

Offline and Real-Time Optimization of EGR Rate and Injection Timing in Diesel Engines

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ABSTRACT

New methodologies have been developed to optimize EGR rate and injection timing in diesel engines, with the aim of minimizing fuel consumption (FC) and NO_x engine-out emissions.

The approach entails the application of a recently developed control-oriented engine model, which includes the simulation of the heat release rate, of the in-cylinder pressure and brake torque, as well as of the NO_x emission levels. The engine model was coupled with a C-class vehicle model, in order to derive the engine speed and torque demand for several driving cycles, including the NEDC, FTP, AUDC, ARDC and AMDC.

The optimization process was based on the minimization of a target function, which takes into account FC and NO_x emission levels. The selected control variables of the problem are the injection timing of the main pulse and the position of the EGR valve, which have been considered as the most influential engine parameters on both fuel consumption and NO_x emissions. The gear number has also been selected for optimization.

One benchmark tool, which is based on the dynamic programming technique, and one real-time tool, which implements a static optimization method, have been developed for the optimization process. A new mathematical technique has been introduced and applied in order to decrease the computational time of the optimizers to a great extent.

It was verified that the real-time method has the potential of being implemented in the engine control unit (ECU), in order to realize an onboard optimization of the selected engine and vehicle parameters.

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INTRODUCTION

Calibration of diesel engines requires a highly time-consuming and expensive experimental activity, also taking into account the high number of parameters that affect the combustion and pollutant formation processes (e.g., injection parameters, EGR rate, swirl ratio, boost pressure, ...). Moreover, for the same engine, the optimal set of engine variables is strictly dependent on the vehicle specifications, as well as on the driving mission.

Engine calibration is usually carried out focusing on specific key-points, which are representative of the engine operation over the vehicle homologation cycle, and identifying the optimal set of engine variables for each key-point through Design of Experiment (DoE) techniques [1]. The DoE approach [2] starts from the identification of the input variables, of the response variables and on the definition of the experimental plan. A response surface model is then chosen for the system response, which is generally of the second-order

polynomial type [1]. This model is tuned on the basis of the experimental tests, and is then used to identify the optimal set of input variables that minimizes a specific target function, which depends on the response variables. Although widely used, this calibration method is expensive as it requires a high number of acquired tests. Moreover, the polynomial models used to characterize the system response are accurate in the selected input variable range, but lack of accuracy outside this range as they are not physically-consistent. Several example of response-surface optimization can be found in the literature [3, 4].

An alternative approach to that based on experiments is represented by virtual engine optimization. The development of a model of the engine system may in fact offer the opportunity of realizing a preliminary optimization of the main engine parameters, with the aim of reducing the experimental effort. Engine simulation can be carried out with different degrees of detail, depending on the specific application.

For example, 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 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 (see [5, 6, 7, 8, 9, 10, 11, 12, 13]).

The complete simulation of the engine system is usually carried out by coupling 1D-CFD models, for the simulation of the pipes and intake/exhaust manifolds, with zero-dimensional models 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 [14, 15, 16, 17]. However, this approach requires a computational time of the order of several minutes to simulate a single operating condition on a modern PC.

Mean-value engine models [18] are based on the modeling of the intake/exhaust engine variables by means of steady-state and mean-value empirical corrections. These approaches offer the opportunity of further decreasing the computational time, while guaranteeing at the same time a good predictive capability at steady-state and mildly transient engine operating conditions.

Finally, artificial intelligence systems, such as artificial neural networks (ANN) and fuzzy logic [19-20], are able to acquire key information patterns in complex systems in a multi-dimensional information domain. The drawback of these methods is represented by the fact that they are of a “black-box” type, and they are therefore not physically consistent.

Virtual optimization can be realized by means of several methods, which can be coupled with engine models characterized by different degrees of detail. One of the most widely adopted techniques for multi-objective optimization is the genetic algorithm (GA) approach. An example of GA-based optimization of engine parameters can be found in [21], where the optimizer is coupled with an ANN-based engine model. An interesting application is also proposed in [22], where 3D-CFD results are used to train an ANN-based model, which is subsequently used for optimization with GA-based methods.

A drawback of the proposed optimization methods is given by the relative high computational time that is required. The latter could not represent a critical aspect for offline optimization, but is of fundamental importance for real-time and onboard applications.

