

Investigations on 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 conditions. The local heat transfer is analyzed with port fuel and direct injection, for different engine parameters and operating points. It is shown that the spatially averaged measured heat loss in HCCI operation represents the global heat loss well. The spatial variations are small in the operation map presuming stable operating points with low cyclic variations and good engine performance.

Furthermore, the heat loss measured in HCCI operation is compared to the heat loss detected in homogeneous and stratified DI-SI operation in the same engine. It is shown that the local heat losses in stratified DI-SI operation show large variations, depending on the direction of the flame propagation. A comparison with the averaged measured heat loss is not valid for this operation mode.

Finally the measured heat losses in HCCI operation are compared with the predictions of existing heat transfer correlations. As major discrepancies can be stated a new model is shown which more precisely describes heat losses in HCCI mode when compared to existing models.

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, 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 local heat transfer is analyzed with port fuel and direct injection. Additionally measurements were carried out for different operating points and engine parameter variations, e.g. the injection and valve timing. It is shown that the spatially averaged measured heat loss in HCCI operation represents the global heat loss well. The spatial variations are small in the operation map presuming stable operating points with low cyclic variations and good engine performance.

Furthermore, heat losses measured in HCCI operation are compared to heat losses detected in homogeneous and stratified DI-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. In stratified DI-SI operation the local heat losses show large spatial variations, depending on the direction of the flame propagation. A comparison with the averaged measured heat loss is not valid for this operation mode.

Finally the measured heat losses in HCCI operation were compared with the predictions of existing heat transfer correlations 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.

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; 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.

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.

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

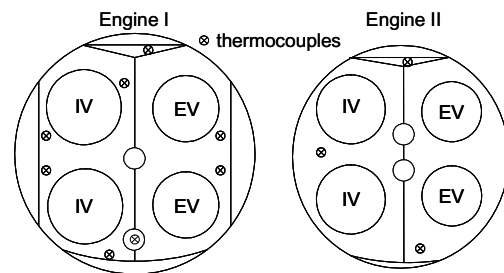


Figure 1: Locations of the thermocouples in both engines

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.

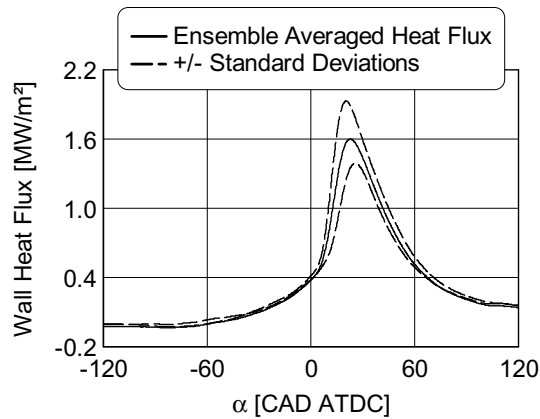


Figure 2: Ensemble-averaged heat flux and standard deviations

RESULTS

OPERATING CONDITIONS – To analyze the heat transfer in HCCI mode and to validate a new model by experimental data, measurements were performed in the operating ranges given in Table 2. Furthermore, the location and start of injection and the valve timings were varied in chosen operating points as well. 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

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. However, in HCCI mode, it is feasible to set up one single operating point with different operating parameters. E.g. depending on the concept, operating points with stoichiometric mixture are preferred to apply conventional three-way catalysts for exhaust gas after-treatment. If DENOX catalysts are used, operating points with lean air/fuel mixture are preferred to reduce fuel consumption. Hence, the influence of internal and external mixture preparation and of different operating parameters (injection and valve timing influencing the EGR rate and mixture composition) on the spatial variations of the heat transfer was investigated in this work.

INJECTION LOCATION – First of all, the influence of the mixture preparation on the spatial variations of the heat flux is analyzed. Figure 3 shows the local temperature oscillations and Figure 4 shows the local heat flux profiles with PFI and DI (SOI=360 CAD BTDC) at the operating point $n=2000$ rpm, IMEP=2 bar and $\lambda=1.1$. With both, PFI and DI, the spatial variations are of the same magnitude as in [17]. 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, the temperature oscillations at the local measurement points and their temperature level vary only slightly. Hence, the global heat flux is well represented by the spatially-averaged heat flux in case of direct injection and early injection timing as well.

