

Investigation of the Parameters Influencing the Spray-Wall Interaction in a GDI Engine - Prerequisite for the Prediction of Particulate Emissions by Numerical Simulation

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ABSTRACT

Due to the EU6 emission standard that will be mandatory starting in September 2014 the particulate emissions of GDI engines come into the focus of development.

For this reason, soot and the mechanisms responsible for the soot formation are of particular importance. A very significant source of particulate emissions from engines with gasoline direct injection is the wall film formation. Therefore, the analysis of soot emission sources in the CFD calculation requires a detailed description of the entire underlying model chain, with special emphasis on the spray-wall interaction and the wall film dynamics.

The validation of the mentioned spray-wall interaction and wall film models is performed using basic experimental investigations, like the infrared-thermography and fluorescence based measurements conducted at the University of Magdeburg. This comparison demonstrates that the used modeling is not only able to correctly reproduce all trends, but also to do quantitatively correct predictions in most instances.

Additionally, on the basis of the measurements carried out on an optically accessible single-cylinder engine, the verification of the presented model chain in an in-cylinder flow simulation is shown. Using a defined multi-component fuel, a good correlation between the calculated mixture formation and the measured particulate emissions is demonstrated.

In essence, the scope of this paper is to present a method, which allows a detailed modeling of the spray-wall interaction as well as the wall film dynamics by numerical simulation, throughout the whole parameter space relevant to an engine process.

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INTRODUCTION

Since the non-premixed diesel combustion can produce high soot emissions, the reduction of particulate emissions has long been in the focus of the development process for compression ignition engines. Due to the premixed, homogeneous combustion in gasoline engines, the soot emissions are on a comparably negligible level. Therefore, the minimization of particulate emissions was only a subordinate development goal. The EU6 emission standard, that will be mandatory starting in September 2014, limits particulate mass (PM) and for the first time also particulate number emissions (PN) for engines with gasoline direct injection. For a three year period, between September 2014

and September 2017, there is the possibility to use a preliminary EU6b limit of 6e12 particulates/km for the NEDC testing cycle. The final EU6c limit, valid starting in September 2017, was set technologically neutral on the limits already mandatory for diesel engines, with a value of 6e11 particulates/km in the NEDC testing cycle. A more detailed overview of the limits and the introduction of the EU6 emission standard can be found in [10]. Compared to the previous limit of particulate mass (EU5 - legislation), the final EU6c limit of the particulate number presents quite a challenge for current GDI engines. Thus, in order to reach the limit of 6e11 particles/km without exhaust gas aftertreatment (see e.g. [7]), the reasons for particulate formation in the

combustion chamber have to be analyzed and understood in detail.

Results of fundamental gasoline engine research show, that providing sufficiently high temperatures, ranging from 1500 K to 2200 K, the starting point for the formation of soot particles in gasoline engines with direct injection is a locally fuel-rich mixture [9]. In stoichiometrically operated gasoline engines the mixture formation time available between the end of injection and the start of ignition should prevent the existence of such locally very fuel-rich zones in the gas phase. According to some earlier investigations, e.g. [17], it is known that the wetting of the combustion chamber walls with liquid fuel leads to an intense soot formation. The fuel deposited on the wall evaporates delayed and cannot be homogenized sufficiently until the start of ignition. These locally fuel-rich zones close to the wall provide ideal conditions for the formation of particulates in the engine. In order to analyse the sources of soot formation by means of numerical simulation a detailed description of the underlying model chain depicted in [figure 1](#) is crucial. Thus, the modeling of the spray-wall interaction and the wall film dynamics come up to play a major role. With the introduced method, published in an earlier publication [8], the processes during the injection can be described well by numerical simulation. Focus of the experimental and numerical investigations shown in this paper was, on the one hand, the deeper understanding of the fundamentals occurring in the spray-wall interaction process. On the other hand, the modeling of the wall film formation and evaporation has to be calibrated and validated, in order to reproduce the spray-wall interaction processes taking place in gasoline engines as detailed as possible.

