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Mass flow estimation with model bias correction for a turbocharged Diesel engine

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ABSTRACT

A systematic design method for mass flow estimation with correction for model bias is proposed. Based on an augmented observable Mean Value Engine Model (MVEM) of a turbocharged Diesel engine, the online estimation of states with additional biases is performed to compute the mass flows for different places. A correction method is applied, that utilizes estimated biases which are in a least-square sense redistributed between the correction terms to the uncertain mass flow maps and then added to the estimated mass flows. An Extended Kalman Filter (EKF) is tested off-line on production car engine data where the combination of an intake manifold pressure sensor, exhaust manifold pressure sensor and turbocharger speed sensor is compared and discussed in different sensor fusions. It is shown that the correction method improves the uncorrected estimated air mass flow which is validated against the airflow data measured in the intake duct.

1. Introduction

Accurate information about the air charge of turbocharged Diesel engines is important for the fueling control where the injected amount of fuel can cause a visible smoke due to incomplete combustion, if the air charge quantity is overestimated. On the other side, the performance can suffer if the fueling is too conservative due to underestimating the air charge quantity. The mass charge induced by the cylinders is defined by the air quantity from the compressor flow and the exhaust gas recirculation (EGR) mass flow. The quality of the induced mass charge is defined by the air–fuel ratio. The information about the true amount of air charge is dimmed by premixing the EGR mass flow to the cylinders, containing the burned gas and unburned fresh air. The percentage of the fresh air in the EGR mass flow can be estimated via knowledge of burned gas fraction dynamics (Diop, Moraal, Kolmanovsky, & Van Nieuwstadt, 1999; Wang, 2008). The most desirable estimation technique combines the mass flow estimation (especially the EGR flow) and the oxygen concentration estimation to estimate the complete information about the air charge for a turbocharged Diesel engine (Kang, Haskara, Chang, & Wang, 2011) or for direct injection spark ignition engine (Stotsky & Kolmanovsky, 2002). These two problems of in-cylinder mass flow estimation and in-cylinder oxygen concentration estimation are often treated as independent estimation problems. The estimation of air charge for turbocharged Diesel engines with no EGR system with measured intake pressure (MAP) and intake temperature is reported in Storset, Stefanopoulou, and Smith (2004). For the same kind of engine, the estimation of air charge through the adaptation of volumetric efficiency based on the upstream compressor airflow measurement (MAF) and MAP measurement is documented in Stefanopoulou, Storset, and Smith (2004). The compressor flow to the intake manifold can be estimated with MAP sensor and exhaust manifold pressure (EXMP) sensor (Kolmanovsky, Jankovic, Van Nieuwstadt, & Moraal, 2000; Polóni, Rohaľ-Ilkiv, Alberer, del Re, & Johansen, 2012; van Nieuwstadt, Kolmanovsky, Moraal, & Stefanopoulou, 2000), where van Nieuwstadt et al. (2000) base the EGR control loop on the estimated compressor flow information. The improved mass flow estimation (in-cylinder mass flow, compressor mass flow and EGR mass flow) accuracy is achieved with additional measurement of the turbocharger speed (NTC) (Höckerdal, Frisk, & Eriksson, 2009; Polóni et al., 2012). The MAF sensor adaptation based on the MAF measurement and turbocharger speed signal is studied in Höckerdal, Frisk, and Eriksson (2011), where the measured air-mass flow with a production MAF sensor is imprecise and requires corrections. A promising approach to estimate the quantity of the induced charge is in application of the in-cylinder pressure measurement, however so far the experiments are reported under steady state conditions (Desantes, Galindo, Guardiola, & Dolz, 2010).

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The precision of the engine's mass flow estimation is characterized by the magnitude of the error. A model-based mass flow estimation algorithm relies on engine model precision which is sometimes uncertain due to map-based steady state Mean-Value-Engine-Model (MVEM) identification principles (Eriksson, Wahlström, & Klein, 2010; Jung, 2003) and engine aging. The adaptation scheme (Stefanopoulou et al., 2004), for engines without the EGR, considers the adaptation of the volumetric efficiency where adaptation scheme for a naturally aspired spark ignition engine with EGR in Kolmanovsky, Sivergina, and Sun (2006) suggests to adapt in-cylinder mass flow based on the MAF and MAP measurements. An approach founded on the linearized state-space model augmentation with variable structure of model bias terms is proposed in Höckerdal et al. (2009) where the augmentation with a given structure of model bias terms is proposed in Polóni et al. (2012). The augmentations (Höckerdal et al., 2009; Polóni et al., 2012) aim to improve the estimation of the intake manifold pressure, exhaust manifold pressure and turbocharger speed by additional estimation of the bias terms entering the nonlinear state-space model.

For the air mass flow estimation problem, different possibilities exist:

- the use of a MAF sensor
- the use of a MAF and MAP sensor with an observer/model
- the use of an Extended Kalman Filter (EKF) and MVEM with fusion of different sensor sets (MAP, EXMP, NTC) (as presented in this paper)
- the use of an exhaust lambda sensor (EGO) and injection signal for estimating the air mass flow
- other combinations of the previous ones

This work presents novelty in utilization of the estimated biases for the mass flow computation and presents the correction method that is experimentally tested on the turbocharged Diesel engine data. As we have focused on the bias modeling and redistribution of the bias error to mass flow maps, the optimal choice of estimation algorithm is not emphasized in our research. The Extended Kalman Filter (EKF) is chosen for estimation since it is a widely established and well known engineering tool. The computation of the mass flow correction vector is formulated as a least-squares problem under the assumption that map-based mass flow computations are the most uncertain parts of the model. Further contribution of this paper is the methodology of using the EKF, MVEM and the sensor fusion for on-board estimation of (not only) upstream compressor airflow, allowing to omit the MAF sensor.

