

Investigation on Multiple Injection Strategies for Gasoline PPC Operation in a Newly Designed 2-Stroke HSDI Compression Ignition Engine

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ABSTRACT

Partially Premixed Combustion (PPC) of fuels in the gasoline octane range has proven its potential to achieve simultaneous reduction in soot and NO_x emissions, combined with high indicated efficiencies; while still retaining proper control over combustion phasing with the injection event, contrary to fully premixed strategies. However, gasoline fuels with high octane number as the commonly available for the public provide a challenge to ensure reliable ignition especially in the low load range, while fuel blends with lower octane numbers present problems for extending the ignition delay in the high load range and avoid the onset of knocking-like combustion. Thus, choosing an appropriate fuel and injection strategy is critical to solve these issues, assuring successful PPC operation in the full engine map.

In this framework, the objective of the present investigation consists of evaluating the use of multiple injection strategies for achieving stable PPC operation, attaining low NO_x and soot emissions together with high efficiencies. This research was carried out in a single-cylinder DOHC 2-stroke HSDI CI engine using 95 Research Octane Number (RON) gasoline fuel. Three different operating conditions in terms of indicated mean effective pressure (IMEP) and speed were investigated: 3.1 bar IMEP and 1250 rpm, 5.5 bar IMEP and 1500 rpm and 10.4 bar IMEP and 1500 rpm. Parametric variations of injection timings, at different rail pressures and different fuel split between injections were experimentally performed to analyze the effect of the injection strategy over the combustion process, exhaust emissions and efficiency levels.

Experimental results confirm how using an appropriate injection strategy helps to achieve stable PPC operation in the selected operating conditions; with competitive combustion stability, lower NO_x and soot levels, and moderate CO and HC emissions with combustion efficiency over 96%, compared to Conventional Diesel Combustion (CDC).

Finally, a detailed analysis of the local cylinder conditions was performed by means of 3D-CFD simulations in order to provide guidelines for further optimization of the gasoline PPC concept, when using multiple injection strategies in the 2-stroke engine under development.

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INTRODUCTION

Conventional compression ignition (CI) engines are well known for their higher thermal efficiency compared to gasoline spark ignition (SI) engines. However, the characteristic mixing-controlled combustion stage of the Conventional Diesel Combustion (CDC) concept still represents an important source of nitrogen oxides (NO_x) and also particulate matter pollutant emissions [1, 2, 3, 4, 5].

In the past decade, increasingly stringent pollutant emission regulations added to the need of decreasing CO_2 emissions while also meeting customer's expectations regarding fuel consumption, has driven research efforts towards further increasing engine thermal

efficiency while simultaneously reducing harmful exhaust emissions [6]. This has considerably accelerated the development process of established and new engine technologies, focused specially on after-treatment and advanced combustion strategies for pollutant control, but also on decreasing mechanical and thermal losses for improving efficiency while increasing the specific power of modern combustion engines [7, 8, 9].

With this motivation, an innovative DOHC 2-stroke HSDI CI engine with scavenge loop through four poppet valves in the cylinder head is being developed for a heavily downsized passenger car application, where high power-to-weight ratio is mandatory. The idea behind this

new engine concept is implementing a 2-stroke cycle to downsize the displacement and obtain a two cylinder (730 cm³) engine with equivalent NVH and torque response than the base 4-stroke four cylinder engine [10].

Previous research carried out by the authors in a single-cylinder research version of this 2-stroke engine operating in CDC, confirmed that the engine architecture under study provides high flexibility in terms of air management settings, to control the cylinder conditions and affect combustion environment and final emissions level [11]. Furthermore, the Homogeneous Charge Compression Ignition (HCCI) combustion concept with diesel fuel was implemented at low load conditions, and its potential for simultaneous reductions of NO_x and soot emissions was experimentally proven [12]. However, the high reactivity of diesel fuel added to the intrinsically high residual gas fraction (IGR), characteristic of the scavenge loop architecture; made it impossible to attain a properly-phased combustion process even when using optimized hardware and engine settings [13]. Therefore, the application of HCCI in this 2-stroke engine concept was discarded.