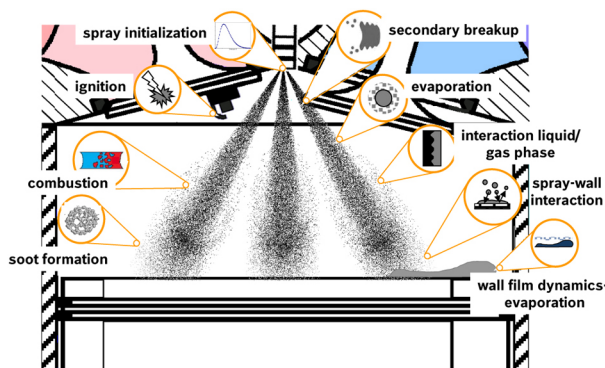


Figure 1. Model chain used to analyze the sources of particulate emissions by numerical simulation

The first part of the article discusses the main parameters influencing the spray-wall interaction. The second part presents the validation and calibration of the employed modeling on the basis of an extensive parameter study. The experimental data set was provided by the University of Magdeburg. Furthermore the influence of the fuel composition on the modeling of the spray-wall interaction in GDI engines will be shown. Based on a comparison with the

experimental data from an optically accessible single cylinder research engine it will be shown, that the developed approach leads to an improved forecast of particulate emissions by means of numerical simulation.

PARAMETERS INFLUENCING SPRAY-WALL INTERACTION

As mentioned in the previous section, the wetting of the combustion chamber walls with liquid fuel is the dominating source of particulate emissions in GDI engines. So, in order to perform an analysis of the soot sources by numerical simulation, the spray-wall interaction processes as well as the wall film dynamics have to be modeled as detailed as possible. The main parameters influencing the spray-wall interaction processes were identified in the framework of CFD calculations of a single cylinder engine.

In this context, the fuel composition could be determined as an important factor. In CFD engine simulations the single-component surrogate fuel n-heptane is often used to model the mixture formation. In terms of its thermo-physical properties, this surrogate fuel depicts the average material properties of gasoline very well. However, the very large range of boiling temperatures of gasoline, with an initial boiling point at $T_{\text{Sat,i}} = 307 \text{ K}$ and a boiling end at $T_{\text{Sat,e}} = 470 \text{ K}$, cannot be represented by a single-component surrogate fuel. But especially the low-volatile components in gasoline fuel, despite of their relatively small mass fraction, are of particular importance for the spray-wall interaction as well as the wall film evaporation. These components evaporate delayed and cause the locally fuel-rich zones, which are responsible for the particulate emissions. This fact is also confirmed by the results of the CFD engine calculations shown in [figure 2](#).

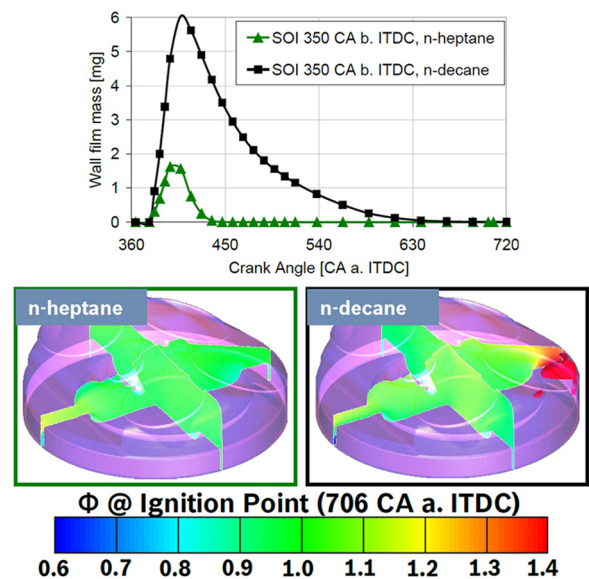
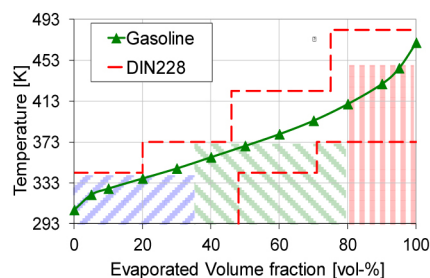


Figure 2. Wall film mass and mixture distribution in dependence of the used surrogate fuel

A part load operating point with a very early start of injection (SOI) at SOI = 350 CA b. ITDC was examined here. The experimental investigations of this operating point showed a strong piston wall wetting with a subsequent non-premixed combustion in the piston area and accordingly high particulate emissions. By using the single-component surrogate fuel n-heptane ($T_{\text{Sat}} = 371 \text{ K}$), these locally fuel-rich zones in the area of the piston could not be reproduced by the numerical simulation. Due to the low boiling temperature, the wall film evaporates considerably before the ignition point. So there is sufficient time available for the homogenization of the fuel evaporating from the wall film. Thus, if the same operating point is calculated with the low-volatile substitute fuel n-decane ($T_{\text{Sat}} = 447 \text{ K}$), there is a strong delay in wall film evaporation. In this case, the wall film remains up to the ignition point, resulting in the correspondingly fuel-rich zones near the piston. On the other hand it turned out that, if an operating point with a start of injection at SOI = 300 CA b. ITDC, which represents an optimum in terms of soot emissions, is calculated by the numerical simulation, the very good homogenization can only be reproduced by using n-heptane as a surrogate fuel. In contrary to the measurement, the calculation with n-decane shows unlikely strong inhomogeneities, mainly due to the higher boiling temperature. So the initially made statement, the average fuel properties and thus the global mixture formation can be mapped well with the single-component surrogate fuel n-heptane, can be confirmed. The spray-wall interaction however is strongly influenced by the low-volatile components in the gasoline fuel. These cannot be represented with n-heptane as a substitute fuel. For this reason, the surrogate fuel shown in [figure 3](#) was defined.



