

Combustion Prediction by a Low-Throughput Model in Modern Diesel Engines

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ABSTRACT

A new predictive zero-dimensional low-throughput combustion model has been applied to both PCCI (Premixed Charge Compression Ignition) and conventional diesel engines to simulate HRR (Heat Release Rate) and in-cylinder pressure traces on the basis of the injection rate.

The model enables one to estimate the injection rate profile by means of the injection parameters that are available from the engine ECU (Electronic Control Unit), i.e., SOI (Start Of main Injection), ET (Energizing Time), DT (Dwell Time) and injected fuel quantities, taking the injector NOD (Nozzle Opening Delay) and NCD (Nozzle Closure Delay) into account. An accumulated fuel mass approach has been applied to estimate Q_{ch} (released chemical energy), from which the main combustion parameters that are of interest for combustion control in IC engines, such as, SOC (Start Of Combustion), MFB50 (50% of Mass Fraction Burned) have been derived. Q_{net} (charge net energy) has been evaluated and the in-chamber pressure during the combustion phase has been determined through the inversion of a single-zone combustion model. Optimized values of the isentropic coefficient have been identified and the compression and expansion phases have been approximated by proper polytropic evolutions. The calculated in-cylinder pressure time-history can be used to estimate PFP (Peak Firing Pressure), IMEP (Indicated Mean Effective Pressure) and CN (Combustion Noise).

The generality of the developed approach has been investigated in order to identify the model parameters that require specific tuning for different engine application. To

this end, the model has been applied to PCCI diesel engines with different CR (Compression Ratio) values, to a twin-stage EURO V diesel engine equipped with piezoelectric injectors, and to three different diesel engines, with displacements of 2.9 dm³, 1.7 dm³ and 1.3 dm³, respectively, equipped with solenoid injectors. The very accurate results that have been obtained highlight the robustness of the proposed approach.

The results have also substantiated the potential of the model to realize a real-time cycle-to-cycle combustion control, which can be applied to improve IC engine emissions, fuel consumption and combustion noise.

INTRODUCTION

Interest in predictive modeling for feed-forward combustion control in IC engines has increased over the last few years in the car manufacturing industry. The main objective of this kind of modeling consists in realizing a cycle-to-cycle combustion control to improve the engine performance, in terms of emissions, fuel consumption and combustion noise.

One of the most common techniques for combustion control involves the closed-loop control of a specific combustion metric, such as the 50% of mass fraction burned (MFB50). This metric is usually evaluated by a real-time processing of the in-cylinder pressure time-history, which is measured using a pressure sensor that is located in the glow plug. The limits of this technique concern the additional cost of the transducer and the fact that closed-loop control can correct non-optimal combustion engine cycles after they have already been occurred.

The main advantage of a reliable feed-forward combustion model is that it can predict the engine cycle combustion parameters a priori, so that the engine operating variables can be set to realize an optimized combustion, in this way avoiding the occurrence of inefficient engine cycles.

The predictive evaluation of the MFB50 quantity requires the estimation of the fuel released chemical energy. To this end, different approaches have been proposed in the literature. The methodologies that appear to be suitable for feed-forward control can roughly be summarized in two types: the accumulated fuel mass approach ([1, 2, 3, 4]) and the Wiebe function approach ([5]). The first of these considers the chemical-energy release rate to be proportional to the mass of the injected fuel that is not yet combusted in the engine cylinder. This approach allows the injection rate to be directly linked to the HRR curve, and thus results to be more physically consistent than the other approach, which usually requires a large number of calibration coefficients to allow an accurate evaluation of the burned mass fraction to be made. Moreover, the Wiebe function approach requires the calibration of mathematical coefficients which are physically difficult to interpret and does not consider the injection rate as an input quantity ([6]). In [6, 7, 8], the Barba model is also suggested for combustion prediction in diesel engines. This model and similar approaches, such as the one presented in [9], have the potential of calculating the diesel heat-release rate in a very accurate way, with “physically tangible” calibration parameters ([6]), but they seem to require excessive throughput for feed-forward control tasks. Thus, the accumulated fuel mass approach was considered by the authors as the most suitable approach to heat release prediction for feed-forward control tasks.

