

Novel Transient Wall Heat Transfer Approach for the Start-up of SI Engines with Gasoline Direct Injection

2010-01-1270

Published
04/12/2010

David Lejsek and André Kulzer
Robert Bosch GmbH

Günter Hohenberg
VKM, Technische Universität Darmstadt

Michael Bargende
IVK, Universität Stuttgart

Copyright © 2010 SAE International

ABSTRACT

The introduction of CO₂-reduction technologies like Start-Stop or the Hybrid-Powertrain and the future emissions limits require a detailed optimization of the engine start-up. The combustion concept development as well as the calibration of the ECU makes an explicit thermodynamic analysis of the combustion process during the start-up necessary. Initially, the well-known thermodynamic analysis of in-cylinder pressure at stationary condition was transmitted to the highly non-stationary engine start-up. There, the current models for calculation of the transient wall heat fluxes were found to be misleading. But with a fraction of nearly 45 % of the burned fuel energy, the wall heat is very important for the calculation of energy balance and for the combustion process analysis. Based on the measurements of transient wall heat transfer densities during the start-up presented in a former work [1], the paper describes the development of adaptations to the known correlations by Woschni [2], Hohenberg [3] and Bargende [4] for the application during engine start-up. To demonstrate the high accuracy of the introduced equations, the results of the cyclic resolved thermodynamic analysis using the presented novel approaches are compared with the results of the measurements. It is shown, that the usage of the novel heat flux models for the engine start-up process make the cyclic resolved thermodynamic analysis good enough to allow an efficient optimization of the engine start-up.

INTRODUCTION

An in-depth look at vehicle emissions shows that the exhaust gas emitted during cold start and in the subsequent warm-up period ([Figure 1](#)) plays a decisive role. For this reason, measures to improve start-up performance have a direct effect on the overall result across the driving cycle. The more stringent limits being placed on emissions and the simultaneously increasing cost pressure - also in exhaust gas after-treatment - implies that improving the start-up process in particular has become extremely important. Also the new CO₂-reduction concepts like Start-Stop-Systems or the introduction of hybrid vehicles cause an increase in the number of engine start-up's during the test cycles. A detailed analysis of the processes in the combustion chamber during start-up is advantageous in understanding the development of the combustion process and also for the data calibration of the engine management system.

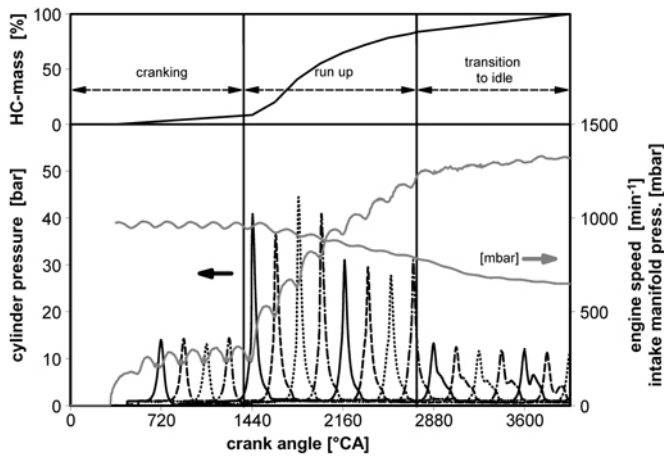


Figure 1. Engine start-up of a 4-cylinder GDI engine

For this, a cycle-by-cycle thermodynamic analysis of the engine start-up is needed. Due to the high speed gradient, the thermodynamic analysis must be calculated on a time basis [5]. Therefore, the description of an internal combustion engine system shown in Figure 2 requires a transient examination over time. Under start-up conditions, in the following derivation all terms refer to temporal changes.

$$d\varphi = 6 \cdot n \cdot d\tau \quad (1)$$

Since during measurement of the start process, both the engine speed n and the length of time $d\tau$ per degree of crankshaft angle φ are recorded, the conversion from the temporal basis to the piston position is possible at any time using equation (1). The thermodynamic analysis of the indicated cylinder pressure investigates the burn rate, i.e., the time gradient for the heat release. Applying the first law of thermodynamics one gets the energy balance (equation 2) for the closed system “combustion chamber”.

