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CONTACT HEAT AND MASS TRANSFER BETWEEN A HOT GAS AND A FLOWING LIQUID FILM

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A mathematical model of contact heat and mass transfer is proposed and the calculations are compared with the experimental data.

One method of utilizing the heat of the flue gases escaping from thermal power plants is to employ them to heat water in film contact apparatus; this has less resistance than a packed column, so that the apparatus can be mounted in the exhaust system of the power plant without it being necessary to install an exhaust fan.

In order to investigate the process of heat and mass transfer in heaters of this kind we designed an experimental set-up [1] in which the film contact apparatus consisted of three modules comprising box-type structures of rectangular cross section measuring 0.5 × 0.3 × 1.0 m with three working plates 0.5 m wide and 0.94 m high, the distance between plates being 0.05 m. In order to eliminate heat losses to the ambient medium, all the modules were insulated on the outside with asbestos lagging.

The liquid film, created by special sprinklers [2], flowed over both sides of the plates and the inner surfaces of the module housing, so that the gas was able to pass through four channels with wetted walls.

The mean temperature of the liquid was measured with thermocouples mounted in the water headers, at the inlet to the equipment, between the modules and at the outlet. The gas temperature was measured at the inlet to the equipment and at the outlet. The moisture content of the gas at the inlet was calculated with allowance for the humidity of the starting air and the moisture formed as a result of combustion of the fuel. The flow rates of fuel and water were measured volumetrically, that of the starting air from the pressure drop in a convergent channel. The final moisture content of the gas was determined by the balance method.

Direct contact between the hot gas and the liquid film is accompanied by simultaneous processes of convective heat transfer in the film and the gas phase and diffusion of vapor in the gas. These processes are interrelated and rather difficult to calculate.

We solved this problem by means of a contact heat and mass transfer model based on averaged differential heat and mass transfer equations, subject to the following assumptions: by virtue of symmetry the heat fluxes at a wetted plate of infinitely small thickness and at the channel axis are equal to zero, the transfer processes are steady-state, the motion of the gas and the liquid is directed along the x axis, there are no heat losses to the ambient medium, changes of flow rates due to phase transitions can be neglected.

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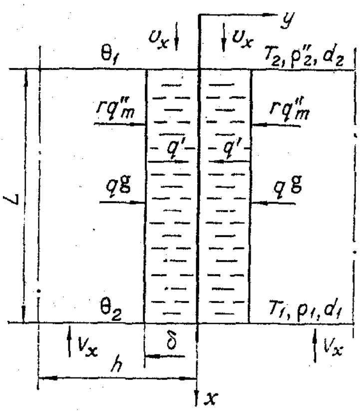

Fig. 1. Calculation scheme for the contact heater.

The calculation scheme for the contact heat and mass transfer problem is set out in Fig. 1. The heat transfer in the liquid film is described by the equation

$$U\frac{\partial\Theta}{\partial x}=\frac{\partial}{\partial y}\left[(a'+a'_t)\frac{\partial\Theta}{\partial y}\right]. \tag{1}$$

For the diffusion of heat and vapor in the gas we have

$$V\frac{\partial T}{\partial x}=\frac{\partial}{\partial y}\left[(a^g+a^g_t)\frac{\partial T}{\partial y}\right],\quad V\frac{\partial\rho''}{\partial x}=\frac{\partial}{\partial y}\left[(D+D_t)\frac{\partial\rho''}{\partial y}\right]. \tag{2}$$

The subscript "t" denotes the turbulent transfer coefficient.

