

Potentialities of Boot Injection Combined with After Shot for the Optimization of Pollutant Emissions, Fuel Consumption and Combustion Noise in Passenger Car Diesel Engines

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ABSTRACT

The present paper illustrates an investigation about the potentialities of injection rate shaping coupled with an after injection. A pilot shot can either be absent or present before the rate-shaped boot injection. The experimental tests have been performed on a partial PCCI Euro 5 diesel engine endowed with direct-acting piezoelectric injectors. Starting from optimized triple pilot-main-after injection strategies, boot injection was implemented by maintaining the direct-acting piezo injector needle open at part lift. The results of two steady state working conditions have been presented in terms of engine-out emissions, combustion noise and brake specific fuel consumption. In addition, in-cylinder analyses of the pressure, heat-release rate, temperature and emissions have been evaluated. Considering the in-cylinder pressure traces and the heat release rate curves, the injection rate shaping proved to influence combustion in the absence of a pilot injection to a great extent. A pilot shot, added before the boot injection, determined a mitigation of the influence of rate shaping. A simultaneous reduction in NO_x and soot can be achieved for small quantities of fuel injected during boot injection, compared to the optimized triple injection strategies. However, a trade-off exists between emissions and combustion noise when boot injection is applied. On the other hand, the variations in $bsfc$ were only minor. The advantages of the application of boot injection are greater for low to medium loads than for medium to high loads. However, it was not possible to find a working condition in which boot injection provided simultaneous advantages on emissions, noise and consumption, compared to the optimized triple injection.

CITATION: D'Ambrosio, S. and Ferrari, A., "Potentialities of Boot Injection Combined with After Shot for the Optimization of Pollutant Emissions, Fuel Consumption and Combustion Noise in Passenger Car Diesel Engines," *SAE Int. J. Engines* 10(2):2017, doi:10.4271/2017-01-9277.

INTRODUCTION

In order to meet the progressively more stringent emission standards of exhaust emissions from combustion engines of passenger cars, many investigation researches have been conducted in recent years about in-cylinder innovative combustion concepts and aftertreatment systems. The combustion development inside the engine cylinder is influenced to a great extent by the instantaneous fuel injection rate; in this perspective, multiple injection and injection rate shaping strategies appear to be promising techniques, and both of them are reviewed in what follows.

A multiple injection strategy is adopted in diesel engines to replace a single fuel injection with discrete injection events of reduced energizing time in order to improve exhaust emissions, combustion noise (CN) and brake specific fuel consumption ($bsfc$) [1, 2, 3]. Modern Common Rail injection systems are able to manage small injected quantities with reduced electrical dwell time (DT) between consecutive injections. The small injections occur before and after the main injection. In particular, the after injection mode is efficient in reducing the engine-out soot emissions [4], which can be up to 40% lower than in the single injection case [5]. In general, after injections are able to oxidize part of the unburned fuel, and can lead to a reduction in engine-out CO, unburned hydrocarbons (HC) and

particulate matter (PM) emissions; the benefits increase when air-fuel mixing is difficult, i.e. at medium to high loads and under high EGR conditions [6,7]. Furthermore, after injection can be implemented to reduce turbo-lag during engine transients and to diminish the time needed to reach the light-off temperature of the diesel oxidation catalyst during engine warmup [8,9].

Soot emissions are affected negatively by prolonged main injections, since both local equivalence ratios and in-cylinder temperatures tend to increase at the end of the main combustion [10]. In these cases, even a small reduction in the quantity injected during the main injection can determine important benefits in terms of soot [11]. When the fuel is introduced into the cylinder with the after shot, the local equivalence ratio is lower than at the end of a single main injection event, because the fuel is not injected at the same location, due to the motion of the piston [12]. As a result, an enhanced air-fuel mixing occurs for main-after injections, compared to single injections. However, the most important effect of the after injection is the increase in the in-cylinder bulk temperatures during the expansion stroke, and this determines a late strong oxidation of the PM in the combustion chamber [13]. On the basis of the after injected mass, an optimum main-after DT value can be set in the electronic control unit (ECU) for the optimization of the interaction between main and after combustion. The subsequent minimization of the soot produced in the overall fuel

