

# Experimental Investigation of Fuel Impingement and Spray-Cooling on the Piston of a GDI Engine via Instantaneous Surface Temperature Measurements

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## ABSTRACT

In order to comply with more and more stringent emission standards, like EU6 which will be mandatory starting in September 2014, GDI engines have to be further optimized particularly in regard of PN emissions. It is generally accepted that the deposition of liquid fuel wall films in the combustion chamber is a significant source of particulate formation in GDI engines. Particularly the wall surface temperature and the temperature drop due to the interaction with liquid fuel spray were identified as important parameters influencing the spray-wall interaction [1].

In order to quantify this temperature drop at combustion chamber surfaces, surface temperature measurements on the piston of a single-cylinder engine were conducted. Therefore, eight fast-response thermocouples were embedded 0.3  $\mu\text{m}$  beneath the piston surface and the signals were transmitted from the moving piston to the data acquisition system via telemetry.

Extensive parameter variations were performed, in order to investigate the influence of e.g. the rail pressure, the engine load and the engine speed on the surface temperature of the piston. In particular the very slow increase of the piston surface temperature in comparison to the fast increase of the engine load could be determined as the main cause of the rise in particulate emissions during dynamic engine operation.

The shown studies can be used as a database from which, important measures to reduce particulate emissions can be derived. Additionally, the numerical simulation can be improved significantly based on the extensive measurement data, representing another important step towards the prediction of particulate emissions by numerical simulation.

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## INTRODUCTION

According to the final stage EU6c of the upcoming new emission standard, a limit of 6e11 particulates/km for the NEDC is postulated. Therefore, the PN-emissions and the mechanisms responsible for the soot formation have to be analyzed in detail.

Recent results of fundamental gasoline engine research are showing that, provided that the temperatures are sufficiently high, ranging from 1500 K to 2200 K, the starting point for the formation of soot particles is a locally fuel-rich mixture [2]. In stoichiometric 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. [3], 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.

Thus, in order to meet the mentioned PN-limit, an efficient optimization of the mixture formation and combustion process is required. This can only be achieved by the complementary use of experimental testing and simulation. Especially the method of computational fluid dynamics (CFD) allows a detailed analysis of the in-cylinder processes. But, in order to be able to analyze soot emission sources by numerical simulation, a detailed description of the entire underlying model chain is a crucial prerequisite. Special emphasis has to be spent on the modeling of spray-wall interaction and the wall film dynamics.

As shown in detail in [1], the validation of the mentioned spray-wall interaction and wall film models based on basic experimental investigations, like the infrared-thermography and the laser-induced fluorescence based measurements, demonstrated that the employed modeling is able to predict quantitatively correct in most instances. Additionally, on the basis of these investigations, the respective surface temperature and especially the temperature drop due to the spray-cooling was determined as an important parameter influencing the spray-wall interaction.

However, the approach conventionally used in engine CFD calculations assumes a constant wall temperature over the whole engine cycle. Thus the spray-cooling cannot be considered. This fact results in a wall film mass calculated considerably too small. For this reason, the approach to calculate the lateral heat conduction in thin walls, used by Koepple et al. [1], was extended on moving meshes. With this approach the CFD calculation is able to consider the temperature drop due to the spray-cooling in engine calculations. So the objective of this work is to advance the understanding of the spray-wall interaction phenomena and to quantify the temperature drop due to the spray cooling in an engine. Therefore surface temperature measurements on the piston of a single-cylinder engine were conducted.

Only few studies have been published investigating the varying piston surface temperature during an engine cycle. Bargende [4], Steeper et al. [5] and Wang [6] measured the instantaneous surface temperature using surface thermocouples, while Buono et al. [7] performed optical measurements of the piston surface temperature during the motored and fired operation of an internal combustion engine. However, there are hardly any studies published investigating the influence of the spray cooling on the surface temperature of the piston. Merely Cho et al. [8] noticed a temperature drop due to the spray cooling of the impinging spray on the piston at early start of injections. Thus the objective of this work is a detailed investigation of the influence of spray cooling in GDI engines. Therefore the temporal and spatial development of the piston surface temperature during an engine cycle and especially during the injection was investigated using measurements with a single cylinder engine.

The paper is structured in the following way. The experimental setup, including the engine setup, the installation of the fast response thermocouples and the description of the data transfer using the telemetry system is shown in the next section. Afterwards the signal processing in order to obtain crank-angle resolved heat fluxes and the influence of soot deposits on the measured temperature signal is discussed. The result section begins with a description of the used methodology. Then various parameters influencing the surface temperature on the piston, like start of injection, engine speed, engine load and the ignition timing are shown. The last section offers a summary and conclusions.

