

Mixing-Controlled, Low Temperature Diesel Combustion with Pressure Modulated Multiple-Injection for HSDI Diesel Engine

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Takaaki Kitamura and Takayuki Ito
Japan Automobile Research Institute

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ABSTRACT

This paper proposes a new mixing-controlled, low temperature combustion (LTC) approach for high-speed direct injection (HSDI) diesel engines. The purpose of this approach is to avoid the excessively high pressure-rise rate (PRR) of premixed, kinetics-controlled LTC and to enable the low nitrogen oxides (NO_x) combustion to operate over the wide speed and load range of the engine. To address the soot/noise trade-off at high load LTC operating conditions, the pressure modulated multiple-injection coupled with swirl control was applied. This injection strategy enables the injection of high pressure (HP) main spray into the local high temperature region of the already burning low pressure (LP) pilot spray injected from the neighboring injection hole. By employing this injection strategy, the equivalence ratio (ϕ) distribution of mixture is drastically varied during main combustion processes. The experimental results showed that by combining LP close-pilot injection, HP main injection near top dead center (TDC), and moderate swirl, the mixing-controlled LTC successfully achieve low engine-out NO_x and soot emissions even at 13.7 bar IMEP while retaining high efficiency and low PRR. Furthermore, the 3D-CFD analysis clarified that the rich mixture of broadband ϕ distribution is formed at the beginning of main combustion followed by rapid homogenization of the mixture occurring as combustion progresses.

INTRODUCTION

The increasingly stringent emissions legislation for diesel engines has accelerated research and development efforts toward innovative combustion systems and exhaust gas after-treatment technologies. One promising approach which offers the potential to simultaneously reduce engine-out NO_x and soot emissions is the premixed LTC strategies [e.g.

1,2,3,4,5,6]. In premixed LTC, the mixture formation process is improved to be either an EGR-diluted, near-homogeneous charge ($\phi < 2$) or a lean homogeneous charge ($\phi < 0.5$) condition prior to auto-ignition; thereby avoiding both NO and soot formation peninsula on the ϕ -T diagram [e.g. 7,8,9]. While low engine-out emissions have been demonstrated, the premixed LTC is very sensitive to many engine parameters (e.g. intake temperature, EGR rate, boost pressure, engine speed) due to chemical kinetics-controlled combustion and consequently, is difficult to control the heat release rates over a wide range of conditions. Therefore, thus far application of premixed LTC has been limited to low through moderate engine-load conditions.

It is desirable to develop a LTC strategy with high efficiency and low NO_x/soot emissions but without the combustion control problems and load limitations. Mixing-controlled LTC, where combustion starts during the injection event, is one such potential strategy. Split-PCI combustion [3] and MULDIC [10] are LTC strategies with multiple fuel injection that produce a combination of premixed LTC and traditional mixing-controlled diesel combustion during the same combustion cycle. With these approaches of early injection, however, the formation of a near-homogeneous mixture without spray-wall interaction is the biggest challenge for achieving high efficiencies and reducing HC emissions. To address the spray-wall impingement, Reitz et al. [11,12,13] proposed AIS (Adaptive Injection Strategies) with multiple injections at both low and high injection pressures. In this strategy, LP early injection is key to minimizing spray-wall impingement. On the other hand, Pickett et al. [14-15] proposed a non-sooting, mixing-controlled LTC concept that consists in promoting the air-fuel mixing upstream of the flame lift-off length of diffusion flame jet. Note that one of the approaches (some call lean-burn mixing-controlled combustion) demonstrates the possibility of low NO_x

combustion over a wide range of engine operating conditions without advanced air management systems such as multi-stage turbocharger or low pressure EGR. Furthermore, Upatnieks et al. [16] demonstrated near-zero engine-out NO_x and smoke emissions up to 12.5 bar IMEP without the combustion control problems of premixed LTC with the highly diluted, mixing controlled LTC approach using high cetane number, highly oxygenated fuels.

