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# DETERMINING THE THICKNESS OF THE BOILER SCALE IN THE PIPE OF A HEAT EXCHANGER BASED ON THE SOLUTION OF THE INVERSE PROBLEM

The temperature inside the heat exchanger pipe in a steam boiler was subjected to analysis. Heating of the pipe with and without the scale was considered. A calculation model was presented describing the inverse problem of a geometrical type. It allows determining the thickness of the boiler scale based on the measurements of the temperature inside the pipe and the heat flow density on t he outer wall of the heat exchanger pipe. The paper analyzes the sensitivity of the obtained temperature distributions in the pipe and in the scale. The temperature measurement error, the inaccuracy of the thermocouple fitting and the measurement error of the heat flow density on the pipe outer wall were taken into account. The thickness of the boiler scale was determined depending on its properties and thermal load of the exchanger element. The calculations were performed for the scales of the heat conductance $\lambda$ from 0.3 to 1 W/mK. The proposed calculation model corresponds to a non-linear problem, in which the change of the heat conductance coefficient was taken into account depending on the temperature.

Key words: boiler scale, inverse problem, steam boilers

## 1. INTRODUCTION

In the components of thermal machines and equipment it is possible to determine the temperature distributions by measuring the temperature in the inner points

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of the object and solving the inverse problem [Ciałkowski 1996]. This method is applied in the analysis of the heat flow in heat exchangers, steam boilers [Taler 1995] or turbine vanes [Frąckowiak , Wolfersdorf, Ciałkowski 2011]. During operation of water boilers, boiler scale deposits on the surface of the boiler components, which results in an increased flow resistance as the thickness of the scale increases. The determination of the scale thickness and its influence on the temperature distribution in the pipe depending on the heat flow through the pipe wall becomes significant. Analyses describing the actual operating conditions of boilers become important when the thermal loads of their heated surfaces grow. Works on the actual conditions of operation of heaters [Urbaniak, Bartoszewicz, Kłosowiak 2015] provide data allowing a validation of the designers' approach to newly developed designs [Jaworski 2015]. A modernization of heaters, hearths and heated components is carried out by experimental research [Urbaniak, Bartoszewicz, Kłosowiak 2015], design calculations including inverse problems [Joachimiak, Ciałkowski, Bartoszewicz 2014, Krajewska, Ciałkowski, Bartoszewicz 2011] and the analysis of causes of malfunctions [Bartoszewicz 2011]. These analyses include the combustion processes and the analysis of the transport of mass momentum and energy in the area of limited flows (boilers) [Bartoszewicz, Bogusławski 2016, Bartoszewicz, Kłosowiak and Bogusławski 2012, Jaworski 2015]. Attempting to reduce the size of a boiler, hence the materials used, results in a series of effects leading to boiler damage. Some of the fundamental reasons for boiler malfunction in the non-professional applications are design shortcomings and non-compliance with the technical specifications regarding the chemical composition of water. These causes are the reasons for premature wear of low and medium power boilers. One of the methods to find the causes of the malfunction and prevent it are analyses of the inverse problems allowing a reproduction of the cause of a malfunction or premature wear. In [Taler et al. 2009] the results of experimental and numerical research were presented related to monitoring of the steam boiler operation. An inverse method was developed allowing the assessment of the following parameters: heat flow, heat transfer coefficient on the inner pipe wall and the temperature of the steam-water mixture. The solution of an inverse problem for the equation of heat conductance in the heat flow analysis of a steam boiler has been presented in [Duda, Taler 2009]. The method of research presented therein may serve as an analysis of the scale deposit on the pipe inner wall and the dust deposits in the flue ducts. The solution of an inverse problem of a geometrical type related to the deposition of boiler scale has been presented in [Joachimiak, Ciałkowski, Bartoszewicz 2014, Krajewska, Ciałkowski, Bartoszewicz 2011]. The presented calculation model allowed determining the thickness of the boiler scale, at its assumed heat conductance coefficient following the temperature measurement inside the pipe with the scale and the heat flow density on the pipe outer wall. In these works, the heat flow was analyzed for pipes with carbonate, sulfate and silicate scales. For the analyzed types of scale, a heat conductance coefficient of 6, 2.5 and 0.3 W/mK was adopted respectively. [Joachimiak, Ciałkowski, Bartoszewicz 2014] describes the influence of the placement of a thermocouple and the accuracy of the heat flow density measurement on the pipe outer wall on the temperature distribution inside the pipe and on the scale. The sensitivity of the solution of temperature distribution to inappropriate fitting of the thermocouple and temperature measurement was also taken into account. In this paper, the inverse problem of the geometrical type was also described for the heated pipe with boiler scale. Scales of the heat conductance coefficient of 0.3 to 1 W/mK were analyzed. This paper is an extension of publication [Joachimiak, Ciałkowski, Bartoszewicz 2014]. This paper includes a calculation model related to the pipe heating with boiler scale and without it. The thickness of the boiler scale was analyzed depending on the heat conductance coefficient of the deposit, heat load of the pipe and the reading of the thermocouple placed in the steel pipe. The obtained temperature distributions in the pipe with the deposit were referred to those obtained for the heating of the pipe without the deposit. The obtained results are significant because of the changing scale thickness during boiler operation.

