CONCEPTUAL MODEL DESCRIPTION OF THE DOUBLE INJECTION STRATEGY APPLIED TO THE GASOLINE PARTIALLY PREMIXED COMPRESSION IGNITION COMBUSTION CONCEPT WITH SPARK ASSISTANCE

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ABSTRACT

New combustion concepts applied to Compression Ignition engines are focused on achieve low temperature combustion together with a lean mixture distribution by allowing extra time from the end of injection to the start of combustion. Recently, the use of gasoline in a Compression Ignition engine under PPC conditions has been demonstrated as a suitable technique to achieve this extra mixing time, however the concept has also demonstrated difficulties under low load conditions using gasoline with octane number up to 95. The use of spark assistance with single injection operation has been found to be an appropriate way to improve the combustion control, providing both temporal and spatial control over the combustion process.

The current paper details the influence of the double injection strategy on the Spark Assisted Partially Premixed Combustion concept compared with the single injection strategy. For this purpose, a reference combustion cycle for both injection strategies is compared in terms of the main parameters derived from the in-cylinder pressure signal as well as OH* and natural luminosity images acquired from the single-cylinder transparent engine. The cylinder head used along the research has been modified including a spark plug. In addition, a detailed analysis of the air/fuel mixing process has been developed by means of a 1-D in-house spray model.

KEYWORDS

Partially premixed combustion

Spark assistance

High octane number gasoline

Combustion control

Double injection

Natural luminosity and OH images

1. INTRODUCTION

The automotive scientific community and manufacturers are currently focusing part of their efforts on the investigation of new combustion modes [1][2] and on the optimization of the current technology with the aim of reducing fuel consumption and emissions in CI diesel engines [3]. Most of these new combustion concepts are achieved by using different strategies that produce a lean air-fuel mixture together with a low temperature combustion. It contributes to decrease drastically the most relevant CI diesel engine-out emissions, NOx and soot [4]. In addition, due to the in-cylinder mixture homogeneity, a fast heat release is obtained when the proper in-cylinder conditions are achieved providing high combustion efficiency.

These combustion concepts based on fully or partially premixed lean mixtures are commonly known as Homogeneous Charge Compression Ignition (HCCI) [5][6]. Even though they achieve important emission benefits [7], these combustion concepts present some practical issues that must be overcome before they can be implemented in CI diesel engines being confined to low engine speeds and loads [8]. The most relevant limitations of this combustion modes consist of achieving an appropriate combustion phasing, the cycle-by-cycle control of the combustion process, spray impingements and its effects on the emissions [9], the noise and operating range extent. Several techniques such as EGR [10], variable valve timing [11][12], variable compression ratio [13] and intake air temperature variation [14] have been investigated in order to overcome these drawbacks. Due to the high chemical reactivity of the diesel fuel, the mentioned techniques cannot provide precise control over the combustion phasing since they require large time scales to achieve cycle-by-cycle control. Thus, not enough mixing time before the start of combustion is provided.

The scientific community is currently trying to overcome these disadvantages by using fuels with different reactivity [15][16][17]. In this sense, Partially Premixed Combustion (PPC) using a low reactivity fuel has been confirmed as promising method to control the heat release rate providing a reduction in

NOx and soot emissions as well. The use of a high ON fuel, such as gasoline, in a CI engine under PPC conditions provides more flexibility to reach lean and low combustion temperatures due to the extra mixing time achieved [18] through the fuel properties. However, the concept has demonstrated difficulties at low load conditions using gasoline with octane number greater than 90, concluding that the use of a low reactivity fuel under PPC conditions provide some control on combustion phasing but still do not solve the possibility of cycle-by-cycle control.

Recent investigations on gasoline engines (SI) running in homogenous or premixed combustion modes such as CAI (always PFI) [19][20], have shown the potential of using the assistance of a spark plug for achieving cycle-to-cycle control and combustion phasing control. The results suggest that this strategy can provide good combustion phasing while the response time is short enough for cycle-by-cycle application. Nevertheless, further research on spark assistance in new combustion modes is necessary for continuing its development with low reactivity fuels [21][22]. Thus, with the aim of integrating phasing and cycle-to-cycle control by means of a spark plug ignition system in a CI engine working in partially premixed charge, the PPC concept with spark assistance fuelled with gasoline has been studied. This engine architecture has a high compression ratio and it is equipped with a common rail injection system that enables high injection pressures. Thus, the Partially Premixed Combustion concept with Spark Assistance has been evaluated in terms of performance and engine-out emissions using a single injection strategy by studying the effect of injection pressure variations and intake oxygen concentration. Under these conditions, spark assistance has been found to be a suitable technique for improving combustion control, providing both temporal and spatial control over the combustion process [23][24]. In spite of its benefits, some drawbacks related to unappropriated mixture distribution and combustion temperatures are attained. Single injection provides excessive rich zones near to the spark plug and excessive lean regions close to the wall chamber resulting in high emission levels as well as deteriorated fuel energy conversion efficiency. Thus, the main objective of the present work is to

analyze the effect of the double injection strategy on the mixture distribution and combustion development under partially premixed compression ignition spark assisted mode. The investigation was performed in an optical engine since it is a suitable tool for performing a basic combustion research combining in-cylinder pressure signal derived parameters and optical combustion images as an experimental sources of information together with a 1-D spray model.

The outline of this paper is as follows: in the first section, the experimental facilities and the different setups used to carry out this research are presented. Specifically, this section describes briefly the methodology, experimental facilities and processing tools used from the acquisition of the raw data during the experimental tests to the final results obtained by means of the post-processing tools. In the following section, which is the base of this paper, a summary of the preliminary results and a description of the combustion event comparing the single and double injection strategies are done. Finally, in section 4, the main conclusions of this research are summarized.

2. EXPERIMENTAL FACILITIES AND PROCESSING TOOLS

This section describes the methodology used to acquire the experimental data and provides a description of the experimental facility, the different devices and systems that were specifically adapted for the study of this combustion mode.

2.1 Experimental setup

This section presents the experimental configuration of the test cell and the main subsystems used in this study. As Figure 1 shows, the single cylinder engine is installed in a fully instrumented test cell, with all the auxiliary facilities required for operation and control.

