

Modeling Post-Injections for HiL-Simulations

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ABSTRACT

In this paper a model of the diesel combustion is introduced, which can simulate post-injections for the regeneration of diesel particulate filters. The combustion process is modeled with two phases. The intended use is a HiL-simulator for engine control units. The model must therefore be executed in realtime.

For comparison a model based on the work of Constien (Constien, 1991) is also described. This model is currently used for HiL-applications.

Evaluation plots of the accuracy of the models are included as well as plots of the execution time on HiL-hardware.

INTRODUCTION

The reduction of pollutants in the exhaust gas is of great interest during engine development. Diesel particulate filters (DPF) reduce significantly the amount of soot which accrues during the combustion.

After a certain time the DPF needs to be regenerated by burning the collected soot. Without further technique the soot particles burn up at over 600°C (Bosch, 2004). At engine operating points with a high load these temperatures can be reached under normal operation. Otherwise – e.g. in city traffic – the temperature of the exhaust gas system must be increased from time to time by late injections during the expansion phase of the combustion process. A detailed description of DPFs and regeneration can be found in (Peck, 2006).

To simulate the exhaust gas system a combustion model is needed which is able to describe the combustion of these late injections accurately, for Hardware-in-the-Loop (HiL) test benches even in real-time.

For this work a model was developed which fulfills these requirements. The in-cylinder pressure is calculated every angle of the crankshaft. As pressure and temperature at the time of post injections are much different from that during main injections, the heat release models which are usually used for real-time simulations can not handle regeneration operating points. Therefore for this work a new model was developed. It is able to calculate the two phases of the diesel combustion, the premixed and the diffusion combustion process. Enhancements especially for late post injections are presented in this paper.

To ensure the accuracy of the model the in-cylinder conditions are compared to measurement data gathered at an engine test bench as well for regeneration mode as for normal operation. To ensure the validity of the model only one parameter set is used for both applications.

As the in-cylinder temperature when the exhaust valve opens is crucial for subsequent calculations the output of the model at this point in time is also compared to measurement data.

Because the model was developed for a HiL test bench the real-time ability of the model must be guaranteed. An execution time analysis is included at the end of the paper.

EXHAUST GAS TREATMENT

To fulfill the law regulations for particulates a treatment of the combustion products outside the cylinder is needed. The system which is regarded in this paper uses a particulate filter, which is located after the catalytic converter. The filter must be regenerated from time to time. For this purpose the engine control unit can inject diesel fuel during the expansion phase of the cylinder, which increases the temperature of the exhaust gas system. In addition the amount of soot in the exhaust gas is increased by post injections, which leads to a higher

temperature of the gas after the catalytic converter. With a temperature of over 600°C in the filter the particulates are burnt up.

For this work up to two injections during the expansion phase are regarded.

ZERO-DIMENSIONAL THERMODYNAMIC MODEL

Due to the limited available computing time at a HiL test bench, a multidimensional model of the cylinder can not be used.

The combustion chamber is regarded as one zone with a homogenous mixture. Spatial pressure and temperature differences in the cylinder are not considered. These presumptions lead to the well-known zero-dimensional thermodynamic model (Figure 1) (Kiencke, 2005) (Pischinger, 2002). The change of the in-cylinder temperature can be written as

$$\frac{dT}{d\phi} = \frac{1}{m \cdot \frac{du}{dT}} \cdot \left[\frac{dQ_F}{d\phi} \cdot \left(1 - \frac{u}{H_u} \right) - \frac{dQ_W}{d\phi} - \frac{dW}{d\phi} - m \cdot \frac{du}{d\lambda} \cdot \frac{d\lambda}{d\phi} \right] \quad (1)$$

$\frac{dQ_F}{d\phi}$ is the heat release rate of the diesel fuel. The amount of fuel which burns is calculated by the combustion model presented in this paper.

