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A Combustion Cycle Model for Stationary and Transient Engine Operation

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Abstract: Since novel approaches of diesel engine control have to balance the emissions and the generated torque most accurately during highly transient engine operation, exact cycle-by-cycle estimates of these engine outputs are mandatory. Accordingly, this paper develops a control-oriented combustion cycle model to determine the indicated mean effective pressure as well as the nitrogen oxide and the smoke emissions. The proposed model has a hybrid structure, i.e. the gas exchange and compression are modeled physics-based with lumped mass and energy balances, whereas the combustion is approximated by a data-based approach due to the complex chemical reaction kinetics. Thus, the model outputs are described with respect to the time-varying signals of the transient air path conditions (intake pressure, oxygen concentration, and temperature) as well as the inputs adjusted by the fuel injection (start position and mass of each fuel injection pulse). Measurement data is used for the parametrization and validation of the model.

Keywords: Combustion modeling, Gaussian processes, Transient engine operation

1. INTRODUCTION

Modern diesel engines must accomplish complex and partially contrary goals in stationary and transient operation, e.g. generating the desired torque, emitting low amounts of emissions and having a high efficiency. Thus, auxiliary components such as turbochargers, exhaust gas recirculation or high pressure fuel injection are utilized (Heywood, 1988; Mollenhauer and Tschoke, 2010). These parts are summarized to subsystems such as the air, the combustion, the injection, and the mechanical system, see Fig. 1.

The control of the entire plant requires the coordination of the various subsystems (Isermann, 2014). During transients this task becomes challenging since each exhibits its own dynamics (Asprion et al., 2014; Grahn et al., 2014; Mrosek et al., 2010). As the fuel injection (i.e. the position and mass of the fuel injection pulses) affects the combustion fast and directly, it represents a cycle-by-cycle manipulated input for the combustion system. In contrast, the air system conditions (i.e. the intake and exhaust pressures, temperatures or oxygen concentrations) change slowly due to turbocharger inertia and gas transport delays (Nakayama et al., 2008; Hillion et al., 2009). For the combustion system operating on a cycle-by-cycle basis, the air system inputs thus represent time-varying conditions. Therefore, the engine control scheme has to determine the fuel injection parameters with respect to the air system state to achieve an optimal trade-off between the transient emissions, torque, and efficiency. Thus, control oriented models are required to determine the engine outputs in dependence of the manipulated inputs and the time-varying signals. Such models are challenging as they must precisely predict the cylinder filling during transients and describe the chemical reaction kinetics during the combustion of multiple fuel injections (Warnatz et al., 2006).

The model presented in this paper predicts the indicated mean effective pressure (IMEP) reflecting the generated torque, the nitrogen oxides (NO_x), and the smoke (SM) on a cycle-by-cycle basis. To do so, a cylinder chamber model is set up with a hybrid structure. This means that the gas exchange and the compression phase are described by physics-based lumped mass and energy balances whereas the combustion phase is approximated by a data-based modeling approach (Gaussian Process Regression) due to the complex reaction kinetics. Thus, an efficient way of describing the combustion of multiple fuel injections utilizing a global non-parametric regression approach (Berger et al., 2011) is combined with a precise physics-based filling estimate, which extends the well-known mean-value filling estimation (Heywood, 1988) and provides an adequate time-continuous interface for models describing the air system delay and dead-time effects (Rowland, 1986; Meddahi et al., 2015; Kotman et al., 2009). The hybrid model scheme and the selection of the inputs of each submodel is motivated by the intrinsic structure of the cyclic cylinder operation interacting with the air and injection system.

The paper is structured as follows: In Section 2 the physics-based cylinder chamber model is introduced with respect to the engine subsystem interaction. In Section 3 the model is analyzed from a system-theoretic perspective and applied to the phases of the engine cycle, whereas the data-based approximation of the combustion is discussed. Section 4 verifies the model based on stationary and transient test bench data. Finally, Section 5 concludes the paper.