This study focuses on controlling the time evolution of the ϕ distribution from start of injection (SOI) to end of combustion (EOC) in order to improve the soot/noise trade-off at high-load LTC operating conditions. That is to say, the ϕ distribution of an EGR-diluted mixture is drastically varied from a “broadband rich mixture” to a “narrowband near-stoichiometric mixture” during ignition and combustion processes by an advanced injection strategy coupled with swirl control. One particular pressure modulated multi-injection strategy was explored where the HP main injection occurs in the local high temperature region of an already burning LP pilot spray. The purpose is to form the rich mixture of broadband ϕ distribution at start of combustion (SOC) and to accelerate the rapid homogenization of the mixture during the later portion of the combustion.

The objective of this study is to determine the fuel injection strategy for achieving the “rich - near-stoichiometric” ϕ distribution history, and to demonstrate the mixing-controlled LTC with improved soot/noise trade-off. Experiments were performed using a single-cylinder research engine equipped with a hydraulically intensified fuel injection system. The hydraulic intensified injector has the capability of delivering multiple injections at both low and high injection pressures through a single injector in the same engine cycle. These investigations showed that by combining LP close-pilot injection, HP main injection near TDC, and moderate swirl, the mixing-controlled LTC successfully achieve low engine-out NO_x and soot emissions even at 13.7 bar IMEP while retaining high efficiency and low PRR. Furthermore, the ϕ -T trajectory of injected fuel was analyzed by the 3D-CFD in order to clarify the mechanisms of the mixing-controlled LTC.

CONCEPT OF MIXING-CONTROLLED LTC

In this section, the concept of the mixing-controlled LTC is presented based on the ϕ -T diagram showing soot and NO_x formation regions. Fig. 1 shows the conceptual diagram of ϕ -T trajectory of the fuel injected into the cylinder for the mixing-controlled LTC compared to a representative premixed LTC. Also shown in Fig. 1 is a spatial spray distribution with multiple-injection strategy coupled with swirl control.

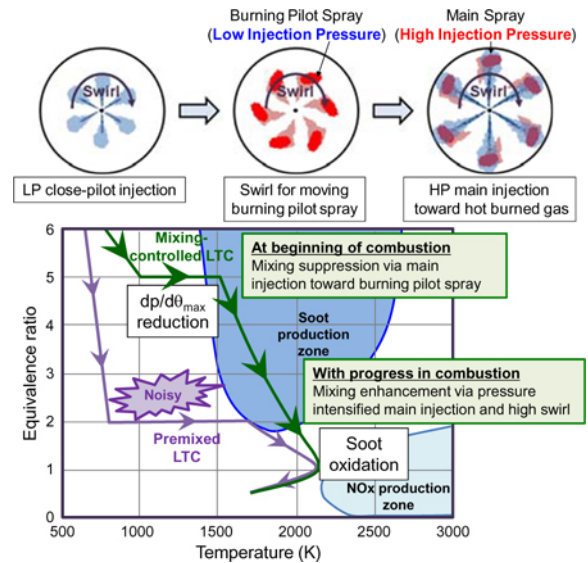


Figure 1. Conceptual diagram of ϕ -T trajectory of injected fuel for mixing-controlled LTC and premixed LTC.

It is well known that NO_x is formed in a near-stoichiometric ϕ and high T region, while soot is formed in a rich ϕ and specific T region [e.g. 7,8,9]. During typical mixing-controlled diesel combustion, a diffusion flame is formed at near-stoichiometric fuel-air mixtures. NO_x formation is high due to the very high diffusion flame temperature at typical diesel operating conditions (exceeding 2600 K) [e.g. 17,18,19]. Soot formation occurs inside the diffusion flame sheath in the fuel-rich regions of the jet [e.g. 20,21,22,23], and any of this soot that is not oxidized in the later phases of combustion becomes engine-out particulate matter emissions. In the case of premixed LTC strategies, fuel-air mixing is enhanced during the ignition delay enough to avoid the soot and NO formation region on the ϕ -T diagram. However, as described in the above section, it is difficult to control the heat release rates due to the chemical kinetics-controlled combustion, leading to combustion control problems and load limitations (intense combustion noise).

