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Numerical analysis of a waste heat recovery process with account of condensation of steam from flue gases

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ABSTRACT

This paper presents modelling of the process of condensation of steam contained in flue gases in systems for waste heat recovery from flue gases. A one-dimensional, non-stationary mathematical model of a heat exchanger was described, and then numerical calculations for flue gases from the combustion of hard coal and brown coal were performed. The results were presented in the form of characteristics with temperature distributions along the axis of the condensing heat exchanger and the degree of cooling of flue gases.

1. Introduction

One of the most important and still current issues in power engineering is so-called "cold end" problem, which concerns mainly older power plants with a classic chimney. This problem consists in the fact that it is not possible to reduce the flue gas temperature below the dew point, as the condensing moisture causes corrosion in flue gas ducts and the chimney. The recovery of heat from flue gases results in a reduction of the biggest loss, which is the loss at the outlet. This is achieved by installing heat exchangers upstream of the flue gas desulphurization system. However, the vast majority of such systems installed so far are based on the heat recovery with the use of non-condensing heat exchangers. The degree of cooling of flue gases is negligible. The waste heat recovered is usually used for heating the intake air, feed water or is directed to district heating systems.

In order to increase the flow of heat recovered from flue gases [4,5,8] it was proposed to use a condensing heat exchanger in the flue gas duct and to cool the flue gas below the dew point [13,14]. Systems of this type are dedicated for new power units, in which a cooling tower performs the function of a traditional chimney, and therefore the problem of corrosion does not exist. The paper presents the numerical modelling of the process of the condensation of steam contained in flue gases. A one-dimensional, non-stationary model of a condensing heat exchanger was used for calculations. It is difficult to solve the

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problem formulated in this way due to the complexity of the model describing the heat exchange in the presence of inert gases, a low concentration of steam, the condensation phenomena, as well as the limitations arising from a considerable heat flow rate and usually also the dimensions of the heat exchanger.

2. Differential one-dimensional model of the heat exchange process, taking into account the steam condensation

In order to model the heat exchange process, a differential one-dimensional mathematical model was used, for which the following assumptions were made:

- the heat transferred to the cooling water during condensation of steam from flue gases is the latent heat generated as a result of the mass transfer (of steam particles) from flue gases, through the condensate layer, to the wall,
- the interface, on which the condensation process takes place, is permeable only to particles of the condensing steam, while flue gases form a layer that hinders the access of the steam to the interface,
- the intensity of the condensation process is determined by the rate of transfer of steam particles from the main mass flow of flue gases to the surface, on which the steam condensation will occur, and by the dissipation rate of the heat released during condensation by the condensate layer,
- the inflow of steam to the cooling surface depends on the difference of the partial pressures of steam in the main core of the flue gas flow and at the interface.

Heat transfer from the condensation surface, through the condensate layer, to the outer surface of the pipe with cooling water takes place as a result of the heat conduction of the condensate layer. Schematic presentation of the layers involved in the flue gas condensation process is shown for the cross-section of the cooling pipe in Fig. 1.

The temperature difference $T_f - T_s$ is the driving force of the heat transfer through the condensate. The total global difference between the temperature of the mixture of dry flue gases and steam T_m and the temperature of cooling water T_w can be represented as the sum of elementary temperature differences (Fig. 1).

$$(T_m - T_w) = (T_m - T_f) + (T_f - T_s) + (T_s - T_z) + (T_z - T_w), \quad (1)$$

where T_f is the temperature between the diffusion and the condensate layers, T_s and T_z are the wall temperatures (Fig. 1).

The heat transfer coefficient is obtained on the basis of the equation

$$\frac{1}{\alpha} = \frac{1}{\alpha_f} + \frac{1}{\alpha_k} + \frac{1}{\alpha_s} + \frac{1}{\alpha_w}, \quad (2)$$

where α_f is the heat transfer coefficient of the flue gas, α_k the equivalent heat transfer conductivity of the condensate layer, α_s the equivalent heat transfer conductivity of the wall, while α_w is the heat transfer coefficient of the water.

