

A Feed-Forward Approach for the Real-Time Estimation and Control of MFB50 and SOI In Diesel Engines

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ABSTRACT

Feed-forward low-throughput models have been developed to predict MFB50 and to control SOI in order to achieve a specific MFB50 target for diesel engines. The models have been assessed on a GMPT-E Euro 5 diesel engine, installed at the dynamic test bench at ICEAL-PT (Internal Combustion Engine Advanced Laboratory at the Politecnico di Torino) and applied to both steady state and transient engine operating conditions.

MFB50 indicates the crank angle at which 50% of the fuel mass fraction has burned, and is currently used extensively in control algorithms to optimize combustion phasing in diesel engines in real-time. MFB50 is generally used in closed-loop combustion control applications, where it is calculated by the engine control unit, cycle-by-cycle and cylinder by-cylinder, on the basis of the measured in-cylinder pressure trace, and is adjusted in order to reduce the fuel consumption, combustion noise and engine-out emissions.

A feed-forward approach has been developed in this paper. This approach is capable of predicting MFB50 on the basis of several parameters, such as, the in-chamber thermodynamic conditions, the injected fuel quantities and timings, the injection pressure, the oxygen concentration and the engine speed and load. The approach has also been inverted in order to predict the start of injection required to achieve a specific MFB50 target in real time. This method can be used in model-based real-time control algorithms to adjust the engine parameters in order to prevent the occurrence of non-optimal combustion cycles.

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INTRODUCTION

The need to comply with the more and more severe emission regulations and the increase in the computational capabilities of modern engine control units (ECU) has led to the development of combustion control algorithms in recent years, in order to reduce fuel consumption, combustion noise and engine-out emissions in real time ([1,2,3,4,5,6,7,8,9,10,11,12,13,14,15]).

One of the most frequently used combustion parameters for this purpose is MFB50, that is, the crank angle at which 50% of the fuel mass fraction has burned.

Most of these algorithms are based on a closed-loop approach: MFB50 is diagnosed by the ECU in real-time, and is then adjusted in order to achieve an optimized value that minimizes a target function, which usually takes into account the engine-out emissions, the combustion noise and the fuel consumption ([1]).

The estimation of MFB50 is usually performed on the basis of the heat release rate that is derived from the measured in-cylinder pressure ([1,2,3]). However, this method is expensive, as it requires the installation of pressure transducers in the combustion chamber of the cylinders.

A novel approach to diagnose MFB50 on the basis of the engine speed measurement has also been presented recently ([4,5,6]). This method has the advantage of not requiring any additional cost, because it is based on the instantaneous engine speed measurement that is already performed in modern engine control systems ([6]).

In general, a closed-loop approach is capable of adjusting the engine conditions in which the diagnosed combustion phasing is non optimal, but suffers from an intrinsic limitation because it cannot avoid the occurrence of low-efficient combustion cycles in advance.

A feed-forward approach, on the contrary, is a methodology that allows MFB50 to be predicted in advance, and could therefore be used to adapt the main engine parameters in real-time in order to achieve a specific MFB50 target. This would prevent the occurrence of inefficient combustion cycles.

Considering that MFB50 is derived from the heat release rate (HRR), a predictive estimation of MFB50 is generally obtained on the basis of a predictive heat release rate model. Several approaches have been presented in the literature to simulate HRR in diesel engines.

For example, the shape of the HRR can be predicted by means of mathematical functions. The best known of these mathematical laws, i.e., the Wiebe function, was originally designed for gasoline engines with external mixture formation([16]). This approach has also been adapted to diesel engines by superposing two Wiebe functions in order to model the premixed and diffusion combustion phases separately ([7, 8]). The Wiebe function does not directly link the injection rate to the heat release rate, as it is based on an empirical mathematical formula, and is therefore devoid of physical consistency.

A model that is based on the apparent combustion time for the central phase of the quasi-steady diesel diffusion combustion was proposed in [17]. This analysis allowed the influence of in-cylinder air density, oxygen concentration, nozzle diameter and injection rate on HRR to be identified and quantified. This 0D phenomenological approach was then generalized in [18] for the final and initial transient phases of diffusion combustion, on the basis of the analysis of the air/fuel mixture at the beginning of the injection process and of the evolution of the turbulent effective viscosity dissipation at the end of the injection process, respectively.

