

Gasoline DICI Engine Operation in the LTC Regime Using Triple- Pulse Injection

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ABSTRACT

An investigation of high speed direct injection (DI) compression ignition (CI) engine combustion fueled with gasoline injected using a triple-pulse strategy in the low temperature combustion (LTC) regime is presented. This work aims to extend the operation ranges for a light-duty diesel engine, operating on gasoline, that have been identified in previous work via extended controllability of the injection process. The single cylinder engine (SCE) was operated at full load (16 bar IMEP, 2500 rev/min) and computational simulations of the in-cylinder processes were performed using a multi-dimensional CFD code, KIVA-ERC-Chemkin, that features improved sub-models and the Chemkin library. The oxidation chemistry of the fuel was calculated using a reduced mechanism for primary reference fuel combustion chosen to match ignition characteristics of the gasoline fuel used for the SCE experiments.

With constraints on a minimum allowable combustion efficiency, maximum allowable noise level (pressure rise rate) and maximum allowable NO_x and soot emissions, engine operation ranges were identified as functions of injection timings and the fuel split ratio (i.e., fraction of total fuel injected in each pulse) with triple-pulse injections. Parametric variation of the engine operating ranges were also investigated with respect to initial (i.e., intake) gas temperature, exhaust gas recirculation ratio, intake boost pressure and injection system rail pressure. Following the modeling, engine experiments were performed under conditions identified through analysis of the numerical results in order to confirm the effectiveness of gasoline direct injection compression ignition (GDICI or GCI) operation with triple-pulse injections at full load.

Based on both computational and experimental results, the role of each pulse in GDICI operation was identified in terms of combustion stability, engine performance and emissions. While maintaining similar emissions characteristics to that of the double-pulse injection cases (~ 0.1 g/kg-f of NO_x and PM, and ~ 173 g/kW-hr of ISFC), the extension of operable conditions using a triple-pulse injection strategy was successfully achieved.

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INTRODUCTION

As part of the effort to reduce fuel consumption and overcome the challenge that current pollutant emission regulations present to conventional diesel engines because of their higher oxides of nitrogen (NO_x) and particulate matter (PM) emissions, various advanced combustion techniques have been developed to control fuel/air mixing and combustion temperature of charge mixtures. Such strategies as early PCCI [1, 2, 3], late PCCI [4, 5, 6, 7] or combination of early pilot and retarded additional injections [8,9] have been reported and each of them has its own disadvantages as well as advantages [10]. In addition to the effort to develop

combustion strategies, a significant effort has been made to research on the characteristics of fuels desirable for the advanced combustion concepts. Due to its high volatility, low cetane index and availability, gasoline emerged as an attractive alternative fuel candidate for LTC engines [8, 11, 12, 13, 14, 15, 16].

Among few reported computational studies of GDICI, Ra, et al. [17] recently conducted a numerical study on GDICI combustion in a conventional heavy-duty diesel engine to show feasibility of GDICI and described the mechanism of emissions reduction in GDICI combustion. Ra, et al. [18] further investigated the effects of parametric variation of engine operating conditions on performance of GDICI

combustion in the same heavy-duty diesel engine as used in [17]. They found that injection timing influences both UHC and CO emissions as well as noise level, and split ratio variation is an effective way to reduce CO emissions and maximum pressure rise rates (PRRmax) at only a slight expense of increased ISFC. EGR ratio variation was also found to be effective to control PRRmax, although the EGR operating window was so narrow that mixtures are likely to misfire at levels of EGR that are within the control variability in conventional diesel-fueled engine operation.

In the authors' previous work [10], the operating conditions of a light-duty engine to meet strict constraints of combustion efficiency and noise level as well as the emissions mandates were successfully identified and operating maps of GDICI operation were generated in the split ratio vs. injection timing domain. The predicted operation maps were supported by experiments that showed reductions of PM and NO_x emissions to levels of about 0.1 g/kg-f while achieving an ISFC of about 180 g/kw-hr. A sensitivity study of the operation map to parametric variation revealed high sensitivity of GDICI operation to changes in EGR ratio, initial gas temperature and boost pressure. In addition, the existence of an optimal injection pressure to maximize the extension of the operation map for a given engine load condition was found both numerically and experimentally.

Although operable ranges in the GDICI engine studied were identified, excessive pressure rise rates still remained as the toughest constraint to meet, which tends to narrow the operable range. This limitation led to a further investigation of possible strategies to attenuate and control PRR, and thus to extend the GDICI operation range using the current advanced injection system technologies.

