TUBULAR TYPE HEAT FLUX METER FOR MONITORING INTERNAL SCALE DEPOSITS IN LARGE STEAM BOILERS

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ABSTRACT

Deposition of scale on the inner surfaces of the water-wall tubes in the high heat flux regions in a steam boiler furnace can cause serious operation problems. In this paper, a numerical technique for determining: heat flux absorbed by the water-wall tubes, water-steam temperature and thermal resistance on the inner tube surface, from a temperature measured at several interior locations of the tube wall is developed. The scale deposition tube is capable of monitoring changes in the flow of heat transfer caused by scale depositions and changes due to varying furnace conditions. It can work for a long time in the destructive high temperature atmosphere of a coal-fired boiler. The scale deposition monitor is an on-line plant monitoring system designed to improve the operation of steam boilers and to enhance tube life.

INTRODUCTION

Internal scale or corrosion deposits are an effective insulating barrier to the transfer of heat from flame to steam.

A thin internal deposit layer will raise the tube-metal temperature into the ash-corrosion range or into the rapid-oxidation range leading to serious furnace-tube problems. In the extreme, furnace wall-tubes can fail by a creep or stress-rupture mechanism due to the overheating. In addition, corrosion under the deposits leads to hydrogen damage, which can lead to premature tube failure. The guidelines for the definition of the need for chemical cleaning are based on the internal deposit loading given in g/m^2, ASTM Standard /4/ or on temperature measurements by chordal thermocouples located at the crown of the tube, e.g. on the outer surface of the water-wall side of the tube, /6, 8, 14/.

The temperature measurement is very simple and it can be used in on-line mode to determine the need for chemical cleaning. However, this technique can be inaccurate since the tube-metal temperature increases are not only due to the larger scale thickness on the internal tube surface. Tube metal temperature increases depend also on many parameters as: the heat flux, the temperature of the water-steam mixture, and the internal heat transfer coefficient.

In this study, three unknown parameters: the heat flux, water-steam temperature and water-side thermal resistance are estimated such that the calculated temperatures agree with measured temperatures at five interior locations. The Levenberg-Marquardt method is used to solve the non-linear least-squares problem.

The temperature distribution over the cross-section of the flux-tube is computed at each iteration step using the Finite Volume Method (FVM).

The technique described in the paper enables effects in heat transfer caused by internal scale deposition to be decoupled from changes due to varying furnace conditions. Internal scale deposition detection is accomplished independent of variable furnace conditions, i.e. load, excess air, etc.

Results are presented that demonstrate capabilities for in situ measurement of scale deposition on the inner tube surfaces, tube temperatures, water-steam temperature and heat flux to the water-wall tube. These parameters when measured are useful in determining chemical cleaning frequency and extending tube life. Flux-tubes described in the paper, strategically placed on the furnace tube wall in the highest heat-flux regions, can be a valuable boiler diagnostic device.

DESIGN OF THE FLUX-TUBE

Whilst several methods exist for the measurement of boiler heat flux /2, 3, 7, 9-11, 18, 19/ and detecting the accumulation of internal deposists, they all have disadvantages in practice. If a heat flux instrument is to measure the absorbed heat correctly, it must resemble the tube as closely as possible so far as radiant heat exchange with the flame and surrounding surfaces is concerned. The two main factors in this respect are the emissivity and the temperature of the absorbing surface. Since the instrument will almost always be coated with ash, it is generally the properties of the ash and not the instrument that dominate the situation. Unfortunately, the thermal characteristics of ash, such as absorptivity and conductivity, can vary widely. Therefore, accurate readings will only be obtained if the deposit on the meter is representative of that on the surrounding tubes. The tubular type instruments, known also as flux-tubes /11/, meet this requirement. In these devices the measured furnace wall metal temperatures are used for the evaluation of heat flux. Several investigators have reported the results of development efforts designed to develop such flux-tubes. It is normal practice to measure the temperature at the front of the tube with two chordal thermocouples placed at the radii r_1 and r_2 (Figures 1 and 2). Chordal thermocouples are placed in holes of known radial coordinates r_1 and $r_2 \leq r_1$. Since both spacing and thermal conductivity k are known, a measurement of temperature difference $(T_1 - T_2)$ gives the heat flux q_m at the outer surface of the tube $r = r_o$

$$q_m = \frac{k\left(T_1 - T_2\right)}{r_o \ln\left(r_1/r_2\right)}. \tag{1}$$

a) b)

