

A New Model to Describe the Heat Transfer in HCCI Gasoline Engines

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ABSTRACT

In this work, heat loss was investigated in two different HCCI single cylinder engines. Thermocouples were adapted to the surfaces of the cylinder heads and the temperature oscillations were detected in a wide range of the engine operation maps. The resultant heat transfer profiles were compared to the heat losses predicted by existing models. As major discrepancies were stated, a new phenomenological model was developed that is well-manageable and describes the heat loss in HCCI mode more precisely than existing models.

To analyze the insulating effect of deposits, the heat transfer equation was solved analytically by an approach that allows consideration of multiple layers with different material properties and thickness. This approach was used for the first time in conjunction with engines to calculate the heat flux at the surface of deposits and the deposit thickness. Additionally, the heat loss measured in HCCI operation was compared to the heat loss detected in homogeneous SI operation in the same engine. The differences in the heat transfer characteristics are demonstrated.

INTRODUCTION

Controlled autoignition in gasoline engines is a promising concept to simultaneously reduce both emissions and fuel consumption of internal combustion engines. Autoignition of gasoline is realized by compressing a fuel/air mixture in the cylinder until ignition occurs. The high temperatures required for autoignition are obtained by trapping hot residual gas in the combustion chamber. With HCCI, it is possible to realize combustion of very lean mixtures. Therefore, peak temperatures in the combustion chamber are reduced and NO_x-formation due to high temperatures can be avoided [1-3].

The autoignition of the mixture depends on the thermodynamic conditions inside the combustion chamber [4-6]. To gain information about these conditions and to estimate the heat release rates, pressure analyses are generally performed using the indicated pressure curves. Thereby, the heat loss is usually calculated by models that were empirically determined with conventional gasoline or diesel engines [7-16]. Hence, the heat loss calculated for HCCI engines is imprecise which leads to uncertain calculations of the heat release rates and the thermodynamic conditions in the combustion chamber [17].

In [17], Chang investigated the heat transfer in an HCCI gasoline engine operated with an exhaust re-breathing strategy. Thermocouples were adapted to the piston and cylinder head surface to measure the instantaneous wall temperatures and heat fluxes. In HCCI mode with premixed fresh charge, the spatially-averaged heat flux represents the global heat transfer well, as the uniformity of the in-cylinder conditions leads to small spatial variations of the measured heat fluxes. The comparison of measured heat flux curves and predictions of existing heat transfer correlations (Woschni, Hohenberg, Annand & Ma) shows major discrepancies. For the correlation of Woschni, Chang traced the problem back to the flame propagation term. Hence, the authors suggest a modification of this model in order to improve its prediction of heat transfer in HCCI gasoline engines. However, the comparison of measured and calculated heat flux profiles still shows discrepancies. A difference in the phasing of the profiles is observed. The peaks of the averaged heat flux curves occur about 5 CAD later than the calculated peaks of the heat transfer, due to dynamics of the transient heat transfer phenomena in the boundary layer.

In this work, the heat loss was investigated in two different HCCI single-cylinder engines with negative valve overlap. Thermocouples were adapted to the combustion chamber surfaces of the cylinder heads and the temperature oscillations were detected in a wide range of engine operation conditions. The resultant heat transfer profiles were compared to the heat losses predicted by existing models during pressure curve analyses. The discovery of major discrepancies guided the development of a new phenomenological model, which is easily managed and more precisely describes heat losses in HCCI mode when compared to existing models. The model is consistent in its units and considers the small mass flows of fresh air when the engine runs at high internal EGR rates. All required parameters are standard parameters and the programming effort is as low as for widely-used models of Woschni [8] or Hohenberg [13].

Furthermore, the heat loss measured in HCCI operation is compared to the heat loss detected in homogeneous SI operation in the same engine. At low load, the heat transfer in HCCI mode shows a different characteristic than the heat transfer determined in homogeneous SI mode. With increasing load, both profiles become more equal as the processes become more similar.

