Performance Analysis and In-cylinder Visualization of Conventional Diesel and Isobaric Combustion in an Optical Diesel Engine

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Abstract

Compared to conventional diesel combustion (CDC), isobaric combustion can achieve a similar or higher indicated efficiency, lower heat transfer losses, reduced nitrogen oxides (NO_x) emissions; however, with a penalty of soot emissions. While the engine performance and exhaust emissions of isobaric combustion are well known, the overall flame development, in particular, the flow-field details within the flames are unclear. In this study, the performance analysis of CDC and two isobaric combustion cases was conducted, followed by high-speed imaging of Mie-scattering and soot luminosity in an optically accessible, single-cylinder heavy-duty diesel engine. From the soot luminosity imaging, qualitative flow-fields were obtained using flame image velocimetry (FIV). The peak motoring pressure (PMP) and peak cylinder pressure (PCP) of CDC are kept fixed at 50 and 70 bar, respectively. The two isobaric combustion cases, achieved using multiple injections, are maintained at the CDC PMP level of 50 bar for the low-pressure case (IsoL) and CDC PCP level of 70 bar for the high-pressure case (IsoH). For each operating condition, soot luminosity signals are captured at a frame rate of 20 kHz, and a semi-quantitative velocity flow-field is obtained from FIV post-processing. Consistent with previous metal engine experiments, isobaric combustion – in particular IsoH, resulted in similar gross indicated efficiency, lower heat losses but higher exhaust losses, compared to CDC. The soot luminosity images of CDC show initial signals originated close to the bowl-wall for certain jets while for the isobaric combustion, the flames corresponding to each jet are clearly distinguished during the earlier flame development process. The vector field distribution within the flames shows the transition of flame-wall impingement to flame-flame interaction regions between the neighboring jets for each combustion mode. Furthermore, higher flame-flame interaction regions and uniform distribution of signals around the combustion chamber for isobaric combustion, justifying higher soot formation and lower heat transfer losses, respectively, compared to CDC.

Introduction

Most of the projections indicate that the future of transportation is constituted of transport technology mix. This mix includes conventional internal combustion engines, hybrid electric, battery-electric, and fuel-cell electric vehicles [1,2]. However, to make an immediate environmental impact, improving the thermal efficiency and reducing the exhaust emissions of internal combustion (IC) engines are an utmost necessity [3]. In particular, the replacement

of IC engines in the heavy-duty transportation sector poses greater difficulties due to extreme power requirements [4]. Over the last few decades, constant volume combustion using low-temperature combustion regimes such as homogeneous charge compression ignition (HCCI) [5–9], reactivity charge compression ignition (RCCI) [10–12], and partially premixed combustion (PPC) [13–15] have been investigated given the efficiency of Otto cycle being higher than other reciprocating engines [16]. However, significant improvement in brake thermal efficiency (BTE) has not been achieved due to various individual energy losses in the process of converting the fuel chemical energy into net work.

One way to reduce the losses – in particular the mechanical and heat transfer losses, is the effective utilization of isobaric combustion, achieved using multiple injections [17–19]. Isobaric combustion controls the heat release profile to attain constant pressure combustion (i.e. theoretical diesel cycle) due to suppressed peak in-cylinder pressure and averaged in-cylinder temperature, which leads to higher brake thermal efficiency. Due to lower peak cylinder temperature, the NO_x emissions can be reduced with a limitation of soot emissions. While several previous studies [18–22] addressed the engine performance and emissions of isobaric combustion using multiple injections from a single injector, in-cylinder visualization remains limited. It was reported in previous studies [23,24] that successive fuel injection events in the high-temperature reaction zones enhanced the localized fuel-air rich mixtures which led to increased soot emissions. However, it was unclear how the interaction of flame-wall and flame-flame localized regions in a heavy-duty engine would impact the soot formation and heat transfer losses.

To enhance the understanding of complicated jet structures, the in-flame flow-fields could be measured by the application of particle image velocimetry (PIV) to flames, i.e. flame image velocimetry (FIV) [25]. Through FIV, the change in the soot luminosity image contrast due to in-flame motion with or without swirl can be effectively captured. Not only the FIV measurement can be applied to line-of-sight integrated soot luminosity images, but also the 2D laser sheet visualization of soot particles [26]. Previous studies applied this approach to look in swirl ratio estimation [27], swirl-induced vortex [28], injection-induced vortex [27,29,30], and diesel knock [31].