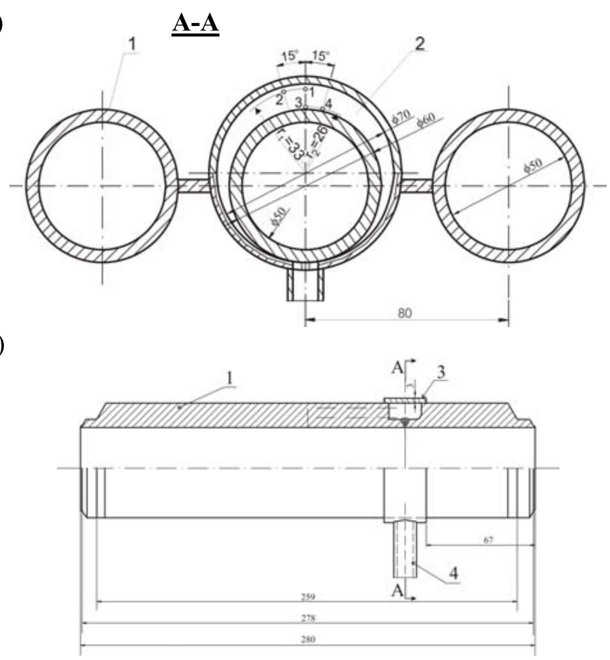

Fig. 1 Tubular type instrument (flux-tube) for heat flux measurement: 1 – water-wall tube, 2 – flux-tube, 3 – thermocouple protecting cover, 4 – thermocouple protecting tube, r_1 – radius at which thermocouples no. 1 and 2 are located, r_2– radius at which thermocouples no. 3 and 4 are located

Tubular type heat flux meters, which operate on a similar principle, have been described in references /6, 14/.

The measuring tube is fitted with two thermocouples in holes of known radial spacing. The thermocouples are led away to a junction box where they are connected differentially to give a flux related e. m. f. These instruments can be used to indicate tube crown surface temperature and to detect water-steam scale in addition to absorbed heat flux. The simple gradient thermocouple method of measuring heat flux has many limitations. Relatively small distance and small differential temperatures exist between inner and outer thermocouple locations. Consequently, any small error in determining temperatures or in distance between thermocouples usually represents a large percentage of this small difference.

Errors in temperature measurements occur also due to a number of sources including the thermal contact resistance between thermocouple and tube wall and the thermal conduction effect along the thermocouple wires and sheath from the hot junction to cooler surroundings at the rear of the tube. With non-uniform heat flux at the front surface of the flux-tube, heat flows by conduction in the circumferential direction to the colder rear that is thermally insulated.

The circumferential heat flow affects the temperature distribution in the flux-tube to such an extent that radial one-dimensional heat flow does not give a good approximation for the exposed portion of the flux-tube and the heat flux cannot be calculated from Equation (1).

The tubular type meters, while capable of monitoring changes in the flow of heat into the boiler tubes, cannot determine the inside heat transfer coefficient or the equivalent heat transfer coefficient taking into account water-steam-side film resistance and scale resistance. In this study, a numerical method for determining the heat flux, the equivalent heat transfer coefficient on the inner tube surface and water-steam temperature, based on experimentally acquired interior flux-tube temperatures, is presented.

The flux-tube is illustrated in Figs 1a and 1b. The tubular type instrument developed here is an improved version of the instrument described in references /15, 16/. The meter is constructed from a short length of eccentric tube containing four thermocouples on the fire side below the inner and outer surfaces of the tube.

The fifth thermocouple is located at the rear of the tube (on the casing side of the water-wall tube).

The boundary conditions on the outer and inner surfaces of the water flux-tube must then be determined from temperature measurements at the interior locations.

Four K-type sheathed thermocouples, 1 mm in diameter, are inserted into holes, which are parallel to the tube axis. The thermal conduction effect at the hot junction is minimized because the thermocouples pass through isothermal holes. The thermocouples are brought to the rear of the tube in the slot (Fig. 1b) machined in the tube wall. An austenitic cover plate with thickness of 3 mm welded to the tube is used to protect the thermocouples from the incident flame radiation. The computer model studies using the program ANSYS show that the effect of the slot on the strength of the tube is small.

A K-type sheathed thermocouple with a pad is used to measure the temperature at the rear of the flux-tube. This temperature is almost the same as the water-steam temperature.

THEORY

The furnace wall tubes in most modern units are welded together with steel bars (fins) to provide membrane wall panels which are insulated on one side and exposed to a furnace on the other, as shown schematically in Fig. 2. A flux-tube has been designed at the Institute for Process and Power Engineering of the Cracow University of Technology.

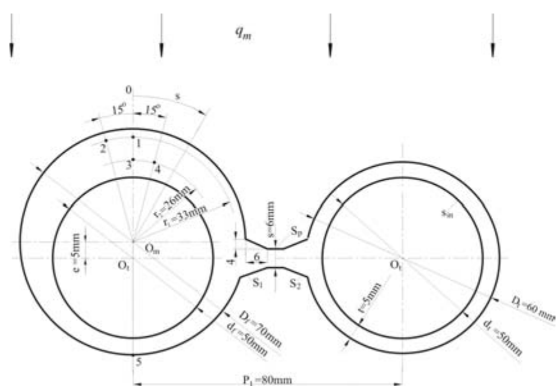

Fig. 2 Flux-tube with adjacent water-wall tube;
1÷5 – locations of thermocouples

The flux-tubes were fabricated in the laboratory and then securely welded to the adjacent water-wall tubes at different elevations in the furnace of the steam boiler The coal fired boiler produces 58.3 kg/s superheated steam at 11 MPa and 540°C.

Because of the symmetry, only the representative water-wall section of width P_t shown in Fig. 2 needs to be analyzed.

