ANALYSIS OF THERMAL FLOW IN WATERWALL TUBES OF THE COMBUSTION CHAMBER DEPENDING ON THE FLUID PARAMETERS

by

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The article presents the method of determining the temperature distribution in waterwall tubes of the combustion chamber.
To simulate the operating conditions of waterwall tubes have been selected the model with distributed parameters, which is based on the solution of equations of the energy, mass and momentum conservation laws. The purpose of the calculations is determining the enthalpy, mass-flow and pressure of the working fluid flowing inside the tubes. The balance equations have been transformed into a form in which spatial derivatives are on the left, and the right side contains time derivatives. Then the time derivatives were replaced with backward difference quotients, and the obtained system of differential equations was solved by the Runge-Kutta method.
The analysis takes into account the variability of fluid parameters depending on the mass-flow at the inlet of the tube and heat flux on the surface of the tube. The analysis of fluid parameters was carried out based on operating parameters occurring in one of the Polish supercritical power plants. Then it was compared with characteristics for systems operating at increased or reduced thermal flux on the walls of the furnace or systems operating at increased or reduced mass-flow of the working fluid at the inlet to the waterwall tube.

Keywords: *heat transfer, combustion chamber, waterwall tube, unsteady flow, computational fluid dynamics*

Introduction

The need to meet the ever-increasing requirements for environmental protection and the growing demand for electricity forced the boiler manufacturers to significantly change the construction of the systems and increase the parameters of produced steam. In order to increase the efficiency of the power plant, are used superheater systems allowing generating supercritical steam, multi-stage turbine systems and capacitors operating at very low pressures. However, raising the parameters of the produced steam is associated with many technical problems - in systems must be used more and more durable materials, and in furnace of boilers, there are more heterogeneous thermal conditions. The main problems occurring in boilers operating on supercritical parameters are the high variability of the heat load on the

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width of the combustion chamber and the variability of the mass-flow of the working fluid flowing through the adjacent waterwall tubes. The occurrence of both of these phenomena results in the occurrence of large non-uniformity of the temperature of the working fluid in adjacent tubes. Monitoring of the temperature distribution in the tubes is very important because uneven heating of the tubes can lead to excessive thermal stresses in the boiler elements. Reducing the value of thermal stresses is, therefore, one of the most important problems that must be overcome to extend the service life and efficiency of boilers.

During the boiler start-up phase, waterwall tubes are exposed to very uneven conditions, resulting in high-value thermal stresses. The variability of thermal stresses in the tubes during operation after the start-up phase occurs from the transient conditions of the flow of the working fluid inside them and the uneven heat flux on the exchanger surface. The dependencies are allowing determining the pressure distribution and mass-flow of the working fluid, taking into account the type of the boiler, are presented in [1]. In this paper were presented the results of numerical calculations carried out using the Runge-Kutta method to solve a system of equations describing the behavior of energy, mass, and momentum. The calculations were carried out for the working fluid flowing inside the waterwall tubes of the furnace operating on supercritical parameters, with a steam capacity of $2400 \cdot 10^3$ kg/h.

High complexity and non-linearity of the processes occurring in the heating surfaces of the combustion chamber make it difficult to accurately and correctly simulate them. The complexity of the analysis is mainly increased by high pressure and temperature values, the variability of the mass-flow of the working fluid flowing inside the waterwall tubes, as well as the size of the heat exchange surface between the fluid and the combustion chamber. Analyzing the operating conditions of individual fragments of the combustion chamber walls may allow reducing the values of thermal stresses occurring in them, and thus to limit the amount of damage resulting from overheating of the boiler elements.

Some authors focus on analyzing the flow through individual waterwall tubes, taking into account the variability of the heat load over the height of the furnace. The articles [2-3] present the results of analyses carried out with the assumption of constant parameters of the factor at the inlet to waterwall tubes, as well as the comparison of simulation results with data obtained experimentally.

In numerous studies, it was found that changing the pitch of the vertical waterwall tubes, by the heat flux changing on the width of the combustion chamber, can reduce the thermal stresses in these tubes. Decreasing the value of thermal stresses and faster stabilization of the heat exchange process can also be obtained by using spirally-rifled tubes in power boilers. The comparison of the heat transfer performance for smooth tubes and internally rifled tubes with significantly larger internal surface is shown in [4-6]. The authors of the articles presented in them the comparison of the heat transfer performance in both types of tubes and compared the obtained results of numerical simulations with experimental data.

In turn, the authors of articles [7-9] focused on the analysis of working conditions of a fluidized bed boiler operating on supercritical parameters, in the case of a decrease in the mass-flow of the working fluid flowing through the system. In papers are presented the temperature distributions of the boiler elements and the fluid flowing through them depending on the mass-flow of the working fluid and the thermal load of the boiler.