EXPERIMENTAL APPARATUS

The experimental investigations were performed with two single-cylinder gasoline engines (see Table 1). To achieve autoignition by trapping hot residual gas in the combustion chambers of both engines, cam phasers were adapted to the exhaust and intake camshafts. Engine I is operated with modified camshafts (valve stroke: intake 2 mm, exhaust 1.5 mm). Engine II is equipped with a variable valve train. In Engine II, a compression ratio of 11.5:1 was used for the current experiments. PFI and DI operation is possible in both engines. In DI operation, the injector is located in the centre of the cylinder head.

Table 1: Engine Specifications

	Engine I	Engine II
Basic Engine	Rotax F650	Ricardo
Compr. ratio	11.5 : 1	9 - 12 : 1
Displacement	652 cm ³	499 cm ³
Bore x Stroke	100 x 84 mm ²	84 x 90 mm ²
Valve train	Cam phasers	VVT

In Engine I, surface temperature measurements were carried out at 8 locations and in Engine II at 3 locations of the cylinder head (see Figure 1). To reduce the computational effort, the heat flux is derived from the cycle-averaged temperature signals of 3 times 100 consecutive cycles for each location and afterwards spatially averaged for each operating point and engine.

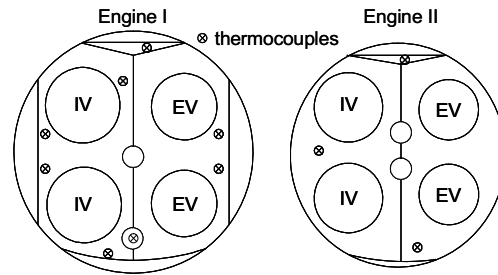


Figure 1: Locations of the thermocouples in both engines

The measurements were carried out with thermocouples that were specially modified for application in internal combustion engines [16-21], as the response time of standard thermocouples is insufficient. The thickness of the sensing end of the thermocouples is less than 2 microns and the response time is on the order of a micro-second. A detailed analysis of the accuracy of these thermocouples is given in [18]. The temperature oscillations as well as the pressure curves were recorded with a temporal resolution of 0.1 CAD. To estimate the heat flux profiles, the surface temperature oscillations are analyzed by an FFT-analysis. The detailed method is given in [16-20]. By this method, the total heat flux is calculated by a transient time-dependent term which coefficients A_k and B_k are determined by FFT-analysis and by a steady-state term which is derived from backside temperature measurements 5 mm below the surface:

$$\dot{q}_w(x, t) = \dot{q}_w + \lambda \sum_{k=1}^N \sqrt{\frac{k\omega}{2a}} \cdot e^{-x\sqrt{\frac{k\omega}{2a}}} \cdot \left[(A_k + B_k) \cos \left(k\omega t - x\sqrt{\frac{k\omega}{2a}} \right) + (B_k - A_k) \sin \left(k\omega t - x\sqrt{\frac{k\omega}{2a}} \right) \right] \quad (1)$$

In equation (1), a is the thermal diffusivity of the medium and x the distance to the surface. The heat flux at the surface of the cylinder wall is derived by setting $x=0$. To give an impression of the temporal variation of the measurement, the ensemble-averaged heat flux profile and the standard deviations are shown in Figure 2. The standard deviation is highest near the peak heat flux due to cyclic combustion variations.

The influence of combustion chamber deposits on the measured temperature oscillations is shown in [22,23]. With increasing thickness of the deposit layer, the measured temperature signal appears damped and shifted in phase. Generally, formation of deposits cannot be completely avoided. To account for the effects of the deposits on the measured temperature signals, two different approaches exist. With the first approach, it is possible to estimate the heat flux that would exist at the cylinder wall surface if there were no deposits. This estimated heat flux is of interest for validating the heat flux models used in cylinder pressure analysis and process simulations. These simplified heat flux models assume that no deposits exist on the cylinder walls. By using the second approach, it is feasible to calculate the

current thickness of the deposit layer and the real heat flux at the deposit surface.

