

Further Development and Application of a Model for the Calculation of Heat Release in Direct Injection Diesel Engines

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ABSTRACT

A model for the calculation of heat release in direct injection Diesel engines is presented. It needs only one engine-specific experimental parameter. In the form the model is presented here it is limited to the medium and upper load range, where Diesel combustion is mainly mixing controlled. The development of the model is based on data from medium speed engines. The applicability to automotive engines is shown in some examples.

The model is based on the theory of single phase turbulent jets. Starting from the balance of momentum and fuel mass flow the stationary part of the jet can be calculated. The propagation of the front of the unsteady jet is determined from a continuity consideration. Heat release is calculated based on the assumptions of the Simple Chemically Reacting System (SCRS). Fuel that is mixed with air is assumed to be burnt instantaneously.

Adjustment of the model to a good correspondence of measured and calculated heat release results in an experimental parameter which is in the same range as the figures published for turbulent jet flames. With this calibration the model is able to calculate the rate of heat release over a wide load range.

The experimental parameter is engine specific but is a constant except for very short injections. This can be explained by the entrainment wave at the end of injection. Its influence is the larger the shorter the injection is. A correlation is presented to reproduce this effect.

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INTRODUCTION

The development process of internal combustion engines is based on simulations to a very large extent. An important group of these is the zero- or one-dimensional calculation of the working process. In contrast to the three-dimensional approach the geometry of the engine is not completely resolved but simplified to one-dimensional pipes and zero-dimensional control volumes. This results in a loss in predictive capability but also in much shorter calculation times.

An important parameter for such simulations is the rate of heat release (ROHR). There are several approaches to provide it. In many cases it is taken from measured data. Then of course the influence of changes for example to the injection system cannot be reproduced. Therefore many models exist which try to calculate the ROHR. They differ widely concerning complexity and number of required input parameters. In some cases the range of application with one parameter set is limited. So there is still room for improvement. This paper is intended to contribute to this.

STATE OF THE ART

There are several groups of models used for 0D/1D-calculations. The first are the purely empirical models. Well known is the method presented by Woschni and Anisits [1]. Here the parameters of a Vibe function are converted dependent on equivalence ratio, engine speed, ignition delay, pressure and temperature. The injection rate is not needed. Its dependence on operating conditions is implicitly contained in the model. This restricts the applicability of the model to mechanical injection systems. Further it is not possible to predict the influence of changes to the injection system. Under these boundary conditions however the results are good.

The second group of models is based on the similarity of a Diesel fuel jet and a steady single phase jet. Normally the mixing of fuel and air are described with a quasi steady jet model. Starting from this basis models for the evaporation of the fuel, the reaction kinetics of ignition and combustion and for pollutant formation are applied. The degree of complexity is very diverse. Most well known in this group is probably the

model of Hiroyasu and Kadota [2], [3]. Other models of this kind were presented by Meguerdichian and Watson [4] and by de Neef [5].

The models of the third group are based on an integral turbulence model. The turbulent kinetic energy density is used to calculate the ROHR. The approach presented by Chmela, Orthaber and Schuster [6] does not use a model of the fuel jet. In the papers presented by Weisser and Boulouchos [7], Chmela et al [8] and Rether et al [9] the turbulence model is combined with information from a model of the jet.

DESCRIPTION OF THE MODEL

The model described in this paper is a further development of the approach presented by one of the authors in [10]. It belongs to the second of the groups described above. In the version shown here it is restricted to the mixing controlled part of combustion and needs only one engine specific model parameter, so the calibration effort is very small. Furthermore the computation times needed are very short. The development of the model is based on data from medium speed engines. The possible applicability to automotive engines is shown in some examples.

The modifications compared to the version presented in [10] consist of the following:

- A new approach for the model constant based on measurements of turbulent jet flames and on the expansion wave after the end of injection, extending the range of applicability
- The possibility to use arbitrary injection profiles
- The possibility to predict the effect of EGR

Steady Jet

The model is based on the theory of single phase, steady, turbulent, circular jets. This can be used as basis for the modeling of Diesel combustion, because the fuel droplets leaving the nozzle transfer their momentum to the surrounding air within a very short distance and entrain it. Thus the two phase jet of liquid in air becomes a single phase gas jet.