$$\frac{dU}{dt} = \frac{dQ_b}{dt} - p \cdot \frac{dV}{dt} + \frac{dQ_w}{dt} + \frac{dH}{dt} \quad (2)$$

Within this, the burn rate and the heat losses through the cylinder walls account for the heat release rate in equation (3). Calculation of the burn rate (equation 4) is the sum of internal energy, work, heat losses through the walls and the enthalpy gradient to be added.

$$\frac{dQ_h}{dt} = \frac{dQ_b}{dt} + \frac{dQ_w}{dt} = \frac{dU}{dt} - \frac{dH}{dt} + p \cdot \frac{dV}{dt} \quad (3)$$

$$\frac{dQ_b}{dt} = \frac{dU}{dt} - \frac{dH}{dt} + p \cdot \frac{dV}{dt} - \frac{dQ_w}{dt} \quad (4)$$

The enthalpy gradient is calculated from the enthalpy of the injected fuel and the leakage which must be taken into consideration (equation 5). For the injected fuel, the entire specific enthalpy is given by the fuel enthalpy h_f minus its evaporation enthalpy Δh_{fv} , and the energy needed to heat up the vaporized fuel mass from a characteristic vaporization end temperature $T_{v,end}$ to the temperature level T of the cylinder charge during the injection.

$$\frac{dH}{dt} = \frac{dm_{f,v}}{dt} \cdot (h_f - \Delta h_{fv} - c_{p,f} \cdot (T - T_{v,end})) + h_{lk} \cdot \frac{dm_{lk}}{dt} \quad (5)$$

Hence, with the change in internal energy expressed in equation (6), the burn rate is given by equation (7).

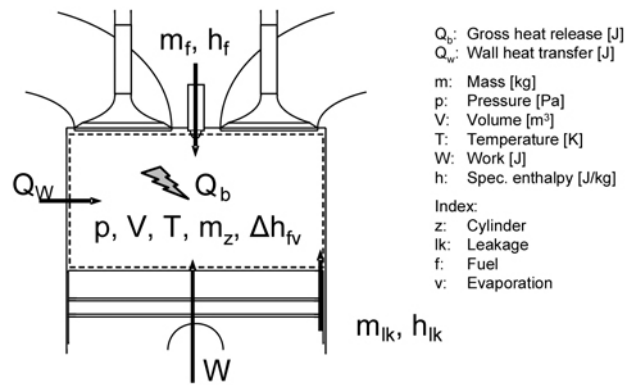


Figure 2. Combustion chamber as a thermodynamic system

The temperature gradient is calculated from the gas equation, considering also the losses due to leakage.

$$\frac{dU}{dt} = m \cdot c_v \cdot \frac{dT}{dt} + u \cdot \left(\frac{dm_{lk}}{dt} + \frac{dm_f}{dt} \right) \quad (6)$$

The analysis during the combustion is modeled by two zones, unburnt and burnt. They differ in temperature and composition of the gas. Using this, the thermodynamic data will be determined. The temperature in the unburnt zone T_{ub} is calculated by equation (8), which describes a polytropic compression of the unburnt zone during the combustion.

$$\frac{dQ_b}{dt} = m \cdot c_v \cdot \frac{dT}{dt} - R \cdot T \cdot \frac{dm_{ik}}{dt} - \frac{dm_{f,v}}{dt} \cdot (\dot{h}_f - \Delta h_{fv} - c_{p,f} \cdot (T - T_{v,end}) - u) + \rho \cdot \frac{dV}{dt} - \frac{dQ_w}{dt} \quad (7)$$

The burnt zone temperature T_b is calculated using the volume balance in the combustion chamber [equation \(9\)](#). Herein, for the unburnt volume determination the temperature T_{ub} is used. Before and after the combustion, T_{ub} and T_b are equal to the mass averaged temperature T .

$$T_{ub} = T_0 \cdot \left(\frac{\rho}{\rho_0} \right)^{\frac{n-1}{n}} \quad (8)$$

The thermodynamic properties for the vaporized fuel, unburnt and burnt gas mixture are determined using the approach by Grill [\[6\]](#).

$$V_b = V - \frac{(m \cdot R \cdot T)_{ub}}{p}$$

$$T_b = \frac{p \cdot V_b}{(m \cdot R)_b} \quad (9)$$

As described in detail in [\[7\]](#), cylinder leakage is calculated by means of an effective cross-sectional leakage gap using the flow rate equation of an isentropic, frictionless flow. However, this simplified modeling does not take into account the increased leakage which occurs predominantly in the change of disposition of the piston rings. For this reason, an empirical curve from Rao [\[8\]](#) is used to describe the dependency of the leakage gap on the engine speed. With this the accuracy in determining the higher leakage losses at low engine speeds increases.

