

**MODELING OF HEAT AND MASS TRANSFER
IN ABSORPTION IN TWO-PHASE BINARY SYSTEMS
USED IN HEAT PUMPS**

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Results of modeling of heat- and mass-transfer processes proceeding simultaneously in vapor absorption on tube banks are described. Theoretical models of film absorption are presented. The calculation results are compared with experimental data on steam absorption by the lithium bromide solution on a vertical tube. In calculation of transfer processes in absorption on horizontal tubes, the possibility of using solutions for the initial thermal length and for the section with a linear temperature profile is substantiated. The calculations are illustrated by the example of a multipass absorber.

In recent years, absorption heat transformers (heat pumps and refrigerators) have gained wide utility in various fields of industry, especially in power engineering. This is related, on the one hand, with the high cost of power resources and, on the other hand, with the fact that such converters can utilize the waste heat of various technological processes, as well as the heat released by natural moderate-temperature sources (solar batteries and geothermal sources) for district heating or room cooling.

An absorber, in which vapor is absorbed by a liquid solution flowing down the surfaces to be cooled or by pre-cooled jets or drops of the solution, is the main component of each absorption heat transformer. Film absorbers with vertical stacks of horizontal tubes cooled by a liquid flowing inside them are the most widespread ones.

A distinctive feature of two-phase binary systems used in absorbers of heat pumps and refrigerating machines is that the absorption process is accompanied by release of a substantial amount of heat. The system in which the aqueous solution of lithium bromide is used as a liquid phase and steam as a gas phase is one of such systems. Obviously, isothermal-diffusion models and models of convective heat transfer without diffusion cannot adequately describe such systems.

Absorption in heat-releasing systems resembles ordinary condensation, and the constructions of film absorbers and condensers are nearly identical. Almost all results on film condensation are usually compared with the well-known Nusselt formulas that describe heat transfer in laminar film condensation of pure saturated vapor on a vertical surface. These simple formulas are used as reference ones in designing condensers, together with other empirical and semi-empirical relations, since up to now the Nusselt solution remains the only analytical solution in the region of heat transfer with condensation, whereas the numerical solutions available in the literature are rather inconvenient for analysis of the process of interest and for engineering calculations.

The problem of film absorption on a vertical surface under assumptions close to those used in the Nusselt problem but in which the heat- and mass-transfer processes are considered as interrelated was formulated and

analytically solved in [1, 2]. In these works, exact solutions in the form of series for a constant film thickness and for a velocity profile uniform across the film, as well as self-similar solutions for the initial length were obtained. Analogous exact solutions in the form of series were obtained in [3] for a parabolic velocity profile. For the initial length and for the section with a linear temperature profile, analytical solutions with a variable film thickness were obtained in [4, 5]. Strictly speaking, these solutions are valid only for low concentrations of the substance to be absorbed in the solution. Further attempts to improve the nonisothermal absorption model, including the case of comparable concentrations of the substance to be absorbed and the absorbent, failed to provide an analytical solution to the problem. Numerical results obtained using these models were reported in [6–8].

In the present work, only analytical solutions are used to analyze the heat- and mass-transfer processes in absorption on tube banks with horizontal tubes.

Film absorption differs considerably from ordinary film condensation, since the liquid phase is two-component, and the equilibrium temperature at the film surface (saturation temperature) depends not only on the pressure but also on the concentration of the dissolved substance. In this case, the heat- and mass-transfer processes are interrelated, and for vapor absorption by the solution film flowing downward a vertical surface, they can be described by the system of heat-conduction and diffusion equations with certain entrance and wall conditions

$$u \frac{\partial T}{\partial x} = a \frac{\partial^2 T}{\partial y^2}, \quad u \frac{\partial C}{\partial x} = D \frac{\partial^2 C}{\partial y^2},$$

$$T = T_0, \quad C = C_0, \quad \delta = \delta_0 \quad \text{for} \quad x = 0, \quad \frac{\partial C}{\partial y} = 0, \quad T(x, 0) = T_w \quad \text{for} \quad y = 0$$

and with conjugate conditions at the interface between the phases at $y = \delta(x)$:

$$C_i = k_1 - k_2 T_i; \tag{1}$$

$$-\lambda \frac{\partial T}{\partial y} = r_a \rho \left(u \frac{d\delta}{dx} - V \right) = 3r_a \rho \langle u \rangle \frac{d\delta}{dx}, \quad \langle u \rangle = \frac{g\delta^2}{3\nu}; \tag{2}$$

$$\rho \left(u \frac{d\delta}{dx} - V \right) = -\frac{\rho D}{1 - C} \frac{\partial C}{\partial y}. \tag{3}$$

