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Investigation of Gasoline Partially Premixed Combustion with External Exhaust Gas Recirculation

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Abstract

The stringent emission regulations for Internal Combustion Engines (ICEs) spawned a great amount of research in the field of innovative combustion approaches characterized by high efficiency and low emissions. Previous research demonstrate that such promising techniques, named Low-Temperature Combustion (LTC), combine the benefits of Compression Ignition (CI) engines, such as high compression ratio and unthrottled lean mixture, with low engine-out emissions using a properly premixed air-fuel mixture. Due to longer ignition delay and high volatility compared to diesel, gasoline-like fuels show good potential for the generation of a highly premixed charge, which is needed to reach LTC characteristics. In this scenario, gasoline Partially Premixed Combustion (PPC), characterized by the high-pressure direct injection of gasoline, showed good potential for the simultaneous reduction of pollutants and emissions in CI engines. However, previous research on gasoline CI highlight that a key factor for the optimization of both efficiency and pollutants is the proper management of Exhaust Gas Recirculation (EGR). This work presents the experimental investigation performed running a light-duty CI engine, operated with gasoline PPC, and varying the mass of recirculated gases trapped in the combustion chamber. To guarantee the stability of gasoline autoignition in all the tested conditions, a specific experimental layout has been developed to accurately quantify the amount of trapped residual gases due to the internal and external EGR. The obtained results clearly highlight the impact of EGR on the combustion process and emissions, demonstrating that optimization of charge dilution with EGR is fundamental to guarantee the optimal compromise between efficiency and emissions over the whole operating range.

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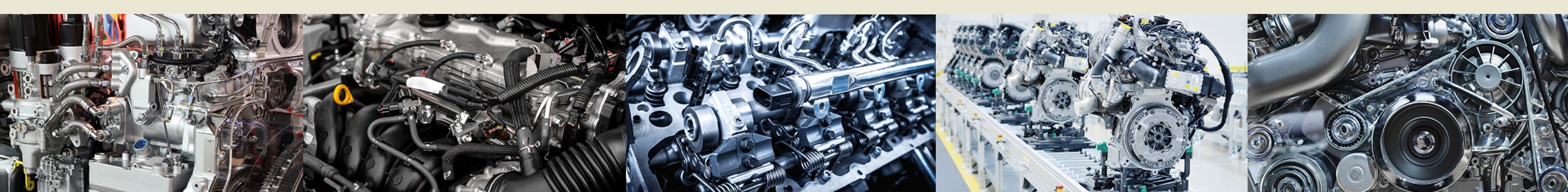


Introduction

Throughout the world, transport accounts for around one-fifth of global carbon dioxide (CO_2) emissions, and over 75% of those come from road vehicles [1]. Therefore, all transport legislation boards have fixed stringent regulations in terms of pollutant emissions going toward zero-carbon transport. To achieve such a challenging aim, transport industries have developed different solutions such as hybrid, plug-in, and electric vehicles [1, 2]. However, considering the current limited autonomy of electric vehicles and their environmental impact from a cradle-to-grave standpoint [3], there is still a need to develop internal combustion engines (ICEs) characterized by high thermal efficiency and low pollutants for hybrid powertrains.

To meet such requirements, many researchers have focused on the benefits provided by new combustion concepts, called low-temperature combustions (LTC), in terms of efficiency and pollutants. Based on the combustion of lean air-fuel mixtures under unthrottled conditions and high compression ratios, those techniques have shown a considerable potential to reach the targets imposed by the legislator. homogeneous-charge compression ignition (HCCI) is the most investigated LTC approach, characterized by compression ignition of fully homogeneous air-fuel mixture [4]. As a result, HCCI has great potential in terms of thermal efficiency and pollutants reduction, but at the same time, its combustion process presents high impulsiveness and low controllability because it is driven by chemical kinetics and thermodynamic conditions which limit its applicability [5, 6].

A suitable approach to overcome HCCI applicability limits is Gasoline partially premixed combustion (PPC). This process is characterized by the CI of gasoline directly injected into the combustion chamber under unthrottled conditions [7, 8, 9]. Many works demonstrate that PPC can be controlled using a proper sequence of high-pressure multiple fuel jets, typically composed by the pilot and main injections [10, 11, 12]. As for conventional diesel combustion (CDC), the combustion of small amounts of gasoline introduced with pilot injections increases both the combustion chamber pressure and temperature, reducing the ignition delay of the following fuel jets. As a matter of fact, only the first fuel jet burns as an HCCI combustion [13], while the following injections burn in stratified conditions, thus generating a smoother combustion shape. As a result, using multiple injections improves the controllability of the whole combustion process, especially the center of combustion and torque delivered [14, 15, 16]. Furthermore, it is important to notice that the use of low-lubricating fuels, such as gasoline, affects the reliability of the standard high-pressure diesel pump, also increasing energy losses and brake-specific fuel consumption (BSFC). To overcome the mentioned reliability-related problems, specifically designed high-pressure gasoline direct injection (GDI) systems (both pump and injectors), suitable to manage closely spaced multiple injections per cycle with optimal dynamics, were developed [17].

Despite the improved controllability given by the use of multiple gasoline injections, the stratification of PPC might have a negative impact on engine-out emissions, especially on nitrogen oxides (NOx) and particulate matter [18, 19]. A large amount of research have been carried out to develop technical solutions suitable to achieve a stable and controllable PPC while avoiding relevant negative impacts on emissions and thermal efficiency. Several works highlight the potential of exhaust gas recirculation (EGR) combined with PPC and demonstrate the effectiveness to achieve a significant NOx reduction without using complex and expensive aftertreatment systems [15, 19].

This work provides a further contribution to this field of research, investigating the effect of EGR on PPC (carried out in a light-duty CI engine) and analyzing the effect of different recirculated masses on efficiency and emissions. The obtained results can be used to identify a proper combination of injection pattern and EGR mass needed to reach the best compromise between emissions and fuel consumption (at different speeds and loads). Finally, the performance in the identified operating conditions can be compared to the performance of the engine run in standard calibrated diesel mode, the goal being to quantify the potential of gasoline PPC with controlled external EGR quantity with regard to conventional diesel combustion.

To test the gasoline PPC, a 1.3L light-duty turbocharged diesel engine has been modified, adding an external supercharger and a diathermic oil heater to control the intake temperature and pressure over the whole engine operating range. Furthermore, the intake system has been modified adding an ultrasonic mass flowmeter, fundamental to accurately estimate the EGR rate (used during the whole experimental activity) through a specifically designed cylinder filling model. The impact of EGR rate on emissions and efficiency has been investigated, varying the EGR valve position while running the engine in gasoline PPC mode at different speeds and loads.

Experimental Setup

The experimental activity was conducted on a 1.3L four-cylinder turbocharged CI engine installed in a test cell. The main technical characteristics of the engine under study are summarized in Table 1.

The engine is equipped with a variable geometry turbine actuator (VGT), suitable to manage the intake pressure, and a high-pressure EGR valve, which recirculates exhaust gases to the intake manifold.

The injection system used for gasoline injection is the standard common-rail multi-jet high-pressure system with four solenoid injectors, center-mounted, one for each cylinder. To obtain a robust PPC, the standard multi-jet system performs multiple gasoline injections (per engine cycle) at high pressure (rail pressures up to 1000 bar were tested while operating the engine with gasoline).

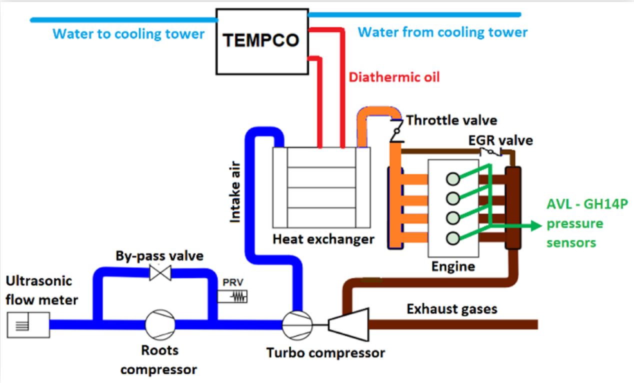
TABLE 1 Engine technical characteristics.

Displaced volume	1248 cc
Maximum torque	200 Nm at 1500 rpm
Maximum power	70 kW at 3800 rpm
Injection system	Common rail, multi-jet
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

As discussed in the literature [15, 19], gasoline PPC needs a high boost pressure and intake temperature to guarantee a robust gasoline autoignition. This aspect becomes crucial especially during cranking, idle, or at low loads when the exhaust gas energy is not enough to drive the turbocharger and achieve the boost pressure needed for a stable PPC. To overcome this problem, the original intake system was modified adding a roots blower (Eaton Compressor M24) upstream of the dynamic compressor. The volumetric compressor is driven by an electric motor (5.5 kW and maximum rotational speed equal to 3000 rpm) controlled by the engine control unit (ECU) to guarantee a sufficient boost pressure before the engine starts. Once the engine overcomes the cranking stage and exhaust gas energy is enough for the centrifugal compressor to reach the speed required to control boost pressure directly with the VGT, the compressor is switched off and bypassed by an auxiliary duct. Therefore, no negative contribution in terms of energy spent on the volumetric compressor needs to be considered in the evaluation of combustion efficiency. Besides boost pressure, many works show that intake temperature also plays a crucial role in gasoline PPC stability [20, 21, 22]. For these reasons, to accurately control the intake air temperature, a high-temperature diathermic oil thermoregulation unit (TEMPCO T-REG HCE 609/15-O) has been installed midway between the centrifugal compressor and the intake manifold. The external unit allowed the accurate management of the intake air temperature in any engine operating conditions, which was also beneficial to stabilize the combustion process in cold-start conditions. Figure 1 shows the integration of these two components and the complete experimental setup of the engine in the test cell.