Today models for calculating the wall heat transfer in combustion engines have neither been developed nor validated for use at very low speeds. The thermodynamic analysis of engine start-up's using the wall heat transfer models by Woschni [\[2\]](#), Hohenberg [\[3\]](#) and Bargende [\[4\]](#) discussed in [\[1\]](#), proves that the above conclusion doesn't lead to desired results.

Hence, for a more accurate thermodynamic analysis and for an efficient application of the engine start-up, new measurements of the transient wall heat flux and a new approach describing the transient heat transfer during the engine start-up are necessary.

MEASURING DURING ENGINE START-UP

Initially, an engine was chosen, which would fulfill the following requirements on a modern gasoline combustion system:

- Gasoline Direct Injection of the 2nd generation, i.e. a central position of the injector in cylinder head.
- Compression Ratio of $\varepsilon \approx 10$ considering the turbo-charged configuration of future downsized engine concepts.

Because of the very good accessibility and short setting-up time a single cylinder test bed assembly in [Figure 3](#) was chosen for the wall heat transfer investigations. The technical data of the used engine are listed in [Table 1](#). Since the high piston speed gradients during the start-up result in very high transmission effort of the measured values, only the local heat flux to the cylinder head wall can be measured reliably. Therefore the heat flux probes for the transient measurement of the wall heat transfer are mounted only within the cylinder head surface of the combustion chamber. Investigations on the local wall heat flux profiles in a constant volume chamber presented in a former work [\[1\]](#) show nearly the same wall heat flux level for the cylinder head, liner and the piston. Because of the same geometry used, obtaining a global heat flux from the measured data of the cylinder head is feasible for the thermodynamic analysis issues.