Accumulated fuel mass approaches are based on the assumption that the rate of released chemical energy is proportional to the energy associated with the fuel quantity made available for combustion at the considered moment ([9, 10, 11, 12, 15, 19, 20, 21, 22, 23, 24]). This energy can be computed at time t as the difference between the chemical energy associated with the injected fuel quantity and the cumulative heat release. This approach has the great advantage of being able to directly relate the injection rate to the combustion rate, and is therefore physically consistent.

Although detailed and accurate, the approaches based on a predictive modeling of the heat release rate to derive MFB50 could be difficult to invert, i.e., it is not easy to derive the optimal engine calibration parameters in advance in order to achieve an MFB50 target. Moreover, the computational time could be an issue when these models are implemented in the ECU. For example, in [15] the authors provide a methodology for the inversion of an accumulated fuel mass-based approach, that is based on the differentiation of the original equations and on the integration of the first-order term, in order to find an

approximated solution. As the coefficients of the first-order equations are based on the evaluation of some integral terms, the method is expected to be time consuming.

The predictive HRR models could be very useful to derive a baseline set of optimal engine variables (e.g., boost pressure, oxygen concentration of the inducted charge, injection parameters) in order to achieve a target of MFB50, by means of an off-line optimization. This baseline set of optimal variables could be stored in the ECU by means of look-up tables, as a function of the engine load and speed, and will represent a setpoint for the controllers of the ECU. However, it is very likely that the actual values of the engine variables will be different from the optimal setpoint values during the engine operation, especially in fast transient conditions, and this justifies the need of developing a fast feed-forward approach capable of correcting MFB50 in real-time.

An alternative predictive approach to estimate MFB50, with respect to the predictive combustion models, has therefore been developed and assessed in the present paper. This method is suitable for feed-forward applications, as it can easily be inverted. The proposed methodology is not based on a heat release predictive model, but it directly relates MFB50 to significant engine parameters, such as the start of injection, the ignition delay and the average combustion rate during premixed combustion.

Inversion of the model, which is based on the linearization of the terms of the model that are a function of the start of injection, has therefore been developed.

In general, MFB50 is also a function of other operating parameters, such as the thermodynamic conditions in the intake manifold, the injected quantities, the oxygen concentration, the injection pressure and the engine speed. However, most of these parameters should be set according to load and emission constraints. In addition, the dynamic response of the different parameters is very different, whereas a feed-forward approach should be capable of imposing the target values of MFB50 cycle-to-cycle and cylinder-by-cylinder in advance, and should therefore act on an input quantity that is able to produce variations in MFB50 in the same time intervals. The main variable of interest for this application is therefore the start of injection.

Finally, the direct and inverse approaches have been tested and assessed at both steady-state and transient conditions.

EXPERIMENTAL SETUP

The experimental tests for the model calibration and validation were carried out on a 2.0 dm³ Euro 5 GMPT-E diesel engine, featuring a piezo-driven injection system ([25]).

The main specifications of the engine are reported in Tab. 1.

Table 1. Main specifications of the Euro 5 diesel engine.

Engine type	2.0L "Twin-Stage" Euro 5
Displacement	1956 cm ³
Bore x stroke	83.0 mm x 90.4 mm
Conrod length	145 mm
Compression ratio	16.5
Valves per cylinder	4
Turbocharger	Twin-stage with valve actuators and WG
Fuel injection system	Common Rail 2000 bar piezo
Specific power and torque	71 kW/l – 205 Nm/l
Injector specifications	d=137 μm, C _d = 0.8, C _a = 0.84, C _v = 0.95

The engine was fully instrumented with piezoresistive pressure transducers and K thermocouples (TCK) to measure the pressure and temperature levels at various engine locations. An 'NGK' UEGO air-fuel ratio sensor was placed inside the exhaust system. A KISTLER 6058A41 high-frequency piezoelectric transducer was fitted to the glow-plug seat to measure the in-cylinder pressure time-histories. The in-chamber pressure traces were referenced on the basis of the pressure in the intake manifold, which was measured by means of a high-frequency KISTLER 4075A10 piezoresistive transducer, located at the inlet runner of the cylinder equipped with the in-cylinder pressure sensor.

The experimental tests were carried out at the ICEAL-PT highly-dynamic test bed, whereas the experimental characterization of the injectors was performed at the hydraulic test rig of ICEAL-PT. A detailed description of the two test beds can be found in [9, 25,26,27].