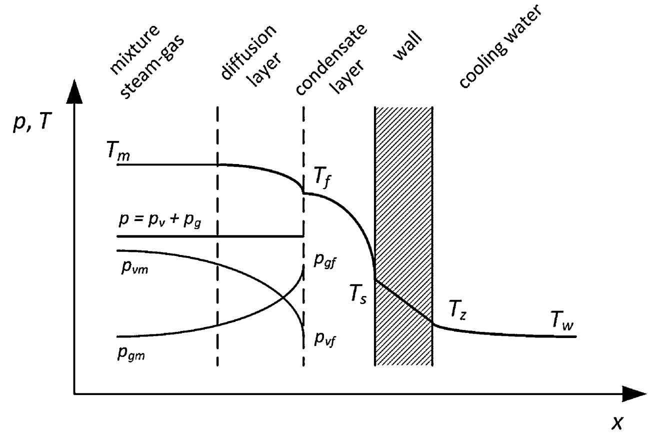

Fig. 1 – Schematic presentation of the layers involved in the steam condensation process in the pipe cross-section.

The equivalent heat transfer conductivity of the wall α_s was calculated on the basis of the equation:

$$\frac{1}{\alpha_s} = R_s = \frac{d_o}{2\lambda_s} \ln\left(\frac{d_o}{d_i}\right), \quad (3)$$

where R_s is the heat resistance of the pipe, λ_s is the thermal conductivity of the wall, d_o and d_i are respectively the outer and inner diameter of the pipe.

The coefficient α_k was calculated from the dependence derived by Nusselt:

$$\alpha_{k,n} = 0.728\left(\frac{\lambda_k^3 \rho_k^2 g\, r}{\mu_k d_o (T_v - T_s)}\right)^{1/4}, \quad (4)$$

where λ_k is the thermal conductivity coefficient of the condensate, μ_k coefficient of dynamic viscosity, ρ_k density of the condensate, g gravitational acceleration, r heat of condensation, and T_v the temperature of steam. The index n refers to the condensation of stationary steam.

A mechanical interaction, caused primarily by friction, occurs between the moving flue gases and the condensate layer. This results in an increase of the velocity in the condensate layer, a decrease in the thickness of the layer, as well as in local turbulence stimulation. The coefficient of heat transfer through the condensate layer increases. Hence it appears that the coefficient α_k will change along with the penetration of steam into the bundle of cooling water pipes, because its velocity will be decreasing. Therefore, the heat transfer coefficient α_k for moving steam will always be greater than the coefficient $\alpha_{k,n}$ for stationary steam. Based on theoretical and experimental studies, a formula has been developed, which takes into account the impact of the steam velocity on the heat transfer coefficient. This formula has the following dimensionless form [9]

$$\frac{\alpha_k}{\alpha_{k,n}} = A \prod_d^k Re_v^m Nu^s, \quad (5)$$

where A, k, m, s denote certain constants, while $\prod_d$ is a criterion number, which takes into account the friction between the moving flue gases and the condensate layer. Eq. (5) with the coefficients given below was adopted for further considerations [9]

$$\frac{\alpha_k}{\alpha_{k,n}} = 28.4 \prod_d^{0.08} Re_v^m Nu^{-0.58}. \tag{6}$$

In Eq. (6), the value $m = 0$, because it was found that in the range $Re = 500$–6000 it has a negligible impact on the change in the quotient of the heat transfer coefficients.

Knowing the dependencies for the heat transfer coefficient, it is possible to formulate equations required for calculations needed to describe the processes of heat and mass transfer in a mixture of dry flue gases and steam. The heat flux density towards the surface of the heat exchanger pipe is expressed by Fourier's law

$$q = -\lambda_k \frac{\partial T}{\partial y} \approx \frac{\lambda_k}{\delta_k}\theta_k, \tag{7}$$

where δ_k is the thickness of condensate layer, $\theta_k = (T_f - T_s)$.