The intake air is supplied by a Roots compressor with an upper pressure limit of 3 bar. Then, the air flows through a filter to remove possible impurities. The heat exchanger and the air dryer allow controlling the temperature and humidity of the intake air independently of the ambient conditions. The

temperature in the inlet settling chamber is maintained constant by using the heater in the intake line. The oxygen concentration variation is performed using a synthetic EGR system. EGR is substituted by nitrogen gas, which greatly simplifies the system ensuring a controllable gas composition without an excessive time to adjust the facility. Despite the limited practical application, it was decided to use this method to have a better control of the variables, which allows studying the underlying phenomena more carefully. The concept is based on decreasing the O_2 concentration at the inlet manifold by increasing the flow of N_2 and keeping constant the total intake mass flow rate (substitution EGR). For this purpose a PID controller is equipped to operate the N_2 valve governed by the intake O_2 meter. With this system, the in-cylinder thermodynamic conditions can be reproduced systematically. To ensure a homogeneous mixture of N_2 and O_2 and to attenuate pressure pulses in the intake manifold, a settling chamber of 500 liters volume is used in the installation. In the exhaust line, after the exhaust analyzer sample probe, a catalyst is mounted to prevent the accumulation of unburned hydrocarbons in the installation. Due to the low temperatures achieved during the combustion event and therefore in the exhaust line, the catalyst is often operating with low efficiency and a cyclone is needed to remove the rest of the hydrocarbons. In the same way as in the intake line, a settling chamber is mounted in order to attenuate pressure pulses. Finally, an exhaust backpressure valve is equipped to maintain a relative pressure of 0.2 bar to the intake pressure, in order to simulate more realistic conditions.

2.1.1 Engine description and cylinder head adaptation

The engine used in the present study is a 4-valve, 0.545 l displacement single cylinder engine with a modified cylinder head for the study of this combustion mode. The bowl dimensions are 45x18 mm (diameter x depth). Table 1 presents the main characteristics of the engine.

In order to characterize the most relevant properties of the gasoline used in this research, various analyses of the fuel properties have been performed following ASTM standards. It is worthy to note that

300 ppm of additive (*Havoline Performance Plus*) was added to improve the lubricity of the gasoline up to diesel fuel level, increasing the service life of the high pressure pump and fuel injector. The addition of the additive does not modify neither density nor the viscosity. The results of the gasoline characterization are summarized in Table 2.

The fuel injection hardware characteristics are depicted in Table 3. The injection control system allows to modify any parameter of the injection event such as timing, duration and rail pressure.

A spark plug is required to implement the partially premixed compression ignition with spark assistance combustion mode. As Figure 2 shows, the cylinder head has been modified by removing an exhaust valve and thus enabling the insertion of the spark plug in the combustion chamber. A standard spark plug (Veru Platinum) with a 1 mm gap is used along with a custom electronic control system. In the standard configuration, the tip protrudes 4.5 mm into the combustion chamber from the cylinder head plane and it is located 17 mm from the cylinder axis. The injector is centered and vertically assembled in the modified cylinder head with a graduated metal circle that can change the relative position between the spark plug and the injector fuel jets by rotating the injector around its vertical axis. This relative position is fixed to make the spray pass between the spark electrodes.

In order to increase the reliability of the combustion mode, a Delphi multicharge ignition system has been used. The high amount of energy released by this ignition system allows igniting the mixture even with local equivalence ratio conditions near their flammability limits with high EGR rates. The spark ignition system is operated at a constant nominal primary voltage of 15 V from the battery and primary current of 25 A, providing around 120 mJ for the typical combustion chamber density test conditions, almost double than of a conventional ignition system.

2.1.2 Optical engine configuration

The engine is equipped with an elongated piston with a cylindrical bowl, which allows optical access to the combustion chamber through a sapphire window placed in its bottom. Below the piston bowl, an elliptical UV mirror is placed on the cylinder axis. In front of the mirror, a beam splitter (50%-50%) is positioned in order to allow the simultaneous acquisition of the OH radical luminosity and the natural luminosity. For the acquisition of the natural luminosity images, a high speed Phantom V12 CMOS camera equipped with a 100 mm focal length Zeiss lens is utilized with an image resolution of 512 x 512 pixels. In order to acquire the OH radical luminosity images a Photron intensified camera equipped with a 100 mm UV lens together with a band pass filter centered at 310 nm is utilized. Figure 3 shows the optical scheme. It is interesting to note that the tests have been performed under skip-fire mode (1 cycle fired per 30) in order to avoid excessive thermal stress in the windows and ensure the same in-cylinder initial thermodynamic conditions for the recorded (fired) cycles.

2.2 Theoretical tools

In order to understand how the results are obtained from the experimental data, a brief description of the principles behind the different tools used for processing and post-processing the results is provided here.

2.2.1 Analysis of pressure signal

The combustion analysis is performed with an in-house one-zone model named CALMEC, which is fully described in [25]. This combustion diagnosis tool uses the in-cylinder pressure as its main input. The in-cylinder pressure was measured with a Kistler 6067C1 pressure transducer. The pressure traces for 200 engine cycles were recorded in order to compensate the cycle-to-cycle variation during engine operation. Then, the first law of thermodynamics is applied between IVC and EVO, considering the

combustion chamber as an open system because of blow-by and fuel injection. The ideal gas equation of state is used to calculate the mean gas temperature in the chamber. Along with these two basic equations, several sub-models are used to calculate instantaneous volume and heat transfer [26], among other things. The main result of the model is the Rate of Heat Release (RoHR). Information related to each cycle can be obtained, such as the IMEP and SoC. Start of Combustion (SoC) is defined as the crank angle position in RoHR where the beginning of the slope rise due to combustion is detected.