In a heat conduction model of the flux-tube the following assumptions are made:

- temperature distribution is two-dimensional and steady-state,
- the thermal conductivity of the flux-tube and membrane wall may be dependent of temperature,
- the heat transfer coefficient h_{in} and the scale thickness δ_s is uniform over the inner tube surface.

The temperature distribution is governed by the non-linear partial differential equation

$$\nabla \cdot [k(T)\, \nabla T] = 0, \qquad (2)$$

where ∇ is the vector operator, which is called nabla, and in Cartesian coordinates is defined by $\nabla \equiv \mathbf{i}\partial/\partial x + \mathbf{j}\partial/\partial y + \mathbf{k}\partial/\partial z$.

The unknown boundary conditions may be expressed as:

$$\left[k(T) \frac{\partial T}{\partial n} \right] \bigg|_s = q(s), \qquad (3)$$

where $q(s)$ is the radiation heat flux absorbed by the exposed flux-tube and membrane wall surface. The local heat flux $q(s)$ is a function of the view factor $F(s)$ (Fig. 3)

$$q(s) = q_m F(s), \qquad (4)$$

where q_m is measured heat flux (thermal loading of heating surface). The view factor $F(s)$ from the infinite flame plane to the differential element on the membrane wall surface can be determined graphically /15/, or numerically.

In this paper, $F(s)$ was evaluated numerically using the finite element program ANSYS /1/, and is shown in Figure 3 as a function of extended coordinate s (Figure 2).

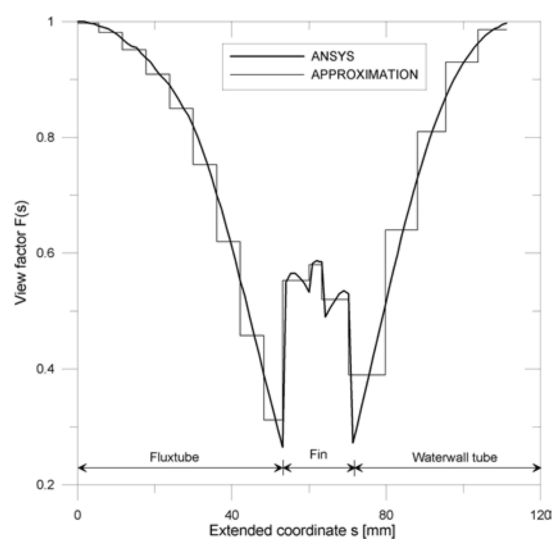

Fig. 3 View factor associated with radiation exchange between elemental surface on the membrane water-wall and flame (infinite plane)

The convective heat transfer from the inside tube surfaces to the water-steam mixture is described by Newton`s law of cooling

$$-\left[k(T) \frac{\partial T}{\partial n} \right] \bigg|_{s_{in}} = h_{in} \left(T\big|_{s_{in}} - T_f \right), \qquad (5)$$

where $\partial T/\partial n$ is the derivative in the normal direction, h_{in} is the heat transfer coefficient and T_f denotes the temperature of the water-steam mixture. The reverse side of the membrane water-wall is thermally insulated.

In addition to the unknown boundary conditions, the internal temperature measurements f_i are included in the analysis:

$$T_e\left(\mathbf{r_i}\right) \equiv f_i, \quad i = 1,...,m, \tag{6}$$

where m = 5 denotes the number of thermocouples (Figure 2). The unknown parameters: $x_1 = q_m$, $x_2 = h_{in}r_{in}/k_{ref}$, and $x_3 = T_f$ were determined using the least-squares method. The second dimensionless parameter x_2 is the Biot number, $r_{in} = d_f/2 = d_t/2$ denotes the inside tube radius, and $k_{ref} = k(T_{ref})$ is the thermal conductivity at reference temperature T_{ref}, which can be chosen arbitrarily, for instance $T_{ref} = f_3$. The object is to choose $\mathbf{x} = (x_1, \ldots, x_n)^T$ for n = 3 such that computed temperatures $T(\mathbf{x},\mathbf{r}_i)$ agree with certain limits with the experimentally measured temperatures f_i. This may be expressed as

$$T\left(\mathbf{x},\mathbf{r_i}\right) - y_i \cong 0, \quad i = 1,...,m, \quad m = 5. \tag{7}$$

The least-squares method is used to determine parameters $\mathbf{x}$. The sum of squares

$$S = \sum_{i=1}^{m}\left[f_i - T\left(\mathbf{x},\mathbf{r_i}\right)\right]^2, \quad m = 5, \tag{8}$$

can be minimized by a general unconstrained method.

