

Quasi-Dimensional Modeling of CI-Combustion with Multiple Pilot- and Post Injections

2010-01-0150

Published
04/12/2010

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ABSTRACT

A new phenomenological CI combustion model was developed. Within this model the given injection rate may contain an arbitrary number of injections during one cycle. Another target was a short computation time of one second per cycle on average. The new approach should also have the ability to simulate a wide engine spectrum from passenger-car engines through to marine engines.

The ignition delay is calculated separately for each single injection. In this way the model depicts the influence of pilot injections on the ignition delay of proximate injections. Each pilot injection is modeled as a single air-fuel mixture cloud with air entrainment. The burn rate of the pilot injection is modeled as a function of flame propagation and of the current local excess air ratio. If the local excess air ratio becomes too lean the pilot combustion stops or does not start at all.

Main and post-injections are calculated by means of a slice approach. The slices are generated isochronously as a function of the injection rate. They propagate through the combustion chamber in the direction of injection. Along the way the entrainment of air into the slices is computed by means of a statistical lambda distribution. The premixed and diffusion combustions are calculated by different approaches. Within the diffusion approach there is a distinction between the fast, nearly stoichiometric combustion and the slow lean

combustion, which is responsible for the characteristic burn-out of CI combustion.

The approach will be shown in detail. On the basis of measurements and simulations, the forecast capability and the limitations of the model are discussed in full.

INTRODUCTION

Between 1990 and today the market penetration of passenger cars with CI engines has risen up to 50% in Europe. The development of common-rail injection systems gave rise to new combustion processes with multiple injections. Multiple pilot injections are state-of-the-art to reduce the ignition delay of the main injection. This causes a smaller premixed combustion, which leads to lower in-cylinder pressure gradients, hence less noise emission. Rate shaping of the main injection is also a common instrument to reduce NO_x formation while simultaneously the engine performance can be increased. To lower the soot emissions sometimes a close-coupled post-injection is applied. Finally a late post-injection is often used to clean the diesel particulate filter (DPF). In this way the injection rate has a big influence on the combustion in CI engines. But the increased variability with regard to the injection rate has also increased the complexity of CI-engine development.

One-dimensional engine simulation tools open up great opportunities for reducing the number of test-bench analyses and hence the development costs. The solutions used to simulate the combustion of CI engines are mostly non-predictive (empirical) models approximating the burn rate for instance with a Wiebe function or a Wiebe hyperbola function. In contrast to empirical models, phenomenological (quasi-dimensional) models are capable of computing the burn rate from a given injection rate and the in-cylinder conditions (pressure, temperature, EGR, ... etc.). In this way they can describe the influence of different operating points or injection strategies on the burn rate. One should consider that the high variability of settings in modern CI engines leads to a large number of different cases, which have to be calculated. In contrast to 3D CFD of engine-combustion, which needs hours or days of CPU-time in order to simulate just one working cycle, the empirical model CPU-times are orders of magnitudes smaller. This makes 3D CFD costly for large parametric studies or for transient simulations of driving cycles in combination with 1D flow simulation tools. The phenomenological models which have been developed often lack the ability to simulate multiple injections and/or their computing time is also much too long. Because of this weakness they are not so widely used. [Reference \[1\]](#) contains a summary about the most common empirical and phenomenological models.

The module introduced in this work shall bridge the gap between fast but non-predictive empirical models and detailed 3D CFD models which are too slow. One opportunity to use the developed model is the integration as a so-called 'compound' in the commercial 1D flow simulation software GT-Power. There it replaces the native DIJet model. In this case GT-Power specifies the in-cylinder conditions based on its flow computations.

The second possibility to use the combustion model is as a stand-alone real working-process calculation tool. In this case the user himself predefines the starting conditions by way of ASCII files. The stand-alone tool is a good choice when one wants to study different injection strategies that do not have a direct effect on the gas-exchange cycle.

