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Benajes Calvo, JV.; Martín Díaz, J.; Novella Rosa, R.; Thein, KJL. (2016). Understanding the performance of the multiple injection gasoline partially premixed combustion concept implemented in a 2-Stroke high speed direct injection compression ignition engine. Applied Energy. 161:465-475. doi:10.1016/j.apenergy.2015.10.034.

The final publication is available at

http://doi.org/10.1016/j.apenergy.2015.10.034

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Additional Information

Understanding the performance of the multiple injection Gasoline PPC concept implemented in a 2-stroke HSDI diesel engine

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Abstract

The newly designed Partially Premixed Combustion (PPC) concept operating with high octane fuels like gasoline has confirmed the possibility to combine low NO_x and soot emissions keeping high indicated efficiencies, while offering a control over combustion profile and phasing through the injection settings. The potential of this PPC concept regarding pollutant control was experimentally evaluated using a commercial gasoline with Research Octane Number (RON) of 95 in a newly-designed 2-stroke poppet valves automotive compression ignition (CI) engine. Previous experimental results confirmed how the wide control of the cylinder gas temperature provided by the air management settings brings the possibility to achieve stable gasoline PPC combustion at low and medium speed conditions (1250-2000 rpm) for the whole load range (3.1-10.4 bar IMEP) with good combustion stability (CoV (IMEP) below 3%), high combustion efficiency (over 97%), and low NO_x/soot levels.

In this context, present research focuses on the two main specific drawbacks of this concept. Firstly, the high BSFC resulting from the work required by the mechanical supercharger to sustain the needs in air management since the turbocharging system is not sufficient at low speeds. Secondly the high level of noise generated by the combustion process is known as a matter regarding customers' expectations, especially at high loads. Therefore, a dedicated analysis has been carried out to fully exploit the benefits of the gasoline PPC concept combined with the innovative 2-stroke engine architecture with the aim of identify and break the most relevant trade-offs.

Keywords:
Compression ignition engine, 2-stroke engine, Gasoline PPC concept, Emission control, Engine efficiency

April 30, 2015

Nomenclature

aTDC	after Top Dead Center
$(A/F)_{st}$	Stochiometric Air to Fuel ration
CA10	Crank Angle for 10% of fuel burnt
CA50	Crank Angle for 50% of fuel burnt
CAD	Crank Angle Degree
CD	Combustion Duration
CDC	Conventional Diesel Combustion
CI	Compression Ignition
ΔP	Pressure difference between intake and exhaust ports
dP/dα_{max}	Maximum pressure gradient
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Opening (angle)
HCCI	Homogeneous Charge Compression Ignition
HSDI	High Speed Direct Injection
IGR	Internal Gas Recirculation
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing (angle)
ISFC	Indicated Specific Fuel Consumption
ISFC$_{corr}$	Corrected ISFC considering energy consumption of the air loop devices (turbocharger and supercharger)
MT	Mixing time
PCCI	Premixed Charge Compression Ignition
P$_{in}$ / P$_{ex}$	Intake/Exhaust Pressure
P$_{max}$	Maximum cylinder pressure
PPC	Partially Premixed Combustion
P$_{rail}$	Injection rail pressure
Φ_{eff}	In-cylinder effective equivalence ratio
RoHR	Rate of Heat Release
SoC	Start of Combustion
SoE	Start of Energizing (injector signal)
SoI	Start of Injection
CoV (IMEP)	Coefficient of variation of indicated mean effective pressure
T$_{in}$ / T$_{ex}$	Intake/Exhaust Temperature
T$_{IVC}$	Mean gas temperature at intake valve closing
TDC	Top Dead Center
TR	Trapping Ratio
VVT	Variable Valve Timing
VVT$_{in}$ / VVT$_{ex}$	Intake/Exhaust Variable Valve Timing
η_{comb}	Combustion efficiency
$\eta_{indicated}$	Indicated efficiency

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1. Introduction

A relatively new combustion process has been developed for the few last past years to operate compression-ignition (CI) engines, as an alternative between fully premixed combustion (typically, HCCI and PCCI) and fully diffusive combustion (classic Diesel combustion). The Partially Premixed Combustion (PPC) concept allows to keep the pollutants emissions at low levels while retaining control over the combustion timing with the injection event. The injection process is advanced towards the compression stroke to be detached from the combustion event, enabling partial mixing of the mixture to avoid over-rich regions where soot is formed, while NO_x emissions are reduced by the introduction of large amounts of EGR allowing to lower the combustion temperatures [1]. It has been confirmed than both NO_x and soot emissions can be simultaneously reduced by the combustion of diesel fuel in a sufficiently premixed cylinder charge, such as HCCI or PCCI premixed combustion strategies [2–5]. But, due to an -often- unavoidable over-mixed mixture and a liquid fuel impinging onto the walls, the HC and CO emission are sharply increased. The main advantage of the PPC concept is its proclivity to present a high indicated efficiency, while still removing the trade-off between the NO_x and the soot emissions, as observed in CDC. Indeed, it has been observed that the combustion at constant-volume of a highly premixed mixture helps to reduce heat transfer during the compression and get an efficient expansion [6, 7].

The operating region of this strategy is however restricted in terms of load range, due to an only indirect control over the combustion process, as the injection events are detached from the combustion. This is needed to get premixed conditions, but the combustion profile is then very difficult to manage properly, and depends mostly on the chamber local conditions. Thus, the suitable range between misfire and knocking-like combustion is very reduced, with a high sensitivity, and the premixed conditions generate a sharp combustion. In order to soften it, a multiple injection strategy has shown clear improvements [8], as well as a fine optimization of the injector nozzle/bowl design matching [9].

Another main issue that appears with this PPC strategy, also limiting the load range, is the RON/load matching. Due to either the strong propensity of auto-igniting of a high cetane fuel along the compression stroke (suitable combustion at low-load, but hard knock at high load), or the low-reactivity of the low cetane fuel, allowing a good combustion at high load, but deteriorating the combustion at low load, or even reaching misfire, it is then almost impossible to reach wide load operating range [10–14]. Previous researches confirmed how the proposed 2-stroke architecture can be a suitable solution for extending this range. Indeed, it allows a higher in-cylinder temperature at IVC thanks to less cooling time, which helps the ignition of a high octane fuel such as gasoline even at low load, providing control on the cylinder conditions and the combustion environment, thus on efficiency and final emissions levels in Conventional Diesel Combustion [15]. Then, a large fraction of cooled EGR can be introduced in order to be able to reach higher loads by reducing the sensibility of the mixture to slow down the chemical process and increase the time between the end of the injection and the start of combustion (SoC), as it has been observed during investigations lead in Lund University [16].

But a really high intake pressure is required to reach both the need in EGR to control the combustion, and the high A/F ratio needed to get a proper combustion. This has dramatic repercussions on the global efficiency of the engine, even considering the high indicated efficiency provided by the PPC concept and demonstrated by the Eindhoven University of Technology (more than 50% in the best conditions) [17]. Indeed, a volumetric compressor is needed to reach these conditions (especially at low load), which punished the BSFC (in the following study, the $\mathrm{ISFC_{corr}}$ represents the ISFC corrected to take these work demands in account). A really clever air loop strategy is then needed to get sustainable conditions, taking also in account the exhaust acoustics to help the scavenging process.

In this framework, previous researches lead by the authors on this newly designed 2-stroke engine combined with a multiple injection strategy confirmed the possibility to extend the load range keeping the same RON gasoline. The pollutant emissions are manageable, and the NO_x / soot trade-off can be controlled and even removed within a wide range of operating conditions. Nevertheless, new trade-offs have been observed between the NO_x/noise levels and the combustion efficiency. NO_x and noise level are linked, as a sharp -and noisy- combustion is the main source of the NO_x generation [18].

As mentioned, one of the major matter of the combustion concept is the sharpness of the heat release rate profile at high load, generating high noise levels, which is a critical aspect for a engine designed for commercial application. The highly premixed condition inside the cylinder results in a very reactive mixture, igniting at quasi-constant volume, arising to a very quick and sharp, knocking-like combustion [19, 20]. Moreover, it may eventually physically damage the engine.

As it has been previously observed, the combustion phasing and the emissions levels are controlled by the injection settings, whereas the air management influences only the performances [8, 15, 21–23]. The following study will then present the different strategies that were explored and defined to get proper air management settings, to determine improvement paths for the global efficiency. Then, a study lead on the noise reduction will be exposed, through a numerical simulation correlated to experiments.

2. Experimental Setup

Engine architecture and test cell characteristics

The engine used along the experimental activities is a single-cylinder research version of an innovative Renault concept consisting of a 2-cylinder DOHC 2-stroke HSDI CI engine with scavenge loop.

The engine is designed to operate with four poppet valves, two intake and two exhaust, driven by a double-overhead camshafts. A specific staggered roof cylinder head geometry has been developed to enhance the 2-stroke scavenge process by 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. 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 [24, 25].

The DOHC are driven by an hydraulic VVT (Variable Valve Timing) system that allows delaying independently intake and exhaust valve timings with a cam phasing authority of +30 degrees from base timing, as it was detailed in previous investigations [22, 25, 26]. The key valve timing angles (EVO/EVC/IVO/IVC) used along the researches are defined at those CAD where the given valve lift was 0.3 mm.

This research engine version has been manufactured by Danielson Engineering and as a reference, Table 1 contains its main geometrical characteristics.