Only the time averaged values of the properties in the jet are considered, not the turbulent fluctuations. A schematic representation is shown in Figure 1. Only the main zone is taken into account, where the profiles of fuel concentration and velocity are similar respectively. The initial and transition zone are neglected, because they are small compared to the length of the total jet. So the conical jet starts from a virtual single point source, as shown in Figure 2.

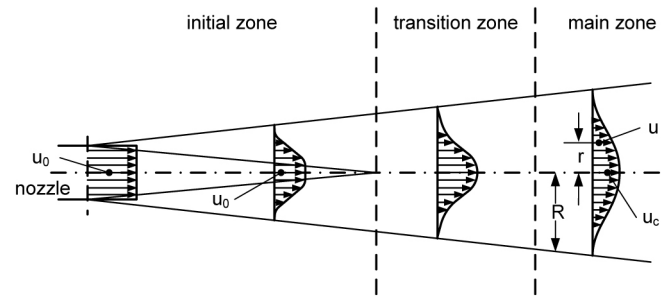


Figure 1. Schematic representation of the jet.

The density in the jet is assumed to be constant and equal to that of the surrounding gas ρ_g . The spray angle is also taken constant. It is assumed that a mixture of liquid and vaporized fuel exits the nozzle [11]. The density of the mixture is set equal to $\mu \cdot \rho_f$, where μ is the discharge coefficient of the nozzle and ρ_f the density of the liquid fuel. The exit velocity u_0 is set equal to the Bernoulli velocity: $u_0 = \sqrt{\frac{2 \Delta p}{\rho_f}}$, where Δp is the pressure difference across the spray hole.

An external flow field, such as swirl, is not taken into account. Medium speed engines typically have no swirl. If engines with swirl are to be calculated the model constant has to be adjusted.

Within the jet the fuel mass flux $\dot{m}_f$ and the momentum flux $\dot{M}$ are conserved: Eq. (1) and (2).

$$\dot{m}_f = \int C \rho_g u \, dA = \text{const.} \quad (1)$$

$$\dot{M} = \int \rho_g u^2 \, dA = \text{const.} \quad (2)$$

The radial profiles for velocity u and fuel concentration (mass fraction) C are those proposed by Abramovich [12]: Eq. (3) and (4). Here $\epsilon = r/R$ is the dimensionless radius and the index c denotes the centerline of the jet.

$$u = u_c (1 - \epsilon^{1.5})^2 = u_c f_u \quad (3)$$

$$C = C_c (1 - \epsilon^{1.5}) = C_c f_c \quad (4)$$

By using Eq. (5) and the momentum balance Eq. (6) can be derived.

$$dA = 2 \pi R^2 \epsilon \, d\epsilon \quad (5)$$

$$2\pi R^2 u_c^2 \rho_g \int_0^1 f_u^2 \epsilon d\epsilon = \text{const.} = \mu \rho_f \frac{\pi}{4} d_N^2 u_0^2 \quad (6)$$

Here u_0 is the exit velocity and d_N the nozzle diameter. This equation can be rearranged into Eq. (8) by using Eq. (7) and solving the integral analytically.

$$R = \frac{dR}{ds} s \quad (7)$$

$$u_c = \frac{1.37}{\frac{dR}{ds} KLA} u_0 d_N \sqrt{\frac{\mu \rho_f}{\rho_g} \frac{1}{s}} \quad (8)$$

Similarly Eq. (9) can be derived from the fuel mass balance. Applying Eq. (7) and (8) leads to Eq. (10).

$$2\pi R^2 C_c u_c \rho_g \int_0^1 f_c f_u \epsilon d\epsilon = \text{const.} = \mu \rho_f \frac{\pi}{4} d_N^2 u_0 \quad (9)$$

$$C_c = \frac{1.02}{\frac{dR}{ds} KLA} d_N \sqrt{\frac{\mu \rho_f}{\rho_g} \frac{1}{s}} \quad (10)$$

Velocity and fuel concentration decrease hyperbolically along the jet axis. In Eq. (8) and (10) the model parameter KLA is introduced. It will be discussed later. For non-reacting jets $KLA = 1$.