Given this background, low-computational demanding optimization techniques, which have recently been developed by the authors for the energy management of hybrid vehicles [23, 24], have been applied in this study for the optimization of the injection timing and EGR rate of an Euro 6 diesel engine, which was coupled with a C-class vehicle. Gear number has also been included in the optimization process. In particular, one global optimizer and one real-time optimizer have been applied.

A recently developed mean-value control-oriented engine model has been used for the modeling of the engine parameter response (i.e., injection strategy and EGR rate) on the output variables (i.e., fuel consumption and NO_x emissions). The advantage of using a mean-value approach for optimization is due to the fact that it is physically-consistent (compared, for example, to black-box models) and it is therefore robust with respect to extrapolation outside the calibration range. At the same time, it is characterized by a very low computational time. 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 emissions.

The optimization process was based on the minimization of a target function, which takes into account FC and NO_x emission levels. The selected control variables of the problem are the injection timing of the main pulse, the position of the EGR valve (which have been considered as the most influential engine parameters on both fuel consumption and NO_x emissions) and the gear number.

One benchmark tool, which is based on the dynamic programming technique, and one real-time tool, which implements a static optimization method, have been applied for the optimization process. A new mathematical technique has been introduced and applied to the vehicle and engine models in order to decrease the computational time of the optimizers to a great extent. First, the vehicle model equations have been written in order to allow a coarse time grid to be used; second, the control variables (i.e., EGR valve position and injection timing) have been discretized, and the values of the main model variables have been evaluated and stored in associated matrices (referred to as configuration matrices), for all the possible combinations of control variables and for each time node, before the optimization process. In this way, the optimizers read the actual values of the relevant variables from the pre-processed data, instead of calculating iteratively during the optimization stage.

The real-time method has the potential of being implemented in the engine control unit (ECU), in order to realize an onboard optimization of the selected engine and vehicle parameters.

ENGINE SETUP AND EXPERIMENTAL ACTIVITY

The experimental tests for the calibration and validation of the engine model were conducted on a 1.6L Euro 6 GM 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.
- 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_f	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 × 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.

[Figure 1](#) reports the engine map tests, in terms of normalized BMEP and engine speed.

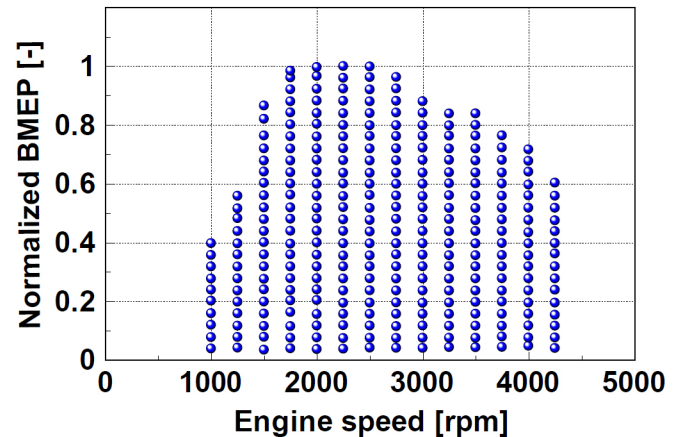


Figure 1. Engine map tests used for the model calibration.

ENGINE MODEL

A control-oriented engine model has been recently developed by the authors and is currently under publication. A summary of the methodology is reported hereafter for the sake of clarity. The approach 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 [26], which was based on the accumulated fuel mass approach [27, 28, 29, 30]. 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. Polytropic evolutions are assumed during the compression and expansion phases. The simulation of the in-cylinder pressure allows several metrics, such as PFP and IMEP, to be evaluated.
3. Friction losses: the Chen-Flynn approach [31] has been re-tuned using the experimental dataset of Tab. 2, 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 the approach proposed in [32], which is based on the opening position of the EGR valve, as well as on the thermodynamic conditions upstream and downstream from the valve. The simulation of the EGR rate is required to evaluate the intake oxygen concentration, which is then used in several correlations.
5. NO_x emission levels: the semi-empirical correlation previously developed by the authors for a 2.0L Euro 5 diesel engine and reported in [33] has been re-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 estimated by means of a correlation that takes into account the intake manifold pressure, the start of injection and engine load and speed; the exhaust manifold pressure has been evaluated by means of a look-up table, which is a function of the pressure in the intake manifold, the engine load and speed. The inducted charge mass has been correlated to the intake manifold pressure and to the engine speed.

The main advantage of the proposed model consists of the low computational demand, which makes it suitable for fast optimization tasks and control-oriented applications.

The proposed method is not however expected to provide an accurate simulation of fast transient conditions, since 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 have not been considered.