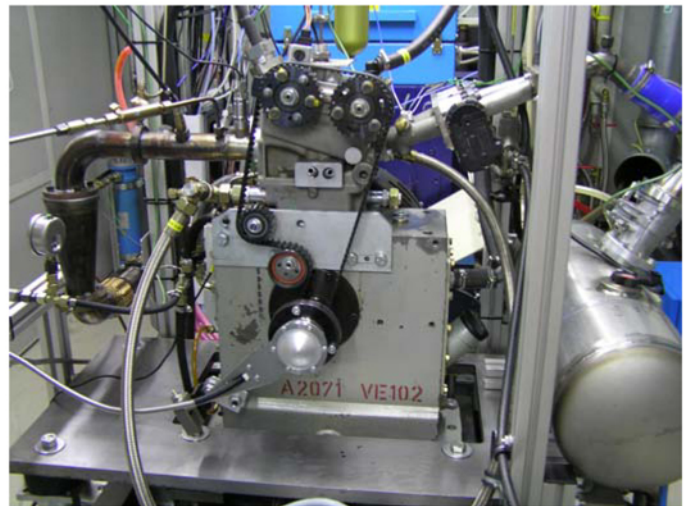


Figure 3. Single cylinder GDI research engine with centrally-mounted injector

Table 1. Engine Data

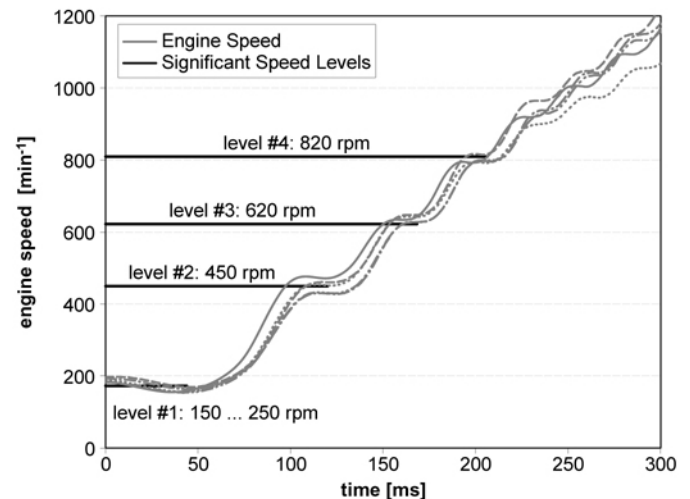
Single Cylinder GDI-Engine

Stroke [mm]	85.0
Bore [mm]	82.0
Compression Ratio [-]	10.5
Displacement [cm ³]	449
Connecting Rod [mm]	145
Reduced Inertia [kgm ²]	1.6
Fuel	EuroSuper 95 RON
Injection System	Bosch Direct
Injector	Bosch HDEV 5.1
Valve timing (@ 2 mm valve lift):	
EVO [degCA bBDC]	25
EVC [degCA bTDC]	29
IVO [degCA aTDC]	26
IVC [degCA aBDC]	18

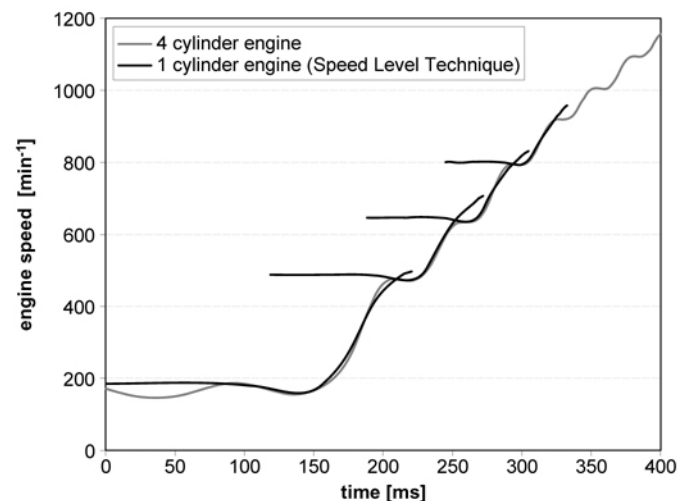
The single-cylinder engine in four-stroke operation is not able to produce the same start-up behavior as a 4- or 6-cylinder engine. Therefore, a new method was developed to condition a single-cylinder engine for the start-up behavior of a multi-cylinder engine [1].

SPEED LEVEL TECHNIQUE

Starting of a multi-cylinder engine results in characteristic speed levels, which are reached after each single combustion during the run-up's. Due to the full-load operation and similar inertia and friction behavior, these characteristic speed levels are reached by any engine with identical number of cylinders and nearly the same displacement. Furthermore, each of them depends on the engine temperature and cranking speed at start-up. Any of these speed levels can be easily set on a single-cylinder test bed by means of a dynamometer. [Figure 4a](#) shows some start-up's of a 4-cylinder engine including significant speed levels reached. After the engine is motored to a significant speed level, the fuel is injected and the ignition induces the combustion.



(a)



(b)

Figure 4. Starting of 4-cylinder engine: Significant speed levels and the comparison with speed gradients demonstrated on the single cylinder research engine using the “Speed-Level-Technique”

However, to reach nearly the same start-up speed gradients of multiple-cylinder engines ([Figure 4b](#)), the single-cylinder engine is further motored by the test bed-dynamometer during the combustion process. The reason for this is the very high inertia of the single-cylinder assembly needed to eliminate resonance vibration at low test bed speeds. The injection and ignition timings, intake air and injected fuel pressure as well as the engine temperature can be controlled by the test bed equipment to match the parameters of multi-cylinder engines. This “Speed-Level-Technique” allows the demonstration of any start-up combustion cycle on a single-cylinder test bed and makes detailed investigations on the

combustion and their thermodynamic analysis during the engine start-up possible [1].

To carry out a thermodynamic analysis the single-cylinder engine test bed was equipped with:

- A conditioned AVL Piezo-electrical pressure transducer for the in-cylinder pressure indication.
- Kistler Piezo-resistive sensors for the intake and exhaust port pressure measurement.
- Injected fuel pressure was measured by a Kistler piezo-resistive high pressure sensor.
- The injection and ignition timings were determined by adapted current probes.
- Temperatures of engine oil and cooling water are registered by thermocouples.

