

# A novel fuel injected mass feedback-control for single and multiple injections in direct injection systems for CI engines

Ferrari A.<sup>(\*,a)</sup>, Novara C.<sup>(b)</sup>, Vento O.<sup>(a)</sup>, Violante M.<sup>(c)</sup>, Zhang T.<sup>(d)</sup>

<sup>a</sup> Energy Department, Politecnico di Torino, Turin, Italy

<sup>b</sup> Electronics and Telecommunications Department, Politecnico di Torino, Turin, Italy

<sup>c</sup> Control and Computer Engineering Department, Politecnico di Torino, Turin, Italy

<sup>d</sup> College of Mechanical and Vehicle Engineering, Hunan University, Changsha, China

(\*) Corresponding author. Email: [alessandro.ferrari@polito.it](mailto:alessandro.ferrari@polito.it) Phone +390110904426

## Abstract

A new feedback-control capable to enhance the fuel injected quantity accuracy has been proposed and tested. The experimental pressure in the rail and that measured along the rail-to-injector pipe, in the vicinity of the injector inlet, have been used as input data to a home-made hydraulic model. By means of this model the pressure downstream of the gauged orifice at the interface between the rail and the rail-to-injector pipe is determined; the mass at the injector inlet can be obtained by means of an integration of the estimated flow-rate entering the injector. A robust mathematical law can be established between the mass that enters the injector and the one injected, thus, a feedback-control based on the error between the target injected quantity stored in the ECU and the predicted injected mass has been designed and implemented for the *ET* correction. The new feedback-strategy has been applied to control both single and double (pilot-main) schedules by using a rapid prototyping hardware. Regarding single shots, the new control results to be capable of reducing the injected mass inaccuracy, which is due to the different thermal regimes experienced by the injector, below 0.6 mg (the standard open loop control can feature an error up to 2 mg when the fuel tank temperature is varied), while for the pilot-main schedules it is possible to dramatically reduce the inaccuracy on the desired overall injected quantity (below 1 mg) when digital or continuous rate shaping programs are implemented.

## 28 **Keywords**

29 Injected mass control; fuel injection system; compression ignition engine, rapid prototyping  
30 hardware; renewable diesel oil.

## 31 **Highlights**

- 32 - A novel feedback-control strategy for the injected quantity is built up.
- 33 - A rapid prototyping hardware is used to test the developed control.
- 34 - Single and multiple injection performance of the prototypal hardware is analyzed.

## 35 **Introduction**

36 In the last years, environmental protection agencies have concentrated on internal combustion engine  
37 (ICE) emissions, due to their effect on environment and health [1]. For this reason, researchers have  
38 been focused on this topic, looking for solutions that are capable to reduce the ICE impact on  
39 environment and living beings [2, 3].

40 Nowadays, diesel engines are still playing a key-role as a power source [4] and numerous studies have  
41 been developed regarding this category of engines, in which almost all the aspects are taken into  
42 account. The shape of the combustion chamber, influencing the fuel-air mixing, can affect the soot  
43 formation and, therefore, the smoke emissions [5]. It has been demonstrated that the consistency of  
44 the engine assembly influences the NO<sub>x</sub> emissions, for example due to a varied combustion  
45 temperature leaded by the different nozzle extension height [6]. A valid approach in the reduction of  
46 pollutant emissions should also consider the refinement of the after-treatment devices such as the  
47 diesel oxidation catalyst, the diesel particulate filter, lean NO<sub>x</sub> trap, and selective catalytic reduction  
48 [7].

49 The Common Rail (CR) fuel injection system has added available parameters to reach an optimum  
50 engine calibration for the widest range of working conditions, thanks to its increasing value of  
51 maximum injection pressure, that can go beyond 3000 bar [8], the elevated number of injection events

52 for combustion cycle and the flexibility in injection timings [9]. The split ratio of a triple injection  
53 strategy has been analysed to determine its effect on the ignition delay, unburned emissions, and  
54 combustion noise (CN) [10]. For a HCCI engine operating with a multiple fuel shot strategy, if the  
55 first injection pulse duration is increased, the diffusion phase combustion will be minimized, reducing  
56 the heat release rate, thereby diminishing  $\text{NO}_x$  and soot [11].

57 Concerning to injected flow-rate pattern, it has been shown that for low and medium loads, the  
58 optimum engine calibration should feature triple or quadruple injections together with possible rate  
59 shaping schedules, leading to an enhancement in the soot- $\text{NO}_x$  trade-off and a reduction in the CN  
60 [12]. In particular, the digital-rate shaping strategy (the two injections are very close, but can still be  
61 distinguished from one another) can improve the overall engine brake specific fuel consumption and a  
62 consequent reduction in the  $\text{CO}_2$  emissions is achieved [13]. These complex injection strategies can be  
63 implemented with either a solenoid or piezoelectric injectors [17].

64 However, if the dwell time (DT) is dramatically reduced to adopt strategies such as the  
65 aforementioned digital-rate shaping or the continuous rate shaping (the two injection events are  
66 merged together), significant cyclic variation in the engine torque occurs, due to injector-to-injector  
67 and cyclic differences regarding the rates of injections, thus representing a potential issue for the  
68 engine implementation [14]. In fact, when the electric DT goes below a particular value, the overall  
69 fuel injected quantity generally features anomalous increasing, hence a small change in the hydraulic  
70 dwell time can lead to an important influence on the fuel injected mass [15, 16].

71 This context justifies the efforts on the development of robust feedback-control strategies for the fuel  
72 injected quantity, which is usually treated by means of an open-loop approach. Indeed, in standards  
73 common rail system, the injected mass is predicted, based on the duration of the electrical current  
74 signal ( $ET$ ) and the nominal rail pressure ( $p_{nom}$ ) [18]: these two values are stored onto the ECU  
75 calibration maps, defined by means of preliminary experimental campaigns and there is no feedback  
76 control on the effective injected mass [19].

77 An estimation technique of the fuel injected mass has been obtained by measuring the in-cylinder  
78 pressure signal [20]. An already validated predictive combustion model has been applied contrariwise.  
79 The overall heat transferred by the charge to the walls over the combustion period is estimated and,  
80 with the acquired in-cylinder pressure, the released net energy is derived. Finally, an estimation of the  
81 injected fuel mass is obtained by calculating the chemical energy of the injected fuel.

82 The feedback strategy proposed in [21] predicts the pilot mass estimation by means of the pilot  
83 misfire ratio. Misfire occurs when the pilot injected quantity is not able to burn and, consequently, an  
84 increased premixed peak of the main injection combustion takes place. Since the misfire probability is  
85 well correlated with the pilot mass [22], the pilot injected mass can be predicted, after a preliminary  
86 characterization of the misfire events, using an in-cylinder pressure transducer applied to detect the  
87 misfire. In [21], the correlation between the pilot injected mass and the pilot misfire ratio is used  
88 together with a Bayesian approach.

