BILATERAL HEAT EXCHANGE IN VAPORIZATION CHANNELS WITH AN INNER SPIRALLY COILED TUBE

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Analytical expressions have been obtained for the profiles of the coolant temperatures and the enthalpy of the heated medium along the length of a vaporization channel with twisted flow, in the case of bilateral heating with forward and reverse motion.

The intensification of heat exchange in heat-exchange equipment, essential to raise the energy levels of the equipment, can be achieved in various ways, including the twisting of the coolant flow.

In vaporization channels with bilateral heating, the twisting of the flow of the heated medium can be accomplished by spirally coiling the inner tube [1]. Thus, the heated medium will move in the direction of the spiral channel formed by the outer and inner tubes in contact with a helical line, while the heating medium will move through the intertube space and through the inner tube. Existing methods of calculating the temperature fields of vaporization channels with bilateral heating [2, 3] do not allow us to take into consideration the exchange of heat between the flows of the heating carrier, said heat exchange occurring at the point of contact between the inner coiled tube and the external tube throughout the entire length of the element.

In this paper we present a method for calculating the temperature fields of the coolant and of the enthalpy of the heated medium in such vaporization channels, with consideration given to the exchange of heat between the coolant flows, for both forward and reverse flows (see Fig. 1).

The one-dimensional equations of energy at the segment dz of the vaporization channel, with the temperature of the heated medium at the inlet equal to the saturation temperature t_s, are described by a system of nonuniform differential equations in whose analytical solution we assumed constancy for the coefficients of heat transfer and the heat capacity of the carrier throughout the length of the vaporizer:

$$\begin{aligned}
-\operatorname{sign}(G_1)\, W_1'\, \frac{dt_1'}{dz} &= K_1 f_1 (t_1' - t_s) - K_3 (1 - f_3)(t_1'' - t_1'),\\
-\operatorname{sign}(G_1)\, W_1''\, \frac{dt_1''}{dz} &= K_2 f_2 (t_1'' - t_s) + K_3 (1 - f_3)(t_1'' - t_1'),\\
G_2 \frac{di}{dz} &= K_1 f_1 (t_1' - t_s) + K_2 f_2 (t_1'' - t_s),
\end{aligned} \tag{1}$$

where the sign for the flow rate of the heating carrier is determined by the direction of motion (the positive direction is assumed to be that which coincides with the 0z axis).

The portions of the surface participating in the convective heat exchange between the flows of the heating and heated media, i.e., f_1, f_2, f_3, are determined from the expressions

$$f_1 = 1 - \frac{p_c}{\pi d_2} k_l; \quad f_2 = 1 - \frac{p_c}{\pi d_3} k_l; \quad f_3 = \frac{2 f_1 f_2}{f_1 + f_2}.$$

Here k_ℓ is a coefficient which shows the extent to which the length of the contact surface is greater than that of the vaporizer as a consequence of its coiled shape: $k_\ell = [(\pi d_3)^2 + S^2]^{1/2}/S$.

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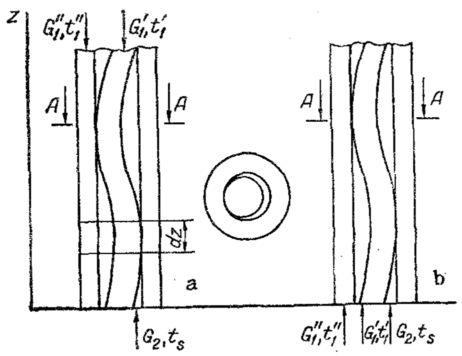

Fig. 1

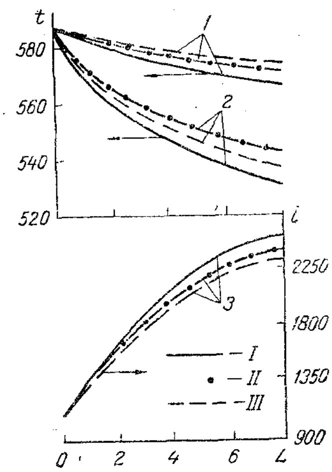

Fig. 2

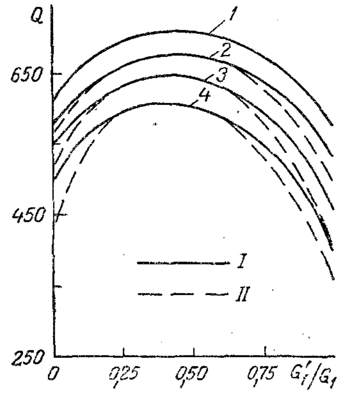

Fig. 3

Fig. 1. Theoretical diagram of vaporization channel: a) counterflow; b) foward flow.

