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A control-oriented approach to estimate the injected fuel mass on the basis of the measured in-cylinder pressure in multiple injection diesel engines

Original

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1	A CONTROL-ORIENTED APPROACH TO ESTIMATE THE INJECTED FUEL MASS
2	ON THE BASIS OF THE MEASURED IN-CYLINDER PRESSURE IN MULTIPLE
3	INJECTION DIESEL ENGINES
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8	
9	Key Words: diesel, control-oriented, injection, pilot, main
10	
11	ABSTRACT
12	A new control-oriented methodology has been developed to estimate the injected fuel
13	quantities, in real-time, in multiple injection DI diesel engines on the basis of the measured
14	in-cylinder pressure.
15	The method is based on the inversion of a predictive combustion model that was previously
16	developed by the authors, and that is capable of estimating the heat release rate and the in-cylinder
17	pressure on the basis of the injection rate. The model equations have been rewritten in order to
18	derive the injected mass as an output quantity, starting from use of the measured in-cylinder
19	pressure as input.
20	It has been verified that the proposed method is capable of estimating the injected mass of pilot
21	pulses with an uncertainty of the order of $\pm 0.15$ mg/cyc, and the total injected mass with an
22	uncertainty of the order of $\pm 0.9$ mg/cyc. The main sources of uncertainty are related to the
23	estimation of the in-cylinder heat transfer and of the isentropic coefficient $\gamma = c_p/c_v$ .
24	The estimation of the actual injected quantities in the combustion chamber can represent a
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25	powerful means to diagnose the behavior of the injectors during engine operation, and offers
26	the possibility of monitoring effects, such as injector ageing and injector coking, as well as of
27	allowing an accurate control of the pilot injected quantities to be obtained; the latter are in fact
28	usually characterized by a large dispersion, with negative consequences on the combustion
29	quality and emission formation.

30 The approach is characterized by a very low computational time, and is therefore suitable31 for control-oriented applications.

32

## **33 NOMENCLATURE**

- 34 A: heat transfer area
- 35 AC: Alternating Current
- 36 B: Bore diameter
- 37 *BDC*: Bottom Dead Center
- 38 BMEP: Brake Mean Effective Pressure
- 39 *BTDC*: Before Top Dead Center
- 40 *c*: coefficient of sensitivity
- 41 CA: crank angle
- 42 *co*: Woschni heat transfer calibration coefficient
- 43 *CLD:* Chemi-Luminescence Detector
- 44  $c_p$ : specific heat at constant pressure
- 45  $c_v$ : specific heat at constant volume
- 46 DI: Direct Injection
- 47 *DoE:* Design of Experiment
- 48 DT: Dwell Time
- 49 ECU: Electronic Control Unit

- 50 EGR: Exhaust Gas Recirculation
- 51 *EOC*: End of Combustion
- 52 EOI: End of Injection
- 53 *ET*: Energizing Time
- 54 EVO: Exhaust Valve Opening
- 55 GMPT-E: General Motors PowerTrain-Europe
- 56 *h: heat transfer convective coefficient*
- 57  $H_L$ : lower heating value of the fuel
- 58 HLM: HRR local minima criterion to estimate the SOC of intermediate pulses
- 59 HRR: Heat Release Rate
- 60 ICEAL-PT: Internal Combustion Engine Advanced Laboratory at the Politecnico di Torino
- 61 *IMEP*: Indicated Mean Effective Pressure
- 62 *IVC*: Intake Valve Closing
- 63 k: coverage factor
- 64 *K*: combustion rate parameter of the heat release model
- 65 *m*: compression phase polytropic coefficient; mass
- 66  $\dot{m}_{f,inj}$ : fuel injection rate
- 67 MFB50: crank angle at which 50% of the fuel mass fraction has burned
- 68 *n*: engine speed
- 69 *NEDC*: New European Driving Cycle
- 70 *p*: pressure
- 71 PCCI: Premixed Charge Compression Ignition
- 72  $p_f$ : injection pressure
- 73 *PFP:* Peak Firing Pressure

- *p*<sub>int</sub>: intake manifold pressure
- $p_m$ : motored pressure
- *q*: generic injected fuel quantity [mm<sup>3</sup>]
- $q_{f,inj}$ : total injected fuel quantity [mm<sup>3</sup>]
- $q_{pil,tot}$ : total injected fuel quantity of the pilot pulses [mm<sup>3</sup>]
- Q: energy
- $Q_{ch}$ : released chemical energy
- $Q_{fuel}$ : chemical energy of the injected fuel
- $Q_{ht}$ : heat exchanged by the charge with the walls
- $Q_{ht,glob}$ : heat globally exchanged by the charge with the walls over the combustion period
- $Q_{net}$ : net energy
- *R*: gas constant
- $R^2$ : squared correlation coefficient
- *RAFR*: relative air-to-fuel ratio
- *RMSE*: root mean square of the error
- 89 SOC: Start of Combustion
- 90 SOI: Start of Injection
- *S<sub>p</sub>*: *Mean piston speed*
- *t*: time
- *T*: temperature
- *TAF*: time average filtering
- *T<sub>int</sub>*: intake manifold temperature
- *T*soi: temperature evaluated at SOI
- $T_w$ : wall temperature
- *TDC*: Top Dead Center

- $u^2$ : variance
- 100 U: Expanded uncertainty
- *V*: volume
- *V*<sub>d</sub>: cylinder displacement
- *WG*: wastegate
- 104 Greek Simbols
- $\gamma = c_p/c_v$ : isentropic coefficient
- $\rho$ : density
- $\rho_{SOI}$ : density evaluated at SOI
- $\tau$  ignition delay parameter of the heat release model
- 110 Subscripts
- *exp*: experimental
- *f, inj*: injected fuel
- *f, evap:* evaporated fuel
- *glob*: global
- *ht*: heat transfer
- *inj*: injected
- *int*: related to the intake manifold
- *main*: related to the main injection
- *pil*: related to the pilot injection
- 121 Superscripts
- *SOC*: calculated from the start of combustion

- 123
- 124 Notation
- 125 \*: constant in time
- 126

#### 127 1. INTRODUCTION

128 Modern DI diesel engines usually adopt multiple injections in order to optimize pollutant 129 formation, combustion noise and efficiency [1-3]. The injected fuel quantity of each pulse is 130 usually set by the ECU (Electronic Control Unit) in open-loop control mode, on the basis of an 131 injector map; this map usually provides the injected quantity as a function of the energizing time 132 (ET) (i.e., the duration of the electric command provided to the injector) and of the injection 133 pressure (p<sub>f</sub>), and is derived from an experimental steady-state characterization of the injectors performed at the hydraulic test bench [4, 5]. However, a real-time control of the actual injected 134 135 quantities has not yet been developed. The actual quantity of fuel that is injected into the 136 combustion chamber can in fact be very different from that resulting from the steady-state 137 injector map, as a consequence of several effects. First, the dynamic pressure effects that are 138 induced in the pipes by multiple injection strategies may lead to a variability of the actual 139 injected quantities even for a constant ET, depending on the time interval between consecutive 140 injections (i.e., on the dwell-time DT) and on the injection pressure [4, 5]. Moreover, effects 141 such as injector ageing and coking [6] may lead to a worsening of the injection system 142 performance, which cannot currently be predicted by ECUs.

A critical situation occurs for pilot injections, in which the injected quantity is very small. The actual amount of injected fuel mass of the pilot pulses may in fact be highly dispersed during engine operation, and this may depend on the injection parameters (i.e., DT, pr) or on injection system ageing. Considering that pilot pulses affect pollutant formation [2, 7], and above all soot formation [8] to a great extent, it could be very useful to be able to perform real-time diagnostics of the actual injected fuel mass in the combustion chamber during engine operation in order to improve the combustion quality and to reduce engine-out pollutant emissions.

