THE NUMERICAL AND EXPERIMENTAL STUDY OF TWO PASSES POWER PLANT CONDENSER

by

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Original scientific paper
DOI:10.2298/TSCI150917011R

The steam condenser is one of the most important element in whole power plant installation. Their proper design and operation makes a significant contribution to the efficiency of electricity production. The purpose of this article is to propose a 2-D mathematical model that allows modelling condenser work. In the model, the tube bundle is treated as a porous bed. The analysis has been subjected to a two passes power condenser with a capacity of 50 MW. The mathematical analysis was compared with the results of experimental studies. The average error between the model and the experiment for difference of cooling water temperatures was 5.15% and 11.60% for the first and second pass, respectively. This allows to conclude that the proposed model is good enough to optimize future work of the condenser.

Key words: steam condenser, numerical and empirical modelling, 2-D porous model, non-condensable gases

Introduction

A condenser is 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.

Mathematical modelling and heat transfer numerical simulations of power condensers have been carried out for many years by a number of research centres and scientists. Zero-dimensional [1-7], 1-D [8], 2-D [1, 9, 10-15], quasi 3-D [16-18], and even 3-D [19-21] models have been developed.

Zero-dimensional models are based on Heat Exchange Institute and ASME standards and have been developed to date [22-26]. The zero-dimensional models are also employed in more comprehensive power plant models, for which fast calculation methods, providing information on dynamic changes in these plants, are required [27-30]. These models also take into account two basic phenomena affecting the performance of condensers: sediment build-up and sucking inert gases (air) into the condenser. Both these phenomena have adverse effect on the

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condenser and steam turbine performance. Numerous models were proposed [2, 8, 31-35] for assessing how inert gases and sediment affect the condenser performance.

Nevertheless, the zero-dimensional models fail to consider all the factors that influence the condenser performance, such as steam pressure losses in the tube bundle, geometric features of the tube bundle, and the presence of devices which suck the steam and gas mixture. Therefore, in order to provide a more comprehensive analysis of the condenser operation, 1- and 2-D models were created and developed based on the balances of mass, momentum and energy. In addition to these balances, some of the proposed models consider a porous deposit model [36, 37].

In power condensers the nature of steam flow is 3-D: flow parameters vary not only in two directions x-y in the cross-sectional plane of the tube bundle but also along the bundle. Due to the 3-D nature of the flow, a 3-D model is the one that most adequately describes processes occurring in the condenser. However, such a model is complex and requires many parameters and long calculation times, which is why 2-D models are most often used for assessing the power condenser performance. It should be noted that no significant differences in calculation results between the quasi-three- and 3-D models were found. This is supported by papers listed in [17, 38], concerning a small condenser and the shape of a *church window* tube bundle [20, 25]. Hence, for analysing the tube bundle in a condenser the paper employs a 2-D model based on a porous deposit concept.

By the use of the 2-D condenser model, pressure distribution on the steam side can be obtained, and thereby areas with major pressure drops can be localized. In addition, the 2-D model enables the localization of areas where inert gas concentration is higher, *i. e.* the areas within the tube bundle where inert gases accumulate, *air pockets*. In such areas heat transfer processes are considerably inhibited.

For analysing an existing condenser in a 50 MW power unit, its 2-D model was developed based on the porous deposit concept. A large volume of steam, local steam pressure changes, and the presence of inert gases make it difficult to measure the steam pressure [39, 40]. Therefore, the accuracy of the proposed model was assessed based on measurements on the cooling water side, *i.e.* for temperature rises in selected tubes of the condenser. In order to verify the correctness of the condenser design and identify any excess pressure losses and areas where inert gases accumulate, the proposed 2-D model was used to obtain distributions of velocities, pressures, and inert gases.

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 non-condensable gases and steam only. The steam is taken as saturated. The mixture of non-condensable gases and steam assumed to be perfect gas, although other equations of state could be considered. The 2-D steady-state porous medium conservation equations of mass, momentum and air mass fraction with flow, heat and mass transfer resistance are written in Cartesian co-ordinate system.