## 2. CALCULATION MODEL

### 2.1. Without deposit

For the stationary equation of heat conductance

$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} = 0 \tag{1}$$

the solution for the pipe layer assumes the form

$$T = C_1 \ln r + C_2 \quad \text{for} \quad r_i \le r \le r_o \tag{2}$$

in order to determine the $C_1$ and $C_2$ constants the following conditions were included:

– heat flow density on the pipe outer wall

$$\dot{\ } \qquad \left.\frac{\partial T}{\partial r}\right|_{r=r_o} \tag{3}$$

– heat flow density on the pipe inner wall

$$\dot{\ } \qquad {}_{P}\left.\frac{\partial T}{\partial r}\right|_{r=r_i} = \alpha_f\left[T\left(r=r_i\right) - T_f\right] \tag{4}$$

The radiuses of the inner wall $r_o$ and outer wall $r_i$ of the pipe were assumed (Fig. 1) as well as the heat conductance coefficient $\lambda_p$ for the pipe material. The measurement of the heat flow density on the pipe outer wall $\dot{\ }$ (on the hot flue side), water temperature in the pipe $T_f$ and the heat transfer coefficient $\alpha_f$ on the pipe outer wall (water side) were assumed.

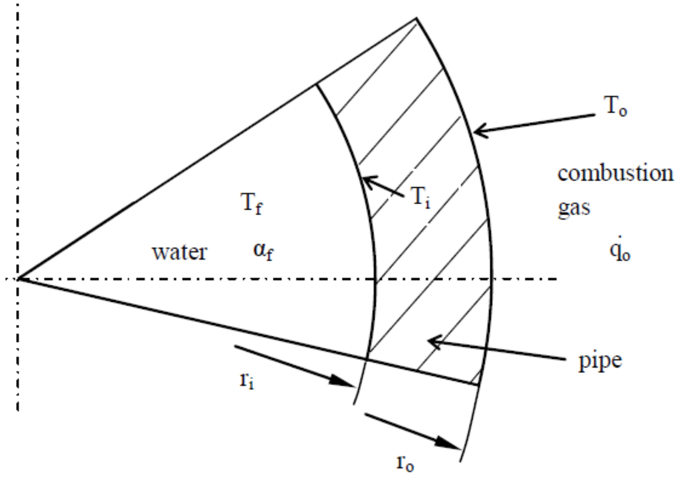


Fig. 1. Cross section scheme of a pipe

Upon considering the form of solution (2) from conditions (3)–(4) the following equalities were obtained

$$\ldots \frac{C_1}{r_o}, \quad -\lambda_p \frac{C_1}{r_i} = \alpha_f \left[ C_1 \ln r_i + C_2 - T_f \right] \tag{5}$$

Hence

$$C_1 = -\frac{\ldots}{\lambda_p}, \quad C_2 = \frac{\alpha_f T_f + \frac{\ldots}{\lambda_p}\left( \frac{\ldots}{r_i} \ldots \alpha_f \ln r_i \right)}{\alpha_f} \tag{6}$$