## EXPERIMENTAL SETUP

### Engine with Optical Access

The main technical data of the optically accessible engine used for the investigations is listed in [table 1](#).

Table 1. Engine Specifications

|                     | Single-Cylinder Engine                        |
|---------------------|-----------------------------------------------|
| Displacement volume | 449 cc                                        |
| Stroke              | 85 mm                                         |
| Bore                | 82 mm                                         |
| Compression ratio   | 9.5:1                                         |
| Injector            | BOSCH HDEV5 6-hole solenoid (Cone Angle: 83°) |
| Fuel Type           | Gasoline (Super - Plus, RON 98)               |

High- and low-pressure indication was applied at this engine. Additionally, the injection process as well as the subsequent combustion process was visualized using an endoscopic access with attached high speed camera. The area of the combustion chamber visible through the borescope is shown in [figure 1](#).

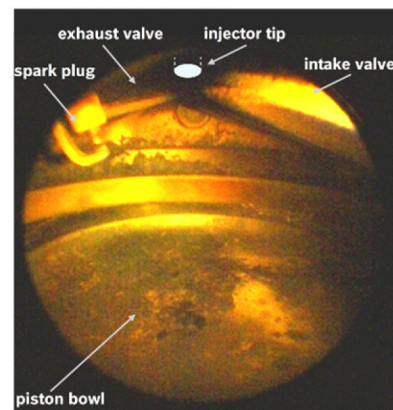


Figure 1. Viewing range of the borescope

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 set to  $f = 6000$  Hz, i.e. at an engine speed of  $n = 2000$  rpm, a picture was recorded with a crankangle resolution of 2 CA



## Instantaneous Surface Temperature Measurements

### Fast Response Surface Thermocouples

The instantaneous piston surface temperatures were measured using fast response surface thermocouples mounted in a specially prepared piston. A very high thermal and mechanical resistance of the thermocouples is required. Additionally they have to be as small as possible, in order to have a very small influence on the temperature field in the piston and in order to be able to mount many thermocouples in a very small area. Furthermore, the proportionally low temperature fluctuations in motored engine operation as well as in lower part load operation require a sufficiently large electrical measurement signal with a high signal-to-noise ratio. Therefore K-type coaxial NiCr-Ni thermocouples with a diameter of 0.5 mm and a temperature range of 0°C to 650°C were used. This very common thermocouple has a high thermal stability at high temperatures and at the same time a sufficiently high resolution of the measurement signal of roughly 40  $\mu\text{V/K}$ . Furthermore, in order to capture the high-frequency temperature changes caused by the injection, the thermocouples must have a very low response time in the order of a microsecond. As shown in [4] this requires a very small wall thickness over the sensing end of the probe of less than 2  $\mu\text{m}$ . For this reason, the thermocouples were glued in the corresponding bore hole in the piston with a small overlap. They were then cut at the piston surface level, exposing the thermocouple elements without electrical contact. By applying a liquid ceramic binder, the MgO-powder used as an insulation material can be cured to a stable ceramic with approximately the same thermal properties as the thermocouple elements. As shown in figure 2, a vacuum-deposited chromium coating with a thickness of 0.15  $\mu\text{m}$  forms a metallurgical bond between the two thermocouple elements, thus forming the thermocouple junction. Additionally in order to protect the thermocouples against mechanical stresses, a layer of fused-silica with a thickness of 0.15  $\mu\text{m}$  was vacuum-deposited on the piston. The response time of this experimental setup is in the order of 0.3  $\mu\text{s}$  [9].

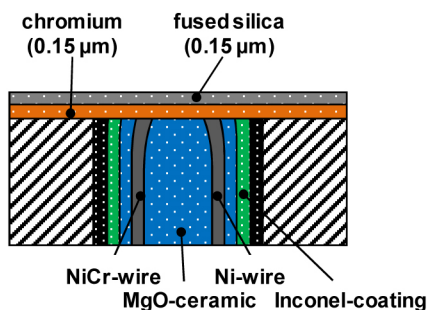


Figure 2. Structure of the NiCr-Ni surface thermocouple

Using the shown technology, a GDI aluminum piston was instrumented with eight fast-response thermocouples. As shown in figure 3, the thermocouples were positioned in the impingement area of one of the six spray-beams of the used multihole-injector. Since this impingement area is strongly

dependent on the start of injection, the thermocouples were arranged as shown in figure 3, covering a large portion of the injection timing dependent impingement area.

