

Thermal effects on Common Rail injection system hydraulic performance

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(Article begins on next page)

1 Thermal effects on Common Rail injection system hydraulic 2 performance

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7 **1. Abstract**

8 The effect of the fuel temperature on the hydraulic performance of a Common Rail diesel injector has
9 been investigated with an integrated experimental-numerical approach. An experimental campaign
10 pertaining to single and double injections has been performed for fuel tank temperatures ranging from
11 28°C to 68°C. In general, an augment in the injected mass has been observed for increasing values of the
12 fuel tank temperature. Moreover, the interaction between the main and after injection changes with the
13 temperature and the dwell time threshold for fusion-free injections increases with the fuel temperature.
14 The temperature at the injector nozzle has been measured and compared with that obtained with a thermo-
15 fluid dynamics simple model, showing that the real temperature and the estimated one correlate well.
16 The influence of the fuel temperature on the internal injector dynamic has been explored by means of a
17 validated 1D numerical model of the injector thermo-fluid dynamics. The main direct effect of the
18 temperature variation concerns the needle lift, which reaches a larger peak value for a higher fuel
19 temperature: this explains the general increment in the injected mass and the augmented value of the
20 injection fusion threshold for the main-after injections. The obtained results could allow more accurate
21 open-loop control strategies for the injected mass, which include thermal effects, to be implemented.

22 **2. Introduction**

23 The fuel injection apparatus has a key role in a powertrain, since engine efficiency [1], combustion [2]
24 and pollutant emissions [3] are significantly influenced by the injection schedule and for this reason the
25 injection system further development continues to be a research challenge even for alternative fuels.

26 The Common Rail (CR) fuel injection apparatus offers great flexibility in terms of number of injection
27 events per engine cycle, dwell-times between shots and high injection pressure (up to 3000 bar) [4]. The
28 designed shape and quantity of every fuel shot affect combustion and engine-out emission control [5]

29 and both design parameters, associated to the injector structure [6], and physical factors, such as the fuel
30 properties [7] or the ambient conditions [8], must be taken into account. Fractioning the injection quantity
31 between multiple shots with close-to-zero hydraulic dwell-times according to digital rate shaping
32 strategies, can allow combustion noise to be reduced significantly [9]. An ultra-high injection pressure
33 coupled with small injection holes allow optimal level of fuel atomization, mixture homogeneity and
34 easy evaporation [10]. Soot and hydrocarbon (*HC*) emission reduction can be obtained, thanks to the
35 diminished liquid length and the consequent reduced impingement on walls, and it can be coupled with
36 a NO_x decrease if high EGR rates are simultaneously applied [11]. Besides, an optimal number of orifices
37 and an ideal angular position of them in the nozzle are able to affect the inter- and near-nozzle flow
38 patterns, thus positively affecting lift-off flame and jet interaction with the squish regions [12].

39 By investigating the influence of low temperature (for both the fuel and the injector body) it has been
40 seen that an injector equipped with a pressure balanced valve shows a pronounced reduction of the fuel
41 injected quantity under cold conditions (the reference temperature was 40 °C for the fuel and 90 °C for
42 the injector body) while a three-way valve injector features negligible differences in the injected mass at
43 800 bar of rail pressure and a progressive increment of the fuel quantity delivered beyond this rail
44 pressure level [13].

45 The effect of the injected fuel temperature has been analysed with reference to the spray development
46 and the combustion evolution. For both diesel fuel and DME (dimethyl ether), the higher is the fuel
47 temperature, the earlier is the evaporation and the lower the fuel plume penetration. This causes a lower
48 maximum combustion pressure and a more delayed start of combustion: both these effects lead to a
49 reduction in the NO_x emissions [14, 15]. Furthermore, a reduction in the fuel temperature can cause a
50 higher non-uniformity degree in the fuel-air mixture, which can increase the time necessary to reach a
51 constant level of evaporation rate and can lead to augmented emissions in unburned *HC* [16]. A
52 significant sensitivity of fuel economy and NO_x emission level to the fuel temperature at the injector
53 nozzle has been noticed even for a biodiesel fuelled engine, especially if low EGR rate is used [17].

54 When a pure biodiesel (B100), characterized by a high viscosity, is considered, the injected fuel
55 temperature can become crucial to improve the biodiesel spray performance. In fact, an increase in the
56 fuel temperature enlarges the spray angle cone and the spray area, which are usually smaller at a fixed
57 temperature if compared with normal diesel fuel. Furthermore, even the pump efficiency improves, due
58 to the reduced viscous friction losses, since fuel viscosity reduces with fuel temperature [18].

59 The flow through the injection system circuit is not isothermal. The major source of heat input to the fuel
60 in the tank during system working is the fuel recirculated from the injectors [19]. The fuel in the tank
61 reaches the injector inlet after the pumping phase, which determines a temperature increment of about 1
62 °C every 100 bar of compression [20]. Concerning the temperature evolution within the injector, it has
63 been often predicted in the literature with an isothermal flow assumption [21, 22]. However, with this
64 hypothesis, the temperature rise, due to the viscous dissipation through the injector restrictions and due
65 to the pressure drop, is not simulated [23]. The variations in density, viscosity [21] and bulk modulus
66 [24] with temperature can affect the injector behaviour. In a ballistic injector, where the needle never
67 reaches the upper stroke-end under the usual operating conditions [25], the needle dynamics significantly
68 depends on the pressure and viscous forces which are affected by the temperature [26]. As far as the
69 nozzle holes are concerned, a complex temperature pattern can be identified with 3D numerical
70 simulation, and a separated behaviour between the boundary layer and the core flow is often highlighted.
71 If a 2000 bar injection is considered, the maximum temperature difference in the flow can reach 80 °C
72 between the fluid at the core region and the one at the walls, which is a remarkable value if the
73 temperature dependence of the fuel properties is taken into account [27]. All this highlights a significant
74 role of thermal aspects in the hydraulic performance with effects on the spray.
75 In the present work, temperature effects on the injector working are analysed. With regard to the
76 hydraulic performance, the temperature at the injector nozzle hole exit has been measured and compared
77 with the one obtained by means of a simple thermodynamic model. Furthermore, a 1D numerical model
78 has been employed to investigate the injector internal dynamics dependence on the fuel temperature.
79 Numerical and experimental tests on single and double injections have been considered in a wide range
80 of nominal rail pressures under different values of the fuel temperature in the tank.

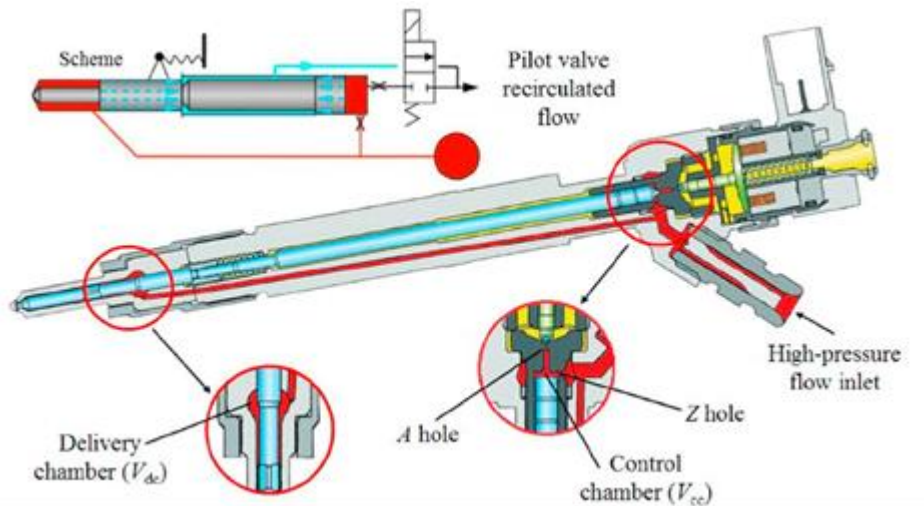
81 **3. Experimental facility**

82 Experimental tests were performed in the Internal Combustion Engines Laboratory at the Politecnico di
83 Torino by means of the Moehwald-Bosch hydraulic test bench for fuel injection systems (maximum
84 speed: 6100 rpm, maximum torque: 100 Nm, nominal power: 35 kW).