Table 1: Main engine specifications

Engine Type	2-stroke compression ignition
Displacement	365 cm^3 (single cylinder)
Bore x Stroke	76 mm x 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_{in}=0)	IVO=161.9 CAD aTDC IVC=251.6 CAD aTDC
Nominal exhaust valve timing (set at VVT_{ex}=0)	EVO=122.6 CAD aTDC EVC=226.9 CAD aTDC
Fuel injection system	Diesel common rail HSDI
Injector nozzle	148°AN, 8 holes, 90 μm

The engine configuration has been kept since the investigations operating with the CDC concept, so it consists of a conventional diesel piston with geometric compression ratio of 17.8 and wide angle injector nozzle fully optimized for the CDC concept. The injection system is a common rail HSDI designed for injecting diesel fuel up to a maximum rail pressure of 1800 bar. The injector nozzle is composed of 8 holes with a diameter of 90 μm, while the included angle is equal to 148°.

Mass flow rate and spray momentum flux measurements have been performed in a dedicated test rig following the methodology described in [9, 27] to compare commercial diesel fuel (used the previous investigations) with the selected gasoline fuel and to determine the limit conditions that can be reached using gasoline in such an injection system. The maximum injection pressure was then limited to 1200 bar, to assure proper functioning of the injection system. Besides, a lubricity additive was added to the calibrated unleaded RON95 gasoline selected for this research. Most important fuel properties are detailed in Table 2.

Table 2: Fuel properties

Test fuel	Unleaded gasoline with lubricity additive
Research Octane Number	94.6
H/C ratio	1.76 mol/mol
O/C ratio	0 mol/mol
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 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 [15, 22, 25, 28].

The auxiliary test cell equipment includes independent water and oil cooling circuits, while the air management is assured by 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. A state-of-the-art gas analyzer is used to measure the most relevant exhaust gas species (O_2 , CO, CO_2 , HC, NO_x , N_2O) as well as the EGR rate. Soot emissions are calculated from the

Smoke, which is measured by a Smokemeter in filter smoke number (FSN) units. Additionally, the trapping ratio -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)- is experimentally measured through a gas tracer method [29, 30] using methane as external gas. The fraction of residual gases retained from the previous combustion cycle in the total trapped mass in the cylinder is called Internal Gas Recirculation (IGR) ratio. Its value, the total trapped mass at IVC and in-cylinder effective equivalence ratio (Φ_{eff}) are estimated in each operating condition using simplified thermodynamic calculations.

A piezoelectric sensor is used to measure the instantaneous cylinder pressure as a relative signal with a resolution of 0.2 CAD, while a different piezoresistive sensor -measuring the absolute pressure- is placed at the cylinder liner close to the bottom dead center to reference the piezoelectric sensor signal at every revolution, with the same frequency. Main global combustion parameters like indicated mean effective pressure (IMEP), peak cylinder pressure (P_{max}), maximum pressure gradient ($dP/d\alpha_{max}$) and combustion stability indicators (CoV IMEP and CoV P_{max}) are directly derived from the analysis of the cylinder pressure signal through quick calculations directly on the test bench. Then, the data post-treatment is performed by an in-house combustion analysis software (CALMEC) [31], resolving the first law of thermodynamics and obtaining the instantaneous evolution of the energy released by the progress of combustion from the measured pressure signal. It can provide a Rate of Heat Release (RoHR) profile, using sub-models for considering heat transfer losses, mechanical deformation of the cylinder and blow-by losses. From this combustion profile, many parameters can be extracted, such as the start of combustion (SoC), ignition delay, combustion angles (CA10, CA50, CA90) and mixing times.

An additional in-house engine simulation software (SICICLO) was used to obtain optimization paths. It is based on the same physical laws as CALMEC but in a reversed way. Then, it generates pressure signals and all the associated data such as noise, IMEP, IFSC... from a given RoHR profile. An experimental reference profile is needed as a first step, then it can be modulated to get a desired combustion shape. Since the previous investigations highlighted the requested conditions to get a given combustion profile, the simulated “optimum” profile is then attempted to be transposed experimentally on the engine to be able to evaluate the real effects.

Finally, combustion noise has been calculated following the classical approach introduced by Austen and Priede [32]. The classical approach is frequently used by engine development engineers to assess the overall engine combustion noise level at steady operating conditions [33, 34]. 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.

Siciclo description

The theoretical part of the study concerning the relation between the combustion profile and the engine efficiency and noise was achieved using an in-house software named SICICLO, which is a 0D single-zone thermodynamic model. From the input data (in our case experimentally obtained RoHR profiles) and by taking into account the heat transfer to the chamber walls, the blow-by leakage, the fuel injection and engine deformations, along with the instantaneous change in gas properties, it can solve the mass and energy conservation equations in order to obtain the instantaneous gas state in the combustion chamber (pressure, temperature...) [35, 36].

As complementary outputs, it can provide the indicated efficiency, the IMEP, the pressure gradient derived from the (theoretical) pressure signal generated, and the noise resulting from these conditions. Then, it is only a matter of reproducing these conditions in the real combustion chamber, which is now a well known combination between air management and injection settings.

3. Methodology

The research work is based on several operating points defining a preliminary engine's map in the most representative area for the NEDC driving cycle: low-to-medium engine speed (1200/1500 rpm) and almost the full load range (3.1/5.5/10.4 bar IMEP). Part of this study has been shown in previous author's publications [8, 23], and here will be presented one particular point, which allows various approaches: 10.4 bar IMEP at 1250 rpm. The objective of this study being the observation of the behavior of the combustion according to parameters, this work was carried out without in-depth optimization of the engine hardware or settings. Indeed, all the engine hardware was kept the same as the previous researches performed with the CDC concept [15, 25, 28]: same injection system and same combustion chamber design. The air management settings were also selected using mathematical models of several engine responses, previously obtained through dedicated Design of Experiment (DoE) methodology with the current engine hardware at similar operating points [15, 22]. This allows us to get fast and easy preliminary results that can be directly compared with the CDC ones in terms of engine efficiency (fuel consumption) and emissions.

Oil and coolant temperatures were kept constant at 90°C while intake air temperature was carefully controlled and kept constant during all tests by using a heater. The injection timing is referred to the Start of Energizing (SoE) current of the injector instead of the actual Start of Injection (SoI), which happens a few degrees ($\approx 150~\mu$s, i.e. 1.5 to 2 CAD) after the SoE due to the hydraulic delay affecting the needle lift.

A triple injection strategy 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 this high load point, this strategy is known for helping in achieving the load target while avoiding/mitigating knock tendency. The total injected quantity was kept constant for all tests along the different studies. Then, in each study the timing of the 2nd or the 3rd injections were swept each 2 CAD in a range defined considering the onset of knocking combustion or smoke limit and the deterioration of combustion stability as the main constraints.

The first part of the work presented here will be based on the 2nd and 3rd injection timings in order to understand better the trends previously reported by the authors [23]. Their individual effects are well described at other loads and speeds, but this particular study helps to improve the knowledge on how to control the combustion process, and more particularly the pollutants emissions, and then how to combine it with other strategies.

The second part of the research was focused on the RoHR profile analysis with the objective of understanding the paths for noise reductions (through the pressure gradient dP/dα_{max}), and see the impact on efficiency and emissions. This was carried out using an in-house simulation software (*Siciclo*), which allows numerical redefinitions of the RoHR profile by changing either its shape (height and length) or its timing, keeping the same total heat release. Then, knowing how to control the RoHR profile with the injection settings, the trends tried to be recovered experimentally, mainly by changing the fuel distribution.

During the last part of the research work, a quick study was dedicated to air management, and more particularly to the exhaust pressure (and pressure drop). This came as an answer to a very low trapping ratio at this working point, due to high pressure drop and low speed, resulting in a lot of wasted compressed air, and so a very high ISFC$_{corr}$.

The most relevant engine settings chosen for each operating condition are detailed below in Table 3.

4. Results

Effects of the injection timing

Due to their relevance, the effects of the timing of the different injection events have been intensively investigated and reported by the authors for various load and speed conditions [23]. Therefore, this section focuses on validating the main trends observed at this high load / low speed condition using different ranges than those selected in previous research works, especially identifying the key trade-offs generated by this PPC concept.

The air management settings for this operating condition were selected from DOEs carried out in previous investigations at equivalent conditions [15, 22], but operating with the CDC concept. It concerns the valves settings and the intake and exhaust pressures, and also the EGR rate. The resulting parameters are not affected by the injection timings, and a constant trapping ratio was obtained during all the study (60%), as well as the IGR rate (36%) or the temperature at IVC ($\approx$ 178˚C). The oxygen mass concentration (at the IVC and EVO) also remained at stable levels of 11% and 2% respectively.

