

Investigations on the Transient Wall Heat Transfer at Start-Up for SI Engines with Gasoline Direct Injection

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ABSTRACT

The introduction of CO₂-reduction technologies like Start-Stop or the Hybrid-Powertrain and the future emissions regulations require a detailed optimization of the engine start-up. The combustion concept development as well as the calibration of the ECU makes it necessary to carry out an explicit thermodynamic analysis of the combustion process during the start-up. As of today, the well-known thermodynamic analysis using in-cylinder pressure traces at stationary condition is transmitted to the highly dynamic engine start-up. Due to this approximation the current models for calculation of the transient wall heat fluxes by Woschni, Hohenberg and Bargende do not lead to desired results. But with a fraction of approximately 40 % of the burnt fuel energy, the wall heat is very important for the calculation of energy balance and for the combustion process analysis during start-up. The paper shows how the transient heat fluxes to the combustion chamber wall can be measured during the engine start-up. Needing an offset correction which can not be carried out properly at engine start-up conditions, the indirect heat flux measuring method can not be used. Due to the high engine speed gradients, a novel heat flux sensor is used for the time-resolved heat flux density measurements. The working principle of this Atomic-Layer-Thermopile sensor is based on a thermoelectric field created by the transverse Seebeck effect. Because of the sensors high frequency response, highly time-resolved local heat flux density measurements up to 1 MHz range can be realized. Results of investigations at homogenous and stratified engine start-up in a DISI engine are presented and the influence of the measured wall heat flux on the cylinder pressure analysis is explained. It is obvious that only a

novel heat flux model for the engine start-up process makes the cyclic resolved thermodynamic analysis good enough to allow an efficient optimization of the engine start-up. The shown experiments help understand the physical effects and set an extensive data base for modeling the wall heat transfer during 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 plays a decisive role. Depending on the engine type and the design of the exhaust gas after-treatment system, the emissions from these two phases can constitute up to 90 % of the total HC-emissions emitted during a NECD test [1]. Especially during start-up, almost all the untreated emissions can be seen as certification relevant, because the catalytic converter is still cold. 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.

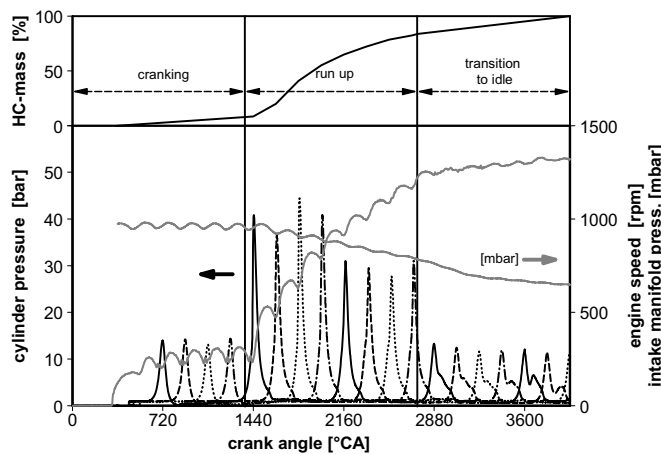


Figure 1 – Engine start-up of a 4-cylinder GDI engine

The start-up process in a conventional engine shown in Figure 1 can be divided into three phases independent of the combustion concept. Meshing and synchronization during the engine cranking, engine run-up and the transition to idling phase. The first combustion is easily recognizable from the cylinder pressure, the intake manifold pressure and the engine speed. Also clearly evident is the drop in cylinder pressure - caused by the late ignition angle at the end of the run-up phase – which is introduced in order to avoid an overshoot in engine speed during start-up. An examination of the cumulative hydrocarbon mass (Figure 1, top) shows that a considerable proportion of the unburnt hydrocarbons emitted comes from the first work cycles. This is due to the unfavorable conditions for mixture preparation in the start-up process. The main factors here are the reduced charge movement in the cylinder due to the low engine speed, the fact that the cylinder walls are still cold and the system-dependent not optimal injection pressure at start-up. These factors lead to the deposition of a significant part of the mixture as a film on the cylinder walls. Since this is either expelled as unburnt hydrocarbons or burnt once the cylinder wall heats up, a direct time-dependency of the emissions to combustion is made more difficult.