85 The selected calibration fluid is the Shell V-Oil 1404 (ISO 4113) since it is capable to satisfactorily
86 reproduce diesel oil properties up to 150 °C over the entire pressure range.

87 Tests were conducted on a Bosch CR injection system for a 4 cylinders passenger car diesel engine. A
88 high-pressure rotary pump, featuring a displacement of 430 mm³/rev and a double effect single piston, is

89 employed to feed the rail. The rail is connected with solenoid-actuated CRI 2.18 injectors, a scheme of
90 which is reported in Fig. 1. The high-pressure fuel enters the injector and a part of it reaches the delivery
91 chamber (volume V_{dc}) upstream the nozzle zone, and the other part, through the Z hole, fills the control
92 chamber (volume V_{cc}). As soon as the electrical current supplied by the Electronic Control Unit (ECU)
93 is switched on, the injector solenoid is energized and the control chamber discharges fuel from the pilot
94 valve through the A hole. Therefore, the pressure forces acting on the needle become unbalanced and the
95 latter moves up, opening the nozzle and allowing the fuel injection. When the ECU switches off the
96 current to the solenoid, the closure of the pilot valve makes the pressure rise in the control chamber,
97 therefore the needle starts its closure phase, which finishes when its tip touches the seat. Hence, the
98 nozzle is again closed and injection ends [9]. The injected fuel quantity depends on both the time-
99 averaged value of the injection pressure and on the hydraulic duration of the injection, which is related
100 to the electrical command to the solenoid, i.e. the energizing time (ET) [28].



101
102 **Figure 1. CRI 2.18 injector.**
103

104 A piezoresistive transducer has been mounted at the injector inlet along the rail-to-injector pipe to
105 measure the pressure time history ($p_{inj,in}$) and this datum will be used as a boundary condition for the
106 numerical model. The transducer measuring range is up to 2000 bar and its linearity error is lower than
107 $\pm 0.1\%$ of the fuel scale value. It has been assumed that the only uncertainty contribution that affects the
108 transducer accuracy is the linearity error, which is lower than ± 2 bar.

109 The injected flow-rate and the injected mass have been measured by means of the HDA flowmeter. In
110 this instrument the pressure in a closed volume is measured, together with the speed of sound (for which

111 an ultrasonic sensor is employed), then, by following Zeuch's method, the injected mass flow-rate is
112 obtained [29].

113 The fuel temperature at the nozzle hole exit has been measured by means of a PT100 temperature sensor
114 mounted at a distance of 25 mm from the injector tip along a duct with a counterpressure of 40 bar where
115 the fuel is injected. The same injector previously mounted in the HDA to detect the injected flow-rate
116 has then been made inject the fuel in the pipe with the PT100 sensor to measure the fuel temperature
117 under the same controlled working conditions.

118 Single injections, together with pilot-main and main-after injections, have been considered in this
119 research investigation. Tests have been performed by varying the ET at different nominal rail pressure
120 (p_{nom}) values for single injections, while for double injections the ET values for the considered injections
121 have been kept constant and the dwell time between the shots has been varied. The fuel temperature at
122 the pump inlet was varied from 28 °C to 68 °C. In particular, 28°C represents the minimum value that
123 can be controlled by the current test bench, 40 °C is usually assumed as the reference temperature for
124 hydraulic tests by the fuel injection system manufacturers and 68 °C represents the maximum
125 temperature that can be reached on the test bench for safety reasons. The hydraulic test bench can heat
126 the fuel in the tank by means of a heating coil and, at the pump inlet, the fuel temperature is controlled
127 by means of a heat exchanger where the cooling water flowrate is regulated in order to fulfil the fuel
128 temperature requirement (the fuel temperature at the pump inlet is measured by means of a temperature
129 probe that represents the feedback signal to modify the cooling water flowrate). Tests have been
130 performed at a pump speed of 2000 rpm.

131 **4. Fuel properties and temperature in the injection apparatus**

132 The temperature can affect the fuel properties significantly. The ones here considered are the density (ρ),
133 the bulk modulus (E), the dynamic viscosity (μ) and the thermal expansivity (β), which is defined as
134 $-\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_p$. Figure 2 reports the trends of the ISO 4113 oil properties, experimentally determined, as
135 functions of the temperature for different nominal rail pressures, according to [21]: symbols stand for the
136 experimental values, whereas the solid lines correspond to the determined polynomial interpolating
137 expressions.

138 The test rig can control the tank oil temperature (T_{tank}). By assuming an isentropic evolution of the fuel
139 through the pump compression, the temperature increase, due to the pressure rise, can be estimated.

140 If the fundamental thermodynamic state relation [30] is considered for an isentropic compression, one
 141 has:

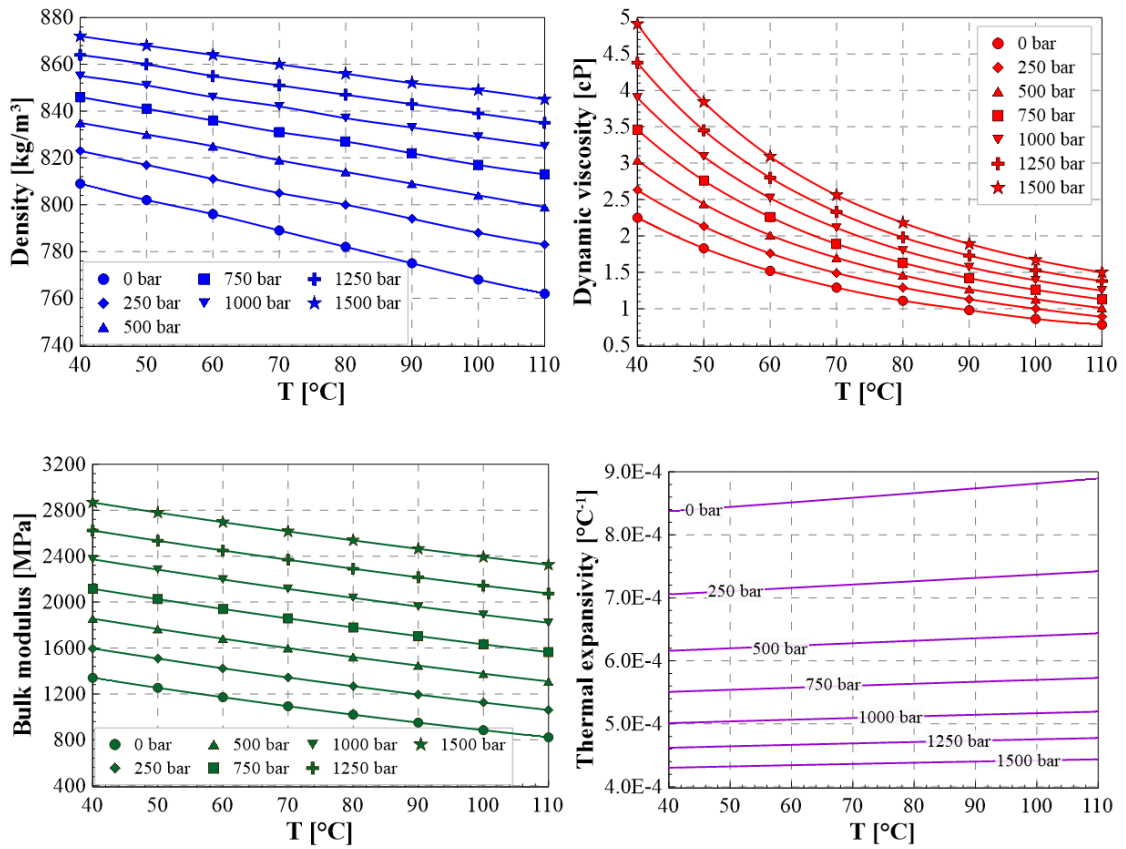
$$142 \quad dh = \frac{dp}{\rho} \quad (1)$$