151 Interest in control-oriented predictive combustion models that simulate the heat release rate and in-cylinder pressure in diesel engines has been growing over the last few years, due to their good 152 153 predictive capability and the increasing computational performance of modern ECUs [9-17]. One of the most widely adopted methods used to simulate the heat release in DI diesel engines is the 154 155 accumulated fuel mass approach [11-17]. This method is based on the assumption that the rate of 156 released chemical energy is proportional to the energy associated with the fuel quantity made 157 available for combustion at the considered instant. This energy can be computed at time t as the 158 difference between the chemical energy associated with the injected fuel quantity and the cumulative heat release. The accumulated fuel mass method has the great advantage of being able to 159 directly relate the injection rate to the combustion rate, and is therefore physically consistent. This 160 approach has been applied extensively and successfully to a wide range of engine hardware setups 161 162 and operating conditions [13-17].

The capability of this model to correctly correlate the injection rate with the heat release rate has 163 164 suggested the possibility of using the same model to derive the injected fuel mass as an output 165 quantity, starting from the experimental heat release rate that is obtained from the measured in-cylinder pressure. The model equations have therefore been rewritten in order to derive the 166 injected fuel mass as a function of the heat release rate. The proposed methodology therefore 167 168 requires the measurement of the in-cylinder pressure, which can be carried out by means of pressure 169 sensors embedded in the glow-plugs (e.g., see [18]); these pressure sensors could also be used for 170 closed-loop combustion control tasks as shown in [18].

171 The investigation has been carried out on a GM 2.0 L diesel engine at several representative 172 key-points of the NEDC (New European Driving Cycle), which feature a pilot-main injection

173 strategy. The approach that is proposed in this study to estimate the actual injected quantities is 174 based on combustion diagnostics; alternative methods based on the measurement of hydraulic 175 variables in the injection system, have already been proposed in the literature, such as those 176 reported in [19-20]. In that study, a miniaturized hot-film anemometer with titanium/platinum 177 metallization was developed on a low-temperature co-fired ceramics substrate and integrated in 178 a Common Rail injection nozzle, so that the injected fluid velocity could be measured. The 179 integration of a ceramic flow sensor chip in the nozzle leads to an additional cost compared to 180 the baseline injector configuration. Moreover, it may cause some disturbances to the fluid 181 properties inside the nozzle and induce a reduction in the injected quantities. However, it offers 182 the advantage of being able to directly monitor the injected fuel velocity in each injector.

The method proposed in the present study is based on the measured in-cylinder pressure. This solution also requires an additional cost, related to the need to install integrated pressure sensors in the cylinders; however, these sensors can also be used for additional tasks, such as closed-loop combustion control of combustion phasing. An advantage of the proposed method is that it does not cause disturbances in the fluid properties inside the nozzle; however, a limit in the minimum dwell-time between consecutive injections is introduced, as the method assumes that any heat release overlapping between consecutive injection pulses is negligible.

In the present investigation, a comparison was made between the injected quantities that are predicted by the model and those which are derived from the ECU on the basis of the injector map. The total injected quantity predicted by the model was also compared with that derived from a fuel flow meter. The investigation was carried out on some representative operating conditions of the NEDC.

A sensitivity analysis on the effects of the main model parameters was also carried out, and the uncertainty of the estimated injected quantities was evaluated.

197 Finally, an analysis of the required computational time was also done.

#### 199 2. TEST ENGINE AND EXPERIMENTAL SETUP

The model has been assessed and applied to a Euro 5 GMPT-E 2.0 L engine equipped with a twin-stage turbocharger and with piezo-driven injectors. The main engine specifications are reported in Table 1.

203

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

Engine type	2.0L "Twin-Stage" Euro 5	
Displacement	1956 cm <sup>3</sup>	
Bore x stroke	83.0 mm x 90.4 mm	
Connecting rod 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/I – 205 Nm/I	
Injector specifications	d=137 $\mu$ m, Cd = 0.8, Ca =0.84, Cv =0.95	

204

The experimental data were acquired in a previous research activity that was financially supported by GMPT-E and which had the aim of investigating the potentialities of innovative piezoand solenoid-driven Common Rail injection systems on engine performance, emissions, fuel consumption and combustion [3]. The experimental tests, related to the engine and to the injectors, were carried out at the highly dynamic test bed and at the hydraulic test rig at ICEAL-PT (Internal Combustion Engine Advanced Laboratory at the Politecnico di Torino).

The experimental characterization of the injectors was carried out at the Bosch-Moehwald hydraulic bench [4-5], which is instrumented for a complete fluid-dynamic characterization of Common Rail fuel-injection systems, and includes an EVI injection-rate meter, an EMI2 device to gauge the oil injected mass as well as thermo-piezo-transducers to measure temperature and 215 pressure traces in the rail and at the injector inlet.

The engine dynamic test rig was equipped with: an 'ELIN AVL APA 100' cradle-mounted AC dynamometer, featuring a power of 220 kW, a nominal torque of 525 Nm, and a maximum speed of 12000 rpm; an 'AVL KMA 4000' system, used to continuously meter engine fuel consumption; an 'AVL AMAi60' raw exhaust-gas analyzer.

All of the abovementioned measuring instruments were controlled by the AVL PUMA OPEN 1.3.2 automation system. The test bed environment was interfaced with AVL CAMEO software to run intelligent engine calibration procedures on the basis of the DoE (Design of Experiment) approach.

224 The engine was fully instrumented with piezoresistive pressure transducers and K and T 225 thermocouples to measure the pressure and temperature levels at various engine locations. An 'NGK' UEGO air-fuel ratio sensor was placed inside the exhaust system. A Kistler 6058A41 226 high-frequency piezoelectric transducer was fitted to the glow-plug seat to measure the 227 228 in-cylinder pressure time-histories. The in-chamber pressure traces were referenced on the 229 basis of the pressure in the intake manifold, which was measured by means of a high-frequency 230 Kistler 4075A10 piezoresistive transducer. The latter was located at the inlet runner of the 231 cylinder equipped with the in-cylinder pressure sensor.

232

#### 233 2.1. Experimental tests

The method was tested on six key-points (expressed in terms of engine speed x BMEP), which were identified in order to characterize the engine operations in a medium-sized passenger car over an NEDC. The key-points are: 1500x2, 1500x5, 2000x2, 2000x5, 2500x8, 2750x12 rpm x bar. Table 2 summarizes the main engine parameters for the six analyzed tests.

238

#### Table 2 – Main specifications of the six analyzed key-points.

V	рf	Intake O <sub>2</sub>	pint	q <sub>pil</sub> (ECU)	qf,inj (ECU)	SOI pil	SOI main
Key-point	bar	%	bar	mm <sup>3</sup>	mm <sup>3</sup>	deg BTDC	deg BTDC
1500x2	458	15.98	0.963	1.7	7.6	10.7	-2.37
1500x5	589	16.73	1.169	1.6	17.3	12.2	-0.75
2000x2	554	16.58	1.051	1.6	9.3	16.2	-0.86
2000x5	753	16.73	1.373	1.4	18.2	17.0	-0.12
2500x8	1198	17.83	2.01	1.2	33.2	25.4	3.84
2750x12	1481	17.69	2.23	1.1	46.7	30.5	6.63

240

#### 241 **3. DESCRIPTION OF THE COMBUSTION MODEL**

A detailed description of the model, including the calibration and validation procedures, has already been given in detail in [13-15]. However, a summary is reported hereafter for the sake of clarity.

The 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. This 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.

