

Some Useful Additions to Calculate the Wall Heat Losses in Real Cycle Simulations

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ABSTRACT

More than 20 years after the first presentation of the heat transfer equation according to Bargende [1,2], it is time to introduce some useful additions and enhancements, with respect to new and advanced combustion principles like diesel- and gasoline- homogeneous charge compression ignition (HCCI).

In the existing heat transfer equation according to Bargende the calculation of the actual combustion chamber surface area is formulated in accordance with the work of Hohenberg. Hohenberg found experimentally that in the piston top land only about 20-30% of the wall heat flux values from the combustion chamber are transferred to the liner and piston wall. Hohenberg explained this phenomenon that is caused by lower gas temperature and convection level in charge within the piston top land volume. The formulation just adds the existing piston top land surface area multiplied by a specified factor to the surface of the combustion chamber.

In this work, it is shown that an analytical calculation of this heat transfer problem is possible. Furthermore, it is also possible to analytically describe the influence of a possible leakage on this wall heat loss component, which results in a significant better description of the wall heat losses in the piston top land area with better results when simulating the wall temperature field of this area.

The combustion-generated convection, which increases the heat transfer during combustion, was formulated empirically by squaring the heat-transfer-increasing effect, taking into account the different driving temperature gradients from the unburned and burned zones to the combustion chamber wall. This - in principle - analytical formulation has only proven to be sufficient for heat-releasing processes close to TDC. When the heat release is moved towards the expansion stroke, with the consequence of a reduced combustion related pressure increase, an underestimation of the combustion-generated convection can be observed. This underestimation occurs by principle in all heat transfer equations, in which the combustion-induced pressure change is used as a measure of the combustion-generated convection (e.g. Woschni and Hohenberg).

This paper presents a new formulation for the combustion-generated convection that avoids this disadvantage and seems to be valid for all common combustion principles, including low temperature combustion without the existence of turbulent flame propagation.

With these adaptations, the modified wall heat transfer equation correctly reflects the transient, spatially averaged wall heat losses for conventional, homogeneous operated SI-engines, stratified combustion in SI-engines, heterogeneous DI-diesel combustion, as well as diesel- and gasoline-HCCI combustion processes without any specific adjustments of empirical calibration factors. The accuracy of this enhanced heat transfer equation was carefully verified by evaluation of intensive in-cylinder pressure and fast-response surface temperature measurements.

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INTRODUCTION

The thermodynamic analysis of the in-cylinder processes, as well as the real working cycle process calculation,

typically used today within the scope of 1D simulations of gas-exchange processes, are effective tools for the application and development of modern combustion processes. Wall heat

losses make up one of the greatest losses in this process. Within the real working cycle calculation, the gas-side heat transfer between the working gas and the combustion chamber wall is considered. In a fast-running combustion engine the heat transfer is dominated by forced convection, and is usually described by the Newton approach.

$$\dot{Q}(\varphi) = \dot{q}(\varphi) \cdot A = \alpha \cdot A \cdot (T_g(\varphi) - T_w) \quad (1)$$

A prerequisite for correctly determining the transient, spatially averaged wall heat losses is a heat transfer equation, which enables the quantitatively correct calculation of the wall heat rate in terms of the energy balance, taking into account the corresponding combustion process. The objective of all models for calculating the gas-side wall heat transfer is the determination of the transient wall heat transfer coefficient α as a function of the crank angle φ . Empirical or phenomenological approaches are based on the Newton approach according to [equation \(1\)](#) and represent the most common models. While the heat transfer coefficient was initially calculated from dimensioned, empirical models, e.g. according to Nusselt [3], Eichelberg [4] and Pflaum [5], the dimensionless approaches based on the similarity theory became state of the art. The advantage of dimensionless models is that they can be applied to 'similar' engine systems. Based on the Re-Nu-correlation, the following generally-valid relationship is used for determining the heat transfer coefficient when forced convection is assumed to be the dominating heat transfer effect:

$$\alpha = C \cdot \lambda \cdot d^{(m-1)} \cdot \left(\frac{w \cdot p}{\eta \cdot R \cdot T_g} \right)^m \quad (2)$$