– 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:

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

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

– conservation of air mass fraction:

$$\frac{\partial(\beta\rho uc_a)}{\partial x}+\frac{\partial(\beta\rho vc_a)}{\partial y}=\frac{\partial}{\partial x}\left(\beta\rho D_a c_a\frac{\partial c_a}{\partial x}\right)+\frac{\partial}{\partial y}\left(\beta\rho D_a c_a\frac{\partial c_a}{\partial y}\right) \tag{4}$$

where the dependent variables are: velocity components u and v, pressure p, and air mass fraction c_g. Local volume porosity, β, is defined as ratio of the fluid volume and total volume of the corresponding element [41]. The concept of an effective viscosity is used, which is defined as the sum of laminar and turbulent viscosities:

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

For simulation, the turbulent viscosity, is assumed to be constant, which is equal from 100 to 1000 times the value of dynamic viscosity [42].

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

$$mLV=\frac{T_s-T_w}{R}A \tag{6}$$

where A is heat transfer area, and R is overall resistance for each control volume – the sum of all individual resistances [20]:

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

For the water side thermal resistance, the Dittus and Boelter [43] relation is employed:

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

The wall resistance for tube R_t is determined by the relation for 1-D steady-state conduction, and it is given by:

$$R_t=\frac{\ln\left(\dfrac{d_0}{d_i}\right)}{2\lambda_t} \tag{9}$$

The fouling resistance, R_f, is taken as $1.4\cdot10^{-4}$ m^{2}K/W, as suggested from experimental by Rusowicz [44]. The fouling deposit formation has significant influence on heat transfer in shell and tube condensers. The investigation [45] has shown that fouling deposit mean heat transfer coefficient h_d in power plant condensers is h_d = 25 kW/(m^{2}K). In individual pipes this value was changed from 10 kW/(m^{2}K) to 35 kW/(m^{2}K). Sometimes, in condensers without automatic cleaning, after a long period of last manual cleaning the value of the heat transfer coefficient decreases to 0.5 kW/(m^{2}K).

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 [46]:

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

where $a = 0.52$, $b = 0.7$ for Re < 350, and $a = 0.82$, $b = 0.6$ for Re > 350.

Diffusion coefficient, D_a, for steam-air mixture is determined on the basis of empirical equation of Fuller:

$$D_a = 0.00011756552T^{1.75}p^{-1} \tag{11}$$

The fluid flow resistance forces are determined by the linear Darcy law, as follows [46]:

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

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 [42]. These equations have the following form:

$$R_u = d_o^{-2} G f \Theta_u, \quad R_v = d_o^{-2} G f \Theta_v \tag{13}$$

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

$$G = -1.017 + \frac{0.3325}{\beta} + \frac{0.3574}{\beta^2} + \frac{0.01348}{\beta^3} \tag{14}$$

The friction factor, f, is given as:

$$f = 350\,\mathrm{Re}^{0.0446} \quad \text{for} \quad \mathrm{Re} < 20 \tag{15a}$$

$$f = 103\,\mathrm{Re}^{0.338} \quad \text{for} \quad 20 < \mathrm{Re} < 300 \tag{15b}$$

$$f = 6.64\,\mathrm{Re}^{0.880} \quad \text{for} \quad \mathrm{Re} > 300 \tag{15c}$$

The 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, $\dot{m}$, 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 non-slip, impervious to flow and adiabatic. Thus, the normal velocity components are equal to zero.

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

Plane of symmetry: Along the centre line the derivate 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 [42]. In first step system of eqs. (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 eq. (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 eqs. (1)-(3) and eq. (4) the stream line upwind

Petrov-Galerkin finite element method is applied. The calculations are performed in mesh using of 852 points and 1542 triangles in the main and cross-flow directions, as shown in fig. 1(b). The geometric and operating parameters for steam condenser are given in fig. 1(a) and in tab. 1.