89 Two robust feedback-control strategies have been developed for both single and pilot-main shots, by  
90 means of the measurement of the flow-rate entering the injector [23, 24]. It has been shown that the  
91 mass entering the injector, obtained through an integration of the measured flow-rate, correlates well  
92 with the effective injected quantity. Hence, a feedback-control to modify the energizing time (*ET*) to  
93 the injector pilot stage has been established, and the current signal is modified based on the error  
94 between the predicted injected mass and the target one.

95 Another technique to estimate the mass entering the injector is based on an artificial neural network  
96 [25]. In this case, the input signal is represented by the pressure time distribution measured in the  
97 vicinity of the injector inlet.

98 A prediction of the injection temporal length (*ITL*) has been obtained by means of an approach  
99 involving the time-frequency analysis [26]. The short-time Fourier transform is applied to the pressure  
100 trace measured in the proximity of injector inlet, then the mean instantaneous frequency (*MIF*) trace  
101 is determined and is used to develop a needle-lift virtual sensor. The injected mass can be estimated

102 by means of the *ITL* prediction, that is given by the distance between two local maximums of the *MIF*  
103 trace.

104 Compensative strategies for the injected mass have been introduced by injection systems  
105 manufacturers, which have produced superior injectors featuring additional devices to mitigate the  
106 inaccuracy of the fuel injected quantity. Bosch developed an injector equipped with a force transducer  
107 (labelled as Needle Closing Sensors), that acquires the pressure force induced by the fuel inside the  
108 control chamber. Some noteworthy instants regarding the needle motion can be recognized by means  
109 of this force time history and they are used as input data to an algorithm which estimates the injected  
110 fuel quantity [27]. The i-Art technology designed by DENSO is able to measure the pressure in the  
111 control chamber by means of a piezoelectric pressure sensor installed to the injector pilot stage. This  
112 pressure signal is used to predict the injected mass with a complex transfer function [28, 29]. Finally,  
113 Delphy has proposed the “Switch” technology. The non-ballistic needle is inserted inside an electric  
114 circuit where the voltage is detected. When the needle closes the nozzle, or it reaches its upper stroke  
115 end the circuit closes in two distinct manners. Therefore, if the voltage is detected, the two  
116 corresponding time instants can be identified, and the injected quantity can be estimated by means of  
117 a needle lift-based submodel [30].

118 In this work, an innovative feedback-control strategy to reduce the inaccuracy of the fuel injected  
119 quantity is presented and tested. The proposed algorithm receives as input datum the measurement of  
120 the pressure traces inside the rail and along the pipe that feeds the injector, in the vicinity of the  
121 injector inlet. The mass which enters the injector can be estimated based on these pressure signals  
122 and, by means of a correlation, the predicted injected mass is obtained. The new feedback-control has  
123 been implemented in a rapid prototyping hardware. Different fuel tank temperatures have been taken  
124 into account to analyse single injections, while for pilot-main shots the total injected mass has been  
125 measured in the short *DT* range. The new strategy represents an evolution of that developed in [23,  
126 24] because the installation of only one additional pressure transducer is required in the commercial  
127 *CR* injection system layout.

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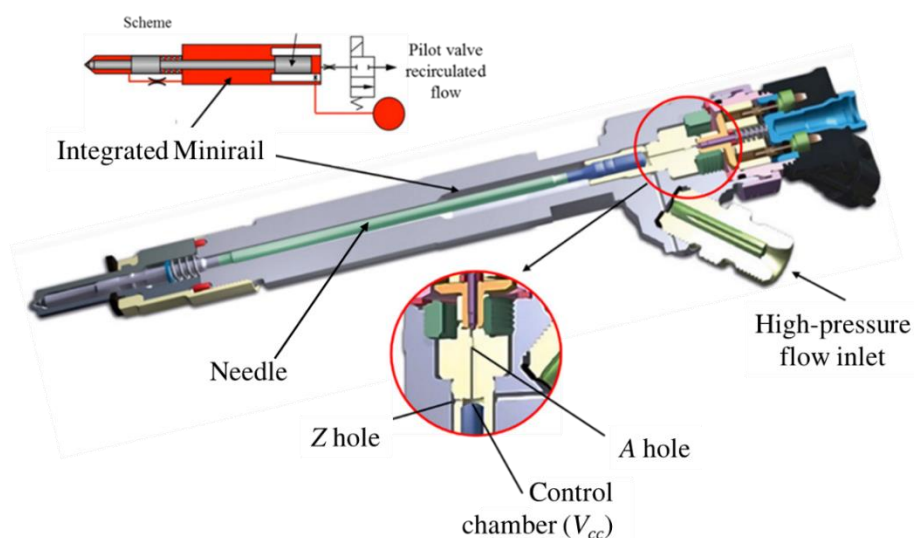
129 **Experimental facility**

130 The hydraulic tests have been performed at the Politecnico di Torino ICE Laboratory, at a Moehwald-  
131 Bosch hydraulic test bench (nominal power: 35 kW, maximum speed: 6100 rpm, maximum torque:  
132 100 Nm). Different thermocouples and piezoresistive pressure transducers are used to monitor the  
133 temperature and the pressure levels in various locations of the analysed high-pressure hydraulic  
134 circuit. A HDA flowmeter [31] measures the injected flow-rate traces that are integrated to obtain the  
135 corresponding masses, while injector fuel leakages are evaluated by means of a KMM flowmeter.

136 The adopted calibration fluid at the test rig is a Shell V-Oil 1404 (ISO 4113), since it is capable to  
137 satisfactorily reproduce the renewable diesel oil properties over a proper range of pressure and  
138 temperature values (at least up to 120°C).

139 The selected injection system for the experimental campaign is a last generation Bosch CR system.  
140 The rail (with a capacitance of 10 cm<sup>3</sup>) is fed by a high-pressure pump featuring a displacement of  
141 430 mm<sup>3</sup> with a double effect single piston and a transmission ratio with the engine equals to 1:1.  
142 Four CRI 2.20 (solenoid-actuated) injectors are connected to the rail by means of high-pressure pipes  
143 with a length of 320 mm and an internal diameter of 2.8 mm.”

144 The injector pilot stages feature a pressure-balanced pilot-valve. Moreover, they present an integrated  
145 Minirail. Figure 1 presents the scheme of the tested injector.

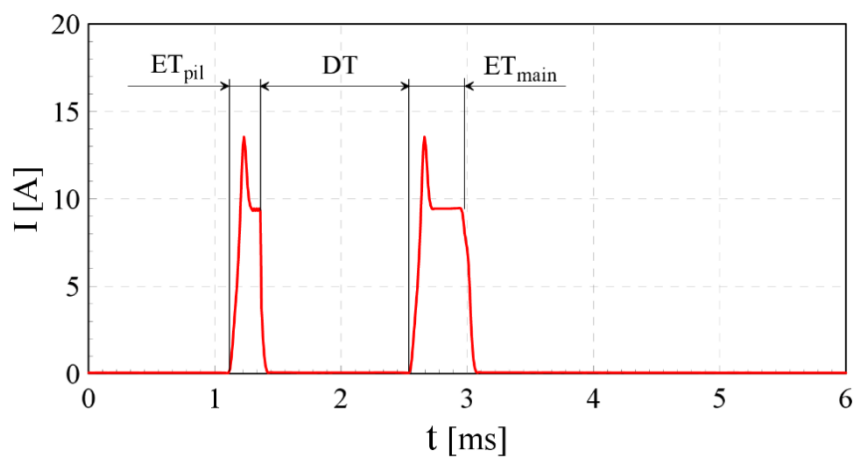


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147

Figure 1. Injector CRI 2.20.