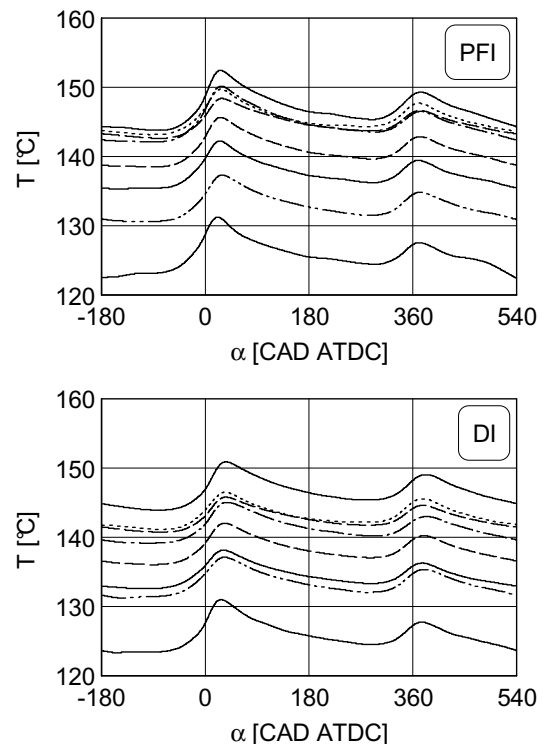


Figure 3: Local temperature oscillations with PFI (top) and DI (bottom), $n=2000$ rpm, IMEP=2 bar, $\lambda=1.1$, Engine I

VALVE TIMING – Figure 5 shows the mean surface wall temperature, the 50% mass fraction burned point (MFB), the air/fuel ratio and the EGR rate depending on the exhaust valve timing at the operating point $n=2000$ rpm, IMEP=2 bar. All these measured variables demonstrate a nearly linear dependency on modifications of the exhaust valve closing (EVC). Retarding the EVC leads to reduced EGR rates and increasing mass flows of fresh

air. The demand for constant load requires an adjustment of the injected fuel mass leading to leaner air/fuel ratios. The temperatures at the end of compression decrease and the combustion is shifted to late. The wall temperatures decrease due to lower mean temperatures and pressures in the combustion chamber resulting in lower heat fluxes.

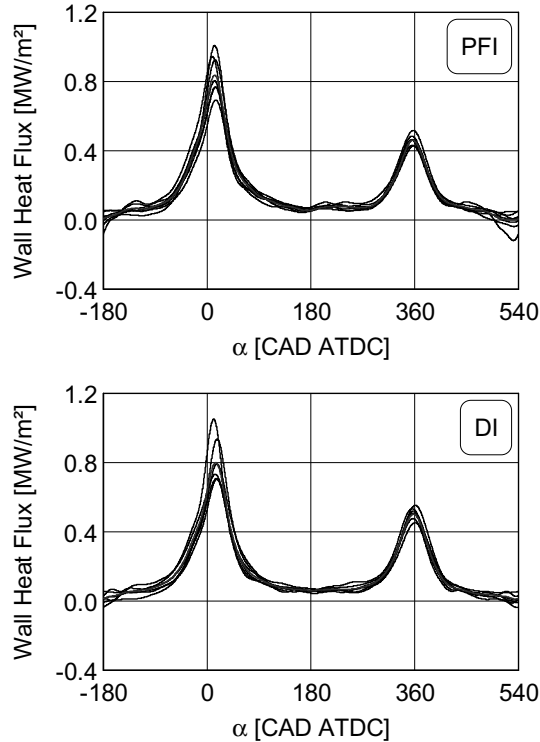


Figure 4: Local heat flux profiles with PFI (top) and DI (bottom), $n=2000$ rpm, IMEP=2 bar, $\lambda=1.1$, Engine I

The earliest and latest EVC show a deviant behavior. At earliest EVC (EVC=265 CAD ATDC) the mixture is stoichiometric. Inhomogeneities lead to incomplete combustion. The lower combustion and exhaust gas temperatures induce later ignition and combustion timings. With latest EVC (EVC=288 CAD ATDC) the injection timing was shifted to early to stabilize the combustion. Due to heat release during the negative valve overlap the temperatures at start and end of compression increase leading to an earlier combustion phasing.