The key of the mixing-controlled LTC approach is the unique ϕ distribution control of an EGR-diluted mixture, where NO_x emissions are reduced with massive EGR which lowers the intake oxygen content below 14% and maintains the peak combustion temperature below 2100 K. Regarding the ϕ control which aims at drastic variation from a “broadband rich mixture” to a “narrowband near-stoichiometric mixture” during ignition and combustion processes, the pressure modulated multiple-injection is utilized along with swirl control. As shown in the upper diagram of Fig. 1, the local high temperature regions are created by LP close-pilot injection and then HP main injection is conducted near TDC to overlap with the burned gas cloud of the pilot spray which is travelling in the combustion bowl circumferentially by

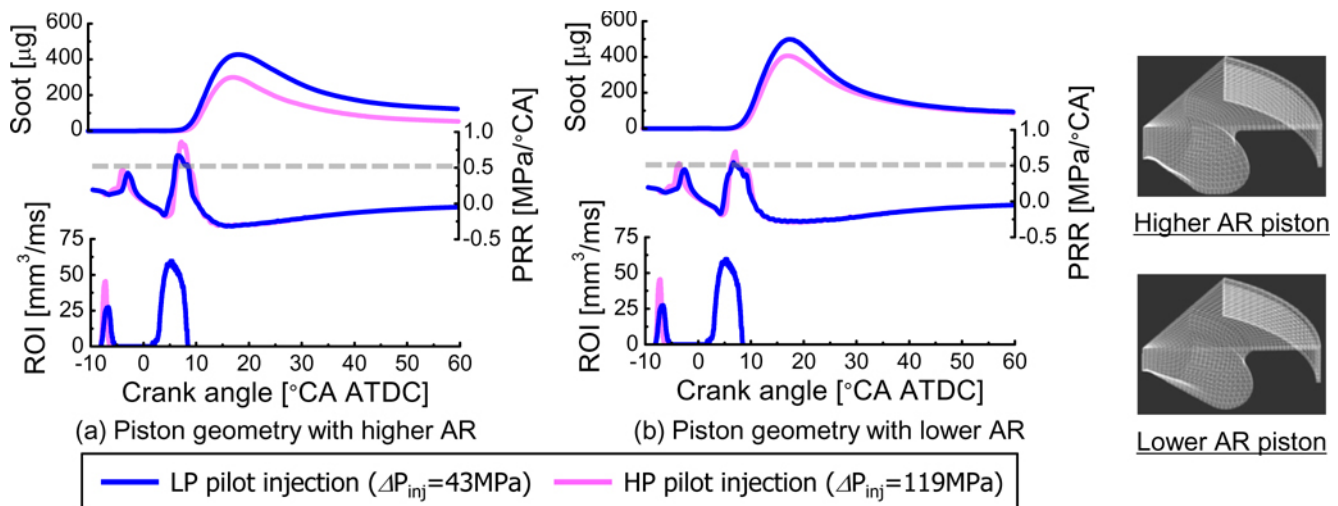


Figure 2. Predicted effect of pilot injection pressure on PRR and soot formation in the mixing-controlled LTC. ($N_e=1500\text{rpm}$, $Q_f=33\text{mm}^3/\text{st}$, $Q_{PI}=4.5\text{mm}^3/\text{event}$, $S/R=2.5$, $\text{Boost}=171\text{kPa}$, $\text{Intake } O_2=13.5\%$)

swirl flow. The main spray entrains the burnt gas shortly after SOI and, consequently, starts to combust with a broadband rich mixture before the injected fuel mixes enough with the in-cylinder oxygen to avoid the fuel-rich regions that form soot particles. After ignition, a diffusion flame is formed which is the same as typical, controllable, mixing-controlled diesel combustion except for the low flame temperature. It is speculated that the injected fuel passes into the soot formation peninsula on the ϕ -T diagram (see lower diagram of Fig. 1) and that abundant soot is formed in the fuel-rich region of the jet flames. Shimo et al. [24] reported that the region where hydroxyl (OH) radical, a very efficient soot oxidizer, forms without NO formation is at lower temperatures of NO formation peninsula on the ϕ -T diagram. Therefore, it is very likely that, even at low temperatures where NOx formation rates are slow, most of the soot formed in the fuel-rich regions could be oxidized if rapid homogenization of the mixture is possible during the later portion of the combustion by HP main injection, high swirl, and so on.