250 The released chemical energy  $Q_{ch,j}$  at time t, for each pulse j  $(1 \le j \le N)$  of the injection event, 251 is therefore evaluated as follows:

252 
$$\frac{dQ_{ch,j}}{dt} = K_j \left[ Q_{fuel,j} \left( t - \tau_j \right) - Q_{ch,j} \left( t \right) \right]$$
(1)

where  $K_j$  and  $\tau_j$  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:

255 
$$Q_{fuel,j} = \int_{t_{SOI,j}}^{t} \dot{m}_{f,inj} H_L dt \qquad t \le t_{EOI,j}$$
(2)

256 
$$Q_{fuel,j} = \int_{t_{SOI,j}}^{t_{EOI,j}} \dot{m}_{f,inj} H_L dt \qquad t > t_{EOI,j} \qquad (3)$$

SOI and EOI denote the start and end of injection, respectively,  $\dot{m}_{f,inj}$  the injection rate and  $H_L$  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:

260 
$$Q_{ch} = \sum_{j=1}^{N} Q_{ch,j}$$
 (4)

The  $K_j$  and  $\tau_j$  parameters have to be appropriately tuned in order to obtain a good matching between the predicted  $Q_{ch}$  curves and those derived from the experimental in-cylinder pressure [13].

The calibration is performed for several steady-state operating conditions, and physically-consistent correlations are then derived as a function of significant engine variables. The tuning procedure of the model for the engine considered in this study is reported in [15].

267 With reference to the combustion rate coefficients, it was found, in [13-15], that a constant value can be taken for the combustion rate parameter of the pilot pulse (i.e., K<sub>pil</sub>). Instead, it 268 269 was shown, in [13-15], that a dependence exists between the combustion rate parameter of the 270 main pulse (i.e.,  $K_{main}$ ) and the engine speed/load conditions, and this dependence can be taken 271 into account either by means of a look-up table or by means of simple correlations such as 272 those shown in [13]. In addition, a reduction factor was introduced in [15] to correctly account 273 for low-density/low-oxygen operating conditions, which may induce a remarkable decrease in 274 the combustion rate.

The ignition delay parameters were studied in detail in [21], and the following correlations were identified for the engine considered in the present study:

277 
$$\tau_{pil} = 0.008977 \rho_{SOI}^2 - 0.36077 \rho_{SOI} + 3.8619$$
(5)

278 
$$\tau_{main} = 3.8E4 \,\rho_{SOI}^{-2.2} \exp^{\left(\frac{2100}{T_{SOI}}\right)} O_2^{-1.9} p_f^{-0.31} q_{pil,tot}^{-0.35} \tag{6}$$

Equation (5) indicates that only the charge density calculated at SOI ( $\rho_{SOI}$ ) is necessary for

a good estimation of the ignition delay of the pilot injections, as the physical contribution is predominant over the chemical one [21]. Instead, the effect of the intake charge oxygen concentration (O<sub>2</sub>), of the charge density and temperature at SOI ( $\rho$ soi, Tsoi), of the injection pressure (pf) and of the total injected pilot quantity (q<sub>pil,tot</sub>) have to be included to accurately predict the ignition delay of the main pulse.

285  $\rho_{SOI}$  and  $T_{SOI}$  are calculated assuming that the in-chamber thermodynamic conditions at BDC 286 are equal to those in the intake manifold (p<sub>int</sub>, T<sub>int</sub>):

287 
$$\rho_{SOI} = \frac{p_{int}}{RT_{int}} \left( \frac{V_{BDC}}{V_{SOI}} \right)$$
(7)

288 
$$T_{SOI} = T_{int} \left( \frac{V_{BDC}}{V_{SOI}} \right)^{m-1}$$
(8)

where *m* is the polytropic coefficient during the compression phase.

The choice of using BDC as the reference angle to estimate  $\rho_{SOI}$  and  $T_{SOI}$  is just a convention. It was in fact verified that the values of  $\rho_{SOI}$  and  $T_{SOI}$  obtained using, for example, IVC as the reference crank angle are very similar (i.e., the root mean square difference is of the order of 0.34 kg/m<sup>3</sup> and 5.3 K, respectively, for the key-points analyzed in the present paper).

294 The values of  $\rho_{SOI}$  and  $T_{SOI}$  obtained using Eq. (7) and Eq. (8) are obviously approximations of the actual values inside the combustion chamber. The actual values of psoi could be estimated by 295 296 dividing the total trapped mass by the in-chamber volume at SOI. The total trapped mass is the sum of the trapped air mass, of the trapped EGR mass and of the residual gas mass. The different 297 298 contributions could be evaluated on the basis of the air-to-fuel ratio and of the EGR rate, which can in turn be estimated with the procedure shown in [22]. The actual T<sub>SOI</sub> values could instead be 299 300 evaluated by applying the ideal gas law at SOI, using the values of the in-cylinder pressure at SOI 301 and of the total trapped mass.

Figure 1a shows a comparison between the actual values of  $\rho_{SOI}$  and those obtained using Eq. (7), while Fig. 1b shows a comparison between the actual values of T<sub>SOI</sub> and those obtained using Eq. (8), for the key-points analyzed in the present study.



Figure 1 – Experimental vs. calculated values of ρ<sub>SOI</sub> using Eq. (7) (a); experimental
 vs. calculated values of T<sub>SOI</sub> using Eq. (8) (b).

305

308 In general, the approximation of the density is quite good, as the average root mean square (RMS) difference is 0.99 kg/m<sup>3</sup> for the p<sub>SOL,pil</sub> term and 1.76 kg/m<sup>3</sup> for the p<sub>SOL,main</sub> term (the 309 310 use of Eq. (7) leads to an overestimation). With reference to T<sub>SOI</sub>, it can be seen that the average root mean square difference between the values obtained with Eq. (8) and the actual 311 ones are 31 K for T<sub>SOI,pil</sub> and 74 K for T<sub>SOI,main</sub>. The underestimation obtained using Eq. (8) is 312 313 mainly due to the heat transfer effects during the intake phase and to the residual gas 314 contribution being neglected. Moreover, the greater overestimation of T<sub>SOI,main</sub> is due to the 315 charge heating caused by the pilot combustion being neglected.

However, it can be seen, in Fig. 1, that the use of Eqs. (7-8) allows the trends of psoI and TsoI to be captured, so that the differences, with respect to the true values, are indirectly taken into account through the tuning of the exponents in Eqs. (5-6). Moreover, Eqs. (7-8) require the intake manifold pressure and temperature, which are known quantities for the ECU, as input, 320 and they can therefore be considered suitable for control-oriented applications.

321 O<sub>2</sub> can be derived from the real-time EGR and intake charge model [23], while  $p_f$  and 322  $q_{pil,tot}$  are known quantities for the engine ECU.

The predicted released chemical energy  $Q_{ch}$  can be used as a starting quantity to evaluate the charge net energy (i.e.,  $Q_{net}$ ) which is given, at each time instant, by the difference between the released chemical energy and the heat exchanged by the charge with the walls, i.e.,  $Q_{ht}$  [24]:

$$Q_{net} = Q_{ch} - Q_{ht} \qquad (9)$$

327  $Q_{net}$  can be approximated by means of a uniform scaling of the  $Q_{ch}$  curve, according to the 328 procedure proposed in [13-15]; the scaling factor includes the global heat exchanged between the 329 charge and the walls over the combustion period, i.e.,  $Q_{ht,glob}$ .