143 where h is the enthalpy per unit of mass. For a liquid, the enthalpy is defined as:

$$144 \quad dh = c_p dT + (1 - \beta T) \frac{dp}{\rho} \quad (2)$$

145 where c_p is the specific heat at constant pressure. By combining Eqs. (1) and (2), one obtains:

$$146 \quad c_p dT = \beta T \frac{dp}{\rho} \quad (3)$$



147

148 **Figure 2. ISO 4113 oil properties with respect temperature and pressure.**

149

150 Hence an estimation of the temperature level in the high-pressure circuit can be obtained for a certain
 151 nominal rail pressure.

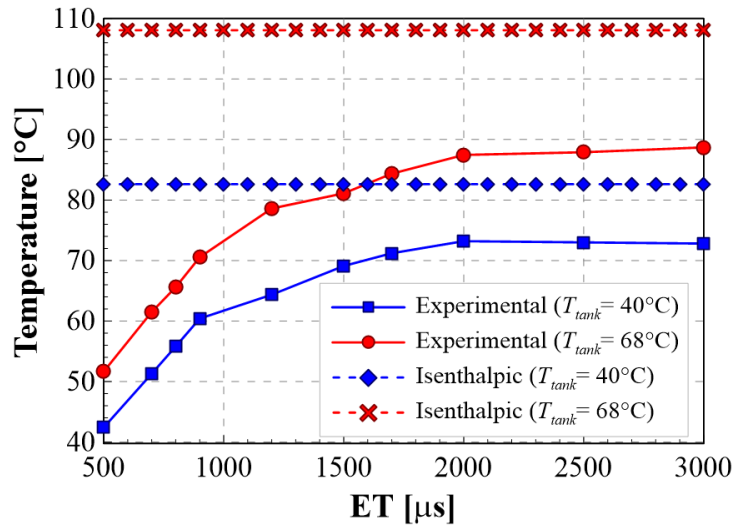
152 The fuel temperature through the injector can be estimated by considering an isenthalpic evolution [21],
 153 if the injector is roughly regarded as a set of localized restrictors. The injected fuel temperature can
 154 therefore be predicted as follows:

$$155 \quad c_p dT = (\beta T - 1) \frac{dp}{\rho} \quad (4)$$

156 **5. Experimental results**

157 Figure 3 reports the trend of the fuel temperature measured at the nozzle hole exit with respect to ET ;
158 data refers to $p_{nom}=800$ bar and to both $T_{tank}=40$ °C and $T_{tank}=68$ °C. Tests were also performed at $T_{tank}=$
159 28 °C and measure fuel temperatures resulted slightly lower than the corresponding ones at
160 $T_{tank}=40$ °C. After a certain value of ET , the measured temperature reaches an asymptotic value. This
161 can be interpreted by considering the measuring procedure used to obtain the temperature value. When
162 ET is small, the injected flow-rate is not high enough to let the sensor measure a steady-state temperature
163 in the pipe where the fuel is injected. Therefore, if the ET is increased, the detected temperature augments
164 and finally reaches an asymptotic value, which is the steady-state one that should be considered in the
165 comparison with the isenthalpic model. The asymptotic values measured for different nominal rail
166 pressures (cf. Fig. 3) have been adopted as the fuel temperatures at the nozzle outlet in the numerical
167 simulations.

168 In Fig. 4, the measured oil temperature at the nozzle exit for $ET=3000$ μ s is plotted (with symbols) with
169 respect to the correspondent isenthalpic temperature obtained for a certain rail pressure under the
170 adiabatic assumption, for both $T_{tank}=40$ °C and $T_{tank}=68$ °C. Although the curves are well distinct from
171 the bisector and the isenthalpic assumption overestimates the real temperature (the injector is less
172 dissipative than a restrictor) the estimations provided can be reasonable. Furthermore, the experimental
173 and isenthalpic temperatures are well correlated, and the correlation is almost linear (cf. the continuous
174 lines in Fig. 4). Hence, for a fixed T_{tank} and a given rail pressure level, the isenthalpic temperature at the
175 nozzle can be obtained by means of Eq. (4) and a more accurate estimation of the real temperature can
176 be obtained by means of a correlation curve which can be determined for each injector typology.

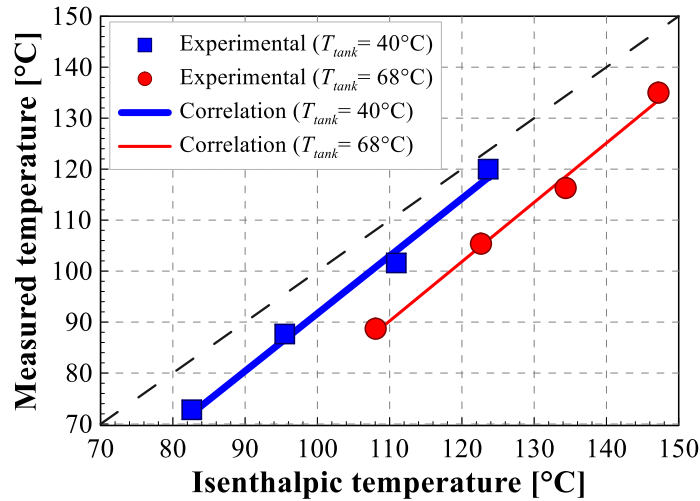


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Figure 3. Fuel temperature at the injector exit for different ET and T_{tank} values ($p_{nom} = 800$ bar).



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Figure 4. Correlation between the measured temperature at the injector outlet and the isenthalpic temperature.