It has been recently verified that this approach is suitable for simulating steady-state and mildly transient conditions, such as the NEDC and the WLTP.

Chemical Energy Release (Q_{ch}) Model

The chemical energy release has been simulated on the basis of an enhanced version [25] of the baseline model presented by the authors in [26], which was based on the accumulated fuel mass approach.

In other words, the chemical energy release rate of each pilot pulse pil,j has been simulated on the basis of the baseline model, 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]:

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

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

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 [26] 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 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]). Finally, 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:

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

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

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

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

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

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 degrees. 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 %.

The adopted correlation variables for ignition delay were chosen in accordance with the study proposed in [34]. In particular, all the engine variables were first included in the correlations. A sensitivity analysis was then carried out in order to select the most influential ones by means of a stepwise regression. This approach was also adopted to identify the engine variables that had to be included in all the correlations reported hereafter.

Net Energy Release (Q_{net}) Model

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 [26]. The net heat release is derived from the chemical release according to the following formulation [26]:

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

$$Q_{net}^{SOI} \cong Q_{net}^{SOC} - Q_{f,evap} \quad (12)$$

where Q_{net}^{SOC} and Q_{net}^{SOI} indicate the net energy release calculated from SOC or SOI, respectively, $Q_{f,evap}$ and $Q_{ht,glob}$ indicate the fuel evaporation heat from SOI to SOC (J) and the heat globally exchanged between the charge and 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. 2 (the procedure is reported in [26]), and the following correlations were identified for the two parameters as a function of the engine speed, load and intake manifold temperature:

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

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

In-Cylinder Pressure Model

The in-chamber pressure was evaluated during the combustion interval using a single-zone model [35]:

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

where the isentropic coefficient $\gamma = c_p/c_v$ was set to be constant and equal to 1.37.

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

$$pV^n = const \quad (16)$$

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

The in-chamber pressure at IVC (Intake Valve Closure), that is, the starting condition, was correlated to the pressure in the intake manifold p_{int} , using a correction factor Δp_{int} , as follows:

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

The experimental values of the n , n' 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 [26] for the complete procedure), and the following correlations were identified for the three parameters, which are a function of the intake manifold thermodynamic conditions and of the engine load and speed:

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

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

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

The in-cylinder pressure values during the intake and exhaust phases were considered as constants and were set to be equal to the pressure in the intake and exhaust manifolds, respectively [26]. 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 metrics, such as PFP and IMEP, to be evaluated.

FMEP Model

The Chen-Flynn approach [31] 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:

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

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:

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

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

Estimation of the EGR Rate, of the Trapped Mass and of the Intake O₂ Concentration

The inputs of the EGR system of the engine are the EGR valve opening signal (i.e., u_{EGR}) and the throttle valve position signal (i.e., u_{th}). The system is also equipped by an EGR cooler by-pass, in order to exclude the cooler for several operating conditions.

For the sake of simplicity, we assume that the throttle valve is not actuated and that the cooler is not bypassed.

As a consequence, the EGR flow is controlled by means of the EGR valve opening signal u_{EGR} .

The procedure illustrated by the authors in [32] to evaluate the EGR rate and the intake O₂ concentration has been tuned for the engine considered in the present paper.

The procedure allows the EGR rate to be estimated on the basis of the thermodynamic conditions measured at the EGR valve inlet and outlet, as well as of the EGR valve opening degree, i.e., u_{EGR} . The EGR rate is then used to estimate the intake charge O₂ concentration.

The analyzed system is a short-route cooled EGR circuit, where the EGR valve is located upstream from the cooler.

First, the EGR volume flow rate was correlated to the parameter W_{EGR} , which is defined as follows [32]:

$$W_{EGR} = u_{EGR} \sqrt{\left(\frac{p_{exh} - p_{int}}{\rho_{int}} \right)} \quad (24)$$

where p_{exh} , p_{int} and ρ_{int} are the pressure in the exhaust manifold, the pressure in the intake manifold and the density in the intake manifold.

The parameter W_{EGR} is the product of a term related to a characteristic EGR flow velocity through the EGR valve by the term u_{EGR} , which is a function of the EGR valve cross sectional area.