Though the two phase jet can be treated as a single phase gas jet, there is however an effect of the atomization process on the spray angle, which is influenced by the nozzle geometry. In this work for jets of liquid fuel the change of radius $\frac{dR}{ds}$ is calculated with an empirical formula based on an equation by de Neef [5], Eq. (11).

$$\frac{dR}{ds} = \frac{0.54}{3 + 0.07 \left(\frac{l_N}{d_N}\right)^{1.6}} \quad (11)$$

Here l_N is the length of the nozzle hole. According to the experience of the authors this formula is applicable for cylindrical spray holes and $\frac{l_N}{d_N} < 5$.

In case a single phase jet is to be calculated, according to Abramovich [12] $\frac{dR}{ds} = 0.22$ is used.

Unsteady Jet

If the velocity on the centerline is assumed to be equal the velocity of the tip of the jet, the calculated spread of the jet is too fast compared with measurements. In fact an unsteady (starting) jet consists of a steady jet and a head vortex or jet front. Eventually, every unsteady jet becomes a steady one, if injection is continued sufficiently long. Therefore a quasi steady treatment is possible if a model for the movement of the jet front is applied.

Here a plane jet front is assumed. This is a simplified approach, which was proposed by Meguerdichian and Watson [4]. The velocity of the jet front can be calculated from a continuity consideration. In one timestep the jet must grow as much as fuel is delivered through the nozzle: Eq. (12) and (13). Integration results in Eq. (14).

$$\dot{m}_f dt = \int_A C \rho dA ds \quad (12)$$

$$2\pi R^2 C_c \rho_g \int_0^1 f_c \epsilon d\epsilon ds = \mu \rho_f \frac{\pi}{4} d_N^2 u_0 dt \quad (13)$$

$$s = \sqrt{\frac{1.14}{\frac{dR}{ds} KLA} u_0 d_N \sqrt{\frac{\mu \rho_f}{\rho_g} t}} \quad (14)$$

For the time after the end of injection the movement of the end of the jet needs to be known. It is assumed that it moves with the same velocity as the jet front did at the respective point. This is also in accordance with the continuity consideration for the jet front. After end of injection the jet is represented by a truncated cone which contains the total injected fuel mass.

An important property of the unsteady jet is the mixing duration t_M which is the time that the jet needs to reach an air ratio of $\lambda = 1$ or $C = C_{st}$ on the centerline. The air ratio is the reciprocal of the equivalence ratio, C_{st} is the stoichiometric fuel concentration. From Eqs. (10) and (14) Eq. (15) can be derived. This parameter is a characteristic number of the unsteady jet.

$$t_M = \frac{0.913}{\frac{dR}{ds} KLA} d_N \sqrt{\frac{\mu \rho_f}{\rho_g} \frac{1}{u_0 C_{st}^2}} \quad (15)$$

Heat Release

Combustion is modeled based on the assumptions of the Simple Chemically Reacting System (SCRS) [13]. This means that fuel, which is mixed with air, burns instantaneously. With this approach combustion in a stationary diffusion flame is calculated based on the mixing of fuel and air [13]. Under the

boundary conditions under discussion here the fuel concentration C calculated with the equations above is equivalent to the mixture fraction.

Heat release in a direct injection Diesel engine is also dominated by the mixing of fuel and air during the main part of combustion. Evaporation and chemistry are much faster during this phase. The range, where mixing is rate controlling is the larger, the larger the engine is and the higher the load. In this paper only this range is discussed.

The equations presented above and the assumption of quasi steadiness results in a fixed concentration field produced by the steady jet. Along the jet axis and towards the boundaries of the jet the fuel concentration decreases. The unsteady jet, displayed by the hatched area in Figure 2, runs through this concentration field. In Figure 2 an instant after the end of injection is shown, the end of the jet has already departed from the nozzle. Because of the movement of the jet the hatched zone leaves the area of high fuel concentration, i.e. the fuel is more and more mixed with the air.

