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Novel direct injection electro-hydraulic model-based controller for high efficiency internal combustion engines

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# 17 NOVEL DIRECT INJECTION ELECTRO-HYDRAULIC MODEL-BASED CONTROLLER FOR 18 HIGH EFFICIENCY INTERNAL COMBUSTION ENGINES

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# 32 ABSTRACT

33 During the past years, automotive industries developed several technologies suitable to increase efficiency and reduce emissions from Internal Combustion Engines (ICEs). Among them, the adoption of high-pressure 34 35 injection systems is considered crucial to optimize air-fuel mixture formation. However, the use of these 36 technologies also promotes the formation of particulate matter (PM), which is a direct result of charge stratification and fluid film on the cylinder walls. Therefore, to obtain a proper mixture formation without the 37 risk of wall impingement, the utilization of consecutive injections is mandatory. Since modern Gasoline Direct 38 Injection (GDI) systems are typically characterized by electrical-actuated injectors connected to a single high-39 40 pressure rail, a deep understanding of electrical and hydraulic effects among two close injection events becomes essential. This paper analyzes the combinations of electrical and hydraulic effects that occur in a 41 42 high-pressure GDI system performing multiple injections. By using a specifically developed open vessel flushing bench, the injection system has been characterized in terms of pressure wave propagation as well as 43 electrical distortions of the driving current profile of the injectors. The analysis of the experimental data has 44 allowed for the calibration of the residual magnetization characteristic map in addition to the development of 45 a pressure wave propagation control-oriented model. Finally, a Magnetization and Pressure Wave (MPW) 46 correction strategy, easily implementable on an Electronic Control Unit (ECU) without the need for additional 47 sensors, has been proposed. By running the MPW strategy, the error between the actual and expected injected 48 49 mass has been reduced below 5% in all tested conditions.

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51

# 52 KEYWORDS

- 53 GDI System
- 54 Multiple Injections
- 55 Residual Magnetization

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- 56
- Pressure Fluctuations
- 57 Injected Fuel Mass Variation
- 58 Control-Oriented Modelling
- 59
- 60 61

SYMBOLS/ABBREVIATIONS					
A <sub>1</sub>	Electric charge on the injection coil in magnetized conditions				
$A_2$	Electric charge on the injection coil in unmagnetized conditions				
BĒV	Battery Electric Vehicle				
A <sub>3</sub>	Difference between magnetized and unmagnetized electric charge				
ĊĊV	Cycle-to-Cycle Variability				
CR	Common Rail				
DI	Direct Injection				
DoF	Degree of Freedom				
DT	Dwell Time				
dt	Time differential				
ECU	Electronic Control Unit				
EOI	End Of Injection				
<b>ET</b> corrected	Corrected Energizing Time to compensate the magnetization phenomenon				
ET <sub>eq</sub>	Equivalent Energizing Time due to the magnetization phenomenon				
ET <sub>1</sub>	Energizing Time of the first injection pulse				
ET <sub>2</sub>	Energizing Time of the second injection pulse				
ET <sub>2corr</sub>	Energizing Time correction of magnetization effect				
ET <sub>2Final</sub>	Energizing Time correction of magnetization and wave effects				
$ET_{2w}$	Energizing Time correction of wave effect				
FEV	<b>EV</b> Fuel Cell Electric Vehicle				
GCI	GCI Gasoline Compression Ignition				
GDI	GDI Gasoline Direct Injection				
GPF	Gasoline Particulate Filter				
HEV	Hybrid Electric Vehicle				
HP	High Pressure				
1(t)	Current behavior in time				
	Internal Combustion Engine				
LP	Low Pressure				
LIC m	Torget mass for first injection				
m	Target mass for second injection				
ш <sub>2</sub> Мррор	Magnetic Proportional				
MPW	Magnetization and Pressure Wave Correction Strategy				
MSD	Mass Spring Damper				
PFI Port Fuel Injection					
<b>PKistIni</b> Pressure from Kistler Sensor mounted on feed duct Injector Side					
<b>PKistRail</b> Pressure from Kistler sensor mounted on feed duct Rail Side					
PM Particulate Matter					
PRail	Rail Pressure Signal				
PWM	Pulse Width Modulation				
Q	Electric charge on the injector coil				
RCP	Rapid Control Prototyping				
RM	Residual Magnetization				
RMSE	Root Mean Squared Error				

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RON	Research Octane Number		
RPM	Revolution Per Minute		
SA	Spark Advance		
SACI	Spark Assisted Compression Ignition		
SNR	Signal to Noise Ratio		
SOI	Start Of Injection		
TWC	Three Way Catalyst		
UHC	Unburnt Hydrocarbon		
ΔΕΤ	Variation of Energizing Time due to the magnetization phenomenon		
с	Equivalent Damping		
ξ	Damping Ratio		
k	Equivalent Stiffness		
m	Equivalent Inertia		
Wa	Damped Natural Frequency		
ω Wn	Natural Frequency		
v <sub>n</sub>	Initial Condition		
Xo	Initial Condition		
··· v			

