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Model Optimization and Flow Rate Prediction in Electro-injectors of Diesel Injection Systems

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Abstract: Modelling and control of the injection process in common rail Diesel engines has attracted the interest of many researches also in recent years. This paper adds a new contribution for accurately modelling electro-injectors. Namely, not only nonlinear dynamics of the fuel flow and electro-mechanical aspects are considered but also a different representation of high-pressure wave propagation in pipes that feed the injector volumes. Model parameters are optimized by an evolutionary technique and a simulation study analyzes the prediction capability of the optimized model. The optimized model shows a superior fitting to experimental data than a nominal and a conventional model, and then it could be profitably used for injection control.

Keywords: Diesel engines, injection systems, electro-injectors, fuel injection control, fractional-order modeling, automotive control.

1. INTRODUCTION

In the past years, ever growing efforts were devoted to new ideas and strategies for reducing fuel and energy consumption as well as polluting emissions of automotive engines. Strict regulations and the contemporaneous competition to propose innovative and performing engines lead manufacturers to investments for developing new components or simplified control software. For the common rail (CR) Diesel injection systems, the advances in technology and combustion processes allowed a more precise metering of the air/fuel mixture demanded by the variations of engine speed and load, then a better performance and lower harmful emissions in every working condition (Guzzella, 2009; Dyntar et al., 2002; Lino et al., 2008).

However, continuous developments are required to improve the metering that poses severe specifications on the fuel injection pressure, which can be controlled with accuracy, and on the injection timing. One main issue is then to develop accurate models for representing and controlling the injectors with designed strategies. This need particularly arises for new electro-injectors that give desired flow rate shaping by multiple and close-up injections. However, complex fuel dynamics are encountered and experimental tuning is hard. Then, a model-based approach is preferable for off-line optimization of control strategies (Lino and Maione, 2013, 2014). Some literature is in (Dongiovanni and Coppo, 2010; Le et al., 2013; Shen et al., 2013).

In this paper, an electro-injector for third-generation CR Diesel engines is considered to propose a complete model for predicting the injected flow rate in many working conditions, even if nonlinearities and complex phenomena occur. The model is optimized by a differential evolution technique that compares model-based simulation with experimental data. The model accuracy could be helpful in reducing the time-consuming calibration and in improving prediction. The level of detail is not excessive to simplify the control design and allow an easy implementation.

2. THE ELECTRO-INJECTOR

The CR Diesel injection systems are based on a CR volume that receives fuel from a high-pressure pump to deliver it to electronically controlled injectors. On one side, controlling the rail pressure p_{CR} is important to accurately meter the injected fuel and reduce emissions. The specifications are met by employing the pump metering unit and by regulating a valve that is situated on the CR. On the other side, the opening/closing time intervals of the injectors must be properly defined. This is possible because the electronic control units allow a precise adjustment of these intervals. Here the focus is on the electro-injector, which ultimately determines the amount of fuel and the shape of the flow rate that is injected into the combustion chamber.

The electro-injector includes a control chamber (CC) with an electro-hydraulic valve (EV) and a feeding circuit, consisting of an accumulation volume (AV) and a small SAC volume, which is connected to the nozzles, and a mechanical coupling between a plunger and a needle (PN) (see Fig. 1). The CC and the AV are fed by the fuel coming from the CR. The fuel flows from the CC to a low-pressure chamber through the A-hole, according to the position of a shutter controlled by the EV. A high-pressure annular pipe brings fuel from the CR to the AV, then to the SAC for the injection in the cylinder through the nozzles. The PN moves up and down inside the injector, and its vertical

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displacement regulates the flow from the AV to the SAC. When the PN is in the lower position, the nozzles close; when it is in the upper position, the nozzles open and fuel is injected. The PN position depends on the resultant of applied forces. In particular, the CC pressure acts on the PN top face, while the AV pressure acts on its bottom face. In normal operations, a preloaded spring pushes the PN down in closing position. When the EV is energized, the flow through the A-hole drops the pressure in the CC, and the higher pressure on the bottom face pushes the PN up and opens the nozzles. To sum up, the injection flow can be suitably regulated by driving the EV.

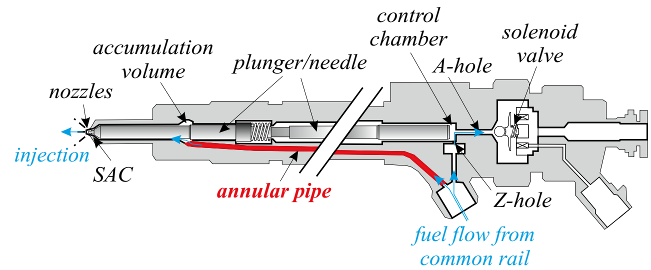

Fig. 1. The electro-injector representation

3. MATHEMATICAL MODEL

To develop an accurate but efficient in prediction and control-oriented model, it is important to select the variables that significantly affect the fuel flow and injection. Once the main independent dynamic elements and their state variables are defined, a state-space model is derived by applying fundamental laws (the continuity equation, the momentum equation, the Newton's second law of motion, etc.). A constant fuel temperature ϑ is assumed in the system, then the dynamics is described by the pressure variations in each part of the injector where the fuel flows.