148 Concerning the injector working principle, a part of fuel entering the injector from the injector-  
149 feeding pipe fills, through the Z hole, the control chamber (volume  $V_{cc}$ ), and the other part reaches the  
150 integrated Minirail. The injection starts after the electronic control unit (ECU) has switched on the  
151 electrical current  $I$  to the pilot valve solenoid. Consequently, the control chamber is discharged  
152 through the A hole and the needle moves up opening the nozzle. The control chamber pressure starts  
153 again to rise at the end of the current signal provided by the  $ECU$ , and the needle closure phase is  
154 induced. The hydraulic end of the injection is achieved when the needle closes the nozzle reaching its  
155 downstroke end [32].



156

157 *Figure 2. Example of a pilot-main injection current signal.*

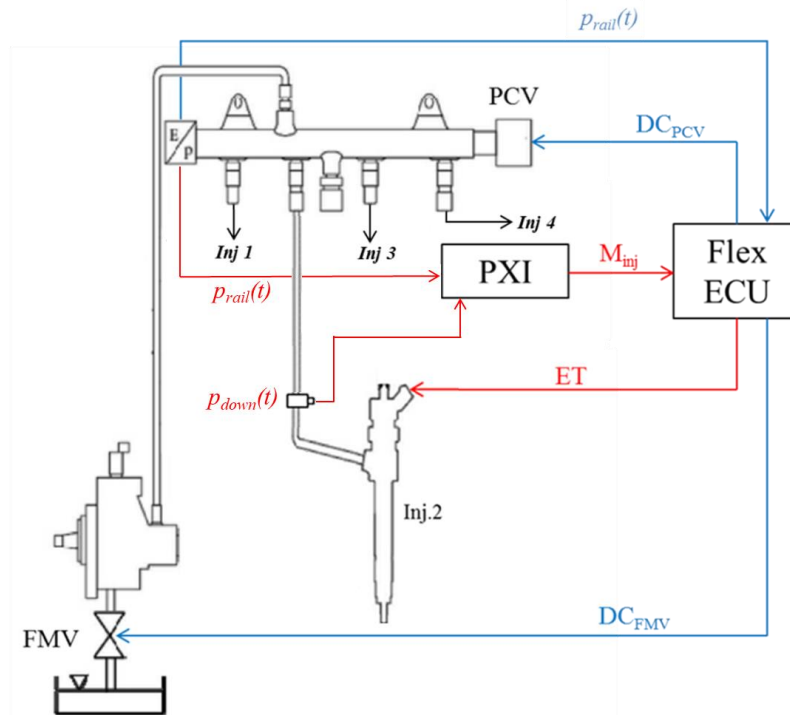
158 Both single and pilot-main injections have been taken into account and the considered nominal rail  
159 pressure levels ( $p_{nom}$ ) varied in the range 500-1800 bar. Tests on single injections have been  
160 performed by considering different  $ET$  values. Regarding the pilot-main injections, different  
161 combination of  $ET_{pil}$ ,  $ET_{main}$  and  $DT$  have been selected, and their meaning is shown in Fig. 2.  
162 Pertaining to pilot-main injections, different rail pressure levels have been selected and, for each of  
163 them, the same main and pilot injected quantities have been chosen: 15 mg, 20 mg or 30 mg for the  
164 main injection and 1.5 mg or 3 mg for the pilot injection. The dwell time has been varied from 100  $\mu$ s  
165 to 500  $\mu$ s.

166 The selected pump speed for the entire experimental campaign was 2000 rpm, corresponding to 2000  
167 rpm of engine speed.

168

169 **Algorithm of the feedback-control and rapid prototyping hardware**

170 Figure 3 shows the injection system layout. A pressure sensor mounted at one rail extremity is used to  
 171 monitor the rail pressure time history  $p_{rail}(t)$ . The nominal rail pressure control strategy and the one  
 172 pertaining to the modulation of the  $ET$  to the injector have been developed on Matlab Simulink.



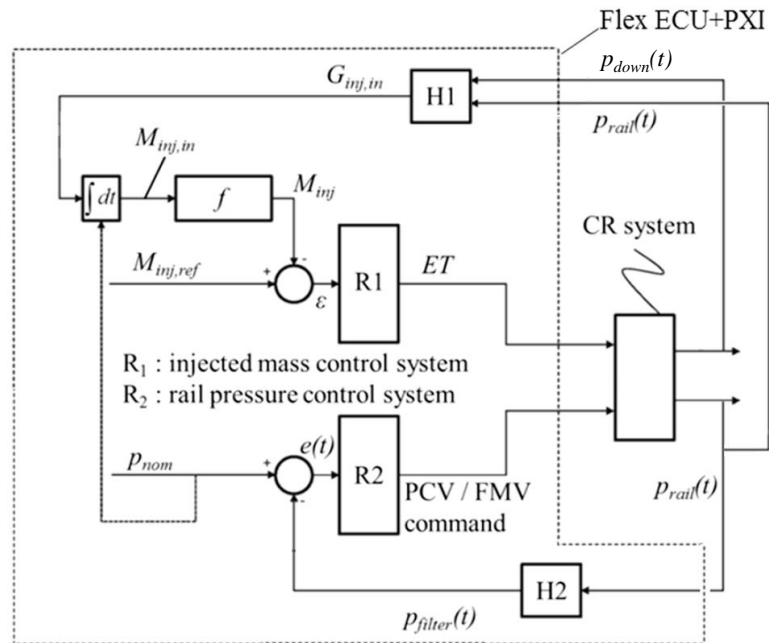
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174 *Figure 3. The layout of the tested injection system (blue refers to  $p_{rail}$  control system and red to  $M_{inj}$  control system).*

175 Opposite the rail pressure sensor, the pressure control valve ( $PCV$ ) is installed (cf. Fig 3). This valve  
 176 is employed to control and maintain the nominal rail pressure  $p_{nom}$  target (provided by the  $ECU$ ). A  
 177 closed-loop control is set up by monitoring  $p_{rail}(t)$ , throttling the excess of the pumped fuel. The rail  
 178 pressure can also be controlled through the fuel-metering valve ( $FMV$ ) that is installed at the pump  
 179 inlet. By acting on the  $FMV$ , the flow-rate entering the pump is modified based on the injector  
 180 requirements and the flow-rate throttled by the  $PCV$  is ideally null [33].