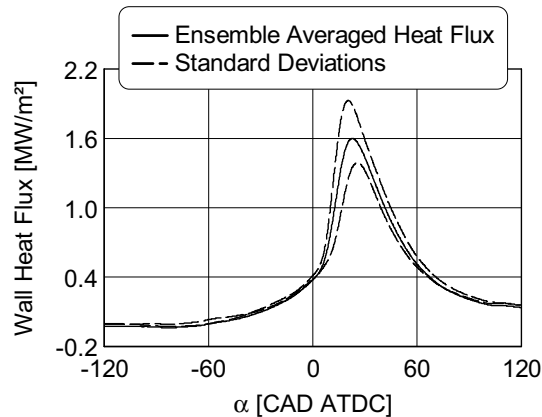


Figure 2: Ensemble-averaged heat flux and standard deviations

The first approach, based on equation (1), makes use of the following assumptions:

- The features of the temperature profile at the deposit surface occur at the same time as those measured by the thermocouple with a clean surface.
- The thermal properties of the sensor and the deposit are quite similar, as the aforementioned solution of the heat transfer equation only applies for homogeneous compounds.

A reference operating point in motored operation is defined and the temperature oscillations are measured under clean conditions. By repeatedly performing these reference measurements between experiments with fired operation, the deposit thickness x can be calculated from the phase shift between the current and the original reference measurements. This thickness x is used in equation (1) to calculate the heat flux at the surface (assuming the semi-infinite cylinder wall and the deposit have the same thermal properties) for the fired conditions [16, 23]. This flux corresponds to the heat flux that would exist if there were no deposits and can be used for the validation of the heat flux model. Figure 3 shows the measured heat flux in the motored reference point at $n=2000$ rpm with and without deposits. Additionally, the heat flux estimated for a clean surface calculated by the measurement with deposits is shown as well. A good accordance between the measurement without deposits and the estimation of heat flux at the clean surface can be stated. The error using this approach is small as long as the deposit thickness is small ($< 50 \mu\text{m}$) [16]. To validate the heat transfer model, this approach was used as the heat flux at the clean surface is of major interest.

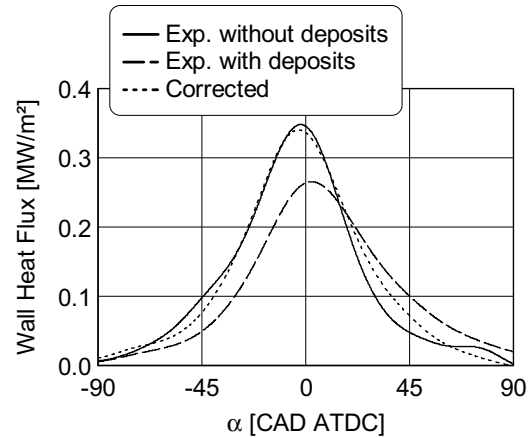


Figure 3: Comparison of measured (Exp.) heat flux with and without deposits and estimation of the corrected heat flux at the clean surface calculated by the measurement with deposits. $n=2000$ rpm, motored, Engine II

This first approach is often used to determine the real thickness of the deposits, to calculate the thermal properties of the deposit, or to determine the real heat flux at the deposit surface. Therefore, this approach is not valid, as the solution of the heat transfer equation in equation (1) only applies for homogeneous compounds and semi-infinite thickness, when in fact the thermal properties of combustion chamber and deposit differ in some magnitudes. The aforementioned second approach, which is explained in [24], is an analytical approach that allows calculation of heat conduction in a composite slab subject with periodic temperatures. The heat transfer equations for multiple layers with different material properties are solved by Laplace-transformation. For the first time, this approach was used to determine the deposit thickness and the heat flux at the surface of the deposit in combustion chambers of engines. The molecular formation and thermal properties of the deposit are given in [25-26].