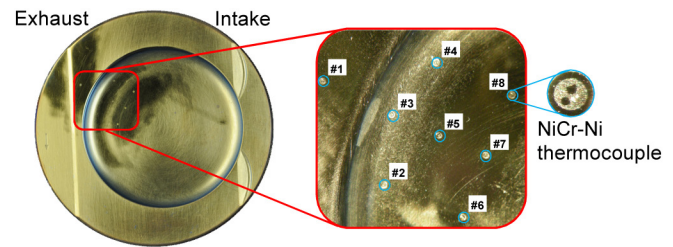


Figure 3. Eight fast-response surface thermocouples positioned in the surface of the coated piston

### Telemetry System

A 4-channel telemetry system with a high-speed PCM technique was used for the acquisition and transmission of the measured temperature data of the moving piston. Figure 4 shows the telemetry system installed in the crankcase.

As illustrated in figure 4, a cylindrical induction coil (rotor antenna) is mounted beneath the piston and attached to the signal amplifier, enabling the wireless energy and data transmission. Additionally, a ferromagnetic metal pin (stator antenna) is attached to the cylinder. When the engine is running the stator antenna enters the rotor antenna for a certain crank angle period and a periodic electric coupling arises. During this time frame the energy needed is transmitted inductively and stored in the telemetry system. Above a rotational speed of  $n = 200$  rpm, this amount of energy transmitted and stored allows a measuring operation of up to 1.4 s. During this time, the analog voltage signals of the thermocouples are recorded and stored with a sampling rate of 40 kHz/channel and a 12-bit accuracy. In the subsequent transmission mode, the measured data and a trigger signal, enabling the synchronization of the measured temperature data with the measurement signals of the in-cylinder pressure measurement of the engine, are transmitted piecewise during the inductive contact to the stator antenna.

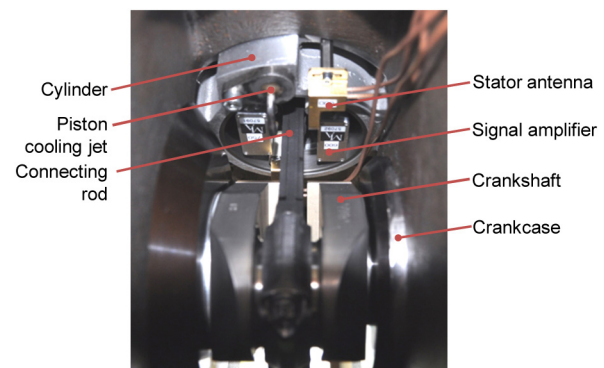


Figure 4. Telemetry system mounted in the crankcase

After completion of this transmission process, which requires approximately 3 s with a maximum transfer rate of 3 Mbit/s, a new measurement can be started.

## Data Processing

### Instantaneous Heat Flux Calculation

Due to the periodical work cycle of the combustion engine a transient heat flux between the fluid and the wall arises, resulting in a non-stationary temperature field in the combustion chamber walls. This transient temperature field can be described using the Fourier differential equation:

$$\frac{\partial T_w}{\partial t} = a \left( \frac{\partial^2 T_w}{\partial x^2} + \frac{\partial^2 T_w}{\partial y^2} + \frac{\partial^2 T_w}{\partial z^2} \right). \quad (1)$$

As shown in [4] a very high damping of the temperature oscillation occurs in the wall. So even in fired engine operation a stationary temperature field can be found in a wall depth of about  $x = 4 - 5$  mm. Therefore a one-dimensional temperature field can be assumed. If furthermore a one-sided half-infinite wall extension is presupposed, a closed-form solution of [equation 1](#) exists for the steady state by using Fourier series:

$$T_w(x, t) = T_{w,m} - \frac{\dot{q}_{w,m}}{\lambda_w} x + \sum_{i=1}^{\infty} e^{-x \sqrt{\frac{i\omega}{2a}}} \left[ A_i \cos \left( i\omega t - x \sqrt{\frac{i\omega}{2a}} \right) + B_i \sin \left( i\omega t - x \sqrt{\frac{i\omega}{2a}} \right) \right] \quad (2)$$

