Effects of fuel injection parameters on PCCI combustion and emissions characteristics in a medium-duty compression ignition diesel engine

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Abstract

From the different power plants, the compression ignition diesel engines are considered the best alternative to be used in the transport sector due to its high efficiency. However, the current emissions standards impose drastic reductions for the main pollutants emitted by this type of engines, i.e. NOx and soot. To accomplish with these restrictions, alternative combustion concepts as the premixed charge compression ignition (PCCI) are being investigated nowadays. The objective of this work is to evaluate the impact of different fuel injection strategies on the combustion performance and engine-out emissions of the PCCI combustion regime. For that, experimental measurements were carried in a single-cylinder medium-duty compression ignition diesel engine at low load operation. Different engine parameters as the injection pattern timing, main injection timing and main injection fuel quantity were sweep. The best injection strategy was determined by means of a methodology based on the evaluation of a merit function. The results suggest that the best injection strategy for the low load PCCI operating condition investigated implies using a high injection pressure and a triple injection event with a delayed main injection with almost 15% of the total fuel mass injected.

Keywords: Premixed charge compression ignition; emissions; heat release; injection settings; diesel engine

1. Introduction

Nowadays, the most important challenge for diesel engines is to meet the strict emissions limits imposed by the future emissions regulations, which force and improvement of the performance with the lowest possible cost penalty. The current injection systems and aftertreatment technologies offer very high performance in reducing the engine-out emissions [1][2]. However, the high price and complexity of these systems threaten the competitiveness of the diesel engine. By this reason, during the last few years, the efforts devoted by the research community in developing new combustion concepts with diesel are growing [3][4]. With this approach, it is intended to reduce significantly the engine-out nitrogen oxides (NOx) and particulate matter (PM) emissions, which will allow to reduce the aftertreatment necessities [5] [6][7]. This could offer a prolonged use of the diesel engines without, or with much reduced aftertreatment requirements [8]. The literature shows two main approaches to achieve a combustion process with simultaneous ultra-low NOx and soot emissions: the single-fuel and the dual-fuel techniques, both of which have as a goal to obtain a lean and highly mixed charge prior to combustion.

The dual-fuel technique so-called reactivity controlled compression ignition (RCCI) [9] has demonstrated to be able to operate with different fuels [10], and promoting ultra-low values of NOx and soot emissions in both stationary tests and driving cycles [11][12]. However, this kind of approaches require important modifications to the engine design and/or the need to carry on board a second fuel, which rests practicality for the real life application of the concept [13].

One of the most important single-fuel techniques is the homogeneous charge compression ignition (HCCI), which relies on achieving a homogeneous air-fuel mixture in the cylinder prior to be ignited due to the high pressure and temperature experienced during compression stroke. HCCI was originally described by Alperstein et al. [14], who used port fuel injection (PFI) with diesel fuel in order to create a fully premixed (or homogeneous) mixture with diesel and air. This type of combustion process, with fully premixed mixture, generates a rapid combustion process characterized by a short and high peak of heat release that minimizes the heat transfer. Therefore, HCCI combustion commonly reaches high thermal efficiency during the combustion process. In addition, due to the enhanced air-fuel mixing process, the local fuel equivalence ratios are lean, which avoids the soot formation. Considering the NOx emissions, the HCCI combustion is characterized to provide a stabilized flame front that mitigates the in-cylinder combustion temperature and thus, reduces the NOx formation. The limitations of HCCI combustion mode are related to its narrow operating range and poor combustion control. The operating range is limited by the fast increase of the in-cylinder pressure at high loads and the misfire phenomenon occurring at low loads. The combustion control is difficult, since the start of combustion depends on several factors including the fuel characteristics, autoignition phenomenon, homogeneity of the air-fuel mixture and the in-cylinder temperature and pressure [15]. In addition, the high dilution and poor in-cylinder temperature promotes the autoignition conditions too late at low load conditions, resulting in a partial burn, which generates high unburned hydrocarbon (HC) and monoxide emissions (CO) emissions and loss of power [16].

The premixed charge compression ignition (PCCI) combustion appears as an alternative to the weaknesses observed in the HCCI operation. PCCI differs from HCCI combustion in the air-fuel mixture preparation. The mixing process in PCCI is premixed and not perfectly homogeneous as in HCCI. Therefore, the PCCI combustion can be conceded as a premixed combustion process dominated by compression ignition combustion. The low volatility of the diesel fuel implies the requirement of high injection pressure strategies with different exhaust gas recirculation (EGR) rates to promote the PCCI combustion [17][18]. In this sense, high injection pressures [19] and high swirl ratios promote a better air-fuel mixing process, generating sufficient premixing time between start of combustion (SoC) to end of injection (EoI) to achieve an efficient PCCI combustion [20]. Initially, a single-stage direct injection of fuel was used to achieve the PCCI combustion, where the early direct injection during the compression stroke favored the air-fuel mixture because of the presence of a higher in-cylinder temperature and longer mixing time. However, too advanced start of injection (SoI) timings provoked an impact of the fuel spray on the walls [21]. This causes the wetting of the cylinder wall, promoting incomplete combustion with high levels of HC and CO emissions and, consequently, lower the thermal efficiency [22][23].

In PCCI combustion, the injection scheme defines the air-fuel mixing process, which is also considerably influenced by the in-cylinder conditions (pressure and temperature) [24]. To control the in-cylinder conditions at the instant of main injection, researchers used split fuel injection strategies. Hashizume et al. [25] proposed a strategy of multiple stage diesel combustion (MULDIC), in which the fuel was injected in two stages to promote the LTC. The same authors reported that the MULDIC strategy is able to promote a simultaneous reduction of NOx and soot emissions. Nevertheless, the fuel consumption was worse than conventional compression ignition (CI) combustion. Neely et al. [26] evaluated the effect of different pilot injections to realize PCCI combustion in light- and heavy-duty diesel engines. Using a single pilot injection, the authors achieved significant reduction in NOx emissions, but higher carbon monoxide (CO) emissions. This CO penalty was mitigated by adding multiple pilot injections. Horibe et al. [27] and Torregrosa et al. [28] analyzed the effect of the split injection strategy on NOx emissions. They concluded that higher efficiency and lower NOx emissions at medium engine loads could be performed using a single pilot injection, but with higher pressure rise rates (PRR) during the combustion process.

Considering the future trend of the emissions regulations, the objective of this study is to determine the possible benefits of using the PCCI combustion in a modern medium-duty compression ignition diesel engine. The ultimate goal would be to reach ultra-low engine-out NOx and soot levels that help fulfilling the upcoming emissions regulations further than EURO

VI to reduce the aftertreatment necessities. To do this, experimental measurements were carried out in a single-cylinder medium-duty diesel engine operated in PCCI combustion mode. A parametric study based on the independent modification of the injection strategy and the injection pressure was carried out at low load conditions. To support the performance results, detailed analysis of the combustion and emission characteristics of PCCI combustion is also performed. Finally, the different injection strategies are evaluated by means of a merit function to determine the best combination of injection settings.