To validate this new approach, FEM simulations were performed. The deposit thickness and the temperature oscillation at the deposit surface were calculated for one of the reference measurements using this new approach. An FEM model including the calculated deposit layer ($x=11.9 \mu\text{m}$) and the combustion chamber wall was set up, and the analytically calculated temperature oscillation was used as a boundary condition. The numerically calculated temperature oscillation at the measurement point was then compared with the measured one, and a good accordance was observed (see Figure 4). For comparison's sake, equation (1) yields a deposit thickness of $x=37.8 \mu\text{m}$ and therefore to a strong deviation between the results of the FEM validation and the measurement. To investigate the influence of the deposits on the heat flux and to calculate the deposit thickness in Figure 8 and Figure 9, this second approach was used.

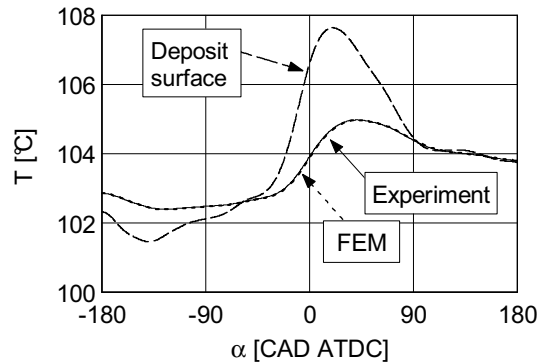


Figure 4: Comparison of measured and simulated temperature oscillations, $n=2000$ rpm, motored, Engine II

RESULTS

OPERATING CONDITIONS – To analyze the heat transfer in HCCI mode and to validate the new model by experimental data, measurements were performed in the operating ranges given in Table 2. Furthermore, the location and start of injection were varied. The coolant and oil temperature was held at 95°C and the intake temperature was set in all experiments to 30°C . Premium gasoline was used for all measurements and the operating points were adjusted for best fuel efficiency.

Table 2: Operating Conditions

Max. operating range in HCCI mode	Engine I	Engine II
Engine speed [rpm]	1500 - 3000	1000 - 4000
IMEP [bar]	1.5 - 3.5	1 - 8

INJECTION LOCATION AND INJECTION TIMING – In [17], it was shown that the heat flux measured at different locations in HCCI mode with exhaust gas re-breathing and premixed fresh charge shows small spatial variations, even among piston and cylinder head. Hence, the spatially-averaged heat flux represents the global heat transfer behavior in HCCI engines well. In this work, the heat transfer with internal and external mixture preparation and different injection timings is compared to get an impression of the spatial variations in heat transfer and of the influence of mixture inhomogeneity. Figure 5 shows the local temperature oscillations and Figure 6 shows the local heat fluxes with PFI and DI (SOI=360 CAD BTDC).

The spatial variation is of the same magnitude as in [17] and with both, PFI and DI, the spatial variations are of the same magnitude. As the port fuel injection is far away from the intake valves, an almost premixed fresh charge and a quite uniform in-cylinder mixture is assumed. Hence, it is assumed as in [17] that the global heat flux is well represented by the spatially-averaged heat flux. In addition, this result was confirmed by heat

balances. Assuming the spatially-averaged heat loss the experimentally-determined fuel conversion is calculated by an error of less than 2%. By switching from PFI to DI, the spatial variations of the local heat fluxes and the temperatures at the local measurement points vary only slightly. Hence, it is assumed that the global heat flux is well represented by the spatially-averaged heat flux in case of direct injection as well.

