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A method of determining local heat flux in boiler furnaces

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Abstract—A method for determining the local heat flux in boiler furnaces from the distribution of temperature in instrumental tube sections, is presented. The solution technique proposed takes into account such factors as scale deposit, tangential wall heat conduction and temperature-dependent conductivity. The method can also be used to determine the inside heat transfer coefficient and the water/steam temperature.

1. INTRODUCTION AND AIMS

THE HEAT flux entering steam generating tubes in power station boilers may be a critical factor in considering the safety of the tubes. There is also a need to know the distribution and the magnitude of this flux to assess boiler performance under widely differing operating conditions and provide data for future plants. The design of a modern boiler furnace requires the computation of furnace wall metal temperatures for proper selection of the tube material and thickness. These temperatures are functions of the heat fluxes and the internal heat transfer coefficients. The heat flux distribution is required for the sizing of the boiler's combustion chamber. Whilst several methods [1–15] exist for the measurement of boiler heat flux, they all have disadvantages in practice. Local heat fluxes in the furnace chambers are usually measured by heat flux meters inserted in inspection ports [1–3].

This type of measurement is, however, becoming more difficult due to the increase in the unit capacity and complexity of boilers. 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, but 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 [1–4]. The tubular type instruments [1, 2, 5–9], known also as fluxtubes [1, 2] meet this requirement. In these devices the measured furnace wall metal temperatures are used for the evaluation of heat flux (see Fig. 1).

Several investigators have reported the results of development efforts designed to develop such flux-

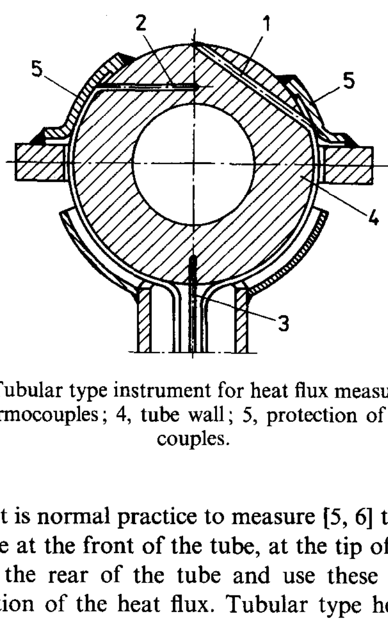

FIG. 1. Tubular type instrument for heat flux measurement: 1–3, thermocouples; 4, tube wall; 5, protection of thermocouples.

tubes. It is normal practice to measure [5, 6] the temperature at the front of the tube, at the tip of the fin and at the rear of the tube and use these for the calculation of the heat flux. Tubular type heat flux meters which operate on a similar principle have been described in refs. [7–9]. 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 arc connected differentially to give a flux related e.m.f. These instruments can be used to indicate tube crown surface temperature in addition to absorbed heat flux.

The heat flux can be obtained using only one thermocouple to measure the tube crown temperature and the second to indicate water/steam temperature [13, 14]. It was assumed that the measured temperature in point 3 (see Fig. 2) coincides with the bulk temperature of fluid [13]. Therefore, for the determination of the heat flux, only one measured tube temperature was used. The heat flux and then the temperature distribution were calculated by means of an iterative procedure in such a way as to again find in point 1 the measured value with a given accuracy. The inside heat transfer coefficient was calculated using the

NOMENCLATURE

a inside radius of tube [m]
b outside radius of tube [m]
Bi Biot number, ha/k
c constant
$(Df(x))^{-1}$ inverse of Jacobian matrix of f
D_0, D_m dimensionless coefficients of the series expansion of the heat flux distribution
$f(x)$ n-dimensional column vector
$\partial_j f$ partial derivative with respect to the jth variable
h heat transfer coefficient [W m^{-2} K^{-1}]
h_j discretization parameter of the jth variable
k thermal conductivity [W m^{-1} K^{-1}]
q heat flux [W m^{-2}]
q_F heat flux conducted from the fin root to the tube [W m^{-2}]
q_p measured heat flux [W m^{-2}]
r radius [m]
$r_1, r_2, \ldots$ radial coordinates of thermocouples [m]
R dimensionless radius, r/a
s thickness of the fin [m]
t pitch of the wall tubes (projected length exposed to radiation) [m]
T calculated temperature [°C]
T_r reference temperature, $T_r \geqslant y_1$ [°C]
u ratio of the outside to the inside radius of the tube, b/a
$x = (x_1, \ldots, x_n)^T$ column vector with components x_i
x^T transpose of x
$y_1, y_2, \ldots, y_n$ measured temperatures [°C]
Y dimensionless measured temperature, y/T_r.

Greek symbols
θ dimensionless calculated temperature, T/T_r
φ angular coordinate [rad]
ψ angle factor
ω angle (Fig. 3) [rad].

Subscripts
f fluid
F fin
in inside
r reference.

Superscripts
(k) iteration number
$\overline{}$ mean.

classical formulae from the literature. The method is suitable for new tubes without an accumulated scale on the inside surfaces.

