Investigation of gasoline compression ignition for combustion control

Published Version:
Investigation of gasoline compression ignition for combustion control / Ravaglioli V.; Ponti F.; De Cesare M.. - In: JOURNAL OF ENGINEERING FOR GAS TURBINES AND POWER. - ISSN 0742-4795. - ELETTRONICO. - 142:9(2020), pp. 091003.1-091003.7. [10.1115/1.4048055]

This version is available at: https://hdl.handle.net/11585/792312 since: 2024-05-14

Published:

DOI: http://doi.org/10.1115/1.4048055

(Article begins on next page)

Investigation of Gasoline Compression Ignition for Combustion Control

Ravaglioli, Vittorio[1]
DIN – University of Bologna
Via Seganti 103, 47121, Forli, Italy
vittorio.ravaglioli2@unibo.it
ASME Member 000100225953

Ponti, Fabrizio
DIN – University of Bologna
Via Seganti 103, 47121, Forli, Italy
fabrizio.ponti@unibo.it
ASME Member 000008020240

De Cesare, Matteo
Marelli Europe SpA
Via Del Timavo 33, 40131, Bologna, Italy
matteo.decesare@marelli.com

ABSTRACT

Future emission regulations for Internal Combustion Engines require increasingly stringent reductions of engine-out emissions, especially NOx and particulate matter, together with the continuous improvement of engine efficiency. In the current scenario, even though compression-ignited engines are still considered the most efficient and reliable technology for automotive applications, the use of Diesel-like fuels has become a critical issue, since it is usually not compatible with the required emissions reduction. A large amount of research and experimentation is being carried out to investigate the combined use of compression-ignited engines and gasoline-like fuels, which proved to be very promising, especially in case the fuel is directly-injected in the combustion chamber at high pressure. This work investigates the combustion process occurring in a light-duty compression-ignited engine while directly injecting only gasoline. A specific experimental setup has been designed to guarantee combustion stability over the whole operating range, that is achieved controlling boost pressure and temperature together with all the injection parameters of the multi-jet pattern. The analysis of the experimental data clearly highlights how the variation of the control parameters affect the ignition process of small amounts of directly injected gasoline and the maximum achievable efficiency. In

[1] Corresponding author

particular, the analysis of the sensitivity to the injection parameters allows identifying an ignition delay model and the key control parameters that might be varied to guarantee a robust control of combustion phasing within the cycle.

1. INTRODUCTION

Compression-ignited engines are still considered the most efficient and reliable technology for road transportation. However, due to the current emission regulations, which severely limit pollutant and greenhouse emissions for internal combustion engines, the conventional use of this technology has become critical [1,2]. As a matter of fact, compression-ignited engines are usually fueled directly injecting Diesel at high pressure. This leads to a combustion process which is heterogeneous by nature, and consequently characterized by the simultaneous production of particulate matter (PM) and nitrogen oxides (NOx).

Over the past years, the need to increase the efficiency and minimize the environmental and health impact of internal combustion engines spread a large amount of research in the field of Low Temperature Combustions (LTC). These innovative combustion techniques overcome the main problems of Conventional Diesel Combustion (CDC) through the auto-ignition of a lean homogeneous air-fuel mixture (usually obtained through high ignition delays) and the use of gasoline-like fuels [3-5]. The best known LTC strategy is probably the Homogeneous Charge Compression Ignition (HCCI), which occurs when a totally pre-mixed air-fuel mixture is compressed to the point of auto-ignition. This combustion methodology proved to be effective to increase the thermal efficiency and simultaneously reduce both NOx and PM [6-9].

Even though the HCCI methodology seems to be promising, its use is severely limited by the fact that the Start of Combustion (SOC) is controlled by chemical kinetics, therefore the ignition delay of the air-fuel mixture is very sensitive to in-cylinder thermal conditions. As a result, the control of the combustion phasing is very difficult, mainly because slight temperature variations significantly modify the ignition delay, and consequently the center of combustion, which is the key factor to maximize the efficiency and avoid knock or misfire [10-12].

The discussed issues, characteristic of the HCCI process, can be mitigated partially modifying the air-fuel mixture preparation. Many works demonstrate that a promising technique suitable to partially overcome the limitations of HCCI combustion is the gasoline Partially Premixed Combustion (PPC), which uses a sequence of injections to properly design the shape of the rate of heat released. In this approach, the amount of fuel introduced with the first injection ignites as an HCCI combustion (high ignition delay strongly dependent on the cylinder thermal conditions) and increase in-cylinder pressure and temperature. Then, the following injections are activated in a very short time, due to the pressure and temperature increase generated by the previous combustion [13].

Even though combustion stability can be improved with the use of a multi-injection strategy, the optimization of the injection pattern and the optimal control of the combustion location within the cycle remain critical. As a matter of fact, the SOC of the first injection still plays a fundamental role, because it increases in-cylinder pressure and temperature, therefore defining the angular position at which the fuel introduced with the second injection will auto-ignite. For this reason, the optimal angular location of the first injection remains fundamental to properly control the entire combustion process.

To investigate the ignition mechanism of the first injection, this work analyzes how the combustion process of a small amount of gasoline (injected in a compression-ignited engine) is affected by the variation of the main control parameters and the environmental conditions. To do so, a 4-cylinder Diesel engine was modified to be run with 3 cylinders fueled with Diesel (CDC) and one cylinder fueled with gasoline (single injection of a small mass of fuel, comparable to the amount that would be injected in the first injection of a multi-jet pattern). To provide gasoline at high pressure, an additional fuel system was set up, together with a system for pressure control and injection management (only in one cylinder). Several experimental tests were carried out to highlight the effects of the injection parameters on the combustion process and the ignition delay of the injected gasoline. The identified correlations were finally used to experimentally identify a model suitable to predict the ignition delay.

2. EXPERIMENTAL LAYOUT

This study was based on the experimental activity run on a modified 1.3-liter compression-ignited turbocharged engine installed in a test cell. In its standard configuration, the engine under investigation is a Common-Rail Multi-Jet Diesel engine, equipped with an injection system designed to operate at a maximum pressure approximately equal to 1600 bar. The technical characteristics of this engine are summarized in Table 1.

TABLE 1: ENGINE TECHNICAL CHARACTERISTICS

Displaced volume	1248 cc
Maximum Torque	200 Nm @ 1500 rpm
Maximum Power	70 kW @ 3800 rpm
Injection System	Common Rail, Multi-
Bore	69.6 mm
Stroke	82 mm
Compression ratio	16.8:1
Number of Valves	4 per cylinder
Architecture	L4
Firing Order	1-3-4-2

The standard injection system, mainly based on a rail connected to the solenoid injectors and a high-pressure volumetric pump, was kept active and used to fuel three cylinders, while one cylinder (cylinder 1) was fueled with gasoline at high pressure.