Triple pulse injection strategy has been employed recently in several experimental studies on diesel engine combustion with gasoline-like fuels mainly to mitigate the combustion rate that would be problematic at mid to high engine load. Manente et al. [19] employed triple pulse injections to control excessive fuel stratification that causes unacceptable pressure rise rates in a heavy-duty diesel engine. They operated the engine at various loads using fixed start-of-injection timings for the three pulses. Kalghatgi et al. [20] tested the effectiveness of triple-injection on controlling heat release rates as well as improving engine performance in a small-bore diesel engine fueled with RON84 gasoline. Sellnau et al. [21] applied a triple-injection method to improve fuel economy in a light-duty diesel engine operated with RON91 gasoline at 6 bar IMEP. They also showed that a triple-injection strategy allowed the use of lower injection pressures than corresponding single and double-injection strategies required. Studies on triple-pulse injection strategies were mostly experimental and numerical simulations of GDICI employing a triple injection strategy have rarely been reported.

In the present study, an investigation of high speed GDICI engine combustion using a triple-pulse injection strategy in the LTC regime is presented. This work aims to extend the previously reported operating ranges for a light-duty diesel engine, running on gasoline, via extended controllability of the injection process. In the current work, the same single cylinder engine (SCE) [10] was operated at full load (16 bar IMEP, 2500 rev/min) and computational simulations were performed using a multi-dimensional CFD code, KIVA-ERC-Chemkin, that features improved sub-models and the Chemkin library. The oxidation chemistry of the fuel was calculated using a reduced mechanism for primary reference fuel combustion chosen to match ignition characteristics of the gasoline fuel used for the SCE experiments.

With constraints of a minimum allowable combustion efficiency, either PRRmax or maximum cylinder pressure (Pmax), and maximum NO_x and soot emissions, engine operation ranges were identified as functions of injection timings and the fuel split ratio (i.e., fraction of total fuel injected in each pulse) with triple-pulse injections. The role of each pulse in triple injections was revealed through the investigation. Parametric variation of the engine operation ranges was also investigated with respect to initial (i.e., intake) gas temperature, exhaust gas recirculation ratio, intake boost pressure and injection-system rail pressure. Following the modeling, engine experiments were performed under conditions identified based on the numerical results in order to confirm the effectiveness of GDICI operation with triple-pulse injections at full load.

NUMERICAL APPROACH

PHYSICAL MODELS

For simulating the spray process and the mixing and combustion of fuel/air mixtures in the combustion chamber, various physical sub-models were employed in the present KIVA-ERC-CHEMKIN code. The code is based on KIVA3V Release 2 [22] coupled with the CHEMKIN II library [23]. The added sub-models include models related to drop breakup, collision and coalescence, drop deformation, drop evaporation, wall impingement and vaporization, etc. [17, 18].

A hybrid primary spray break-up model that is computationally efficient as well as comprehensive enough to account for the effects of aerodynamics, liquid properties and nozzle flows was employed [24]. In this model, the injected fuel "blobs" are tracked by a Lagrangian method while the break-up of each blob is calculated from considerations of jet stability from Kelvin-Helmholtz (KH) instability theory. For the secondary and further break-up processes, a Kelvin Helmholtz (KH) - Rayleigh Taylor (RT) hybrid model was used. In the present study, the model constants were used as suggested by Beale and Reitz [24] since, due to high volatility of gasoline, the variation of fuel distribution in the cylinder is not as sensitive to spray model constants as in diesel spray cases.



Documents

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A droplet collision model based on the stochastic particle method [22] was used, in which the collision frequency is used to calculate the probability that a drop in one parcel will undergo a collision with a drop in another parcel, assuming all drops in each parcel behave in the same manner. The probability of coalescence is determined considering the Weber number that includes the effects of density and surface tension of the liquid droplets.

Droplet deformation in terms of its distortion from sphericity is modeled using a forced, damped harmonic oscillator model, where the surface tension and viscosity of the droplet are the major properties used in the restoring force and damping terms, respectively [25]. Distortions of the droplets affect the momentum exchange between the droplets and the ambient gas, and subsequently the drop velocities (or relative velocity between the drop and the gas) that are the governing parameters in the breakup and evaporation processes.

The droplet vaporization model [26] considers the evaporation of spray droplets using the Discrete Multi-Component (DMC) approach under temperatures ranging from flash-boiling conditions to normal evaporation. The improved model accounts for variable internal droplet temperatures, and considers an unsteady internal heat flux with internal circulation, and a model for the determination of the droplet surface temperature. The model uses an effective heat transfer coefficient model for the heat flux from the surrounding gas to the droplet surface. Also, the variable density of the gasoline-surrogate fuel as a function of temperature is considered in the governing equations and the relevant sub-models. The effective heat transfer coefficient calculated in the model is also used to determine the amount of fuel to be treated as vapor when the drop surface temperature reaches the critical temperature while the drop interior is still in the sub-critical condition. The model has been well tested for evaporation of sprays as well as single drops at various pressure and temperature conditions including flash-boiling.