For this reason, the state-of-the-art analysis of a global consideration of the emitted hydrocarbon mass (total engine emission of hydrocarbons during start-up) for optimizing the ignition and injection calibration is misleading. Combined with the growing number of parameters provided by the gasoline direct injection system, individual application settings for each combustion during engine start-up become necessary. Therefore a cycle-by-cycle thermodynamic analysis during the start-up is needed. Due to the high speed gradient, the thermodynamic analysis must be calculated on a time basis [2]. The standard description based on position (degree of crankshaft angle) is unreliable for start-up. Because of variable engine speed during start-up, the elapsed time per degree of crankshaft angle changes with different engine speed patterns. Thus, a crank angle based treatment leads to differences in results because of the strong individuality

(speed gradient) of each start-up process. A comparison of the heat release rate based on position would only be justified if the increase in engine speed was identical in each case, something which hardly occurs in practice.

THERMODYNAMIC ANALYSIS DURING ENGINE START-UP

As described above, the thermodynamic description of an internal combustion engine shown in Figure 2, under start-up conditions, requires a transient examination over time. Therefore, 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 a temporal to a positional basis using equation (1) is possible at any time. 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) then requires the variation in 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 minus its evaporation enthalpy and the leakage which must be taken into consideration (equation 5).

$$\frac{dH}{dt} = \frac{dm_f}{dt} \cdot (h_f - \Delta h_{fv}) + h_{lk} \cdot \frac{dm_{lk}}{dt} \quad (5)$$

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

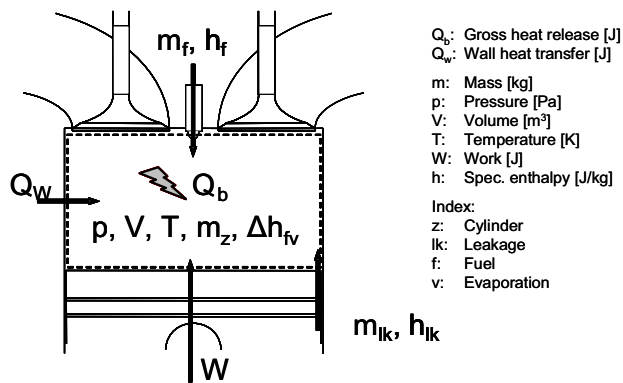


Figure 2 – Combustion chamber as a thermodynamic system

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. According to this, the thermodynamic data and the wall heat flux are determined. The temperature in the unburnt zone T_{ub} is calculated according to equation (8), which describes a polytropic compression of the unburnt zone during the combustion.

$$\frac{dQ_b}{dt} = m \cdot c_v \cdot \frac{dT}{dt} - R \cdot T \cdot \frac{dm_{lk}}{dt} - \frac{dm_f}{dt} \cdot (h_f - \Delta h_{fv} - u) + p \cdot \frac{dV}{dt} - \frac{dQ_w}{dt} \quad (7)$$

The burnt zone temperature T_b is determined using a novel empirical approach shown in equation 9, resulting from investigations on flame propagation in a constant volume chamber [3]. With this, a temperature offset to the mass averaged temperature T is calculated considering the combustion process by the normalized burn heat X and the mixture air-fuel ratio λ . Further, an engine-specific factor A describes the temperature level in the burning zone. For engines up to 500 cm³ displacement per cylinder, good results can be reached with $A \approx 850$ K. Before and after the combustion, T_{ub} and T_b are equal to the mass averaged temperature T .

$$T_{ub} = T_0 \cdot \left(\frac{p}{p_0} \right)^{\frac{n-1}{n}} \quad (8)$$

The caloric data for real gases are determined from the tables by DeJaegher [4], thermodynamic properties for

the vaporized fuel are taken from Pischinger [5]. Further, the thermodynamic properties for the unburnt and burnt mixture can be derived using the approach by Grill [6].

$$T_b = (A^* \cdot (1 - X) + X \cdot A) \cdot (1 - X) + T \quad (9)$$

$$A^* = (0.9 - 2.26 \cdot \lambda + 1.9 \cdot \lambda^2 - 0.5 \cdot \lambda^3)^{-1}$$

As described in detail in [2], cylinder leakage is calculated by means of an effective leakage gap cross-sectional as per 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 [7] 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.

