

Development and Validation of a Real-Time Model for the Simulation of the Heat Release Rate, In-Cylinder Pressure and Pollutant Emissions in Diesel Engines

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ABSTRACT

A real-time mean-value engine model for the simulation of the HRR (heat release rate), in-cylinder pressure, brake torque and pollutant emissions, including NO_x and soot, has been developed, calibrated and assessed at both steady-state and transient conditions for a Euro 6 1.6L GM diesel engine.

The chemical energy release has been simulated using an improved version of a previously developed model that is based on the accumulated fuel mass approach. The in-cylinder pressure has been evaluated on the basis of the inversion of a single-zone model, using the net energy release as input. The latter quantity was derived starting from the simulated chemical energy release, and evaluating the heat transfer of the charge with the walls. NO_x and soot emissions were simulated on the basis of semi-empirical correlations that take into account the in-cylinder thermodynamic properties, the chemical energy release and the main engine parameters.

The model is also referred to as “steady-state” as the main thermodynamic properties in the intake and exhaust manifolds, as well as the inducted mass flow rate, have been evaluated by means of correlations that were calibrated under steady-state conditions. This has allowed a very low computational time to be obtained, so that the approach is suitable for implementation in an ECU for real-time applications.

The model has been calibrated over several experimental tests carried out at a dynamic test bench at GMPT-E (General Motors powertrain-Europe). The tests include a complete engine map as well as several full-factorial variation lists of the main engine parameters, which have been conducted for seven representative operating conditions of the NEDC (New European Driving Cycle).

The validation has been carried out under transient conditions over NEDC and WLTP missions, which were simulated at the dynamic test bench for a C-class vehicle.

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INTRODUCTION

The increasing computational capabilities of modern ECUs (Engine Control Units) in diesel engines are offering the opportunity of implementing more and more complex model-based algorithms in order to control the combustion and pollutant formation processes in real time.

The development of real-time models that focus on these aspects is therefore of great interest for car manufacturers. However, these fast and reliable simulation tools should also be able to realize an offline optimization of the main engine parameters, in order to reduce the experimental effort and the related costs.

In general, the simulation of the fluid-dynamics in the pipes, manifolds and in the combustion chamber, as well as of the combustion and emission formation processes, can be carried out with different degrees of detail, using multidimensional, one-dimensional or zero-dimensional approaches.

3D-CFD (Computer Fluid-Dynamics) calculation methods a priori have the potential of reproducing the physical and chemical processes that take place in the engine pipes, as well as in the chamber during the injection-combustion process, but they require a considerable computational time and suffer from the drawback of still being conditioned by a lack of precise knowledge on the physics of some processes. These methods therefore usually focus on the simulation of specific engine sub-systems. For example, 3D CFD simulations have been applied to identify the optimal geometry of the intake-

manifold in [1], or to optimize the engine water cooling jacket in [2], where the CFD code has been used in conjunction with an FEM (Finite Element Method) tool for the structural analysis. However, the main application fields of 3D-CFD simulations in diesel engines are mainly focused on the in-cylinder processes, including spray formation [3], mixture preparation [4], as well as combustion and pollutant formation [5,6,7,8,9].

The complete simulation of the pipes and intake/exhaust manifolds of the engine system is usually carried out with 1D-CFD models coupled with a zero-dimensional modeling of the cylinders, injectors, valves, compressors and turbines. The 1D approach is based on the application of the conservation equations for the unsteady compressible flows in the pipes, and it allows the wave propagation phenomena, as well as the inertial effects, to be captured in steady-state and transient conditions. Several examples of studies based on this approach can be found in the literature [10,11,12,13]. However, this approach cannot be applied for real-time applications, as it requires a computational time of the order of several minutes to simulate a single operating condition on a modern PC. The modeling of the intake and exhaust manifolds could be simplified using lumped parameter approaches, such as those presented in [14], in which the manifolds are treated as single control volumes to which mass and energy conservation laws are applied. This approach requires a much lower computational effort than the 1D approach. However, differential equations for the control volumes still have to be solved.

Given this background, a mean-value zero-dimensional engine model for real-time applications has been developed in this study.