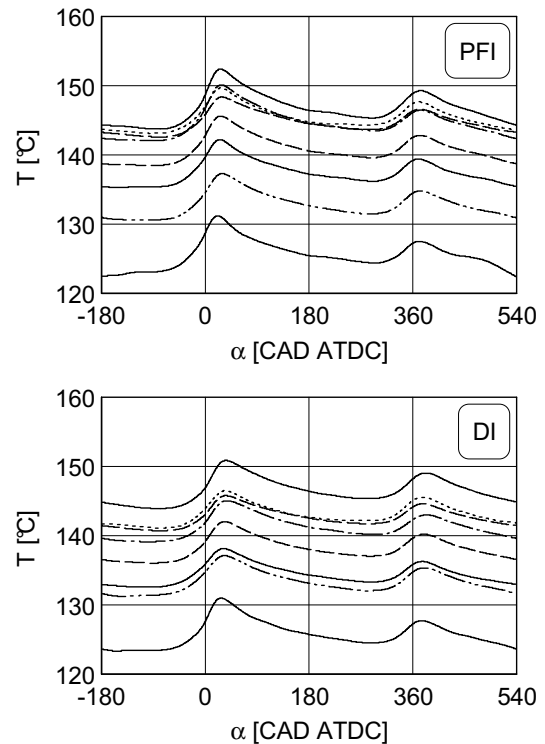


Figure 5: Spatial variations of the local temperature oscillations with PFI (top) and DI (bottom), $n=2000$ rpm, IMEP=2 bar, Engine I

For further investigation of the influence of mixture inhomogeneity on the spatial variation of the heat transfer, a variation of the injection timing was performed. By shifting the start of injection, the combustion process can be influenced; the homogeneity of the mixture influences the spatial and temporal progress of the reactions [6,27-29]. The top plot in Figure 7 shows the local heat fluxes for early injection timing and homogeneous mixture. On the bottom, the heat fluxes for late injection timing with a more inhomogeneous mixture are shown. Again, the spatial variation of the heat flux is in both cases of the same magnitude. Hence, the spatially-averaged heat flux again represents the global heat transfer into the cylinder head well.

The conditions in the boundary layer, especially the pressure and the temperature, are crucial for heat transfer. It can be assumed that the pressure in the boundary layer follows the global pressure in an engine. It is suggested that in HCCI mode, the temperature in the boundary layer depends more on the global conditions in

the combustion chamber and on the local ignition delay time, which is a function of wall temperature and local mixture distribution. The temporal and spatial progress of the reactions in the combustion chamber seems to have a minor effect on the conditions in the boundary layer, as molecular transport processes can be neglected in HCCI mode [30]. Furthermore, the differences in temperature between burned and unburned zone are significantly smaller than in SI operation, which leads to lower variations of the local temperature difference between gas and wall.

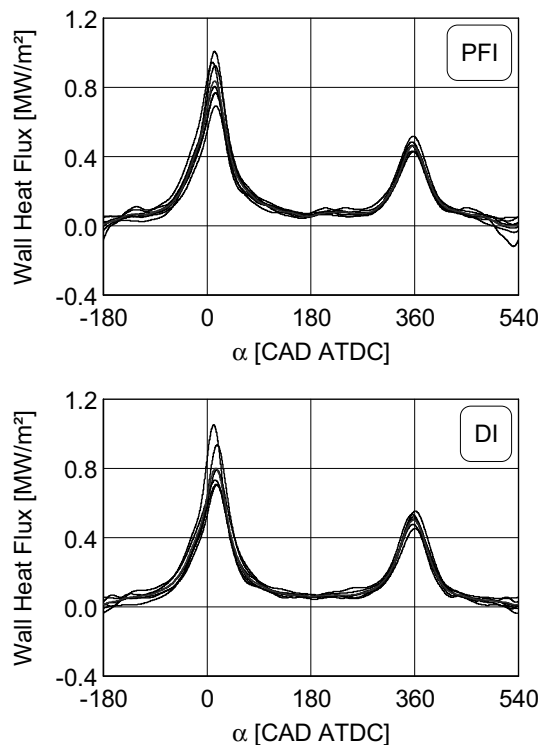


Figure 6: Spatial variations of the local heat flux with PFI (top) and DI (bottom), $n=2000$ rpm, IMEP=2 bar, Engine I