However, the properties of (8) make it worthwhile to use methods designed specifically for the nonlinear least-squares problem. In this paper the Levenberg-Marquardt method /13, 17/ is used to determine the parameters x_1, x_2 and x_3. The Levenberg-Marquardt method performs the k^{th} iteration as

$$\mathbf{x}^{(k+1)} = \mathbf{x}^{(k)} + \delta^{(k)}, \tag{9}$$

where

$$\delta^{(\mathbf{k})} = \left[\left(\mathbf{J}^{(k)}\right)^T \mathbf{J}^{(k)} + \mu^{(k)}\mathbf{I_n}\right]^{-1}\left(\mathbf{J}^{(k)}\right)^T\left[\mathbf{f} - \mathbf{T}\left(\mathbf{x}^{(k)}\right)\right], \tag{10}$$

$$k = 0,1...,$$

where μ is the multiplier and $\mathbf{I_n}$ is the identity matrix. The Levenberg-Marquardt method is a combination of the Gauss-Newton method ($\mu^{(k)}\rightarrow 0$) and the steepest-descent method ($\mu^{(k)}\rightarrow\infty$). The $m \times n$ of $T(\mathbf{x}^{(k)},\mathbf{r}_i)$ is given by

$$\mathbf{J}^{(k)} = \left.\frac{\partial \mathbf{T}\left(\mathbf{x}\right)}{\partial \mathbf{x}^T}\right|_{\mathbf{x}=\mathbf{x}^{(k)}} = \left.\begin{bmatrix} \frac{\partial T_1}{\partial x_1} & \cdots & \frac{\partial T_1}{\partial x_n} \\ \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots \\ \frac{\partial T_m}{\partial x_1} & \cdots & \frac{\partial T_m}{\partial x_n} \end{bmatrix}\right|_{\mathbf{x}=\mathbf{x}^{(k)}}, m = 5, n = 3 \tag{11}$$

where $\mathbf{T}(\mathbf{x}^{(k)}) = (T_1^{(k)}, ..., T_m^{(k)})$. The iterative procedure is continued until the changes in $x_i^{(k)}$, $i = 1, \ldots, n$ are less than some small amount ε.

At every k-th iteration step the temperature distribution $T(\mathbf{x}^{(k)},\mathbf{r}_i)$ is calculated. The boundary value problem given by Equation (2) and boundary conditions (3) and (5) can be solved by the finite volume method (FVM) or finite element method (FEM).

The uncertainties of the determined parameters $\mathbf{x}$ were estimated using the error propagation rule of Gauss, Press et al. /18/.

ON-LINE MONITORING OF WATER-WALL TUBES

The method for solving the inverse heat conduction problem described above is time consuming because at every iteration step the temperature distribution has to be determined in the whole domain. In addition, the solution of direct heat conduction problem needs also many iteration steps, because the thermal conductivity $k(T)$ depends on temperature.

Fast computation of the temperature field is required for on - line determination of the parameters x_1, x_2 and x_3. This can be achieved by assuming constant, temperature-independent thermal conductivity k. The temperature distribution in the flux-tube and adjacent membrane wall tube was computed using FVM software package FLUENT /5/.

In order to check that the results are satisfactory, the temperature calculations were also conducted using the FEM code ANSYS. The domain discretizations for the FVM and FEM computations are shown in Figs 4 and 5, respectively. The same results are basically obtained with both methods. The small differences between the results are caused by different approximations of $F(s)$ used in the FVM and FEM analyses.

In the first method, $F(s)$ is approximated by the step-wise function, while the piecewise linear interpolation is used in the second method. Figures 6 and 7 show dimensionless metal temperatures

$$\theta = \frac{T - T_f}{\left(q_m r_{in}/k\right)} \equiv \frac{T - x_3}{\left(x_1 r_{in}/k\right)}, \quad (12)$$

as a function of the dimensionless parameter $x_2 = h_{in} r_{in}/k$.

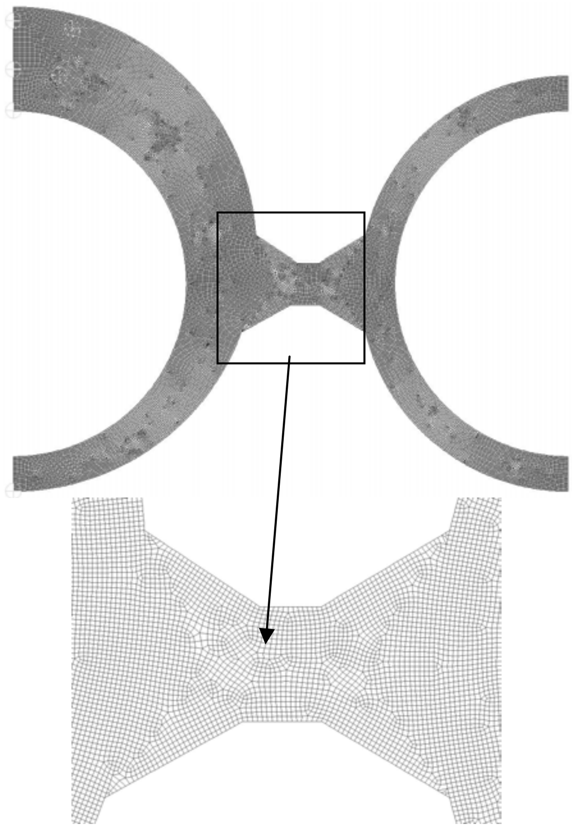

Fig. 4 Mesh of finite volumes for the calculation of temperature distribution using Fluent program

Since the direct heat conduction problem is linear for $k = const$, then the membrane wall temperatures θ_i, $i = 1, \ldots, 5$, the mean temperature over the whole domain $\bar{\theta}$, and the mean temperature over the flux-tube wall thickness $\bar{\theta}_w$ at $s = 0$ (Fig. 2) can be approximated by the following function:

$$\theta = 0.00894\left(A + \frac{B}{x_2} + \frac{C}{x_2^2} + \frac{D}{x_2^3} + \frac{E}{x_2^4} + \frac{F}{x_2^5} - 320\right), \quad (13)$$

where the constants $A, \ldots, F$ were determined by approximation of the results obtained from FLUENT using the least-squares method.