Fig. 2 presents the results of preliminary simulations with KIVA/CHEMKIN code coupled with a detailed soot kinetic model [25] to confirm what is the benefit of using LP injection for pilot event in the mixing-controlled LTC. The diagram shows the calculated soot, the calculated PRR, and the tested rate of fuel injection (ROI). The ROI trace consists of close-pilot injection and main injection near TDC, where both low and high injection pressures for pilot event ($\Delta P_{inj} = 43 \text{ MPa}$ vs. 119 MPa) were examined using two piston geometries with different aspect ratio (AR). It is found that LP pilot injection would be beneficial in lowering PRR during main combustion regardless of piston geometry. This is because LP pilot injection minimizes spray-bowl impingement and the resulting hot burned gas is not widely

dispersed. The negative effect of LP pilot is to increase the soot production. However, the piston geometry with lower AR shows a possibility to enhance the soot oxidation process even in the LP pilot case. On the basis of these results, all experiments reported in this paper were made with a lower AR piston.

<figure 2 here>

EXPERIMENTAL SETUP

SINGLE-CYLINDER RESEARCH ENGINE

The exhaust gases and the combustion characteristics for the mixing-controlled LTC were examined using a supercharged, single-cylinder research engine based on the Honda i-CTDi diesel engine. The specifications are presented in Table 1. The bowl shape with a lower AR is of a re-entrant type as shown in Fig. 3, and the compression ratio is set to 16.7:1. This research engine is equipped with a hydraulically intensified common rail fuel injection system (Sturman Industries, S1.3 fuel injector) together with a hydraulic valve actuation system (Sturman Industries, HVA Research Module). Furthermore, a swirl control valve (SCV) placed upstream of one of the intake ports was used to control the swirl ratio.

Table 1. Specifications of test engine.

Engine Type	4-stroke Single-Cylinder HSDI Diesel Engine
Bore x Stroke	ϕ 85 mm x 97.1 mm
Displacement	551 cm ³
Compression Ratio	16.7
Combustion Chamber	Re-entrant Type
Valve Train	Hydraulic Valve Actuation System 2-Inlet, 2-Exhaust
Swirl Ratio	Continuously Variable
Fuel Injection System	Hydraulic Intensified CR System
Intensification Ratio	2.25, 2.94
Nozzle Type	ϕ 0.139 mm x 6 holes, $\beta=153^\circ$

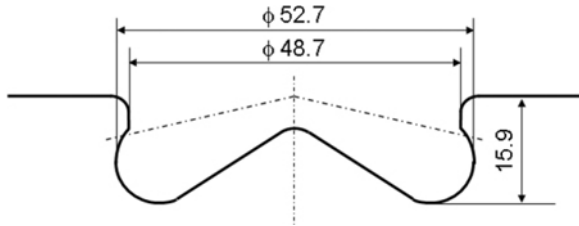


Figure 3. Schematic of combustion bowl with lower AR.

