DIAGNOSIS OF HEAT EXCHANGER TUBE FAILURE IN FOSSIL FUEL BOILERS THROUGH ESTIMATION OF STEADY STATE OPERATING CONDITIONS

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Abstract

Estimation of operating conditions for fossil fuel boiler heat exchangers is often required due to changes in working conditions, design modifications and especially for monitoring performance and failure diagnosis. Regular heat exchangers in fossil fuel boilers are composed of tube banks through which water or steam flow, while hot combustion (flue) gases flow outside the tubes. This work presents a top-down approach to operating conditions estimation based on field measurements. An example for a 350 MW unit superheater is thoroughly discussed. Integral calculations based on measurements for all unit heat exchangers (reheaters, superheaters) were performed first. Based on these calculations a scheme of integral conservation equations (lumped parameter) was then formulated at the single tube level. Steady state temperatures of superheater tube walls were obtained as a main output, and were compared to the maximum allowable operating temperatures of the tubes material. A combined lumped parameter - CFD (Computational Fluid Dynamics, FLUENT code) approach constitutes an efficient tool in certain cases. A brief report of such a case is given for another unit superheater. We conclude that steady state evaluations based on both integral and detailed simulations are a valuable monitoring and diagnosis tool for the power generation industry.

Introduction

In the estimation of operating conditions for fossil fuel boiler heat exchangers, the level at which diagnosis is necessary depends on the nature of the change in operating conditions or the type of failure; for example, a typical tube failure would require tube temperature estimation, etc.

Two cases will be discussed in this work. In the first one, tube wall temperatures are calculated for a 350 MW unit superheater under steady state operating conditions. Diagnosis at the tube level appeared to be necessary due to repeated failure of some of the tubes. Furthermore, it was necessary to estimate the influence of proposed design changes aimed at lowering the temperature of critical tubes. The first issue (failure) was addressed by adopting a maximum allowable temperature failure criterion, i. e., the obtained temperatures were compared to the maximal allowable value for the material in question. The second issue (design geometry changes) was examined by comparing the results for the existing geometry and the new, proposed one, namely, temperatures and mass flow rates. The scheme renders, in addition to the wall temperatures, steam and gas outlet temperatures for each tube, as well as the corresponding heat flow superheater distribution. The top-down (sequential) approach works as follows. At first the results from a measurement based integral heat balance of the boiler are obtained. The boiler is modeled as an assembly of heat exchangers some of them in parallel (superheaters - evaporators) and others in series. These results provide the input for the heat balance at the superheater-reheater level, which in turn constitute the input for the single tube calculations. In addition an overall heat balance of the boiler is performed as a self consistency check of the above calculations. This check takes into account: the heat produced by coal combustion, the heat absorbed by the water and steam in the different heat exchangers and the heat losses, namely, gases coming out of the chimney, incomplete combustion, heat losses through the boiler walls, etc.

In the second case, repeated tube failures were observed in the final superheater of a 72 MW unit. In this case no correlations were provided by the manufacturer in order to calculate the heat transfer. Therefore, a detailed CFD simulation of flow and heat transfer had to be performed in order to calculate the heat transfer coefficients.

Superheater geometry and flow data

A scheme of the heat exchanger in the 350 MW boiler is shown in fig. 1. The primary and secondary reheaters are designated as RH1 and RH2 respectively. The corresponding notation for the primary, secondary and final superheaters is indicated by SH1, SH2 and SH3 respectively. The economizer is denoted as EC. The problem of interest concerned the final superheater SH3. This heat exchanger operates in a cross-flow regime (vertical gas flow - horizontal tubes). The superheater is composed of 34 vertical sections (tube banks parallel to the plane of fig. 1). Each section is composed of ten tubes which in their way upwards undergo a series of eight horizontal folds. Steam is supplied from two different headers to two groups of 17 sections each. The headers locations for the two groups are different. Two typical hydraulic paths arise then in this analysis as steam flows from all tubes into the outlet header. The influence of changes in the internal diameter and length of some of the tubes was examined.