330 The net energy of the charge can therefore be approximated as follows [13]:

331 
$$Q_{net}^{SOC} \cong Q_{ch} \frac{m_{f,inj}^* H_L - Q_{ht,glob}}{m_{f,inj}^* H_L}$$
(10)

where  $m_{f,inj}^*$  is the total injected fuel mass and H<sub>L</sub> the lower heating value of the fuel; the 332 333 superscript (SOC) indicates that  $Q_{net}$  is evaluated from the start of combustion onwards. The  $Q_{ht,glob}$ 334 parameter must be known in order to apply Eq. (10). In general, it is necessary to derive a 335 physically consistent correlation for this parameter, such as that proposed in [13], as a function of the ratio between the injected fuel quantity and the engine speed. The experimental values of *Q*<sub>ht,glob</sub>, 336 which are necessary to identify this correlation, are calculated as the difference between the 337 chemical energy of the total injected mass (i.e.,  $m_{f,inj}^*H_L$ ) and the value of the experimental net 338 energy curve evaluated at the end of combustion. The experimental net energy curve  $(Q_{net,exp}^{SOC})$  can 339 340 be derived from the measured in-cylinder pressure using a single zone approach [24]:

341 
$$dQ_{net,exp}^{SOC} = \frac{\gamma}{\gamma - l} p_{exp} dV + \frac{1}{\gamma - l} V dp_{exp} \qquad (t \ge t_{SOC}) \qquad (11)$$

342 where  $\gamma = \frac{c_p}{c_v}$  is the ratio of the specific heats, and was evaluated as a function of the

343 relative air-to-fuel ratio, according to the correlation that was proposed in [13]:

344 
$$\gamma = -0.0064RAFR^2 + 0.0563RAFR + 1.2151 \quad (12)$$

The experimental values of  $\gamma$  used to derive Eq. (12) are shown in Fig. 2, and were derived in [13] for a 1.9L GM diesel engine using different values of the compression ratio. The values of the squared correlation coefficient R<sup>2</sup> and of the root mean square error (RMSE) for the predicted vs. experimental values of  $\gamma$  are also reported in the chart.





relative air-to-fuel ratio [13].

352 The experimental value of  $Q_{ht,glob}$  is therefore calculated as follows:

349

351

353 
$$Q_{ht,glob,exp} = m_{f,inj}^* H_L - Q_{net,exp}^{SOC}(t_{EOC})$$
(13)

It is convenient to use an accurate estimate of the total injected mass in Eq. (13) (e.g., that derived from a fuel flow meter) in order to accurately evaluate the global heat transfer parameter. The calibration of the  $Q_{ht,glob}$  parameter was carried out in [13-15] using the values of the fuel flow rate that were measured by means of a fuel flow meter installed at the test bed; this flow rate 358 was split into equal parts over the 4 cylinders, and converted into an average injected mass per cycle.

359 It has in fact been verified that, for the present engine, the performance of the four injectors is similar360 (see section 3.1).

It was found in [13-15] that  $Q_{ht,glob}$  is well correlated with the  $q_{f,inj}/n$  parameter, where  $q_{f,inj}$ indicates the total injected quantity of fuel and *n* the engine speed. The following correlation was derived from engine map tests acquired on a GM 1.9L diesel engine with a similar hardware setup to that of the present study, but equipped with solenoidal injectors (see [14]):

365 
$$Q_{ht,glob} = 7656 \cdot \frac{q_{f,inj} \left[ mm^3 \right]}{n \left[ rpm \right]} + 14.4 J \qquad (14)$$

Figure 3 shows the experimentally-derived values of  $Q_{ht,glob}$  that were used to identify Eq. (14) (blue triangles), as well as the experimentally-derived values of  $Q_{ht,glob}$  for the six key-points considered in this study (red circles). It can be noted that the values of the key-points are in trend.

369 The values of the squared correlation coefficient  $R^2$  and of the root mean square error (RMSE) 370 for the predicted vs. experimental values of  $Q_{ht,glob}$  are also reported in the chart.





17

376 It will be shown, in Section 5, that a correct estimation of this parameter is fundamental in order377 to accurately predict the injected fuel quantities.

The estimation of the net energy of the charge, i.e.,  $Q_{net}$ , allows the in-cylinder pressure to be predicted in the interval between SOC and EOC, using a single zone approach, as follows:

380 
$$dp = \left(\frac{\gamma - l}{V}\right) \left(dQ_{net} - \frac{\gamma}{\gamma - l} p dV\right) \quad (15)$$

The reconstruction of the complete in-cylinder pressure trace also includes the compression and expansion phases, which are modeled by means of polytropic evolutions, and the gas exchange phases, which are modeled as constant pressure evolutions (see [13-15]). In particular, the following correlations were used to estimate the polytropic coefficients of the compression and expansion phases:

$$386 m = 0.0011 q_{f,inj} (mm^3) + 1.30 (16)$$

387 
$$m' = -0.0016 q_{f,inj} (mm^3) + 1.38$$
 (17)

388

#### 389 *3.1 Analysis of the cylinder-to-cylinder dispersion of the injectors*

Although only a single cylinder was instrumented for the pressure measurements, the cylinder-to-cylinder dispersion of the injectors installed on the present engine was verified indirectly on the basis of the analysis of the measured gas temperatures in the four exhaust runners. In general, the temperatures of the exhaust gases outflowing from each cylinder depend on combustion phasing (related to the injection phasing), on the air-to-fuel ratio (related to the injected fuel quantity) and on heat transfer effects in the cylinders and in the runners. Outer cylinders are in general characterized by higher heat transfer rates than inner ones.

Figure 4 reports the values of the exhaust gas temperatures in the four runners (Fig. 4a), as

398 well as the differences between the temperatures of the cyl1-cyl4 and cyl2-cyl3 exhaust runners (Fig.399 4b).

It can be noted that the outer cylinders are characterized by systematically lower temperatures than the inner ones, due to a higher heat transfer rate. However, by comparing the temperatures of the outer cylinders (cyl1, cyl4), as well as those of the inner cylinders (cyl2, cyl3), it can be noted that the differences are small (i.e., of the order of 5-10°C)



Figure 4 – Values of the gas temperatures in the exhaust runners of the four cylinders (a);
 differences between the temperatures of the gases in the cyl1 and cyl4 exhaust runners, as well
 as between the cyl2 and cyl3 ones (b).

408

404

This indirectly confirms that the behavior of the injectors installed in cyl1 and cyl4, as well as of those installed in cyl2 and cyl3, is very similar in terms of injection phasing and total injected quantity. In order to verify this, the combustion model was applied to the six key-points to test the impact of a deviation of  $\pm 2$  deg in the injection phasing of the main pulse, as well as of a deviation  $\pm 1$ mg in the injected quantity, on the predicted in-cylinder temperatures at EVO. The results are shown in Tab. 3.

415

# 416 Table 3 – Impact of variations in SOI<sub>main</sub> and in the injected fuel mass on T<sub>EVO</sub>, using the 417 combustion model

Var astat	SOI <sub>main</sub> +2deg	SOI <sub>main</sub> -2deg	m <sub>f,inj</sub> +1mg	m <sub>f,inj</sub> -1mg	
Key-point	ΔT <sub>EVO</sub> [°C]	ΔT <sub>EVO</sub> [°C]	ΔTevo [°C]	ΔTevo [°C]	
1500x2	-4.6	5.3	36.8	-37.5	
1500x5	-7.7	8.6	26.9	-27.0	
2000x2	-6.0	6.5	37.0	-37.8	
2000x5	-8.6	9.2	25.5	-26.2	
2500x8	-10.5	11.3	18.1	-18.2	
2750x12	-13.6	14.5	15.5	-15.6	

It can be noted that the experimental differences in the exhaust gas temperatures shown in Fig. 4b correspond to a lower dispersion of the injected quantities than 1 mg/cyc for the same SOI<sub>main</sub>, or to a lower dispersion of SOI<sub>main</sub> than 2 deg for the same m<sub>f,inj</sub>.

Moreover, in a recent research activity devoted to the refinement of the combustion model [17], each cylinder of a 1.6L GM engine has been instrumented with pressure transducers. It has been verified that the average cylinder-to-cylinder dispersion of IMEP was of the order of 0.6 bar (corresponding to a dispersion of the total injected quantity of about 0.5 mg/cyc), and that the average cylinder-to-cylinder dispersion of MFB50 was of the order of 0.5 deg. This confirms that the injector dispersion in modern diesel engines is very low.