burning process [14]. The maximum smoke reduction is obtained when the start of the after injection is phased within a tiny window of the main diffusive flame [12]. The best dwell times for soot minimization for passenger car diesel engines are in the 600-1000 μs range, and tend to augment when the mass of the after-injected fuel increases. The optimum quantities of the after injection are within the 1.5-3 mm^3 range, and are determined on the basis of a trade-off. In fact, an excessively small quantity produces an unappreciable effect, whereas if the fuel quantity is excessively high, the after injection mass burns in the presence of too high equivalence ratios, with the risk of increased soot emissions [12].

Main-after injection strategies also affect *bsfc*, and engine-out nitrogen oxide (NO_x) and *CO* emissions. A small after injection, close to the main shot, can determine a faster combustion process with respect to the without after shot case. This can explain why engine efficiency can improve and exhaust temperatures can reduce when a closely coupled after injection is added to the main shot [15]. Furthermore, the combustion temperature can rise, thus leading to a higher level of NO_x emissions and lower *CO* emissions than in the absence of the after shot. On the other hand, an after injection that occurs far from the main injection, can improve the NO_x emissions, even though *bsfc* and *CO* can undergo penalties, especially for larger after injections.

Injection rate shaping is a very attractive strategy, because it enables a superior control of the combustion event [16]. It consists of the modulation of the injected flow-rate time history to improve the trade-off between engine performance, engine-out emissions and combustion noise [17, 18, 19]. The fuel injection rate increases to the maximum value and decreases to zero very rapidly when a square pattern of the injected flow-rate is considered. This pattern maximizes the amount of fuel injected for a fixed energizing time, and thus improves *bsfc*. The injection performed at part lift, with the control of the position of the needle, is usually referred to as boot injection. When a boot-shaped main injection is implemented, the injection rate first rises to a lower flow-rate level, and this is followed by another rise to a second higher injected flow-rate level [18]. As a consequence, a small amount of fuel is introduced into the combustion chamber at the beginning of the main injection event, thus leading to a rate-shaped main injection. This main injection pattern allows the diesel fuel to be introduced into the combustion chamber at the rate required for the standard combustion process, avoids premature heat release before *TDC* [20] and limits the intensity of the premixed combustion phase [16,21]. The latter is responsible for the peak heat release rate (*HRR*) value that is normally observed at fuel ignition and which causes the typical noise problems of diesel engines as well as the emissions of NO_x during the subsequent diffusive phase.

When the boot injection is applied [16,21], both NO_x and combustion noise decrease over the whole start of injection (*SOI*) angle range [22]. Furthermore, the injection rate shaping allows the length of the liquid fuel plume to be reduced along the injection event, because of the reduced fuel momentum during the initial stage of the injection. This yields innovative strategies that can prevent liquid fuel wall

impingement and flame quenching (these phenomena occur due to an interaction between the fuel jet and piston bowl) [16], which increase *HC* and *CO* emissions at low loads [23].

Boot injection has generally been recognized as an effective means of reducing NO_x , without a huge detriment in fuel consumption and particulate matter [22]. As the injection process is prolonged by the presence of a boot phase, the combustion time is longer, and, consequently, engine efficiency is expected to decrease [3,24]. Furthermore, the partial opening of the needle causes a pressure drop in the needle-seat flow passage, and thus a reduction in the effective nozzle sac pressure. Lower injection pressures lead to slower air-fuel mixing rates, because the spray cone-angle increases with a subsequent reduction in the liquid lift-off length [23]. Therefore, higher soot emissions can be expected, especially for long boot injections performed at very low values of the needle lift [21].