Supercritical steam boilers are designed as Benson type once-through single-pass or double-pass boilers. The boiler with the Benson type evaporator consisting of vertical waterwall tubes gives the possibility to determine the mass-flow of the working fluid flowing through each of the waterwall tubes depending on the required heat load of the boiler. Its construction and working conditions are presented in [10]. An advantage of this variant is the possibility of reducing thermal stresses and hydraulic losses in the waterwall tubes, especially if the boiler is to be operated at a reduced mass-flow of the working fluid. In turn in boilers with a spiral arrangement of the tubes, each tube passes through all four walls of the combustion chamber, which strongly reduces the difference in length of tubes. Additionally, each tube can absorb a nearly equal quantity of heat, so that fluid with similar temperature is obtained at their outlets.

In the article [11] the problem of the analysis of the influence of the variable pitch of the vertical waterwall tubes on the boiler operation parameters was presented. The operation parameters and the heat load distribution were compared for the boiler in which evaporator was made of the 13CrMo4-5 steel – in one variant, the waterwall tubes were vertical, and in the other, the tubes were arranged spiral around the combustion chamber. To compare the efficiency of both systems, simulations of the temperature of the working fluid at the outlet of waterwall tubes in three different variants of the boiler's operation were presented.

During the modeling of the temperature distribution and thermal stresses in waterwall tubes, should be taken into account the dynamics of changes in the material parameters of tubes in response to a rapid shift in thermodynamic conditions – a step change in the temperature of the working fluid at the inlet to the tube or a quick change of the heat stream falling on the furnace wall. The solution to this problem, using a differential scheme in explicit and implicit form, is presented in [12].

In order to accurately analyze the operating conditions of combustion chamber walls, it is necessary to determine not only how change the parameters of the working fluid and the parameters of the steel from which the waterwall tubes are made, but also to determine the temperature distribution in these tubes and the fins between them. For this purpose, the authors of the publications [13-15] created a new model for dividing the cross-section of the boiler waterwall tube with the fin on the control volumes. The articles present the results of their work – temperature distributions in the cross section of waterwall tubes generated for parameters obtained in a supercritical boiler operating in one of the Polish power plants. The temperature distributions obtained with this method can be used to determine the distribution of thermal stresses in the walls of the furnace chamber.

Problem analysis

The article presents an analysis of the temperature distribution in a smooth waterwall tube, which is mounted on the furnace wall of a supercritical steam boiler. The proposed model allows, among others, to determine the fluid enthalpy, mass-flow and pressure distribution in relation to the height above the inlet to the tube. Parameters of the fluid flowing inside the tube have been determined in the model using the International Association for the Properties of Water and Steam (IAPWS IF97) parameters table [16].

General principles of the model allowing determining fluid parameters are presented in [6]. In order to increase the accuracy of calculations, we assume that the waterwall tube has been divided into infinitesimal control volumes.

In the following subsections, will be presented the equations describing the principles of mass, momentum, and energy conservation. Then, equations will be solved using the method proposed in [1].

Mass conservation law

The equation of mass balance for the control volume can be written:

$$\frac{\partial \rho}{\partial \tau} = -\frac{\partial}{\partial z}(\rho w) \text{ or } \frac{\partial p}{\partial \tau} = -\frac{1}{A}\frac{\partial \dot{m}}{\partial z} \tag{1}$$

where $\dot{m} = A\rho w$ denotes the mass-flow of the flowing working fluid.

Momentum conservation law

In order to correctly write the equation of momentum balance for the analyzed control volume, we must take into account not only the rate of changes of momentum accumulated in the control volume and the momentum flux flowing in and out of the control volume but also take into account all forces affecting the control volume. Among these forces, we can distinguish surface forces – forces exerted by pressure on the walls of the control surface, opposite to the normal to the inlet and outlet surfaces, and friction force on the surface of the tube. The force exerted by the wall on the fluid in the control volume is equal to the product of pressure p and the wall surface area ($A|_z=U|_z \cdot z$, where U denoted the circumference of the tube) – in the analyzed case, the force component in the axis z-direction is zero because $A|_{z-\Delta z/2}=A|_{z+\Delta z/2}$.