Besides the dimensionless constant C and the choice of the exponent m , the individual models basically differ in their definition of the characteristic length d and the characteristic velocity w . Furthermore, the gas composition and its temperature influence on the thermal conductivity λ and the dynamic viscosity η are weighted differently. The modeling of the combustion-generated convection - or generally: heat-transfer-related effects due to combustion - is done in almost all models by using a more or less empirical correlation with the combustion-induced pressure change. For diesel engines, the calculation approach according to Woschni, which was developed in 1965 [6] and modified in 1970 [7], as well as the heat transfer equation according to Hohenberg, published in 1980 [8], are both particularly widespread, while the Woschni-equation also became an international standard for spark-ignition engines. Various inadequacies in the Woschni-equation led to many corrections and extensions over the years, of which only a few actually developed to form the standard. Despite these well-known weaknesses, the original form of 1970 is still used in most applications. Bargende introduced a heat transfer equation for spark-ignition engines,

also based on the Re-Nu correlation, in 1991 [1,2]. This heat transfer equation has been extensively verified at different research centers with very good agreement compared to measurement results and has therefore become a widely used heat transfer equation, both for spark-ignition engines as well as for diesel engines [9].

In recent years some approaches have been proposed for alternative combustion modes [10,11]. These approaches are based on the Woschni-equation and tuned to fit the wall heat losses in gasoline-HCCI combustion mode.

Due to the continuous and new development of the various combustion principles, this paper presents an enhanced calculation approach based on the equation according to Bargende in order to improve the general validity for determining the transient wall heat losses. Using this new approach, it is possible to calculate wall heat losses without tuning any constants for different combustion principles.

ORIGINAL FORM OF THE HEAT TRANSFER EQUATION ACCORDING TO BARGENDE

Based on the general approach (2), the existing heat transfer equation according to Bargende is given by [equation \(3\)](#):

$$\alpha = C^* \cdot V^{-0.073} \cdot T_m^{-0.477} \cdot p^{0.78} \cdot w^{0.78} \cdot A$$

$$\text{for : } C^* = 27.97 \cdot 10^3 \cdot \frac{1.15 \cdot r + 2.02}{[R \cdot (2.57 \cdot r + 3.55)]^{0.78}} \quad (3)$$

Since the heat transfer occurs in the boundary layer between the working gas and the combustion chamber wall, a mass-weighted mean temperature made up of the spatially averaged gas temperature T_g and the wall temperature T_w is introduced as a heat-transfer-related temperature T_m . This mean temperature T_m is used for calculating the temperature-dependent material constants λ , η as well as for the density ρ . Furthermore, the dependence of the material constants on the current gas composition is taken into account by the air content r ($0 \leq r \leq 1$). Considering the change in the combustion chamber geometry during a working cycle, the characteristic length is represented by the actual combustion chamber volume V according to Hohenberg [8].

In order to describe the constructive conditions of the combustion chamber properly, Bargende considers the transient flow field in the characteristic velocity w . This includes even the influence of the quench flow which exists in trough-shaped combustion chambers.

The heat-transfer-related velocity w for the transient flow field arises due to the turbulent kinetic energy k from a simplified, global k - ε model, as well as the actual piston speed c :

$$w = \frac{1}{2} \cdot \sqrt{\frac{8}{3} \cdot k + c^2} \quad (4)$$

See [Appendix](#) for definitions of individual terms.

As opposed to Woschni, where the influence of the combustion is considered in the flow term, Bargende introduced a multiplicative combustion term Δ .

$$\Delta = (A + B)^2 \quad (5)$$

This combustion term includes the different driving temperature differences between the burned ($T_b - T_w$) and unburned ($T_{ub} - T_w$) fractions of the working gas and the related combustion chamber wall fraction.

$$A = X \cdot \frac{T_b}{T_g} \cdot \frac{T_b - T_w}{T_g - T_w} \quad (6)$$

$$B = (1 - X) \cdot \frac{T_{ub}}{T_g} \cdot \frac{T_{ub} - T_w}{T_g - T_w} \quad (7)$$

The mass fraction burned X is given by:

$$X = \frac{Q_b}{Q_{b, total}} \quad (8)$$

[Equation \(6\)](#) and [\(7\)](#) contain a simplified relationship for the combustion chamber wall fraction, which is already defined as burned fraction:

$$\frac{A_b}{A} = \frac{V_b}{V} \quad (9)$$

The temperature of the unburned mass fraction T_{ub} is calculated in a simplified way by means of polytropic compression starting from the point of ignition (IP) - or the beginning of injection in DI-diesel engines - according to [equation \(10\)](#). The temperature of the burned mass fraction T_b is determined using a 2-zone model. To simplify, it is assumed that the gas constants in the burned and unburned fractions are identical [\(11\)](#):

$$T_{ub} = T_{g, IP} \cdot \left(\frac{p}{p_{IP}} \right)^{\frac{n-1}{n}} \quad (10)$$

$$T_b = \frac{1}{X} \cdot T_g + \frac{X-1}{X} \cdot T_{ub} \quad (11)$$

This formulation of the combustion-generated convection includes an empirical squaring of this analytical term which has been proven to be sufficient for heat-releasing processes close to TDC. However, when the heat release is moved towards expansion stroke, with the consequence of a reduced combustion related pressure increase, an increasing underestimation of the combustion-generated convection can be observed. This underestimation occurs by principle in all heat transfer equations, in which the combustion-induced pressure change is used as a measure of the combustion-generated convection (e.g. Woschni and Hohenberg).