Differentiating this temperature field with respect to  $x$  and inserting it with  $x = 0$ , i.e. at the surface of the combustion chamber wall, into the one-dimensional heat conduction equation, the heat flux causing the surface temperature oscillation can be calculated as:

$$\dot{q}_w(t) = \dot{q}_{w,m} + \sum_{i=1}^{\infty} \sqrt{\frac{i\omega}{2a}} \cdot \lambda_w \cdot [(A_i + B_i) \cos(i\omega t) + (B_i - A_i) \sin(i\omega t)] \quad (3)$$

$A_i$  und  $B_i$  are Fourier coefficients,  $\lambda_w$  is the heat conductivity and  $a$  is the conductivity of temperature. These two material dependent constants are replaced by the heat penetration coefficient  $b$ , with

$$b = \sqrt{\lambda \rho c_p} = \frac{\lambda}{\sqrt{a}}. \quad (4)$$

However, as shown in [figure 2](#), the thermocouple junction consists of a combination of different materials, with significantly different material properties. Therefore a simple determination of a representative mean value of the heat penetration coefficient  $b$  is not possible. For this reason, the progress of the heat penetration coefficient of a NiCr-Ni thermocouple during an engine cycle was calculated by Bargende by means of a finite-element model. According to these results, the heat penetration coefficient can be assumed to be constant over the whole engine cycle, with a value of  $b = 6000 \text{ J/m}^2\text{Ks}^{0.5}$ .

Furthermore, the mean heat flux  $\dot{q}_{w,m}$  is needed to determine the heat flux according to [equation 2](#). As shown in [4] this mean heat flux can be determined with an acceptable accuracy by using a simplified method, assuming a quasi-stationary behavior of the thermal boundary layer, which was also applied in [10]: Following the Newtonian heat transfer equation, the instantaneous heat flux at the combustion chamber surface ( $x = 0$ ) has to be equal to zero at the time at which the local gas temperature outside the thermal boundary layer equals the wall temperature. Using [equation 3](#) and [4](#), the mean heat flux can be calculated as:

$$\dot{q}_{w,m} = -b \cdot \sum_{i=1}^{\infty} \sqrt{\frac{i\omega}{2}} \cdot [(A_i + B_i) \cos(i\omega t_0) + (B_i - A_i) \sin(i\omega t_0)] \quad (5)$$

Assuming a combustion chamber surface without deposits the transient heat flux can now be calculated according to [equation 3](#). However, in the gasoline engines with direct injection like the one considered here, deposits on the piston surface cannot be avoided. Thus their influence must be taken into account when calculating the heat flux. Hence a brief overview of the formation mechanisms of deposits and their effect on the measured surface temperature will be given in the following section.

### Influence of Deposits

Due to some recent studies ([11], [12], [13]), the basic mechanisms of soot deposit formation are well known, but the details of the formation mechanism are still not completely understood. Bargende [4] showed, that the soot formation on the combustion chamber surfaces is increasing initially very rapidly and is then reaching steady state, with a nearly constant soot layer thickness. The increasing insulation effect of the soot layer causes an increase in the respective surface temperature. Hence mainly the low-volatile fuel components can be deposited on the existing soot layer [14].

Previous investigations showed that two soot deposit zones with different properties are formed on the piston, respectively the plate surface ([15], [16], [13]). A shiny black, lacquer-like soot layer, mainly composed of coked unburned droplets, is

formed in the area of direct spray-wall contact. Hutfleiß et al. [13] showed that this zone is strongly influenced by the fuel composition. As the typical surface temperature of about  $T_w = 433$  K is above the Leidenfrost temperature of the high-volatile fuel components, these are in the film-boiling regime with no direct liquid-wall contact. However, the low-volatile fuel components are in the nucleate boiling regime with a direct liquid-wall contact whereby deposits with a coke-like structure are formed.

The deposits in the outer area have a foggy black structure and are mainly composed of soot [16]. Due to the low flow velocities in the wall boundary layer the soot particles in this outer area have sufficient time to accumulate at the wall. Additionally Suhre and Foster [17] showed that in this zone the physical process of thermophoresis plays an important role. As a result of the higher momentum on the hotter gas side, the soot particles present in the thermal boundary layer move along the temperature gradient in the direction of the cold wall and accumulate there. The combustion chamber deposits formed by the mechanisms explained above can be removed by increasing thermal stresses. These can be obtained by increasing the gas temperature, respectively the soot surface temperature, e.g. due to the increasing insulation effect with increasing soot layer thickness [14]. Additionally, if there is enough oxygen available at this time, the soot particles can be oxidized again.