Here T is the temperature, C is the mass concentration of water in the solution, x is the coordinate along the film, y is the transverse coordinate, δ is the film thickness, a is the thermal diffusivity, D is the diffusivity, λ is the thermal conductivity, ρ is the solution density, ν is the kinematic viscosity of the solution, u is the longitudinal velocity, V is the transverse velocity, and r_a is the specific absorption heat; the subscripts 0, w , and i refer to the conditions at the entrance, at the wall, and at the interface, respectively.

Condition (1) is the simplest condition for an equilibrium state of the “solution–vapor” system. It is valid, in particular, for the “lithium bromide–steam” system within the ranges of temperatures and concentrations typical of heat pumps and refrigerating machines. The coefficients k_1 and k_2 depend on the steam pressure. Equalities (2) and (3) represent the balance of energy and the equality of the mass-flux densities at the film surface provided that all the heat of absorption is released at the interface between the phases and is spent on heating the solution. Expression (3) for the mass flux follows from the Fick law in the case one of the components of the solution is not spent and does not enter the solution (see [9]). In the case considered, lithium bromide is such a component. Since a solution with comparable concentrations of lithium bromide and water is used in absorbers of heat pumps and refrigerating machines, and the water concentration varies only within a narrow range (less than 10%), the expression for the mass-flux density can be simplified by replacing it with the following approximate expression:

$$m = \rho \left(u \frac{d\delta}{dx} - V \right) = -\frac{\rho D}{1 - C_0} \frac{\partial C}{\partial y}.$$

The film thickness that enters the equations and boundary conditions is unknown.

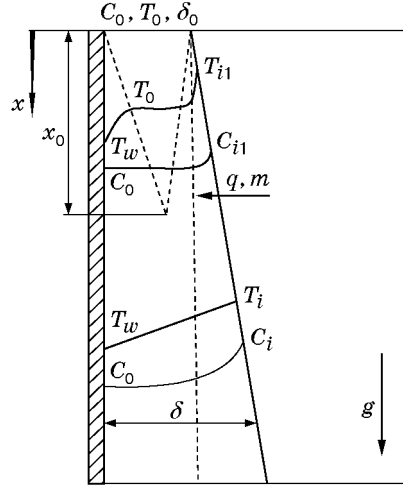


Fig. 1. Schematic representation of the problem.

Exact solutions of the above-formulated problem were obtained in [1–3] under the assumption that the film thickness is uniform. These results have clarified some special features of the absorption process under operating conditions of apparatus where lithium bromide was used. Two characteristic sections were shown to exist: an initial length of heat-flux development near the wall and near the film surface and a section with a linear temperature profile across the film. Throughout the initial length, the equilibrium temperature and concentration on the film surface are spatially uniform, and outside it they vary along the film, closely following the equilibrium curve. Moreover, over both sections the concentration varies predominantly inside a thin layer near the film surface (since the Lewis number is $Le = D/a \ll 1$). It was also shown in [1–3] that the choice of a particular shape of the velocity profile (either parabolic or stepwise) only weakly affects the solution.

The assumption that the film thickness is uniform never holds perfectly. This circumstance leads to some contradictions in the model and restricts the area of application of the solutions obtained. The problem of film absorption with allowance for the change in the volume of the liquid phase, i.e., for a nonuniform film thickness, was solved in [4, 5] for the initial length and for the section with a linear temperature profile. For the sake of simplicity, the problem was solved for a uniform velocity distribution ($u = 1.5\langle u \rangle$) inside the thermal layer that develops near the film surface, over the initial length, and inside the diffusional layer throughout the entire film length. Inside the thermal layer which develops near the wall, a linear velocity profile is used within the initial length. Thus, a semi-parabolic velocity profile is substituted by a piecewise linear one. A schematic representation of the problem is shown in Fig. 1.

