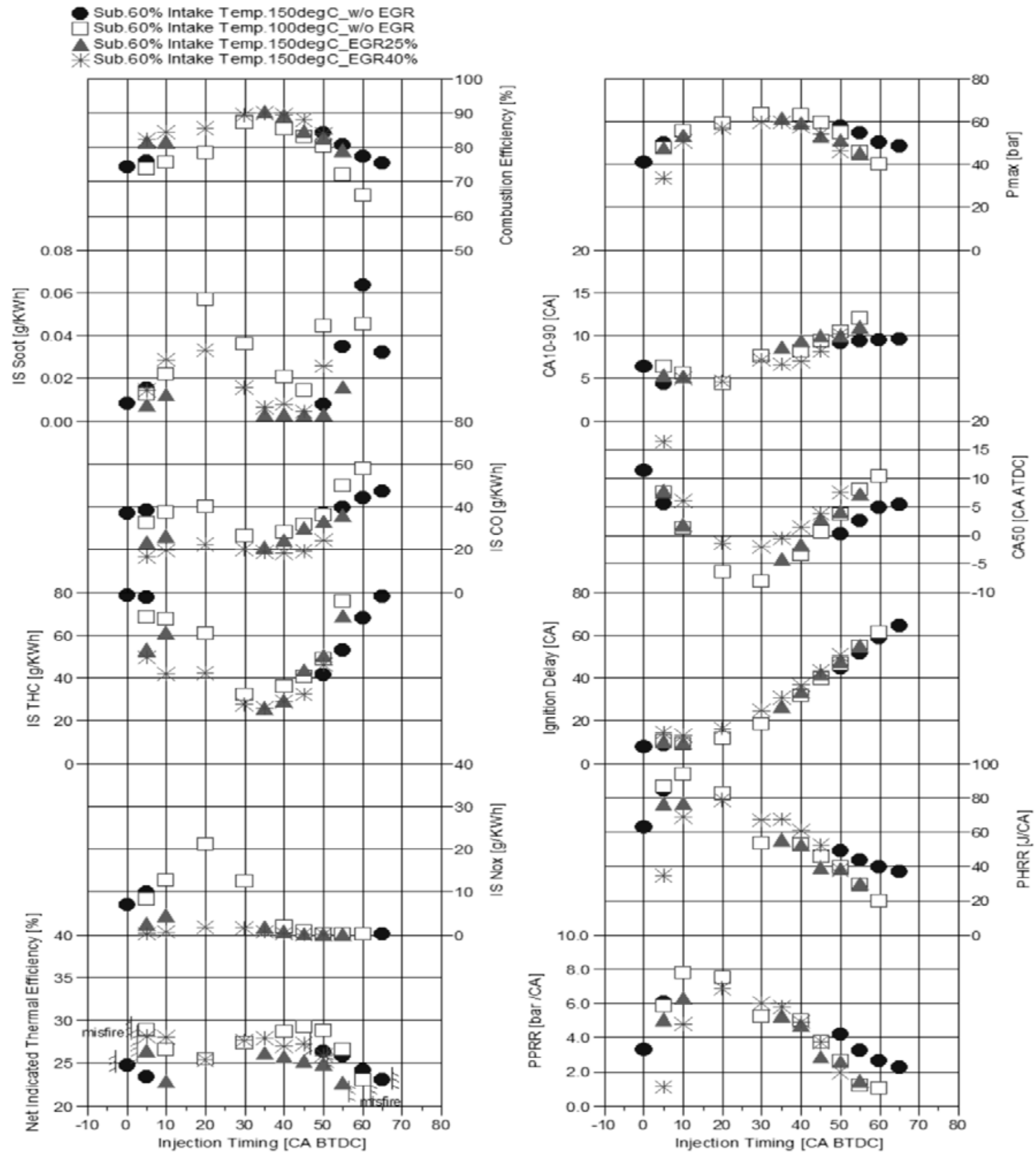

Figure 4 Comparison of engine performance, emission and combustion characteristics at different intake conditions with 60% gasoline substitution ratio and single diesel injection

Soot emissions are generally lower at earlier SOI with higher intake temperatures and higher levels of EGR. That said, there is an increase in soot levels at SOI timings earlier than 50^o CA BTDC, specially for lower intake temperatures and EGR levels. It can be observed that there is a peak in soot levels at diesel SOI timing of 20^o CA BTDC without any EGR when the intake temperature is 100^{o}C. This could be caused by the diesel fuel not being injected into the combustion bowl properly at this diesel injection timing.

Combustion efficiency, at higher intake temperature, is higher because of the higher combustion temperature. Where the data for all conditions is available in figure 4, it can be observed that EGR does not have a significant effect on combustion efficiency at early injection timing, i.e. 50° BTDC, and slightly improves combustion efficiency at late injection timing, i.e. 5° BTDC, even though oxygen concentration is decreased. Ignition delay is longer at lower intake temperature and with EGR because of lower in-cylinder temperature by their lower heat specific ratio and lower oxygen concentration. Both decreasing intake temperature and using EGR, led to later CA50, longer CA10-90, lower P_{max}, and lower PRRR, because of slower chemical reaction in lower temperature or lower oxygen ratio.

Figure 5 shows the in-cylinder pressure and HRRR when SOI is sweeping at 100°C intake temperature and 60% of gasoline substitution ratio without EGR. SOI from 5^o CA to 30^o CA BTDC, advances combustion. As SOI of diesel is advanced, combustion timing is advanced except at 40^o, 50^o and 60^o CA BTDC of SOI, with the SOI of 0^o CA BTDC having its combustion much later starting at around 15^o CA ATDC. Amongst these SOI conditions, 10^o CA BTDC of SOI of diesel, has the highest PHRR and shortest combustion duration, which is caused by rapid combustion around TDC and moderate homogeneity of diesel mixture created. The beginning of HRRR at 50^o CA BTDC of SOI is slower than others and even more so in the case of 60^o CA BTDC, due to more diluted mixture and lower charge temperature. On the other hand, it is also observed that at 5^o CA BTDC of SOI, heat release of late combustion, is more like diesel combustion, and as expected the heat release for the SOI of 0^o CA BTDC is as low as the heat release rate for SOI of 50^o CA BTDC and starts faster but very late in comparison. Further analysis of the combustion process will be performed by the high-speed spray and combustion imaging later. It is also observed from figure 5 that as the SOI of diesel gets closer to TDC, i.e. after 30^o CA BTDC, the HRRR is retarded, which could be due to the fact that diesel and gasoline have less time to create a homogenies fuel mixture, and a more diffusion like combustion of diesel rich areas create this trend for the later diesel SOI.

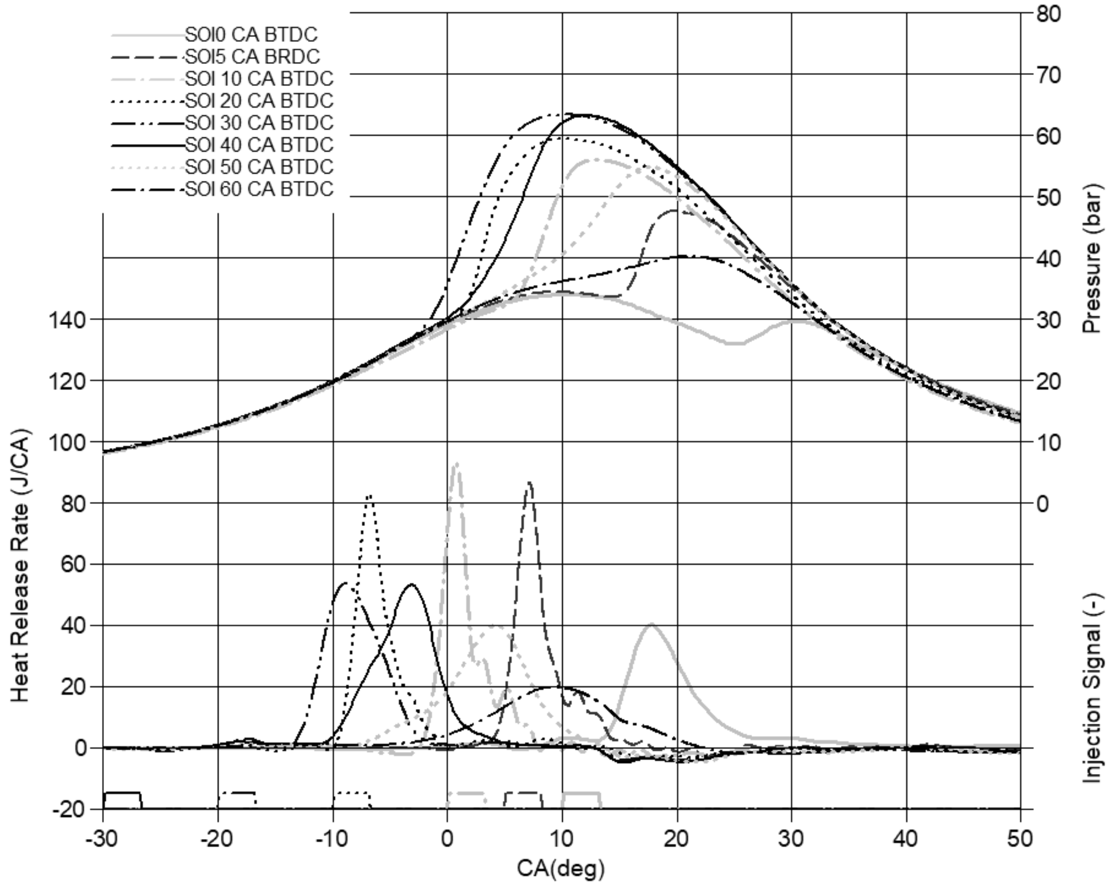

Figure 5 In-cylinder pressure and HRRR at different single SOI at intake temperature 100°C and gasoline substitution ratio of 60%

Figure 6 shows in-cylinder pressure and HRRR with the same diesel SOI of 5° CA BTDC at different intake conditions. As the intake temperature increases, PHRR becomes lower, combustion duration is shorter and ignition delay is shorter. Increasing EGR to 25% at the same SOI and intake temperature increases the ignition delay as well as the heat release rate with similar combustion duration. At intake temperature of 100°C without any EGR, the start and duration of the combustion, as well as the heat release rate and peak pressure, are almost the same as that of the intake temperature of 150^{o}C with 25% of EGR with the 100^{o}C intake temperature without EGR being slightly higher. But as the EGR increases to 40% with 150^{o}C intake temperature, it also shows the similar effect to cooler intake as to heat release characteristics, with 40% EGR at 150^{o}C intake temperature having the least heat release rate at a much later time in comparison. At the conditions with EGR and the one with 100°C intake temperature, heat release occurs slightly just before main heat release. This may be LTHR (low temperature heat release) of diesel fuel due to the longer ignition delay. Similar observations were made when comparing different intake conditions at the same early SOI timing.