A major weakness of most of the aforementioned methods is the calculation of the inside heat transfer coefficients using known correlations from literature instead of experimentally determining their values.

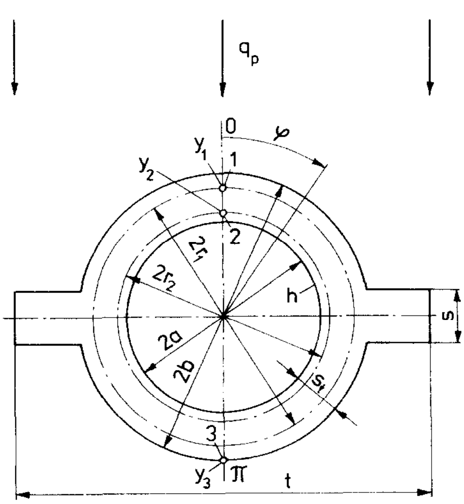

FIG. 2. Membrane-wall cross section: 1–3, temperature sensors (thermocouples).

Moreover, these methods are not appropriate for measuring deposition of scales on the inner surfaces of the water tubes. The tubular type meters, while capable of monitoring changes in the flow of heat into boiler tubes, cannot determine the inside heat transfer coefficient. Therefore, an improved method of determining the local heat flux is needed. In this study, a numerical technique for determining the heat flux in boiler furnaces, based on experimentally acquired interior tube temperatures, is presented.

2. THEORY

A heat transfer problem is considered as direct when boundary conditions are prescribed. On the other hand, the inverse problem is concerned with the determination of the boundary conditions from temperature measurements inside a heat-conducting body. That is, given a system description and a response, find the input, which caused it. Typically, these problems arise because measurements can only be made in easily accessible locations or perhaps a desired variable can only be measured indirectly. The problem of determining the heat flux in the boiler tube wall is an example of the inverse heat conduction problem. In order to determine radiant heat flux q_p in the boiler furnace, it is necessary to calculate, additionally, the inside heat transfer coefficient h and water/steam temperature T_f from a temperature measured at several interior locations of the tube wall.

In a general case, there are n dimensionless parameters: x_1, x_2, ..., x_n to be determined such that the calculated T_i and measured temperatures y_i at n locations inside the tube wall are equal

$$\begin{aligned} f_1 &= \theta_1 - Y_1 = 0 \\ f_2 &= \theta_2 - Y_2 = 0 \\ &\dots \\ f_n &= \theta_n - Y_n = 0 \end{aligned} \quad (1)$$

where

$$\theta = T/T_r \quad \text{and} \quad Y = y/T_r.$$

Equations (1) represent a system of n nonlinear equations in the unknowns $(x_1, x_2, \ldots, x_n)^T$ and can be solved by any appropriate technique for nonlinear algebraic equations. One means of solving the nonlinear set of equations is to apply the Newton–Raphson method [16]

$$x^{(k+1)} = x^{(k)} - (Df(x^{(k)}))^{-1} f(x^{(k)}), \quad k = 1, 2, 3, \ldots \quad (2)$$

where k is the iteration number, $Df(x^{(k)})$ represents the Jacobian matrix of system (1) and $(Df(x^{(k)}))^{-1}$ its inverse.

With the improved $x^{(k)}$ found, new values of the functions and derivatives can be computed, and a new set of $x^{(k+1)}$ unknowns determined. The process is repeated until convergence is satisfactory. This can be expressed as

$$\left| \frac{x_i^{(k+1)} - x_i^{(k)}}{x_i^{(k+1)}} \right| \leqslant e, \quad i = 1, 2, \ldots, n \quad (3)$$

where e is some specified tolerance.

At each step, not only the n components of $f(x^{(k)})$ but also the n^2 entries $\partial_j f_i(x)$ are needed.

The most direct approach to avoiding the computation of $Df(x^{(k)})$ analytically is simply to approximate the partial derivatives by difference quotients

$$\partial_j f_i(x) \approx \frac{f_i(x_1, \ldots, x_j + h_j, \ldots, x_n) - f_i(x_1, \ldots, x_j, \ldots, x_n)}{h_j} \quad (4)$$

2.1. *Measurement of local heat flux distribution at boiler finned tube wall*

The tubular type meter has been designed to provide a very accurate measurement of absorbed heat flux, inside heat transfer coefficient, water/steam temperature and, additionally, to provide measurements of boiler tube crown temperature. The meter is illustrated in Fig. 1. It is constructed from a short length of boiler tube containing two thermocouples just below (or at) the inner and outer surfaces of the tube.

The third thermocouple, which is located at the rear of the tube, is used to evaluate water/steam temperature. The boundary conditions on the outer and inner surfaces of the water tube must then be determined from temperature measurements at the interior locations. The heat flux q_p, heat transfer coefficient h and, water/steam temperature T_f, written in dimensionless form, are determined such that the calculated and measured temperatures at these three interior locations are equal.

