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The analyzis of the stream-air mixture during a flow through a tube bank inside a condenser

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Abstract

The paper presents the results of calculations related to the flow analysis of a tube bank taking into account the partial pressures of gases flowing through the condenser. Different mass share of the gas not condensing in the heat exchanger upstream to the first row of tubes and the partial pressures of water vapor and air have been taken into account. The value of the partial pressures of gases, pressures drop, temperature, velocity, Reynolds number, and air share in the subsequent rows of tubes have been calculated. This study is a basis for the analysis of heat exchange in the condenser considering the influence of the noncondensing gas. The increase in the share of air in the following rows of tubes results in an increase in the thermal resistance and condensation of increasingly lower mass of water vapor.

Keywords: Condenser; Pressure drop; Steam-air mixture; Water vapor condensation in a solution with air; Partial pressure

Nomenclature

A	–	flow area of gases, m^2
c	–	velocity, m/s
D	–	tube diameter, m
k	–	mass share of air
m	–	number of rows of tubes

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$\dot{m}$	–	mass flow, kg/s
p	–	pressure, Pa
R	–	gas constant, J/(kg K)
Re	–	Reynolds number
T	–	temperature, K
X	–	distance between the tube centers, m

Greek symbols

Δ	–	difference
v	–	kinematic viscocity, m^2/s
ξ	–	accumulative coefficient of loses
ρ	–	density, kg/m^3

Subscripts

z	–	external
v	–	water vapor
a	–	air
i	–	ith row of pipes

1 Introduction

Increasing restrictions about greenhouse gases are forcing electric energy producents to search a way to raise efficiency of energy conversion while reducting level of pollution. Effective, economical and pure technologies are the way to sustainable development which has to have significant role in multiscale energetics [11].

One of the way to get the effective electric energy production is increasing efficienty of cycle or only keeping it on constant level. Cycle elements which have significant influence on efficiency of conventional powerhouses are boiler, turbine, condenser and pump [12]. In this article we will focus on possibility to improve the work of the power plant condensers. As is well known, the main task of these devices is to maintain an appropriate level of vacuum for the last stage of steam turbine and the condensation of water vapour. The condenser is a tank with absolute pressure range (4–12) kPa inside. It is necessary to keep low pressure in order to provide low temperature steam condensation what is advisable from thermodynamic view. Steam condensation temperature while this low pressure is respectively (28.97–49.42) °C [13].

Condensation of water vapor in a condenser occurs when the temperature of the condensate film is lower than the saturation temperature at a given pressure [6]. Under actual conditions of operation of the condensers, the condensation of water vapor occurs in a solution with a non-condensing gas (air). Then the pressure in the condenser is the sum of the partial pressure of water vapor and the partial pressure of air. The saturation pressure drops with the partial pressure of water vapor, which makes the condensation difficult [1, 4]. The contact of gases with the condensate film of sufficiently low temperature results in condensation of water vapor and accumulation of air around the condensate film. The inert gas causes an additional heat resistance [1, 6]. This significantly influences the heat exchange inside the condenser as described in [3]. In [2] a flow analysis of the tube bank and the share of air in the subsequent rows of tubes was performed not including the heat exchange and partial pressure of the gases.

The paper [8] presents a mathematical model of a steam condenser in changed conditions. A list of independent parameters on which the water temperature at the outlet from the steam condenser depends was selected and by means of the Buckingham $\prod$ theorem a functional relation between two dimensionless quantities was obtained. The exact form of the function was determined by Laskowski [8] on the basis of data from the characteristics of the steam condenser for a 200 MW power plant and actual measurement data for a different condenser operating in a 200 MW power plant. Specific features pertinent to the flow of steam-gas mixture are discussed in paper [9] as applied to the inclined tubes of air-cooled condensers used in steam turbine units, and the condensate film annular flow and 'streamlet' models are analyzed. The calculated results from [9] are compared with the data obtained by other researchers. Artemov *et al.* [9] shown that with relatively small flow rates of steam gas mixture typical for air-cooled condensers, the restoration of pressure along the channel may exceed the losses connected with hydraulic resistance. Data on the influence of cooling air temperature on the condensation process are also presented in [9]. Rhodes and Else's [10] analyzing of condenser performance based on computational fluid dynamics (CFD). The paper [10] has described the background to the modeling approach, and gave examples of water-cooled condensers in which substantial reductions in turbine back pressure have been achieved by implementing modifications assessed using the computer model. The application to air-cooled condenser was demonstrated by way of an example of wind effect on a hypothetical set [10].