#### 64 INTRODUCTION

65 Over the past years, to develop increasingly efficient powertrains, many solutions have been proposed. Among these, hybrid vehicles, which normally involve the combined use of an Internal Combustion Engine (ICE) and 66 67 an electric motor, are often a good compromise solution, mainly because of the current technological limitations of Battery Electric Vehicles (BEVs), such as long charging time and limited rangeability [1-2]. In 68 this scenario, the development of less polluting and more efficient engines remains crucial to ensure optimal 69 70 utilization of energy resources during the current and future energy transition. Many authors [3-6] 71 demonstrated the potential of HEVs based on modern Spark Ignited (SI) ICEs, usually equipped with a highpressure Gasoline Direct Injection (GDI) system. The adoption of a GDI system allows injecting gasoline at 72 high pressure directly into the combustion chamber. This system results beneficial to improve combustion 73 74 efficiency, phasing, duration and injected fuel quantity [7,8]. Conversely, Liang et al. [9] demonstrated that GDI engines are characterized by high Particulate Matter and Unburned Hydrocarbon (UHC) emission. As 75 76 explained by Catapano et al. [10], high PM production is typically related to an incomplete vaporization of 77 some fuel droplets. Moreover, other works in the literature [11] report that the high protrusion capability of 78 the fuel jet might increase the PM production due to the formation of a thin gasoline film on the cylinder walls. 79 Therefore, to mitigate engine-out emissions in GDI engines, complex aftertreatment systems [12], typically 80 composed of the Three-Way-Catalyst (TWC) and Gasoline Particulate Filter (GPF), have been adopted. Unfortunately, the backpressure increase produced by the GPF raises the pumping work and knock tendency, 81 82 affecting the overall engine efficiency. A possible approach to mitigate PM production is strictly connected to 83 the injection strategy management (increasing the injection pressure and optimizing the Start of Injection (SOI) 84 [13]). Yamaguchi et al. [14] demonstrated that a higher injection pressure results beneficial to obtain smaller fuel droplets, thus shortening the vaporization process. On the other hand, the fuel jet's protrusion is directly 85 related to the injection pressure: the higher the injection pressure, the longer the penetration promoting the 86 cylinder wall impingement. The adoption of a multiple injection strategy is crucial to overcome this limitation 87 88 and maximize the benefits of high-pressure GDI systems. Injecting the required fuel mass through multiple 89 injections promotes mixture homogenization and reduces the risk of wall impingement [15-16]. Moreover, 90 multiple injections proved to be essential for the management of innovative combustion concepts, such as Low

91 temperature Combustions (LTC) [17-19].

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92 As widely described in the literature [20-23], in a GDI system operated with consecutive fuel jets, the first 93 injection could strongly influence the behavior of the following, and the overall injected mass could significantly differ from the desired quantity. This result is due to the superposition of two effects: an electro-94 magnetic phenomenon in the injector coil (the magnetization of the coil due to the previous injection has not 95 completely disappeared before the SOI of the following) and the propagation of pressure waves in the feed 96 97 duct between rail and injector [24]. With regard to the electro-magnetic phenomenon, Viscione et al. [25] 98 demonstrated that the residual energy content on the secondary coil of the injector is directly related to the shape of the current driving profile and Dwell Time (DT) between the injections. In particular, the higher the 99 100 residual energy in the coil (e.g., when small DT are actuated) the higher the slope of the subsequent current profile. Such electrical interference results in a greater mass of fuel injected. With regard to the fuel pressure 101 102 fluctuation (well known in high-pressure common rail injection systems [22]), it is triggered every time the 103 injector is opened, and it occurs in the feed duct between the rail and the injector until its energy is dissipated. 104 Thus, if the following injection occurs before the wave is completely dissipated, the injector will experience 105 an instantaneous upstream pressure deviation from the target value (typically the rail pressure) and it will 106 deliver a varying mass, depending on the difference between rail pressure and the average of actual pressure 107 at the injector inlet during the second injection.

108 This paper presents an innovative injection management strategy aimed at improving standard GDI injection 109 controllers by modelling and coupling both the electrical and hydraulic behavior of the injection system (by means of properly calibrated look-up tables to represent both phenomena), while performing coupled 110 injections. In the present work, a GDI injection system capable of delivering fuel at pressures up to 750 bar 111 has been studied. Extensive experimental activity aimed at investigating the effects of different injection 112 113 strategies (injections duration, number, and spacing) and rail pressures on the behavior of the GDI system has been conducted using an open vessel flushing bench. Since previous work analyzes how the electro-magnetic 114 115 phenomenon impacts the behavior of the injection system, this paper focuses on the analysis and modeling of 116 the high-pressure wave in the injection system. Moreover, the detailed knowledge of the GDI injection system 117 behavior has allowed to develop an innovative model-based GDI injection controller (MPW). In order to demonstrate the practical application of the presented approach, the developed injection controller has been 118 implemented in a Rapid Control Prototyping (RCP) system. Finally, the MPW strategy has been 119 120 experimentally validated and the improvements in terms of injection mass controllability have been demonstrated by comparing the injected mass running the injection system in standard configuration and with 121 122 the innovative model-based injection controller.

123

### 124 1. EXPERIMENTAL SETUP

125 The experimental activity has been carried out on a specifically designed high-pressure open vessel flushing bench fueled with Research Octane Number (RON) 95 commercial gasoline. In this layout, the high-pressure 126 127 system [22] has been connected to a set of GDI injectors [26] provided by Marelli Europe S.p.A. through standard feed ducts. The experimental activity has been carried out by injecting fuel into air at atmospheric 128 129 conditions (pressure and temperature were not conditioned/controlled). The consumption has been measured 130 through an AVL balance 733s. Figure 1 shows a schematic of the hydraulic bench layout, where the Low-131 Pressure (LP) and High-Pressure (HP) lines are represented in yellow and red respectively, while the water cooling is depicted in blue. Moreover, the main characteristics of the injection system are summarized in Table 132 133 1.

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Figure 1: Flushing bench hydraulic layout

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The LP line, between the LP pump and the HP pump, has been kept at a constant pressure of 4.5 barA through a mechanical pressure regulator. The return line of the pressure regulator has been connected with the return of the HP pump and water-cooled before mixing with the fuel coming from the AVL Balance. This configuration limits the increase in fuel temperature while the bench is operating. Moreover, to properly maintain at target value of the rail pressure, the HP pump has been equipped with a normally-opened solenoid metering valve Magnetic Proportional (MPROP). The return line of the MPROP has been directly sent to the AVL Balance.