The hydraulic intensified injector uses a portion of the fuel as a working fluid to elevate the injection pressure prior to the specific injection event, and supports an injection pressure up to 250 MPa (a 300 MPa version is in development). Two different control valves are installed in the injector body. One is a needle control valve (NCV), which controls the amount of fuel injected and the injection timing upon command from the laboratory computer. The other is an intensifier control valve (ICV), which controls the operation of intensifier piston/plunger assembly to provide high injection pressures. Two intensification ratios of 2.25:1 and 2.94:1 were used for the experiments presented here. The injection strategy combined with LP pilot injection and HP main injection was realized as follows: pilot injection uses common rail (non-intensified) mode to allow for better small quantity control, and the intensifier is then activated selectively for the main event to allow for very high pressure. The hydraulic intensified injector also enables injection rate shaping by opening the NCV before the ICV (see Fig. 4). All measurements reported in this paper were made with a typical 6-hole (equally spaced at 60°) nozzle. The hole outlet diameter was 0.139 mm with a k-factor of 1.5 and the nominal angle of the fuel-jet axis was 13.5° downward from horizontal.

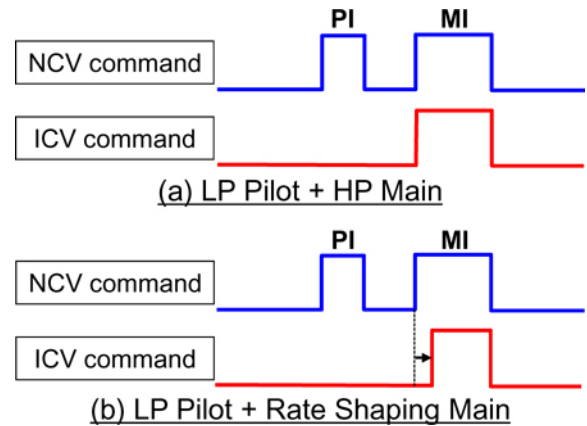


Figure 4. Injection command pulses for LP (non-intensified) pilot injection followed by HP (intensified) main injection.

The HVA system provides full flexibility of the valves, allowing individual control of valve lift, opening and closing timing and, thereby, duration. The illustration of engine head equipped with a hydraulic intensified injector and HVA units is shown in Fig. 5. The major parts of the HVA system are the engine valve modules, a free-standing hydraulic supply unit (HSU) to supply the high pressure hydraulic oil to the actuators, the oil rails, the valve control module (VCM) responsible for processing the valve position signals for control, the valve drive module (VDM) to control the actuation of each valve, and a laboratory computer to communicate with the VDM. A two-position, three-way (2P3W) digital valve is used to control the compound actuator that opens the engine valve. The engine valve is hydraulically returned by way of return pistons. The housing of the return pistons is also the mechanical stop that limits the maximum lift of the engine valve. Under normal valve operation, the engine valves move between the seated position and the mechanical stop which controls the maximum lift of the valve. In this system, the position of the mechanical stop can be varied to change the maximum lift during engine operation. For all experiments presented here, however, the valve lift trace is kept constant in order to focus on the ϕ history control using an advanced fuel injection strategy. From the preliminary study of valve timing effects on charging efficiency and indicated work, the valve profile shown in Fig. 6 was selected.

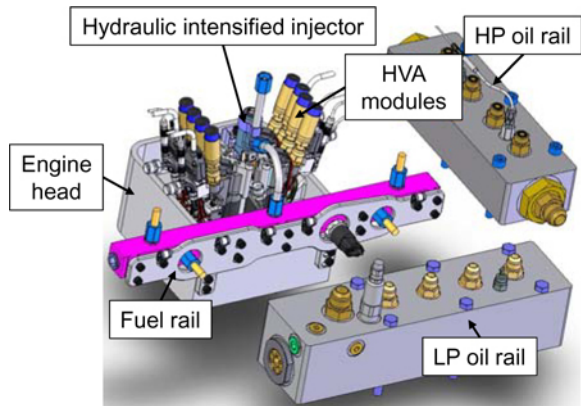


Figure 5. Appearance of the engine head equipped with a hydraulic intensified injector and an HVA system.

