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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.

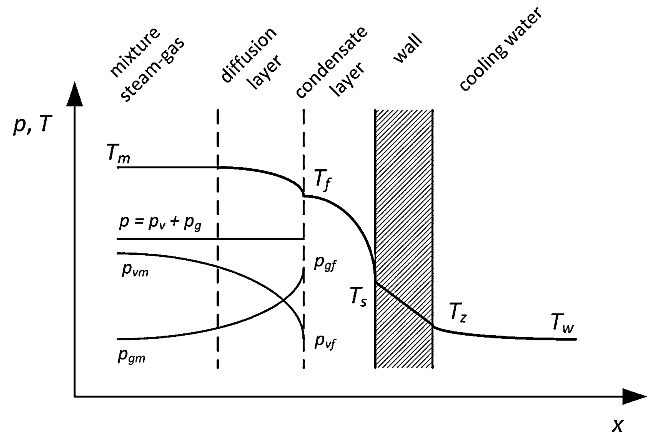

Fig. 1 – Schematic presentation of the layers involved in the steam condensation process in the pipe cross-section.

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