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DETERMINATION OF THE LOCAL PARAMETERS OF HEAT- AND MASS TRANSFER IN DEEP COOLING IN A TUBE BUNDLE OF FUEL COMBUSTION PRODUCTS OF THERMAL POWER PLANTS

V. I. Baikov, V. A. Borodulya,
V. L. Malevich, and A. E. Sinkevich

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The specific features of heat and mass transfer of flue gases deeply cooled below the dew point of the steam contained in them have been analyzed. A method of determining the local parameters of heat and mass transfer in surface condensation heat utilization units is suggested. An analysis of the local characteristics of cooled and condensing combustion products allows one to study the relationship between the heat and mass transfer parameters and justify the optimum geometric characteristics of condensation heat utilization units.

Keywords: *vapor-gas flow, heat utilization, heat and mass transfer, condensation.*

Introduction. The poor provision of Belarus with its own energy carriers requires an increase in the efficiency of using natural gas. In present-day boiler plants, a considerable portion of heat (>10%) is lost with effluent gases, whose temperature attains 110–150^{o}C [1, 2]. These losses can be considerably reduced by deep cooling of combustion products below the dew point in surface condensation heat-utilization units.

The methods of calculation and analysis used at the present time by various authors [3, 4] are based on the determination of the average parameters of heat and mass transfer in cooling the products of combustion and condensation of steam, contained in them, on finned surfaces. The local parameters of heat and mass transfer along the length of the tube and in depth of a tube bundle may differ considerably from average ones, and this exerts its influence on the determination of the basic geometric characteristics of heat-utilization units.

Statement of the Problem. A flux of a vapor-gas mixture entering a heat exchanger contains up to 90% of noncondensable gases. The main components of the flue gases outgoing from boilers are steam, carbon dioxide, and nitrogen. Based on the fact that nitrogen is the main noncondensable component, it is worthwhile to computationally reduce the totality of the outgoing gases to a binary vapor-gas mixture.

The process of heat removal from outgoing flue gases can be conventionally divided into two stages. At the first stage, the combustion products are cooled to the temperature of their dew point; this corresponds to convective heat transfer without a change in the moisture content of the flow and at the second stage, there occurs deep cooling of the vapor gas mixture with condensation of water vapor contained in the flow. This sequence of processes is conventional and, as will be shown below, condensation of a superheated vapor actually takes place in the bundle.

The presence of noncondensable gases substantially alters the mechanism of the process of condensation heat- and mass transfer. As a result of the removal of thermal energy from the flow during condensation, there always exists a convective gas flow to the phase interface that causes accumulation of the noncondensable components. As a result, on the phase interface, an excessive concentration (as compared with the bulk one) of noncondensable components is established that satisfies the condition of equality of convectively supplied and diffusely removed fluxes:

$$g_k = \mathbf{j}_k \,, \quad g_k = \rho_{k0\text{int}} \frac{dG}{dx} \,, \quad \mathbf{j}_k = \beta_k \left(\rho_{k0\text{int}} - \rho_{k0}\right) .$$

As a result, the partial pressure of condensed vapor on the interface decreases as compared to the volume, and the saturation (condensation) temperature, which depends on the partial pressure, decreases. This in turn leads to a de-

A. V. Luikov Heat and Mass Transfer Institute, National Academy of Sciences of Belarus, 15 P. Brovka Str., Minsk, 220072, Belarus; email: bor@itmo.by; malevich@telegraf.by. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 82, No. 2, pp. 289–295, March–April, 2009. Original article submitted October 8, 2008.

crease in the temperature head (under the same cooling conditions) and, as a consequence, to a decrease in the heat transfer intensity. The decrease in the saturation temperature on the phase interface as compared to the saturation temperature in the flow leads to general condensation of the superheated vapor. The process of vapor condensation from a vapor-gas mixture in a bundle of tubes will favor an increase in the concentration of noncondensable gases in a flow, which in turn causes weakening of the condensation process.

The indicated features of the condensation of a vapor-gas mixture lead to great mathematical difficulties in a rigorous description of the process. Therefore an analysis of heat and mass transfer in deep cooling of the combustion products of thermal power plants is carried out in the present work on the basis of a one-dimensional model that describes the process of vapor-gas mixture condensation in a tube bundle.

Mathematical Model. The second-order partial differential equations of heat and mass transfer are transformed into a system of first-order ordinary differential equations that describe a lengthwise change in the radius-averaged condensing flow parameters. The transformed conservation equations are supplemented by thermal boundary conditions and by expressions of coupling between the concentrations of components.