To provide gasoline at high pressure, an additional fuel system was designed and installed. It consists in another volumetric pump, kept in motion by the crankshaft, and an additional high-pressure rail, connected only to the injector of cylinder 1 (same solenoid injector already used for CDC operation). Figure 1 reports the installation of the additional high-pressure injection system in the engine.

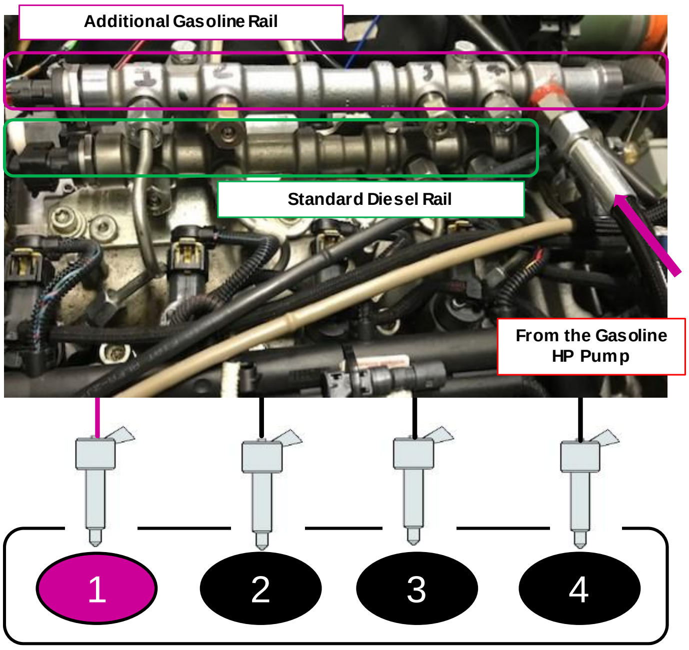

FIGURE 1: INSTALLATION OF THE ADDITIONAL HIGH-PRESSURE FUEL SYSTEM

The optimal management of gasoline injection has been obtained integrating the additional hardware in the testing setup shown in Figure 2. To guarantee the proper control of gasoline pressure and injected mass, which has to be completely flexible and independent from the mass of diesel injected in the other cylinders, a specifically designed control strategy was implemented in a programmable Rapid Control Prototyping system (RCP) based on a National Instruments cRio 9082. The new controller managed both gasoline injection pressure and energizing time (ET), i.e. the injected mass. Gasoline pressure was real-time controlled varying the duty of a PWM command provided to a by-pass solenoid valve, which was opened or closed according to the difference between target and measured gasoline pressure. The proper management of the injection command required the real-time knowledge of the crankshaft angular position, which was calculated by the RCP system processing the signal coming from the standard crankshaft speed sensor. Once the RCP determined the crank angle, it output the desired logical command for the injector, characterized by start (SOI) and duration (ET) of the injection (respectively start and duration of injector energizing). These quantities were communicated via CAN bus to the standard ECU, which converted the logical commands into the electrical command for the injector of cylinder 1.

To achieve a good level of flexibility in the investigation of gasoline auto-ignition and understand how the operating conditions affect the ignition delay of the mixture, it is necessary to have a good control of the intake air conditions, i.e. boost temperature and pressure. Intake temperature was varied using a controller that changed the mass flow of the liquid coolant introduced in the air cooler located after the centrifugal compressor (the coolant mass flow was varied according the difference between target and measured boost temperature). The boost pressure control was based on the indirect control of the exhaust pressure, that can be performed changing the load of the three cylinders operated with diesel. To control the load of these cylinders, the RCP communicated to the ECU (via CAN bus) both diesel energizing time (ET_D) and diesel start of injection (SOI_D) overwriting (when necessary) the reference values. Once the exhaust pressure reached a proper value, the position of the actuator that controlled the mass flow through the turbine (variable geometry turbine) was finally adjusted to keep the boost pressure at its target value.

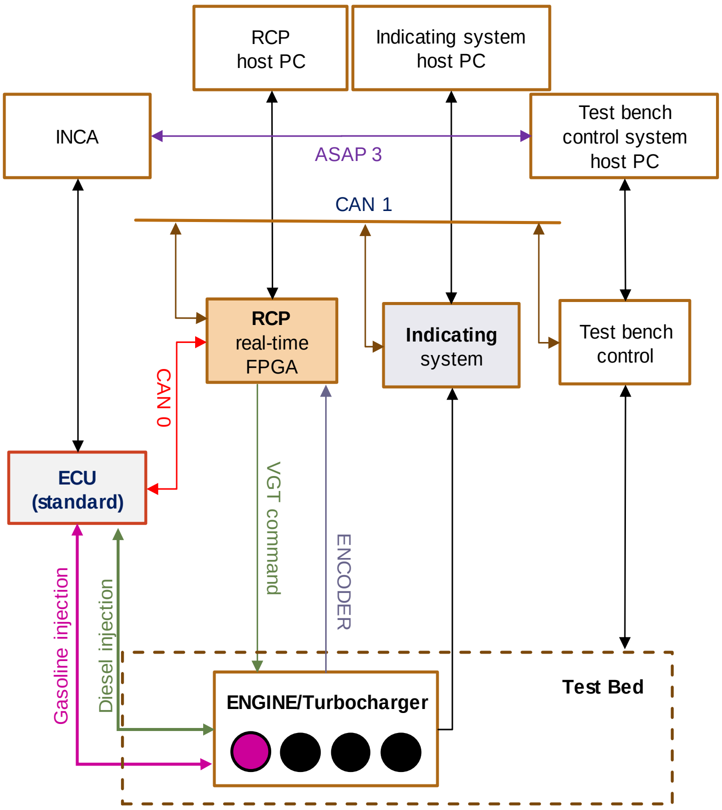

FIGURE 2: SCHEME OF THE EXPERIMENTAL SETUP USED TO INVESTIGATE GASOLINE PPC COMBUSTION

During the experimental tests, the signals coming from several standard and additional sensors installed in the engine were acquired. The standard sensors can be monitored using INCA software and ETAS hardware (connected to the ECU for engine control). In this work, the main additional sensors were 4 in-cylinder pressure sensors (AVL GH14P, one per cylinder), real-time acquired and processed by an indicating system that sampled the cylinder pressure signals at high frequency (100 kHz) and real-time calculated the main combustion indexes (center of combustion, indicated torque and pressure peak, all communicated to the RCP via CAN bus).

The discussed experimental setup was used to investigate gasoline auto-ignition in the engine under study, focusing the attention on the sensitivity of the ignition delay to the variation of several control parameters, such as intake temperature, intake air pressure and fuel pressure.

3. ANALYSIS OF THE COMBUSTION PROCESS

Once the engine layout has been modified to inject gasoline at high pressure into cylinder 1, several tests have been run to investigate how the variation of the main control parameters affect the gasoline ignition process. As already mentioned, this work focused the attention on the auto-ignition mechanisms which occur in gasoline PPC combustion, i.e. in the combustion process that follows a pattern with multiple injections of gasoline. In PPC combustions, the most critical aspect to be managed is that the ignition delay of the mass introduced with the first injection is strongly influenced by the cylinder thermal conditions. Once the first injected mass has been burned, the ignition delay of the following injections is significantly reduced by the increased pressure and temperature. As a result, the only way to keep the center of combustion at a proper target value is to properly correct the SOI of the first injection in a way that maintains reasonably stable its Start of Combustion.