Figure 6 shows the local temperature oscillations with early (EVC=265 CAD ATDC) and late exhaust valve closing (EVC=280 CAD ATDC). The local temperature oscillations only show minor variations. The changes of the mean temperatures at the particular sensors and of the temperature distribution are insignificant. The variation of the intake valve timing is not shown here, as any influence on the temperature distribution and spatial variations of the heat loss could not be determined as well. Hence, the valve timing does not influence the spatial variation of the heat transfer and in all cases the global heat flux is well represented by the spatially averaged heat losses.

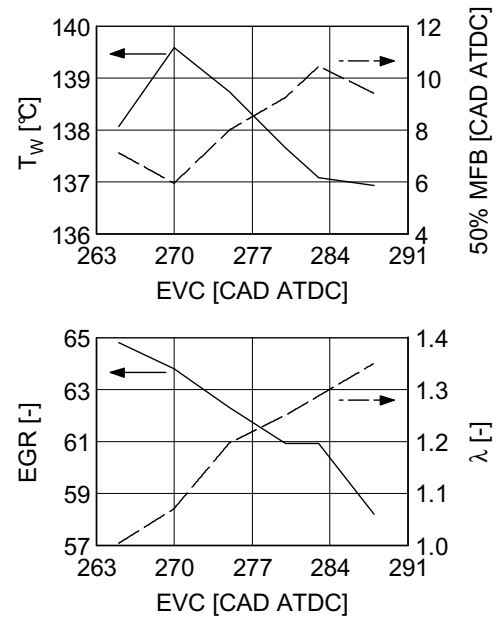


Figure 5: Mean surface wall temperature, 50% mass fraction burned point, air/fuel ratio and EGR rate depending on the exhaust valve timing, $n=2000$ rpm, IMEP=2 bar, Engine I

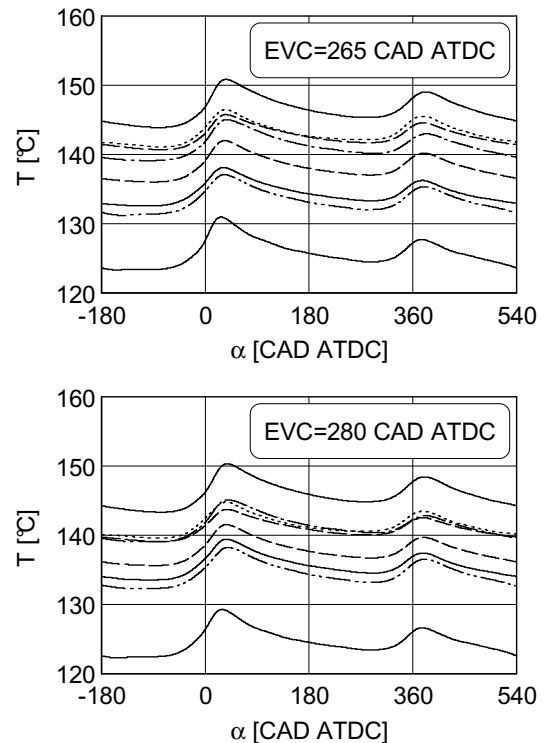


Figure 6: Local temperature oscillations with early and late exhaust valve closing, $n=2000$ rpm, IMEP=2 bar, Engine I

INJECTION TIMING – 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 at the operating point $n=2000$ rpm, IMEP=2 bar. 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 [6,24-26]. Figure 7 shows the mean surface temperature, the 50% mass fraction burned point, the maximum pressure gradient depending on the start of injection, for constant air/fuel ratio ($\lambda=1.15$). Retarding the SOI leads to later combustion phasing and decreasing pressure gradients. A very late SOI leads to earlier combustion phasing due to hot spots that ignite earlier.