In the present study, conventional diesel combustion (CDC), low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion were investigated based on engine performance

coupled with 1D GT-Power simulations. A double-injection strategy for CDC and three injection events for isobaric combustion were executed to achieve stable combustion, consistent with previous studies. For a given combustion mode, the high-speed imaging of Mie-scattering and soot luminosity were conducted for non-reacting and reacting conditions, respectively. Finally from the soot luminosity images, the flow-field distribution within the flames was analyzed using FIV to assess the flow development associated with flame-wall and flame-flame interaction. These experiments were performed in a single-cylinder, heavy-duty optical diesel engine with n-heptane fuel.

Methodology

Engine test facility

Figure 1 shows the schematic of the single-cylinder optical diesel engine utilized for the conventional diesel and isobaric combustion experiments. The in-line six-cylinder Volvo engine has been modified with five de-activated cylinders i.e. no intake/exhaust valves assembly and only one active cylinder. As listed in Table 1, the heavy-duty engine has a bore and stroke of 131 mm and 158 mm, respectively, resulting in a displacement volume of 2130 cm³. An ω -shape piston mounted on the titanium piston-holder, which in combination with Bowditch extended piston, a metal liner, and a windows-holder provides optical access to the combustion chamber. The windows-holder has three side quartz windows, one of which was utilized for the background illumination during the Mie-scattering imaging, the rest of the windows allow laser access and side-view imaging, which has not been used in this study. The geometrical compression ratio of the engine is 15 by utilizing a piston with a squish height of 2.3 mm; however, due to blow-by

Figure 1. Schematic of single-cylinder, heavy-duty optical diesel engine

losses, the effective compression ratio was reduced to 11.7.

The intake air and nitrogen pressures were adjusted using mass flow controllers (Brooks Instrument SLA5853), while the exhaust pressures were controlled using a backpressure valve (AMM 730). For measuring the total injected fuel mass, a mass balance system (RADWAG PM) was used. Several pressure sensors and thermocouples were installed in the cooling water system, lubricating oil circuit, fuel line, intake, and exhaust lines for the constant monitoring of local pressure and temperature. The piezoelectric pressure sensor (AVL GH01D) installed in the windows-holder, in combination with a charge amplifier (AVL FI PIEZO) was used to measure the in-cylinder pressure data using a rotary encoder (Linde RSI 503) with a crank angle resolution of 0.2 degrees. A distributed pump common-rail system (Delphi F2E) was used for the direct-injection of fuel through a centrally-mounted solenoid injector. The injector nozzle has a six-hole configuration with a nominal hole diameter of 240 μ m. A LabVIEW FPGA-based real-time embedded control system (NI 9038 CompactRIO) was used for the engine operation, and the data acquisition.

Table 1. Engine specifications

Displacement volume [cm ³]		2130
Bore [mm]		131
Stroke [mm]		158
Connecting rod [mm]		255
Compression ratio		11.7 (effective)
Valve timings [°CA aTDC]	Intake (open)	-360
	Intake (close)	-177
	Exhaust (open)	151
	Exhaust (close)	338
Fuel injection system		Delphi F2E common-rail
Number of nozzle holes		6
Nominal hole diameter [μ m]		~240

Table 2. Engine operating conditions

Engine speed [rpm]		1200
Intake/Exhaust pressure [bar]	CDC	~1.5/1.8
	IsoL	~1.5/1.8
	IsoH	~2.1/2.4
Intake air temperature [°C]		~25
Coolant temperature [°C]		~60
Injection pressure [bar]		1500
Fuel		n-heptane
Fuel injection strategy	CDC	Double
	IsoL	Triple
	IsoH	Triple
Injection timing [°CA aTDC] and duration [μ s]	CDC	-3 (270), 1 (700)
	IsoL	-5 (270), 0 (300), 7.7 (600)
	IsoH	-5 (270), 0 (300), 7 (575)
Overall air-excess ratio [λ]	CDC	3
	IsoL	3
	IsoH	4.1
Fuel MEP [bar]		15.8 $\pm$ 0.5
Total injected fuel mass [mg/cycle]		76 $\pm$ 2