## 2.2. Allowing for the boiler scale

Allowing for the boiler scale deposited on the inner side of the pipe leads to a heat flow through two layers, on the boundary of which compatibility conditions occur. For the stationary equation of heat conductance

$$\frac{d^2T}{dr^2} + \frac{1}{r}\frac{dT}{dr} = 0 \tag{7}$$

the solution for the pipe layer assumes the form

$$T = C_1 \ln r + C_2 \quad \text{for} \quad r_i \le r \le r_o \tag{8}$$

and for the scale it assumes the form

$$+ D_2 \quad \text{for} \quad r_d \le r \le r_i \tag{9}$$

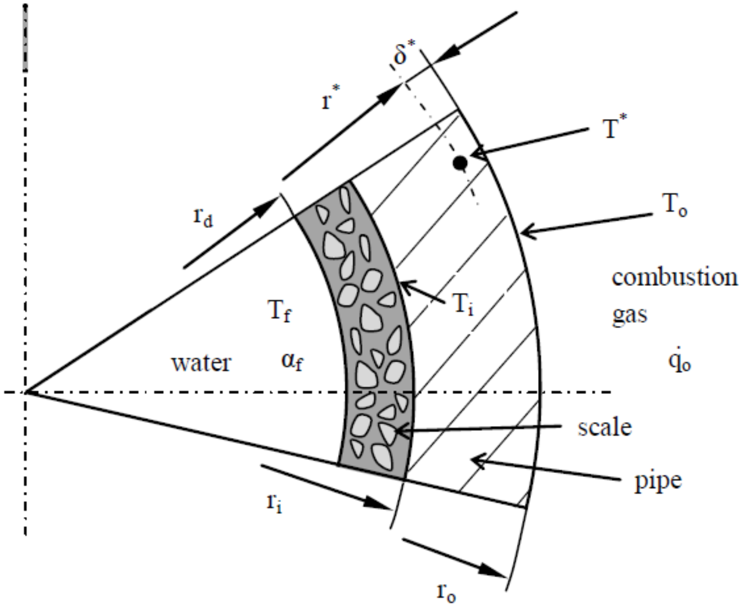


Fig. 2. Cross section scheme of a pipe with boiler scale [Joachimiak, Ciałkowski and Bartoszewicz 2014]

In order to determine constants $C_1$, $C_2$, $D_1$ and $D_2$ the following conditions were adopted:

– heat flow density on the pipe outer wall

$$\left.\frac{\partial T}{\partial r}\right|_{r=r_o} \tag{10}$$

– measurement of the temperature inside the pipe on radius $r^*$ (at a distance of $\delta^*$ from the pipe outer edge)

$$T\left(r=r^*\right)=T^* \tag{11}$$

– heat flow density at the contact point of the scale with the pipe

$$\left.\frac{\partial T}{\partial r}\right|_{r=r_i} = -\lambda_d \left.\frac{\partial}{\partial r}\right|_{r=r_i} \tag{12}$$

– temperature at the contact point of the scale with the pipe

$$T\left(r=r_i\right)= \tag{13}$$

Based on expressions (10)–(13) an the form of solution (8)–(9) of the equation (7) the following were obtained

$$\ldots, \; T^* = C_1 \ln r^* + C_2, \; \lambda_p \frac{C_1}{r_i} = \lambda_d \frac{D_1}{r_i}, \; C_1 \ln r_i + C_2 = D_1 \ln r_i + D_2 \tag{14}$$

Based on equations (14) the constants were determined

$$C_1 = \frac{\cdot}{\lambda_p}, \; C_2 = T^* - \frac{\cdot}{\lambda_p} \ldots, \; D_1 = \frac{\cdot}{\lambda_d}, \; D_2 = T^* + \cdot \left( \frac{\ln\left(\frac{r_i}{*}\right)}{\lambda_p} - \frac{\ln r_i}{\lambda_d} \right) \tag{15}$$

In the next stage of the calculations, the thickness of the boiler scale was determined based on the condition related to the heat flow density on the scale wall

$$\lambda_d \frac{\partial}{\partial r}\Big|_{r=r_d} = \alpha_f \left[ \quad - T_f \right] \tag{16}$$

Condition (16) based on the form of solution (9) takes a form

$$-\lambda_d \frac{D_1}{r_d} = \alpha_f \left[ D_1 \ln r_d + D_2 - T_f \right] \tag{17}$$

By substituting constants (15) a non-linear equation was obtained with unknown $r_d$ (on the radius of the contact of water and scale, Fig. 2) in the form

$$f(r_d) = \frac{\cdot}{r_d} \; \alpha_f \left[ T^* + \cdot \left( \frac{\ln\left(\frac{r_i}{*}\right)}{\lambda_p} + \frac{\ln\left(\frac{r_d}{r_i}\right)}{\lambda_d} \right) - T_f \right] = 0 \tag{18}$$