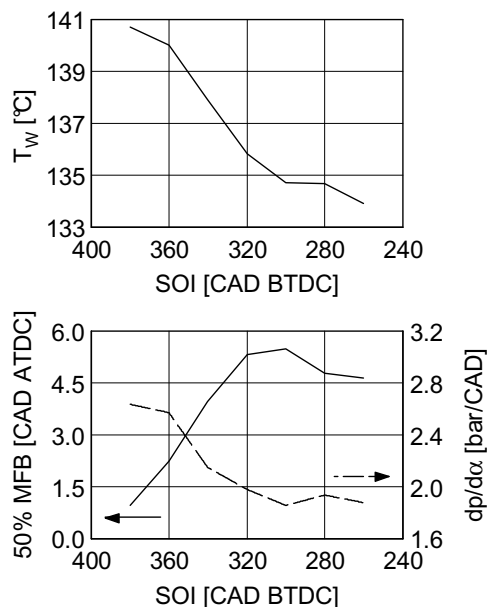


Figure 7: Mean surface wall temperature, 50% mass fraction burned point, maximum pressure gradient depending on the start of injection, $n=2000$ rpm, IMEP=2 bar, $\lambda=1.15$, Engine I

The mean surface temperatures follow the thermodynamic conditions in the cylinder. Retarding the combustion leads to lower pressures and mean temperatures in the combustion chamber. Hence, the heat fluxes into the wall decrease. The latest injection timings lead to local hot spots that ignite earlier, resulting in earlier 50% mass fraction burned points. As the inhomogeneities lead to incomplete combustion the mean temperatures and pressures decrease leading to lower heat losses and surface wall temperatures.

Figure 8 and Figure 9 show the local temperature oscillations and local heat flux profiles for early and late injection timing. As shown for the valve timing, the start of injection has no influence on the temperature distribution as well. The spatially measured temperature oscillations do not significantly change relative to one another by shifting the injection timing. It was shown that the spatially averaged measured heat flux represents the

global heat flux well for early injection timing. As the retarding of the injection timing has no influence on the temperature field and on the spatial variations of the heat flux, this approach is also valid for operating points with later start of injection.

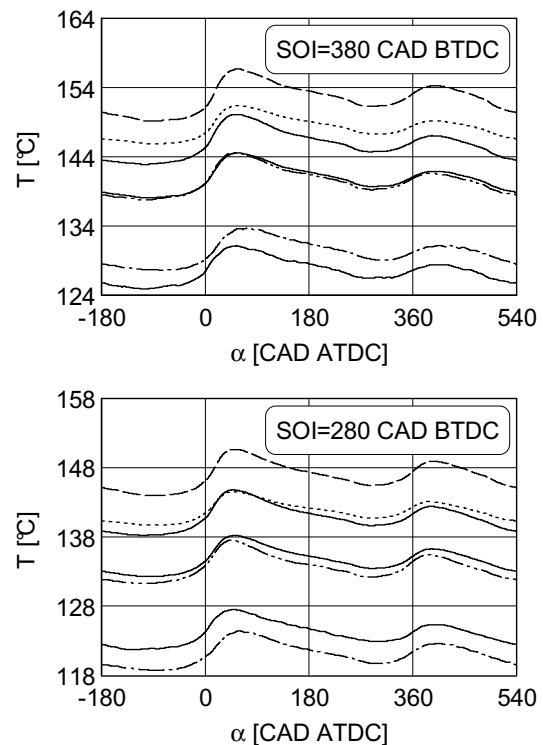


Figure 8: Local temperature oscillations for different injection timings, DI, $n=2000$ rpm, IMEP=2 bar, $\lambda=1.15$, Engine I

The conditions in the boundary layer, especially pressure and temperature, are crucial for the heat transfer. It can be assumed that the pressure in the boundary layer follows the mean pressure in the combustion chamber. It is suggested that in HCCI mode, the temperature in the boundary layer depends on the global conditions in the combustion chamber and on the local ignition delay time. The temporal and spatial progress of the reactions in the combustion chamber seem to have a minor effect on the local conditions in the boundary layer, as molecular transport processes can be neglected in HCCI mode [27]. Furthermore, the differences in temperature between burned and unburned zone are significantly smaller than in SI operation, leading to lower variations of the local temperature difference between gas and wall.

Figure 10 shows the mean averaged pressure curves and the spatially averaged heat flux profiles for early and late injection timing. The later combustion phasing due to later injection timing can be observed in the pressure curves and the heat flux profiles. The spatially averaged heat flux follows the thermodynamic conditions in the combustion chamber. Hence, the spatially averaged heat flux is well suited for the validation of heat flux curves calculated during pressure analyses.