Nevertheless, the effects of the boot injection on *bsfc* and smoke have a different impact, depending on load and speed of the engine. When the initial injection rate is controlled by throttling the needle-seat passage, the premixed combustion is inhibited and the injection timing can be advanced: as a result, fuel consumption can even improve at medium load conditions, even though black smoke emission remains a drawback [25]. On the other hand, the use of boot injection does not seem to be very efficient at medium speeds and low loads [24], because the increase in the injection time aggravates the issue of the longer combustion duration that is due to the relatively low pressures and temperatures within the cylinder, and shifts the heat release toward the expansion stroke. Engine calibrations that combine different injection strategies in the working area of the New European Driving Cycle (*NEDC*) can be implemented in order to adapt the mixture preparation, the ignition delay and the heat release to the different operative conditions, and thus obtain a generalized optimal engine performance.

The simultaneous application of a boot shaped main injection with an after shot closely phased to the main injection could be a promising strategy for optimizing the trade-off between engine-out emissions, fuel consumption and combustion noise at medium engine loads and speeds. In fact, the penalties of injection rate shaping (primarily the increase in soot and *bsfc*) could be balanced by the advantages of the after injection. On the other hand, there is a lack of investigations in the literature about the possible benefits of a synergy between these two strategies.

The present paper explores the effects of the combined application of boot and after injections at medium load and speed conditions, and highlights the cause-and-effect relationships for exhaust emissions, combustion noise and fuel consumption. Starting from a triple pilot-main-after (*pMa*) injection strategy, optimized with a design of experiments (*DoE*) statistical technique, boot injections, characterized by different flow-rate levels and durations, have been compared on the basis of the engine test results. The combination of the considered innovative strategy with an additional pilot injection has also been analyzed.

EXPERIMENTAL FACILITIES AND TEST ENGINE

The experimental campaign on the engine has been carried out on the highly-dynamic test bed at the Politecnico di Torino ICE laboratories [26,27]. The test bed is equipped with an 'ELIN AVL APA 100' cradle-mounted AC dynamometer, with an AVL AMAi60 raw exhaust-gas analyzer, which is basically made up of three analyzer trains: two of them are complete in order to simultaneously measure pollutant emissions upstream and downstream of the engine aftertreatment system, and the third train is only used to measure the CO_2 levels in the inlet manifold and calculate the EGR mass ratio in the engine. The two complete trains feature the following modules: one heated flame ionization detector for the HC analysis, one heated chemiluminescence detector for the analysis of the NO_x emissions, three nondispersive infrared analyzers for the measurement of the CO_2 concentration and of the low as well as high CO concentration levels, and one paramagnetic oxygen detector for the O_2 levels. Since the aftertreatment system was not present in the test bed installation, and was replaced by a throttle valve, used to simulate pressure drops in the real on-vehicle application, the second complete train was installed in the inlet manifold to measure the O_2 concentration in the inlet manifold. An AVL 415S smokemeter is also used to calculate the exhaust soot emissions. Finally, an 'AVL KMA 4000 Methanol' measuring system is able to continuously meter the engine fuel consumption with a reading accuracy of 0.1% in the 0.28-110 kg/h range.