428

## 429 4. ESTIMATION OF THE INJECTED MASS ON THE BASIS OF THE MEASURED

## 430 IN-CYLINDER PRESSURE

The combustion model inversion procedure, which allows the injected fuel mass to be calculated as a model output, starting from the measured in-cylinder pressure time-histories, is reported hereafter.

A pilot-main injection strategy has been considered for the development of the method, which is however of general application. It has been assumed that the dwell-time between consecutive injections is not too short, so as to prevent overlapping of the heat release contributions that stem from the different injection pulses. The dwell time threshold generally depends on the specific engine operating condition. In order to give some general indications, a dwell-time sweep was simulated using the combustion model for the six key-points. A threshold was identified for each key-point, under which a significant overlapping of heat release occurs between the pilot and main pulses. The thresholds are reported in Tab. 4.

443 Table 4 – Estimated dwell-time threshold necessary to avoid significant overlapping between
 444 the pilot and main heat release

	DT Threshold [µs]
1500x2	300
1500x5	400
2000x2	500
2000x5	500
2500x8	600
2750x12	600

445

446 Starting from Eq. (1), the  $Q_{fuel}(t-\tau_j)$  term is made explicit as a function of  $Q_{ch,j}$  for each injection 447 pulse j, as follows:

448 
$$Q_{fuel,j}(t-\tau_{j}) = Q_{ch,j}(t) + \frac{1}{K_{j}} \frac{dQ_{ch,j}}{dt}(t) \quad (18)$$

449 It is therefore necessary to obtain an experimental estimation of the  $Q_{ch,j}$  curve for each 450 injection pulse in order to evaluate the  $Q_{fuel}(t-\tau_j)$  term. The estimation can be made as follows.

451 First, Eq. (11) is applied to evaluate the whole trend of the experimental charge net energy (i.e.,

452  $Q_{net,exp}^{SOC}$ ) on the basis of the measured in-cylinder pressure.

The calculation is performed from SOC to EOC (a conventional EOC angle, e.g., 430°, can be assumed). The estimation of SOC is therefore necessary; this can be done according to the procedure illustrated in [21].

456 Once the estimation of  $Q_{net,exp}^{SOC}$  has been carried out, the experimental trend of the released

457 chemical energy (i.e.,  $Q_{ch,exp}^{SOC}$ ) is drawn by scaling the  $Q_{net,exp}^{SOC}$  curve as follows:

458 
$$Q_{ch,exp}^{SOC} \cong Q_{net,exp}^{SOC} \frac{Q_{net,exp}^{SOC}(t_{EOC}) + Q_{ht,glob}}{Q_{net,exp}^{SOC}(t_{EOC})}$$
(19)

This is the inverse procedure to that used to estimate the  $Q_{net}$  curve on the basis of the  $Q_{ch}$ curve predicted by the combustion model (see Eq. (10)). The  $Q_{ht,glob}$  parameter was introduced in Eq. (10) and represents the heat globally exchanged between the charge and the walls during combustion; it can be evaluated through Eq. (14), which was derived from experimental tests. As an example, Fig. 5 reports, for the 1500x5 key-point, the experimental  $Q_{net}$  curve (dotted red line) and the experimental  $Q_{ch}$  curve obtained with Eq. (19) (black line). The  $Q_{ht,glob}$ parameter is also indicated in the figure. In addition, the experimental  $Q_{ch}$  curve obtained using

466 a detailed heat transfer model is also reported with a blue chain line.

467

468

469

471



470 key-point. The chemical energy release curves obtained from Eq. (19) and using a

detailed heat transfer model (Eq. (20)) are reported.

473 The detailed heat transfer model is based on a convective formula [25]:

474 
$$Q_{ht} = hA(T - T_w) \qquad (20)$$

475 where *h* is the convective heat transfer coefficient, *A* is the instantaneous heat transfer area, *T* is 476 the instantaneous gas temperature and  $T_w$  is the wall temperature.

477 The convective heat transfer coefficient  $h [W/m^2K]$  was modeled using the Woschni correlation 478 [26]:

479 
$$h[W/m^{2}K] = c_{0}B[m]^{-0.2} p[bar]^{0.8}T[K]^{-0.53} w[m/s]^{0.8}$$
(21)

480 
$$w = \left[ c_1 S_p[m/s] + c_2 \frac{V_d[m^3]T_r(IVC)}{p_r(IVC)V_r[m^3](IVC)} (p - p_m) \right]$$
(22)

where *B* is the bore diameter,  $S_p$  is the mean piston speed,  $V_d$  is the cylinder displacement, while  $T_r$ ,  $p_r$  and  $V_r$  represent the in-cylinder temperature, pressure and volume at the moment of intake valve closure, respectively.  $c_0$  was used as a calibration factor, while  $c_1$  and  $c_2$  were kept constant and equal to nominal values of 0.0039 and 2.31, respectively.

It can be observed, in Fig. 5, that the simplified procedure used to estimate  $Q_{ch}$  (i.e., Eq. (19)) leads to a good approximation of that obtained with the detailed heat transfer model (i.e., using Eq. (20)), even though a slight overestimation occurs for intermediate CA values between SOC and EOC. Similar trends were obtained for the other key-points, but the results are not reported here for the sake of brevity.

The approach based on the detailed heat transfer model requires the estimation of the instantaneous in-cylinder temperature, which is evaluated through the ideal gas law, using the measured in-cylinder pressure and the total trapped mass in the cylinder as input data. The latter quantity includes the contributions of air, EGR and residual gas.

494 This approach requires a higher computational time effort than the simplified approach (i.e., Eq.

495 (19)), and computational time is a critical parameter for control-oriented applications.

496 Once the  $Q_{ch,exp}^{SOC}$  curve has been evaluated, it is necessary to split this curve into the single 497 contributions that stem from each injection pulse. As previously stated, it is assumed that 498 consecutive injections are scheduled using a moderately long dwell-time, in order to reduce the 499 overlapping of the heat release contributions that stem from the different pulses.

First, it is necessary to identify the SOC of each injection pulse. This can be done by means of the methods reported in [21]. The overall SOC (i.e., that of the most advanced injection pulse) is evaluated as the crank angle at which the first chemical energy fraction is released. The SOC of the subsequent injection shots is detected by identifying the relative minima of the HRR curve. This procedure is referred to as the "HRR local minima" (HLM) method.

At this point, the experimental chemical energy release of the different injection pulses j (i.e.,  $Q_{ch,j,exp}$ , where j ranges from 1 to N, 1 indicates the most advanced pulse and N indicates the most delayed one) can be calculated as follows. With reference to the most advanced pulse (i.e., j=1):

510  

$$Q_{ch,l,exp} = Q_{ch,exp}^{SOC} \quad (t_{SOC} \le t < t_{SOC,2})$$

$$Q_{ch,l,exp} = Q_{ch,exp}^{SOC} (t_{SOC,2}) \quad (t \ge t_{SOC,2})$$
(23)

511 where SOC is the overall start of combustion and SOC,2 is the SOC of the second 512 injection pulse.

513 The contribution of the chemical energy release of the subsequent pulses, excluding the 514 last one (i.e.,  $2 \le j \le N-1$ ), is instead calculated as follows:

515  

$$Q_{ch,j,exp} = Q_{ch,exp}^{SOC} - Q_{ch,exp}^{SOC}(t_{SOC,j}) \quad (t_{SOC,j} \le t < t_{SOC,j+1})$$

$$Q_{ch,j,exp} = Q_{ch,exp}^{SOC}(t_{SOC,j+1}) - Q_{ch,exp}^{SOC}(t_{SOC,j}) \quad (t \ge t_{SOC,j+1})$$
(24)

516 Finally, the contribution of the chemical energy release of the most delayed pulse (i.e.,

517 j=N), is derived as follows:

518 
$$Q_{ch,N,exp} = Q_{ch,exp}^{SOC} - Q_{ch,exp}^{SOC}(t_{SOC,N}) \qquad (t_{SOC,N} \le t \le t_{EOC})$$
(25)

519 The procedure is illustrated in Fig. 6 for the 2000x5 key-point featuring a pilot-main injection520 strategy.



521

Figure 6 – Trends of the global experimental chemical energy release and contributions of
 the pilot and main injections evaluated with Eqs. (23-25) for the 2000x5 key-point. The trend
 of HRR is also reported.