To determine x_1, x_2, and x_3, a temperature distribution in the tube has to be calculated. Equations (1) are expressed in dimensionless form. Therefore, the parameters q_p, h and T_f have been reduced to the following parameters:

$$x_1 = \frac{q_p b}{k T_r}; \quad x_2 = Bi = \frac{ha}{k}; \quad x_3 = \frac{T_f}{T_r} \quad (5)$$

and

$$\theta = \frac{T}{T_r}.$$

In this way all necessary parameters have been made dimensionless for subsequent application.

In the method of analysis that follows, the tube and the fin are disjoined and considered independently of each other. This type of approach has the advantage of treating the heat flux measurement for geometries, both with and without fins, in the same computer program.

An outside heat flux boundary condition (see Fig. 3)

$$\left.\frac{\partial \theta}{\partial R}\right|_{R=u} = \frac{q(\varphi)a}{kT_r} = \frac{q(\varphi)x_1}{q_p u} \quad (6)$$

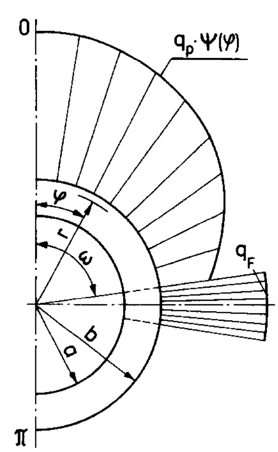

FIG. 3. Variation of the heat flux over the tube circumferences.

is assumed with a uniform heat transfer coefficient inside the tube (i.e. $h(\varphi) = h$)

$$\left.\frac{\partial \theta}{\partial R}\right|_{R=1} = x_2(\theta|_{R=1} - x_3). \quad (7)$$

The local heat flux $q(\varphi)$ on the tube circumference is a function of the angle factor $\psi(\varphi)$ [8]

$$q(\varphi) = \begin{cases} q_p \psi(\varphi), & 0 \leqslant \varphi \leqslant \omega \\ q_F, & \omega < \varphi < (\pi - \omega) \\ 0, & (\pi - \omega) \leqslant \varphi \leqslant \pi. \end{cases} \quad (8)$$

It should be noted that the measured heat flux value q_p represents the mean thermal loading on the entire projected finned tube surface. Taking into account that the reverse side of the finned tube wall is thermally insulated, the heat flux q_F conducted from the fin root to the boiler tube by means of the attachment weld is

$$q_F = \frac{q_p \left[t - 2 \int_0^{\omega} b\psi(\varphi)\,\mathrm{d}\varphi \right]}{2s}. \quad (9)$$

The total heat absorption by the tube surface and fins is given by

$$Q = q_p t.$$

The temperature distribution in the tube is obtained by separation of variables [17].

Assuming that the incidence of heat is symmetrical, in relation to the plane through the tube axis perpendicular to the fins, the temperature distribution in the tube is

$$\theta = \frac{T}{T_r} = x_3 + x_1 \left\{ D_0 \left(\ln R + \frac{1}{x_2} \right) + \sum_{m=1}^{\infty} \frac{D_m u^m \left[(x_2 + m) R^m - (x_2 - m) \left(\frac{1}{R} \right)^m \right] \cos(m\varphi)}{m[x_2(u^{2m} + 1) + m(u^{2m} - 1)]} \right\} \quad (10)$$

where

$$D_0 = \frac{q_0}{q_p} \quad \text{and} \quad D_m = \frac{q_m}{q_p}.$$

Approximating the angle factor $\psi(\varphi)$ by

$$\psi(\varphi) \approx \cos(c\varphi), \quad 0 \leqslant \varphi \leqslant \omega$$

with

$$c = \frac{1}{\omega} \arccos \psi_F$$

where $\psi_F = \psi(\omega)$ is the exact angle factor at $\varphi = \omega$, the dimensionless coefficients in the series expansion of the heat flux $q(\varphi)$ can be expressed in closed form as [8]

$$D_0 = \frac{1}{\pi c} \left[\sin(c\omega) + c(\pi - 2\omega) \frac{q_F}{q_p} \right]$$

and

$$D_m = \frac{2}{\pi(c^2 - m^2)} [c \sin(c\omega) \cos(m\omega) - m \cos(c\omega) \sin(m\omega)] + \frac{4 q_F}{\pi m q_p} \times \cos\left(\frac{m\pi}{2} \right) \sin\left[m \left(\frac{\pi}{2} - \omega \right) \right], \quad m = 1, 2, \ldots$$

where

$$\frac{q_F}{q_p} = \frac{t - \frac{2b}{c} \sin(c\omega)}{2s}.$$

If the circumferential heat conduction in the tube wall is neglected compared to the radial heat flux, the heat flux and the inside heat transfer coefficient are given by

$$x_1^{(1)} = \frac{q_p^{(1)} b}{k T_r} = \frac{y_1 - y_2}{T_r} \frac{1}{\ln \frac{r_1}{r_2}} \quad (11)$$

and

$$x_2^{(1)} = \frac{q_p^{(1)} b}{k} \frac{1}{y_1 - x_3^{(1)} T_r - \frac{q_p^{(1)} b}{k} \ln \frac{r_1}{r_2}} \quad (12)$$