Firing mode	5-skip firing
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The engine was coupled to an AC motor for the fixed engine speed tests at 1200 revolutions per minute (rpm). The engine operating conditions are summarized in Table 2. Three combustion cases of CDC, IsoL, and IsoH combustion were tested. The peak motoring pressure (PMP) and peak-cylinder pressure (PCP) of CDC were maintained at 50 and 70 bar, respectively, similar to our previous studies [19,21]. Corresponding to the pressure boundaries of CDC combustion, the PCP for IsoL and IsoH was fixed at 50 and 70 bar, respectively. These in-cylinder pressures were accomplished using absolute intake air pressure of 1.5 bar for CDC and IsoL, and 2.1 bar for IsoH combustion. Noting that the exhaust pressure was kept 0.3 bar higher than the intake pressure for all the conditions to mimic turbocharger conditions as that of the metal engine. The intake air temperature was fixed at 25°C while for thermally stable engine conditions. The coolant temperature was maintained at 60°C. The injection pressure was fixed at 1500 bar and the experiments were performed using n-heptane fuel. A double-injection for CDC and a triple-injection strategy for both isobaric combustion cases were implemented, with the respective injection timings and durations are shown in Table 2. The effect of the number of injections on the isobaric combustion can be found in our previous work [19,21]. The fuel mean effective pressure (FuelMEP) and total injected fuel mass per firing cycle were fixed at around 15.8 bar and 76 mg, respectively, which resulted in an overall air-excess ratio of 3 for CDC and IsoL; and 4.1 for IsoH combustion.

The engine was operated for 50 cycles using a 5-skip firing mode, i.e. the fuel injection occurred every 5 continuous firing cycles followed by 5 motoring cycles. The skip firing operation is vital for minimizing the thermal stresses on quartz windows. Out of 5 continuous firing cycles (and 25 total firing cycles) for each combustion mode, the first firing cycle did not result in stable combustion and therefore the rest of the 20 firing cycles were ensemble-averaged to determine the rate of heat release (RoHR). The RoHR was used to determine the combustion phasing, represented by the crank angle location for 10% (CA10), 50% (CA50), and 90% (CA90) heat release.

GT-Power simulations

The software GT-Power, version 2019 was used for 1D simulations of the combustion cylinder by replicating the geometrical parameters of the Volvo engine, including the intake/exhaust valve lift profiles. Figure 2 shows a non-predictive combustion model to adjust the in-cylinder pressure and heat release rate to match the experimental data. The model is used to calculate the heat transfer and exhaust losses which cannot be estimated from the experiments due to the unavailability of exhaust emission analyzer. The estimation of the air-excess ratio (λ) is also confirmed with the model.

As in our previous studies [32–34], Woschni GT heat transfer model, that is based on the classical Woschni equation [16], was implemented in the simulations to account for the in-cylinder heat transfer losses. For all tested conditions, the heat transfer coefficient (ranging from 1.0 to 1.5) was adjusted during all four strokes to closely match the in-cylinder pressure and heat release rates. The exhaust flow rate and temperature were determined from the model, and then integrated to estimate the exhaust energy.

High-speed imaging and flame image velocimetry (FIV)

The bottom-view images were captured through the quartz piston and a 45° UV mirror mounted on the hollow space of the extended piston using a high-speed camera (Photron Fastcam SA-X2) equipped with a camera lens (Nikon Nikkor 50 mm f/1.2). The frame rate of the camera was set to 20 kHz which resulted in a 0.35°CA temporal resolution between frames (at an engine speed of 1200 rpm) for both the liquid phase fuel-injection (i.e. Mie-scattering) and soot luminosity imaging. For the given frame rate, the maximum image resolution of 768 x 768 was selected. The exposure time was fixed at 2.5 μ s, with different lens aperture of f/1.2 for Mie-scattering and f/5.6 for soot luminosity imaging. These camera and lens settings maximized the signal strength during the non-reacting and reacting conditions. Additionally, a neutral density filter (ThorLabs ND2) was used to avoid the signal saturation and attaining the desirable contrast for flame image velocimetry (FIV) post-processing.