The aim of this work is to study the effect of EGR on gasoline PPC in terms of combustion stability and pollutants. To guarantee maximum flexibility in the management of both engine and actuators, all the necessary control strategies have been implemented in a fully programmable ECU (SPARK by Alma Automotive). The use of this open ECU, based on National Instruments hardware and programmable via LabView software, allowed managing the engine (and all the subsystems) to overcome the limitations that usually occur when a production ECU with standard control software is used. Standard sensors were acquired by the ECU, and all the actuators mounted on the engine (i.e., VGT position,

FIGURE 1 Scheme of the experimental setup developed to run gasoline PPC in all four cylinders [27].



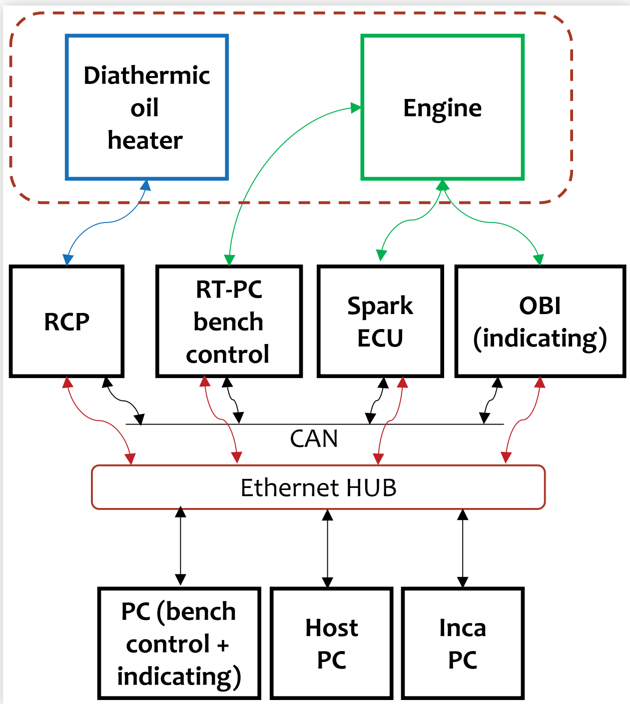
injections, fuel pressure) were managed following the implemented strategies. To improve the testing operations, the engine was controlled and all standard sensors were logged using INCA software and ETAS hardware, directly communicating with the ECU. To thoroughly investigate gasoline PPC, besides standard sensors, the engine was also equipped with pressure and temperature sensors placed on the intake and exhaust pipes. All additional signals have been acquired by the test bench control system used as input/feedback for the control algorithms.

To monitor the combustion process, the engine has been equipped with four in-cylinder pressure sensors (AVL GH14P), one per cylinder, acquired at 200 kHz and real-time analyzed by the indicating system to calculate the main combustion indexes such as indicated mean effective pressure (IMEP), CA50, pressure peak value, and location. To further improve combustion stability during the tests, the calculated combustion indexes have been sent to the ECU via the controller area network (CAN) bus. The availability of the calculated combustion indexes was necessary to run specific closed-loop combustion control strategies, suitable to operate the engine at the target center of combustion and load.

As well known [19], the use of EGR in gasoline PPC strongly affects the combustion process and, consequently, the engine-out emissions. To measure the emissions and investigate their sensitivity to the EGR rate, a Continental (SNS14) NOx sensor and an AVL Smoke Meter (415S) have been mounted in the exhaust line. To accurately quantify the combustion thermal efficiency, the instantaneous fuel consumption was measured using an ultrasonic fuel flowmeter (FlowSonic FFM LF DP-010-02) installed in the low-pressure fuel line.

Finally, the evaluation of the EGR mass flow was performed indirectly, as the difference between the total mass flow introduced in the cylinders (air and external EGR) and the air mass flow. The fraction of fresh air aspirated by the engine was directly measured using an ultrasonic air mass flowmeter (FEV AirRate 100) installed at the beginning of the intake system, before the additional volumetric supercharger. With regard to the total mass flow introduced, it was estimated

FIGURE 2 Scheme of an engine at test bed control systems [27].



using a specifically developed cylinder filling model, which will be discussed in detail in the following section.

Figure 2 shows the control systems and test cell communication layout developed to properly manage the gasoline PPC.

Cylinder Filling Model

The above-described testing layout, specifically developed for this work, was used to run the engine with gasoline PPC, the goal being to study EGR effects on thermal efficiency and pollutants.

$$EGR_{Rate} = \frac{m_{EGR}}{m_{Cyl}} \quad \text{Eq. (1)}$$

To properly investigate the effect of EGR under different revolutions per minute (rpm) and load conditions, it was necessary to quantify the EGR mass and calculate the so-called EGR rate defined in Equation 1, in which m_{Cyl} and m_{EGR} are the total mass trapped inside the cylinder and the mass of recirculated gases, respectively.

To determine the defined EGR rate, it is first necessary to quantify the total mass trapped in the cylinder (m_{Cyl}). For this purpose, a filling model able to predict the total amount of air and external EGR trapped in the cylinder has been developed. The model is based on the measurements of in-cylinder pressure and airflow (from the additional ultrasonic flowmeter). In general, the in-cylinder trapped mass is composed of the sum of fresh air m_{Air}, internal residual gases m_{RG} and external EGR m_{EGR}, as shown in Equation 2.

$$m_{Cyl} = m_{Air} + m_{RG} + m_{EGR} \quad \text{Eq. (2)}$$

First, the model was developed to keep the EGR valve closed. Applying the well-known ideal gas equation, the contributions of fresh air and internal residuals have been estimated. As shown in Equations 3 and 4, these masses can be determined by referring to specific pressures and temperatures measured in different stages of the engine cycle. Residual gases were calculated as the mass trapped inside the combustion chamber at the end of the exhaust stroke (V_{CC}), when the piston reaches the top dead center (TDC). In this condition, it is possible to consider the in-cylinder pressure and temperature approximately constant and equal to the values acquired by the pressure and temperature sensors mounted on the exhaust line, P_{Exh} and T_{Exh}, respectively.

$$m_{RG} = \frac{P_{Exh} V_{CC}}{R T_{Exh}} \quad \text{Eq. (3)}$$

Once the internal residuals have been quantified, the estimation of the total charge can be based on the information provided by the in-cylinder pressure sensors. During the intake stroke, different contributions impact the cylinder filling process. Previous research show that residuals and valve timing have a strong impact on cylinder filling, mainly by reducing the effective volume compared to the maximum geometrical volume (V_{Cyl}) [23, 24]. For these reasons, two volume reduction contributions (with respect to V_{Cyl}) have been introduced, which represent the volume loss as a result of the expansion of the hot residuals from the exhaust to the intake pressure (ΔV_1) and the volume loss due to the late intake valve closure (ΔV_2, difference between the cylinder volume in correspondence to the bottom dead center and the volume in correspondence with the intake valve closure). Equations 4 and 5 define these two volume reductions.

$$\Delta V_1 = V_{CC} \left(\frac{P_{Exh}}{P_{Int}} \right)^{1/\gamma} - V_{CC} \quad \text{Eq. (4)}$$

$$\Delta V_2 = V_{BDC} - V_{IVC} \quad \text{Eq. (5)}$$

In the discussed model, P_{Int} was considered constant and equal to the mean value of the measured in-cylinder pressure trace during the intake stroke (intake valve open), while the charge temperature was set equal to the measurement provided by the sensor installed in the intake manifold (T_{Int}). Finally, the gas constant R was set at 287.05 J/(kg K). Equation 6 summarizes the cylinder filling model used to estimate the mass of fluid introduced in the cylinder at intake valve closing (IVC) (from the intake manifold), i.e., the sum of fresh air and external EGR.

$$m_{Air} + m_{EGR} = \frac{P_{Int} \left(V_{Cyl} - \Delta V_1 - \Delta V_2 - V_{CC} \right)}{R T_{Int}} \quad \text{Eq. (6)}$$

FIGURE 3 Cylinder filling model behavior and validation during EGR valve sweeps and charge composition (fresh air, residuals, and external EGR). The VGT actuator movement allows to keep the boost pressure at its target value.

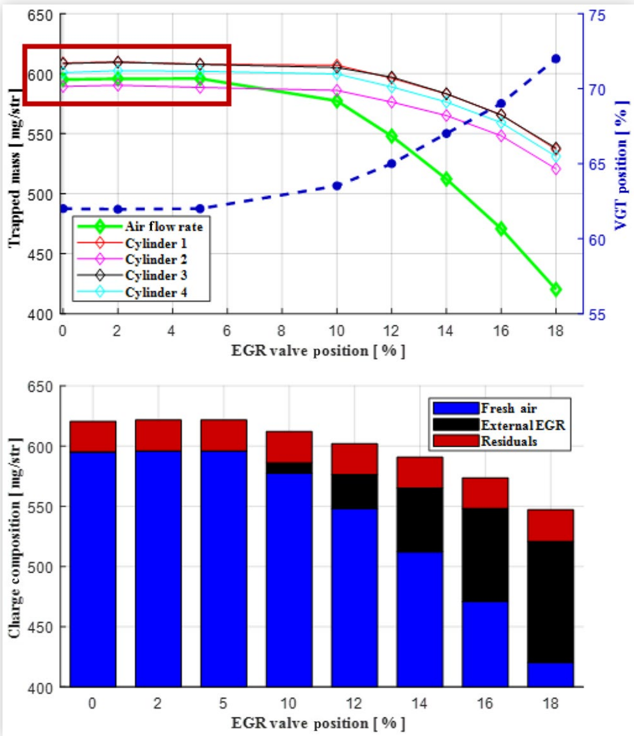


Figure 3 shows a comparison between the total mass trapped in the cylinder (estimated using the model reported in Equation 6) and the measurement performed by the air flowmeter as a function of the EGR valve position. It is important to notice that, while performing the EGR valve scan reported in Figure 3, the position of the VGT actuator has been continuously adjusted to maintain the boost pressure approximately constant. As it can be observed, when the EGR valve position is lower than 5%, both the VGT position and measured amount of fresh air are almost constant. Consequently, the mass of external EGR does not change with respect to the condition at EGR equal to zero.

