DISTRIBUTION OF MASS VELOCITIES OF WATER-DROP AND STEAM FLOWS IN A HEATED TUBE WITH A TURBULENT AIR FLOW AND INJECTION OF WATER ALONG THE WALL

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The results of measurements of the mass velocities of a flow of water drops and steam in air in the case of a turbulent flow in a cold and electrically heated tube of diameter 20 mm and length 900 mm are presented.

There are numerous sources [1] describing the distribution and evaporation of a liquid in tubes with heated walls and with a simultaneous flow of air. Regimes in which the mass flow of air is a small fraction of the liquid flow (the air is admitted to the channel only to increase the liquid velocity) have been investigated. The experiments have shown that the heat fluxes and pressure differences can be calculated from the parameters of the liquid phase with appropriate allowance for the cross section of the channel actually occupied by it. In the present work we investigated experimentally the distribution of drops of water and steam in a tube in which water was injected in the form of an annular film along the wall and the mass flow of water was a small fraction of the total flow of the two-phase mixture. Injection of water in this way is an advantage in cases where the cold flow of air in the tube does not cool its externally heated walls sufficiently.

The experiments were conducted with 1Kh18N9T steel tubes of diameter d = 20 mm, wall thickness 1.1 mm, and different lengths (l/d = 13, 25, and 45). The investigated section (Fig. 1) was vertical and the air for producing the flow within the tube was admitted at the top. At the initial cross section of the tube the water entered round its internal perimeter through small tubes of internal diameter 0.8 mm (31 in all) distributed evenly round the circumference. The air and water entered the tube at an ambient temperature of 290-295°K. The nonslip condition for water on the tube wall after its perpendicular outflow was satisfied [2], and, hence, after the water jets spread over the perimeter of the tube, an annular water film would be formed on its internal surface. In addition, to monitor the process we carried out special experiments involving visualization of the flow of water in the region of its outflow onto a flat plate over which an open stream of air flowed, with the same system and velocity of injection of the water as in the investigated tube (attempts to visualize the flow near the outflow of the water jets in a transparent tube were unsuccessful owing to sweating of the walls). On the basis of the results of [2] and these experiments we can infer that in the investigated range of parameters the water film adhered to the walls.

The tube walls were heated by the passage of a direct electric current through them. The heated portions of the tubes were thermally insulated on the outside. The heat flux from the tube wall into the two-phase flow was determined from the electrical power released in the tube minus the heat lost by leakage through the insulation.

During the experiments wemeasured the air flow rate by means of specially calibrated orifice plates (24 to 52 g/sec), water flow rate by a sampling gauge with a calibrated capacity of 146 cm^3 (2 to 15 g/sec), current from 0 to 1500 A, and voltage on the ends of the heated section, and wall temperature by means of Chromel–Alumel thermocouples welded to the outside of the tube.

To measure the mass velocities of the flows of water drops and stream, and also the temperature of the two-phase flow, we used a specially constructed annular cap which fitted onto the end of the tube and carried four-point measuring combs: a comb of tubes of external diameter 2.2 mm and wall thickness 0.2 mm for taking samples of the two-phase mixture and a comb of Chromel–Copel thermocouples held in similar tubes with BF-4 cement. The centers of the sampling tubes and thermocouple junctions were located at the following distances from the tube center: 1.3, 3.6, 6.2, and 8.7 mm, to an accuracy of ±0.1 mm. The "mouths" of the

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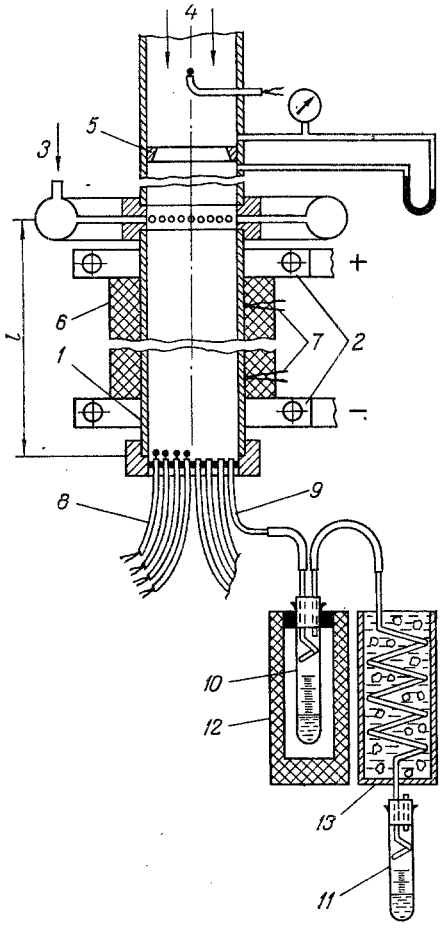