Effects associated with spray/wall interactions, including droplet splash, film spreading due to impingement forces, and motion due to film inertia were considered in a wall film sub-model, in addition to calculations of film transport on complex surfaces with heating and vaporization of the film, and separation and re-entrainment of films at sharp corners [27]. For the turbulence calculation, the RNG k - ϵ model [28] was used.

In the two-phase transport equations, droplets are treated as point sources and the wall film fuel flow is not resolved on the computational grid. Therefore, it is assumed that in a computational cell where droplets or wall film parcels exist, the liquid vaporizes under the prevailing mixture conditions and the vapor mixes completely with the gaseous mixture within the cell. Thus, stratification of gaseous species within a single cell is not resolved.

The physical models employed in the present study have been extensively validated for diesel spray injections.

Although the models were not extensively validated with gasoline sprays in the present study. The performance of the models in gasoline application was tested by the authors in the previous study [10].

COMBUSTION MODELS

The ignition/combustion characteristics of automotive fuels are often represented using blends of two hydrocarbons (e.g., primary reference fuel), i.e., iso-octane (iC_8H_{18}) and n-heptane (C_7H_{16}). It is widely accepted that the oxidation processes of n-heptane and iso-octane well represent the ignition and combustion characteristics of diesel and gasoline fuels, respectively. In the present study, a skeletal reaction mechanism for primary reference fuel oxidation with 41 species and 130 reactions [29] was used to calculate the detailed chemical kinetics of combustion. The mechanism has been well validated using data from HCCI [30] and direct injection engine experiments [14, 31], as well as with the ignition delay time data obtained in shock tube tests [32] for various temperatures, pressures and fuel compositions. In the present study, gasoline was modeled as PRF 87, i.e., 87 % iso-octane and 13 % n-heptane for physical properties and chemistry calculation.

For the calculation of NO_x formation, a 4 species (N, NO, N_2O and NO_2) and 12 reaction NO_x mechanism was used that has been reduced from the GRI NO_x mechanism (available online) and added to the PRF reaction mechanism. A phenomenological soot model, modified from the Hiroyasu soot model [33] was employed to predict soot emissions. For oxidation of soot, the Nagle-Strickland-Constable (NSC) model was employed in the soot model. Assuming a well-stirred reactor in each cell, changes of species concentration were obtained from the chemical reaction calculations, which are directly integrated with the transport calculations in the CFD code.

COMPUTATIONAL CONDITIONS

A small-bore light-duty diesel engine with a compression ratio of 16.5 [34] was used for the simulations. Triple injections through a 7-hole injector with an included spray angle of 155° were modeled. In the computations, the first pulse injection is assumed to be made such that the injected fuel vaporizes completely and mixes with air uniformly before IVC timing. This assumption corresponds to operation of the SCE engine where the first pulse was injected during valve overlap leaving the remainder of the intake stroke to vaporize and mix before intake valve closure. Injection timings of the second and third pulses were varied between -80 and TDC and -30 and $+4^\circ$ ATDC, respectively. The same initial and boundary conditions for a baseline engine operation as used in the previous work [10] were used, which were well validated with the measured GDICI engine operations.

The simulated cylinder gas pressure and temperature at intake valve closure (IVC) were set to 3.25 bar and 361 K,

respectively, regardless of the timing of the second and third injections. The gas temperature at IVC was intentionally increased from that of the baseline case employed in the previous work in order to choose an operating condition beyond the operation map, since the objective of the present study is to extend the operation range obtained using a double-pulse injection strategy. Note that, instead of intake port conditions, in-cylinder mixture conditions at IVC are used in this paper. It is also notable that, due to heat transfer from the cylinder walls, mixing of fresh air with hot internal residual gases, and a slight compression of the gas mixture during the period between BDC and IVC, the gas temperatures at IVC are normally higher than the intake port gas temperatures. Detailed specifications of the engine, injector and computed operating conditions are listed in Table 1.

Table 1. Engine and injector specification and simulated operating conditions. Numbers in parenthesis are the parametric variation.

Engine	
Type	light-duty, reentrant bowl piston
Bore x Stroke [mm]	82 x 90.4
Connecting rod length [mm]	161
Compression ratio	16.5
Valve timing [° ATDC]	IVO=344, IVC= -132, EVO=112, EVC=388
Injector	
Number of holes	7
Included angle [degree]	155
Nozzle hole diameter [μm]	141.4
Injection timings [° ATDC]	1 st pulse: -350 2 nd pulse: -80 ~ tdc 3 rd pulse: -34 ~ +6
Injection system rail pressure [bar]	1160 (860, 1500)
Injection amount [mg/cycle]	43.6
Operating conditions	
Engine speed [rev/min]	2500
Target net IMEP [bar]	16.0
Swirl ratio	2.2
Temperature at IVC [K]	361 (371)
Pressure at IVC [bar]	3.25 (3.5)
O ₂ fraction @ IVC [%]	10.9 (11.9)
fuel	Gasoline (ON=87)
Injection strategy	triple
Pilot pulse split ratio [%]	40 ~ 100
Crank angle range of computation [° ATDC]	IVC ~ EVO

The oxygen fraction at IVC was set to 10.9 %, which is equivalent to about 47% EGR ratio using the simulated exhaust gas composition for a stoichiometric mixture. Note again that the O₂ fraction at IVC was intentionally increased from that of the previously reported double-injection study [10] to choose an operating point out of the operation map for the double-injection operations.