Furthermore, the existing approach describes the heat-transfer-related surface area A , used in [equation \(1\)](#). In general, this surface area is determined by addition of the surface areas of the cylinder head $A_{Cyl.Head}$, piston A_{Piston} , the actual surface area of the cylinder liner A_{Liner} and the surface of the piston top land surface area A_{PTL} multiplied by a specified factor:

$$A = A_{Cyl.Head} + A_{Piston} + A_{Liner} + 0.25 \cdot A_{PTL} \quad (12)$$

In addition to the actual combustion chamber surface area $A_{Comb.Chamber} = A_{Cyl.Head} + A_{Piston} + A_{Liner}$ (whereby any influences which increase the surface area, such as trough-shaped piston bowls or roof combustion chambers, are to be taken into account), the surface area of the piston top land A_{PTL} was first time taken into account by Hohenberg [\[8\]](#). This surface area is calculated by:

$$A_{PTL} = 2 \cdot \pi \cdot d \cdot h_{PTL} \quad (13)$$

Hohenberg could experimentally detect a heat transfer in the piston top land gap which was only about one third of the heat flux in the combustion chamber. The reason for this phenomenon is the lower temperature and gas velocity (25 - 40%) in this area. The reduced wall heat transfer can be easily adjusted by an empirical correction of the relevant surface area. For this reason, only 30% of the piston top land surface area is taken into consideration [\[8\]](#). Bargende adopted this consideration from Hohenberg, but adjusted the factor to 0.25 [\[1,2\]](#).

ANALYTIC DESCRIPTION OF THE HEAT TRANSFER IN THE PISTON TOP LAND

To describe the wall heat losses in the piston top land area, an analytical approach including the description of the influence of a possibly present leakage should be introduced. According to [equation \(1\)](#) and [\(12\)](#), this approach is based on the assumption:

$$\frac{dQ_{W,PTL}}{d\phi} = 0.25 \cdot A_{PTL} \cdot \alpha \cdot (T_g - T_w) \quad (14)$$

The material interaction of the combustion chamber and piston top land gap can be described by the exchange of enthalpy. In reality, the piston top land gap also interacts with the crankcase due to blow-by losses. Also, the piston top land gap thermally interacts with the adjacent surfaces of the piston and cylinder liner. Figure 1 shows the 'piston top land' system and its interaction with its environment. Due to the very large surface/volume ratio in the piston top land [12], a thermodynamic equilibrium is established very quickly between the gas temperature in the piston top land volume and the temperature of the piston top land boundary on the piston and cylinder lining sides, T_{Piston} and T_{Liner} [10]. Thus, as a simplification, an isothermal state in the piston top land volume can be assumed. This leads to a temperature T_{PTL} in the piston top land calculated by an average temperature of T_{Piston} and T_{Liner} .

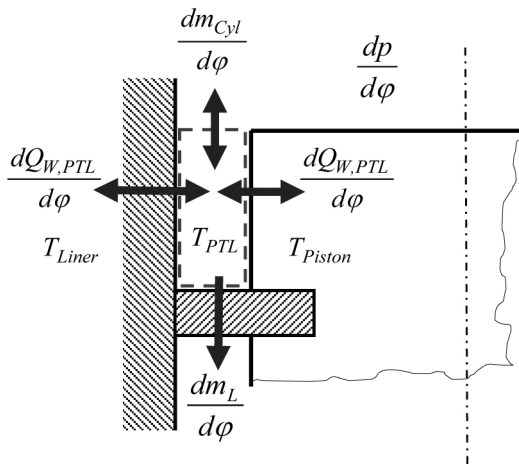


Figure 1. Interaction in the piston top land

Together with the assumption of a constant piston top land volume, i.e. neglecting effects due to the position change of the piston, the wall heat loss in the piston top land area can be determined analytically, as Bargende notified the author of [13] and later assumed in [10]. Here, these approaches should be supplemented with an analytical consideration of the leakage.