181 A scheme of the implemented injection system control strategies is reported in Fig. 4. The rail  
 182 pressure signal is acquired with a frequency of 200 Hz by the  $ECU$ , giving the filtered signal  $p_{filter}(t)$   
 183 (block H2). The difference  $e(t)$  between  $p_{nom}$  and  $p_{filter}(t)$  is computed and is received as an input by  
 184 block R2, where the  $PID$  controller for the rail pressure is usually implemented in commercial

185 systems. This controller generates the duty cycle value to be provided to the valve selected for the  
 186 pressure control (FMV or PCV).



187

188

Figure 4. The injection system control strategies.



189

190

Figure 5. Scheme of the hydraulic model.

191 The new control strategy shown in Fig. 4 consists in the injected mass feedback-control and the  
 192 installed hardware has been simplified compared to that installed in [23, 24]. The measured pressure  
 193 signals  $p_{down}(t)$  (acquired near the injector inlet, cf. Fig. 3) and  $p_{rail}(t)$  are sufficient to determine the  
 194 mass flow-rate time history entering the injector. Unlike the prototype in [23, 24], as is shown in Fig.  
 195 3, it is therefore enough to install only one pressure sensor along the injector-feeding pipe.

196 In block H1 (cf. Fig. 4), which receives  $p_{rail}(t)$  and  $p_{down}(t)$  as input (acquired with a frequency of 30  
 197 kHz, that is much higher than the standard sampling frequency used in the H2 block for the rail  
 198 pressure control), a simple hydraulic model is implemented. In this model, represented in Fig. 5, the  
 199 rail is assumed as a zero-dimensional chamber at which the measured pressure  $p_{rail}(t)$  is imposed as a

200 boundary condition. The rail is connected to the injector-feeding pipe by means of a gauged orifice  
 201 (its used in the commercial layout as a passive damper for the free pressure waves triggered by the  
 202 nozzle closure [34]), characterized by a restricted area  $A_{res}$  (obtained by measuring the gauged orifice  
 203 diameter, equals to  $d_{res}=1$  mm) and a flow coefficient  $C_d=0.8$  (this value has been fitted on flow-rates  
 204 values at the injector inlet obtained by applying other techniques presented in [23, 24]). Close to the  
 205 downstream extremity of the high-pressure pipe with cross-section area  $A$ , the measured pressure time  
 206 history  $p_{down}(t)$  is assigned as the second boundary condition at a distance  $L$  from the orifice outlet.  
 207 The pressure  $p_{up}(t)$  represents the pressure trace measured at the inlet of the injector-feeding pipe (cf.  
 208 Fig 5) and is an unknown quantity. The instantaneous flow-rate  $G_{or}$  flowing through the gauged  
 209 orifice can be expressed as:

$$210 \quad \begin{cases} G_{or} = -C_d \cdot A_{res} \cdot \sqrt{2 \cdot (p_{up} - p_{rail})\rho} & \text{if } p_{up} > p_{rail} \\ G_{or} = C_d \cdot A_{res} \cdot \sqrt{2 \cdot (p_{rail} - p_{up})\rho} & \text{if } p_{rail} > p_{up} \end{cases} \quad (1)$$

211 where  $\rho$  represents the flow density.

212 The continuity equation and the momentum balance equation for the piece of pipe with length  $L$  have  
 213 been combined. By neglecting the wall shear stress and with the assumption of 1D incompressible  
 214 flow, one obtains

$$215 \quad \frac{\partial u}{\partial t} + \frac{1}{\rho} \frac{\partial p}{\partial x} = 0 \quad (2)$$

216 where  $u$  and  $p$  stand for the 1D velocity and pressure, respectively, and  $x$  represents the spatial  
 217 coordinate along the pipe axis.

218 Multiplying Eq. (2) by  $\rho A$  ( $\rho$  is the flow density), integrating over the distance  $L$  and dividing by the  
 219 same distance  $L$ , on has:

$$220 \quad \frac{d\bar{G}}{dt} = \frac{A}{L} (p_{up} - p_{down}) \quad (3)$$

221 where  $\bar{G}$  represents the space-averaged instantaneous flow-rate. By integrating Eq. (3) with respect to  
 222 the time, one obtains:

$$223 \quad \bar{G} = \int_0^t \frac{A}{L} \cdot (p_{up} - p_{down}) dt \quad (4)$$

224 It can be assumed that  $G_{or} \approx \bar{G}$  [35]. Eqs. (1) and (4) can therefore be solved together and the two  
 225 unknown variables, namely the pressure  $p_{up}$  and the flow-rate  $G_{or}$  traces, can be evaluated.

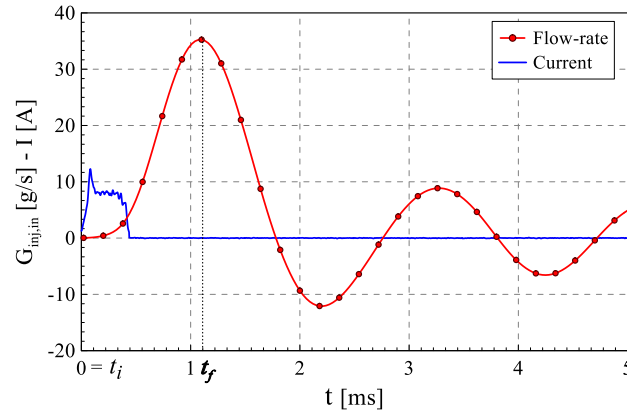
226 The flow-rate that reaches the injector is evaluated by means of [23, 24]:

$$227 \quad G_{inj,in} = \frac{A}{L} \int_0^t \Delta p dt - \frac{A}{L} \langle \Delta p \rangle \cdot t \quad (5)$$

228 where  $\Delta p = p_{up} - p_{down}$  and  $\langle \Delta p \rangle$  stands for the  $\Delta p$  time-averaged value for the considered  
 229 injection cycle. This flow-rate represents the output of block H1 (cf. Fig. 4) and by integrating it over  
 230 time instants  $t_i$  and  $t_f$ , the injector inlet mass  $M_{inj,in}$  is obtained:

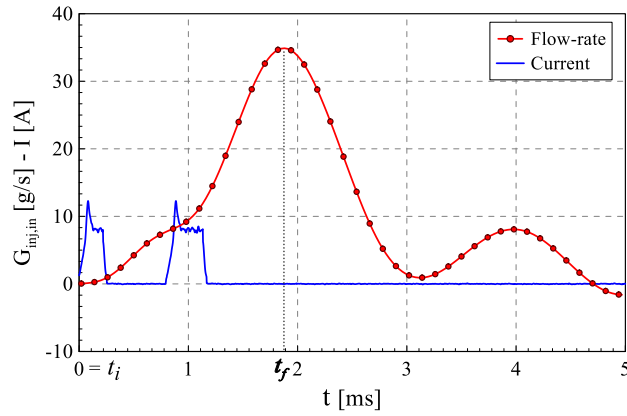
$$231 \quad M_{inj,in} = \int_{t_i}^{t_f} G_{inj,in} dt \quad (6)$$

232 In Fig. 6, two different flow-rates for single and double injection are represented. The integration  
 233 performed in Eq. 6 to obtain  $M_{inj,in}$  starts in correspondence of the time instant at which the electrical  
 234 current begins to rise (in Fig. 6 one has  $t_i = 0$ ), and it stops at the time instant where the  $G_{inj,in}(t)$  trace  
 235 features its absolute maximum [23, 24]. The negative flow-rates that can be inferred in Fig. 6 are due  
 236 to the pressure waves travelling back and forth along the pipe that feeds the injector, which are caused  
 237 by the nozzle closure. These physical contributions are not taken into account since they occur after  $t_f$ .