The baseline injection pressure was chosen to be 1160 bar at which detailed injection rate measurement data is available since the optimal injection pressure was found to be around 1200 bar in the authors' previous work [10].

For triple injection cases, the proportion of fuel injected during the first pulse (hereafter, denoted as 1st split ratio, S1), which was assumed to be completely vaporized and uniformly mixed with air before IVC, was varied between 40 and 100 %.

In order to investigate the effects of parametric variation, the initial gas temperature, boost level, EGR ratio and injection pressure were varied from the baseline values. When one parameter was changed the remaining parameters were kept the same as the baseline values. Detailed computational conditions for the parametric variations studied are also listed in Table 1.

In Figure 1, the injection rate shape profile of a triple pulse injection with 60/10/30 % split ratios is shown for the baseline injection pressure of 1160 bar at engine speed of 2500 rev/min. The start of injection commands (SOIC) of the three pulses are -340°, -24° and -10°, respectively. In the figure, the rates are normalized by the maximum value during the injection. When the injection amount for each pulse was changed as the split ratio varied, the duration of each pulse was changed accordingly based on the measured injection rate profiles. The pressure wave interactions between the 1st and 2nd pulses can be neglected since the pulse dwell is long enough (~20 ms) for such wave interactions to damp out. However, possible interactions between the 2nd and 3rd injection pulses was not considered in the present numerical simulations when the dwell between the 2nd and 3rd pulses was short.

Three dimensional computational grids with the crevice volume resolved as an elongated top land region were employed. A 7-hole injector with an included angle between sprays of 155 degrees was modeled. To save computation load, a 1/7th sector of the full 360° mesh with periodic boundaries (corresponding to one plume from the seven-hole injector nozzle) was used. The average cell dimensions were 1.2 to 1.8 mm and 0.6 to 4.1 mm in the radial and vertical directions, respectively, with twenty cells azimuthally (see Figure 2). To resolve the crevice region, i.e., the gap between the piston and cylinder wall above the top ring, two radial cells were used with three vertical cells. This grid resolution was found to be fine enough to ensure grid insensitivity of the spray sub-models and the combustion model.

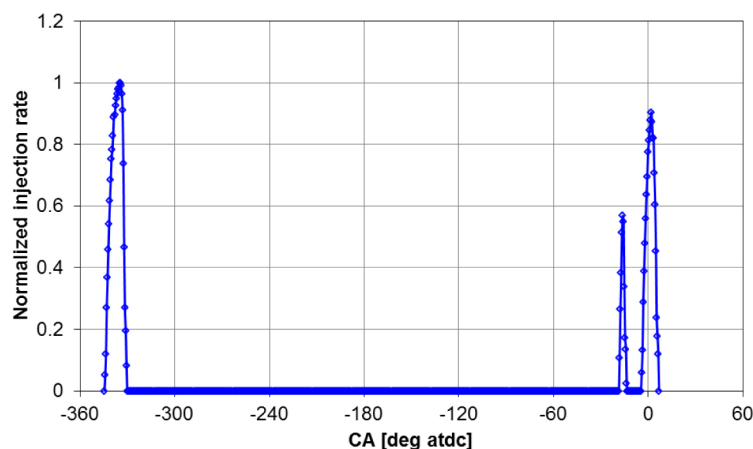


Figure 1. Injection rate shape of a triple pulse injection for split ratios of 60/10/30 %. Injection pressure was 1160 bar and engine speed was 2500 rev/min. Rates are normalized by the maximum value during the injection.

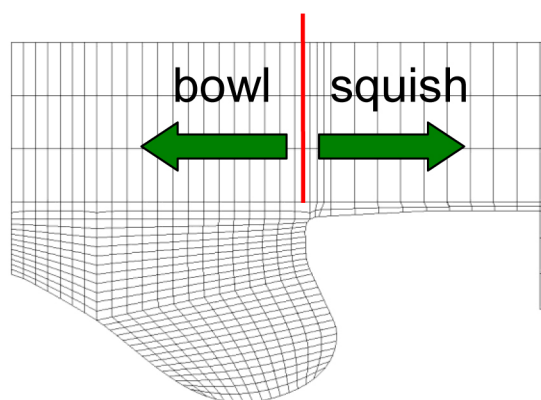


Figure 2. Vertical cross-section view of the computational grid with crevice volume resolved as a thin annulus. Azimuthal angle of the sector span was 51.4 degrees.

