Characterization of Deposits at Water Wall Panels of Steam Generators by Using Heat Flux Measurement

Paper # 27

Sebastian Grahl, Michael Beckmann, Gabriele Magel, Wolfgang Spiegel

Sebastian Grahl, Michael Beckmann, Technical University Dresden, Institute of Power Engineering, Chair for Combustion, Heat and Mass Transfer, 01062 Dresden – Germany;

e-mail: sebastian.grahl@tu-dresden.de

Gabriele Magel, Wolfgang Spiegel, CheMin GmbH, Am Mittleren Moos 46A, 86167 Augsburg – Germany

1 ABSTRACT

The combustion of biomass and heat-recoverable waste products often leads to corrosive deposits at the evaporator finned tube walls, which subsequently cause material damage, greater exhaust gas losses or unacceptably high material stress. For prevention the deposits are being cleaned regularly, largely mechanical, during the plant shutdowns. To reduce the maintenance and service efforts and thus increase the economic efficiency, plant shutdowns for cleaning and removal of corrosion damages should be minimized.

Due to the obstruction of heat extraction caused by deposits as well as the large temperature gradient between the finned tube panel surface and the exhaust gas on one side and the appearance of a deposit material with corrosive, temperature-dependent properties on the other side, there exists a qualitative relationship between formation and structure of deposits and the heat flux hitting the evaporator wall. The latter can therefore be used to characterize deposits at finned tube walls of steam generators as well as to determine the point in time of cleaning and to assess the corrosion potential of the deposit.

For this purpose a method was developed that can use the Fourier transform of the heat flux signal to determine the temperature-dependent material properties of the deposits. Currently, the change of heat flux density of a finned tube wall with deposits compared to the clean state is experimentally demonstrated and a further development takes place by means of signal processing to derive information about the deposit situation in practical application. The results are discussed in the following paper.

2 INTRODUCTION

The combustion of biomass and heat-recoverable waste products often leads to particularly corrosive deposits at the evaporator fin tube wall, which subsequently cause material damage, greater exhaust gas losses or unacceptably high material stress in other places. This must be prevented by appropriate measures. For that purpose the deposits are being cleaned regularly, largely mechanical, during the plant shutdowns. To reduce the maintenance and service efforts and thus increase the economic efficiency, plant shutdowns for cleaning and removal of corrosion damage should be minimised.

Due to the obstruction of the heat extraction caused by deposits as well as the large temperature gradient between the fin tube panel surface and the exhaust gas on one side and the appearance of a deposit material with corrosive, temperature-dependent properties on the other one, there exists a qualitative relationship between formation and structure of deposits and the heat flow hitting the evaporator wall. The latter can therefore be used firstly to characterise deposits at the fin tube walls of steam generators as well as to determine the point in time of cleaning and the other to assess the corrosion potential of the deposit.

Under the assumption of known temperature fluctuation amplitude, which can be determined from the temperature profile of the boiler using Fourier transformation, heat flux density is solely dependent on the deposit properties of density, specific heat capacity and thermal conductivity as well as the heat transfer coefficient. If the surface temperature of the deposit layer is known, the heat transfer coefficient can be determined from the heat flux density. The temperature-dependent material properties density, specific heat capacity and thermal conductivity of the deposit as well as its thickness have an attenuation influence on the applied external temperature fluctuation of the fin. From the signal profile of the heat flux density measurement – the amplitudes and frequency spectrum and from the phase shift at higher temporal resolution of the measured signal as well – it is therefore possible to derive information about the deposit situation.

To determine the heat flux density a non-invasive procedure was developed in earlier works. [1] With that the heat flux density on the membrane wall can be determined over the temperature difference between the soffit and the fin welded between the evaporator tubes.

Currently, the change of heat flux density of a fin tube wall with deposits compared to the clean state is experimentally demonstrated and will then be further developed for practical application.

3 THE PRINCIPLE OF HEAT FLUX MEASUREMENT

Membrane walls consist of parallel boiler tubes in which the fins have to be welded in between. By known construction of the walls it is principally possible to evaluate the heat flux density on the membrane walls from the temperature difference between two characteristic positions. Practically it is worthwhile to consider the section between the positions at the soffit and the middle of the fins. The reason being that at the both ends it is possible to mirror the existing geometry as well as the physical characteristics and hence the applicability of the boundary conditions. In here the fin of the membrane wall behaves like a single-sided heated fin, in which respectively the ends are kept at a constant temperature level by evaporating water inside the boiler tube. Depending on the construction of the wall, that results in a defined distribution of the heat between the fin and the soffit area.

With the use of mathematical models it is possible to evaluate the temperature profile and respectively the distribution of the heat flux density for a defined wall construction. Therefore by means of modeling, a calculation is possible for the entrant heat flux density into the membrane walls through convection and radiation.

Practically the heat flux density can be evaluated in dependency to the temperature difference between the middle of the fin and the soffit at the outer side of evaporator walls and thus offers the possibility of a non-invasive measurement.

Principally the written method from Krüger [2] for heat flux density measurement was validated for practical use and can be applied to all membrane walls. This is only under the condition that the geometrical conditions and the heat conductivity of the used material for the wall construction are known.

4 INFLUENCE OF THE DEPOSITS ON THE HEAT FLUX DENSITY

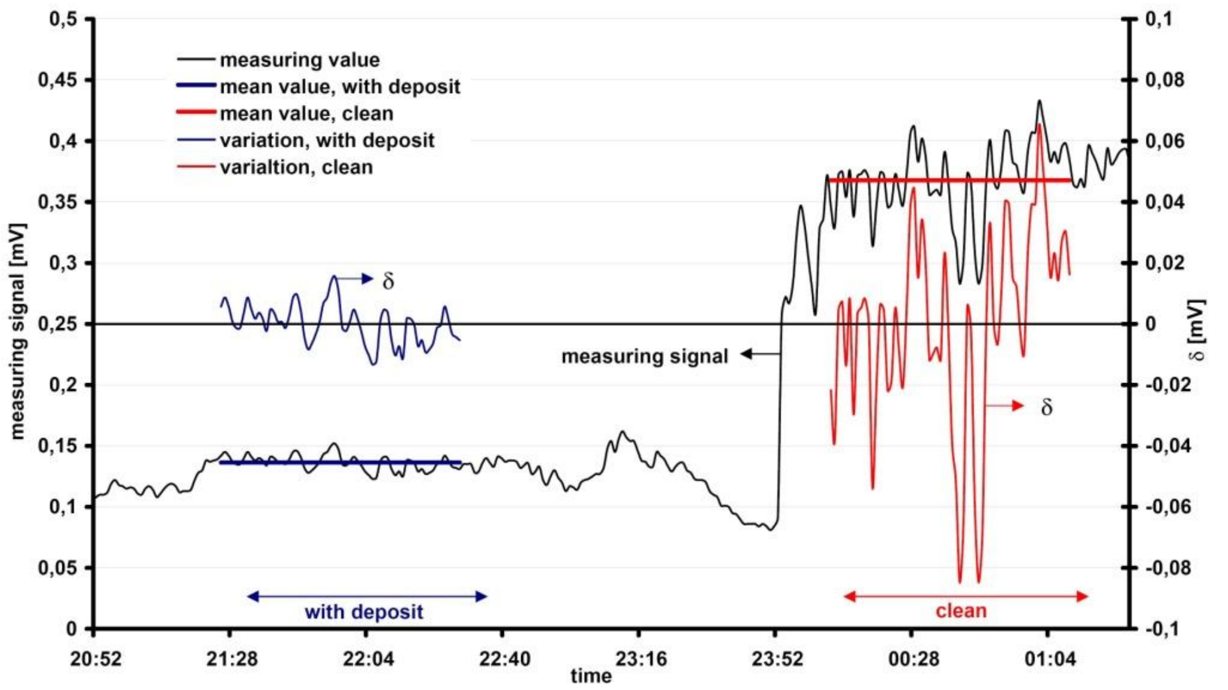
The gas in the combustion chamber of a steam power plant emits heat continuously into the ambient boiler walls, such that the water in the tube walls is brought to boiling point. The heat transport occurs due to convection and radiation. With the constant build up of deposition layers at the boiler walls there is variation of the parameters for the heat transport during operation. In one way there is normally an increase of the emissivity on the wall surface as long as there is a deposition layer and in the other way it results in increasing thermal resistivity. For any increase in the deposition layer, consequently there is a reduction in the transmitted heat flux density and generally the surface temperature increases. Large heat transfer resistance and smaller temperature differences between the combustion chamber and the surface temperature of the deposition – result in a decrease in the dissipated heat flux density from the gas to the boiling water.

The influence of the deposition on the heat flux density was quantified by Krüger in [2]. In the work, the relationship was shown between the heat flux density for a wall construction with deposits and for a clean tube wall to the layer thickness, and the dependency to the thermal conductivity of the deposition layer.

5 RESULTS FROM FIELD STUDIES

Depending on the measuring signal of the thermocouple voltage in millivolts shown in Figure 1, the temperature difference between fin and soffit of the membrane wall is measured. With that signal, the heat flux density $\dot{q}$ is almost linearly related (see also [2]).

Figure 1: The influence of the deposition on the measuring signal dynamics.



From the signal progression of the heat flux density, it is possible to derive information about the layer deposition situation as shown in Figure 1. Specifically that is from the amplitude and frequency spectrum as well as at higher temporal resolution, also from the phase shift of the measuring signal. That is because, from the fluctuating temperature of the gas, there is also a resultant fluctuating heat flux density. On the membrane wall, the damping of the amplitude as well as the displacement of the signal of the temperature is dependent on the material properties (heat conductivity λ, density ρ and specific heat capacity c) also the thickness of the deposition layer δ_D. At the left side of Figure 1 it is possible to recognize the isolation and heat storage effect of a deposition layer through the comparable low heat flux density and the large signal damping. Contrarily as for the right side in Figure 1 for clean conditions – the measuring signal reaches a comparable higher level and damping of the amplitude is smaller.

Figure 2: The influence of the deposition layer on the measuring signal dynamics – Fourier analysis.

a. with deposit b. clean surface

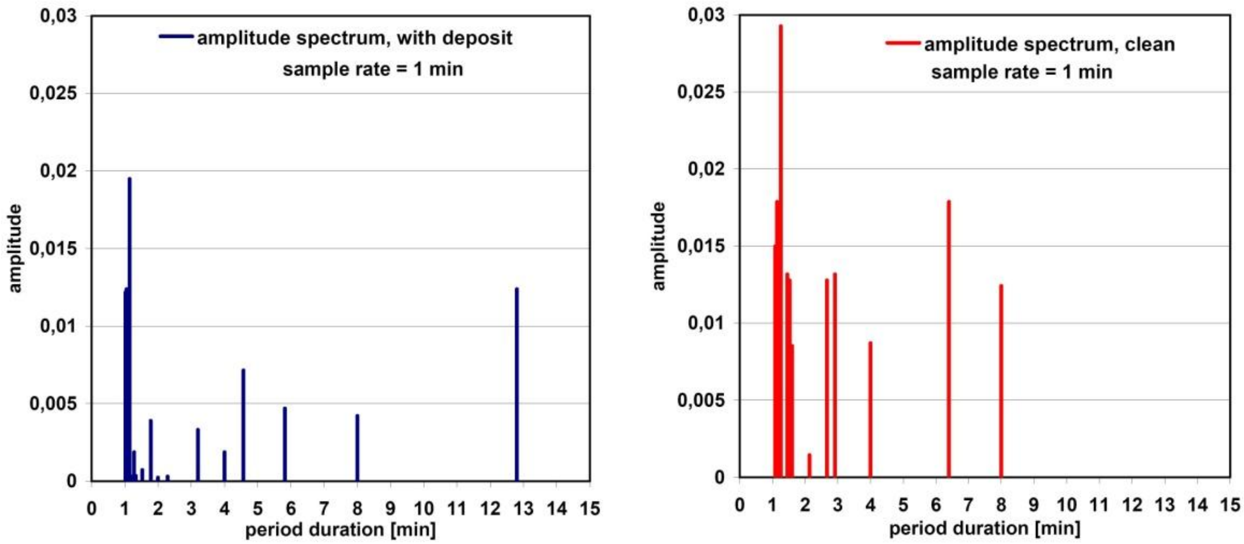


Figure 2 shows the amplitude spectrum of a discrete Fourier analysis. The fluctuations from Figure 1 are shown in detail for the deposited (Figure 2a) and the clean (Figure 2b) membrane wall. In contrary to the deposition case (Figure 2a) the analysis of the clean case (Figure 2b) shows a higher number of frequencies (with similar amplitudes) between one and two minutes. From the heat flux density fluctuations, in the clean case (Figure 2b) there is a pronounced fluctuation of the amplitude (also for frequency) in comparison to the deposition case (Figure 2a).

6 STRATEGY FOR DEPOSIT CHARACTERIZATION AND DEPOSIT-SPECIFIC CLEANING

The investigation into the periodic variable temperature is a subject matter of the theoretical considerations e.g. for the investigations into the temperature fluctuations in the earth or for the heat transfer from the combustion gas into the cylinder wall in engines [3]. These basic approaches can principally be used for the membrane wall even then when the fluctuation of the temperature is aperiodic, whereby normally Laplace Transformation is used and respectively the Fourier transformation.

Generally it is valid initially for the periodic, time dependent heat input in a system, that the temperature of the adjacent fluids, see equation (1), on the surface of a solid and with it the transferred heat flux density changes according to the periodic temporary law.

$$\vartheta_F = \vartheta_m + \Delta\vartheta \cdot \cos(\omega \cdot \tau) \tag{1}$$

At this juncture ϑ_m denotes a constant median temperature, $\Delta\vartheta$ the temperature fluctuation amplitude, ω the angular frequency of the fluctuation, τ the period (from an arbitrary point t_0 until to arbitrary point t). The cosine function can be transformed into a sine function at similar conditions through a phase shift of $\pi/2$ and therefore for further calculation steps only the cosine function can be considered.

If one assumes that the deposition layer is a single sided infinite long solid, at homogeneous and isotropic conditions, then for the temperature field of the deposit layer the Fourier differential equation (2) with the location coordinate x and the thermal diffusivity a is applicable.

$$\frac{\partial\vartheta}{\partial t} = a\frac{\partial^2\vartheta}{\partial x^2} \tag{2}$$

The assumption that a single-sided infinitely long solid at small deposition layers can be made, since the temperature fluctuations rapidly occur and the boiling water temperature in the evaporator tubes can be considered as a constant value.

This consideration occurs in close reference to the literature from [3] and [4], in which the statement refers to examples of single sided infinite elongated solids. For these examples the solid always has an infinitely large thermal resistance as result of the significance of the heat conductivity and thus the temperature fluctuations are completely damped inside the solid. This leads to the fact that there is no heat transferred through the solid. In the presented case however the heat from the combustion chamber is transferred to evaporating boiling water into the boiler tubes, while for the evaporation of water (phase transition of first order) an infinitely large specific heat capacity exists. Thereby the same effect occurs as for an infinitely elongated solid, in which the temperature fluctuations until the surface of the boiling water are almost completely damped. The difference between the both cases is the temperature gradient for the median temperature ϑ_m in the wall construction from which the transferred median heat flux can be calculated. When the gradient of the temperature is zero, there is no heat transported through the solid as in the initial case. Equation (1) has therefore to be extended to include the dependency of the median temperature ϑ_m to the location state of the solid and the median transferred heat flux density $\dot{q}_m$ on the boiling water. As an approach equation (2) is used, however the left side of the equation is zero when a quasi-stationary state exists, in which the median temperatures at the entry and exit of the solid do not change over time.

The solution of the equation for the temperature field after calculation steps eventually results to a shortened form:

$$\vartheta(x,\tau)_{\tau\to\infty} = \vartheta_{m,D} - \frac{x}{\lambda}\dot{q}_m + \frac{\Delta\vartheta \cdot exp(-\xi)}{\sqrt{1+2\beta+2\beta^2}} cos(\omega\cdot\tau-\varepsilon-\xi). \quad (3)$$

In that $\vartheta_{m,D}$ is the middle temperature of the deposition surface and that yields

$$\varepsilon = arctan\left(\frac{\beta}{1+\beta}\right) \quad (4)$$

with

$$\beta = \frac{\lambda}{\alpha}\sqrt{\frac{\omega}{2a}} = \frac{b}{\alpha}\sqrt{\frac{\omega}{2}} \quad (5)$$

and

$$\xi = x\sqrt{\frac{\omega}{2a}} = \frac{x}{\lambda}b\sqrt{\frac{\omega}{2}}. \quad (6)$$

The modification of the equation (5) includes the thermal effusivity coefficient

$$b = \sqrt{\lambda\cdot\rho\cdot c}\,. \quad (7)$$

The progression of the temperature field is eventually dependent on the median temperature in the combustion chamber, the temperature fluctuation of the amplitude, the period duration, the heat transfer coefficient α and the material properties heat conductivity λ, density ρ as well as the specific heat capacity c of the deposition layer, which can be summarized from the thermal diffusivity a. If it is possible to evaluate the thermal conductivity from the temperature and heat flux density fluctuations, then a statement can be made about the characteristics of the deposit. In the next step there are at least two of the three material properties to determine that are included in the thermal diffusivity, such that a more accurate statement can be made about the type of the deposition.

For the investigation of the deposition layer the surface temperature is particularly important. When in equation (3) $x = 0$, that reduces to:

$$\vartheta_0 = \vartheta_{m,D} + \frac{\Delta\vartheta}{\sqrt{1+2\beta+2\beta^2}} cos(\omega\cdot\tau-\varepsilon)\,. \quad (8)$$

It therefore results, that the surface temperature similar to the ambient temperatures shows a harmonic fluctuation. Its period duration has the same value as that for the fluctuation of the surrounding temperature. The surface temperature follows thereby the ambient temperature with a time delay, i.e. a phase shift of the value ε. Furthermore there occurs a damping of the fluctuation with a value of $\beta > 1$. For the total temperature field and with a growing penetration depth x i.e. the value ξ, an additionally damping of the fluctuations with a phase shift is generated. However the frequency of the temperature fluctuations on the wall is similar to that one of the surroundings. Independent of the influence of the range x, the heat transfer coefficient α effects on the amplitude and the phase shift of the temperature fluctuations. For the border case of an infinitely great value for α this influence disappears. Practically the heat transfer coefficient α however by far is much lower. With a reducing value (α) there is an increase in the phase shift as well as the damping of the amplitude for the temperature fluctuations. To better understand the said influences a temporal and location resolved temperature field as an example for a 20 mm thick porous concrete layer as the deposit is shown in Figure 3.

Figure 3: Time- and location-resolved temperature field in a deposition layer.

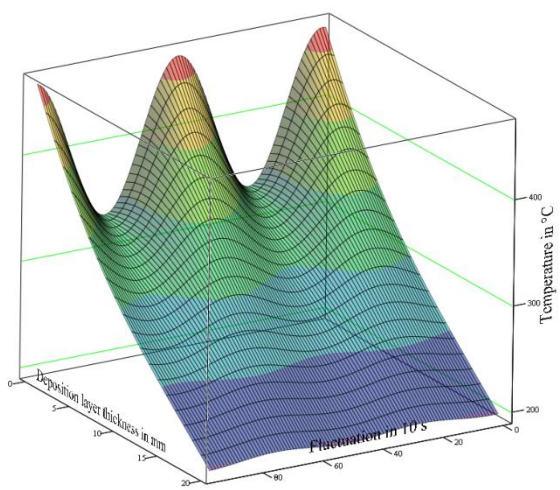
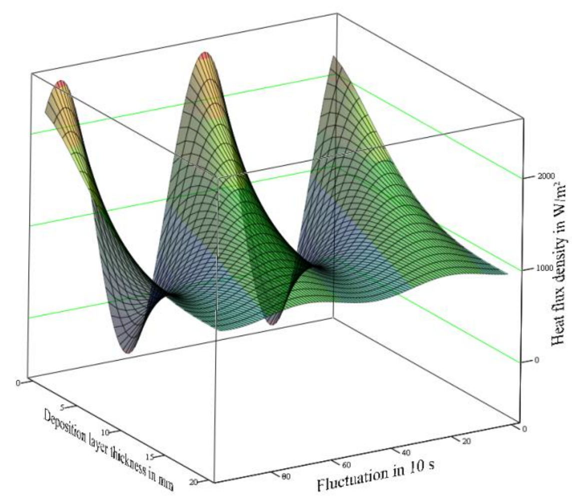


Figure 4: Temporal and local heat flux density progress in a deposition layer.



- Example deposit material: porous concrete
- Deposition layer thickness: 20 mm
- Period duration of the fluctuation: 480s
- Median surface temperature: 400 °C
- Temperature fluctuations amplitude: 80 K
- Median heat flux density: 1000 W/m^2

At the test facility for heat flux density measurements (as similar in practical operations) the heat flux density is determined at the outer side of the evaporator wall and that is in dependency to the temperature difference between the fin and the soffit. Hence it is very important – the dependency of the heat flux density and the fluctuation of the bordering fluid temperature on the deposited membrane wall. For the calculation of the fluctuating heat flux density it is therefore initially assumed that a temperature gradient lies on the surface $(x = 0)$ and thus there is heat conduction at the deposition layer. It therefore results in:

$$\dot{q} = -\lambda \left(\frac{\partial \vartheta}{\partial x}\right)_{x=0} \quad , \tag{9}$$

whereby under the use of equation (3) and eventually a differentiation after x leads to the formulation of the equation (10) for the heat transfer density.

$$\dot{q}(x,\tau) = \dot{q}_m + \frac{\Delta\vartheta \cdot b \cdot \sqrt{\omega} \cdot e^{-\xi}}{\sqrt{1 + 2\beta + 2\beta^2}} cos\left(\omega \cdot \tau - \varepsilon - \xi + \frac{\pi}{4}\right) \tag{10}$$

The progress of temporal and local resolved heat flux density is shown in Figure 4 analogous to the temperature field in Figure 3. A special investigation of the temperature change behavior at the fin through a numerical model and the influence on the heat flux density process has already been investigated by Krüger [2]. Hence the subject would not be discussed further at this point. Practically it is likely that the temporal curve progression of the combustion temperature is neither sinus- nor cosine-shaped i.e. it is a non harmonic fluctuation but corresponds to a different temporal progression. Also here the indicated theory does not lose its validity but rather this special temporal progression of the combustion chamber temperature must be subjected to harmonic analysis. Here it can one way result in a simplification through the superposition of cosine lines and the other way from leap and bounds changes the progression can be resolved by decomposition in harmonics by the principle of Fourier series. [4] The Laplace – Transformation can then simplify the calculation and with the use of electronic computer processing the objective can be reached much faster.

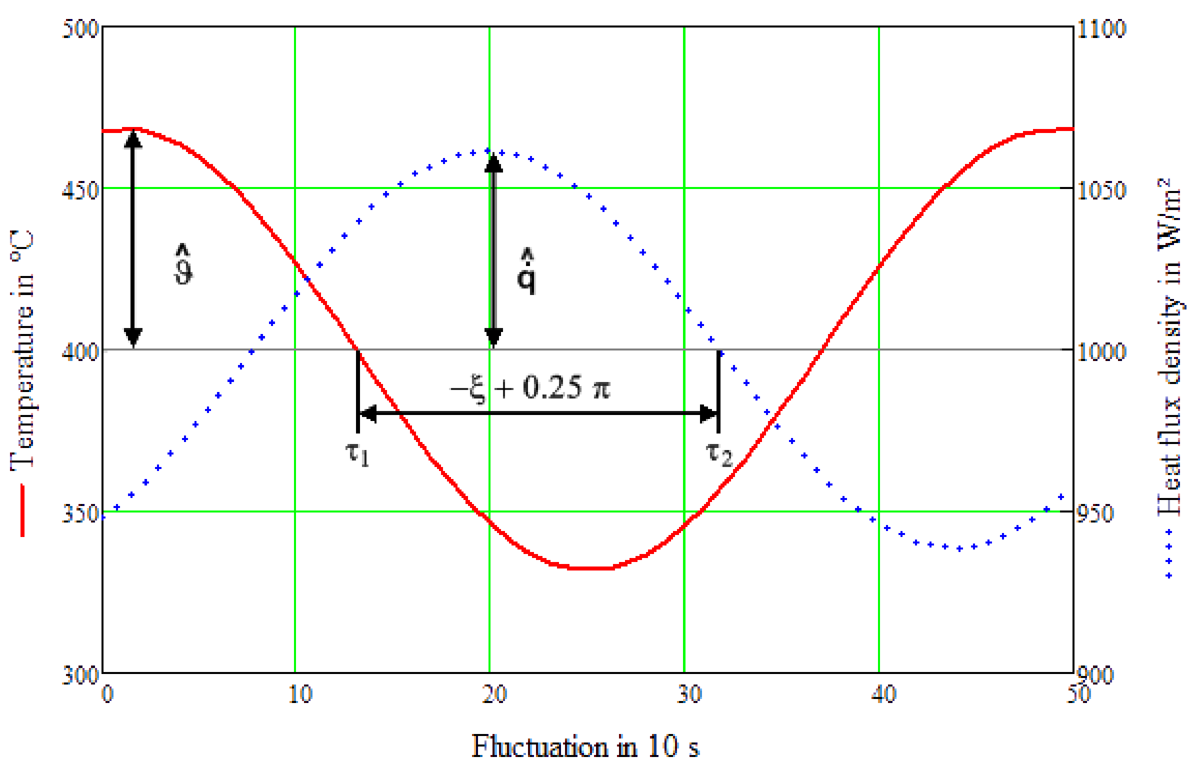
For the experiments at laboratory scale there is measuring signal into the determination of the heat flux density (Figure 5). This signal follows equation (10) and has a phase shift of $-\varepsilon - \xi + 0{,}25\pi$ compared to the temperature fluctuations in the combustion chamber after equation (3). Through the imprinting of temperature fluctuations in the combustion chamber with the aid of secondary air in the burner, a cosines-shape progression can be generated. If during the experiments the surface temperature of the deposit is determined for example with a high-speed infrared camera, then in equation (8) $-\varepsilon$ is inapplicable. This is because from the boundary condition 3 the boundary condition 1 occurs with $\alpha \rightarrow \infty$, i.e. comparison of equation (5) and (6). The phase shift between the surface temperature of the deposit and the heat flux density signal (see

Figure 5) is therefore $-\xi + 0.25\pi$. With ξ the thermal effusivity b from the equation (8) and (10) can be evaluated. The calculation equation is:

$$b = \frac{\hat{q}(\delta_D, \tau_2)}{\hat{\vartheta}(0, \tau_1)} \sqrt{\frac{1}{\omega}} e^{\xi} \tag{11}$$

with the amplitude of the heat flux density $\hat{q}(\delta_D, \tau_2)$ at the transition between the deposit and the membrane wall, the amplitude of the surface temperature of the deposition $\hat{\vartheta}(0, \tau_1)$, the proportionality factor between both amplitudes $\omega^{-0.5}$ as well as the correction of the damping through the deposit between temperature and heat flux fluctuations e^{ξ}.

Figure 5: Phase progression into the determination of phase shift for the heat flux density at the point $\delta_D = 20\ mm$ in comparison with the temperature fluctuation from the example of Figure 3 and 4.



From thermal effusivity b and phase shift ξ the heat conductivity can be iteratively determined in relationship to the layer thickness. In the first step the heat conductivity of the deposit is evaluated from the conversion of the equation (6) with an assumption of the deposition layer δ_D which leads to:

$$\lambda = \frac{\delta_D}{\xi} b \sqrt{\frac{\overline{\omega}}{2}}. \quad (12)$$

In the second step with equation (12) the ascertained heat conductivity λ and also the deposition layer thickness δ_D can be calculated through the mean heat flux density $\dot{q}_m$ through the membrane wall and the median temperature on the deposit layer surface $\vartheta_{m,D}$ as well as between the deposit and the membrane wall construction $\vartheta_{m,P}$ after equation (13). Therefore equation (3) can be rearranged and a quasi-stationary state can be taken, in which the right side of the equation that contains the fluctuations is neglected.

$$\delta_D = \frac{\lambda}{\dot{q}_m} \left(\vartheta_{m,D} - \vartheta_{m,P}\right) \quad (13)$$

The steps 1 and 2 have to be repeated until the evaluated deposition layer thickness from the equation (12) and (13) are identical, and such the heat conductivity λ and the layer thickness of the deposition δ_D can be obtained. After the respective rearrangement of the equation (6) and (7) the thermal diffusivity a and the volumetric capacity $\varrho \cdot c$ can be determined.

7 SUMMARY

For the heat flux density (analogous to temperature field), it shows that there is a dependency to the deposition properties namely density ρ, specific heat capacity c, thermal conductivity λ and the heat transfer coefficient α. That is under the condition that the temperature fluctuation amplitude $\Delta\vartheta$, can be determined from the temperature profile of the combustion chamber through the use of the Fourier transformation.

If the surface temperature of the deposition layer is known, then the heat transfer coefficient α can be determined from the heat flux density. The determination of the surface temperature can for example be done through a timely high resolution infrared camera.

In further steps the heat conductivity λ can be iteratively calculated in relation with the dependency of the deposition layer thickness δ_D. The product of the remaining material properties i.e. specific heat capacity c and density ρ can be calculated from the thermal diffusivity a or the thermal effusivity b.

The attained information can then be used for the online measurements of deposition and aid the plant operator in the optimization of the process, the fixing of the waste gas cycles and the determination of the corrosion behavior of deposits.

8 REFERENCES

[1] **Spiegel, Wolfgang; Beckmann, Michael; Magel, Gabriele and Krüger, Sascha** *Entwicklung eines Messsystems für die Online-Bestimmung des Wärmestromes auf Membranwände von Dampferzeugern in Biomasseverbrennungsanlagen zur Erfassung der Beeinträchtigungen der Anlageneffizienz aufgrund von Korrosion und Verschmutzung.* December 2008. Final Report DBU-Project AZ: 23893-24.

[2] **Krüger, Sascha.** *Wärmestromdichtemessung an Membranwänden von Dampferzeugern.* Neuruppin : TK Verlag Karl Thomé-Kozmiensky, 2009. Dissertation at the Technical University of Dresden.

[3] **Grigull, Ulrich and Sandner, Heinrich.** *Wärmeleitung.* 2nd Edition. Berlin, Heidelberg, New York : Springer-Verlag, 1990. 3-540-52315-4.

[4] **Gröber, Heinrich; Erk, Siegmund and Grigull, Ulrich.** *Die Grundgesetze der Wärmeübertragung.* 3rd Edition. Berlin, Heidelberg, New York : Springer-Verlag, 1988. 3-540-02982-6.