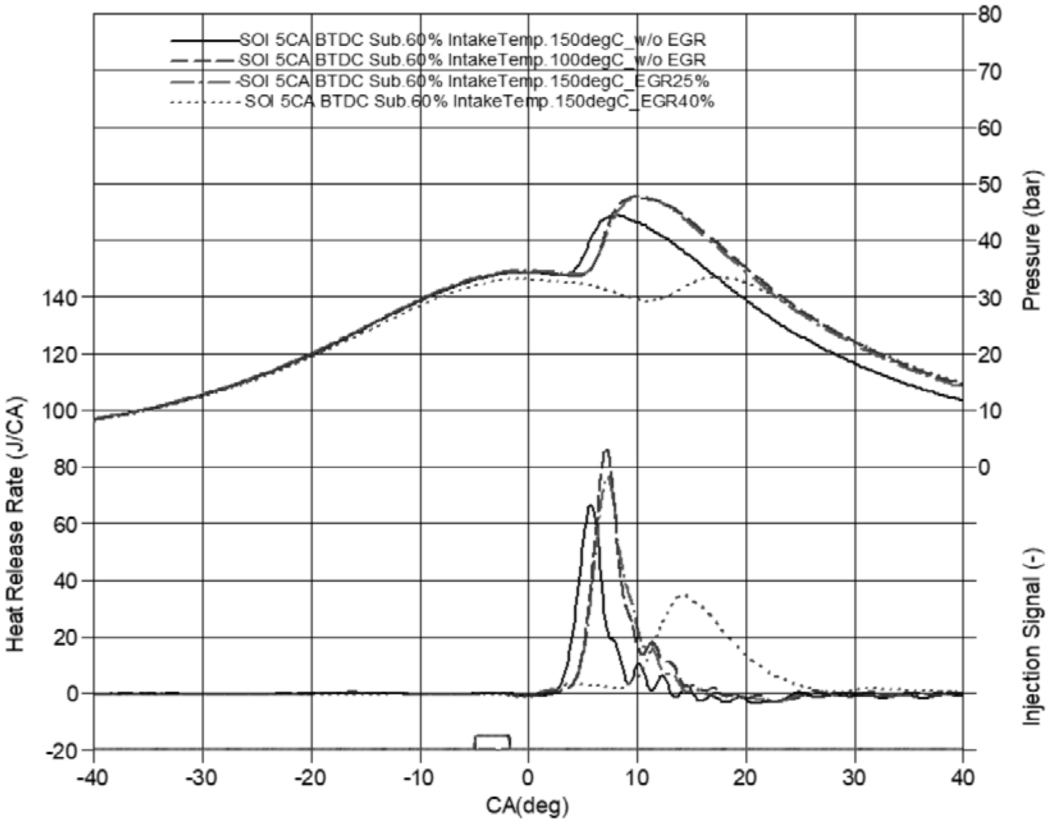

Figure 6 In-cylinder pressure and HRRR at different intake conditions with SOI at 5 CA BTDC and gasoline substitution ratio of 60%

Figure 8 demonstrates engine performances, emission and combustion characteristics with the optimized SOI as to thermal efficiency and NOx at different intake conditions, while displaying the effect of the injection timing for the same exact condition with a later injection as well. Figure 7 shows in-cylinder pressure and HRR for the same conditions as Figure 8. The intake conditions are: 50^{o} CA BTDC of SOI at 150°C of intake temperature without EGR, 45^{o} CA BTDC of SOI at 100°C of intake temperature without

EGR, 40° CA BTDC of SOI at 150°C of intake temperature with 25% EGR, and 35° CA BTDC of SOI at 150°C of intake temperature with 40% EGR.

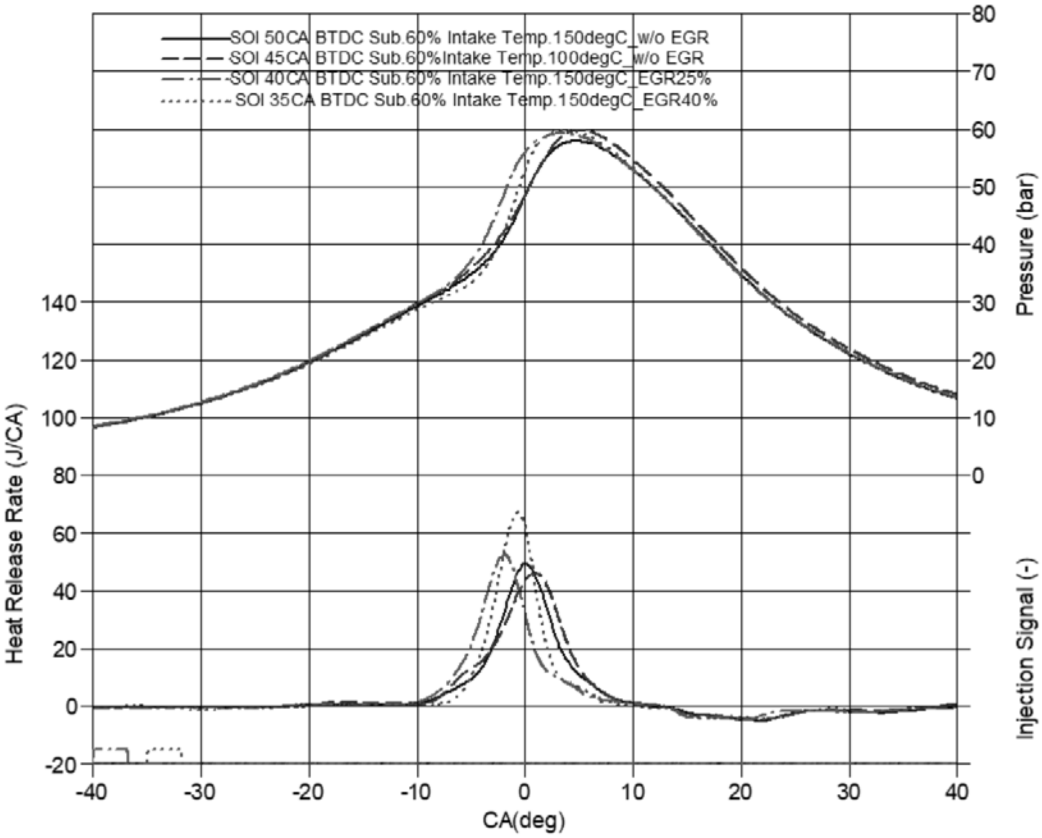

Figure 7 in-cylinder pressure and HRRR with the optimized SOI at different intake conditions with gasoline substitution ratio of 60%

The results illustrate that increased intake temperature leads to a higher thermal efficiency because of higher combustion efficiency, due to lower THC. EGR increases the thermal efficiency because of lower heat loss from lower combustion temperature. This is more apparent in later injection timings. Combustion efficiency or THC is not decreased by EGR due to higher overall equivalent ratio, caused by less fresh air. It can be observed that NOx is generally low at early SOI timings, but it can be seen from the later diesel SOI timings that NOx is lower at 150°C of intake temperature without EGR because the fuel mixture is more homogeneous with longer ignition delay, by hotter intake condition without the EGR gas.

Increasing intake temperature decreases soot without EGR, and combustion duration (CA10-90) is longer with higher intake temperature, or with EGR conditions. The starts of combustion are similar to each other in Figure 7, with diesel SOI of 35^{o} CA BTDC with 150°C of intake temperature and 40% EGR having the highest heat release rate, at just before TDC, probably due to less time being available for heat loss between start of injection and the start of combustion at this condition. The slope of HRR in the condition with SOI of 45^{o} CA BTDC and 100°C intake temperature and no EGR, changes around TDC. It is considered that gasoline combustion is accelerated by diesel combustion at this point.

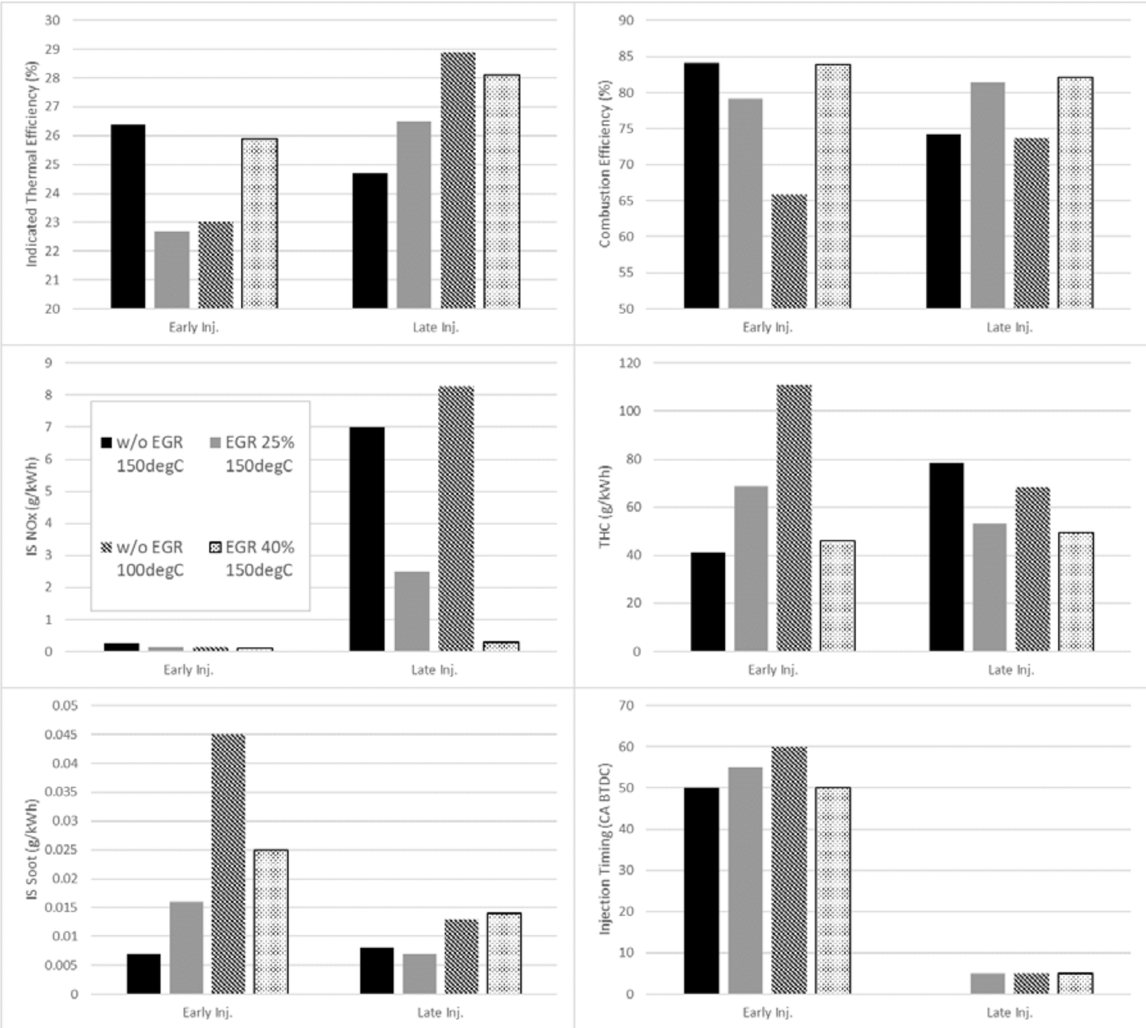

Figure 8 Engine performances and Emission with the optimized SOI at different intake conditions

The results of this section of the study shows that, increasing in-cylinder temperature and using EGR, which in this study due to the use of heated air intake and injection of N_2 and CO_2 into air intake from an external source have the similar effect as using internal EGR, are effective to simultaneously reduce emissions and increase thermal efficiency at the part-load conditions examined.

6.1.2. Comparison of Single and Split Diesel Injection with 60% Substitution Ratio and Intake Air Temperature of 150°C

Figure 9 from top-left to bottom-right, illustrates Combustion Efficiency, Soot, CO, THC, and NOx levels, Indicated Thermal Efficiency, P_{max}, CA10-90 (Combustion Duration), CA50 (combustion centre), Ignition Delay (from Injection Signal to CA5), PHRR (Peak Heat Release Rate), PPRR (Peak Pressure Rise Rate), start of diesel fuel injection signal timing, for split injection operations.

Combustion characteristics are quite similar between the single and split injection strategies. The combustion with the split injection strategy is primarily controlled by the second injection. In particular, the split injections enable the engine to operate without knocking combustion when the second SOI is varied from 10° CA to 45° CA BTDC, due to more stratified mixture. Misfire limit with split injections is slightly narrower than the single injection, due to the smaller amount of second injection, creates leaner diesel mixture.