Results confirm how the combustion phasing is controlled mainly by the injection events, and more particularly the 2nd injection [23], while the 3rd injection has much less influence. Indeed, it is evident that at this conditions an early main injection generates a late Start of Combustion (SoC), and at the contrary delaying the injection event advances its onset. This have been explained in

Table 3: Main injection study at 10.4 bar IMEP: Engine settings

a) SoE2 study - 10.4 bar IMEP / 1250 rpm						
Parameters	T_{in} [˚C]	$VVT_{(in,ex)}$ [CAD]	*Olap* [CAD]	*EGR* [%]	P_{rail} [bar]	m_{fuel} [mg/st]
All Tests	45	(5,20)	78.4	43.5	850	19.8
Parameters	P_{in} [bar]	ΔP [bar]	*SoE1* [CAD]	*SoE2* [CAD]	*SoE3* [CAD]	$\%_{fuel}$ [%]
Studies	2.755	**0.600**	-60	**-46/-38**	-2	17/66/17
	2.755	**0.725**	-60	**-42/-36**	-2	17/66/17
	2.755	0.600	-60	**-46/-38**	-2	**17/66/17**
	2.755	0.600	-60	**-38/-34**	-2	**17/60/23**

b) SoE3 study - 10.4 bar IMEP / 1250 rpm						
Parameters	T_{in} [˚C]	$VVT_{(in,ex)}$ [CAD]	*Olap* [CAD]	*EGR* [%]	P_{rail} [bar]	m_{fuel} [mg/st]
All Tests	45	(5,20)	78.4	43.5	850	19.8
Parameters	P_{in} [bar]	ΔP [bar]	*SoE1* [CAD]	*SoE2* [CAD]	*SoE3* [CAD]	$\%_{fuel}$ [%]
Studies	2.755	**0.600**	-60	-40	**-4/0**	17/66/17
	2.755	**0.725**	-60	-40	**-6/0**	17/66/17
	2.755	0.600	-60	-36	**-6/0**	**17/66/17**
	2.755	0.600	-60	-36	**-8/-2**	**17/60/23**

details by CFD calculation, and presented in the THIESEL conference of 2014 [22], showing how the homogeneity of the mixture (and so the local richness) influences its reactivity, and then determines the combustion timing.

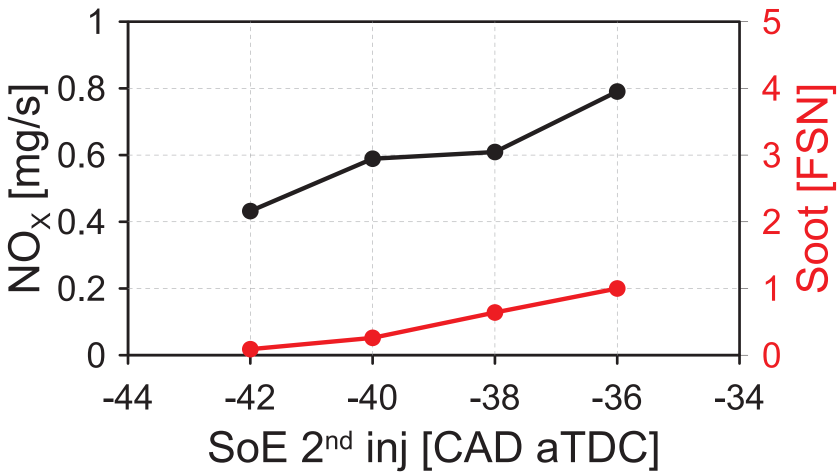

Figure 1: NO_x & Soot emissions

The 2nd injection timing also affects NO_x and soot emissions, and noise level. The combustion timing strongly impacts the pressure evolution so an early onset attained by retarding the 2nd injection timing trends to generate a constant volume combustion with high P_{max} and $dP/d\alpha$, which is directly linked to an increment in noise level. As a result of the high P_{max}, the local temperature of the reaction zones increases enhancing the NO_x generation along the combustion process. However, the NO_x level is still acceptable (below 0.8 mg/s at most), as shows *Fig. 1*, but is doubled within an injection timing range of only 6 CAD. Additionally, the same trend is observed for soot emissions due to the advanced combustion onset linked to the late 2nd injection, resulting in a reduced mixing time for the 3rd event and the generation of a diffusion-like combustion [22]. This last point is easily manageable by advancing the 3rd injection to extend the mixing time. But, as shown by *Fig. 2*, a too early 3rd injection generates a too premixed mixture and a sharper RoHR, increasing NO_x emissions (and pressure gradient and noise level). This last injection's timing does not influence any other parameter (almost constant values), and thus is an easy lever to tune finely the combustion process without interacting with the other settings (no additional trade-offs observed).

Therefore, since there is no trade-off between NO_x and soot emissions for the 2rd injection timing but it exists for the 3rd injection timing, the well-known NO_x / soot trade-off intrinsically arising operating with the CDC concept is potentially solved even in high load conditions by implementing this PPC concept with a suitable injection pattern, while at lower loads it is even possible to reach a zero-NO_x / zero-soot combustion [8, 21, 23]. As a negative counterpart, these results highlight the extremely high sensitivity of the PPC concept to the injection pattern that controls the in-cylinder local conditions in which the combustion process starts and develops.

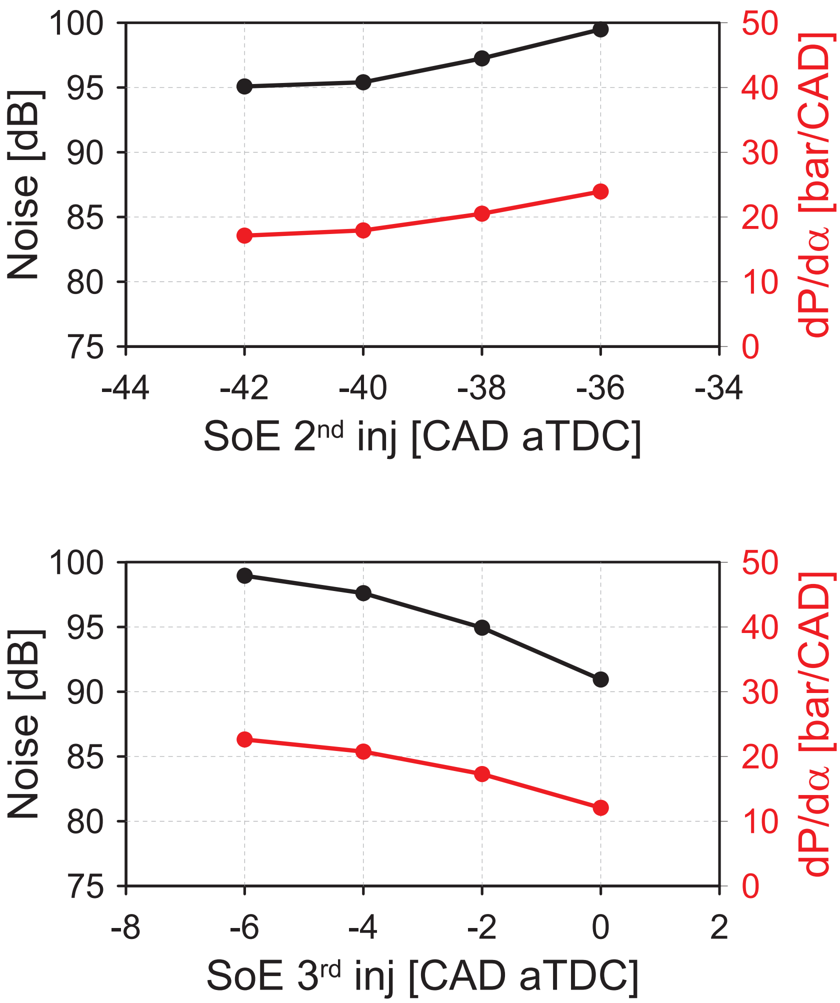

Figure 2: Noise level

The HC and CO emissions are also controlled mainly by the 2nd injection timing. Indeed, advancing this event puts a larger mass of fuel inside the squish and eventually promotes the liquid fuel impingement onto the cold combustion chamber walls. Thus, a large part of the fuel trapped in the squish is hardly reached by the diffusion flame during the combustion process and then it can not burn properly, increasing the HC and CO emissions. This is reflected by the combustion efficiency in *Fig. 3*. This has direct impact on the indicated efficiency (ISFC) and proportionally on $ISFC_{corr}$ as the air management remains constant. However, besides the combustion efficiency fluctuation generated, the effect of the injection timings on these parameters are low (less than 5% on the ISFC).