144

145

Table	1:	Injection	System	Characteristics
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Number of Injectors	4
CR Pump	Bosch CP1
Injector Type	Marelli IVPH 700 bar [26]
Feed Duct Internal Length	29 mm
Feed Duct Internal Diameter	3 mm
CR External Length	19 mm

146

147 To monitor fuel pressure and temperature, additional transducers have been installed in the low-pressure line.

148 With regard to the high-pressure line, where the only standard transducer available is the rail pressure sensor 149 (PRail), to study pressure fluctuations and Residual Magnetization (RM) one of the high-pressure feed ducts

150 (that delivers fuel from the rail to a specific injector) has been equipped with two piezoresistive high-pressure

151 sensors Kistler 4067A: the first close to the rail (PKistRail) and the second close to the injector (PKistInj) (the

distance between the two sensor is 19.9 cm) [25].

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153 154

Figure 2: Position of pressure sensors and HP pump

156 To monitor the current driving profile, a current clamp Hioki CT6846A has been located in correspondence of 157 the coil of the injector. The high-pressure sensors (PKistRail, PKistInj and PRail) and the current clamp have 158 been acquired at 100 kHz to capture a higher-frequency content in the acquired signals, thus preventing aliasing 159 in the sampling of pressure waves and injection commands. Moreover, to measure the HP pump speed, an 160 optical encoder has been installed on the pump driveshaft. To properly control the flushing bench, and collect data coming from sensors, an RCP system based on a National Instruments cRIO 9082 has been developed. 161 162 As shown in Figure 2, where the open vessel flushing bench is depicted, the high-pressure pump is 163 mechanically connected through a toothed timing belt, with a fixed transmission ratio of 0.5 (replicating the 164 on board GDI pump connection), to an electric motor (5.5 kW maximum power @3000 rpm). During the 165 whole testing campaign, the speed of the HP pump has been kept constant at 500 rpm directly controlling the 166 speed of the electric motor. This value has been selected since higher frequencies lead to an electro-magnetic 167 interference between the modulation of the inverter and the signals detected by the RCP system. However, the presented approach remains valid also for different rotational speeds. As regards the rail pressure, it has been 168 169 kept at a target value through proper management of the Pulse Width Modulation (PWM) command of the 170 MPROP valve. To ensure a flexible control of the whole injection pattern, both in terms of rail pressure, 171 number of injections, duration, and relative distance between the fuel jets, a fully programmable Electronic 172 Control Unit (ECU), SPARK by Alma Automotive, has been adopted. Finally, to log the parameters of the 173 injection controller and improve testing operations, INCA software (by ETAS) has been used. Figure 3 shows

the complete layout of the control and acquisition systems.

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# 177

#### **RESIDUAL MAGNETIZATION MODELING** 178 2.

In this section, the behavior of the injector under a double injection strategy in terms of total injected mass as 179

180 a function of DT has been investigated. Figure 4 depicts the trend of the total injected mass performing two 181 consecutive injections (same pulse duration on both injections equal to 700 µs) as a function of DT for three

levels of rail pressure: 300 bar, 500 bar, and 700 bar. 182



183

184 Figure 4: Total injected mass as a function of DT for first and second nominal injection duration of 700 µs for PRail of 300 bar 185 (black line), 500 bar (blu line), 700 bar (red line) and hydraulic overlap (in yellow), transition region (in green) and stable region 186 (in purple)



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By looking at Figure 4, three different regions can be identified: DT from 0 to 350 µs called "hydraulic overlap", where the injector operates in the hydraulic fusion region and the second injection starts before the injector needle comes back to the closing position; "transition region", where DT ranges between 350 µs and 1500 µs, with a clear decrease of the measured injected mass; "stable region", DT from 1500 µs up to 4000 µs, shows a stable value of the injected mass in the whole range of DTs.



193 194

Figure 5: Current profiles for different DTs synchronized with the EOI of the second injection

195

196 To highlight the influence of electro-magnetic effects on the current driving profiles performing double 197 injections, four different DTs (both injections have the same duration) have been compared and synchronized 198 with respect to the descendent slope of the second injection (End of Injection, EOI), as shown in Figure 5. As 199 it is possible to see, the lower the DT, the steeper the driving profile during the opening phase of the second 200 injection. This aspect is mainly related to the amount of energy not yet dissipated in the injector coils and, as 201 it is possible to notice, it leads to an increase in the effective opening time of the injector during the whole 202 injection process [25]. Focusing the attention on the range of maximum interaction (DT = 50  $\mu$ s to 500  $\mu$ s), the 203 response time of the electrical opening command of the injector (time to reach 100% of the driving profile) is 204 significantly lower with respect to the one needed in unmagnetized condition ( $DT = 1500 \mu s$ ), resulting in a 205 higher injected mass. This behavior can be explained by referring to the residual energy content in the coils, 206 which is high for close injection, then damps out as the dwell time exceeds 1500 µs. To characterize the amount 207 of energy stored in the injector coils, the charge (Q) has been calculated by Equation 1, where i represents the 208 acquired current of the injector driving profile:

$$Q = \int_{SOI}^{EOI} i(t)dt \tag{1}$$

209

Differences between the charges of the driving unmagnetized and magnetized profiles have been calculated asa function of DT. The equivalent unmagnetized driving profile has been obtained by rigidly translating the

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- 212 electrical profile of the first injection pulse (which is unaffected by the magnetization) using the EOI of the 213 second injection for synchronization. Figure 6 reports the signal manipulation process used to compare different driving profiles (unmagnetized and magnetized) at  $DT = 50 \mu s$ . As shown in Figure 6, the yellow area 214 215 represents the nominal overlap between the two profiles. The green region, which lies between the equivalent 216 unmagnetized profile (blue trace) and the magnetized profile (black trace), as well as the reference signal (red 217 trace), stands for the RM effect. As DT increases the hydraulic fusion region ends and the magnetized profile 218 reaches 0% current during the closure phase: in this situation the RM is calculated as the area between the 219 magnetized profile and the equivalent unmagnetized one.
- 220



221 222

Figure 6: Example of magnetized (black), unmagnetized (blue), overlap area (yellow), RM area (green)

In Figure 7, the magnetized equivalent charge (A1: blue curve) is compared with the equivalent unmagnetized (A2: black dashed). As previously described, the overlap reduces the effective charge in the coil, and its effect is appreciable in the first part of the A2 curve. Finally, in order to keep into account both effects (overlap and magnetization), the difference between the two curves (A3: red dotted), has been calculated. As expected, the

A3 curve shows a decreasing trend with the increase of the DT.