Fig. 1. Diagram of experimental section: 1) tube; 2) busbars; 3) injection of water through annular collector and small tubes; 4) admission of air; 5) measuring section for measuring air flow rate; 6) heat insulation; 7) thermocouples on wall; 8) comb of thermocouples in flow; 9) comb for sampling two-phase mixture; 10) test tube for sampling liquid phase; 11) test tube for sampling condensed steam; 12) heat-insulating block; 13) tank with ice.

sampling tubes were sharpened by beveling round the perimeter of the outside. Thus, the samples were taken by tubes of diameter 1.8 mm and practically zero wall thickness. After passage through these tubes, which were 60 mm long, the samples of two-phase mixture were directed through rubber piping into calibrated glass test tubes for separation of the liquid and gas phases; for this purpose the conducting channel in the test tubes was helical. The internal diameter of this channel, like that of the rubber piping, was 4 mm, and their total length was 250 mm. The quality of separation of the water in the test tube was checked by mounting two test tubes in series. The absence of moisture in the second test tube in the cold tests (without heating of the tubes) indicated good removal of moisture in the first test tube. To exclude the possibility of condensation of steam in the test tubes the latter were accommodated in a foam-plastic heat-insulating block. The absence of moisture in the test tube in the case of experiments in which the tube was electrically heated and the water was completely evaporated (in the case of a low flow rate) indicated the absence of condensation in this test tube.

The gas mixture (air and steam) remaining after the first test tube entered a coil contained in an ice bath (the internal diameter of the coil was 7 mm and the tube length was 1000 mm). The steam from the gas sample condensed in the coil and the newly obtained two-phase mixture entered a second test tube, similar to the first, for separation of the liquid phase. Air at a temperature close to 0°C, containing an insignificant amount of steam, emerged from the second test tube.

In calculation of the mass velocity of the flow of water drops it was assumed that none of the drops were deflected in their approach to the sampling tube, i.e., all the liquid in a current jet of diameter 1.8 mm opposite the mouth of the sampling tube entered the tube. Samplers of such type have been used before for sampling of the liquid phase in cold flows [3]. To ensure that all the liquid from a current jet of diameter 1.8 mm entered the sampler, in the early experiments we connected the sampling channel after the two test tubes to a vacuum pump with air flow measurement. This allowed isokinetic sampling of the gas through the tube at a velocity equal to the gas velocity in the investigated tube and also sampling at a much lower velocity. For instance, reduction of the gas flow through the sampling channel

by a factor of 4 in comparison with the gas flow in isokinetic sampling had practically no effect on the amount of separated water in the first test tube and at the same time reduced by a factor of 4 the amount of condensed water in the second test tube. This test indicated that in weight proportion practically all the drops entered the sampling tube without being deflected. Thus, the local value of the mass velocity of the water drops can be calculated with satisfactory accuracy from the formula

$$\rho u = Q/(\tau f), \tag{1}$$

where Q is the amount of liquid in the first test tube; τ, sampling time; f, area of the sampling aperture. All the results of measurements of the mass velocity of water drops will henceforth be represented in the form of relations c(r), where $c = \rho u/(\rho u)_o$; $(\rho u)_o = G/(\pi R_o^2)$ is the mean mass velocity of water over the tube cross section; G, initial mass flow rate of water; R_o, tube radius; $r = R/R_o$, dimensionless radius of the position of the center of the sampling tube. The random error of measurement and calculation of c could reach 6.5%. It is made up of errors of 4% in measurement of the amount of water in the test tubes (2.5-25 g), 2% in measurement of the flow rate in the tube, and 0.5% in determination of the sampling time (250-500 sec). The systematic errors could be 2.5% in measurement of the area of the samplers and 0.5% in the area of the investigated tube. Thus, the largest possible error in measurement of c was 10%.