The model includes the simulation of the HRR (heat release rate), of the in-cylinder pressure and of the related combustion parameters, including peak firing pressure (PFP), indicated mean effective pressure (IMEP), friction mean effective pressure (FMEP), brake mean effective pressure (BMEP), as well as of the NO_x and soot emissions.

Several approaches have been proposed in the literature to simulate the HRR. For example, the shape of the HRR can be predicted by means of mathematical functions, including those of Wiebe [15,16,17]. Although widely used, this is a purely mathematical approach that lacks physical consistency. Another proposed methodology to simulate HRR in diesel engines is based on the apparent combustion time [18-19]. This is a phenomenological approach that is capable of taking into account the effects of the influence of the in-cylinder air density, oxygen concentration, nozzle diameter and injection rate. Finally, one of the most widely adopted methods to simulate heat release is the accumulated fuel mass approach [20,21,22,23,24]. This method is 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 instant. 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 method has the great advantage of being able to directly relate the injection rate to the combustion rate, and is therefore physically consistent.

A predictive real-time combustion model, based on the accumulated fuel mass approach, has already been presented by the authors in [24]. In that case, the HRR model was coupled with a single zone approach to simulate the in-cylinder pressure, but it required the measurement of the intake/exhaust manifold thermodynamic quantities, as well as of the inducted mass flow rate, and did not include a friction model to predict BMEP.

The engine model proposed in this study starts from the approach proposed in [24] and includes the following novelties:

- An enhanced version of the HRR model has been used. This model has recently been developed and proposed by the authors in [25]; in the present study, the improvements in the prediction of the in-cylinder pressure and pollutant emissions have been evaluated using the enhanced heat release model instead of the previous model presented in [24].
- Correlations calibrated at steady-state engine operating conditions have been introduced in order to evaluate the intake and exhaust manifold thermodynamic conditions, as well as the inducted charge mass.
- A correlation has been identified to evaluate the EGR (Exhaust Gas Recirculation), starting from the EGR valve and throttle valve opening signals, as well as from the thermodynamic quantities in the intake/exhaust manifolds; this has allowed the intake oxygen concentration to be evaluated.
- A friction model, which is based on the Chenn-Flynn approach, has been implemented in order to predict FMEP; this has allowed BMEP to be evaluated on the basis of the IMEP.
- Semi-empirical models, previously developed in [26,27], have been assessed and tuned in order to predict NO_x and soot emissions, which are the most critical pollutant species in diesel engines.

The model was calibrated over several experimental tests carried out at a dynamic test bench at GMPT-E (General Motors powertrain-Europe). The tests include a complete engine map as well as several full-factorial variation lists of the main engine parameters, which were conducted for seven representative operating conditions of the NEDC (New European Driving Cycle).

After the tuning, the model performance was evaluated under steady-state and transient conditions. In particular, the main combustion parameters, i.e., MFB50 (crank angle at which 50% of the fuel mass has burnt), PFP, IMEP, as well as the NO_x emissions were compared with the experimental trends over NEDC and WLTP missions. The results obtained with the developed engine model were compared with those obtained with the same combustion and NO_x formation models, but using the measured intake/exhaust variables during the transients instead of the values obtained with the steady-state correlations. The aim of this comparison was to verify whether the assumption of using steady-state correlations to estimate the intake/exhaust variables could be accepted when simulating moderately transient conditions, such as the NEDC/WLTP cycles.

Finally, a sensitivity analysis was carried out on the calculation step and on the number of experimental tests required for model calibration.

The developed model can be useful for several applications, such as offline calibration of engine parameters, engine simulation in Hardware-in-the-Loop (HiL) or Model-in-the-Loop (MiL) systems, as well as development of model-based feed-forward control algorithms.

ENGINE SETUP AND EXPERIMENTAL ACTIVITY

The experimental tests for the calibration and validation of the models were conducted on a 1.6L Euro 6 diesel engine. The main engine technical specifications are summarized in [Tab. 1](#).

Table 1. Main engine specifications.