Several problems commonly encountered when operating with HCCI and high cetane diesel fuels, can be alleviated by switching towards slightly retarded injection timings compared to those required to achieve a fully homogeneous mixture [14, 15, 16, 17, 18, 19]. This approach, known as Premixed Compression Ignition (PCI) or Partially Premixed Combustion (PPC), was designed to operate CI engines in between fully premixed and fully diffusive combustion, where simultaneous reduction in NO_x and soot emissions can be still attained, while retaining the control over the combustion timing by the injection event [20, 21].

The first strategies reported in the literature to achieve PPC with high cetane diesel fuels, were focused on a combination of high fractions of cooled EGR with comparatively low compression ratio; and/or high injection pressures combined with high swirl ratios to speed up the mixing process [20, 21, 22, 23]. These strategies aim to extend the ignition delay for premixing the fuel-air charge as much as possible before combustion starts, avoiding over-rich regions where soot is formed; while NO_x formation is decreased by reducing combustion temperatures due to the dilution effect of EGR. However, the operating range for simultaneous reduction of NO_x and soot emissions is narrow and a sharp decline in combustion efficiency is often unavoidable.

Later, research work performed by Kalghatgi et al. both in heavy-duty and light-duty size engines, demonstrated how gasoline-like fuels, having a higher resistance to auto-ignition, are better suited for extending mixing times before the onset of combustion compared to diesel-like fuels [24, 25, 26]. As a result, low values of engine-out soot and NO_x emissions can be obtained in a wider range of loads compared to PPC of diesel fuels.

Since this early work, many research groups from Lund University [27, 28, 29, 30, 31, 32], University of Cambridge [33], Argonne National Laboratory [34, 35, 36], University of Wisconsin Madison [37] and Delphi Corporation [38, 39, 40] have performed additional experimental and numerical investigations operating with PPC using

different fuels in the octane range of gasoline and ethanol. Different injection strategies have been explored, with various EGR rates, boost pressures, intake temperatures and swirl ratios at different engine loads and speeds. In general, reported results confirmed how it is possible to implement PPC with very high efficiency, very low NO_x emissions and also lower soot levels compared to CDC in a wide range of load operation.

During PPC operation, the reactivity of the fuel-air charge is mainly controlled by the ignition characteristics of the fuel, the cylinder thermo-chemical conditions, and the mixture stratification in terms of equivalence ratio prior to the start of combustion (SoC). For instance, gasoline fuels with high octane number (ON) provide a challenge to ensure reliable ignition especially in the low load limit, while fuel blends with lower ON present problems for extending the ignition delay in the high load range and avoid the onset of knocking combustion. This supposes that the PPC concept requires different fuel reactivity and/or advanced valvetrain and boost/EGR systems to assure proper ignition control, and optimize emissions and efficiency in the full engine map [39]. Thus, there are still many practical issues which remain under investigation before reaching a production-viable powertrain; such as the injection system requirements, combustion chamber design, air charging strategy definition, among others.

In this framework, the flexibility of the 2-stroke architecture for assuring stable PPC operation in medium/low load conditions was already demonstrated by the authors using a single injection strategy with RON95 gasoline [41]. At 5 bar IMEP and 3 bar IMEP it was possible to achieve low NO_x emissions (below 0.4 g/kWh) with extremely low soot emissions, while retaining 98% of combustion efficiency and proper combustion stability with a coefficient of variation (CoV) in the IMEP under 3%. However, at higher load (10 bar IMEP) a transition between premixed and mixing-controlled combustion was observed depending on the particular in-cylinder conditions, and the conventional trade-off between NO_x and soot emissions was recovered [42].

Recent investigations have shown that the use of a multiple injection strategy allows precise control of the fuel-air stratification before the SoC, which affects the timing and strength of auto-ignition as well as the rate and completeness of fuel oxidation throughout the combustion chamber [35, 43, 44, 45]. Moreover, Sellnau et al. confirmed the potential of a triple injection strategy for increasing thermal efficiency compared to single injection strategies thanks to reduced heat losses during the expansion stroke given by a more favorable fuel distribution during combustion which results in less contact between hot combustion gases and chamber walls [38, 39].