As a result of the introduction of the heat transfer coefficient, the following was obtained

$$q = \alpha_k (T_f - T_s) = \alpha_k \theta_k. \tag{8}$$

On the basis of Fick's law, the mass flux density can be expressed by the equation

$$d_k = -D_c \frac{dC}{dy}. \tag{9}$$

where D_c is the diffusion coefficient, while the gas concentration C is expressed in kg/m^3 and is equal to the specific gravity of the gas. The concentration gradient can be expressed by the partial pressure gradient [6]

$$d_k = -D_p \frac{dp}{dy}, \quad d_k = \frac{D_c}{R_v T_v}. \tag{10}$$

where R_v is gas constant for steam and T is absolute temperature. By taking steps similar as with the Fourier law and the determination of the heat flux, the mass flux density can be expressed as [2,6]

$$d_k = \beta_r (p_v - p_f) = \beta_r \Delta p_v. \tag{11}$$

In this formula, β_r is the mass transfer coefficient. The specific heat flux associated with the mass transfer is

(12)$q = d_k \Delta i = \beta_r \Delta p_v \Delta i$,where Δi is driving force related to transfer mass by diffusion and Δp_v is difference in partial pressures in the diffusion layer.

As all the heat emitted in the process of condensation of steam from flue gases is transferred through the condensate layer to the wall of the cooling water pipe, the following equation was obtained

$$q = \beta_r (p_v - p_f) \Delta i = \alpha_k (T_v - T_s), \tag{13}$$

or

$$q = \beta_r \Delta p_v \Delta i = \alpha_k \theta_f, \tag{14}$$

The equality between the heat flow caused by the condensation of steam and the heat flow transferred through the condensate layer depends on the steam pressure and temperature t_f at the interface and the difference $T_f - T_s$. In order to determine the coefficient β_r, the following dependence was used [3]

$$Sh_D = \frac{\beta_r d_0}{D_p} = C_1 \left(\frac{\Delta p_v}{p_m}\right)^{-1/3} \varepsilon_0^{-x}, \tag{15}$$

where for $Re > 350$, the value of the constant $C_1 = 0.82$ and the index exponent $x = 0.6$, while for $Re < 350$, the value of $C_1 = 0.52$ and $x = 0.7$; Sh_D is the diffusion Nusselt number. The symbol e_0 denotes the mass fraction of dry flue gases in wet flue gases. The diffusion coefficient D_p is determined by the Rossi's formula [1,6]

$$D_p = \frac{6.27 \times 10^{-6}}{p_m} C_1 \left(\frac{T_m}{273}\right)^{0.8} \tag{16}$$

where T_m is absolute temperature of the mixture of steam and flue gases and p_m is absolute pressure of the mixture expressed in bars. Reynolds number is defined in relation to flue gas parameters. By substituting the coefficient β_r with D_p in the criteria expression, the following is obtained

$$\beta_r = \frac{C_2}{p_m \varepsilon_0^x d_0 10^6} \left(\frac{T_m}{273}\right)^{0.8} Re^{0.5} \left(\frac{\Delta p_v}{p_m}\right)^{-1/3} \tag{17}$$

where $C_2 = 5.15$, $x = 0.6$ for $Re > 350$ and $C_2 = 3.26$, $x = 0.7$, for $Re < 350$ [10]. The expression (17) for the coefficient β_r must not be extrapolated to the range of values close to zero for ε_0 and $\Delta p_v/p_m$, because the expression (17) quickly tends to infinity.

The proposed mathematical model is a one-dimensional non-stationary model, in which independent variables are the length x and the time t [7,11]. This model allows determining the instantaneous distributions of temperatures of water, flue gases as well as the pipe wall running along the heat exchanger, on which the heat exchange process takes place. Modelling of transient heat transfer processes becomes particularly important, if condensation occurs. Due to the complexity of this process and because of a large number of parameters affecting the condensation process, a simplified one-dimensional model was used. It was assumed that the process of heat exchange between water and flue gases takes place as shown in Fig. 2.