The main features of the tested engine are reported in Table 1. It is a Euro 5 engine, fueled with conventional diesel oil and equipped with direct-acting piezoelectric (DAP) injectors. A high-frequency piezoelectric transducer has been installed in an adaptor that replaced the glow plug in cylinder #2 in order to measure the pressure time-history of the gases in the corresponding combustion chamber, and a high-frequency piezoresistive transducer has been used to evaluate the absolute pressure in the inlet runner in cylinder #2 to perform the pegging of the in-cylinder pressure. An AVL 365C crank-shaft driven encoder generates the time base for the automatic data-acquisition system, which is based on the AVL Indimodul and is capable of acquiring up to 8 channels of data. The pressure was acquired on a crank base every 0.1 °CA (crank angle degrees). A Tektronix measurement system, made up of a probe and its amplifier, was used to acquire the injection current in cylinder #2. The acquisition system is managed by AVL Indicom software, in order to allow both the online analysis of the high-frequency signals and data storage for post-processing with a previously validated three-zone combustion diagnostic tool [28,29]. In this model, the combustion chamber content is divided into three different zones: an unburned gas zone (which contains fresh-air, EGR and residual gas from previous cycles), a mixture zone, (which is created when the fuel is injected into the unburned zone, and in which air and fuel are mixed in an approximately stoichiometric ratio), and a burned gas zone (which stems from the combustion of the mixture zone). Mass and energy conservation equations are applied to the three zones and are numerically solved, starting from the experimental in-cylinder pressure. The model allows the temperatures of the three zones to be calculated as functions of the engine crank angle. Furthermore, thermal and prompt NO mechanisms have been implemented in the code by considering Zeldovich and Fenimore submodels,

respectively. Soot formation and oxidation are modeled by means of empirical formulas, based on the temperature of the burned gas zone [30].

Table 1. Main specifications and schematic of the tested engine.

Engine type	2.0 L diesel Euro 5
Displacement	1956 cm ³
Bore × stroke	83.0 mm × 90.4 mm
Compression ratio	16.3
Valves per cylinder	4
Turbocharger	Double-stage with valve actuators and WG
Fuel injection system	Common Rail 2000 bar direct-acting piezoelectric injectors
Specific power and torque	71 kW/l – 205 Nm/l
EGR system type	Short-route cooled EGR

A Moehwald Bosch hydraulic test rig, installed in the ICEAL laboratory at the Politecnico di Torino, was used to perform experimental hydraulic tests on the injection system and estimate the instantaneous in-cylinder injection rate [31, 32, 33, 34]. The test rig is equipped with several instruments that are used to detect the injected quantity, instantaneous injected flow-rate, pressure time histories at different locations in the high-pressure injection system circuit and the electric driving signals to the injectors. In particular, the EMI and EVI flowmeters [31,33] have been used to measure the injected mass and the instantaneous injected flow-rate, respectively. EMI determines the volume injected into the injection chamber on the basis of a piston displacement [31]. The simultaneous temperature measurement in the EMI injection chamber allows the fluid density to be evaluated and, hence, the injected mass can also be calculated. Injection into the EVI takes place in an oil-filled measuring tube: the fuel discharge generates a pressure wave, the amplitude of which is measured by means of a piezoelectric transducer. The oil temperature measurement in the tube is used in the algorithm that converts the pressure signal into a flow-rate time history, according to the procedure described in [31].

BOOT INJECTION

The boot injection, performed with the needle intentionally maintained at part lift, can be efficiently implemented with direct-acting injectors, in which the needle position can be controlled on the basis of the charge quantity stored in the piezo stack. The internal hydraulic circuit and the dynamics of the direct-acting piezoelectric injector considered in the present investigation have been described in detail in [35]. A schematic of the injector driving system is reported in Fig. 1. The high-pressure fuel coming from the rail-to-injector pipe enters the injector and, after passing a mechanical filter, goes through pipe P_1 and reaches chamber C_1 . Three drilled passages have been machined in the balance plate to allow a hydraulic connection between chamber C_1 and chamber C_2 ; these passages are referred to as P_2 in the schematic.

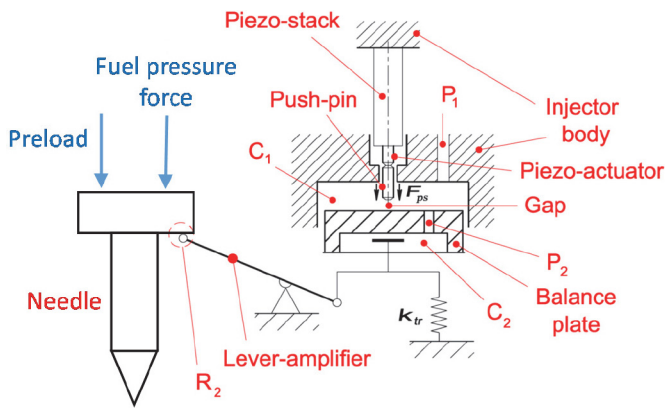


Figure 1. Driving system of the direct-acting piezoelectric injector [35].