EXPERIMENTAL ACTIVITY

The experimental data were acquired in the frame of a research project carried out with GMPT-E and explained in [25]. The ultimate goal of the activity was to assess the potentiality of innovative piezo- and solenoid-driven Common Rail injection systems on engine performance, emissions, fuel consumption and combustion. Particular attention was paid to the effects of the advanced injection patterns enabled by each Common Rail system.

A standard calibration set, featuring an injection schedule with a pilot shot and a main pulse, was used as the baseline, and three advanced injection patterns were taken into account, namely:

- PPM: two pilot pulses and a main injection;
- PMA: a pilot pulse and a main injection followed by an after pulse;

- PPMA: two pilot pulses and a main injection followed by an after pulse.

The injection strategies were optimized at six key-points, which were identified in order to characterize engine operation for a medium-sized passenger car during an NEDC cycle, in terms of engine speed and BMEP (1500×2, 1500×5, 2000×2, 2000×5, 2500×8, 2750×12 rpm × bar). Different calibration sets were identified for each key-point and for each considered injection strategy by means of a specifically developed DoE approach ([25]).

Table 2. Minimum and maximum range of variation of the main engine parameters for the variation lists and the EGR-sweep tests.

Key-point	No. of tests		p _t	Intake O ₂	λ	q _{pil2}	q _{pil1}	q _{aft}	q _{f, inj}	SOI pil2	SOI pil1	SOI main	SOI aft
			bar	%	-	mm ³	mm ³	mm ³	mm ³	CAD	CAD	CAD	CAD
1500x2	148	max	600	16.48	1.93	2.0	2.5	-	14.4	357.8	362.7	366.9	-
		min	300	13.88	1.31	0.8	0.8	-	8.9	328.0	342.9	360.4	-
1500x5	133	max	900	17.59	1.66	1.5	2.0	-	21.6	352.1	359.5	366.3	-
		min	600	14.63	1.18	0.8	0.8	-	18.6	329.9	345.8	362.0	-
2000x2	144	max	700	16.99	2.08	2.0	2.5	-	16.5	355.5	362.0	367.7	-
		min	400	13.87	1.16	0.7	0.7	-	9.6	316.2	338.6	361.0	-
2000x5	134	max	1150	17.30	1.67	1.5	2.0	-	21.6	347.4	356.6	366.0	-
		min	750	15.06	1.31	0.8	0.8	-	19.3	319.0	340.0	361.5	-
2500x8	130	max	1400	18.42	1.77	-	1.5	4.0	35.4	-	351.1	362.2	394.7
		min	1000	15.55	1.3	-	0.7	0.7	29.5	-	330.6	357.1	376.4
2750x12	160	max	1800	18.21	1.41	-	1.3	4.0	52.9	-	344.2	359.6	392.0
		min	1500	15.82	1.12	-	0.7	0.7	43.9	-	326.5	355.5	376.3
EGR-sweep	70	max	1482	20.89	4.25	-	1.7	-	44.6	-	351.7	364.8	-
		min	453	14.96	1.32	-	1.1	-	9.5	-	332.9	356.8	-

The PPM injection strategy gave the best results for medium-low key-points (1500×2, 1500×5, 2000×2, 2000×5), and showed the best potential to reduce CN (combustion noise), with limited penalties in soot emissions. The PMA strategy showed the best results for the 2500×8 and 2750×12 key-points, as after injection allows soot oxidation to be controlled.

The engine calibration parameters were optimized by means of the DoE technique at each key-point. In particular, several variation lists were identified and tested for each key-point. The main engine parameters that were considered in the variation lists are the rail pressure, the intake charge oxygen concentration, the injection quantities and the timings of the different injection shots. In the present investigation, the experimental tests were used for a detailed study of the ignition delay of the different injection shots.

Table 2 summarizes the maximum and minimum values of the engine parameters considered in the variation lists. Additional experimental data, related to EGR-sweeps, were also included in the dataset for the six key-points. About 900 experimental tests were carried out.

MFB50 MODEL

Definition of the Model

The MFB50 model was defined on the basis of an analysis of a typical heat release curve, which is shown in Fig. 1.