Equations (1) and (2) are solved for the following boundary conditions:

$$\Theta=\Theta_1 \quad \text{at} \quad x=0,\quad T=T_1,\quad \rho''=\rho_1 \quad \text{at} \quad x=L,$$

$$\frac{\partial T}{\partial y}=0,\quad \frac{\partial\rho''}{\partial y}=0 \quad \text{at} \quad y=h,\quad \frac{\partial\Theta}{\partial y}=0 \quad \text{at} \quad y=0, \tag{3}$$

$$T=\Theta=\Theta_\delta \quad \text{at} \quad y=\delta,\quad q'=q_g+rq''_m \quad \text{at} \quad y=\delta.$$

The last of relations (3) takes into account the fact that the heat flow from the phase interface into the liquid film q' is composed of the heat flux from the gas q^g and the heat of phase transition rq''_m.

The relation between the vapor density at the interface and temperature is given by the expression [3]

$$\log\rho''_\delta=6.9(\Theta_\delta-273.16)/(\Theta_\delta-43)-2.314,\quad y=\delta. \tag{4}$$

Integrating (1), (2) with respect to the y coordinate, we obtain

$$\langle U\rangle\delta\frac{d\langle\Theta\rangle}{dx}=(a'+a'_t(\delta))\frac{\partial\Theta}{\partial y}\bigg|_\delta,$$

$$\langle V\rangle(h-\delta)\frac{d\langle T\rangle}{dx}=-(a^g+a^g_t(\delta))\frac{\partial T}{\partial y}\bigg|_\delta,$$

$$\langle V\rangle(h-\delta)\frac{d\langle\rho''\rangle}{dx}=-(D+D_t(\delta))\frac{\partial\rho''}{\partial y}\bigg|_\delta, \tag{5}$$

where $\langle\varphi\rangle=\int v\varphi dy/\int vdy$ is the mean-flow value of the quantity φ; $\langle v\rangle$ is the corresponding mean-flow velocity.

The turbulent transfer coefficients at the interface entering into (5) are not known. We express the values of the heat and mass fluxes (right sides in system (5)) in terms of the heat and mass transfer coefficients:

$$q'=\alpha'(\langle\Theta\rangle-\Theta_\delta),\quad q^g=\alpha^g(\Theta_\delta-\langle T\rangle),\quad q''_m=\beta(\rho''_\delta-\langle\rho''\rangle). \tag{6}$$

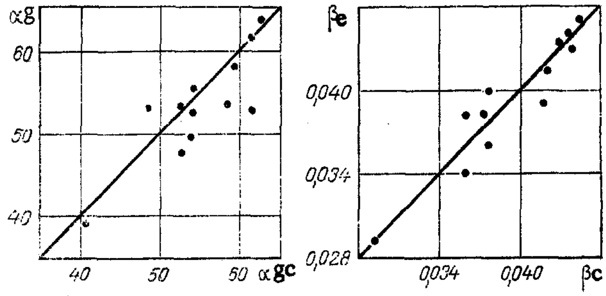

Fig. 2. Comparison of the calculated and experimental values of the heat and mass transfer coefficients α^g and β. α^g, α^{gc}, $W/m^2 \cdot K$; β_e, β_c, m/sec.

Substituting (6) in (5) and taking into account the fact that $\langle V\rangle < 0$ (the gas moves in a direction opposite to that of the x axis), we arrive at the following problem:

$$\frac{d\langle\Theta\rangle}{dx} = -\frac{\alpha'\Omega}{G'c_p'}(\langle\Theta\rangle - \Theta_\delta), \tag{7}$$

$$\frac{d\langle T\rangle}{dx} = -\frac{\alpha^g\Omega}{G^g c_p'}(\Theta_\delta - \langle T\rangle), \tag{8}$$

$$\frac{d\langle\rho''\rangle}{dx} = -\frac{\beta\rho^g\Omega}{G^g}(\rho_\delta'' - \langle\rho''\rangle). \tag{9}$$

The local heat balance condition (the last of relations (3)) is rewritten in the form

$$\alpha'(\langle\Theta\rangle - \Theta_\delta) = \alpha^g(\Theta_\delta - \langle T\rangle) + r\beta(\langle\rho''\rangle - \rho_\delta''). \tag{10}$$

