ТЕПЛОЕНЕРГЕТИКА

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THE NUMERICAL MODELING AND MEASUREMENTS FOR POWER PLANT CONDENSER.

Наведено порівняння числових прогнозів двовимірної моделі конденсатора електростанції та експериментальних вимірювань. Виконано аналіз реального пучка труб конденсатора потужністю 50 МВт. Розв'язано рівняння збереження маси і імпульсу з урахуванням частки повітря. Результати числового моделювання містять поля швидкості, тиску і концентрації повітря. Вимірювання визначало різницю між вхідною і вихідною температурою охолоджувальної води.

This paper presents a comparison between numerical predictions of the two-dimensional model of power plant condenser and measurements. The real ribbon tubes bundle of plant condenser of 50 MW is analyzed. The equations governing the conservation of mass, momentum, and air mass fraction are solved. The results of numerical simulation presents velocity, pressure and air concentration fields. The measurements program included determination of difference between inlet and outlet water cooling temperature in tubes.

1. Introduction. A condenser ia an important component that affects the efficiency and performance of power plants. Development of advanced numerical methods for shell-side flows in condensers is a critical step in improving current condenser design techniques. The advantage of numerical simulation is that they can provide a more detailed information on fluid flow and heat transfer in the tube bundle. This information may eliminate, in early stages of the design process, problems related with flow induced vibration and flow distribution and improve overall heat transfer coefficients and increase the unit performance. Until now, these simulation have been carried out by using single-phase two-dimensional model.

2. Mathematical model. The shell-side and water-side flows are treated as steady state and the steam-side flow is assumed to behave as an ideal mixture made up of noncondensable gases and steam only. The steam is taken as satured. The mixture of noncondensable gases and steam asumed to be perfect gas, althought other equations of state could be considered. The two-dimensional stady-state porous medium conservation equations of mass, momentum and air mass fraction with flow, heat and mass transfer resistance are written in Cartesien coordinate system. Local volume porosity β is defined as ratio of the fluid volume and total volume of the corrasponding element [7,8].

Mass conservation equation for the mixture:

$$\frac{\partial(\beta\rho u)}{\partial x}+\frac{\partial(\beta\rho v)}{\partial y}=-\beta\dot{m} \tag{1}$$

Momentum conservation equations for the mixture:

$$\frac{\partial}{\partial x}(\beta\rho uu)+\frac{\partial}{\partial y}(\beta\rho vu)=\frac{\partial}{\partial x}\left(\beta\mu_{eff}\frac{\partial u}{\partial x}\right)+\frac{\partial}{\partial y}\left(\beta\mu_{eff}\frac{\partial u}{\partial y}\right)-\beta\frac{\partial p}{\partial x}-\beta\dot{m}u-\beta F_u \tag{2}$$

$$\frac{\partial}{\partial x}(\beta\rho uv)+\frac{\partial}{\partial y}(\beta\rho vv)=\frac{\partial}{\partial x}\left(\beta\mu_{eff}\frac{\partial v}{\partial x}\right)+\frac{\partial}{\partial y}\left(\beta\mu_{eff}\frac{\partial v}{\partial y}\right)-\beta\frac{\partial p}{\partial y}-\beta\dot{m}v-\beta F_v \tag{3}$$

Conservation of air mass fraction

$$\frac{\partial}{\partial x}\left(\beta\rho u c_g\right)+\frac{\partial}{\partial y}\left(\beta\rho v c_g\right)=\frac{\partial}{\partial x}\left(\beta\rho D_a c_g\frac{\partial c_g}{\partial x}\right)+\frac{\partial}{\partial y}\left(\beta\rho D_a c_g\frac{\partial c_g}{\partial y}\right) \tag{4}$$

where the dependent variables are: velocity components u and v, pressure p, air mass fraction c_g.

The concept of an effective viscosity is used, which is defined as the sum of laminar and turbulent viscosities:

$$\mu_{eff} = \mu + \mu_t \tag{5}$$

For simulation, the turbulent viscosity, is assumed to be constant, which is equal from 100 to 1000 times the value of dynamic viscosity [2].