		n-hexane	iso-Octane	n-decane	Gasoline
Density	[kg/m³]	661	699	721	740
Saturation Temp.	[K]	342	372	447	307 - 470
Kinemat. Viscosity	[m²/s]	4.64e-07	2.95e-06	1.29e-06	-
Mass fraction	[-]	0.35	0.45	0.2	

Figure 3. Saturation curve and material properties of the different fuels [5]

To keep the computational effort as small as possible, the thermo-physical properties of gasoline should be considered with a low number of fuel components. In particular the range of boiling temperatures of gasoline could be mapped well with the three-component surrogate fuel shown here.

The influence of this fuel composition on the spray-wall interaction process, in comparison to the previous single-component modeling, will be shown in the next part of the paper.

The respective surface temperature represents another important parameter influencing the spray-wall interaction. In CFD engine calculations often a surface temperature, constant over the whole engine cycle, is assumed for each component. But with this assumption both, temporal and spatial fluctuations of the surface temperature are neglected. In the context of the basic examinations it will be shown, that considering the temperature drop due to the impinging fuel represents an important parameter influencing the spray-wall interaction and the wall film evaporation. As pointed out in [figure 4](#), a reduction of the mean wall temperature of about 8 K arises due to the spray-cooling. This calculated temperature drop corresponds to the reduction of the mean wall temperature observed in the experimental investigations. As explained in the next section, locally even much higher temperature drops occur.

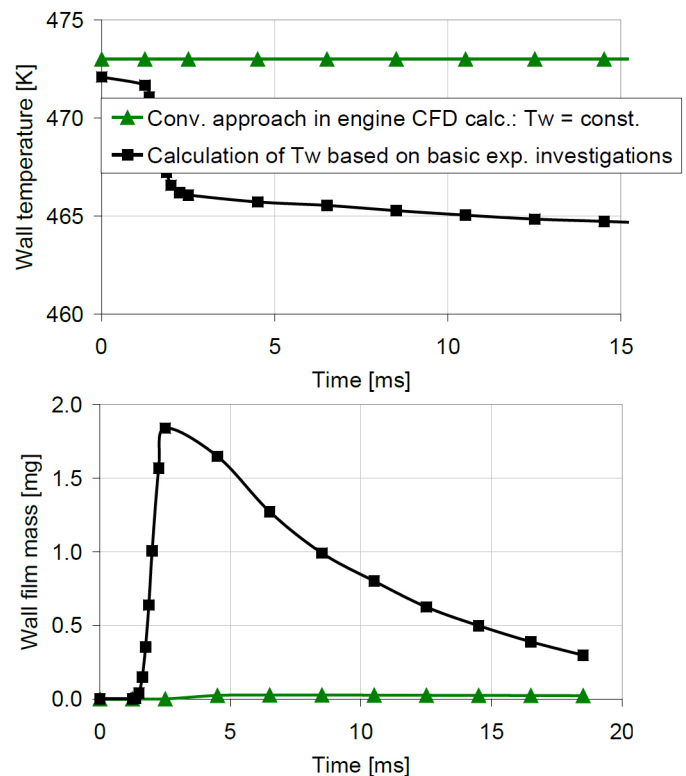


Figure 4. Influence of the spray cooling on the calculated wall temperature and the calculated wall film mass

The high wall temperature of $T_{\text{Wall}} = 473 \text{ K}$ is slightly above the Leidenfrost temperature of the used fuel. Thus, liquid fuel can be deposited on the wall only after a corresponding temperature drop due to the spray-cooling effect. Applying the conventional approach, the assumption of a constant wall temperature, this effect can not be

reproduced. As shown in [figure 4](#), this results in a wall film mass calculated considerably too small.