With the prospect of setting up a model of the ignition delay (first injection), the analysis has been focused on the behavior of small amounts of fuel, i.e. the ones used in Pilot injections. In this work, the ignition of two different amounts of gasoline has been investigated: 2 and 4 mg/stroke. For each fuel mass, the effect of three different fuel pressures has been investigated, i.e. 300, 500 and 700 bar. The most critical aspect to be managed was the injection of gasoline, performed using the same solenoid injectors used in the standard operating mode for diesel injection. Consequently, the standard injector map (stored in the ECU) could not be used to accurately control the injected mass at the three investigated pressure levels (the ET stored in the standard map is not correct when gasoline is injected). Therefore, to accurately control the gasoline mass flow, gasoline consumption has been measured using a high accuracy flow meter (FlowSonic LF), characterized by a measurement range compatible with the small mass flow rates introduced inside cylinder 1 (the only cylinder in which gasoline is injected).

Each combination of gasoline mass and injection pressure has been tested with different combinations of intake pressure and temperature: two levels of intake pressure (1450 and 1550 mbar, the boost pressure is controlled real-time adjusting the position of the VGT actuator) and two levels of intake temperature (20 and 75 °C, controlled using the air cooler installed in the middle between the compressor and the intake manifold) have been investigated. During all the mentioned tests, several steady-state tests have been run changing the SOI from 50 to 10 deg BTDC. The whole set of experimental tests has been analyzed, the goal being to highlight the effect of each control parameter on heat release and ignition delay.

3.1 Effect of the Injection Parameters on Heat Release and Ignition

The calculation of the heat released during the combustion process is a useful way to quantify the effects of the ignition parameters on the ignition process. A commonly used approach for the calculation of the energy released during the combustion process, based on the experimental measurement of in-cylinder pressure, is reported in Eq. (1). Here, θ is the crankshaft angle, γ is the specific heat ratio, *p* and *V* are in-cylinder pressure and volume respectively.

$$ROHR = \frac{1}{\gamma-1} \cdot V \cdot \frac{dp}{d\theta} + \frac{\gamma}{\gamma-1} \cdot p \cdot \frac{dV}{d\theta} \qquad (1)$$

The rate of heat released (*ROHR*) does not take into consideration the losses due to heat transfers through the walls and blow-by. This aspect was not negligible in this study, because when small amounts of fuel are injected, the amount of heat exchanged through the crevices or the cylinder walls can be comparable to the one which produces a pressure increase captured by the transducer. If the amount of (positive) energy released during the combustion process is comparable to the negative contribution (blow-by and heat transfer through the walls), it might be difficult to perform detailed studies of the ignition process, because some of the phenomena of interest might be hidden by the losses.

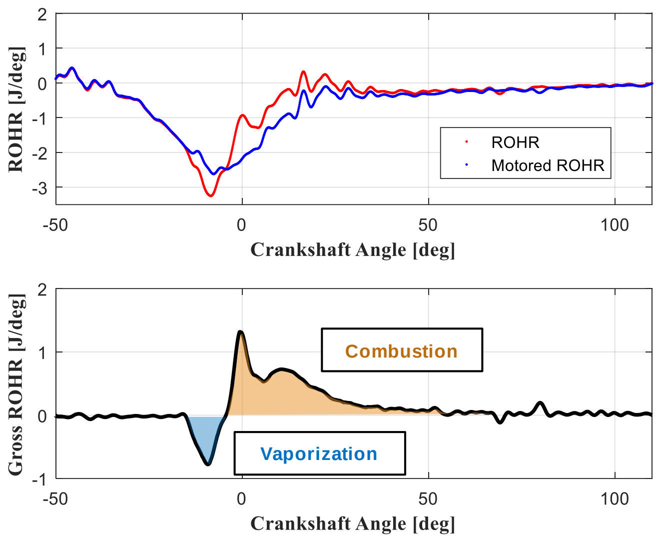

FIGURE 3: GROSS ROHR CALCULATION FOR A TEST RUN AT 2000 rpm, BOOST PRESSURE EQUAL TO 1550 mbar, INTAKE TEMPERATURE EQUAL TO 75°C AND pRail = 300 bar (4 mg/stroke).

To automatically compensate the mentioned negative contributions, each *ROHR* trace calculated from the pressure measurements (through Eq. (1)), acquired during the experimental activity, has been corrected removing the heat release trace calculated during a motored test (no fuel injected) run in the same condition of rotational speed, intake pressure and temperature. This procedure was applicable because it was possible to keep the boost pressure at its target value also when the injection was deactivated in cylinder 1 (this is done using the three cylinders fueled with diesel). Figure 3 demonstrates that the discussed procedure is suitable to compensate the effects of the losses, returning the gross *ROHR* [14] released during the combustion process. In addition, it is interesting to notice that the gross *ROHR* in Figure 3 shows a negative and a positive region. The negative part corresponds to the vaporization stage (negative because the fuel receives heat), while the positive part is the combustion stage (positive because the fuel releases heat).

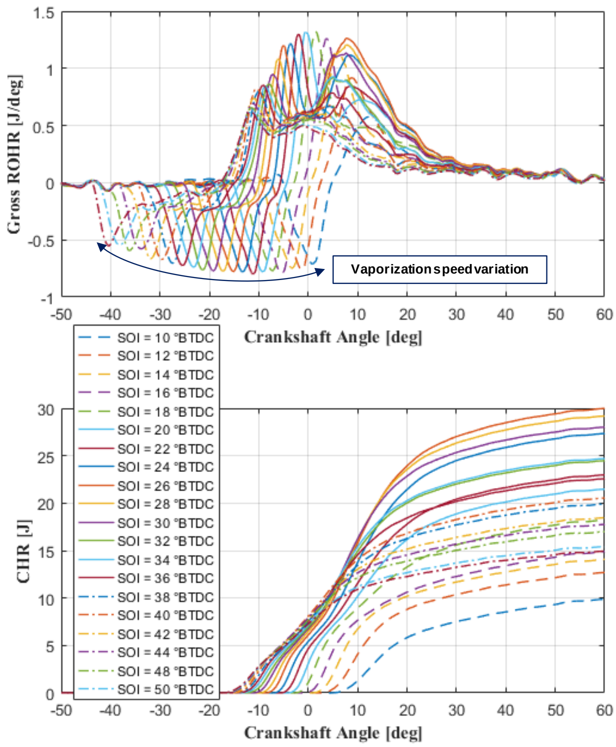

FIGURE 4: GROSS ROHR AND CHR VARIATIONS DURING A SOI SWEEP (2000 rpm, 4 mg/stroke, BOOST PRESSURE 1550 mbar, INTAKE TEMPERATURE 75°C AND RAIL PRESSURE 300 bar).