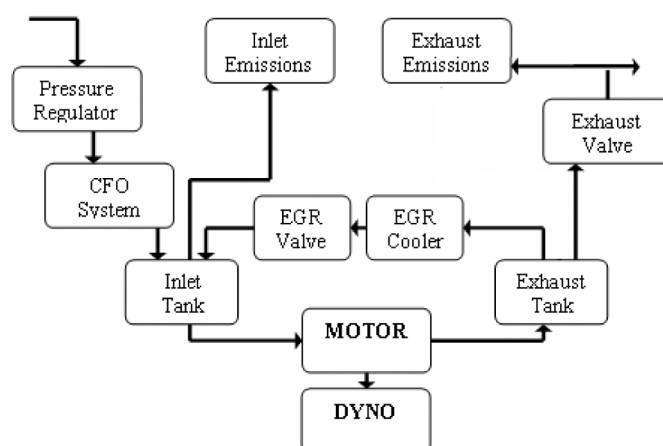


Figure 3. Schematic of test cell layout. CFO stands for Critical Flow Orifice. Exhaust Valve stands for exhaust back-pressure valve.

EXPERIMENTAL SETUP

The engine used in this study is a single-cylinder version of a GM 1.9L, 4 cylinder light-duty diesel engine; the specifications of which can be seen in [Table 1](#). The production cylinder head and valve-train were used in conjunction with a Labeco CLR crankcase. A re-entrant bowl design with lower compression ratio than the current Euro-4 compliant production configuration was used along with a slightly longer connecting rod length; all other cylinder geometry was matched to the equivalent 4-cylinder engine. The single-cylinder engine has the capability to run up to 4000 rev/min at full-power conditions using pressurized inlet air and an exhaust backpressure valve to simulate the behavior of a turbocharger. Full capability for variation of EGR rate, coolant and oil temperature and respective inlet and exhaust pressures and temperatures exists in the laboratory. [Figure 3](#) shows the schematic of the engine and test cell layout.

The injection system utilizes a BOSCH CP3.3 pump for supplying fuel rail pressure. ETAS INCA calibration software used in conjunction with the production ECU allows for up to 5 injections per cycle. A production CRIP2-MI injector was used with an experimental $7 \times 155^\circ$ mini-sac nozzle tip with a flow number of 440 cc/30sec. [Table 2](#) summarizes the injection system specifications.

The measurement of cylinder pressure was accomplished via a piezoelectric KISTLER 6125B pressure transducer. The resolution of data was 0.1 CAD, corresponding to 3600 counts per revolution; 200 cycles of data were recorded for all heat release analysis. The combustion noise was recorded using an AVL Combustion Noise Meter (Model A01); meter output was in decibels. Injector current was measured using a Tektronix A622 AC/DC current probe. Exhaust emissions were measured using a Horiba emissions bench capable of recording CO₂, CO, (Model AIA 220), O₂ (Model FMA 220), THC (Model FIA 236 1, propane span gas) and NO_x.

(Model CLA 220). In addition, a reflectivity-based PM measurement was taken using an AVL Smoke Meter (Model 415S) at each operational point. The emissions sample was extracted downstream of both the exhaust surge tank and the exhaust backpressure valve. In order to obtain the real-time mass percentage of EGR, measurements on the same Horiba emissions bench were done for inlet CO₂ (Model AIA 210) and inlet O₂ (Model MPA 21A). All emissions equipment used heated lines and filters except the intake CO₂ and O₂.

Table 2. Injection System Specifications

Fuel Injection Equipment		
Design	Common Rail	-
Injector	Bosch CRIP 2	-
Included Cone Angle	155	Deg
Nozzle Type	Mini-Sac	-
Nozzle Flow Number	440	cc/30sec
Nozzle Hole Diameter	0.141	mm
Number of Holes	7	-
Operated Rail Pressures	1000(1100)	Bar
Peak Rail Pressure	1600	Bar
Fuel Pump	Bosch CP3.3	-

The fuel used in the experiments was intended to be a typical Gasoline with Octane number of 87. Infineum R655 was used as lubricity additive in the experiments. It provided sufficient protection to the fuel injection hardware. The general properties of this fuel can be seen in Table 3.