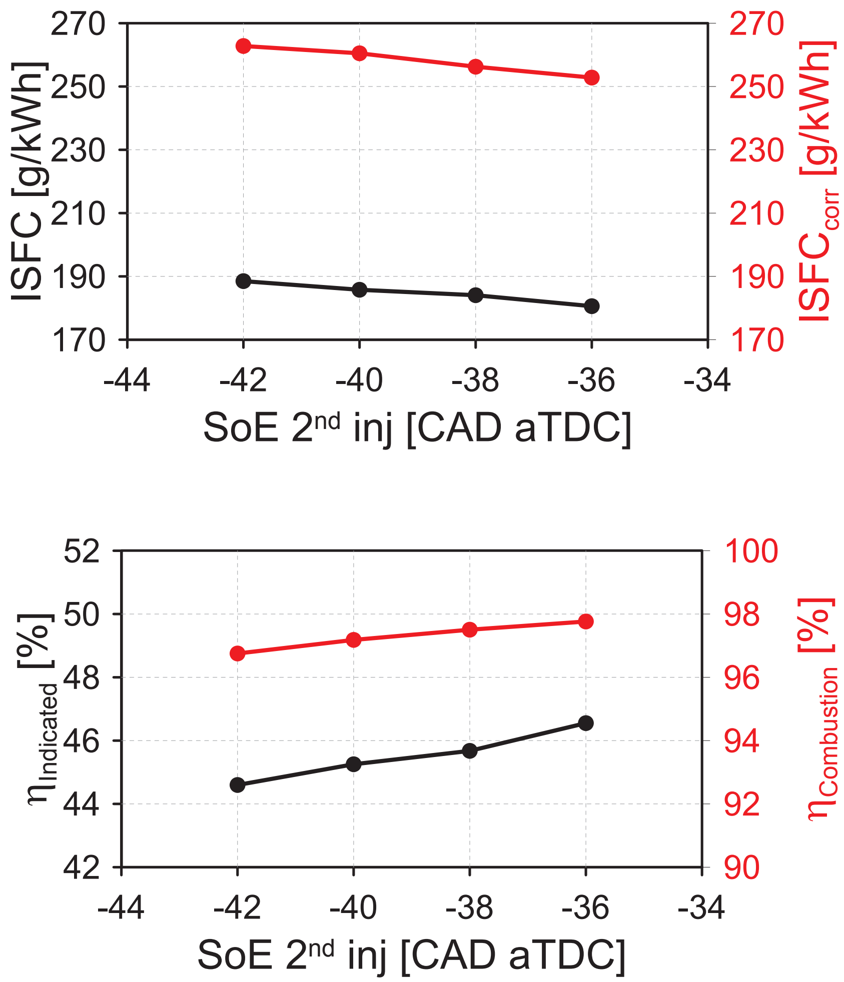

Figure 3: ISFC & Key engine efficiencies

Nevertheless, the first plot of *Fig. 3* shows a very high ISFC$_{corr}$ level. This operating point requires a high air mass flow, but the low speed of the engine does not allow the turbine to be efficient enough to provide this needs. Thus, this load has to come from the volumetric compressor, which costs a lot in terms of energy and worsens the final ISFC$_{corr}$ level. Hence, the air provided by the volumetric compressor has to be efficiently used, with the least wasted mass as possible, so the TR needs to be high. A straight solution consists of decreasing the pressure drop across the engine, between the intake and the exhaust (ΔP) manifolds, by increasing the exhaust pressure to avoid short circuit. This strategy's drawback is a possibly punished scavenge, so it should be analyzed in detail.

Strategy for improving ISFC$_{corr}$

Experiments were then carried out to observe the impact of decreasing the pressure drop across the engine, between the intake and the exhaust (ΔP) manifolds on ISFC and ISFC$_{corr}$. The reference value of ΔP was set at 0.725 bar (see previous section), then it was decreased down to 0.600 bar. The impact on TR was significant, as well as on ISFC$_{corr}$, as shown on *Fig. 4* and *Fig. 5*.

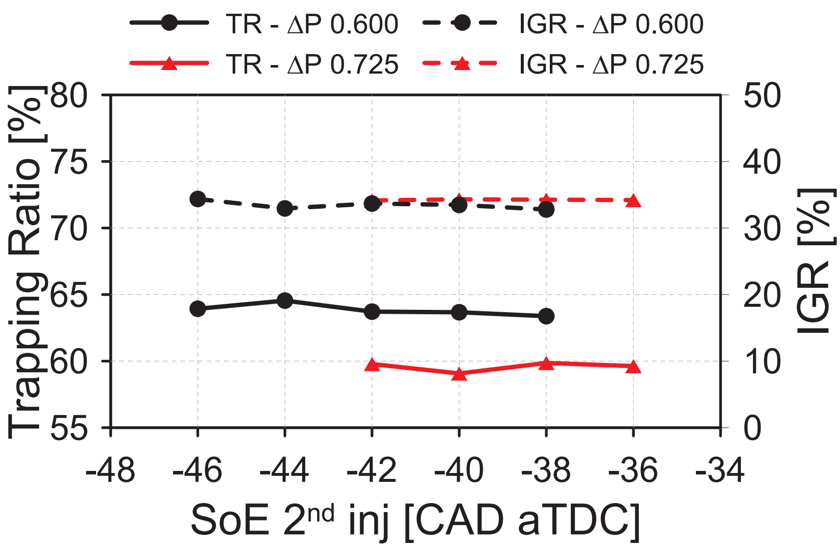

Figure 4: Trapping Ratio & IGR

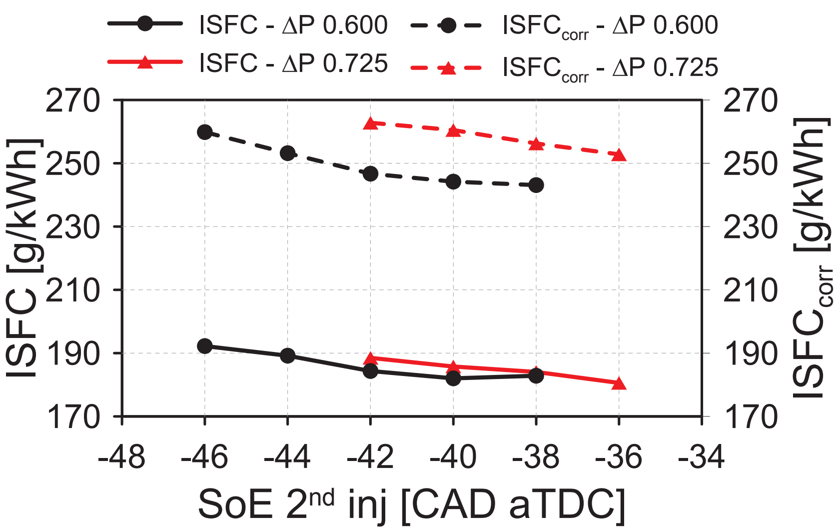

Figure 5: ISFC & ISFC$_{corr}$

According to previous research activities performed by the authors focused on air management optimization, the best TR considering the trade-off between ISFC and ISFC$_{corr}$ is around 70-75%. Increasing TR from the previous 60 to 65% provides important benefits, and the ISFC$_{corr}$ is optimized by 10 to 15 g/kWh. This improvement is even more significant considering that the impact on the other parameters is very low. The combustion phasing and profile are slightly advanced, without damaging stability. This earlier onset explains the main drawback of this strategy: a higher noise level, by 1 to 2 dB. These effects were predictable, as a reduced short circuit results in a higher IGR level in terms of mass (same proportion in a higher total mass), implying an increased T_{IVC}, and so a more reactive mixture.

The pollutant emissions are also barely punished (slightly higher soot level), the CO and HC emissions are even improved (Fig. 6). This was also intended, as a higher trapped mass results in a higher density, so a lower spray penetration, putting less fuel into the squish. Additionally, the higher temperature along the closed cycle caused by the increment in T_{IVC} and the earlier onset of combustion, which is shifted towards the TDC, also helps to promote the conversion of CO into CO_2.

Nevertheless, even if the gains are not negligible, this ISFC$_{corr}$ level is still high and needs even

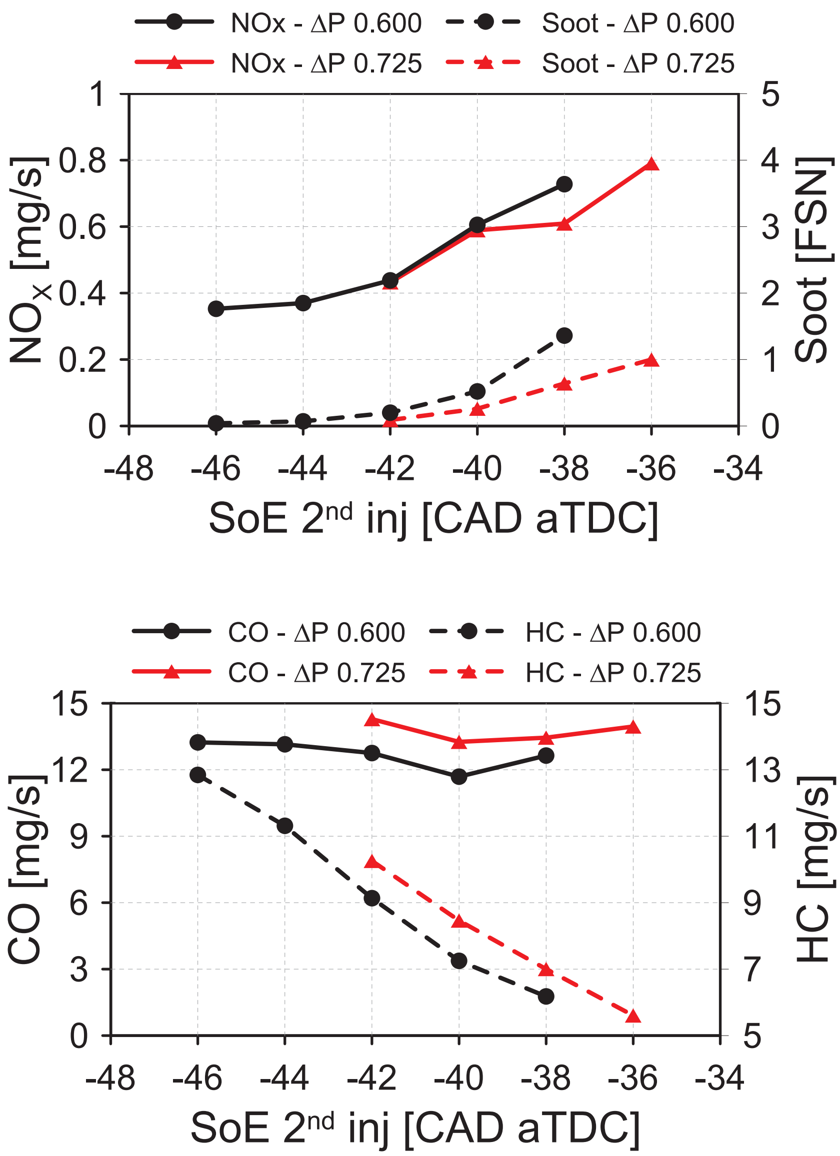

Figure 6: Pollutants

more improvement. A ΔP of 0.500 bar was tested, but the stability was appreciably damaged at this point, and the combustion was too delayed (close to misfire) due to the deteriorated in-cylinder thermo-chemical conditions resulting from the very wrong scavenge (too much IGR together with very low fresh air flow rate). Then, the results can not be fairly compared in this study. However, it would be interesting to explore paths to increase TR even more without punishing the scavenge process.