2. Material and methods

2.1. Test cell and engine description

The experimental study was carried out on a single-cylinder diesel engine derived from a production medium-duty engine. The production engine is a multi-cylinder VOLVO D5K developed for urban freight distribution purposes. The main characteristics of the engine are depicted in Table 1.

Table 1. Main characteristics of the base engine.

Characteristic	Value
Style	4 stroke, DI diesel engine
OEM calibration	EURO VI
Maximum power	177 kW @ 2200 rpm
Maximum brake torque	900 Nm@1200-1600 rpm
Maximum in-cylinder pressure	190 bar
Maximum injection pressure	2000 bar
Bore x Stroke	110 mm x 135mm
Connecting rod length	212.5 mm
Crank length	67.5 mm
Total displaced volume	5100 cm^3
Number of cylinders	4
Compression ratio	17.5:1

As Figure 1 shows, the single-cylinder engine was installed in a fully instrumented test cell with all the necessary auxiliary facilities to perform the experimental test campaign. To achieve the intake airflow conditions required, an externally driven screw compressor is used to supply the required boost pressure before passing through an air dryer. The compressor can provide an inlet pressure up to 3.7 bar. After the dryer, the air mass flow was measured by means of a volumetric flow meter. The air pressure and temperature were adjusted in the intake settling chamber. The backpressure at the exhaust line was controlled using a valve positioned after the EGR line, which controls the pressure inside the exhaust settling chamber. The required EGR rate was regulated using a valve positioned between the EGR settling chamber and the intake line. This allowed the exhaust gas to be mixed to the intake air flow along the intake runner. To regulate the temperature of the EGR-fresh air mixture, a temperature regulator was used in the intake manifold. To gain regulation capacity during the tests, the coolant circuits of the EGR and oil were independent from the engine coolant circuit. The facility also included fully instrumentation to acquire several temperatures and mass flows. A combination of thermocouples and resistance temperature detectors (PT100) was selected to measure the liquid and gas temperatures.

The engine control is done by an in-house control software, which is based on a real time National Instruments powertrain control system, combining a field-programmable gate array (FPGA) based on the injection synchronization, and a peripheral component interconnect (PCI) extensions for instrumentation (PXI) system is used for the in-cylinder pressure acquisition and processing. The in-cylinder pressure was measured with a Kistler 6125C10 glow-plug piezoelectric transducers and Kistler 4603B10 charge amplifiers. A resolution of 0.5 CAD was used to acquire the in-cylinder pressure. To remove cycle-to-cycle combustion scattering, mean data set of 120 consecutive combustion engine cycles was used for in-depth combustion analysis.

A five gas Horiba MEXA 7100D analyzer was used for the exhaust gases (NOx, CO, HC, O_2, CO_2) analysis, where the volume fractions of each pollutant was sampled and averaged over a 40 seconds once the steady-state operation of the engine was reached.

An AVL 415 variable sampling smoke meter was used to measure the smoke emissions. The filter smoke number (FSN) values shown in the paper are the average of three consecutive measurements at the same operating condition. These measurements were transformed into mg/m3 by means of the Equation 1 [29]:

$$[mg/m^3] = \frac{1}{0.405} \cdot 4.95 \cdot FSN \cdot e^{0.38 \cdot FSN} \qquad (1)$$

Considering the injection system, a common-rail direct-injection system with a solenoid injector was used. The main characteristics of the diesel injector are depicted in Table 2. An AVL 733S fuel balance measures the fuel mass flow. Commercially available European diesel fuel was used for this work. Table 3 shows the main characteristics of the fuel used.

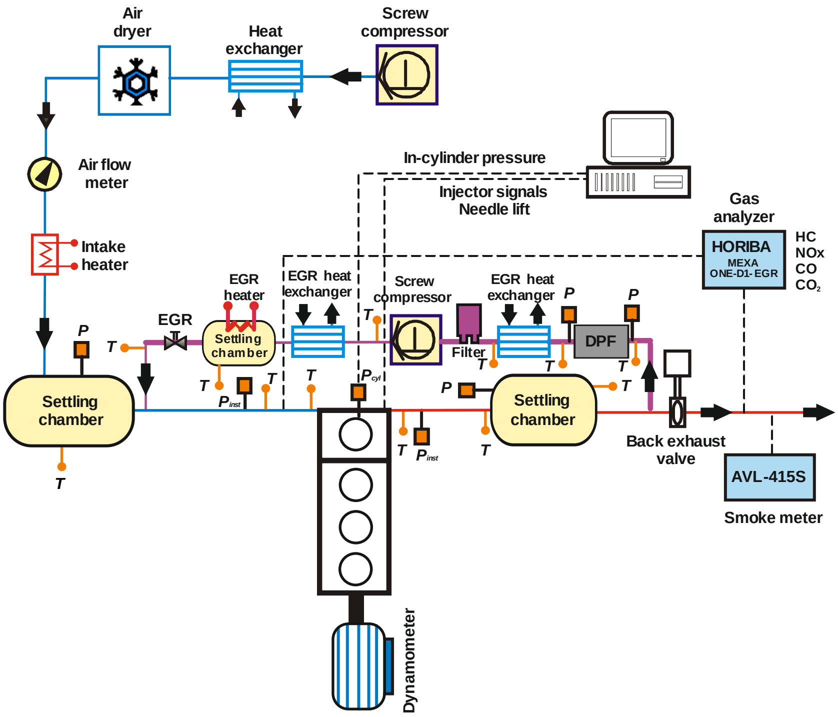

Figure 1. Scheme of the test bench.

Table 2. Diesel fuel injector characteristics.

Diesel injector	
Actuation type	Solenoid
Steady flow rate @ 100 bar [cm^3/min]	1300
Number of holes	7
Hole diameter [µm]	177
Included spray angle [°]	150

Table 3. Physical and chemical properties of the diesel fuel used.

Properties	Diesel
Density [kg/m^3] (T=15°C)	820
Viscosity [cm^2/s] (T=40°C)	2.8
Molecular weight [kg/kmol]	215.42
C/H by weight [-]	6.25
Cetane number [-]	55.77
Lower heating value [MJ/kg]	42.44

2.2. In-cylinder pressure analysis

A 0D combustion diagnosis code (CALMEC) has been used to study the combustion process, which is described in detail in [30]. The main input for this tool is the in-cylinder pressure. After the acquisition of the in-cylinder pressure signal, a Fourier series low-pass filter has smoothed each experimental combustion cycle pressure signal. Once filtered, the 100 combustion cycles collected have been averaged together to produce a representative cylinder pressure trace, that has been used to execute the combustion analysis. Then, the first thermodynamics law has been implemented between intake valve close (IVC) and exhaust valve open (EVO), taking into account the combustion chamber as an open system due to blow-by and injection process.

$$RoHR = mc_v\frac{dT}{d\alpha} + \frac{dQ}{d\alpha} + p\frac{dV}{d\alpha} - \left(h_{f,inj} - u_{f,g}\right)\frac{dm_{f,ev}}{d\alpha} + RT\frac{dm_{bb}}{d\alpha} \quad (2)$$

being α the crank angle degree, m, dT, dV and dQ are the instantaneous in-cylinder mass, bulk gas temperature difference, volume difference and instantaneous evolution of the heat transfer (HT) to the chamber walls respectively, c_v is the specific heat coefficient which depends on the instantaneous gas temperature and characteristics, $dm_{(f,ev)}$ is the variation of fuel injected mass, $h_{(f,inj)}$ and $u_{(f,g)}$ are the injected fuel enthalpy and internal energy of the evaporated fuel, R is the ideal gas constant and dm_{bb} is the blow-by difference. In addition, the HT to the chamber wall (Q) together with a lumped conductance model permitted an accurate RoHR estimation.