The friction force is directed opposite to the direction of the fluid flow and equals the product of the tangential stress σ_τ and wall surface area $A=U \cdot z$. The equation of momentum balance for the control volume can be written:

$$\frac{\partial \dot{m}}{\partial \tau} = -\frac{1}{A}\frac{\partial}{\partial z}\left(\frac{\dot{m}^2}{\rho}\right) - A\left(\frac{\partial p}{\partial z} + \frac{\partial p_\tau}{\partial z} + \rho \mathrm{g} \sin\beta\right) \tag{2}$$

Energy conservation law

The equation of energy balance for the control volume can be written as a comparison of the internal energy, heat transfer and kinetic energy of the fluid. The initial equation is presented in [1], after its transformations, we can write the equation of energy balance:

$$\frac{\partial h}{\partial \tau} = \left(1 - \frac{1}{\rho}\frac{\partial p}{\partial h}\right)^{-1}\left[\frac{\dot{m}}{A\rho}\left(\frac{1}{\rho}\frac{\partial p}{\partial z} - \frac{\partial h}{\partial z} + \frac{1}{\rho}\frac{\partial p_\tau}{\partial z}\right) + \frac{4\dot{q}}{d_{\mathrm{in}}\rho} - \frac{1}{A\rho}\frac{\partial p}{\partial \rho}\frac{\partial \dot{m}}{\partial z}\right] \tag{3}$$

Solving balance equations

The method basec on eqs. (1)-(3) has been proposed in order to determine the mass-flow, pressure and enthalpy distributions of the working fluid. These equations, written in the appropriate form, will make up a mathematical model allowing for the analysis of the thermal and flow phenomena occurring in boiler waterwall tubes.

After some transformations and simplifications eqs. (1)-(3) can be brought to a form in which spatial derivatives are on the left and time derivatives on the right side [17]:

– the equation of mass conservation:

$$\frac{\partial \dot{m}}{\partial z} = -A\frac{\partial \rho}{\partial \tau} \tag{4}$$

– the equation of momentum balance:

$$\frac{\partial}{\partial z}\left(\frac{\dot{m}^2}{A^2\rho} + p\right) = -\frac{1}{A}\frac{\partial \dot{m}}{\partial \tau} - \frac{\partial p_\tau}{\partial z} - \rho \mathrm{g} \sin\beta \tag{5}$$

– the equation of energy balance:

$$\frac{\partial h}{\partial z}=\frac{\rho A}{\dot{m}}\left(-\frac{\partial h}{\partial \tau}+\frac{4\dot{q}}{d_{\text{in}}\rho}\right) \tag{6}$$

The time derivatives on the right side of the equations were replaced with backward difference quotients. The obtained system of differential equations can be solved by the Runge-Kutta method. After the transformations, we obtain the equation of the energy balance in the following form:

$$\frac{\mathrm{d}h_j^\tau}{\mathrm{d}z}=\frac{\rho_j^{\tau-\Delta\tau}A}{\dot{m}_j^{\tau-\Delta\tau}}\left(-\frac{h_j^\tau-h_j^{\tau-\Delta\tau}}{\Delta\tau}+\frac{4\dot{q}_j^{\tau-\Delta\tau}}{d_{\text{in}}\rho_j^{\tau-\Delta\tau}}\right) \tag{7}$$

The density of the fluid is found as a function of enthalpy and pressure:

$$\rho_j^\tau = f\left(h_j^\tau, p_j^{\tau-\Delta\tau}\right) \tag{8}$$

By solving mass and momentum conservation equations, we get respectively:

$$\frac{\mathrm{d}\dot{m}_j^\tau}{\mathrm{d}z}=-A\frac{\rho_j^\tau-\rho_j^{\tau-\Delta\tau}}{\Delta\tau} \tag{9}$$

$$\frac{\mathrm{d}}{\mathrm{d}z}\left[\frac{\left(\dot{m}^2\right)_j^\tau}{A^2\rho_j^\tau}+p_j^\tau\right]=-\frac{1}{A}\frac{\left(\dot{m}\right)_j^\tau-\left(\dot{m}\right)_j^{\tau-\Delta\tau}}{\Delta\tau}-\frac{\mathrm{d}p_\tau}{\mathrm{d}z}-\rho_j^\tau \mathrm{g}\sin\phi \tag{10}$$

The temperature of the fluid is found as a function of enthalpy and pressure:

$$t_j^\tau = f\left(h_j^\tau, p_j^\tau\right) \tag{11}$$

Subscript j in eqs. (7)-(11) signifies the number of analyzed cross-sections and varies in the range of 2, …, M. All thermophysical properties of the fluid are determined at each time step of the calculations.