The use of the calculated gross release was fundamental to properly analyze the effects of intake conditions, injection pressure and SOI variation on the vaporization and combustion stage (for each amount of fuel injected). Given an investigated steady-state condition, the first aspect to be noticed is that SOI variation has a remarkable impact on both vaporization and combustion phase. To clarify this consideration, Figure 4 reports the result of the SOI sweep performed injecting 4 mg/stroke, keeping boost pressure at 1550 mbar, intake temperature at 75°C and rail pressure at 300 bar. As it can be observed, to obtain a good efficiency of the combustion stage, i.e. a high Cumulated Heat Release (CHR) (which is calculated as the cumulated sum of the positive ROHR region), it is necessary to select the optimal SOI (nearly 26 deg BTDC in this case). It is easy to notice that SOI also influenced the vaporization phase, since the vaporization speed (represented by the minimum values of the negative ROHR portion) is slow for very advanced injections (high SOI), reaches its maximum at intermediate SOI values and decreases again when a retarded SOI (near the TDC) is applied. This result is obviously due to the different average temperatures experienced by the fuel mass during its vaporization (given the injected mass, faster vaporizations correspond to higher temperatures).

The results reported in Figure 4 clarify the importance of the optimal SOI selection to guarantee a reasonable combustion efficiency for the small amounts of gasoline injected (with the prospect of using it as first injection of a multi-injection pattern), but the effect of this quantity has to be combined with the effects due to the variation of the other control parameters. As an example, Figure 5 shows that even if SOI was kept at a constant value (26 deg BTDC) the measured gross ROHR varied with the variation of the intake conditions. With high boost pressure and temperature, the efficiency of the combustion process was significantly higher (a diffusive combustion portion followed the first pre-mixed peak). When pressure or temperature were reduced, the combustion efficiency dropped (and the diffusive combustion portion disappears), probably because combustion propagation was slower and, consequently, the injected fuel had more time to propagate within the combustion chamber. As a result, the flame was quenched when it reached these too lean regions. This result suggests that, with the perspective of a multiple injection pattern, the first injection should be performed in operating conditions that guarantee good combustion stability and efficiency, i.e. the ones in which the first pre-mixed combustion is able to further propagate within the combustion chamber. To do so, a minimum level of intake pressure and temperature needs to be guaranteed (when possible), together with the proper choice of the injection phase.

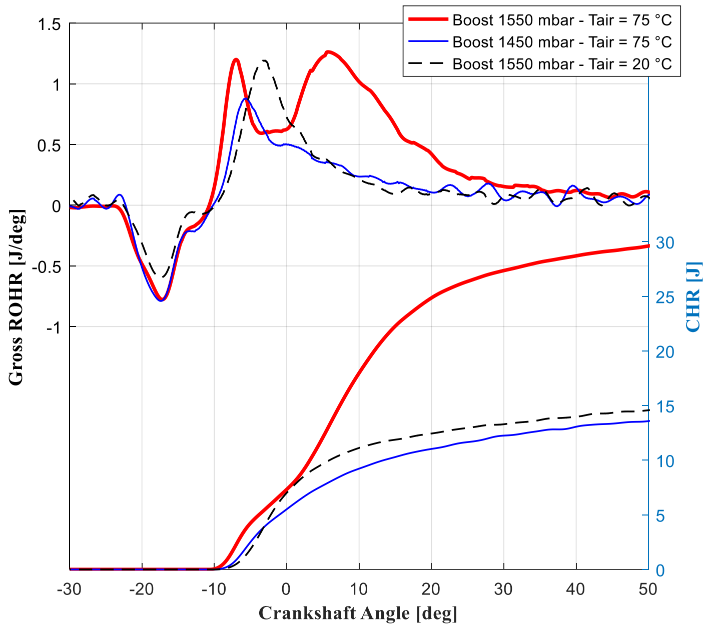

FIGURE 5: ROHR AND CHR TRACES FOR 3 TESTS RUN AT 2000 rpm, SOI EQUAL TO 26 deg BTDC, ALL RUN INJECTING 4 mg/stroke AT 300 bar AND CHANGING THE INTAKE CONDITIONS (PRESSURE AND TEMPERATURE).

The comparison of similar tests performed at different injection pressures highlights also the effect of this parameter on the combustion process. Figure 6 reports a comparison between three tests in which SOI and intake conditions have been kept approximately constant while varying only the injection pressure. As it can be observed, the pressure increase significantly accelerated the vaporization process, which resulted in more negative peaks of the calculated gross ROHR (higher fuel pressures guarantee a better air-fuel mixing and smaller fuel drops).

Another aspect to be noticed is that faster vaporization and better jet penetration also affected combustion efficiency (by varying the quality of the local air-fuel mixture). In this case, when the fuel pressure was increased, the diffusive combustion portion disappeared because the better air-fuel mixing accelerated the formation of ultra-lean regions in which the combustion process did not propagate. On the contrary, all the gross *ROHR* traces showed similar premixed portions, located in different angular position according to injection pressure. Given a fixed SOI, the first part of the combustion process was anticipated when the injection pressure was increased, which means that higher pressures reduced the ignition delay of the air-fuel mixture.

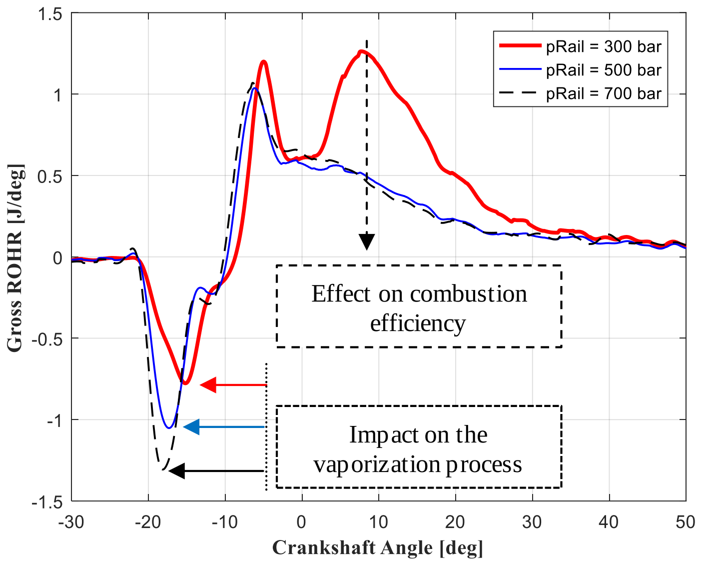

FIGURE 6: ROHR TRACES FOR 3 TESTS RUN AT 2000 rpm, SOI EQUAL TO 26 deg BTDC, ALL RUN INJECTING 4 mg/stroke AND CHANGING THE FUEL PRESSURE (INTAKE CONDITIONS ARE KEPT APPROXIMATELY CONSTANT).