The ideal gas equation of state has been used to estimate the averaged bulk gas temperature in the chamber. Accompanied by these two basic equations, several sub-models: a filling/emptying model has been used to estimate the trapped mass [31]; the specific gas heat is related on both gas temperature and characteristics [32]; blow-by model has been based on the progression of the gas in an isentropic nozzle [33]; the chamber volume deformation because of pressure and inertia has been calculated based on a simple deformation model [34];the HT coefficient in the chamber walls has been estimated by a modified Woschni-like model [35] :

$$h = CD^{-0.2}p^{0.8}T^{-053}\left[C_{W1}c_m + C_{W2}c_u + C_2\frac{V_d T_{IVC}}{V_{IVC}p_{IVC}}(p - p_0)\right]^{0.8}$$

being C and C_2 are constants with values are 0.12 and 0.001 respectively, c_m is the averaged piston speed, c_u is the instantaneous in-cylinder gas tangential velocity which has been fitting by means of CFD simulations [35], p_0 is the in-cylinder pressure during motoring conditions considering a polytrophic evolution, and finally C_{W1} and C_{W2} are constants. The values of both constants have been adjusted for each engine using a mixture of modelling and experimental methodology [36].

2.3. Test matrix

The engine was run under PCCI combustion mode at low load condition (3.8 bar IMEP) and constant engine speed of 1000 rpm. For that propose, three different injection parameters were swept: the timing of all the injections train (Study 1), the main injection timing (Study 2) and the fuel mass quantity of the main injection (Study 3). These sweeps were done over an injection strategy with three events (2 pilot injections and a main injection). Moreover, two injection pressure levels were used in the first two sweeps (1000 and 1500 bar) and only one injection pressure level was studied in the last sweep (1500 bar). The total fuel mass injected was equal between the studies. In the case of the Study 1, the main injection timing was varied from 40° to 55° before top dead center (bTDC) while maintaining constant dwell between the injection events. Moreover, the injected mass at each of the three events was equal. In the Study 2, the main injection timing was varied and the pilot injection timings were fixed at 57.5° and 55° bTDC. As in the previous sweep, the injected fuel mass for each injection event was the same. Finally, an additional sweep of the fuel mass main quantity in the main injection was performed varying the mass from 15% to 47% of the total fuel mass injected. The rest of the fuel mass was split between the two pilot injections, whose injection timings were also fixed at 57.5° and 55° bTDC.

In this case, the main injection timing was tested at 10°, 0° and -5° bTDC. For all the engine conditions measured, the air mass flow was kept constant at 10.4 g/s, the EGR rate at 50% and fuel mass needed to obtaining the desired load was approximately 0.275 g/s. Table 4 shows the test matrix of the three studies evaluated in this study.

Table 4. Experimental test matrix.

	IP [bar]	Sol_{P1} [ºbTDC]	Sol_{P2} [ºbTDC]	Sol_{main} [ºbTDC]	Fuel $mass_{P1}$ [% total mass]	Fuel $mass_{P2}$ [% total mass]	Fuel $mass_{main}$ [% total mass]
$Sol_{pattern}$ sweep	1000	45, 50, 55, 60	42.5, 47.5, 52.5, 57.5	40, 45, 50, 55	33	33	33
	1500						
Sol_{main} sweep	1000	57.5	55	52.5, 45, 35, 25, 15, 5, -5	33	33	33
	1500						
$Fuel_{main}$ quantity sweep	1500	57.5	55	10, 0, -5	42.5, 40, 35, 30, 25	42.5, 40, 35, 30, 25	15, 20, 30, 40, 50

2.4. Merit function

A merit function was defined to assess the potential of each injection strategy on the results obtained. The merit function includes a limit imposed for each parameter and a weighting factor (F_i) to define the importance of each parameter on the merit function result. The emissions limits selected to calculate the merit function are those imposed by the EURO VI regulation (NOx_{limit}=0.4 g/kWh, CO_{limit}=1.5 g/kWh, HC_{limit}=0.13 g/kWh and $Soot_{limit}$=0.01 g/kWh). The value selected for the ISFC and exhaust gas temperature limit were that obtained from the OEM calibration CDC operation (ISFC is a confidential value and $T_{exh,limit}$= 170°C). If the value obtained from the calculations in brackets is negative (i.e. the actual value is lower than the limit), the result of this operation is forced to be zero. Thus, the best injection strategy will be that which minimizes the merit function. It is interesting to note that the weighting factors must be selected in the order of magnitude of the variable, if not, the merit function result will be unbalanced. In this case, the weighting factors are F1=10, F2=10, F3=0.2, F4=0.02, F5=0.1 and F6=0.1, so that the most important factors to be minimized are ISFC, NOx and soot because they weight double than the others.

$$MF = F1 \cdot \left(\frac{ISFC}{ISFC_{limit}} - 1\right) + F2 \cdot \left(\frac{T_{exh}}{T_{exh,limit}} - 1\right) + F3 \cdot \left(\frac{NOx}{NOx_{limit}} - 1\right) + F4 \cdot \left(\frac{Soot}{Soot_{limit}} - 1\right) + F5 \cdot \left(\frac{CO}{CO_{limit}} - 1\right) + F6 \cdot \left(\frac{HC}{HC_{limit}} - 1\right)$$

3. Results and discussion

This section studies the influence of different injection strategies on the PCCI combustion process and emissions. As described in section 2.3, three different studies were carried out. The first subsection presents the combustion analysis based on the heat release rate (HRR) and other parameters used to describe the combustion process. Later, the average values in terms of engine-out emissions and performance for the three studies are evaluated. Finally, a merit function analysis is presented and discussed to choose the best combination of injection settings.