525

Figure 6 reports the overall experimental chemical energy curve (thick black line), as well as the contributions of the pilot (blue chain line) and main (pointed red line) pulses evaluated by means of Eqs. (23-25). The SOC instants evaluated with the above-mentioned procedures are also indicated, as is the HRR curve (thin black line).

530 The  $Q_{fuel,j}$  term can now be estimated at time *t* through the following equation, which is derived 531 from Eq. (18) taking the  $\tau_j$  parameter into account:

532 
$$Q_{fuel,j}(t) = Q_{ch,j}\left(t + \tau_j\right) + \frac{1}{K_j} \frac{dQ_{ch,j}}{dt} \left(t + \tau_j\right)$$
(26)

533 The  $\tau_i$  parameters are estimated by means of Equations (5-6); when input variables related 534 to the injected quantities are used in the correlations to evaluate the model parameters (i.e.,  $\tau_i$ ), the values derived from the injector maps can be used, to avoid iterative procedures; it was in 535 536 fact verified that the resulting error in the evaluation of the ignition delay parameters is negligible. 537

The injected fuel mass curve of each injection pulse j can be estimated as follows: 538

539 
$$m_{f,inj,j} = \frac{Q_{fuel,j}}{H_L}$$
(27)

Finally, the total injected quantity of each pulse can be estimated evaluating the 540 541 steady-state final value of the injected mass curve evaluated by means of Eq. (27).

542 The overall injected fuel mass curve, i.e.,  $m_{f,inj}$ , is obtained by summing the contributions 543 of all the injection pulses:

544 
$$m_{f,inj} = \sum_{j=1}^{N} m_{f,inj,j}$$
 (28)

546

Figure 7 shows a scheme that summarizes the proposed methodology. 545



## 548 **5. RESULTS AND DISCUSSION**

549 The procedure explained in Section 4 was applied to the six key-points whose 550 specifications are reported in Tab. 2. In particular, a comparison was made between the injected 551 fuel mass estimated by means of the proposed method and those derived from the injector map; 552 moreover, a sensitivity analysis was carried out on the main model parameters, and the uncertainty 553 of the pressure-derived values of the injected fuel mass was evaluated.



554 555

Figure 8 – Trends of the measured in-cylinder pressure, of the pressure-derived HRR and of the injection rate, for the six key-points.

557 Figure 8 reports the measured in-cylinder pressure trends (black lines), the HRR curves 558 derived from the pressure traces using a single-zone model (thin red lines) and the injection

rate profiles (pointed blue lines) for the six key-points. The in-cylinder pressure traces shown in Fig. 8 were obtained as the average of 100 consecutive cycles, which were filtered using the TAF (Time-average filtering) procedure shown in [27]. This procedure is based on the following steps. First, the time average of the instantaneous pressure is evaluated in the discrete consecutive CA intervals into which the engine cycle is divided, and the resulting averaged pressure values, taken in the middle of the respective intervals, are interpolated through cubic spline fitting to obtain the original in-cylinder pressure curve.

It can be noted, in Fig. 8, that all the key-points feature a pilot-main injection strategy. Moreover, it can be observed, from the HRR traces, that the 1500x2, 2000x2 and 1500x5 key-points show dominant premixed combustion, while the remaining key-points show a premixed-mixing controlled combustion type. It can be noted that HRR premixed and mixing controlled combustion peaks for the latter key-points are of the same order of magnitude, due to the adoption of a pilot injection which leads to a reduction in the premixed combustion contribution. A single injection strategy would lead to much higher HRR peaks during the premixed combustion phase.

573 With reference to the model outcomes, Figure 9a reports, for the six key-points, the values of the pressure-derived total injected fuel mass, as well as the values obtained from the injector map 574 575 and those derived from the fuel flow meter, while Fig. 9b reports the differences between the values 576 obtained from the injector map/fuel meter and those derived from the measured pressure. Figure 9c, 577 instead, reports the values of the pressure-derived pilot injected quantities as well as those derived from the injector map, while Fig. 9d shows the differences between the values obtained from the 578 579 injector map and those derived from the measured in-cylinder pressure. Figures 6b-d also report the 580 average uncertainty bands of the proposed method. The procedure to estimate the uncertainty is 581 explained in Section 5.2.



Figure 9 – Comparison between the total (a) and pilot (c) fuel injected quantities derived from the in-cylinder pressure and those derived from the injector map (total, pilot) and fuel meter (total); difference between the injected quantities from the injector maps/fuel meter and those obtained from the in-cylinder pressure (b, d). The average uncertainty bands in the estimation of the pressure-derived injected quantities are also reported.

First, it can be noted that the average uncertainty in the evaluation of the injected mass of the pilot pulses is very small (Fig. 9d), as it is of the order of 0.15 mg. The average uncertainty in the evaluation of the total injected quantity is instead of the order of 0.9 mg, and it will be shown, in Section 5.2, that this value mainly depends on the uncertainty in the evaluation of the heat transfer parameter ( $Q_{ht,glob}$ ) when Eq. (14) is used.

597 From the results reported in Fig. 9a-b, it can be observed that the pressure-derived total 598 injected quantities are comparable with those derived from the fuel meter, as the difference is below 599 the uncertainty band of the method. It can also be noted that the proposed method diagnoses several 600 deviations in the total injected quantities estimated by the injector map, (i.e., by the ECU): the 601 quantities of the medium-low load key-points (i.e., 1500x2-5, 2000x2-5) are underestimated, while 602 those of the medium-high load key-points (i.e., 2500x8, 2750x12) are overestimated. In particular, 603 the deviations of the ECU-derived total injected quantities are of the order of +3 mg for the 604 2750x12 key-point, of +1.7 mg for the 2500x8 key-point, of -1.5 mg for the 1500x2 and 2000x5 605 key-points and of -2 mg for the 2000x5 key-point. These deviations can be considered significant, 606 compared with the uncertainty of the method.

With reference to the pilot injected quantities, it can be observed that the method diagnoses significant deviations in the quantities evaluated by means of the injector map at the 2750x12 key-point (-1.3 mg), at the 2000x5 (-0.8 mg) key-point, but also at the 1500x5 and 2500x8 operating conditions (about -0.4/-0.5 mg).

611

#### 612 5.1 Sensitivity analysis

A sensitivity analysis was carried out in order to verify the effect of the main model parameters on the pressure-derived injected quantities. The first three parameters that were considered in this analysis are the values of the K<sub>j</sub> coefficients of the combustion model (see Eq. (26)), the value of the isentropic coefficient  $\gamma$  used in Eq. (11) to derive the experimental  $Q_{net}$  trace from the in-cylinder pressure and the  $Q_{ht,glob}$  parameter used in Eq. (19) to obtain the experimental  $Q_{ch}$  trace, starting from the experimental  $Q_{net}$  trace; finally, the effect of horizontal and vertical shifts in the in-cylinder pressure trace were also investigated.

620 Each parameter was varied separately, keeping the remaining ones unchanged. The analysis



was made for the all the key-points. Reasonable variation ranges were chosen for theparameters considered in the sensitivity analysis.

differences are also reported (b, d, f).



Figure 11 – Effect of a variation in the values of the K<sub>j</sub>, γ and Q<sub>ht,glob</sub> parameters (with
 respect to the nominal values) on the estimated pilot injected mass (a, c, e). The differences are
 also reported (b, d, f).