Equation (18) is a non-linear equation that will be solved with the Newton's method [Björck and Dahlquist 1983]. Radius $r_d$ was determined in the iterative way

$$r_{d,j+1} = r_{d,j} - \frac{f\left(r_{d,j}\right)}{f'\left(r_{d,j}\right)}, \; j = 0,1,\ldots \tag{19}$$

where index j denotes subsequent iterations of the solution, and the derivative of function f (18) is determined by the formula

$$f'(r_d) = \frac{\cdot}{r_d} \left( \frac{\cdot}{r_d} + \frac{\alpha_f}{\lambda_d} \right) \tag{20}$$

In the first iteration step, it was assumed that the radius, on which the deposit ends, equals the radius of the pipe inner wall $r_{d,0} = r_i$. The following condition of calculation completion was adopted

$$\left| r_{d,j+1} - r_{d,j} \right| < 10^{-12} \tag{21}$$

## 2.3. Sensitivity of the solution

The temperature distribution error in the pipe and in the scale was estimated with the inequality

$$\left|\Delta T(r)\right| \le \left|\frac{\partial T}{\partial T^*}\right|\left|\Delta T^*\right| + \left|\frac{\partial T}{\partial r^*}\right|\left|\Delta r^*\right| + \left|\frac{\partial T}{\partial \dot{[illegible]}}\right| [illegible] \tag{22}$$

$$\left|\Delta \tilde{[illegible]}\right| [illegible] \left|\frac{\partial \tilde{T}}{\partial T^*}\right|\left|\Delta T^*\right| + \left|\frac{\partial \tilde{T}}{\partial r^*}\right|\left|\Delta r^*\right| + \left|\frac{\partial \tilde{T}}{\partial \dot{[illegible]}}\right| [illegible] \tag{23}$$

where $\Delta T^*$ denotes the pipe temperature measurement error, $\Delta r^*$ thermocouple fitting error and $\Delta \dot{[illegible]}$ heat flow density measurement error in the pipe inner wall. Based on the solution form of temperature distribution in the pipe (8), in the scale (9) and the constants (15) inequalities were obtained

$$\left|\Delta T(r)\right| \le \left|\Delta T^*\right| + \frac{\dot{[illegible]}}{\lambda_p r^*} [illegible] \frac{r_o}{\lambda_p}\ln\left(\frac{r}{r^*}\right)\left|\Delta \dot{[illegible]}\right| \tag{24}$$

$$\left|\Delta \tilde{[illegible]}\right| [illegible] \left|\Delta T^*\right| + \frac{\dot{[illegible]}}{\lambda_p r^*} [illegible] r_o\left(\frac{\ln r}{\lambda_d} + \frac{\ln\left(\frac{r_i}{r^*}\right)}{\lambda_p} - \frac{\ln r_i}{\lambda_d}\right)\left|\Delta \dot{[illegible]}\right| \tag{25}$$

## 2.4. Extension of the calculation model to a non-linear problem

For the heat conductance equation

$$\operatorname{div}\left(\lambda(T)\nabla T\right) = 0 \tag{26}$$

Kirchhoff's substitution principle was applied [Gdula 1984]

$$\vartheta = \int_{T_0}^{T} \lambda(u)\,du \tag{27}$$

where $T_0$ is a constant. Upon differentiating the equalities (27) the following was obtained

$$\frac{\partial \vartheta}{\partial x} = \lambda(T)\frac{\partial T}{\partial x} - \lambda(T_0)\frac{\partial T_0}{\partial x} \tag{28}$$

Therefore, taking into account $\frac{\partial T_0}{\partial x} = 0$ we have

$$\nabla \vartheta = \lambda(T)\nabla T \tag{29}$$

By substituting (29) to equation (26) we obtain

$$\mathrm{div}\left(\lambda\left(T\right)\nabla T\right)=\mathrm{div}\left(\nabla\vartheta\right)=\Delta\vartheta=0 \tag{30}$$

Taking the relation of the heat conductance coefficient and the temperature (27) into account, the analyzed non-linear problem gets down to an equation, whose form has been shown in chapter 2.3.

## 3. RESULTS OF CALCULATIONS

The data adopted in the calculations related to the pipe geometry ($r_o, r_i, r^*$), the heat transfer coefficient on the water side ($\alpha_f$), the water temperature ($T_f$), the heat flow density on the pipe outer wall ($\dot{q}$), the heat conductance coefficients of the pipe ($\lambda_p$) and the scale ($\lambda_d$), the temperature measured in the pipe ($T^*$) and the accuracies of the measured ($\Delta T^*, \Delta\dot{q}$) and estimated values ($\Delta r^*, \Delta\alpha_f$) have been presented in table 1.