Figure 2 shows the results of measurements of the mass velocity of the water drops over the length and radius of cold ($q_w = 0$) and heated ($q_w = 2.5 \cdot 10^5$ W/m^2) tubes. The parameter is the water mass flow rate G. As Fig. 2 shows, reduction of the water flow in the tube reduces the relative amount of liquid in the dispersed state. In the investigated range of air velocities with G < 5.9 g/sec for the cold tube and with G < 3.5 g/sec for the heated tube water drops could not be detected in the air flow by means of the described samplers. This fact is in good agreement with the results of visual observations in experiments with transparent tubes [4, 5] and with a plate [6] in the case of liquid films in a turbulent air flow ($U_g = \sim 100$ m/sec). In [6] a relation was given for the critical Reynolds number (determined from the thickness δ and velocity U of the liquid film on the boundary on the assumption of a linear velocity profile), at which only short-wave disturbances appear on the film surface (without appreciable dispersion of the liquid):

$$\mathrm{Re}^* = \frac{\rho U \delta}{\mu} = \frac{2G}{\pi d \mu} = \begin{cases} \alpha^2 & \text{for } \mu/\mu_g \leqslant 50 \\ \alpha^2 \cdot 50/(\mu/\mu_g & \text{for } \mu/\mu_g \geqslant 50, \end{cases} \tag{2}$$

where μ and μ_g are the coefficients of dynamic viscosity of water and air; $\alpha = 12$ is the von Karman constant. According to relation (2) and the conditions of the conducted experiments ($T_w = 287°K$, $\mu/\mu_g = 65$ for cold tube and $T_w = 363°K$, $\mu/\mu_g = 16$ for $q_w = 2.5 \cdot 10^5$ W/m^2), the critical Reynolds numbers Re* are 110 and 144, respectively, and the critical flow rates G* = 4 and 1.5 g/sec. Visual observations [6, 7] showed that at increased flow rates of the liquid the latter is dispersed from the film surface; the degree of dispersion increases with increase in the gas velocity and reduction of the surface tension of the liquid. For Re_s, at which appreciable dispersion of the liquid occurs, we proposed the relation [6]

$$\mathrm{We}_s [1 - (\mathrm{Re}^*/\mathrm{Re}_s)^{3/2}] = 6, \tag{3}$$

where Re* is the critical number given by formula (2); $\mathrm{We}_s = \rho V^2 \delta/\sigma = \mathrm{Re}_s^{3/2} \mu \sqrt{\tau_w/\rho}/\sigma$ is the Weber number, determined from the film parameters and the friction stress τ_w, calculated from the air flow parameters. According to relation (3) and the conditions of the conducted experiments (given in Fig. 2), appreciable dispersion of the water should be observed when G = 7 g/sec with $q_w = 0$ and G = 4 g/sec with $q_w = 2.5 \cdot 10^5$ W/m^2 (the friction stress was calculated for a turbulent gas flow from the Blasius formula [8], derived for a hydrodynamically stabilized section of a smooth-walled tube). As Fig. 2 shows, dispersion of water from the film on the cold wall was detected at a value G = 6 g/sec, which is less than the calculated value from formula (3) (for a heated tube the comparison of these values is approximate owing to evaporation of the water). This disagreement between the results of visual observations and the results of these experiments with a cold wall can be attributed to the following causes: 1) dispersion of water in the tube at G = 6 g/sec occurs at the site of inflow of the water onto the internal surface; 2) when the breakaway of water drops from the surface of the established film (far from the inflow) is detected visually, part of the liquid is already in a dispersed state.

These circumstances were established by visual observations of formation of the liquid film and corresponding measurements by the same samplers on a flat plate at a specific flow

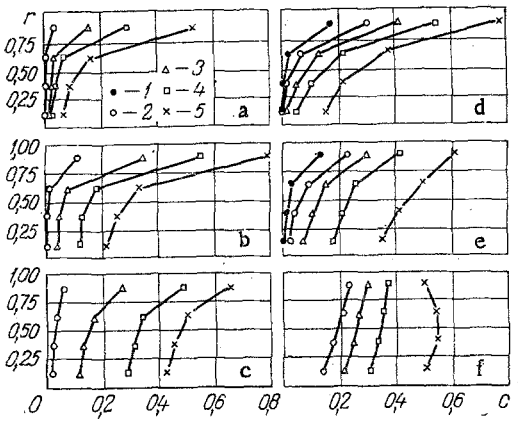

Fig. 2. Distribution of mass flow velocity of drops over the radius (G_g = 35 g/sec): a, b, c) $q_w = 0$; d, e, f) q_w = 2.5·10 W/m²; a, d) l/d = 13; b, e) 25; c, f) 45; 1) G = 4.2 g/sec; 2) 5.9-6.3; 3) 7.1-7.7; 4) 9.5-10.1; 5) 14.7-15.5.