The paper [7] presents bank of tubes analysis taking into account gases mixture component pressures and has been fundamental for following publications.

2 The mathematical model

Modeling of gas mixture flow and steam flow in condenser is quite complicated and it requires taking into account the process of bank of tubes flow. We are solving this problem in several stages. The first one is to define parameters. Those values are fundamental to designate heat exchange values. This article describes the first step in modeling process that kind of flow – steam flow through condenser's bank of tubes. In order to keep change of water vapor flowing through condenser, there is adopted simplified assumption that in every tier of tube mounted in condenser the out-dropping mass of steam is identical to mass of steam letting into the bank of tubes. So the stream is dropped out after passing hole bank of tubes. Calculation process presented below is an introduction to modeling heat exchange in another stage of tier of tube. In calculations there were assumed different air mass fraction in inlet mixture to the condenser.

In the calculation, the following dependencies of density and water vapor temperature on the pressure have been assumed [5]

$$p_{iv} = 0.00007p_i + 0.000975 \ , \tag{1}$$

$$T_i = 16.117 \ln(p_i) + 168.45 \ , \tag{2}$$

for $i = 1, ..., m{+}1$. It has been assumed that the mass share of air in the condenser upstream of the first row of tubes was, $k_1 = 0.01$;0.02;0.03;.0.04 or 0.05, the pressure of the mixture of gases at the inlet to the condenser $p_1 = 5000$ Pa and the velocity $c_1 = 50$ m/s. The calculations were performed for $m = 50$ rows of serial and parallel tubes of the external diameter $D_z = 28$ mm and the distances between the tube centers in both directions of $X = 35$ mm (Fig. 1). It has also been assumed that the kinematic viscosity of water and air in the condenser was $\nu_v = 9.8 \times 10^{-6}$ m^2/s and $\nu_a = 19 \times 10^{-6}$ m$^2/s$, respectively [5]. The viscosity of the mixture in the subsequent rows of tubes was calculated based on relation:

$$\nu_i = (1 - k_i)\nu_v + k_i\nu_a \ . \tag{3}$$

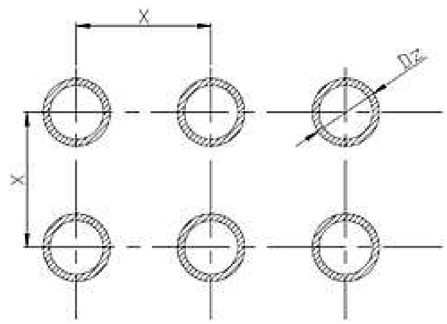


Figure 1. Distribution of the tubes in the tube bank.

The cross section of the analyzed part of the condenser is 1 m $\times$ 1 m, hence the flow area of the gases is A=1 m^2. In the calculations, the authors assumed the value of the gas constants for water vapor $R_v = 461.91$ J/(kg K), and air $R_a = 287.05$ J/(kg K). The authors also assumed that the temperature of the water vapor and air are the same and in accordance with Eq. (2). The calculations of the parameters in the subsequent rows of tubes were made iteratively for i=1,2,...,m+1. The pressure in the ith row of tubes was determined as a difference in the pressure of the previous row of tubes and the drop of pressure following the flow resistance:

$$p_i = p_{i-1} - \Delta p_{i-1} \; . \tag{4}$$

The density of air was calculated based on the Clapeyron equation

$$\rho_{ia} = \frac{p_{ia}}{R_a T_i} \; . \tag{5}$$

It was assumed that on each row of tubes the same mass flow of water vapor is condensed. Downstream of the last row m of tubes the entire water vapor is condensed, hence

$$\dot{m}_i = \dot{m}_{i-1} \left(\frac{\dot{m}_1 (1 - k_1)}{\dot{m}} \right) \; . \tag{6}$$