Fig. 2. Distribution of coolant temperatures and the enthalpy of the heated medium along the length of the vaporization channel: 1-3) the heating medium in the intertube space, in the inner spirally coiled tube, and the heated medium, respectively; I) without consideration of contact; II) with allowance for contact; III) with allowance only for the shielding of the heat-exchange surface. i, kJ/kg.

Fig. 3. Thermal power of vaporizer as function of flow-rate relationships: 1) p_c = 0; 2) 0.003; 3) 0.005; 4) 0.007 m; I) with allowance for contact; II) with allowance only for surface shielding. Q, kW.

The linear heat-transfer coefficients K_1, K_2, K_3 are found from the familiar relationships, for which

$$K_3 = \pi \Big/ \left(\frac{1}{\alpha_1 d_1} + \frac{1}{2\lambda_1} \ln \frac{d_2}{d_1} + \frac{1}{2\lambda_2} \ln \frac{d_4}{d_3} + r_c + \frac{1}{\alpha_2 d_4} \right).$$

Having denoted $k_1 = K_1 f_1$, $k_2 = K_2 f_2$, and $k_3 = (1 - f_3)K_3$, by substitution of $t_1' = y_1' + t_s$ and $t_1'' = y_1'' + t_s$, we reduce the first two equations of system (1) to the uniform

$$-\operatorname{sign}(G_1)\, W_1' \frac{dy_1'}{dz} = (k_1 + k_3)\, y_1' - k_3 y_1'', \quad -\operatorname{sign}(G_1)\, W_1'' \frac{dy_1''}{dz} = -k_3 y_1' + (k_2 + k_3)\, y_1''. \qquad (2)$$

The characteristic equation of system (2), transformed to

$$\gamma^2 + \mathrm{sign}(G_1)\,\sigma\gamma + \beta = 0,$$

has the following roots:

$$\gamma_{1,2} = -\mathrm{sign}(G_1)\frac{\sigma}{2} \pm \frac{1}{2}\sqrt{\sigma^2 - 4\mathrm{sign}(G_1)\,\beta}, \tag{3}$$

where

$$\sigma = \frac{W_1''(k_1+k_3) + W_1'(k_2+k_3)}{W_1' W_1''}; \quad \beta = \frac{k_1k_2 + k_1k_3 + k_2k_3}{W_1' W_1''}.$$

We will write the solution of system (2) in the form

$$t_1'(z) = A_1\exp(\gamma_1 z) + B_1\exp(\gamma_2 z) + t_s, \quad t_1''(z) = A_2\exp(\gamma_1 z) + B_2\exp(\gamma_2 z) + t_s. \tag{4}$$

Having substituted (4) into (2), with consideration of the initial conditions (z = 0), we obtain the following relationships between the integration constants:

$$A_1 = A, \quad A_2 = \varkappa A, \tag{5}$$

$$B_1 = B, \quad B_2 = \varkappa^* B, \tag{6}$$

where

$$\varkappa = \frac{k_3}{k_2 + k_3 + \mathrm{sign}(G_1)\,W_1''\gamma_1}; \quad \varkappa^* = \frac{k_3}{k_2 + k_3 + \mathrm{sign}(G_1)\,W_1''\gamma_2}.$$

Bearing in mind expressions (5)-(6), we will write the general solutions (4) of the system of equations (2) in the form

$$t_1'(z) = A\exp(\gamma_1 z) + B\exp(\gamma_2 z) + t_s, \quad t_1''(z) = A\varkappa\exp(\gamma_1 z) + B\varkappa^*\exp(\gamma_2 z) + t_s. \tag{7}$$

The integration constants A and B are found from the following boundary conditions for forward and reverse heat-carrier motions, respectively:

$$t_1'(0) = T_1, \quad t_1''(0) = T_1; \tag{8}$$

$$t_1'(L) = T_1, \quad t_1''(L) = T_1. \tag{9}$$

Having sequentially substituted (8) and (9) into (7), and solving the system of equations for A and B, we find the integration constants of system (2), and their substitution into (7) allows us to derive expressions for the calculation of the temperature fields in flows of the heating heat carrier as these move in the forward and reverse directions:

$$t_1'(z) = t_s + (T_1 - t_s)\frac{(\varkappa^* - 1)\exp(\gamma_1 z + \gamma_2 Ln) - (\varkappa - 1)\exp(\gamma_1 Ln + \gamma_2 z)}{(\varkappa^* - \varkappa)\exp(\gamma_1 + \gamma_2)\,Ln}, \tag{10}$$

$$t_1''(z) = t_s + (T_1 - t_s)\frac{\varkappa(\varkappa^* - 1)\exp(\gamma_1 z + \gamma_2 Ln) - \varkappa^*(\varkappa - 1)\exp(\gamma_1 Ln + \gamma_2 z)}{(\varkappa^* - \varkappa)\exp(\gamma_1 + \gamma_2)\,Ln}, \tag{11}$$

where n = 1 indicates the counterflow; n = 0 indicates the forward flow.