A counterflow heat exchanger, in which flue gases with the temperature T_m move with the velocity v_m, water with the temperature T_s flows with the velocity v_w, and T_w is the

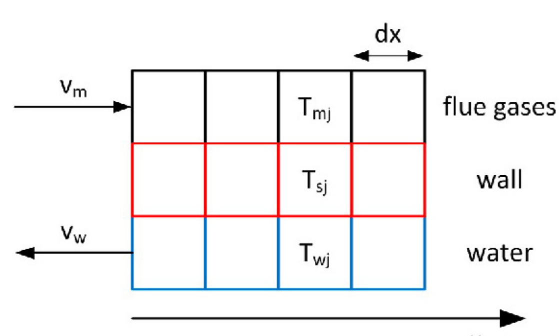

Fig. 2 – Calculation cell for the heat exchanger.

temperature of the wall, was adopted for the calculations. The heat exchanger was replaced with a single pipe, which is perfused by flue gases. The pipe was then divided into sections with the length $dx = L/n$, where L is the length of the pipe, and n is the number of sections it was divided into. The calculations were performed for each element of the pipe with the length of dx. Index j denotes the jth cell; $j = 0, 1, 2, \ldots, n$. Each cell contains a certain mass of water m_w, mass of the pipe m_s, and mass of flue gases m_m.

The equations describing the process of heat transfer between flue gases, the wall, and water are presented in the following form

$$c_m m_m \frac{\partial T_m}{\partial t} + c_m m_m \left(v_m \frac{\partial T_m}{\partial x} \right) = \alpha_m F_s (T_s - T_m) + \gamma \dot{Q}_k, \tag{18}$$

$$c_m m_m \frac{\partial T_w}{\partial t} = \alpha_m F_m (T_w - T_s) + \alpha_w F_w (T_w - T_s) + \gamma \dot{Q}_k, \tag{19}$$

$$c_w m_w \frac{\partial T_w}{\partial t} + c_w m_w \left(v_w \frac{\partial T_w}{\partial x} \right) = \alpha_w F_w (T_s - T_w) + \gamma \dot{Q}_k. \tag{20}$$

The left side of Eq. (18) describes transfer of heat with the velocity c along the variable x, while the right side expresses the penetration of heat from the wall to flue gases and the condensation heat $\dot{Q}_k$. The second equation describes a change in the temperature of the wall as a result of heat flow from flue gases, the transfer of heat to water, and additional heat flow from steam condensing in flue gases. Eq. (20) is the heat transfer equation for water. Heat is carried with the velocity v, and the source term in the equation is the heat received from the wall and the heat of condensation. Due to a small thickness of the pipe wall, the thermal resistance of the pipe was omitted. The flow of heat $\dot{Q}_k$ resulting from condensation of steam in flue gases is included in Eq. (18) as an additional internal heat source of wet flue gases.

The system of Eqs. (18)–(20) was solved by approximating the temperature derivative along the direction x using a second-order backward scheme. The time derivative was approximated using a first-order implicit scheme, which can be expressed in the following form

$$T^{k+1} = T^k + f(T^{k+1})\Delta t \tag{21}$$

where k denotes a subsequent time layer, while Δt is the time step. Thanks to the use of the implicit scheme, partial differential equations were reduced to a system of algebraic equations with a tridiagonal matrix, which can be solved using the Thomas algorithm [10]. An advantage of an implicit formulation is a lack of the time step restrictions resulting from the Courant condition [10,12].

The flow of heat $\dot{Q}_L$ resulting from condensation of steam in flue gases was added when the temperature of flue gases in the cell dropped below the saturation temperature for steam in flue gases $T < T_n$. The symbol γ denotes a parameter that takes the value of 0 when $T > T_n$ and the value of 1 when $T < T_n$.

An assumption was made that the process of condensation of steam from flue gases will run similarly to the process of cooling moist air shown in the Mollier diagrams (Fig. 3).

The heat transfer coefficient for water α_w was calculated from the formula (22)

$$Nu = \frac{\alpha_w d_i}{\lambda_w} = C \times Re^a \times Pr^b, \tag{22}$$

for which the following values of coefficients were assumed: $C = 0.023$, $a = 0.8$ and $b = 0.4$. Since the Prandtl number for water depends on the temperature, the coefficient α_w changes along the length of the pipe.