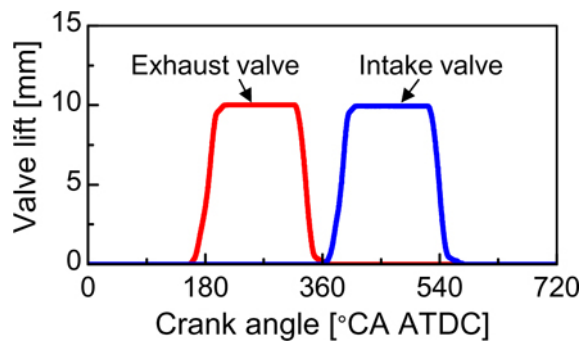


Figure 6. Valve lift trace for all experiments.

OPERATING CONDITIONS AND FUEL

Table 2 shows the experimental conditions for the demonstration of the mixing-controlled LTC concept described in the previous section. All data presented in this paper were taken at an engine speed of 1500 rpm using a commercially available JIS#2 diesel fuel (LHV = 45.9 MJ/kg, density = 829 kg/m³ at 288 K) with a sulfur content less than 10 ppm. Two different engine-load conditions were selected. The total fuel quantity of 33 mm³/st is representative for moderate load condition (9.2 bar IMEP), in which the effects of engine parameters (e.g. pilot injection quantity, dwell time between pilot and main injection, swirl ratio, injection pressure) on the engine emissions and performances were investigated. The next step was to apply the mixing controlled LTC to a high-load condition where the total amount of fuel injected was 47 mm³/st (13.7 bar IMEP). In order to maintain the peak combustion temperature below 2100 K, EGR rates were adjusted to keep the intake oxygen concentration at 13.5 %, the same level of EGR dilution as EPA's Clean Diesel Combustion technology [26] and Ricardo's ACTION strategy [27], while the intake gas temperature was held at 333 K. Furthermore, the intake gas was pressurized to 171 kPa and 244 kPa at each engine-load condition respectively to keep the air-to-fuel (A/F) ratio

consistent. Note that the A/F ratio changes with the port swirl ratio slightly in the range from 18.6 to 19.0. Before conducting the experiments, the engine was heated to 353 K by means of electrical heaters on the cooling water and lubricating oil circulation systems.

Table 2. Operating conditions and fuel.

Test Fuel	JIS #2 Diesel Fuel
Engine Speed	1500 rpm
Fuel Quantity	33, 47 mm ³ /st
Boost Pressure	171, 244 kPa (A/F=18.6 ~ 19)
Intake O ₂	13.5 vol%
Intake Temperature	333 K

DATA ACQUISITION

For engine testing, Meidensha's 110kW electric motor/generator dynamometer was used to control the engine speed and to measure the engine torque. The intake and exhaust gas components, such as carbon dioxide (CO₂), carbon monoxide (CO), NO_x, unburned hydrocarbon (HC) and oxygen (O₂), were measured using an exhaust gas analyzer (Horiba Instruments, MEXA-9100D), which consists of non-dispersive infrared detector (NDIR), chemiluminescence detector (CLD), flame ionization detector (FID) and magneto pneumatic detection (MPD). Soot emissions were measured using an AVL 483 micro soot sensor [28]. The soot sensor is based on a near-infrared photoacoustic principle and is sensitive for soot without a reasonable interference to other components. The amount of fuel injected into the cylinder was measured using an ONO SOKKI FP213 volumetric-type fuel flow detector in conjunction with a DF-2410 digital flowmeter.

The time-series data (e.g. in-cylinder pressure, intake and exhaust pressure, injection command pulse and engine valve lift) were digitized and recorded at 0.1 °CA increments and ensemble-averaged over 200 engine cycles using an ONO SOKKI DS-2000 series multi-channel data station. For the in-cylinder pressure data acquisition, a Kistler Model 6055 piezoelectric transducer was used adopting a glow plug adapter mounting. Apparent heat release rates (ROHR) were calculated from the ensemble-averaged in-cylinder pressure trace using the typical first law. Prior to calculating the ROHR, the experimental pressure data were filtered in the frequency domain using a low-pass filter with a Gaussian roll-off function. These cut-off frequencies were selected to remove acoustic ringing in the cylinder pressure data, while retaining the general features of the ROHR. The intake and exhaust pressure were measured using a Kistler Model 4055 piezoresistive absolute pressure sensor for the 1D/3D gas exchange simulation. In addition, ROI was measured by the Bosch long tube method based on the principle of pressure