Wimmer [18] showed in his investigations on a diesel engine, that the soot deposits reduce the wall heat flux. This is caused by insulation effects resulting in a higher surface temperature and a reduced temperature difference between the wall and the gas temperature. On the other hand, with increasing soot layer thickness, the temperature is measured in a corresponding wall depth rather than at the piston surface. As shown in figure 5 this results in an attenuation and phase shift of the measured heat flux curve.

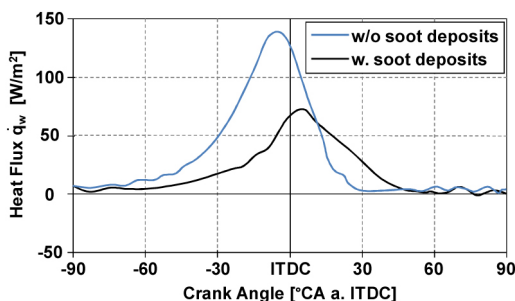


Figure 5. Influence of soot deposits on the wall heat flux [18]

In order to correct the shown effect of the deposit layer and thus determining the temperature on the piston surface without deposits a correction method was introduced by Bargende [4]. Therefore, after a certain time of fired engine operation, a reference point in motored operation with the correspondingly sooted thermocouples is chosen. Comparing those

temperature curves to the ones measured at the same reference point in motored operation with a deposit-free combustion chamber the penetration depth  $x$  is varied iteratively until there is an agreement between the two mean heat fluxes calculated at the time at which the mean gas temperature equals the wall temperature (eq. 5). Finally the temperature curve of the fired operating point is corrected with the calculated penetration depth  $x$ , using equation 2. However, since the presented solution of the differential equation of the temperature field is valid only for homogeneous materials, a prerequisite for the application of this correction method is that the soot deposits and the thermocouple materials have the same thermal properties. Thus the ratio of the thermal conductivity  $\lambda$  and the heat penetration coefficient  $b$  has to remain constant for the sooted as well as for the clean thermocouple.

## RESULTS AND DISCUSSION

### Methodology

First of all, in order to check the plausibility of the surface temperature measurement, the temperature curves in motored operation, as shown in figure 6, were analyzed.

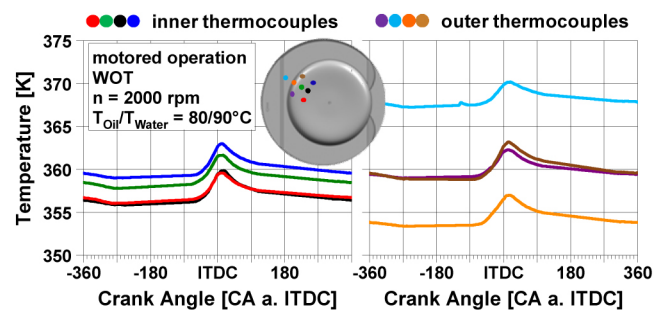


Figure 6. Temperature curves in motored operation at  $n_{Mot} = 2000$  rpm

Thus an assessment of the temperature curves measured by the individual thermocouples is possible, independently of the local influence of the injection and the combustion. As expected, the temperature levels of the thermocouples positioned in the piston bowl are within a narrow temperature range (cf. figure 6 left). The slight increase of the average temperature towards the thermocouples marked in green and blue is due to the different influence of the piston cooling, which is discussed in a subsequent section (cf. also figure 28). However, as illustrated on the right side of figure 6, the temperature curves of the thermocouples located in the outer area of the piston show a significant difference in the average temperature level. Hence the average temperature level of the thermocouple marked in blue is significantly higher than the others, whereas the average temperature level of the thermocouple marked in orange is well below the average temperature levels of the remaining elements. The higher temperature level of the thermocouple marked in blue is due to the arrangement outside of the piston bowl in the direction of

the liner. On the one hand, the influence of the piston oil cooling (oil is sprayed to the bottom of the piston) is reduced because of the higher wall thickness in this area (cf. [10]), on the other hand the influence of the frictional heat is increasing in the direction of the liner. The lower temperature level of the thermocouple marked in orange is probably caused by a slight offset of the considered thermocouple, although a detailed calibration process of the entire measurement setup was performed before the experimental investigations.

As explained above, even after a short time of fired engine operation a soot layer is deposited on the piston surface, resulting in an attenuation and a phase shift of the measured temperature curve. In order to investigate these effects, the surface temperature curves of the piston were measured at regular intervals during a measurement run of five-hour continuous engine operation (cf. figure 7). In order to be able to additionally perform the deposit correction method (cf. eq. 5 and 2) explained in the section above, a temperature measurement in motored operation was carried out after each temperature measurement in fired operation.