In [5–7], the initial two-dimensional conservation equations of the convective diffusion of the kth component

$$\rho W_x \frac{\partial \rho_{k0}}{dx} + \rho W_r \frac{\partial \rho_{k0}}{\partial r} = \frac{1}{r}\frac{\partial}{dr}(-r\mathbf{j}_k)$$

and of energy

$$\rho W_x \frac{\partial h}{dx} + \rho W_r \frac{\partial h}{\partial r} = \frac{1}{r}\frac{\partial}{dr}(-rq_{\text{ef}})$$

with allowance for the introduction of the parameters of averaging over the section and use of the continuity equation

$$\frac{\partial(\rho W_x)}{\partial x} + \frac{1}{r}\frac{\partial}{\partial r}(r\rho W_r) = 0 \tag{1}$$

and the condition of the impermeability of the kth noncondensable component at the interface

$$\rho_{k0}\rho_{\text{int}}\left(-W_r + W_x \frac{dr_{\text{int}}}{dx}\right)_{\text{int}} = \mathbf{j}_{k\text{int}} \tag{2}$$

are transformed into a system of first-order ordinary differential equations:

$$\frac{d\bar{\rho}_{k0}}{dx} = -\frac{\bar{\rho}_{k0}}{G}\frac{dG}{dx}, \tag{3}$$

$$\frac{d\bar{T}}{dx} = \frac{1}{c_p G}\left[-\Pi_{\text{int}} q_{\text{int}} - (\bar{h} - h_{\text{int}})\frac{dG}{dx}\right]. \tag{4}$$

Equations (3) and (4) were obtained for an axisymmetric problem, and they describe a change in the concentration and temperature over the channel length. Having divided both equations by the channel perimeter, we will pass from the change of parameters along the length to a change over the surface $\Pi dx = dF$. Then Eqs. (3) and (4) will take the form

$$\frac{d\bar{\rho}_{k0}}{dF} = \frac{\bar{\rho}_{k0}}{G}\frac{dG'}{dF}, \tag{5}$$

$$\frac{d\bar{T}}{dF} = \frac{1}{c_p G}\left[-q_{\text{int}} + (\bar{h} - h_{\text{int}})\frac{dG'}{dF}\right]. \tag{6}$$

The mechanism of the process of heat- and mass transfer in condensation from a vapor-gas mixture is independent of the shape of the channel but is determined by the conditions of the removal of heat through the surface and therefore Eqs. (5) and (6) can be applied also to the outer problem, where the element of the surface dF for a finned tube is defined as $dF = 2\pi d_{\text{out}} dL\varphi$.

Equations (5) and (6) are closed by the expressions for determining the heat and mass fluxes.

A change in the flow rate (intensity of condensation) is

$$\frac{dG'}{dF} = \frac{q_{\text{ph}}}{\Delta i} . \tag{7}$$

The intensity of condensation is related to a decrease in the vapor phase flow rate by the relation

$$\frac{dG'}{dF} = -\frac{dG}{dF} .$$

The phase transition heat flux is

$$q_{\text{ph}} = q - q_{\text{vg}} , \tag{8}$$

where $q = k(T_{\text{s.int}} - T_{\text{liq}})$ is the heat flux on the wall; $q_{\text{vg}} = \alpha_{\text{vg}}(\overline{T}_{\text{vg}} - T_{\text{s.int}})$ is the heat flux due the superheating of the vapor-gas mixture relative to the saturation temperature.

The calculation of the coefficients of convective heat transfer of a vapor-gas flow and water are made from dimensionless relations recommended in [8].

The mass transfer coefficient is determined on the basis of the triple analogy [3] by the relation of the form

$$\beta_k = \alpha_{\text{vg}} \left(\frac{\text{Sc}}{\text{Pr}} \right)^{0.43} \left(\frac{D_2}{\lambda} \right), \tag{9}$$

where $\text{Sc} = \mu / D_2$; $D_2 = \rho D_{12}$; $D_{12} = D_0 (T/T_0)^{1.8}$; $D_0 = 2.16 \cdot 10^{-4}$ at $T_0 = 273.16$ K.