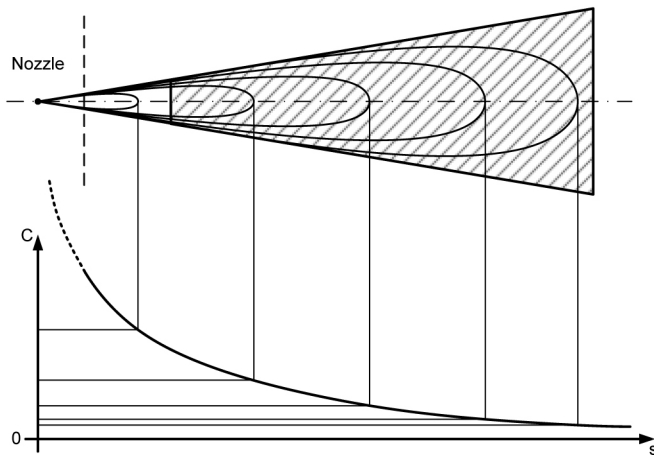


Figure 2. Mixture formation.

Differing from the assumptions of the SCRS a lower limit of the air ratio is taken into account, below which no heat release takes place. It is taken as $\lambda_l = 0.32$ [5]. An upper limit λ_u can also be taken into account. Presently this is de facto not done by choosing a very large value for λ_u .

Heat release is calculated by integrating the fuel mass which is mixed with air within the air ratio limits defined above. This fuel mass is called the prepared fuel mass m_{fprep} : Eq. (16).

$$\frac{dm_{fprep}}{ds} = \int_{A_{prep}} C \rho_g dA \quad (16)$$

Here A_{prep} is the area where the air ratio is within the upper and lower limits discussed above. From the distribution of the fuel concentration and its relation with the air ratio the dimensionless radii ϵ_l , ϵ_u and ϵ_{st} can be determined. The first and second correspond to the air ratio limits and the third to

stoichiometric conditions. In case the fuel is mixed locally with an air ratio less than 1, the local conversion efficiency $\eta_{con,loc} = \lambda_{loc}$ is applied [5]. This results in Eqs. (17) and (18).

$$C_c \leq C_{st}: \frac{dm_{fprep}}{ds} = C_c \rho_g K L A^2 2\pi R^2 \int_{\epsilon_{st}}^{\epsilon_u} f_C \epsilon d\epsilon \quad (17)$$

$$C_c > C_{st}: \frac{dm_{fprep}}{ds} = C_c \rho_g K L A^2 2\pi R^2 \left(\int_{\epsilon_{st}}^{\epsilon_u} f_C \epsilon d\epsilon + \int_{\epsilon_l}^{\epsilon_{st}} \eta_{con,loc} f_C \epsilon d\epsilon \right) \quad (18)$$

The integrals can be solved analytically which results in the very short calculation times of the model.

The time derivative required for the process simulation results from multiplication with the jet front velocity which can be achieved by differentiation of Eq. (14).

$$u_F = \frac{0.572}{\frac{dR}{ds} K L A} u_0 d_N \sqrt{\frac{\mu \rho_f}{\rho_g} \frac{1}{s}} = \frac{ds}{dt} \quad (19)$$

$$\frac{dm_{fprep}}{dt} = \frac{dm_{fprep}}{ds} u_F \quad (20)$$

Up to now the calculation yields the fuel mass prepared in a jet with continuous injection. To treat a jet in an engine, where the injection has a finite duration, the assumption made above is used, that the end of the jet at time t moves like the front at time $t - t_{ID}$. Here t_{ID} is the injection duration. The mass in the first jet is reduced by the mass in a second jet, which is delayed by the injection duration [5]. The resulting mass is called fuel mass between (air ratio) limits m_{fbl} : Eqs. (21) and (22).

$$t \leq t_{ID}: \frac{dm_{fbl}(t)}{dt} = \frac{dm_{fprep}(t)}{dt} \quad (21)$$

$$t > t_{ID}: \frac{dm_{fbl}(t)}{dt} = \frac{dm_{fprep}(t)}{dt} - \frac{dm_{fprep}(t - t_{ID})}{dt} \quad (22)$$

Because in the version of the model described here instantaneous combustion of the fuel mixed with air is assumed, the ROHR is calculated simply by Eq. (23), where LHV is the lower heating value of the fuel.

$$\frac{dQ_B}{dt} = LHV \frac{dm_{fbl}}{dt} \quad (23)$$