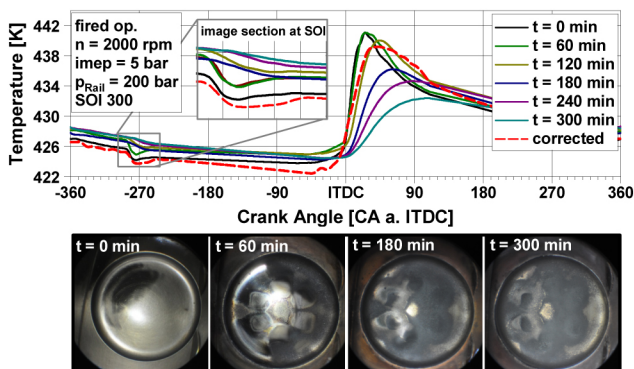


Figure 7. Time-dependent influence of soot deposits on the cycle-averaged (23 cycles) temperature curves measured at the thermocouple marked in red

Thereby, the increasing attenuation and phase shift of the temperature curve due to an increasing deposit layer thickness with time, as mentioned in the literature, is clearly visible. Particularly in the period between the first and the second hour of operation, a sharp increase in the attenuation of the temperature drop during the injection, as well as in the attenuation of the temperature rise during the combustion is visible. As can be seen from figure 7, this behavior correlates well with the soot deposits shown in the corresponding pictures of the piston. Thus even after a one hour operation of the engine the piston remains almost free of deposits on the exhaust side of the piston and therefore in the area of the thermocouples, whereas a remarkable deposit layer is build up on the inlet area of the piston. After two hours of operation a first deposit formation can be recognized on the exhaust side of the piston, corresponding to the strong attenuation and phase shift of the temperature curve at this time.

The overall higher temperature in the exhaust area might be one of the reasons for the delayed deposit formation on the exhaust side of the piston. Thereby the liquid fuel deposited on the piston evaporates before the onset of combustion and only a relatively small portion of the coke-like deposits can be formed. Another reason might be the cylindric charge motion, rotating counter-clockwise and thus generating a flow from the exhaust area towards the inlet area. This causes the fuel as well as the soot particles to be deposited preferably on the intake area of the piston.

The deposit-correction-method according to [4] is one possibility to correct the shown effects of the deposit layer and thus to determine a temperature curve corresponding to the deposit-free state of the piston. By applying this correction, the attenuation and the phase shift of the temperature curve can be corrected and a temperature signal almost corresponding to the temperature signal of the piston surface free of deposits can be calculated (cf. figure 7). However, mainly because of the assumption that the soot deposits and the thermocouples have similar thermal properties, no complete correction of the temperature curve is possible. Therefore, the measurements shown in this article were conducted with a piston surface almost free of deposits.

As explained before, the soot deposits can be removed thermally and mechanically. Additionally, if there is enough oxygen available, the soot particles can be oxidized again. For that purpose, ethanol ( $C_2H_6O$ ) was temporarily used as a fuel in order to remove the formed deposits. In contrast to gasoline fuel, ethanol has no aromatic compounds, has a lower C/H-ratio and thus has a smaller sooting tendency. In addition, due to the oxygen atom present in the ethanol molecule, the soot oxidation is intensified. The influence of the deposit degradation process on the temperature curve is shown in figure 8.

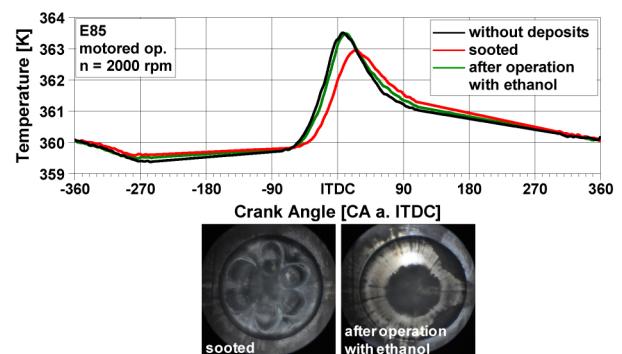


Figure 8. Removal of soot deposits by a temporary fired operation with E85 injected with an outward opening piezo injector as well as their influence on the temperature curves in motored operation, measured at the thermocouple marked in black