634 The results are shown in Figs. 10-13. Figures 10-11 report the effect of a variation in the values 635 of the K<sub>j</sub> (Figs. 10a, 11a),  $\gamma$  (Figs. 10c, 11c) and Q<sub>ht,glob</sub> (Figs. 10e, 11e) parameters, with respect to 636 the nominal values, on the estimated total injected quantities (Fig. 10) and pilot injected quantities

637 (Fig. 11). The differences are also reported in Figs. 10b, 10d, 10f and 11b, 11d,11f.

It can be noted that a variation in the values of the  $K_j$  parameter (Figs. 10a,b and 11a,b) does not influence the pressure derived injected quantities to any great extent; a variation in the values of the  $\gamma$  parameter has a remarkable effect on both the pressure-derived total injected quantity (Figs. 10c, d) and on pilot injected quantity (Figs. 11c, d), and the effect increases with the engine load. Finally, it is interesting to note that a variation in the value of  $Q_{ht,glob}$  mainly affects the diagnosed total injected quantities (the deviations are independent of the engine load/speed), but has little influence on the diagnosed pilot injected quantities.

This analysis suggests that is important to accurately select the  $\gamma$  parameter to evaluate the charge net energy  $Q_{net}$  (see Eq. (11)), as it can significantly affect the estimation of the total and pilot injected quantities. Moreover, a good prediction of the heat transfer parameter  $Q_{ht,glob}$  is also required, as it directly affects the estimated total injected quantity. However, the impact of this parameter on the estimated values of the pilot injected quantities can be considered negligible, and this leads to low uncertainty, as will be shown in Section 5.2.

The estimation of the injected quantities could also be affected by an error in the measured in-cylinder pressure trace, which is the main model input. Figures 12-13 report the effect of horizontal and vertical shifts in the in-cylinder pressure trace on the estimated total injected quantities (Fig. 12) and pilot injected quantities (Fig. 13).



estimated total injected mass (a, c). The differences are also reported (b, d).



659 660

Figure 13 – Effect of a horizontal and vertical shift of the measured pressure on the estimated total injected mass (a, c). The differences are also reported (b, d).

662

It can be observed, in Fig. 12, that both horizontal and vertical shifts affect the pressure-derived total injected quantities to a great extent, and that the effect of a horizontal shift increases with engine load, while that of a vertical shift is somewhat constant.

666 With reference to the estimated pilot injected quantities (Fig. 13), it should be noted that 667 a horizontal shift leads to significant deviations, which increase with the engine load, while a 668 vertical shift has little influence.

As a result, attention should be paid to the correct pressure phasing, with respect to the crank angle, and to the correct in-cylinder pressure referencing. The latter may be realized comparing the values of the in-cylinder pressure around BDC with the values measured in the 672 intake manifold by means of an absolute pressure transducer.

673

## 674 5.2 Evaluation of the experimental uncertainty

675 The procedure applied for the evaluation of experimental uncertainties is based on the 676 recommended practices reported in [28] and hereafter summarized.

677 Given an output quantity y, which is dependent on N independent  $x_i$  variables, the associated 678 variance  $u_c^2(y)$  is calculated through the following relation:

679 
$$u_c^2(y) = \sum_{i=1}^N \left(\frac{\partial y}{\partial x_i}\right)^2 u^2(x_i) = \sum_{i=1}^N c_i^2 u^2(x_i) \quad (29)$$

Equation (29) is consistent if the mutual effects between independent variables are neglected. $c_i$ is the "coefficient of sensitivity" of the output quantity y with respect to the *i*-th independent variable  $x_i$ .

Once  $u_c^2(y)$  has been evaluated, it is possible, by assuming a level of confidence (e.g., 95%) and a correspondent coverage factor k (e.g., equal to 2), to evaluate the expanded combined uncertainty of y,  $U_c(y)$ :

686 
$$U_c(y) = k \sqrt{u_c^2(y)}$$
 (30)

687 If the independent variables  $x_i$  are the result of a measurement, the associated variance should 688 be estimated by taking into account the accuracy of the instrument declared in the calibration 689 certificate, the repeatability, the instrument resolution and the master uncertainty.

690 The procedure was used to estimate the uncertainty in the predicted values of the injected pilot 691 mass and of the total injected quantity for the six analyzed key-points.

The main parameters that could affect the estimation of the injected quantities were identified and analyzed in Section 5.1. However, most of those parameters can be excluded from the uncertainty calculation as they are not sources of error. First, it was shown that the K<sub>j</sub> parameters 695 used in the heat release model do not affect the predicted injected quantities to any great extent 696 (see Fig. 10a, 11a). The γ coefficient has remarkable influence on the predicted quantities (see 697 Fig. 10b, 11b), and it should therefore be selected accurately. The correlation developed in [13], 698 as a function of the RAFR parameter, was used in the present study (see Fig. 2 and Eq. (12)). 699 The  $Q_{ht,glob}$  parameter also has a significant effect on the estimated total injected quantities (see 697 Fig. 10c) but has little influence on the estimated pilot injected quantities (see Fig. 11c).

701 Vertical and horizontal shifts in the in-cylinder pressure affect the estimation of  $Q_{net}$  and 702 therefore of the pressure-derived injected quantities (see Fig. 11-12). However, if the 703 in-cylinder pressure is referenced correctly (using, for example, an absolute pressure sensor in 704 the intake manifold) and phased correctly (on the basis of the thermodynamic loss angle, see 705 [29]), this error contribution can be excluded from the uncertainty calculation procedure.

706 The only parameters whose effect on the uncertainty of the method has been considered 707 significant are therefore  $\gamma$  and  $Q_{ht,glob}$ . These parameters were evaluated, in this study, by means of the correlations expressed in Eq. (12) and Eq. (14), and shown in Fig 2 and Fig. 3, 708 709 respectively. There are two main sources of error for each of the two parameters; the first one is 710 related to the uncertainty in the input variables that are used in the correlations (i.e., RAFR in 711 Eq. (12),  $q_{f,inj}$  and n in Eq. (14)). The second source of error is related to the dispersion of the 712 predicted values of  $\gamma$  and  $Q_{ht,glob}$  (through Eq.(12) and Eq. (14), respectively) with respect to 713 the experimental ones; this dispersion was quantified by means of the RMSE parameter, which 714 is equal to 0.006 for the  $\gamma$  parameter and to 18 J for the  $Q_{ht,glob}$  parameter, using the proposed correlations. The first source of error was evaluated for each parameter ( $\gamma$ ,  $Q_{ht,glob}$ ) by applying 715 716 Eq. (29) to the analytical expression of  $\gamma$  and  $Q_{ht,glob}$  expressed by Eqs. (12, 14), and assuming reasonable values for the variance of the RAFR,  $q_{f,inj}$  and n parameters. The second source of 717 718 error was evaluated on the basis of the RMSE values. However, it has been verified that the

first source of error is two orders of magnitude lower than the second one. Moreover, it has also been verified that the contribution of  $\gamma$  is low, compared to that of  $Q_{ht,glob}$ , but it increases with the engine load (as can also be seen in Figs. 10d, 11d).

Table 5 summarizes the expanded combined uncertainty of the pressure-derived pilot and total injected quantities. It can be noted that the uncertainty in the evaluation of the pilot quantity is very small, as its average is of the order of 0.15 mg. The average uncertainty in the evaluation of the total quantity is instead of the order of 0.9 mg, and is larger at higher engine loads. This uncertainty is mainly related to the dispersion of the predicted values of  $Q_{ht,glob}$  by means of Eq. (14), and could be reduced if a more refined method were to be identified to estimate heat transfer.