MODEL DESCRIPTION

REQUIREMENTS OF THE MODEL

In order to use the model one needs an injection rate for every operating point which is to be simulated. Furthermore, the in-cylinder conditions at IVC must be predefined by the user. One can either use a 1D flow simulation to calculate these values or obtain them from test-bench measurements.

Calibration of the combustion model should be performed at several operating points from all over the engine map. Every aspect of the engine working-process strategy should be

represented in the chosen operating points. From the measured in-cylinder pressure one can obtain the burn rate by performing a pressure trace analysis. One should recognize that this analysis is always made under certain thermodynamic conditions. The comparison between simulated burn rate and that calculated from measurements is only valid if the same thermodynamic estimates are made.

GENERAL MODEL DESCRIPTION

The present model has been integrated into the 'cylinder module' developed by Grill [2, 3]. The 'cylinder module' is a modular-designed concept for object-oriented modeling of in-cylinder processes of internal combustion engines. Hence all the usual approaches to calculating the calorics and the wall heat transfer can be combined with the new combustion model.

The further explanations are confined to the description of the combustion model.

Ignition delay

In order to simulate the combustion of a CI engine, it is necessary to compute the ignition delay of each injection. With regard to fast computation, approaches with detailed or reduced reaction mechanisms are not practicable. For this reason, a simplified Arrhenius [equation \(1\)](#) is often used to calculate the ignition delay.

$$r_{Arr} = k_{Arr} \cdot c_f \cdot c_{ox} \cdot e^{\frac{-k_1 \cdot T_{act}}{T}} \quad (1)$$

Because this equation is only valid for constant boundary conditions (T, c_f, c_{ox}), normally the reaction rate is calculated in every time step and then integrated from SOI until the actual time step. When this integral exceeds a fixed limit value, the fuel-air mix starts to burn. [Equation \(1\)](#) represents the influence of temperature and local excess air ratio on the ignition delay. Because of the wide spread of temperature over the whole cycle in this work, it was necessary to modify the Arrhenius equation. Otherwise it is not possible to use one formula to calculate the ignition delay of pilot injections as well as the delay of the main and post-injections. The changes made by the authors are shown in [equation \(2\)](#)

$$r_{Arr} = k_{ID1} \cdot k_{Arr} \cdot c_f \cdot c_{ox} \cdot e^{\frac{-k_1 \cdot T_{act}}{T + Q_b \cdot k_{ID2}}} \quad (2)$$

The additional term in the denominator of the exponent is intended to depict the influence of previous combustions/injections on the ignition delay of subsequent combustions. This was necessary, as the ignition-delay calculation is made with the mass mean temperature. With the usually low injection rates, the precombustions however mostly result

only in a comparatively faint rise in the mass mean temperature. Instead, there arise during precombustions local, clearly hotter areas which give rise in the subsequent combustions to a correspondingly shorter ignition delay. These local, hot areas are taken into account in the Arrhenius equation by way of the already released combustion energy Q_B of previous injections.

k_{ID1} is a prefactor with which the reaction rates of the main and post-injections are reduced compared with the pilot injections. According to [4] vaporization with increasing injection rates has a greater influence on the local temperature. However, as the ignition-delay calculation is made by way of the mean cylinder temperature, it must be adapted during the main injection and all subsequent fuel injections with the aid of k_{ID1} .

Analogously to [5] the influence of the injection turbulence on the ignition delay is also taken into account by way of equation (3).

$$r_{Mag} = k_{ID1} \cdot k_{Mag} \cdot c_{ox} \cdot \frac{\sqrt{k}}{\sqrt[3]{V_{Cyl}}} \quad (3)$$

Calculation of the specific turbulence is made by way of equation (4). The parameter k_{turb} describes the amount of kinetic energy which is converted into turbulent energy. Determination of the required velocity v in Eq. (4) differs between pilot and main or post-injections and will be explained in the later sections.