All the above discussed comparisons show some of the main effects caused by the injection parameters' variation on ignition and combustion. As already discussed, one of the most critical aspects to be managed in PPC combustions is the prediction of combustion location, since the ignition process is strongly affected by slight variations of the control parameters. Therefore, based on the discussed experimental acquisitions, the following section focuses the attention on the analysis of the ignition delay, the goal being to set up a model suitable to compensate the variations of the injection parameters and correct SOI to keep nearly constant the start of the combustion process.

3.2 Identification of an Ignition Delay Model

Given a fuel, the auto-ignition delay (calculated as the time interval between SOI and SOC) is significantly affected by the local quality of the air-fuel mixture and the average pressure and temperature experienced by the injected fuel [14]. The analysis of the experimental data acquired from cylinder 1 can be used to model and summarize these dependencies.

To practically calculate the ignition delay from the gross ROHR waveforms, it has been computed as the time corresponding to the angular distance between SOI and the position in which the gross ROHR overcomes a fixed threshold, equal to 0.2 J/deg in this study.

If both intake conditions and fuel pressure are fixed, the ignition delay is mainly influenced by the injection phasing, because SOI variation changes the average cylinder pressure and temperature at which the fuel is exposed before its auto-ignition. If intake temperature and pressure are varied (together with SOI), also the ignition delay corresponding to one specific SOI will be varied, since the average pressure and temperature experience by the injected fuel will be modified. To clarify this consideration, Figure 7 reports the ignition delay sensitivity to simultaneous variations of SOI, intake pressure and temperature (test run injecting 4 mg/stroke at 300 bar). As expected, the minimum ignition delay was obtained during the sweep run at higher pressure and temperature.

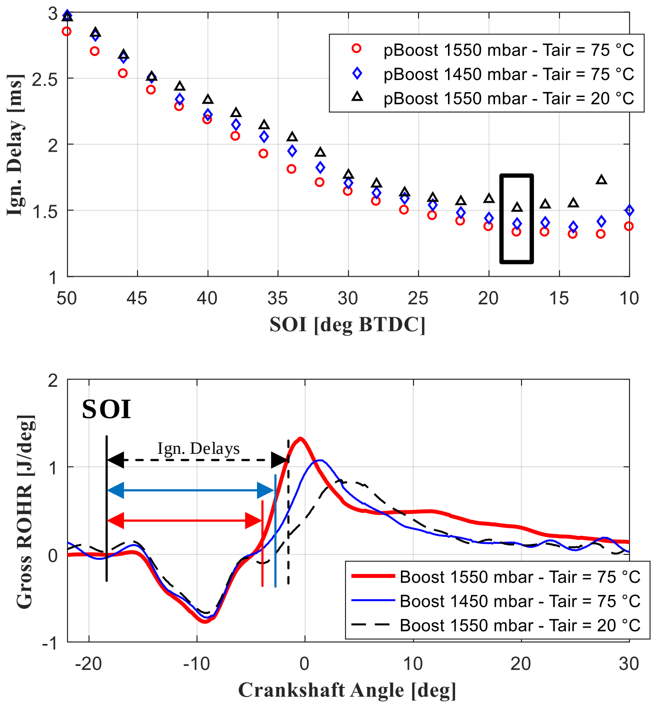

FIGURE 7: SOI SWEEPS RUN AT 2000 rpm, INJECTING 4 mg/stroke AND CHANGING THE INTAKE CONDITIONS (PRESSURE AND TEMPERATURE).

Even though a SOI retard usually reduces the ignition delay (motored pressure and temperature are increased), it is interesting to notice that for very retarded SOI values (lower than 15 deg BTDC in this work) the ignition delay tends to increase again. This is because part of the time interval between SOI and SOC occurs after the TDC, i.e. during the expansion stroke.

Bearing in mind the above considerations, the analysis of the ignition delay has been performed trying to correlate this quantity with an evaluation of the in-cylinder pressure and temperature experienced by the fuel mass after its injection. The temperature estimation can be performed combining the effects of a polytopic compression with the contribution provided by the hot residual gases trapped inside the cylinder after the previous cycle [15]. To do so, the cylinder pressure measured in correspondence of the Exhaust Valve Opening (EVO) can be used to calculate the corresponding temperature (T_{EVO}) through Eq. (2).

$$T_{EVO} = \frac{P_{EVO} V_{EVO}}{R m_{tot}} \qquad (2)$$

Then, the measurement of the average exhaust pressure (P_{exh}) can be used to determine the volumetric fraction of residual gases trapped inside the cylinder ($V_{\%EGR}$), it yields:

$$V_{\%EGR} = \frac{V_{cc}\left(\frac{P_{exh}}{P_{boost}}\right)^{\frac{1}{\gamma}}}{V_{cc}+V_d} \qquad (3)$$

Here, V_{cc} is the volume of the combustion chamber and V_d is the displaced volume. The calculated quantities (T_{EVO} and $V_{\%EGR}$) can be used to estimate the average charge temperature in correspondence of the intake valve closure (T_{IVC}) through Eq. (4):

$$T_{IVC} = \frac{T_{man} m_{AirQ} + T_{EVO} V_{\%EGR} m_{tot}}{m_{AirQ} + V_{\%EGR} m_{tot}} \qquad (4)$$

Finally, considering compression as polytropic allows determining the temperature in correspondence of the start of injection through Eq. (5).

$$T_{SOI} = T_{IVC}\left(\frac{V_{IVC}}{V_{SOI}}\right)^{\gamma-1} \qquad (5)$$

The calculated T_{SOI} provides information about the thermal conditions of the cylinder when the injection starts. Consequently, it is reasonable to expect that this quantity will be correlated with the ignition delay of the air-gasoline mixture. As it can be observed in Figure 8, the correlation between T_{SOI} and the ignition delay exists (the ignition delay usually decreases when the estimated temperature increases), but the level of correlation between the 2 quantities is very poor. As a matter of fact, the SOI sweeps performed with different intake conditions (boost pressure equal to 1450 or 1550 mbar and boost temperature equal to 20 or 75 °C, respectively "Tcold" and "Thot") show different trends, and these trends are not always coherent (no stable correspondence between temperature and delay).

In addition, the delays corresponding to the highest estimated temperatures tend to decrease with respect to the ones which correspond to lower temperatures. This is because the ignition delay is not only influenced by the temperature in correspondence of the SOI, but from the average temperature experienced from the fuel in the time interval between SOI and SOC.