Table 1. Data adopted for the calculations

| | | | | | |
|---|---|---|---|---|---|
| $r_o$ | 0.025 | $\dot{q}$ | 20, 30, 40, 50 | $\Delta T^*$ | 2 |
| $r_i$ | 0.017 | $\lambda_p$ | 50 | $\Delta r^*$ | 0.0005 |
| $r^*$ | 0.023 | $\lambda_d$ | 0.3, 0.5, 0.7, 1 | $\Delta\dot{q}$ | 1000 |
| $\alpha_f$ | 6000 | $T^*$ | 500, 510, 520, 530, 540, 550, 560, 570, 578, 580 | $T_f$ | 120 |

During boiler operation, due to inappropriate chemical composition of water, boiler scale deposits on the water-steam side of the heated space. A deposit of the thickness of $\delta_d = 1$ mm results in a significant increase in the temperature at the point of measurement to the value of approx. 219 ºC, 268 ºC, 318 ºC and 368 ºC for the heat flow density on the pipe outer wall of 20, 30, 40 and 50 kW/m$^2$ respectively for the scale of the lowest heat transfer coefficient $\lambda_d = 0.3$ W/mK (Fig. 3a). For the pipe without a deposit, the measured temperature does not exceed 125 ºC. The temperature of the pipe outer wall $T_o = 580\,^{o}C$ and the temperature measured inside the pipe $T^* = 578\,^{o}C$ at a distance of $\delta^* = 2$ mm from the edge of the pipe were assumed as values, for which the heated element is damaged. The values of the measured temperatures $T^*$ for the scale of the thickness of $\delta_d = 2$ mm (Fig. 3b) and $\delta_d = 3$ mm (Fig. 3c) reach 637 ºC and 922 ºC.

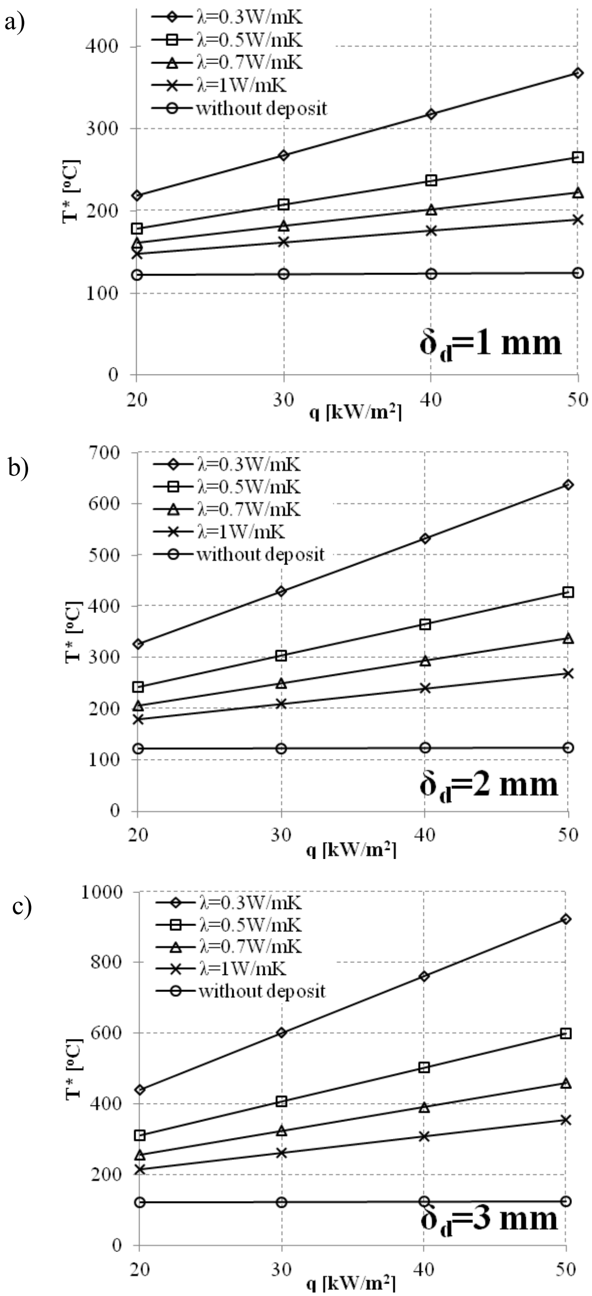


Fig. 3. The temperature values recorded at a point 2 mm from the edge of the pipe for the heat flow density from 20 to 50 kW/m$^2$ for the scale of the thickness: a) $\delta_d$ = 1 mm b) $\delta_d$ = 2 mm, c) $\delta_d$ = 3 mm and the scale of the heat transfer coefficient $\lambda_d$ = 0.3, 0.5, 0.7 and 1 W/mK