where

$$q_p^{(1)} = \frac{x_1^{(1)} k T_r}{b}.$$

Although equation (11) is often used to calculate heat flux, it can cause significant errors. Since the circumferential wall conduction affects both the determined heat transfer coefficient and heat flux, a nondimensional parameter μ will be used to characterize this effect (the peripheral wall heat conduction)

$$\mu = \frac{q_{in}}{q_p u} \quad (13)$$

where

$$q_{in} = k \left.\frac{\partial T}{\partial r}\right|_{r=a}$$

is the heat flux at the front of the tube ($\varphi = 0$) on its inside surface. In order to determine the inside heat flux q_{in}, it was necessary to solve the two-dimensional conduction equation for the tube wall, taking into account both radial and tangential heat transfer. Thus, equation (10) was used to calculate q_{in} in equation (13). The circumferential heat conduction parameter μ is presented in Fig. 4 as a function of the dimensionless heat transfer coefficient x_2. It is evident that there is a significant deviation of the inside heat flux q_{in} from the one-dimensional solution, for which $\mu = 1$.

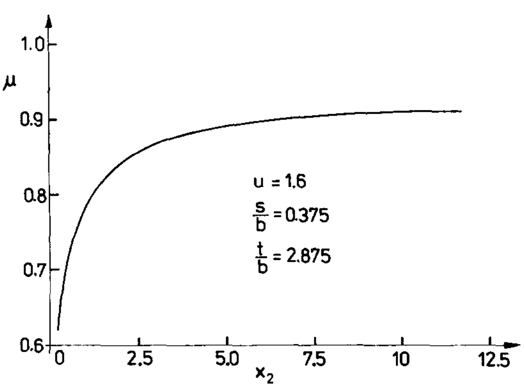

FIG. 4. Circumferential wall conduction parameter.

Therefore, it is not appropriate, to use equation (11) to determine heat flux in boiler furnaces.

In the present analysis, the above quantities $x_1^{(1)}$ and $x_2^{(1)}$ are only taken as the starting values. The starting value of the third unknown is taken as

$$x_3^{(1)} = \frac{y_3 - 5}{T_r}. \tag{14}$$

In order to increase the accuracy of the calculation of the parameters, it was taken into consideration in the temperature distribution analysis of the tube that the thermal conductivity of the tube material is a function of the temperature. The variable conductivity can be accounted for without serious complication of the presented method, using the linearization technique by Dixmier [18]. In the present study, to simplify the presentation, the thermal conductivity was calculated at the mean temperature of the whole cross section of the tube. As the temperature distribution (10) does not depend explicitly on the thermal conductivity, the value of k can be chosen at first arbitrarily, for instance $k = k(T_r)$. Having determined the parameters x_i, $i = 1, \ldots, 3$ using the Newton–Raphson method, a mean temperature of the tube

$$\bar{T} = \frac{2}{\pi(b^2 - a^2)} \int_0^{\pi} \int_a^b T(r, \varphi) r \, dr \, d\varphi \tag{15}$$

is calculated iteratively from the equation

$$\bar{T} = D_0 \frac{x_1 T_r k(T_r)}{k(\bar{T})} \left(\frac{u^2}{u^2 - 1} \ln u - \frac{1}{2} + \frac{k(\bar{T})}{x_2 k(T_r)} \right) + x_3 T_r. \tag{16}$$

Thus, the resulting expressions for the dimensional parameters are

$$q_p = x_1 \frac{k(\bar{T}) T_r}{b}, \quad h = x_2 \frac{k(\bar{T})}{a},$$

$$T_f = x_3 T_r.$$

2.2. *Determination of the circumferential distribution of heat flux absorbed by smooth tubes*

On the basis of the temperature measured at the wall interior, the wall temperature and heat flux distribution on the outside of the tube can be determined by means of a developed method. In fossil-fired boilers the heat is supplied to only one side of the smooth tubes. The one-sided heating can also be simulated by means of resistance heating of silver plating tube (over one half of the tube circumference [19]). The other possibility to achieve nonuniform heating of the tube is the application of a semicircumferential heating system with a silicon carbide heater [20] (Fig. 5). The test tube is radiantly heated on its circumference while being covered on its opposing semicircumference with ceramic wool and fire bricks [20]. The test tube is either a test tube for heat flux measurement or a sample tube for measurements of thermal conductivity of scale deposits on its inside surface.

In such situations [19, 20], the heat flux distribution on the outside of the tube can be described fairly well by (see Fig. 5)

$$q(\varphi) = q_p, \quad 0 \leqslant \varphi \leqslant \pi/2$$

$$q(\varphi) = q_p \exp\left[\frac{1}{x_4}\left(1 - \frac{2\varphi}{\pi}\right)\right],$$

$$\frac{\pi}{2} \leqslant \varphi \leqslant \pi \tag{17}$$

where x_4 is a constant.