Figure 2. A layout of Volvo D13 engine using GT-Power model

Figure 3. The FIV procedure of vector field calculation using the raw soot luminosity images for an image pair of low-pressure isobaric (IsoL) combustion

For FIV analysis, the raw high-speed soot luminosity images were processed using LaVision Davis 8.4.0 software. The contrast variations in luminosity signals for the consecutive image pairs were used to estimate the in-flame flow-field distribution. An example of the FIV procedure is depicted in Figure 3. The raw image pair was pre-processed using a 3×3 Gaussian smoothing and sharpening filters for minimizing the background noise and contrast enhancement, respectively. This was followed by PIV time-series operation which includes the following steps: circular geometric mask set up for the region of interest, a sequential cross-correlation algorithm applied to consecutive image frames using multi-pass Discrete Fourier Transform (DFT), and a universal outlier detection median filter for removing the erroneous vectors.

The multi-pass DFT cross-correlation algorithm is used to track the pixel displacement of the signal contrast variations for the consecutive image pairs (i.e. 0+1, 2+3, etc.). This step is computed within sub-regions of the original image, i.e. interrogation window size. The window size is selected based on the quarter rule [35], i.e. it should be at least four times bigger than the largest pixel displacement. For this study, the initial and final window sizes of 96×96 and 24×24 were selected using 12 passes of the iteration process with the vectors calculated for 50% overlap in each pass. Finally, the inaccurate vectors were deleted using the median smoothing filter which was set at 1 with a region of 3×3 , similar to previous studies [36,37], and the empty spaces in the image were filled using interpolation. Following the optimization procedure, the instantaneous vectors describing the flow structure details within flames for the given image pair are shown in Figure 3 (right) with a maximum pixel displacement resolution of 64.

Results and Discussion

In-cylinder pressure and performance analysis

Figure 4 shows the individual and averaged in-cylinder pressure, rate of heat release (RoHR) traces, and the injector current signals for CDC (black lines), IsoL (blue lines), and IsoH (green lines) combustion. CDC shows a single-peak ROHR corresponding to the main (i.e. second) injection while IsoL and IsoH combustion shows

Figure 4. In-cylinder pressure, rate of heat release (RoHR), and the injector current signals for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

Figure 5. The start of combustion (CA10), combustion phasing (CA50) and burn duration (CA10-90) for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

two peaks of RoHR due to the last two injection events. The pilot (i.e. first) injection was implemented to increase the pre-combustion mixing therefore reducing the pressure rise rate and soot emissions [38–40]. Compared with previous studies conducted on the metal engine with diesel fuel [19,21], the isobaric combustion was difficult to achieve in the optical engine. This could be attributed due to four factors: lower intake air and coolant temperature to prevent the wear-out of piston rings, ignition delay differences of n-heptane compared to diesel fuel, different rate of injection (ROI) characteristics of the central injector, and skip-firing compared to continuous firing engine operating mode.

Figure 5 shows the start of combustion (i.e. CA10), combustion phasing (i.e. CA50), and burn duration (i.e. CA10-90) for all three combustion modes. Compared to CDC, both IsoL and IsoH show an early start of combustion and it is evident by a first small peak

Figure 6. The energy distribution of gross indicated efficiency, heat losses, and exhaust losses for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

Figure 7. The estimated in-cylinder temperature for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

of heat release, corresponding to the second injection of isobaric combustion cases (see Figure 4). For maintaining the combustion isobaric, the second and third injections are selected such that the pressure fluctuations can be minimized. Both the isobaric combustion cases resulted in a higher CA50 and stretched-out heat release profile (i.e. higher burn duration) than CDC combustion probably due to the delivery of fuel from the late third injection in the hot mixtures of the second injection. Compared to IsoL, IsoH combustion shows lower burn duration, i.e. faster burning due to short injection dwell period between the last two injections. This resulted in a higher effective expansion ratio for IsoH combustion.

Figure 6 shows the impact of different combustion modes on the energy distribution of gross indicated efficiency, heat transfer, and exhaust losses, assuming 100% combustion efficiency. Late combustion phasing of IsoH combustion did not result in a gross indicated efficiency lower than CDC. This could be due to an optimized combination of late combustion phasing, higher lambda and a slower heat release rate for IsoH combustion than CDC. However, IsoL combustion has resulted in the lowest efficiency due to an extended burn duration. Higher burn duration for both isobaric combustion cases has also resulted in higher exhaust losses in comparison with CDC. The heat transfer losses show an opposite trend as that of exhaust losses, i.e. isobaric combustion show lower heat losses than CDC. This can be explained with a lower peak RoHR (see Figure 4), resulting in lower localized flame temperature, which is confirmed from Figure 7. The higher air-excess ratio of IsoH compared to IsoL and CDC for the same FuelMEP has resulted in even lower in-cylinder temperature.

