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DETERMINATION OF THE COEFFICIENT OF HEAT TRANSFER AT THE INNER SURFACE OF A TWO-PHASE HEAT EXCHANGER

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Heat transfer at the inner surface of a two-phase system of thermal regulation is investigated. A method of conjugate gradients for an inverse problem of nonsteady heat conduction is used. The time dependence of the heat transfer coefficient is found for the startup regime of the thermal regulation system, and the calculation accuracy is estimated.

The creation of new engineering models requires extensive experimental research with processing of test data, including that on thermal regimes. Heat exchangers based on a closed evaporation–condensation cycle are now being created, and one of the main problems, even in the preliminary design stage, is the determination of heat-transfer coefficients.

Let us consider the hydraulic loop of a thermal regulation system (Fig. 1) designed to stabilize the temperature of radio apparatus. The heat-releasing elements 6 are located on a heat plate 1 made of PK 01309 aluminum extrusion with a capillary structure in the form of rectangular grooves on the inner surface. The grooves 7 and the condensate pipe 5 are filled with acetone. The heat exchanger works on the heat-pipe principle. The heat released in the operation of instruments goes into heating and evaporating the coolant, the vapor goes through the pipe 3 into the condenser 4 where it condenses, and the liquid goes through the main 5 into the capillary structure of the heat plate 1. Heat and mass transfer occur due to capillary pressure generated at the phase interface of the capillary structure.

The high efficiency of the heat exchanger is achieved by heat transfer of the latent heat of vaporization of the coolant. The absence of moving mechanical parts, a power supply, or a system of automatic regulation makes evaporation–condensation devices more reliable than traditional devices and improves the weight characteristics.

The experimental determination of the heat-transfer coefficient in the evaporation zone is complicated by the fact that the height of a vapor channel is fairly small ($b = 0.004$ m). The methods of inverse problems of nonsteady heat conduction [1] therefore seem the most suitable for finding α. Here the boundary conditions are specified from experience.

The problem consisted in an investigation of the start-up regime of the thermal regulation system. The heat-releasing elements were simulated by pumping hot water through an auxiliary heat exchanger (not shown in Fig. 1). In the course of the experiment, we measured the applied heat flux q, the temperature T_W of the outer wall, and the vapor temperature T_V in the inner cavity (Fig. 2).

The amount of heat was determined by calorimetry and the temperatures were measured with Chromel–Copel thermocouples (wire diameter 0.003 m) and KSP-4 potentiometers.

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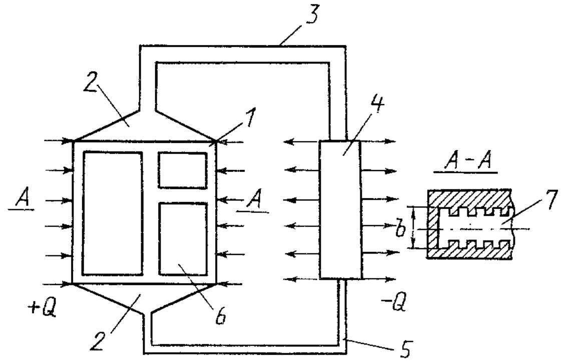

Fig. 1. Diagram of the two-phase heat exchanger: 1) heat plate; 2) collector; 3) vapor pipe; 4) condenser; 5) condensate pipe; 6) heat releasing elements; 7) capillary structure filled with coolant.

The heat-exchange surface consists of a flat wall with rectangular grooves on the inner surface (see Fig. 1). Since the transverse dimensions of a groove are fairly small (0.5 × 0.5 mm), the profile of this surface does not significantly affect the heat transfer, and we took a flat wall for the heat-transfer model. The mathematical statement of the inverse problem of nonsteady heat conduction has the following general dimensionless form:

$$\frac{\partial\Theta}{\partial\tau} = \frac{\partial^2\Theta}{\partial\xi^2}, \tag{1a}$$

$$\tau = 0,\ \Theta = 0, \tag{1b}$$

$$\xi = 0,\ P(\tau) = \frac{\partial\Theta}{\partial\xi}, \tag{1c}$$

$$\xi = 1,\ \frac{\partial\Theta}{\partial\xi} = Q,\ \Theta = \Phi(\tau), \tag{1d}$$

where $\xi = x/b$, $\tau = at/b^2$, $\Theta = (T - T_0)/T_0$, and $Q = bq/(\lambda T_0)$ are the dimensionless coordinate, time, temperature, and heat flux; $\Phi(\tau) = (T_W - T_0)/T_0$ is the dimensionless temperature of the outer surface; $P(\tau) = \alpha b/(\lambda T_0)(T_0 + T_0\Theta - T_V)$ is a function to be determined.

The boundary condition (1c) is a condition of the inverse problem. We shall treat the system of equations (1) as a problem of optimum control, and $P(\tau)$ will then be the control function. For the solution we use the method of iterative regularization developed by O. M. Alifanov [2].