rate g = 0.95 g/(cm·sec), corresponding to a flow of 6.0 g/sec in the tube. The fact is that at the site of inflow of the water jets, Re determined from the parameters of an individual jet is greater than Re determined from the parameters of the film spreading over the whole perimeter of the tube. In addition, We for an individual jet also increases not only due to increase in Re for individual jets, but also due to increased friction stress at points where the air flows over the hillocks that are formed when the horizontal velocity of the liquid is still low. It should be noted that the presence of small water drops in the air flow, which because of their small total volume are not detected by the samplers owing to the low accuracy of measurements by the latter, was discovered by a comparison of the thermocouple readings at the end of the tube (lower in the case of wetting of the thermocouples by drops) with the readings of the dry thermocouples situated upstream of the water inlet. Such comparisons revealed water drops in the flow at values of G from 2 to 6 g/sec. Visual observations of the flow of the film on a plate showed that detachment of small drops occurred mainly at the site of inflow of the water into the air flow.

Measurements of the mean water-drop diameters at the outlet of heated tubes in the range G = 7-15 g/sec by the method of laser light scattering [9] gave the following results: The mean drop diameter decreased from 70 to 25 μm with increase in air flow rate in the tube from 24 to 52 g/sec and was practically independent of the water flow rate. The Stokes parameter [10] $k = \rho U_g b^2/(18\mu D)$, where b is the drop diameter, and D is the diameter of the tube for sampling the two-phase mixture, in the conditions of the conducted experiments was many times greater than unity. This means [10] that the water drops before the sampling tube move in practically a straight line, which was indirectly confirmed by the previously described independence of the measured mass velocity of the liquid drops on the velocity of sampling of air through the sampling tube.

The initial dispersion of the liquid (at its site of inflow) largely depends on the system by which it is delivered to the surface in the gas flow and, as visual observations showed, is insignificant in the overall balance of the dispersed liquid at large flow rates (G > 10 g/sec for our experiments). The relative amount of dispersed liquid at the end of the tube

$$G_d/G = 2\int_0^1 crdr \qquad (4)$$

was calculated by the coarsely approximate numerical integration of the presented distributions of c with a step of 0.25 in r and measured values of c within each step.

From the values of G for experiments with an unheated tube we could calculate the Reynolds number for the undispersed part of the film by analogy with definition (2),

$$\mathrm{Re}_f = 2(G - G_d)/(\pi d\mu), \qquad (5)$$

and these values of Re_f for the flow parameters indicated in Fig. 2a were 158, 193, 225, and 293; in Fig. 2b, 152, 164, 180, and 202; and in Fig. 2c, 150, 161, 167, and 184, respectively, in order of increase in water flow rate. As an analysis of these values indicates, the flow rates of water in the film over tube lengths l/d = 25 to l/d = 45 were practically constant (within the limits of accuracy of the measurements and approximate calculations). Hence, all that happens on this length is leveling out of the profile of the mass velocities of the dispersed water over the tube cross section. The maximum values of Re_f on this part of the tube (with

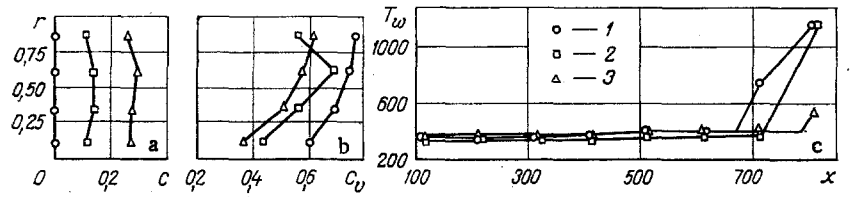

Fig. 3. Distribution of mass velocities of flows of water drops (a) and steam (b) over tube radius and of wall temperature along tube (c) (l/d = 45, G_g = 35 g/sec: 1) q_w = $1.2 \cdot 10^5$ W/m^2, G = 1.76 g/sec; 2) $2.5 \cdot 10^5$ and 4.9; 3) $3.8 \cdot 10^5$ and 9.1. T_w, °K; x, mm.

stabilized flow rate of water in the film) were close in value to Re_s determined from formula (3). In the cross section of a tube with l/d = 13 the Re_f values for the film were greater than its stabilized value. Perhaps the dispersion of the film before its flow rate was stabilized proceeded until this section was reached, but the dispersed liquid was located in such a thin wall layer that the measured maximum value of the mass velocity of the water by the sampler near the wall (r = 0.87), which mainly determines Re_f, was much less than the mean value of c in this layer, and, hence, the Re_f was greater than the stabilized value.