The 1st law of thermodynamics in differential form for the piston top land, taking leakage into account, results in:

$$\frac{dU_{PTL}}{d\phi} = -p \frac{dV_{PTL}}{d\phi} + \frac{dQ_{W,PTL}}{d\phi} + \frac{dQ_b}{d\phi} + \frac{dm_{Cyl}}{d\phi} \cdot h + \frac{dm_L}{d\phi} \cdot h_L \quad (15)$$

For the isothermal and isochoric state without heat release in the piston top land, equation (15) is simplified to:

$$-\frac{dQ_{W,PTL}}{d\phi} = \frac{dm_{Cyl}}{d\phi} \cdot h + \frac{dm_L}{d\phi} \cdot h_L - \frac{dm_{PTL}}{d\phi} \cdot u_{PTL} \quad (16)$$

The mass balance in the piston top land can be written as:

$$\frac{dm_{PTL}}{d\phi} = \frac{dm_{Cyl}}{d\phi} + \frac{dm_L}{d\phi} \quad (17)$$

As well as the ideal gas law:

$$p \cdot V_{PTL} = m_{PTL} \cdot R \cdot T_{PTL} \quad (18)$$

According to [13] the isothermal and isochoric states, the ideal gas law can be expressed as:

$$\frac{dm_{PTL}}{d\phi} = \frac{V_{PTL}}{T_{PTL} \cdot R} \cdot \frac{dp}{d\phi} \quad (19)$$

With equation (17) and $h_L = h_{PTL}$ (for $p_{Cyl} > p_{Crank}$) equation (16) results in:

$$-\frac{dQ_{W,PTL}}{d\phi} = \left(\frac{dm_{PTL}}{d\phi} - \frac{dm_L}{d\phi} \right) \cdot h + \frac{dm_L}{d\phi} \cdot h_{PTL} - \frac{dm_{PTL}}{d\phi} \cdot u_{PTL} \quad (20)$$

This can be written as:

$$-\frac{dQ_{W,PTL}}{d\phi} = \frac{dm_{PTL}}{d\phi} \cdot (h - u_{PTL}) + \frac{dm_L}{d\phi} \cdot (h_{PTL} - h) \quad (21)$$

When equation (19) is inserted in (21), this results in:

$$-\frac{dQ_{W,PTL}}{d\phi} = \frac{V_{PTL}}{T_{PTL} \cdot R} \cdot \frac{dp}{d\phi} \cdot (h - u_{PTL}) - \frac{dm_L}{d\phi} \cdot (h - h_{PTL}) \quad (22)$$

Without leakage, there is an enthalpy flow into the piston top land gap in the high-pressure phase until the peak pressure in the combustion chamber is reached ($\phi_{IVC} \leq \phi < \phi_{Pmax}$). In this case, the specific enthalpy h corresponds to the specific enthalpy of the combustion chamber h_{Cyl} , which can be calculated using the spatially averaged gas temperature in the combustion chamber. By definition, an isothermal state in the piston top land is reached due to the wall heat rate. Considering leakage, a blow-by mass flow results depending on the different pressure conditions between the combustion chamber (or piston top land volume) and the crankcase. Thus, the flow transition no longer occurs when the peak pressure p_{max} is reached, but even later with increasing blow-by flow, as shown in Figure 3 and Figure 4. After the flow transition,