Thus, the problem reduces to the solution of system of equations (4), (7)-(10) with the following initial conditions:

$$\langle\Theta\rangle = \Theta_1, \quad x = 0, \quad \langle T\rangle = T_1, \quad \langle\rho''\rangle = \rho_1, \quad x = L.$$

Since the initial conditions for the liquid and gas phases are given at different ends, this system was solved by iteration. Knowing the values of $\langle T\rangle$ and $\langle\rho''\rangle$ in the n-th iteration step, by integration from 0 to L from (7) and (10), using (4), we find Θ_δ and $\langle\Theta\rangle$. Substituting Θ_δ in (8), (9) and integrating from L to 0, we find the new values of $\langle T\rangle$ and $\langle\rho''\rangle$ in the (n + 1)-th step. This iteration procedure converges. The calculations were terminated upon satisfaction of the condition

$$|\Theta_{\delta_n} - \Theta_{\delta_{n+1}}| < \varepsilon, \quad x = 0. \tag{11}$$

The calculations show that if ε is varied from 0.1 to 0.01, the temperature values change by less than 0.1°. For such ε condition (11) is satisfied after 3-5 iterations. As a result of solving this system of equations we obtain the parameters of the gas and the liquid film along the length of the contact zone.

The coefficient of heat transfer from the surface of the film into the liquid is calculated from the expression [4]

$$\alpha' = 0.71\,\lambda'(\nu^2/g)^{-\frac{1}{3}}\mathrm{Re}'^{-0.282}.$$

In Fig. 2 the coefficients of heat transfer from the gas to the surface of the liquid film α_c^g, calculated in accordance with [5], and the mass transfer coefficients β_c, calculated from the Chilton–Colbourn equation [6], are compared with the experimental values α_e^g and β_e:

$$\alpha_c^g = 0.0018\frac{\lambda^g}{d_e}(\mathrm{Re}')^{0.33}\mathrm{Re}^g\left(\frac{V}{V_i}\right)^{0.3}\gamma^{0.5},$$

$$\gamma = \frac{\delta}{\delta_i} = 1 + 1.5\cdot 10^{-2}(V - V_i)/\sqrt{gd_e}, \tag{12}$$

$$\alpha_e = \frac{G^g(c_p^g + c_p''d_1)(T_2 - T_1)}{\Omega L\Delta\overline{T}_l},$$

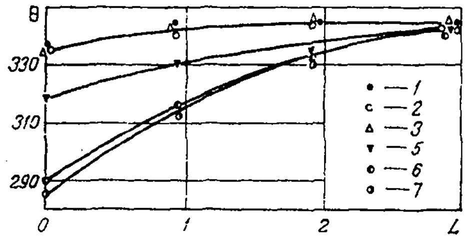

Fig. 3. Comparison of the calculated and experimental values of the liquid temperatures in the contact zone. The figures correspond to the numbers of the experiments in Table 1. θ, K, L: m.

TABLE 1. Comparison of the Calculated and Experimental Values of the Gas Outlet Temperature and Moisture Content

Parameter	Number of expt.						
	1	2	3	4	5	6	7
Gas-liquid ratio, kg/kg	1,64	1,36	1,33	1,77	1,1	2,03	2,2
Inlet gas temp., °K	668	638	668	673	631	663	633
Inlet gas moisture content, kg/kg	0,108	0,088	0,087	0,078	0,070	0,071	0,097
Outlet gas temp., °K:							
exptl.	415	398	405	401	385	400	393
calc.	396	385,9	389,9	397	375,4	392,6	393,4
Outlet gas moisture content, kg/kg:							
exptl.	0,239	0,208	0,216	0,209	0,168	0,163	0,155
calc.	0,285	0,214	0,213	0,181	0,133	0,142	0,139
Error, %	−19,2	−2,8	+1,4	+13,4	+20,1	+12,9	+10,3

$$\beta_c = \frac{M^g}{M} \alpha^g \frac{1}{\rho^g c^g_c} \left(\frac{\Pr_c}{\Pr_t}\right)^{-\frac{2}{3}}, \tag{13}$$

$$\beta_e = \frac{G^g(d_2 - d_1)}{\Omega L \Delta\overline{\rho_t}}.$$

An analysis of the curves presented in Fig. 2 shows good agreement between the calculated and experimental values of α^g and β, which makes it possible to use expressions (12) and (13) for solving the system of equations (4), (7)-(10).