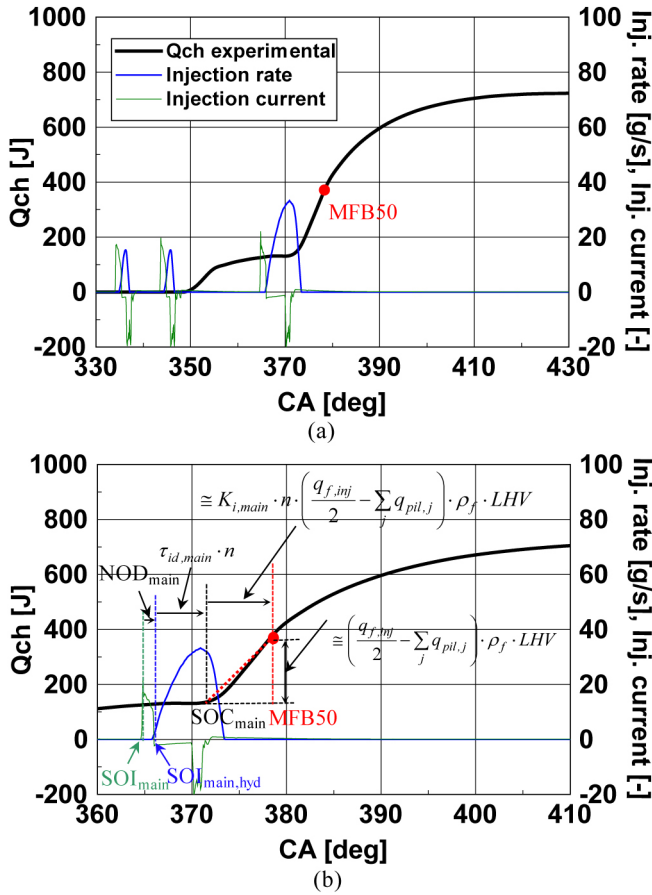


Figure 1. Injection current, injection rate and Q_{ch} for a test case at the 2000x5 key-point that features a PPM injection strategy.

Figure 1a reports the injection current (dashed green line), the injection rate (blue line) and the chemical heat release Q_{ch} (thick black line) for a test case at an engine speed of 2000 rpm and BMEP equal to 5 bar that features a PPM injection strategy.

MFB50 is highlighted in the figure by means of a red circle. MFB50 is in the crank angle interval during the burning of the main pulse for the selected test case, and this has been verified to usually occur for most of the operating conditions. The scale of the x-axis has been resized in Fig. 1b in order to show the proposed methodology more clearly. Figure 1b also reports the electric start of injection of the main pulse (SOI_{main}), the hydraulic start of injection ($SOI_{main,hyd}$), i.e., the instant at which the injector needle opens, the start of combustion of the main pulse (SOC_{main}) and MFB50. It can be observed that MFB50 can be derived, starting from SOI_{main} , by adding three contributions: the nozzle opening delay of the main injection (NOD_{main}), i.e. the delay between SOI_{main} and $SOI_{main,hyd}$; the ignition delay of the main injection, expressed in crank angle degrees ($\tau_{id,main} \cdot 6n/1000$ where $\tau_{id,main}$ is expressed in ms and n in rpm); the product between the average of the inverse heat release rate of the main pulse in the (SOC_{main} , MFB50) interval,

$$\left(\frac{dt}{dQ_{ch}} \right)_{main}^{def} = K_{i,main} = \frac{t_{MFB50} - t_{SOC_{main}}}{Q_{ch}(MFB50) - Q_{ch}(SOC_{main})},$$

expressed in ms/J), and the difference between the chemical heat release at MFB50 and the chemical heat release at SOC_{main} . This quantity is then converted in crank angle degrees. For the sake of clarity, we define K_i as the average of the inverse of the combustion rate from the start of injection of the generic pulse and MFB50.

Assuming that the combustion efficiency is equal to 1, a condition that is generally true in diesel engines because of the high adopted values of the relative air-to-fuel ratio λ , $Q_{ch}(MFB50)$ can be expressed as follows:

$$Q_{ch}(MFB50) \cong \frac{q_{f,inj}}{2} \rho_f LHV \quad (1)$$

where $q_{f,inj}$ is the total injected fuel quantity, expressed in mm^3 , ρ_f is the fuel density, assumed equal to 0.835 mg/mm^3 , and LHV is the lower heating value of the fuel, which is assumed equal to 42.5 J/mg .

Moreover, assuming that the chemical heat of the pilot pulses had been released completely before the start of combustion of the main pulse, the term $Q_{ch}(SOC_{main})$ can be expressed as follows:

$$Q_{ch}(SOC_{main}) \cong \sum_{j=1}^J q_{pil,j} \rho_f LHV \quad (2)$$

where $q_{pil,j}$ is the injected fuel quantity of the generic pilot pulse j , expressed in mm^3 , and J is the total number of pilot pulses that are adopted. As j increases, the start of injection of the pilot pulse advances. It has been verified that, for the considered experimental dataset, that the conditions in which most of the chemical heat of the pilot pulses is released before the start of combustion of the main pulse occur for higher values of the pilot-main dwell-time (DT) than $300 \mu s$.