3.1 Fuel Properties and Dynamics

The fuel thermodynamic properties depend on the operating conditions. The fuel volume V changes by the motion of the plunger and needle and also depends on the intake and outtake flow rates. Moreover, the fuel density, the kinematic and dynamic viscosity, the bulk modulus (the pressure increment for a decrease of a unitary volume), and the speed of sound change with the instantaneous pressure p inside each volume (Dongiovanni and Coppo, 2010):

$$\begin{cases} \rho(p,\vartheta) = k_{\rho_1} + \left(1 - e^{-\frac{p}{k_{\rho_2}}}\right) k_{\rho_3}\,(p)^{k_{\rho_4}} \\ \nu(p,\vartheta) = \left[k_{\nu_1} + k_{\nu_2}\,(p)^{k_{\nu_3}}\right] 10^{-6} \\ \mu(p,\vartheta) = \nu(p,\vartheta)\,\rho(p,\vartheta) \\ K_f(p,\vartheta) = \left[k_{b_1} + \left(1 - e^{-\frac{p}{k_{b_2}}}\right) k_{b_3}\,(p)^{k_{b_4}}\right] 10^{6} \\ c(p,\vartheta) = \sqrt{K_f(p,\vartheta)/\rho(p,\vartheta)} \end{cases} \quad (1)$$

where p is in bar, $\vartheta \approx 80°$C, and $k_{\rho_1} \approx 782.2$, $k_{\rho_2} \approx 131.3$, $k_{\rho_3} \approx 0.3935$, $k_{\rho_4} \approx 0.7156$, $k_{\nu_1} \approx 1.178$, $k_{\nu_2} \approx 0.0002$, $k_{\nu_3} \approx 1.151$, $k_{b_1} \approx 1010$, $k_{b_2} \approx 75.54$, $k_{b_3} \approx 1.204$, $k_{b_4} \approx 0.9760$. The previous relations were obtained by applying a least-squares method to fuel data (an ISO4113 air-free test oil). With a pressure of 160 MPa (1600 bar), $\rho \approx 859.5$ Kg/m^3, $\nu \approx 2.250 \cdot 10^{-6}$ m^2/s, $\mu \approx 0.0019$ Pa s, $K_f \approx 2.624 \cdot 10^9$ Pa, $c \approx 1.747 \cdot 10^3$ m/s. The nonlinearities in (1) allow better prediction than the constant parameters or linear functions (Lino et al., 2016).

In large accumulation volumes, a lumped parameters representation establishes that (Lino et al., 2005):

$$\frac{dp}{dt} = -\frac{K_f}{V}\left(\frac{dV}{dt} - Q_l + \sum_i Q_i\right) \quad (2)$$

where Q_i is an intake or outtake flow rate and Q_l is the leakage flow rate. Then, the pressure p is determined by flow rates, instantaneous volumes, and initial conditions. A flow rate Q is given by (Lino et al., 2005):

$$Q = \text{sgn}(\Delta p)\, c_d\, A_0\, n_h \sqrt{\frac{2\,|\Delta p|}{\rho}} \quad (3)$$

where A_0 is the flow section area through the holes, n_h is the number of holes, and Δp is the pressure gradient across A_0. c_d is a discharge coefficient depending on the difference between the actual and ideal flows, because the flow rate is reduced by large pressure gaps and narrow orifice sections. The leakage between mechanical elements that are in relative motion and lubricated by the fluid is:

$$Q_l = \text{sgn}(\Delta p)\, \frac{\pi\, d\, g^3}{12\, l\, \rho\, \nu}\, |\Delta p| \quad (4)$$

where d is the mean diameter of the cross-section flow area, g is the radial gap, l is the mechanical coupling length, and Δp is the pressure drop (Streeter et al., 1998).

3.2 The Annular Pipe

In the annular pipe between the CR and the AV, fuel is forced to propagate by the rail pressure, which should be kept constant to the reference value but is subject to variations determined by other components and by the injection itself. Then this variation and the mechanical action by the PN originate a pressure wave in the pipe that inevitably interacts with the injection process.

If the fluid is Newtonian and incompressible, then the wave propagation of high-pressure fuel can be represented by Navier-Stokes equations including nonlinearities that can be solved by approximation techniques. Linearization of the continuity and momentum equations yields:

$$\frac{\partial p}{\partial t} + c^2\,\rho\,\frac{\partial u}{\partial x} = 0 \quad (5)$$

$$\frac{\partial u}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial x} - (\lambda + 2\mu)\frac{\partial^2 u}{\partial x^2} = 0 \quad (6)$$

where $u = u(t,x)$ and $p = p(t,x)$ are the velocity and pressure of the wave, x is the longitudinal direction of wave propagation (the radial transversal propagation is neglected), λ is the dilatation viscosity coefficient. Known and constant fluid parameters give a simplified model (Lino et al., 2015). Here pressure dependence is considered (for $0.1 \le p \le 200$ MPa and $10 \le \vartheta \le 120\,°$C) by (1).

The Stokes' assumption ($\lambda \approx -2\mu/3$) brings to

$$\frac{\partial u}{\partial t} + \frac{1}{\rho}\frac{\partial p}{\partial x} - \frac{4}{3}\,\mu\,\frac{\partial^2 u}{\partial x^2} = 0 \quad (7)$$

Since several fluids showed dependencies of power type (Uchaikin, 2013), noninteger-order time derivatives are