

HRR and MFB50 Estimation in a Euro 6 Diesel Engine by Means of Control-Oriented Predictive Models

Roberto Finesso, Ezio Spessa, and Yixin Yang
Politecnico di Torino

Vincenzo Alfieri and Giuseppe Conte
General Motors Powertrain Europe Srl

ABSTRACT

The paper has the aim of assessing and applying control-oriented models capable of predicting HRR (Heat Release Rate) and MFB50 in DI diesel engines. To accomplish this, an existing combustion model, previously developed by the authors and based on the accumulated fuel mass approach, has been modified to enhance its physical background, and then calibrated and validated on a GM 1.6 L Euro 6 DI diesel engine.

It has been verified that the accumulated fuel mass approach is capable of accurately simulating medium-low load operating conditions characterized by a dominant premixed combustion phase, while it resulted to be less accurate at higher loads. In the latter case, the prediction of the heat release has been enhanced by including an additional term, proportional to the fuel injection rate, in the model.

The already existing and the enhanced combustion models have been calibrated on the basis of experimental tests carried out on a dynamic test bench at GMPT-E. A comparison has been made between the models, in terms of accuracy in the prediction of HRR and MFB50, as well as of the required computational time and calibration effort, at several steady-state operating conditions as well as over NEDC and WLTP cycles.

The values of MFB50 predicted by means of the two approaches, for the same steady-state tests and driving cycles, have been compared with those obtained from a low-throughput invertible MFB50 predictive model that has recently been developed by the authors, which is characterized by an extremely low computational time.

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INTRODUCTION

Numerical simulation has become a powerful way of realizing the optimum configuration for both advanced conventional combustion systems [1] and LTC/PCCI (Low-Temperature Combustion/Premixed Charge Compression Ignition) [2] diesel engines, as it allows a significant reduction to be obtained in the experimental calibration effort and in the related costs. In particular, the simulation of the in-cylinder combustion process, including the HRR (Heat Release Rate), is of fundamental importance in order to predict engine performance [3]. Moreover, combustion instabilities have become a critical issue in modern low-temperature diesel engines, particularly under transient operation [4]. A breakthrough concept which can be adopted to avoid such instabilities is the development of predictive low-throughput HRR models to be implemented in the engine ECU (engine control unit) for feed-forward combustion control. 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. In general, the evaluation of this parameter

requires the implementation of a predictive HRR model, although direct approaches have also recently been developed to evaluate MFB50 [5].

Therefore, interest in predictive HRR models is becoming more and more widespread in the car manufacturing industry to both speed-up the search for the optimum engine configuration and to develop suitable model-based control approaches.

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 [6]. 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 quasi-steady diesel diffusion combustion was proposed in [9]. This analysis allowed the influence of in-cylinder air density, oxygen concentration, nozzle diameter and injection rate to be identified and quantified on HRR. This 0D phenomenological approach was then generalized in [10] 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.

The accumulated fuel mass approach is one of the most widely used methods to derive HRR in diesel engines [11, 12, 13, 14, 15, 16, 17, 18, 19, 20]. This approach is based on the assumption that the rate of chemical energy released by the fuel is proportional to the energy associated with the fuel quantity made available for combustion at the considered time 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.

Considering this background, a collaborative research project has been carried out between the Energy Department of the Politecnico di Torino and GMPT-E (General Motors Powertrain Europe), with the aim of developing and assessing control-oriented combustion models to predict HRR and MFB50 in diesel engines.

An existing combustion model, previously developed by the authors and based on the accumulated fuel mass approach [15], has been assessed and modified within this project to enhance its physical background. It has in fact been verified that the accumulated fuel mass approach is suitable for the accurate simulation of medium-low load operating conditions, while it has resulted to be less accurate for higher loads.

The performance of the baseline and enhanced HRR models has been evaluated and discussed under both steady-state and transient conditions, over NEDC (New European Driving Cycle) and WLTP (Worldwide Harmonized Light-duty Testing Procedure) missions. The NEDC is currently the reference homologation cycle for passenger cars in Europe, but it will be replaced by WLTP in January 2017.

The MFB50 values predicted by means of the two approaches, for the same steady-state tests and driving cycles, have been compared with those obtained by means of a low-throughput invertible MFB50 predictive model that has recently been developed by the authors in [5], which is characterized by an extremely low computational time and is therefore suitable for implementation in the engine ECU.