In this case q_p represents the heat flux at the outside tube surface. The Fourier representation of the function $q(\varphi)$ known throughout the interval 0 to π is

$$\frac{q(\varphi)}{q_p} = D_0 + \sum_{m=1}^{\infty} D_m \cos(m\varphi) \tag{18}$$

where the coefficients are given by the expressions

$$D_0 = \frac{1}{2} + \frac{1}{2} x_4 \left[1 - \exp\left(-\frac{1}{x_4}\right)\right]$$

and

$$D_m = \frac{2}{\pi}\left\{\frac{1}{m}\sin\left(m\frac{\pi}{2}\right) + \frac{\pi^2 x_4^2}{4 + \pi^2 x_4^2 m^2} \cdot \left[\frac{2}{\pi x_4}\cos\left(m\frac{\pi}{2}\right) - \frac{2}{\pi x_4}\exp\left(-\frac{1}{x_4}\right)\cos(m\pi) - m\sin\left(m\frac{\pi}{2}\right)\right]\right\},$$

$$m = 1, 2, 3, \ldots. \tag{19}$$

The expression for the temperature distribution in the tube wall is still of the form of equation (10). To determine the parameters x_1, x_2, x_3, and x_4 on the basis of measured temperatures y_1, y_3, and y_4, the system of equations (1) for $n = 4$ must be solved. Starting with trial values of $x_1^{(1)}$, $x_2^{(1)}$, $x_3^{(1)}$ (equations (11), (12) and (14)) and

$$x_4^{(1)} = \frac{\pi - 2\varphi_4}{\pi \ln\left(\frac{Y_4 - x_3^{(1)}}{x_1^{(1)}} \frac{x_2^{(1)}}{x_2^{(1)} \ln R_4 + 1}\right)} \tag{20}$$

iterations (2) are carried out.

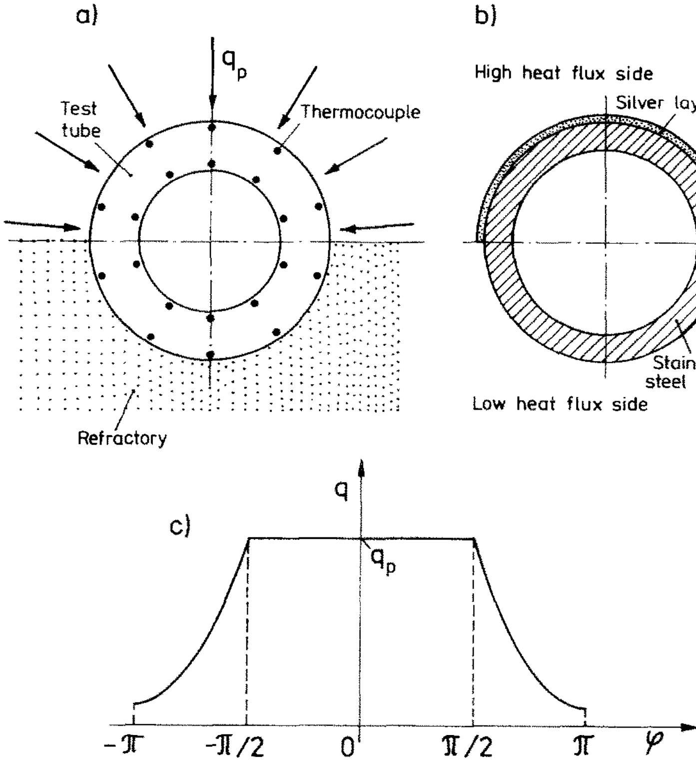

FIG. 5. Evaporator tubes with circumferential non-uniform heating; (a) radiant heating; (b) electric resistance heating; (c) model for the circumferential distribution of the heat flux.

3. PRACTICE

Two applications, corresponding to the analyses of Sections 2.1 and 2.2, are presented to illustrate the effectiveness of the present method. The first numerical example was selected to evaluate the performance of the technique in determining the heat flux to the finned tube walls. In the second example, the results of Section 2.2 were applied to a set of experimental data obtained by Ishikawa *et al.* [20].

3.1. *Determination of the local heat flux to the membrane wall of the boiler furnace*

In order to demonstrate the application of the developed method a typical finned tube wall will be investigated for the following conditions: inside radius ($a = 10$ mm), outside radius ($b = 16$ mm), fin thickness ($s = 6$ mm), tube pitch ($t = 46$ mm) and thermal conductivity ($k = 44$ W m^{-1} K^{-1}). For a specific application of the presented method, the following measured temperatures were used:

$$y_1 = 428.1°C \quad \text{at} \quad r_1 = 0.0157\,\text{m}, \varphi = 0$$

$$y_2 = 392.7°C \quad \text{at} \quad r_2 = 0.0117\,\text{m}, \varphi = 0$$

and

$$y_3 = 352.4°C \quad \text{at} \quad r_3 = 0.016\,\text{m}, \varphi = \pi.$$

The results are reported below

$$q_p = 349\,997\,\text{W m}^{-2}, h = 20\,000\,\text{W m}^{-2}\,\text{K}^{-1}$$

$$\text{and} \quad T_f = 350°C.$$

The calculated temperatures at measurement points are

$$T_1 = 428.2°C, T_2 = 392.7°C$$

$$\text{and } T_3 = 352.4°C.$$

The tolerance $e = 1 \times 10^{-4}$ was achieved after $k = 5$ iterations. An estimated temperature distribution pattern across a tube section is shown in Fig. 6. On account of the symmetry the isotherms are depicted only for one half of the cross section.