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Figure 7: Equivalent charge for magnetized and unmagnetized profiles

Since the RM produces a higher injected mass, its effect can be described as an increment in ET with respect to the nominal injection duration. The Equivalent ET  $(ET_{eq})$ , defined in Equation 2, represents the ET for the second injection (in a two injection pattern with a standard duration equal to  $ET_1$ ) that keeps into account the RM [24]. Figure 8 and Figure 9 report examples of  $ET_{eq}$  trend as a function of DT with different injection pressures and durations (both injections have the same ET). It is worth pointing out that the different profiles in Figure 9 collapse in a single curve if each profile is normalized with respect to the  $ET_{eq}$  at regime (when it reaches the nominal ET value).

$$ET_{eq} = \left[\frac{A_3(DT)}{\max(A_3(DT))} + 1\right] * ET_1$$
(2)

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<sup>231</sup> 





Figure 8: Equivalent ET as a function of DT for pulses nominal duration of 700 μs for PRail of 300 bar (black line), 500 bar (blue line) and 700 bar (red line)



243

Figure 9: Equivalent ET as a function of DT for PRail equal to 500 bar and pulses nominal duration of 400 μs (black line), 510 μs
 (blue line) and 700 μs (red line)

As Equation 2 describes,  $ET_{eq}$  depends only on RM, overlap and first injection duration, while any dependence with respect to the pressure can be neglected [25]. In Figure 10, the map of  $ET_{eq}$  is presented as a function of

the nominal pulse duration of the second injection ( $ET_2$ , since  $ET_2 = ET_1$ ) and DT.

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Figure 10: Equivalent ET map



In order to validate the presented method, the estimated injected mass for the second injection pulse in a double injection pattern has been calculated, with the procedure schematized in Figure 11. At first, the target mass of the second injection  $(m_2)$  has been converted into the injection durations  $(ET_2)$  through the injector map interpolation (actual PRail has been considered) neglecting the RM effect. Thus, by using the map shown in Figure 10, the RM effect has been taken into account with the  $ET_{eq}$  calculation (greater than  $ET_2$ ) as a function of DT and  $ET_2$  (with  $ET_2 = ET_1$ ). At that point, the obtained equivalent ET and the acquired PRail have been used to interpolate the injector map estimating the mass injected during the second injection.

260



261

262

Figure 11: Procedure for the calculation of the injected mass under magnetized conditions

263

In Figure 12, the comparison between acquired (through the flushing bench, black solid curve with squares) total injected mass ( $m_1 + m_2$ , where  $m_1$  represents the target injected mass in the first injection and  $m_2$  in the second injection) and estimated (blue solid curve with dots) performing double injection strategy (where  $m_1 = m_2$ ) has been reported.

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Figure 12: Comparison between experimental consumption under magnetized conditions (black curve), estimated one (blue curve), relative error (red curve) and hydraulic fusion region (in yellow)

272 As Figure 12 shows, the relative error (red dotted curve) is always below 6%, except for the hydraulic fusion 273 region (yellow area, between 50 µs and 500 µs) where a peak value of 14% is reached. However, even at high 274 DT (higher than 1500 µs), the model cannot accurately predict the behavior of the injector; in fact, the injected mass predicted by the model is about 2% higher than that measured one. As a matter of fact, since the model 275 276 relies on the value of the rail pressure, which could be slightly higher than the actual the injector is subject to 277 during the second injection (due to the pressure fluctuations in the feed duct of the injector), the modelled 278 injected mass is usually overestimated. Therefore, to improve the estimation of the injected mass, it is critical 279 to also develop a model that predicts the instantaneous pressure inside the injector feed port. Such a model will 280 be discussed in the next section.

281

### 282 **3. PRESSURE WAVES PHYSICAL MODEL**

In this section, a detailed analysis of pressure dynamics that take place in the feed ducts of the injectors is 283 284 reported. Several authors [25–28] already analyzed the pressure fluctuations that occur in high-pressure injection systems. However, most of these studies are focused on CR high-pressure injection systems for 285 286 compression ignited engines. To model the hydraulic behavior of the system reported in Figure 2 [22], a wide 287 experimental activity, summarized in Table 2, has been carried out performing both single and double injections (different DT with the same injection durations,  $ET_1 = ET_2$ ). The tests have been run activating one 288 289 single injector, while the remaining 3 injectors of the system have been kept inactive (but connected to the 290 rail).

291

Table 2: Summary of experimental activity tests

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Injection Type	Rail Pressure [bar]	ET1[µs]	ET2[µs]	DT [μs]
Single	200:100:700	350:100:1950	-	-
Single	750	350:100:1950	-	-
Double		400	400	
Double	300	450	450	
Double		700	700	
Double		400	400	
Double	500	510	510	50:50:1500
Double		700	700	
Double		400	400	
Double	700	600	600	
Double		700	700	

293 Figure 13 reports an example of acquired pressure signals: PRail (grey), PKistInj (red) and PKistRail (blue).



294

Figure 13: Experimental pressure traces of the three signals for a single injection at 500 bar, ET 750 μs: PRail (grey line), PKistInj
 (red line), PKistRail (blue line), injection command (black line)

297

298 Analyzing the acquired signals immediately after the SOI, it is possible to notice a sudden drop of the pressure 299 in both PKistRail and PKistInj traces, while such information is not clearly visible in the PRail signal. This 300 aspect can be easily explained by the amount of fuel mass contained in the rail (compared to the feed duct) 301 which limits the effects of the wave propagation. Even if PKistRail and PKistInj signals show the same 302 behavior in terms of oscillations, the drop phase in PKistRail is slightly retarded with respect to PKistInj. In 303 fact, the high-pressure wave (generated by the injector opening), moves backward to the rail. As a 304 consequence, the PKistRail sensor experiences a pressure drop slightly later. Focusing the attention on the 305 PKistInj signal, three different phases can be identified [31]: the first stage (point 1 in Figure 13) refers to the 306 abrupt pressure drop caused by the injection, the second (point 2) features the high-pressure oscillation where 307 the wave triggered by the injection propagates back in the feed duct, while the last (point 3) is the pressure 308 recovery stage, where the high-pressure wave is almost completely dissipated. Due to the position of PKistInj

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309 with respect to the injector, this sensor perceives the injector's upstream pressure: for this reason, the modelling 310 of the hydraulic behavior has been based on the information contained in the corresponding signal. In addition, 311 to increase the robustness of the modelling approach by analyzing only the pressure wave caused by the 312 injection matrix.