Steam condensation rate per unit volume m, is determined according to the equation:

$$m\,\mathrm{L}\,\mathrm{V}= a\,(\mathrm{T}-\mathrm{T_w})/R \tag{6}$$

where a is heat transfer area, and R is overall resistance for each control volume, R is the sum of all individual resistances [4]:

$$R = R_w\left(\frac{d_o}{d_i}\right)+R_t+R_f+R_a+R_v \tag{7}$$

For the water side thermal resistance, the Dittus and Boelter [3] relation is employed:

$$\frac{1}{\mathrm{R_w}} = 0.023\cdot\frac{k_w}{d_i}\mathrm{Re}^{0.8}\cdot\mathrm{Pr}^{0.4} \tag{8}$$

The wall resistance for tube R_p is determined by the relation for one-dimensional steady-state conduction, and it is given by:

$$R_t = \frac{\ln\left(\frac{d_o}{d_i}\right)}{2\cdot\lambda_t} \tag{9}$$

The fouling resistance, R_f, is taken as $1.4\mathrm{x}10^{-4}$ $\mathrm{m^2K/W}$, as sugested from experimental by Rusowicz [6]. The fouling deposit formation has significant influence on heat transfer in shell and tube condensers. The investigation [6] has shown that fouling deposit mean heat transfer coefficient h_d in power plant condensers is h_d=25 kW/($\mathrm{m^2K}$). In individual pipes this value was changed from 10 kW/($\mathrm{m^2K}$) to 35 kW/($\mathrm{m^2K}$). Sometimes, in condensers whitout automatic cleanig, after a long period of last manual cleanig the value of the heat transfer coefficient decreases to 0.5 kW/($\mathrm{m^2K}$).

The resistance, to account for condensing steam having to diffuse through an air film close to tube surface, is evaluated by the berman and fuks relation [1] was used.

$$\frac{1}{R_a} = a\frac{D_a}{d_o}\mathrm{Re}_s^{1/2}\left(\frac{p}{p-p_s}\right)^b p^{1/3}\left(\rho_s\frac{L}{T_v}\right)^{2/3}\frac{1}{\left(T_v-T_{sc}\right)^{1/3}} \tag{10}$$

Where a = 0,52, b = 0,7 for Re<350, a = 0,82, b = 0,6 for Re>350

Difusion coefficient D for steam-air mixture is determined on the basis of empirical equation of Fuller

$$D_a = 0{,}00011756552\cdot\mathrm{T}^{1{,}75}\cdot\mathrm{p}^{-1} \tag{11}$$

The fluid flow resistance forces are determined by the linear Darcy law, as follows [5]:

$$F_u = \mu R_u u,\ F_v = \mu R_v v \tag{12}$$

where the flow resistance R_u and R_v are determined by the adequate empirical relations. The equations proposed here are the approximations experimental results for local flow resistance of a two-phase flow across the tube bundle [2]. these equations have the following form:

$$R_u = \mathrm{d_o}^{-2}\,\mathrm{G}\cdot f\cdot\theta_u\quad R_v = \mathrm{d_o}^{-2}\,\mathrm{G}\cdot f\cdot\theta_v \tag{13}$$

where: G is the coefficient which takes in account influence of a tube bundle geometry

$$\mathrm{G} = -1{,}017 + 0{,}3325/\beta + 0{,}3574/\beta^2 + 0{,}01348/\beta^3 \tag{14}$$

The friction factor f is given as:

$f = 350\ \mathrm{Re}^{0{,}0446}$ for Re < 20

$f = 103\ \mathrm{Re}^{0{,}338}$ for 20<Re < 300

$f = 6{,}64\ \mathrm{Re}^{0{,}880}$ for Re > 300

θ_u and θ_v are correction factors which take into account the influence of the condensation rate, i.e. two-phase flow, on the flow resistance forces. they are functions of the condensation rate m, Reynolds number and the direction of the flow (upward or downward).

The boundary conditions for the inlet, vent, walls and baffles and plane of symmetry are:

Inlet: The velocity, pressure and air mass fraction are specified at inlet boundary.

*Walls:*The shell walls and baffles of condenser are assumed to be nonslip, impervious to flow and adiabatic.Thus, the normal velocity components are equal to zero.

Vent: At the outlet of the condenser, the bondary conditions is velocity of steam-air mixture, which is calculated on the basis of the charakteristic of venting apparatus.

Plane of symetry: Along the centre line the derivates which respect to the cross steam direction of all field variables are set to zero.