The share of inert gas was determined using the formula calculating the ratio of the mass of air at the inlet to the condenser to the mass of the mixture in the ith row of the tube bank:

$$k_i = \frac{k_1 \, \dot{m}_1}{\dot{m}_i} \; . \tag{7}$$

The partial pressures of the gases, calculated based on the Dalton's law were given by the formulas [1, 6]

$$p_{ia} = \frac{k_i \, p_i \, R_a}{k_i \, R_a + (1 - k_i) R_v} \; , \tag{8}$$

$$p_{iv} = p_i - p_{ia} \; . \tag{9}$$

The density of the gas mixture can be written as

$$\rho_i = \frac{\rho_{iv} R_v + \rho_{ia} R_a}{R} = \frac{\rho_{iv} R_v + \rho_{ia} R_a}{(1 - k_i) R_v + k_i R_a} \; . \tag{10}$$

The velocity was calculated based on the equation of flow continuity:

$$c_i = \frac{\dot{m}_i}{A \, \rho_i} \; . \tag{11}$$

Hence, the Reynolds number and accumulative coefficient of loses are [2]

$$\mathrm{Re}_i = \frac{D_z c_i}{\nu_i} \; i \, \xi_i = (6 + 9i) \big(\frac{X}{D_z} \big)^{-0.13} \mathrm{Re}_i^{-0.26} \; . \tag{12}$$

The calculation iteration was ended by determining the pressure losses resulting from the flow resistance in the ith row of tubes, according to relation [1]

$$\Delta p_i = \frac{\rho_i c_i^2}{2} (\xi_i - \xi_{i-1}) \; , \tag{13}$$

assuming $\xi_0 = 0$.

Calculation model presented above leads to iterative calculation process which designates characteristic flow quantity. To do this calculation were performed with an input program wrote in FORTRAN software [7].

3 Result and discution

In the tube bank under analysis, a drop of pressure of both gases in the following rows of tubes takes place, as shown in Fig. 2. The greater the mass share of air upstream of the first row of tubes the lower the values of pressure (Fig. 2b). Those diagrams present pressure as another stage of tier of tube m.

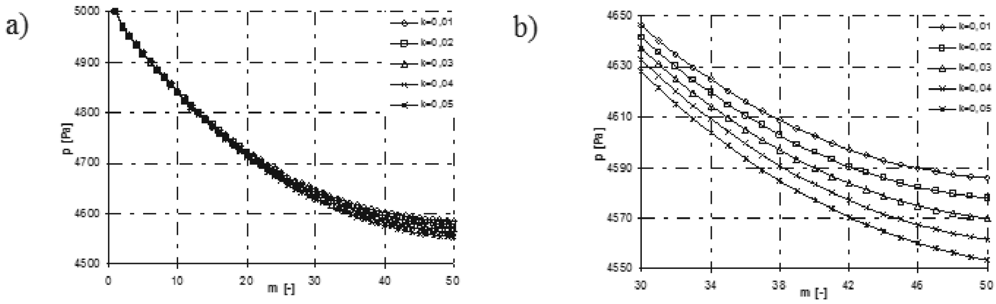


Figure 2. Total pressure in the following rows of the tube bank of the condenser for the initial share of air $k = 0.01$, 0.02, 0.03, 0.04, and 0.05.

The drop in the pressure in the following rows of tubes is getting smaller (Fig.3) and reaches values from 30.8 Pa to 0.2 Pa per each row of the tube bank. The biggest pressure drop is in first tier of tube.

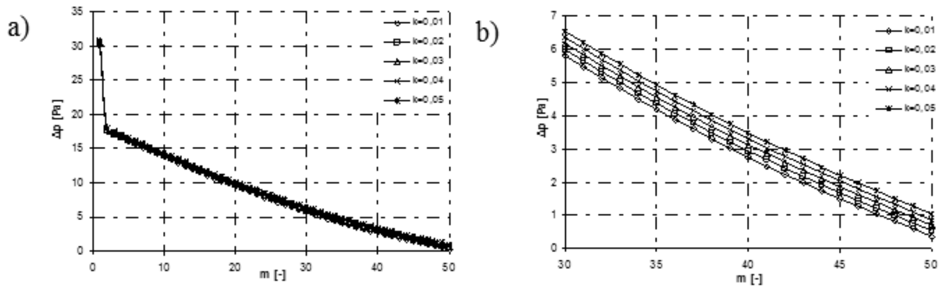


Figure 3. Drop in the total pressure in the following rows of the tube bank of the condenser for the initial share of air $k = 0.01$, 0.02, 0.03, 0.04, and 0.05.