MFB50 can therefore be modeled as follows:

$$MFB50 = SOI_{main} + NOD_{main} + \tau_{main} \frac{6n}{1000} + K_{i,main} \frac{6n}{1000} \left(\frac{q_{f,inj}}{2} - \sum_{j=1}^J q_{pil,j} \right) \rho_f LHV \quad (3)$$

The different contributions of Eq. (3) are shown in Fig. 1b.

Equation (3) is valid for the cases in which $\sum_{j=1}^J q_{pil,j} \leq \frac{q_{f,inj}}{2}$. This

has been verified to occur for most of the operating conditions reported in Tab. 2 for the considered engine.

However, Equation (3) can theoretically be extended to the cases in which $\sum_{j=1}^J q_{pil,j} > \frac{q_{f,inj}}{2}$

For example, in the case of an injection strategy that employs two pilot pulses, the following equations should be adopted to predict MFB50:

$$MFB50 = SOI_{pil1} + NOD_{pil1} + \tau_{pil1} \frac{6n}{1000} + K_{i,pil1} \frac{6n}{1000} \left(\frac{q_{f,inj}}{2} - q_{pil2} \right) \rho_f LHV \quad \left(q_{pil2} < \frac{q_{f,inj}}{2} \right) \quad (4)$$

$$MFB50 = SOI_{pil2} + NOD_{pil2} + \tau_{pil2} \frac{6n}{1000} + K_{i,pil2} \frac{6n}{1000} \frac{q_{f,inj}}{2} \rho_f LHV \quad \left(q_{pil2} \geq \frac{q_{f,inj}}{2} \right) \quad (5)$$

Equations (4) and (5) are only valid if the heat release of the different pilot pulses does not overlap. By analyzing several operating conditions on the considered engine, it has been verified that the ignition delay of the pil2 shot is usually very long because of the advanced start of injection, and the related chemical heat release partially overlaps that of the pil1 shot in most cases. The double-pilot strategy with $\sum_{j=1}^J q_{pil,j} > \frac{q_{f,inj}}{2}$ is in

fact usually adopted at a low-load condition, i.e., at the 1500×2 and 2000×2 key-points for the considered engine (see Tab. 2).

The method used to estimate MFB50 in the cases in which

$\sum_{j=1}^J q_{pil,j} > \frac{q_{f,inj}}{2}$ and the heat release of the pilot pulses

overlaps has therefore been identified as follows:

$$MFB50 = SOI_{pil2} + NOD_{pil2} + \tau_{pil2} \frac{6n}{1000} + K_{i,pil} \frac{6n}{1000} \frac{q_{f,inj}}{2} \rho_f LHV \quad (6)$$

where:

$$K_{i,pil} = \frac{t_{MFB50} - t_{SOC_{pil2}}}{Q_{ch}(MFB50)} \quad (7)$$

and $K_{i,pil}$ is the average combustion rate of the two pilot pulses.

An example of this situation is shown in Fig. 2, which reports the injection current (green dashed line), the injection rate (blue line) and the heat release Q_{ch} (thick black line) for a test case with a PPM injection strategy at an engine speed of 1500 rpm

and BMEP equal to 2 bar. MFB50 is highlighted with a red circle in the figure. The scale of the x-axis has been resized in Fig. 2b in order to have a better picture of the methodology.