3.2. *Application of the method to the smooth tube heated radiantly on its semicircumference*

In the second example, the method is applied to actual temperature data taken from an experiment conducted by Ishikawa *et al.* [20].

The experiment was performed with a radiantly heated smooth tube in order to determine the thermal conductivity of scale deposits under boiling conditions typical for supercritical boilers. The cross section of the test tube is shown in Fig. 5. The austenitic steel tube (SUS 304 steel), with an outside diameter of 31.8 mm and a wall thickness of 6.5 mm, was equipped with ten thermocouples which measured the outer surface temperature profile at an interval of 36°. Ten other thermocouples were attached to the inside wall of the tube opposite the corresponding, outer wall thermocouples. The outside and inside wall temperatures were measured at two radial locations,

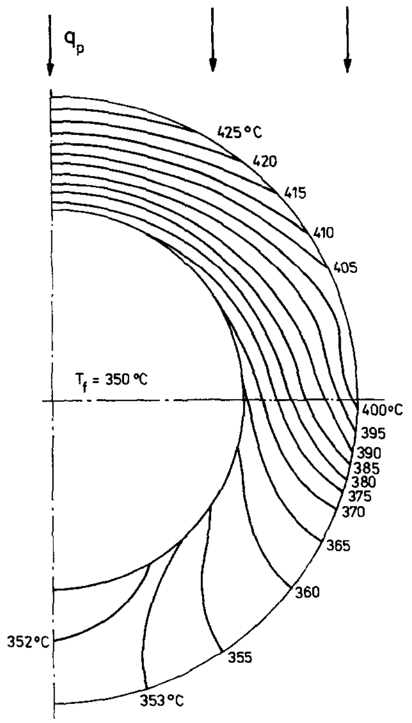

FIG. 6. Temperature distribution in the furnace wall tube.

$r_1 = 15.4$ mm and $r_2 = 9.9$ mm, respectively. The thermal conductivity of tube material varies with temperature

$$k = 14.65 + 0.0144T \text{ W m}^{-1} \text{K}^{-1}$$

where T is in °C.

The calculations were carried out for three different thermocouple locations (Fig. 7).

An example of the circumferential wall temperature distribution of the test tube at 20 points is shown in Fig. 8 [19]. As can be seen from the figure, there is no significant difference between measurements at symmetric angles on the left and right portions of the test tube. The parameters x_1, x_2, x_3, and x_4 were

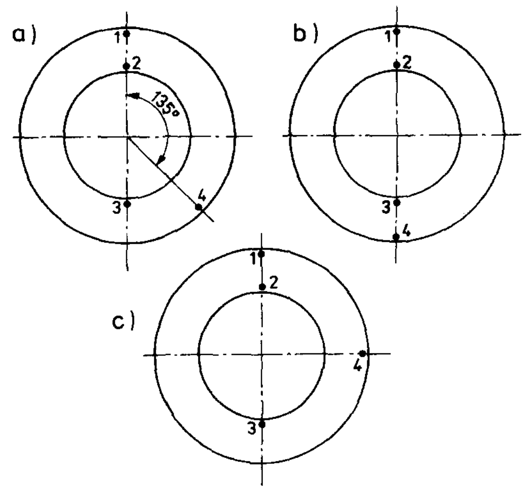

FIG. 7. Analysed locations of temperature sensors in the test tube.

determined from the experimental curves of the circumferential temperature distribution, illustrated in Fig. 8. A cosine series was used to approximate the measured tube wall temperatures. The analyses were conducted to determine whether there is a significant difference in the determined parameters, when the location of the fourth thermocouple changes. The fluid temperature was assumed to be 100°C, because its value is not specified in ref. [20]. This value of the fluid temperature was added to the temperature difference shown in Fig. 8. The temperatures obtained in this way, were used as the experimental data.

The results of the application of the present method to the data given are shown in Table 1.

An examination of Table 1 shows that the determined parameters are not significantly affected by the location of the fourth thermocouple. The calculated circumferential temperature distribution of the tube temperature at locations $r_1 = 15.4$ mm and $r_2 = 9.9$ mm are in good agreement with the measured values (Figs. 9–11).