Optimization of the RoHR profile to control the noise level

It has been proven in a previous section how the injection timing (2nd and 3rd event) can easily control the combustion phasing and emissions, without influencing the indicated efficiency or the air management parameters (hence, no impact on the previous ISFC$_{corr}$ study). From these conclusions, a study focused on the combustion profile (RoHR) has been carried out to define paths to control noise, theoretically as a first approach. This was performed using the in-house simulation software (*Siciclo*), which allows to manipulate numerically the RoHR by changing either its shape (height and length) or its timing, keeping the same total heat release and then the total injected fuel since this theoretical study keeps constant the combustion efficiency. However, a reference experimental RoHR profile is requested, so the baseline test used for all the following simulations was set as described in *Table 4* and the reference RoHR profile is shown in *Fig. 7*.

Table 4: Reference point: Engine settings

n	*IMEP*	P_{in}	ΔP	T_{in}	*VVT*$_{(in,ex)}$	*Olap*
[rpm]	*[bar]*	*[bar]*	*[bar]*	*[˚C]*	*[CAD]*	*[CAD]*
1250	10.4	2.755	0.600	45	(5,20)	78.4
EGR	m_{fuel}	P_{rail}	*SoE1*	*SoE2*	*SoE3*	$\%_{fuel}$
[%]	*[mg/st]*	*[bar]*	*[CAD]*	*[CAD]*	*[CAD]*	*[%]*
43.5	19.8	850	-60	-40	-2	17/66/17

The two paths investigated here were focused on:

- **The RoHR shape:** extending the combustion duration (CA10 $\rightarrow$ CA90) from 100 to 200%, by 20% steps (100% being the reference). In order to keep the same total heat release (constant integral), the height of the profile is accordingly modified.

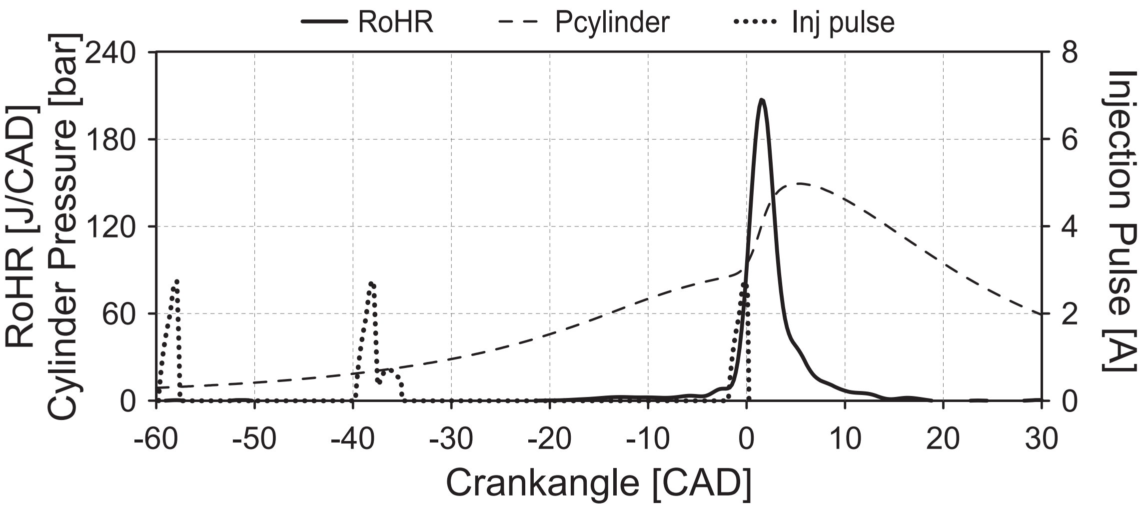

Figure 7: Reference RoHR

- **The combustion phasing:** moving the RoHR profile from -5 to $+5$ CAD from the reference, by 2.5 CAD steps (0 being the reference).

Table 5: Reference point: Main characteristics

SoC	*CD*	η_{ind}	η_{comb}
[CAD]	*[CAD]*	*[%]*	*[%]*
-1.7	8.0	46.18	97.54

As discussed, the efficiency is hardly affected by changing the combustion profile or moving its onset (within a given but quite large range). The simulation is able to reproduce very similar results, as shown in *Fig. 8*. Despite the very different RoHR profiles considered in the present study, the efficiency keeps within a restricted range of less than 2 points (from 45 to 47%). In practice, this range is even more restricted due to the impossibility for the engine to reproduce some of these RoHR profiles: a too early SoC leads to hard knocking conditions, and a diffusion-like combustion is required too get a very wide RoHR profile, which generates too much soot.

It seems essential to note that the combustion efficiency is not taken in account in this calculation. Its experimentally observed variation is quite small (but not negligible), and affects directly the indicated efficiency (in fact, only partially, the other part being mixed with the heat losses). This effect is neglected in the simulations, so the results provided by the model are slightly optimistic and they will be validated with experimental data.

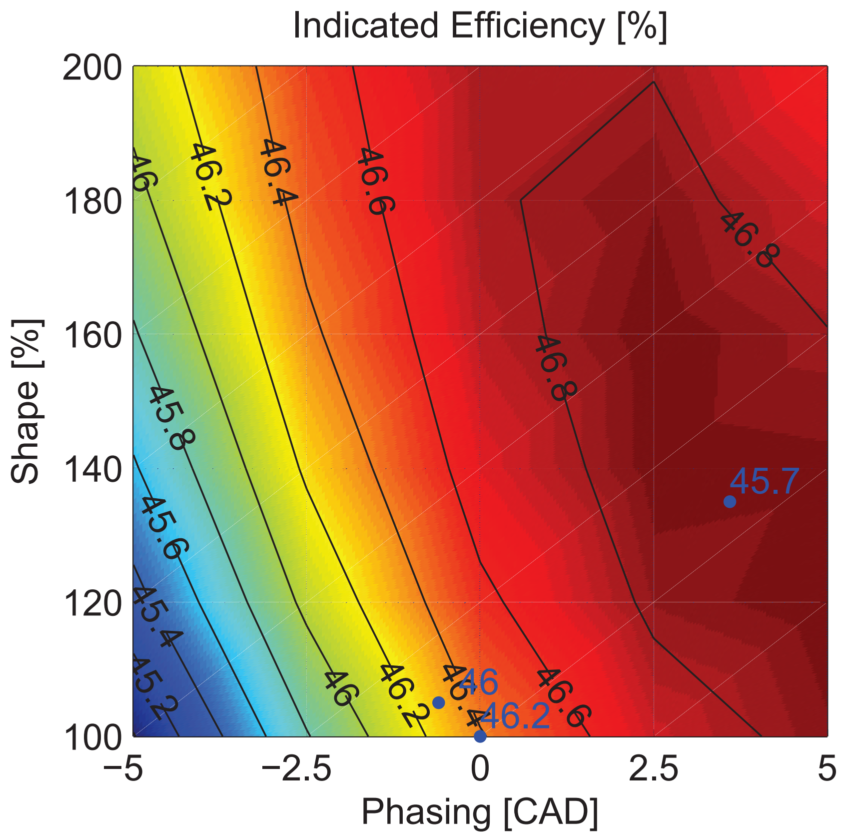

Figure 8: Indicated efficiency

It is noticeable how the indicated efficiency is directly linked to the heat losses, comparing *Fig. 8* and *9*. Indeed, except for late and wide RoHR profile where expansion work clearly decreases, the heat transfer is the only factor that affects the efficiency (aside from the combustion efficiency itself, as previously seen). Moreover, the level of these losses is quite critical at this operating condition since about 15% of the total fuel energy is lost directly by heat exchanges from the combustion chamber.

The other main exploration allowed by the simulation is the noise reduction path, which is a

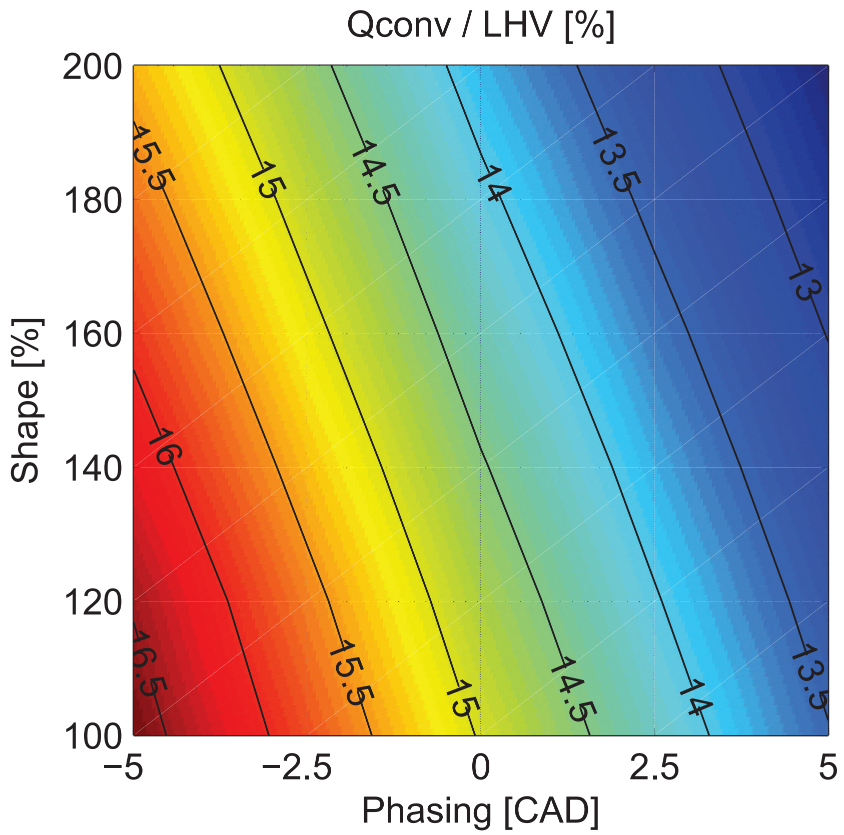

Figure 9: Proportion of energy release compared to introduced energy

real matter for the user's comfort. The noise level is directly linked to the pressure gradient during the combustion process, which is mainly controlled by the shape of the RoHR profile, more than by the combustion phasing (at least inside the studied range). *Fig. 10* displays the pressure gradient variation calculated from the simulations, and points out the low levels that can be reached by smoothing the RoHR profile. Indeed, a sharp combustion developing at quasi-constant volume results in a very fast pressure rise, generating hard knock in the worst cases. However, extending the profile and decreasing the maximum RoHR peak is a key alternative to overcome this intrinsic drawback of the gasoline PPC concept.