2.2.2 Analysis of mixing process

A 1-D in-house spray model DICOM is used to estimate equivalence ratio distributions in the fuel jet in order to get better insight into the variations in mixture distribution associated with the variations in the parameters studied in the experimental tests. The start of combustion and the combustion development have an extreme dependency on the local mixture conditions at Start of Spark (SoS) timing. The inputs of the DICOM model are the in-cylinder thermodynamic conditions (pressure, temperature and density), the spray cone angle, the fuel mass flow rate and the spray momentum. The model solves the general conservation equations either in a transient or steady state formulation for axial momentum and fuel mass along the center line. The results can be used to calculate values of spray velocity, species mass fractions and other values of the mixing process [27]. Finally, with some other assumptions described in [28], the model is used to obtain different temporal evolutions such as the spray liquid and vapor penetration, maximum spray velocity, equivalence ratio along the center line of the spray and the fuel mass fraction which has mixed to different equivalences ratios. The fuel mass fraction is the main variable used in this research.

2.2.3 Image processing tool

In order to complement the information about the differences in the combustion development for both injection strategies, time resolved parameters were calculated for every image in each sequence following a well-defined methodology. First, segmentation was performed for every image by calculating

a threshold value, which is equal to the minimum digital level in the image plus 15% of the difference between the maximum and the minimum. This percentage was set, based on previous experience, as a compromise to remove light reflected off the liquid spray and the chamber walls without losing much information from the combustion event [29]. After segmentation, the flame area is defined as the summation of the number of pixels which belong to the flame (above the threshold). Thus, the digital levels of all pixels containing the combustion radiation (those above the threshold) are accumulated and averaged over the number of pixels of the flame area obtaining a single mean flame intensity parameter named I_{mean}. Additionally, an apparent combustion velocity (ACV) is calculated as Equation 1 [20], where *A* is the flame area, *L* is the perimeter of the flame area and *t* is the time:

$$\mathrm{ACV} = \mathrm{d}(\mathrm{A}/\mathrm{L})/\mathrm{dt} \qquad (1)$$

3. BASIS OF GASOLINE PARTIALLY PREMIXED COMPRESSION IGNITION SPARK ASSISTED COMBUSTION CONCEPT

As noted above, one of the main drawbacks of the new combustion concepts is to control the combustion phasing as well as cycle-to-cycle variation. The use of gasoline coupled with the spark assistance has been studied in order to minimize the influence of the thermodynamic conditions on the ignition process for CI engines confirming its suitability to improve the cycle-to-cycle control. Figure 4 shows the coefficient of variation of IMEP (COV IMEP) versus IMEP for single and double injection tests using gasoline with spark assistance and a batch of single injection tests using gasoline without spark assistance (PPC).

On one hand, comparing the single injection cases with and without the use of the spark (points 3, 4 and 5) in figure 4, it is possible to state that in all cases the COV IMEP is reduced using the spark assistance. Focusing on low load (point 3), it is possible to observe how the case without spark assistance has an unacceptable COV IMEP. However, by means of the spark assistance COV IMEP is halved in the same operating condition widening the operating range. Unfortunately, for low-medium loads and

considering the same fuel injection quantities for points 3, 4 and 5 in figure 4 the IMEP values are decreased due to the nature of this combustion type (less constant volume-like coupled with a longer combustion duration).

On the other hand, comparing only single and double injection tests with spark assistance, it is possible to state that the use of the double injection strategy allows to increase the IMEP values with lower fuel injected mass. This, suggests that the fuel consumption will be diminished. In addition, a reduction at the coefficient of variation is also achieved.

3.1 Description of the reference case combustion event

The reference cases for the single and double injection strategies will be used to describe the partially premixed compression ignition with spark assistance combustion process. In both cases, the injection pressure was maintained constant at 900 bar, intake O_2 concentration at 19.6% and injected fuel mass at 21 mg/stk. The temporal evolution of the mass flow rate for both strategies is represented in Figure 5.

Figure 6a shows the crank angle evolution of different variables. From the top to the bottom, the figure shows the mass flow rate, the mean unburned gas temperature, the in-cylinder pressure and the rate of heat released. In all cases, the spark plug discharge was set at EoI and it determines the SoC. Moreover, to further understand the air/fuel mixing process and the equivalence ratio conditions at SoS and autoignition time, an analysis was carried out using a 1-D jet mixing model. Thus, it is possible to understand how the fuel mass is distributed throughout different equivalence ratios as a function of time. By processing the 1-D model results, a distribution of the fuel mass mixture fraction using bins of 0.2 ϕ width at experimental SoC and autoignition is obtained. Figure 6b shows, for the single and double injection strategies, the fuel mass mixture distributions calculated using the 1-D jet mixing model described above. Figure 7 and Figure 9 show the temporal evolution of the natural luminosity (NL) and

OH radical for the single and double injection reference combustion cycles represented in Figure 6. Each image corresponds to the crank angle degree showed above them. In addition, the spark plug location (SP) and the swirl motion is depicted in the first image of the sequence. In Figure 8, the temporal evolution of the non-dimensional flame area (the flame area divided by the combustion chamber area), the apparent combustion velocity (ACV) and the mean intensity in the combustion chamber are shown for both injection strategies.

A detailed description of the PPC Spark Assisted combustion development under single injection conditions will be explained in order to take it as a reference to compare with the double injection strategy. Considering the red color lines in Figure 6a, once the injection process has finished at -9.8 CAD, the spark plug discharge takes place initiating the combustion process. At this time (-9.8 CAD in Figures 7 and 9), it is possible to observe that the first kernel near the spark plug presents intensity values above the minimum threshold of the correlated color scale. Therefore, it is confirmed that the start of combustion is controlled by the spark plug and not by the in-cylinder thermodynamic conditions. The kernel growth generates a partially premixed flame propagation. As Figure 6a shows, an almost linear-sustained heat release from -9.8 CAD to -3.5 CAD is attained. The flame propagation is guided by the swirl motion which can be clearly observed along the whole combustion sequence presented in Figure 7. It should be noticed that without the spark assistance the combustion process is not achieved at all under these operating conditions.