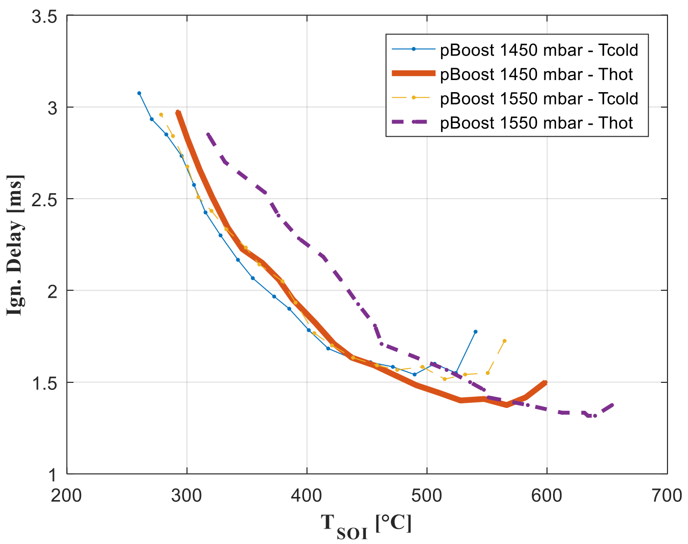

FIGURE 8: CORRELATION BETWEEN T_{SOI} AND IGNITION DELAY FOR TESTS RUN AT 2000 rpm, 300 bar, INJECTING 4 mg/stroke AND CHANGING THE INTAKE CONDITIONS (PRESSURE AND TEMPERATURE).

To overcome the discussed limitations, the temperature calculation has been modified to calculate the average temperature experienced by the fuel between SOI and SOC. The new estimated temperature T_{IGN} has been calculated through Eq. (6).

$$T_{IGN} = T_{IVC}\left(\frac{V_{IVC}}{\frac{\int_{\theta_{SOI}}^{\theta_{SOC}} V d\theta}{(\theta_{SOI}-\theta_{SOC})}}\right)^{\gamma-1} \qquad (6)$$

Furthermore, the same approach has also been used to estimate the average pressure in the angular range $(\theta_{SOI} - \theta_{SOC})$, starting from the pressure P_{IVC} measured by the cylinder pressure sensor in correspondence of the IVC, it yields:

$$P_{IGN} = P_{IVC}\left(\frac{V_{IVC}}{\frac{\int_{\theta_{SOI}}^{\theta_{SOC}} V d\theta}{(\theta_{SOI}-\theta_{SOC})}}\right)^{\gamma} \qquad (7)$$

Once P_{IGN} and T_{IGN} have been calculated, it is possible to plot the ignition delay as a function of both quantities. As it can be observed in Figure 9, the new quantities capture all the main effects due to the simultaneous variations of intake conditions (pressure and temperature) and injection phasing. This is valid for all the experimental tests run at constant amount of gasoline injected and constant fuel pressure. In particular, the experimental results reported in Figure 9 have been properly fitted using the polynomial correlation reported in Eq. (8), which quantifies, for the engine under study, how the variation of P_{IGN} and T_{IGN} affects the ignition delay (τ_{IGN_4}, in ms) of 4 mg of gasoline directly injected at 300 bar.

$$\tau_{IGN\_4} = \tau_{300\_4} - 0.0086 \cdot T_{IGN} - 0.026 \cdot P_{IGN} + 0.00032 \cdot T_{IGN} \cdot P_{IGN} + 0.0036 \cdot {P_{IGN}}^2 - 4.69 \cdot 10^{-6} \cdot T_{IGN} \cdot {P_{IGN}}^2 - 1.74 \cdot 10^{-6} \cdot {P_{IGN}}^3 \qquad (7)$$

Here, τ_{300_4} is a constant value approximately equal to 8.22.

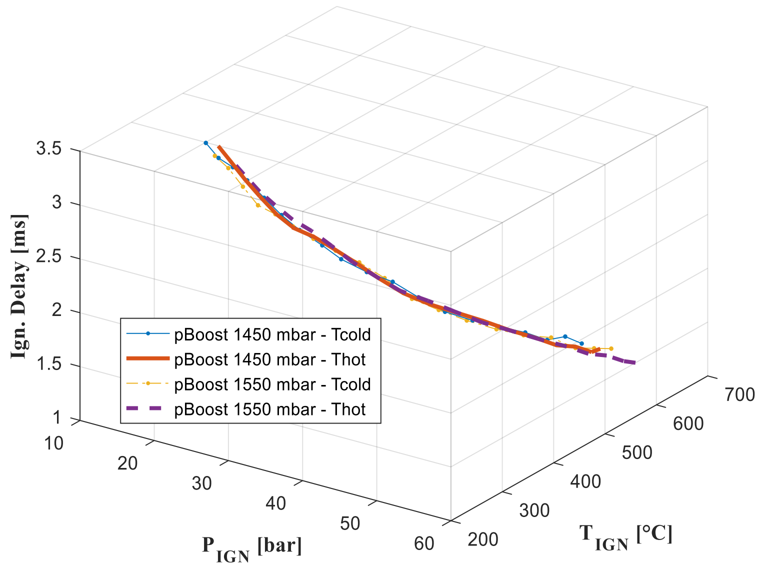

FIGURE 9: CORRELATION BETWEEN T_{IGN}, P_{IGN} AND IGNITION DELAY FOR TESTS RUN AT 2000 rpm, 300 bar, INJECTING 4 mg/stroke AND CHANGING THE INTAKE CONDITIONS (PRESSURE AND TEMPERATURE).

Similar correlations (as the one reported in Figure 9) can be obtained for all the investigated levels of rail pressures. As clearly visible in Figure 10, the main effect of a rail pressure increase is the reduction of the ignition delay, mainly due to the acceleration of the vaporization process already discussed in Figure 6. When the injection pressure is varied, the shape of the curves reported in Figure 10 remains nearly constant. For this reason, to easily quantify the impact of the injection pressure, the experimental results obtained with a rail pressure of 500 and 700 bar have been fitted using curves with the same shape reported in Eq. (8) and varying the constant coefficient (τ_{300_4} for the tests run injecting gasoline at 300 bar) to minimize the Root Mean Squared Error (RMSE) between measured and modeled ignition delay. This procedure led to the identification of τ_{500_4} and τ_{700_4}, respectively equal to 7.95 and 7.68. Using the discussed model, for the 3 investigated levels of rail pressure, the RMSE between measured and modeled ignition delay remained always lower that 0.037 ms.