As mentioned, combustion characteristics are highly dependent on the mixture preparation prior to ignition; therefore, both the timing and the fuel quantity injected in each injection must be carefully optimized depending on the operating condition. The objective of the present investigation focuses on evaluating the use of multiple injection strategies for achieving stable PPC operation in the single-cylinder 2-stroke CI engine using RON95 gasoline. Parametric variations of the timing of the main injection, with different rail pressures and different fuel split between injections were

experimentally performed at three operating conditions, to analyze the effect of the injection strategy on the combustion process, exhaust emissions and efficiency levels.

EXPERIMENTAL SETUP

Engine architecture and test cell characteristics

Experimental activities were performed in the single-cylinder research version of an innovative Renault engine concept, consisting of a two-cylinder DOHC 2-stroke HSDI CI engine with scavenge loop, which is currently under development.

The combustion chamber has four poppet valves with double-overhead camshafts and a staggered roof geometry, specifically designed for masking the flow of air between the intake and exhaust valves, allowing proper scavenging of the burnt gases while keeping short-circuit losses as low as possible during 2-stroke operation. The definition of the engine architecture, boost system requirements, combustion chamber geometry and scavenging characteristics of this newly designed engine were reported by the authors in previous publications [10, 46].

A hydraulic cam-driven Variable Valve Timing system allows delaying intake and exhaust valve timings with a cam phasing authority of +30 degrees from base timing, as it was detailed in a previous investigation [41, 42]. In this research, the key valve timing angles (EVO/EVC/IVO/IVC) were defined at those crank angle degrees (CAD) where the given valve lift was 0.3 mm. The single cylinder research version of the Renault 2-stroke engine concept has been manufactured by Danielson. As a reference, Table 1 lists detailed engine specifications.

Table 1. Main engine specifications.

Engine type	2-stroke compression ignition
Displacement	365 cm ³ (single cylinder)
Bore × Stroke	76 mm × 80.5 mm
Connecting Rod Length	133.75 mm
Compression ratio	17.6:1
Number of Valves	4 (2 intake & 2 exhaust)
Type of scavenge	Poppet valves with scavenge loop
Valvetrain	DOHC with VVA
Nominal intake valve timing (set at VVT=0)	IVO=161.9 CAD aTDC IVC=251.6 CAD aTDC
Nominal exhaust valve timing (set at VVT=0)	EVO=122.6 CAD aTDC EVC=226.9 CAD aTDC
Fuel injection system	Diesel common rail HSDI
Injector nozzle	148° AN, 8 holes, 90µm

The single-cylinder engine is very flexible and parts can be easily interchanged. However, for this first evaluation of the PPC concept, a conventional diesel piston with geometric compression ratio equal to 17.6 and wide angle injector nozzle non-optimized for the strategy were kept on the engine. The injection system is a common rail HSDI designed for injecting diesel up to a maximum rail pressure of 1800 bar. The injector is equipped with a 8 holes nozzle, with hole diameter of 90 µm and a spray cone angle of 148°. A detailed

optimization operating with the PPC concept is expected to provide a better piston/nozzle match in terms of number of holes, hole diameter, spray included angle and bowl geometry.

The injector mass flow rate and spray momentum flux were measured in a dedicated test rig at a suitable range of operating conditions in terms of injection pressure, injector back-pressure and injection duration; following the methodology described in [47, 48] using commercial diesel fuel and also the selected gasoline fuel. The maximum injection pressure when injecting gasoline is limited to 1200 bar, to avoid cavitation in the return line and assure correct measurement of the fuel flow. However, in the conditions studied in this research, extremely high injection pressures are not expected to be interesting, since lower pressures help reducing the tendency for spray impingement on surfaces of the combustion chamber. A lubricity additive was added to the RON95 gasoline, in a small proportion compared to the total blend, to ensure proper operation of the injection equipment without affecting the ignition characteristics of the gasoline. Most important fuel properties are listed in Table 2.