The results of summation of the values of c, presented in Figs. 2d-f, according to formula (4), show that the dispersed part of the water G_d/G varies weakly with tube length l/d from 13 to 45. The water evaporates mainly from the annular ring formed at the start of the tube. As soon as all the film has evaporated, the wall temperature sharply increases. This, for instance, prevented us from obtaining results of measurements with a heat load of $2.5 \cdot 10^5$ W/m^2 for tube length l/d = 45 and G = 4.2 g/sec. All the experimental results presented in Fig. 2 correspond to the case where the water film was present up to the end of the tube.

An analysis of the corresponding results of experiments with different air flow rates showed that an increase in air flow rate leads to more intense dispersal of the film. The value of the stabilized water flow in the film is given approximately by formula (3) and occurs at shorter tube lengths.

Before we presented the results of measurements of the mass velocity of steam in the tube we carried out the methodological experiments required to determine the coefficient of inflow of air and steam into the sampling tubes, which expresses the ratio of the sampling velocity to the velocity of the gas flow in the tube. The actual values of the steam mass velocity are equal to the measured values divided by the inflow coefficient. In our experiments without a vacuum pump the inflow coefficient was always less than unity. Figure 3 shows the results of measurements of the mass velocities of the flows of liquid drops and steam at the outlet of a tube of length 900 mm, and also of the tube wall temperature. Since in these experiments there was no liquid film on the wall at the end of the tube (the wall temperature was significantly higher than the boiling point of water), the inflow coefficient on the basis of the presented data can be calculated from the water-balance equation

$$\psi = 2\int_0^1 c_v r dr \Big/ \left(1 - 2\int_0^1 c r dr\right), \qquad (6)$$

where $c_V = \rho u_V (\pi R_0^2/G)$ is the measured dimensionless mass velocity of the stream, corresponding to the quantity c for the liquid phase. Numerical integration of (6) was performed with a step of 0.25 in r and with measured values of c and c_V within each step. Experiments with a laminar flow of film on the wall were very convenient and more accurate for calculation of ψ. In this case the water in the film completely evaporates and there is no liquid phase at all in the flow (Fig. 3, experiment with G = 1.76 g/sec, c = 0, $\psi = 2\int_0^1 c_v r dr$). As a result of such approximate calculation of ψ for many experiments we obtained the following values: ψ = 0.56-0.68, 0.68-0.77, 0.73-0.88 for G_g = 24, 35, and 52 g/sec, respectively. We subsequently used the mean values of ψ in the indicated ranges. The increase in ψ with increase in airflow in the tube can be attributed to blockage of the tube outlet section by the measuring combs. No dependence of ψ on the water flow rate was detected. Calculation of the hydraulic resistance of the whole sampling channel by the Blasius formula [8] with due consideration of the measured static pressure in the tube in front of the sampling tubes gave approximately the same values for ψ.

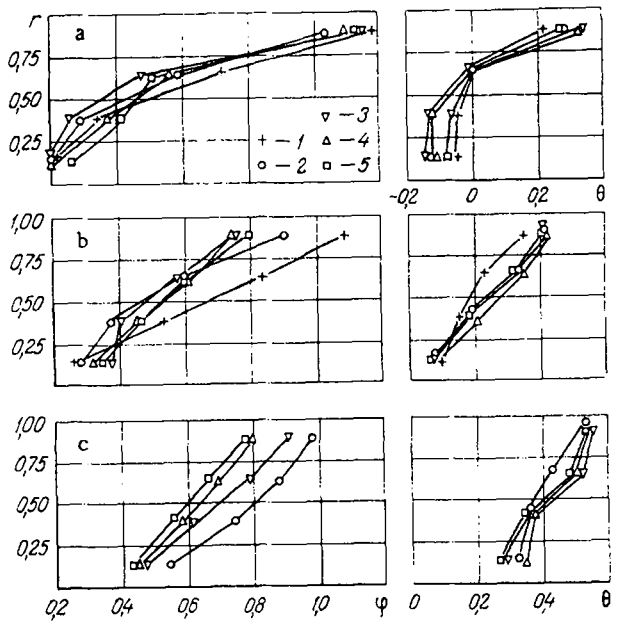

Fig. 4. Distribution of mass velocity of steam (Ψ) and temperature of two-phase flow (θ) in different cross sections of tube ($q_w = 2.5 \cdot 10^5$ W/m^2, G_g = 35 g/sec): a) l/d = 13; b) 25; 3) 45; 1) G = 2.8-3.0 g/sec; 2) 5.8-6.0; 3) 7.2-8.0; 4) 9.5-10.1; 5) 14.7-15.2.