It is observed how the RoHR profile shape has an important effect over the pressure gradient and noise level much less over the engine indicated efficiency, while the opposite is observed for the combustion phasing. The interesting correlation here is the lack of trade-off between noise

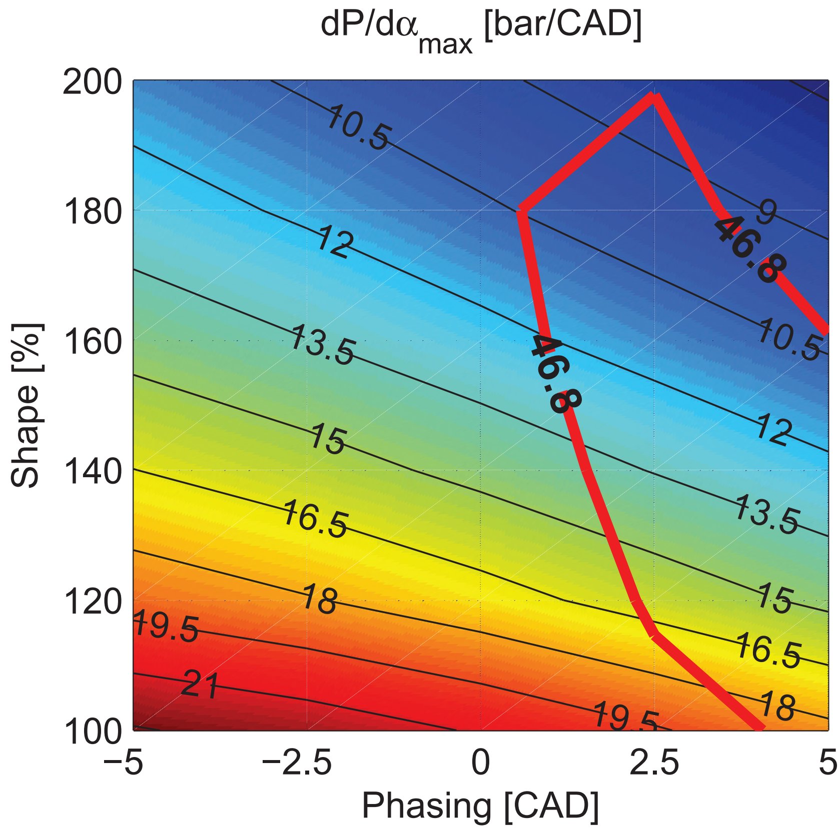

Figure 10: Maximum pressure gradient

level and indicated efficiency. However, there are two main restrictions to this strategy. First, generating a suitable combustion profile while keeping the premixed combustion process results in a strong deterioration of the combustion stability, even reaching the misfire limit, because the combustion must be shifted towards the expansion stroke to get a wide RoHR profile with a short maximum peak. Second, optimizing the combustion profile by switching to a diffusive-like combustion, with a wide RoHR profile with moderate maximum peak, impacts negatively the soot emissions and then the NO_x / soot trade-off is recovered. The objective then is to set up a strategy to generate an optimized combustion profile allowing to reach the low pressure gradient conditions, keeping an acceptable combustion stability and/or soot emission level.

Strategy for improving Noise/emissions/efficiency trade-offs

As previously defined, a multiple injection strategy is selected for this PPC concept [23], and the fuel quantity needed at high load requires a 3-events strategy. As the 2nd injection timing controls the combustion phasing, the 3rd injection close to TDC controls its profile and final soot emissions. But the fuel quantity injected during this 3rd event can also be managed in addition to the timing, in order to control the combustion profile to reduce noise going toward a diffusion-like combustion, but allowing to advance the SoI to increase the mixing time. These hypotheses were then tested on the engine, according to the parametric study presented in *Table 6*.

Fig. 11 shows the RoHR profiles obtained with the reference settings and also with those adjusted for performing this analysis of the impact of the fuel distribution between the 2nd and the 3rd injection events. The points selected here are based on the previous reference defined for the *Siciclo* study. The test in black represents this reference, the test in red has the 2nd injection event delayed by 2 CAD, and finally the point in blue keeps the same settings as the red one, but with a different fuel distribution. The key settings and results are shown in *Table 6*.

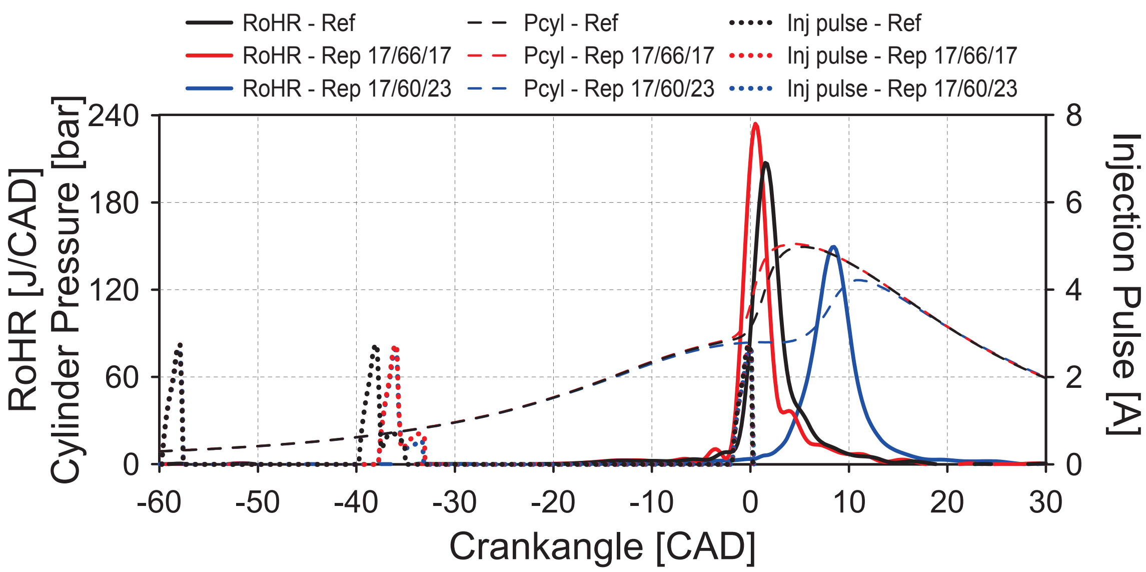

Figure 11: Fuel Distribution Study

The impact of the fuel distribution is critical to adapt the RoHR profile following the trends ob-

Table 6: Results of the fuel distribution comparison

	dP/dα [bar/CAD]	*Noise* [dB]	*NO_x* [mg/s]	*Smoke* [FSN]	*CO* [mg/s]	*HC* [mg/s]	η_{comb} [%]
SoE2 -40 - SoE3 -2 - 17/66/17 [1]	20.3	97.0	0.6	0.52	11.7	7.2	97.5
SoE2 -38 - SoE3 -2 - 17/66/17	23.8	99.2	0.7	1.36	12.6	6.2	97.8
SoE2 -38 - SoE3 -2 - 17/60/23	11.1	90.4	0.2	0.30	20.7	10.3	96.2
	SoC [CAD]	*CD* [CAD]	*$RoHR_{max}$* [J/CAD]	*ISFC* [g/kWh]	*$ISFC_{corr}$* [g/kWh]	η_{ind} [%]	*MT 3rd inj* [CAD]
SoE2 -40 - SoE3 -2 - 17/66/17 [1]	-1.7	8.0	207.3	182.0	244.2	46.2	4.3
SoE2 -38 - SoE3 -2 - 17/66/17	-2.3	8.4	234.2	182.9	243.1	46.0	5.0
SoE2 -38 - SoE3 -2 - 17/60/23	1.9	10.8	149.5	183.8	243.3	45.7	1.2

[1] *Siciclo reference point*

served with the previous theoretical study. Increasing the fuel quantity of the 3rd injection without moving its phasing helps to generate a smoother combustion process. Decreasing the fuel quantity injected early during the compression stroke (1st and 2nd injection events) decreases the reactivity of the mixture, delaying the SoC. The newly observed trade-off between noise and soot can then be broken. Looking at the results shown in *Table 6*, all output parameters are improved (noise is reduced to the target of 90 dB, soot and NO_x are both reduced too) or kept stable (efficiencies, ISCF / $ISFC_{corr}$). In general, only CO and HC levels increase.