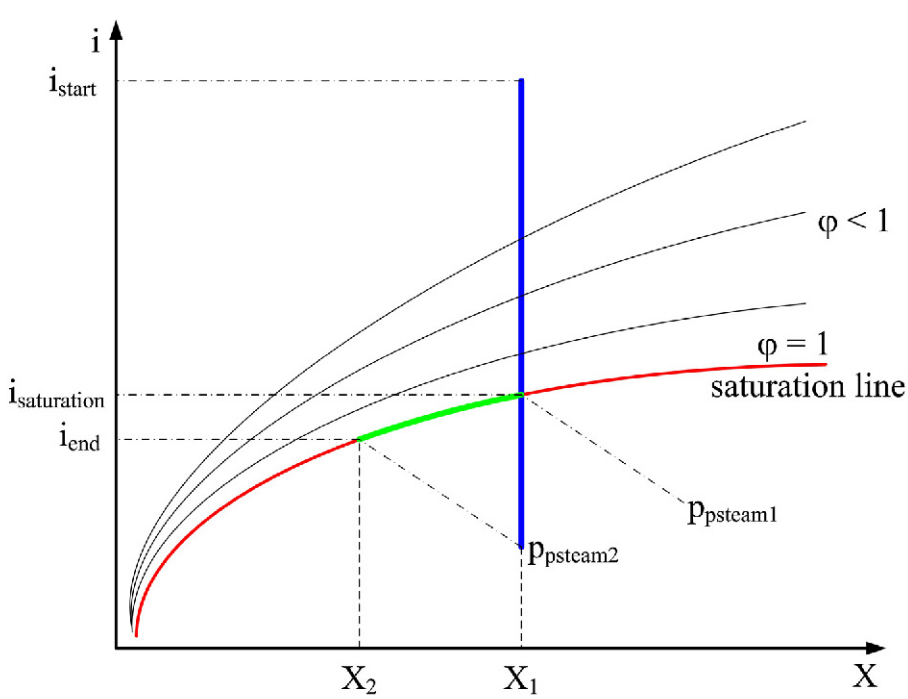

Fig. 3 – Isobaric cooling of the wet gas at a constant moisture content coefficient.

3. Calculation results obtained by solving the mathematical model

A counterflow heat exchanger with copper pipes (diameter: 13.5 mm, wall thickness: 2 mm) was adopted for the calculations. The calculations were performed for flue gases from the combustion of hard coal and brown coal. The main difference resulting from the fuel used is the mass share of moisture in coal, which directly translates into the share of moisture in flue gases and further into the saturation temperature for steam in flue gases. Saturation temperatures were determined for the assumed composition of hard coal and brown coal. The saturation temperature for hard coal is $t_n = 42.1\,^\circ C$, while for brown coal it is $t_n = 65.0\,^\circ C$. The mass flow rates of flue gases adopted for the calculations correspond to a power unit with a capacity of 900 MW. For brown coal the mass flow rate is 1090.2 kg/s and for hard coal it is 830 kg/s. The flue gas temperature at the inlet to the heat exchanger for brown coal and hard coal is $t_{sp1} = 170\,^\circ C$ and $t_{sp1} = 120\,^\circ C$, respectively.

3.1. *Results of calculations for brown coal*

The calculations were performed for two variants. The first variant assumes that the entire mass flow of flue gases is directed to the heat exchanger, while in the second variant the flue gas flow supplies two parallel heat exchangers. Temperature distributions along the heat exchanger were obtained on the basis of the proposed mathematical model, as shown in Fig. 4.

The calculations were performed until the heat transfer process stabilized and stationary distributions of temperatures along pipes of the heat exchanger were obtained. Stationary distributions were obtained for the time $t > 170$ s. The length of the pipe of the heat exchanger was assumed to be $L = 28$ m and then was divided into $n = 50$ cells of equal length. The heat transferred by flue gases by the time of condensation is $Q = 153.53$ MW. Cooling the flue gases to a temperature below the saturation temperature results in an additional latent heat flow $Q_L = 124.83$ MW. The condensation of steam contained in flue gases takes place near the outlet of the heat exchanger, on the last 5.8 m of the heat exchanger pipes. The length of the non-condensing part of the heat exchanger is $L_{bk} = 22.2$ m.