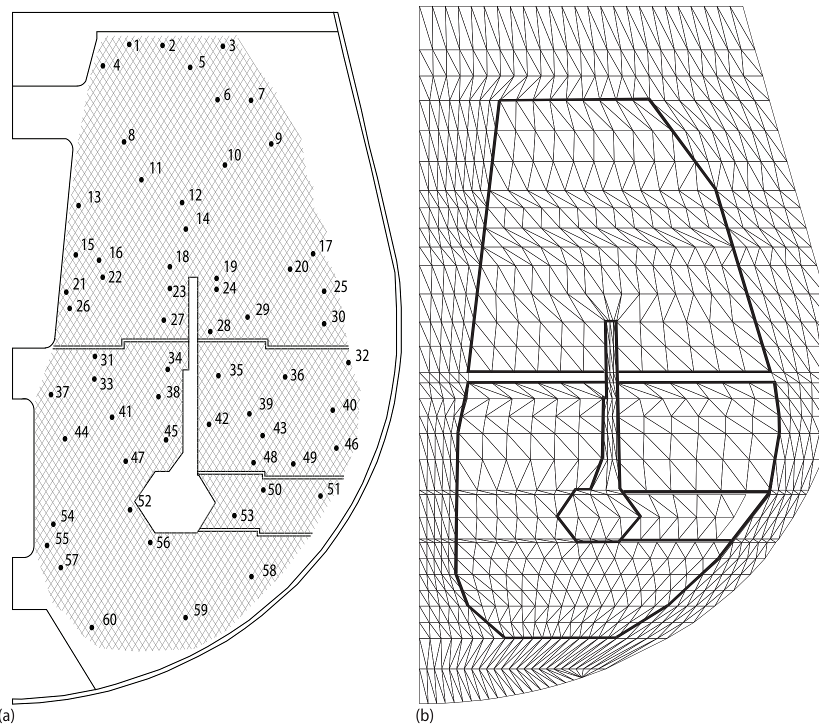

Figure 1. (a) Cross-section of the condenser with the points of measurement, (b) grid used for the simulation

The condenser under consideration has two tube passes. Water flows through the tubes in the lower part (first pass), then it is reversed in the reversing chamber and flows through the tubes in the upper part (second pass) of the condenser.

Table 1. Geometric and operating parameters for a 50 MWe condenser

Geometrical parameters	
Number of tube bundles	7642
Number of tubes per I bundle	3812
Number of tubes per II bundle	3830
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 [t/h]	140

Results

The proposed 2-D model was verified by comparing cooling water temperature rises in 60 tubes, with 30 tubes of the first pass and 30 tubes of the second one. Numbers and locations of the points of measurement across the condenser bundle are shown in fig. 1(a). Table 2 compares values obtained experimentally by measuring water temperature at the inlet and outlet of the tubes considered with the results of numerical simulations.

Table 2. Comparison of calculated and experimental difference of cooling water temperatures

II Bundle				I Bundle			
Location	Experiment	Calculation	Error [%]	Location	Experiment	Calculation	Error [%]
1	5.1	4.97	2.55	31	4.6	4.53	1.52
2	5.2	4.96	4.62	32	5.1	5.04	1.18
3	5.0	4.90	2.00	33	5.7	5.64	1.05
4	4.8	4.46	7.08	34	4.7	5.01	6.60
5	4.7	4.52	3.83	35	5.0	5.18	3.60
6	4.4	4.37	0.68	36	5.1	5.04	1.18
7	5.3	4.24	20.0	37	4.6	5.28	14.78
8	4.7	4.09	12.98	38	5.3	5.36	1.13
9	4.6	3.86	16.09	39	4.5	5.00	11.11
10	4.8	3.88	19.17	40	4.9	5.08	3.67
11	4.9	3.93	19.80	41	5.0	5.36	7.20
12	4.3	3.77	12.33	42	4.8	4.93	2.71
13	5.4	3.74	30.74	43	4.9	5.06	3.27
14	4.3	3.88	9.77	44	5.6	5.60	0.00
15	4.9	3.76	23.26	45	5.0	4.99	0.20
16	4.7	3.68	21.70	46	4.9	4.98	1.63
17	4.9	4.00	18.37	47	4.1	4.25	3.66
18	3.5	3.65	4.29	48	4.7	4.62	1.70
19	3.5	3.70	5.71	49	4.7	4.34	7.66
20	4.0	3.67	8.25	50	5.6	5.17	7.68
21	5.0	3.67	26.60	51	4.4	4.75	7.95
22	4.3	3.62	15.81	52	5.1	4.67	8.43
23	3.5	3.66	4.57	53	5.0	5.04	0.80
24	3.6	3.54	1.67	54	5.6	5.00	10.71
25	4.1	4.05	1.22	55	5.7	5.36	5.96
26	4.9	3.65	25.51	56	4.2	5.11	21.67
27	3.7	3.60	2.70	57	5.0	4.90	2.00
28	3.8	3.62	4.74	58	5.4	5.26	2.59
29	3.1	3.53	13.87	59	4.5	4.99	10.89
30	4.3	3.96	7.91	60	4.6	4.69	1.96
	Average error		11.59		Average error		5.15

By comparing the values measured with those obtained from the 2-D model, we can consider that the model describes the processes in the condenser with a satisfactory accuracy. The average errors of measurements and calculated values for I bundle are about 5,15%, and for II bundle about 11,6%. The agreement with the experimental results is good in most regions of the tube region. Larger deviations in the second pass may result from insufficient mixing of water in the reversing head and from slightly different velocities of water in the tubes due to variations in water pressure at the tubes inlet.