312 injection event (i.e., removing noise), the average of the pressure signal acquired over 500 consecutive cycles

has been considered. The average pressure trace presents a well-defined shape, typical of the underdamped
 mass-spring-damper system, as shown in Figure 14.



315

**316** *Figure 14: PKistInj signals of 500 consecutive cycles (grey lines) and the mean one (red line) for 500 bar, ET 750 μs* 

317

The behavior of the hydraulic system can be described as a free response of a one-degree-of-freedom Mass Spring Damper (MSD) [22] system described in Equation 3, where x represents the pressure, m represents the fuel inertia, c is the equivalent damping and k the equivalent stiffness.

$$m\ddot{x} + c\dot{x} + kx = 0 \tag{3}$$

321

322 It is possible to define the natural frequency  $\omega_n$  of the system and the damping ratio  $\xi$ , as in Equations 4 and 323 Equation 5 respectively.

$$\omega_n = \sqrt{\frac{k}{m}}$$
(4)

$$\xi = \frac{c}{2\sqrt{km}} \tag{5}$$

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325 Moreover, it is important to define the oscillating frequency  $\omega_d$  in under-damped conditions as reported in 326 Equation 6:

$$\omega_d = \omega_n \sqrt{1 - \xi^2} \tag{6}$$

327

The free response of the system is described by Equation 7 where  $x_0$  and  $v_0$  represent the initial conditions at t = 0.

$$x(t) = e^{-\xi\omega_n} \left[ x_0 \cos(\omega_d t) + \frac{\nu_0 + \xi\omega_n x_0}{\omega_d} \sin(\omega_d t) \right]$$
(7)

330

To understand how many free responses are needed to describe the hydraulic system under study (the overall system behavior will be the result of the superposition of several MSD free responses [22]), the power spectrum
of experimental pressure waves under different conditions has been studied. Figure 15 shows an example of
the power spectrum for three different values of rail pressure obtained running single injection at a constant
ET of 700 μs.



336

Figure 15: Power spectrum for ET equal to 700 µs for three different values of the rail pressure: 300 bar (black line), 500 bar (blue line) and 700 bar (red line)

339

From the analysis of the power spectrum reported in Figure 15, four characteristic frequencies can be clearly identified: the first frequency at 8.33 Hz is related to the injection frequency; the second, between 520 Hz and 590 Hz; the third, between 650 Hz and 740 Hz, and the fourth, between 5 kHz and 6.5 kHz. The first frequency can be neglected because it does not represent the dynamic system response but only the external trigger (injection event). The second frequency can be considered as the main carrier of the system, because it shows

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- 345 the highest amplitude in all the operating conditions. On the opposite, since it does not exhibit a high amplitude
- and its relative frequency distance from the main carrier is limited (as previously demonstrated by Silvagni et
- al. [22]), the third carrier can be disregarded. Finally, despite its low amplitude, the fourth main frequency
- cannot be neglected, mainly because such contribution is clearly visible in a relatively high-frequency range.
- Therefore, in order to properly model the pressure fluctuations both second and fourth harmonics are required.
- 350 Moreover, since the frequencies of the pressure waves are proportional to the speed of sound (which is affected 351 by the bulk modulus of the fluid), the power spectrum shifts to higher frequencies with the increase of the rail
- by the bulk modulus of the fluid), the power spectrum shifts to higher frequencies with the increase of the rat
- **352** pressure [31].

In order to reconstruct the single-injection pressure wave propagation as the sum of free responses of the system, the characteristic parameters ( $\omega_n$ ,  $\xi$ ,  $x_0$ ,  $v_0$ ) of each characteristic carrier (second and fourth frequencies) have been calculated through a numerical procedure using MATLAB code. The numerical algorithm minimizes the distance between the modulus between the actual pressure trace, filtered across the considered carrier (one at a time), and the 1 Degree of Freedom (DoF) MSD parametric free response reported in Equation 7. This procedure has been applied for each pressure trace for the two identified characteristic carriers.

- 360 Figure 16 and Figure 17 show the maps of the 1-DoF MSD system characteristic parameters for each carrier
- 361 as a function of injection pressure and ET.



362

**363** Figure 16: Characteristic parameters for the first carrier as a function of PRail and ET:  $\omega_{n1}$  (a),  $\xi_1$  (b),  $x_{01}$  (c),  $v_{01}$  (d)

364

By looking at the maps of the first carrier, it can be noticed that both  $\omega_{n1}$  and  $v_{01}$ , Figure 16 a) and d)

respectively, are linearly dependent on the injection pressure while the dependence on the injection duration is significantly lower. Figure 16 c) shows the trend of  $x_{01}$  as a function of ET and rail pressure. The map of  $\xi_1$ 

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368 (Figure 16 b) does not report a clear tendency. However, due to the very limited range of variation of  $\xi_1$ , its

369 value can be considered constant for the following modeling process.

370



371

**372** Figure 17: Characteristic parameters for the second carrier as a function of PRail and ET:  $\omega_{n2}$  (a),  $\xi_2$  (b),  $x_{02}$  (c),  $v_{02}$  (d)

373

Analyzing the maps of the second carrier also  $\omega_{n2}$ , as expected, shows a linear trend with the rail pressure,  $\xi_2$ which can be considered constant for all the conditions as well as  $v_{02}$ . Finally,  $x_{02}$  becomes different from zero at intermediate values of rail pressure and pulse duration.