The mathematical model of steam flow and heat transfer in power plant condenser is solved in two step procedure [2]. In first step system of equations (1)-(3) is solved. As a result of this step the values of mixture velocity and pressure are obtained. In second step, the equation for conservation of air mass fraction (4) is solved. As a result of this step the values of air mass fraction in mixture are obtained. The whole procedure is repeated until the satisfactory accuracy is achieved. In both steps, for solving system equation (1)-(3) and equation (4) the Stream line Upwind Petrov-Galerkin finite element method is applied. The calculations are performed in mesh using of 1982 points and 3686 triangles in the main and crossflow directions, as shown in Fig.1.b. The geometric and operating prarameters for steam condenser are given in Fig.1.a and Table 1.

Table 1

Geometric and operating parameters for a 50 MWe condenser

Geometrical parameters	
Number of Tube Bundles	7160
Number of Tubes Per I Bandle	3640
Number of Tubes Per II Bandle	3520
Condenser Length [m]	6,58
Tube Outer Diameter [mm]	24
Tube Inner Diameter [mm]	22
Tube Pitch [mm]	32
Tube material	brass
Operating parameters	
Inlet Temperature of Cooling Water [^{o}C]	24
Inlet Velocity of Cooling Water [m/s]	1,57
Total Steam Condensation Rate [kg/s]	140 t/h

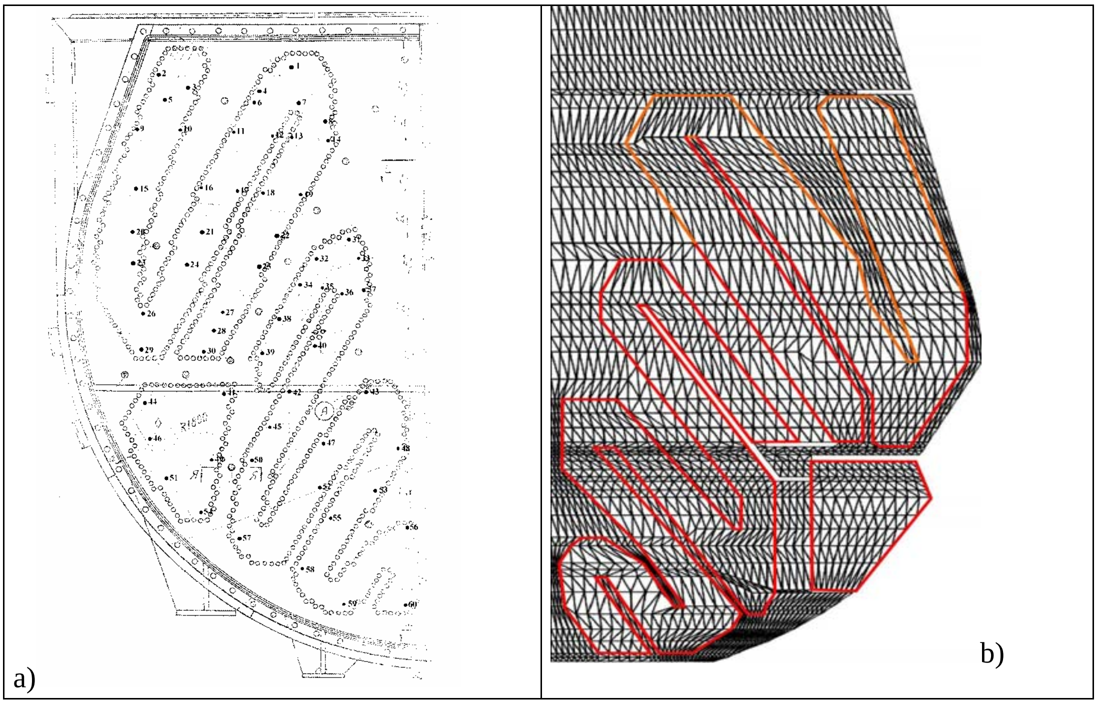

Fig.1. Cross-section of the condenser with the points of measurement, b) grid used for the simulation

3. Presentation of results. The results are presented in the form of plots showing, velocity and pressure contours (Fig.2) and velocity vectors and air concentration contours (Fig.3).