Engine type	Euro 6 diesel engine
Displacement	1598 cm ³
Bore x stroke	79.7 mm x 80.1 mm
Rod length	135 mm
Compression ratio	16.0
Valves per cylinder	4
Turbocharger	VGT type
Fuel injection system	Common Rail
Specific power and torque	71 kW/l – 205 Nm/l

The engine is equipped with a short-route cooled EGR system, in which the EGR valve is located upstream from the cooler. A throttle valve is installed upstream from the intake manifold and EGR junction, in order to allow high EGR rates to be obtained when the pressure drop between the exhaust and intake manifolds is not sufficient. Moreover, the EGR circuit is equipped with an EGR cooler bypass, in order to prevent EGR gases from flowing across the cooler under certain driving conditions, e.g., during cold start phases.

The test engine was instrumented with piezoresistive pressure transducers and thermocouples to measure the pressure and temperature at different locations, such as upstream and downstream from the compressor, turbine and intercooler, and in the intake manifold. Thermocouples were also used to measure the temperatures in each exhaust runner. Piezoelectric transducers were installed to measure the pressure time-histories in the combustion chamber of the cylinders.

The experimental tests were carried out on a dynamic test bench at GMPT-E, in the frame of a research project between the Politecnico di Torino and GMPT-E, pertaining to the assessment of control-oriented heat release predictive models [25]. To this aim, several tests were conducted, including:

- Full-Factorial variation tests of p_{int} (intake manifold pressure), SOI_{main} (start of injection of the main pulse), O_2 (intake oxygen concentration) and p_f (injection pressure) at several representative key-points of the NEDC cycle.

- A full engine map with baseline operating parameters.

Table 2 summarizes the experimental test matrix used for the calibration of the models, and reports the minimum and maximum values of the main engine parameters for the different operating points.

Table 2. Experimental test matrix.

Operating point	Tests		P _r	P _{int}	O ₂	λ	X _r	q _{pil2}	q _{pil1}	q _{f,inj}	SOI pil2	SOI pil1	SOI main
		bar		%	-			mm ³	mm ³	mm ³	CAD	CAD	CAD
1500x2	100	max	500	1.26	20.6	4.78	50.1	1.3	1.6	9.18	332.7	342.6	353.2
		min	250	0.95	14.7	1.65	0.98	1.3	1.5	7.74	339.4	348.5	358.2
1500x5	100	max	680	1.45	20.7	2.81	36.5	1.3	1.6	17.8	331.5	340.7	352.9
		min	380	0.95	14.4	1.14	0.53	1.3	1.6	15.4	339.5	348.5	358.2
1500x8	100	max	800	1.68	20.8	2.18	17.9	1.3	1.6	28.1	331.4	340.0	355.5
		min	400	1.10	18.1	1.11	0.32	1.3	1.6	23.4	337.9	346.4	360.0
2000x2	100	max	580	1.58	20.73	5.13	48.2	1.3	1.5	10.1	327.4	339.8	354.7
		min	250	0.85	15.3	1.63	0.71	1.3	1.5	7.90	332.8	343.8	356.6
2000x5	100	max	680	1.87	20.7	3.48	30.5	1.3	1.5	17.1	326.8	338.3	354.8
		min	380	0.97	16.4	1.3	0.47	1.3	1.5	15.0	329.8	340.7	355.4
2000x8	100	max	1100	2.09	20.8	2.64	21.0	1.3	1.5	26.6	325.1	335.9	356.1
		min	700	1.34	17.2	1.17	0.45	1.3	1.5	23.4	326.8	337.2	356.4
2000x12	100	max	1040	2.46	20.8	2.18	0.47	1.3	1.8	36.1	326.2	338.7	351.5
		min	640	1.84	20.8	1.75	0.33	1.3	1.8	33.5	327.3	339.3	351.5
Engine map	285	max	1936	2.73	20.8	5.77	50.6	1.3	3.1	74.0	311.5	306.3	337.5
		min	305	1.01	16.8	1.08	0.14	0	1.3	5.47	360.0	353.4	360.2

The key-points are identified by the speed and BMEP (rpm x bar) values. X_r is the EGR rate, defined as the ratio between the inducted EGR mass flow rate and the total inducted mass flow rate; λ is the relative air-to-fuel ratio, q_{pil2} and q_{pil1} indicate the injected fuel quantity of the pilot2 and pilot1 injections, respectively, while $q_{f,inj}$ denotes the total injected fuel quantity. All injected quantities are expressed in mm³/cyc.