A zero-dimensional predictive combustion model to be implemented in an engine ECU for model-based control has been developed and assessed on a PCCI diesel engine ([10]). On the basis of the injection parameters and a few quantities measured in the intake manifold, the model is capable of calculating the released chemical energy Q_{ch} , the charge net energy Q_{net} , the in-chamber pressure and the related combustion parameters, such as MFB50, p_{max} and IMEP. The approach stems from physically consistent correlations for the ignition delay and the chemical energy release rate, and therefore it is effective in capturing cause and effect relationships between injection and combustion. It is also very simple and fast to apply, and thus suitable for implementation in the engine ECU.

The present work has the aim of assessing to which extent these correlations can be applied to different engines, featuring different CR values and injection systems. Moreover, the key parameters requiring specific engine dependent calibration to maintain an excellent result accuracy have been found.

MODEL DESCRIPTION

A detailed description of the model can be found in [10] and only the main equations are summarized hereafter. For each pulse j ($1 \leq j \leq n$) of the injection event, the released chemical energy rate $Q_{ch,j}$ at time step t was evaluated as:

$$\frac{dQ_{ch,j}}{dt} = \sum_{j=1}^n K_j [Q_{fuel,j}(t - \tau_j) - Q_{ch,j}(t)] \quad (1)$$

where K_j and τ_j are model calibration quantities and $Q_{fuel,j}$ is the chemical energy associated to the injected fuel mass:

$$Q_{fuel,j}(t) = \int_{t_{SOI,j}}^t \dot{m}_{f,inj}(t) LHV_p dt \quad t \leq t_{EOI,j} \quad (2)$$

$$Q_{fuel,j}(t) = \int_{t_{SOI,j}}^{t_{EOI,j}} \dot{m}_{f,inj}(t) LHV_p dt \quad t > t_{EOI,j} \quad (3)$$

SOI and EOI are the start and end of injection, respectively, $\dot{m}_{f,inj}$ is the injection rate of the fuel mass and LHV_p is the fuel lower heating value at constant pressure. The total released chemical energy can be calculated as:

$$Q_{ch} = \sum_{j=1}^n Q_{ch,j} \quad (4)$$

In Eq. (1) the rate of released chemical energy, for each pulse j , is proportional to the mass of the injected fuel that is not yet combusted in the cylinder. With respect to the approach proposed in [1,2], the energy associated to the injected fuel is not evaluated at instant t , but at instant $(t - \tau_j)$, in order to take the ignition delay effects into account. According to [2], K_j can be interpreted as a spray or atomization characteristic number. As K_j increases, the droplet size reduces and/or spray penetration is enhanced, so that the combustion rate is increased. For a given spray, K_j can be related to the swirl ratio or to the in-chamber turbulence.

The combustion phase simulation was then carried out using a single-zone model, starting from the charge net energy (i.e., the apparent heat release) Q_{net} :

$$dp = \left(\frac{\gamma - 1}{V} \right) \left(dQ_{net} - \frac{\gamma}{\gamma - 1} p dV \right) \quad (5)$$

where $\gamma = c_p/c_v$.

A reliable estimation of the γ coefficient is fundamental to achieve accurate results. In [10] γ was shown to depend mainly on RAFR and a correlation between γ and RAFR was found by analyzing the experimental in-cylinder pressure time histories with the multizone premixed-diffusion combustion model [11], for different working conditions on the same PCCI engine referred to in the present paper:

$$\gamma = -0.0064 \cdot \text{RAFR}^2 + 0.0563 \cdot \text{RAFR} + 1.2151 \quad (6)$$

As for the evaluation of Q_{net} in Eq. (5):

$$Q_{\text{net}} \cong Q_{\text{net}}^{\text{SOC}} - \Delta Q_{\text{f, evap}} \quad (7)$$

$$Q_{\text{net}}^{\text{SOC}} \cong Q_{\text{ch}} \frac{m_{\text{f, inj}}^* \text{LHV}_p - \Delta Q_{\text{ht, glob}}}{m_{\text{f, inj}}^* \text{LHV}_p} \quad (8)$$