The energy released during the flame propagation phase described above causes an increase in the unburned gas pressure and temperature (Figure 6a), leading to a second phase of combustion governed by the autoignition of the rest of the mixture from -3.5 CAD to +2.4 CAD. Aside from the increase in the unburned gas pressure and temperature, it is also possible to appreciate this second stage during the combustion development by observing the RoHR profiles in terms of maximum peak (76 J/CAD for the flame propagation and 175 J/CAD for the autoignition) as well as in the combustion duration (7.8 CAD

279 for the flame propagation vs 4.3 CAD for the autoignition) shown in Figure 6a. This second phase is also
280 noticeable observing the changes in the flame pattern shown in the images of Figure 7 and Figure 9. This
281 change in the flame pattern leads to a higher combustion area and luminosity as it can be appreciated in
282 the mean luminosity profile presented in Figure 8. In this sense, another way to confirm that the
283 autoignition phase is faster than the flame propagation phase is the comparison of the ACV peaks of
284 both phases for the single injection strategy in Figure 8.

285 Regarding double injection strategy, the RoHR profile in Figure 6a shows two combustion stages, as well.
286 However, some differences are found comparing with the single injection case. It is possible to
287 appreciate in both, Figure 7 and 9, how the first kernel near the spark plug (-2.3 CAD) is fainter than the
288 one acquired in the single injection case (-9.8 CAD). The luminosity is related with the mixture
289 distribution at SoC shown in Figure 6b. The comparison of the mixture distributions for both strategies
290 reveals clear differences in the distribution of the fuel mass at SoC. Two zones are observed in the
291 double injection strategy. The first zone ($0.2<\phi<0.8$) is attributed to the fuel injected during the pilot
292 injection. In this zone, the fuel mass is distributed under low equivalences ratios because of the longer
293 mixing time provided by the dwell between the injections. In the second zone ($\phi>0.8$), attributed to the
294 main injection, the fuel mass is distributed under more reactive equivalences ratios providing the local
295 conditions needed to ignite the mixture with the spark plug assistance. Comparing the mixture
296 distributions for both strategies at SoC, it can be appreciated how the single injection strategy leads a
297 considerably higher amount of fuel mass mixed up to rich equivalence ratios in comparison with the
298 double injection strategy. Particularly, the more fuel mixed in this range of equivalence ratios (from
299 $\phi=0.5$ to $\phi=3.5$) ensures that some part of the mixture distribution is inside the range of flammability
300 limits. Therefore, the spark is capable of promoting the start of combustion and the progression of the
301 premixed flame. This higher quantity of fuel mass under stoichiometric and rich equivalences ratios in
302 the case of the single injection strategy becomes in a first kernel with high temperature once the spark

303 plug has discharged the energy. It is possible to confirm the higher temperature in the case of the single
304 injection by comparing the OH radical luminosity in single injection strategy (-9.8 CAD) and double
305 injection strategy (-2.3 CAD) in Figure 9. Thus, comparing the mean intensity profiles shown in Figure 8,
306 it can be noticed that the flame intensity in the kernel (just after the peak observed during the spark
307 discharge) in the double injection strategy is lower than in a single injection strategy (220 a.u. vs 128
308 a.u.).

309 Considering the comparison between flame propagation phases for single and double injection
310 strategies, it can be stated that the flame propagation phase in the double injection strategy from -2.3
311 CAD to +7.6 CAD in Figures 7 and 9 is slower and less intense than in the single injection case. It can be
312 confirmed taking into account the lower ACV peak in the double injection (at +2.2 CAD) compared with
313 the single injection case (at -6.8 CAD) in Figure 8, as well as the maximum RoHR values obtained during
314 this phase (double injection: 60 J/CAD versus single injection: 77 J/CAD) in Figure 6a. Both parameters,
315 apparent combustion velocity and heat release, are highly related with the in-cylinder local equivalence
316 ratio distribution. Figure 6b shows a lower amount of fuel mass within the range of reactive conditions
317 in the double injection case, slowing down the flame propagation. This slowing down can be observed
318 comparing the RoHR duration in both injection strategies attributed to the flame propagation phase in
319 Figure 6a (9 CAD for the double injection vs 7 CAD for the single injection). These differences in the
320 equivalence ratio distribution can be also noticed in Figure 7 by means of the luminosity acquired in the
321 first instants of the flame propagation phase. Taking into account the color scale in Figure 7, the
322 luminosity obtained in the single injection (from -6.4 CAD to -3.5 CAD) is more intense than in the
323 double injection case (from +2.2 CAD to +5.1 CAD). This fact is ratified comparing both mean intensity
324 profiles in Figure 8. By contrast, due to the more homogeneous fuel-air distribution attained with the
325 pilot injection, a larger area is covered in the case of the double injection during this period as Figure 8
326 shows.

Once the flame propagation phase has been compared between both injection strategies, the differences found during the autoignition time are explained. At the autoignition time, the fuel mass distributions presented in Figure 6b are quite similar in both injection strategies. The extra mixing time (around 8 CAD in both cases) achieved from the EoI up to the autoignition time provides a leaner and more homogeneous in-cylinder mixture distribution. This homogeneity in the mixture, together with the increase in the in-cylinder pressure and temperature due to the first combustion phase, promotes a faster combustion development during this autoignition phase compared with the flame propagation phase. Comparing the mixture distributions in both injection strategies, it is possible to appreciate that in both cases all the mixture is distributed below the stoichiometric equivalence ratio being the more remarkable difference the higher fuel mass amount available in the double injection strategy. It promotes a more powerful and faster autoignition compared with the single injection strategy. In the same way, it is possible to confirm this behavior making a comparison between the maximum RoHR peaks in Figure 6a (268 J/CAD versus 176 J/CAD in the single injection case) as well as comparing the autoignition phase ACV peaks in both injection strategies (at -5 CAD in single injection and +4.8 CAD in double injection) in Figure 8. It is interesting to note that the double injection strategy improves the region where the combustion takes place, covering the whole combustion chamber. This can be clearly confirmed by looking the non-dimensional flame area profile depicted in Figure 8, as well as comparing the images of Figure 7 (single injection: +2.4 CAD vs double injection: +9.1 CAD or single injection: +5.6 CAD vs double injection: +10.1 CAD).