Figure 1 reports the engine map tests, in terms of normalized BMEP and engine speed.

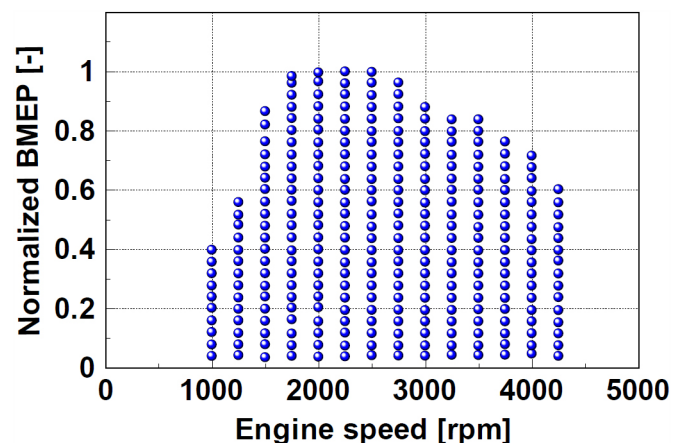


Figure 1. Engine map tests used for the calibration of the models.

REAL-TIME ENGINE MODEL

The real-time model proposed in this study includes the simulation of:

1. Chemical energy release: the approach is based on an enhanced version (see [25]) of the model previously presented by the authors, which was based on the accumulated fuel mass approach [24]. The input data of the model are the injection

parameters, as well as the main thermodynamic conditions in the intake manifold and the engine operating parameters.

2. In-cylinder pressure: the approach is based on the inversion of a single-zone heat release model which requires the net energy release as input; the latter is derived starting from the predicted chemical energy release and estimating the heat transfer between the charge and the walls. Polytopic evolutions are assumed during the compression and expansion phases. The simulation of the in-cylinder pressure allows several parameters, such as PFP and IMEP, to be evaluated.
3. Friction losses: the Chenn-Flynn approach has been used to predict FMEP on the basis of the engine speed and PFP; the simulation of FMEP allows BMEP to be evaluated starting from IMEP.
4. EGR rate: the EGR rate has been modeled by means of an empirical correlation that estimates the in-cylinder trapped EGR volume on the basis of the opening position of the EGR valve, of the thermodynamic conditions upstream and downstream from the valve, of the cooler by-pass activation flag and of the throttle valve position. The simulation of the EGR rate is required to evaluate the intake oxygen concentration, which is then used in several correlations.
5. NOx and soot emission levels: the semi-empirical correlations previously developed by the authors for a 2.0L Euro 5 diesel engine and reported in [26, 27] have been tuned and validated for the 1.6L Euro 6 engine considered in the present study.
6. Intake/exhaust manifold thermodynamic conditions and inducted mass flow rate: a steady-state approach has been applied in the present study, and neither the turbocharger nor the intake and exhaust manifolds have been simulated. The pressure in the intake manifold (p_{int}) has been set equal to the set-point value that was derived from the baseline engine map, as a function of the engine load and speed; the intake manifold temperature has been evaluated by means of an enthalpy balance equation at the EGR mixing node; the exhaust manifold temperature has been evaluated by means of a correlation that takes into account the in-cylinder temperature at exhaust valve opening, as well as the engine load and speed; the exhaust manifold pressure has been evaluated by means of a look-up table that takes into account the pressure in the intake manifold, the engine load and speed. The inducted charge mass has been evaluated on the basis of the volumetric efficiency, which was evaluated on the basis of a look-up table.

The main advantage of this approach consists in the high computational time saving, which makes it suitable for implementation in an ECU for real-time applications.

The main drawbacks of such an approach are related to the fact that the lags due to the delayed response of the turbocharger, as well as the effects related to the filling and emptying processes of the manifolds and the high pressure EGR delays are not taken into account. The proposed method is therefore not expected to provide an accurate simulation of fast transient conditions. However, one of the objectives of the present study is to verify whether this approach is suitable for simulating moderately transient conditions, such as the NEDC/WLTP cycles.