$$k = k_{turb} \cdot \frac{E_{kin}}{m} = k_{turb} \cdot \frac{1}{2} \cdot v^2 \quad (4)$$

The characteristic reaction time of a process is proportional to the reciprocal value of its reaction rate:

$$\tau \sim \frac{1}{r} \quad (5)$$

It is accepted that the characteristic time of the ignition delay is obtained from the sum total of the characteristic times for the Arrhenius and Magnussen approaches:

$$\tau_{ID} = \tau_{Arr} + \tau_{Mag} \quad (6)$$

Putting Eq. (5) into Eq. (6) produces the following equation for calculating the reaction rate of the ignition delay:

$$r_{ID} = \frac{r_{Arr} \cdot r_{Mag}}{r_{Arr} + r_{Mag}} \quad (7)$$

For each individual injection there is a separate integration of this reaction rate against the crank angle, see Eq. (8). Combustion begins as soon as the integral value R of the respective injection exceeds an established limit value.

$$R = \int_{t_{SOI}}^{t_{curr}} r_{ID} \cdot dt \quad (8)$$

With the aid of the modifications presented the combustion model is able to forecast the ignition delay of pilot, main and if necessary post-injections of many different operating points of an engine with just one set of parameters (k_{Arr} , k_{Mag} , k_{ID1} , k_{ID2}).

Pilot injections

Pilot injections serve first and foremost to shorten the ignition delay of the main injection. The shorter this is, the smaller will be the premixed amount in the main combustion. In this way the pressure gradients in the combustion chamber get smaller and thus the engine noise emissions and component loads decrease. It is known that small pilot-injection rates at operating points at high rotational speed or long ignition delay do not start to burn and are only converted with the main injection. It is therefore important to be able to forecast this behavior with the combustion model. For this reason each pilot injection is modeled as a separate air-fuel mixture cloud whose excess air ratio leans out with time by way of air admixture. During the injection the mixture cloud is added proportionally to the injected fuel mass combustion-air gas, see Eq. (9). Analogously to [6] this equation describes the fuel-spray behavior before spray breakup with an empirical approach. 0.65 has proven effective as the admixture ratio k_2 .

$$\frac{dm_{Entr.,1}}{dt} = k_2 \cdot \frac{dm_{f,inj}}{dt} \cdot a_{stoich} \quad (9)$$

After the end of injection a constant entrainment rate is assumed against the crank angle, this rate being calculated with formula (10). The admixture ratio k_3 was set at 0.17. This is a simplification of the detailed entrainment calculation within the Barba model, which rest upon an approach for the air diffusion through the mixture cloud surface.

$$\frac{dm_{Entr.,2}}{dt} = k_3 \cdot m_{inj,Pl} \cdot a_{stoich} \quad (10)$$

Fig. 1 shows the exemplary characteristic of the fresh excess air ratio in the mixture clouds of two separate pilot injections.