The discussed cylinder filling model has been validated by comparing the estimated fresh charge (fresh air only) with the measurement provided by the ultrasonic flowmeter during tests operated without EGR, returning an accuracy of approximately 2%. It is important to underline that EGR also changes the charge composition and temperature, which vary the gas constant and, consequently, introduce a small error in the in-cylinder mass evaluation. Due to the absence of a gas analyzer, the chemical composition of the charge cannot be evaluated using the discussed experimental layout. Furthermore, another small error in the estimation of the masses is due to the use of a constant specific heat ratio γ (it should be varied with the temperature of the charge). However, it is reasonable to consider the model acceptable during the whole EGR valve sweep since the mentioned errors are small (compatible with requirements for control-oriented models) and do not affect the relative comparison between operating tests run in different opening positions of the EGR valve.

By using the cylinder filling model, it was possible to properly quantify the EGR rate under different engine conditions and analyze the effect of different amounts of EGR mass on pollutants and thermal efficiency.

Operating Conditions

Based on the discussed experimental layout, this work highlights the effect of EGR with gasoline PPC under different operating conditions, summarized in Table 2.

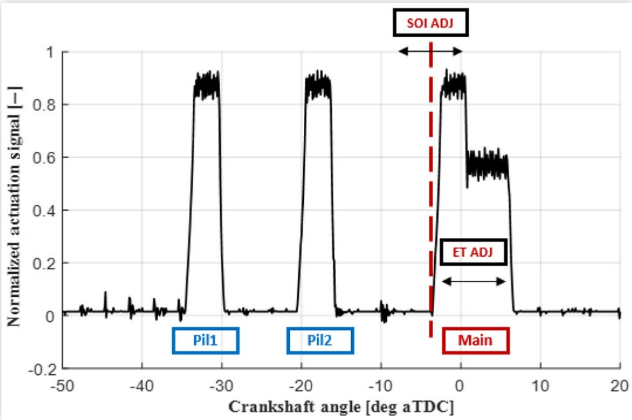
To investigate the impact of EGR on efficiency and pollutants reduction, different EGR rates were taken into account. Such variations were obtained by performing several EGR valve sweeps ranging from 0% to the maximum value still to ensure combustion stability. To highlight the impact of EGR on the combustion process, wide CA50 scans were performed for each investigated EGR valve position, starting from the misfire limit (retarded combustion) and moving the center of combustion to very advanced values. The minimum CA50 achieved during the scans was selected as the maximum between 0 (TDC) and the CA50 at which the peak pressure rise rate (PPRR) exceeded a calibrated threshold (equal to 10 bar/deg in this work).

To guarantee combustion stability in all the operating conditions investigated during the EGR valve sweeps, PPC was operated using a properly calibrated multiple injections pattern [14, 26]. Based on previous works [25], both the number and angular position of each injection were set to ensure combustion stability and controllability. The total number of injections was fixed to three, i.e., two Pilot injections (each one with a mass approximately equal to 1 mg) and one Main injection. The autoignition of the Pilot injections (guaranteed by optimal injection phasing and accurate control of the intake conditions) releases small amounts of energy, which increase the in-cylinder pressure and temperature,

TABLE 2 Engine operating conditions tested during the experimental activity.

Engine speed [rpm]	IMEP [bar]	Rail pressure [bar]	Boost pressure [barA]	CA50 [deg aTDC]	Air temperature upstream manifold [°C]
2000	8	500/700/900	1.5/1.6	6 to 14	50
2000	14	500/700	1.9/2.0	8 to 14	50
3000	10	500/700	1.9	8 to 20	50

FIGURE 4 Normalized injection command used to run gasoline PPC in all the operating conditions tested. The start of injection angle (SOI) mainly controls the center of combustion and ET mainly controls the load.

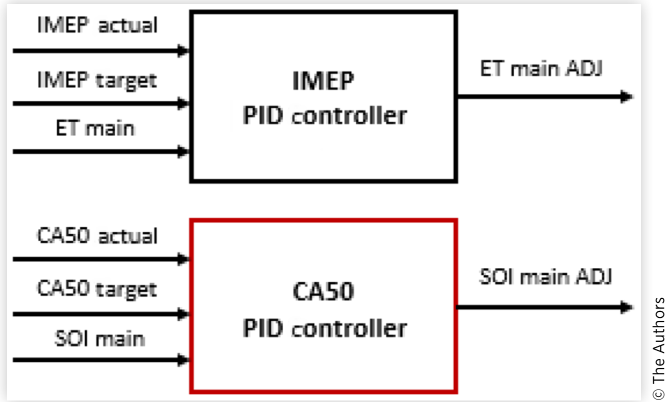


therefore reducing the ignition delay of the following Main injection. Thus once the angular location of the Pilot injections has been optimized, the combustion process can be controlled by changing the position and duration of the Main injection. Figure 4 shows an example of the normalized injection command used during the experimental activity.

To perform the previously mentioned CA50 sweeps, a closed-loop combustion controller was implemented on the ECU. Since mass and location of the Pilot injections are fixed (for a specific operating condition) and based on an experimental optimization [25, 28], the developed combustion controller manages only the injection parameters of the Main injection. Based on the combustion indexes calculated by the indicating system and sent to the ECU via the CAN bus, two independent proportional integral derivative (PID) controllers vary the Start of the Main injection to keep constant the center of combustion (CA50) at a proper target value, and the Energizing Time (ET), i.e., the amount of fuel injected, to keep constant the measured IMEP. The structure of combustion controllers is shown in Figure 5. Mainly due to the very low PID gains, the impact of the closed-loop controller on the cycle-to-cycle variability of the measured in-cylinder pressure signal is negligible. The goal of the control structure is to obtain experimental data at different EGR rates and CA50 that are comparable in terms of IMEP. As a matter of fact, the contribution of the PID controllers remains approximately constant during each performed steady-state test (rpm, load, CA50, EGR rate kept constant during each operating point).

Furthermore, to maintain the boost pressure at a proper target value during the EGR valve sweeps, an additional boost pressure controller was implemented in the ECU. As mentioned before, when the EGR valve opens, the exhaust gas flow rate through the turbine decreases. For this reason, the VGT position needs to be adjusted to compensate for the effects of exhaust pressure and mass flow variations on the power delivered by the turbine. In particular, the boost pressure controller reads the signal coming from the intake

FIGURE 5 Scheme of the IMEP and CA50 closed-loop controller implemented in the ECU.



manifold pressure sensor and, based on the difference between target and measured boost pressure, changes the VGT position so that boost pressure is brought back to its target value. Figures 6 and 7 show the boost pressure controller scheme and its behavior during an EGR valve sweep, respectively.

To deeply investigate the EGR effect on the gasoline PPC process, EGR valve sweeps were also tested with different rail and boost pressures, as reported in Table 2. During each test, the thermoregulation unit has been used, as an air-cooler, to keep the air temperature upstream of the intake manifold at 50°C. As a result, the difference between 50°C (reference without EGR) and the actual temperature in the intake manifold will depend on the amount of exhaust gases recirculated toward the intake manifold (hot external EGR will increase the intake temperature).

All the experimental results in terms of combustion behavior, pollutants, and thermal efficiency will be described in the following section.

Results and Discussions

Based on the above-discussed experimental layout, first, the experimental activity focused on investigating gasoline PPC performance without EGR in three different operating conditions, as reported in Table 2. The aim was to show gasoline PPC potential in terms of thermal efficiency improvement and to measure the engine-out emissions (especially NOx and

FIGURE 6 Scheme of the boost pressure closed-loop controller implemented in the ECU.

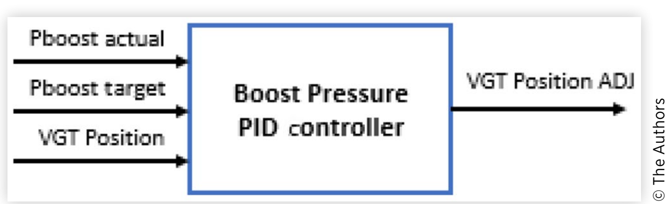


FIGURE 7 Boost pressure closed-loop controller (VGT position) and closed-loop combustion controllers (IMEP and CA50) behavior during the test.