A summary of each sub-model is reported hereafter.

In general, it should be pointed out that the HRR and in-cylinder pressure models were assessed on the basis of the measured in-chamber pressure of one of the four cylinders, which was considered as the representative one. It was in fact verified that the cylinder-to-cylinder dispersion of the main combustion parameters is low.

Estimation of the Chemical Energy Release Q_{ch}

The chemical energy release has been simulated on the basis of an enhanced version [25] of the baseline model presented by the authors in [24], which was based on the accumulated fuel mass approach.

The accumulated fuel mass approach assumes that, at any time instant, the rate of chemical energy released by the fuel is proportional to the energy associated with the in-cylinder accumulated fuel mass. Such an energy can be calculated at time instant t as the difference between the chemical energy of the injected fuel mass and the released chemical energy.

It was verified in [25] that this approach leads to accurate results for the pilot injections, for which the chemical energy release rate has therefore been evaluated as follows:

$$\frac{dQ_{ch,pil,j}}{dt}(t) = K_{pil,j} [Q_{fuel,pil,j}(t - \tau_{pil,j}) - Q_{ch,pil,j}(t)] \quad (1)$$

where $K_{pil,j}$ and $\tau_{pil,j}$ are model calibration quantities related to the combustion rate and to the ignition delay, respectively, and $Q_{fuel,pil,j}$ is the chemical energy associated with the injected fuel mass.

The chemical energy release of the main pulse has instead been simulated by means of a modified formulation that was proposed in [25], and which leads to a higher accuracy especially at medium-high load conditions:

$$\begin{aligned} \frac{dQ_{ch,main}}{dt}(t) = & K_{1,main} [Q_{fuel,main}(t - \tau_{main}) - Q_{ch,main}(t)] \\ & + K_{2,main} \frac{dQ_{fuel,main}(t - \tau_{main})}{dt} \end{aligned} \quad (2)$$

The formulation proposed in Eq. (2) needs an additional calibration parameter with respect to the baseline approach of Eq. (1) (i.e., $K_{2,main}$).

From a physical point of view, the term proportional to the injection rate takes into account the effect, on the heat release, of the turbulence induced by the fuel injection, which is not negligible in diesel sprays.

For each injection pulse j , the chemical energy Q_{fuel} associated to the injected fuel quantity is defined as follows:

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

$$Q_{fuel,j}(t) = \int_{t_{SOI,j}}^{t_{EOI,j}} \dot{m}_{f,inj}(t) H_L dt \quad t > t_{EOI,j} \quad (4)$$

where t_{SOI} is the start of the injection time, t_{EOI} the end of the injection time, H_L the lower heating value of the fuel and $\dot{m}_{f,inj}$ the fuel mass injection rate.

The total chemical energy release is given by the sum of the contributions of all the injection pulses:

$$Q_{ch} = \sum_{j=1}^n Q_{ch,j} \quad (5)$$

It can be noted, from Eqs. (1-2), that the baseline approach described in [24] has been used for the pilot shots, as it had been verified that it provides satisfactory results [25].

Instead, an enhanced formulation, which includes an additional term proportional to the injection rate, has been developed for the main pulse. It had been verified in [25] that the addition of this term leads to an increase in the accuracy of the predicted heat release trend and of MFB50, especially at high-load operating conditions, where the combustion of the main pulse is of the premixed/mixing-controlled type.

The term proportional to the injection rate takes into account the effect of the injection-induced turbulence on the heat release, which is a non-negligible term in diesel sprays which influences the mixing controlled combustion phase that usually occurs during the oxidation of the main pulse.

The model was assessed for the steady-state conditions reported in Tab. 2. In particular, the optimal τ and K parameters were identified by comparing the predicted and experimentally-derived heat release profiles, and minimizing the sum of errors and the MFB50 prediction error by means of a genetic algorithm (see [25]). The adopted correlation variables for ignition delay were chosen in accordance with the study proposed in [28]. In particular, at the beginning, all the engine variables were included in the correlations, and a sensitivity analysis was carried out in order to exclude the least influential ones, thus a stepwise regression was adopted. This approach was also adopted to identify the engine variables that had to be included in all the correlations reported hereafter.