EXPERIMENTAL AND NUMERICAL INVESTIGATIONS OF THE SPRAY-WALL INTERACTION PROCESS

In order to deepen the understanding of the phenomena occurring during spray-wall interaction and in order to calibrate and validate the modeling of the wall film formation and evaporation, basic experimental and numerical investigations of the spray-wall interaction process were conducted. In addition, to validate the used model chain, experimental and numerical investigations were carried out at an optically accessible single-cylinder research engine. These will be presented in the following section.

Experimental Setup

A precise measurement of the wall film properties within an operating gasoline engine is not really feasible. For this reason a generic setup was defined, consisting of a plate and a high-pressure fuel injector in a well defined position to the plate. Infrared thermography, as well as LIF measurements were carried out at the University of Magdeburg (cf. [\[11\]](#), [\[14\]](#), [\[15\]](#)).

For the single cylinder engine the soot luminosity was used as a qualitative indicator.

Generic setup: spray impinging on a plate

First the wall temperature was measured by using the infrared thermography. Here, the main goal was the determination of the temperature distribution starting during spray-wall interaction up to the complete evaporation of the wall film. Second, the local wall film heights were measured by using laser-induced fluorescence.

Infrared thermographic investigations

As mentioned in the previous section, the temperature of the piston surface is a parameter influencing the whole process, starting with the spray-wall interaction, the wall film formation and finally the wall film evaporation. For this reason, the knowledge of the varying wall temperature is of great importance for the numerical prediction. The experimental setup, used to determine the spatial- and time-resolved temperature distribution on the piston surface is shown in [figure 5](#). Here, the piston is replaced by an electrically heated plate with a smooth surface. The initial temperature of the plate before injection can be accurately controlled via the electrical current. The temperature distribution under the plate can then be measured using an infrared camera. The bottom of the plate is additionally coated to achieve a high emissivity and to minimize the error of the thermography system.

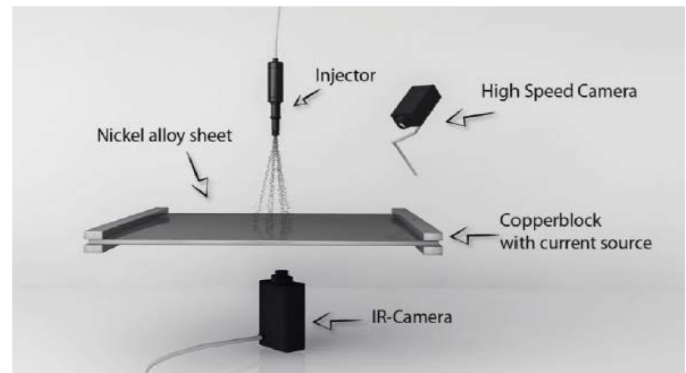


Figure 5. Experimental setup of the infrared thermographic investigations [14]

A more detailed description of the experimental setup and the calibration process can be found in [\[11\]](#). As a typical example of the obtained results, the temperature distribution 20 ms after the start of injection of a six-hole injector, starting with an initial wall temperature of 353 K, is shown in [figure 6](#). The strong decrease in temperature at the spray-impingement points and the resulting wall films are clearly visible. As explained in the previous section and shown in [figure 4](#), this reduction of the surface temperature is the reason for the increased time needed to evaporate the wall film.

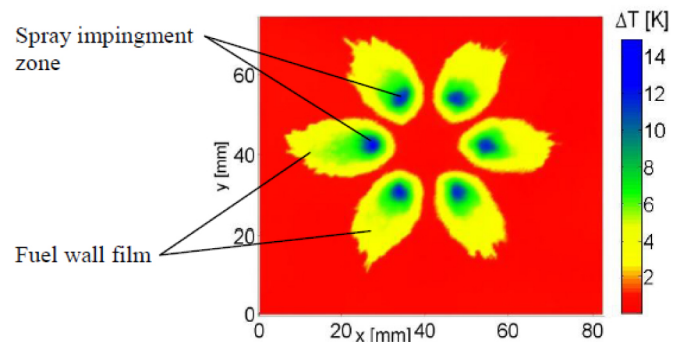


Figure 6. Temperature variations caused by the wall film [14]

Laser-induced fluorescence investigations

The film mass and its spatial distribution are among others the most relevant properties of wall films. The exact values can be determined using the method of laser-induced fluorescence (LIF). The experimental setup used in the present case is shown in [figure 7](#). For the experiments, iso-octane with 5 vol.% of the tracer 3-Pentanone was used as a surrogate fuel. A beam of the spray is injected onto a quartz glass, representing the piston surface. Both, the Nd/YAG laser as well as the camera are located beneath the quartz glass. After the injection, the tracer in the wall film is excited to fluorescence by light pulses with a wavelength of 266 nm.

Conclusions on the film height are drawn based on the intensity of the fluorescence signal.