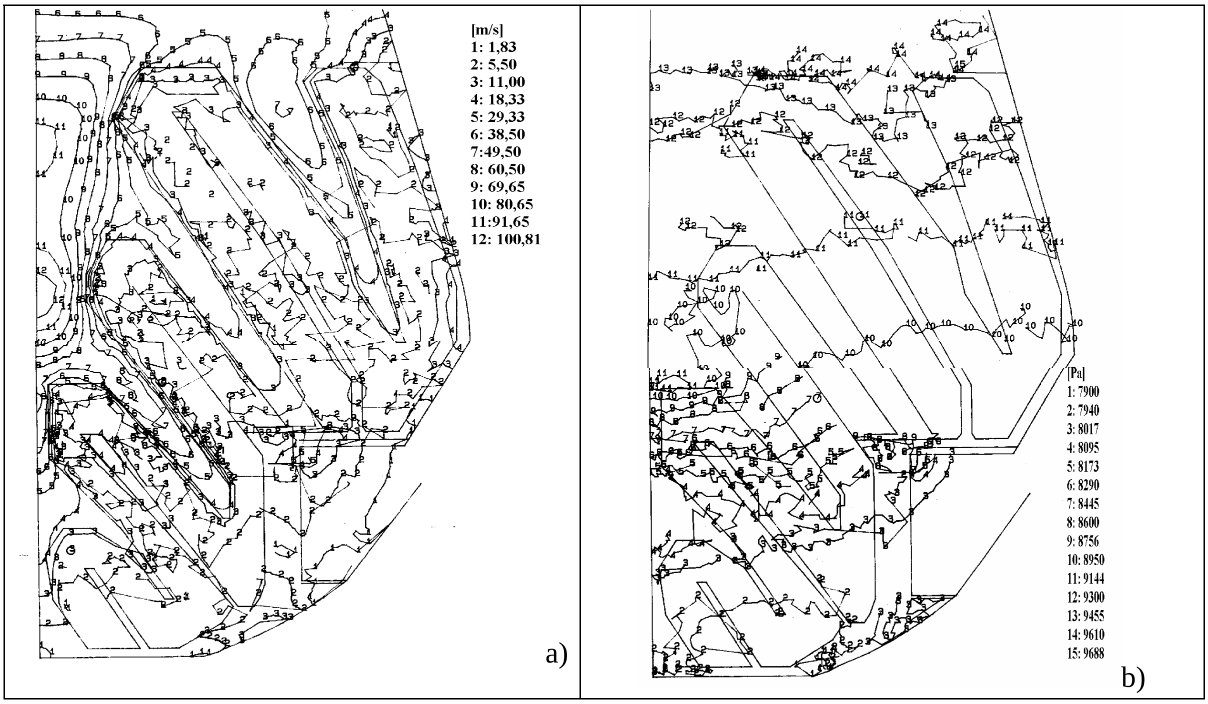

Fig.2. Results of numerical simulation: a)velocity distribution, b) pressure distribution

The air mass fraction contour is given in Fig.3.a, which shows a sharp increase of air mass fraction in the vent region. There is one region of air bubbles, lower part of I tube bundle. Steam velocities are lower in this region. The velocity vector plot is shown in Fig.3.b. It can be seen from this figure that the velocity distribution in the inlet and vicinity of the bundle is nearly "parallel". Table 2 provides comparison between predicted and experimental results for the nominal condensation rate. The measurements program included determination of difference between inlet and outlet water cooling temperature in 60 tubes.