The single-cylinder engine test cell is equipped with independent water and oil cooling circuits, an external compressor unit with its dryer for providing water-free compressed air to simulate the required boosting conditions, and an additional low pressure EGR circuit to provide arbitrary levels of cooled EGR even at high intake pressures. The fuel consumption of the engine is measured with an accuracy of 0.2% using a gravimetric dynamic fuel meter. Measurements of O₂, CO, CO₂, HC, NO_x, N₂O, and EGR rate are performed for all the tests with a state-of-the-art HORIBA 7100 DEGR gas analyzer. Soot emissions traced by the filter smoke number (FSN) are measured with an AVL 415 Smokemeter.

The laboratory setup used in the experimental test campaign, as well as the required instrumentation and the accuracy of most important measurement equipment, were fully described in previous publications [41, 42].

Table 2. Fuel properties.

Test fuel	Unleaded gasoline with lubricity additive
Research Octane Number (RON)	94.6
Motor Octane Number (MON)	84.8
H/C ratio	1.76 mol/mol
O/C ratio	0 mol/mol
Aromatics	36.1 %Vol
Benzene	0.3 %Vol
Oxygen content	< 0.17% (m/m)
(A/F) _{St} (by mass)	14.37
LHV	42.82 MJ/kg
Density (15°C)	758.1 kg/m ³
Kinematic viscosity (40°C)	0.44 cSt

The trapping ratio is defined as the mass of delivered charge that has been trapped in the cylinder at IVC divided by the mass of delivered charge supplied to the cylinder (fresh air plus EGR). It is measured experimentally in every point of the test matrix by means of a tracer

gas method [49, 50], using methane (CH_4) as external gas. First, a controlled quantity of CH_4 (around 1000 ppm) is homogeneously injected in the intake flow, when operating at the stabilized point with the desired engine settings. Next, the CH_4 concentration is measured at the intake and also at the exhaust manifolds with a dedicated analyzer. Assuming that all the CH_4 trapped in the cylinder will burn completely during combustion, an accurate estimation of the short-circuited mass from the intake flowing directly to the exhaust is obtained by means of a set of mass balances in the intake, exhaust and in-cylinder gases.

The internal gas recirculation (IGR) ratio is then defined as the fraction of residual gases retained from the previous combustion cycle in the total trapped mass in the cylinder. The IGR ratio, total trapped mass at IVC and in-cylinder effective equivalence ratio (ϕ_{eff}) are estimated in each measured test using simplified thermodynamic calculations. This estimation is based on an enthalpy balance, where the enthalpy of the total trapped mass at the IVC equals to the enthalpy of the residual mass plus the enthalpy of the intake delivered trapped mass, both estimated also at the IVC

It can be presumed that the air management characteristics are not affected by the negligible change in the combustion process that could be derived from the addition of the tracer gas, when is injected in such a small proportion. However, for accurate measurement of the exhaust emissions and precise combustion diagnosis, all the required instantaneous signals and time-averaged measurements are performed after the trapping ratio has been measured, thus, after removing the injection of tracer gas in the intake flow.

Cylinder pressure is measured using a piezoelectric sensor, while a different piezoresistive pressure sensor is placed at the cylinder liner close to the bottom dead center to reference the piezoelectric sensor signal. All high frequency signals are sampled with a resolution of 0.2 CAD. Main global combustion parameters like indicated mean effective pressure (IMEP), peak cylinder pressure (P_{max}), maximum pressure gradient (dP/da_{max}) and combustion stability indicators (CoV IMEP and CoV P_{max}) are directly derived from the analysis of the cylinder pressure signal. An in-house combustion analysis software (CALMEC) is used to resolve the first law of thermodynamics and obtain the instantaneous evolution of the energy released by the progress of combustion from the measured pressure signal. Then, the start of combustion (SoC), combustion angles (CA10, CA50, CA90), ignition delay and mixing times are obtained from the calculated rate of heat release (RoHR). The RoHR calculation includes sub-models for considering heat transfer losses, mechanical deformation of the cylinder and blow-by losses.