For the scale of the heat transfer coefficient $\lambda_d = 0.3$ W/mK , the deposit below 2 mm results in damage of the heat exchanger pipe. For $\lambda_d = 0.5$ W/mK the temperature $T^* = 578\,^{\circ}$C is exceeded when less than 3 mm of the boiler scale deposits. From the point of view of the boiler operation, the results show the significance of the knowledge on the thickness and properties of the boiler scale. Fig. 4 presents the thickness of the boiler scale depending on the thermocouple $T^*$ reading for the scale of the heat transfer coefficient from 0.3 to 1 W/mK and for the heat flow density on the pipe outer wall ˙ W/m$^2$. For the thermocouple reading $T^* = 578\,^{\circ}$C and $\lambda_d = 0.3,\ 0.5,\ 0.7,\ 1$ W/mK the thickness of the boiler scale is 2.9, 4.5, 6 and 7.9 mm respectively (Fig. 4).

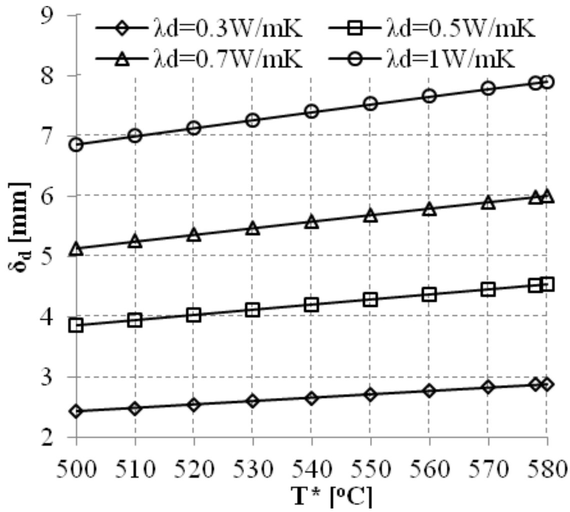


Fig. 4. Thickness of the boiler scale depending on the reading of the thermocouple for the heat conductance coefficient $\lambda_d$ = 0.3, 0.5, 0.7 and 1 W/mK and the heat flow density on the pipe outer wall ˙ W/m$^2$

The distribution of temperature in the pipe and in the scale for the thermocouple reading $T^* = 578\,^{\circ}$C, the heat conductance coefficient $\lambda_d = 0.5$ W/mK and the heat flow density on the pipe outer wall ˙ from 20 to 50 kW/m$^2$ has been shown in figure 5. The temperature of the pipe assumes values from 570 to 580 ºC and the scale from 105 to 575 ºC. The drop of temperature in the pipe and in the scale is the fastest for the highest heat flow density ˙ W/m$^2$.