Moreover, in the case of the proposed method, a stability condition should be satisfied to ensure that the numerical solution moves at higher speed, $\Delta z/\Delta\tau$, than the physical speed, w. This condition is called the Courant condition and described by the dependence:

$$\Delta\tau \le \frac{\Delta z}{w} \tag{12}$$

The temperature distribution in the tube

In order to increase the accuracy of the analysis and to determine the temperature distribution in tubes connected with fins, the method of division into the control volumes described in [12-14] was applied. Half of the tube cross-section was divided into control volumes, in which 20 characteristic points were determined, fig. 1:

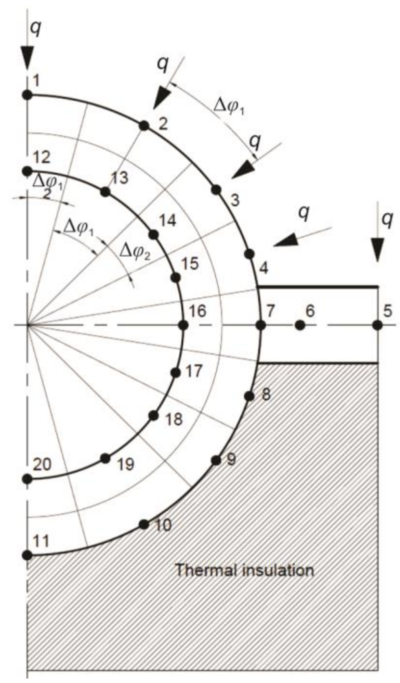

Figure 1 Division of the half of the tube cross-section into control volumes

The temperature distribution in the tube wall was determined using the equation described in [1]:

$$\theta_j^{\tau+\Delta\tau} = \left(\frac{D_j^\tau}{D_j^\tau + \Delta\tau}\right)\theta_j^\tau + \left(\frac{\Delta\tau}{\Delta\tau + D_j^\tau}\right)\left(t_j^{\tau+\Delta\tau} + G_j^\tau \dot{q}_j^{\tau+\Delta\tau}\right); \quad j = 1,\ldots,M \tag{13}$$

where

$$D = \frac{c_\mathrm{w}\rho_\mathrm{w} d_\mathrm{m} \mathrm{g}_w}{\alpha d_\mathrm{in}}, \quad d_\mathrm{m} = \frac{d_\mathrm{o} + d_\mathrm{in}}{2} \tag{14}$$

$$G = \frac{1}{\alpha\pi d_\mathrm{in}} \tag{15}$$

The equation allowing determining the temperature distribution in the tube cross-section, for point 1, can be written:

$$\begin{aligned} c_{1,j}\rho_{1,j}\frac{\Delta\hat{\varphi}_1}{2}(r_\mathrm{o}^2 - r_\mathrm{m}^2)\Delta z\frac{\mathrm{d}\theta_{1,j}}{\mathrm{d}\tau} &= \lambda_{1,j}\frac{\theta_{2,j} - \theta_{1,j}}{\Delta\hat{\varphi}_1 r_\mathrm{o}}\Delta r\Delta z + \\ &+ \lambda_{1,j}\frac{\theta_{12,j} - \theta_{1,j}}{\Delta r}\Delta\hat{\varphi}_1 r_\mathrm{m}\Delta z + \lambda_{1,j}\frac{\Delta\hat{\varphi}_1}{2}(r_\mathrm{o}^2 - r_\mathrm{m}^2)\frac{\theta_{1,j-1} - \theta_{1,j}}{\Delta z} + \\ &+ \lambda_{1,j}\frac{\Delta\hat{\varphi}_1}{2}(r_\mathrm{o}^2 - r_\mathrm{m}^2)\frac{\theta_{1,j+1} - \theta_{1,j}}{\Delta z} + \dot{q}_{1,j}\Delta\hat{\varphi}_1 r_\mathrm{o}\Delta z \end{aligned} \tag{16}$$

Equations for the remaining cross-section points are created in an analogous manner, based on geometrical relationships between nodes.

Simulation of thermal and flow phenomena occurring in the supercritical boiler waterwall tubes for various flow parameters

This section presents the results of numerical simulations performed for the waterwall tubes of a supercritical boiler. Computations were carried in accordance with the assumptions of the method presented in Section *Problem analysis*. The fluid flowing in the waterwall tubes is single-phase because its parameters exceed the parameters for the critical point of water. Bearing this in mind were developed functions that allow on-line calculations of the fluid parameters in a supercritical state. The computations were carried out with the use of the MATLAB code [18]. Simulations were carried out assuming a variable heat load value at the height of the combustion chamber, fig. 2. Heat flux is determined to base on the boiler operating conditions in one of the Polish power plants (live steam parameters: p = 26.6 MPa, T = 554 °C, the nominal power output capacity P = 858 MW). In order to compare the operating conditions of waterwall tubes at different

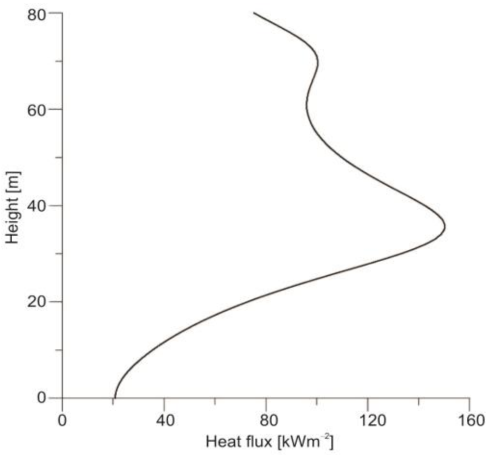

Figure 2 Distribution of the thermal load in the furnace waterwalls

heat load values, simulations were carried out for 80%, 90%, 100%, 110%, and 120% of the reference thermal load.