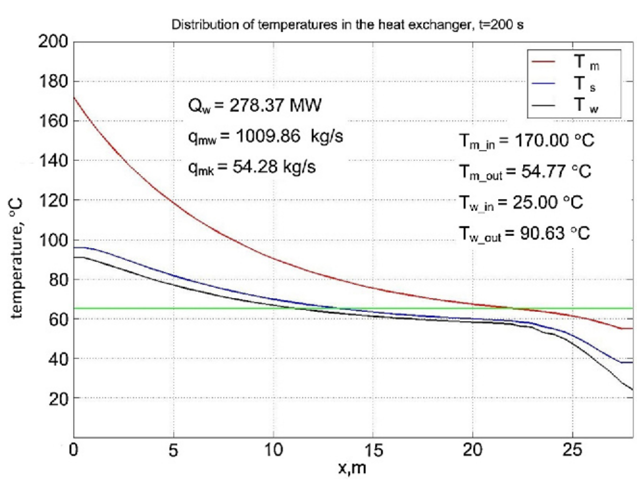

Fig. 4 – Distribution of the flue gas and water temperatures along the heat exchanger, at the mass flow rate of flue gases from the combustion of brown coal $q_{mm} = 1090.2$ kg/s. Q_w is the thermal power of the heat exchanger, q_{mw} mass flow rate of cooling water, and q_{mk} mass flow rate of the condensate, $T_{m1,2}$, which is respectively the temperature of flue gases at the inlet and at the outlet of the heat exchanger, and $T_{w1,2}$, which is respectively the temperature of water at the inlet and at the outlet of the heat exchanger. Green colour indicates the saturation temperature for steam contained in flue gases, $T_n = 64.97\,^\circ C$.

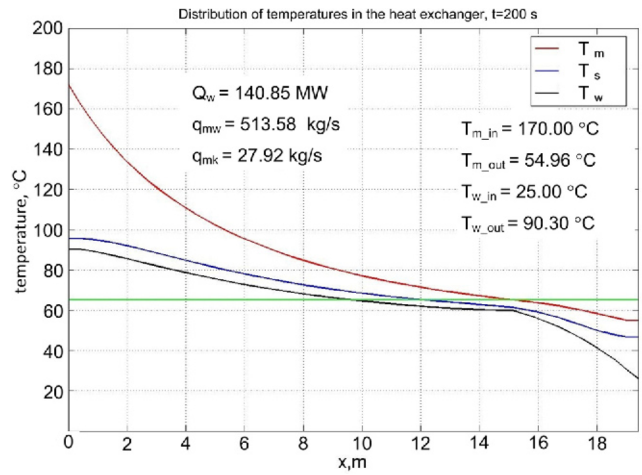

Fig. 5 – Distribution of the flue gas and water temperatures along the heat exchanger, at the mass flow rate of flue gases from the combustion of brown coal $q_{mm} = 513.58$ kg/s.

The condensation of steam in flue gases generates an additional heat flow, which is then carried in accordance with the direction of the velocity of water. The occurrence of an additional heat flow causes locally a significant increase in the water temperature and a decrease in the temperature differences between water and flue gases. The temperature distributions for the heat exchanger supplied with half of the flue gases are shown in Fig. 5. For the variant used for the analysis, the pipe length $L = 19.4$ m was adopted and divided into $n = 50$ cells. The process of condensation begins at the 15.2th-m of the pipe and therefore the length of the condensing part of the heat exchanger is approx. $L_{kond} = 4.2$ m.

3.2. *Results of calculations for hard coal*

Subsequently, similar calculations were performed for hard coal. Fig. 6 shows the temperature distributions along the heat exchanger supplied with the full stream of flue gases.

Fig. 7 shows the temperature distribution in the heat exchanger supplied with half of the stream of flue gases. The pipe length was reduced to 30 m. The process of condensation of steam contained in flue gases is barely noticeable and takes place in the pipe section for $L > 28.6$ m, which gives the length of the condensing part $L_{kond} = 1.4$ m.