wave propagation in a column of liquid. For the ROI measurement, a Kistler Model 211B pressure sensor was used. Injection velocity for each injection event was directly estimated from the ensemble-averaged ROI trace and hole outlet diameter without taking into account the loss of flow area in an orifice due to cavitation and hydraulic flip [e.g. 29, 30]. For all data presented in this paper, 0 °CA is defined as compression TDC so that exhaust TDC is at 360 °CA.

3D-CFD MODEL

CFD CODE

3D-CFD was performed using the CONVERGE code developed by Convergent Science, referred to as MoSES code previously, to predict the in-cylinder gas mixture conditions such as local ϕ and T . The features such as a user-specified, fixed grid embedding technique and/or a self-adaptive mesh refinement (AMR) technique allow for efficient simulations while maintaining the grid resolution necessary for accurate engine modeling. The basic features of the CONVERGE code are well documented [31] and will not be repeated here for brevity. The sub-models used in this study are summarized in Table 3. The SAGE parallel detailed chemistry solver [32] is used for the present study. A skeletal reaction mechanism for n-heptane [33] is used to model diesel fuel. This mechanism includes 34 species and 74 reversible reactions representing hydrocarbon combustion and NO formation [34]. Note that n-heptane is often used as a surrogate for diesel fuel due to its similar cetane number. In addition, the low temperature chemistry and two-stage ignition phenomena have been shown to be captured by this mechanism [33]. While this reduced mechanism simulated the combustion reactions of the diesel fuel, the physical properties of the fuel for use in the spray model were taken as those of tetradecane ($C_{14}H_{30}$). Soot emissions are predicted by a phenomenological soot model that uses competing formation and oxidation rates. The formation is described by the Hiroyasu soot formation model while the oxidation is predicted by the Nagle-Strickland-Constable oxidation model [35]. In the model, acetylene (C_2H_2) is used as the soot formation species instead of “fuel”, since it is an important precursor species.

Table 3. Sub-models used in 3D-CFD calculations with CONVERGE code.

Spray breakup	KH-RT (Kelvin-Helmholtz/Rayleigh-Taylor)
Drop drag	Dynamic drag model
Collision and Coalescence	NTC collision
Wall impingement	Rebound/slide model
Turbulence	Rapid distortion RNG k-epsilon
Heat transfer	Law-of-the-wall
Ignition and combustion	SAGE detailed chemistry solver n-heptane skeletal mechanism (30 species, 65 reactions)
NOx formation	Reduced NOx mechanism (additional 4 species, 9 reactions)
Soot formation	Hiroyasu soot formation model Nagle-Strickland-Constable oxidation model

MODELING METHODOLOGY

In the present engine simulations, intake calculations from intake valve opening (IVO) to intake valve closing (IVC) were first performed using a full engine model. These cases were simulated using a relatively coarse base grid of 2.8 mm. Much finer resolution is needed locally around the valves, however, for accurate prediction of intake gas flow rates. Both fixed grid embedding and AMR were used to adequately resolve the flow around the valves, resulting in cells of 0.35 mm in these regions (see left side of Fig. 7). After intake calculations, spray/combustion calculations from IVC to exhaust valve opening (EVO) were then performed using a sector model with the in-cylinder gas conditions (e.g. swirl ratio, temperature and pressure) obtained from the intake calculations. In the sector model, 60° symmetry is assumed due to the 6-hole nozzle. An example of grid resolution is shown in the right side of Fig. 7, where the spray is injected from left to right. As shown in the figure, high resolution is achieved around the spray. The total number of cells during a sequence of calculations in addition to a comparison of the measured and predicted cylinder averaged pressure traces is given in Fig. 8. To keep the runtime reasonable, a maximum overall number of cells was maintained below 100,000 in the sector case. This condition still provided good correlation between the predictions and measured data.