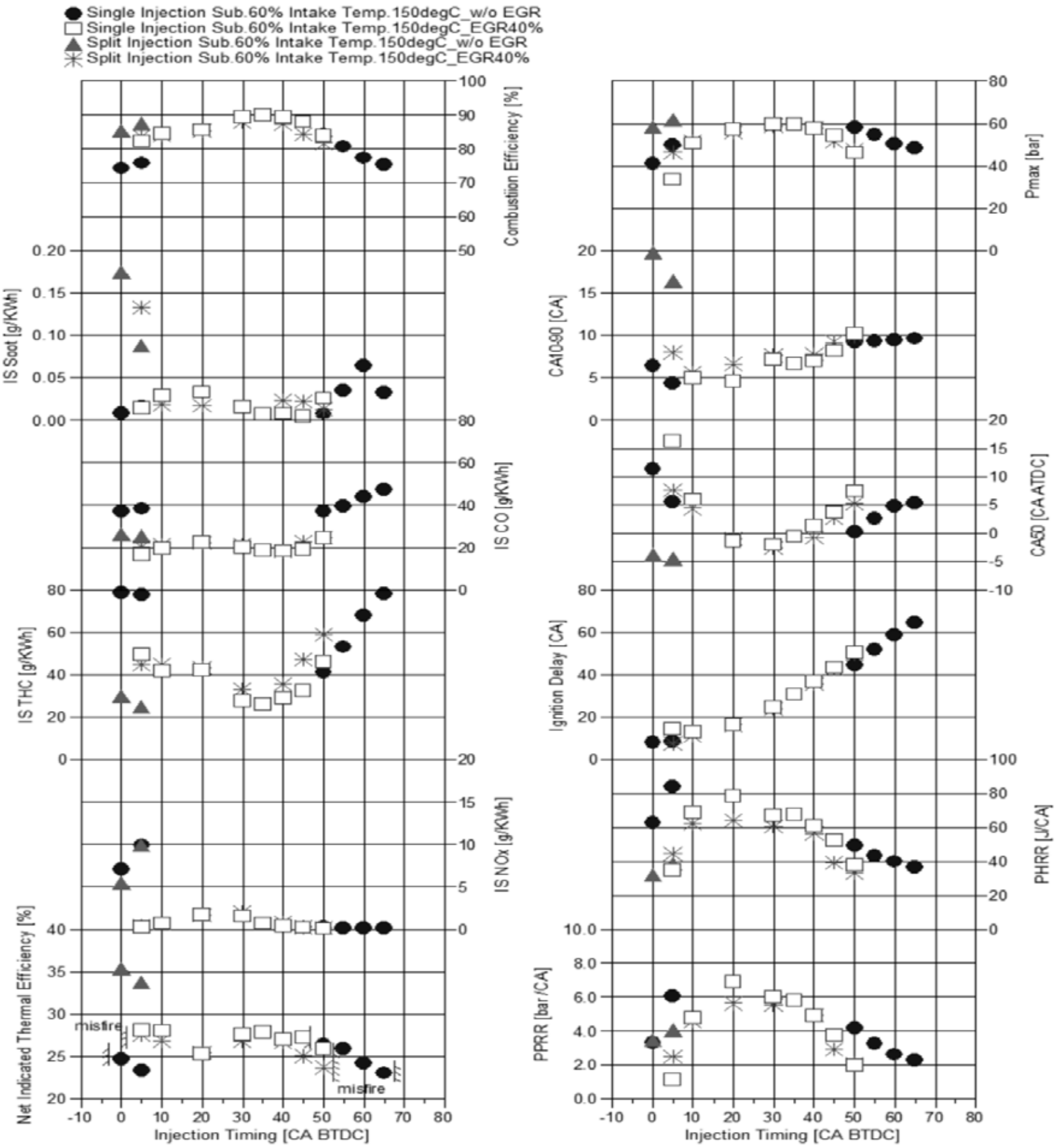

Figure 9 comparison of engine performance, emission and combustion characteristics between single and split diesel injection, as well as EGR effect, when substitution ratio is 60%, intake temperature is 150°C

It is also evident from this figure, that although there is practically no NOx emission for any of the conditions with early injection timing of the diesel fuel, there are increasing levels of THC, CO and soot as the diesel injection timing is advanced, with the exception of soot for the single injection condition with no EGR, which decreases again when advanced to 65^o BTDC. This increase in emissions with the earlier injection timing is understood to be as a result of the fuel trapped in the crevices, cylinder liner and over lean zones of the combustion chamber, where gas with low temperature hardly combusts or is quenched by lower temperature zone, due to the long ignition delay.

Figure 10 presents and compares the results obtained with the 5^o CA BTDC of SOI for single injection of diesel and 5^o CA and 40^o CA BTDC of SOI for split injection of diesel. This figure shows that the split injection strategy has an earlier, but lower heat release rate with a slower combustion than the single injection when there is no EGR. With EGR, the heat release rate of the split injections is still earlier than the single injection heat release rate at a slightly higher rate. This is because the retarded 1st injection can assist the combustion start by reducing the ignition delay of the second injection, which results in combustion starting during or just soon after second injection timing.

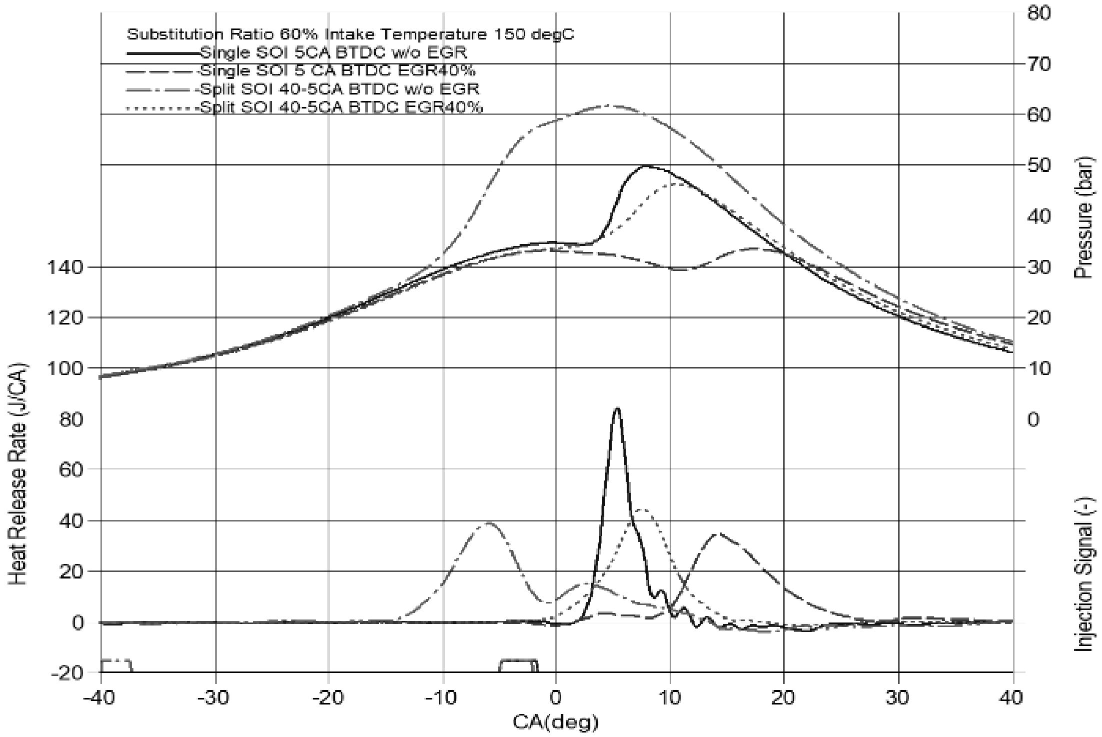

Figure 10 in-cylinder pressure and HRRR with single injection and split injection when SOI of the single and the split are 5 and 40-5 CA BTDC respectively, intake temperature is 150°C and gasoline substitution ratio is 60%, at 0 and 40% EGR

Figure 11 illustrates the in-cylinder pressure and HRRR with the SOI of single diesel injection at 50^{o} CA BTDC and 5^{o} CA BTDC, along with split diesel injection at 5^{o} CA and 40^{o} CA BTDC. It is observed that 150°C of intake temperature and no EGR, late single injection of diesel has the highest heat release rate and shorter ignition delay whilst a single early diesel SOI timing having the longer ignition delay with combustion starting at around 10^o CA BTDC. The split diesel injections have the later, lower PHRR and slower combustion because of leaner and more premixed fuel mixture given by the earlier first injection and less amount of second injection.

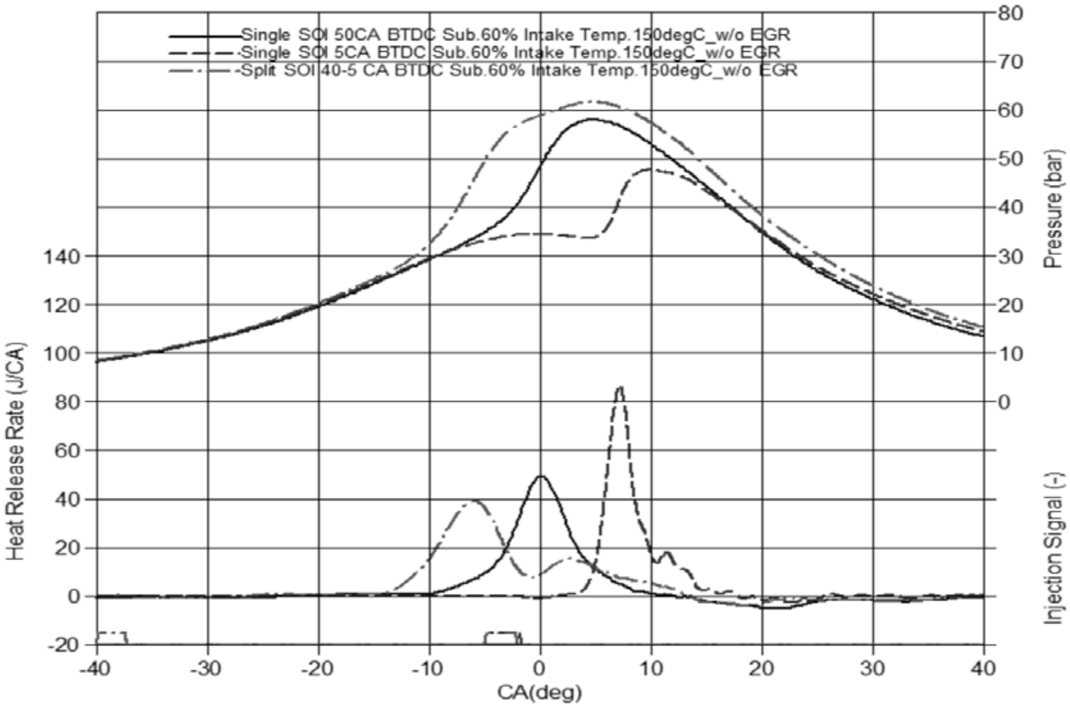

Figure 11 in-cylinder pressure and HRRR with single injection and split injection when SOI of single and split are 5 and 40-5 CA BTDC respectively, intake temperature is 150°C and gasoline substitution ratio is 60%

Figure 12 shows the in-cylinder pressure and HRRR at different split SOI when intake temperature is 150°C and gasoline substitution ratio is 60% with 40% EGR, where with 35^o CA dwell between the first and second diesel injection, operation with initial diesel injection at 65° CA BTDC has the shortest ignition delay evident by the early heat release, and also has the highest in-cylinder pressure, and the late injection operation with initial diesel injection at 40° CA BTDC has a longer combustion duration with a much lower in-cylinder pressure rise. Therefore, result show that split injection causes more diffusion-like combustion, which might be helpful to enhance thermal efficiency but worsen soot.