377

## 378 4. RESULTS AND DISCUSSION

In the following sections, an innovative injection management strategy aimed at improving standard GDI
 injection controllers, based on the combined modeling of both electrical and hydraulic behavior of the injection
 system, is described.

## 382 4.1 Pressure Wave Reconstruction

383 In order to accurately calculate the  $ET_{eq}$  described in section 3, the pressure fluctuation in the feed duct of the 384 injection system under study has to be reconstructed.

# 385 4.1.1 Pressure Wave Reconstruction for Single Injection

As reported in previous works [22], the injection system hydraulic behavior performing multiple injectionscan be described as the superposition of two reconstructed injection pressure traces triggered by a single

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388 injection event. In the case of single injection event, an estimation of the pressure fluctuation can be obtained 389 using the scheme shown in Figure 18: the rail pressure and ET of the injection triggering the pressure wave 390 are used to interpolate the maps of the characteristic parameters for the 1-DoF MSD equivalent systems of the 391 2 selected carriers. Moreover, Gain and Offset maps are interpolated in the same way to obtain a complete 392 reconstruction of the pressure trace. The Offset map represents the mean value of the pressure wave (lost 393 during the reconstruction, since the fundamental component is not modelled) [22] which is lower than the 394 PRail since the pressure in the duct is not recovered even after several milliseconds, as appreciable in Figure 395 13. As regards the Gain map, it is necessary to compensate for the loss of information due to the fact that only 396 two harmonics of the spectrum are considered [22].

397



398

### 399

Figure 18: Single injection reconstruction strategy

400

In order to evaluate the accuracy of the proposed methodology, the percentage error between acquired and modelled pressure traces has been calculated. Figure 19 compares the predicted and the experimental pressure trace from the PKistInj in different conditions of rail pressure and ET. As it can be seen, the error, defined as the difference between the acquired and the estimated pressure oscillation, is always between +/-5 bar with the exception of the zone immediately after the injector closing (such area is out of the nominal operating range

406 of GDI injection systems).

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408 Figure 19: Experimental pressure traces (blue curves) and modeled (purple curves) for 300 bar (a), 500 bar (b) and 700 bar (c), for
 409 ET of 400 μs (numbers 1) and 700 μs (numbers 2)

410

#### 411

#### 4.1.2 Pressure Wave Reconstruction for Double Injection

412 As mentioned in the previous section and described by several authors [17–19, 30, 31], the proper management 413 of the injection pattern is crucial to guarantee proper combustion stability and controllability [34]. However, 414 the amount of fuel injected through the injection process after the first injection event is strongly affected by 415 the pressure wave triggered by the first injection pulse (if the pressure wave is not yet completely damped). It 416 is important to underline that, even if extremely high DTs are applied (i.e., 5 ms where the magnetization 417 phenomena can be neglected), the pressure in the feed duct in the considered GDI injection system is still not 418 recovered, resulting in a lower injected mass, compared to doubling a single injection event (if the same ET is 419 applied).

420 In order to obtain the pressure traces generated in the feed duct when a double injection pattern is actuated, the 421 acquired PKistInj signal during a double injection test has been deeply analyzed. Figure 20 shows the 422 comparison between the measured pressure traces when a double injection (with  $ET_1 = ET_2$ , red trace) and a 423 single injection are actuated (blue trace). Moreover, Figure 20 also reports the reconstruction of the pressure 424 wave (magenta trace) for the double injection strategy, obtained superimposing to the rail pressure signal PRail 425 two single injection pressure oscillation traces: the first wave starts from SOI<sub>1</sub>, while the following occurs in 426 correspondence of the considered DT (equal to1000 us). The analysis of the relative error between the 427 experimental (PKistInj) and reconstructed (PRail + pressure oscillations model) pressure traces confirms the 428 accuracy of the presented modelling approach even in the case of multiple injection strategies.

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430 Figure 20: Comparison of single injection at 700 bar (blue curve) with an ET equal to 700  $\mu$ s, a double injection at 700 bar with 431  $ET_1 = ET_2 = 700 \ \mu$ s and  $DT = 1000 \ \mu$ s (red curve), and the reconstructed pressure trace of the double injection obtained 432 superimposing to the single injection wave itself, at the DT (purple curve)

A scheme of the methodology for the reconstruction of the pressure waves when a double injection pattern isused, is reported in Figure 21.



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Figure 22 shows the comparison between reconstructed and experimental pressure oscillations for double injection strategy at four different DTs of 300  $\mu$ s, 500  $\mu$ s, 750  $\mu$ s, 1000  $\mu$ s respectively (PRail approximately equal to 700 bar, ET<sub>1</sub> = ET<sub>2</sub> = 700  $\mu$ s). For each condition, the analysis of the absolute error reported in Figure 22 demonstrates the capability of the reconstruction strategy to predict the pressure waves in the feed duct during a double injection pattern. As it is possible to see, during the second injection the error is always between +/-5 bar. As a result, the model can reliably estimate pressure fluctuations and the actual injection pressure even running double injection patterns.