4. CONCLUSIONS

The influence of the double injection strategy on the Spark Assisted Partially Premixed Compression Ignition combustion concept fuelled with high ON gasoline at low load operating conditions has been studied combining theoretical and practical tools. An analysis of the parameters derived from in-cylinder pressure measurement has been combined with the 1-D jet model calculations. In order to complement

the research, images of the natural luminosity and OH radical as well as different parameters derived from these images have been also analysed.

The spark assistance with double injection strategy has been found to be capable of extending the limits of the gasoline CI combustion mode at low-medium load conditions to lower fuelling levels than was possible with single injection strategy. Another important finding obtained by means of the in-cylinder images acquired with the high speed cameras is that the spark assistance is capable to provide spatial and temporal control over the combustion process in the double injection strategy conditions tested. Regarding the general development of the combustion process, the combustion phenomenology described here is consistent with previous Spark Assisted PPC work using a single injection strategy. The spark plug discharge takes place initiating the combustion process by means of a first kernel. The kernel growth generates a partially premixed flame propagation that rises the unburned gas pressure and temperature leading to a second phase of combustion governed by the autoignition of the rest of the mixture. Important differences on the different combustion phases have been found in the double injection operating conditions tested:

- A fainter *kernel* is produced due to the leaner equivalences ratios at SoC. This leaner equivalence ratio distribution, presented by the 1-D jet model calculations, results in a lower in-chamber luminosity (mean natural luminosity) and temperature (OH radical).

- The leaner mixture conditions during the first instants of the combustion development makes the *flame propagation phase* of the double injection strategy slower compared with the single injection strategy, as the apparent combustion velocity values shows. That is also possible to appreciate by comparing the RoHR durations. This leaner equivalence ratio distribution results in a lower in-chamber luminosity (mean natural luminosity) and temperature (OH radical). In

addition, the double injection strategy is less energetic than the single injection case as the comparison of the maximum peaks of the RoHR denotes.

- With reference to the *autoignition* phase, the improvement in the spatial mixture distribution associated to the pilot injection improves the region in which the combustion takes place, covering the whole combustion chamber. It has been demonstrated with the images acquired as well as with the non-dimensional area calculation. Due to the higher mixture amount available to be burned at this time, a more intense autoignition is attained (higher maximum RoHR peak). In addition, it is possible to appreciate higher maximum peaks in the apparent combustion velocity besides a lower RoHR duration.

It is interesting to remark that the use of the double injection strategy allows an increase in the IMEP values with lower fuel injected mass. In addition, a reduction of the coefficient of variation is obtained. Thus, a reduction in the fuel consumption is expected.

ACKNOWLEDGMENTS

The authors would like to thank General Motors for supporting this research.

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ABBREVIATIONS

ASTM: American Society of Testing Materials

bTDC: before Top Dead Center

BDC: Bottom Dead Center

CAD: Crank Angle Degree

CI: Compression Ignition

CMOS: Complementary Metal Oxide Semiconductor

COV: Coefficient Of Variation

DI: Direct Injection

EOI_{main}: End of main injection

EOI_{pilot}: End of pilot injection

EVO: Exhaust Valve Opening

FeCE: Fuel energy Conversion Efficiency

HCCI: Homogeneous Charge Compression Ignition

IVC: Intake Valve Closing

IMEP: Indicated Mean Effective Pressure

LTC: Low Temperature Combustion

ON: Octane Number

PCCI: Premixed Charge Compression Ignition

PID controller: Proportional-Integral-Derivative controller

PPC: Partially Premixed Charge

SOI_{main}: Start of main injection

SOI_{pilot}: Start of pilot injection

SoC: Start of Combustion

SoS: Start of Spark

TDC: Top Dead Center

HIGHLIGHTS

- Double injection improves the combustion control in low load.
- Double injection enhances both phases of the combustion mode.

- Double injection increases the combustion area in the chamber.
- Combustion process described by mixing process, OH* and natural luminosity images.

Tables

Type	CI, 4stroke, DI
Max. engine speed [rpm]	4500
Cylinder number	1
Displacement [cm^3]	544.75
Bore [mm]	85
Stroke [mm]	96
Compression ratio [-]	14.7:1
Bowl diameter [mm]	45
Bowl depth [mm]	18
Bowl volume [cm^3]	25

Table 1. Main single cylinder engine characteristics

	Gasoline
Density (T=15°C) [kg/m^3]	722
Viscosity (T=40°C) [mm^2/s]	0.37
RON [-]	98
Lower heating value [kJ/kg]	44542

Table 2. Characteristics of the gasoline used in the study

Type	Common-Rail
Injector	Bosch CRIP 3.3
Hole number	7
Included angle [deg]	154^o
Hole diameter [um]	97 um

Flow capacity [cm^3/30s]	210

Table 3. Injection system characteristics used in the study

Figure captions

Figure 1. Complete test cell setup

Figure 2. Image of the modified cylinder head with spark plug and injector hole (left). Diagram of the relative position between the injector and spark plug (right)

Figure 3. Optical configuration for the image acquisition

Figure 4. IMEP vs. COV IMEP for single and double injection strategy and cases with and without spark assistance

Figure 5. Single and double injection strategy mass flow rate for the reference cases

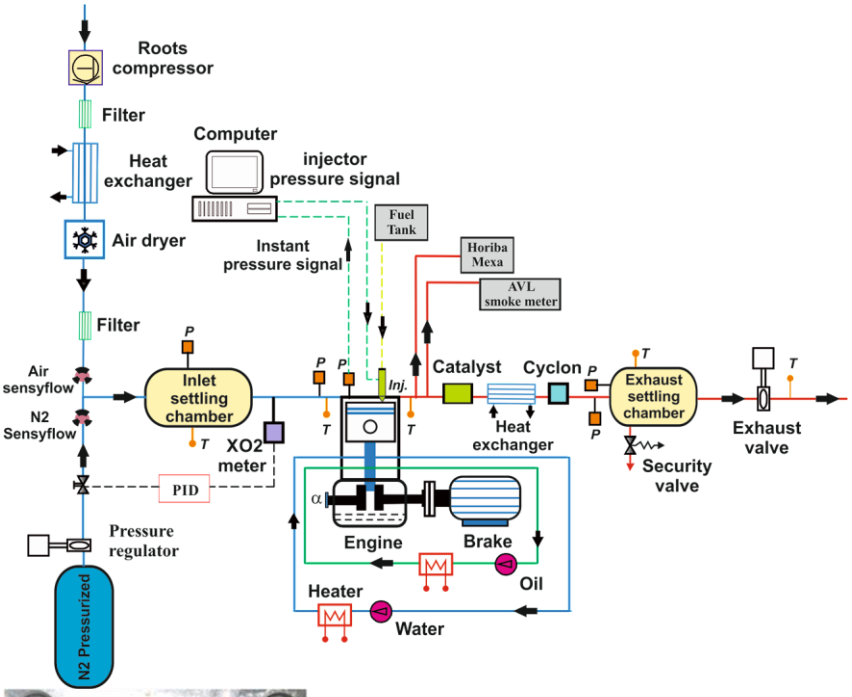
Figure 6. Crank angle evolution of the mass flow rate, unburned gas temperature, in-cylinder pressure, and rate of heat released for the single injection strategy and double injection strategy (6a). Distribution of fuel mass vs. φ in experimental SoS and autoignition time for the single and double injection strategies (6b).