Finally, combustion noise has been calculated following the classical approach introduced by Austen and Priede [51]. The classical approach is frequently used by engine development engineers to assess the overall engine combustion noise level at steady operating conditions [52, 53]. This method is based on calculating the 'structural attenuation' curve, which is the difference between the cylinder pressure and the radiated noise 1/3-octave band spectra. In this theory, since a linear response of the engine structure is assumed,

its characteristic attenuation curve can be used as a transfer function to estimate the sound pressure level spectrum of the engine noise from the cylinder pressure trace.

Multi-Dimensional Engine Model

A computational fluid dynamic (CFD) model was built in the CONVERGE CFD platform to perform full coupled open and closed cycle calculations using the full intake/exhaust and cylinder geometries. The CFD code uses a structured Cartesian grid with base cell size of 3 mm. Three additional grid refinements linked to flow velocity and temperature were performed by means of an adaptive mesh refinement (AMR) as well as a fixed three level refinement within the spray region.

The injection rate profile was generated from the experimental database available after the injector characterization. The diesel-like injection of gasoline is simulated by the standard Discrete Droplet Model (DDM) [54]. Gasoline fuel physical properties are defined using iso-octane as surrogate. Spray atomization and break-up are simulated by means of the Kelvin-Helmholtz-Rayleigh-Taylor (KH-RT) model [55]. Turbulent flow is modeled by means of the renormalization group (RNG) k- ϵ model with wall functions in order to account for wall heat transfer [56]. Concerning combustion modeling, a direct integration of detailed chemistry approach was used by means of the CONVERGE code and the SAGE solver.

The chemical mechanism of a primary reference fuel (PRF) blend of n-heptane (5%) and iso-octane (95%) is used as fuel surrogate to reproduce the ignition characteristic of the RON95 gasoline. A well-validated skeletal reaction mechanism for PRF oxidation derived from the ERC-Multichem mechanism [57], with 45 species and 152 reactions was used for the fuel chemistry representation. In particular, the mechanism includes the thermal path for NO_x formation according to the extended Zeldovich mechanism, and the N_2O path for accounting the NO_x formation at low temperatures, where this path gains relevance. A classical 2-step phenomenological soot model is used, with the Hiroyasu soot formation mechanism combined with a Nagle-Strickland-Constable (NSC) model for soot oxidation [58, 59].

The full description of the model setup performed at the reference case with 10.4 bar IMEP and 1500 rpm operating with the gasoline PPC concept and the three injection strategy was described in detail in a previous publication [44].

METHODOLOGY

The research work, divided into the three different load points described below, corresponds to basic studies of the gasoline PPC concept, without in-depth optimization of the engine hardware or settings. The engine operating conditions chosen for this experimental test campaign correspond to one low speed/load point, at 1250 rpm and 3.1 bar of IMEP; and two low speed and medium load points, at 1500 rpm and 5.5 and 10.4 bar IMEP respectively.

A dedicated Design of Experiment (DoE) campaign, using a Central Composite Design with four factors and five levels, was previously performed in the engine, to optimize the air management conditions using response surface models for different engine outputs (such as fuel consumption or emissions), while operating in CDC at the selected operating points [11, 44]. The intake pressure (P_{int}), the EGR rate, the pressure difference between the intake and exhaust (ΔP), and the valve overlap (Olap) are the four air management settings selected as input factors in the DoE. The resulting second order mathematical models were used to find proper in-cylinder conditions, mainly in terms of oxygen concentration ($YO_{2,IVC}$) and temperature at IVC (T_{IVC}), to assure proper ignition around TDC when operating with the PPC concept.

A multiple injection strategy (triple or double injection) was used in all studies presented in this research, with a fixed fueling rate which provided the required IMEP target at the baseline case with the optimum CA50 at each load. The total injected quantity was kept constant for all tests along the different studies. Then, the injection timing of the main injection, which is the one primarily controlling combustion onset, was swept each 2 CAD for each study; in a range defined considering the onset of knocking combustion or smoke limit and the deterioration of combustion stability as the main constraints.