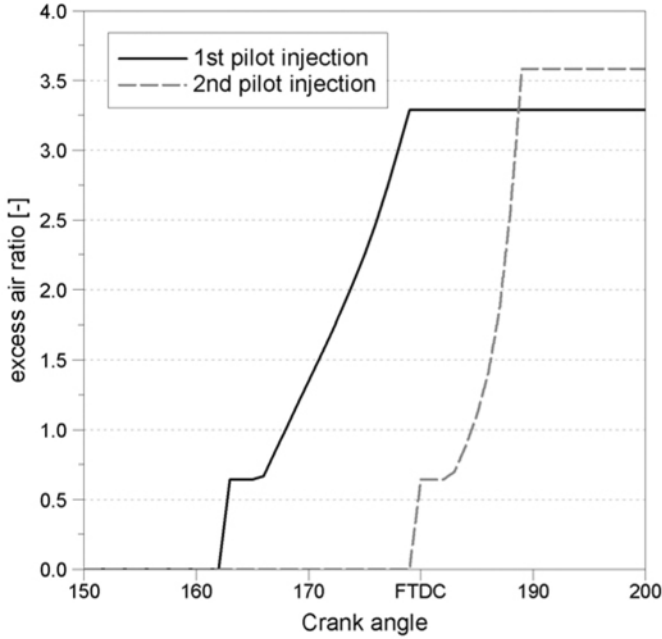


Fig. 1. Excess air ratio inside mixture cloud against crank angle

One can easily recognize that the excess air ratio of the mixture cloud is constant during the injection. After the end of injection the excess air ratio increases exponentially or linearly, depending whether there is a combustion or not.

If a mixture cloud leans out too dramatically, it is no longer possible for ignition to occur inside the cloud or a combustion already occurring is extinguished. An excess air ratio of 3 was fixed as the upper limit value, based on similar observations as in [4] and [6].

The precombustion was modeled consistently as a premixed combustion. At the start of combustion the fresh air and the fuel inside the mixture cloud are thus fully mixed and prepared. In order to translate the effects of the leaning out of pilot injections on combustion as exactly as possible, the approach of Barba [4, 6] was used to calculate premixed combustions. In this approach the laminar flame velocity is also included in the calculation of the burn rate. The laminar flame velocity can be calculated as a function of the excess air ratio by way of equation (11). If the excess air ratio in the mixture cloud is bigger than 3 or smaller than 0.536 the laminar flame speed gets negative which correspond to flame quenching. With the aid of Eq. (14) and using Eq. (12) and

Eq. (13) the calculated velocity is adapted as a function of temperature, pressure and residual-gas content in the combustion chamber.

$$s_{lam,0} = 0,276 - 0,47 \cdot \left(\frac{1}{\lambda_{mc}} - \frac{1}{0,91} \right)^2 \quad (11)$$

$$\gamma = 2,18 - 0,8 \cdot \left(\frac{1}{\lambda_{mc}} - 1 \right) \quad (12)$$

$$\delta = -0,16 + 0,22 \cdot \left(\frac{1}{\lambda_{mc}} - 1 \right) \quad (13)$$

$$s_{lam} = s_{lam,0} \cdot \left(\frac{T}{T_0} \right)^\gamma \cdot \left(\frac{p}{p_0} \right)^\delta \cdot (1 - 2,1 \cdot x_{EGR}) \quad (14)$$

Calculation of the turbulent flame velocity follows:

$$s_{turb} = s_{lam} \cdot \left(1 - k_{fs} \cdot \left(\frac{c_m}{s_{lam}} \right)^{0,8} \right) \quad (15)$$

The parameter k_{fs} serves to calibrate the rise of the burn rate at the start of precombustion. In the subsequent characteristic initially a spherical propagation of the combustion from a single ignition point is assumed. The change in the sphere radius corresponds here to the turbulent flame velocity. By disregarding the time-based change in density in the combustion chamber the current flame-sphere surface (16) is determined, with the aid of which it is possible to calculate the current mass flow which flows into the flame sphere because of the flame propagation, see Eq. (17).

$$A_{flm} = 4 \cdot \pi \cdot r_{flm}^2 \quad (16)$$

$$\frac{dm_{fsphere}}{dt} = \rho_{mc} \cdot A_{flm} \cdot s_{turb} \quad (17)$$

This mass flow has the current mixture-cloud composition. To calculate the combustion, however, only the mass flow of unburnt fuel which can be calculated from the mixture-cloud composition is relevant:

$$\frac{dm_{b,1}}{dt} = \frac{dm_{fsphere}}{dt} \cdot \frac{m_f}{m_f + m_a + m_{f,b} + m_{a,b}} \quad (18)$$

However, it must be assumed from this that ignition occurs not just at one local point of the combustion chamber, especially when combustion is already advanced. Following [4, 6] a second approach to calculating the burn rate was therefore implemented:

$$\frac{dm_{b,2}}{dt} = k_{comb} \cdot \frac{1}{\Lambda_{mc}^2} \cdot \frac{s_{turb}}{r_{mc}} \cdot m_f \quad (19)$$