These three points are represented in blue in *Fig. 8*. As discussed before, the indicated efficiency calculated through *Siciclo* does not take in account the combustion efficiency. Yet, in this last case (more fuel in the 3rd injection), it is worsened by 2%, which is approximately the difference observed between the estimation and the test result.

The main constraint of this strategy was observed during the experiments: the combustion stability. Indeed, the range for the injection timings (2nd and 3rd events) is really restricted between

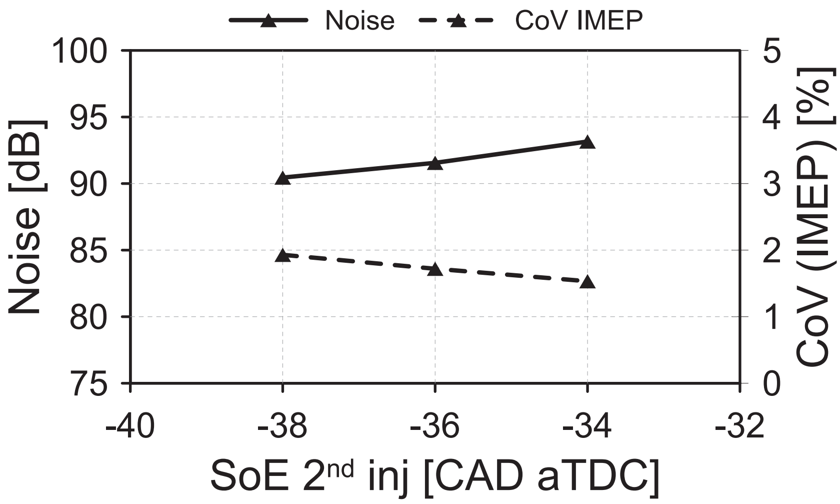

Figure 12: Noise level with more fuel in the 3rd injection

knock and misfire, and the sensibility to all other parameters (air management) increases exponentially. Thus, such a study has to be lead with a lot of care to determine stable settings, providing a suitable combustion process.

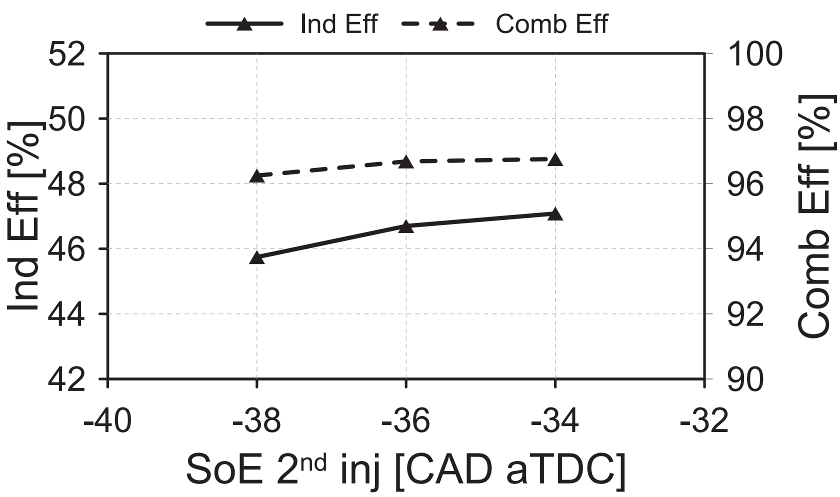

Figure 13: Efficiencies with more fuel in the 3rd injection

A short preliminary study has been conducted on the effect of the 2nd and 3rd injection timings to observe the sensibility and the trends obtained with this strategy (see *Table 3* on page 12). With this fuel repartition (17/60/23), the 2nd injection timing has now a small impact on almost all the parameters. At it is now well known, the air management is not affected along this kind of study. Its influence on the in-cylinder conditions is also limited: combustion phasing and maximum pressure are constant, while pressure gradient (noise) is only slightly affected (see *Fig. 12*). The stability is quite correct in the testing range, but falls drastically by advancing the injection event by 2 CAD more, to -40 CAD aTDC.

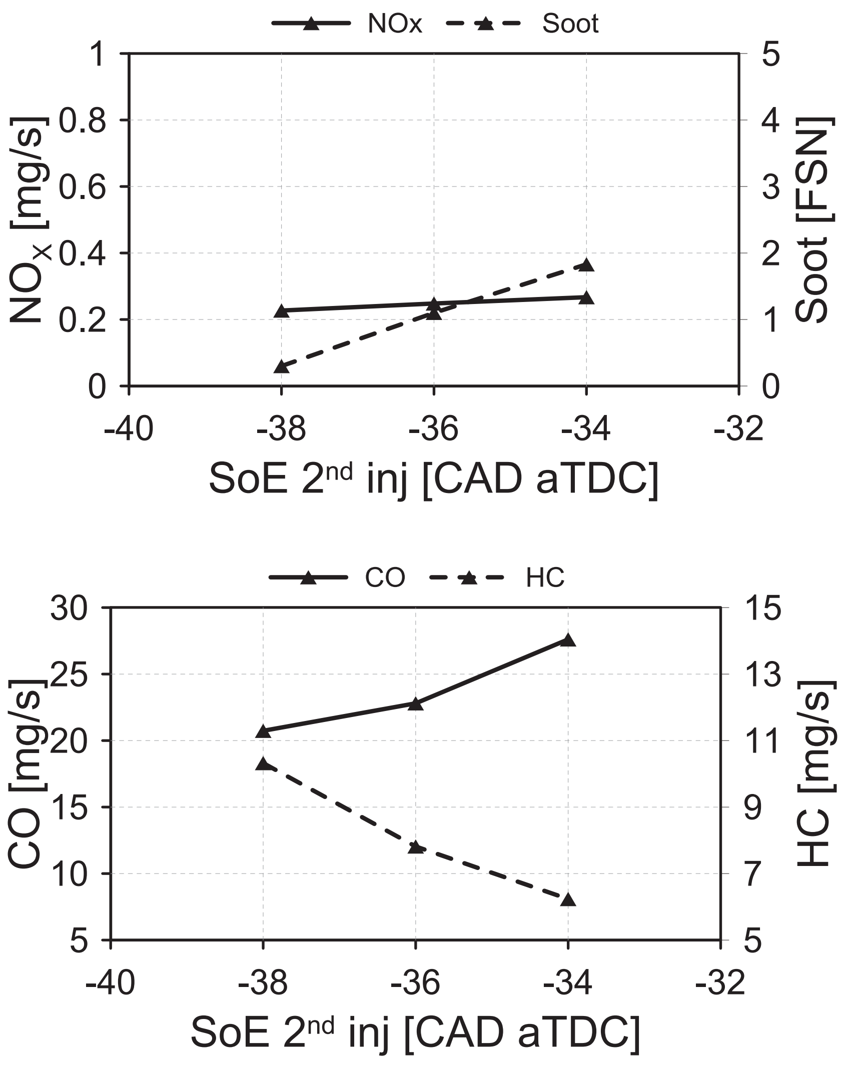

Figure 14: Pollutants level with more fuel in the 3rd injection

Additionally, the performance (consumption / efficiency) and emissions are both affected by the 2nd injection timing. As shown in *Fig. 13* and *Fig. 14*, the indicated efficiency increases by around 2% by delaying the 2nd injection, while CO and soot emissions also increase. But the HC production decreases by reducing the liquid fuel impingement onto the combustion chamber walls, helping to keep the combustion efficiency constant. However, this 2nd injection timing has no impact on the combustion profile or on its phasing (*Fig. 15*). It can then be set to control emissions, almost independently from the other parameters.

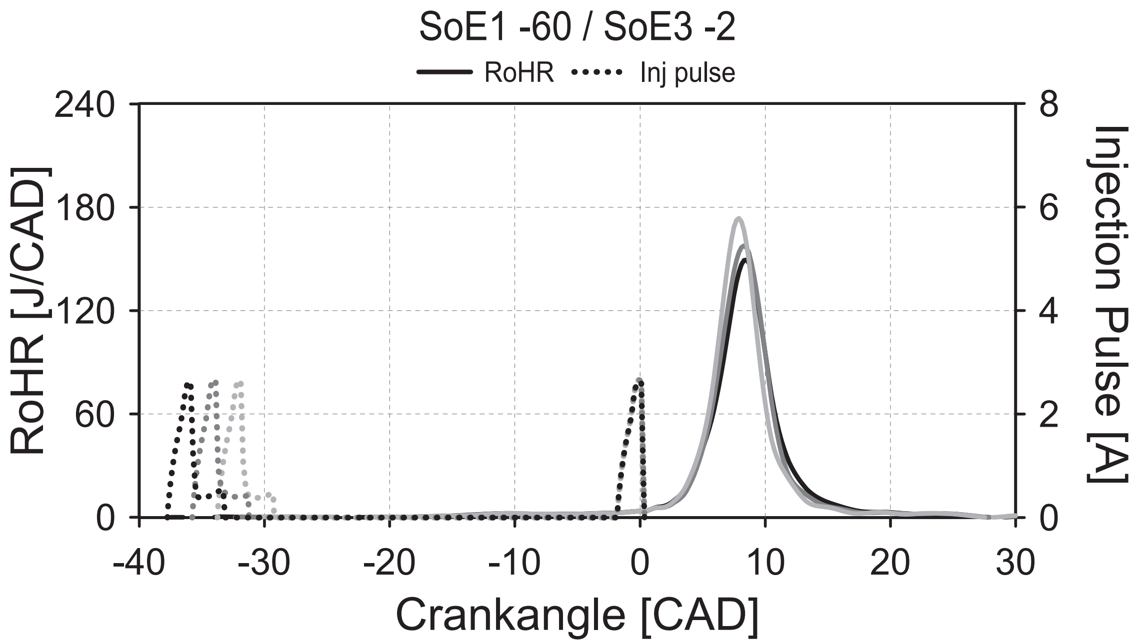

Figure 15: RoHR profiles with the new fuel distribution (2nd injection sweeping)

The 3rd injection timing still has no influence on the air management, but all the other parameters are sensible to its variation. When delaying this event, the maximum in-cylinder pressure, the pressure gradient and the noise (linked effects) drastically decrease. Indeed, as it was observed during the previous experiments and through the *Siciclo* simulations, it controls the combustion profile, and this effect is accentuated increasing the quantity of fuel injected during this last event. Thus, noise ranges from 95 to 85 dB within an injection variation range of only 6 CAD as confirmed by *Fig. 16*, while NO_x/soot levels are kept constant and HC/CO levels moderately increase.