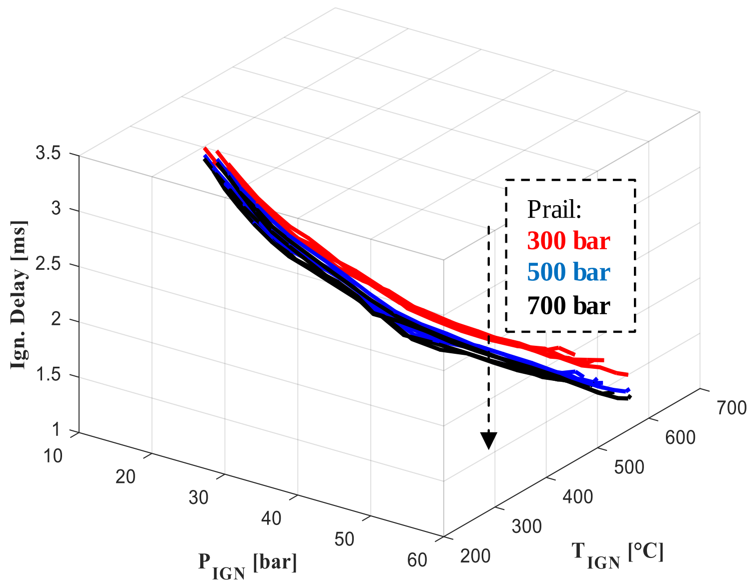

FIGURE 10: CORRELATION BETWEEN T_{IGN}, P_{IGN} AND IGNITION DELAY FOR TESTS RUN AT 2000 rpm, INJECTING 4 mg/stroke (ALL THE RAIL PRESSURES) WHILE CHANGING THE INTAKE CONDITIONS (PRESSURE AND TEMPERATURE).

Similar considerations are valid for the tests run injecting 2 mg/stroke, which also show a very strong correlation between T_{IGN}, P_{IGN} and the ignition delay. The reduction of the injected mass leads to a variation of the ignition delay (with respect to the injection of 4 mg/stroke), but the overall trends are similar to the ones summarized in Figure 9 and 10. For the investigated engine/layout, Eq. (9) quantifies how the variation of P_{IGN} and T_{IGN} affects the ignition delay (τ_{IGN_2}, in ms) of 2 mg of gasoline directly injected at 300 bar.

$$\tau_{IGN\_2} = \tau_{300\_2} - 0.0024 \cdot T_{IGN} - 0.23 \cdot P_{IGN} + 3.8 \cdot 10^{-5} \cdot T_{IGN} \cdot P_{IGN} + 0.005 \cdot {P_{IGN}}^2 - 1.78 \cdot 10^{-6} \cdot T_{IGN} \cdot {P_{IGN}}^2 - 2.84 \cdot 10^{-5} \cdot {P_{IGN}}^3 \qquad (9)$$

Here, τ_{300_2} is a constant value approximately equal to 6.35. To properly fit the ignition delay of the tests run with a rail pressure nearly equal to 500 and 700 bar, the constant value reported in Eq. (9) has been reduced to 6.12 (τ_{500_2}) and 6.01 (τ_{700_2}) respectively.

CONCLUSION

This work analyzes the ignition mechanisms of small gasoline quantities directly injected in a compression ignited engine, the goal being to define a methodology suitable to model the ignition delay of the fuel. The obtained results show the existence of a strong correlation between the ignition delay of the fuel and the in-cylinder temperature and pressure experienced by the air-fuel charge. In particular, fixed the fuel injection pressure, it is possible to accurately model the variations of the ignition delay through the variations of average pressure and temperature in the angular range between start of injection (SOI) and start of combustion (SOC). In case of injection pressure variations, the identified model needs to be adjusted (mainly through a constant offset) to compensate the main effects of rail pressure variations on the ignition delay, such as the acceleration of fuel vaporization due to an increase of the injection pressure (higher rail pressures generate smaller fuel drops, which need less time to vaporize and, consequently, to ignite). Since the fuel quantities investigated in this paper are the ones normally used in the first injection of a multi-jet pattern for gasoline PPC combustion, the identified correlations could be used to set-up a real-time ignition delay model aimed at automatically compensating the effects of the variations of control parameters and intake conditions. This could be done correcting the start of gasoline injection in a way that keeps reasonably constant the start of combustion.

NOMENCLATURE

Acronyms

BTDC	Before the top dead center
CAN	Controller area network
CDC	Conventional diesel combustion
CHR	Cumulated heat release
ECU	Electronic control unit
EVO	Exhaust valve opening
ET	Energizing time
HCCI	Homogeneous charge compression ignition
LTC	Low temperature combustion
PM	Particulate matter
PPC	Partially premixed combustion
RCP	Rapid control prototyping
RMSE	Root mean squared error
ROHR	Rate of heat release
SOC	Start of combustion
SOI	Start of injection

Symbols

γ	Specific heat ratio
p	In-cylinder pressure
V	In-cylinder volume
θ	Crankshaft angle
τ_{ign}	Ignition delay
τ_{300_2}	Ignition delay offset for 2 mg and *pRail*=300 bar
τ_{500_2}	Ignition delay offset for 2 mg and *pRail*=500 bar
τ_{700_2}	Ignition delay offset for 2 mg and *pRail*=700 bar
τ_{300_4}	Ignition delay offset for 4 mg and *pRail*=300 bar
τ_{500_4}	Ignition delay offset for 4 mg and *pRail*=500 bar
τ_{700_4}	Ignition delay offset for 4 mg and *pRail*=700 bar
ET_D	Diesel energizing time
m_{tot}	Total mass trapped in the cylinder
$pRail$	Rail pressure
P_{EVO}	Pressure at the exhaust valve opening
P_{exh}	Exhaust pressure
P_{SOI}	Pressure at the start of injection
P_{IGN}	Average pressure during the ignition delay
R	Universal gas constant
SOI_D	Diesel start of injection

T_{IGN}	Average temperature during the ignition delay
T_{EVO}	Temperature at the exhaust valve opening
T_{SOI}	Temperature at start of injection
V_{SOI}	Volume at the start of injection
$V_{\%EGR}$	Volumetric fraction of residual gases
V_{cc}	Volume of the combustion chamber
V_d	Displaced volume
V_{EVO}	Volume at the exhaust valve opening

REFERENCES

(Samples of the most commonly referenced materials are provided. If in doubt, please refer to the latest editor of the *Chicago Manual of Style*. DOIs should be provided whenever possible for the greatest accuracy.

[1] Torregrosa, A.J.; Broatch Jacobi, J.A.; García Martínez, A.; Monico Muñoz, L.F. (2013). “Sensitivity of Combustion Noise and NOx and Soot Emissions to Pilot Injection in PCCI Diesel Engines,” Applied Energy. **104**:149-157.

[2] Kolbeck, A.F., "Closed Loop Combustion Control - Enabler of Future Refined Engine Performance Regarding Power, Efficiency, Emissions and NVH under Stringent Governmental Regulations", SAE Technical Paper 2011-24-0171, 2011, doi:10.4271/2011-24-0171.

[3] Curran, S., Hanson, R., Wagner, R., and Reitz, R., "Efficiency and Emissions Mapping of RCCI in a Light-Duty Diesel Engine," SAE Technical Paper 2013-01-0289, 2013, doi:10.4271/2013-01-0289.

[4] Wissink, M. and Reitz, R., "Direct Dual Fuel Stratification, a Path to Combine the Benefits of RCCI and PPC," SAE Int. J. Engines **8**(2):878-889, 2015, doi:10.4271/2015-01-0856.

[5] Li, C., Xu, L., Bai, X.-S., Tunestal, P., and Tuner, M., “Effect of Piston Geometry on Stratification Formation in the Transition from HCCI to PPC,” SAE Technical Paper 2018-01-1800, 2018, doi:10.4271/2018-01-1800.