728

## 729 Table 5 – Evaluation of the expanded uncertainty U of the predicted injected quantities.

Vor noint	Pilot	Pilot+main		
Key-point	mg	mg		
1500x2	0.15	0.85		
1500x5	0.10	0.87		
2000x2	0.15	0.85		
2000x5	0.12	0.86		
2500x8	0.15	0.90		
2750x12	0.24	1.02		

730

#### 731 5.3 Analysis of the cycle-by-cycle variations and considerations on the in-cylinder pressure quality

The proposed methodology has also been tested on single, independent cycles in order to verify its capability of estimating cycle-by-cycle variations in the injected quantities. Moreover, the same analysis has been carried out using the acquired raw pressure, without TAF filtering, in order to verify the effect of signal noise on the outputs. This is an important aspect, as the use of the proposed method in commercial engines would require integrated pressure sensors, which are characterized by a higher signal noise than the Kistler sensor used in the present investigation.



738 Figures 14-15 report the estimated total and pilot injected quantities, respectively, for 10

739 consecutive cycles using both filtered and raw pressure, for the six key-points.



Figure 14 – Estimated total injected quantity for 10 consecutive cycles using both
filtered and raw pressure. The RMS cyclic dispersion (calculated using the filtered
pressure) and the RMS error of the raw pressure-derived quantities with respect to those





747 Figure 15- Estimated pilot injected quantity for 10 consecutive cycles using both filtered 748 and raw pressure traces. The RMS cyclic dispersion (calculated using the filtered pressure) 749 and the RMS error of the raw pressure-derived quantities with respect to those derived from

the filtered pressure are also reported.

751

750

The RMS cyclic dispersion (calculated using the filtered pressure) and the RMS error of the injected quantities obtained from the raw pressure, with respect to those derived from the filtered pressure, are also reported in the graphs.

It can be noted that, in general, the cycle-by-cycle dispersion of both the pilot and totalinjected quantities is very low.

The method is also robust with respect to the use of a raw in-cylinder pressure signal. In this case, the errors depend on the specific engine operating conditions. The largest RMS errors occur at the 2750x12 and 2000x5 key-points, and are of the order of 0.36 and 0.17 mg/cyc, respectively, for the main pulse (see Fig. 14) and of the order of 0.12 and 0.16 mg/cycle, respectively, for the pilot pulses (see Fig. 15).

The use of a raw in-cylinder pressure is therefore suitable for estimating average injected quantities for all the operating conditions, while an accurate evaluation of cycle-by-cycle dispersion in general requires filtering of the pressure trace.

765

## 766 5.4 Analysis of the computational time

The computational time is basically expected to depend on the crank-angle step used for the in-cylinder pressure; this step should be chosen on the basis of a trade-off between the calculation time and the prediction accuracy.

An analysis has therefore been carried out in order to evaluate the impact of the crank-angle step on the computational time and on the prediction accuracy of the injected fuel quantities.

773 The elaboration was performed with the Labwindows CVI software, using a Pentium-D

774 PC.

The effect of the crank-angle step on the average computational time necessary to run a single operating condition is reported in Tab. 6.

777

778

Table 6 - Effect of the crank angle step on the computational time.

Calculation	0.1	0.5	1.0	2
step [deg]				
Computational				
time: [ms]	3.4	0.9	0.6	0.3

Figure 16, instead, shows the effect of the crank angle step used for integration on theestimated total and pilot injected quantities.



Figure 16 – Effect of the crank angle step used for integration on the estimated total and
 pilot injected quantities. The differences with respect to the case at 0.1 deg are also reported

## (b, d).

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The analysis shows that the procedure is very robust with respect to the crank angle step used for integration. It should be specified that the net energy of the charge was calculated according to Eq. (11), using the trapezoidal rule for integration; it has been verified that this integration rule provides a nearly-independent trend of the net energy with respect to the crank angle step used for integration. The computational time that is required for a crank angle step of 2 deg is of the order of 0.3 ms. Therefore, the method can be considered suitable for control-oriented applications.

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## 795 **6. FUTURE WORK**

In addition to the injected quantities, the proposed methodology also has the potential of estimating the injection rate curve on the basis of the measured in-cylinder pressure. This can be done by deriving Eq. (28) with respect to time.

An example is reported in Fig. 17 for the 1500x5 and 2000x5 key-points.

800 In particular, Fig. 17 reports a comparison between the injection rate profiles obtained 801 with the measured in-cylinder pressure (black lines) and the experimental ones (dotted red 802 lines).

803

- 805



Figure 17 – Comparison between the pressure-derived injection rate profiles and the
 experimental ones for the 1500x5 and 2000x5 key-points.

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810 It should be noted that the evaluation of the shape of the injection rate is quite good for the 811 1500x5 key-point, while it shows a double-peak characteristic for the main injection of the 2000x5 812 key-point. The accuracy of the prediction of the injection rate depends exclusively on the capability 813 of the heat release model to correctly reproduce the HRR shape on the basis of the injection rate 814 profile. It can therefore be observed, in Fig. 17, that the accumulated fuel mass approach (i.e., Eq. 815 (1)) provides good results if the HRR shape is mainly of the premixed type, but is less accurate if 816 the HRR shape is of the premixed-mixing controlled type (see Fig. 8). The result is that the 817 injection rate shape derived from the in-cylinder pressure reflects the premixed-mixing controlled 818 characteristics of the HRR (see Fig. 17b). A refinement in the prediction of the injection rate shape 819 could be realized by refining the heat release model, using, for example, a crank-angle dependent 820 value of the K<sub>i</sub> parameters of Eq. (1). This investigation is currently ongoing.

821 Moreover, the applicability of the proposed method to highly premixed combustion modes, 822 such as PCCI (Premixed Charge Compression Ignition) should also be verified.

#### 824 **7. CONCLUSION**

A new methodology has been developed to estimate injected fuel quantities on the basis of the measured in-cylinder pressure in multiple injection DI diesel engines.

The method is based on the inversion of a predictive combustion model that was previously developed by the authors, and which is based on the accumulated fuel mass approach.

830 A sensitivity analysis has been carried out in order to verify the effects of variations in the 831 main input parameters on the calculated injected quantities. It has been found that errors in the 832 estimation of the global heat transfer of  $\pm 50J$  can lead to errors in the estimation of the total 833 injected quantities of about 1 mg/cyc, but the effect on the estimated pilot quantity error is 834 much smaller (below 0.2 mg/cyc). Moreover, errors in the estimation of the isentropic 835 coefficient  $\gamma$  of  $\pm 0.05$  lead to large errors for both the estimated total injected quantity (of up to 2 mg/cyc) and for the estimated pilot quantity (of up to 1 mg/cycle). It has also been verified 836 837 that horizontal shift errors on the in-cylinder pressure of  $\pm 0.5$  deg lead to errors in the 838 estimated total and pilot quantities of up to 1 mg/cyc and 0.5 mg/cyc, respectively, and vertical 839 shift errors on the in-cylinder pressure of  $\pm 0.5$  bar lead to errors in the estimated total quantity 840 of up to 0.8 mg/cyc, while the impact on the pilot quantity estimation error is negligible.

An uncertainty analysis has been carried out on the basis of the previous results. It has been verified that the proposed method is robust when used to estimate the injected quantity of the pilot pulses, as the average uncertainty is of the order of 0.15 mg. The average uncertainty in the prediction of the total injected quantity is of the order of 0.9 mg, and depends mainly on the accuracy of the evaluation of the heat transfer of the charge with the walls.

It has also been shown that the method is suitable for evaluating cycle-by-cycle variations of the injected quantities, provided that the input pressure is filtered. However, it has been verified that the use of the raw in-cylinder pressure allows the average injected quantities to be 849 estimated.

Finally, the approach is characterized by a very low computational time, that is, of the order of 0.3 ms, when the elaboration is run on a PC and a crank angle integration step of 2 deg is used, and is therefore suitable for control-oriented applications.

853

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856

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