For simulation was used the data for a boiler in which a combustion chamber was designed as a 23.16 × 23.16 m square, in which smooth waterwall tubes are arranged in a spiral and are divided into two parts. The lower part of each tube is made of 16Mo3 steel and have approximately 32 meters length. The tubes in lower part of furnace have dimensions ($d_o \times g_w$ = 33.7 × 6.1 mm), are spaced with a pitch s = 50 mm and set at an angle φ = 24.62°. The tubes in the upper part are located above the pulverized fuel burners and are made of 13CrMo4-5 steel – material which can operate at higher temperatures. They have larger diameter and pitch ($d_o \times g_w$ = 38.0 × 6.3 mm, s = 57 mm) and are set at a greater angle of inclination (φ = 28.36°).

For the computations were assumed the following data [1]:

- number of waterwall tubes: n = 768,
- length of tubes: L = 166 [m],
- spatial size of control volume: z = 0.75 [m],
- total mass-flow: $\dot{m} = 2400 \cdot 10^3$ [kgh^{-1}],
- water pressure at the waterwall inlet: p_{inlet} = 29.96 [MPa], and
- feed water temperature at the combustion chamber waterwalls inlet: t_{inlet} = 313.4 [°C].

In order to increase the accuracy of the simulation, the time step size was each time determined from the Courant condition based on the assumed control volume length and the highest velocity value in the previous iteration. Dependence (12) has been transformed into equation ($\Delta\tau = 0.8 \cdot \Delta z/w$). The time step value for iteration was calculated from this equation, using the highest value of velocity in the previous iteration and the constant length of the control volume.

The precise determination of the temperature distribution of the fluid at any time requires the determination of the heat transfer coefficient. Its value on the inner surface of tubes, with supercritical fluid flow, can be determined from one of the empirical formulae available in the literature - in this case, was used the dependence determined by Kitoh *et al.* [19]:

$$\text{Nu} = 0.015\,\text{Re}^{0.85}\,\text{Pr}^{m} \tag{17}$$

where

$$m = 0.69 - \frac{8100}{200G^{1.2}} + f_c \dot{q}$$

and

$$f_c = 29 \cdot 10^{-8} + \frac{0.11}{200G^{1.2}} \;\text{ for } 0 \le h_b \le 1500 \tag{18}$$

$$f_c = -8.7 \cdot 10^{-8} - \frac{0.65}{200G^{1.2}} \;\text{ for } 1500 \le h_b \le 3300 \tag{19}$$

$$f_c = -9.7 \cdot 10^{-7} + \frac{1.3}{200G^{1.2}} \;\text{ for } 3300 \le h_b \le 4000 \tag{20}$$

The equation gives correct results under the following conditions:

- fluid bulk temperature: t_b = 20-550 [°C],
- fluid bulk enthalpy: h_b = 100-3300 [kJkg^{-1}],
- mass flux: G = 100-1750 [kgm^{-2}s^{-1}],
- thermal load of waterwalls: $\dot{q}$ = 0.0-1.8 [MWm^{-2}]

The following, figs. 3-8, presents the results of simulations carried out on the waterwall tube and the fluid flowing through it, at different values of heat flux incident on the surface of the tube.

According to the data presented in fig. 3, increasing the heat load from 80% to 120% of the reference value, causes an increase in temperature of the fluid at a distance of 40 m from the beginning of the tube by about 12 °C and at a distance of 80 m by about 17 °C.