Table 3. Gasoline fuel properties for testing

Fuel Properties for the Typical Gasoline	
Spec Gravity	0.737
LHV [MJ/kg]	43.2
H/C ratio	1.88
Ethanol [%]	0
(RON+MON)/2	87
Temp at 10% Distilled [°C]	69.4
Temp at 50% Distilled [°C]	105.0
Temp at 90% Distilled [°C]	160.6

TEST CONDITIONS

The baseline GDICI engine operational conditions were guided by numerical simulations and are listed in Table 4. The injection delay between the SOIC and actual injection timings was about 360 μ s, which is equivalent to 5.4 °CA at 2500 rev/min. Injection-system rail pressure was 1000 bar, which was found to be an optimal value in the previous study [10]. Engine operation was at full load (targeting 16 bar IMEP).

Table 4. Baseline engine operation conditions

Engine Speed	rev/min	2500
Net IMEP	bar	16.0
Fuel Rail Pressure	bar	1000
SOIC 1 st	° ATDC	-350
SOIC 2 nd	° ATDC	-21
SOIC 3 rd	° ATDC	-8
Fuel Mass% 1 st /2 nd Injection	%	57/39
Inlet Pressure	kPa abs	280
Exhaust Pressure	kPa abs	301
Swirl Ratio	-	2.2
Inlet O ₂	%	11.3
EGR	%	48
Intake Temperature	° C	50
Coolant Temperature	° C	90
Oil Temperature	° C	92

RESULTS AND DISCUSSION

Using the computational model already described, operation ranges of a light-duty diesel engine operating with GDICI combustion with constraints of combustion efficiency, PRRmax (or Pmax) and emissions were identified as functions of injection timings and the fuel split ratio of triple-pulse injections. To determine the operation range combustion efficiency of 85% or higher, PRRmax of 10 bar/deg or lower (or Pmax of 16 MPa, whichever comes first), PM and NO_x emissions of 1.0 g/kg-fuel or lower, respectively, were considered. PRRmax of 10 bar/deg was chosen as the limit because it is representative of the upper range of acceptable pressure rise rate at full load operation. Note that the limit was simply used as a screening criterion to quantify the size of the acceptable operating range of the engine and it is not a limit that would be carried into production unchanged or to be used as a single constraint across an engine speed-load map.

GENERATION OF COMPUTED OPERATION MAP

Effects of split ratio: extension of PRRmax-line

Figure 4a shows predicted pressure profiles for the engine operating with HCCI (i.e., totally premixed) combustion with partial fuel amounts between 60 and 100 % of total fuel to be injected. As the premixed fuel amount increases, PRRmax and Pmax increase. Considering the constraints of PRRmax (=10bar/deg) and Pmax (=16MPa), a fuel quantity of 70% of the total fuel quantity is predicted to yield HCCI operation