3.1. Combustion analysis

Figure 2 presents the temporal evolution of the in-cylinder pressure and HRR for the different injection timings (of the complete pattern) at two injection pressures (1000 and 1500 bar). At each injection pressure, the injection pattern was sweep from SoI_{main} 40° to 55° bTDC while maintaining constant the fuel mass and dwell time between of each injection event. During experimental tests, the EGR rate was fixed at 50% to contribute to the NOx reduction. The results show that the combustion process is delayed as the injection pattern timing advanced. This occurs because the extra time available for the fuel-air mixing process results in highly diluted conditions, reducing the local equivalence ratios and needing for higher in-cylinder temperatures (nearer to the TDC) to promote the autoignition process. As the HRR is moved towards the TDC, the fuel-to-work conversion efficiency is improved, leading to lower ISFC (Figure 7). At advanced injection pattern timings, the combustion duration is also reduced (shorter width of the HRR curve) because the chemical kinetics is accelerated. These tendencies are similar to those found by Zhao et al. [37]. For both injection pressure cases, a two-stage heat release trace was observed: the first stage was associated with low temperature reactions in which the magnitude of the heat released is approximately 15% of the total, and the second stage was associated with high temperature reactions where most of the fuel energy is released. In terms of injection pressure, the increase of the injection pressure promotes a faster combustion development (steeper HRR slope) and shorter combustion duration due to an increase of the total air entrained into the fuel, which results in an enhancement in the fuel-air mixing process [38].

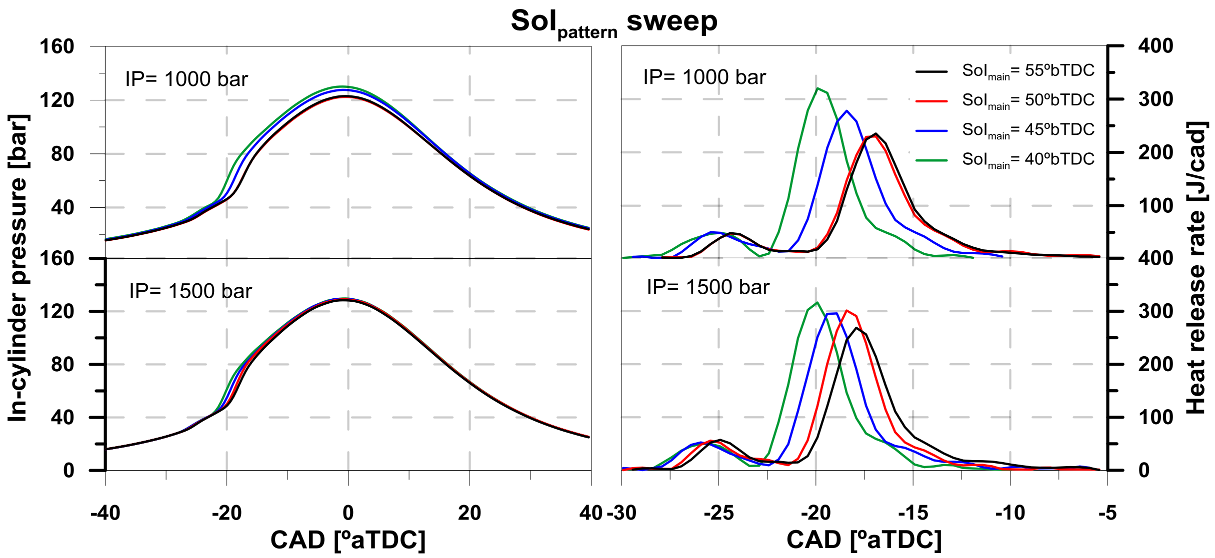

Figure 2. Temporal evolution of the in-cylinder pressure and the HRR for the different injection pattern timings and injection pressures.

Figure 3 presents the temporal evolution of the in-cylinder pressure and HRR at different Sol_{main} timings for two injection pressures (1000 and 1500 bar). At each injection pressure, the Sol_{main} timings were varied from -5° to 52.5° bTDC maintaining constant the fuel mass of each injection event. Sol_{P1} and Sol_{P2} timings were fixed at 57.5° and 55° bTDC, respectively, and the EGR rate was fixed at 50%. The effects of advancing the Sol_{main} timings on the combustion process present similar behavior to those observed in the previous study. When the Sol_{main} timings were advanced (from 35° to 52.5° bTDC), the in-cylinder pressure peak (P_{max}) and HRR peak increased and shifted towards TDC and the combustion duration was reduced. This behavior is characteristic of a highly premixed combustion process. Delaying the Sol_{main} more than 35° bTDC, the in-cylinder pressure and HRR peak decrease considerably and the HRR shape is split into two-stages: the first stage is referred to the fuel injected in both pilot injections because the main injection event has not started when the HRR peak is observed. This fact causes inferior in-cylinder conditions due to mainly poorer fuel-air mixing process in addition to lesser available fuel, resulting in lower P_{max} and peak HRR curve. The second stage is directly related to the main injection event since this combustion phase takes place from the SoI_{main} timings. This indicates that the SoI_{main} timing controls the chemical kinetics of the fuel-air mixture and therefore it can be used to control the PCCI combustion.

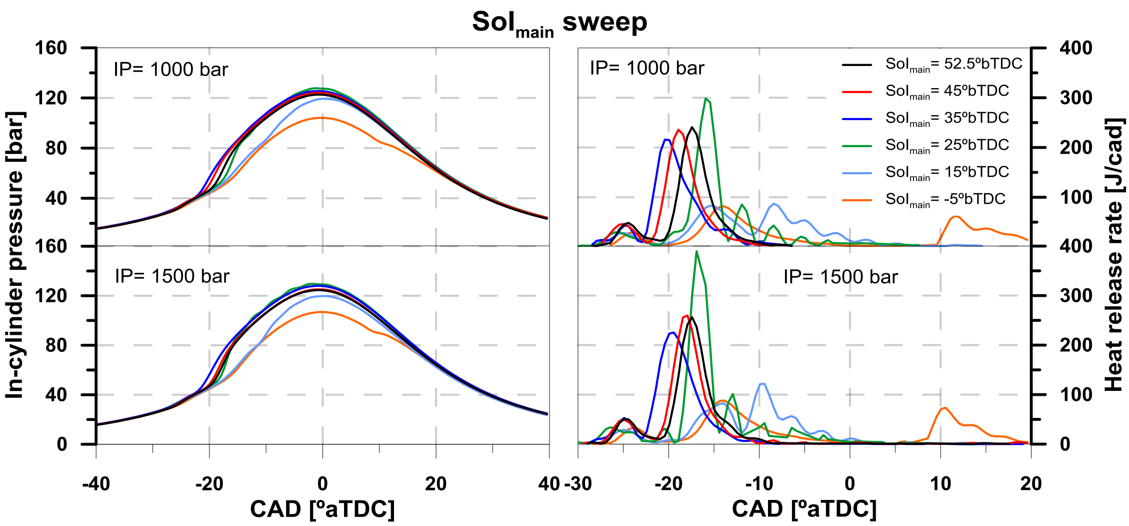

Figure 3. Temporal evolution of the in-cylinder pressure and the HRR for the different SoI_{main} timings and injection pressures.