ENGINE SETUP AND EXPERIMENTAL ACTIVITY

The experimental tests for the calibration and validation of the models have been run on a 1.6L Euro 6 diesel engine. The main technical specifications of the engine 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 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. To this end, several tests were conducted, including:

- Full-Factorial variation tests of pint (intake manifold pressure), SOI_{main} (start of injection of the main pulse), O₂ (intake oxygen concentration) and p_f (injection pressure) at several representative key-points of the NEDC cycle.
- A full engine map with the baseline engine operating parameters.

Table 2. Experimental test matrix.

Operating point	Tests		P _r bar	P _{int} %	O ₂ %	λ	X _r -	q _{pi2} mm ³	q _{pi1} mm ³	q _{r,inj} mm ³	SOI pil2 CAD	SOI pil1 CAD	SOI main 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

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. The key-points are identified by the speed and BMEP (rpm x

bar) values. X_r is the EGR rate, which is 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, and $q_{f,inj}$ denotes the total injected fuel quantity.

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

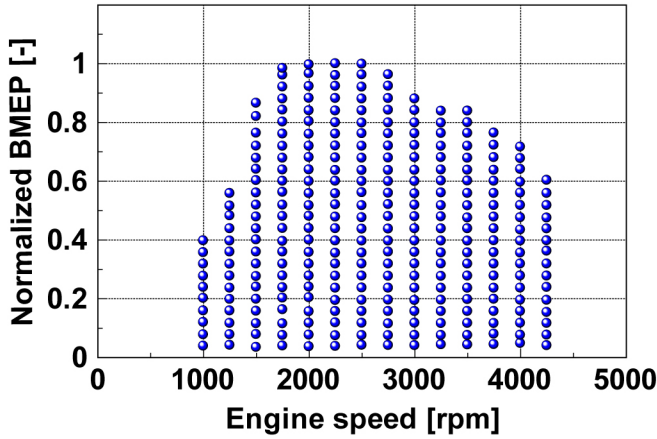


Figure 1. Engine map tests used for the calibration of the models.

DESCRIPTION OF THE MODELS

A synthetic description of the baseline predictive heat release model developed by the authors in [15], of the enhanced heat release model and of the low-throughput invertible MFB50 model developed in [5] is reported in this section.

Baseline Heat Release Model

The baseline HR (heat release) model is based on the accumulated fuel mass approach, which 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.

The released chemical energy $Q_{ch,j}$ at time t , for each pulse j ($1 \leq j \leq n$) of the injection event, is therefore evaluated as follows:

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

where K_j and τ_j are model calibration quantities, related to the combustion rate and to the ignition delay, respectively, and $Q_{fuel,j}$ is the chemical energy associated with the injected fuel mass:

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

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

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

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

The approach requires the injection rate as input. A simple triangular injection rate, which is calculated on the basis of the injection parameters (injection timings/quantities and injection pressure), has been used [15], taking into account the nozzle opening delay (NOD) and nozzle closure delays (NCD) of the injector.

Model calibration requires the identification of the optimal values of K_j and τ_j in Eq. (1), in order to obtain an accurate matching between the predicted and experimentally-derived chemical energy release curves.

It has been verified that the accumulated fuel mass approach is capable of accurately predicting the heat release trend at medium-low load operating conditions, but is less accurate for high load operating points. This can be observed in Fig. 2, which reports the experimentally-derived and the predicted trends of the chemical energy release Q_{ch} , for a medium-low load operating point (2000×5, Fig. 2a) and for a high load operating point (2750×14, Fig. 2b). The operating points were taken from the engine map tests. The injection rate is also reported.

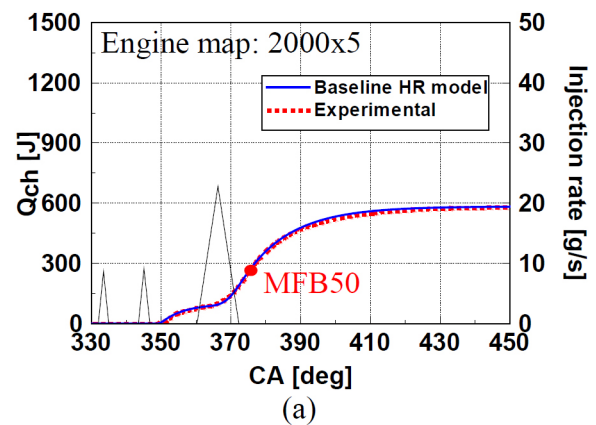


Figure 2. Experimental and predicted trends of Q_{ch} , obtained with the baseline HR model, for a medium-low load operating point (a) and for a high load operating point (b) taken from the engine map.