Rearranging Equation (12) gives the dimensional temperatures

$$T_i = T\left(\mathbf{x}, \mathbf{r}_i\right) = x_3 + \frac{x_1 r_{in}}{k}\theta_i\left(x_2\right), \quad i = 1, \ldots, 5, \quad (14)$$

$$\bar{T} = x_3 + \frac{x_1 r_{in}}{k}\bar{\theta}\left(x_2\right), \quad (15)$$

$$\bar{T}_w = x_3 + \frac{x_1 r_{in}}{k}\bar{\theta}_w\left(x_2\right). \quad (16)$$

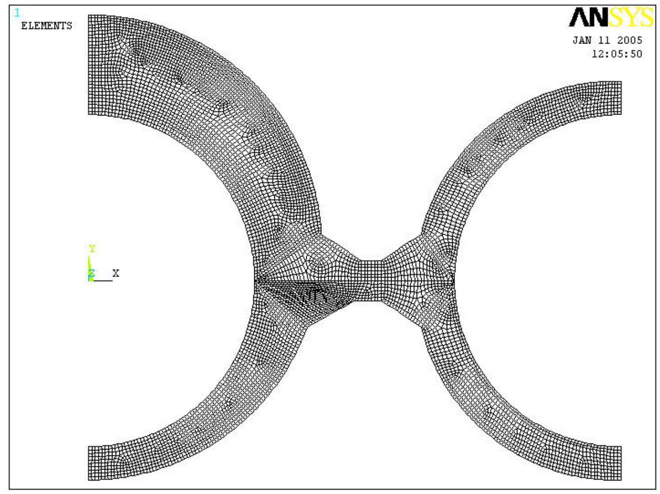

Fig. 5 Mesh of finite elements for the calculation of temperature distribution using ANSYS program

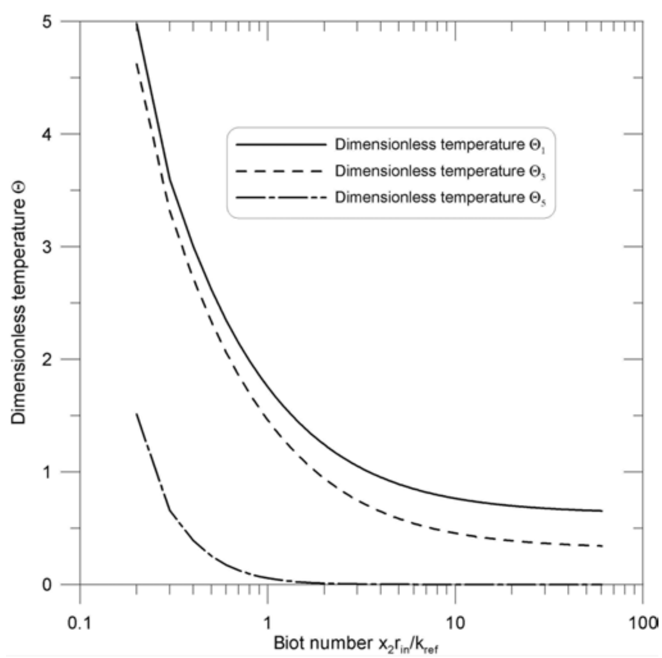

Fig. 6 Dimensionless flux-tube – metal temperatures at locations no. 1, 3 and 5 versus Biot number

The simplified procedure described above can also be used for temperature dependent thermal conductivity. In the first iteration $k^{(0)} = k_{ref} = k(f_3)$ may be assumed. Having determined the parameters $x_1^{(0)}$, $x_2^{(0)}$ and $x_3^{(0)}$ using the Levenberg-Marquardt method, the computations are repeated for $k^{(1)} = k(T_w^{(0)})$.

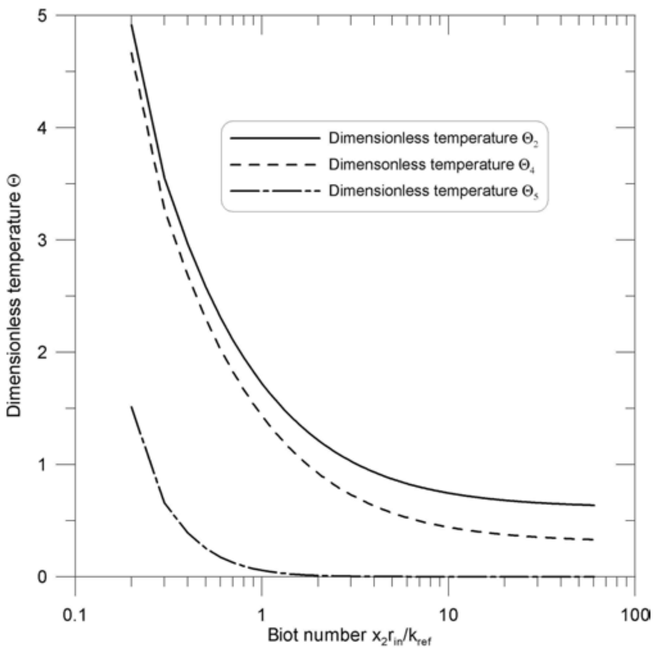

Fig. 7 Dimensionless flux-tube – metal temperatures at locations no. 2, 4 and 5 versus Biot number