The following correlations were identified in [25] as a function of the in-chamber thermodynamic quantities at SOI/SOC (start of injection/start of combustion) and other engine variables:

$$K_{pil} \left[\frac{1}{s} \right] = 1.109 \exp \left(\frac{-10254}{T_{SOC,pil}} \right) O_2^{3.80} n^{0.724} q_{pil,tot}^{-0.866} q_{f,inj}^{1.207} \quad (6)$$

$$K_{1,main} \left[\frac{1}{s} \right] = 19.91 p_f^{0.37} \rho_{SOC,main}^{0.575} n^{0.225} q_{f,inj}^{-0.69} \quad (7)$$

$$K_{2,main} \left[\frac{s}{deg} \right] = 0.296 p_f^{0.84} \rho_{SOC,main}^{0.59} n^{-0.507} K_{1,main}^{-2.12} \quad (8)$$

$$\tau_{pil} [deg] = 257 \rho_{SOI,pil}^{-0.757} \exp \left(\frac{1559}{T_{SOI,pil}} \right) O_2^{-1.17} \quad (9)$$

$$\tau_{main} [deg] = 508 p_f^{-0.648} \rho_{SOI,main}^{-0.95} O_2^{-1.265} n^{0.827} q_{f,inj}^{-0.323} \quad (10)$$

In equations (6, 7, 8, 9, 10), ρ_{SOI} , T_{SOI} , ρ_{SOC} and T_{SOC} indicate the in-chamber densities and temperatures evaluated at the start of injection or combustion, respectively, and are expressed in kg/m³ and K. The injection pressure p_f is expressed in bar, the engine speed n in rpm, the total injected fuel quantity $q_{f,inj}$ (used as a load parameter) in mm³/cyc/cyl, the total injected fuel quantity of the pilot shots $q_{pil,tot}$ in mm³/cyc/cyl and finally the intake oxygen concentration O_2 in %. For the combustion rate parameters, the in-chamber thermodynamic conditions evaluated at SOC were selected as being more representative than those evaluated at SOI. It should be noted that it is necessary to convert the ignition delay values in 's' before their utilization in Eqs. (1-2).

Estimation of the Net Energy Release Q_{net}

It is necessary to derive the net energy release, starting from the chemical release, in order to simulate the in-cylinder pressure during the combustion phase, taking into account the heat transfer and fuel evaporation heat effects [24]. The net heat release Q_{net} is derived from the chemical release according to the following formulation [24]:

$$Q_{net,ht} \cong Q_{ch} \frac{m_{f,inj} H_L - Q_{ht,glob}}{m_{f,inj} H_L} \quad (11)$$

$$Q_{net} \cong Q_{net,ht} - Q_{f,evap} \quad (12)$$

where $Q_{net,ht}$ is the result of the scaling of the chemical energy release according to the global heat exchanged by the charge with the walls, $Q_{f,evap}$ and $Q_{ht,glob}$ indicate the fuel evaporation heat from SOI to SOC (J) and the heat globally exchanged by the charge with the walls over the combustion cycle (J), and $m_{f,inj}$ is the total injected fuel mass per cycle/cylinder. The experimental values of the $Q_{f,evap}$ and $Q_{ht,glob}$ parameters were identified on the basis of the measured in-cylinder pressure traces for all the tests shown in Tab. 3 (the procedure is

reported in [24], and the following correlations were identified for the two parameters as a function of the engine speed, load and intake manifold temperature:

$$Q_{f, \text{evap}}[J] = 2.22E-11 T_{\text{int}}^{3.76} n^{0.584} q_{f, \text{inj}}^{0.270} \quad (13)$$

$$Q_{\text{ht, glob}}[J] = 117 n^{-0.642} q_{f, \text{inj}}^{1.50} \quad (14)$$

Estimation of the in-cylinder pressure

The in-chamber pressure was evaluated during the combustion interval using a single-zone model [15]:

$$dp = \left(\frac{\gamma - 1}{V} \right) \left(dQ_{\text{net}} - \frac{\gamma}{\gamma - 1} p dV \right) \quad (15)$$

where the isentropic coefficient $\gamma = c_p / c_v$ was set constant and equal to 1.37. An explicit integration method was used to solve Eq. (15), as it was verified that it leads to a good accuracy and is stable for crank angle integration steps ranging from 0.1 to 2 deg. Therefore, the adoption of implicit integration methods is not necessary.