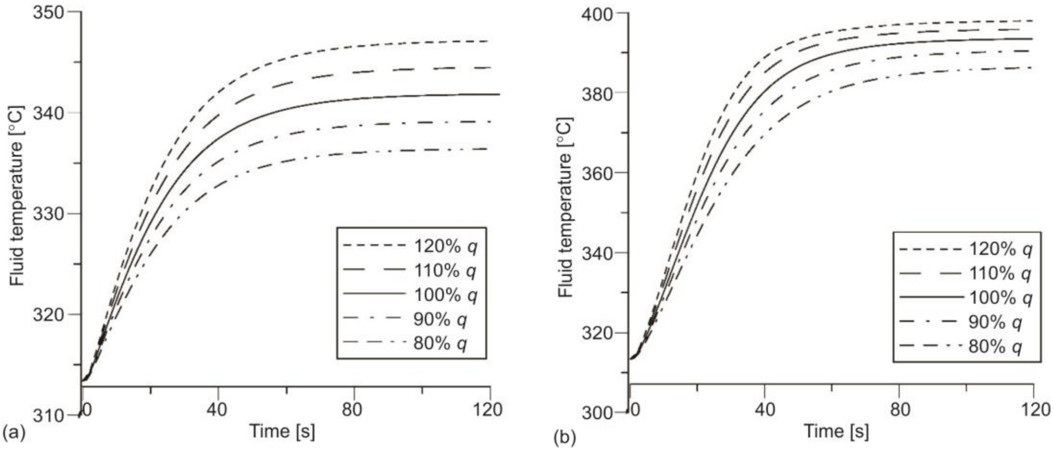

Figure 3. Changes of the temperature of the working fluid at a length of; (a) 40 m, (b) 80 m

Figures 4 and 5 shows the temperature distributions at half the cross-section of the tubes 40 m and 80 m from the inlet to the tube for 80%, 100%, and 120% reference load. As shown, at the height of 40 m temperature difference for the point on the face of the tube (point 1) is 18.9 °C, and at the height of 80 m – 21.24 °C. The temperature at the point in the middle of the fin that connects the tubes also changes significantly - by 33.48° and 45.64 °C, respectively.

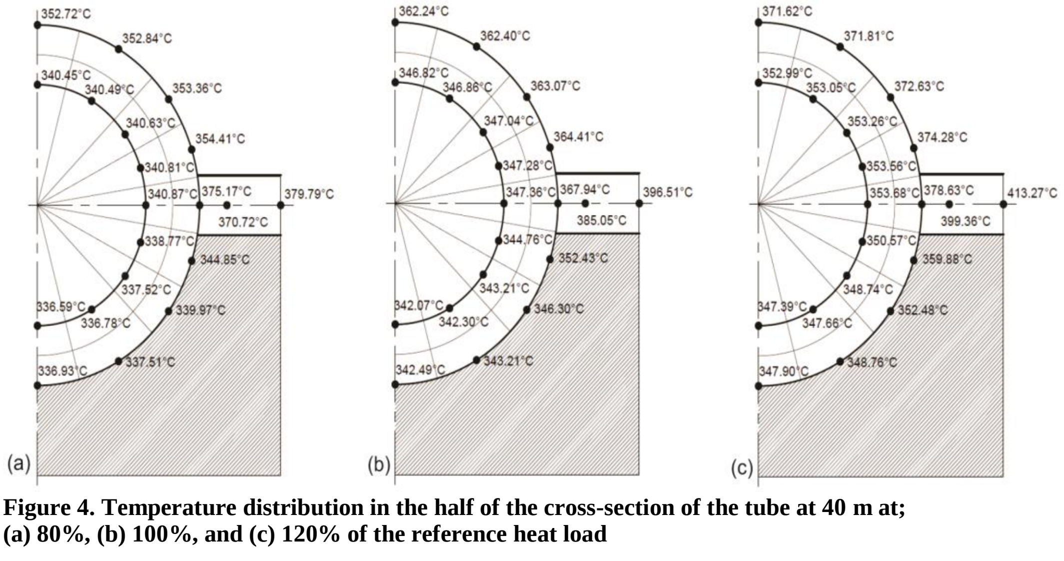

Figure 4. Temperature distribution in the half of the cross-section of the tube at 40 m at; (a) 80%, (b) 100%, and (c) 120% of the reference heat load

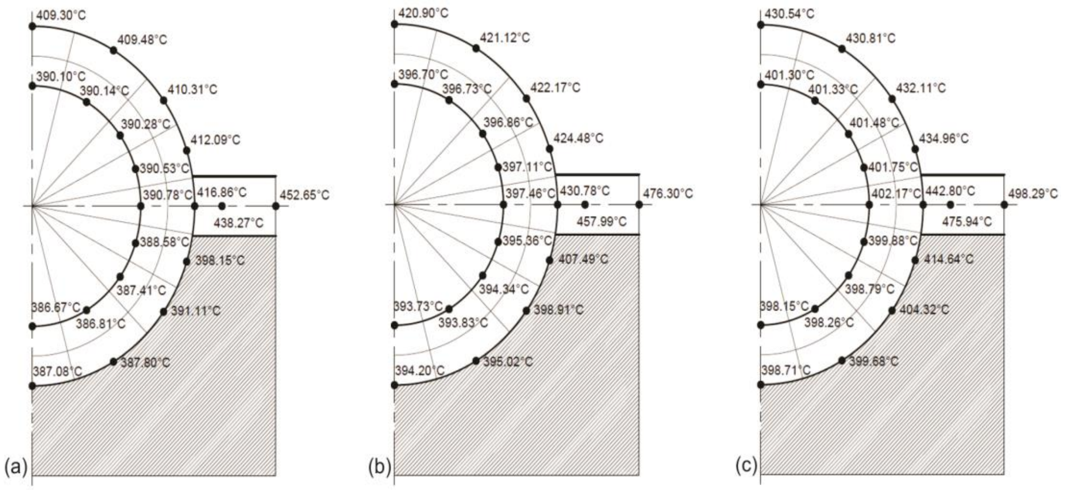

Figure 5. Temperature distribution in the half of the cross-section of the tube at 80 m at; (a) 80%, (b) 100%, and (c) 120% of the reference heat load