From the impermeability condition (2) the mass concentration of the noncondensable component on the interface surface is determined:

$$\rho_{k0\text{int}} = \frac{\rho_{k0}}{1 - \frac{dG}{dF} \Big/ \beta_k} . \tag{10}$$

The partial vapor pressure on the interface or the saturation pressure is

$$P_{\text{s.int}} = P m_{\Sigma} \frac{\rho_{10\text{int}}}{m_{10}} , \quad \rho_{10\text{ï}} = 1 - \rho_{k0\text{int}} .$$

The saturation temperature of the interface is defined as the function of the partial vapor pressure on the interface $T_{\text{s.int}} = f(P_{\text{s.int}})$.

The heat transfer coefficient in condensation on a bundle of finned tubes can be determined from the data of [9] with allowance for the influence of the finning parameters on heat transfer:

$$\alpha' = \psi_{\text{f}} \varphi \alpha'_{\text{out}} , \tag{11}$$

where α'_{out} is the heat transfer coefficient defined by the Nusselt formula for condensation on the outside surface of the tube [9].

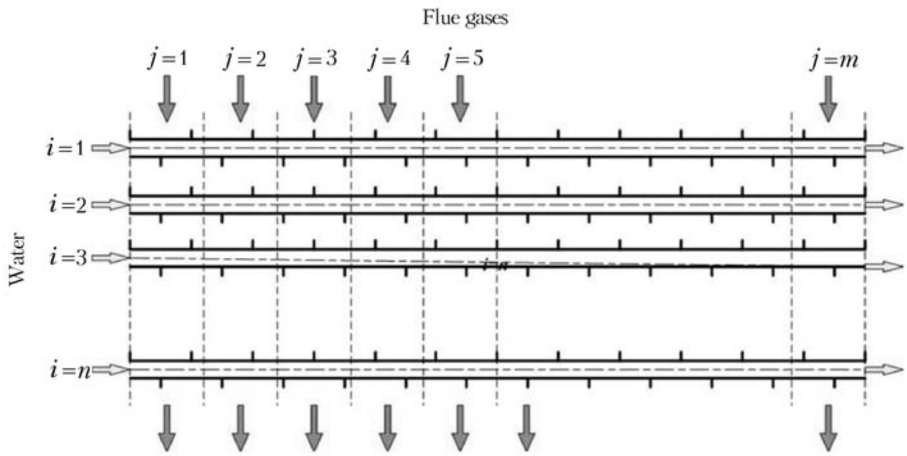

Fig. 1. Computational scheme of the heat utilization unit.

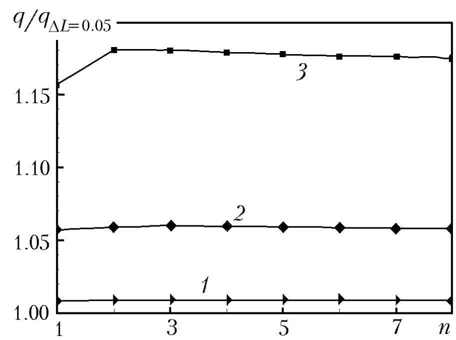

Fig. 2. Influence of the length of the integration section in a many-row bundle on the relative value of the specific heat flux: 1) $q_{\Delta L=0.2}/q_{\Delta L=0.05}$; 2) $q_{\Delta L=1.0}/q_{\Delta L=0.05}$; 3) $q_{\Delta L=2.5}/q_{\Delta L=0.05}$.

Having determined the values of the thermal and mass fluxes from Eqs. (8)–(11), we can easily calculate the value of the intensity of condensation from Eq. (7) and the values of the derivatives $d\rho_{k0}/dF$ and dT/dF from Eqs. (5) and (6). These derivatives allow one to calculate new values of the parameters in the flow in passing from row to row in the depth of the bundle:

$$\rho_{k0}^* = \rho_{k0} + \frac{d\rho_{k0}}{dF}\Delta F\,, \quad T^* = T + \frac{dT}{dF}\Delta F\,, \quad G^* = G + \frac{dG}{dF}\Delta F\,.$$

Discussion of Results. Based on the suggested mathematical model, the procedure and computational program have been worked out for calculating the surface condensation unit that utilizes the heat of outgoing flue gases. The procedure is based on the determination of the local characteristics of a cooled vapor-gas flow and of heated water.

The simplified computational scheme of the heat-utilization unit is presented in Fig. 1. A vertical row consisting of n tubes along the vapor-gas flow is considered. Along the tubes, the heat exchanger is divided into a finite number of m sections.

As the initial data we adopt: a) the velocity of the incoming flow of flue gases, their temperature, and composition at the inlet into the heat exchanger; b) the velocity of the heated water and its temperature at the inlet to the tube, and c) the parameters of the tube bundle, tubes, and finning, the type of the material of tubes and fins.