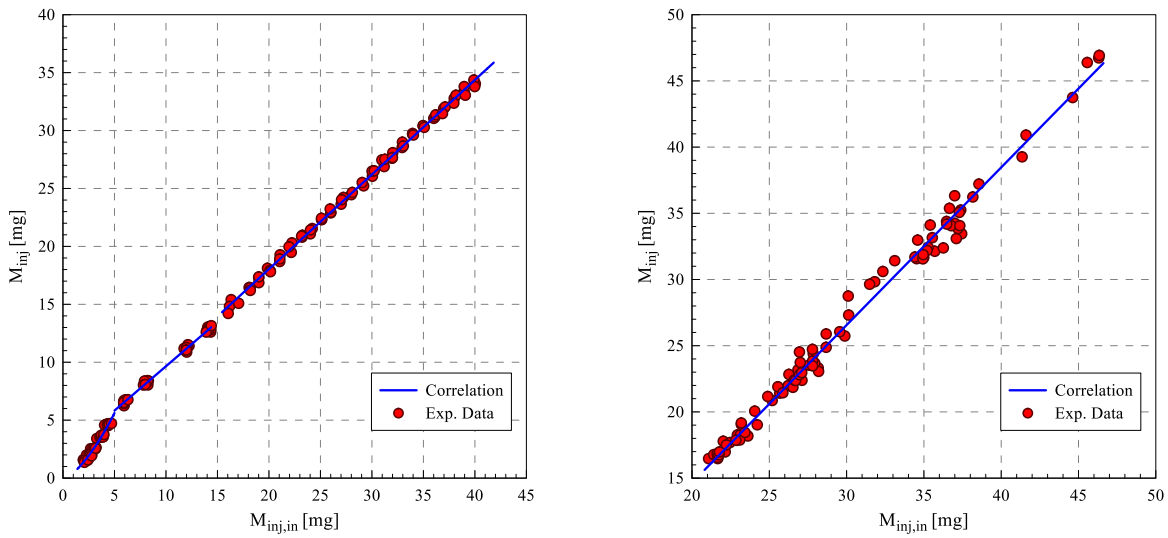
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(a): Single injection ( $p_{nom} = 1200$  bar,  $M_{inj,ref} = 20$  mg).



(b): Pilot-main injection ( $p_{nom} = 1600$  bar,  $M_{inj,pil} = 3$  mg,  $M_{inj,main} = 30$  mg,  $DT = 500$   $\mu$ s)

Figure 6. Flow-rates time histories entering the injector for a single (a) and a pilot-main (b) injection.



(a): Single injections.

(b): Pilot-main injections

Figure 7. Mathematical laws to determine the injected fuel quantity from the injector inlet one.

Figure 7 shows that the fuel injected quantity ( $M_{inj}$ ) correlates well with the fuel amount that enters the injector ( $M_{inj,in}$ ). This is verified for both single (cf. Fig. 7a) and pilot-main injections (cf. Fig. 7b), where the  $M_{inj}$  data refer to the overall injected quantities during the different injection events). Mathematical laws, namely  $M_{inj}=f(M_{inj,in})$ , reported in Fig. 7 have been obtained by means of a fitting of the experimental points (126 for single injections and 90 for pilot-main schedules, both referring to different rail pressure levels and injected masses). Thanks to these correlations the measurement of  $M_{inj,in}$  allows the estimation of  $M_{inj}$  during the real system operation. Furthermore, the presented correlations result to be independent with respect to the fuel tank temperature ( $T_{tank}$ ) [23, 24]. Quantity  $\varepsilon$  is calculated by subtracting the prediction of the injected amount  $M_{inj}$  to the desired target value ( $M_{inj,ref}$ ), stored in the ECU, This difference reaches, as an input, block R1 (cf. Fig. 4), which contains

256 a *PID* controller that calculates the corrected *ET* that can mitigate the inaccuracy of the fuel injected  
257 quantity for the next injection event. In Fig. 3, the hardware components employed for the new  
258 control strategy are represented. A National Instrument PXI platform has been used to receive the  
259 signals from the sensors of  $p_{rail}$  and  $p_{down}$  and to implement Eqs. (1), (4)-(6) and the mathematical laws  
260 given by  $M_{inj}=f(M_{inj,in})$ . A CAN interface cable is used to deliver the injected mass prediction from the  
261 PXI platform to the ETAS Flexible ECU, which generates the energizing time to the injectors.  
262 A *ECU* software is usually split into two parts: the first, generally provided by the injection system  
263 manufacturer, is the basic software, which contains all the basic information (e.g., the instructions to  
264 connect the different pins with the injection system actuators) and the second, called application  
265 software, contains the control strategies, such as the pressure and the *ET* ones. ETAS EHOOKS  
266 software is used to merge the two aforementioned software and to provide a single software that is  
267 finally flashed in the Flexible ECU.

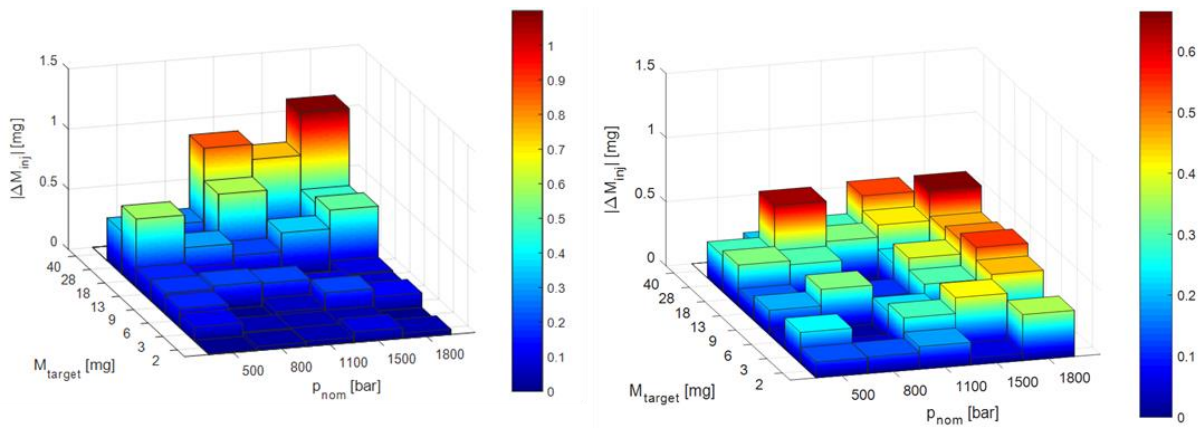
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## 269 **Results and discussion**

270 For a selected working condition, in terms of different values of nominal pressure and target  
271 injected mass, the HDA flowmeter has measured the effectively injected fuel quantity ( $M_{inj}$ )  
272 averaged over 100 consecutive engine cycles. Two different fuel tank temperatures have been  
273 selected: 40°C, which is usually selected as reference value from the injection system  
274 suppliers, and 68°C, that represents the maximum temperature available in the test bench, for  
275 safety reasons.