The injection dynamics of the *DAP* can be summarized as follows: when the piezo-stack is energized by the *ECU*, it elongates and moves a push-pin. A contact force can therefore be transferred from the piezoactuator to the needle-valve through the kinematic chain, which is made up of the push-pin, the balance plate and the lever-amplifier, which is in contact with the needle. As soon as this opening force prevails over both the needle spring preload and the fuel pressure action on the rear of the needle, which tends to maintain the nozzle closed, the needle rises and allows the fuel to be injected. The lever-amplifier has been designed to magnify the lift of the needle with respect to the elongation of the piezo-stack [35]. At the end of the injection phase, the piezo-stack is discharged, and the force exerted by the lever-amplifier on the needle suddenly drops and the needle-valve closes, because of the spring action and the fuel pressure force. The considered *DAP* injector can guarantee variable shaping of the injected flow-rate through an accurate and flexible modulation of the time history of the current provided to the piezo-stack. As a result, the needle can be prevented from opening completely during the early phase of the injection. This occurs when the needle is stationary at part lift in order to perform a boot injection.

Figure 2 reports the electrical current and p_{EVI} time histories (pressure p_{EVI} is proportional to the injected flow-rate and is the output signal of the *EVI* device) for $M_{inj}=45$ mg and a nominal pressure equal to $p_{rail}=1200$ bar. Boot injection is characterized by means of two parameters: T_{CHA} ($=230$ μ s in Fig. 2), which basically defines the needle part-lift value, and therefore determines the value of flow-rate G_{boot} , and T_{HLD} ($=500$ μ s in Fig. 2), which defines the temporal extension of the boot injection. The higher the T_{CHA} value, the larger the restricted flow-area at the needle-seat passage during the boot injection phase, and the higher the pressure in the sac: this increases the G_{boot} value. The needle is stationary during T_{HLD} , and maintains the position reached at the end of T_{CHA} (Fig. 2).

A new positive current pulse is then given to the piezo-stack at the end of the boot injection. As a consequence, the needle reaches its stroke-end (needle fully open), remains in mechanical contact with a stop-plate (this defines the stroke-end of the needle) and keeps this position till the end of *ET* (the corresponding flow-rate level starts at G_{max} in Fig. 2). When either T_{CHA} or T_{HLD} grows at fixed M_{inj} , the boot-injected fuel quantity (M_{boot}) increases, compared to the whole M_{inj} mass ($M_{inj}=M_{boot}+M_{Main}$). In general, the T_{CHA} and T_{HLD}

parameters offer additional degrees of freedom for a more efficient management of G_{boot}/G_{max} and M_{boot}/M_{inj} at the different engine working conditions.

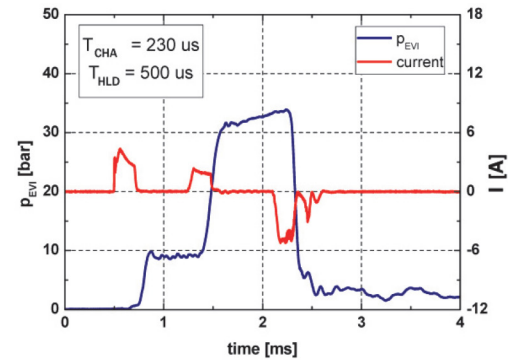


Figure 2. Injected flow-rate pattern and current time history ($M_{inj}=45$ mg, $p_{rail}=1200$ bar) [35].