The iteration process is terminated after a few iterations. Thermal conductivity $k(T)$ is calculated at temperature T_w, because most thermocouples are located at the front section of the flux-tube

TEST CASES

The first numerical experiment with simulated data is presented to demonstrate the effectiveness of the present method. The dimensions of the flux-tube are shown in Figure 2. To simulate "exact" measurement data the direct heat conduction problem was solved using: q_m = 200000 W/m^2, h_{in} = 40000 W/(m^{2}K), T_f = 320°C. The thermal conductivity of tube material was assumed constant: k = 44.7 W/(mK). The temperature distribution was computed using FLUENT code. The temperatures at the five thermocouple locations are then obtained as f_1 = 397.63°C, f_2 = 395.62°C, f_3 = 362.80°C, f_4 = 361.32°C, and f_5 = 320.01°C. Based on these "experimental data" an inverse analysis was conducted using the method proposed in this paper.

The estimated results: q_m = 200007.2 W/m^2, h_{in} = 40032.27 W/(m^{2}K), T_f = 320.01°C are in excellent agreement with the input data.

The second example is used to demonstrate the accuracy of the same method for the temperature-dependent thermal conductivity. The thermal conductivity of tube material (carbon steel) varies with temperature

$$k = 53.26 - 0.023778T \tag{17}$$

where k is expressed in W/(mK) and T in °C.

The "measured" temperatures: f_1 = 417.0°C, f_2 = 415.52°C, f_3 = 382.78°C, f_4 = 381.02°C, and f_5 = 320.08°C were obtained from the FLUENT simulation for the following input data: q_m = 200000 W/m^2, h_{in} = 10000 W/(m^{2}K), and T_f = 320.00°C. The results of the application of present method (for on line monitoring) are: q_m = 200565.9 W/m^2, h_{in} = 10390.3 W/(m^{2}K) and T_f = 320.02°C. It can be seen that the determined parameters are not significantly affected by the variable conductivity.

BOILER TESTS

Four flux meters with the design shown in Fig. 1 were installed at a 50 MW coal fired steam boiler. The temperatures measured at the flux-tube situated at the elevation of 15.4 m were: f_1 = 405.1°C, f_2 = 402.4°C, f_3 = 366.8°C, f_4 = 364.1°C, and f_5 = 318.2°C. The tube thermal conductivity is given by Eq. (17). The 95% confidence limit uncertainties for the measured temperatures and the thermal conductivity were: ±0.2°C and ±0.5 W/(mK).

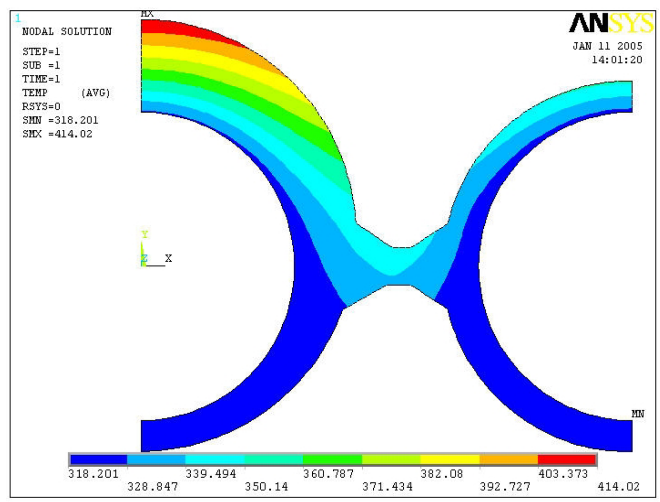

Fig. 8 Computed temperature distribution (in °C) in the water-wall without scale on the inner surface of the flux-tube and water-wall tube; q_m = 220235.3 W/m^2, h_{in} = 37105.3 W/(m^{2}K), T_f = 318.2°C

The method proposed in this paper gives: q_m= 220235.3±5913.3 W/m^2, h_{in}= 37105.5±3084.8 W/(m^{2}K), and T_f = 318.2±0.0001°C. The minimum sum of squares is S = 0.6637K^2, and $k\left(\overline{T}_w\right)$ = 44.37 W/(mK).

The temperature distribution and isotherms at the cross-section of the membrane wall are shown in Figs 8 and 9, respectively.