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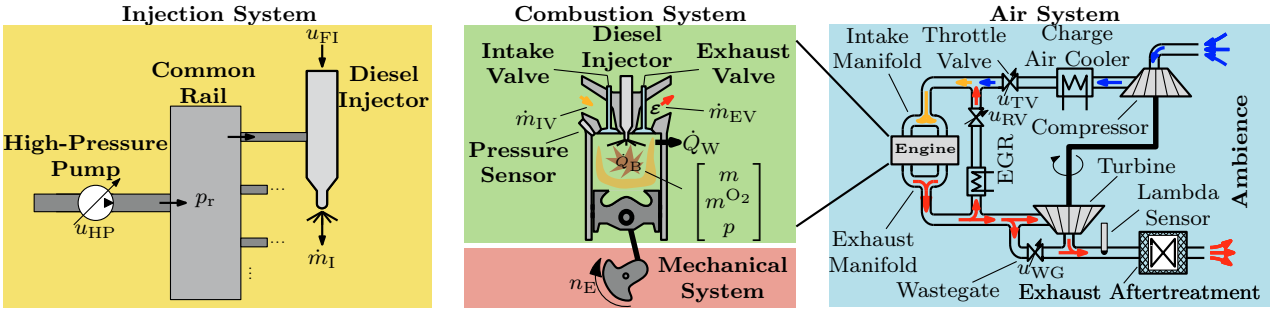

Fig. 1. Scheme of the internal combustion engine with the combustion, air, injection, and mechanical subsystems.

2. PHYSICS-BASED CYLINDER MODEL

The control-oriented cylinder chamber model requires on the one hand an appropriate description of the inputs and internals to accomplish the desired accuracy and on the other hand needs to be set up as a low-order model with respect to implementation conditions and controller design. A corresponding cylinder chamber model is described in Subsection 2.1, whereas Subsection 2.2 discusses the calculation of the IMEP p_{mi}, the nitrogen oxide emissions $\mathcal{E}_{\mathrm{NO_x}}$, and the smoke emissions $\mathcal{E}_{\mathrm{SM}}$.

2.1 First-Principle Cylinder Chamber Model

The model considered in this paper describes the cylinder chamber by means of a single, ideally mixed thermodynamic zone (Pischinger et al., 2009). This lumped parameter approach comprises an overall and a component mass balance as well as an energy balance to describe the cylinder state variables mass $m(t)$, oxygen mass $m^{\mathrm{O_2}}(t)$, and pressure $p(t)$, see Fig. 1. Here, two chemical fractions $f \in \{\mathrm{O_2}\,, \overline{\mathrm{O}}_2\}$ are distinguished, namely oxygen and non-oxygen. Since the data-based approximation of the combustion phase does not describe the fuel oxidation over time and it is assumed that no fuel remains after the stoichiometric combustion, diesel fuel (fl) has no balance equation. Further, the impact of the emissions $\mathcal{E}_{\mathrm{NO_x}}$ and $\mathcal{E}_{\mathrm{SM}}$ on the overall cylinder mass balance is neglected. Nevertheless, their generation is considered within the equation of the outputs discussed in Subsection 2.2.

The overall mass balance is given by

$$\frac{\mathrm{d}m(t)}{\mathrm{d}t} = \dot{m}_{\mathrm{IV}}(t) - \dot{m}_{\mathrm{EV}}(t) + \dot{m}_{\mathrm{I}}(t) \tag{1}$$

with the mass flows $\dot{m}_v\,,\ v \in \{\mathrm{IV}, \mathrm{EV}\}$, through the intake valve (IV) and exhaust valve (EV) as well as the injected fuel mass flow $\dot{m}_{\mathrm{I}}$. The oxygen mass balance is

$$\frac{\mathrm{d}m^{\mathrm{O_2}}(t)}{\mathrm{d}t} = \dot{m}^{\mathrm{O_2}}_{\mathrm{IV}}(t) - \dot{m}^{\mathrm{O_2}}_{\mathrm{EV}}(t) - \dot{r}^{\mathrm{O_2}}(t) \tag{2}$$

with the oxygen mass flow $\dot{m}^{\mathrm{O_2}}_v = X^{\mathrm{O_2}}_v\,\dot{m}_v$ and the oxygen mass fraction $X^{\mathrm{O_2}}_v$ which depend on the flow direction of $\dot{m}_v$. The oxygen demand of the combustion is described by $\dot{r}^{\mathrm{O_2}} = \mu^{\mathrm{O_2}}\,\dot{m}_{\mathrm{I,b}}(t)$ with the stoichiometric factor $\mu^{\mathrm{O_2}}$ and the burned fuel mass $\dot{m}_{\mathrm{I,b}}$.