In the variables ξ_1 and η , the following self-similar solutions with uniform equilibrium distributions of temperature and concentration at the interface between the phases are valid inside the thermal and diffusional layers near the film surface over the initial length $x \leq x_0$ (Fig. 1):

$$\theta = \theta_{i1} \left[1 - \operatorname{erf} \left(\frac{\sqrt{3}\eta}{2\sqrt{2}\xi_1} \right) \right], \quad \gamma = \gamma_{i1} \left[1 - \operatorname{erf} \left(\frac{\sqrt{3}\eta}{2\sqrt{2}\operatorname{Le}\xi_1} \right) \right].$$

Here $\theta_{i1} = (T_{i1} - T_0)/(T_e - T_0) = K_a\sqrt{\operatorname{Le}}/(1 + K_a\sqrt{\operatorname{Le}})$, $\gamma_{i1} = (C_{i1} - C_0)/(C_e - C_0) = 1/(1 + K_a\sqrt{\operatorname{Le}})$, $\eta = 1 - y/\delta$, $\Delta = \delta/\delta_0$, $\xi = x/(\operatorname{Pr}\operatorname{Re}\delta_0)$, $\xi_1 = \int_0^\xi \frac{dt}{\Delta^4}$, $\theta = (T - T_0)/(T_e - T_0)$, $\gamma = (C - C_0)/(C_e - C_0)$, $C_e = k_1 - k_2T_0$, $C_0 = k_1 - k_2T_e$, $Le = D/a$; $\operatorname{Pr} = \nu/a$, $\operatorname{Re} = \langle u \rangle_0\delta_0/\nu$, $\langle u \rangle_0 = g\delta_0^2/(3\nu)$, $K_a = r_a(C_e - C_0)/(c_p(T_e - T_0)(1 - C_0))$, Le , Pr , and Re are the Lewis, Prandtl, and Reynolds criteria, respectively, K_a is the absorption analog of the phase-transition criterion, and c_p is the specific heat capacity; the subscript e marks equilibrium values. Condition (2) with allowance for these solutions determines the film thickness. Passing to the ordinary variable ξ yields the following expression for the inverse function:

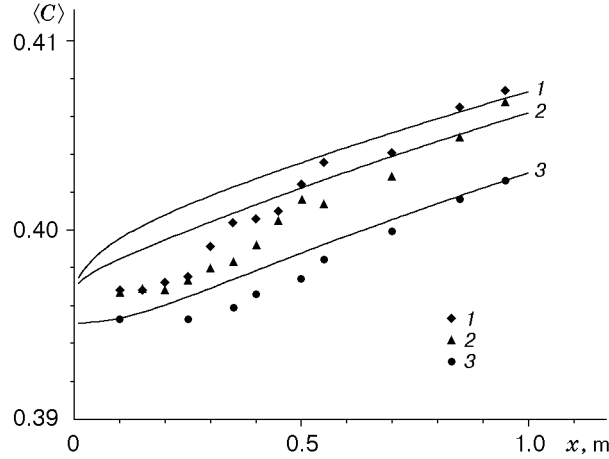


Fig. 2. Distribution of the mean concentration of water in the lithium bromide solution along the vertical tube ($Re = 69$, $P = 0.96$ kPa, and $T_w = 24^\circ\text{C}$): the points are the experimental data and the curves are the calculated data for $C_0 = 0.397$ and $T_0 = 24^\circ\text{C}$ (1), $C_0 = 0.397$ and $T_0 = 33^\circ\text{C}$ (2), and $C_0 = 0.395$ and $T_0 = 44.6^\circ\text{C}$ (3).

$$\xi(\Delta) = \frac{\Delta^4(4 \ln \Delta - 1) + 1}{32A^2}, \quad A = \frac{\sqrt{Le}(C_e - C_0)}{2\sqrt{\pi}(1 + K_a\sqrt{Le})}.$$