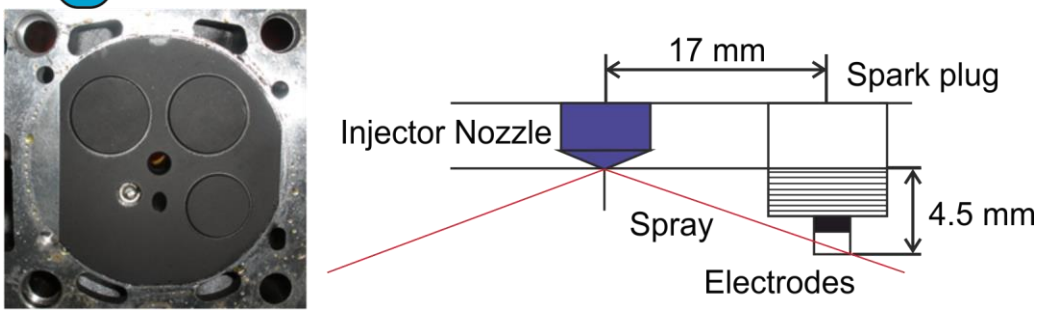
Figure 7. Temporal evolution of the natural luminosity (NL) for the reference combustion cycle for the single injection (upper) and double injection (lower). Each image corresponds to the crank angle degree showed above them

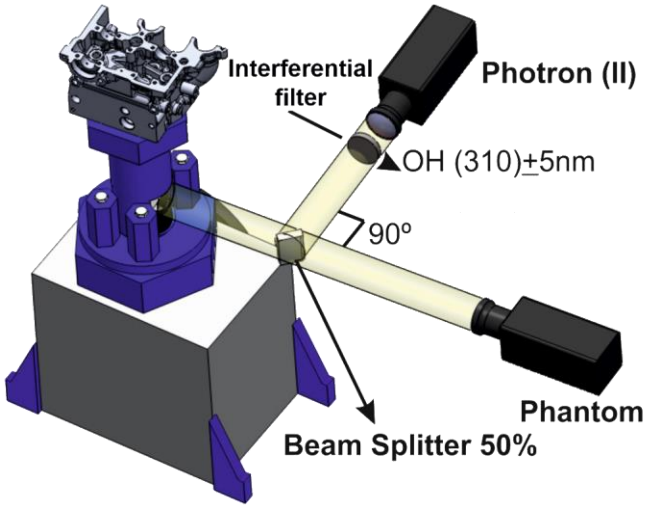
Figure 8. Temporal evolution of the non-dimensional flame area, apparent flame velocity and mean flame intensity for the single (red lines) and double (blue lines) injection cases presented above

Figure 9. Temporal evolution of the OH radical luminosity for the reference combustion cycle for the single injection (upper) and double injection (lower) cases. Each image corresponds to the crank angle degree shown above them