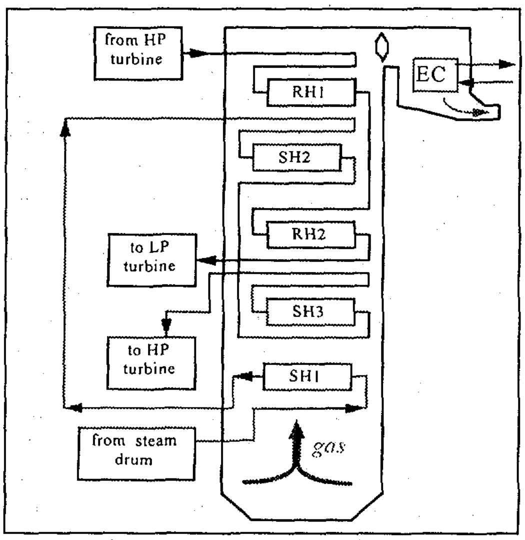

Fig. 1: Scheme of the boiler

The mathematical model

No coupling is assumed between the hydraulic and heat transfer calculations, due to the negligible influence of buoyancy forces. The above is validated at the end of the computation, since the ratio of buoyancy to inertia forces is smaller than 1 [1]:

$$(Gr/\mathrm{Re}^2) << 1, \tag{1}$$

where Gr denotes the Grashoff number and Re the Reynolds number. Incompressible flow was assumed due to the small pressure variations obtained and the steam velocities range. The input for this step of the calculation comes from the integral balance and some of the measured values. In addition to the geometry, the relevant input variables for the detailed (tube level) calculation are: superheater steam inlet temperature, overall steam mass flow rate, heat flow absorbed by the superheater, heat flow absorbed by the evaporator (in parallel with the superheater), superheater gas inlet temperature, gas mass flow rate. Loss coefficients (steam) for bends and other accessories are defined according to the boiler manufacturer estimations [2]. The calculations are performed using Matlab and it is therefore convenient to work with the equivalent length coefficients in the corresponding matrices. The expressions for the hydraulic computation are:

$$\Delta p = R\,\dot{m}^2 \Leftarrow \begin{cases} \dot{m} = \rho\, A\, \mathrm{v} \\ R = \dfrac{f\, L}{2\,\rho\, A^2\, D} \end{cases}, \tag{2}$$

where ρ denotes the density of steam, A the cross sectional area, v the velocity, f the friction coefficient, D the internal diameter, R the hydraulic resistance and Δp the pressure loss. The length L is given below with N_b as a loss coefficient.:

$$L = \begin{cases} L_{tube} & \text{for straight tubes,} \\ \dfrac{N_b D}{f} & \text{for bends and other accesories.} \end{cases} \tag{3}$$

The resistance for elements in series is:

$$R_s = \sum_i R_i\,, \tag{4}$$

and for elements in parallel:

$$R_P = \left(\prod_{i=1}^{n} R_i\right) \Bigg/ \left[\sum_{i=1}^{n} \prod_{\substack{j=1 \\ j\neq i}}^{n} \sqrt{R_i}\right]^2 . \tag{5}$$

The heat transfer formulation scheme is illustrated for a typical control volume. The control volume for the two first folds, includes a single tube, for the rest of the superheater, tubes are stacked in pairs at the same height (identical flue gas conditions). The control volume then includes two tubes. A parabolic scheme is used, i. e. the equations for each control volume are solved by using the solution of the previous tube (flue gas direction) as input. The heat transfer coefficients correlations are given in [2], the convection coefficient between the steam and the tube:

$$h_i^l = \left[0.023\frac{G_i^{s\,0.8}}{D_i^{in\,0.2}}\right]\left[\frac{c_p^{s\,0.4}\,k^{0.6}}{\mu^{s\,0.4}}\right]\left[\frac{T_i^s}{T_i^f}\right]^{0.8}, \tag{6}$$

where $T_i^s = \dfrac{T_i^{S_{in}} + T_i^{S_{out}}}{2}$, $T_i^f = \dfrac{T_i^s + T_i^{ws}}{2}$, are the average (inlet-outlet) steam temperature in the i[th] tube and the film temperature respectively. T^{ws} denotes the inner (steam side tube temperature. Indices s and g designate steam and gas, respectively. The intermediate brackets in the above expression denote the influence of steam physical properties. G

stands for the mass flow rate per unit area (obtained from the hydraulic calculation) and D^{in} for the internal diameter. Convection heat transfer between gas and the tube external wall (cross flow) is characterized by the following heat transfer coefficient:

$$h_i^c = \left[0.287\frac{G_i^{g\,0.61}}{D_i^{o\,0.39}}\right]\left[\frac{c_p^{g\,0.33}k^{0.67}}{\mu^{g\,0.28}}\right], \qquad (7)$$

where D^o stands for the outer tube diameter. The relations for convection heat transfer from the gas to the tube and from the tube to the steam are:

$$\begin{aligned} \dot{Q}_i &= h_i^c \pi\, L_i D_i^o \left(T_i^g - T_i^{Wg}\right), \\ \dot{Q}_i &= h_i^l \pi\, L_i D_i^{in} \left(T_i^{WS} - T_i^S\right) \qquad (8a,b,c) \\ T_i^g &= \frac{T_i^{g_{out}} + T_i^{g_{in}}}{2}. \end{aligned}$$