In contrast to the initial length, the equilibrium temperature and concentration vary along the film surface over the section with a linear temperature profile as follows:

$$\theta_{i2} = \frac{b\sqrt{\pi}}{\sqrt{p}} (1 - \operatorname{erf} \sqrt{p(\xi_1 - \xi_0)}) \exp(p(\xi_1 - \xi_0)) - \theta_0, \quad \gamma_{i2} = 1 - \theta_{i2}, \quad \theta_0 = \frac{T_0 - T_w}{T_e - T_0},$$

$$\xi_0 = \frac{1}{RePr} \frac{x_0}{\delta_0}, \quad p = \frac{1}{K_a^2 Le}, \quad b = \frac{K_a\sqrt{Le} + \theta_0(1 + K_a\sqrt{Le})}{\sqrt{\pi} Le (1 + K_a\sqrt{Le}) K_a}.$$

At the boundary between the initial length and the section with a linear temperature profile ($\xi = \xi_0$) the temperatures and concentrations at the film surface are identical. The complete equation permitting estimation of the initial length ξ_0 can be found in [4]; it can be replaced with the approximate equality $\xi_0 = 0.3$.

The thickness of the film in the section with a linear temperature profile is given by the equation

$$\Delta = \exp \left\{ \frac{C_e - C_0}{2K_a} \left[\frac{b\sqrt{\pi}}{p\sqrt{p}} (\exp[p(\xi_1 - \xi_0)] - 1) - \frac{b\sqrt{\pi}}{p} \left(\frac{1}{\sqrt{p}} \operatorname{erf} \sqrt{p(\xi_1 - \xi_0)} \exp[p(\xi_1 - \xi_0)] - \frac{2(\xi_1 - \xi_0)}{\sqrt{\pi}} \right) \right] \right\}.$$

The solution of the problem with a uniform film thickness is a particular case for $\Delta = 1$ ($\xi_1 = \xi$) and, simultaneously, it can be used as the first approximation for solving the equation for the film thickness.

The calculations show that the film thickness changes little over distances comparable with the half-perimeter of the tubes for the “lithium bromide aqueous solution–steam” system during operation of heat pumps and refrigerating machines. Therefore, in engineering calculations, one can use simple formulas for a uniform film thickness ($\delta = \delta_0$, $\Delta = 1$, and $\xi_1 = \xi$).

Figure 2 shows the predicted distributions along the vertical surface of the concentration of water averaged across the film in comparison with the experimental data obtained in [10]. The average concentrations were found by numerical integration of the concentration profiles over the film thickness at the initial length and at the section with a linear temperature profile.

Inside the thermal boundary layer near the wall, one can use the well-known solution of the heat-conduction equation with a linear velocity profile across the film [9]

$$T = T_w + \frac{T_0 - T_w}{\Gamma(4/3)} \int_0^{\eta_1} \exp(-t^3) dt,$$

where $\Gamma(4/3) \simeq 0.893$ is the gamma-function and $\eta_1 = y(g\delta/(9\nu ax))^{1/3}$. This solution yields the following formulas for the local heat-flux density and the wall heat-flux density averaged over the film length:

$$q_{w1} = -\lambda \frac{\partial T}{\partial y} = \lambda \frac{T_w - T_0}{\Gamma(4/3)} \left(\frac{g\delta}{9\nu ax} \right)^{1/3},$$

$$\langle q \rangle_{w1} = \frac{1}{x_0} \int_0^{x_0} q_{w1} dx = 1.165 \frac{\lambda(T_w - T_0)}{\delta} \left(\text{Re Pr} \frac{\delta}{x_0} \right)^{1/3} = 1.165 \frac{\lambda(T_w - T_0)}{\delta} \xi_0^{-1/3}.$$

In the section with a linear temperature profile $\theta = \eta(\theta_i + \theta_0) - \theta_0$, the heat-flux density is determined by the equilibrium temperature at the interface between the phases:

$$q_2 = -\lambda \frac{\partial T}{\partial y} = \frac{\lambda(T_i - T_w)}{\delta} = \frac{\lambda(\theta_i + \theta_0)}{\delta} (T_e - T_0).$$