INFLUENCE OF DEPOSITS – Figure 8 shows the estimated heat flux at the clean surface, at the surface of the deposit, and the measured heat flux for a chosen operating point ($n=1000$ rpm, IMEP=4 bar). The heat flux at the surface of the deposit is calculated by the method presented in [24]. The deposit thickness was calculated as only $x=12\text{ }\mu\text{m}$ and a strong insulating effect is observed. The heat losses through the cylinder head and the piston surface are 6.1 % lower than in the clean condition, assuming similar heat transfer at the piston and cylinder head. Referring to the complete combustion chamber, the heat losses are 4.2 % lower, as the heat loss for a clean surface is assumed on the cylinder running surface.

In [16], the deposit layer is formed very quickly and thereafter remains constant at approximately $x=50\text{ }\mu\text{m}$. Figure 9 shows the spatially-averaged deposit thicknesses calculated for both engines. For Engine I,

the deposit thicknesses during the reference measurements are shown on the top of Figure 9. On average, the deposit thickness is less than in [16] and no constant deposit layer is formed as the engine was operated at different loads and speeds among the reference measurements shown. After the first two hours the deposit thickness reaches values of approximately $x=31\text{ }\mu\text{m}$ and decreases in the next two hours to nearly $x=15\text{ }\mu\text{m}$. Hence, the deposit layers consistently increase and decrease during HCCI operation. The signals of single thermocouples even show sudden phase shifts as the deposit on the top of a thermocouple may chip off sporadically.

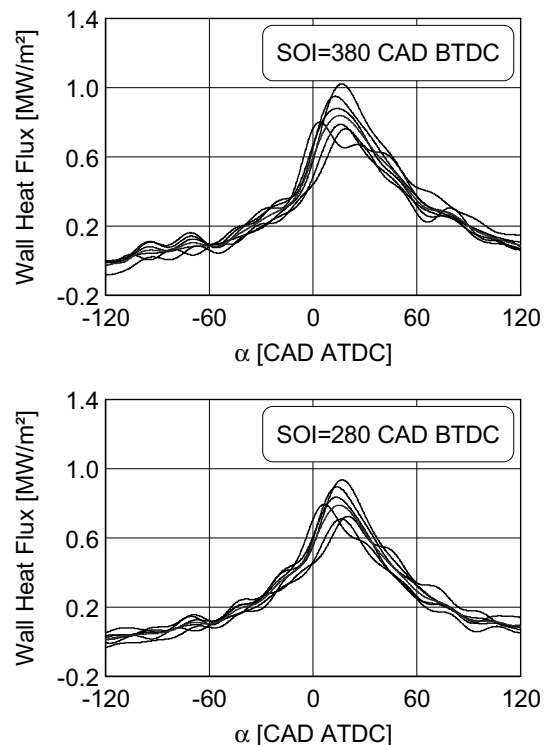


Figure 7: Spatial variations of the local heat flux for different injection timings, DI, $n=2000$ rpm, IMEP=2 bar, Engine I

As the operating map of Engine I is restricted to low loads and speeds, the deposit thicknesses calculated for the consecutive measurements performed in Engine II are shown in Figure 9 at the bottom. As this engine was also operated at different loads and speeds the deposit layer changes as well. Thicker deposits of about $40\text{ }\mu\text{m}$ were measured at operating points near idle. With increasing load, decreasing thicknesses of the deposit layers were observed. However, modifications of the deposit layers were identified due to variations of operation parameters, such as the start of injection, as well. Therefore, clear correlations between the operating point and the deposit thickness were not found. Hence, it is only feasible to generate a heat transfer model and to compare existing models for engine conditions without deposits.