The partial pressure of water vapor in the following rows of tubes drops from 4970 Pa to 3520 Pa for the share of air $k = 0.01$ (Fig. 4a). As the share of air grows upstream of the first row of tubes the drop in the partial pressure is getting greater and reaches values from 4850 Pa to 1760 Pa (Fig. 4a) for $k = 0.05$. The partial pressure of air increases in the following rows of tubes of the tube bank from 30 Pa to 1070 Pa for $k = 0.01$ and from 150 Pa to 2790 Pa for $k = 0.05$, which confirms the accumulation of the inert gas in the final rows of the tube bank.

The temperature in the condenser drops by approximately 1 K from 305.7 K to 304.2 K (Fig. 5a). An increase in the share of air upstream of the first row of tubes results in a greater drop of the temperature inside the condenser (Fig. 5b).

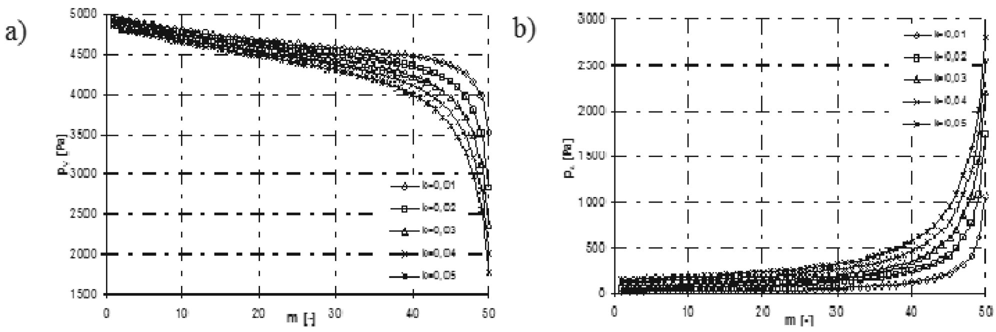


Figure 4. Partial pressure of water vapor (a) and air (b) in the following rows of the tube bank of the condenser for the initial share of air $k = 0.01$, 0.02, 0.03, 0.04, and 0.05.

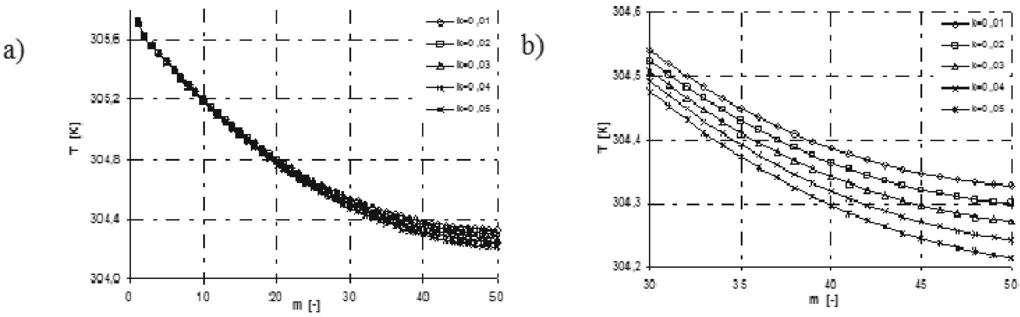


Figure 5. Temperature of gases in the following rows of the tube bank of the condenser the initial share of air $k = 0.01$, 0.02, 0.03, 0.04, and 0.05.

As the gases flow through the condenser, their velocity drops from 50 m/s to almost 2 m/s and the Reynolds number from approximately 140 000 to 3 000 (Figs. 6–7). The values of velocity and the Reynolds number slightly differ for $k = 0.01$–0.05 (mass share of air). The density of gases reaches 0.034 kg/m^3 to 0.046 kg/m^3. As the condensation of the water vapor continues the density of the gases drops from first row of the tube bank to $m = 40$. The density of gases increases in the last ten rows of tubes (Fig. 8). The share of air k increases along with the flow of gases through the model tube bank of the condenser, as shown in Fig. 9. The greater the share of air upstream of the first row of tubes the less abrupt the increase in the share of the inert gas in the final rows of tubes.