$Q_{\text{net}}^{\text{SOC}}$ is the charge net energy evaluated from SOC; $\Delta Q_{\text{ht, glob}}$ is the global heat transfer from the in-cylinder gases to the combustion chamber wall; $\Delta Q_{\text{f, evap}}$ is the energy associated to fuel evaporation, and $m_{\text{f, inj}}^*$ is the total injected fuel mass per cylinder and per engine cycle. The part of the Q_{net} curve in the time interval between SOI and SOC can be approximated by a parabolic profile, in order to guarantee the continuity of the first-order derivative of Q_{net} with respect to time at SOC.

The in-chamber pressure at SOI was evaluated by assuming the following polytropic evolution

$$pV^m = \text{const} \quad (9)$$

from the in-chamber pressure at IVC (p_{IVC}). The latter was correlated to the pressure p_{map} in the intake manifold, which is measured by the ECU using a correction factor as follows:

$$p_{\text{IVC}} = p_{\text{map}} + \Delta p_{\text{int}} \quad (10)$$

where Δp_{int} represents the pressure losses between the intake manifold and combustion chamber at IVC. The expansion phase after the EOC was simulated using the following polytropic evolution:

$$pV^{m'} = \text{const} \quad (11)$$

The gas exchange processes can be approximated by considering the in-chamber pressure during the intake phase

as being equal to the intake manifold pressure and the in-chamber pressure during the burned gas expulsion phase as being equal to the exhaust manifold pressure. The approach is actually justified by the fact that pumping losses in diesel engines are low compared to the total energy associated to the whole engine cycle.

TUNING OF THE MODEL PARAMETERS

The combustion model tuning was carried out in [10] with reference to a 16V engine having a central-dome combustion chamber and CR = 15.5:1. Physically-consistent correlations were found for the model parameters that were introduced into Eq. (1) and Eqs. (7, 8, 9, 10, 11); these are summarized hereafter, for the sake of completeness.

TUNING OF K AND τ FOR Q_{ch} ESTIMATION

K_j and τ_j were appropriately tuned ([10]) in order to have a good matching between the Q_{ch} curves predicted by Eq. (1) and the experimental values of the released chemical-energy that were diagnosed from the experimental in-chamber pressure traces using the multizone tool described in ([11]). The matching was optimized to achieve the best accuracy, not only in terms of MFB50, but also of P_{max} , when the whole model is applied.

A robust correlation was found between τ_j and ρ_{TDC} by processing a very wide range of data ([10]), where ρ_{TDC} is the charge density at TDC evaluated as:

$$\rho_{\text{TDC}} = \frac{p_{\text{map}}}{RT_{\text{map}}} \left(\frac{V_{\text{BDC}}}{V_{\text{TDC}}} \right) \quad (12)$$

Where V_{BDC} and V_{TDC} are the camber volumes at BDC and TDC, T_{map} is the temperature measured in the intake manifold, ρ_{TDC} was shown to be effective in taking the EGR effect (through T_{map} and p_{map}) and the boost effect into account at the same time, and also, implicitly, the effect of the rail pressure (p_{rail}), because higher p_{rail} values are associated to higher p_{map} . In addition, the engine load resulted to have a significant influence on τ_j . However, a simple input parameter is required by the ECU to identify the load value at a certain working condition, because the model should be used for feed-forward combustion control. It has been shown in [10] that the total injected fuel quantity $q_{\text{f, inj}}$, which is available from the ECU at each instant, is a good estimate of the engine load. All these effects were lumped together in the following WF (weight function) parameter:

$$WF = C \cdot (\rho_{TDC} / \rho_{TDC,0})^a + D \cdot (q_{f,inj} / q_{f,inj,0})^b$$

$$C = 0.5, D = 0.3, a = b = 2,$$

$$\rho_{TDC,0} = 35 \text{ kg/m}^3, q_{f,inj,0} = 80 \text{ mm}^3$$
(13)