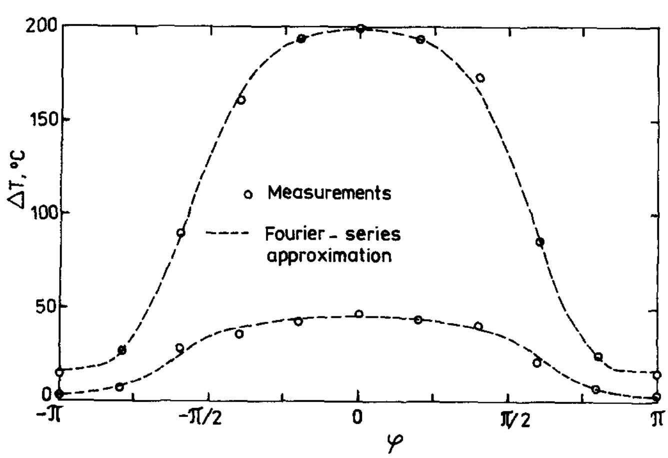

FIG. 8. Variation of the measured temperatures over the tube circumference at locations r_1 and r_2.

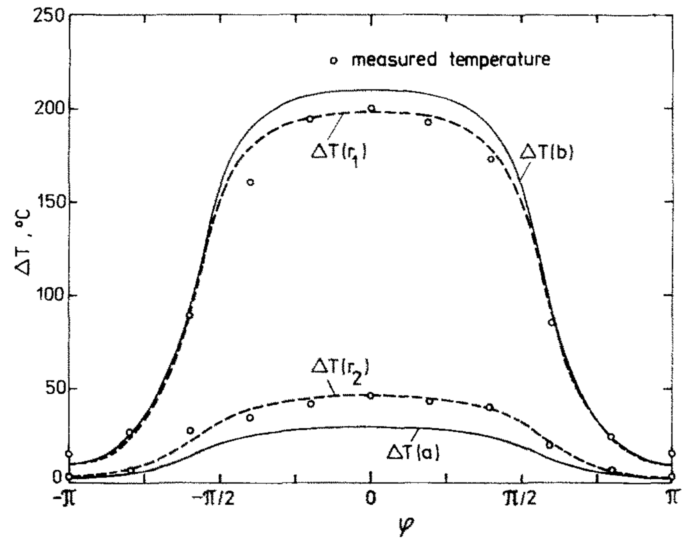

FIG. 9. Results for the circumferential distribution of the temperature (case (a) in Fig. 7).

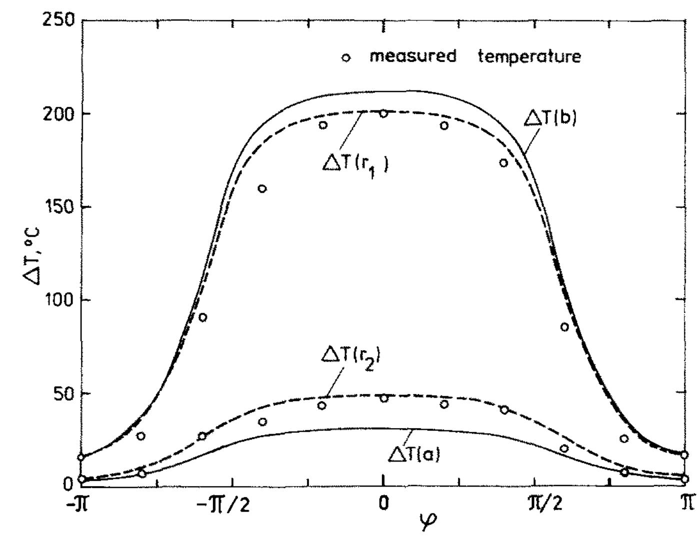

FIG. 10. Results for the circumferential distribution of the temperature (case (b) in Fig. 7).

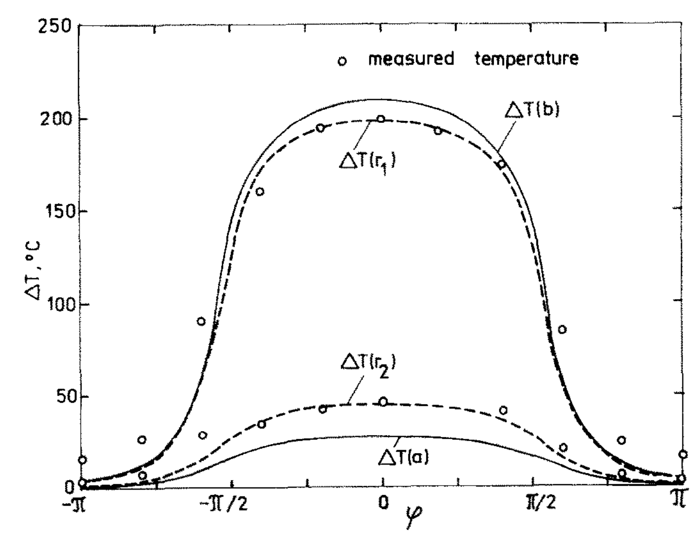

FIG. 11. Results for the circumferential distribution of the temperature (case (c) in Fig. 7).