276 To assess the injected mass accuracy of the fuel injection system, one can introduce the  
277 accuracy  $|\Delta M_{inj}|$ , given by the absolute value of the difference between the effective-injected  
278 mass and the target value. This accuracy is represented along the vertical axis of the 3D  
279 diagrams in Figs. 8a, 8b and 8c with respect to both  $p_{nom}$  and  $M_{inj,ref}$  for single injections. In  
280 particular, Figs. 8a, 8b and 8c all refer to the fuel tank temperature  $T_{tank}= 40^\circ\text{C}$  and they plot  
281 data for the standard system with the standard open-loop technique used to control the

282 injected mass (cf. Fig. 8a), for the previously developed [23] system featuring the closed-loop  
 283 strategy in which two pressure sensors are involved (cf. Fig. 8b), and for the new system,  
 284 where the closed-loop control requires only one pressure signal measured along the injector-  
 285 feeding pipe (cf. Fig. 8c). For all the systems, the pressure is *FMV* controlled based on the  
 286 difference between  $p_{nom}$  and  $p_{filt}$ . In Fig. 8, the system with one pressure sensor features  
 287 similar performance as that of the layout with two additional pressure sensors and its  
 288 accuracy, which is generally lower than 0.6 mg, slightly improves that of the standard  
 289 apparatus.

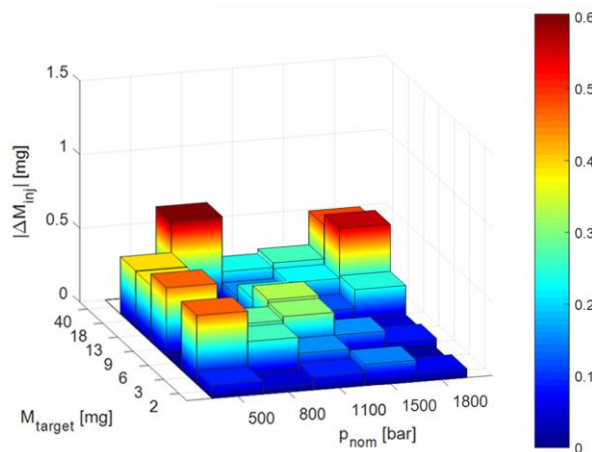


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291

(a): Standard control strategy.

(b): Control strategy with two pressure sensors.



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(c): Control strategy with one pressure sensor.

294

Figure 8. Control strategies performance at  $T_{tank} = 40^{\circ}\text{C}$ .

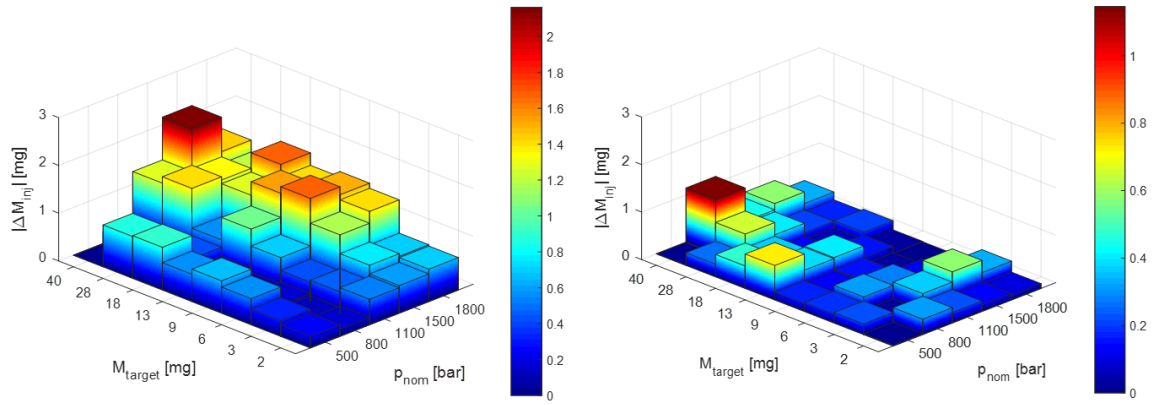
295 The advantages of the feedback-control strategy for the injected quantity become obvious as

296  $T_{tank}$  increases, as reported in Fig. 9, where  $T_{tank}=68^{\circ}\text{C}$ . In this case, the standard injection

297 system (open-loop control for the injected mass) features an inaccuracy of the injected mass  
298 of around 1.5-2.2 mg for significant target mass and high  $p_{nom}$  level, as shown in Fig. 9a,  
299 while when the new closed-loop control acts the error is below 0.9 mg within the entire range  
300 of working condition (cf. Fig. 9c). Furthermore, the performance is in line with that in Fig.  
301 9b, where the results of the two additional pressure sensors technique are represented. The  
302 fuel temperature affects the injector internal dynamic and modifies the injected quantity for a  
303 certain set of  $ET$  and  $p_{nom}$ , since the  $ET$ -based injector characteristic is temperature dependent  
304 [26]. This is evident from the comparison of Figs. 8a and 9a: the standard injection system  
305 has the best performance when the  $T_{tank}$  value is set to the one at which the standard open-  
306 loop control strategy has been tuned, while the accuracy worsens when the fuel temperature  
307 augments.

308 Referring to the new control strategy (cf. Figs. 8c and 9c), the inaccuracy is mainly related to  
309 the error in the injected mass prediction given by the  $M_{inj,in}-M_{inj}$  correlation (cf. Fig. 7a,  
310 where all the experimental data are inside a band of  $\pm 1$  mg of error), while the augment of  
311 injected mass lead by the increased fuel temperature can be efficiently contrasted.

312 The normalized standard deviation pertaining to the fuel injected quantity is plotted in the  
313 vertical axis of the 3D diagram in Fig. 10 ( $\sigma$  is normalized with respect to the average value  
314 of the injected mass and is expressed as a percentage) is reported for the three different  
315 control strategies. The standard deviation has been evaluated by means of the HDA over 100

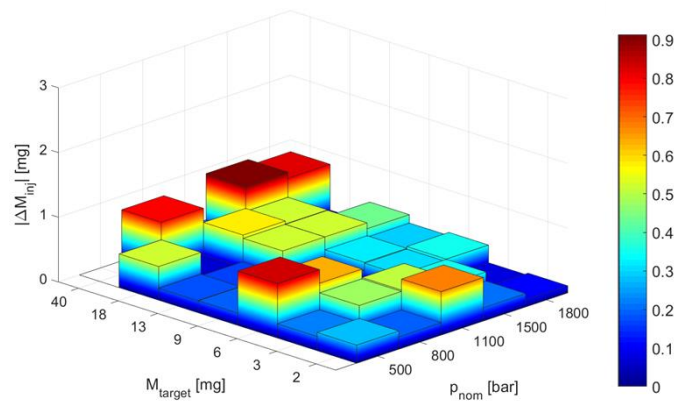


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(a): Standard control strategy.