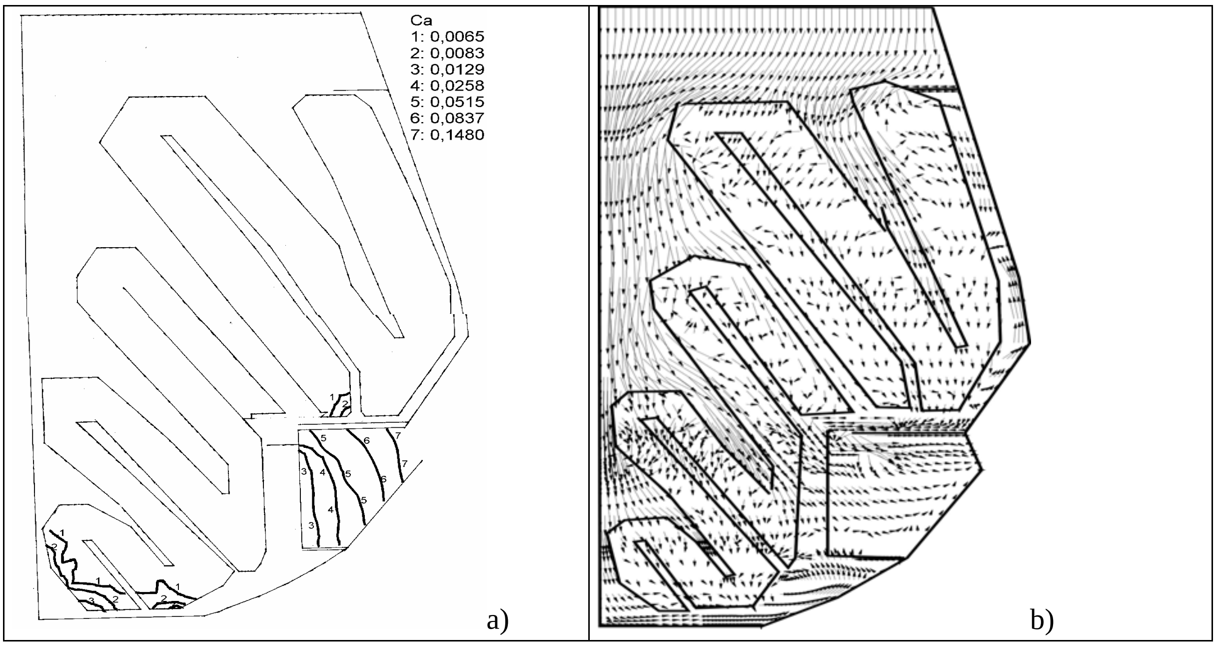

Fig.3. Results of numerical simulation a) air concentration distribution, b) velocity vector plot

Table 2

Comparison of calculated and experimental difference of cooling water temperatures

II Bandle				I Bandle			
Location	Experiment	Calculation	Error [%]	Location	Experiment	Calculation	Error [%]
1	5,7	5,75	0,9	31	7,8	8,49	8,8
2	4,8	5,57	16,0	32	7,8	7,82	0,3
3	5,2	5,90	13,5	33	10,0	8,72	12,8
4	5,4	5,90	9,3	34	10,2	8,38	17,8
5	4,9	5,30	8,2	35	7,3	8,22	12,6
6	5,4	5,80	7,4	36	8,3	8,18	1,4
7	5,0	5,69	13,8	37	9,3	8,61	7,4
8	4,6	5,42	17,8	38	7,3	7,53	3,2
9	4,6	5,36	16,5	39	8,2	8,06	1,7
10	5,0	5,54	10,8	40	9,0	7,84	12,9
11	5,5	5,61	2,0	41	6,8	7,69	13,1
12	5,0	5,63	12,6	42	8,6	7,31	15,0
13	4,9	5,41	10,4	43	9,9	8,60	13,1
14	4,3	5,62	30,7	44	2,7	6,27	132,2
15	5,7	5,50	3,5	45	8,0	8,00	0,0
16	5,9	5,70	3,4	46	2,5	6,54	161,6
17	5,1	5,45	6,9	47	10,7	7,56	29,3
18	4,0	5,35	33,8	48	7,6	7,37	3,0
19	5,4	5,37	0,6	49	7,3	7,11	2,6
20	5,9	5,56	5,8	50	9,4	7,10	24,5
21	5,5	5,36	2,5	51	3,9	6,56	68,2
22	4,2	5,10	21,4	52	8,7	7,22	17,0
23	4,9	4,93	0,6	53	9,0	7,30	18,9
24	5,1	5,28	3,5	54	6,6	6,60	0,0
25	3,9	5,13	31,5	55	7,6	7,01	7,8
26	3,4	5,12	50,6	56	7,8	7,30	6,4
27	4,0	5,03	25,8	57	6,4	6,80	6,2
28	4,3	4,95	15,1	58	7,2	6,80	5,6
29	5,2	4,90	5,8	59	5,9	6,73	14,1
30	3,8	5,19	36,6	60	8,5	6,57	22,7
	Average error		13,9		Average error		21,4

4. Conclusion. Two-dimensional numerical model to predict the fluid flow and heat transfer in condenser has been developed. The simulations have been carried out for real condenser. As a result, comparisons can be concluded that the average errors of measurements and calculated values for I bundle are about 21%, and for II bundle about 13%. However, in case I bundle the largest errors were observed in the area of the outlet air-cooler (points 44, 46, 51). If omitted the comparison of these points, the errors were received at around 10% for II bundle of the condenser. The agreement with the experimental results is good in most regions of the tube region.

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