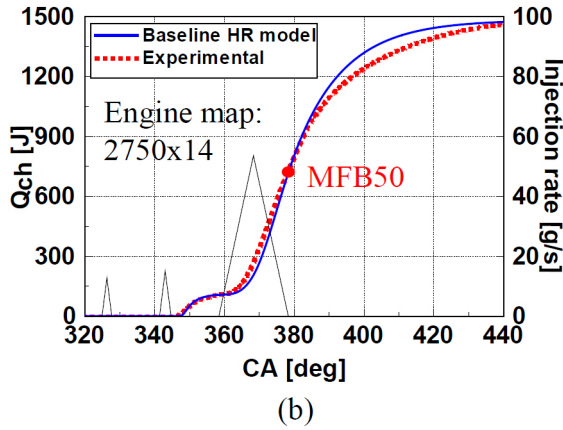


Figure 2. (cont.) Experimental and predicted trends of Q_{ch} , obtained with the baseline HR model, for a medium-low load operating point (a) and for a high load operating point (b) taken from the engine map.

It can be noted that the prediction of the heat release of the pilot pulse is accurate in both cases, but the prediction of the main pulse is less accurate for the high load operating point. From a physical point of view, this can be ascribed to the fact that the combustion of the main pulse at medium-low load operating points is mainly of the premixed type, as the ignition delay is long [21], as can also be observed in the figure, therefore a single calibration coefficient, related to the combustion rate (i.e., K_{main} in Eq. (1)), is sufficient to accurately predict the heat release shape. Instead, as the load increases, the combustion of the main pulse results to be constituted by an initial premixed phase with a subsequent mixing-controlled phase, and therefore a single calibration coefficient related to the combustion rate is not suitable to accurately describe the heat release trend of the main pulse. The combustion of the pilot pulses is mainly of the premixed type [22], therefore the accumulated fuel mass approach allows a good prediction of the Q_{ch} shape to be obtained.

However, it can be noted, in Fig. 2, that the prediction of MFB50 is good for both the medium-low load and high load cases.

Enhanced Heat Release Model

The baseline heat release model has been improved in this study, considering a separate formulation for the pilot and main pulses.

In particular, and in accordance with the discussion reported in the previous section, the baseline approach featuring the accumulated fuel mass has been used for each pilot pulse pil,j :

$$\frac{dQ_{ch,pil,j}}{dt}(t) = K_{pil,j} [Q_{fuel,pil,j}(t - \tau_{pil,j}) - Q_{ch,pil,j}(t - \tau_{pil,j})] \quad (5)$$

Instead, the chemical energy release of the main pulse has been simulated by means of an improved formulation, defined by the following equation, which includes an additional term proportional to the chemical energy associated with the fuel injection rate:

$$\begin{aligned} \frac{dQ_{ch,main}}{dt}(t) = & K_{1,main} [Q_{fuel,main}(t - \tau_{main}) - Q_{ch,main}(t - \tau_{main})] \\ & + K_{2,main} \frac{dQ_{fuel,main}(t - \tau_{main})}{dt} \end{aligned} \quad (6)$$

The formulation proposed in Eq. (6) needs an additional calibration parameter compared to the baseline approach in Eq. (1) (i.e., $K_{2,main}$).

The total chemical energy release is given again by the sum of the contributions of the pilot and main pulses (see Eq. (4)).

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.

Invertible MFB50 Model

A summary of the feed-forward invertible MFB50 model, developed by the authors in [5] for a Euro 5 2.0L diesel engine, and which in this study was assessed for the Euro 6 1.6L engine, is reported in this section.

This approach is capable of estimating MFB50, starting from the injection timing, without the need of simulating the heat release profile.

The method is illustrated in Fig. 3 [5].

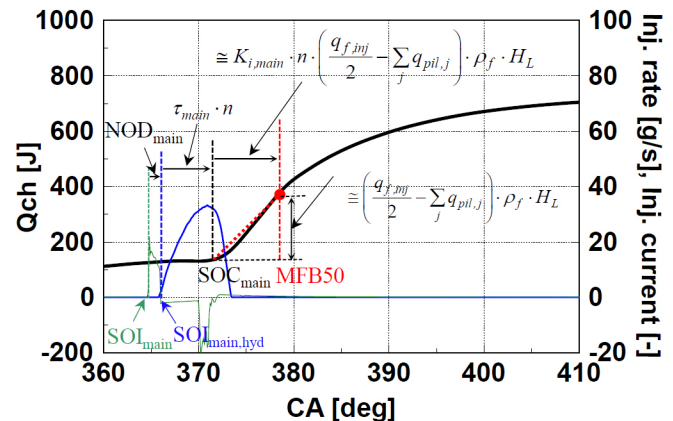


Figure 3. Injection current, injection rate and Q_{ch} for a test case at the 2000x5 key-point that features a pilot-pilot-main injection strategy ([5]).