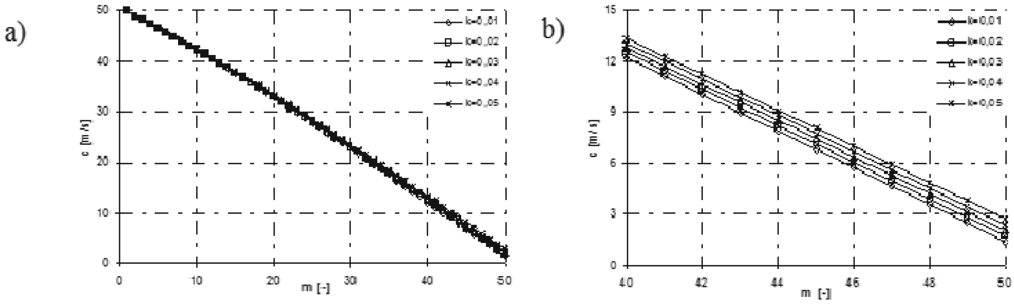
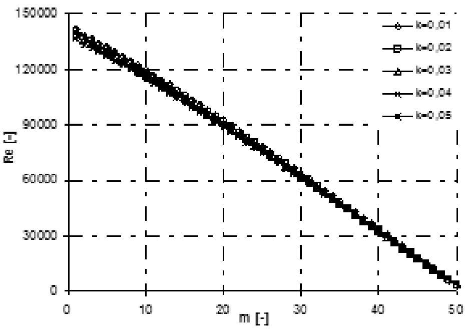


Figure 6. Velocity of gases in the following rows of the tube bank of the condenser for the initial share of air $k = 0.01$, 0.02, 0.03, 0.04, and 0.05.



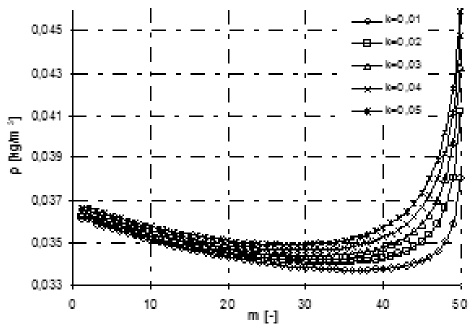


Figure 7. Reynolds number in the following rows of the tube bank of the condenser for the initial share of air k=0.01, 0.02, 0.03, 0.04, and 0.05.

Figure 8. Density of gases in the following rows of the tube bank of the condenser for the initial share of air $k = 0.01$; 0.02; 0.03; 0.04 and 0.05.

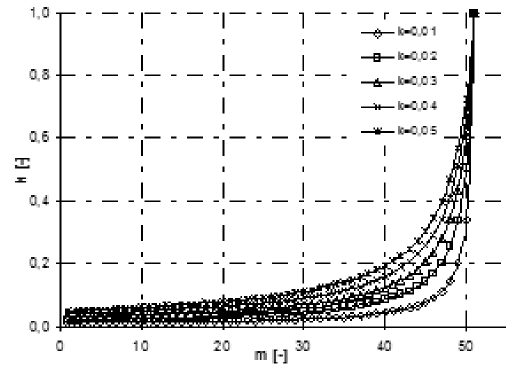


Figure 9. The share of air in the following rows of the tube bank of the condenser for the initial share of air k=0.01, 0.02, 0.03, 0.04, and 0.05.

4 Conclusion

In the paper, a flow analysis of a tube bank taking into account the partial pressures of gases flowing through the condenser have been presented. These calculations are a basis for the heat exchange to be included in further research. The increase in the share of air in the following rows of tubes results in an increase in the thermal resistance and condensation of increasingly lower mass of water vapor. Taking the heat exchange into account we need to note the significant changes of the partial pressures and the resulting changes of the temperatures of saturation and increased thermal resistance.

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