The results of the sweep of the main injection fuel quantity are shown in Figure 4. As the previous two figures, the temporal evolution of in-cylinder pressure and HRR curve for the different fuel injection main quantity at two SoI_{main} timings tested (10° and -5° bTDC). At each SoI_{main} timing, the main fuel quantity was varied from 15% to 47% respect to total fuel mass injected maintaining constant the EGR rate at 50%. As it is possible to see in the figure, the shape of the HRR curve does not change for the different main injection fuel mass. Only the maximum value of the HRR peak curve varies depending on the main fuel mass injected. However, the start of combustion remains constant in all the cases. This indicates that, under the conditions studied, the chemical kinetics of the air-fuel mixture is controlled only by the instant in which the fuel is injected.

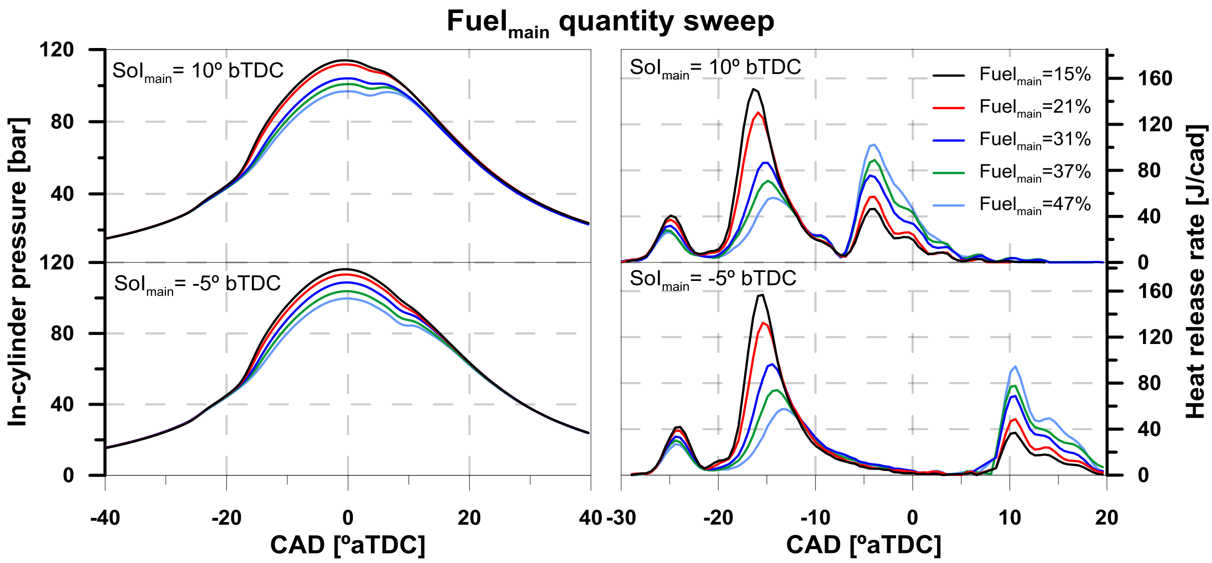

Figure 4. Temporal evolution of the in-cylinder pressure and the HRR (bottom) for the different fuel injection main quantity at two SoI_{main} timings tested.

Figure 5 shows the variations in the start of combustion (SoC), combustion phasing (CA50) and combustion duration (CA90-CA10) for the three studies. The crank angle position corresponding to 10% of the mass fuel burned (CA10) was considered as SoC. In the PCCI combustion mode, the combustion process typically starts before the TDC because of the advanced injection strategy timings and fast chemical kinetics associated to the high reactivity of the diesel fuel. In the conditions analyzed in this research, the SoC varied from 16° to 24° bTDC. The results of the first study (injection pattern timing sweep), reveal that the SoC is delayed when the injection pattern timings is advanced. This trend was also visible in the in-cylinder pressure and HRR curves (Figure 2). This fact is attributed to a worsened fuel-air mixing due to the poor in-cylinder thermodynamic conditions when the fuel was injected [39]. In the second study, about the main injection timing sweep, it is shown that advancing the main injection timing from 35° to 55° bTDC the SoC was delayed. Nevertheless, for SoI timings more delayed than 35° bTDC, this trend changes drastically and then, advancing the injection pattern timing results in advanced SoC. This showed that the chemical kinetics was not significantly affected by fuel-air premixing, but was significantly affected by the in-cylinder thermodynamic conditions. This could be clearly observed in the combustion phasing results. In both studies (1 and 2), the effect of the injection pressure is similar. Higher injection pressure implies an earlier SoC, mainly caused by an improvement of fuel-air mixing process [40]. In the study 3, about the fuel main quantity sweep, the trend is as expected. Higher main fuel injected provokes the SoC shifting towards the TDC. The crank angle position corresponding to 50% of the total fuel mass burned (CA50) was used as parameter to track the combustion phasing. This parameter is relevant because too advanced combustion phasing reduces the combustion efficiency under PCCI conditions. The use of an advanced combustion phasing results in a higher heat release rates, resulting in excessive knocking. Nevertheless a late combustion phasing results in a poor combustion process, which results in higher HC and CO emissions. As shown in Figure 5, the combustion phasing followed similar trend at the three different sweeps. However differences between combustion phasing were relatively lower compared to that of SoC. In PCCI combustion mode, the CA50 was significantly advanced compared to compression ignition combustion [41]. This is consequence of the premixed combustion phase, much faster combustion than a diffusive combustion. Combustion duration is defined as the crank angle difference between 10 and 90% of mass fuel burned (CA90-CA10). The fluctuation in combustion duration represents the rapidness of combustion. The results obtained show that the combustion duration of PCCI combustion mode is about 10-20 CAD, significantly shorter compared to CI combustion (~30-40 CAD) due to the absence of diffusion combustion phase in PCCI mode. However, the combustion duration increases when the main injection event was closer to TDC because this promotes a diffusive combustion process.