In-cylinder visualization for reacting and non-reacting conditions

Figure 8 (middle) shows an example of raw soot luminosity image for IsoL combustion in which the injector position is represented by a yellow circle in the centre while the piston bowl-wall (i.e. 92 mm diameter) and squish region thickness (i.e. 38 mm diameter) are shown in red circles. The crank angle location (°CA aTDC) is shown in the top-left corner of the image. Figure 8 (left) shows the

Figure 8. Mie-Scattering (left), a raw image (center) and soot luminosity imaging (right) for CDC, low-pressure isobaric (IsoL) and high-pressure isobaric (IsoH) combustion. Shown at the center is an example of raw luminosity image highlighting the piston-bowl wall and squish region

selected liquid-phase fuel injection process using Mie-scattering imaging for CDC (top row), IsoL (middle row), and IsoH (bottom row) cases. Irrespective of the combustion regime, the first signals were observed at 2.5°CA after the start of injection, meaning an injection delay of 2.5° between the electronic start of injection (eSOI) and the actual start of injection (aSOI), commonly known as hydraulic delay. This imaging was useful in deciding the minimum injection duration needed for the first injection such that enough pre-combustion mixing could be achieved without resulting in combustion before the top dead center (TDC).

Figure 8 (right) shows the soot luminosity images for all three combustion modes. For a given condition, the selected images were chosen from one of the five firing cycles of a 5-skip firing mode, which was closest to the ensemble-averaged pressure trace. The selection of individual images compared to the ensemble-averaged image was preferred as the latter case has resulted in smeared-out flame boundaries, similar to previous studies [41,42]. As shown later, this is critical to understand the flow-field details for different combustion modes. For CDC, the initial soot luminosity signals were found close to the bowl-wall for some of the jets from the second injection event. This could be due to the distribution of local equivalence ratios from the early first-injection which enriched the mixtures near the wall. These jets show an indication of flame-wall and flame-flame interaction at 15°CA aTDC, which corresponds closely to the peak RoHR phase (Figure 4). In comparison, for both the isobaric combustion cases, the luminosity signals first emerge on the jet axis, close to nozzle holes, and flames for each jet are clearly distinguished during the earlier flame development process. This could be due to the close-coupled injections, which resulted in lesser pre-combustion mixing for isobaric combustion compared to CDC. The flames corresponding to the second injection were not visible for IsoL as opposed to IsoH combustion, although Mie-scattering imaging shows similar fuel injection, as shown in Figure 8 (left). This could

be either due to the small first-peak RoHR (Figure 4) or the neutral density filter used which reduced the signal strength during the earlier flame development process. In IsoH combustion, the luminosity signals from the second injection originated earlier at 6.5°CA aTDC due to increased fuel-air mixing process due to higher λ . This resulted in an advanced combustion phasing with reduced burn duration (Figure 5) of IsoH combustion compared to IsoL combustion. This is well complemented by Figure 9, where the averaged soot intensity counts were found to be at earlier crank angles for IsoH than IsoL combustion. At 11.2°CA aTDC, the flame interaction between the second and third injections is seen, which merges and progressed towards the wall, resulting in flame-wall interaction. At 17.7°CA aTDC, flame-flame interaction

Figure 9. The averaged soot intensity counts as a function of crank angle degree for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

Figure 10. The flow-field distribution for CDC, low-pressure isobaric (IsoL), and high-pressure isobaric (IsoH) combustion

regions were initiated between the neighbouring jets which at later crank angles dominate the combustion chamber.

Strong flame-flame interaction regions for isobaric combustion suggest higher soot formation due to limited fuel-air mixing, compared to CDC. From Figure 9, the soot intensity counts for isobaric cases are 2.5-fold higher than CDC, thereby would result in higher soot emissions and has been confirmed from our previous metal engine experiments [19,21]. We also observe that the luminosity signals were not only limited within the piston-bowl but also found in the squish region (seen at 17.7°CA aTDC) for isobaric combustion compared to CDC, which would result in higher carbon monoxide (CO) and unburnt hydrocarbon (UHC) emissions [19].