184 Tests for single injections have been performed over a wide range of nominal rail pressure (between 600

185 bar and 1800 bar) and of ET (between 350 μs and 1000 μs) in order to explore the entire range of working

186 conditions for a diesel passenger car. Figure 5 shows the injector characteristics pertaining to different

187 nominal rail pressure levels and to two distinct fuel tank temperatures, namely 40 °C and

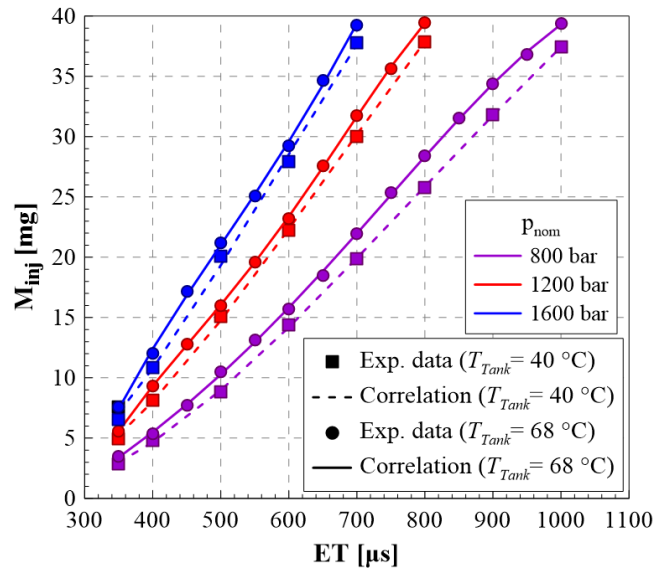
188 68 °C. As can be inferred from Fig. 5, the injected mass generally increases with the fuel temperature for

189 a fixed working condition in terms of p_{nom} and ET .

190 Figure 6 reports the injected flow-rates for $p_{nom}=1600$ bar and $ET= 600 \mu s$, at two different T_{tank} values.

191 It can be observed that the injection temporal length (ITL) increases with the tank fuel temperature, and

192 this leads to an augmented injected fuel quantity for fixed ET and p_{nom} . [31].

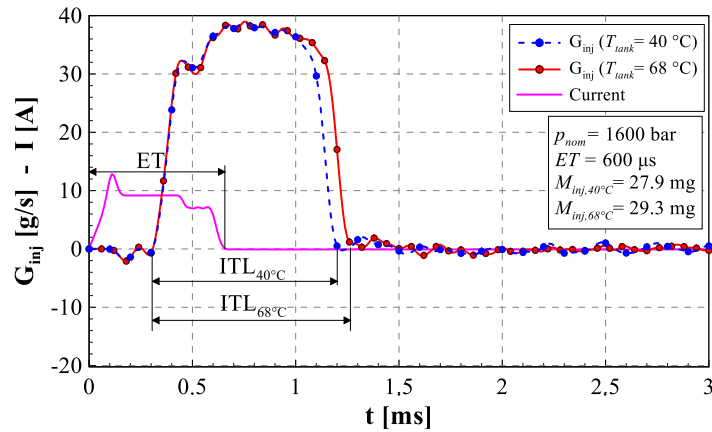


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Figure 5. Injector characteristics for different nominal rail pressure levels and fuel tank temperatures.



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Figure 6. Injected flow-rate for different fuel tank temperature values ($p_{nom} = 1600$ bar, $ET = 600$ μs).

199 As far as the double injections experimental campaign is concerned, a CRI 2.20 injector has been

200 employed. This injector, differently from the CRI 2.18 used for the analysis on single injections, features

201 a Minirail in place of the delivery chamber (the latter corresponds to volume V_{dc} in Fig. 1) in order to

202 reduce the injected mass fluctuation of the second shot with respect to the dwell time [25]. It has been

203 already verified that also the CRI 2.20 injector is affected by a thermal drift similar that of the CRI 2.18

204 injector, and an augment in the injected mass can be detected as the fuel temperature increases [31].

205 Pertaining to pilot-main injections, a DT sweep has been reported in Fig. 7, referring to $p_{nom} = 800$ bar,

206 $ET_{pil} = 230$ μs and $ET_{main} = 500$ μs . The overall injected mass $M_{inj,total}$ (given by the sum of pilot and main

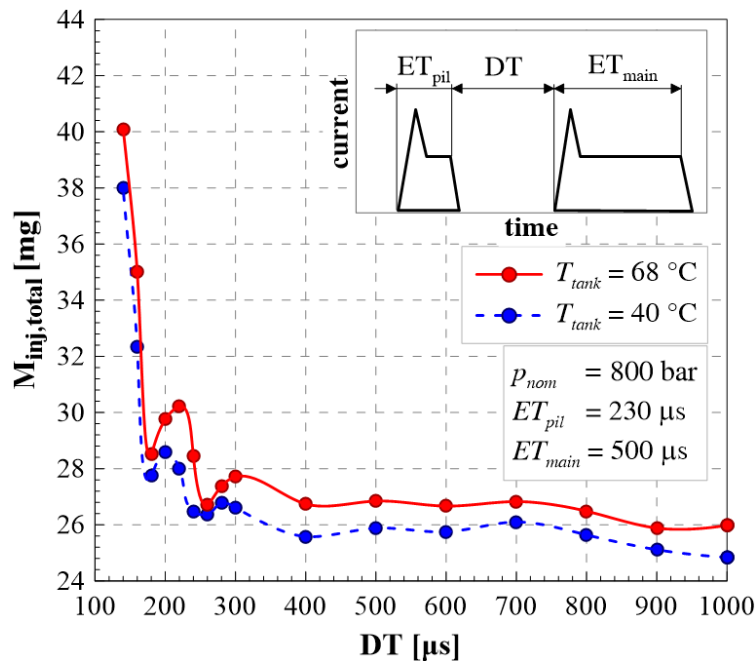
207 injected masses), measured by means of the HDA, has been reported at different DT values for both T_{tank}

208 $= 40^\circ C$ (blue curve) and $T_{tank} = 68^\circ C$ (red curve). The injected mass increases with the temperature, as

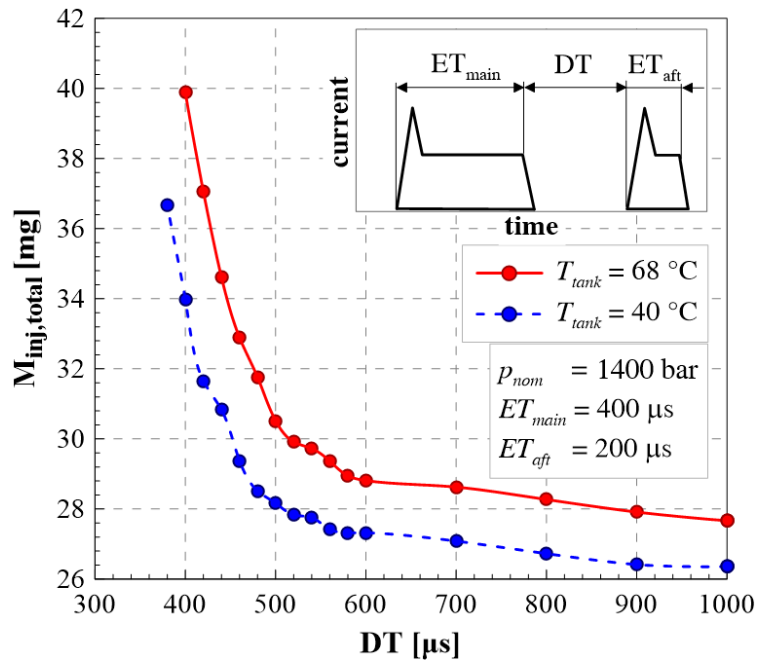
209 occurs for single injections, while the dynamic features of the system, such as the injection fusion
 210 threshold, do not vary appreciably.