For medium-to-high load points (as in the case of 10.4 bar IMEP), a triple injection strategy is expected to help in achieving the load target while avoiding/mitigating knock tendency, as in the case of medium load operation (5.5 bar IMEP) both triple and double injections are evaluated in order to find the most suitable injection pattern. Finally, in the case of the lower load condition (3.1 bar IMEP), a double injection strategy was selected in advance, since a relatively small fuel quantity is injected. So, the 1st early injection placed at -60 CAD aTDC is removed, to avoid excessively high HC emissions, and the fuel is split between the remaining two injections.

The influence of the injection pressure (P_{rail}) and fuel split between the injections (%fuel) is evaluated at the medium load points (5.5 and 10.4 bar IMEP), aside from the straight effect of sweeping the injection timing of the main injection. Specifically in the case of the low load point (3.1 bar IMEP), it was only possible to measure two different %fuel at one level of injection pressure. At this low load condition, the high resistance to auto-ignition given by the RON95 gasoline increases the sensitivity of the combustion process to the injection strategy.

Oil and coolant temperatures were kept at 90°C, while intake air temperature (T_{int}) was carefully controlled during all tests by using a heater. The injection timing is referred to the Start of Energizing (SoE) current of the injector instead the actual Start of Injection (SoI), which happens a few degrees (1.5 to 2 CAD) after the SoE due to the hydraulic delay affecting the needle lift. The most relevant engine settings chosen for each operating condition are detailed below in Table 3.

Table 3. Engine settings for experiments at a) 1500 rpm - 10.4 bar IMEP, b) 1500 rpm 5.5 bar IMEP and c) 1250 rpm and 3.1 bar IMEP.

a) 1500 rpm / 10.4 bar IMEP						
Air management	T_{int} [°C]	P_{int} [bar]	ΔP [bar]	$VVT_{(1,2)}$ [CAD]	Olap [CAD]	EGR [%]
ALL TESTS	35	2.75	0.71	(5,20)	78.4	43.5
Injection process	m_{fuel} [mg/st]	P_{rail} [bar]	SoE1 [CAD]	SoE2 [CAD]	SoE3 [CAD]	%fuel [%]
Baseline	18.8	850	-60	-42/-34	-2	20/64/16
$P_{rail}=750$ bar	18.8	750	-60	-44/-38	-2	20/64/16
$P_{rail}=950$ bar	18.8	950	-60	-38/-34	-2	20/64/16
%fuel=20/69/11	18.8	850	-60	-44/-40	-2	20/69/11
%fuel=20/56/24	18.8	850	-60	-38/-34	-2	20/56/24

b) 1500 rpm / 5.5 bar IMEP						
Air management	T_{int} [°C]	P_{int} [bar]	ΔP [bar]	$VVT_{(1,2)}$ [CAD]	Olap [CAD]	EGR [%]
ALL TESTS	45	1.7	0.32	(8,8)	63.4	33.5
Injection process	m_{fuel} [mg/st]	P_{rail} [bar]	SoE1 [CAD]	SoE2 [CAD]	SoE3 [CAD]	%fuel [%]
Baseline	10.8	600	-60	-42/-34	-4	16/68/16
$P_{rail}=400$ bar	10.8	400	-60	-46/-38	-4	16/68/16
$P_{rail}=800$ bar	10.8	800	-60	-40/-34	-4	16/68/16
%fuel=0/84/16	10.8	600	w/o	-46/-38	-4	0/84/16
%fuel=0/68/32	10.8	600	w/o	-48/-40	-4	0/68/32
%fuel=16/84/0	10.8	600	-60	-46/-40	w/o	16/84/0
%fuel=16/68/0	10.8	600	-60	-46/-40	w/o	32/68/0

c) 1250 rpm / 3.1 bar IMEP						
Air management	T_{int} [°C]	P_{int} [bar]	ΔP [bar]	$VVT_{(1,2)}$ [CAD]	Olap [CAD]	EGR [%]
ALL TESTS	45	1.295	0.21	(14,13)	62.4	15
Injection process	m_{fuel} [mg/st]	P_{rail} [bar]	SoE1 [CAD]	SoE2 [CAD]	SoE3 [CAD]	%fuel [%]
%fuel=0/60/40	6.5	400	w/o	-36/-26	-4	0/60/40
%fuel=0/40/60	6.5	400	w/o	-40/-30	-4	0/40/60