Figure 4 shows profiles of the steam mass velocity and temperature of the two-phase medium in different cross sections of the tube. In these measurements the water film completely covered the heated part of the tube. The steam mass velocity is represented by the dimensionless quantity

$$\varphi = (\rho u_v/\psi)/(4lq_w/dH).$$

The mean value $\bar{\varphi}$ over the tube cross section characterizes the part of the heat flux which goes entirely towards the evaporation of water; the other part is expended on heating of the two-phase medium, whose temperature is given by the dimensionless quantity $\theta = (T - T_o)/(T_s - T_o)$, where T_o is the initial temperature of the air and water; T_s is the boiling point of water.

As Fig. 4 shows, the greater the water flow rate, the less steam is produced by the same supplied heat load. This dependence is weak, however, and in a first approximation can be neglected. The results obtained for G = 2.8 g/sec correspond to a flow of water film without dispersion, and these data differ slightly from the other results, which relate to cases where significant dispersion of the liquid occurred (in such a case the 900-mm tube was destroyed).

It should be noted that the profile of the steam mass velocity over the tube cross section levels out along the tube. The mean values $\bar{\varphi}$, however, obtained by the previously described approximate numerical integration were close to one another for all cross sections and were equal to 0.7-0.8.

The mean temperature of the two-phase medium increases along the tube mainly as a result of increase in stream concentration in the flow (Fig. 4). The temperature profiles level out over the length of the tube (Fig. 4). The negative values of θ at the tube center for l/d = 13 correspond to the readings of the wet thermocouples in the still unheated flow.

NOTATION

R_o, d, l, radius, diameter, and length of the tube; ρ, density; V, velocity; T_w, wall temperature; q_w, specific heat flux; σ, coefficient of surface tension; c, c_v, dimensionless local values of mass velocity of water drops and steam referred to mean mass velocity of water in tube; φ, dimensionless mass velocity of steam referred to mean calculated value; H, specific heat of heating and evaporation of water; θ, dimensionless mixture temperature; Re, We, Reynolds and Weber numbers determined from liquid film parameters; ψ, coefficient of inflow of gas into sampling tube. Subscript g denotes air.

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MATHEMATICAL MODEL OF CONTACT HEAT EXCHANGE OF A GAS AND WATER DURING ADIABATIC EVAPORATION

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It is shown that the gas temperature distribution in a contact heat exchanger is characterized by the presence of two zones during adiabatic evaporation — an initial zone with a reduced temperature and a final zone with an elevated temperature — as compared with the heat exchange through a separating surface.

Heat exchange with direct contact between the high-temperature gas and water is used in different industrial apparatus (moistening columns, scrubbersof "wet" gas cleansers). In all these cases, adiabatic evaporation of the water is observed.

To estimate the modes of operation of technological equipment it is necessary to know the nature of the heat-carrier temperature distribution along the heat exchange surface. The temperature distribution in a contact heat exchanger differs from that which holds in ordinary heat exchangers with separating surfaces which, as is known, is described sufficiently accurately by a logarithmic curve [1, 2].

To find the nature of the change in gas temperature during adiabatic evaporation, we use the fundamental heat exchange equation

$$G_g(c_g + c_v x)\,dt = \alpha\,(t - t_f)\,dF + dG\,(c_{v1}t - c_{v2}t_f). \tag{1}$$

This relationship has two distinctions from the ordinary heat exchange equation through a separating surface: 1) the second term on the right side takes account of the additional expenditure of heat on reheating the connecting couple from the fluid temperature to the gas temperature; 2) during heat exchange the moisture content of the gases, their mass and the mass specific heat ($c_g + c_v x$) vary.

If we find

$$dG = \frac{\alpha\,(t - t_f)\,dF}{r}, \tag{2}$$

determine x from the expression $\alpha(t - t_f) = \beta r(x'' - x)$,

$$x = x'' - \frac{\alpha}{\beta r}\,(t - t_f), \tag{3}$$

substitute (2) and (3) into (1), and take into account that

$$c_g = c_{gf} + \Delta_c(t - t_f) \text{ and } c_v = c_{vf} + \Delta_m(t - t_f),$$

then after manipulation we obtain

$$\left(\frac{C_1}{t_1 - t_f} - \frac{C_2}{t^2 + bt + C} - \frac{C_3 t}{t^2 + bt + C}\right) dt = dF.$$

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