211 Regarding main-after injections, the overall injected mass with respect to the dwell time value is
 212 represented in Figure 8 for the case $p_{nom} = 1400$ bar, $ET_{main} = 400$ μ s, $ET_{aft} = 200$ μ s at two different fuel
 213 tank temperatures, i.e. $T_{tank} = 40^\circ$ C (blue curve) and $T_{tank} = 68^\circ$ C (red curve). The most evident effect is,
 214 even in this case, an augment in the injected mass when the fuel temperature increases. Moreover, it can
 215 be observed that the injection fusion threshold, that is the minimum dwell-time value for which two
 216 injections are still hydraulically distinct, increases with the temperature. This is obvious in Figure 9,
 217 where the injected flowrates for the $DT = 540$ μ s case is reported for both $T_{tank} = 40^\circ$ C (blue curve, the
 218 main and after injection are still separated), and $T_{tank} = 68^\circ$ C, (red curve, where the needle never closes
 219 the nozzle holes during the entire injection event).

220 Many strategies can be implemented on the ECU to correct the injection schedules in order to compensate
 221 for the abovementioned fuel temperature effects. As an example, the main energizing time can be
 222 modified according to the engine water cooling temperature, by means of an open-loop strategy,
 223 through an empirical coefficient. The presented thermo-fluid dynamic model allows the temperature of
 224 the injected fuel, which considers the temperature increase due to both the fuel compression in the high-
 225 pressure pump and the fuel expansion across the injector, to be used to refine the open-loop control
 226 strategy of the injected mass.



227
 228 **Figure 7. Pilot-main injection DT sweep for different fuel tank temperatures.**
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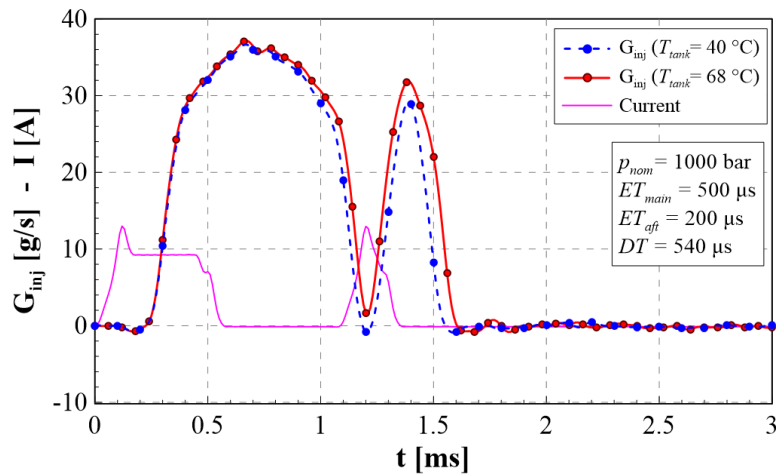


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Figure 8. Main-after injection DT sweep for different fuel tank temperatures.



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Figure 9. Injected flow-rate for a main-after injection for different fuel tank temperatures.

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6. Model description

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The analysis regarding single injection is based on a 1D injector numerical diagnostic tool developed in [32, 33] and further validated at different fuel temperatures. A scheme of the 1D model is represented in Fig. 10. The circuit generally consists of a network of 0D chambers connected by means of 1D pipes [34]. The boundary conditions consist of the measured pressure time history at the injector inlet along the rail-to-injector pipe and of the electrical current signal provided to the solenoid of the pilot valve. By selecting these boundary conditions, any inaccuracy, due to the modelling of the rail, the injector feeding pipe, the pump and the ECU, can be avoided.

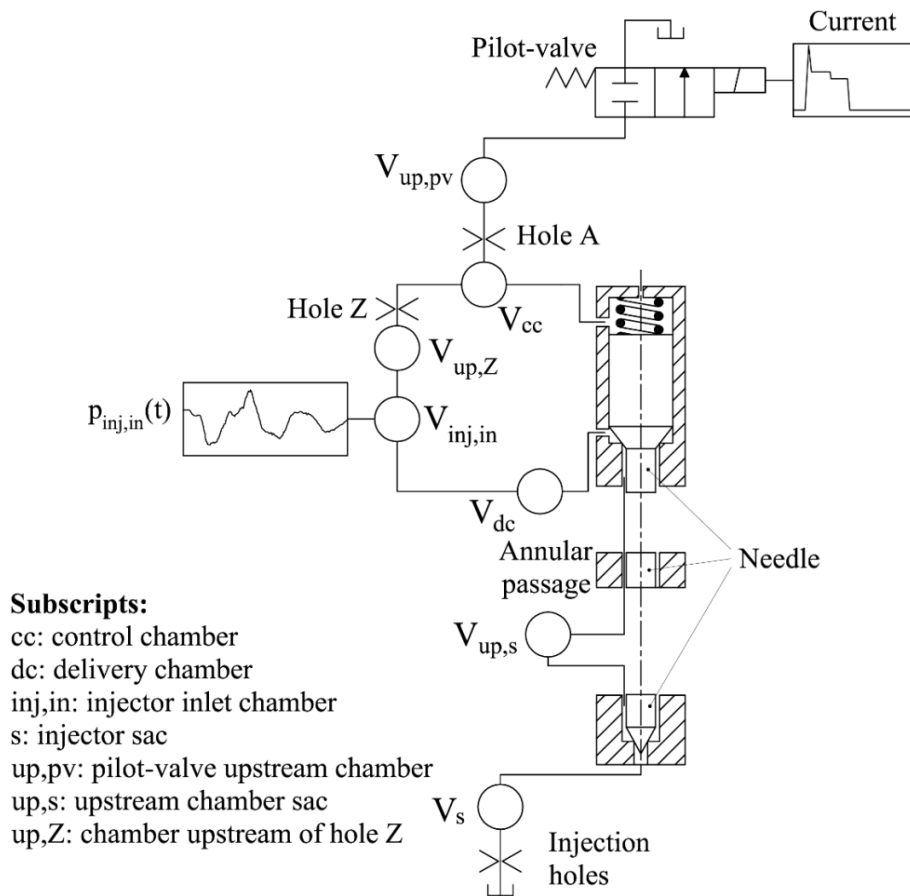
244 During the needle movement, a viscous force is generated between the needle and the injector body, due
 245 to the presence of a thin boundary layer. The flow in the small annular passage (cf. Fig. 10) is laminar
 246 and follows Poiseuille's law. The viscous friction force in this zone can therefore be evaluated as [35]:

$$247 \quad F_{fric} = \frac{\mu L_c v \pi \cdot d_n}{\delta_r} \quad (5)$$

248 where d_n is the needle diameter, L_c represents the length of the annular passage, δ_r is the radial distance
 249 between the needle and the injector holder and v is the velocity of the needle.

250 The needle movement is related to the pressure variation in the control chamber, that is to the control
 251 chamber discharge and refilling through the A and Z holes, respectively (cf. Fig. 10). The Z hole is in a
 252 non-cavitating condition, whereas the A hole can experience cavitation conditions [36]. One important
 253 point in thermal models concerns the discharge coefficient of the holes.

254



255

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Figure 10. Scheme of the 1D numerical model.

258 The discharge coefficient of a hole is defined as the ratio of the effective flow-rate to the ideal one,
 259 obtained by means of the Bernoulli equation for a liquid flow [37].