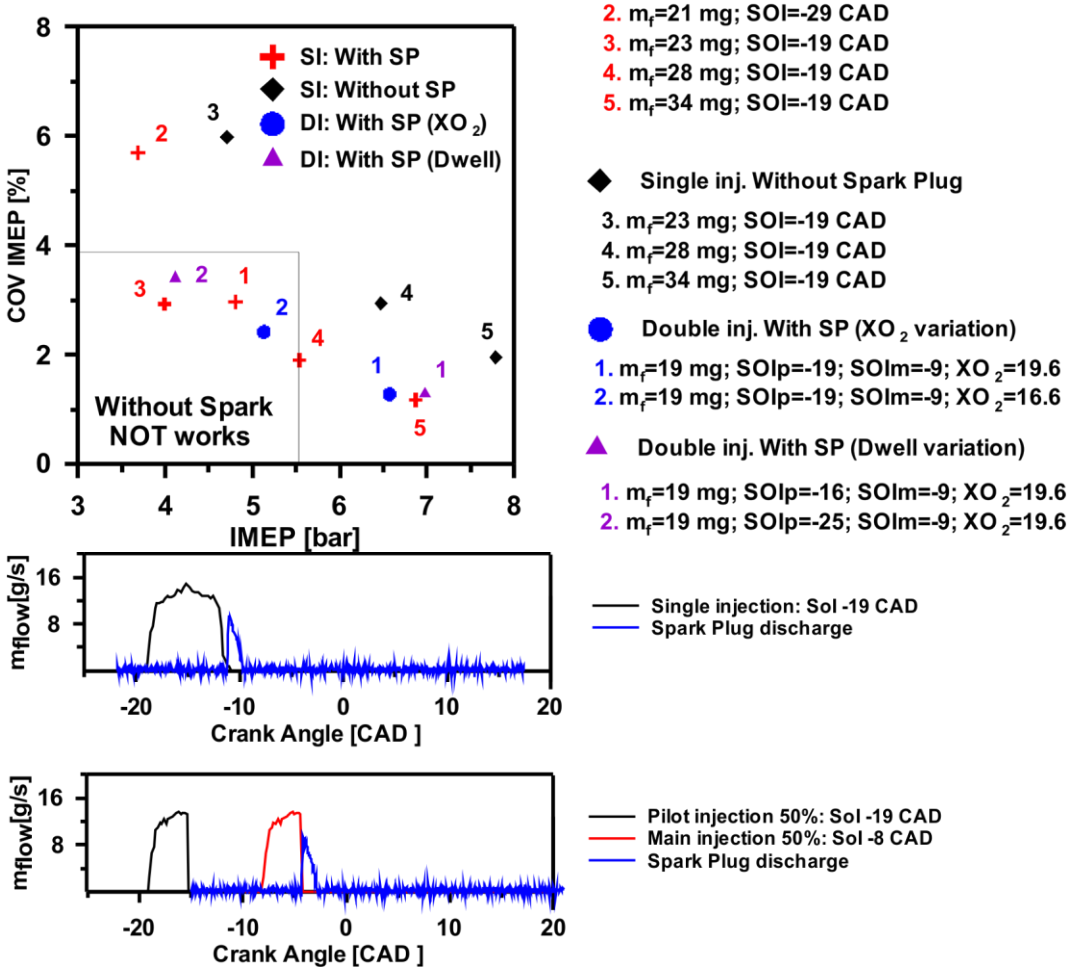
Figures


Roots compressor
Filter
Computer
Heat exchanger
injector pressure signal
Air dryer
Instant pressure signal
Fuel Tank
Horiba Mexa
AVL smoke meter
Filter
Air sensyflow
N2 Sensyflow
Inlet settling chamber
XO2 meter
PID
Pressure regulator
N2 Pressurized
Inj.
Catalyst
Cyclon
Heat exchanger
Exhaust settling chamber
Security valve
Exhaust valve
Engine
Brake
Oil
Heater
Water