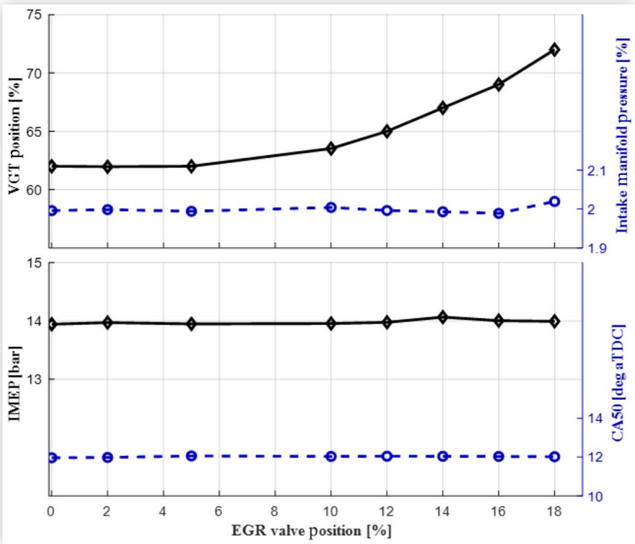
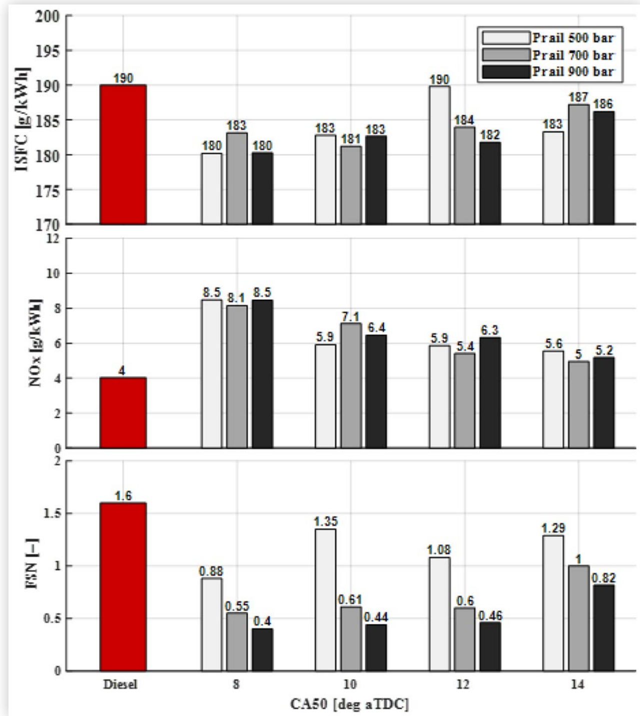


FIGURE 8 Comparison of ISFC, NOx, and soot emissions between Diesel Reference and gasoline PPC (without EGR) during CA50 sweep for the mid-load operating condition at 2000 rpm, IMEP = 14 bar.



particulate matter) generated with respect to the same engine operated with CDC.

Starting from previous works [25, 27, 28], several injection parameters were selected to stabilize the combustion process, i.e., the number of injections, the angular position of the pilot injections, and rail pressure. In addition, boost pressure was chosen to reduce both fuel consumption (ISFC) and pollutants (NOx and Filter Smoke Number FSN). For each operating point, CA50 sweeps were performed, highlighting the impact of the combustion phase on combustion performance and emissions.

Gasoline PPC Combustion Investigation

In this work, the experimental investigation has been focused on three engine operating points, characterized by different loads and speeds. Two tests were run at 2000 rpm, one at high load (14 bar IMEP) and the other at low load (8 bar IMEP). The third operating condition was run at a higher rotational speed (3000 rpm) and intermediate load (10 bar IMEP). The selected engine conditions span a significant portion of the load-speed range of the engine under study and can therefore provide useful information about the potential related to the use of gasoline PPC in a modern homologation cycle (with respect to the standard CDC). The conclusions drawn analyzing the operating points will be useful to estimate the performance of the engine in the intermediate (in terms of load and speed) operating conditions.

Figure 8 reports pollutants and thermal efficiency measured during the CA50 sweep performed at 2000 rpm and 14 bar IMEP (no external EGR). Observing Figure 8, the trend of NOx and FSN with respect to CA50 variation clearly emerges. As a matter of fact, for all the tested gasoline injection pressures, retarding the center of combustion results in a NOx reduction and an increase in FSN. Since it is well known that retarded combustions generate lower pressure and temperature peaks during the combustion process, the trends visible on the measured pollutants were expected and can be easily explained. On the other hand, it is interesting to highlight that the indicated specific fuel consumption (ISFC) measured in gasoline PPC mode is always lower or equal to the one measured with standard CDC. The optimal thermal efficiency is achieved in correspondence with the most advanced values of CA50 (8 and 10 deg) at all the tested injections pressures.

As regards rail gasoline pressure effects, as widely discussed in previous works [27, 28], due to the impact exerted on the mixing process, specific values of gasoline pressure must be chosen to generate a stable and efficient combustion and avoid high levels of pollutants in each operating condition. Considering the tests reported in Figure 8, the lowest value of gasoline pressure (500 bar) generates high levels of FSN if compared with the higher gasoline pressure values analyzed.

As a means of explaining the need for EGR in gasoline PPC, Figure 8 also shows the Diesel Reference values set for each parameter (ISFC, NOx, and FSN) to provide information about the need to improve PPC management, making this combustion methodology convenient to both in terms of emissions and efficiency with respect to the standard calibrated

CDC engine. It is important to underline that the reported CDC conditions (diesel) were obtained by testing the above-described 1.3L engine in standard conditions, both in terms of engine layout and calibration, defined by the engine manufacturer. It is worth mentioning that the base (standard) calibration for CDC operation (rail pressure, CA50, boost pressure, EGR rate) is mainly determined by the need to comply with the pollutant emission regulations (tailpipe emissions), and is not necessarily aimed at achieving the maximum possible thermal efficiency. As a result, the reference CDC conditions (diesel) do not represent the most efficient (minimum ISFC) CDC calibration reachable with this engine, but can be considered a reference for gasoline PPC. To guarantee the coherence between efficiency and emissions measured both in CDC and gasoline PPC mode, all the experimental tests were run using the same engine and test bench layout (in-cylinder pressure sensors, fuel consumption sensor, NOx probe, and FSN measurement sensor were not changed).

From such comparison, it is possible to observe that, despite efficiency and FSN showing promising potential, gasoline PPC (operated without EGR) always produces higher amounts of NOx with respect to diesel combustion. This result suggests that the use of EGR might be crucial, in gasoline PPC to mitigate NOx emissions.

It is interesting to notice that the CDC reference was obtained by injecting the fuel at approximately 1150 bar, while the tests run in PPC mode were operated at significantly lower pressures. This means that the work spent to increase the fuel pressure, with gasoline PPC, might be lower; therefore, an extra benefit in terms of BSFC might be obtained because of the injection pressure reduction. Since, in this work, the experimental tests have been performed using the standard common-rail injection system (not optimal to manage GDI in terms of efficiency and reliability over time), the measured BSFC is not compared. Further work is currently being performed to replace the common-rail injectors with the newly developed Marelli GDI injectors.

Figure 9 shows that similar considerations also apply to the test run at 3000 rpm and IMEP = 10 bar (high revs). In this case, the investigated CA50 range has been extended (from 8 to 18 deg) to better highlight the effect of combustion retardation on efficiency. The observed trend confirms that the minimum ISFC is obtained in correspondence with the most advanced CA50 values. In particular, the ISFC obtained at 8-10 deg is significantly lower than the ISFC for the calibrated standard CDC condition. PPC seems to be very promising also to reduce particulate matter, especially using an injection pressure approximately equal to 700 bar. On the contrary, without external EGR, the NOx emissions are always too high, also for the retarded combustion phases.

Similar conclusions can also be drawn by analyzing the tests carried out at 2000 rpm and IMEP = 8 bar (low load), but in this case, the impact of gasoline pressure on pollutants is less evident, as shown in Figure 10. Low-load conditions are characterized by the combustion of small quantities of fuel in large amounts of air (in this test the air-fuel ratio was close to 2.4, mainly due to the need to maintain the boost pressure higher than 1.5 bar to guarantee a robust gasoline autoignition

FIGURE 9 Comparison of ISFC, NOx, and soot emissions between Diesel Reference and gasoline PPC (without EGR) during CA50 sweep for the high-revs operating condition at 3000 rpm and IMEP = 10 bar.

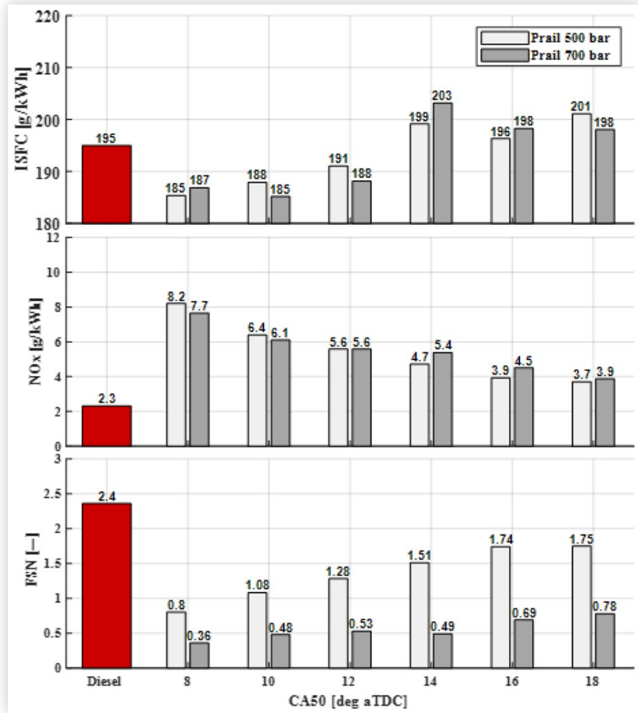
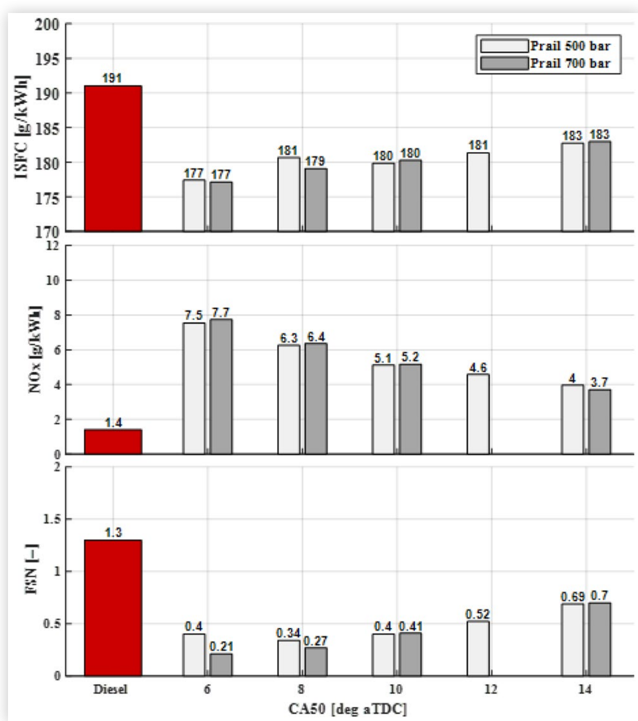


FIGURE 10 Comparison of ISFC, NOx, and soot emissions between Diesel Reference and gasoline PPC (without EGR) during CA50 sweep for the low-load operating condition at 2000 rpm and IMEP = 8 bar.