The effect on the combustion profile is observed in *Fig. 17*, where the softening action of delaying the 3rd injection is evident. Delaying further the 3rd injection generates a reverse trend, so the combustion starts earlier and overlaps with the 3rd injection. The assumption here is a cooling effect of the 3rd injection event that decreases the reactivity of the mixture, delaying its ignition. When this event starts too late, the mixture reaches its ignition point without being affected by this cooling effect and then the onset of combustion advances. The final result is a sharp increment in soot emissions and noise levels caused by the earlier and faster combustion process.

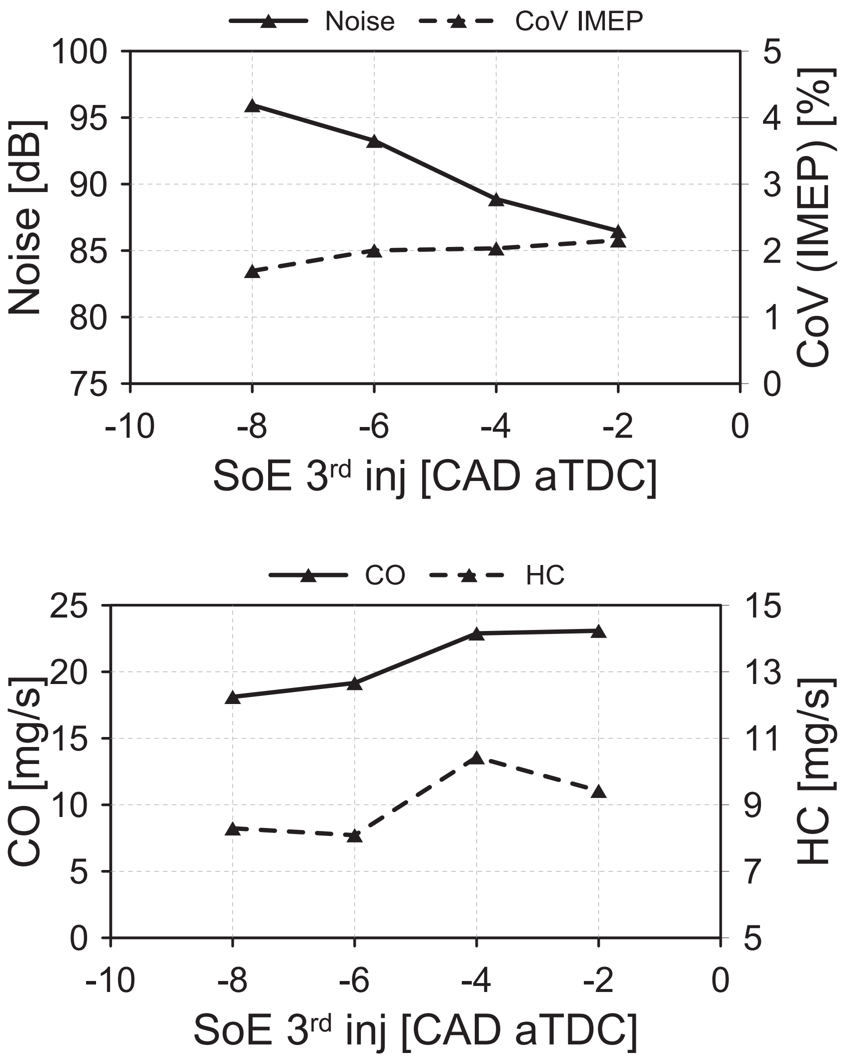

Figure 16: Noise and pollutant levels with more fuel in the 3rd injection

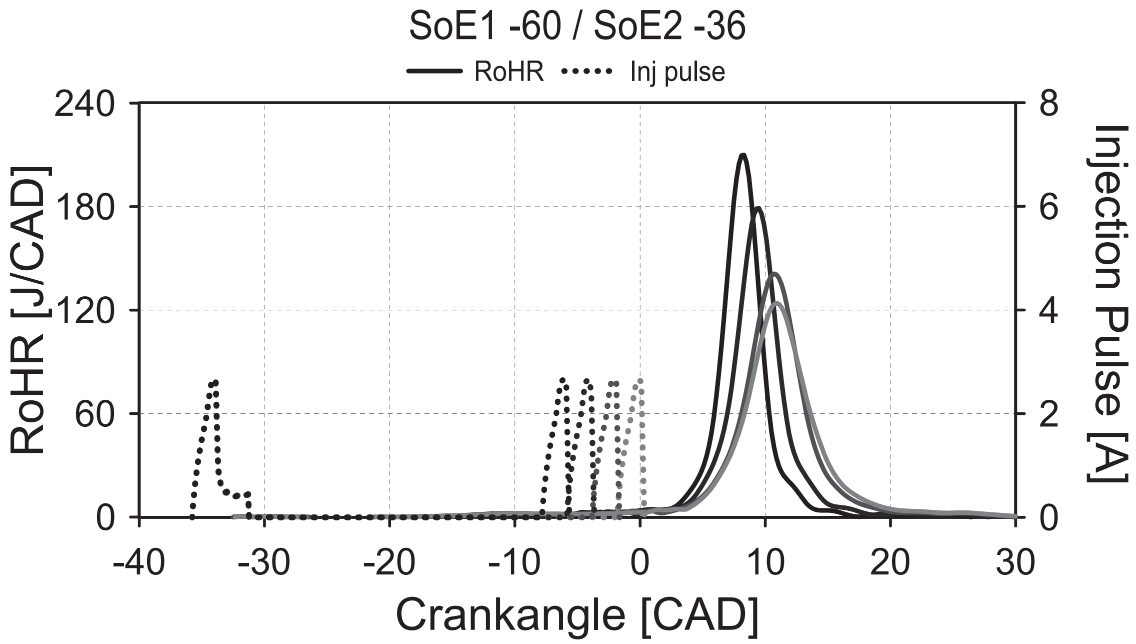

Figure 17: RoHR profiles with the new fuel distribution (3rd injection sweeping)

5. Conclusion

The research work reported in this paper was focused on the optimization of the combustion process development in a 2-stroke poppet valves HSDI CI engine, operating with the gasoline PPC concept. This engine architecture offers a large operating range for this combustion concept and also a high flexibility on the different settings, allowing the quasi-independence between air management settings (influencing in-cylinder conditions and performances) and injection settings (controlling the combustion process evolution and emissions). The ISFC previously obtained was in a satisfactory range, but the ISFC$_{corr}$ (taking in account the power demanded by the supercharger to provide the needed intake pressure, especially at low load) was too high. Also, the other observed drawback was the combustion noise level, unacceptably high, because of the sharp knocking-like combustion (especially at high load).

The ISFC$_{corr}$ was reduced by influencing the Trapping Ratio (TR). Increasing it means trapping more intake air and wasting less work from the supercharger. This was obtained by increasing the exhaust pressure to reduce the pressure drop (ΔP) across the engine. The gains were significant, but not sufficient to reach the expected levels without damaging combustion (stability, emissions). But the strategy implemented here has demonstrated through a new way the independence between the air management and the combustion process and emissions, within a given range. The next objective would be to improve even more the ISFC$_{corr}$, by increasing the TR without worsening the scavenge, or by decreasing the contribution of the supercharger. Another approach could be the reduction of the EGR rate, but an alternative should be presented to reduce both NO_x emissions and mixture reactivity.

The combustion noise is also one of the main known problems of the gasoline PPC concept. The multiple injections strategy adopted by the authors allows some flexibility concerning the fuel distribution. Then, the study carried out focused on the mass injected during the 3rd event shows very promising results, providing control over the noise, by switching only a little quantity of fuel from the main 2nd injection to the 3rd event. It also helps to reduce significantly both soot and NO_x emissions, but deteriorates combustion efficiency by generating more CO and HC emissions. But the indicated efficiency and ISFC / ISFC$_{corr}$ trade-off are hardly affected from one distribution to another. This confirms the possibility of controlling NO_x/soot emissions keeping attractive fuel consumption levels in a relatively wide range of injection settings, while noise control at high loads demands a fine tuning of both injection and air management settings to generate a suitable combustion process.

These experimental observations come as a correlation with a numerical study performed as a prediction, that reveals trends that were not -could not be- observed experimentally, allowing the researchers to define new investigations paths.

Acknowledgments

The authors kindly recognize the technical support provided by Mr. Pascal Tribotté from RENAULT SAS in the frame of the DREAM-DELTA-68530-13-3205 Project.

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