(b): Control strategy with two pressure sensors.



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(c): Control strategy with one pressure sensor.

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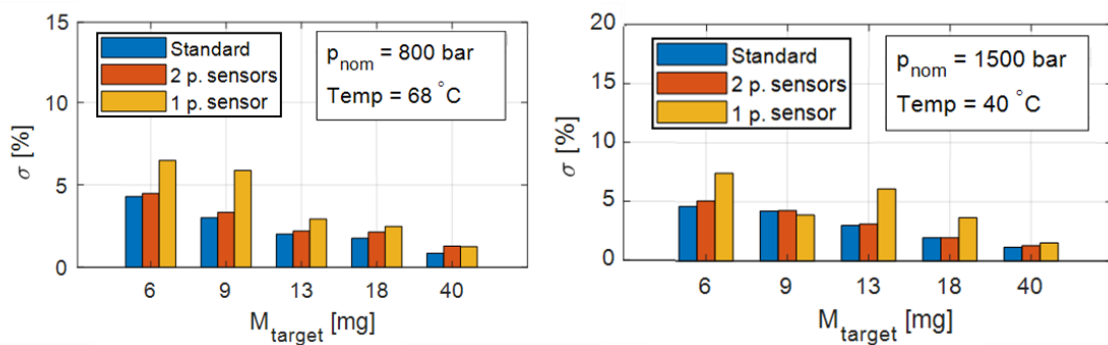
Figure 9. Control strategies performance at  $T_{tank} = 68^\circ\text{C}$ .

321 consecutive injections, for different single injection schedules (in terms of  $ET$ ,  $p_{nom}$  and  $T_{tank}$ ).

322 The severe manufacturing tolerances adopted for the CRI 2.20 injector and its Minirail are

323 capable to keep the injection stable, reaching a very satisfactory precision. Regarding the new

324 feedback-control, the precision slightly worsens with respect to the



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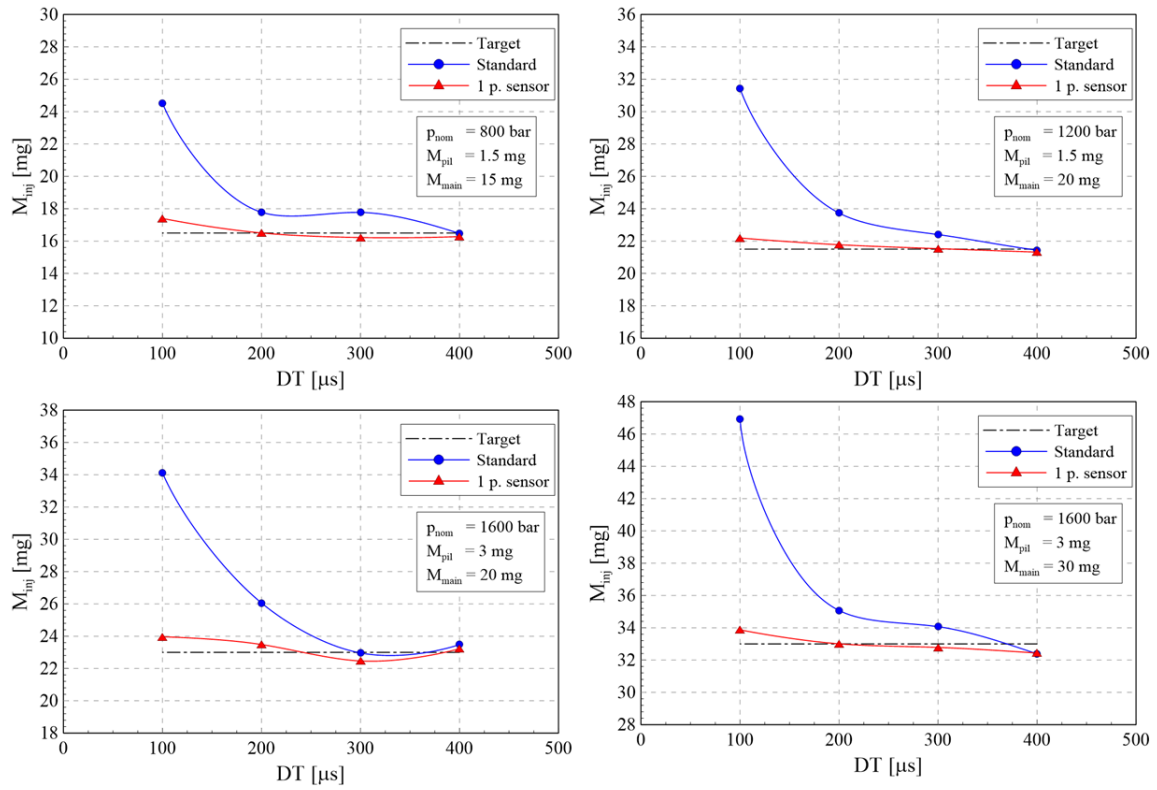
Figure 10. Injected mass normalized standard deviation for different  $p_{nom}$  and  $T_{tank}$  conditions.

327 control presented in [23]. However, it must be considered that the purpose of the proposed  
328 strategy is aimed at improving the accuracy of the injection system, which can deteriorate,  
329 due to physical drifts (such as the thermal one), rather than at enhancing the precision, which  
330 affects the small injected masses to a great extent.

331 Concerning the pilot-main injection, dwell time sweeps have been performed. Figure 11  
332 reports the pattern of the overall injected mass as  $DT$  varies for different  $M_{pil}$ ,  $M_{main}$  (the pilot  
333 and the main injected masses, respectively) and  $p_{nom}$  conditions when the injector is  
334 controlled by means of the standard open-loop strategy (continuous curves with circle  
335 symbols) and the new control-strategy (continuous curves with triangle symbols), compared  
336 with the target value (dash dot lines), which is given by the sum of the pilot and main  
337 nominal masses. The overall injected mass value is given by summing average values of  $M_{pil}$   
338 and  $M_{main}$  over 100 consecutive cycles measured by the HDA flowmeter. As can be observed,  
339 the standard injection system cannot achieve satisfactory accuracy below  $DT=300 \mu s$ , hence  
340 the possibility of applying efficient digital or continuous rate shaping is strongly limited [24].  
341 On the contrary, the new injection system features lower accuracy than 1 mg over the entire  
342  $100 \mu s \div 400 \mu s$   $DT$  range.