As the target function we write the rms discrepancy

$$J(P) = \int_0^{\tau_{max}} [\Theta(P,\ 1,\ \tau) - \Phi(\tau)]^2\, d\tau, \tag{2}$$

enabling us to estimate the departure of the temperature $\Theta(P, 1, \tau)$, calculated for the heat-flux density $P(\tau)$, from the measured temperature $\Phi(\tau)$.

To regularize the inverse problem, we use the iterative algorithm

$$P^{k+1} = P^k - \Delta P^k,\ k = 0,\ 1,\ 2, \ldots,$$

where P^0 is the initial approximation. We determine the correction ΔP^k at each step from the condition of a decrease in the functional $J(P^k) < J(P^{k-1})$.

Starting from the initial approximation P^0, we construct an iterative sequence

$$P^{k+1} = P^k - \beta_k S^k. \tag{3}$$

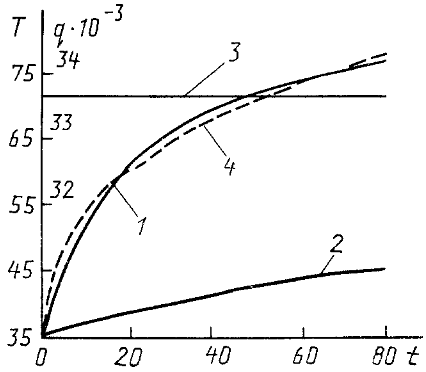

Fig. 2

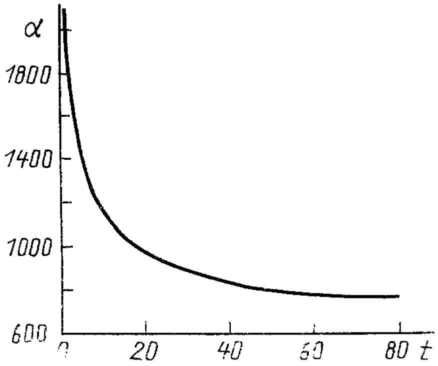

Fig. 3

Fig. 2. Experimental values of the heat-transfer characteristics and reconstructed temperature of the surface as a function of time t, sec: 1) temperature T_w of outer surface of wall, °C; 2) vapor temperature T_v, °C; 3) applied heat flux q, W/m^2; 4) reconstructed wall temperature T_w, °C.

Fig. 3. Heat-transfer coefficient α, W/(m·°C), as a function of time t, sec.

The numerical parameters B_k are determined from the condition min $J(P^k - \beta_k S^k)$ in each iteration. Hence, we will have

$$\beta_k = \left. \frac{\int\limits_0^{\tau_{\max}} \Delta\Theta\,[\Theta - \Phi(\tau)]\,d\tau}{\int\limits_0^{\tau_{\max}} (\Delta\Theta)^2 d\tau} \right|_{\xi=1}. \tag{4}$$

The values of S^k are calculated in accordance with [3],

$$S^k = J'(P^k) + \gamma_k S^{k-1}, \tag{5}$$

where

$$\gamma_0 = 0,\ k = 1, 2, 3, \ldots;$$

$$\gamma_k = -\frac{\int\limits_0^{\tau_{\max}} J'(P^k)\,[J'(P^{k-1}) - J'(P^k)]\,d\tau}{\int\limits_0^{\tau_{\max}} [J'(P^{k-1})]^2\,d\tau}. \tag{6}$$

To use this algorithm, we must calculate the derivative $J'(P^k)$ of the functional. For this, we consider a problem conjugate to (1):

$$\begin{gathered} -\frac{\partial\Psi}{\partial\tau} = \frac{\partial^2\Psi}{\partial\xi^2}; \\ \tau = \tau_{\max},\ \Psi = 0; \\ \xi = 0,\ \frac{\partial\Psi}{\partial\xi} = 0; \\ \xi = 1,\ \frac{\partial\Psi}{\partial\xi} = 2\,[\Theta(1, \tau) - \Phi(\tau)], \end{gathered} \tag{7}$$

from which we obtain the value of $\Psi(0, \tau)$, and then

$$J'(P^k) = \Psi(0, \tau). \tag{8}$$

To find the boundary value of the temperature, we write the initial-value problem

$$\begin{gathered}\frac{\partial \Theta}{\partial \tau}=\frac{\partial^2 \Theta}{\partial \xi^2};\\ \tau=0,\ \Theta=0;\\ \xi=1,\ \frac{\partial \Theta}{\partial \xi}=Q(\tau);\\ \xi=0,\ \frac{\partial \Theta}{\partial \xi}=P(\tau).\end{gathered}\tag{9}$$

We find the temperature increase from the solution of the boundary-value problem

$$\begin{gathered}\frac{\partial \Delta\Theta}{\partial \tau}=\frac{\partial^2 \Delta\Theta}{\partial \xi^2};\\ \tau=0,\ \Delta\Theta=0;\\ \xi=1,\ \frac{\partial \Delta\Theta}{\partial \xi}=0;\\ \xi=0,\ \frac{\partial \Delta\Theta}{\partial \xi}=\Delta P,\end{gathered}\tag{10}$$

where $\Delta P = \beta_k S^k$.