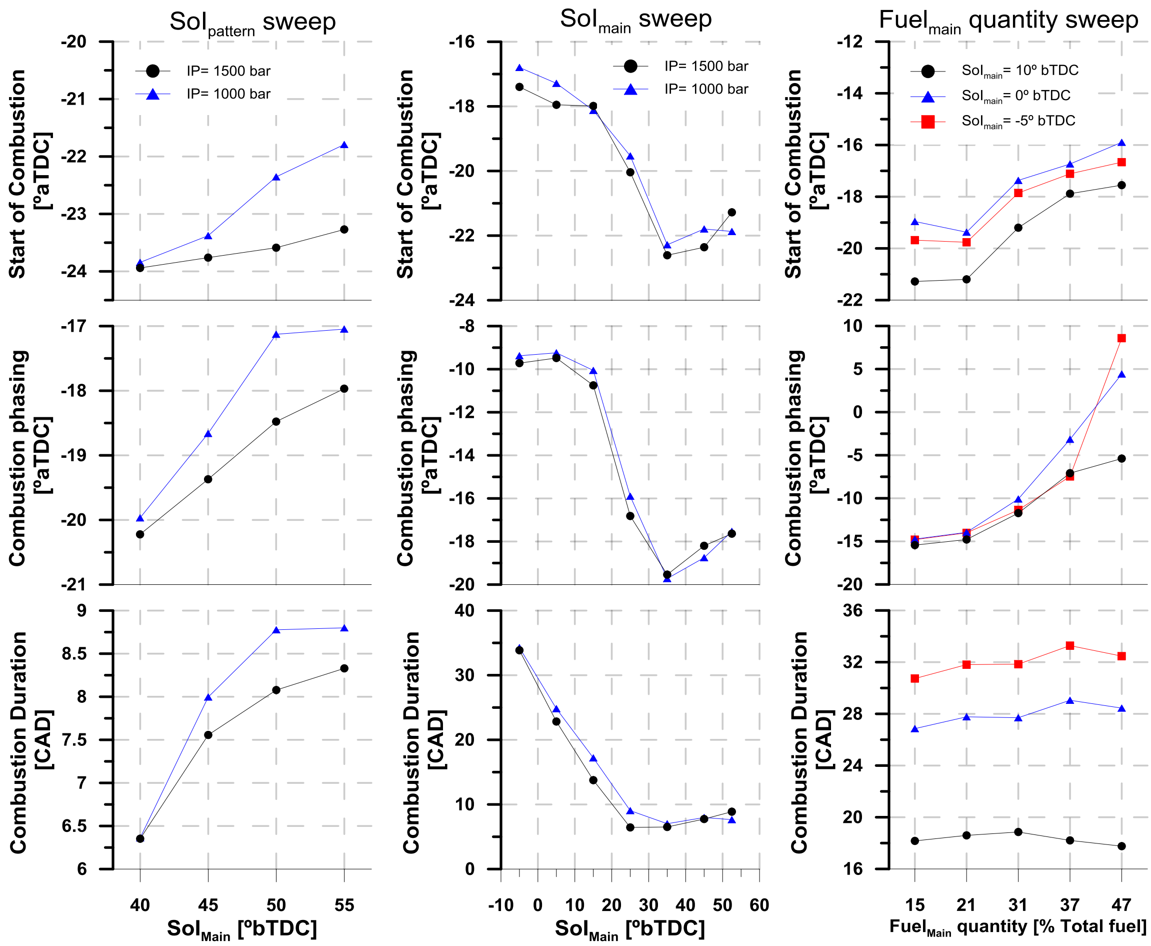

Figure 5. Start of Combustion, combustion phasing and combustion duration for the three studies.

3.2. Emissions and performance

Figure 6 presents the NOx, HC, CO and soot emissions for the three different sweeps tested in this research. It is important to remark that the same EGR rate was used in all the cases. The main advantage of the PCCI combustion is the possibility of reaching ultra-low NOx emissions. The NOx formation is a characteristic process of the diffusion combustion, where the in-cylinder local temperatures are high. The PCCI combustion process is considered as low temperature combustion (LTC) due to the absence of a prominent diffusion combustion phase. As shown in Figure 6, the NOx emissions decrease when the injection pattern timings are advanced and the main injection fuel mass is reduced. This occurs because both actions contribute to promote a more premixed in-cylinder charge, thus avoiding a diffusive combustion process. In the case of the SOI_{main} sweep, a change in the NOx trend is observed, which is coherent with the trends of SoC, CA50 and combustion duration. Regarding CO and HC, several researches have reported higher values of both emissions under PCCI combustion [42][43][44]. CO emission is an intermediate combustion process, essentially generated by the incomplete conversion of CO to CO_2. This conversion process is limited by two factors: lower combustion temperatures that decrease the conversion reactions and the lack of oxygen in the reaction zone [45]. Advanced injection pattern and/or main injection timings and/or higher fuel mass in the main injection lead to relatively higher CO emissions. Relatively advanced combustion phasing at advanced injection pattern timings and the deterioration of combustion process at delayed main injection were the main reasons for this behavior, leading to relatively lower in-cylinder temperature. The HC formation is expected to occur by three reasons: (i) incomplete combustion (shorter combustion duration), (ii) trapped fuel droplets in crevice volume (early injection at higher fuel injection pressures), and (iii) in-cylinder wall quenching (lower in-cylinder temperature). Figure 6 shows very similar HC emissions obtained in all cases (between 0.4-1 g/kWh), which are low results under PCCI combustion. The higher values were obtained at advancing injection pattern timings, which result in homogeneous fuel-air mixture and promote premixed combustion. This leads to lower in-cylinder temperature and increase the HC formation due to incomplete combustion and wall quenching. In addition, at injection timings of 50°-55° bTDC, relatively larger fraction of fuel enters into the crevice volume resulting in higher HC emissions. Fuel spray impingement at advanced injection timings may be another reason for higher HC emissions, wherein fuel spray comes in contact with cold in-cylinder walls and combustion is quenched. Finally, the values of soot emissions are very low in all cases due to the operating condition measured it was a low load (3.8 bar IMEP).

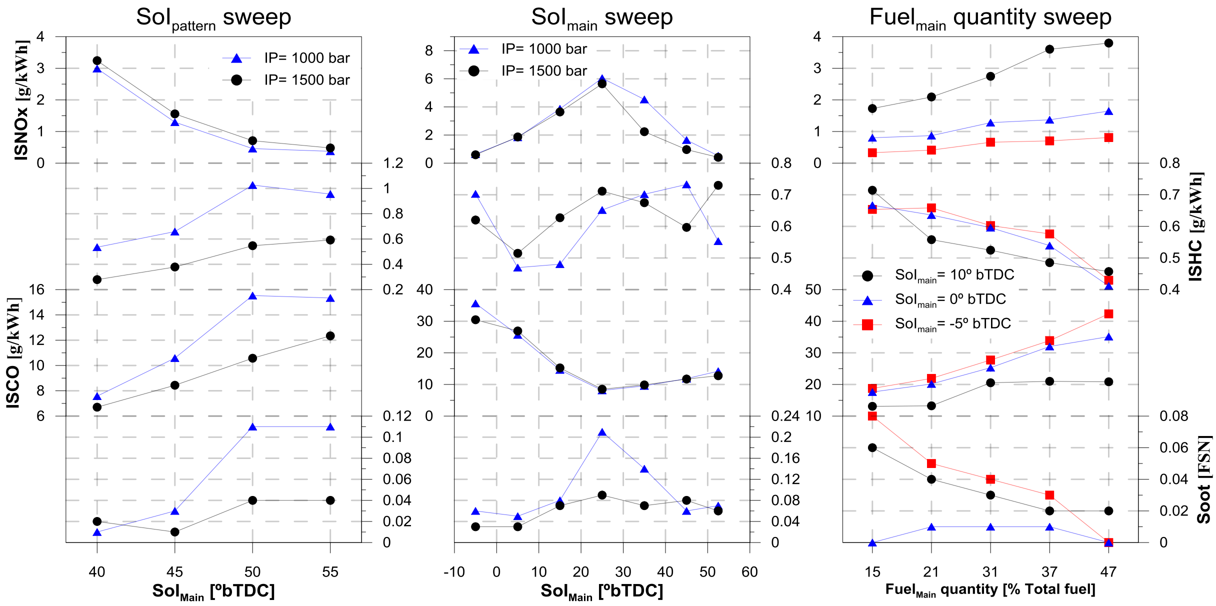

Figure 6. NOx, HC, CO and soot emissions for the three studies.