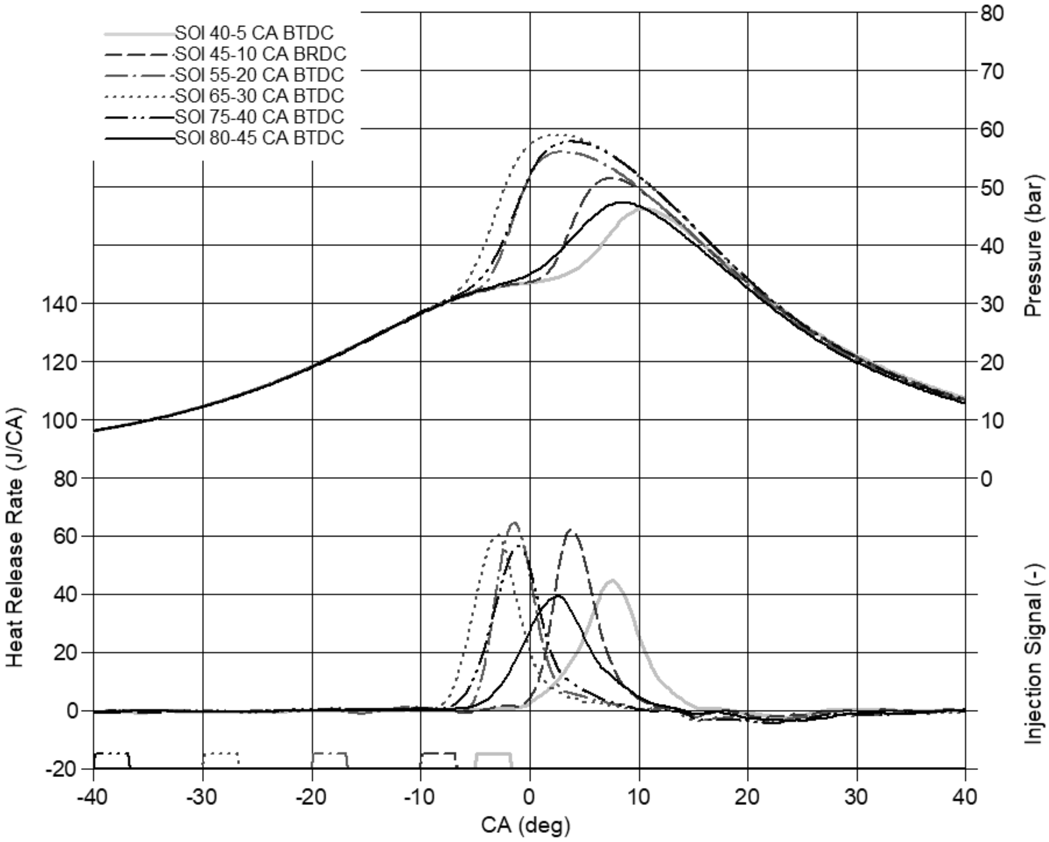

Figure 12 in-cylinder pressure and HRRR at different split SOI when intake temperature is 150°C and gasoline substitution ratio is 60% with 40% EGR

6.1.3. Effect of Substitution Ratio with Single Diesel Injection, Intake Air Temperature of 150°C, and no EGR

The effect of gasoline substitution ratio on combustion and emissions are presented in Figure 13. Several experiments were performed with different gasoline substitution ratios and the results demonstrated that are shown for gasoline substitution ratio of 60% and 75% because of their higher combustion efficiency.

Knocking combustion occurs between 45° CA BTDC and 10° CA BTDC with 60% substitution ratio and between 35° CA BTDC and 10° CA BTDC of diesel SOI with 75% substitution ratio having knock. Misfire occurs at early SOI, with 60% substitution ratio having misfire as early as 70° CA BTDC of diesel SOI, and the 75% substitution ratio having misfire as early as 55° CA BTDC of diesel SOI, as well as when the diesel SOI is delayed after TDC.

When SOI for diesel is near TDC, NOx emissions are similar in all conditions, including diesel only condition. The lowest NOx is obtained with earlier SOI of diesel with lower gasoline substitution ratio, because of the longer ignition delay for these conditions.

Diesel baseline operation produces the lowest THC and CO and hence the highest combustion efficiency. Diesel condition also has higher PRRR and PHRR than dual-fuel condition due to higher reactivity of diesel fuel. Dual-fuel operation produces higher THC and CO due to gasoline fuel trapped in crevices and the presence of over-lean zones.

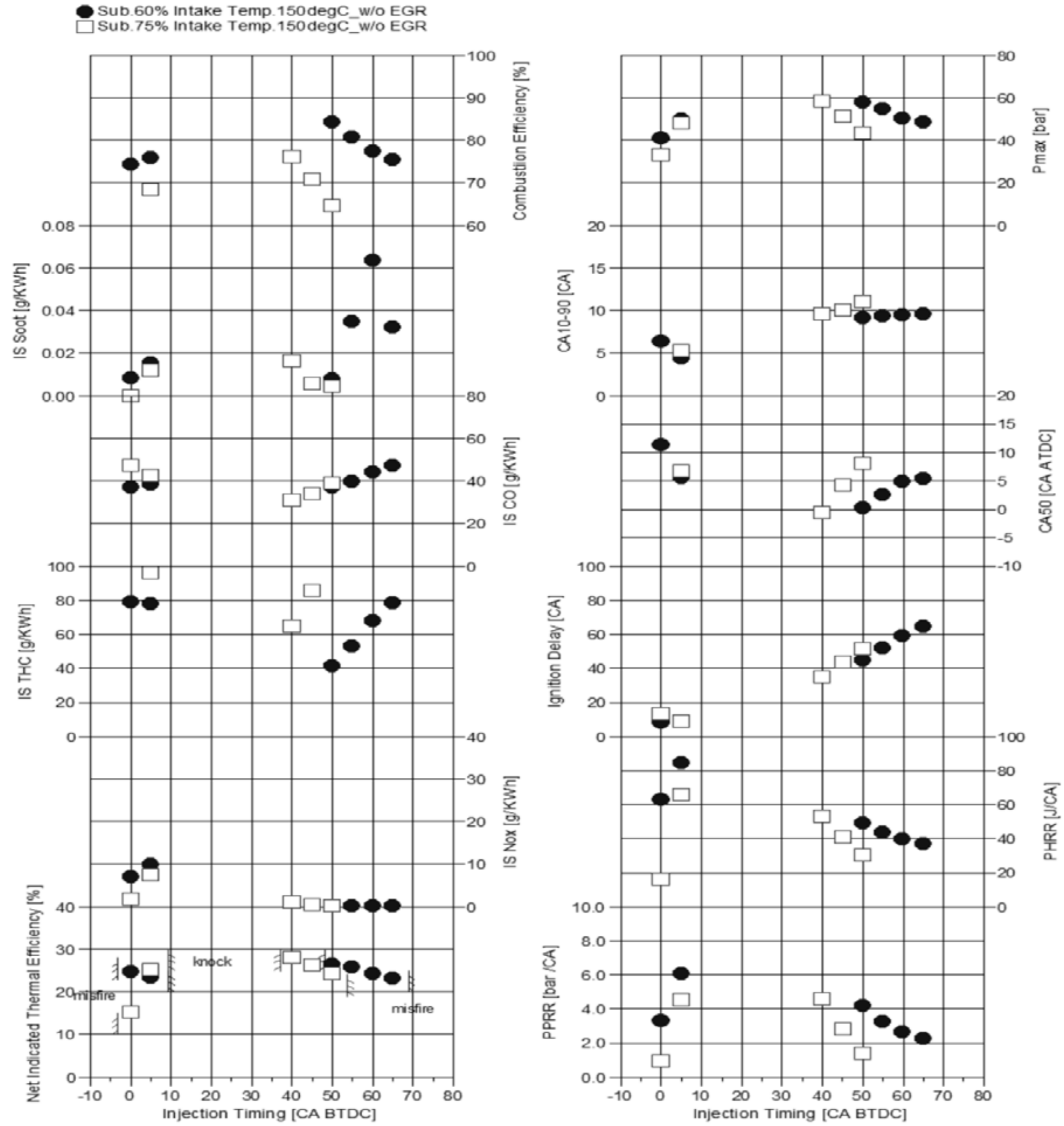

Figure 13 comparison of engine performance, emission and combustion characteristics at different gasoline substitutions without EGR, when single diesel injection timing is sweeping, and intake temperature is constant 150°C

Baseline diesel operation produces the highest soot. The smoke level decreases with increasing gasoline substitution ratio. Combustion efficiency of all dual-fuel conditions increases up to a maximum as SOI is advanced before it starts to decrease. The 60% substitution ratio operations have much higher combustion efficiency than the 75% substitution ratio at the same SOI of diesel injection, for both early and late diesel inject timings.

Figure 14 shows the in-cylinder pressure and HRRR with different substitution ratios of gasoline, at the same diesel SOI timing of 5^o CA BTDC with 150^o intake temperature and no EGR. Heat release process begins at similar times at each condition and to that of diesel only condition, with combustion duration being slightly longer with 75% substitution ratio, as evidenced by its higher pressure during the expansion stroke, while the heat release rate being higher at 60% substitution ratio compared to that of 75% substitution ratio, and the diesel only condition having the highest heat release rate. In-cylinder pressure of 60% gasoline substitution ratio has a higher peak pressure, but after that, both substitution ratios have the same pressure.

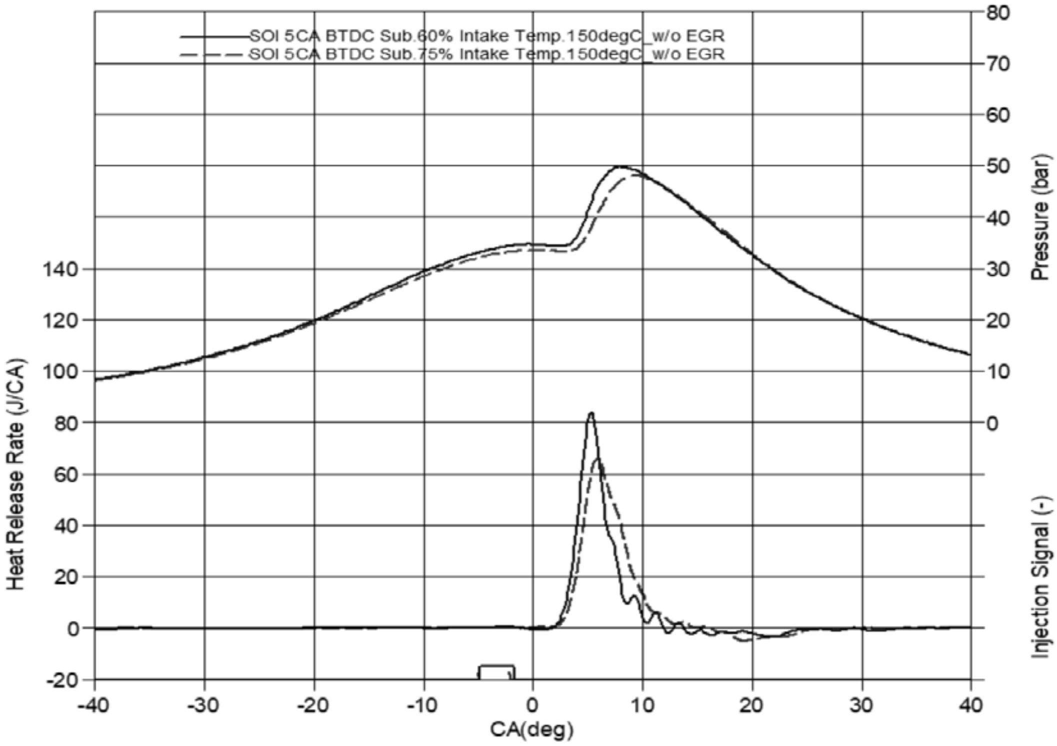

Figure 14 in-cylinder pressure and HRR with the late SOI at different gasoline substitution ratio

Figure 15 illustrates in-cylinder pressure and HRR with the optimized SOI for emission and efficiency at different gasoline substitution ratios, with 150^{o} intake temperature and no EGR. Figure 16 shows engine performance and emission with the same conditions as Figure 15, while displaying the effect of the injection timing for the same exact condition with a later injection as well. It must be noted that the 45% gasoline substitution ratio is ignored for the discussion in this section due to its very low thermal efficiency, and high levels of NOx and soot.