260 For a cavitation-free hole the discharge coefficient C_d has been defined as follows [38]:

$$261 \quad C_d = C_{d,max} \cdot \tanh\left(\frac{2 \cdot Re}{Re_{crit}}\right) \quad (6)$$

262 where $C_{d,max}$ is the maximum value of the discharge coefficient, Re is the Reynolds number and Re_{crit}
263 is the critical Reynolds number, where the transition between laminar and turbulent flow occurs. The
264 Reynolds number is calculated in the following way:

$$265 \quad Re = \frac{D}{\mu} \cdot \sqrt{2\Delta p\rho} \quad (7)$$

266 where D is the hole diameter and Δp the pressure drop across the hole.

267 When the flow through the hole can be subjected to cavitation phenomena, the cavitation number CN can
268 be introduced [39]:

$$269 \quad CN = \frac{p_{in} - p_v}{p_{in} - p_{out}} \quad (8)$$

270 where p_{in} , p_{out} and p_v are the pressure value at the inlet of the hole, at the outlet of the hole and the
271 vapour pressure of the fuel, respectively. When CN is lower than a critical cavitation number (CN_{crit})
272 which is experimentally obtained, cavitation occurs, and the discharge coefficient is calculated by
273 considering the actual cavitation number CN and the contraction coefficient C_c according to this
274 expression [40]:

$$275 \quad C_d = C_c \cdot \sqrt{CN} \quad (9)$$

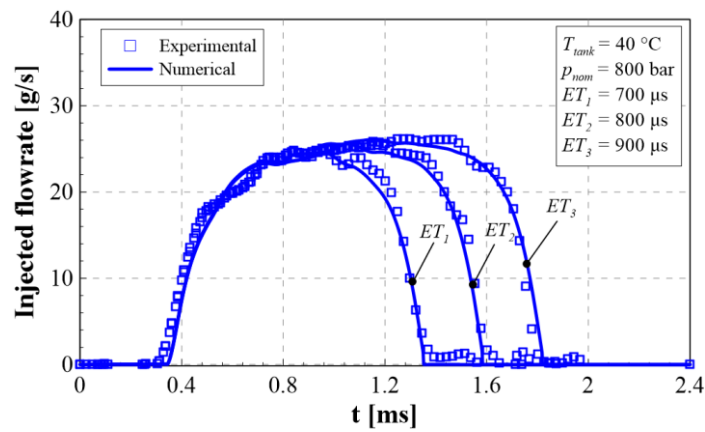
276 In the present investigation, the flow coefficient pertaining to Z hole was evaluated with Eqs. (6) and
277 (7), whereas the complete model of C_d which includes Eqs. (6-9) was implemented for A hole and for
278 the injection holes in order to simulate possible cavitation occurrences.

279 **6.1 Numerical model results**

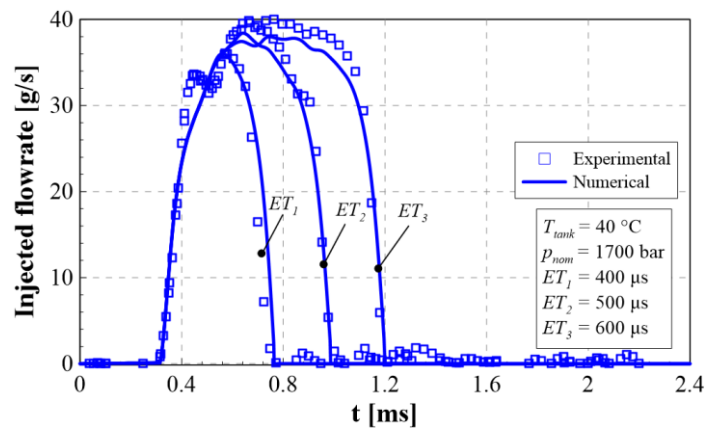
280 Figure 11 shows the comparison between the experimental injected flow-rates (plotted with symbols)
281 and the numerical ones (plotted with continuous line) for the two pressure levels 800 bar (Fig. 11a), and
282 1700 bar (Fig. 11b). The T_{tank} was set at 40°C and the results pertaining to different ET s have been
283 reported in the plots. The corresponding results referring to $T_{tank} = 68$ °C have been plotted in Fig. 12. It
284 can be observed that the numerical results match the experimental ones accurately.

285 In Fig. 13, the numerical control chamber pressure traces have been reported for both $T_{tank} = 40$ °C and
286 $T_{tank} = 68$ °C ($p_{nom} = 800$ bar and $ET = 800$ μs). As can be inferred, the difference in fuel temperature does
287 not affect significantly the pressure time history in the control chamber because both the flow-rates

288 through the Z and A holes are almost the same for the two cases. This is in line with the results available
 289 in the literature, where the effect of the fuel temperature on the control chamber pressure and flow-rates
 290 through A and Z holes are significant only when the fuel temperature within the injector is below 0° C
 291 [36]. From the numerical model it results that the A hole is experiencing cavitating condition for both
 292 the tested T_{tank} temperatures and only a small increase in the CN can be noticed for $T_{tank} = 40^{\circ}\text{C}$, leading
 293 to a slight augment in the C_d value. Instead, the Z hole doesn't undergo any cavitating regime and the
 294 flow is turbulent for both $T_{tank} = 40^{\circ}\text{C}$ and $T_{tank} = 68^{\circ}\text{C}$.
 295 Figure 14 reports the numerical pressure time histories detected in the delivery chamber (p_{dc}), in the
 296 upstream-sac chamber ($p_{up,s}$) and in the injector sac (p_{sac}) (cf. Fig. 10 for the chambers location), as well
 297



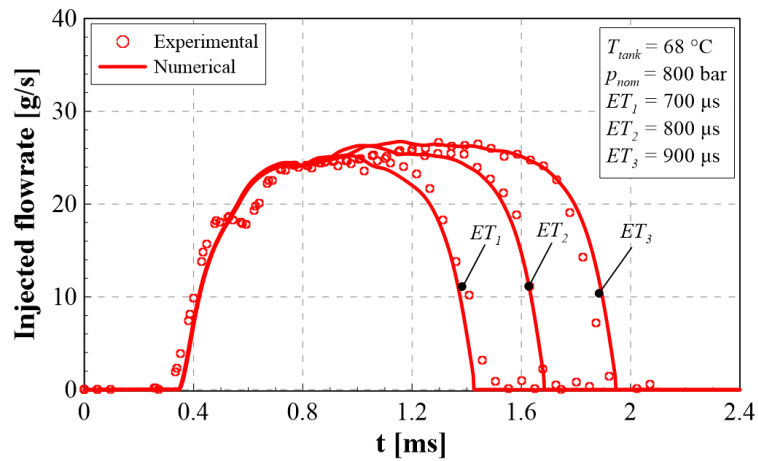
(a)



(b)

301 **Figure 11. Experimental injected flow-rates compared with the numerical ones with $T_{tank} = 40^{\circ}\text{C}$,**
 302 **$p_{nom} = 800$ bar (a), $p_{nom} = 1700$ bar (b).**
 303 as the needle lift numerical trace ($p_{nom}=800$ bar, $ET=1000$ us, $T_{tank}=40^{\circ}\text{C}$). The temperature at the injector
 304 inlet (cf. $V_{inj,in}$ in Fig. 10) is provided as an input datum to the injector model and has been obtained by
 305 means of Eq. (3). For the other injector chambers, starting from the injector inlet temperature, the
 306