An assumption was made that the length of the heat exchanger pipe was $L = 40$ m, while the number of calculation

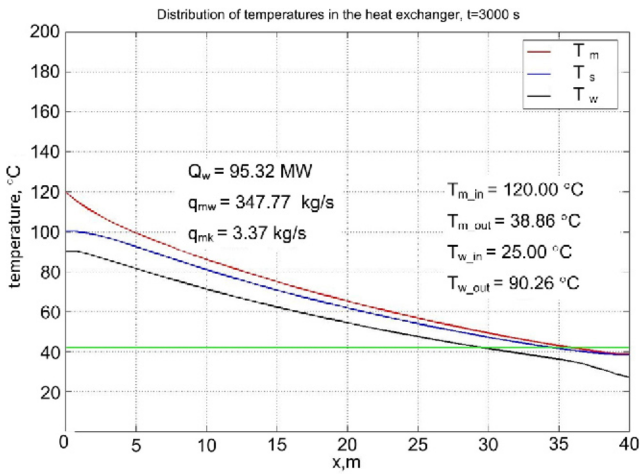

Fig. 6 – The temperature distribution in the flue gas–water heat exchanger for flue gases obtained from the combustion of hard coal, q_{mm} = 830 kg/s. Green colour indicates the saturation temperature for steam contained in flue gases, at which the condensation process starts.

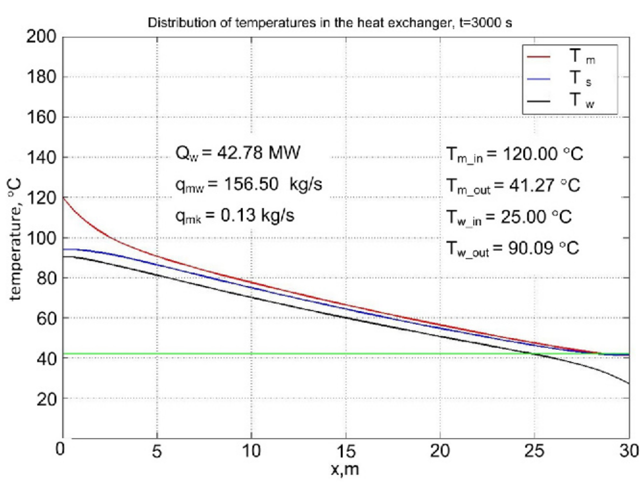

Fig. 7 – The temperature distribution in the flue gas–water heat exchanger for the gas obtained from the combustion of hard coal at the mass flow rate of flue gases q_{mm} = 415.5 kg/s.

cells was n = 50. Steam condensation begins at the 35.8th-m of the pipe. The length of the pipe section which operates in the condensing steam conditions is approx. 4.2 m. In the event of low velocities of water in the pipe and a small value of the coefficient of heat transfer from water; it takes much more time to determine temperature distributions in the heat exchanger. In the given case, the steady state was achieved after approx. 2500 s. Thermal powers obtained for hard coal are also much lower. For the flue gas mass flow rate 830 kg/s and the cooling water mass flow rate q_{mw} = 347.7 kg/s, the thermal power of the non-condensing part was obtained at the level of 87.7 MW, while the thermal power of the heat exchanger was 95.3 MW, i.e. about 34% of the power reached in the heat exchanger supplied with flue gases from the combustion of brown coal.

4. Summary

The paper presents modelling of the process of condensation of steam contained in flue gases from the combustion of hard coal and brown coal. A one-dimensional, differential mathematical model was used for calculations. It takes into account the heat transfer between flue gases, the pipe wall and water for cooling the pipes of the condensing heat exchanger. The formulas used to calculate the heat transfer coefficients were given and discussed and the process of mass transfer that accompanies the phenomenon of condensation of steam in the presence of inert gases was described. Numerical calculations were performed, on the basis of which the temperature distributions along the length of the condensing heat exchanger were obtained. As it appears from the diagrams, cooling down flue gases to a temperature below the dew point greatly increases the flow of the heat recovered. This is particularly evident in the case of flue gases from the combustion of brown coal, which are cooled from the temperature of 170 °C down to a temperature below 65 °C. Just a slight decrease below the saturation temperature causes a considerable release of latent heat flows, which increases the flow of recovered heat. This process looks differently in the case of flue gases from hard coal. In order to induce steam condensation, flue gases should be cooled down from the temperature of 120 °C to a temperature below 42 °C. This is a significant problem which is associated with the possibility of a further use of the recovered heat as it has a very low temperature. Thus it appears that in the case of flue gas from hard coal the heat recovery below the dew point does not make much sense.

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