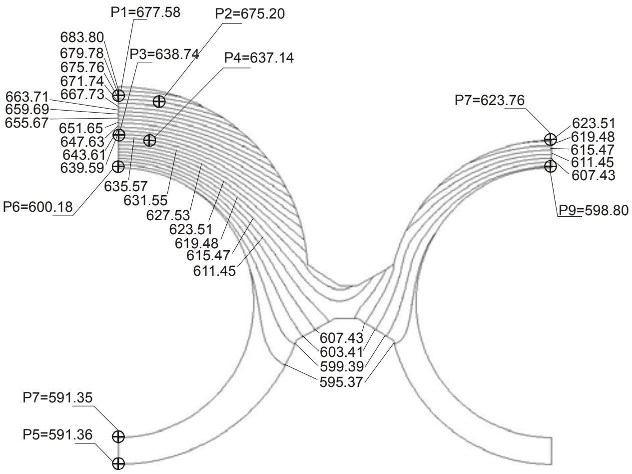

Fig. 9 Computed isotherms (in K) in the water-wall without scale on the inner surface of the flux-tube and water-wall tube; q_m = 220235.3 W/m^2, h_{in} = 37105.3 W/(m^{2}K), T_f = 318.2°C

The second example is the same as the first one, except that the scale is deposited on the inner surfaces of the flux-tube and water-wall tubes.

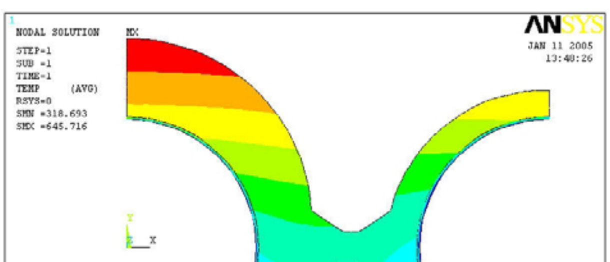

Fig. 10 Computed temperature distribution (in °C) in the water-wall with scale on the inner surface of the flux-tube and water-wall tube; q_m = 220235.3 W/m^2, h_{in} = 37105.3 W/(m^{2}K), T_f = 318.2 °C, δ_s = 0.5 mm, k_s = 0.5 W/(mK)

Figs 10 and 11 show the temperature distribution and isotherms for fouled tube surfaces. The thickness and the thermal conductivity of the scale are: δ_s = 0.5 mm and k_s = 0.5 W/(mK).

The scale layer affects the tube-metal temperature. The maximum flux-tube temperature is 645.72°C, and exceeds the allowable temperature for the carbon steel.

With the assumption that conduction through the scale layer is essentially one-dimensional, an equivalent heat transfer coefficient h_e will be introduced to account for the thermal resistance of the scale

$$\frac{1}{h_e} = \frac{r_{in}}{k_s} \ln \frac{r_{in}}{r_{in} - \delta_s} + \frac{r_{in}}{r_{in} - \delta_s} \frac{1}{h_{in}}. \tag{18}$$

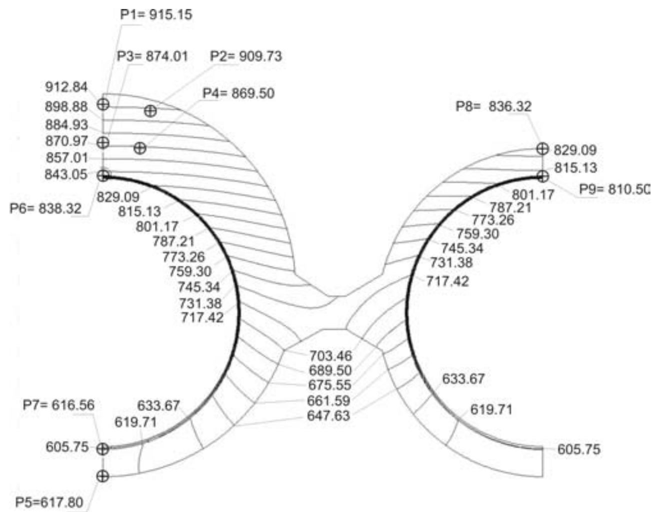

Fig. 11 Computed isotherms (in K) in the water-wall with scale on the inner surface of the flux-tube and water-wall tube; q_m = 220235.3 W/m^2, h_{in} = 37105.3 W/(m^{2}K), T_f = 318.2°C, δ_s = 0.5 mm, k_s = 0.5 W/(mK)

If $\delta_s << r_{in}$, then Equation (18) can be simplified to

$$\frac{1}{h_e} = \frac{\delta_s}{k_s} + \frac{1}{h_{in}} \tag{19}$$

Taking into account, that: r_{in} = 0.025 m, δ_s = 0.0005 m, k_s = 0.5 W/(mK) and h_{in} = 37105.5 W/(m^{2}K), the equivalent heat transfer coefficient obtained from Equation (19) is: h_e = 963.73 W/(m^{2}K). Equation (19) gives h_e = 973.76 W/(m^{2}K).

The temperature distribution in the analyzed domain obtained for q_m = 220135.3 W/m^2, h_{in} = 37105.5 W/(m^{2}K) and T_f = 318.2°C is shown in Figure 12.

The discrepancies between the results shown in Figures 10 and 12 are very small.

In order to show the effectiveness of the flux-tube as a scale detector, the inverse analysis was conducted for the "measured" temperatures shown in Figure 11: f_1 = 915.015K = 642°C, f_2 = 909.73K = 636.58°C, f_3 = 874.01K = 600.86°C, f_4 = 869.50K = 596.36°C, and f_5 = 617.80K = 344.65°C.