According to the data presented in fig. 6, increasing the heat load from 80% to 120% of the reference value, causes an increase in temperature of the fluid at a distance of 120 m from the beginning of the tube by about 30 °C and at a distance of 166 m by about 50 °C.

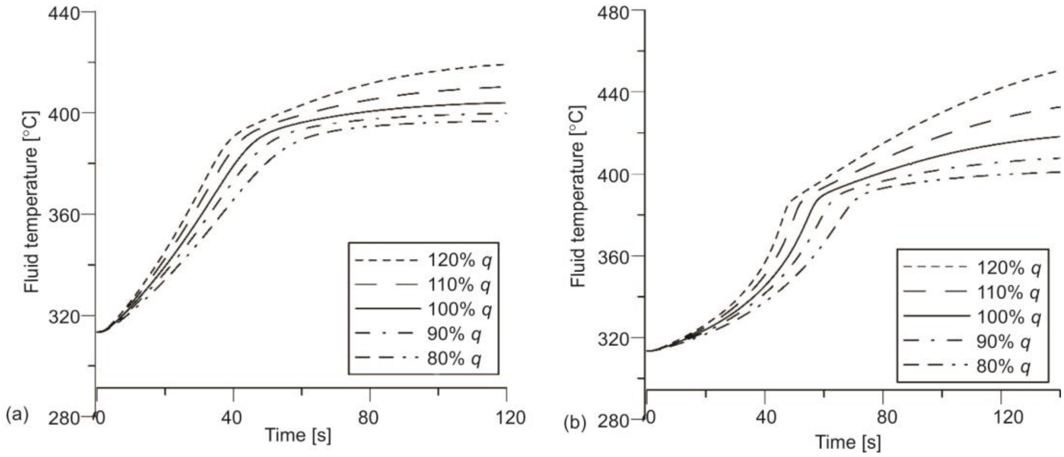

Figure 6. Changes of the temperature of the working fluid at a length of; (a) 120 m, (b) 166 m

Figures 7 and 8 show the temperature distributions at half the cross-section of the tubes 120 m and 166 m from the inlet to the tube for 80%, 100%, and 120% reference load. As shown, at the height of 120 m temperature difference for the point on the face of the tube (point 1) is 38.02 °C, and at the height of 166 m – 64.24 °C. The temperature at the point in the middle of the fin that connects the tubes also changes significantly – by 59.52 °C and 65.50 °C, respectively.

The occurrence of such large differences in temperature results in the formation of significant thermal stresses on the boiler's furnace width.

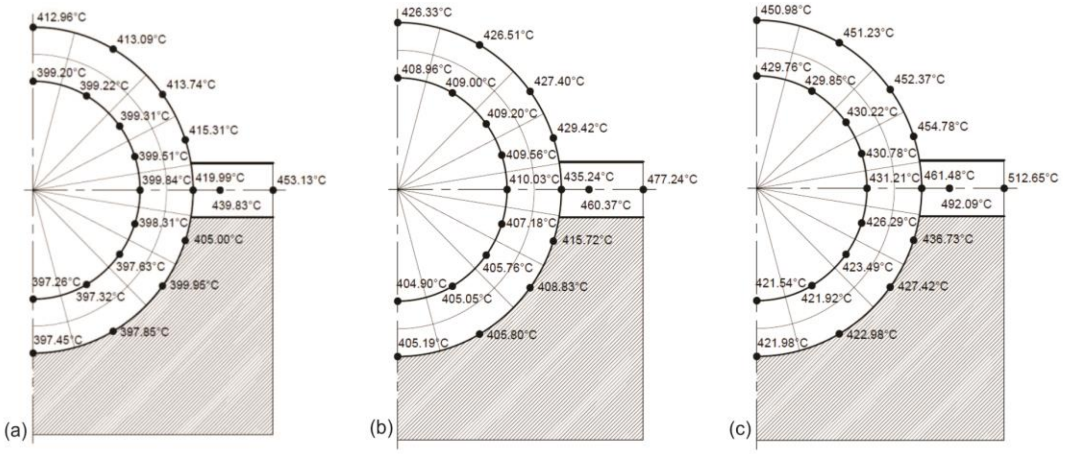

Figure 7. Temperature distribution in the half of the cross-section of the tube at 120 m at; (a) 80%, (b) 100%, and (c) 120% of the reference heat load