Polytropic evolutions were assumed to calculate the in-cylinder pressure during the compression and expansion phases:

$$pV^m = \text{const} \quad (16)$$

$$pV^{m'} = \text{const} \quad (17)$$

The starting condition for the evaluation of the compression phase, i.e. the in-chamber pressure at IVC (Intake Valve Closure), was correlated to the pressure in the intake manifold p_{int} , using a correction factor Δp_{int} , as follows:

$$p_{\text{IVC}} = p_{\text{int}} + \Delta p_{\text{int}} \quad (18)$$

The experimental values of the m , m' and Δp_{int} parameters were identified on the basis of the measured in-cylinder pressure traces for all the tests shown in Tab. 2 (reference can be made to [24] for the complete procedure), and the following correlations were identified for the three parameters, which are functions of the intake manifold thermodynamic conditions and of the engine load and speed:

$$m = 3.02 T_{\text{int}}^{-0.173} n^{0.0167} q_{f, \text{inj}}^{0.0148} \quad (19)$$

$$m' = 2.78 T_{\text{int}}^{-0.0919} n^{-0.0176} q_{f, \text{inj}}^{-0.0209} \quad (20)$$

$$\Delta p_{\text{int}} [\text{bar}] = 0.163 p_{\text{int}}^{1.06} n^{-0.000825} q_{f, \text{inj}}^{-0.0429} \quad (21)$$

The in-cylinder pressure values during the intake and exhaust phases were considered as constants and were set equal to the pressure in the intake and exhaust manifolds, respectively [24]. It has in fact been verified that this assumption leads to a very small error in the prediction of IMEP.

The simulation of the in-cylinder pressure traces allows several combustion parameters, such as PFP and IMEP, to be evaluated.

Estimation of FMEP

The Chen-Flynn approach [29] was adopted to estimate FMEP. This correlation takes into account the effect of the engine speed and of the peak firing pressure on the friction losses.

The experimental values of FMEP were evaluated as the difference between the experimental values of the net IMEP (obtained by means of the integration of the experimental pressure trace over the entire cycle) and the measured values of BMEP, as follows:

$$FMEP = IMEP - BMEP \quad (22)$$

As previously stated, the values of IMEP of one of the four cylinders were taken as being representative of all the cylinders, due to the low cylinder-to-cylinder dispersion.

The following correlation was identified to evaluate FMEP:

$$FMEP[\text{bar}] = -1.38E-04 n + 6.67E-08 n^2 + 0.0126 PFP + 1.04 \quad (23)$$

where the engine speed is expressed in rpm and PFP is expressed in bars.

The squared correlation coefficient R^2 between the predicted and experimental values of FMEP is the order of 0.89.

Estimation of the EGR Rate, of the Trapped Mass and of the Intake O_2 Concentration

An empirical correlation was identified to estimate the in-cylinder trapped volume of the EGR gas (taking the EGR gas density upstream from the valve, i.e., ρ_{exh} , as the reference one), as a function of the pressure drop across the EGR valve ($\Delta p = p_{\text{exh}} - p_{\text{int}}$), of the EGR flow density at the EGR valve inlet and outlet ($\rho_{\text{int}}, \rho_{\text{exh}}$), of the engine speed (n), of the EGR valve opening signal (i.e., u_{EGR}), of the cooler by-pass activation flag and of the throttle valve position (i.e., u_{th}).

The proposed correlations are:

$$V_{\text{EGR}} \left[\frac{m^3}{\text{cyc cyl}} \right] = 6.65E-12 \Delta p^{0.199} \rho_{\text{int}}^{-1.20} \cdot \rho_{\text{exh}}^{-0.889} u_{\text{EGR}}^{0.0821} u_{\text{th}}^{-0.197} n^{2.33} (\text{bypass}_{\text{EGRCooler}} = \text{OFF}) \quad (24)$$