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Accurate dynamic modeling of an electronically controlled CNG injection system

Paolo Lino, Guido Maione

Dept. of Electrical and Inform. Eng., Politecnico di Bari, Bari, Italy
(e-mail: {paolo.lino,guido.maione}@poliba.it)

Abstract: Compressed natural gas engines provide a non-standard solution for reducing emission of polluting gases and particulate matter. However, the gas compressibility and the complex nonlinear fluid-dynamics make the task of metering the air-gas mixture very difficult. To address this issue and to synthesize effective strategies for controlling the common rail pressure, it is necessary to develop an accurate control-oriented model that describes the relevant components, processes and dynamics. This paper presents a model-based approach that can be used for prediction, analysis of performance and control design. The developed model is validated by comparison of simulation results with experimental data from a real injection system.

Keywords: Fuel injection modelling, Compressed natural gas, Common rail injection system, Injection pressure control, Dynamic modelling.

1. INTRODUCTION

Automotive engines employing compressed natural gas (CNG) can lower the impact of harmful emissions and adhere to more and more strict regulations. Even if the combustion process in Diesel engines greatly reduced the emissions of gas and particulate matter, the interest in CNG is because of its wide availability and simple delivery by well spread infrastructure (International Gas Union, 2005; Geok et al., 2009). Moreover, reducing fuel consumption is a shared objective by manufacturers and users. At the same time, certain levels of performance must be maintained or increased. These objectives can be reached by accurately metering the air-fuel mixture (Baratta et al., 2015), which can be obtained in two ways in the common rail injection systems. One is to precisely control the opening/closing time intervals of the injectors. Another way is to control the injection pressure, even if an accurate and efficient regulation is difficult mainly because of gas compressibility. Namely, the speed variations, the load changes and the power requirements may determine substantial (and frequent) changes of the injection system working points that are necessary to inject the proper fuel amount. Moreover, parametric variations that affect the system performance and disturbances must be compensated (Baratta et al., 2015).

In this context, the aim is to improve the robustness and performance of the controllers by advanced schemes and by model-based design approaches. Namely, accurate modeling the CNG injection system is useful both for behavior analysis and prediction and for designing controllers that better regulate the rail pressure. The model must be simple enough for control design. However, even if many control-oriented models have been developed for diesel and gasoline injection systems, few regard CNG injection systems (Lino et al., 2008; Lino and Maione, 2013a,b, 2014). Instead, existing accurate models that are able to replicate the real system behavior are not suitable for model-based control design. These models are useful to improve the knowledge of the injection process, to evaluate and validate the effects of operational conditions, and to explore different configurations and alternative functional designs (see Lino and Maione (2007); Dellino et al. (2009); Misul et al. (2014); Baratta et al. (2015), and references therein).

The aim of this work is then to develop an accurate model of an electronically controlled CNG injection system, by trading off between accuracy, simplicity and suitability for control development. The approach is based on the physical equations underlying the involved processes, and requires the tuning of a minimal set of parameters. The starting point is the simple second-order nonlinear model presented in Lino et al. (2008), which includes many simplifying assumptions. Then, to improve the prediction accuracy, a new higher-order nonlinear model is developed by properly including the electro-magnetic forces, inertia of moving parts, load losses, propagation delays, and volume changes due to motion of mechanical elements.

The paper is organized as follows. Section 2 presents the CNG injection system and Section 3 describes the modeling approach. Section 4 reports on a validation analysis performed by comparing simulation and experimental data to evaluate the prediction capabilities, either in time and frequency domains. Finally, conclusions are presented in Section 5.

2. THE METHANE INJECTION SYSTEM

This paper considers an innovative CNG fuel feeding system developed by the FIAT Research Center, Valenzano branch, Italy (Amorese et al., 2004; Lino et al., 2008) shown in Fig. 1. The elements that can be distinguished

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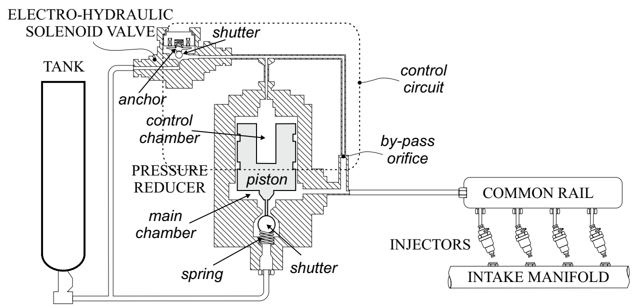

Fig. 1. Block scheme of the CNG injection system

in the system are a gas tank, a mechanical element for pressure reduction, an electro-hydraulic solenoid valve, a metering system composed of a common rail (CR) and a set of four electro-injectors. An electronic control unit (ECU) processes the sensors information and determines the commands to the valve and the injectors. Gas flows from the tank to the rail by passing through the mechanical reducer, and then is metered by the injectors for a proper air-gas mixture in the intake manifolds.