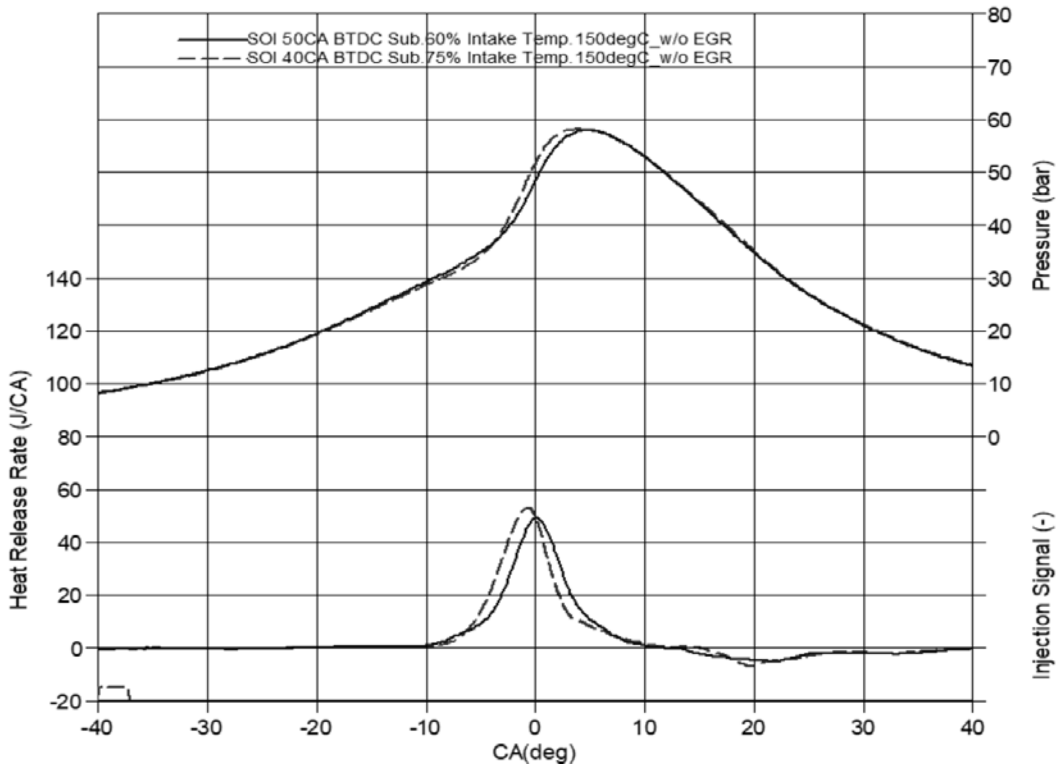

Figure 15 in-cylinder pressure and HRR with the early SOI at different gasoline substitution ratio

When the gasoline substitution ratio is lower, optimized SOI is advanced further, so that richer diesel mixture needs longer ignition delay to have proper combustion timing, and combustion is milder and PHRR is slightly lower due to less local diesel rich mixture area by means of earlier injection timing. In terms of emissions, lower gasoline substitution ratio, decreases NOx with more homogeneous diesel mixture, and same can be said for THC. The thermal efficiency at lower substitution ratio is worse, be it very minor, due to too advanced combustion timing. Again, when neglecting the 45% Gasoline substitution ratio, increasing the substitution ratio from 60% to 75%, increases the soot emission, probably due to having a lower combustion temperature to oxidise soot. 45% substitution ratio has high level of soot due to overall diesel rich mixture.

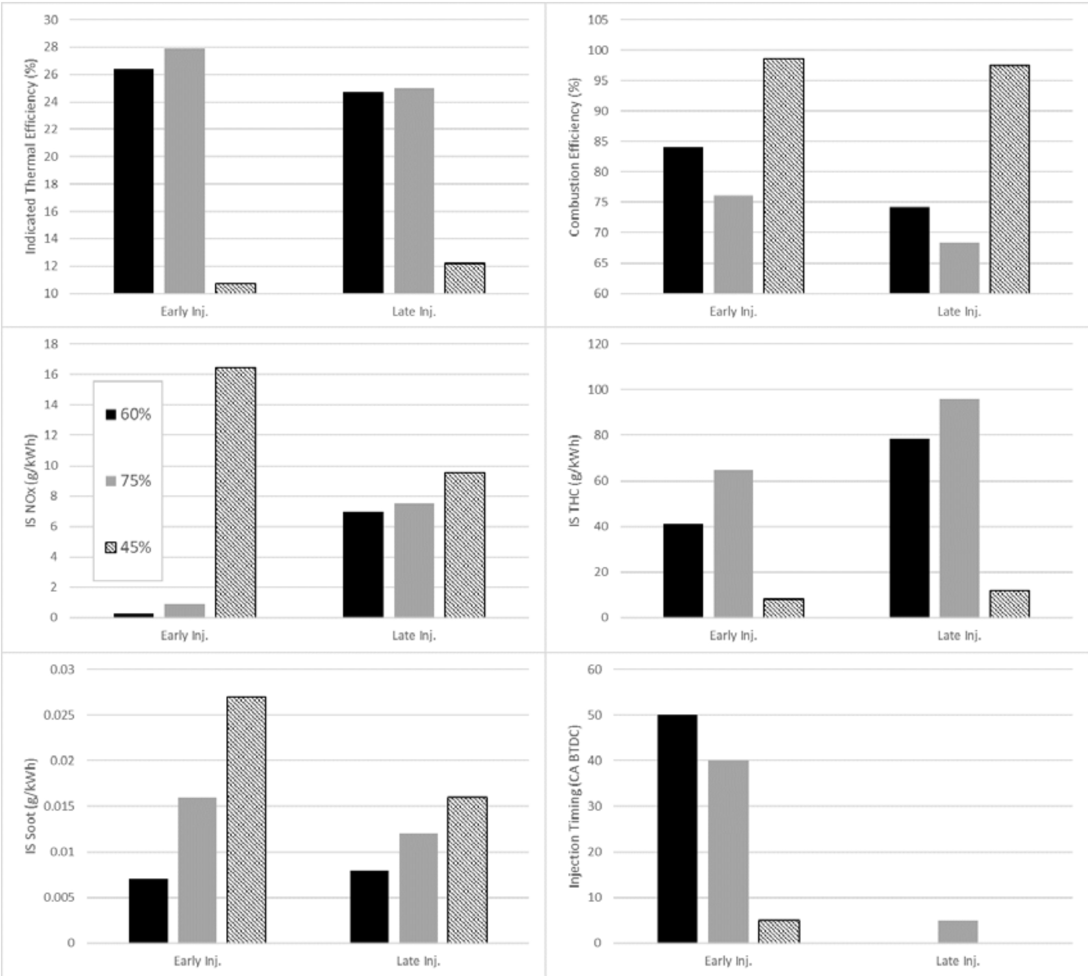

Figure 16 Engine performances and Emission with the optimized SOI at different gasoline substitution ratio

6.2. High speed imaging of fuel spray and combustion of dual-fuel operations

Table 5 shows the conditions chosen for further analysis using optical diagnosis with the glass piston head, based on the thermodynamic analysis completed using the metal piston.

Test Condition	Diesel Injection Strategy	Gasoline Substitution Ratio	Diesel SOI (°CA BTDC)
1	Single	60%	5
2	Single	60%	50
3	Split	60%	40-5
4	Single	75%	40

Table 5 Optimized conditions selected for optical analysis

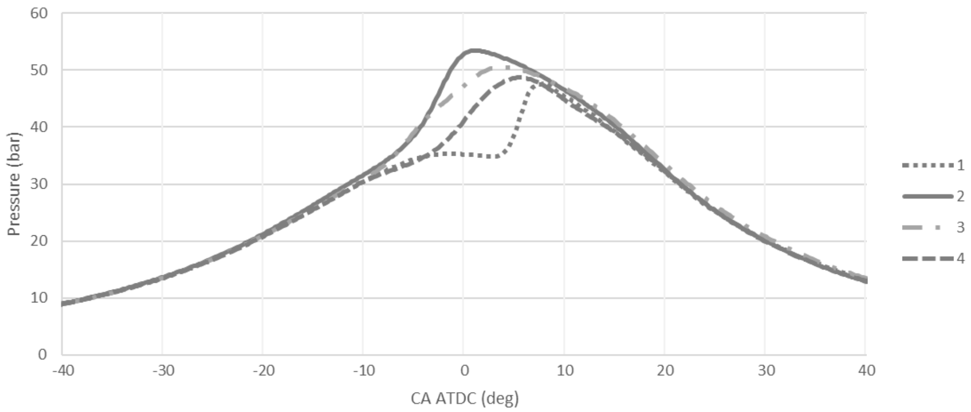

Figure 17 Pressure vs crank angle (P-theta) diagram

Figure 17 illustrates the in-cylinder pressure for the optimized operation conditions selected for optical analysis, as presented in Table 5, against the engine crank angle, showing their maximum in-cylinder pressure, and combustion duration.

Figure 18 shows in-cylinder pressure, HRR, combustion area and the image sequence obtained simultaneously through high speed video imaging of spray and diesel combustion without image intensifier during a single diesel injection operation, when SOI is 5^{o} CA BTDC. With an in-house MATLAB code, the displayed image is obtained by subtracting the background image from the original one.

1st picture at -1.2^{o} CA ATDC shows the start of fuel spray jets where fuel sprays are uneven due to uneven pressure inside of injector holes, a characteristic of the VCO type nozzle. The difference in the SOI and the optical image is caused by the delay in actuating the solenoid and lifting of the needle in the injector. At the next image, fuel spray jets developed more, and their penetration length becomes longer reaching almost the side wall of combustion chamber. 3rd image at 8.3^{o} CA ATDC shows the first combustion site at the tip of one of fuel sprays, which matches the beginning of main heat release rate. However, the heat release starts slightly around 5^{o} CA ATDC and is not visible in the combustion image because of the lower temperature. At 4th frame, combustion developed primarily with blue flame combustion due to premixed combustion and a little bit luminous flame due to diffusion combustion. Combustion is separated into 6 groups from the 6 fuel sprays. At 5th frame at 11.9^{o} CA ATDC, the area of luminous flame spreads around the chamber but the centre of chamber remains dark with no combustion.