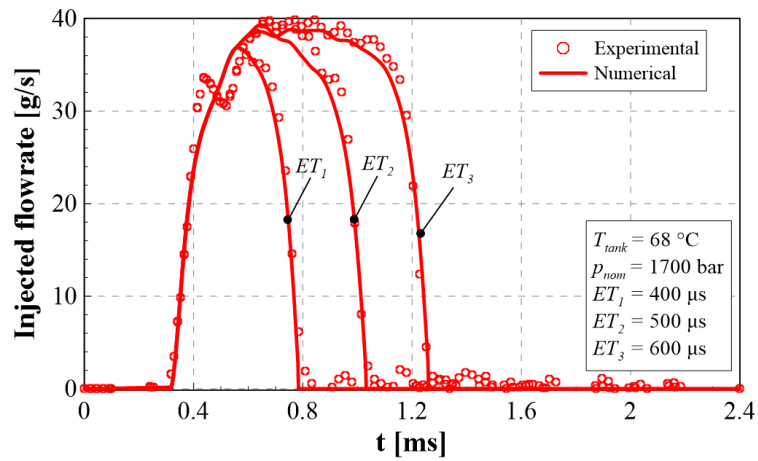
307 instantaneous pressure difference between two adjacent chambers (cf. $\Delta p_{dc \rightarrow up,s}$, $\Delta p_{up,s \rightarrow sac}$ and $\Delta p_{sac \rightarrow cyl}$
 308 in Fig. 14) is used to estimate the temperature rise under the $h=const$ hypothesis
 309 and, by means of the correlation reported in Fig. 4, an estimation of the real temperature can be obtained.
 310 If the isenthalpic temperature results to be outside from the considered range in Fig. 4, a linear
 311 extrapolation has been performed. In fact, based on data of fuel temperature measured at the nozzle exit
 312 for a wide range of p_{nom} values in [41], it was possible to extend the linear correlations reported in Fig. 4
 313 up to about 160 °C (for $T_{tank}=40$ °C) and to about 180 °C (for $T_{tank}=68$ °C).



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(a)



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(b)

318 **Figure 12. Experimental injected flow-rates compared with the numerical ones with $T_{tank} = 68$ °C,**
 319 **$p_{nom} = 800$ bar (a), $p_{nom} = 1700$ bar (b).**

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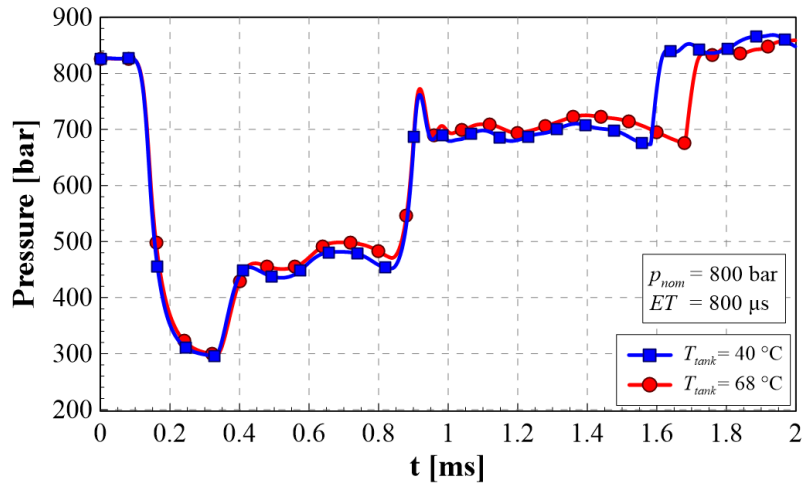


Figure 13. Pressure in the control chamber for $T_{tank} = 40\text{ °C}$ and 68 °C , $p_{nom} = 800\text{ bar}$, $ET = 800\text{ μs}$.

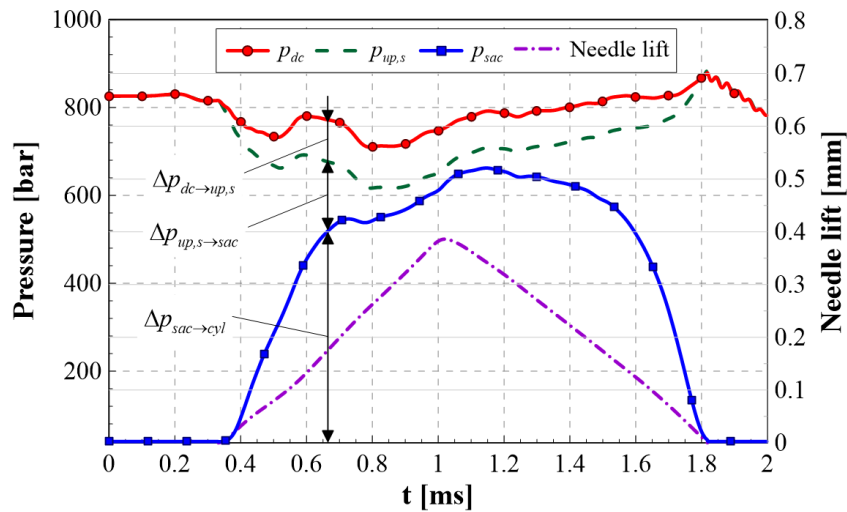
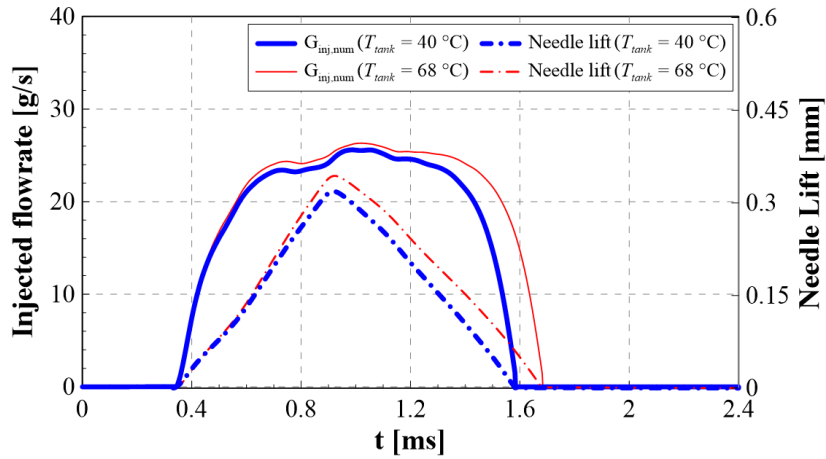


Figure 14. Pressure time histories and needle lift from the numerical model, $p_{nom} = 800\text{ bar}$, $ET = 800\text{ μs}$, $T_{tank} = 40\text{ °C}$.