and the following correlations τ_j versus WF were identified for both main and pilot injection pulses

$$\begin{aligned} \tau_{main} = & 35118 \cdot WF^6 - 129332 \cdot WF^5 + \\ & + 194421 \cdot WF^4 - 153979 \cdot WF^3 + \\ & + 68524 \cdot WF^2 - 16443 \cdot WF + 1801.3 \mu\text{s} \quad WF < 0.6 \end{aligned}$$
(14)

$$\tau_{main} = 120 \mu\text{s} \quad WF \geq 0.6$$
(15)

$$\begin{aligned} \tau_{pil} = & 20456 \cdot WF^4 - 47847 \cdot WF^3 + \\ & + 40369 \cdot WF^2 - 14544 \cdot WF + 2273.4 \mu\text{s} \quad WF < 0.6 \end{aligned}$$
(16)

$$\tau_{pil} = 400 \mu\text{s} \quad WF \geq 0.6$$
(17)

As is reported in [10], coefficients a, b, C and D in Eq. (13) were identified to optimize the correlations τ_{main} (WF) (Eqs. 14-15) and τ_{pil} (WF) (Eqs. 16-17), which are expressed by means of simple polynomial expressions, so as to be easily implemented in the engine ECU. The coefficient values reported in Eq. (13) were identified for the CR 125.5 PCCI diesel engine, described in [10], over the whole engine map. Nevertheless, in the following sections it will be shown that Eqs. (13, 14, 15, 16, 17) are of quite general application, when used for engines featuring different displacement values or equipped with different injection systems.

When after-injections are enabled, τ_j of the after-pulses can be considered virtually constant and equal to 500 μs . With reference to multiple pilot injections, equations (16) and (17) can only be applied for the closest pilot pulse to the main shot. Instead, τ_j correlations are currently under investigation for the other pilot pulses and the related results will be presented in a future companion paper.

With reference to K_j , the analysis carried out in [10] showed that K_{main} decreases as the load (i.e., $q_{f,inj}$) increases and as the engine speed N decreases. The following correlations were identified:

$$K_{main} = 0.4819 \cdot n[\text{rpm}] + 848.45 \text{ s}^{-1} \quad q_{f,inj} < 20 \text{ mm}^3$$
(18)

$$K_{main} = 0.2992 \cdot n[\text{rpm}] + 532.17 \text{ s}^{-1} \quad q_{f,inj} > 40 \text{ mm}^3$$
(19)

$$K_{main} = \frac{(0.4819 \cdot n[\text{rpm}] + 848.45) - (0.2992 \cdot n[\text{rpm}] + 532.17)}{20} \cdot (q_{f,inj}[\text{mm}^3] - 20) + 0.4819 \cdot n[\text{rpm}] + 848.45 \text{ s}^{-1}$$

$$20 \text{ mm}^3 \leq q_{f,inj} \leq 40 \text{ mm}^3$$
(20)

It was shown that an average K_j value of 2500 s^{-1} can be used for pilot and after shots.

TUNING OF $\Delta Q_{ht, glob}$ AND $\Delta Q_{f, evap}$ FOR Q_{net} ESTIMATION

The values of $\Delta Q_{ht, glob}$ and $\Delta Q_{f, evap}$ used in Eqs. (7) and (8) for Q_{net} estimation were experimentally evaluated in [10] by processing the experimental in-cylinder pressure traces through multizone ([11]) and one-zone diagnostic models, for a large number of engine operating conditions.