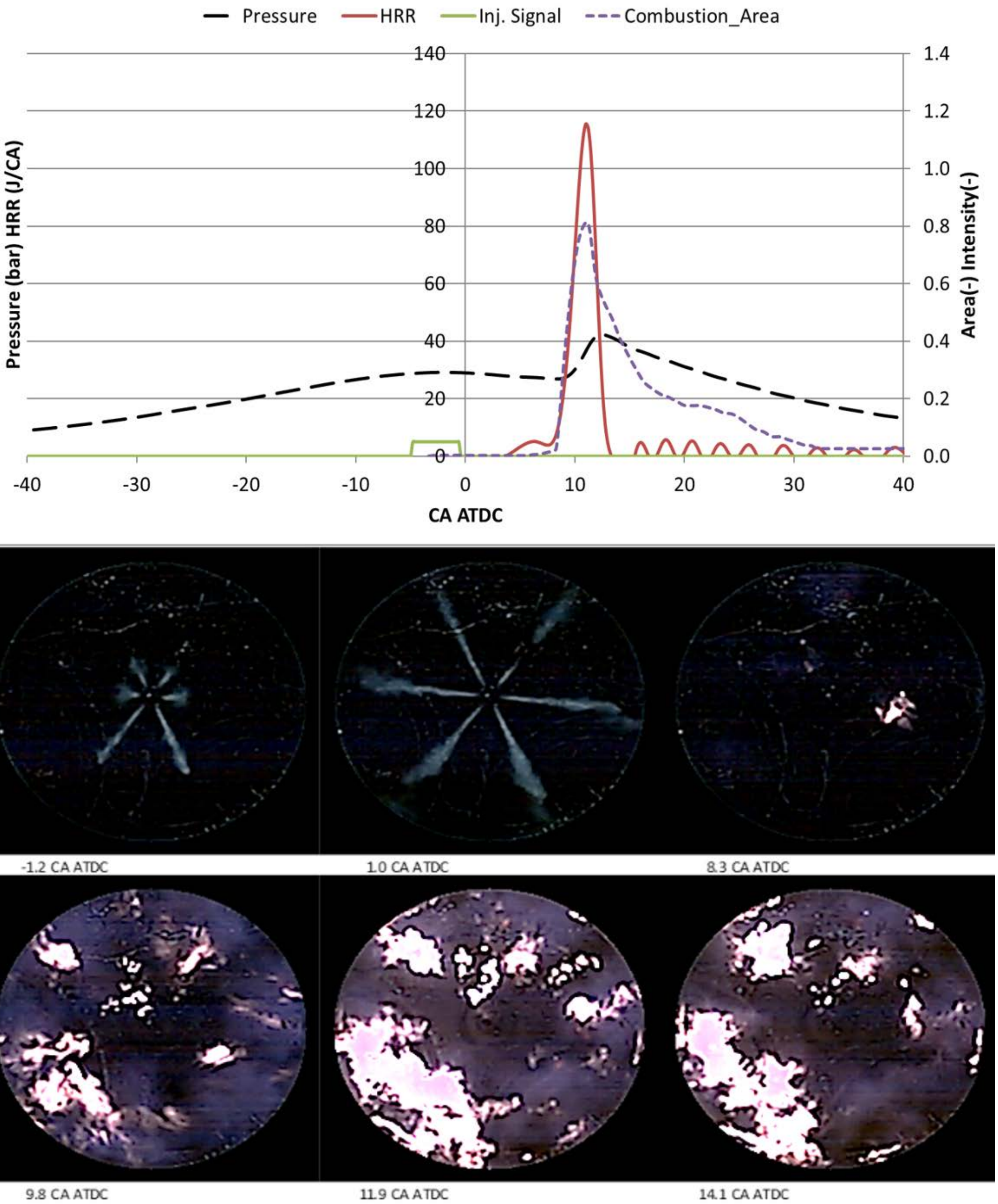

Figure 18 Direct combustion Image of diesel combustion without intensifier at 5CA BTDC of SOI

At 6th frame at 14.1^{o} CA ATDC, combustion is still observed even though HRR is finished when more visible light emission is produced by the combustion of soot with much less additional heat released.

Figure 19 shows in-cylinder pressure, HRR, combustion area, intensity of combustion luminosity of single dual-fuel combustion when SOI is 5^{o} CA BTDC, substitution ratio is 60%, by means of the intensified high-speed video imaging system. The HRR graph shows that around 1^{o} CA ATDC HRR occurs, which can be observed by the luminosity at the upper right side of the combustion chamber in the first image, possibly due to the higher gas temperature around exhaust valves. In the second frame of 2.2^{o} CA ATDC, combustion luminosity is increased, and the combustion almost spread throughout the entire combustion chamber except the centre and gaps between each of diesel fuel sprays. The timing of peak combustion area and intensity in the image correspond with peak HRR in the graph. Here, a lot of strong luminosity spots are observed. At this condition through all frames, combustion is not seen to spread to the entire combustion chamber. Although there is homogeneous gasoline-air mixture distributed throughout the cylinder, the lack of combustion luminosity in the black region indicates the lack of high temperature combustion in the regions of premixed lean gasoline/air mixture. This cause unburned fuel, meaning high THC and low combustion efficiency at duel fuel condition compare to diesel combustion. Later combustion like diesel combustion is seen at the last frame and in the combustion area and intensity curve as well.

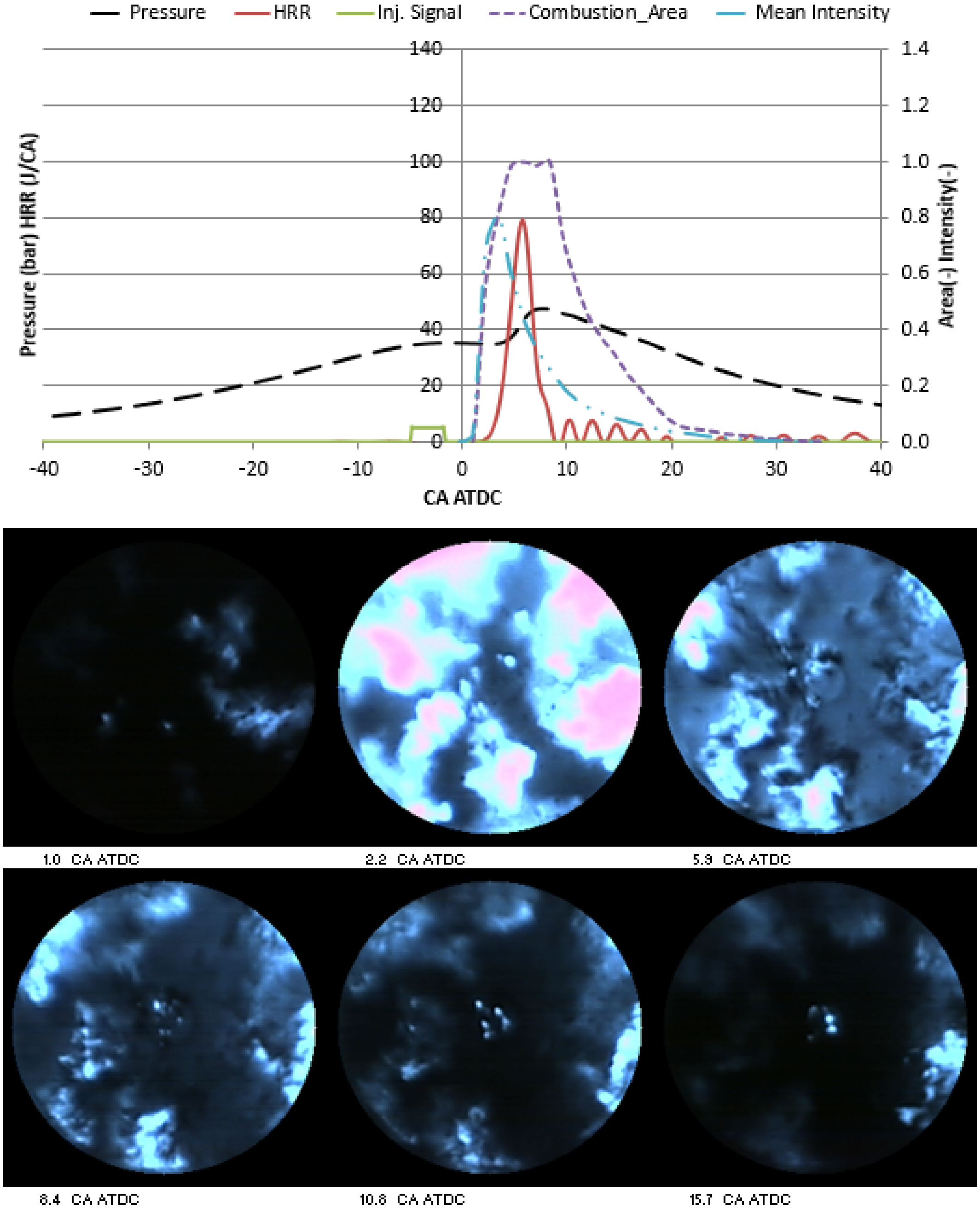

Figure 19 Direct combustion Image with intensifier with gasoline substitution ratio of 60% and diesel SOI at 5CA BTDC

Figure 20 shows the images from the same condition when no image intensifier is used for collecting the high-speed video, and therefore, only the luminosity from the diesel combustion is recorded. When compared with Figure 19, it shows a much smaller value of the total combustion area, with the same pressure and HRR curve. At the first frame at 1.7^o CA BTDC, the spray from the diesel injection can also be observed.

Figure 21 illustrates in-cylinder pressure, HRR, combustion area, intensity of combustion luminosity and the image sequence with image intensifier when the single diesel SOI is 50^o CA BTDC, with gasoline substitution ratio of 60%.

Compared to Figure 19 , the location of combustion beginning is at exhaust valve side but closer to wall of combustion chamber in the first and second frames due to longer penetration length due to the longer ignition delay of diesel fuel. Combustion spreads towards the centre from the periphery in the second to fourth frame as HRR and combustion area increase. Combustion luminosity is seen in the whole combustion chamber at CA 3.9^o to 1.5^o CA BTDC. Compared to the late single injection, less unburned fuel is present around the centre of chamber. In the last frame, late combustion is observed around the periphery. Overall, intensity of combustion luminosity is much lower with early injection than that with late injection because of lower combustion temperature and more uniformed mixture during combustion.

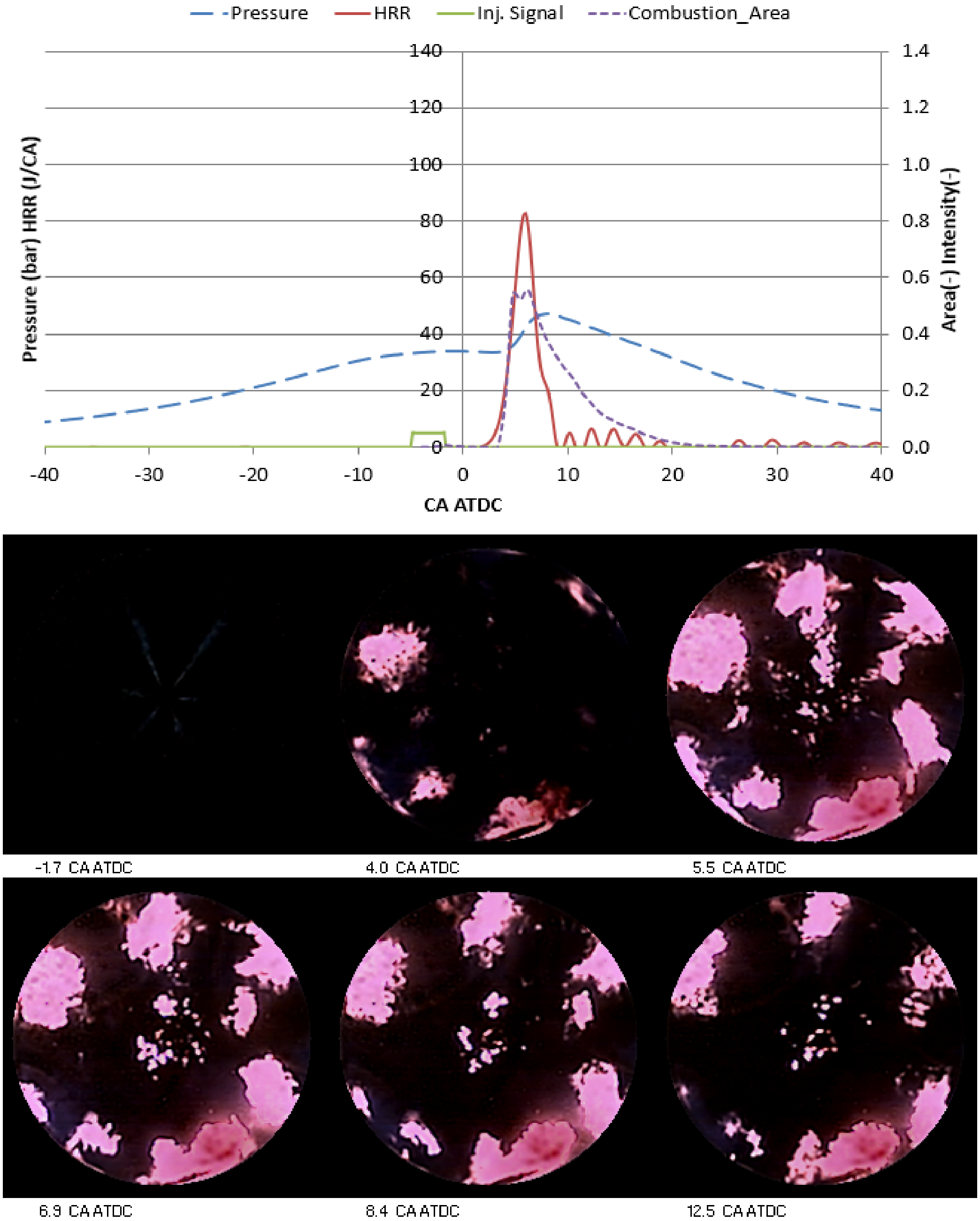

Figure 20 Direct combustion Image without intensifier with gasoline substitution ratio of 60% and diesel SOI at 5CA BTDC