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329 At the beginning and at the end of the injection event, the pressure drop $\Delta p_{up,s \rightarrow sac}$ is larger than $\Delta p_{sac \rightarrow cyl}$
330 (cf. Fig. 14) because the restricted flow area is at the needle-seat passage. Therefore, the fuel heating
331 phase mainly occurs when the fuel flows across the passage between the needle and its seat. On the other
332 hand, when the needle lift is sufficiently high and the flow-rate is controlled by the nozzle holes, $\Delta p_{sac \rightarrow cyl}$
333 becomes much larger than $\Delta p_{up,s \rightarrow sac}$, hence the fuel heating mainly occurs through the nozzle holes.
334 In Fig. 15 the numerical needle lift time distribution is reported, for both the considered T_{tank}
335 temperatures, together with the numerical injected flow-rates ($p_{nom} = 800\text{ bar}$, $ET = 800\text{ μs}$). Figure 16
336 plots the same quantities as in Fig. 15 but with reference to a different working condition ($p_{nom} = 1700$
337 bar, $ET = 600\text{ μs}$).



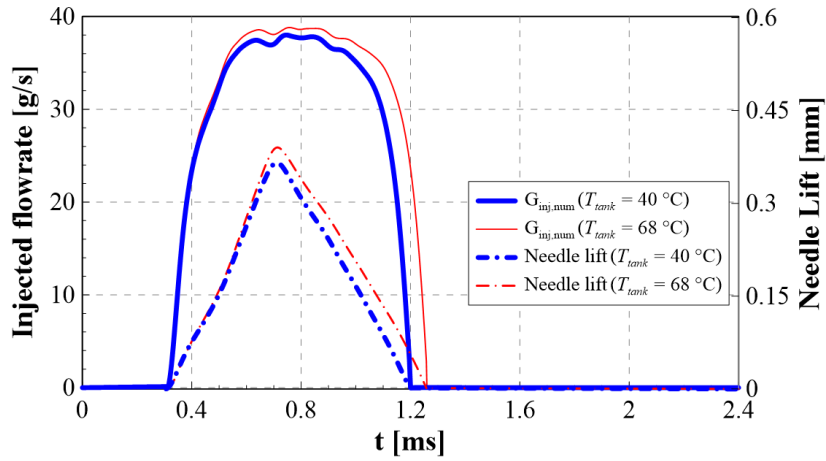
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Figure 15. Injected flowrate and needle lift for $T_{tank} = 40\text{ }^{\circ}\text{C}$ and $68\text{ }^{\circ}\text{C}$, $p_{nom} = 800\text{ bar}$, $ET = 800\text{ }\mu\text{s}$.



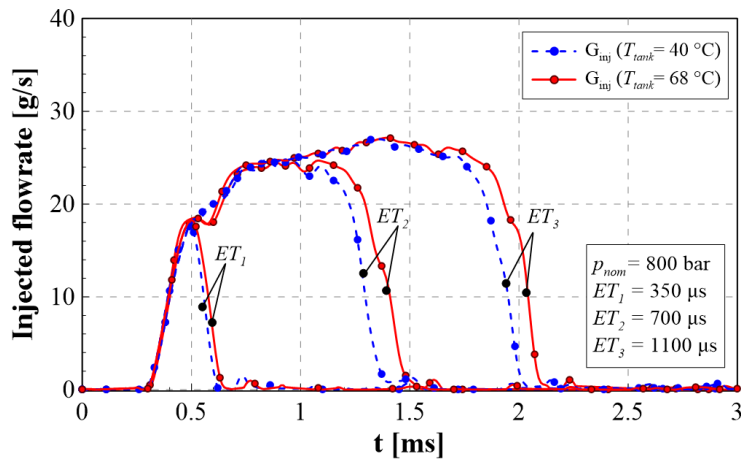
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Figure 16. Injected flowrate and needle lift for $T_{tank} = 40\text{ }^{\circ}\text{C}$ and $68\text{ }^{\circ}\text{C}$, $p_{nom} = 1700\text{ bar}$, $ET = 600\text{ }\mu\text{s}$.



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Figure 17. Injected flow-rates for different ET values with $T_{tank} = 40\text{ }^{\circ}\text{C}$ and $68\text{ }^{\circ}\text{C}$, $p_{nom} = 800\text{ bar}$. In general, since the fuel viscosity reduces with the temperature (the decreasing trend is exponential in Fig. 2), the viscous force acting on the needle diminishes as the fuel temperature augments. Therefore,

350 due to the reduced viscous force, the needle velocity increases during the opening phase as T_{tank} augments,
351 leading to a higher needle peak lift value under a fixed ET . Then, after the electrical current is switched
352 off, the needle starts its closure phase and, the higher is the temperature, the larger is the needle
353 downstroke required to close the nozzle, and therefore the longer are the nozzle closure delay and the
354 injection duration.

355 The effect of the fuel temperature is less obvious for injections characterized by small ET s, as is shown
356 in Fig. 17. In fact, a shorter- ET injection is characterized by a smaller needle lift peak value and the
357 absolute variation of the latter with T_{tank} is small. Hence, the higher is the needle lift peak value, the more
358 evident is the effect of the fuel temperature in the final part of the injected flow-rate.

359 In [36] it has been verified that, even for a CRI 2.20 injector, the main effect of the fuel temperature, in
360 the considered range, is given by a reduced viscous force acting on the needle, which leads to a higher
361 needle lift peak. Therefore, for the pilot-main injections, the higher needle lift peak obtained at higher
362 fuel temperature leads to a general increment in the mass, although the injection fusion threshold does
363 not change considerably, as shown in Fig. 7. Instead, for main-after injections, when the tank fuel
364 temperature grows, an augment in the injection fusion threshold can be detected, together with an
365 increment in the injected mass (cf. Figs. 8-9). The injection fusion threshold increment is justified, since
366 the tendency to reach higher needle-lift peak values for larger fuel temperatures becomes more evident
367 for longer injections (cf. Fig 17): unlike pilot-main schedules, the shot with the larger ET is the former
368 in main-after injections.

369

370 **7. Conclusions**

371 The fuel temperature effect on the hydraulic performance of a CR injection system has been analyzed by
372 means of both experimental and numerical results. The major findings can be summarized as follows:

373 - the experimental campaign on single and double injections (pilot-main and main-after schedules) has
374 shown that, when the fuel temperature increases, the injected mass generally grows. Furthermore, the
375 fuel temperature appreciably affects the injection fusion threshold of the main-after injections;

376 - the injected fuel temperature has been measured along a pipe where the fuel is injected, at a distance of
377 25 mm from the injector tip. This experimental temperature has been compared successfully with the one
378 obtained by means of a newly developed simple thermo-fluid dynamics model: starting from the fuel
379 tank temperature, and by considering an isentropic compression through the pump, the fuel temperature

380 at the injector entrance has been obtained. Furthermore, by means of an isenthalpic evolution assumption
381 across the injector, the final temperature at the nozzle exit has been calculated;
382 - a 1D numerical model has been used to investigate the injector dynamics at different fuel temperatures.
383 The most relevant effect of the fuel temperature increase is the reduction in the fuel viscosity, which
384 leads to a diminished viscous force that acts against the needle movement. When the fuel temperature
385 increases, the needle reaches an augmented peak value, which explains the experimental differences with
386 respect to the fuel temperature for both single and double injections;
387 - the 1D numerical model shows that the fuel heating inside the injector occurs primarily in
388 correspondence of the needle seat passage, with an effect on the beginning and end of the injection law,
389 as well as across the nozzle holes with an effect on the central part of the injection law;
390 - an innovative open-loop control strategy of the injected mass, based on the estimated nozzle exit fuel
391 temperature, could be designed to improve the compensation of the injector thermal drifts during engine
392 operation (the current strategies are usually based on the water cooling temperature).

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