To investigate the effects of different injection strategies on PCCI engine performance, the indicated specific fuel consumption (ISFC) was calculated and plotted for the three studies. Figure 7 shows the normalized ISFC values with respect to the conventional diesel combustion (CDC) at the same low load operating condition. The results show that the normalized ISFC decreases when advancing of injection pattern timings and/or retarded main injection timings and/or increasing the fuel mass in the main injection. Looking at the combustion phasing shown in Figure 5, it is possible to remark that the best ISFC cases in all the studies correspond to the cases with a combustion phasing closer to the TDC. However, the cases in which the combustion takes place during the compression stroke show an important penalty in the ISFC, i.e. a low fuel-to-work conversion efficiency. To avoid this disadvantage, several researches have reported the reduction of the compression ratio engine as a possible solution. The aim of the reduced compression ratio is to reduce the in-cylinder thermodynamic conditions, hence the ignition delay and combustion phasing is delayed in addition to reduce flame temperatures during the combustion to suppress NOx emissions [46]. However, in the current work this was not possible because one of the imposed constraints was to use a currently under production diesel engine without modifications. Nonetheless, it is interesting to remark that the best case of the study 3

(SOI_{main}=0° bTDC with 47% main fuel quantity) shows a penalty of only 2% in ISFC, while engine-out NOx and soot emissions are below the levels imposed by the EURO VI regulation.

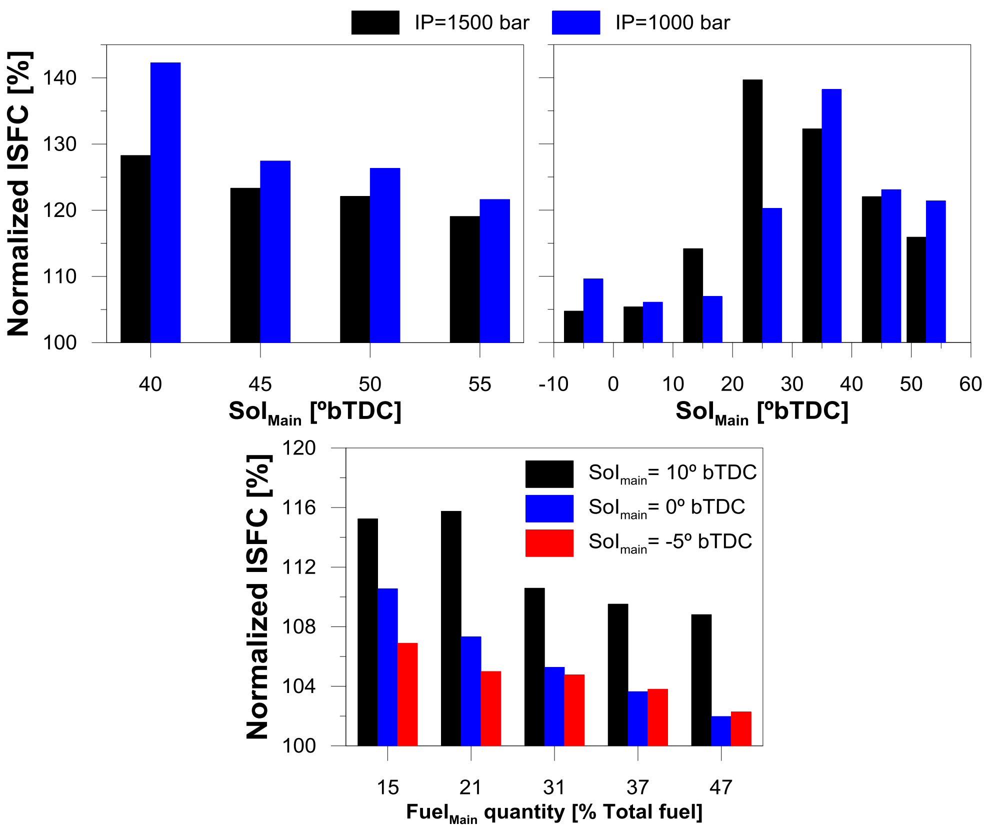

Figure 7. Normalized ISFC for the three studies.

3.3. Merit function

Until now, the results described in Results and discussion section do not stand out which injection strategy is best option to be used for low load condition selected under PCCI combustion mode. Given that the combustion parameters, performance and emissions do not follow the same trend, the analysis performed is not enough to conclude with the best injection strategy. To choose the best which injection strategy would put up the best results in terms of performance and emissions, the methodology described in section 2.4 was used. The results of the merit function analysis as a function of the different injection strategies evaluated are presented in Figure 8. In this figure, there are three plots corresponding each injection sweeps analyzed. In the injection pattern sweep, the merit function values show a decrease when injection pattern timings are advanced. In addition, the results are very similar for both injection pressure levels being the lowest one at 55° bTDC with 1500 bar of injection pressure. Respect to main injection sweep, the lowest merit function values are presented at advanced and retarded main injection timings being the best timings at 5° aTDC with also 1500 bar of injection pressure. Finally, regarding the main injection quantity sweeps, main injection timing at 5° aTDC and lower main injection quantity (15% of the total fuel injected) shows lessen merit function value in all cases evaluated.

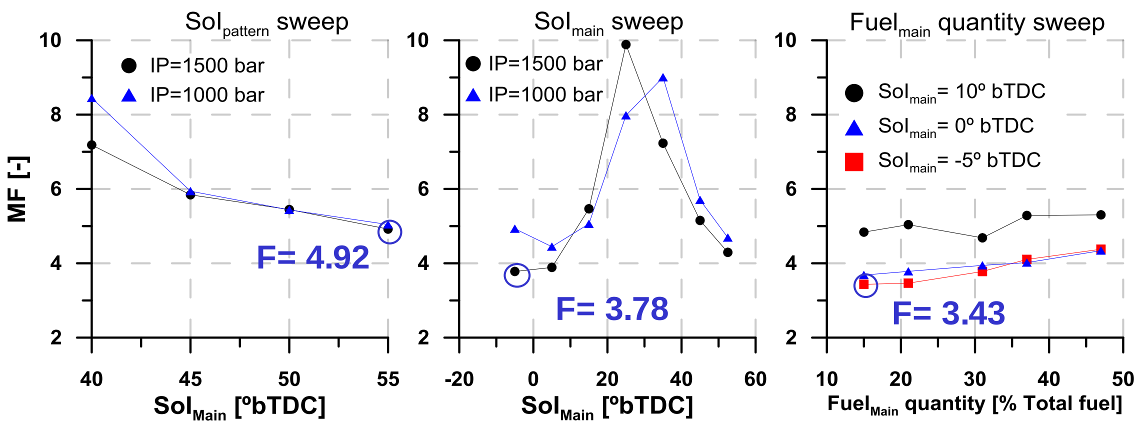

Figure 8. Merit function results for the three studies.