The following results are obtained: q_m = 219983.9±6021.09 W/m^2, h_e = 1012.1±17.84 W/(m^{2}K), and T_f = 323.78±0.79°C, S = 0.0697K^2, and $k\left(\overline{T}_w\right)$ = 38.77 W/(mK).

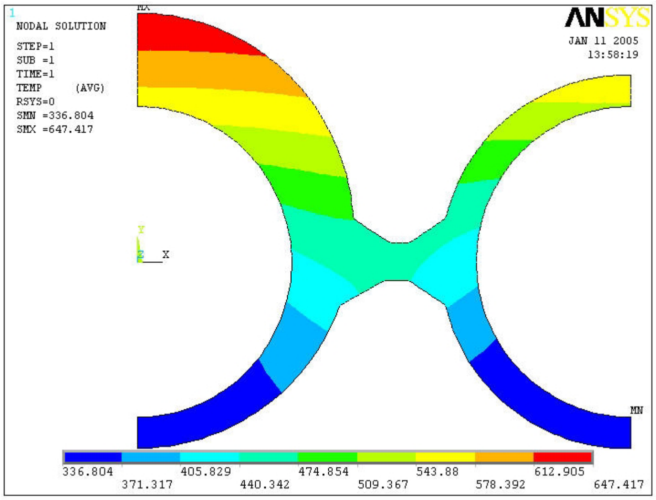

Fig. 12 Computed temperature distribution (in °C)– thermal conductivity and thickness of the scale on the inner surface of the flux- and water-wall-tube are accounted for by calculation an equivalent heat transfer coefficient h_e; q_m = 220235.3 W/m^2, h_e = 963.73 W/(m^{2}K), T_f = 318.2 °C

With the known value of h_{in} = 37105.5 W/(m^{2}K) for the clean flux-tube, the thermal resistance of the scale layer δ_s/k_s can be evaluated from Equation (19)

$$\frac{\delta_s}{k_s}=\frac{1}{h_e}-\frac{1}{h_{in}}=\frac{1}{1012.1}-\frac{1}{37105.5}=9.611\cdot 10^{-4}\,\frac{m^2K}{W} \qquad (20)$$

The relative difference between the input (exact) value (δ_s/k_s = $1\cdot 10^{-3}$ m^{2}K/W) and the obtained result is only 3.89%.

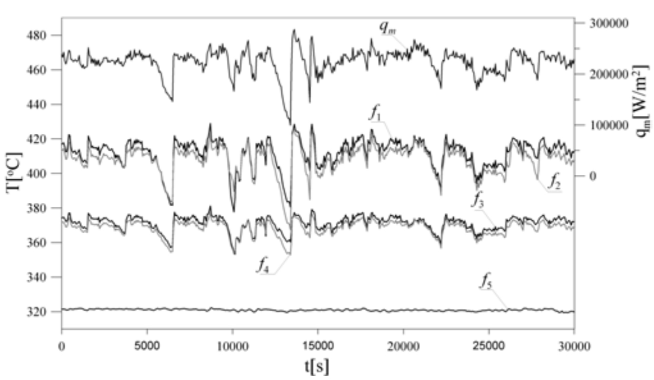

Fig. 13 Measured flux-tube – metal temperatures f_1÷f_5 and determined heat flux in the middle of the front furnace wall at elevation of 15.4 m

The last example shows the measured temperatures and the estimated heat flux obtained in on line mode (Fig. 13). The flux-tube is installed at the 50 MW steam boiler on the front membrane wall at the elevation of 15.4 m.

CONCLUSIONS

It has been demonstrated that the tubular type instrument and the mathematical method described here can monitor the following parameters: the absorbed heat flux to water tubes on the membrane wall and the temperature of the water-steam mixture. The presence of the scale on the inside surface of the wall tube and its thermal resistance can also be detected. The scale deposition tube is capable of monitoring changes in the flow of heat transfer caused by scale depositions and changes due to varying furnace conditions. It can work for a long time in the destructive high temperature atmosphere of a coal-fired boiler. The scale deposition monitor is an on-line plant monitoring system designed to improve the operation of steam boilers and to enhance tube life.

NOTATION

A, ..., G	constants
Bi	Biot number, $h_{in}r_{in}/k$
f_i	measured flux-tube temperature, °C or K
F	view factor
h_e	equivalent heat transfer coefficient, W/(m^{2}K)
h_{in}	heat transfer coefficient on the inner surface of the tube, W/(m^{2}K)
J	Jakobian matrix of **T**
k	thermal conductivity, W/(mK)
m	number of measuring points
n	number of unknown parameters
r_{in}	inner radius of the flux-tube, m
r	position vector
s	extended surface coordinate, m
T_f	fluid temperature, °C or K
T_i	calculated temperature at the location $\mathbf{r_i}$, °C or K
T	m-dimensional column vector of calculated temperatures
x_i	unknown parameter
x	n-dimensional column vector of unknown parameters
δ_s	scale thickness, m
Θ	dimensionless temperature

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