ENGINE TEST RESULTS

The present analysis has been performed at the following working conditions: $bmeP=5$ bar, $n=2000$ rpm (2000×5) and $bmeP=8$ bar, $n=2500$ rpm (2500×8). These can be considered as representative engine key-points of medium-low and medium-high speeds and loads along a *NEDC* homologation cycle [36].

A triple pilot-main-after injection strategy was preliminarily optimized by means of the *DoE* statistical technique, following the procedure that was described in [28,29] for an indirect-acting piezoelectric injection system. The following parameters were considered as the most relevant input variables for the procedure: rail pressure (p_{Rail}), swirl actuator position (Sw), dwell times between consecutive injections (DT_{Pil} between the pilot and main shots, DT_{Aff} between the main and after shots), main injection timing (SOI_{Main}), the injection quantities in the pilot or after shots (q_{Pil} and q_{Aff}), the inducted air per stroke and per cylinder (m_a). An appropriate number of levels were selected for these variables in order to obtain accurate results with a reasonable number of tests for each variation list. The quantity of fuel in the main injection is set by the test bed control system in order to guarantee the *bmeP* value, and is therefore not present as an input parameter. The *EGR* ratio is not considered as input as the *ECU* does not evaluate this parameter directly, but measures m_a , which is intimately connected to *EGR*. The considered output variables were the engine-out specific NO_x , CO , HC and soot emissions, the *bsfc* and the *CN*. The triple injection strategy, the triple injection calibrations, obtained using the Matlab Calibration

Generation toolbox, was aimed at minimizing engine-out NO_x emissions and combustion noise. The input parameter values of the optimal calibration obtained by the *DoE* for the triple pilot-main-after injection are reported in Tables 2 and 3 for 2000×5 and 2500×8, respectively. Some engine parameters (not relevant for the definition of the optimization, but depending on the input variables and useful to fully characterize the working point) are also included in these Tables.

Starting from the optimized calibration that implements the triple injection, two different boot injection strategies have been considered: a boot injection without a pilot shot, in order to obtain a boot-main-after (*bMa*) injection strategy, and a boot injection with a pilot shot, in order to obtain a pilot-boot-main-after (*pbMa*) injection strategy. Boot injections have been tested with different T_{CHA} and T_{HLD} values: $T_{CHA} = 150, 160, 170, 180, 200 \mu s$ and $T_{HLD} = 200, 300, 400, 500 \mu s$. A full factorial test plan of T_{CHA} and T_{HLD} has been considered for both the *bMa* and *pbMa* strategies, whereas the following quantities remained unchanged with respect to the *DoE* optimized triple *pMa*: m_a , Sw , p_{Rail} , q_{Aft} and DT_{Aft} . Whenever necessary, the main injection timing SOI_{Main} was modified slightly in order to maintain roughly the same value (within 1 °CA) of the angle at which 50% of the fuel has burned (*MFB50*) for the reference *pMa* strategy. In the case of the *pbMa* injection schedule, q_{Pil} and DT_{Pil} , which is defined in this case as the time interval between the end of the pilot electrical current and the beginning of the boot-main electrical current, have been maintained constant in addition to the abovementioned input parameters.

For the 2000×5 key-point, it was not possible to explore all the desired values of T_{CHA} and T_{HLD} , especially in the case of the *pbMa* injection strategy. In fact, if a significant fixed amount of fuel is injected during the pilot and after shots, as well as during the boot phase (this means that M_{boot} is high), the total quantity of fuel M_{inj} , which is necessary to reach the *bmepp* value, should be obtained with a main injection performed with the needle always at part lift. This situation is not significant, since the needle does not arrive at its stroke-end during the main injection, and has not been investigated.

Table 2. Input parameters of the optimal *pMa* calibration and other major engine parameters (dependent on the inputs) at the 2000×5 working point.