The results of the simulation based on the 2-D model for the condenser of the 50 MWe power unit are presented in the form of plots showing velocity and pressure contours, fig. 2, and velocity vectors and air concentration contours, fig. 3.

Figure 2(a) shows the steam velocity field, steam velocities vary from 80-87 m/s to 3 m/s. The highest steam velocities occur in the channel beside the symmetry axis, in the middle of the

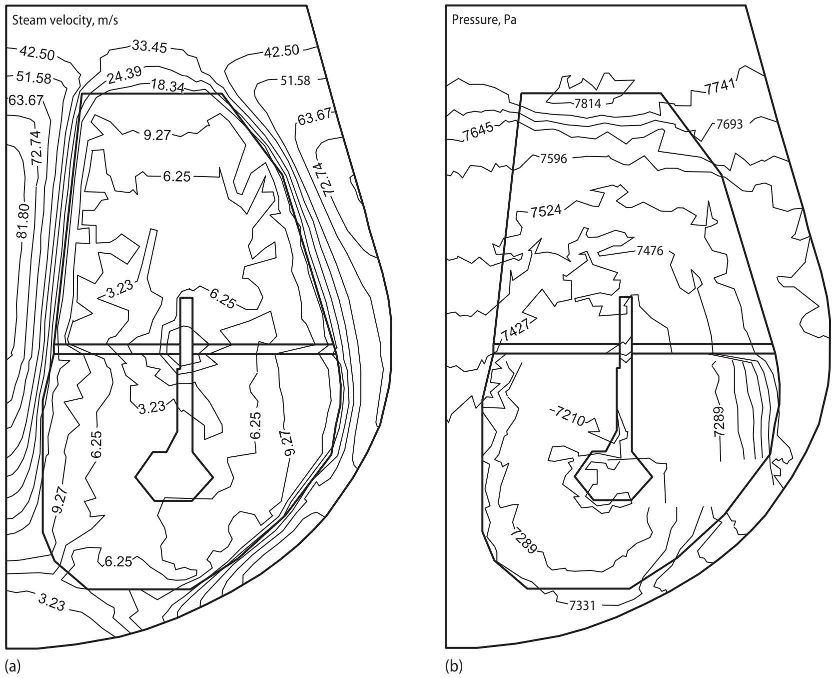

Figure 2. Results of numerical simulation: (a) velocity distribution, (b) pressure distribution

second pass, and by the condenser wall. At the edge of the tube bundle a high rate of deceleration of steam flowing from the channels to the bundle area, from 25-30 m/s to about 18 m/s, is observed. In the bundle body a rather uniform velocity of 6-9 m/s is found. The velocity of steam both in the channel and flowing onto the bundle can be reduced by applying a shell of a larger cross-section, as is the case with *church window* condensers. Additionally, in order to protect the tubes from high steam velocities at the bundle edge, two rows of tubes with thicker walls can be used.

Figure 2(b) shows the pressure field across the condenser cross-section. Based on measurements, pressure of 7778 Pa at the condenser inlet was assumed. The increase in pressure by about 400 Pa is observed in the area where steam flows into the middle of the upper part of the second pass. The pressure distribution features bands, gradual pressure drop in the channel and bundle can be seen. A pressure drop of about 350 Pa can be found across the first pass of the tube bundle, and of about 400 Pa across the second one. The shape of the bundle is correctly selected, as uniform pressure drops in both passes can be seen. This should also provide uniform steam inflow to the two condenser passes.

Figure 3(a) illustrates the distribution of calculated inert gas concentrations. A major increase in concentration can be seen at the outlet of the sub-cooler. A slight increase in the concentration of these gases is observed also in the middle part of the condenser. A way to decrease the concentration of inert gases in this area could be replacing trays around the middle channel with trays having numerous openings. This improves sucking inert gases and limits flooding the channel with condensate.

