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Modelling of a Fuel Supply System for Model-based Calibration

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Abstract: The amount of functions and calibration parameters contained in engine management systems has increased dramatically in recent years, leading to high effort and time for the calibration process. Model-based calibration methods using physical models can significantly improve the efficiency of the calibration process. In this contribution a model of the high pressure part of a fuel supply system implemented in a gasoline direct injection engine is presented. The proposed model captures the basic dynamics of the fuel pressure inside the rail and is successfully used to reduce the large amount of measurements necessary for the calibration of a rail pressure controller, due to the high nonlinearity of the process.

1. INTRODUCTION

The current situation in the development of engine management systems is marked by an exponential increase in the amount of functions and calibration parameters, due to more stringent regulations concerning onboard diagnosis, the implementation of more complex engine actuators (e.g. variable valve trains) and the use of advanced control strategies to satisfy increasing requirements concerning fuel consumption and exhaust emissions.

The conventional calibration method based on grid measurements requires a large amount of measurements and thus leads to high effort and time for calibration. Model-based calibration methods can significantly improve the efficiency of the calibration procedure and save resources. Commonly used techniques include artificial neural networks (e.g. Meyer et al. 2002) and statistical methods, such as Design of Experiments (e.g. Brooks et al. 2005), which generate mathematical models of the process output using a considerably smaller subset of test bench measurements.

Here, a different approach is used to obtain the maps for proportional and integral gain of a rail pressure controller. The controller uses engine speed and load dependent maps instead of constant values to account for the high nonlinearity of the process. The calibration method is based on a physical model of the high pressure part of a fuel supply system implemented in a gasoline direct injection engine. This model is used to simulate the behaviour of the rail pressure in different operating points. Subsequently, the estimation of the maps is carried out by means of appropriate optimisation routines, independent of the way (i.e. measurement or simulation) the process data has been obtained. The estimated maps provide a basis for a self-tuning rail pressure controller. Using the presented method, the amount of cost-intensive test bench measurements can further be reduced, because most of the necessary process data is obtained by simulation and only a few measurements are necessary to adapt the model to the engine.

This contribution deals with the model used for the model-based calibration of the fuel pressure controller.

2. MODELLING

The purpose of the fuel supply system shown in figure 1 is to transport fuel from the tank into the combustion chamber with a certain pressure. The key components of the high pressure part of this system are: high pressure pump, rail and fuel injectors. The high pressure pump pressurises the fuel and delivers it into the rail. The rail serves as a storage volume and feeds the injectors, which spray fuel into the combustion chamber.

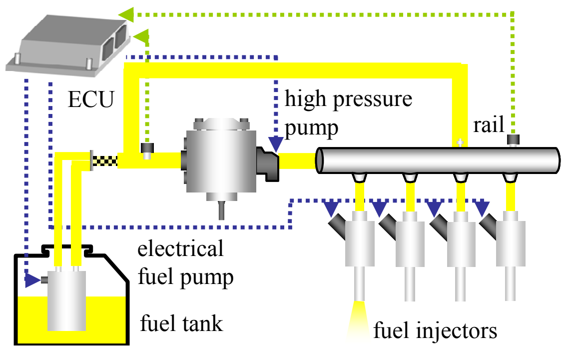

Fig. 1: Fuel supply system

Hence, the mass balance for the amount of fuel stored within the rail is given by:

$$\frac{dm_{rail}}{dt} = \dot{m}_{hpp} - \dot{m}_{inj}. \tag{1}$$

Here m_{rail} denotes the fuel mass stored within the rail, the fuel mass flow rate of the high pressure pump and through the injectors are represented by $\dot{m}_{hpp}$ and $\dot{m}_{inj}$, respectively. Considering the definition of the density ρ and assuming constant temperature $\square_{rail}$, the pressure p_{rail} of the fuel stored within the rail volume V_{rail} is given by (Blath 2006):

$$\frac{dp_{rail}}{dt} = \frac{1}{V_{rail}\dfrac{\partial \rho_{fuel}}{\partial p_{rail}}}\left(\dot{m}_{hpp} - \dot{m}_{inj}\right). \tag{2}$$

A relationship between mass density and pressure found in (Jelali et al. 2003) is used to calculate the fuel mass density ρ_{fuel} with respect to rail pressure p_{rail}. Combining this expression with a new approach based on DIN 51757 to account for the temperature dependence of the mass density leads to:

$$\frac{\partial \rho_{\mathrm{fuel}}}{\partial \mathrm{p}_{\mathrm{rail}}} = \frac{\rho_0(\vartheta_{\mathrm{rail}}) \cdot \left[\frac{\partial \kappa_{\mathrm{fuel}}}{\partial \mathrm{p}_{\mathrm{rail}}} \cdot (\mathrm{p}_{\mathrm{rail}} - \mathrm{p}_0) + \kappa_{\mathrm{fuel}} \right]}{\left[1 - \kappa_{\mathrm{fuel}} \cdot (\mathrm{p}_{\mathrm{rail}} - \mathrm{p}_0)\right]^2}. \tag{3}$$

Here $\rho_0(\vartheta_{rail})$ denotes mass density at reference pressure p_0 and current fuel temperature within the rail. It is calculated by multiplying the known mass density at reference temperature by a volume correction factor (see DIN 51757). The compressibility of the fuel is represented by κ_{fuel}.