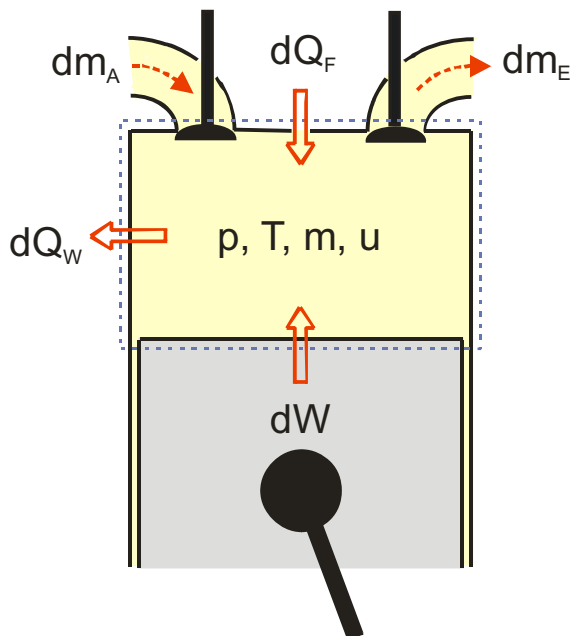


Figure 1 - thermodynamic system

For the calculation of the internal energy of the fuel-air mixture in the cylinder an approach of Justi (Justi, 1938) is used. It is dependent on the temperature and the air-fuel ratio.

$$u(T, \lambda) = 144.5 \left[- \left(0.0975 + \frac{0.0485}{\lambda^{0.75}} \right) (T - T_0)^3 10^{-6} + \left(7.768 + \frac{3.36}{\lambda^{0.8}} \right) (T - T_0)^2 10^{-4} + \left(489.6 + \frac{46.4}{\lambda^{0.93}} \right) (T - T_0) 10^{-2} + 1356.8 \right] \quad (2)$$

As heat transfer coefficient for the cylinder walls formula (3), which was introduced by (Hohenberg, 1983), is used. It can be seen as a mean coefficient for the walls, the piston and the cylinder head.

$$\bar{\alpha} = 130 \cdot V^{-0.06} \cdot p^{0.8} \cdot T^{-0.4} \cdot (\bar{c} + 1.4)^{0.8} \quad (3)$$

With the additional temperature of the cylinder walls the heat dissipation rate can be calculated.

The derivation of the zero-dimensional model can be found e.g. in (Constien, 1991) and (Torkzadeh, 2003).

When diesel is injected into the cylinder it does not burn immediately. The fluid fuel decays to drops and evaporates. These processes are very complex and depend on many influences, which can not be regarded in detail in a realtime model. Therefore an abstraction is used which phenomenologically summarizes these processes and describes the combustion of the diesel fuel injected into the cylinder. In the following two models are described and compared to each other.

COMBUSTION MODEL INTRODUCED BY CONSTIEN

The combustion model which was used previously is based on a work of Constien. In the next paragraphs a short description of the calculation and its physical basis is given. A detailed description can also be found in (Constien, 1991) or (Torkzadeh, 2003).

First, due to the discrete execution time of the model, the fuel is divided into packets. The amount of fuel which is injected during the current simulation step is calculated by integration of the injection rate

$$\Delta m_{inj,i} = \int_{t_{i-1}}^{t_i} \dot{m}_{inj}(t) dt \quad (4)$$

As injection rate a trapezoid-function is used.

It is assumed, that the fluid fuel decays to droplets depending on fuel properties and in-cylinder states. A mean diameter of the droplets can be calculated.

$$SMD = 8.7 \cdot 10^{-6} \cdot D_{inj} \cdot (Re \cdot We)^{-0.28} \quad (5)$$

It is dependent on the diameter of the injector orifice (D_{inj}) and the Reynolds'- and Weber number. The Reynolds' number (Re) is the fraction between the inertial forces and the viscous forces. The Weber number describes the influence of the surface tension (Pischinger, 2002).