The experimental values of the EGR volume flow rate $\dot{V}_{EGR}$ have been evaluated, for the tests listed in Tab. 2, starting from the experimental values of the EGR mass flow rate, which in turn were evaluated on the basis of the procedure explained in [36], which is

based on the CO₂ measurement in the intake manifold. The density in the intake manifold ρ_{int} was assumed as the reference one. The identified correlation is the following one:

$$\dot{V}_{EGR} \left[\frac{m^3}{h} \right] = 0.4454 W_{EGR} + 4.5478 \quad (25)$$

An empirical correlation was used to evaluate m_{trap} , which is a function of the intake manifold pressure and engine speed:

$$m_{trap} = m_{air} + m_{EGR} = 430 p_{int}^{1.10} N^{-0.0264} \quad (26)$$

Once the trapped mass and the EGR mass flow rates are known, the EGR rate X_r can be evaluated as follows:

$$X_r = \frac{m_{EGR}}{m_{trap}} = \frac{m_{EGR}}{m_{air} + m_{EGR}} \quad (27)$$

Finally, the intake oxygen concentration was evaluated as a function of the parameter X_r/λ [37-38], and the following correlation was identified:

$$O_2 = -20.353 \cdot \frac{X_r}{\lambda} + 20.786 \quad (28)$$

where λ is evaluated on the basis of the mass of injected fuel and of the mass of trapped air. The latter quantity can be obtained as the difference between the total trapped mass (Eq. (26)) and the trapped mass of EGR.

Estimation of the NO_x emissions

The NO_x emissions were evaluated starting from the semi-empirical model developed by the authors in [33]. This correlation was re-tuned for the engine considered in the present study, on the basis of the experimental tests reported in Tab. 2.

The following correlations were obtained from the tuning of the NO_x emission model:

$$NO_x [ppm] = 24.6 E6 p_f^{0.326} \exp \left(\frac{-24500}{T_{b,MFB50}} \right) X_r^{-0.827} N^{-0.722} q_{f, inj}^{-0.0534} \quad (X_r > 20\%) \quad (29)$$

$$NO_x [ppm] = 2970 p_f^{0.307} \exp \left(\frac{-39000}{T_{b,MFB50}} \right) O_2^{4.95} N^{-0.521} q_{f, inj}^{-0.00772} \quad (X_r \leq 20\%) \quad (30)$$

Equations (29-30) were derived considering all the data related to the engine map tests and variation list tests, as it was verified that this led to satisfactory results for both datasets. Moreover, all the main engine variables related to NO_x formation were first included in the correlations, and a sensitivity analysis was carried out in order to exclude the least influential ones.

The temperature of the burned gases evaluated at MFB50 (i.e., $T_{b,MFB50}$) was used instead of the maximum temperature during the combustion of the main pulse (i.e., $T_{b,max,main}$, see [33]). In fact, it was found that the utilization of the $T_{b,MFB50}$ parameter leads to the same accuracy as $T_{b,max,main}$, but it is much easier to identify. It is worthwhile recalling that the temperature of the burned gases was evaluated by means of the real-time thermodynamic three-zone model presented in [39], which is summarized in the next subsection.

Three-Zone Thermodynamic Model

The real-time three-zone thermodynamic model presented in [39] has been used in order to evaluate the temperature of the burned gases, for the subsequent implementation of the NO_x formation model (see Eqs. (29-30)).

The combustion chamber has been divided into: a vapor-fuel zone (f), an unburned gas zone (u), made up of fresh-air, residual gas and EGR, and a burned gas zones (b) obtained from a globally stoichiometric combustion. The combustion process can be modeled as follows. The in-cylinder content from the IVC to the SOI, is made up of unburned gas, which is considered as a homogeneous mixture of fresh air, EGR and residual gas. A vapor-fuel zone is generated after the SOI, as a consequence of the heating of the injected fuel. The parcels from the vapor fuel zone react with the parcels from the unburned gas zone at stoichiometric ratio, thus generating the burned gas zone (b) after the SOC.

The energy conservation equations are written in Eulerian nonstationary formulation for the different zones:

$$\delta Q_f + V_f dp = d(m_f h_f) - dm_{f, inj} h_{f, inj} + dm_{f \rightarrow b} h_f \quad (31)$$

$$\delta Q_u + V_u dp = d(m_u h_u) + dm_{u \rightarrow b} h_u \quad (32)$$

$$\delta Q_b + V_b dp = d(m_b h_b) - dm_{u \rightarrow b} h_u - dm_{f \rightarrow b} h_f \quad (33)$$

The mass conservation equations can be written as follows:

$$dm_f = dm_{f, inj} - dm_{f \rightarrow b} \quad (34)$$

$$dm_u = -dm_{u \rightarrow b} \quad (35)$$