Figure 3(b) is intended to verify whether, according to the physical characteristics of the phenomenon, the steam spreads correctly across the condenser cross-section and there are no areas where steam stops.

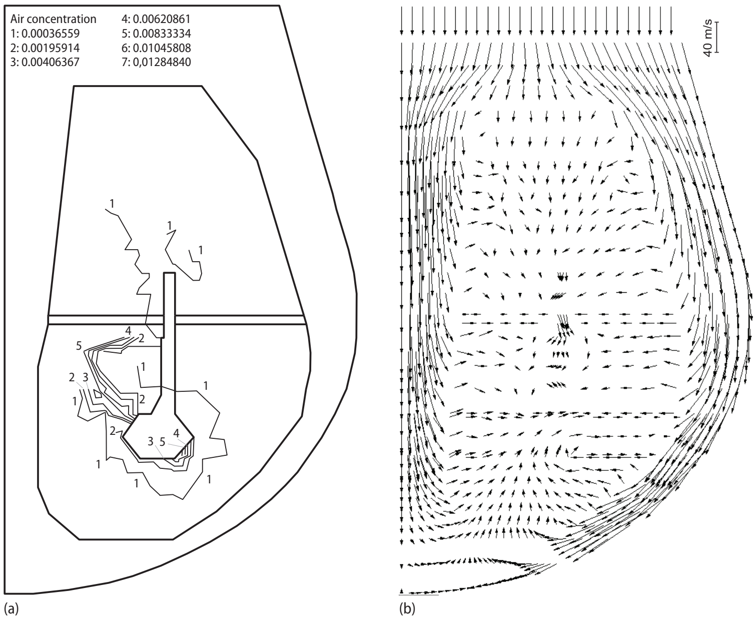

Figure 3. Results of numerical simulation (a) air concentration distribution, (b) velocity vector plot

Conclusions

A 2-D numerical model to predict the fluid flow and heat transfer in a condenser was developed based on a porous deposit concept. Simulations of an existing condenser operating in a 50 MWe power unit were carried out. The proposed model was verified against measurements of cooling water temperatures at the inlet and outlet of selected tubes. The agreement with the experimental results is good in most regions of the tube region. The comparison allows to draw a conclusion that the average errors of measurements and calculated values for the first pass are about 5.15%, and for the second pass about 11.6%.

The analysis employing the proposed 2-D model provided a series of quality and quantity information concerning the condenser, allowing the design of a more effective condenser. Uniform pressure drops in both passes of the condenser were observed. Areas where steam velocities are the highest were identified. In these areas channels can be widened for the steam flow, or the tubes in the bundle can be reinforced by using tubes with thicker walls. Additionally, the area with higher concentration of inert gases within the first pass was identified. In order to eliminate this area, the collecting channel in the middle of the bundle was re-designed: the enclosure of the channel was replaced with perforated sheet.

Nomenclature

A – heat transfer area, [m^2]
c_a – air mass fraction, [–]
D_a – diffusivity coefficient of air in vapour, [m^{2}s^{-1}]
d_i – inner diameter of tube, [m]
d_o – outer diameter of tube, [m]
F_u, F_v – flow resistance forces in momentum equations, [Nm^{-3}]
f – friction factor, [m]
L – latent heat of condensation, [Jkg^{-1}]
$\dot{m}$ – total steam condensation rate, [kgs^{-1}]
Pr – Prandtl number, [–]
p – pressure, [Pa]
R – thermal resistance, [m^{2}KW^{-1}]
Re – Reynolds number, [–]
T – temperature, [°C]

u – velocity component in the x-direction, [ms^{-1}]
V – volume, [m^3]
v – velocity component in the y-direction, [ms^{-1}]
x – main flow direction co-ordinate, [m]
y – cross-stream co-ordinate, [m]

Greek symbols

β – local volume porosity, [–]
λ – thermal conductivity, [$Wm^{-1}K^{-1}$]
μ – dynamic viscosity, [$kgm^{-1}s^{-1}$]
ρ – mass density, [kgm^{-3}]

Subscrips

a – air
eff – effective
f – fouling
i – inner diameter
o – outer diameter
s – steam
sc – surface
t – turbulent, tube
w – water

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Paper submitted: September 17, 2015
Paper revised: January 9, 2016
Paper accepted: January 20, 2016