$\dot{Q}_i$ denotes the heat flow transferred to the ith control volume and T^{Wg} the external wall temperature. Conservation of energy for gas and steam is formulated as:

$$\begin{aligned} \dot{Q}_i &= \dot{m}_{SH3}^g \left(H_i^{g_{out}} - H_i^{g_{in}}\right), \\ \dot{Q}_i &= \dot{m}_i^s \left(H_i^{s_{out}} - H_i^{s_{in}}\right). \end{aligned} \qquad (9a,9b)$$

H denotes the enthalpy and $\dot{m}_{SH3}^g$ the part of the gas mass flow rate which transfers heat to the superheater (recall that it is in parallel with the steam generator on the wall), given by:

$$\dot{m}_{SH3}^g = \frac{\dot{Q}_{SH3}}{\dot{Q}_{SH3} + \dot{Q}_{wall}}\left(\dot{m}_{SH3}^g + \dot{m}_{wall}^g\right). \quad (10)$$

The heat flows are obtained from the integral heat balance. Finally heat conduction through the wall yields:

$$\dot{Q}_i = \frac{2\pi\, k\, L_i}{\ln\left(D_i^o / D_i^{in}\right)}\left(T_i^{Wg} - T_i^{Ws}\right). \qquad (11)$$

The above constitutes a system of five nonlinear equations with five unknowns, namely, $\dot{Q}_i, T_i^{s_{out}}, T_i^{g_{out}}, T_i^{WS}, T_i^{Wg}$ for each tube. In the upper (outlet) bank, the elementary control volume is split and the number of unknowns raises to ten. The overall heat balance is checked once the solution is obtained. Some of the results will be displayed to illustrate the model capabilities. The estimated maximum tube temperatures normalized to the maximum allowable value are shown in fig. 2 for two different tube thickness series of data.

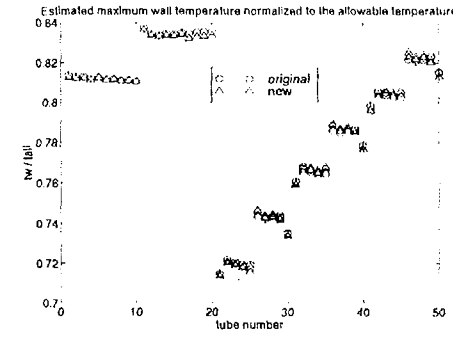

Fig. 2: Normalized wall temperatures.

Heat transfer coefficients estimation

A different approach must be used when the correlations for the heat transfer coefficients are not provided (72 MW unit). The solution would require a detailed CFD simulation of the flow and heat transfer. However, the size and geometry of the superheater made a full CFD treatment too large for practical purposes. The steam flow in the 168 tubes was controlled by orifices of different diameters, which makes the geometry of the pipe not relevant for pressure drop (mass flow rate) calculations. The heat flow was calculated for a typical single tube using a CFD code. The superheater was modeled using an equivalent geometry which allowed for fully developed flow (equivalent pipe length) and both the pressure drop and the heat transfer were introduced as localized sources (lumped parameters within the CFD code). Heat balances were checked by calculating for each seven tubes panel (24 of them).

Conclusions

Modern computational means enable the building of combined engineering models, in the whole range of detail, from integral balance, through lumped parameter to full field CFD simulation. In this work, some of these options were used. The model used for the large unit allows for a detailed mapping of the heat flux, steam, gas and metal temperatures under steady state conditions operation. Superheater wall temperatures were below the allowable limit for both designs. Failure was attributed to the graphytization tendency of the steel used for the tubes in question found in the specimens extracted from the failed tubes. The influence of design changes can be evaluated as shown in the work.

References

[1] Burmeister, L. C., 1983. *"Convective Heat Transfer"*. John Wiley & Sons, Inc.

[2] Babcock & Wilcox, 1992. *"Steam, its Generation and Use"*, 40th edition.