RESULTS AND DISCUSSION

Focusing on the air management characteristics, Table 3 summarizes the main settings selected for the injection studies. The goal is to control the in-cylinder conditions, by balancing both the trapping ratio and the amount of residual hot gases (IGR ratio), to search for conditions with high air trapped mass, which allow using the high EGR rates needed to decrease NO_x emissions and control the combustion rate. In addition, the requirements in combustion stability are assured by affecting the temperature at the beginning of the closed cycle. Previous research work focused on the implementation of the PPC concept using a single injection strategy in the current 2-stroke engine, already demonstrated the flexibility for controlling oxygen concentration ($YO_{2,IVC}$) and temperature (T_{IVC}) at the IVC by using EGR, Δp and valve overlap as main control levers [41].

At the higher load point (10.4 bar IMEP), where temperatures are inherently high, the introduction of high rates of EGR (43.5%) is mandatory for decreasing $YO_{2,IVC}$. This requires an intake pressure of 2.75 bar, 0.71 bar of ΔP , and an overlap of 78.4 CAD. With these settings, the air management parameters correspond to 67% of trapping ratio, 35% of IGR ratio, 0.83 of ϕ_{eff} and 12% and 18% of $YO_{2,IVC}$ and T_{IVC} respectively. The low oxygen concentration at IVC

is necessary to avoid the onset of knocking-like combustion, reduce NO_x emissions by decreasing combustion temperatures, and also to extend the ignition delay until reaching the mixing times needed to decrease soot emissions. The air management characteristics remained constant along the performed parametric tests, due to their weak relation with the injection settings.

When decreasing the engine load, decreasing ΔP and valve overlap is mandatory to increase the IGR ratio and then T_{IVC} until reaching the auto-ignition temperature of the RON95 gasoline (around 950 K) close to TDC and sustain the combustion process. Moreover, the combustion process becomes more sensitive and less tolerant to high EGR rates, so both the intake temperature and T_{IVC} are used to compensate the overall decrease in charge reactivity. At the medium load point (5.5 bar IMEP), the decrease in valve overlap and ΔP to 63.4 CAD and 0.32 bar, made possible to increase trapping ratio and IGR ratio to 82% and 43%, which added to a relatively high P_{int} and EGR of 1.7 bar and 33.5%, resulted in 12% of $\text{YO}_{2,IVC}$ with 220°C of T_{IVC} and 0.8 of ϕ_{eff} .

Finally, at the lower load point (3.1 bar IMEP) the main priority was to achieve stable PPC operation and assure reliable ignition. The chosen air management settings provided a trapping ratio/IGR combination of 81% and 54% respectively, while the EGR level is limited to 15% to maintain high combustion stability. The final in-cylinder conditions at this low load operation correspond to 0.66 of ϕ_{eff} and 13.8% and 235°C of $\text{YO}_{2,IVC}$ and T_{IVC} respectively. This intrinsic flexibility of the 2-stroke engine architecture is a key advantage to achieve PPC conditions over a wide operating range and assure stable combustion even at low loads, where auto-ignition of high octane fuels can be difficult in conventional 4-stroke engines.

Regarding the combustion process, Figure 1 shows the effect of the second injection timing, denoted as SoE2, over the RoHR profile for the three operating conditions at the baseline or reference study. The range of SoE2 is limited by poor combustion stability and misfire in the case of early timings, and by the onset of knocking-like combustion or smoke level above 4 FSN in the case of late timings.