Calculations are performed successively for each element (i, j). The tabulated data on the thermophysical properties of water [10] and flue gases [11] are approximated by polynomials within the ranges 0–100°C for water and 0–300°C for flue gases, and they are determined from the average temperatures on each element refined by the iteration method.

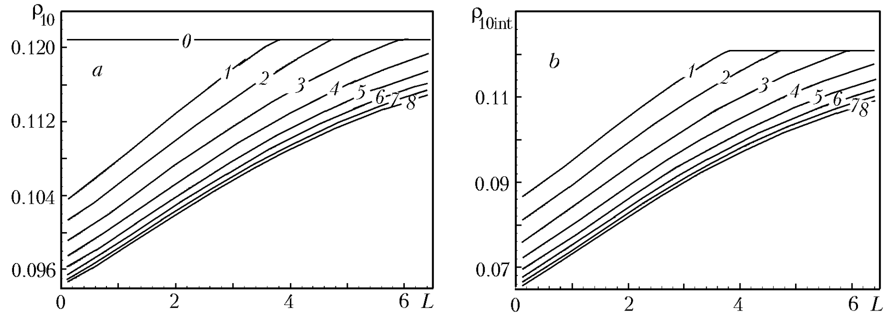

Fig. 3. Change in the concentration of water vapor in a flow (a) and on the phase interface (b) over the rows of tubes and along the length (0 is the concentration at the flow inlet; 1, 2, ..., 8 denote the number of a row). L, m.

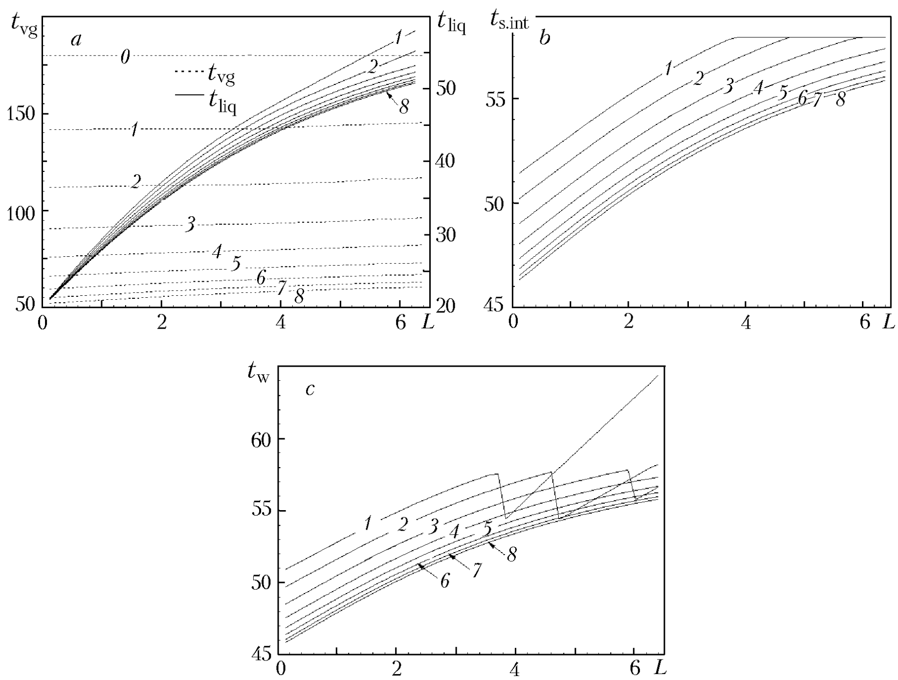

Fig. 4. Local distributions of the temperatures of a vapor-gas flow and water (a), of saturation on the interface (b) and of the tube wall (c) over the rows (1, 2, ..., 8) and along the length of tubes. t, ^{o}C; L, m.

Figures 2–6 present the results of calculation for the 8-row bundle of bimetallic tubes with crosswise helical finning of staggered arrangement for a flow of flue gases with the inlet temperature of 180^{o}C, velocity at the front of the bundle 1.9 m/sec, water with an inlet temperature of 20^{o}C, and tube inlet flow velocity of 1 m/sec. The carrying tube is made of steel, and the finning is made by knurling an aluminum tube fitted on the carrying one. The finning coefficient is ~13.