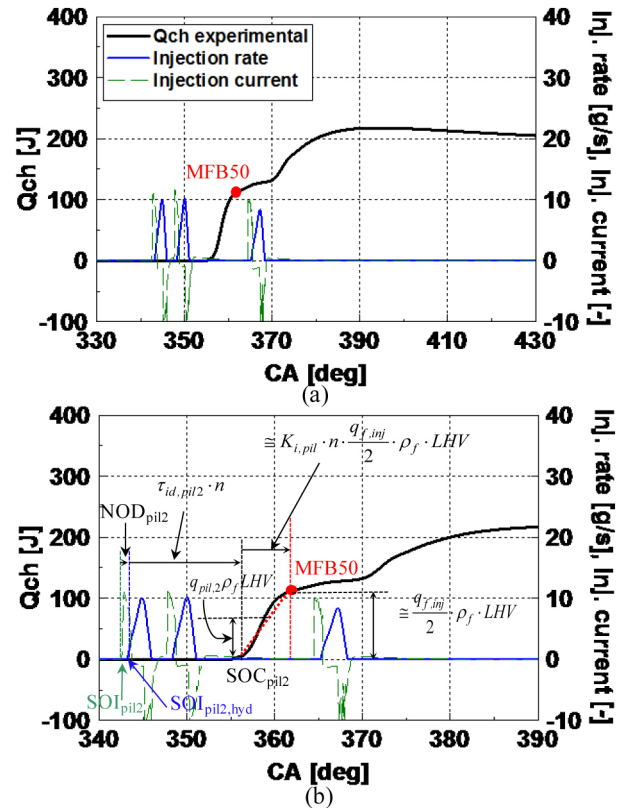


Figure 2. Injection current, injection rate and Q_{ch} for a test case at the 1500×2 key-point that features a PPM injection strategy.

Figure 2 shows that the start of combustion of the pil2 pulse is after the SOI of the pil1 pulse, and that the heat release of the two pulses overlaps, as the shape of the heat release in the (SOC_{pil2} , MFB50) interval does not reflect the contribution of the two pilot shots. This is also confirmed by the fact that the chemical heat release at MFB50 is higher than the $q_{pil2} \rho_f LHV$ product.

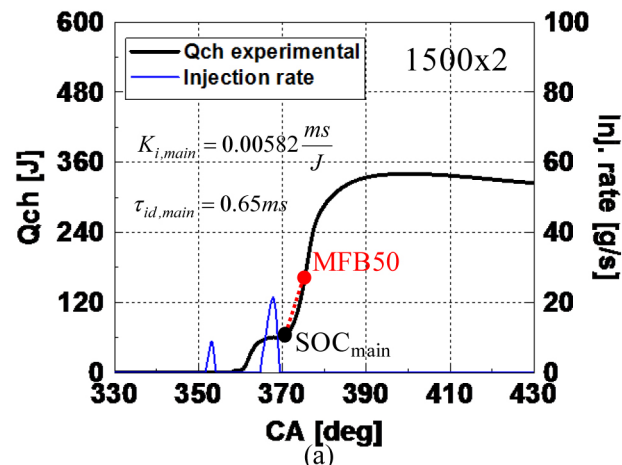


Figure 3. Chemical heat release Q_{ch} and injection rate for 1500×2, 1500×5 and 2500×8 key-points that feature a PPM injection strategy. SOC_{main} and MFB50 are also indicated, along with the connecting line.