In context of the development of the 'direct start' [2], the burn rates determined from the cylinder pressure using the wall heat transfer models of Woschni, Hohenberg and Bargende, were analyzed for their plausibility at zero engine revolution and at low engine speeds. As discussed in [2], all named models were found not to lead to the desired results. Further thermodynamic analysis of engine start-up's proves this conclusion. As of today models for calculating the wall heat transfer in combustion engines have neither been developed nor validated for use at very low speeds.

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 model describing the 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 $\epsilon \approx 10$ considering the turbo-charged configuration of future downsized engine concepts.

Because of the high complexity of the heat transfer phenomena, the measurements were divided into two phases.

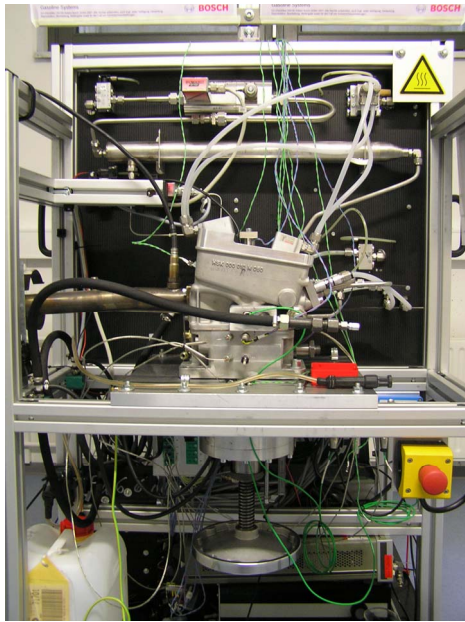


Figure 3 – Assembly of the constant volume chamber

In Phase-I, measurements on a constant volume chamber were made. At first, it was very important to verify the novel ALTP heat flux density probes [8] used in this work at exact defined conditions. These can be realized only in a stationary system. Second reason for the investigations on the combustion chamber was the elimination of inner charge flow, introduced in a cranking engine by the intake process and piston movement. In the constant volume chamber any charge flow is caused by the combustion itself. This in turn is relevant for modeling the heat flux to the walls. The used assembly is shown in Figure 3. The main features of the chamber are:

- Main geometry based on an research single-cylinder engine:
 - Single-cylinder engine head
 - Bore: 82 mm
 - Stroke (max): 85 mm
- Piston position variable and fixable.
- Operation with premixed Methane-Air-Mixture to provide homogenous charge with variable air-fuel-ratio.
- Intake and exhaust valves are pneumatically operated.
- Temperature of the chamber walls can be conditioned by an external heating and cooling system.
- The test procedure is full automated by LabView-based software.

The combustion chamber assembly allows the introduction of equipment for the transient measurement of all values needed for the thermodynamic analysis of the system. There are two pressure transducers installed, one in cylinder head to measure the

combustion pressure and the second one in the piston behind the ring-package for determining the leakage mass flow during the combustion process. Six thermocouples are mounted to verify the homogeneity of the desired wall temperature setting. 15 ALTP wall heat density probes are fastened in the chamber to detect the local heat flux to the wall.