An accurate prediction of the overall and the oxygen mass requires an exact calculation of the mass flows through the inlet and the outlet valves. Therefore, both mass flows are described by means of detailed throttle equations

$$\dot{m}_v = \alpha_v\,A_v\,p_{\mathrm{u}}\sqrt{\frac{2}{R\,T_{\mathrm{u}}}}\,\psi(p_{\mathrm{u}}, p_{\mathrm{d}}, \kappa) \quad \text{with} \quad p_{\mathrm{u}} > p_{\mathrm{d}} \tag{3}$$

$$\text{and} \quad \psi(p_{\mathrm{u}}, p_{\mathrm{d}}, \kappa) = \sqrt{\frac{\kappa}{\kappa-1}\left(\left(\frac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\right)^{\frac{2}{\kappa}} - \left(\frac{p_{\mathrm{d}}}{p_{\mathrm{u}}}\right)^{\frac{\kappa+1}{\kappa}}\right)}$$

depending on the up- and downstream pressure p_{u} and p_{d} as well as on the upstream temperature T_{u}. Further parameters are the adiabatic index κ, the cross-sectional area A_v and the flow coefficient α_v (Guzzella and Onder, 2004). It is worth noting that equation (3) changes sign for an inverse pressure gradient. With respect to (1)-(3) the intake and exhaust pressures $p_{\mathrm{IM}}, p_{\mathrm{EM}}$, the temperatures $T_{\mathrm{IM}}, T_{\mathrm{EM}}$ and the oxygen mass fractions $X^{\mathrm{O_2}}_{\mathrm{IM}}, X^{\mathrm{O_2}}_{\mathrm{EM}}$ are assumed to be known e.g. by measurements as depicted in the test bench scheme in Fig. 6 or by respective models.

The fuel mass flow $\dot{m}_{\mathrm{I}}$ consisting of two pulses, see Fig. 2, is approximated by a sequence of heavy-side functions $\Theta(\cdot)$

$$\dot{m}_{\mathrm{I}} \approx \sum_{i=1}^{2} \frac{m^{\mathrm{I}i}}{\Delta t^{\mathrm{SI}i}}\left(\Theta(t - t^{\mathrm{SI}i}) - \Theta(t - (t^{\mathrm{SI}i} + \Delta t^{\mathrm{SI}i}))\right). \tag{4}$$

The variables $t^{\mathrm{SI}i}$ and $\Delta t^{\mathrm{SI}i}$ define the start and duration of the fuel pulses with $\Delta t^{\mathrm{SI}i} = f_{\Delta t}(m^{\mathrm{I}i}, p_{\mathrm{r}}(t))$ depending on the fuel mass $m^{\mathrm{I}i}$ and the fuel pressure $p_{\mathrm{r}}(t)$, see Fig. 1. The start time $t^{\mathrm{SI}i}$ refers to the piston top dead center (TDC). It is worth noting that the parametrization of the fuel mass flow $\dot{m}_{\mathrm{I}}$ (4) may equivalently be expressed by means of the crank angle $\varphi^{\mathrm{SI}i}$ instead of $t^{\mathrm{SI}i}$.

The ODE (ordinary differential equation) of the cylinder pressure $p(t)$ is derived from the energy balance as well as the ideal gas law and holds as

$$\frac{\mathrm{d}p(t)}{\mathrm{d}t} = \frac{R}{V\,c_{\mathrm{v}}}\Big[\underbrace{\dot{Q}_{\mathrm{B}}}_{(A)} + \underbrace{\Psi + \Omega}_{(B)} - \underbrace{\frac{c_{\mathrm{p}}}{R}\,p\,\dot{V}}_{(C)}\Big], \tag{5}$$

$$\psi = \dot{m}_{\mathrm{I}}\left(h^{\mathrm{fl}}(\tilde{T}) - h^{\mathrm{fl}}(T) + \frac{c_{\mathrm{p}}\,R^{\mathrm{fl}}\,T}{R}\right),$$

$$\Omega = \textstyle\sum_v \sum_f \dot{m}^f_v\left(h^f(\tilde{T}) - h^f(T) + \frac{c_{\mathrm{p}} R^f T}{R}\right).$$

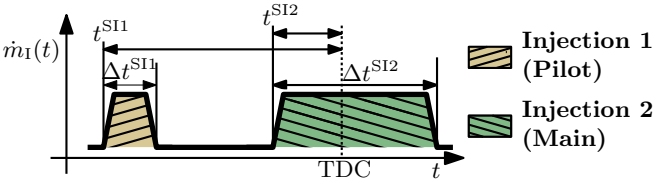

Fig. 2. Fuel mass flow profile with two injection impulses.