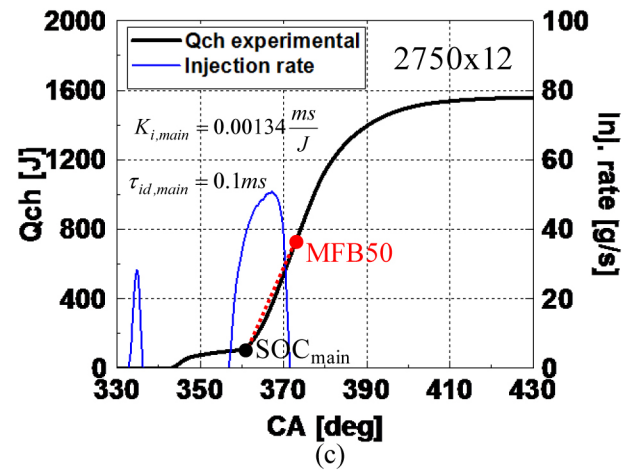
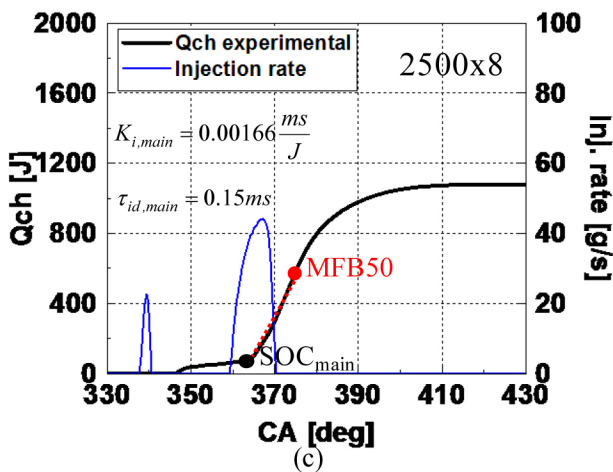
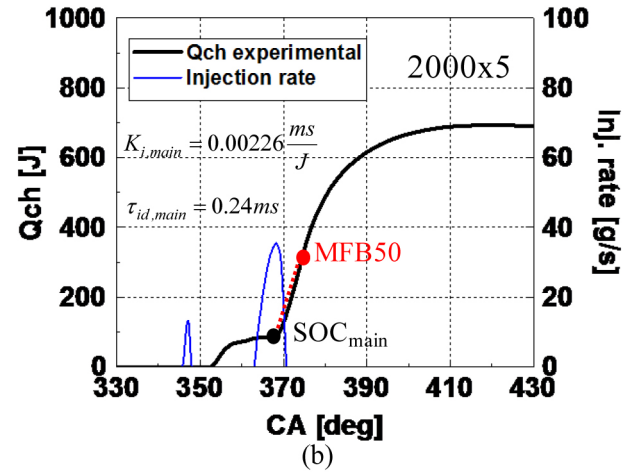
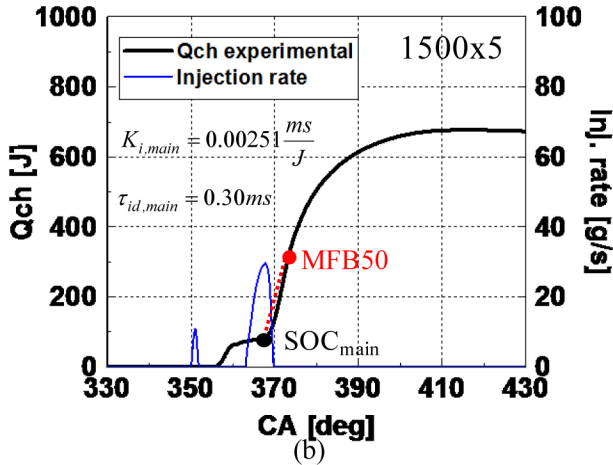


Figure 3. (cont.) Chemical heat release Q_{ch} and injection rate for 1500x2, 1500x5 and 2500x8 key-points that feature a PM injection strategy. SOC_{main} and MFB50 are also indicated, along with the connecting line.

Figure 4. (cont.) Chemical heat release Q_{ch} and injection rate for 2000x2, 2000x5 and 2750x12 key-points that feature a PM injection strategy. SOC_{main} and MFB50 are also indicated, along with the connecting line.

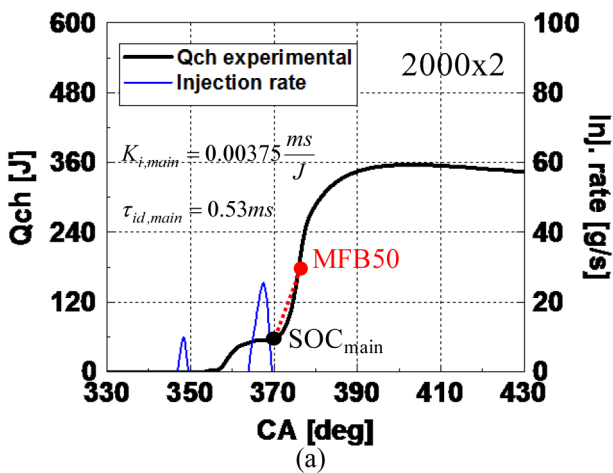


Figure 4.

Figures 3-4 report the injection rate (blue line) and the chemical heat release Q_{ch} (thick black line) for the six nominal baseline NEDC key-points that feature a pilot-main (PM) injection strategy. MFB50 is reported with a red circle, and SOC_{main} with a black circle. The injection rate profiles plotted in Figs. 1, 2, 3, 4 were derived on the basis of the interpolation of experimental profiles, which were acquired at the hydraulic test bench of ICEAL-PT as a function of the energizing time and injection pressure.

The slope of the line that connects SOC_{main} to MFB50 is the average heat release rate (expressed in J/deg) in the (SOC_{main} , MFB50) interval, that is related to the $K_{t,main}$ parameter.

The values of $\tau_{id,main}$ and $K_{t,main}$ are also reported for the six NEDC key-points. It can be observed that both $\tau_{id,main}$ and $K_{t,main}$ reduce to a great extent when the engine load increases.

It has been verified, by analyzing all the test cases considered in Tab. 2, that MFB50 is in the burning interval of the main pulse in almost all cases, i.e., Eq. (3) applies.