The crank angle position is derived from an AVL angle encoder with a resolution of 0.25 °CA. The instantaneous engine speed can be obtained from this signal by transient recording and usage of equation (1). The angle encoder also provides one TTL-signal per revolution for detecting the exact piston position.

4 heat flux probes within the cylinder head surface were implemented to detect the local wall heat flux. Figure 5 shows the mounting position of this ALTP probes.

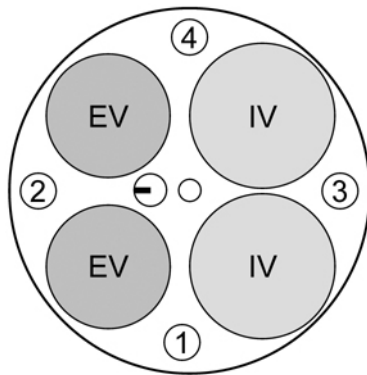


Figure 5. Position of the 4 ALTP probes within the cylinder head surface for local wall heat flux measurement

All the measured values are stored by a DeweTron transient recorder with a transient resolution of 10 μs (100 kHz) per channel. This allows a time based thermodynamic analysis of the combustion process as described above.

HEAT FLUX MEASUREMENT

In this work, a novel heat flux density probe is used. This Atomic-Layer-Thermopile (ALTP) was presented in [9] and

is not a novel invention of this paper. The working principle of the ALTP is based on a thermoelectric field created by the transverse Seebeck effect within the active sensor layer. Figure 6 shows a schematic cross section of an active ALTP layer.

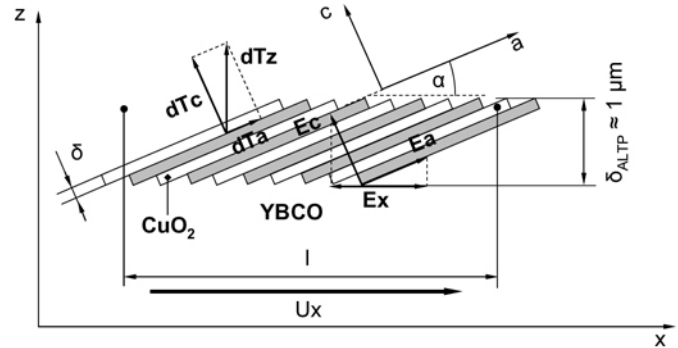


Figure 6. Schematic cross section of an active ALTP layer [9]

$$\vec{E} = \vec{S} \cdot \nabla \vec{T} \quad (10)$$

The thermopile consists of an YBCO film with intermediate CuO₂ layers. The crystalline structure of this material causes strong anisotropic thermoelectric properties. The films are grown by laser ablation on SrTiO₃ substrates. The substrates can be cut at a certain angle to its crystallographic plane. Therefore, the YBCO film grows with a certain tilt angle α between the c-axis of the crystal and the surface normal z of the film.

$$U_x = E_x \cdot l \quad (11)$$

Usage of the common definition of thermo electrical voltage (equation 10) leads to an expression for the thermo electrical voltage along the x-axis U_x (equation 11). Introduction of the Seebeck-coefficients S_a and S_c for the a- and c-axis together with the distance between the contact points l yields an expression for U_x depending on the temperature gradient along the surface normal z , dT_z in equation (12).

$$U_x = \frac{1}{2} \cdot \sin 2\alpha \cdot (S_a - S_c) \cdot \frac{\partial T_z}{\partial z} \cdot l \quad (12)$$

Hence, the heat flux density induced by the temperature gradient dT_z (equation 13) can be expressed with equation (12).

$$\dot{q}_w = \frac{\dot{Q}_w}{A} = \frac{\lambda}{s} \cdot dT_z = \lambda \cdot \frac{\partial T_z}{\partial z} \quad (13)$$