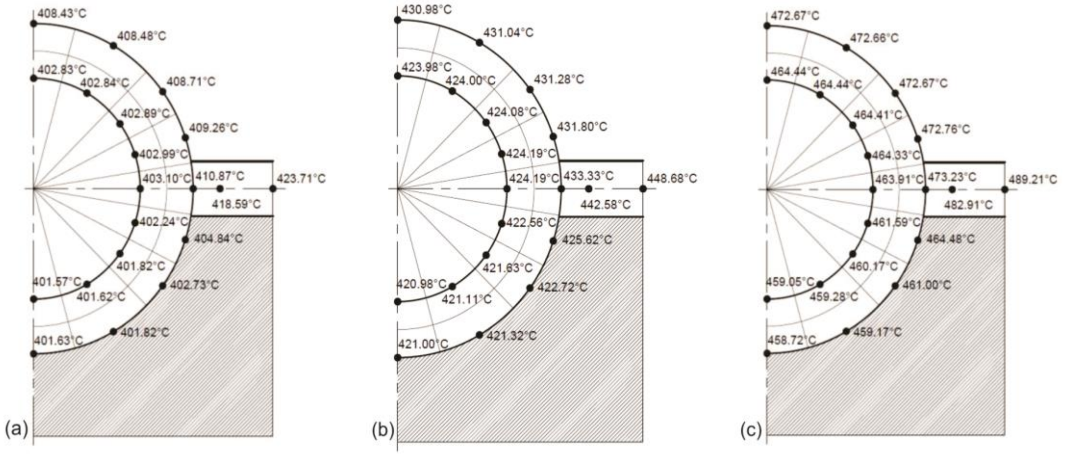

Figure 8. Temperature distribution in the half of the cross-section of the tube at 166 m at; (a) 80%, (b) 100%, and (c) 120% of the reference heat load

Conclusions

The article presents the results of numerical simulation of operating conditions of waterwall tubes installed in a supercritical boiler operating at variable thermal load. The temperature distributions of the tubes with fins and the working fluid flowing through them are compared depending on the heat load value for 80%, 90%, 100%, 110%, and 120% of the reference load. At the extreme heat load values (in the analyzed range), the temperature differences of the working fluid at the outlet of the tube reach about 45 °C, and in the material constituting the tubes even 70 °C (for the face of the tube). The temperature of the tube at its height also changes significantly – depending on the heat load, the differences are 60-80 °C for the face of the tube and 10°-20° on the inner surface. Due to the fact that the conditions in which the operation of the tubes was analyzed, correspond to the variability of the conditions occurring in the combustion chamber, it can be expected that significant thermal stresses will occur between neighboring tubes. The results of the simulations presented in this article will be used to determine stresses in waterwall tubes. Based on the obtained stress values, an analysis will be carried out to determine the conditions that will reduce the value of thermal stresses in order to reduce the amount of damage occurring in the power boilers. The highest temperature variation is observed in the fins of the tubes, so it can be expected that the change in their dimensions may partially reduce the value of the stresses occurring in them.

Nomenclature

A – cross-sectional area, [m^2]
c – specific heat, [$Jkg^{-1}K^{-1}$]
d – diameter, [m]
g – gravitational acceleration, [ms^{-2}]
G – mass flux, [$kgm^{-2}s^{-1}$]
h – enthalpy, [$kJkg^{-1}K^{-1}$]
L – tube length, [m]
$\dot{m}$ – mass-flow rate, [kgs^{-1}]
Nu – Nusselt number ($=\alpha d/\lambda$), [–]
n – number of tubes, [-]
Pr – Prandtl number ($=\nu/\alpha$), [–]
p – pressure, [Pa]
$\dot{q}$ – heat flux, [Wm^{-2}]
Re – Reynolds number ($=wd/\nu$), [–]
r – radius, [m]
Δr – wall thickness, [m]
t – fluid temperature, [°C]
U – circumference, [m]
w – velocity, [ms^{-1}]
z – axial co-ordinate, [m]
Δz – control volume length, [m]

Greek letters

α – heat transfer coefficient, [$Wm^{-2}K^{-1}$]
β – angle of inclination, [°]
ϕ – angle
$\Delta\hat{\varphi}$ – characteristic angle, [rad]
θ – wall temperature, [°C]
λ – thermal conductivity, [$Wm^{-1}K^{-1}$]
ρ – density, [kgm^{-3}]
τ – time, [s]
ν – kinematic viscosity, [m^2s^{-1}]

Subscripts

b – fluid bulk parameter
in – inner
m – medium
o – outer
τ – tangential
w – wall

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