[29, 30]). Consequently, increasing the injection pressure at very high levels is less crucial to obtain a good air-fuel mixing.

From the analysis of the three investigated points, it appears clear that improving the diesel performance by reducing simultaneously the ISFC, NOx, and FSN would not be possible in running the gasoline PPC without EGR, and as a result, the impact of external recirculated gases on gasoline PPC must be investigated. This aspect will be thoroughly discussed in the following section.

Impact of EGR on Gasoline PPC

As explained in the section above, EGR investigation is necessary, for gasoline PPC to improve engine-out emissions (i.e., the trade-off between NOx and particulate matter) and try to improve the performance with respect to the standard CDC, used in this work as a reference condition. Starting from the information collected during tests without EGR, for each engine operating condition, an optimal set of injection parameters has been selected: optimal injection pattern and rail and boost pressures.

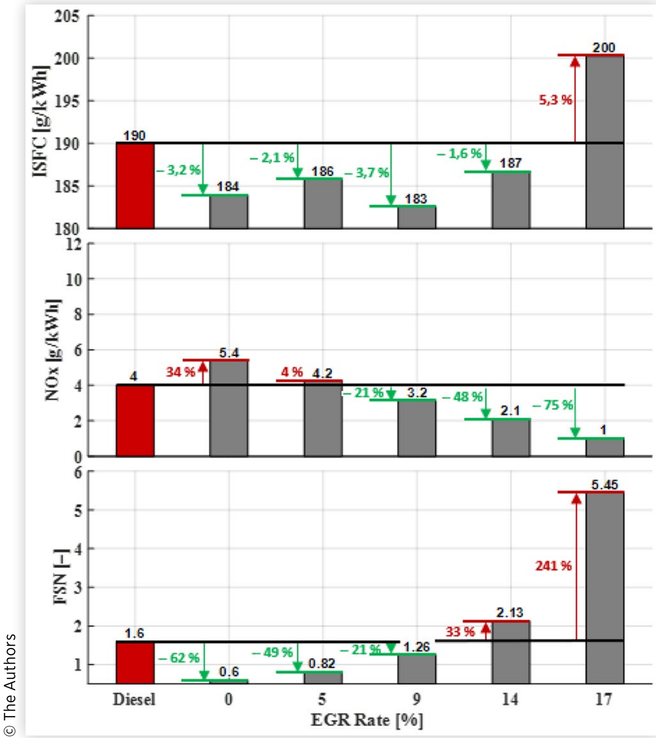
To investigate the impact of EGR on the combustion process and pollutant emissions production, several EGR valve sweeps were carried out for each one of the three engine operating conditions previously described. In addition, to correctly identify the most convenient operating condition in terms of both pollutants and efficiency, gasoline PPC was tested, taking into account different CA50 values. As explained above, during these experimental tests, both IMEP and CA50 were kept at their target values by activating the combustion controllers schematized in Figure 5, while the boost pressure controller (Figure 6) kept the intake pressure at a constant value by compensating the turbine power reduction generated by the (partial) exhaust flow recirculation. Furthermore, using the developed cylinder filling model and the measured airflow, it was possible to calculate the EGR rate for each test and, consequentially, to identify the thermal efficiency/pollutants trade-off condition. Table 3 summarizes the best operating conditions obtained for each engine operating point analyzed, hereafter referred to as optimal conditions.

The effect of EGR on efficiency and pollutants for the high-load operating point run at 2000 rpm—14 bar IMEP is reported in Figure 11. As a matter of fact, increasing the EGR rate until 14% strongly reduces the NOx concentration, while efficiency appears to remain almost constant and FSN slowly increases from 0.6 to 2.13. A comparative analysis of the ISFC and pollutants with respect to the Diesel Reference condition (Figure 11) clearly shows that 9% might be selected as the EGR rate, assuring the efficiency/pollutants trade-off for this condition. In this case, ISFC, NOx, and PM are all significantly improved with respect to the calibrated CDC, while higher EGR rates would produce a quick rise in FSN and ISFC, also exceeding the Diesel Reference thresholds.

TABLE 3 Optimal engine operating conditions considered for EGR rate sweeps.

Engine point name	Engine speed [rpm]	IMEP [bar]	Rail pressure [bar]	Boost pressure [barA]	CA50 [deg aTDC]
Low load	2000	8	500	1.5	14
High load	2000	14	700	2.0	12
High revs	3000	10	700	1.9	12

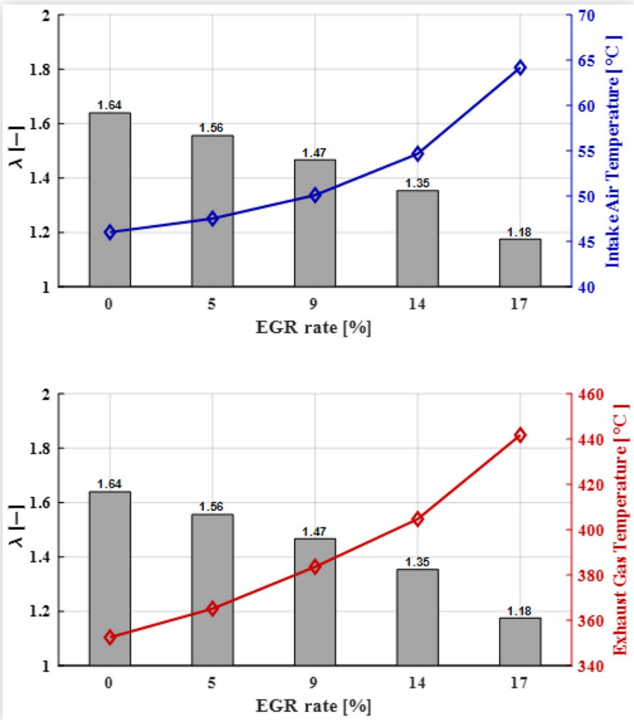
FIGURE 11 Comparison of ISFC, NOx, and soot emissions between gasoline PPC with external EGR for the high-load operating condition at 2000 rpm, IMEP = 14 bar, CA50 12 deg, and Diesel Reference.



As for the standard CI combustion process, charge dilution with external EGR decreases the amount of oxygen available for fuel oxidation and the peak combustion temperature; consequently, the quantity of NOx produced decreases. Furthermore, particulate matter increases, mainly because oxygen and air-fuel mixing are not enough to fully oxidize the injected fuel until the end of combustion. This is clearly shown in Figure 12, which reports the air-fuel ratio measured during the EGR rate sweep.

Furthermore, it is interesting to focus on the intake and exhaust temperatures during EGR sweeps. Figure 12 clearly shows the rise in both temperatures as the EGR rate increases. In fact, the temperature of the recirculated gases is usually higher than the temperature of the fresh air; as a result, when EGR is recirculated, the intake manifold temperature increases. Due to the engine layout, the EGR comes from the exhaust to the intake manifold directly through the cylinder head. Therefore, since the air temperature before the intake manifold has been kept constant, the rise of the intake manifold (and then exhaust) temperature, shown in Figure 12, during EGR

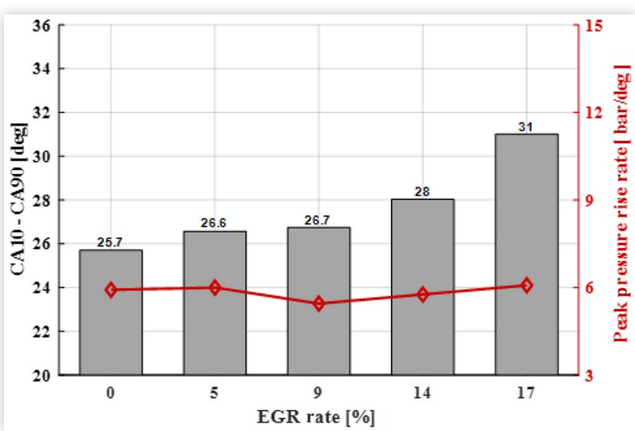
FIGURE 12 Lambda, intake air temperature, and exhaust gas temperature during EGR rate sweep for the high-load operating condition at 2000 rpm, IMEP = 14 bar, and CA50 12 deg.



rate sweeps is a consequence only of the thermal mixing of fresh air (constant 50°C) and exhaust gases. Thus, if hotter fresh air is used at the end of the combustion process, exhaust gases will become even hotter, resulting in an increase in the exhaust gas temperature (bottom subplot of Figure 12).

Figure 13 shows the effect of EGR on combustion duration (defined as CA10-90) and PPRR. It is important to notice that an increase in the EGR rate causes longer combustions for the same reasons explained above; however, it seems to be ineffective on PPRR. This aspect will be better discussed in the following section, where the impact of EGR on PPRR will be verified, comparing two different CA50 conditions.

FIGURE 13 Combustion duration and PPRR during EGR rate sweep for the high-load operating condition at 2000 rpm, IMEP = 14 bar, and CA50 12 deg.