Integration of the last equation in (1) over length with consideration of (10) and (11), as well as of the boundary condition $[i(0) = i_s]$, gives us an expression to calculate the enthalpy of the heated medium through the height of a vaporization channel with an inner coiled tube with bilateral heating for the most common heat-exchange processes:

$$i(z) = i_s + \frac{T_1 - t_s}{G_2(\varkappa^* - \varkappa)\gamma_1\gamma_2\exp(\gamma_1+\gamma_2)\,Ln}\{\gamma_2(k_1 + k_2\varkappa)(\varkappa^* - 1)\times$$

$$\times[\exp(\gamma_1 z) - 1]\exp(\gamma_2 Ln) - \gamma_1(k_1 + k_2\varkappa^*)(\varkappa - 1)[\exp(\gamma_2 z) - 1]\exp(\gamma_1 Ln)\}.$$

Realization of this mathematical heat-exchange model on a computer allows us to calculate the distribution of temperatures over the length of the vaporizer by segments, and within the limits of the segment under consideration the thermophysical parameters of the heat carrier and the structural materials, as well as the heat-transfer coeffcients, are averaged over the length of the segment and assumed to be constant. At the next stage of the calculation, the thermophysical properties are calculated from the corresponding temperatures at the outlet from the preceding segment, while the coefficients of heat transfer to the heated medium are calculated from the magnitudes of the heat flow and the vapor content of that flow.

Variation calculations of the vaporization module were carried out to study the effect of contact between the inner and outer tubes on the exchange of heat in vaporization channels; the vaporization module was 8 m long and exhibited the following geometric and regime parameters: $d_1 = 0.006$ m, $d_2 = 0.01$ m, $d_3 = 0.013$ m, $d_4 = 0.018$ m, $S = 0.04$ m, $p_c = 0.003$ m, and $G_1' = 0.8$ kg/sec, $G_1'' = 3.7$ kg/sec, $G_2 = 0.5$ kg/sec.

The distribution of the enthalpy of the heated medium and of the temperatures of the flows of the heating heat carrier in the case of forward flow is shown in Fig. 2. Analysis of the calculation data shows that shielding of the heat-exchange surface and the exchange of heat at the point of contact stand out as two factors of opposite effect: the shielding of the heat-exchange surface reduces the quantity of transferred heat within the vaporization channels as a result of the reduction in the heat-exchange surface by a magnitude equal to that of the shielding area, while the exchange of heat between the flows of the heating media, reducing the temperature difference between these flows, leads to a more efficient utilization of the thermal energy of these flows. In this case, the first factor exerts greater influence on the exchange of heat than does the second; as a result, the enthalpy of the heated medium within the vaporization module is smaller when we allow for contact than is the case when no consideration is given to contact.

Figure 3 shows the magnitude of the power at the outlet from the vaporization module as a function of the flow-rate relationships for the flow rates of the heating medium. These results make it possible for us to isolate the optimum relationship of flow rates through the inner tube and the intertube space which, for the design under consideration, is found to range from 0.41 to 0.5.

Analysis of the results shown in Fig. 3 demonstrates that in calculating bilateral heat exchange allowance only for the factors of heat-exchange surface shielding, e.g., by increase in the reserve factor reduces the magnitude of the transmitted power over a broad range of relationships for the heating medium.

NOTATION

W is the water equivalent, W/K; t, temperature, K; i, enthalpy, J/kg; z, spatial coordinate, m; K_1 and K_2, linear coefficients of heat transfer for the inner and outer tubes, W/(m·K); p_c, portion of the surface participating in the convective heat exchange; α_1, α_2, coeffcients of heat transfer from the inner surface of the inner tube and from the outer surface of the outer tube, W/(m²·K); λ_1 and λ_2, coefficients of thermal conductivity for the inner and outer tubes, W/(m·K); T_1, temperature at the inlet of the heating carrier; d_1 and d_2, inside and outside diameters of the inner tube, m; d_3 and d_4, the inside and outside diameters of the outer tube, m; S, coil pitch of the inner tube, m; L, length of the vaporization channel, m; G, mass flow rate, kg/sec; subscripts for t, W, and G: 1) heating medium; 2) heated medium; ') medium within the inner tube; ") medium within the intertube space.

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