In order to determine the influence of the length of the integration portion on the output parameters of heat and mass transfer, a series of calculations with its different values were made. As a result of the analysis of the data obtained, the section $\Delta L = 0.05$ m was taken as a standard. It is seen from Fig. 2 that the choice of the value of ΔL can substantially influence the results of calculation of the heat exchanger. At $\Delta L = 2.5$ m the heat flux is on the average 18% higher than at $\Delta L = 0.05$ m. The difference in the results at $\Delta L = 0.05$ and 0.2 m amounts to about 1%. Thus, the local computational model allows one to obtain more accurate results. The data presented in Figs. 3–6 were obtained at the length of the integration section equal to $\Delta L = 0.125$ m.

A change in the concentration of water vapor in the vapor-gas flow (Fig. 3a) and on the phase interface (Fig. 3b) over the rows of tubes and along their length shows that accumulation of noncondensable gas components

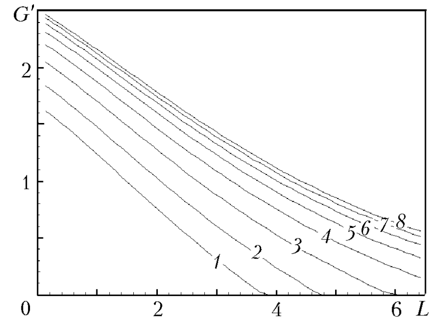

Fig. 5. Local distribution of the condensed water vapor flow rate over the rows (1, 2, ..., 8) and along the length of tubes. G', mg·sec/m^2; L, m.

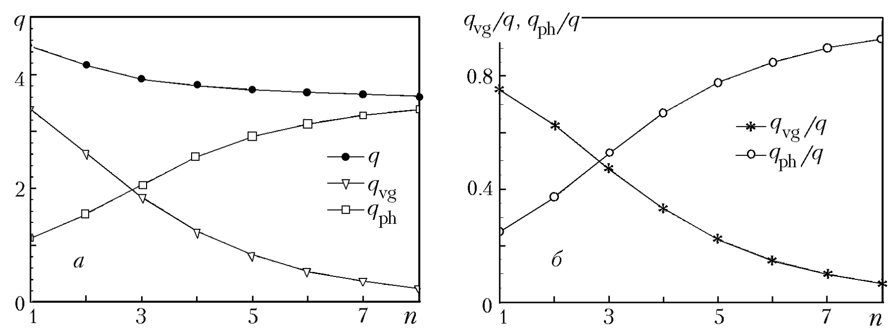

Fig. 6. Distribution of the specific heat fluxes (a) and the ratio of the heat fluxes of overheating and phase transition (b) over the rows of tubes (q_{vg} is the superheating heat flux; q_{ph} is the phase transition heat flux, $q = q_{vg} + q_{ph}$). q, kW/m^2.

on the phase interface decreases the vapor concentration. The process of condensation leads to accumulation of noncondensable components near the phase interface, as a result of which the concentration of water vapor on the phase interface is smaller than in the flow. From the data given in Fig. 3 it follows that, for example, for the section of the start of condensation on the third row the concentration of water vapor on the phase interface is approximately 30% smaller than in the flow and also that the concentration of water vapor from row to row decreases, whereas along the tube length it increases as the cooling water is heated. The concentration of water vapor on the phase interface determines the partial pressure and, as a consequence, the saturation temperature.

The local distributions of the temperatures of the vapor-gas flow and of heated water, the saturation temperature, and the tube wall temperature are presented in Fig. 4. It is seen from Fig. 4a that at an inlet temperature of flue gases of 180^{o}C it amounts on the average to about 55^{o}C at the outlet of the eighth row, which is lower than the dew point at the inlet of the apparatus (58^{o}C). On the first row, the temperature of the vapor-gas flow decreases by 35–38^{o}C, and on the eight row by about 3^{o}C. The water is heated by 38^{o}C in the first row and by 31^{o}C in the eighth row, which corresponds to a decrease in the thermal power of the eighth row by 18%, as compared to the first row.

The distribution of the saturation temperature repeats the profiles of the change in the concentration of water vapor on the phase interface. The saturation temperature increases along the tube length as the cooling water is heated.

A joint analysis of the data of Fig. 4a and b shows that practically for each row of tubes there occurs condensation of the superheated water vapor. The superheating heat depends on the difference between the gas temperature in the flow and the saturation temperature. Thus, for the first row the temperature of the flue gases at the outlet changes along the tube length from 141 to 143^{o}C, whereas the saturation temperature changes from 48 to 57.4^{o}C, which corresponds to a superheating of about 90^{o}C. For the fourth row the superheating amounts to about 26^{o}C, and for the eighth one to 4–5^{o}C.