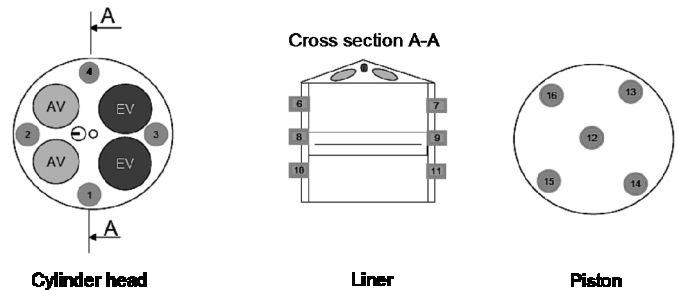


Figure 4 – Position of the ALTP wall heat density probes within the constant volume chamber

As shown in Figure 4, the steady-state measurements were performed using 5 probes on piston surface, 6 on the cylinder liner and 4 probes within the cylinder head surface. The measured values are stored by a transient recorder with a time resolution of 10 μ s (100 kHz) per channel.

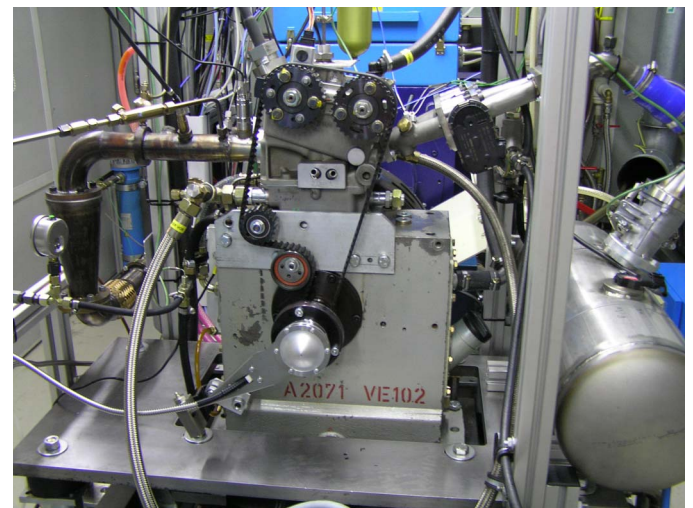


Figure 5 - Single cylinder GDI research engine with centrally-mounted injector

During Phase-II, tests were conducted with dynamic operation. Figure 5 shows a single cylinder research engine using the same geometry as that of the constant volume combustion chamber. The technical data of the used engine are presented 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. However, the positions of ALTP heat flux probes within the cylinder head surface are identical to the constant volume combustion chamber.

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.

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, this characteristically 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 this speed levels can be easily set on a single-cylinder testbed by means of a dynamometer. Figure 6 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.

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 Injection
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

However, to reach nearly the same start-up speed gradients of multiple-cylinder engines, 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.

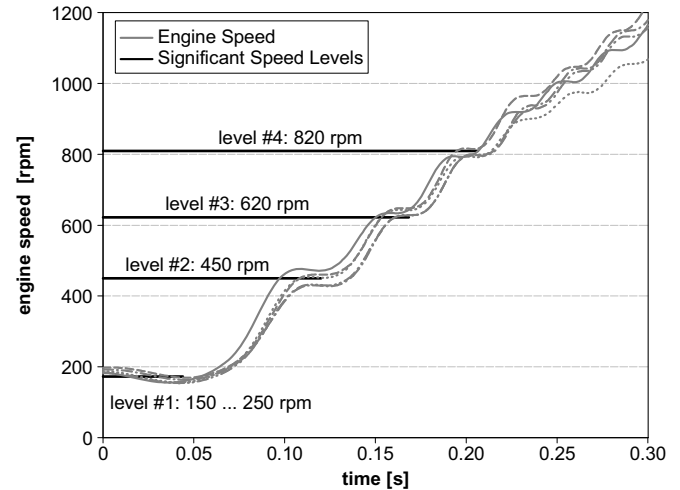


Figure 6 – Starting of 4-cylinder engine: Significant speed levels

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 is measured by a Kistler piezo-resistive high pressure sensor.
- The injection and ignition timings are measured by adapted current probes.
- Temperatures of engine oil and cooling water are registered by thermocouples.
- The crank angle position is derived from a AVL angle encoder with the resolution of 0.25 °CA. The instantaneous engine speed can be obtained from this signal by transient recording. 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 are implemented to detect the local wall heat flux.