Figure 22 shows the in-cylinder pressure, HRR, combustion area, intensity of combustion luminosity and the image sequence with image intensifier at split dual-fuel combustion when SOI is 40º-5º CA BTDC, substitution ratio is 60%, the second injection timing is the same as Figure 19. At the first frame -1.8º CA ATDC, combustion luminosity begins at the upper side of the injection nozzle. In the following frame, the luminance spreads, with stronger luminosity occurring around the injector nozzle close to the exhaust valves, and fuel spray jet is visualized thanks to the strong luminosity. At the third frame, the area with strong luminosity expands. This strong luminosity is due to diffusion combustion of diesel fuel mixture due to the shorter ignition delay time caused by higher in-cylinder temperature of the combustion of first injection. At 4th frame the combustion can be seen in the entire combustion chamber, where the area of the strong luminosity peaked and then decreased through fourth to sixth frame. In the graph, the intensity, pressure and combustion area peak at around the same point, but the HRR peaks at two different stages, one of which earlier that intensity, due to the initial diesel injection, and again at TDC, just before the intensity peak, caused by the second diesel injection, with the start of this second increase in HRR, matching the start of intensity and combustion area increasing just before the first frames CA. Strong luminosity from diffusion combustion indicates more soot formation, illuminating strongly during combustion. In the diffusion combustion flame, NO may not be generated so much due to the lack of oxygen. This diffusion combustion can enhance the temperature around the centre of combustion

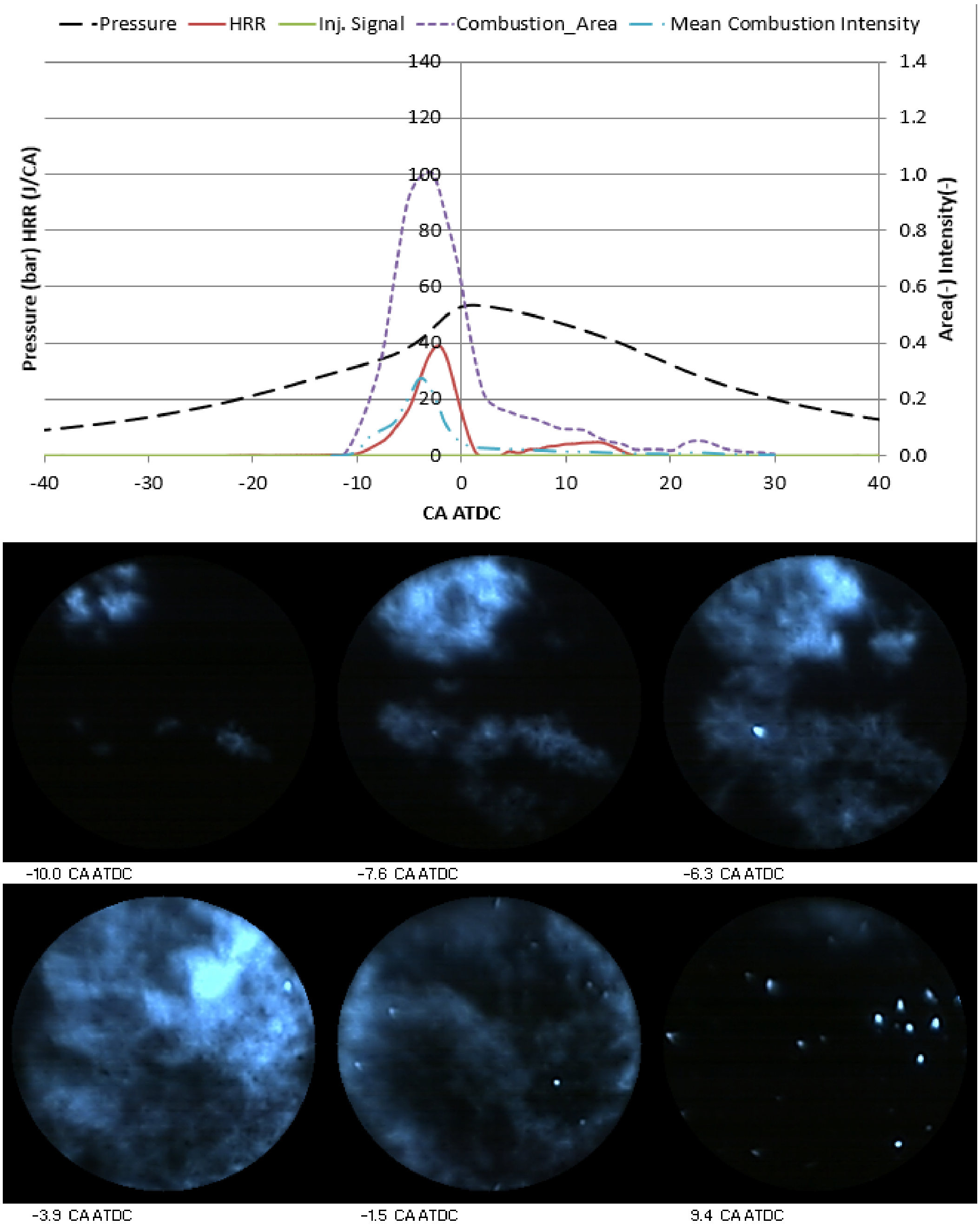

Figure 21 Direct combustion Image with intensifier with gasoline substitution ratio of 60% and diesel SOI at 50CA BTDC

chamber and reduce the temperature around the wall with shorter spray penetration length at combustion beginning before end of injection, which may reduce heat loss.

Figure 23 shows the images from the same condition when no image intensifier is used for collecting the high-speed video, thus, only the luminosity from the diesel combustion is recorded. Compared to Figure 22, it shows a much smaller value for the total combustion area, with the same pressure curve and similar, but not identical HRR curve, which could be caused by a slight difference in testing conditions between the two operations.

In the frames included in this figure, the path of the diesel injection is well illustrated, and it is easy to see how the combustion starts with the start of the second diesel injection and following the path of the injection spray as it develops, until the fourth frame at 0.3^o CA ATDC, where the entire combustion chamber is illuminated, but the luminosity of the diesel spray still dominating most of the image. The fifth frame shows the combustion area, and the intensity beginning to fade away, and at a very minimum by the last frame included in the figure at 15^o CA ATDC.

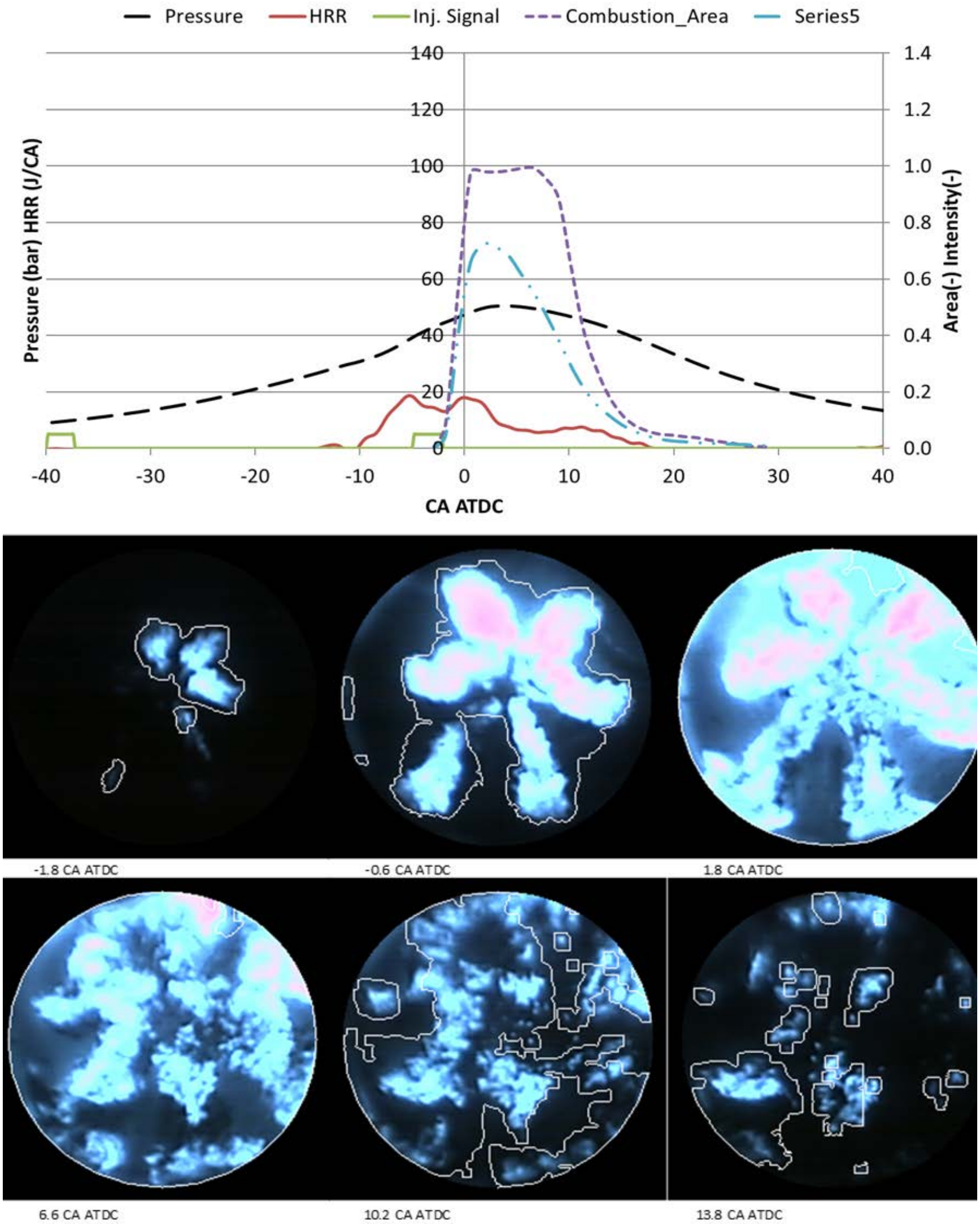

Figure 22 Direct combustion Image with intensifier when substitution ratio is 60%, diesel injection is split injection and SOI is 40-5CA BTDC