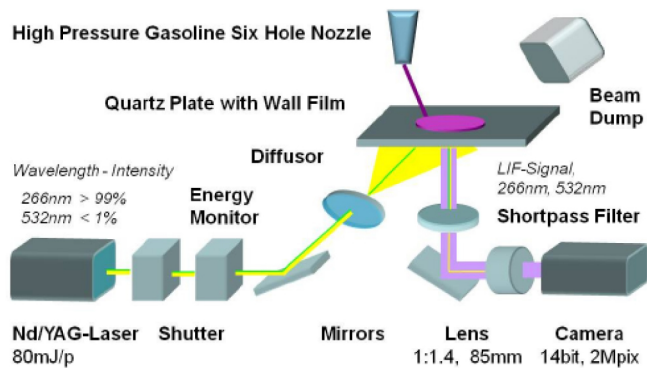


Figure 7. Experimental setup of the Laser-induced fluorescence investigations [15]

In the context of the LIF-experiments shown here, two aspects must be taken into account particularly. First, the fluorescence signal is very sensitive to disturbances. Among other things, the correct filter and short camera shutter times help to receive a meaningful LIF-signal. Secondly, quantitative results can only be obtained after a detailed calibration. For the present study, the drop calibration method (cf. [4]) was used for the evaluation. The detailed description of the experimental setup, the signal-wavelengths used and the calibration can be found in [15].

The optical engine

The main technical data of the optical engine used for the investigations is listed in table 1.

Table 1. Engine Specifications

	Single-Cylinder Engine
Displaced volume	449 cc
Stroke	85 mm
Bore	82 mm
Compression ratio	9.5:1
Injector	BOSCH HDEV5 6-hole

High- and low-pressure indication was applied at this engine. Additionally, there was the possibility to visualize the injection process as well as the subsequent combustion process via an endoscopic access and the corresponding high speed camera. The area of the combustion chamber visible through the endoscopy is shown in figure 8.

All areas which are of relevance to soot formation, such as the piston and the fire land are clearly visible. In this case, the image frame rate was $f = 6000$ Hz, i.e. at an engine speed of $n_{\text{Mot}} = 2000$ rpm, a picture was recorded every 2 CA. A part load operating point, with an engine speed of $n_{\text{Mot}} = 2000$ rpm and an indicated mean effective pressure of $\text{imep} = 5$ bar was considered for the verification of the numerical simulations presented in the framework of this paper.

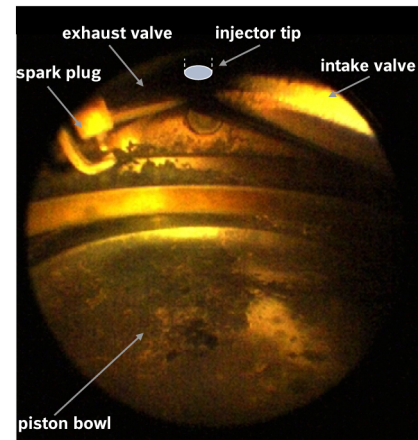


Figure 8. Viewing range of the endoscopy

Numerical Setup

Modeling spray-wall interactions

The coupling of the dispersed phase with the wall film is done for each drop impinging on the wall according to the corresponding spray-wall interaction model. Generally several different physical phenomena can be observed here, depending on the impact condition. It is common to divide the phenomena occurring during the impingement of the droplets on the wall into different regimes, see e.g. Bai and Gosman [2]. The drop properties, like drop speed or drop diameter, the liquid properties and also the wall properties like wall temperature or wall roughness are generally distinguishing between a deposition, breakup or splashing of the impinging droplet. Therefore, based on the criteria mentioned above, the spray-wall interaction model has to determine the corresponding regime first. In a further step, the properties of the secondary droplets have to be determined, e.g. in the case of a droplet breakup, the secondary drop mass, drop speed and drop size as well as the ejection angle and in case of deposition the exchange of mass and enthalpy with the wall film.

In the context of this contribution, the spray-wall interaction was modeled according to a model originally developed by Kuhnke [12] and further refined by Birkhold [3]. As can be seen in figure 9, the basic regimes are similar to the classification of Bai and Gosman [2], but the quantitative definition of the regime limits is based on a dimensionless analysis of a comprehensive database. This database included single drop as well as real spray investigations under engine-relevant conditions. Additionally different fluids, including also some hydrocarbons, were taken into account. Compared to the commonly used spray-wall interaction models, the main advantage of this model is the consideration of the wall temperature. As shown in figure 9, the regime classification is additionally based on a dimensionless wall temperature T^* , with: