IMPLEMENTATION OF HEAT TRANSFER MONITORING SYSTEMS IN SOLID FUEL FIRED BOILERS

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Abstract. *One of the problems associated with the operation of coal or biomass fired boilers is the slagging and fouling of heat exchange surfaces which changes with fuel quality and hence is difficult to predict. The deposits growth can be analylsed directly using specific probes measurements at critical locations or can be viewed by CCD cameras. A cheaper alternative to the direct observation of deposits is boiler heat transfer monitoring, where plant signals are used to generate a complete thermal analysis of the steam and gas circuits, identifying the heat transfer resistances in individual heat exchangers. The paper presents the application of boiler monitoring heat transfer models to biomass fired boilers. The model is shown to represent the evolution of heat transfer resistance along time in individual heat exchangers, mainly in the superheaters. The model calculates furnace exit gas temperature that is compared with experimental values obtained by plant continuous instrumentation and with measurements performed using a fine thermocouple or acoustic pyrometry. The calculated values are shown to follow the plant data but are larger and in closer agreement with the values measured in specific measuring campaigns. The further development and use of the monitoring and simulation models is discussed in the paper.*

Keywords*. Heat transfer, Boiler, Modeling.*

1. Introduction

The formation of deposits in solid fuel fired boilers is a complex phenomena and the formulation of a model to represent this process is difficult, involving flow, heat transfer and ash chemistry (e.g. Lee and Lockwood, 1999). The characteristics of the deposits change depending on the type of fuel and operating conditions and although some data is available they are difficult to simulate. Valero and Cortés (1996) reviewed the state of art on monitoring heat transfer in boiler walls using specific heat transfer probes. There are two types of probes aiming respectively to the measurement of the incident heat flux and the absorbed heat flux. The incident heat flux depends mainly on gas conditions within the furnace and has a smaller influence from the slag deposits. On the other hand, absorbed heat fluxes are directly influenced by deposits, but they also change due to boiler load, so the interpretation is better if the two values are known. The use of heat flux probes requires an extensive instrumentation of the boiler furnace. The values can only be observed in specific locations and are restricted to boiler walls. Heat flux measurements provide local information so the identification of the slag deposits distribution requires the use of interpretation tools (Afgan et al, 1999). Slagging problems often occur in super-heater tube panels where the installation of probes is difficult because of lack of access. For super heater tube panels, the use of CCD cameras in cooled probes is a possibility (Fontes, 2003), although the interpretation of the images is a difficult task.

Valero and Cortés (1996) identified an alternative procedure for the identification of heat transfer degradation in heat exchanger surfaces in a boiler, based on calculations of the gas temperature in the boiler, from the boiler exit flue gas temperature. These methods have been applied using dedicated sets of thermocouples mounted in a grid but they can also be applied using the boiler exit gas temperature that is usually monitored in all boilers. The backwards gas temperature calculations are based on plant data, thermodynamics and heat transfer. The heat transfer monitoring models provide a resolution of the boiler similar to the resolution in the plant instrumentation. Therefore the calculated values for the heat transfer characteristics are average values to a set of heat exchanger including eventually several tube banks or several water tube panels.

Instituto Superior Técnico (IST) developed a monitoring model for a Portuguese coal fired boiler in Sines power plant (Azevedo, 1998). Heat transfer correlations for the main heat exchangers were derived from plant data corresponding to the higher values for the global heat transfer coefficient. The differences between convection heat transfer coefficients derived from the plant data and correlations from literature were within 15%, which are of the same order as the accuracy of the heat transfer correlations. Using the heat transfer correlations from literature, sometimes leads to negative heat transfer resistances but the important aspect to be considered is its variation along time. The Potuguese boilers in power stations have two paths in the convection section, requiring the calculation of the gas flow separation between both paths. The monitoring model developed was coupled with the automate system of operating the soot blowing during a short test period. The decision criteria to operate soot blowers were a mix of rules from the operation and trigger values for the heat transfer resistances. The setting for the trigger values was some how arbitrary and was based on the observed values for an acceptable boiler efficiency loss.

Later, IST enhanced the monitoring tool with a boiler simulator where the boiler operator could simulate the effect of lowering the heat transfer resistance on individual heat exchangers on the calculated boiler efficiency. This simulation tool was installed in the Sines power plant as part of the ACORDE project. From the data generated in this project, the rate of heat transfer degradation was analyzed as a function of the coal quality (Silva and Azevedo, 2000) but no correlation could be established due to the small variation in ash quality in the period analyzed. The monitoring model was further adapted and used for three other boilers operating on coal, oil and natural gas.

The present paper addresses the application of heat transfer monitoring models to biomass fired boilers. Section 2 presents the basis of the monitoring model and section 3 describes the three boilers considered in this study. Section 4 presents results from the application of the monitoring model, discussing the use of flue gas temperature measurements, comparing calculated gas temperature with measurements and showing the evolution of heat transfer resistances. Section 4 contains also a discussion on the use of the monitoring model results to assist boiler operation and section 5 presents the conclusions from the present work.

2. Heat transfer monitoring model

The boiler model considered in this study considers steady state conditions for the flue gas circuit, while in the water/steam circuit transient effects are considered to calculate the heat exchange rate. The boiler is represented by elements on the air/flue gas circuit and elements in the water/steam circuits and their connections. The identification of the elements to be considered is performed according to the plant data available on-line and differs from boiler to boiler. The water/steam circuit is divided in heat exchangers corresponding in the sequence of flow to pre-heaters, economizers, boiler drum, evaporators (water panels and tube banks), super-heaters, re-heaters and atemperators. The turbine is not part of the boiler domain but its working conditions are considered to evaluate the reheated flow that is obtained after deducing the steam extraction flow for pre-heaters, that is not the case in any of the boilers considered here. As the plant instrumentation normally includes most inlet and outlet water temperature measurements energy balances are performed to the water circuit to calculate the heat absorbed in individual heat exchanger surfaces from:

$$\dot{Q}_{Abs} = \dot{m}_w \Delta h_w + m_w \, \Delta u / \Delta t \qquad (1)$$

where $\dot{m}_w$ and m_w are the water mass flow rate and total mass in the heat exchanger. h and u are the enthalpy and internal energy calculated using the International Association for the Properties of Water and Steam in 1997 (Wagner et al, 2000) and Δt is the time interval between the measurements considered. The application of the monitoring model therefore, requires measurements of the water and steam temperature at the inlet/outlet of individual heat exchangers. When some of the intermediate water/steam temperatures are not measured in the boiler an assumed value is used to calculate the heat absorption that is later estimated based on heat transfer correlations.

The heat absorbed in individual evaporators can also not be estimated directly from plant data as they are not usually monitored. Models for the natural circulation of water in the boiler were developed but their calibration to specific boilers is difficult due to the complex flow network and ultimately the heat transfer is mainly determined from the gas side. Therefore for evaporators heat transfer correlations are used or alternatively when flue gas temperature is measured, heat balances to the flue gases are performed. In the application of the model the variation of the mass of water in the boiler drum was considered from the measurement of the boiler drum. The variations were found to be well correlated with observed variations between the feed water and live steam flow rates.

Boiler exit gas temperature (BEGT) is measured in all plants as it is an important parameter to control boiler efficiency, during operation. Further thermocouples to measure flue gas temperature are also installed in some boilers but in general as temperature increase the accuracy of the measurements decrease. Therefore the use of other flue gas temperature measurements further to BEGT are only considered when there is no other possibility to reconstruct the heat balances. The flue gas temperature is calculated starting from the BEGT through all heat exchangers back to the furnace from the following energy balance:

$$h_{g,in} = h_{g,out} + \left(\sum \dot{Q}_{Abs} - \dot{Q}_{Rad} \right) / \dot{m}_g \qquad (2)$$

applied to all zones including tube banks as well as furnace zones.

In Eq. (2) $\dot{m}_g$ is the gas flow rate and h_g is the gas enthalpy. The total gas flow is calculated as the sum of the air flow rate, that is measured although with large errors, and the biomass flow rate estimated from the measured flue gas oxygen concentration. For coal fired units the coal feed rate is often measured but the composition is not known on-line so the model presents some errors due to these factors. For oil and natural gas firing, the error in the fuel flow rate is within 3% but for solid fuel firing it may be larger.

The heat balance in Eq. (2) includes the heat exchanged by radiation $\dot{Q}_{Rad}$ from the zone considered to the neighbour zones. The heat exchanged either by convection or radiation in the gas zone with all heat exchanger surfaces is represented by the heat absorption term $\dot{Q}_{Abs}$. The heat transfer model has two different approaches, depending on the main mechanism being radiation or convection heat transfer. For all the regions, both mechanisms are considered but the algorithm is different assuming that one mechanism is dominant.

For tube banks, convection heat transfer is assumed to be the main mechanism and radiation is considered from an equivalent convection heat transfer coefficient. The heat exchanged by convection $\dot{Q}_{Conv}$ for individual heat exchangers is calculated as the absorbed heat, discounting the radiation heat transfer exchanged with the inlet and outlet planes of the tube bank. Based on the heat exchanged by convection and on the calculated values of the flue gas temperature and water/steam temperature, the effective global heat transfer coefficient (U) is calculated from:

$$U = \dot{Q}_{Conv} / (F \Delta T_{ln}) \tag{3}$$

where ΔT_{ln} represents the logarithmic mean temperature difference, F the correction factor for the flow configuration. In some cases relations using the ε-NTU method are used for convenience of calculation to solve Eq. (2) and Eq. (3) when no radiation heat transfer is considered. The application of these methods is restricted to the case of only two streams exchanging heat, so when considering the evaporator walls due to their lower area compared with the tube banks, the surface temperature is assumed to be similar. When heat exchangers with equivalent area are in parallel, the model considers the actual water/steam heat exchanged in each one and the gas flow is considered with the same inlet and outlet temperature, divided in two streams. The heat transfer resistance in the convection tube banks is calculated from the global heat transfer coefficient where it is the single assumed unknown.

$$R_f = U^{-1} - \left[\frac{1}{h_e} + \frac{D}{2k} \ln\left(\frac{D}{D_i}\right) + \frac{D}{D_i}\left(R_{ox} + \frac{1}{h_i}\right) \right] \tag{4}$$

where h_i is the convection heat transfer coefficient inside the tubes calculated with the water/steam flow, h_e is the convection heat transfer coefficient between the gas and the deposit surface calculated using the correlations from Zukauskas et al (1998) and a contribution from radiation of the gas, R_{ox} is the heat transfer resistance due to the deposits inside the tubes, k is the conductivity of the tube material and D_i and D are the inner and external tube diameters.

The procedure for the boiler furnace and tube panels is different due to the larger importance of radiation heat transfer. The domain here is divided in zones including the furnace and zones with tube panels and cavities until the flue gas inlet to tube banks. Each zone is separated from the neighbours by ficticious planes across which the incident radiation (Irradiation - G) from one zone (I) is made equal to the radiosity (J) to the next zone (I+1). The irradiation onto a surface is calculated from:

$$J_{i,I+1} = G_{i,I} = \left(\sum_{j=1}^{N} J_j \cdot F_{ij} \cdot \tau_I \right)_I + \varepsilon_I \cdot \sigma T_I^4 \tag{5}$$

where the sum in the second term includes the radiosities from all surfaces in zone I and the emission of radiation from the gas particle mixture assumed at an average temperature T_I in the zone. In the above expressions F_{ij} are shape factors and ε_I and τ_I are the emissivity and transmissivity of the gas (Leckner, 1971) and particle mixture. $\sigma = 5.67 \cdot 10^{-8}$ W/m^{2}K^4 is the Stefan Boltzman constant. The irradiation at the end of the radiation zone is assumed to be totally absorbed by the tube bank that is considered as a black body. The radiosities for all other heat exchanger surfaces are calculated from an energy balance considering the emission and reflection of radiation.

$$J_i = \varepsilon_i \cdot \sigma T_i^4 + (1 - \varepsilon_i) \cdot G_i \tag{6}$$

where ε_i is the surface emissivity and T_i is the surface external temperature. This surface temperature is calculated iteratively from the absorbed heat flux obtained from the following heat balance:

$$\dot{Q}_{Abs,i} = \frac{A_i \varepsilon_i}{1-\varepsilon_i} \cdot \left(J_i - \sigma T_i^4\right) + A_i h_e \left(T_g - T_i\right) \tag{7}$$

where h_e is the external convection heat transfer coefficient and A_i is the heat transfer area. The surface temperature can be used to calculate the heat transfer resistance from:

$$R_f = \frac{Q_{Abs,i}}{T_i - T_w} - \left[\frac{D}{2k} \ln\left(\frac{D}{D_i}\right) + \frac{D}{D_i}\left(R_{ox} + \frac{1}{h_i}\right)\right] \tag{8}$$

As for the evaporator surfaces only the total heat absorbed can be estimated from the energy balances so Eq. (7) can not be applied to individual heat exchanger surfaces. For the evaporator surfaces an iterative procedure is used to change the heat transfer resistance equally in all surfaces to fit the total absorbed heat flux calculated from the heat transfer model and from the energy balance.

The application of the boiler heat transfer monitoring models to each of the plants considered follows the general principles described above but some other assumptions are made to build up the calculation algorithms that are specific for each case and are mentioned in the following section.

3. Boiler configurations considered.

This section presents briefly the characteristics of the boilers where the monitoring heat transfer model was applied. The methodology was applied to three different biomass fired boilers of different sizes and combustion systems. The application was done for a grate fired biomass boiler installed in Mortágua in Portugal with 9 MWe and two cogeneration plants in Sweden, one is a fluidized bed boiler in Nykopping (90MWth/30MWe) and a pulverized fired boiler in Uppsalla (200MWth/125MWe).

Figure 1 presents a sketch of the flue gas circuit from the Mortágua and Upsalla boilers. The Mortágua boiler consists of the furnace, followed by three convective paths in sequence passing through super heaters, evaporators and economizers. The furnace walls and the walls from the first convective path are evaporating surfaces working in parallel with the evaporator tube banks. For this boiler, due to the presence of two evaporators with tube banks Ev1 and Ev2, the temperature measured between these two was considered in the calculations further to the boiler exit gas temperature. The fouling resistance of Ev1 was considered to be similar to the value for Ev2. The result of the monitoring model using also the other two temperatures are reported in section 4.1 and as will be verified does not produce good results.

The Upsalla boiler was a supercritical unit readapted to fire a mixture of biomass with peat prepared in briquettes. After the furnace, the gases flow upwards through tube panels corresponding in sequence to the intermediate super heater (SH2), final super heater (SH3) and reheater (RH2) in parallel, primary reheater (RH1) and finally through the primary super heater (SH1) tube bank. The gas then passes downward through the economizers (EC2, EC1) tube banks before exiting the boiler. The furnace walls are evaporating surfaces that are connected with the boiler walls considered in parallel with the main heat exchanging surfaces. For Upsalla the temperature of the steam between the first and second super heater is not measured, therefore some assumptions were introduced to estimate the heat transferred in the primary superheater (SH1). As this heat exchanger is a tube bank, the heat transfer resistance was considered to be similar to the one for the economizer (EC2). In this plant thermocouples to measure the flue gas temperature were installed before all the water panels (SH2), (RH2/SH3) and (RH1) and an alternative would be to use the measured flue gas temperature before the first reheater (RH1) and the heat balance to this component to estimate the flue gas temperature after the primary superheater (SH1). The Upsalla boiler is used to generate electricity as well as district heating and therefore has a seasonal operation. Some of the thermocouples installed did not last the whole period of boiler operation and the measurements are also affected by deposits on the probes, so for this boiler, only the boiler exit gas temperature is used in the calculations, while the measured values are used for comparison purposes.

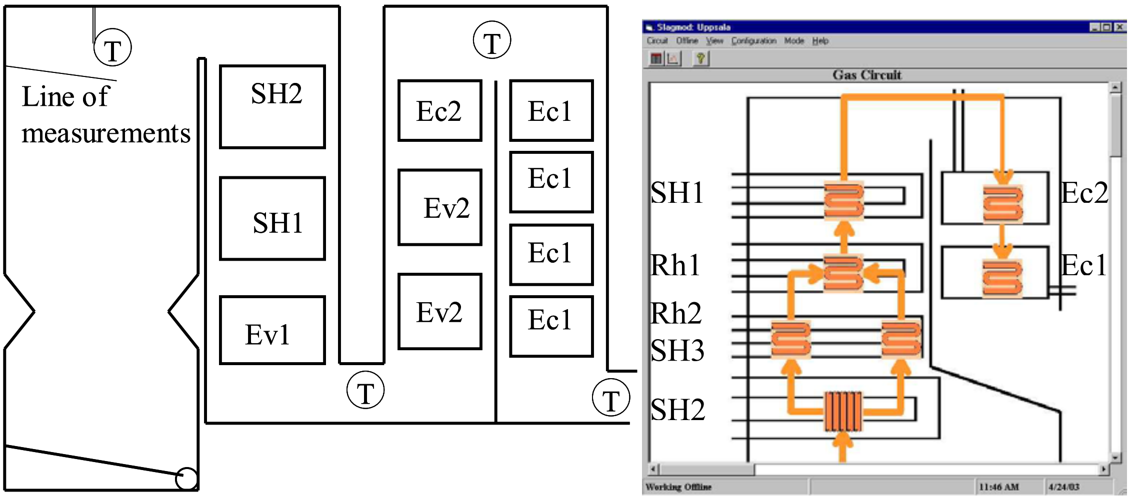

Figure 1 - Sketch of the gas circuits from plants. a) Mortágua (Indication is given of locations where temperature is measured), b) Upsalla (Sketch taken from the graphical interface installed at the plant).

Figure 2 presents the gas circuit of the Nykoping boiler and all the evaporator surfaces considered in the furnace and boiler walls and tube banks. The sketch in Fig. (2) is included in the graphical interface where each component in the flue gas or water/steam circuits is represented by components. The secondary super heater (SH2) consists of water panels and is included in the radiation zone. All the relevant temperature measurements are available for the water circuit and only for the evaporators, energy balances are not possible. After the furnace, the flue gas flows in sequence through two convective paths through the primary super heater (SH1), evaporator (Ev), catalytic converter (Not represented in the figure), economizer (Ec) and air pre-heater (APH). Temperature is measured at the top of the furnace, before and after the secondary super heater, after the evaporator and after the economizer. The heat balances to the water and flue gas in the Economizer do not provide consistent values as there should be some temperature change across the catalytic converter. The calculations done for the economizer were therefore performed only with the flue gas exit temperature. For the rest of the boiler, the calculations were based on the flue gas temperature after the Evaporator, considering the heat transfer resistance from this component similar to the first superheater (SH1). Some calculations were also performed using the temperature measured after the secondary superheater, but the results as will be shown in section 4.1 are not fully consistent for the evaporator.

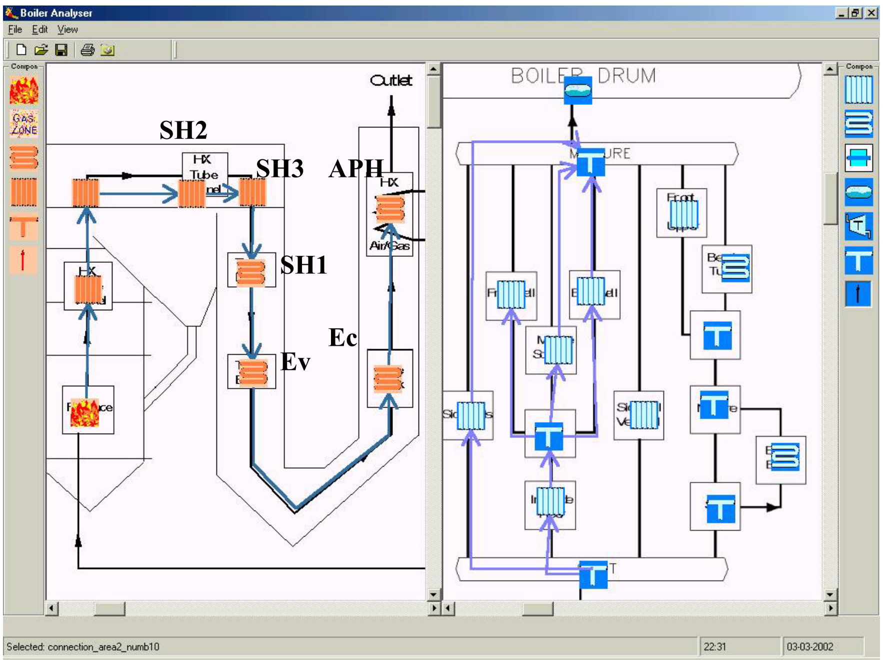

Figure 2 - Sketch of the gas circuit and water panels and tube banks for the evaporator in Nykoping.

4. Monitoring model results

4.1 Results obtained using flue gas temperature measurements

The normal plant instrumentation from Mortágua and Nykoping includes flue gas temperature measurements during normal operation in continuous, while the thermocouples in Upsalla close to steam panels can not stand a long operation period, being used within the SLAGMOD project. For Mortágua and Nykoping the flue gas temperature measurements are useful to calculate the heat exchanged in the evaporator tube banks, as there is no other plant data to evaluate the heat exchanged in each tube bank of the evaporator. This section presents the results of using all the measured gas temperatures.

Figure 3 shows for the Mortágua boiler a comparison between the values obtained in the heat balances and the values of the rate of heat transfer calculated assuming no heat transfer resistance in all exchangers. From this figure it can be observed that there is a good correlation between the calculated and measured values for the super heaters where the heat balance was obtained from measured data on the water circuit. For the economizer there is already some spread in the data as the split between the two economizers EC1 and EC2 are based on flue gas temperature. For the boiler banks the heat balances are obtained from differences between values obtained from the flue gas circuit and from the water circuits, showing a large spread of data and even some negative values obtained for the heat balance.

Figure 4 presents the evolution of heat transfer resistances calculated along a heating season, showing a continuous increase along time. This result seems feasible with higher degradation for the furnace walls and secondary and final super heaters (SH2 and SH3). The results for the evaporator tube bank however do not show a clear tendency and they are a direct consequence of the measured gas temperature values that are used to calculate the heat balances. The results from Fig. (3) and Fig. (4) show the inadequacy of using flue gas temperature measurements for heat balances.

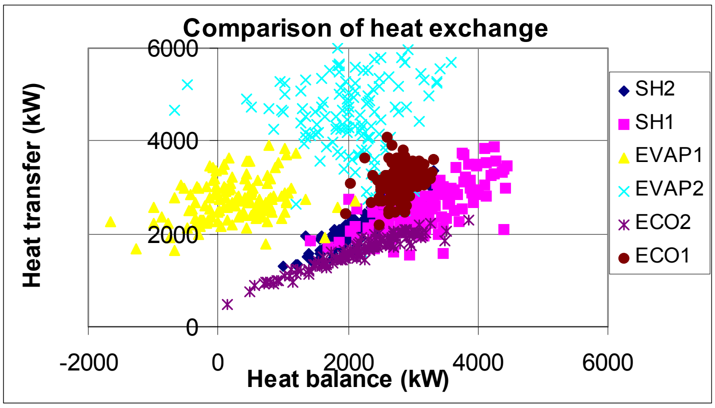

Figure 3 – Comparison between heat exchanged in different tube banks from heat balances and the rate of heat transfer considering zero heat transfer resistance.

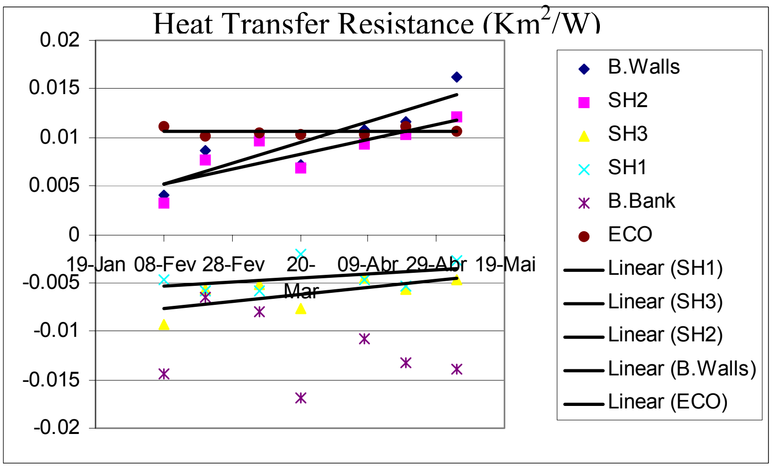

Figure 4 – Evolution of calculated heat transfer resistance along a heating period using gas temperatures.

4.2 Comparison of calculated and measured gas temperature

Due to the inconsistencies found for the evaporator tube banks, the calculation algorithms were modified to minimize the consideration of flue gas temperature measurements. The calculation of the natural water circulation in the tubes requires the knowledge of the heat exchanged, while this depends on the evaporator tube surface that in the presence of nucleate boiling is close to the steam temperature. With this assumption the heat exchanged in the evaporators can be calculated assuming values for the heat transfer resistance that was considered similar to neighbor heat exchangers. For Mortágua the temperature between the evaporator tube banks was considered and the heat transfer resistance of both evaporators was considered similar. For Nykoping the temperature after the evaporator tube bank was considered also because of the catalytic converter and the heat transfer resistance was assumed similar to the first superheater.

Using the modified calculation algorithms the furnace exit gas temperature is a result of the model and therefore can be compared with the plant data for validation. For Mortágua gas species concentrations and temperature were measured inside the furnace at several locations, including a port close to the furnace exit where temperature was found to be around 850ºC. Figure 5 presents a comparison between the calculated and plant data values for the day of the measurements (26/02/03) showing that the calculated value follows the trend from the plant data. The calculated values are in the range between 800 and 900ºC in agreement with the fine thermocouple measurements. Figure 5 (b) presents a comparison between calculated and plant data for the furnace exit gas temperature using very different types of biomass, showing again a correlation between these values.

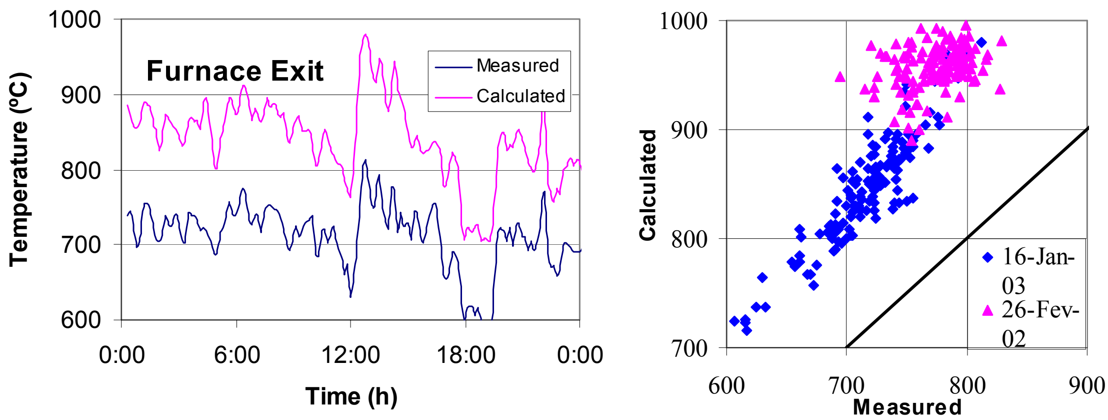

Figure 5 – Comparison between calculated and plant data furnace exit gas temperature in Mortágua boiler. a) Time evolution, b) Comparison for two different periods.

For Nykoping the measured gas temperature before and after the second superheater (SH2) is similar from the plant instrumentation, while from the calculations for full load a difference of about 100ºC is calculated. For the calculations performed starting from the evaporator flue gas exit, the calculated value before the second superheater is about 100ºC higher than the measured value and the difference between the calculated and measured value before this superheater (SH2) is about 200ºC. This result is in line with the values observed by Blug (2003) using acoustic pyrometry that also exceed by about 200ºC the plant data. The plant has a thermocouple installed in the furnace roof where the value observed is about 100ºC higher a difference similar to the one calculated. These comparisons enhanced confidence in the methodology used to estimate the heat removal along the furnace zones. The calculated evolution of the heat transfer resistances over the heating season have a smaller increase than the one shown in figure 4, but as the value for the evaporator was taken similar to the first superheater (SH1) the results are more consistent.

For Upsalla the calculated flue gas temperature is compared with the values measured by industrial type thermocouples installed in the plant in Fig. (6), (a-c). The calculated values are in general higher than the measured values as can be observed from Fig. (6), (b-c). The values observed at the furnace exit show a large difference between both sides of the furnace. The value at the right side is lower than after the superheater at the furnace exit the measured values show large differences between both sides of the furnace. The value observed at the furnace exit right side during this day is not realistic as it is lower than the value measured after super heater 2 and therefore should not be working properly for this day. The values observed at the left side are comparable to the calculated values. Unfortunately the acoustic pyrometry measurements were not performed at the same time so no direct comparison can be performed.

The values observed by acoustic pyrometry were in the range between 1000ºC and 1300ºC, with variations up to 400ºC due to soot blowing. These values are in line with the values observed at the left side and the results of the model. The thermocouple in the left side is possibly clean as it is exposed to the action of soot blowers that could also be responsible of damaging the thermocouple in the right side. The thermocouples between the tube panels can not measure accurately gas temperature either because they are not properly exposed to the gases or because they can build up deposits not cleaned by soot blowing.

Figure (6) (d) presents a comparison between the calculated and average measured values, showing that the model follows the plant data, as could also be observed from Fig. (6) (b-d), although the calculated values have a lower sensitivity than the measured values. The calculated temperature values after the upper water panels (after SH3/Rh2) present a better correlation that is mostly loss at the furnace exit. It should be stressed however that the calculated values are average in time and space.

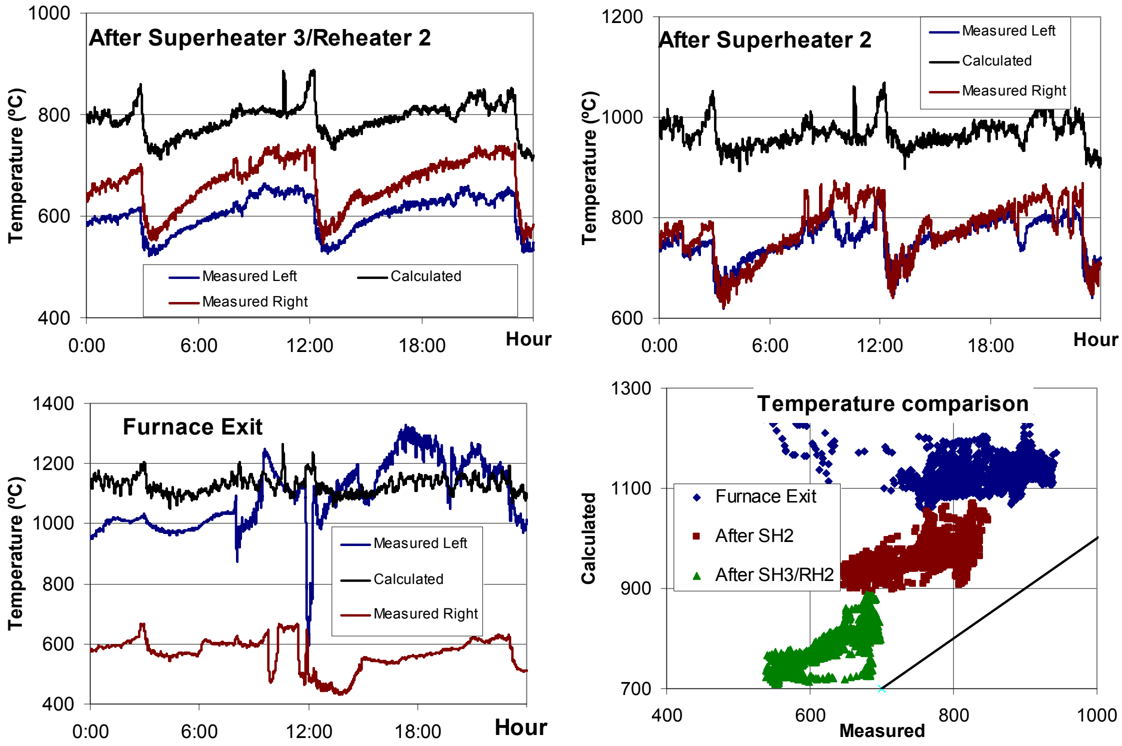

Figure 6 – Comparison between calculated and measured temperature in Upsalla a) Furnace exit, b) Above first tube panel, c) Above second tube panel, d) Comparison of calculated with average measured.

4.3 Evolution of calculated heat transfer resistance

The use of the monitoring model to calculate flue gas temperature was demonstrated in the previous section. The motivation of the model application however is the possibility to generate information about the effects of slagging and fouling. The heat transfer resistances calculated from Eq. (4) and (8) are a measure of these effects. As the model does not exactly represent the reality the calculated values are not accurate and some values are even negative. Convection heat transfer correlations have typically an error of about 10% that have to be added to equivalent errors from the total flue gas flow rate. For the furnace the radiosity calculations performed have a small resolution and the specification of the particle concentration and wall emissivity has larger errors. Therefore the objective of the analysis to be performed is not the absolute evaluation of the heat transfer resistance, but its evolution along time. The values can be compared with the minimum values that are observed when the boiler is started or following soot blowing during normal operation.

For Mortágua the variation of the heat transfer resistance along time has in general small variations, with the exception when using olive stone. The evolution of the heat transfer resistance for a day using this fuel is presented in Fig. (7), when a soot blowing operation occurred at 14h. The figure shows that in the super heaters there is a slight continuous increase of heat transfer resistance that is not affected by soot blowing, while for the economizer there is recover in heat transfer following soot blowing. This result may indicate that soot blowing is not very effective in removing the higher temperature deposits from this type of fuel.

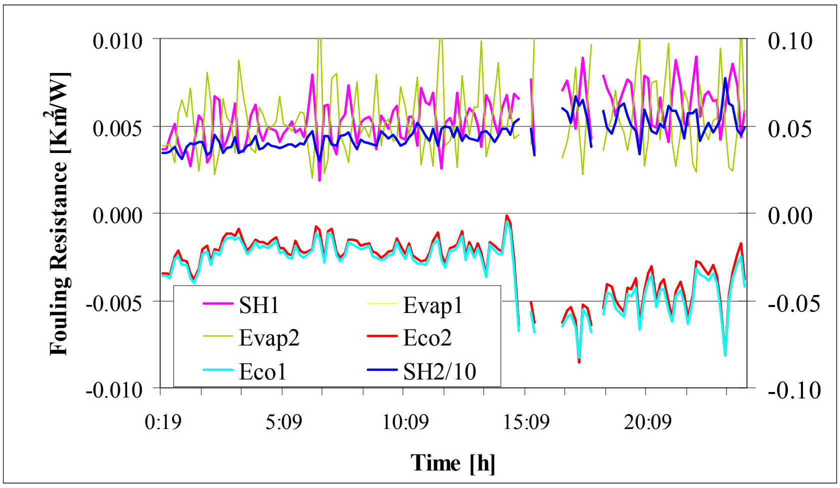

Figure 7 – Variation of heat transfer resistance during a day for Mortágua using olive stones (26/02/02).

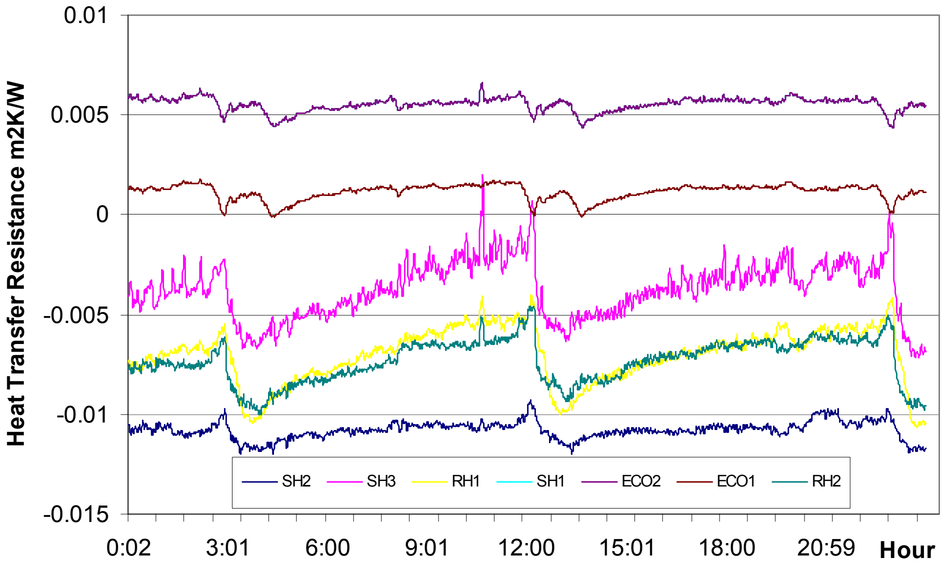

Figure 8 - Variation of heat transfer resistance during a day for Upsalla.

Figure (8) presents the evolution of heat transfer resistance in heat exchanger surfaces for Upsalla boiler. From this figure the effects of slagging and fouling can be clearly observed as well as the effect of soot blowing operations. The economizer is blown firstly to decrease boiler exit gas temperature and afterwards a complete cycle is performed following the flue gas path. Figure (8) shows that when cleaning the super heaters and reheaters, heat transfer resistance in the economizers increase, being reduced when they are cleaned.

The evolution of the heat transfer resistance in heat exchangers for the Nykoping boiler is shown in Fig. (9) as displayed by the interface developed by IST. As for the other boilers the absolute values present large differences but the evolution along time presents the characteristic cycles with an increase during deposits build up and a recover following soot blowing operations. The variation is higher for the secondary super heater in this case as is in general for the heat exchanger closer to the furnace exit. For this super heater the value is calculated considering mainly radiation.

Figure 9 – Evolution of heat transfer resistance in Nykoping. (From the IST interface).

4.4 Further model development and utilization

The evolution of average heat transfer resistances in individual heat exchangers is useful information to support boiler operation. The effects of deposits in heat exchangers can be recognized in the plant data by an increase in the boiler exit gas temperature and a reduction of the atemperator water flow rate. These values however do not show where the deposits have more effect and the monitoring model provides this information. The relative change of heat transfer resistance is a new type of information for boiler operation and its interpretation is not straightforward. The degradation in boiler efficiency during the heat degradation process can be accessed but the effect of the individual heat exchangers is only possible using a boiler simulation model. This approach was implemented in the ACORDE project but requiring the operator to perform these type of simulations. In the current OREMA project where IST is participating simulations are performed automatically by the system to optimize operating conditions. The OREMA project considers a boiler using natural gas or fuel oil where the formation of deposits is not a main issue.

In the initial project developed by IST to the Sines boiler an optimization study was performed to improve boiler efficiency. Taking into account the steam consumption during soot blowing it was concluded that the frequency of soot blowing should be increased to a time interval of about four hours. This optimization took only in consideration the thermal behavior of the boiler but other criteria are important, such as the erosion promoted by the soot blowers in the tubes. It can thus be concluded that the role of the monitoring models is to provide further information to boiler operation. The use of a simulation model is also very important and it should have a similar resolution as the monitoring model to allow the direct use of the heat transfer resistances calculated in the monitoring model.

The monitoring model presented in this paper has specific calculation algorithms for specific boilers. This procedure requires an engineering task to analyze the plant data and prepare the model. This approach was already used for other type of boilers, namely with two parallel paths of the flue gases in the convective section. This was possible for the Sines boiler because the flue gas temperature was measured at the outlet of each of the parallel paths. The application of the model to other plants (Setúbal and Carregado) in Portugal, required the use of a different numerical algorithm to solve a set of non-linear equations by the Newton method (Canhoto, 2003). Although this type of algorithm can be more time consuming for simpler cases, it may reduce the engineering work as it is prepared in a general format and includes the efficiency analysis. This is one of the aims of the work currently being developed at IST, in order to enable the provision of a configured model within a short time period.

5. Conclusions

The present paper presents the methodology of heat transfer monitoring models and its application to three different biomass fired boilers. The use of flue gas temperature measurements for the calculations is shown to lead to inconsistencies in energy balances and on the evolution of heat transfer resistances in the evaporator tube banks.

The application of the monitoring model with normal plant data is shown to calculate flue gas temperature, including the furnace exit gas temperature, following plant data and in better agreement with measurements from fine thermocouples. This is one of the applications of the monitoring model to provide a virtual temperature probe.

The use of the monitoring model to calculate heat transfer resistances along time is shown to produce coherent evolutions of these parameters, reflecting the effects of deposits growth and cleaning due to soot blowing. The paper also discusses the use of the monitoring models to assist boiler operation.

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