More specifically, the tank stores gas at high pressures (up to 200 bar) and delivers it to the pressure reducer (the pressure for the rail is about $5 \sim 20$ bar). To this aim, the volume of the reducer is partitioned in an upper control chamber, in which gas arrives by regulation of the electro-hydraulic valve, and a lower main chamber, in which gas arrives through an orifice that is in communication with a pipe from the tank. The flow into the control chamber changes the gas pressure on a piston moving up and down and displacing a shutter in the main chamber, whose inflow section changes by the position of this shutter. In other words, opening the valve increases the pressure in the control chamber, pushes the piston down and opens the shutter below, so that more fuel enters the main chamber and increases the rail pressure. Instead, closing the valve diminishes the pressure on the piston that moves up: in this case the shutter is closed by a preloaded spring and the rail pressure decreases. To summarize, on one side the ECU is able to regulate the rail pressure by properly driving the electro-hydraulic valve. On the other side, the ECU drives the injectors on the basis of the required speed and applied load. Since the injection flow depends on the rail pressure and the injection duration depends on the injector opening time intervals, simultaneous control of both of them allows accurate metering of the injected fuel.

3. MATHEMATICAL MODEL

To develop a model that is an efficient predictor, it is important to select the physical variables and significant phenomena that affect the rail pressure. The injection system is represented by an interconnection of control volumes in which the fuel flows, and is characterized by a uniform, time-varying, pressure distribution. The pressure in these subsystems is different and changes according to different dynamics. Once the main independent dynamic elements are determined and state variables are defined for each of them, a state-space model is derived by applying fundamental laws (the ideal gas law, the continuity equation, the momentum equation, the Newtons second law, etc.). The same temperature is assumed in all the considered volumes, because it is slowly varying and nearly constant, as it can be experimentally verified. Since the tank pressure is almost constant over a large time interval and does not change significantly during the injection process, it is considered as an input. Then state variables are: the pressures in the control chamber, the main chamber, the CR, and the pipes, the position and velocity of moving parts, and the magnetic flux of electro-valve. The main output is the rail pressure, whose value is used for feedback control.

3.1 The Pressure Reducer

The injection system control volumes are modeled by coupling the continuity equation and the perfect gas law (Zucrow and Hoffman, 1976). In particular, the perfect gas law is $p = \rho R T$, where p is the gas pressure in the control volume, ρ is the gas density, T is the temperature, R is the gas constant. Then, the time derivative of the perfect gas law and the conservation of mass inside the time varying volume V give:

$$\frac{dp}{dt} = \frac{R T}{V} \left(\dot{m}_{in} - \dot{m}_{out} - \rho \frac{dV}{dt} \right), \tag{1}$$

where $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass inflow and outflows, respectively. The mass flows through orifices and crossing sections can be assumed as isentropic transformations, and then described by the momentum equation (Zucrow and Hoffman, 1976):

$$\dot{m} = c_d \, A \, \rho_1 \, \Phi(r), \tag{2}$$

where A is the minimal crossing section, which is perpendicular to the flow direction, ρ_1 is the gas density before the crossing section, and $r = p_1/p_2$ is the ratio between the upstream and downstream pressures. c_d is the discharge coefficient that takes into account losses due to local friction and loss of kinetic energy, and it is expressed by the Perry's law as a function of the pressure ratio r (McCloy and Martin, 1980). Moreover, given the heat capacity ratio γ, Φ is a nonlinear function that increases as the ratio r decreases to describe subsonic flow for $r \geq [0.5(1+\gamma)]^{\frac{\gamma}{1-\gamma}}$:

$$\Phi(r) = \sqrt{\frac{2\gamma RT}{\gamma - 1} \left(r^{\frac{2}{\gamma}} - r^{\frac{\gamma+1}{\gamma}} \right)}. \tag{3}$$

If the ratio is below the critical value (i.e. the critical flow condition holds), i.e. $r < [0.5(1+\gamma)]^{\frac{\gamma}{1-\gamma}}$, then the flow is characterized by the sound speed, and it holds:

$$\Phi(r) = \sqrt{\gamma RT \left(\frac{2}{\gamma + 1} \right)^{\frac{\gamma+1}{\gamma-1}}}. \tag{4}$$

The previous equations can be suitably specified for each control volume. Note that the temperature is almost constant inside the injection system, then a perfect gas behaviour can be assumed. In fact, the isentropic calorific factor is almost constant and equals the inverse of the specific heat ratio (McCloy and Martin, 1980). Under this assumption, (1)-(4) give a good approximation of the gas behavior, without complicating the model unnecessarily.

The state equation for the main chamber of the pressure reducer considers the mass inflow from the tank, the mass outflow towards the CR, and volume changes due to piston motion. In this case, eq. (1) becomes