Quantity	Value
Inputs	
SOI_{Main} [°CA bTDC]	-0.5
m_a [mg/(stk·cyl)]	371
q_{Pil} [mm ³]	1.8
DT_{Pil} [μs]	640
q_{Aft} [mm ³]	2.8
DT_{Aft} [μs]	610
Sw [%]	35
p_{Rail} [bar]	770
Other engine parameters	
X_{EGR} [%]	35.9
$Global \lambda$ [-]	1.51
$MFB50$ [°CA aTDC]	19.1

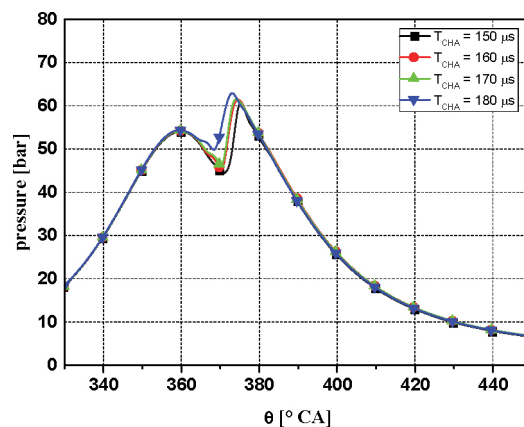
Table 3. Input parameters of the optimal *pMa* calibration and other major engine parameters (dependent on the inputs) at the 2500×8 working point.

Quantity	Value
Inputs	
SOI_{Main} [°CA bTDC]	2.6
m_a [mg/str]	560
q_{Pil} [mm ³]	0.7
DT_{Pil} [μs]	637
q_{Aft} [mm ³]	2.7
DT_{Aft} [μs]	805
Sw [%]	24
p_{Rail} [bar]	1160
Other engine parameters	
X_{EGR} [%]	27.2
$Global \lambda$ [-]	1.53
$MFB50$ [°CA aTDC]	19.0

Boot Injection at Medium-Low Speeds and Loads

2000×5 *bMa*

Figures 3 and 4 report the in-cylinder pressure (p_{cyl}) versus crank-angle (θ) distributions (a) and the corresponding heat release rate traces, together with the schematic injected flow-rates (b), at $n = 2000$ rpm and *bmepp* = 5 bar, for a *bMa* injection schedule. Each figure refers to a specific holding time of the boot injection, i.e. $T_{HLD} = 300 \mu s$ in Fig. 3 and $T_{HLD} = 500 \mu s$ in Fig. 4, and to different T_{CHA} values. The other investigated cases have not been reported for the sake of conciseness. In general, higher variations in the diagrams of p_{cyl} and *HRR* have been found with T_{CHA} than with T_{HLD} . Curves related to $T_{CHA} = 160$ and $170 \mu s$ appear closer than the other curves. This is probably due to the difficulty of the prototypal injection system in controlling the steady-state needle position at low lift values. The delayed SOI_{Main} (after TDC) reported in Table 2 is adopted to reduce NO_x emissions, and is typical of late premixed charge compression ignition (PCCI) engines [36]. It determines similar pressure traces in the compression stroke and a local maximum of p_{cyl} around TDC (Figs. 3a and 4a). Differences are found after TDC, especially for the higher T_{HLD} value (Fig. 4a). In general, the combustion related to the boot injection can be perceived from the p_{cyl} and the *HRR* traces, especially for high values of T_{CHA} , in the $365 \leq \theta \leq 370$ °CA interval for $T_{HLD} = 300 \mu s$ and in the $360 \leq \theta \leq 370$ °CA interval for $T_{HLD} = 500 \mu s$.



a. p_{cyl} crankshaft angle distributions

Figure 3. Effects of T_{CHA} on the in-cylinder pressure and heat release rate curves at $T_{HLD} = 300 \mu s$ (*bmepp* = 5 bar, $n = 2000$ rpm) for the *bMa* injection strategy.