2. Turbocharged engine system description and Mean-Value Engine model

The simplified schematic turbocharged Diesel engine setup considered in this study is shown in Fig. 1. According to this figure, the engine is modeled with the compressor (c), turbine (t), final volume intake manifold (i), final volume exhaust manifold (x), intercooler (ic), EGR cooler (co), EGR pipe and the engine cylinders (e). The model has five states. The first four states represent the mass dynamics and they are: p_i intake manifold pressure [kPa], m_i intake mass [kg], p_x exhaust manifold pressure

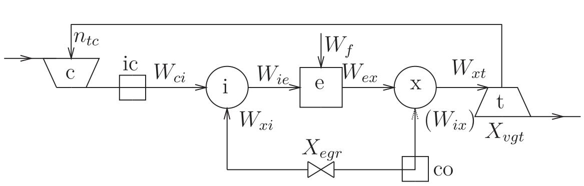

Fig. 1. Schematic turbocharged Diesel engine model representation.

[kPa] and m_x exhaust mass [kg]. The fifth state is the turbocharger speed n_{tc} [rpm]. The inputs are EGR valve position X_{egr} [% open], VGT actuator position X_{vgt} [% open], injected fuel mass W_f [kg s^{-1}] and the engine speed n_e [rpm].

The engine model is represented by the following mass and energy balance differential equations in the intake and exhaust manifolds, and by the torque balance at the turbocharger shaft (Kolmanovsky, Moraal, van Nieuwstadt, & Stefanopoulou, 1998)

$$\dot{p}_i = \frac{R\kappa}{V_i}\left(W_{ci}T_{ci} - W_{ie}T_i + W_{xi}T_{xi} - \frac{\dot{Q}_1}{c_p}\right) \tag{1}$$

$$\dot{m}_i = W_{ci} - W_{ie} + W_{xi} \tag{2}$$

$$\dot{p}_x = \frac{R\kappa}{V_x}\left((W_{ie} + W_f)T_e - (W_{xi} + W_{xt})T_x - \frac{\dot{Q}_2}{c_p}\right) \tag{3}$$

$$\dot{m}_x = W_{ie} + W_f - W_{xi} - W_{xt} \tag{4}$$

$$\dot{n}_{tc} = \left(\frac{60}{2\pi}\right)^2 \frac{1}{J}\left(\frac{P_t - P_c}{n_{tc}}\right) \tag{5}$$

The symbols in these equations are either input variables, state variables, constants or functions of the five states and inputs. The index associated with each variable defines the location of the variable. In the case of two indexes the first one is the upstream and the second one is the downstream location. The mass flows are denoted as W [kg s^{-1}]. The EGR mass flow is modeled as W_{xi} where alternatively the backflow can be considered as W_{ix} however, under standard operating conditions $W_{ix} = 0$. The temperatures [K] in the intake and exhaust manifolds are T_i and T_x. The differences of the static and dynamic pressures and temperatures are neglected because of the low mass flows. The constant parameters for the model are the intake manifold volume V_i [m^3], exhaust manifold volume V_x [m^3], specific heats at constant pressure and volume [J kg^{-1} K^{-1}] c_p, c_v, isentropic exponent of air $\kappa = c_p/c_v$, specific gas constant of air $R = c_p - c_v$ and the turbocharger inertia J [kg m^2]. The heat losses in the Eq. (1), $\dot{Q}_1$ and Eq. (3), $\dot{Q}_2$ are assumed to be zero. For the intake manifold, this is a reasonable assumption since the temperature is not much different than the ambient temperature. However in the case of exhaust manifold the heat loses are significant. The exhaust gas heat loss is implicitly captured in the steady state (look-up table) out-flow engine temperature model T_e that is discussed later.

The following equations summarize the dependencies of intermediate variables in Eqs. (1)–(5). Some of these dependencies are expressed as the look-up tables obtained by fitting the steady-state experimental engine data to a second order polynomial surface with parameter vector Θ. The others are obtained by physical relations. The air mass flow from compressor to the intake manifold W_{ci}, pressure after compressor p_c and the compressor efficiency η_c are mapped as (Jung, 2003)

$$W_{ci} = f_{W_{ci}}(p_i, n_{tc}, \Theta_{W_{ci}}) \tag{6}$$

$$p_c = f_{p_c}(W_{ci}, n_{tc}, \Theta_{p_c}) \tag{7}$$

$$\eta_c = f_{\eta_c}(W_{ci}, p_i, \Theta_{\eta_c}) \tag{8}$$

with fitted polynomial surfaces characterizing the turbocharged Diesel combustion engine. The look-up tables are 2-D functions with a linear interpolation capability with grid data generated from the fitted polynomial surfaces.

The temperature after the compressor is given by

$$T_c = T_a + \frac{1}{\eta_c}T_a\left[\left(\frac{p_c}{p_a}\right)^{(\kappa-1)/\kappa} - 1\right] \tag{9}$$