For every fluid fuel packet in the cylinder the fraction of evaporating fuel is derived by the formula

$$\Delta m_{evap,i} = C_{evap} \cdot \theta \cdot p_{cyl}^{-0.9} \cdot \frac{1}{SMD} \cdot \frac{\Delta \varphi}{n} \quad (6)$$

Beside the entire surface area of the fuel (θ), the mean diameter (SMD) and the in-cylinder pressure (p_{cyl}) an empirical parameter (C_{evap}) must be set by the help of measurement data gathered from an engine test bench.

As the evaporated diesel fuel does not burn immediately an ignition delay is taken into account. Pressure, temperature and the exhaust gas recirculation have influence on the delay.

$$\tau_{id} = a \cdot \frac{(p_{inj} - p_{cyl})^{0.15}}{p_{cyl}^b} \cdot \exp\left(\frac{c}{T_{cyl} - d}\right) + e \cdot EGR^f \quad (7)$$

In contrast to the two-phase combustion model described later, no integration is used. For every fluid fuel packet the currently calculated ignition delay is compared to the time since the packet is injected into the cylinder. If it is longer than the ignition delay, the packet is assumed to ignite.

The heat release rate is calculated by multiplication of the ignited fuel mass with the net calorific value of diesel fuel.

$$\dot{Q}_F = H_u \cdot \sum_i \Delta m_{ig,i} \quad (8)$$

EVALUATION OF THE MODEL

For HiL test benches the model is a good trade off between accuracy and computation time. Execution times of around 60µs even for a 6-cylinder engine are possible, when the in-cylinder pressure and additional characteristic values like the mean effective pressure are calculated.

A big advantage of the model is that the parameters can be set by automated scripts. This is important when the model is used in the industry.

The model was first developed for engines with only one main injection. With some extensions it was possible to enhance the model for pre-injections. Engines with up to two pre-injections were simulated with a good reproduction of the measurement data.

However, some major influences on the heat release rate are not taken into account:

- In the model, the composition of the in-cylinder gas has no influence on the burning of the fuel. As the total mass of the pre-injections is low this influence can be neglected for the main injection. On the other hand a high amount of fuel of the main injection is burnt when post injections occur. Therefore the composition of the in-cylinder gas should be regarded for post injections.
- The diesel combustion can be divided into three phases: the premixed, the diffusion combustion and the soot oxidation. The premixed combustion is the combustion of nearly homogenous mixed air and fuel while the diffusion combustion accrues at the boundary of fuel and air. A detailed description can be found in (Merker, 2001) and (Oppenheim, 2004). The model introduced by Constien is not able to simulate these phases separately.
- Unburnt fuel mass can not be simulated with the model.

As these constraints limit the accuracy of the combustion model especially for post-injections, we decided to implement a model which is capable of simulating both the premixed and the diffusion combustion.

TWO-PHASE COMBUSTION MODEL

PHENOMENOLOGICAL DESCRIPTION OF THE DIESEL COMBUSTION

In figure 2 the in-cylinder pressure and the heat release rate of a main injection can be seen. The combustion process can be divided into three parts:

1. The first phase of the combustion is characterized by a temporary high gradient of the heat release rate. The diesel fuel, which was conditioned during the ignition delay, burns up. Temperature and pressure in the cylinder rise fast. This phase is called premixed combustion.

2. In the second phase, fuel mixes with the air and the combustion products. The gas mixture is highly inhomogeneous. The flames propagate through lean and rich air fuel mixtures. Soot accrues in the rich areas. This phase is called diffusion combustion and is mixing-controlled.
3. The soot, which formed during the second phase, is burnt in lean areas due to the turbulence in the cylinder. This process is limited by the decreasing temperatures due to the expansion.