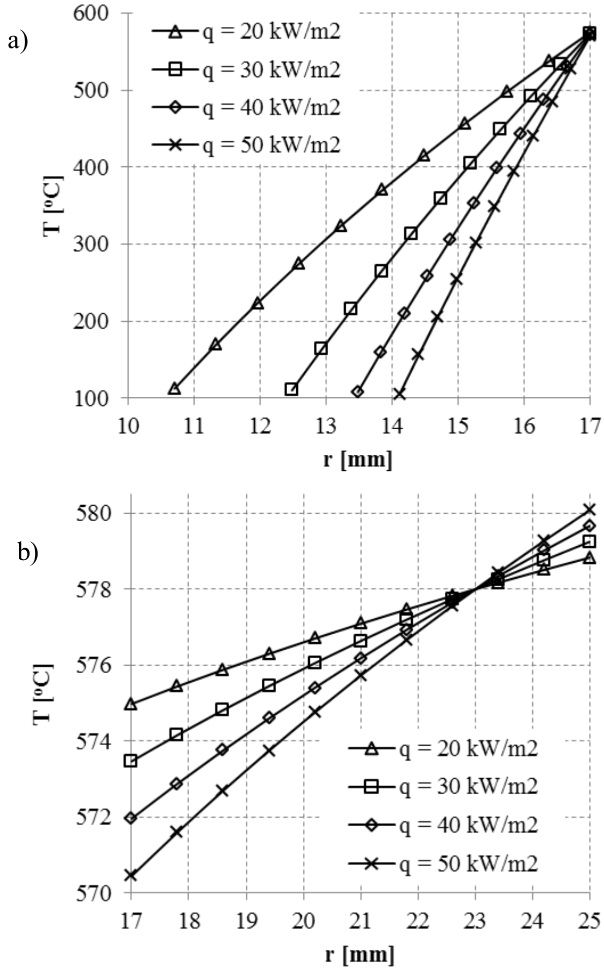


Fig. 5. Temperature distribution a) in the scale and b) in the pipe for [illegible] 20, 30, 40 and 50 kW/m$^2$ for the thermocouple reading $T^*$ = 578 °C and the heat conductance coefficient for the scale $\lambda_d$ = 0.5 W/mK

The temperature distribution in the pipe and in the scale, taking into account the sensitivity of the solution for $T^* = 578\,^{\circ}\mathrm{C}$, $\lambda_d = 0.5$ W/mK and [illegible]/m$^2$, has been presented in figure 6. The error of temperature determination on the edge of the pipe, where it is the highest, does not exceed 2.4 ºC. On the scale, this error reaches 17.9 ºC. As the heat conductance coefficient for the scale grows from 0.3 to 1 W/mK the thickness of the scale increases from 4.1 to 10.3 mm for the heat flow density on the pipe outer wall of 20 kW/m$^2$ and the thermocouple reading

578 ºC (Fig. 7). For ˙ ) oraz 50 kW/m$^2$ a growth in $\lambda_d$ results in an increased thickness of the scale from 2.9 to 7.9, from 2.2 to 6.3 and from 1.8 to 5.3 mm respectively.

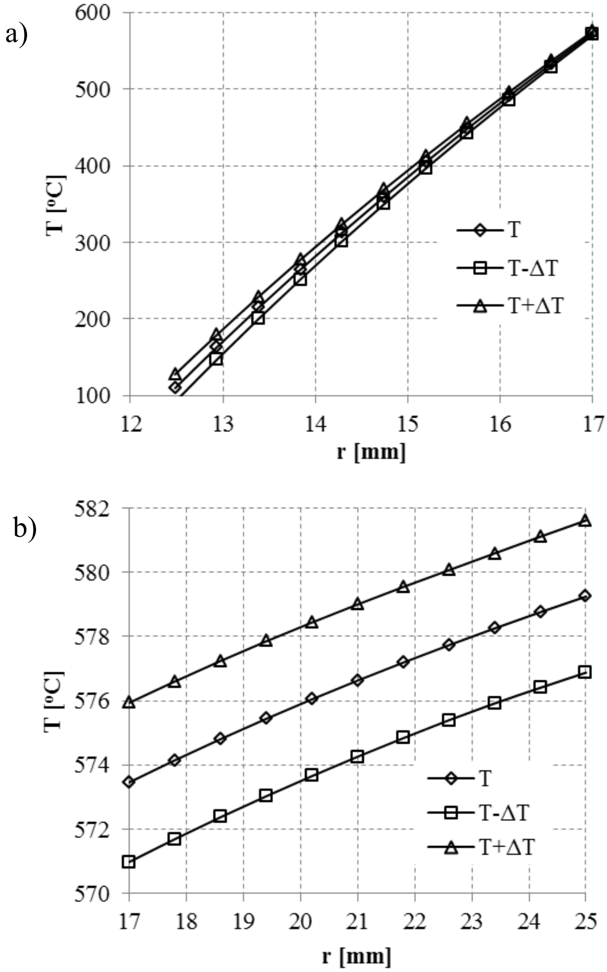


Fig. 6. Temperature distribution a) in the scale and b) in the pipe taking into account the sensitivity for the heat flow density on the pipe outer wall ˙ 30kW/m$^2$ for the thermocouple reading $T^* = 578$ ºC and the scale heat conductance coefficient $\lambda_d = 0.5$ W/mK