Equations (3, 6) have the advantage of directly relating MFB50 to the electric start of injection of the main or pilot pulses, which is one of the main input signals of the engine control unit. Moreover, the model takes into account the effects of the injected fuel quantities, of NOD, of the engine speed n , of the ignition delay and of the average of the inverse of the heat release rate in the (SOC, MFB50) interval on MFB50. The ignition delay and the average of the inverse of the heat release rate will be correlated to the main engine quantities that are measured by the ECU in Section "Model calibration", in order to derive a formulation for MFB50 that is suitable for inversion.

The main limitations of Eqs. (3, 6) arise in the cases in which the dwell-time between the main pulse and the closest pilot injection is very short, and the heat release of these two injection shots overlaps before MFB50 is reached. In this case, it is difficult to identify the start of combustion of the main pulse on the basis of the heat release rate. However, by analyzing all the test cases considered in Tab. 2, it has been verified that this condition occurs in a very limited number of test cases, which are usually not of interest for the optimized calibration strategy used for the considered engine.

Determination of the Experimental Dataset of the Model Calibration Parameters

τ_{id} and K_i are the calibration parameters of the MFB50 model that is defined by Eq. (3) for most of the operating conditions, and by Eq. (6) in the few cases in which the injected quantities of the pilot shots are higher than one half of the total injected quantity.

It is necessary to derive physically-consistent correlations for both parameters, as a function of significant engine quantities, in order to use the MFB50 model for predictive purposes. A dataset of the experimental values of both calibration parameters should therefore be determined for calibration.

The identification of both parameters requires the calculation of the start of combustion of the pilot and main pulses and the evaluation of the chemical heat release curve, as shown in Figs. 1, 2, 3, 4.

First, the experimental chemical energy release was calculated by means of a single zone model, starting from the measured in-cylinder pressure ([16]).

The procedures to estimate the start of combustion of the pilot and of the main pulse have been explained in detail in [28], but a summary is reported hereafter.

The overall SOC, which is the start of combustion of the most advanced injection shot, usually a pilot injection, can be defined as the crank angle at which the first chemical energy fraction is released. This method is based on the application of the energy conservation equation to the whole combustion chamber, and has been reported by the authors in ([29, 30]).

The SOC of the subsequent injection shots, and therefore also of the main injection, was detected by identifying the relative minima of the HRR curve, as also suggested in [31]. This procedure was referred to as the "HRR local minima" (HLM) method in [28].

Once the SOC has been determined for both the pilot and main injection shots, the value of τ_{id} can be calculated as follows:

$$\tau_{id} [ms] = \frac{SOC[deg] - SOI_{hyd}[deg]}{n[rpm] \cdot 360 / 60} \cdot 1000 \quad (8)$$

The hydraulic start of injection for each injection pulse was calculated, starting from the measured injector current, taking into account the NOD, by means of correlations as a function of the energizing time (ET) and of the injection pressure p_f . These correlations were calculated on the basis of a similar methodology to that illustrated in [27].

The experimental values of the start of combustion of the main and of the pilot pulses were also used to determine the experimental values of the K_i parameter, as follows:

$$K_i \left[\frac{ms}{J} \right] = \frac{MFB50[deg] - SOC[deg]}{\frac{Q_{ch}(MFB50)[J] - Q_{ch}(SOC)[J]}{1000}} \cdot \frac{1000}{n[rpm] \cdot 360 / 60} \quad (9)$$

where MFB50 is detected as the crank angle at which 50% of the maximum chemical energy of the fuel has been released.

A dataset of the experimental values of τ_{id} and K_i for the pilot and main injections, was determined for all the operating conditions reported in Tab. 2.

MODEL CALIBRATION

It is necessary to derive robust physically-consistent correlations for the model calibration parameters (τ_{id} and K_i) in order to use Eqs. (3, 4, 5, 6) for predictive purposes.

A detailed analysis of the ignition delay was carried out in [28], for the same engine, over the same steady-state operating conditions considered in the present paper. A global-mechanism approach, based on the separate modeling of the physical and chemical contributions, and a control-oriented approach were developed for both the main and pilot pulses.

In the present work, the control-oriented approach has been used, considering that the proposed model must be simple enough to be suitable for feed-forward control applications.

It has been verified, in [28], that a robust correlation exists for the pilot injections between the ignition delay and the charge density calculated at SOI. The charge density affects both the