The fissure quantity of the mixture cloud Λ_{mc} is defined as follows:

$$\Lambda_{mc} = \frac{m_a + m_{a,b} + m_{f,b}}{a_{stoich} \cdot m_{inj,PI}} \quad (20)$$

It puts the current fresh-air and residual-gas mass of the mixture cloud in proportion to the air mass which would be necessary for a stoichiometric combustion. The fissure quantity thus continues to increase proportionally to the entrainment, and also as combustion advances.

The conversion rate of a precombustion is obtained from the smaller value of the two equations (18) and (19):

$$\frac{dm_{b,PI}}{dt} = \min\left(\frac{dm_{b,1}}{dt}, \frac{dm_{b,2}}{dt}\right) \quad (21)$$

The conversion rate is limited at the start of the precombustion by equation (18) and at the end of the combustion by equation (19). If a precombustion cannot burn completely because of leaning out, the unburnt fuel contained in it is added to the first fuel slice of the main injection.

To conclude, there will be an explanation of calculating the mixture-cloud velocity which is needed to determine the specific turbulence with the aid of (4). It is assumed that the mixture cloud has a uniform velocity which decreases over the course of time. Starting out from the mean injection velocity which is determined by way of formula (22), calculation of the velocity in the further characteristic is made by way of (23). Neglecting the influence of different injection profiles on the injection velocity is permissible with regard to the generally short duration of pilot injections.

$$u_{mc,inj} = \frac{k_{noz} \cdot m_{inj,PI}}{n_{noz} \cdot t_{inj,PI} \cdot \rho_{f,liq} \cdot \left(\frac{\pi \cdot d_{noz}^2}{4}\right)} \quad (22)$$

$$u_{mc} = \min\left(u_{mc,inj}, k_v \cdot u_{mc,inj} \cdot \frac{d_{noz}}{x_{mc}}\right) \quad (23)$$

Main injection

The slices are created in constant time steps. During initialization a fixed amount of combustion-air gas is admixed with each slice. Before the ignition delay elapses the fuel inside the slices is diffused to two fuel pools. The first is the so-called “premixed pool”, while the second is known in the further characteristic as the “diffusion pool”. Slices which are created after the ignition delay has elapsed contain only one diffusion pool.

It is assumed that the mixing of fuel and air inside the premixed pool of each slice is extensively concluded at the ignition point. The upshot of this is that the fuel of the premixed pools is converted relatively quickly after the start of combustion. The combustion is therefore, along similar lines to [5], modeled by means of an Arrhenius approach (24). The definition of the fuel concentrations can be taken from formula (25).

$$r_{Pre} = k_{Pre} \cdot c_f \cdot c_{ox} \cdot e^{\frac{-k_f \cdot T_{act}}{T}} \quad (24)$$

$$c_f \cdot c_{ox} = \underbrace{\frac{m_{f,Pre}}{V_{mix}}}_{c_f} \cdot \underbrace{\frac{\lambda \cdot a_{stoich} \cdot m_{f,Pre}}{V_{mix}}}_{c_{ox}} \quad (25)$$

The mass conversion rate of the premixed combustion is finally obtained using the following equation:

$$\frac{dm_{b,Pre}}{dt} = r_{Pre} \cdot V_{mix} \cdot (t - t_{SOC})^2 \cdot (1 - x_{EGR})^{k_x} \quad (26)$$

V_{mix} corresponds here to the volume of the stoichiometric fuel-air mixture with the fuel mass of the premixed pool. The term at the end describes the decrease in the laminar flame velocity at high EGR rates.

The fuel slices created “fly” through the combustion chamber in the injection direction. As they propagate the admixture of combustion-chamber gas to the slices is stipulated by means of an empirical distribution of the excess air ratio “lambda”