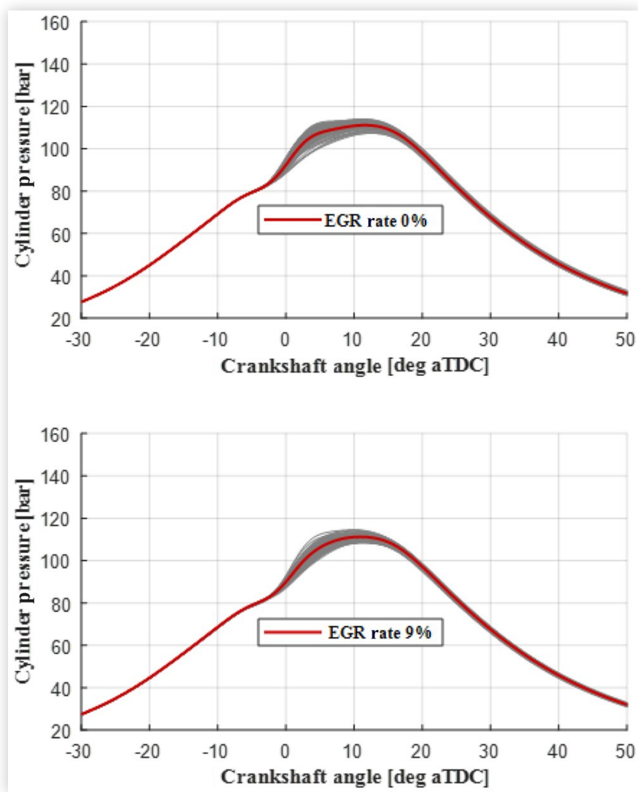


Combustion Analysis

To further explain how EGR affects the behavior of gasoline PPC, the acquired cylinder pressure traces have been compared. As an example, Figure 14 shows 100 consecutive pressure traces and its average for the high-load operating point (2000 rpm—14 bar IMEP—CA50 12 deg) operated without (top subplot) and with external EGR (bottom subplot).

As it can be observed, cycle-to-cycle variability is very low in both cases (with and without EGR). Despite the average pressure traces looking reasonably similar, the major impact of external EGR is visible at the start of the combustion process, i.e., on the ignition mechanism of the Pilot injections. Focusing the attention on the pressure traces in Figure 14, especially on the angular region between 0 deg and 5 deg, it is easy to notice that the first stage of the combustion process is strongly affected by the recirculated gases. In fact, the pressure trace obtained without EGR is characterized by a rapid pressure rise until 5 deg, and then the combustion process becomes slower until the maximum pressure peak is

FIGURE 14 In-cylinder pressure measured for the high-load operating condition at 2000 rpm, IMEP = 14 bar, and CA50 12 deg. EGR rate effect on the combustion process. EGR rate 0% (top subplot) and 9% (bottom subplot).



reached (approximately at 10 deg). When the external EGR is used (EGR rate approximately 9% in this case), the above-described rapid pressure rise disappears, highlighting that EGR slows down the ignition process of gasoline PPC.

To further highlight the effect of EGR on the start of the combustion process (especially on the ignition of the Pilot injections), the Rate of Heat Release (RoHR) has been calculated from −60 deg aTDC to 80 deg aTDC, using Equation 7 [32], for all the average pressure traces measured during the EGR sweep summarized in Figure 11.

$$\text{RoHR} = \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta} + \frac{\gamma}{\gamma - 1} \cdot p \cdot \frac{dV}{d\theta} \qquad \text{Eq. (7)}$$

Figure 15 compares the obtained ROHR traces.

It is interesting to notice that the combustion process clearly shows two main stages: a premixed portion (impulsive, long ignition delay) mainly related to the combustion of the amount of fuel introduced during the Pilot injections and a diffusive portion mainly related to the combustion of the fuel introduced with the Main injection (ignition delay reduced by the energy released by the combustion of the Pilot injections).

The analysis of Figure 15 clearly shows that the presence of exhaust gases strongly affects the combustion of Pilot injections (and, consequently, the whole combustion process). Charge composition and stratification strongly impact the ignition mechanisms of Pilot injections (chemically driven) in CI engines [4, 6, 9]. With the aim of highlighting the effects of EGR on gasoline PPC, Figure 15 shows the RoHR of the "high-load" operating condition with different EGR rates.

Regarding the ignition phase, the analysis of the heat released confirms that, when charge dilution with the exhaust gases is increased, the first combustion stage becomes more and more impulsive (while all the other control parameters, i.e., injection pattern, rail pressure, and boost pressure are kept constant). Despite the intake temperature rise would suggest a decrease of the ignition delay for the Pilot injections [12, 17, 27], Figure 15 shows that the ignition delay reduction due to the temperature rise is lower than the effects related to the chemical inertia of EGR (which retards the autoignition). As a result, the ignition delay of the pre-injections increases with the EGR rate. Furthermore, focusing the attention around the TDC, it is possible to observe that longer ignition delays contribute to the impulsiveness of the premixed (first stage) combustion portion since the tests that run with higher EGR rates show higher RoHR peaks. Longer ignition delays mean that air and fuel have more time to mix; therefore, the amount of charge that autoignites becomes higher. The above-described behavior can be also observed in Figure 16, which reports the PPRR measured with CA50 equal to 5 deg aTDC (same speed and load of the tests shown in Figure 15). In this case, the EGR rate equal to 5% slightly hastens the combustion process with respect to the reference test without EGR (the temperature increase due to the hot residuals is dominant and reduces the ignition delay). Then further EGR rate increases lead to longer ignition delays and more impulsive combustions.

In addition, the effect of EGR can be observed also on the diffusive stage of the combustion process, approximately from 10 deg aTDC to 60 deg aTDC in Figure 15. The exhaust gases recirculated in the combustion chamber slow down the mixing process between air and the fuel introduced with the Main injection. As a result, the diffusive combustion portion becomes slower (lower values of RoHR peak) and longer compared to the reference condition without EGR. Figures 13 and 16 summarize the effect of EGR on combustion duration. All the considerations about the combustion process confirm

FIGURE 15 RoHR traces during EGR rate sweep for the high-load operating condition at 2000 rpm, IMEP = 14 bar, and CA50 12 deg.

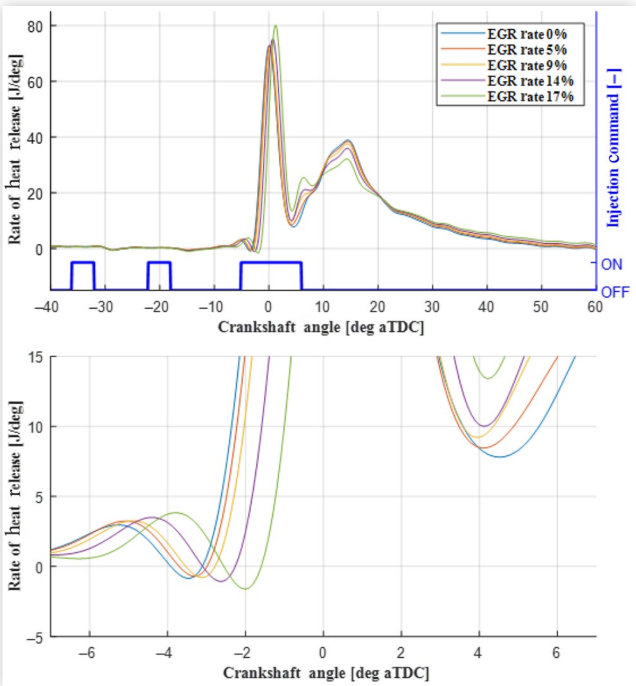
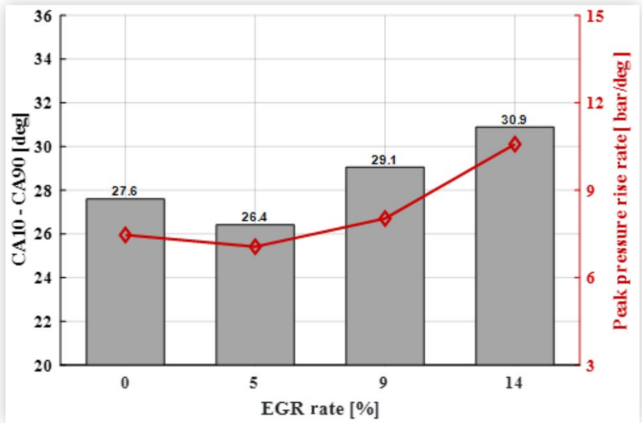


FIGURE 16 Combustion duration and PPRR during EGR rate sweep for the high-load operating condition at 2000 rpm, IMEP = 14 bar, and CA50 5 deg.



the gasoline PPC pollutant emissions (NOx and soot) trends reported in Figure 11 for the high-load operating point.

The above considerations, based on the analyzed experimental test, confirm that a proper EGR rate calibration is crucial to operating the engine at the best compromise between efficiency and pollutants (both can be improved with respect to standard CDC) while guaranteeing engine reliability (PPRR lower than 10 bar/deg [31]). As a matter of fact, the use of recirculated gases affects the RoHR shape and combustion location within the cycle, as clearly highlighted by the steady-state tests run varying the EGR rate and keeping constant the start and duration of the injections. Based on this analysis, future activities will be focused on the compensation of the ignition delay deviations introduced by the recirculated gases, the goal being to properly vary the start of the pre-injections and stabilize the angular location of the premixed combustion (for different EGR rates).

High-Revs Condition

All the above-described considerations can be applied also to the "high-revs" operating point (3000 rpm—10 bar IMEP).

Figure 17 shows pollutants and ISFC for the "high-revs" condition. In this operating point, the impact of EGR (EGR rate from 5% to 20%) on combustion efficiency seems to be marginal (ISFC quite constant). A comparative analysis of ISFC and pollutants with respect to the Diesel Reference condition (Figure 17) shows that an EGR rate approximately equal to 18% might be selected as the EGR rate reaching the efficiency/pollutants trade-off for this condition (both fuel consumption and emissions are improved with respect to the calibrated CDC).

With regard to the emissions of NOx and FSN (Figure 17), the "high-revs" engine condition shows trends similar to the ones already discussed for the "high-load" operating point.

The comparison between the average pressure traces with and without EGR (Figure 18) still highlights the impact of EGR on the ignition delay. By the analysis of the pressure traces between 0 deg aTDC and 10 deg aTDC, it arises that the first stage of the combustion process is slightly retarded when an EGR rate equal to 18% is used (selected as the optimal compromise between pollutants and efficiency).