Figure 22: Comparison of experimental (blue curve) and simulated (purple curve) pressure traces for double injection strategy at
 PRail of 700 bar with ET1 and ET2 equal to 700 μs for DT equal to 300 μs (a), 500 μs (b), 750 μs (c) and 1000 μs (d)

449

450 Since the injected mass of the second fuel jet depends on the actual pressure acting on the injector during the 451 second injection event (which can significantly differ from the pressure value in the rail, if a pressure wave is 452 still present) and ET2, the average of the acquired and the estimated pressure oscillation during the second 453 pulse (in the analyzed condition, PRail = 700 bar and  $ET_1 = ET_2 = 700 \ \mu s$ ) has been calculated (Figure 23). 454 As it is possible to notice, the average pressure during the second injection is always lower than the rail pressure 455 (which corresponds to a pressure value of 0 bar). This aspect explains the behavior of Figure 12, where the 456 modeled mass shows an offset with respect to the experimental one (since the RM model relies on PRail). 457 Therefore, the effect of the pressure oscillation is always to decrease the injected mass, with respect to the sum

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458 of two single injections performed under the same conditions of ET and rail pressure. The absolute error 459 between the two pressure traces is always between  $\pm -3$  bar, confirming the reliability of the presented method. 460 However, the error between the two pressures slightly increases at higher DTs owing to an overestimation of 461 the second pressure drop. This aspect is related to the procedure applied for the reconstruction of the wave. An 462 offset is applied before the sum of the second wave to the first one, and it can be modelled as a constant value or as a function of DT. For the purpose of the present work, to perform accurate estimation below DT = 800463 464 us where the absolute value of the error is below 2%, such quantity has been considered as a constant being 465 the objective



466

467 Figure 23: Mean pressure oscillation during the second injection for experimental (red crosses) and simulated (blue dots) pressure
 468 traces for PRail of 700 bar and pulses duration of 700 μs (a) and absolute error (b)

469

### 470 **4.2 Equivalent Energizing Time Modeling for Pressure Wave**

In the previous sections, the effect of the hydraulic interaction between two consecutive injections on the total
amount of injected fuel has been deeply discussed. As a result, to optimize the combustion process when
multiple injection pattern is used, a strategy which compensates for the injected mass deviations is mandatory.

474 In Figure 24, a schematic of a possible injection quantity controller for a double injection pattern (two small 475 pilots are actuated with  $ET_1 = ET_2$ ) which compensates the pressure wave effect is presented. The first injection 476 duration,  $ET_1$ , is directly calculated from the injector map using the PRail signal and the requested mass  $m_1$  as 477 inputs. On the contrary, since the injector map has been characterized in standard condition (a single injection 478 pulse was actuated),  $ET_2$  can not be obtained in the same way because of the hydraulic dynamic interference

479 between the injections. As a matter of fact, while for the first injection the PRail signal and the pressure at the

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480 injector inlet are nearly identical, when the second injection occurs the pressure facing the injector differs from 481 the PRail due to the pressure wave propagating in the feed duct. Therefore, to properly compensate for such 482 an effect, the second injection duration (ET<sub>2</sub>) needs to be calculated using as inputs the target fuel mass for the 483 second injection,  $m_2$ , and the actual average pressure for the second pulse. It is easy to understand that the cost 484 of the sensors aimed at capturing the fuel pressure near the injector (similar to the PKistInj sensor) is not 485 compatible with on-board mounting. As a result, an alternative approach, aimed at estimating the actual 486 pressure during the injection process is mandatory. As previously described, the pressure wave can be 487 accurately reconstructed starting from the 1-DoF MSD characteristics parameters mapping. Therefore, the 488 presented methodology relies only on the 3D maps previously generated which can be implemented in a 489 standard ECU without the need for any additional sensor.



490

491

Figure 24: ET2w strategy for pressure wave compensation on the injected mass on the second injection

492

#### 493 4.3 Equivalent Energizing Time Modeling for Residual Magnetization

Once the strategy for compensating pressure wave effects has been defined, to ensure proper control of the total fuel mass injected with the GDI injector, the effect of residual magnetization (RM) has to be accurately compensated. As explained in the literature [25], the electro-magnetic interactions can be compensated using a strategy based on the inversion of the previously discussed RM model, Figure 25. For each DT, the duration of the second injection needed to compensate for the residual magnetization ( $ET_{2\_corrected}$ ) has been calculated through the map reported in Figure 25, with the current DT and  $ET_2$ .

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Figure 25: Map of ET corrected

500

503 Figure 26 shows an example of RM correction for a PRail of 500 bar and  $ET_1 = ET_2$  equal to 700 µs. The DT sweep shown in Figure 26 has been performed by applying ET<sub>2 corrected</sub> as the duration for the second 504 injection pulse. It is possible to see that the RM correction methodology is capable of mitigating the 505 506 incoherences in terms of total injected mass with respect to the uncorrected condition. However, although the 507 RM compensation strategy is able to bring the value of the injected mass very close to the target value (defined 508 as twice the mass introduced with the first injection), the injected mass remains below the reference even for 509 DT values of 1500  $\mu$ s ("stable region"). As previously discussed in paragraph 4.3, this phenomenon is 510 generated by the pressure variation within the injector supply line (with respect to the PRail value). Therefore, 511 even at high DT values, the overall injected mass remains below the reference mass. The following sections 512 describe how the presented approach can be further improved through proper modelling of the pressure 513 fluctuation in the injector feed duct.

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515 Figure 26: Comparison between not corrected and corrected fuel injected mass with 500 bar injection pressure and ET of 700 μs

516

#### 517 4.4 MPW Correction Strategy

518 Cavicchi et al. [24] discussed the coupled effect of RM and pressure wave of a high-pressure GDI injector on 519 the total injected fuel in a multiple injection pattern. As mentioned before, such aspects might become critical 520 when running innovative high-efficiency combustion methodologies, characterized by a very small operating 521 range. Therefore, a novel injection management strategy aimed at compensating at the same time RM and 522 pressure waves effect (MPW correction strategy, Figure 27) has been developed and implemented in the RCP 523 system.

524 The first step is the definition of the injection pattern parameters: rail pressure, target injection mass for both 525 injection pulses and DT. From the target masses and PRail, ET<sub>1</sub> and ET<sub>2</sub> are calculated. The MPW correction 526 strategy workflow starts compensating the RM effect: the corrected ET for the second injection (ET<sub>2 corrected</sub>) 527 can be estimated starting from the ET<sub>2</sub> and the DT through the magnetization characteristic of the injector 528 (determined as described in section 4.3, when  $ET_1 = ET_2$ ). Therefore, the injector duration during the second 529 pulse is now lower than ET<sub>2</sub>, but the target mass of the second injection is guaranteed. Once the RM effect has 530 been compensated, from PRail, ET1, ET2\_corrected and DT, it is possible to estimate the pressure wave that 531 would take place in the double injection pattern. Lastly, the final injection duration to be applied (ET<sub>2 Final</sub>) 532 can be defined and sent to the injection electrical driver.