[6] Dempsey, A. B., Curran, S. J., and Wagner, R. M. (2016). “A Perspective on the Range of Gasoline Compression Ignition Combustion Strategies for High Engine Efficiency and Low NOx and Soot Emissions: Effects of In-cylinder Fuel Stratification”. International Journal of Engine Research, **17**(8), 897–917. doi:10.1177/1468087415621805.

[7] Kimura, S., Aoki, O., Kitahara, Y., and Aiyoshizawa, E., "Ultra-Clean Combustion Technology Combining a Low-Temperature and Premixed Combustion Concept for Meeting Future Emission Standards," SAE Technical Paper 2001-01-0200, 2001, doi:4271/2001-01-0200.

[8] Kokjohn, S., Hanson, R., Splitter, D., and Reitz, R., "Experiments and Modeling of Dual-Fuel HCCI and PCCI Combustion Using In-Cylinder Fuel Blending," SAE Int. J. Engines **2**(2):24-39, 2010, doi:10.4271/2009-01-2647.

[9] Ravaglioli, V., Ponti, F., De Cesare, M., Stola, F., Carra, F., and Corti, E., "Combustion Indexes for Innovative Combustion Control," SAE Int. J. Engines **10**(5):2371-2381, 2017, doi:10.4271/2017-24-0079.

[10] Masurier, J., Waqas, M., Sarathy, M., and Johansson, B., "Autoignition of Isooctane beyond RON and MON Conditions," SAE Int. J. Fuels Lubr. **11**(4):459-468, 2018, doi:10.4271/2018-01-1254.

[11] Gentz, G., Dernotte, J., Ji, C., and Dec, J. "Spark Assist for CA50 Control and Improved Robustness in a Premixed LTGC Engine Effects of Equivalence Ratio and Intake Boost," SAE Technical Paper 2018-01-1252, 2018, doi:10.4271/2018-01-1252.

[12] An, Y., Mubarak Ali, M.J., Vallinayagam, R., AlRamadan, A., Sim, J., Chang, J., Im, H., Johansson, B., "Compression Ignition of Low Octane Gasoline under Partially Premixed Combustion Mode," SAE Technical Paper 2018-01-1797, 2018, doi:10.4271/2018-01-1797.

[13] Matsuura, K. and Iida, N., "Effect of Temperature-Pressure Time History on Auto-Ignition Delay of Air-Fuel Mixture," SAE Technical Paper 2018-01-1799, 2018, doi:10.4271/2018-01-1799.

[14] Heywood, John B. *Internal Combustion Engine Fundamentals*. New York: McGraw-Hill, 1988.

[15] Ravaglioli V., Ponti F., Carra F., De Cesare M., "Heat Release Experimental Analysis for RCCI Combustion Optimization". *ASME Internal Combustion Engine Division Fall Technical Conference*, doi:10.1115/ICEF2018-9714

APPENDIX A. UNCERTANTIES

Further information about the additional sensors used by the authors to measure:

1. In-cylinder pressure: necessary to determine ignition delay (through the calculation of the gross rate off heat release), T_{IGN} and P_{IGN}.

TABLE 2: TECHNICAL CHARACTERISTICS OF THE IN-CYLINDER PRESSURE SENSORS

Sensor Name	AVL GH14P
Measuring range	0-250 bar
Overload	300 bar
Sensitivity	15 pC/bar
Linearity	≤ ± 0.3%
Calibrated ranges	0 ... 80 bar 0 ... 150 bar
Natural frequency	115 kHz

2. Fuel consumption: necessary to control the amount of fuel injected (2 or 4 mg/stroke at different rail pressures).

TABLE 3: TECHNICAL CHARACTERISTICS OF THE FUEL FLOW METER

Sensor Name	Flowsonic LF
Repeatability	+/- 0.15% of reading
Uncertainty	+/- 0.5% of reading
Measurement flow range	8-4000 ml/min
Measurement rate	2.2 kHz
Fluid temperature range	-20°C to +120°C
Ambient temperature	-40°C to +120°C

Figure Captions List

Fig. 1	Installation of the additional high-pressure fuel system.
Fig. 2	Scheme of the experimental setup used to investigate gasoline PPC combustion
Fig. 3	Gross ROHR calculation for a test run at 2000 rpm, boost pressure equal to 1550 mbar, intake temperature equal to 75°C and pRail = 300 bar (4 mg/stroke).
Fig. 4	Gross ROHR and CHR variations during a SOI sweep (2000 rpm, 4 mg/stroke, boost pressure 1550 mbar, intake temperature 75°C and rail pressure 300 bar).
Fig. 5	ROHR and CHR traces for 3 tests run at 2000 rpm, SOI equal to 26 deg BTDC, all run injecting 4 mg/stroke at 300 bar and changing the intake conditions (pressure and temperature).
Fig. 6	ROHR traces for 3 tests run at 2000 rpm, SOI equal to 26 deg BTDC, all run injecting 4 mg/stroke and changing the fuel pressure (intake conditions are kept approximately constant).
Fig. 7	SOI sweeps run at 2000 rpm, injecting 4 mg/stroke and changing the intake conditions (pressure and temperature).
Fig. 8	Correlations between T_{SOI} and ignition delay for tests run at 2000 rpm, 300 bar, injecting 4 mg/stroke and changing the intake conditions (pressure and temperature).

Fig. 9 Correlation between T_{IGN}, P_{IGN} and ignition delay for tests run at 2000 rpm, 300 bar, injecting 4 mg/stroke and changing the intake conditions (pressure and temperature).

Fig. 10 Correlation between T_{IGN}, P_{IGN} and ignition delay for tests run at 2000 rpm, injecting 4 mg/stroke (all the rail pressures) while changing the intake conditions (pressure and temperature).

Table Caption List

Table 1 Engine technical characteristics

Displaced volume	1248 cc
Maximum Torque	200 Nm @ 1500 rpm
Maximum Power	70 kW @ 3800 rpm
Injection System	Common Rail, Multi-
Bore	69.6 mm
Stroke	82 mm
Compression ratio	16.8:1
Number of Valves	4 per cylinder
Architecture	L4
Firing Order	1-3-4-2

Table 2 Technical characteristics of the in-cylinder pressure sensors

Sensor Name	AVL GH14P
Measuring range	0-250 bar
Overload	300 bar
Sensitivity	15 pC/bar
Linearity	≤ ± 0.3%
Calibrated ranges	0 ... 80 bar 0 ... 150 bar
Natural frequency	115 kHz

Table 3 Technical characteristics of the fuel flow meter

Sensor Name	Flowsonic LF
Repeatability	+/- 0.15% of reading
Uncertainty	+/- 0.5% of reading
Measurement flow range	8-4000 ml/min
Measurement rate	2.2 kHz
Fluid temperature range	-20°C to +120°C
Ambient temperature	-40°C to +120°C