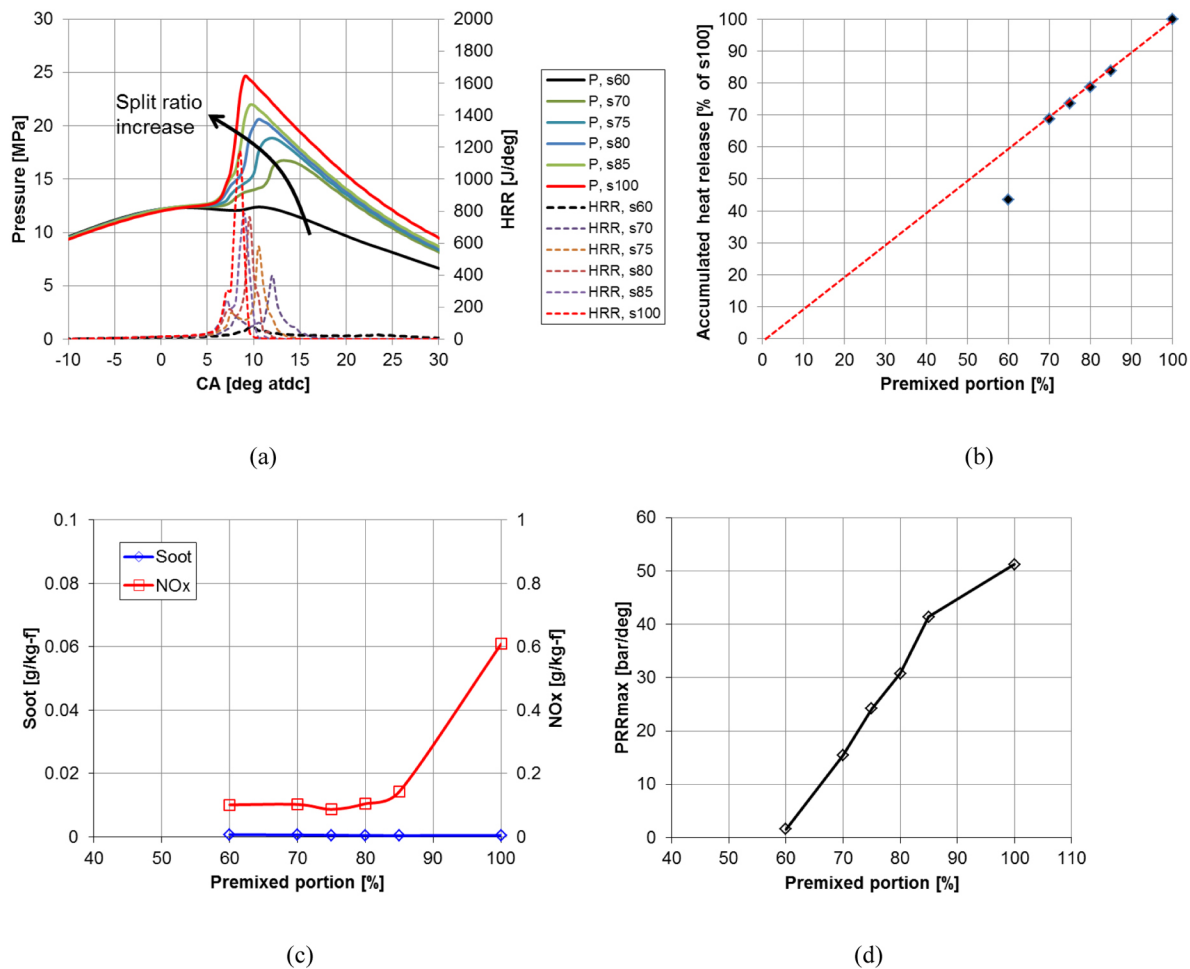


Figure 4. Predicted pressure and HRR profiles and emissions of HCCI engine operation with various partial fuel amounts. For all the cases, O_2 concentration, gas temperature and pressure at IVC were 10.9%, 361 K and 3.25 bar, respectively. (a) pressure and HRR profiles, (b) combustion efficiency, (c) soot and NO_x emissions, (d) PRRmax.

which has borderline acceptability for preventing excessive pressure rise. It is also predicted that the accumulated heat releases of HCCI operation start dropping from complete reaction of the fuel in the chamber, at approximately 70% of the total fuel being introduced in the first injection (Figure 4b). This implies that engine operation with more than 70% of total fuel introduced in the first injection to form the initial premixed mixture is likely to result in stable combustion, but with an excessive PRRmax. Predicted NO_x and soot amounts are shown in Figure 4c. The soot emissions were extremely low. The NO_x emissions were predicted to increase rapidly beyond 85% of the total fuel, but stayed within the constraint value of 1.0 g/kg-f. The corresponding maximum pressure rise rates were predicted to increase with the premixed portion, as shown in Figure 4d. For fuel amounts greater than about 66% of the total fuel, PRRmax becomes higher than the constraint value used in the present study.

Simulations of stratified mixtures with 70% of the total fuel with the same initial conditions indicated that ignition occurred before that shown in Figure 4, where the mixture

was homogeneous. We used this result to formulate the strategy for the triple injection. The first injection, establishing the premixed base charge would be limited to less than threshold point shown in Figure 4, i.e. 70% of the total fuel. This would allow the second injection to manipulate the heterogeneity and thus promote ignition but still keep the rates of pressure rise and peak pressure within our constraints. However, to minimize particulate matter and NO_x we would still try to keep a substantial portion of the total fuel in the first injection, as shown in Figure 4c.

Using 60% 1st pulse split ratio, which results in significant incomplete burning in HCCI combustion (refer to Figure 4b), the effect of a 2nd pulse injection was investigated for various SOIC2 and fuel amounts. Figure 5a shows PRRmax variation for the baseline O_2 fraction, Tivc, and Pivc. In the figure, various double-injection runs with S1=60% and S2= 40% are also shown for comparison. For the complete fuelling case, S1=60%, S2=40%, an operable SOIC2 window is predicted for a retarded timing range (-13 ~ +2° ATDC). It is seen that the earliest and latest SOI2 are