All the quantities required for the iterative process (3) are determined in this way.

The boundary-value problems (7)-(10) were solved by the finite-difference method. A purely implicit-difference scheme with an advance was used. For the differential equation (9) we have

$$\frac{\Theta_{i,j}-\Theta_{i,j-1}}{\Delta\tau}=\frac{1}{h^2}\left(\Theta_{i+1,j}-2\Theta_{i,j}+\Theta_{i-1,j}\right),\tag{11}$$

where $\Theta_{i,\ j}$ is the difference analog of the temperature; the indices i and j correspond to the numbers of the grid node with respect to the coordinate and time.

The function $\Theta_{i,\ j}$ being sought was found by an inverse sweep using the well-known recursion relations of [4]. The calculations were carried out for an aluminum wall 4 mm thick with the following input parameters of the problem: c = 921 J/(kg·deg), λ = 204 W/(m·deg), ρ = 2670 kg/m, T_0 = 35°C, and q = 33,738 W/m^2; the time step was $\Delta\tau$ = 0.1 and the coordinate step was h = 0.01. We limit the iterative sequence to the number k for which the discrepancy criterion

$$2J(P^k-\beta_k S^k)\approx\delta^2$$

is satisfied, where δ is the error of the input data.

The results of a calculation are given in Fig. 3. Since the heat flux is initially supplied to the wall in a step, the heat-transfer coefficient, as one would expect, varies from infinity to some steady-state value. To estimate the reliability of the results obtained, we solved the direct heat-transfer problem using the value of α found.

A comparison of the reconstructed wall temperature with experimental data shows good agreement (the deviation is within 5%), which indicates the adequacy of the calculated heat-transfer coefficient to the physical process.

The method of conjugate gradients thus enables us to obtain reliable results for a flat wall. The calculated heat-transfer coefficient can be interpreted as a characteristic of the nonsteady heat-transfer process corresponding to acetone evaporation from the surface of the capillary structure in the start-up regime of the heat exchanger.

NOTATION

T, temperature; T_0, initial temperature; α, heat-transfer coefficient; x, coordinate; t, time; λ, thermal conductivity; a, thermal diffusivity; c, heat capacity; ρ, density; $\Delta\tau$, time step; h, coordinate step.

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SPECIAL FEATURES OF FLOW AND HEAT TRANSFER IN STAGGERED BUNDLES OF TRANSVERSELY FINNED TUBES

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The special features of flow and heat transfer in staggered bundles of tubes with external circular fins are investigated in a wide range of their geometrical characteristics.

The flow in bundles of transversely bathed tubes with external circular fins is distinguished by considerable complexity, because of which it is very difficult to investigate its characteristics. In this situation, the acquisition of even qualitative results is justified from the standpoint of a deeper physical understanding of the process and the possibility of explaining features of heat transfer in the system under consideration.

The present paper continues the investigation of [1], carried out using a method of visualization of the air stream with a kerosene-soot suspension. In the case under consideration, we posed the problem of obtaining flow patterns in the bundle as a whole, for which the image was recorded on the surfaces, polished and coated with white nitroenamel, of the lower tube plates of bundles with different cross-sectional sizes and longitudinal spacings. The experimental procedure is similar to that described in [1]. In Figs. 1 and 2 we show the results of experiments with tubes having the following geometrical characteristics: $d = 21$ mm, $h = 30$ mm, $t = 4.0$ mm, $\delta = 1.2$ mm, $\psi = 38.3$.

Because the flow patterns were recorded near the wall of the working section rather than in the stream core, they were obviously affected by processes related to the development of the intrinsic boundary layer of the wind tunnel. This effect can be assumed to be slight, however, particularly because the boundary layer at the wall was broken up by the tube array of the bundle, and the flow conditions in the gaps between the end fins and the tube plates were similar to the conditions in the gaps between fins, since these gaps were made equal in constructing the tubes and the working section of the stand. This is confirmed by the fact that the images on sections of the tube plates corresponding to fin boundaries, which are denoted on the photographs by increased stream velocities (with respect to the velocities in the gaps between tubes) in the channels between fins, agree fairly completely with the images obtained directly on the fins [1].

A study of the photographs reveals certain qualitative regularities in the bathing of tubes in bundles. The main source of disturbances in the stream bathing the tubes in a bundle is the cylinder carrying the fins. The large-scale, three-dimensional vortex formations originating from the separation from it of swirled shear layers [1], immediately after emerging beyond the channels between fins, break up into a small-vortex structure, forming a clearly defined, dark turbulent wake behind a finned tube. The dimensions of this wake and the nature of its interaction with tubes lying downstream, for constant fin parameters, depend on the spacings S_1 and S_2 of the bundle and are taken into account fairly well by the parameter S_1/S_2.

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