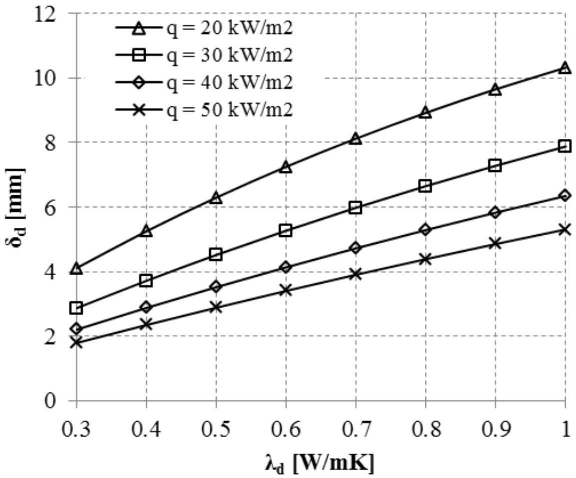


Fig. 7. Thickness of the boiler scale for the heat flow density on the pipe outer wall [illegible] 20, 30, 40 and 50 kW/m$^2$ depending on the value of the scale heat conductance coefficient $\lambda_d$ for the thermocouple reading $T^* = 578$ °C

## 4. CONCLUSIONS

A small growth in the scale thickness significantly influences the changes in the temperature of the heat exchanger pipe of a steam boiler. This results from the fact that the heat conductance coefficient for the scale is $\lambda_d = 0.3 \div 1$ W/mK while for the pipe material it is $\lambda_p = 50$ W/mK. The increase in the scale thickness is the most frequent cause of reduced durability of the heated surfaces of the boiler. The increase in the temperature of the steel wall leads to its reduced durability. As a consequence, the disadvantageous conditions resulting from the aerodynamic impact of the flue and water on the weakened wall of the boiler heated surface lead to its mechanical damage, the consequence of which is the boiler inoperativeness. As was indicated in the paper, the information on the thickness of the depositing scale is very important for proper boiler operation. The attempts to reduce the heat exchange area in the boiler must be accompanied by even greater efforts to provide water with appropriate technical parameters. The results obtained in the research work allow determining the safe admissible temperature of the boiler operation. The results have been obtained based on the geometrical solution of the inverse problem.

## NOMENCLATURE

$C_1$, $C_2$, $D_1$, $D_2$ – constants, [-]
$\dot{q}$ – heat flux density, [W/m$^2$]
r – radius, [m]
T– temperature in the pipe, [$^o$C]
$\tilde{T}$ – temperature in the deposit, [$^o$C]
Greek symbols:
α – heat transfer coefficient, [W/m$^2$K]
δ – thickness, [m]
λ – heat conduction coefficient, [W/mK]
Subscripts:
f – water
d – deposit
i – inner wall of the pipe
o – outer wall of the pipe
p – pipe
Superscripts:
* – location of the thermocouple

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## OKREŚLENIE GRUBOŚCI KAMIENIA KOTŁOWEGO W RURZE WYMIENNIKA CIEPŁA NA PODSTAWIE ROZWIĄZANIA ZAGADNIENIA ODWROTNEGO

### Streszczenie

W artykule zanalizowano rozkład temperatury w rurze wymiennika ciepła kotła parowego. Uwzględniono nagrzewanie rury z kamieniem kotłowym oraz bez osadu. Przedstawiono model obliczeniowy opisujący rozwiązanie zagadnienia odwrotnego typu geometrycznego. Pozwala on na wyznaczenie grubości kamienia kotłowego na podstawie pomiaru temperatury w rurze oraz gęstości strumienia ciepła na ściance zewnętrznej rury wymiennika ciepła. Zanalizowano wrażliwość uzyskiwanych rozkładów temperatury w rurze i kamieniu. Uwzględniono błąd pomiaru temperatury, niedokładność zabudowy termoelementu oraz błąd pomiaru gęstości strumienia ciepła na ściance zewnętrznej rury. Wyznaczono grubość kamienia kotłowego w zależności od jego właściwości oraz obciążenia cieplnego elementu wymiennika. Obliczenia wykonano dla kamieni o współczynniku przewodzenia ciepła $\lambda$ od 0.3 do 1 W/mK. Zaproponowany model obliczeniowy odpowiada również zagadnieniu nieliniowemu, w którym uwzględniono zmianę współczynnika przewodzenia ciepła w zależności od temperatury.

Słowa kluczowe: kamień kotłowy, zagadnienie odwrotne, kotły parowe