$\Delta Q_{ht, glob}$ is the heat globally transferred to the walls and is therefore expected to be proportional to the product of the heat transfer rate and the average combustion temporal length. Moreover, the first of these two contributions should be proportional to the combustion temperature (namely, to the engine load) whereas the second one is inversely proportional to the engine speed, so that a correlation for $\Delta Q_{ht, glob}$ can be sought as a function of the load/speed parameter ([10]). Since $q_{f,inj}$ has already been considered as a load parameter, one obtains:

$$\Delta Q_{ht, glob} = 9000 \frac{q_{f,inj}[\text{mm}^3]}{N[\text{rpm}]} \text{ J}$$
(21)

$\Delta Q_{f, evap}$ has been shown to decrease as the load increases, most likely because of the ignition delay reduction ([10]). Thus one has:

$$\begin{aligned} \Delta Q_{f, evap} = & 0.0052 (q_{f,inj}[\text{mm}^3])^2 + \\ & - 0.6421 (q_{f,inj}[\text{mm}^3]) + 20.509 \text{ J} \end{aligned}$$
(22)

TUNING OF Δp_{int} , m, m' FOR THE ESTIMATION OF THE PRESSURE TIME-HISTORIES

The pressure drop from the intake manifold to the combustion chamber at IVC is expected to be a function of the engine speed and load. Therefore, from the analysis of experimental values of p_{map} and p_{IVC} obtained in [10], one can write:

$$\Delta p_{int} = 0.0042917 + 0.000077366 \cdot n [\text{rpm}] + 0.0072429 \cdot q_{f, inj} [\text{mm}^3] \quad (23)$$

Table 1. Summary of the predictive combustion model

Target	Input data required	Correlation parameters	Calculation procedure
Injection rate	$q_{f, inj}$ SOI, ET, DT (ECU map)	Injector NOD and NCD	Calculation of a triangular profile
Q_{ch} , MFB50	Injection rate p_{map} (measured) T_{map} (measured)	$K(q_{f, inj})$; $\tau(p_{TDC}, q_{f, inj})$; $p_{TDC}(p_{map}, T_{map})$.	Q_{ch} model (K, τ)
Q_{net}	Q_{ch} $q_{f, inj}$ N (measured)	$\Delta Q_{ht, glob}(q_{f, inj}, N)$, $\Delta Q_{f, evap}(q_{f, inj})$	Q_{net} model ($\Delta Q_{ht, glob}$, $\Delta Q_{f, evap}$)
Pressure trace, p_{max} , IMEP	Q_{net} $q_{f, inj}$ N , p_{exh} (meas.), RAFR (meas.)	$\Delta p_{int}(q_{f, inj}, N)$, γ (RAFR), $m, m'(q_{f, inj})$	Polytropic laws; inversion of single zone model (Q_{net}, γ)

Finally, m and m' were experimentally determined by fitting polytropic processes to the measured pressure traces during compression and expansion strokes. It has been found that ([10]):

$$m = 0.0011 \cdot q_{f, inj} [\text{mm}^3] + 1.3001 \quad (24)$$

$$m' = -0.0016 \cdot q_{f, inj} [\text{mm}^3] + 1.383 \quad (25)$$

Table 1 gives a summary of the model structure and input/output parameters. For a given target output quantity in column 1, columns 2 and 3 show the required input and correlation quantities, respectively, whereas column 4 explains which submodel is used for the calculation of the examined output quantity. Thus, for example, the calculation of Q_{ch} and MFB50 (second row in Table 1) requires the injection rate, T_{map} and p_{map} (column 2) as input parameters. These two quantities allow the K and τ parameters (column 3) to be estimated as is required by the Q_{ch} predictive model (column 4).

TEST ENGINE

The low-throughput combustion model has been applied to a number of GMPT-E DI diesel engines, with different combustion and injection systems.

The first engine (Tab. 2) features a 16V central-dome combustion chamber and is equipped with a solenoidal Common Rail injection system, a VGT turbocharger, as well as with a cooled EGR system. Three prototypes were considered in this investigation presenting compression ratios (CR) of 16.5:1, 15.5:1 and 15:1, respectively. The prototypes with CR = 15.5:1 and CR = 15:1 were capable of running either in conventional combustion mode or in Premixed Charge Compression Ignition (PCCI) mode ([12, 13]).