The calculation of the heat-flux density averaged over the film length in the section with a linear temperature profile

$$\begin{aligned} \langle q \rangle_2 &= \frac{1}{L - x_0} \int_{x_0}^L q_2 dx = \frac{\lambda(T_e - T_0)}{(L - x_0)\delta} \int_{x_0}^L \theta_i dx + \frac{\theta_0(T_e - T_0)\lambda}{\delta} \\ &= \frac{\lambda(T_e - T_0) \text{Re Pr}}{L - x_0} \int_{\xi_0}^{\xi_L} \theta_i d\xi + \frac{\lambda(T_0 - T_w)}{\delta} \end{aligned}$$

reduces to integration, which can be performed analytically:

$$\begin{aligned} I &= \int_{\xi_0}^{\xi_L} \theta_{i2} d\xi = \frac{b\sqrt{\pi}}{p\sqrt{p}} \{ \exp[p(\xi_L - \xi_0)] - 1 \} \\ &\quad - \frac{b\sqrt{\pi}}{p} \left\{ \frac{1}{\sqrt{p}} \exp[p(\xi_L - \xi_0)] \text{erf} \sqrt{p(\xi_L - \xi_0)} - \frac{2\sqrt{\xi_L - \xi_0}}{\sqrt{\pi}} \right\} - \theta_0(\xi_L - \xi_0). \end{aligned}$$

Using this integral, one can calculate the temperature averaged over the film length in the section with a linear temperature profile:

$$\langle T \rangle_{i2} = \frac{1}{L - x_0} \int_{x_0}^L T_{i2} dx = \frac{1}{L - x_0} \int_{x_0}^L [T_0 + \theta_{i2}(T_e - T_0)] dx = T_0 + \frac{(T_e - T_0) \text{Re Pr} \delta}{L - x_0} \int_{\xi_0}^{\xi_L} \theta_{i2} d\xi.$$

The density of the mass flux through the film surface can be also calculated for the initial length and for the section with a linear temperature profile. For the initial length, we have

$$m_{i1} = -\frac{\rho D}{1 - C_0} \frac{\partial C}{\partial y} = \frac{\rho D(C_e - C_0)}{(1 - C_0)\delta} \frac{\partial \gamma}{\partial \eta} = \frac{\sqrt{3}}{\sqrt{2\pi}} \frac{\rho D(C_e - C_0)}{(1 - C_0)\delta} \gamma_i (\text{Le} \xi)^{-1/2},$$

$$\langle m \rangle_{i1} = \frac{1}{x_0} \int_0^{x_0} m_1 dx = \frac{\sqrt{6}}{\sqrt{\pi} \text{Le}} \frac{\rho D(C_e - C_0)}{(1 - C_0)\delta} \gamma_i \xi_0^{-1/2}.$$

Since the mass and heat fluxes at the film surface are related by formula (2), the equalities $q_{i1} = r_a m_{i1}$ and $\langle q \rangle_{i1} = r_a \langle m \rangle_{i1}$ hold at the initial length, and the equalities $m_{i2} = q_2/r_a$ and $\langle m \rangle_{i2} = \langle q \rangle_2/r_a$ are valid at the section with a linear temperature profile.

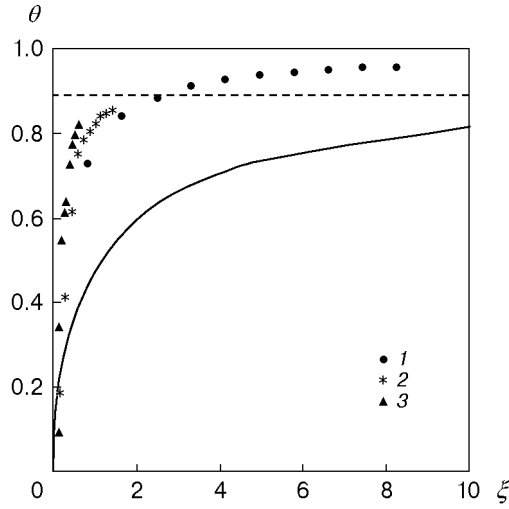


Fig. 3. Distribution of the temperature averaged across the film along the vertical surface ($P = 0.96$ kPa, $Le = 0.014$, and $K_a = 7.6$): the solid and dashed curves show the calculated (see [1]) and asymptotic values of θ , respectively; points 1, 2, and 3 refer to $Re = 12$, 44, and 94, respectively.