In Fig. 3 the experimental values of the liquid temperature are compared with the results of computer calculations based on Eqs. (4), (7)-(10). Table 1 gives the calculated and experimental values of the outlet temperature and moisture content of the gas. An analysis of the tabulated data and the graphs shows that the experimental data are in good agreement with the results of the calculations. Thus, the deviations of the calculated values of the temperatures of the liquid and the gas and the moisture content of the gas do not exceed 5, 20, and 20% respectively.

Thus the proposed mathematical model makes it possible to calculate, with sufficient accuracy, the parameters of the gas and the liquid film under conditions of direct contact. Accordingly it can be recommended for designing film contact heaters.

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VAPOR CONDENSATION ON HORIZONTAL WIRE-PROFILED TUBES

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The effect of the profiling parameters and the nonisothermality of the walls on the vapor condensation rate is analyzed for the case of a horizontal tube profiled by means of a wire spiral.

The degree of intensification of the process of vapor condensation on profiled as compared with smooth horizontal tubes depends to a large extent on the flooding by the condensate of the lower surface of the tube [1] and, moreover, on the thermal conductivity and thickness of the wall [2]. Our investigations [1] of the bottom flooding of horizontal tubes showed that tubes with spiral wire fins have a much smaller bottom flooding angle φ_f than do tubes profiled by deformation of the wall, for similar values of the We number which in this case characterizes the ratio of the forces of surface tension drawing the condensate to the base of the fins to the force of gravity.

In [3] an approximate theoretical solution of the problem was given for the case of vapor condensation on a horizontal wire-profiled tube. In the present article certain assumptions of [3] are refined and the effect of the profiling parameters and the physical properties of the condensate on the vapor condensation enhancement is analyzed in detail.

The assumptions made in [3] for the purposes of a theoretical solution of the problem of vapor condensation on a horizontal tube with spiral wire finning are analogous to the Nusselt theory for the condensation of stationary vapor on a vertical wall. Here we will consider the case where We > 5 and over the entire perimeter of the tube, except for the zone with the bottom layer, the condensate flows from the center between the wires under the action of surface tension predominantly along the normal to the wires and, under the force of gravity, drains along the wires to the bottom of the tube (Fig. 1).

In order to refine the inequality We > 5 we will estimate the radius of curvature of the film R, which enters into We, from the expression obtained in [4] by studying the geometry of the liquid film forming the meniscus (near the wire):

$$R = l_2^2/2d. \tag{1}$$

From the solution of the differential equation for the growth of the stream width (see below), over a broad interval of variation of the heat flux q and the tube diameter and for various liquids $l_2 \leqslant d/2$. Taking $l_2 = d/2$, we determine the pitch of the spiral S ensuring $We \geqslant 5$:

$$S \leqslant 3.2\sigma/\rho g d. \tag{2}$$

At d = 1.5 mm for water vapor and ammonia $S \leqslant 13$ mm, for freons $S \leqslant 2.5$-4 mm. For this pitch and the tube diameters D > 16 mm encountered in practice the maximum inclination of the spiral relative to the vertical for a uniform single-thread winding is ≈16°. Accordingly, in the solution of the problem we can neglect the inclination of the spiral and treat the finning as annular (transverse). In [3] the following differential equation was obtained for the flooding by the condensate of the interwire space in terms of the angular coordinate φ for a tube with spiral wire fins at a wall temperature constant with respect to φ and x:

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