Table 1. Experimental temperature data and calculated parameters for the test tube

	Case		
	(a)	(b)	(c)
φ_4	$3\pi/4$	π	$\pi/2$
y_1 [°C]	299.4	299.4	299.4
y_2 [°C]	146.7	146.7	146.7
y_3 [°C]	103.7	103.7	103.7
y_4 [°C]	134.5	116.1	232.1
q_p [W m^{-2}]	376 855	378 299	375 349
h [W m^{-2} K^{-1}]	22 148	20 568	23 460
x_4	0.1895	0.2628	0.0850
T_f [°C]	100.5	98.1	102.3

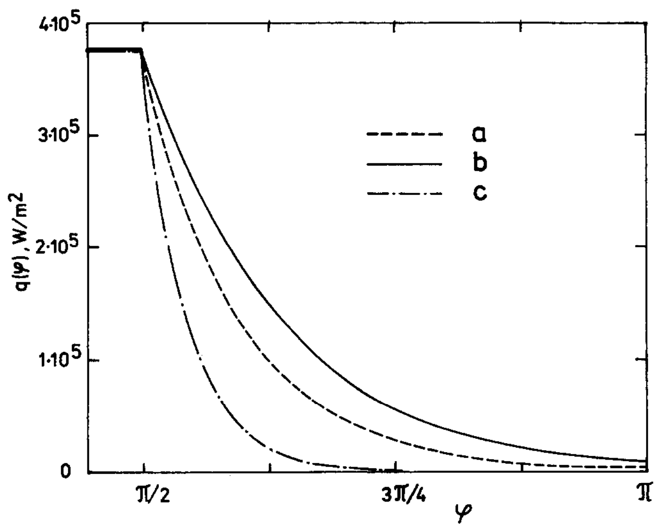

FIG. 12. Results for the circumferential distribution of the heat flux on the rear side of the test tube.

The numerical results for the outside and inside surfaces of the tube are also presented graphically in Figs. 9–11. The heat flux distribution on the rear side of the tube is shown in Fig. 12. Inspection of the computational results illustrates that it can hardly be satisfactory for φ_4 to be $\pi/2$ (case c), since the tube temperature at that angular location is less sensitive to x_4 than in the two other cases. This is illustrated by the results in Table 1 and Fig. 12.

4. CONCLUSION

The numerical method has been developed for the measurement of heat flux in boiler furnaces, which utilizes the tube temperature data from interior thermocouples.

It is suited both to membrane construction walls and to waterwalls from smooth tubes.

Further, the method can be used to determine inside heat transfer coefficient and water/steam temperature.

The above-mentioned parameters are calculated from the governing system of nonlinear algebraic equations using the Newton–Raphson method. Numerical examples are presented as verification of the method.

The comparison of the calculated values with the experimental data shows good agreement.

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MÉTHODE DE DÉTERMINATION DU FLUX DE CHALEUR LOCAL SUR LES SURFACES DE CHAUFFE DES CHAUDIÈRES

Résumé—L'objet de cet article est une méthode de la détermination des flux de chaleur locaux sur les surfaces de chauffe dans les foyers des chaudiers. Elle nécessite de connaître les lois de températures dans des points intérieurs des tubes-écrans. Celles températures sont fonction du flux de chaleur absorbé et utiles pour la détermination de celui-ci. La méthode permette de déterminer aussi le coefficient d'échange sur la surface intérieur du tube et la température du mélange eau–vapeur. Elle se caractérise par une bonne précision. Les paramètres mentionnés ci-dessus sont obtenues en utlisant la méthode Newton–Raphson pour la résolution d'un système des équations algébriques non linéaires. Les résultats des calculs sont comparés avec ceux des études expérimentales.

METHODE ZUR BESTIMMUNG DER LOKALEN WÄRMESTROMDICHTEN IN FEUERRÄUMEN VON DAMPFERZEUGERN

Zusammenfassung—In der vorliegenden Arbeit wurde ein Verfahren zur Bestimmung der lokalen Wärmestromdichten in Feuerräumen von Dampferzeugern auf der Basis der in inneren Punkten gemessenen Rohrwandtemperaturen dargestellt. Die gemessenen Rohrwandtemperaturen hängen von der aufgenommenen Wärmestromdichte ab und können deshalb zu ihrer Bestimmung dienen. Außerdem können der innere Wärmeübergangskoeffizient und die Temperatur des Wasser/Dampf Gemisches mit Hilfe des entwickelten Verfahrens genau ermittelt werden. Die oben genannten Grössen ergeben sich aus der Lösung eines Systems von nichtlinearen algebraischen Gleichungen nach der Methode von Newton–Raphson. Zur Überprüfung des Verfahrens wurden die berechneten Resultate mit den experimentallen Daten verglichen.

МЕТОД ОПРЕДЕЛЕНИЯ ЛОКАЛЬНОГО ТЕПЛОВОГО ПОТОКА В ТОПКАХ КОТЛОВ

Аннотация—Описывается метод определения локального теплового потока в топках котлов по распределению температур на различных участках инструментальных труб. Предложенный метод учитывает такие факторы, как отложение накипи, теплопроводность стенки в продольном направлении и ее зависимость от температуры. Метод может также использоваться для определения коэффициента теплообмена и температуры воды (водяного пара).