In order to deep into the description of the merit function, Figure 9 presents the merit function value sectioned in different combustion performance and emission terms. In particular, NOx, CO, HC and soot at emissions and ISFC and exhaust gas temperature with respect to combustion performance. In this case, the plotted sweeps have been chosen with lower merit function values. In the two injection timings sweeps, ISFC and exhaust gas temperature have more importance than emission terms. Fuel consumption and premixed combustion (and thus, reduced exhaust gas temperature) penalized enough respect to CDC combustion. In fuel main quantity cases, CO emission is the predominant term especially when fuel main injected correspond at almost 50% of the total injected fuel mass.

In summary, the best injection strategy for this low load operating condition under PCCI combustion is high injection pressure and triple injection events in which, the main injection is delayed with almost 15% of the total fuel injected.

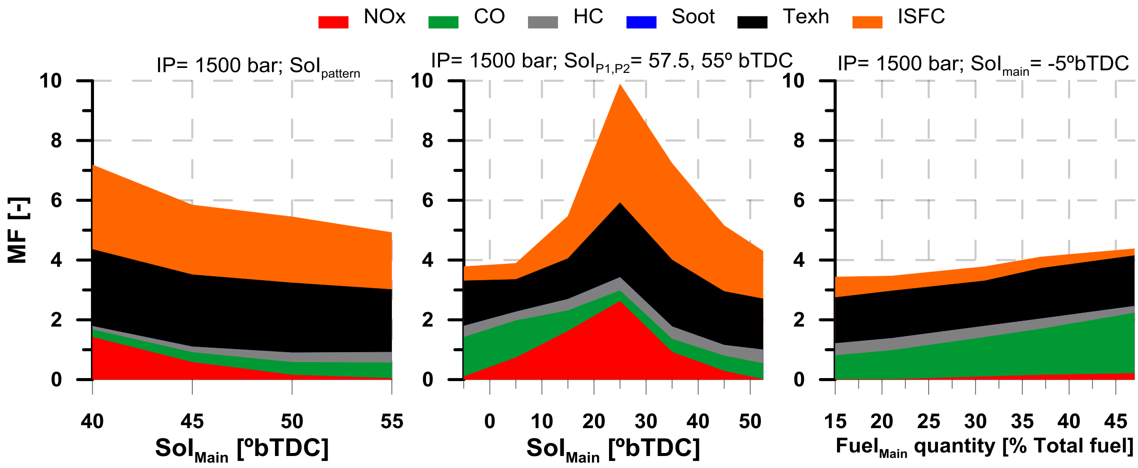

Figure 9. Merit function results divided into combustion performance and emission terms for the three studies.

4. Conclusions

In this research, the effect of different fuel injection strategies on the PCCI combustion process, performance and engine-out emissions at low load engine load have been evaluated experimentally in a single-cylinder medium-duty CI diesel engine. For that, the experimental measurements were carried out at three different sweeps: injection pattern timing (from 40° to 55° bTDC), main injection timing (from 52.2° to -5° bTDC) maintaining constant the SoI_{P1} and SoI_{P2}, and fuel quantity injected in the main injection (from 15 up to 45% of the total fuel mass).

The results have shown that, at different injection pattern timing, the combustion process improves when the injection pattern timing is advanced. The start of combustion is delayed and peak of in-cylinder pressure and HRR curve is higher due to higher premixed combustion behavior and thus, faster air/fuel chemical kinetics. In addition, a better ISFC is achieved because the combustion phasing is shifted towards TDC and then, the fuel to work conversion efficiency is improved. In terms of engine-out emissions, lower NOx emissions are reached at 55° bTDC due to the lower in-cylinder temperature. Nevertheless, this temperature promotes higher HC and CO emissions due to incomplete combustion. Finally, the values of soot emissions are very low in all the cases due to the low load operating condition measured.

From the main injection timing sweep, it was found that advancing the SoI_{main} timings results in similar behavior to that observed in the cases of injection timings sweep. For advanced SoI_{main} timings, the in-cylinder pressure (P_{max}) and HRR peak increased and shifted towards TDC and the combustion duration was reduced. From SoI_{main} at 35° bTDC, the peak in-cylinder pressure and peak of HRR decrease considerably and the HRR shape is split into two-stages. For these cases, the best fuel consumption is obtained on the lower NOx emissions at 5° aTDC. Similarly to the previous sweep, higher HC and CO emissions are obtained when the NOx levels are lower.

From the main injection fuel quantity sweep, it was found that the modification of the main fuel mass injected does not change the HRR shape. Only the maximum value of the HRR curve varies depending on the fuel mass injected. As expected, a higher fuel mass injection in the main injection delayed the start of combustion and the combustion phasing independently on the main injection timing. In addition, an increase of the fuel mass injected increases the combustion duration, which sifts the combustion towards TDC, promoting a diffusive combustion process and consequently, improving the fuel consumption. Considering the engine-out emissions, these cases follow the same trend that the two other studies: better fuel consumption for lower NOx emissions and higher HC and CO emissions. The soot emissions are negligible in all the cases evaluated.

Finally, the global effect of the different injection strategies have been evaluated by applying a methodology based on a merit function for obtaining the best strategy. The merit function have been sectioned in NOx, CO, HC and soot at emissions and ISFC and exhaust gas temperature with respect to combustion performance. For the cases evaluated, , the best injection strategy for this low load operating condition under PCCI combustion is high injection pressure and triple injection events in which, the main injection is delayed with almost 15% of the total fuel injected.

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Abbreviations

aTDC after top dead center

CA10 crank angle for 10% of fuel burnt

CA50 crank angle for 50% of fuel burnt

CA90 crank angle for 90% of fuel burnt

CA90-CA10 combustion duration

CAD Crank angle degree

CDC Conventional Diesel Combustion

CI compression-ignition

CO Monoxide emissions

CO_2	Carbon dioxide
EGR	Exhaust gas recirculation
EoI	End of injection
EVO	Exhaust valve open
FPGA	Field-programmable gate array
FSN	Filter smoke number
HC	Unburned hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
HRR	Heat release rate
HT	Heat transfer
IMEP	indicated mean effective pressure
IVC	Intake valve close
IP	injection pressure
ISFC	Indicated specific fuel consumption
LTC	Low Temperature Combustion
MULDIC	Multiple stage diesel combustion
NOx	Nitrogen oxides
O_2	oxygen molecular
PCI	Peripheral component interconnect
PXI	Peripheral extensions for intrumentation

Pmax	maximum in-cylinder pressure
PFI	port fuel injection
PCCI	Premixed charge compression ignition
PM	particulate matter
PRR	Pressure rise rate
RoHR	rate of heat release
RCCI	Reactivity Controlled Compression Ignition
SOI	start of the injection
SoC	start of combustion
Texh	Exhaust temperature