All the measured values are stored by a DeweTron transient recorder with a time 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

There are two established methods for measuring the wall heat flux in combustion engines. The indirect measurement method provides a temperature equivalent signal of the wall surface. This wall temperature oscillation can be converted by the Fourier-Transformation into the heat flux. This method needs an offset correction of the calculated heat flux. In

combustion engines, this offset can be obtained by assuming an adiabatic state during the compression stroke. For this, the wall temperature has to be higher than the intake-air. However, during cold start of engines this condition can not be fulfilled.

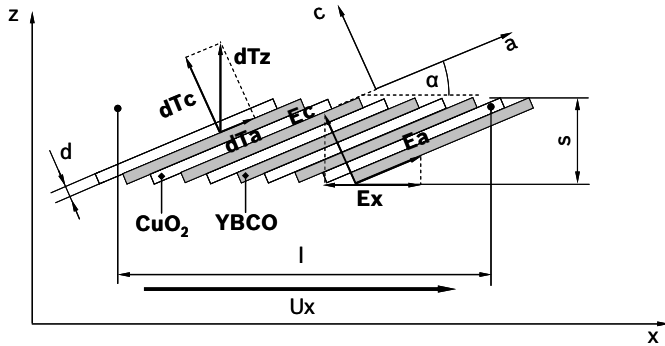


Figure 7 – Schematic cross section of the active ALTP layer [8]

For this reason, the indirect method is not suitable to be used for thermodynamic analysis during engine start-up. Direct methods of measuring provide an equivalent signal to the wall heat flux. In this work, a novel heat flux density probe of this category is used. This Atomic-Layer-Thermopile (ALTP) was presented in [8] 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 7 shows a schematic cross section of an active ALTP layer.

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

The thermopile consists of CuO_2 layers and intermediate YBCO layers. The films are grown by laser ablation on SrTiO_3 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.

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

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 number of active layers along the distance l , n , yields an expression for U_x depending on the temperature gradient along the surface normal z , dT_z in equation (12).

$$U_x = (S_a - S_c) \cdot \frac{l}{d} \cdot \sin \alpha \cdot dT_z \quad (12)$$

Hence, the heat flux density induced by the temperature gradient dT_z (equation 13) can be expressed with equation (14).

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

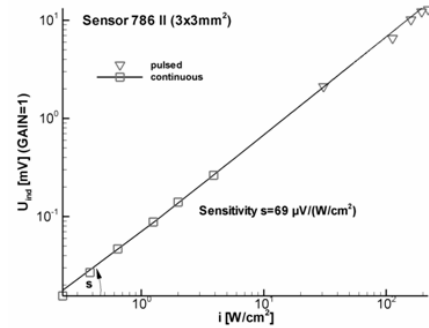
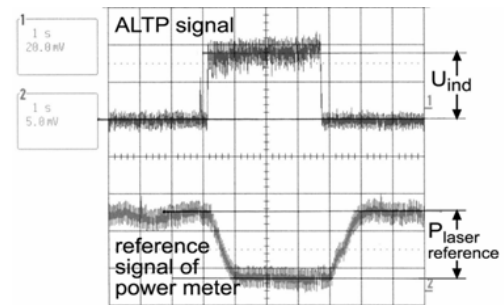


Figure 8 – ALTP-probe calibration [8]

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 sensor. Investigations on ALTP sensors in [8] show a constancy of F_{ALTP} in the range of about $10^1 - 10^3$ of the measured heat flux density.

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

In this work, the calibration was made by means of a CO_2 -laser, which induced predefined heat flux density to the sensor. Figure 8 shows an example of the ALTP calibration. Further information on the ALTP-sensor and his calibration can be found in [8].

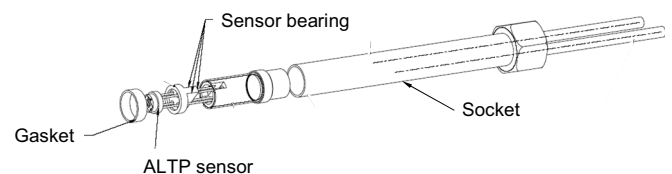


Figure 9 – Assembly of the ALTP probe