17 mm
Spark plug
Injector Nozzle
4.5 mm
Spray
Electrodes

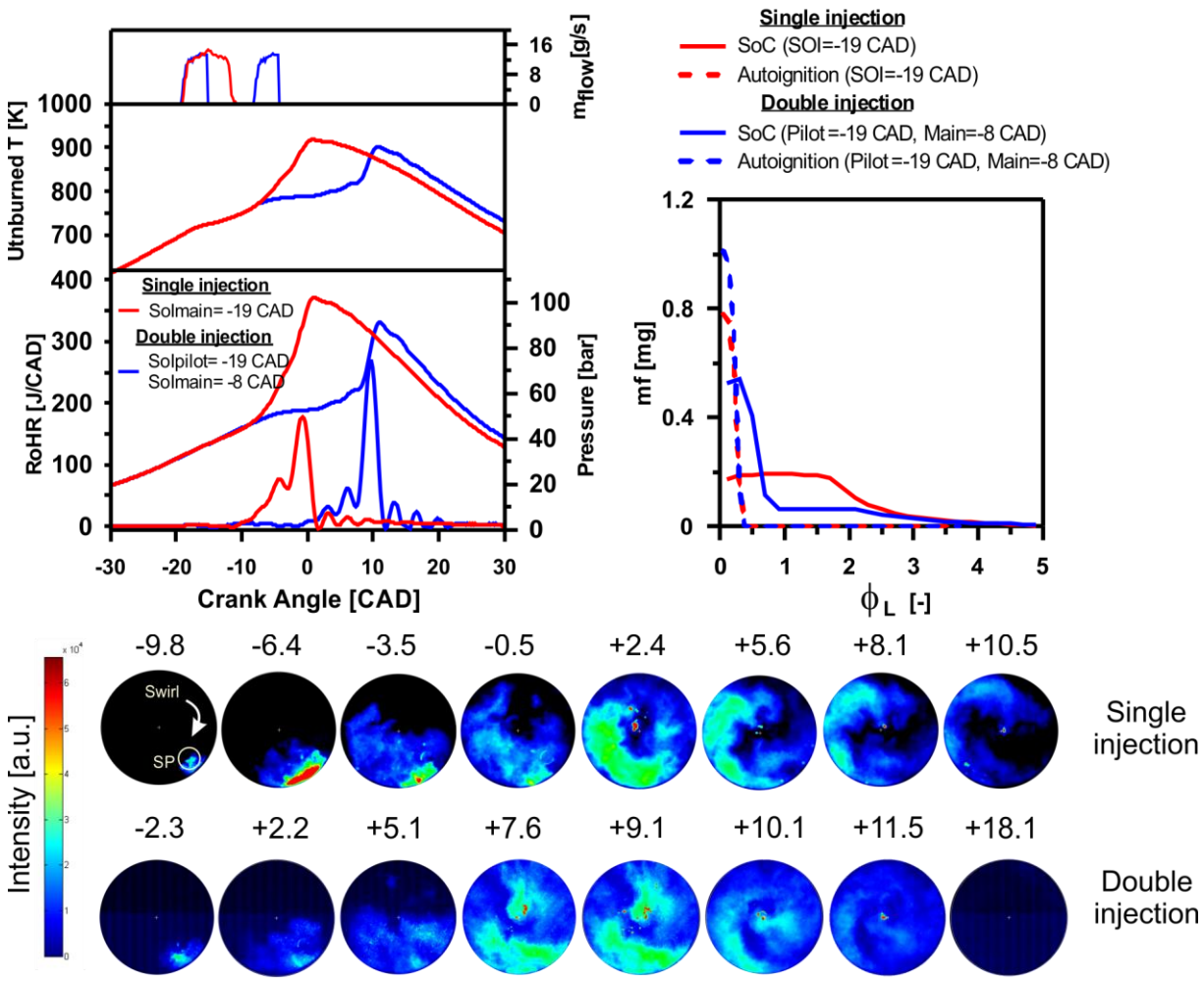

Interferential filter
Photron (II)
OH (310)±5nm
90º
Phantom
Beam Splitter 50%

543

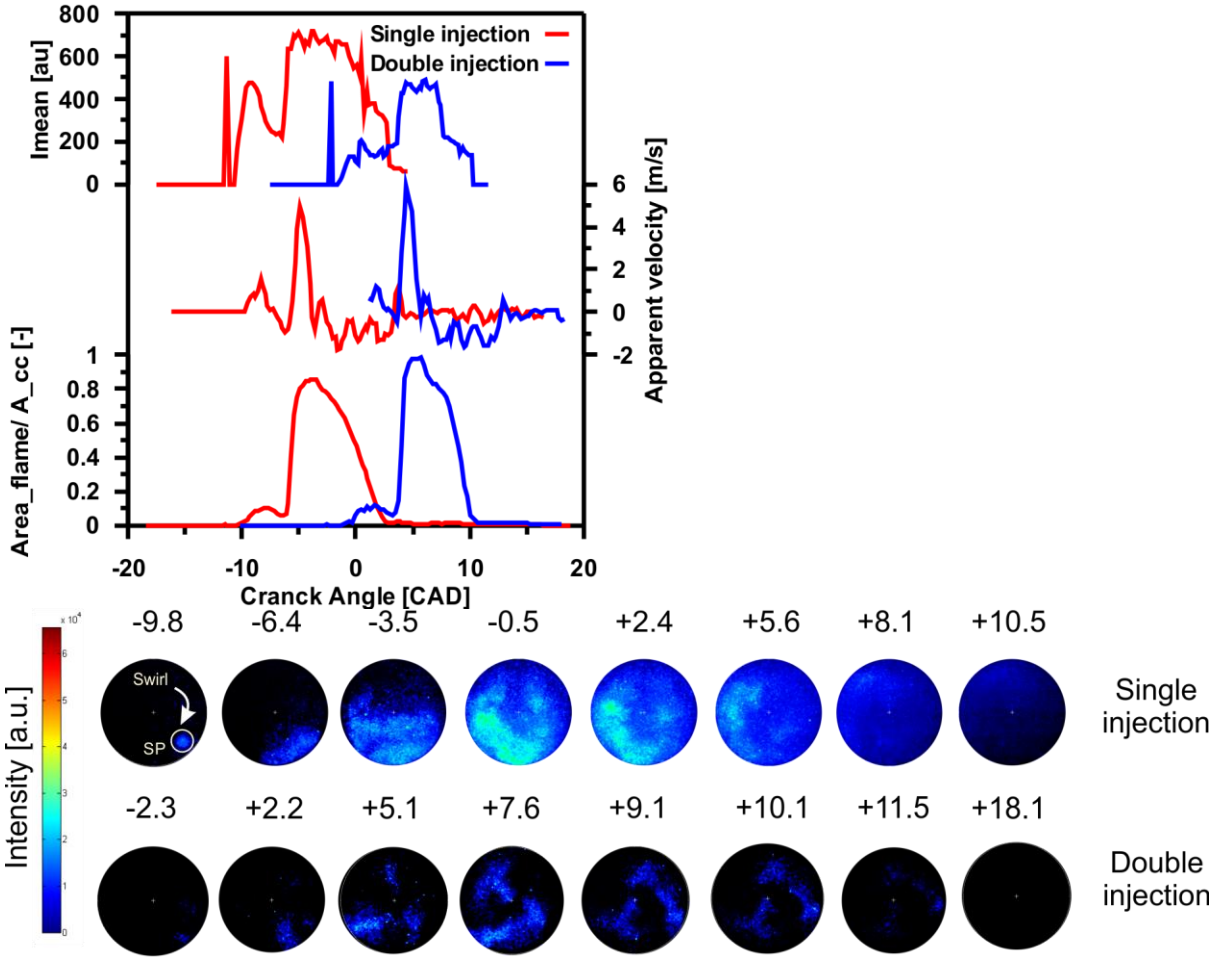

SI: With SP
SI: Without SP
DI: With SP (XO 2)
DI: With SP (Dwell)
COV IMEP [%]
IMEP [bar]
Without Spark NOT works
Single inj. with Spark Plug
1. mf=21 mg; SOI=-24 CAD
2. mf=21 mg; SOI=-29 CAD
3. mf=23 mg; SOI=-19 CAD
4. mf=28 mg; SOI=-19 CAD
5. mf=34 mg; SOI=-19 CAD
Single inj. Without Spark Plug
3. mf=23 mg; SOI=-19 CAD
4. mf=28 mg; SOI=-19 CAD
5. mf=34 mg; SOI=-19 CAD
Double inj. With SP (XO 2 variation)
1. mf=19 mg; SOIp=-19; SOIm=-9; XO 2=19.6
2. mf=19 mg; SOIp=-19; SOIm=-9; XO 2=16.6
Double inj. With SP (Dwell variation)
1. mf=19 mg; SOIp=-16; SOIm=-9; XO 2=19.6
2. mf=19 mg; SOIp=-25; SOIm=-9; XO 2=19.6
mflow[g/s]
Crank Angle [CAD]
Single injection: SoI -19 CAD
Spark Plug discharge
Pilot injection 50%: SoI -19 CAD
Main injection 50%: SoI -8 CAD
Spark Plug discharge

544

545


Utnburned T [K]
RoHR [J/CAD]
Pressure [bar]
$\dot{m}_{flow}$[g/s]
Crank Angle [CAD]
Single injection
Solmain= -19 CAD
Double injection
Solpilot= -19 CAD
Solmain= -8 CAD
Single injection
SoC (SOI=-19 CAD)
Autoignition (SOI=-19 CAD)
Double injection
SoC (Pilot=-19 CAD, Main=-8 CAD)
Autoignition (Pilot=-19 CAD, Main=-8 CAD)
mf [mg]
ϕ_L [-]
Intensity [a.u.]
Swirl
SP
-9.8
-6.4
-3.5
-0.5
+2.4
+5.6
+8.1
+10.5
Single injection
-2.3
+2.2
+5.1
+7.6
+9.1
+10.1
+11.5
+18.1
Double injection

548


Single injection
Double injection
Imean [au]
Area_flame/ A_cc [-]
Apparent velocity [m/s]
Cranck Angle [CAD]
-9.8
-6.4
-3.5
-0.5
+2.4
+5.6
+8.1
+10.5
Swirl
SP
Intensity [a.u.]
Single injection
-2.3
+2.2
+5.1
+7.6
+9.1
+10.1
+11.5
+18.1
Double injection

549