The above solutions may be used in calculating heat- and mass-transfer characteristics for absorption on a single tube and also on a tube bank provided that each tube has a distinct initial length and a well-defined section with a linear temperature profile. Our estimates show that the initial length can amount to $(1/3-1/2)\pi R$, where R is the tube radius. For a tube bank, the calculation procedure should include successive calculations to be performed beginning from the upper tube and then passing to that located below. The initial parameters T_0 and C_0 of the lower tube are calculated as values at the exit of the upper tube averaged across the film.

In the present work, we propose two schemes to calculate heat- and mass-transfer in absorption on a tube bank. In the first scheme, it is assumed that there is no absorption in the space between the tubes, and the rates of heat and mass transfer at each tube can be calculated using the formulas for the initial length and for the section with a linear temperature profile, with subsequent averaging of the heat- and mass-transfer characteristics over the tube half-perimeter:

$$\langle q \rangle_w = \frac{\langle q \rangle_{w1}\xi_0 + \langle q \rangle_{i2}(\xi_L - \xi_0)}{\xi_L}, \quad \xi_L = \frac{1}{Re Pr} \frac{\pi R}{\delta},$$

$$\langle m \rangle_i = \frac{\langle m \rangle_{i1}\xi_0 + \langle m \rangle_{i2}(\xi_L - \xi_0)}{\xi_L}, \quad \langle T \rangle_i = \frac{\langle T \rangle_{i1}\xi_0 + \langle T \rangle_{i2}(\xi_L - \xi_0)}{\xi_L}.$$

The second scheme rests on the assumption that intense absorption on jets and drops occurs between the tubes under adiabatic conditions with attaining an equilibrium state in each local region. In this case, a saturated solution comes to the tube located below. For this reason, on each tube located below, the temperature of the solution decreases first, and then absorption starts. This assumption is supported by the analogy between the condensation and absorption processes and by the results of temperature measurement in the tube space in condensation on tube banks [11]. However, unlike the case of condensation, for which the saturation temperature attained between the tubes depends only on pressure, the equilibrium temperature and concentration for a given total pressure are not known beforehand in the case of absorption. The “solution–steam phase” diagram determines only the relationship between these quantities. To find the equilibrium values of temperature and concentration, we propose to use asymptotic formulas obtained on the basis of exact solutions of heat- and mass-transfer problems for film absorption with an adiabatic condition at the wall and adiabatic jets [12]: $\theta = K_a/(1 + K_a)$ and $\gamma = 1/(1 + K_a)$.