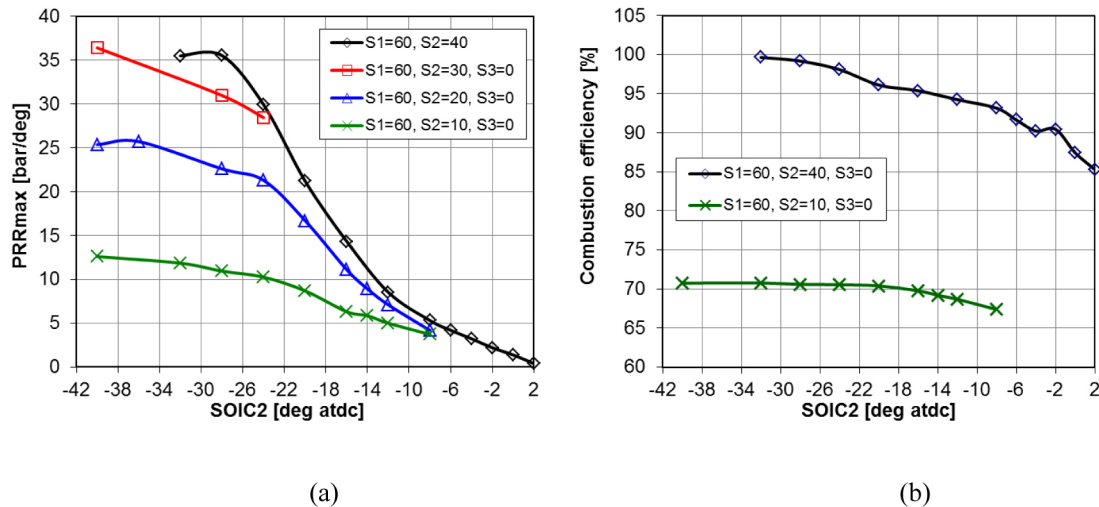


Figure 5. Effects of 2nd pulse injection timing variation for various partial fuel amounts. For all the cases, O₂ concentration, gas temperature and pressure at IVC were 10.9%, 361 K and 3.25 bar, respectively. (a) PRRmax vs. SOIC2, (b) Combustion efficiency vs. SOIC2.

bounded by PRRmax (Figure 5a) and combustion efficiency (Figure 5b), as was reported in the authors' previous study [10]. As the 2nd pulse amount is reduced, the maximum pressure rise rate is reduced for the same SOIC2, thus the SOIC2 at which PRRmax becomes the threshold value (hereafter called as 'PRRmax-line') advances. Note that, as SOIC2 further advances, PRRmax approaches the values of the corresponding HCCI operation (Figure 4d) since the mixture becomes more uniform due to more time available for fuel/air mixing. On the contrary, the combustion becomes more unstable and combustion efficiency decreases as SOIC2 is retarded, as shown in Figure 5b. Thus, the latest SOIC2 of the operable SOIC2 range can be bounded by this combustion stability such that combustion efficiency with the remaining 3rd pulse injection becomes a threshold value at the constraint value of 85%. For example, for S1=60%, S2=10%, and S3=30%, the latest SOIC2 that has operable SOIC3 window was predicted to be near -10° ATDC. For SOIC2s later than -10° ATDC, any operation with SOIC3 after the end of 2nd pulse injection plus the physical dwell time could not meet the combustion efficiency constraint (results not shown).

For a given S1, it was predicted that the combustion stability line (hereafter called the 'CS-line') of the 2nd pulse around the SOIC2 that has about 95% of heat release of the HCCI operation case. This CS-line slightly advances with decreasing 2nd pulse split ratio (results not shown here), and thus the window of operable SOIC2 is increased with decreasing S2 fuel quantity. Note that the latest possible SOIC2 is found in the double-pulse injection case. For example, the stability criterion for S1=60%, S2=40% and S3=0 gives the most retarded SOIC2 for stable combustion.

Effects of split ratio: extension of CS-line

To explore the effects of split ratio on the combustion stability line the condition of PRRmax for S1=60%, S2=10%, S3=0 was chosen as an initial starting point. The SOIC2 was set to -24° ATDC. This corresponded to the earliest SOIC2 that meets the PRRmax constraint of 10 bar/deg in Figure 5a. Note that early SOIC2 is preferred because not only one of the objectives of the present study is to extend the operable injection timing window, but also combustion efficiencies increase with advancing SOIC2 (Figure 5b). At this condition the remainder of the fuel was injected, S3=30%, with a start of injection command varying from -10 to +2° ATDC. The earliest SOIC3 was predicted to be limited by the physical minimum dwell time between the 2nd and 3rd pulses for engine operation with a single injector. In the present study, the minimum dwell time of 0.33 ms (5 deg CA at 2500 rev/min) was used.

Figure 6a compares the pressure profiles of triple injection operation of S1=60%, S2=10% and S3=30%, with different SOIC3, to the corresponding double-injection of S1=60% and S2=40%. Also shown for reference is the double injection case S1=60%, S2=10% and S3=0 (partial fuel case in Figure 5a). It is clearly seen that the addition of the 3rd injection does not change the PRRmax and Pmax much from that of double pulse injection with partial fuel (S2=10%), while the ignition timings are only slightly changed. The expansion pressures are increased by the addition of the 3rd pulse, which increases the engine power. The overall heat release was always higher than the threshold value of 85% for the SOIC3s considered in the study. However, the maximum combustion efficiency for the triple injection operation was predicted to be slightly lower than that of the double injection S1=60% and S2=40%, which