To analyze more in depth the effect of EGR in the "high-revs" condition, the average RoHR traces, measured during the EGR sweep, have been reported in Figure 19. As discussed for the "high-load" condition, exhaust gas recirculation affects both gasoline PPC combustion stages (premixed and diffusive). From the observation of Figure 19, it is possible to notice that, at a higher engine speed (3000 rpm), the impact of EGR on the ignition delay varies with the amount of EGR rate. In particular, the charge temperature increase due to the hot

FIGURE 17 Comparison of ISFC, NOx, and soot emissions between gasoline PPC with external EGR for the high-revs operating condition at 3000 rpm, IMEP = 10 bar, CA50 12 deg, and Diesel Reference.

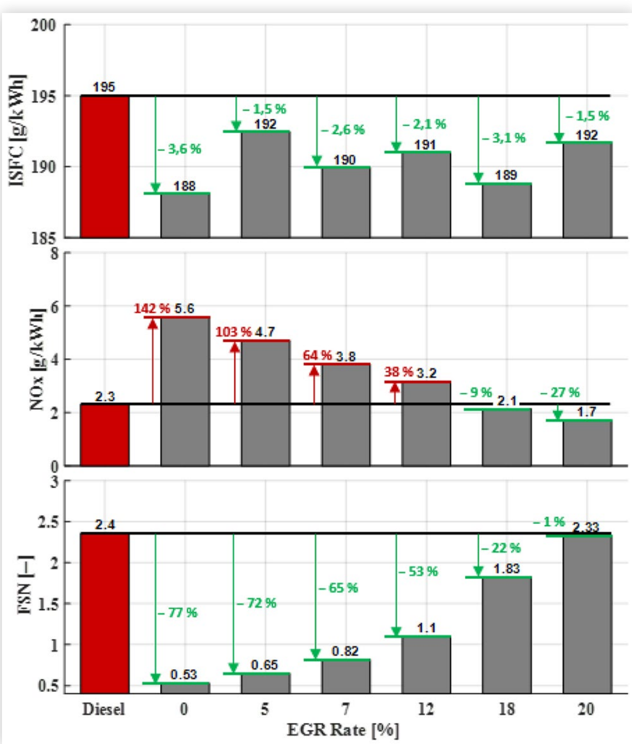


FIGURE 18 In-cylinder pressure measured for the high-revs operating condition at 3000 rpm, IMEP = 10 bar, and CA50 12 deg. EGR rate effect on the combustion process.

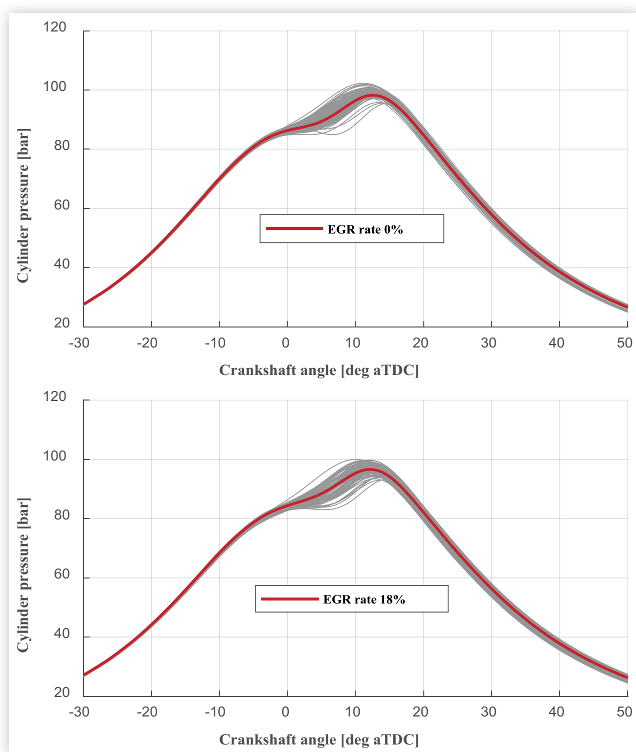
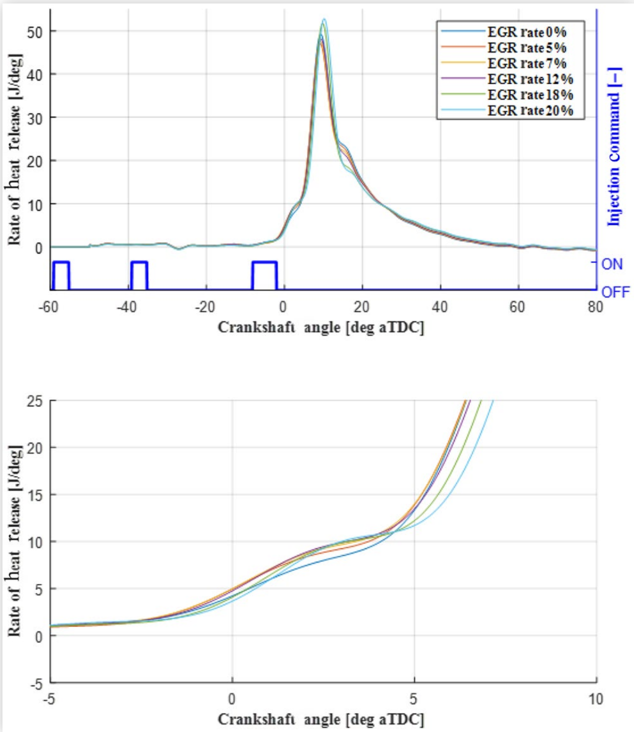


FIGURE 19 RoHR races for the high-revs operating condition at 3000 rpm, IMEP = 10 bar, and CA50 12 deg. EGR rate effect on the combustion process.



recirculated gases hastens the spontaneous ignition of the air-fuel mixture when the EGR rate is lower than 12%, which means that (when the mass of recirculated gases is low) the temperature increase is dominant with respect to the higher chemical inertia due to the exhaust gases. This effect, not clearly visible in the tests operated at 2000 rpm and IMEP = 14 bar, might be (at least partially) explained by the higher turbulence in the cylinder at the higher engine speed. When the EGR rate overcomes 12%, the effect of the recirculated gases becomes similar to the one discussed for the "high-load" condition, i.e., the ignition delay soars together with the impulsiveness of the first premixed combustion stage (visible by observing the maximum RoHR peak).

As highlighted in the analysis of the "high-load" tests, the presence of the recirculated gases in the combustion chamber also affects the diffusive stage of the combustion. As clearly visible in Figure 19, approximately in the angular range between 15 deg aTDC and 20 deg aTDC, increasing the EGR rate makes the diffusive portion (of the heat release) slower and longer (the presence of exhaust gases slows down the mixing process between air and fuel).

Low-Load Condition

The last engine operating point analyzed in this study is the "low-load" condition (2000 rpm—8 bar IMEP), where the engine is operated at the same rotational speed of the high-load point, but with less injected fuel. As already discussed, these sweeps were operated at a leaner air-fuel ratio (approximately 2.4) compared to the "high-load" condition (approximately 1.5). When the engine is operated in PPC mode, the air-fuel ratio needs to be increased to maintain the boost pressure high enough to guarantee a robust gasoline autoignition. For the engine under study, this results in a minimum boost pressure around 1.5-1.6 bar (the lower limitation varies with the intake temperature [21, 33]), which limits the minimum load at which combustion can be operated in stable conditions nearly at IMEP = 5 bar (if the load is further reduced, the air-fuel ratio becomes too lean).

As for the previously analyzed operating conditions, Figure 20 shows ISFC and pollutants for the "low-load" engine operating point. In this case, the use of recirculated gases has a remarkable impact also on the measured fuel consumption, probably because adding EGR to a barely stable autoignition (operated at very lean air-fuel ratio) immediately reduces combustion effectiveness. Regarding the pollutant emissions, trends similar to the ones discussed for the previous operating conditions can be observed: particulate matter increases with the EGR rate, while NOx emissions are reduced. Unfortunately, in this operating point, it is no longer possible to identify an EGR rate clearly beneficial both in terms of combustion

FIGURE 20 Comparison of ISFC, NOx, and soot emissions between gasoline PPC with external EGR for the low-load operating condition at 2000 rpm, IMEP = 8 bar, CA50 14 deg, and Diesel Reference.

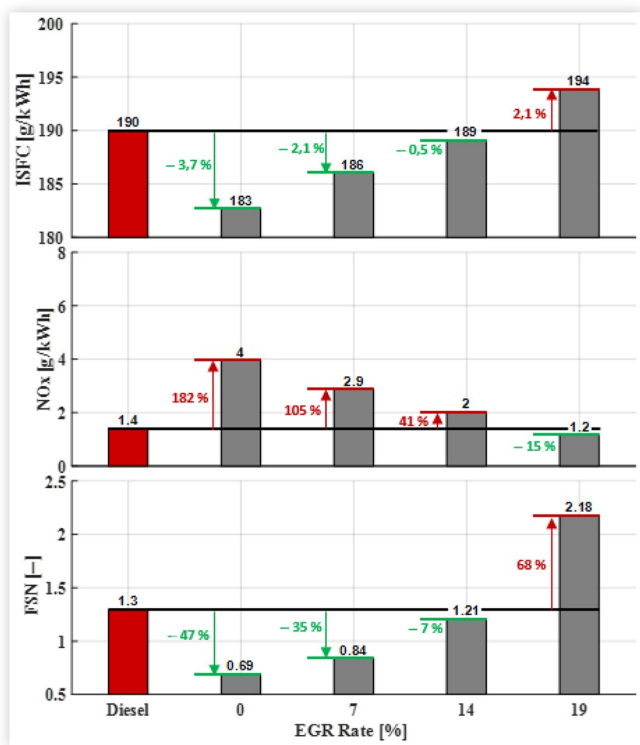


FIGURE 21 In-cylinder pressure measured for the low-load operating condition at 2000 rpm, IMEP = 8 bar, and CA50 14 deg. EGR rate effect on the combustion process.