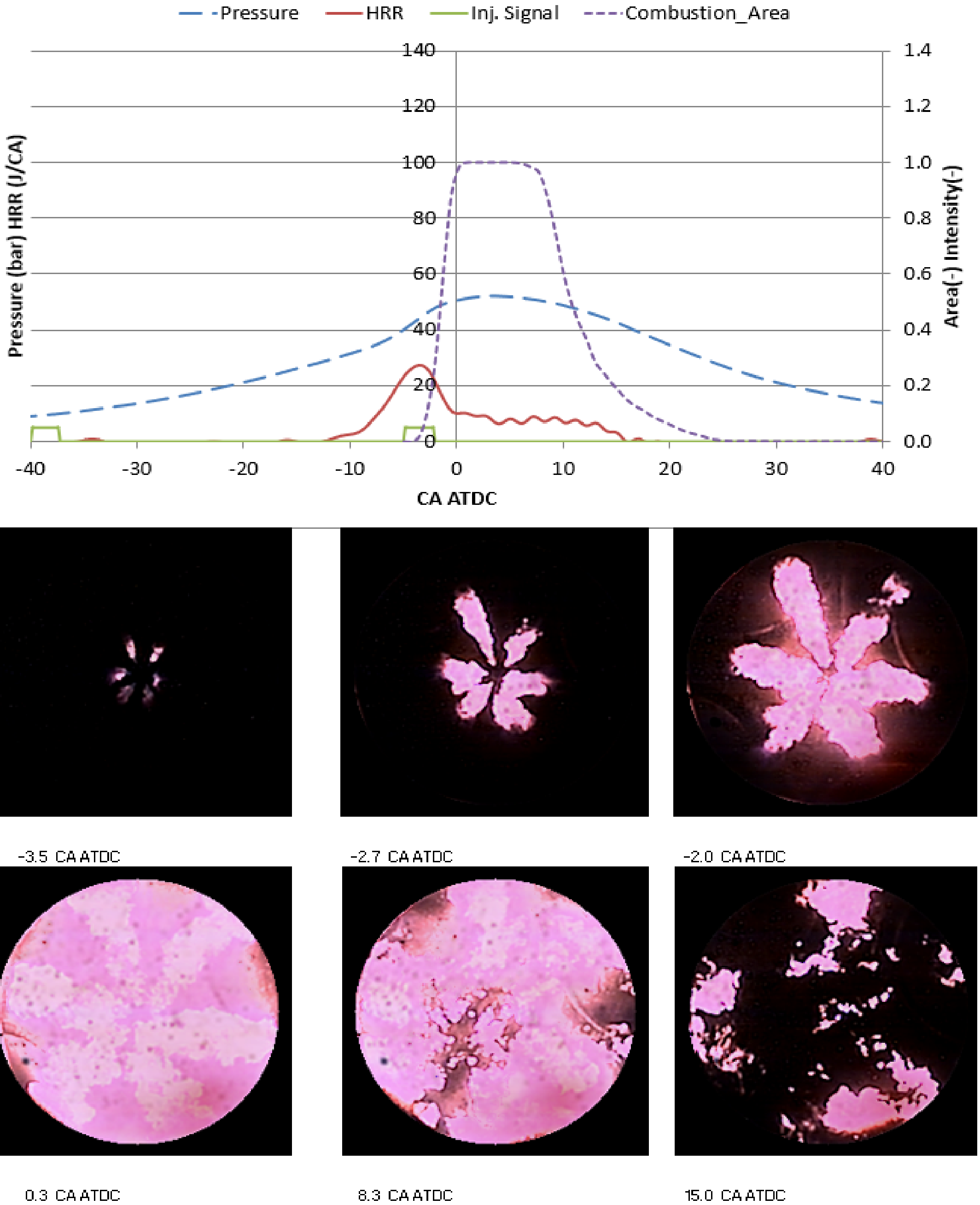

Figure 23 Direct combustion Image without intensifier when substitution ratio is 60%, diesel injection is split injection and SOI is 40-5CA BTDC

Figure 24 illustrates the in-cylinder pressure, HRR, combustion area, intensity of combustion luminosity and the image sequence with image intensifier at single dual-fuel combustion when diesel SOI is 40^{o} CA BTDC, gasoline substitution ratio is 75%. At first frame, combustion starts at the upper sides where the exhaust valves are located, and then at the next frame shows the advancement of the combustion still at the top and close to the exhaust valves.

Through the second frame to the fourth frame, which is -3.5^{o} CA ATDC to 2.6^{o} CA ATDC, combustion area spreads and mean intensity of combustion luminosity increases and peaks in the graph. Although combustion area, intensity and HRR reach the peak by this point, combustion luminosity is still very faint in some area around lower right side of the combustion chamber. At the fifth and sixth frames, some bright spots are seen that seems to be luminous flame of soot caused by injector dribbling.

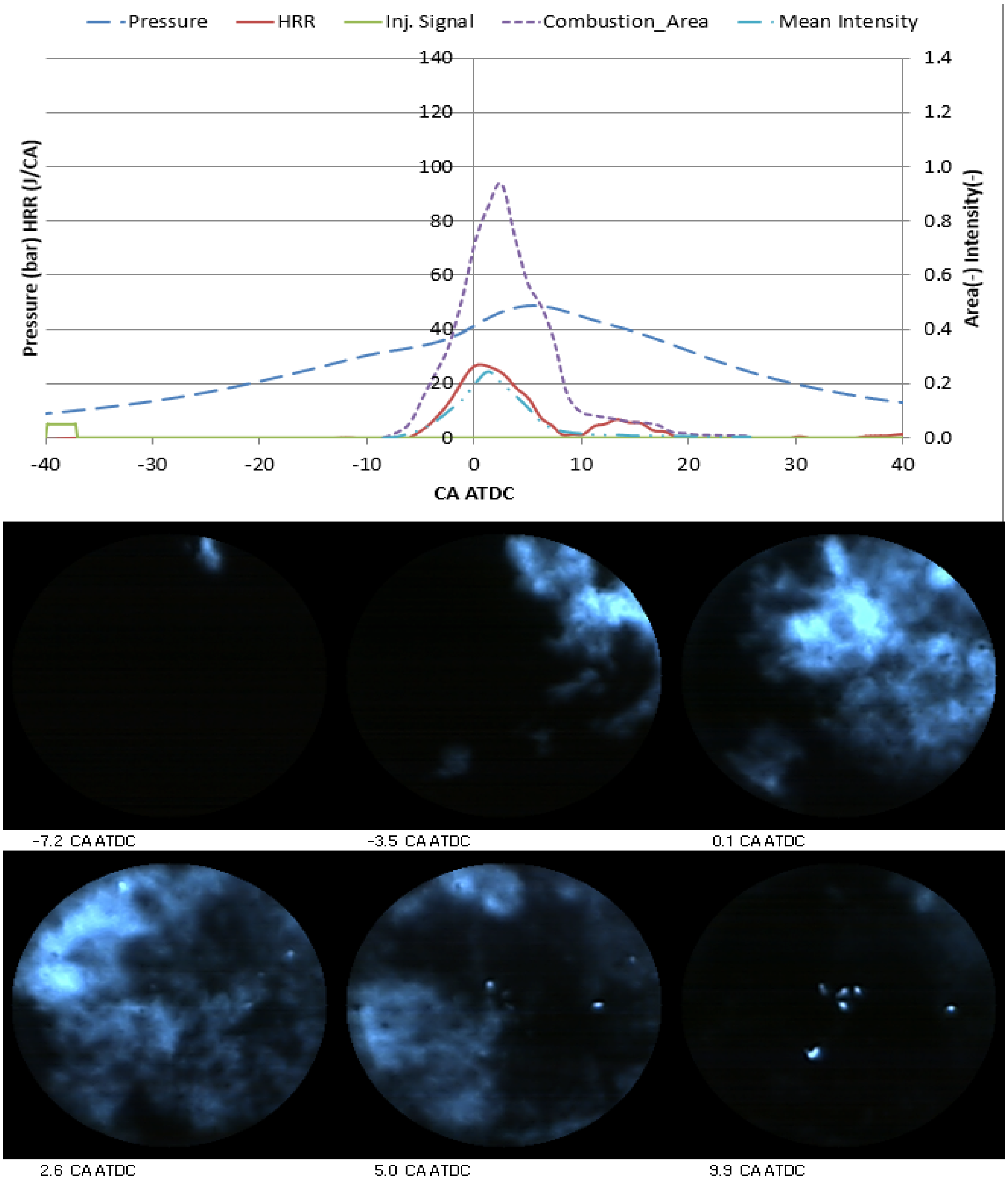

Figure 24 Direct combustion Image with intensifier when substitution ratio is 75%, diesel SOI is 40CA BTDC

7. Conclusions

This study investigated combustion characteristics, engine performance, and exhaust emission of diesel-gasoline dual-fuel operation in an optical DI CI engine by thermodynamic analysis, emission measurements, with high-speed imaging of in-cylinder processes. Different substitution ratios Effects and diesel injection strategies were studied with constant total fuel energy. Substitution ratios used, 45% 60%, and 75%, indicate gasoline fuel energy, and remainder from diesel.

Single late diesel injection conventional dual-fuel combustion, EGR reduces NOx, because of lower oxygen concentration and higher heat capacity, and increased thermal efficiency due to less heat loss and less THC. Lower intake temperature lowers NOx and increases thermal efficiency, suggesting external cooled EGR can benefit emission and thermal efficiency at part-load.

Early and late diesel injection dual-fuel combustion, EGR reduces THC and increase thermal efficiency. Increased intake temperature improves efficiency, THC and NOx because of higher in-cylinder temperature and homogeneous combustion, suggesting external-hot/internal EGR improves emission and efficiency at part-load.

Split injection Dual-fuel combustion is controlled by second diesel injection. Second injection timing and single injection timing, have similar effects on combustion characteristics, like ignition-delay, CA50, PHRR. The difference is due to heat release and/or distribution of diesel from first injection. With same second injection and single injection timing, NOx, THC and CO trends are similar, with slightly less NOx for split injection.

At same second injection and single injection timing, split injection has higher thermal efficiency. Late second injection improves thermal efficiency and increases soot compared to late single injection, because it makes more diffusion-like combustion with shorter second injection ignition delay, since first injection combustion starts before or during second injection. This combustion makes milder heat release rate, lower heat loss and more soot.

Single diesel injection combustion is more intense than dual-fuel, with higher PHRR and shorter combustion duration, due to diesel's higher chemical reactivity. Dual-fuel has higher THC and CO, hence lower combustion efficiency, due to gasoline homogeneous mixture in piston crevice and around cylinder liner, with low temperature gas hardly combusts or is quenched by lower temperature zone.

High-speed imaging of single injection dual-fuel operations reveal, combustion initiates around combustion chamber wall, downstream of diesel sprays, then spreads towards centre. Late injection results in stronger luminosity, because of diesel's greater diffusion combustion. With injections near TDC, diesel sprays surroundings dominate luminosity. With earlier injection, combustion is observed throughout combustion chamber. Combustion of early split injection produces weaker luminosity than early single because of leaner, more homogeneous mixture. Late split injection has much stronger luminosity than late single because of diesel from second injection combusts immediately; because higher in-cylinder temperature at second injection, reduces ignition delay, causing more diffusion-like combustion. Increased substitution ratio, requires later injection timing for similar combustion timing. Therefore, stronger luminosity of diffusion combustion of less mixed diesel, characterises higher substitution ratio combustions. For both diesel and dual-fuel combustion with single diesel injection, areas near centre or between diesel sprays remain dark indicating no combustion.

Abbreviations

AC	Alternating Current
ATDC	After Top Dead Centre
BTDC	Before Top Dead Centre
CA	Cam Angle
CFD	Computational Fluid Dynamics
CI	Compression-Ignition
CMOS	Complementary Metal Oxide Semiconductor
CO	Carbon Monoxide
CO_2	Carbon Dioxide
DC	Direct Current
DI	Direct Injection
DME	Dimethyl Ether
DMF	Dimethylformamide
DPF	Diesel Particulate Filter
EGR	Exhaust Gas Recirculation

EU	European Union
GHG	Greenhouse Gas
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat Release Rate
HSDI	High-Speed Direct Injection
IC	Internal Combustion
IMEP	Indicated Mean Effective Pressure
LED	Light-Emitting Diode
LTHR	Low Temperature Heat Release
NI	National Instruments
NO	Nitric Oxide
NO_X	Nitrogen oxide
PCCI	Premixed Charge Compression Ignition
PFI	Port Fuel Injector
PHRR	Peak Heat Release Rate
PPC	Partially Premixed Combustion
RCCI	Reactivity Controlled Compression Ignition

SI	Spark Ignition
SOI	Start Of Injection
TDC	Top Dead Centre
THC	Total Hydrocarbon
UHC	Unburnt Hydrocarbons
UK	United Kingdom
USA	United States of America
VBA	Visual Basic for Applications
VCO	Valve Covered Orifice

List of Figures

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