As described in a previous investigation performed at medium-to-high load conditions using a triple injection strategy, the timing of the 2nd injection is controlling both start of combustion (SoC) and phasing, while the timing of the 3rd injection controls combustion rate and duration, but has a very small influence in the onset of combustion [44].

Figure 2 illustrates the effect of SoE2 over the SoC, and over combustion phasing traced by the angle at 50% of the mass burnt fraction (CA50) for the three operating conditions.

In the case of medium/high load conditions (5.5 bar and 10.4 bar IMEP), combustion onset and phasing shift towards the expansion stroke when advancing SoE2, while the RoHR becomes smoother with longer duration and lower peak, as confirmed by Figure 1. On the contrary, retarding SoE2 closer to TDC decreases ignition delay and mixing time, so the local equivalence ratio stratification increases

enhancing the reactivity of the mixture. Consequently, SoC and CA50 advance towards the TDC, while combustion is faster with a higher peak in the RoHR.

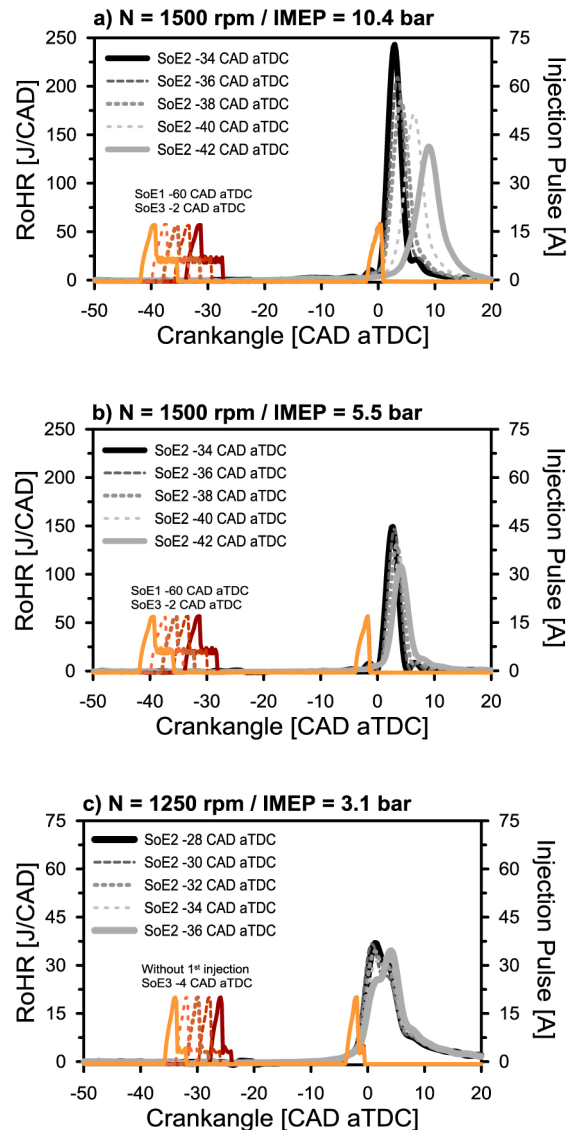


Figure 1. Effect of SoE2. RoHR and injection pulse for a) N=1500 rpm / IMEP=10.4 bar Baseline, b) N=1500 rpm / IMEP=5.5 bar Baseline and c) N=1250 rpm / IMEP=3.1 bar %fuel=0/60/40 case.

At the lower load point (3.1 bar IMEP) the RoHR exhibits a combined structure, with an initial premixed phase followed by a mixing-controlled stage, as shown in Figure 1.c. The stratification in local ϕ , given by the relatively retarded SoE2 and higher fuel amount in the late injection close to TDC, allows assuring higher combustion stability and proper control over CA50 compared to a highly premixed combustion. Since a higher percentage of the fuel is injected in the late injection, the observed effect of SoE2 over the SoC and CA50 become less influential compared to the medium/high load cases, as confirmed by Figure 2.c, while the effect of the late injection (not presented in this investigation) starts to gain relevance.