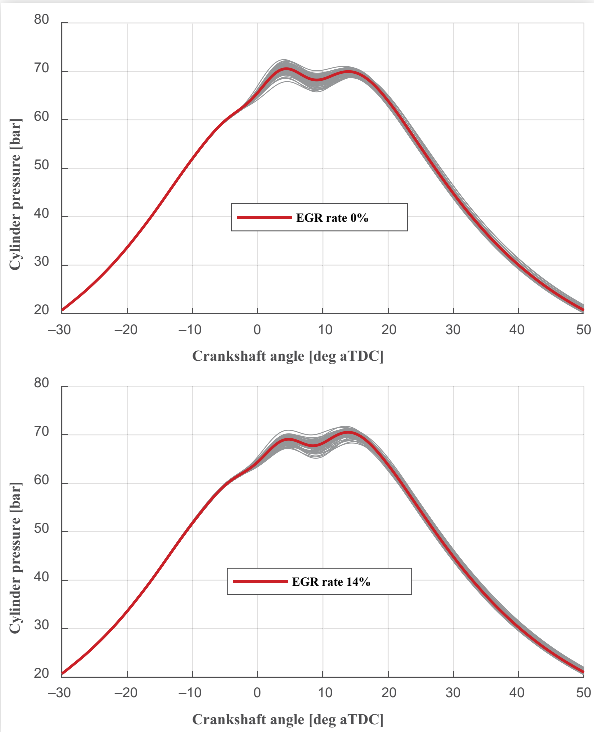
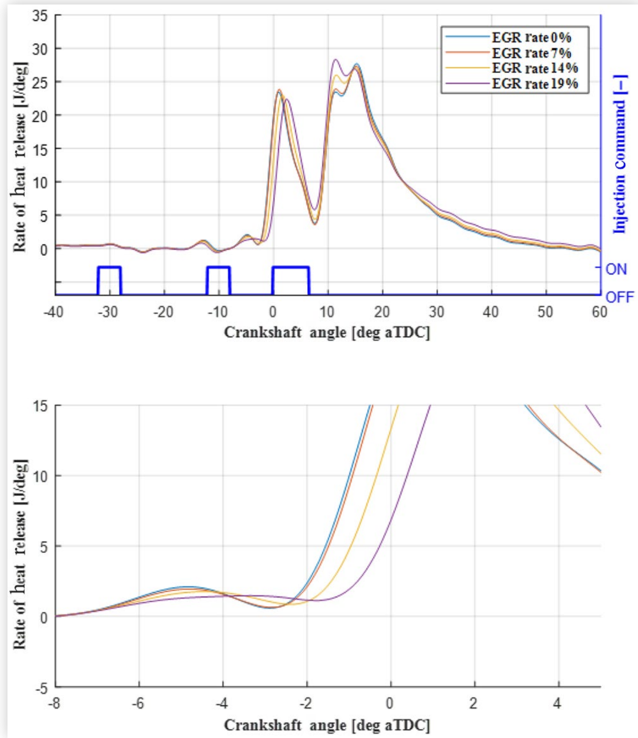


FIGURE 22 RoHR traces during EGR rate sweep for the low-load operating condition at 2000 rpm, IMEP = 8 bar, and CA50 14 deg.



efficiency and emissions. As a matter of fact, the use of an EGR rate approximately equal to 14% returns a slight improvement in ISFC and particulate matter, but measured NOx emissions exceed the level of the calibrated CDC.

The observation of the pressure traces obtained without and with EGR (Figure 21) confirms that EGR increases the ignition delay of the Pilot injections. Focusing the attention on the angular interval between 0 deg and 10 deg, it is easy to notice that the first stage of the combustion process is retarded when the EGR rate is equal to 14% (compared to EGR rate = 0%). This is even more visible in Figure 22, where the RoHR calculated from the average pressure traces of the whole EGR rate sweep are compared.

All the previously described effects of the EGR on the gasoline PPC are visible also in the average RoHR traces of the "low-load" tests. As it can be observed in Figure 22, moving toward higher EGR rates, the ignition delay of the first premixed stage increases (chemical inertia effect due to the additional exhaust gases). Furthermore, increasing the EGR rate, the diffusive combustion, reported in Figure 22 approximately in the range between 10 deg aTDC and 20 deg aTDC, becomes slower and longer compared to the reference condition without EGR.

Summary/Conclusions

This work discusses the experimental activity carried out to investigate the impact of high-pressure external EGR on gasoline PPC, performed in a light-duty CI engine. To guarantee combustion stability and a flexible investigation of different intake/exhaust conditions, the engine was modified by adding a volumetric supercharger (before the standard dynamic compressor). Furthermore, specific control strategies, suitable to manage the actuators of the intake system and achieve specific pressures and temperatures in the intake and exhaust manifolds, were implemented in the ECU.

Three main operating conditions, characterized by different engine speeds and loads, were first investigated without EGR to quantify the performance of gasoline PPC and compare it with the efficiency and emissions of the same engine operated with diesel (calibrated by the constructor). The obtained results clearly show that gasoline PPC might potentially reduce fuel consumption and soot emissions. However, without EGR, the obtained NOx emissions would be too high with respect to the NOx emissions obtained during standard CDC operation.

Several scans were run, varying the rate of recirculated EGR, while keeping constant the center of combustion. EGR proved to be a useful way to mitigate NOx emissions also with gasoline PPC, but the use of high EGR rates might have a negative impact on efficiency and PM emissions. However, for the two operating conditions characterized by the highest loads (IMEP equal to 14 bar and 10 bar), it was clearly possible to identify a combination of injection parameters suitable to achieve a significant improvement of both efficiency and emissions with respect to the standard CDC operation. On the contrary, for the third analyzed operating point (IMEP equal to 8 bar), the efficiency/pollutants trade-off between returned results is similar to the optimal performance obtained in standard CDC mode.

Further investigations are currently being performed to try to extend gasoline PPC operation at low engine loads (below IMEP equal to 5 bar) and investigate its performance over the whole engine operating range. In addition, since PPC can be operated using injection pressures lower than 1000 bar, the engine will be also modified to test the recently available Marelli GDI injectors (up to 1000 bar), to provide further understanding of the combustion process with components (suitable for production) that can overcome the limitation of the standard gasoline injectors, avoiding the complex adaption of the diesel injector to operate with gasoline. Finally, since the use of EGR affects the ignition delay of gasoline PPC, with the aim of improving combustion controllability and extending the engine operating range using relevant EGR rates, an ignition delay model suitable to predict the start of combustion is under development.

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Definitions/Abbreviations

BSFC - Brake-specific fuel consumption
CA10 - Start of combustion
CA50 - Center of combustion
CA90 - End of combustion
CAN - Controller area network
CDC - Conventional diesel combustion
CO_2 - Carbon dioxide
dP - Derivative pressure
$d\theta$ - Derivative angle
dV - Derivative volume
ECU - Electronic control unit
EGR - Exhaust gas recirculation
ET - Energizing time
EGR rate - Fresh air dilution factor
FSN - Filter smoke number
GCI - Gasoline compression ignition
GDI - Gasoline direct injection
HCCI - Homogeneous charge compression ignition
IMEP - Indicated mean effective pressure
ISFC - Indicated specific fuel consumption
LTC - Low-temperature combustion
m_{EGR} - Exhaust gas recirculated mass
NOx - Nitrogen oxides
P_{exh} - Exhaust gas pressure
P_{Int} - Intake air pressure
PPC - Partially premixed combustion
PPRR - Peak pressure rise rate
Prail - Gasoline injection pressure
R - Air gas constant
RoHR - Rate of heat release
SOI - Start of injection angle
TDC - Top dead center angle
T_{exh} - Exhaust gas temperature
T_{Int} - Intake air temperature
V_{BDC} - Volume at bottom dead center angle
Vcc - Combustion chamber volume
$Vcyl$ - Maximum geometrical volume of the cylinder
VGT - Variable geometry turbine
V_{IVC} - Volume at intake valve closing angle
ΔV_1 - Re-expansion reduction volume contribution
ΔV_2 - Late intake valve reduction volume contribution
γ - Adiabatic index
λ - Lambda

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Appendix A: Uncertainties

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

1. In-cylinder pressure: necessary to calculate all combustion indexes (IMEP, CA50, PPRR)

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 0 … 250 bar
Natural frequency	115 kHz

2. Fuel consumption: necessary to evaluate the total fuel injected mass for each condition and, therefore, to calculate combustion efficiency (ISFC)

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 range	−40°C to +120°C

3. NOx: necessary to evaluate the gasoline PPC NOx production

Sensor name	Continental Uni-NOx SNS14
Measurement principle	ZrO_2 based multilayer sensor with integrated heater and 3 oxygen pumps
Measured value output	NOx, linear λ and binary λ
Measurement range	NOx 0-500 ppm Lin λ, 0.75 to 14 Bin λ, > 0.75 V at λ = 0.9; < 0.2 V at λ = 1.1
Time for readiness NOx	<100 s
Operating gas temperature	100-800°C

4. FSN: necessary to evaluate the gasoline PPC particulate matter production

Sensor name	AVL Smoke Meter 415s
Measurement principle	Filter paper blackening
Measured value output	FSN (filter smoke number), mg/m^3(soot concentration)
Measurement range	0 to 10 FSN
Detection limit	0.0002 FSN or 0.02 mg/m^3
Exhaust pressure ranges	0 to 3000 mbar (high-pressure option)
Maximum exhaust temperature	800°C (long probe configuration)
Repeatability	±(0.005 FSN + 3% at 10 seconds intake time)