Table 2. Main specifications of the 16V engine with the central-dome combustion chamber

Engine type	1.9 l – EURO 5
Displacement	1910 cm ³
Bore x stroke	82.0 mm x 90.4 mm
Stroke-to-bore ratio	1.10
Compression ratio (baseline)	16.5
Valves per cylinder	4
Turbocharger	Single-stage with VGT
Fuel injection system	Common Rail 2nd generation 1600bar
Specific power and torque	57.6 kW/l – 167 Nm/l

The second engine (Tab. 3) was derived from the GMPT-E 2.0 dm³ 16V twin-stage turbocharged engine, equipped with piezoelectric injectors ([14]). The oxidation catalyst and particulate filter of the exhaust line were removed from this engine. The correct exhaust backpressure was then set by means of a throttle valve that was fitted to the exhaust line.

For both engines, all of the experimental tests were carried out on the AVL highly dynamic test bed at the ICEAL of Politecnico di Torino.

Furthermore, the model development was supported with additional experimental data supplied by GMPT-E for three different engines (Tab. 4) equipped with solenoidal injectors and presenting a total displacement of 2.9 dm³, 1.7 dm³ and 1.3 dm³, respectively.

Table 3. Main specifications of the twin-stage turbocharged engine

Engine type	2.0L “Twin-Stage” EURO V
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

Table 4. Main specifications of the three diesel engines tested at the GMPT-E facilities

Compression ratio [-]	16.5
Number of cylinders	6
Displacement [dm ³]	2.9
Injection system	Common Rail

(a)

Compression ratio [-]	16.5
Number of cylinders	4
Displacement [dm ³]	1.7
Injection system	Common Rail

(b)

Compression ratio [-]	16.8
Number of cylinders	4
Displacement [dm ³]	1.3
Injection system	Common Rail

(c)

EXPERIMENTAL SETUP

The test rig was equipped with: an ‘ELIN AVL APA 100’ cradle-mounted AC dynamometer, which has a power of 220 kW, a nominal torque of 525 Nm and a maximum speed of 12000 rpm, an ‘AVL KMA 4000’ system to continuously measure the engine fuel consumption, and an ‘AVL AMAi60’ raw exhaust-gas analyzer. The latter is basically made up of three analyzer trains. Two of these trains have: one heated chemiluminescence detector to analyze NO_x - NO; a couple of heated flame ionization detectors to analyze THC/CH₄; three nondispersive infrared analyzers to measure low and high CO and CO₂ concentrations; one paramagnetic oxygen detector. These trains allow the pollutant levels to be contemporarily measured upstream and downstream from the aftertreatment equipment. The third train is made up of a detector which is used to measure the CO₂ levels in the intake manifold, in order to evaluate the engine EGR ratios. The dynamic test bed was also equipped with: an AVL 715S smokemeter, an AVL 439S opacimeter and an AVL SPC472 Smart Sampler to measure the PM emissions from the diesel engines.

All of the abovementioned measuring instruments are controlled by the PUMA OPEN 1.3.2 automation system, which also includes ISAC 400 software to simulate not only the vehicle, but also the driver's behavior.

The test engine was instrumented with piezoresistive pressure transducers and K thermocouples (TCK) in order to measure the pressure and temperature at the following locations: upstream and downstream from the compressor and turbine, at the manifold inlet, and downstream from the aftertreatment system. The TCKs were also used to measure the temperatures in the intake and exhaust runners, downstream and upstream from the EGR cooler, and downstream from the intercooler. An ‘NGK’ UEGO air-fuel ratio sensor was located at the exhaust. One KISTLER 6058A41 piezoelectric transducer was installed at the glow-plug seat to measure the pressure time-histories in one of the cylinder combustion chambers; a high-frequency KISTLER 4075A10 piezoresistive transducer was used to detect the pressure levels in the inlet runner of the same cylinder to reference the in-chamber pressure. An AVL 365C crankshaft driven encoder generated the time base for an automatic 14 bit data-acquisition system (based on the AVL Indimodul 620 system), which is capable of acquiring up to 8 channel data with a maximum frequency of 800 kHz per channel. The acquisition system was managed by AVL Indicom software and interfaced with PUMA OPEN 1.3.2, so that both online analysis of the indicated cycle and data storage for postprocessing with specifically developed homemade diagnostic tools could be carried out.