Figure 4c presents the distribution of the temperature of the outside wall of the tubes. A comparison of the data presented in Fig. 4b and c shows that the wall temperature during condensation is lower than the corresponding saturation temperature, which precisely ensures the process of vapor condensation from a vapor-gas mixture. For the 1–3rd rows of tubes, when the temperature of the vapor-gas flow is still high and the water temperature approaches the saturation one in view of the stoppage of condensation, one observes a certain decrease in the wall temperature because of the convective removal of the mixture superheating heat.

The distribution of the specific flow rate of condensed water vapor over the rows of rubes with allowance for the accumulation of condensate on the previous rows is presented in Fig. 5. On the first row the process of condensation ceases at a length of 3.8 m, and further only the cooling of the vapor-gas flow occurs. On the succeeding rows of tubes this boundary is displaced to a greater length, and beginning from the fourth row, vapor condensation occurs along the entire length of the tubes. A change in the total condensate flow rate along the length of the tube bundle corresponds to the values given for the 8th row.

The integral parameters that characterize the ratio of heat fluxes over tube rows (Fig. 6) show that on the first and second rows the main portion of the total heat flux is attributable to the convective superheating flow (60–75%); on the third row the convective and condensation flux are commensurable, and on the 5th–8th rows the condensation flux amounts to 77–93%.

Conclusions. A mathematical model of the process of heat and mass transfer on deep cooling of fuel combustion products in a condensation heat-utilization unit from bimetallic tubes with external helical finning exposed to a transverse vapor-gas flow has been worked out. The computer program written on the basis of the model allows one to determine the local parameters of heat and mass transfer along the length of each of the tubes of the heat utilization unit.

The calculations performed have shown that virtually on all the rows of tubes there occurs vapor condensation from a vapor-gas flow. On the first two rows the main portion of the removed heat flux is attributable to the convective component due to the superheated vapor, whereas on the fourth and subsequent rows the main contribution to the utilized power is made by the condensation of water vapor from flue gases.

NOTATION

c_p, specific isobaric heat capacity, J/(kg·K); d, diameter of a tube, m; dL, length of the surface element dF, m; D_0, D_{12}, coefficient of diffusion, kinematic at T_0 = 273.16 K and at the vapor-gas flow temperature, respectively, m^2/sec; D_2, dynamic coefficient of diffusion, kg/(m·sec); F, surface, m^2; g, convective heat flux, W/m^2; G, flow rate, kg/sec; h, enthalpy, J/kg; $\mathbf{j}$, diffusion heat flux, kg/(m^2·sec); k, heat transfer coefficient, W/(m^2·K); L, length, m; m, number of elements along the tube length; m_{10}, molecular mass of water vapor; m_Σ, molecular mass of vapor-gas mixture; n, number of rows of tubes over the depth of a tubular bundle; P, pressure, Pa; Pr, Prandtl number; q, heat flux, W/m^2; r, radial coordinate; Sc, Schmidt number; T, temperature, K; t, temperature, ^{o}C; W, velocity, m/sec; x, axial coordinate; α, heat transfer coefficient, W/(m^2·K); β, mass transfer coefficient, m/sec; ΔF, surface element, m^2; ΔL, length of calculated element, m; Δi, phase transition heat, J/kg; φ, finning coefficient; λ, thermal conductivity, W/(m·K); μ, coefficient of dynamic viscosity, Pa·sec; Π, perimeter, m; ρ, density, kg/m^3; ρ_{k0}, $\rho_{k0\mathrm{int}}$, relative mass concentration of uncondensing component in a flow and on the interface, respectively; ρ_{10}, $\rho_{10\mathrm{int}}$, relative mass concentration of water vapor in a flow and on the interface, respectively; ψ_f, ratio of the surface of fins to the total surface of the finned tube. Subscripts: ef, effective; f, fin; int, phase interface; k, noncondensable component of the mixture; k0, noncondensable component in a flow; k0int, noncondensable component on the phase interface; liq, water; out, outside; ph, phase transition; s, saturation line in a flow; s.int, saturation line on the phase interface; vg, vapor-gas flow; w, tube wall; 10, water vapor in a flow; 10int, water vapor on the phase interface; Σ, total; *, new value of quantities; ′, condensate; ¯, average value.

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