Figure 3 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 ($\tau_{main} \cdot 6n/1000$, where τ_{main} is expressed in ms and n in rpm); the product between the inverse of the average heat release rate of the main pulse in the (SOC_{main} , MFB50) interval, i.e.:

$$\left(\frac{dt}{dQ_{ch}} \right)_{main}^{def} = K_{i,main} = \frac{t_{MFB50} - t_{SOC_{main}}}{Q_{ch}(MFB50) - Q_{ch}(SOC_{main})} \quad (7)$$

(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 into crank angle degrees. For the sake of clarity, K_i is defined 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 for the pilot and main injections is 1, MFB50 can be modeled as follows:

$$MFB50 = SOI_{main} + NOD_{main} + \tau_{main} \frac{6n}{1000} + K_{i,main} \frac{6n}{1000} \left(\frac{q_{f,inj}}{2} - \sum_{j=1}^J q_{pil,j} \right) \rho_f LHV \quad (8)$$

The different contributions of Eq. (8) are shown in Fig. 3.

Equation (8) 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. However, Equation (8) can theoretically be extended to the cases in which $\sum_{j=1}^J q_{pil,j} > \frac{q_{f,inj}}{2}$ (the reader can refer to [5]).

The calibration coefficients of this model are the ignition delay of the main pulse τ_{main} and the parameter $K_{i,main}$.

This approach is not only capable of predicting MFB50 on the basis of the injection timing, but also of identifying the optimal injection timing to achieve a target of MFB50 [5], and is therefore suitable for inversion.

The invertible MFB50 model is simpler than the heat release models. However, the model does not calculate the instantaneous heat release rate. Owing to its simplicity, this model requires a very low computational time and is particularly suitable for implementation in an ECU for feed-forward combustion control applications.

CALIBRATION OF THE MODELS

Heat Release Models

The calibration of the heat release models requires the optimal values of K_j and τ_j (see Eqs. (1, 5-6)) to be identified for each experimental test, in order to obtain an accurate matching between the predicted and the experimentally-derived chemical heat release rate (see Fig. 2). The latter is estimated on the basis of the measured in-cylinder pressure trace. First, the experimental net energy release is evaluated on the basis of a single zone approach [6]:

$$\frac{dQ_{net}}{dt} = \frac{\gamma}{\gamma-1} p \frac{dV}{dt} + \frac{1}{\gamma-1} V \frac{dp}{dt} \quad (9)$$

where $\gamma = c_p/c_v$ is the ratio of the specific heats, p is the in cylinder pressure and V is the cylinder volume. γ is considered constant and set at 1.37.

Subsequently, the start of combustion (SOC) is identified, and the experimental chemical energy release is derived, starting from the experimental net energy release, with the procedure outlined in [15].

A completely automatic procedure has been developed in order to identify the optimal values of the K_j and τ_j coefficients for the baseline and enhanced HR models.

The ignition delays τ_j were calculated on the basis of the evaluation of the SOC of the pilot and main pulses with the procedures illustrated in [21], while the combustion rate coefficients were tuned in order to obtain an accurate matching between the predicted and experimental heat release trends.

Given the importance of correctly predicting MFB50, a weight function that accounted for both MFB50 and the heat release curve prediction error was defined and used as a target function to be minimized during the calibration process. This weight function was defined as follows:

$$WF = a \cdot SSD + b \cdot \Delta MFB50 \quad (10)$$

where SSD is the sum of the squared differences of the experimental and predicted heat release trends over the cycle, while $\Delta MFB50$ is the MFB50 prediction error. The parameters a and b are weighting factors, which are introduced to balance the accuracy in the prediction of MFB50 and the heat release trend during the calibration process.

A genetic algorithm (GA) technique was used to minimize the weight function.

The set of optimal values of the K_j and τ_j parameters was then used to identify physically-consistent correlations that are functions of the main engine variables.

Invertible MFB50 Model

The calibration procedure of the invertible MFB50 model has already been explained in detail in [5]. However, a summary is reported hereafter. First, the start of combustion of the main pulse is derived through the procedure reported in [21], and this allows the experimental values of τ_{main} to be evaluated for each calibration test. The experimental values of $K_{i,main}$ are then estimated on the basis of Eq. (7).

Physically-consistent correlations are then identified for the τ_{main} and $K_{i,main}$ parameters, as a function of the main engine variables.

Identified Correlations

The following correlations have been identified for the ignition delay parameters τ_{pil} and τ_{main} :