The behaviour of the injection valves is modelled using Bernoulli's law. Assuming a constant pressure drop and constant valve opening area A_{inj}, the instantaneous fuel mass flow rate through the injectors is given by the following equation:

$$\dot{m}_{inj} = \begin{cases} c_{inj} \cdot A_{inj} \cdot \sqrt{2 \cdot \rho_{fuel} \cdot \left(p_{rail} - p_{cyl}\right)} & ,\alpha_{ib} \le \alpha_{cs} \le \alpha_{ie} \\ 0 & ,\text{otherwise}, \end{cases} \tag{4}$$

in which c_{inj} denotes the discharge coefficient, α_{cs} denotes the crankshaft angle, α_{ib} and α_{ie} represents beginning and ending of injection, based on the crankshaft angle. If the cylinder pressure p_{cyl} has not been measured, it can be approximated by the manifold pressure, in case of fuel injection during the intake stroke (according to Tomforde 2006). If p_{cyl} is neglected, the error is about one percent at the smallest rail pressure in normal operating mode (3 MPa). The discharge factor has been estimated by means of fitting (4) to the fuel mass flow rate calculated by the ECU.

The working cycle of the high pressure pump consists of the suction stroke and the discharge stroke. During the suction stroke the piston of the pump moves down from the top dead centre (TDC) to the bottom dead centre (BDC) and fuel is drawn into the pump chamber. When the piston reaches BDC, the control valve closes and the discharge stroke begins. The piston moves up and the pump's outlet valve opens, once the fuel pressure within the pump chamber has exceeded the rail pressure. The fuel flow into the rail ends with the opening of the control valve. Thus, the fuel leaves the pump chamber towards the low-pressure side of the pump and the outlet valve is closed, due to the pressure within the rail. A new suction stroke begins, when the piston has reached the TDC again. The following expression is used to calculate the instantaneous mass flow rate of the high pressure pump:

$$\dot{m}_{hpp} = \begin{cases} \rho_{fuel} \cdot \pi \cdot r^2 \cdot \dot{\alpha}_{cs} \cdot \dfrac{ds}{d\alpha_{cs}} & ,\left(\alpha_{cl} + \alpha_{cmp}\right) \le \alpha_{cs} \le \left(\alpha_{op} + \alpha_{off}\right) \\ 0 & ,\text{otherwise}. \end{cases} \tag{5}$$

Here α_{cl} and α_{op} denote opening and closing angle of the control valve, respectively. The piston radius is represented by r and s denotes the piston stroke. The quantity α_{cmp} accounts for the difference between closing of the control valve and beginning of fuel delivery into the rail, caused by the compressibility of the fuel. A new approach, based on an equation for the calculation of the pressure in a fluid filled cylinder with variable volume found in (Beater 1999), is used to estimate α_{cmp}. The quantity α_{off} accounts for differences between commanded opening angle and actual one. Its values have been estimated by fitting (5) to the fuel mass flow rate through the injectors calculated by the ECU. In case of constant rail pressure, both mass flow rates must be equal.

3. MODEL VALIDATION

Figure 2 shows measured and simulated rail pressures following a stepwise change of the control valve's opening angle. The experiments were performed at constant engine speeds and loads in open loop, i.e. with deactivated pressure controller. Measured and simulated pressures show a good agreement, especially the nonlinearity of the process is captured very well. This nonlinearity can be recognized in figure 2 by means of different rail pressure variations at 2000 and 4000 rpm due to the same change in opening angle.

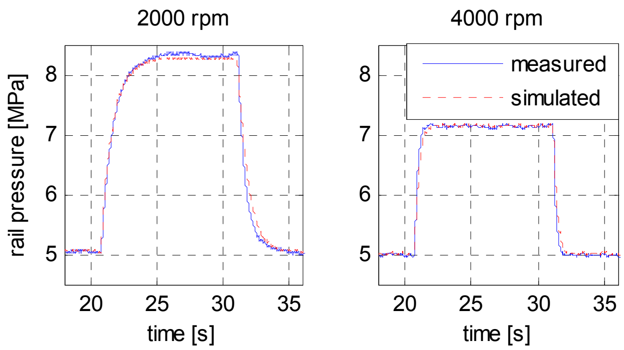

Fig. 2: Measured and simulated rail pressures

4. CONCLUSIONS

A physical model of the fuel supply system of a gasoline direct injection engine has been derived and validated. This model captures the basic dynamics of the rail pressure and requires less computing time, due to its simplicity. Currently, the presented model is successfully used for model-based pre-calibration of a self-tuning rail pressure controller and serves as a basis for the derivation of new control strategies.

REFERENCES

Beater, P. (1999) *Entwurf hydraulischer Maschinen*. Springer-Verlag.

Blath, J. P. (2006). Modelling of Fuel Pressure Dynamics. 3rd ASIM Workshop, Wismar, 67-75.

Brooks, T., G. Lumsden, H. Blaxill (2005). Improving Base Engine Calibrations for Diesel Vehicles Through the Use of DoE and Optimization Techniques. SAE 2005-01-3833.

Jelali, M., A. Kroll (2003). *Hydraulic Servo-systems*. Springer-Verlag.

Meyer, S., A. Greff (2002). New Calibration Methods and Control Systems with Artificial Neural Networks. SAE 2002-01-1147

Tomforde, M. (2006). Entwicklung von Simulationsmodellen für verbrennungsmotorische Prozesse. Diploma thesis, University of Wismar.