Figure 10 shows the flow-field distribution for CDC (top row), IsoL (middle row) and IsoH (bottom row) combustion cases. The FIV analysis is reliant upon the combustion luminosity, which is dominated by incandescence signals of hot soot particles within the high-temperature flames [30,37]. Therefore, the FIV is applied near the peak heat release rate region (Figure 4) where the averaged soot luminosity signal (Figure 9) is higher. For CDC, strong flame wall-interaction regions are seen at 6 and 8 o'clock

positions (shown in red boxes), where these flames traveled along the bowl-wall. At 17.5°CA aTDC, the flame-flame interaction occurred between these jets, and the vector-fields within the flames are directed towards the center of the combustion chamber. For isobaric combustion, clear flame-wall interaction regions for all six jets are seen. For example, the flames at 10 and 2 o'clock position after hitting the bowl-wall impingement point progressed in clockwise and anti-clockwise direction along the bowl-wall. This is followed by the interaction of adjacent flame-wall regions traveling in the opposite direction, i.e. flame-flame interaction and the flames progressed in the inward direction.

Because of larger flame-flame interaction regions for isobaric combustion than CDC, the soot-formation is higher. However, due to uniformly distributed signals for isobaric combustion, the heat release rate is lower and is more stretched out (Figure 4), which resulted in lower heat transfer losses (Figure 6).

Conclusions

In this study, the performance analysis and in-cylinder visualization for reacting and non-reacting conditions were

investigated for CDC, IsoL, and IsoH combustion. The heat transfer and exhaust losses were estimated using the 1D GT-Power model. For each combustion mode, the flow-field details within the flames were also derived from the consecutive soot luminosity image pairs. The experiments were conducted in a single-cylinder, heavy-duty optical diesel engine using n-heptane fuel. The main findings of the present study are summarized as follows:

- Compared to CDC, IsoL and IsoH combustion show lower peak RoHR, with extended heat release till later crank angles. The optimized combination of combustion phasing and burn duration for IsoH leads to similar gross indicated efficiency as that of CDC, while IsoL shows the lowest efficiency. For both isobaric combustion cases, the heat transfer losses decrease due to the lower flame temperature while the exhaust losses increases due to slower burning and later combustion phasing than CDC.
- For CDC, the initial soot luminosity signals emerge close to the bowl-wall for certain jets while for the isobaric combustion cases, the flames corresponding to each jet are clearly distinguished during the earlier flame development. The flame-wall interaction regions for each combustion mode are followed by flame-flame interaction between the neighboring jets.
- Both the soot luminosity images and flow-field distribution within the high-temperature flames show lesser flame-flame interaction for CDC which would result in lower soot formation compared to isobaric combustion cases. However, the flame interaction regions for isobaric cases are uniformly distributed in the combustion chamber, thereby resulting in lower heat transfer losses than CDC.

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Acknowledgments

This paper is based on work supported by Saudi Aramco Research and Development Center FUELCOM program under Master Research Agreement Number 6600024505/01. FUELCOM (Fuel Combustion for Advanced Engines) is a collaborative research undertaking between Saudi Aramco and KAUST intended to address the fundamental aspects of hydrocarbon fuel combustion in engines, and develop fuel/engine design tools suitable for advanced combustion modes.

We would like to thank Prof. Sanghoon Kook and Dr Lingzhe Rao from the University of New South Wales (UNSW), Sydney, Australia for their assistance on FIV analysis.

Abbreviations

1D – One Dimensional
°CA – Crank Angle Degree
aSOI – after Start of Injection
aTDC – After Top dead Center

BTE – Brake Thermal Efficiency
CDC – Conventional Diesel Combustion
CO – Carbon Monoxide
eSOI – electronic Start of Injection
FIV – Flame Image Velocimetry
FuelMEP – Fuel Mean Effective Pressure
HCCI – Homogenous Charge Compression Ignition
IC – Internal Combustion
IsoL – Low-pressure Isobaric Combustion
IsoH – High-pressure Isobaric Combustion
LTC – Low Temperature Combustion
NO_x – Nitrogen Oxides
PCP – Peak Cylinder Pressure
PIV – Particle Image Velocimetry
PPC – Partially Premixed Combustion
PM – Particulate Matter
RCCI – Reactivity Controlled Compression Ignition
RoHR – Rate of Heat Release
ROI – Rate of Injection
TDC – Top Dead Center
UHC – Unburnt Hydrocarbon