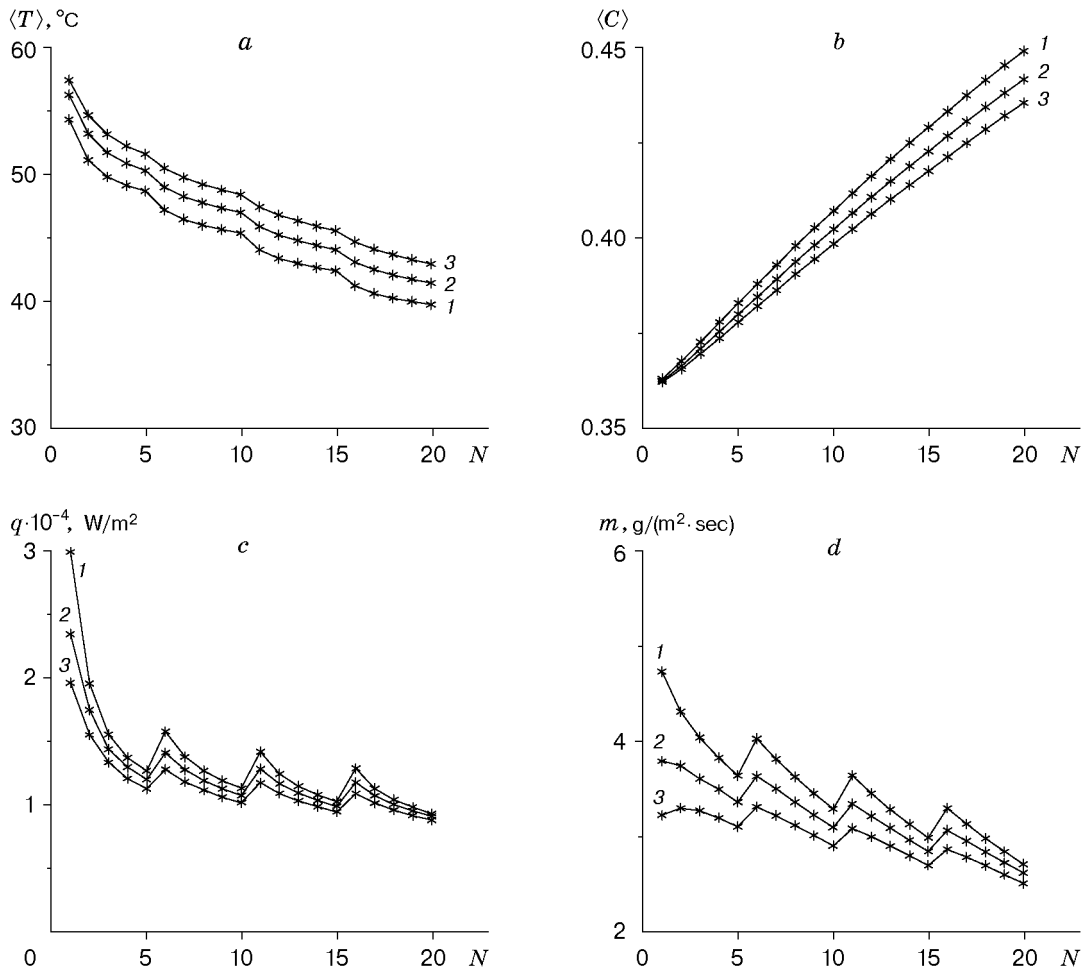


Fig. 4. Heat- and mass-transfer characteristics for steam absorption on a tube bank (N is the tube number): (a) mean temperature of the solution; (b) mean concentration of the solution; (c) heat-flux density; (d) mass-flux density; $R_s = 0$ (curve 1) and $1/R_s = 5000$ (curve 2) and $2500 \text{ W}/(\text{m}^2 \cdot \text{K})$ (curve 3).

It follows from a comparison of the exact solutions obtained in [1] with the experimental data on steam absorption by the aqueous solution of lithium bromide on a vertical tube without cooling (adiabatic condition at the wall) reported in [13] (Fig. 3) that the limiting (asymptotic) value of the equilibrium temperature of the solution sufficiently far from the entrance is predicted rather accurately by the above simple formula. However, in experiments the limiting value of θ is attained at smaller distances. The latter indirectly confirms the hypothesis about rapid saturation of the solution in the tube space.

It should be noted that real absorbers, as a rule, are multipass ones, i.e., cooling water passes in succession several tubes arranged in a vertically oriented tube bank. The system in which cooling water is supplied from tubes located below to those located above is believed to be the most effective one. In this situation, the temperature of cooling water in the tubes and, hence, the wall temperature of each tube is determined by heat-transfer processes both on tubes located below and those located higher. Therefore, in designing apparatus in which the supply of cooling water is organized in this manner, two iteration procedures that involve the heat-transfer coefficient K are obviously required:

$$K = \left(\frac{1}{\alpha_1} + \frac{1}{\alpha_2} \frac{d_1}{d_2} + \frac{d_1}{2\lambda_w} \ln \frac{d_1}{d_2} + R_s \right)^{-1}, \quad \alpha_1 = \frac{\langle q \rangle_w}{\langle T \rangle - T_w}.$$

Here α_1 is the heat-transfer factor of the film, d_1 and d_2 are the outer and inner tube diameters, respectively, and R_s is the thermal resistance caused by sedimentation in the tubes. In calculating heat-transfer coefficients inside the tubes (α_2), empirical formulas for various flow modes may be used [14].

As an example, Fig. 4 shows the calculated heat- and mass-transfer characteristics for steam absorption by the lithium bromide solution for one stack of twenty horizontal tubes divided into sections with five tubes in each. Cooling water is supplied into the lower section with a temperature and velocity identical for each of the five tubes, and then, passing through this section, the water gets mixed and enters the section located above the first one.

The procedure proposed can be applied to predict the heat- and mass-transfer characteristics at each tube of a tube bank (temperature and concentration of the solution, heat-flux densities at the wall and at the film surface, mass flux at the interface between the phases, etc.) and find the optimum parameters in designing absorbers of heat pumps and refrigerating machines.

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