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535

Figure 27: MPW correction strategy to compensate for both effect of RM and pressure wave



Figures 28 and 29 show a comparison of different compensation methodologies with the reference injected 537 538 mass, using a pattern of two consecutive injections: no correction (purple dashed lines), only RM compensation 539 (black dashed lines), and both RM and pressure waves compensations (blue dashed lines). As it is possible to 540 see, the main correction contribution is related to the RM effect, while a secondary but not negligible 541 correction, related to the pressure wave propagation, brings back the fuel consumption across the target value 542 reported in the red dashed line for all the tested conditions (PRail = 500 bar and m<sub>tot</sub>39 mg/cycle). A maximum 543 error (calculated as the percentage distance between the estimated and the desired total mass, Figure 29) of 544 10% is achieved when only the magnetization correction is applied, and approximately 5% with both 545 magnetization and pressure wave corrections.



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547 Figure 28: Final compensation for 500 bar with a target total injected mass of 39 mg/cycle of the injected mass with magnetization
 548 and pressure wave correction (blue curve), only correction of magnetization (black curve), not corrected consumption (purple
 549 curve), reference consumption (red curve) and hydraulic fusion region (in vellow)



550

Figure 29: Final compensation error 500 bar with a target total injected mass of 39 mg/cycle of the injected mass with magnetization
 and pressure wave correction (blue curve), only correction of magnetization (black curve), not corrected consumption (purple curve)
 and hydraulic fusion region (in yellow)

554

The obtained results show that the MPW strategy is not only able to properly correct the ET value to compensate for the local pressure reduction within the supply duct and the RM effect at the same time but also capable to damp the mass oscillation (especially visible across  $DT = 600 \ \mu$ s and  $DT = 900 \ \mu$ s). In fact, the mean error when only magnetization is corrected is -5.5 % (with an RMSE of 6.1 %) while when both corrections are applied it drops to -0.9 % (with an RMSE of 1.9 %). The presented approach works properly in all the operating points outside the hydraulic fusion region (the study of hydraulic fusion is outside the scope of this paper).

562

#### 563 CONCLUSIONS

This paper presents a model-based control strategy to optimize the control of the injected fuel mass when a GDI injector performs multiple injections per cycle. To characterize the system, both electrically and hydraulically, a specially designed open vessel flushing bench was used. A wide experimental campaign was carried out to investigate several operating conditions under single and multiple injection patterns, obtained varying ET, injection pressure and DT.

The analysis of the results obtained in the case of double injections, carried out by changing rail pressure and dwell time, and keeping the duration of the two injections ( $ET_1$  and  $ET_2$ ) equal and unchanged, showed the high

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- 571 impact of dwell time on the total fuel mass injected. In particular, it was verified that the reduction of dwell
- time causes an increase in the total injected mass compared to the nominal value (i.e., to that which can be
- 573 predicted using the injector map). This aspect is mainly related to the residual energy content of the coils of
- the injector when the second pulse is triggered. The study of this residual magnetization was further
- 575 investigated by measuring, the excitation current on the coil of the injector. The experimental data showed that 576 the RM makes the current profile of the second injection rise faster during the opening phase, thus leading to
- an increase in injected mass. To improve the control of the injected mass, the impact of magnetization was
- 578 modelled (by means of a look-up table capable of representing the effect), and the model was inverted to
- 579 determine an equivalent ET, function of dwell time between injections and nominal ET of the second injection:
- this approach can compensate for the distortion of the current profile.
- The hydraulic characteristics of the system were also analyzed, mainly based on high-frequency measurements of the instantaneous pressure within the injector supply line. In particular, the pressure fluctuations generated by the first injection were modeled using the mathematical structure of an MSD system. The characteristic parameters of the MSD system were identified (from experimental data) and mapped, as a function of PRail and dwell time, for the two main carriers. The developed model, once calibrated, makes it possible to predict with good accuracy the instantaneous value of the pressure inside the injector feed line without the use of
- additional sensors (compared to those in the standard engine layout).

588 Finally, the authors developed a strategy to compensate for the combined effects of residual magnetization and 589 pressure fluctuations at the same time running multiple injections. Specifically, the RM compensation strategy is able to determine an equivalent  $ET_2$  that takes into account the distortion of the current profile, while the 590 591 instantaneous pressure modeling allows to estimate the real value of pressure upstream of the injector during 592 the second injection, which is normally lower than the pressure in the rail. The proposed approach has been validated in the DT range between the end of the hydraulic fusion region and DT equal to 1500 µs. The 593 594 presented method, easily implementable in an ECU without the need for supplementary sensors, has been 595 capable to reduce the error on the injected mass to values always lower than 5%.

596

# 597 DATA AVAILABILITY

- 598 Data will be available upon reasonable request.
- 599

# 600 UNCERTAINTIES

This section describes the information about the most important sensors used by the authors during the presented study.

## 603 - Pressure wave propagation inside the injector pipes.

Element	Value
Sensor name	Kistler 4067A
Measuring range	0-2000 bar
Overload	500 bar
Sensitivity	5 mV/bar
Linearity	$\leq \pm 0.5$
Natural frequency	>100 kHz

604

605 - Fuel Injected mass.

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Element	Value
Sensor name	AVL Balance 733s
Measuring range	0-150 kg/h
Measurement uncertainty	$\leq \pm 0.12 \%$
Maximum measurement frequency	10 Hz

#### 607 Driving current profiles.

Element	Value
Sensor name	Hioki CT6846A
Rated current	1000 A AC/DC
Frequency bandwidth	DC – 100 kHz
Max allowable input	± 1900 Apeak
Accuracy	DC: 0.2 % + 0.02%
	DC < f < 100 kHz: 0.2% + 0.01%
Linearity	<u>±</u> 20 ppm

608

609

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611

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