343 Finally, Fig. 12 shows a dynamic test regarding a pilot-main injection with a target pilot mass  
344 ( $M_{pil}$ ) equals to 2 mg, a target pilot mass ( $M_{main}$ ) equals to 20 mg, a dwell time of  $100 \mu s$ , at a  
345 nominal pressure of 1000 bar and  $T_{tank}= 40 \text{ }^\circ\text{C}$ . Fig. 12a reports the overall injected mass  
346 measured for 150 consecutive injections by the HDA (continuous line), compared with the  
347 target value (dash-dot line). In the left part of the graph, where the standard open-loop  
348 strategy is acting, the error on the total injected mass is around 15 mg (an injection fusion  
349 event is considered). When the new control-strategy is switched on (around injection no. 50),  
350 after a quick transient, the instantaneous injected mass approaches the target value, leading to  
351 a dramatic reduction in the error. Figs. 12b and 12c show two different instantaneous injected

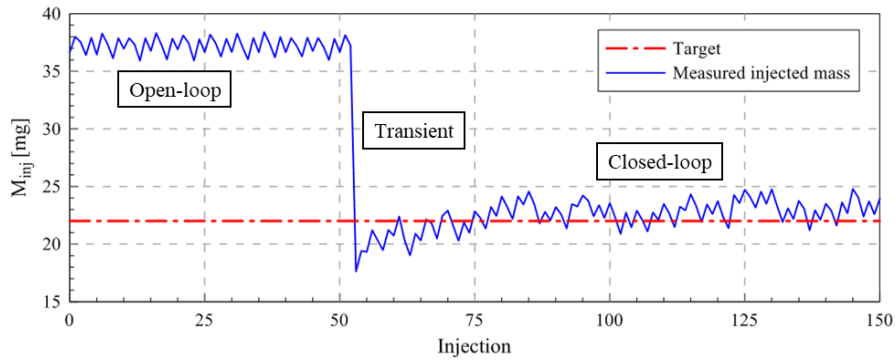
352 flow-rate traces pertaining to the standard control strategy (injection no. 25) and to the new  
 353 closed-loop control strategy (injection no. 102), respectively.



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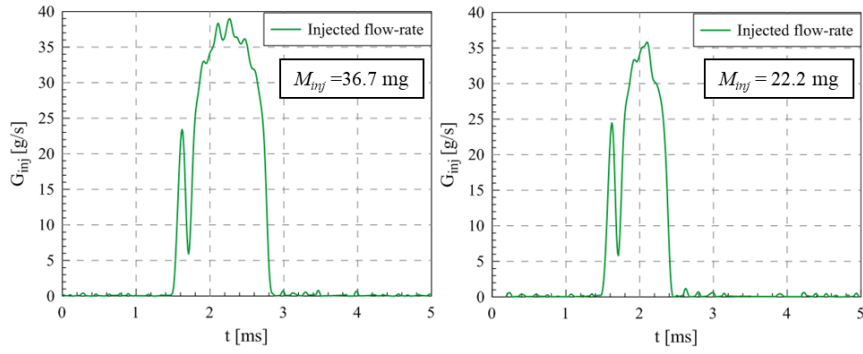
Figure 11. Performance of the new control strategy for pilot-main injections.



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(a): Overall injected mass for 150 consecutive injections.



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(b): Injected flow-rate time history (standard control).

(c): Injected flow-rate time history (closed-loop control).

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Figure 12. Feedback-control activation effect on the injected flow-rate ( $M_{inj,pi} = 2$  mg,  $M_{inj,main} = 20$  mg,  $DT = 100$   $\mu$ s,  $p_{nom} = 1000$  bar,  $T_{tank} = 40$   $^{\circ}$ C)

362

## Conclusion

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A new algorithm has been developed to calculate the flow-rate that enters the injector, based

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on two measured pressure traces: the one inside the rail (on which the rail pressure level in

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the commercial CR system is based) and the one detected along the injector-feeding pipe, in

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the proximity of the injector inlet. The developed algorithm for the flow-rate calculation is

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based on a simple hydraulic model. The rail is modelled as a zero-dimensional chamber, in

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which the corresponding measured  $p_{rail}$  time history is imposed, and this chamber is

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connected to the injector-feeding pipe by means of a gauged orifice, with specified area and

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discharge coefficient. The other measured pressure signal is  $p_{down}$ , imposed as the second

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boundary condition. The momentum balance and the continuity equations are solved in

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conjunction with a steady-state relation and this set of equations gives as outcomes the flow-

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rate across the orifice and the pressure time history at the orifice exit, namely  $p_{up}$ . This

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pressure signal is used, together with  $p_{down}$  to determine the flow-rate trace entering the

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injector, i.e.  $G_{inj,in}$ . By integrating  $G_{inj,in}$  with respect to time, the mass at the injector inlet is

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obtained. This mass correlates well with the effective fuel injected quantity, hence two

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mathematical laws  $M_{inj} = f(M_{inj,in})$  (one for single injections and one for pilot-main shots) have

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been determined. Owing to this correlation, by measuring the mass that enters the injector,

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one can obtain an estimated value of the real injected mass. A PID controller receives, as

380 input datum, the error  $\Delta M_{inj} = M_{inj} - M_{inj,ref}$ , and, based on it, a corrected value of the energizing  
381 time acting on the injector is determined, to mitigate the injected quantity inaccuracy.

382 The presented control strategy has been tested by means of a rapid prototyping hardware  
383 consisting on a Flexible ECU and a PXI platform. The software flashed into the ECU has  
384 been obtained by means of ETAS EHOOKS, which merges application and basic software.

385 Experimental tests have been carried out with a passenger car Common Rail injection system  
386 equipped with solenoid CRI 2.20 injectors of latest generation and both single and pilot-main  
387 schedules have been considered. The performance of the newly designed closed-loop control  
388 strategy has been compared with a previously developed control, in which two pressure  
389 signals were measured along the pipe feeding the injector to measure the mass at the injector  
390 inlet.

391 The results on single injections showed that the new control strategy is capable of reducing  
392 the error in the injected quantity below 0.6 mg for all the considered working condition. The  
393 control system is capable of compensating the drift, due to the different thermal regime  
394 experienced by the injector. If the fuel tank temperature is varied from 40 °C to 68 °C an  
395 error in the injected mass up to 2 mg can be noticed for the standard system, while the new  
396 closed-loop control features an accuracy below 1 mg, in line with the previously developed  
397 closed-loop control applying two additional pressure sensors.

398 Regarding the precision, the new control strategy is not capable to give any appreciable  
399 benefits since the CRI 2.20 injection apparatus features high performance in terms of  
400 repeatability, even when is equipped with the standard control.

401 Finally, tests on pilot-main shots have shown the capability of the presented control to  
402 dramatically reduce the overall injected mass error when the two injection events are close to  
403 be fused. The  $DT$  sweeps, for fixed set of  $p_{nom}$ ,  $M_{pil}$  and  $M_{main}$  values, present an error in the  
404 overall injected mass which is always below 1 mg.

405 In summary, the innovative architecture reaches the same control performance as in the case  
406 of a previous prototype with two additional high-pressure transducers, but with significantly  
407 lower manufacturing costs.

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