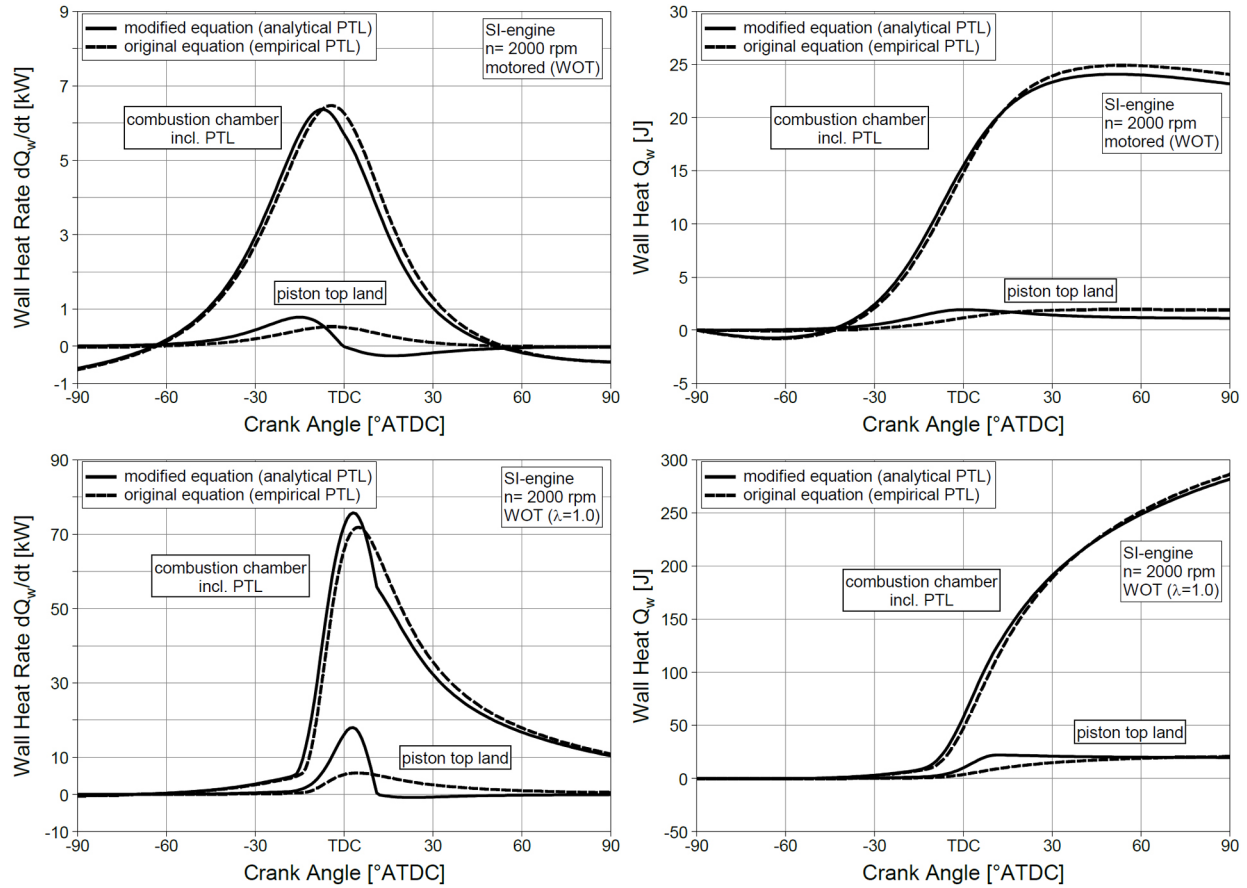


Figure 2. Wall heat transfer without leakage

there is an enthalpy flow until the end of the high-pressure phase out of the piston top land volume back into the combustion chamber, reduced by the blow-by mass flow which is still flowing into the crankcase. In this case the specific enthalpy h is equivalent to the specific enthalpy of the piston top land h_{PTL} , which is calculated using the mean wall temperature of the piston and cylinder liner in the piston top land.

$$\frac{dm_{Cyl}}{d\phi} = \frac{V_{PTL}}{T_{PTL} \cdot R} \cdot \frac{dp}{d\phi} - \frac{dm_L}{d\phi} \begin{cases} > 0: & h = h_{Cyl} \\ & h_{PTL} \end{cases} \leq 0: \quad (23)$$

For $dm_{Cyl}/d\phi > 0$, equation (24) applies for the wall heat rate in the piston top land:

$$\frac{dQ_{W,PTL}}{d\phi} = \frac{dm_L}{d\phi} \cdot (h_{Cyl} - h_{PTL}) - \frac{V_{PTL}}{T_{PTL} \cdot R} \cdot \frac{dp}{d\phi} \cdot (h_{Cyl} - u_{PTL}) \quad (24)$$

For $dm_{Cyl}/d\phi \leq 0$, equation (25) applies for the wall heat rate in the piston top land:

$$\frac{dQ_{W,PTL}}{d\phi} = -V_{PTL} \cdot \frac{dp}{d\phi} \quad (25)$$

The total wall heat rate can be calculated using equations (24) and (25):

$$\frac{dQ_W}{d\phi} = \alpha \cdot A_{Comb,Chamber} \cdot (T_g - T_w) + \frac{dQ_{W,PTL}}{d\phi} \quad (26)$$

Figure 2 shows the result of this analytical calculation method for application in motored operation mode and SI-combustion, assuming an ideally sealed combustion chamber, i.e. $m_L = 0$. The wall heat rate out of the combustion chamber is identical for both approaches by definition and thus not shown here. The total wall heat rate results as the sum of the wall heat rates from the piston top land and combustion chamber according to equation (26).

Both in motored operation mode, as well as with SI-combustion, the curve of the wall heat rate from the piston top land according to the analytic method and the original empirical description do not agree. As a matter of fact, in the analytical procedure the direction of the heat rate changes when flow transition is reached. The empirical description, where only a factor is assigned to the combustion chamber wall heat rate, cannot have this change in direction by principle.

As already determined experimentally by Hohenberg [8], the wall heat loss in the piston top land in motored operation