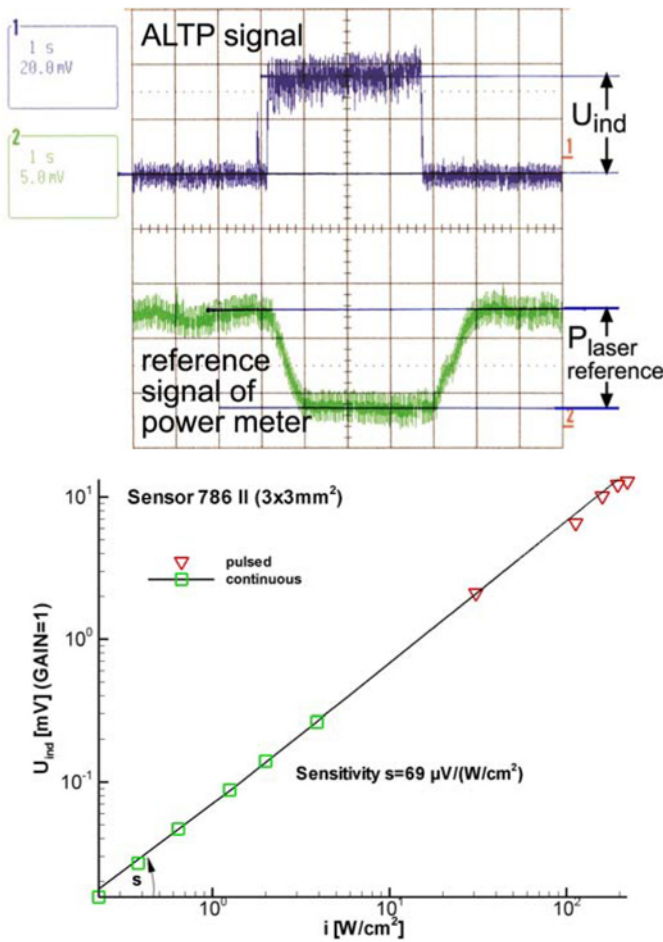


Figure 7. ALTP-probe calibration [9]

In equation (14), F_{ALTP} describes the geometry of the active layer and its thermo electrical properties. This factor must be obtained by calibrating every single probe. Investigations on ALTP sensors in [9] show a constancy of F_{ALTP} in the range of about $1 \dots 10^3$ of the measured heat flux density.

$$\dot{q}_w = \frac{2 \cdot \lambda}{\sin 2\alpha \cdot (S_a - S_c) \cdot l} \cdot U_x = F_{ALTP} \cdot U_x \quad (14)$$

Calculating the mean wall heat flux for use in equation (7), an appropriate partition of unburnt and burnt heat flux is needed.

$$\gamma = \frac{A_b}{A_{W,system}} \quad (15)$$

According to the combustion chamber division in unburnt and burnt parts, the burnt wall surface fraction γ (equation 15) is used to determine the mean global heat flux. For this, γ was obtained by using CAD data of the single-cylinder engine assuming hemispherical flame propagation during the combustion. Hence, the mean wall heat flux is given by equation (16)

$$\frac{dQ_w}{dt} = (1 - \gamma) \cdot \frac{dQ_{w,ub}}{dt} + \gamma \cdot \frac{dQ_{w,b}}{dt} \quad (16)$$

where $\frac{dQ_{w,ub}}{dt}$ and $\frac{dQ_{w,b}}{dt}$ are provided by the measurement.

For the calibration of the used probes extensive investigations were carried out in [9] to ensure the measurement accuracy. An additional study on a constant volume chamber was made in [1] to verify the novel ALTP heat flux density probes used in this work at exact defined combustion conditions. The results of the thermodynamic analysis shown in [1] verify the measured data and also confirm the feasibility of the global heat flux determination using the equations (15) and (16).

WALL HEAT MODEL ADAPTATION FOR ENGINE START-UP

The thermodynamic investigations on the transient wall heat transfer during the engine start-up in [1] using the models by Woschni [2], Hohenberg [3] and Bargende [4] show the potential to adapt these models to start-up conditions. However, the development of a completely new description of the wall heat transfer phenomena is not necessary, since:

- The mentioned models describe the gas properties very good. For the application on engine start-up, the effect of pressure and temperature on these properties has to be the same as in any other engine conditions.
- An important issue for modeling the wall heat loss at run-up conditions is the characterization of the heat transfer relevant gas speed, i.e. the relation of piston speed and combustion effects, within the boundary layer. Known models already specify this influence by different terms of piston speed and combustion propagation. Therefore an adaption of these equations to the engine start-up conditions should lead to satisfying results.

The presented investigations in [1] set an extensive data base including

- characteristic start-up measurements regarding to 4- and 6-cylinder engine run ups