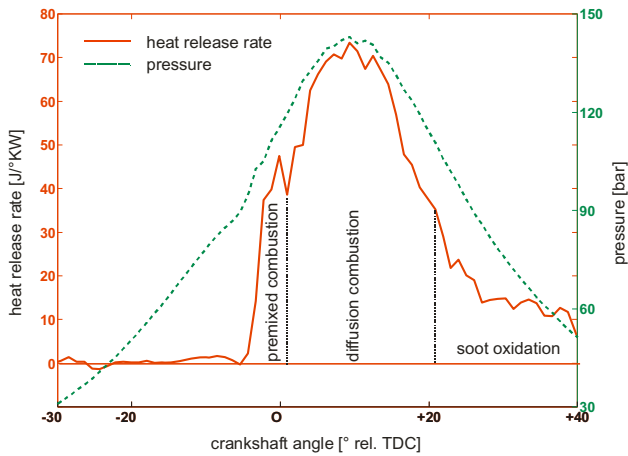


Figure 2 - phases of the diesel combustion

EVAPORATION

Because of the limited execution time of the model a detailed physical calculation of the evaporation is not possible (Boulouchos, 2000). Therefore these processes must further on be described by empirical formulas.

Detailed measurements of the spray formation and evaporation were not available for the injection system, thus approaches found in literature have been used and evaluated. The calculation of the initial mean diameter of the droplets by equation (5) is also used for the two-phase model.

Out of the initial mean diameter every simulation step a new mean diameter is derived by

$$SMD^2 = SMD_0^2 - K_{evap} \cdot t \quad (9)$$

The evaporating fuel can be calculated out of the change of the mean diameter.

If the evaporation constant is in the range of $10^{-6} m^2/s$ the evaporation time of a droplet with the size of $\approx 20 \mu m$ is around 60ms.

The evaporated fuel mass is not immediately available for the subsequent model, which calculates the mixture

of fuel and air. It is assumed that the evaporated fuel mass needs some time before it is ignitable. In contrast to other models this applies not only for the diffusion but also for the premixed combustion.

$$\frac{dm_{F,ig}}{dt} = \frac{1}{\tau_{prep}} \cdot m_{F,evap} \quad (10)$$

PREMIXED COMBUSTION

The premixed model calculates the combustion of fuel and air mixed during the ignition delay. The model is based on an approach of Barba (Barba, 2001) and is divided into two parts. The first part describes the mixing of air and evaporated fuel, while the second part models the combustion.

After evaporation, the fuel mass mixes with the in-cylinder gas, which consists of fresh air and residual gas. The ratio is assumed to be constant

$$\Lambda_0 = \frac{\Delta(m_{air} + m_r)}{L_{st} \cdot \Delta m_{F,ig}} \quad (11)$$

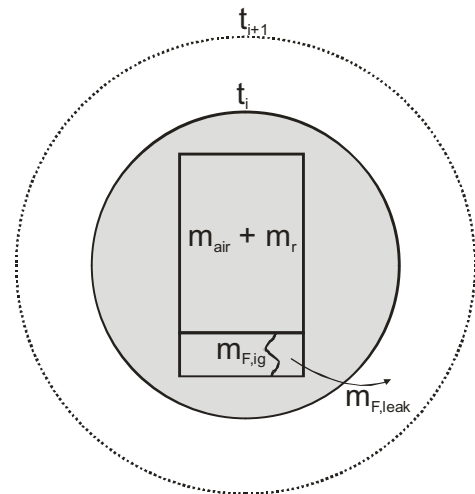


Figure 3 - expansion of the sphere

While Barba used a value of 0.8 for all engines, it appeared that especially for post-injections Λ_0 has to be reduced to 0.5.

The air-fuel mixture is now assumed to form a sphere, which expands by time (see figure 3). It has to be noted, that this sphere is only used for the premixed combustion model and does not affect the zero-dimensional thermodynamic model.

Because of the different fuel concentration between the sphere and the ambient air, fuel leaks from the sphere. A turbulent transport equation is used: