COMPARATIVE STUDY OF THERMODYNAMIC EFFICIENCIES OF EVAPORATION SCHEMES IN CAPILLARY STRUCTURES

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A procedure is described and principal results are presented for an experimental comparison of the thermodynamic efficiencies of the classical evaporation scheme in capillary structures and the scheme with heat and mass counterflows used in contour heat pipes.

Statement of Problem. The organization of highly efficient heat transfer over considerable distances in a mass force field is one of the important problems of modern heat engineering. The use of heat pipes (HP) is an optimal solution to this problem. They are of interest owing to the possibility of their use in new technology as well as to the principles of heat-exchange intensification by phase transitions realized in capillary structures (CS) of heat pipes. Since the problem of heat-transfer organization is encountered in all areas of industry, heat pipes are used or can be used in radio engineering, electrical engineering, power engineering, metallurgy, the chemical, light, and food industries, mechanical engineering, etc. [1, 2]. Along with the well-studied classical heat pipes, exhaustive reviews of which have been published [3, 4], contour heat pipes (CHP), which employ a substantially different evaporation scheme, have recently gained wide use [5-7]. This scheme is also used in new heat-pipe designs – namely, in heat pipes that employ the principle of the "inverted meniscus" (IMHP) [8].

The structural characteristics of these heat pipes, such as separation of vapor and liquid flows and the use of a high-head capillary structure localized in the heating zone, require an evaporation scheme in the capillary structure that has heat and mass counterflows. A number of experiments with contour heat pipes as well as with inverted-meniscus heat pipes [5-8] do not provide a clear answer to the fundamental question: Is the scheme more efficient thermodynamically than that employed in classical HP's? It is therefore important to perform a careful comparative experiment to determine the thermodynamically efficient evaporation schemes for various types of capillary structures used as HP wicks.

Experimental Setup. A measuring cell was constructed that, with slight changes, could simulate a classical evaporation scheme as well as a scheme with heat and mass counterflows. The experiment was organized differently in each case.

The measuring cell shown in Fig. 1 was used to study heat and mass transfer in evaporation in a capillary structure organized by the classical scheme. When the thermal load from a resistance heater 2 is delivered to the capillary structure 1, a vapor is formed and is carried upward through large pores, breaking through the thin coolant layer above the CS (h = 1-3 mm), into a collector 3. Then the vapor passes through a line 4 into a condenser 5. The condensed coolant is returned through a line 6 to the evaporation zone, where it is fed to the structure 1. A slow increase in the supplied thermal power caused a gradual transition from individual vapor bubbles that penetrated the coolant layer to chains of bubbles that were emitted from the large pores. But with a further increase in heating, evaporation became so strong that it was impossible to determine the thickness of the coolant layer – more accurately, that of the layer of boiling vapor–water mixture above the studied capillary structure.

The setup was somewhat different for the study of heat and mass transfer in evaporation in a capillary structure with heat and mass counterflows (see Fig. 2). In this case, when the thermal load from a resistance heater 3 was delivered to the studied CS 1, the vapor formed was collected in special channels 2 and carried by a line 4 to a condenser 6. The

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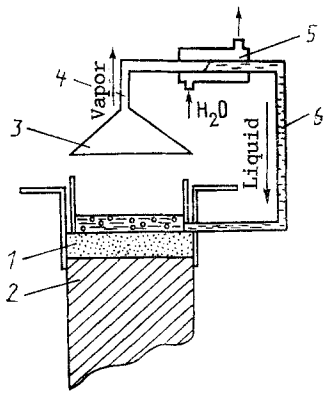

Fig. 1

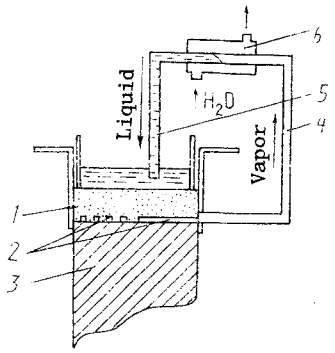

Fig. 2

Fig. 1. Experiment setup for study of evaporation by classical scheme.

Fig. 2. Experiment setup for study of evaporation by scheme with heat and mass counterflows.

TABLE 1. Principal Parameters of Specimens

Number	Capillary-structure type	CS material	d (part) × 10^{-6}, m	d (pore) × 10^{-6}, m	Π, %	K, 10^{-13} × m^2
1	Porous powdered materials	Titanium	16,0	3,42	57,6	4,14
2		»	16,0		49,0	
3		»	16,0		41,1	
4		Nickel	0,45	1,03	71,3	0,405
5		»	0,45	0,90	70,3	0,283
6		»	0,90	1,21	60,1	0,247
7	High-porosity cellular materials	Monel metal		600/10	90,0	5200
8		Copper		600/10	84,0	
9		»		600/10	87,0	
10	Porous screen materials	Brass		120	66,0	490
11		»		80	66,0	164
12		Stainless steel		130	45,0	14,5
13				130	41,0	13,2

condensed coolant passed through a line 5 into the evaporation zone, where it adds to the pool of coolant above the specimen capillary structure 1 (h = 1-3 mm). The coolant is fed directly to the evaporation region by the hydrostatic pressure of a small column of liquid above the specimen as well as by capillary forces. Here we shall present only the results of experiments performed under conditions that were most favorable for the classical evaporation scheme – conditions under which additional capillary feeding of the specimen was not required, i.e., the heating zone was located below the condensation zone and the coolant flowed into the evaporation zone under the influence of mass forces. Doubly distilled water was used as the coolant in all of the experiments described below.

Experiment Procedure. In order to detect a common tendency, comparative studies were performed on three different types of capillary structures: porous screen materials (PSM), porous powdered materials (PPM), and high-porosity cellular materials (HPCM). The principal parameters of the structures studied are listed in Table 1. It must be noted that the surfaces of the PSM and HPCM specimens were oxidized beforehand to improve their wettability.

The experiment consisted of plotting the heat-transfer coefficient as a function of the supplied thermal power for the different evaporation schemes followed by comparison of the results. The experiment was organized so that the maximum relative error did not exceed 7% for determination of the heat-transfer coefficient and 6% for determination of the supplied thermal power. All experimental studies can be divided into several cycles, which are described in detail below.

Firstly, to check the reliability of our data, the results obtained with the experimental apparatus were compared with those of other authors. The check consisted of obtaining a curve of boiling on the smooth surface of the heater and comparison of the results with similar results. The curve is shown in Fig. 3; it almost completely coincides with the curve plotted by the equation

$$\mathrm{Nu} = 75\,\mathrm{K}^{0,7}\,\mathrm{Pr}^{-0,2}, \tag{1}$$

which was proposed by Tolubinskii [9] on the basis of generalization of a large quantity of experimental data. Moreover, this experimental curve is in good agreement (slightly overstated) with that obtained by the widely used boiling-curve equation

$$\alpha = 2{,}6q^{0,7}, \tag{2}$$

which was proposed by Kovalev and Len'kov [10].

The second step consisted of selection of a type of capillary structure from the available types (see Table 1) for further study. The studies consisted of the structure that provided the thermodynamically most efficient evaporators for identical dimensions. The evaporaton scheme with heat and mass counterflows as well as the scheme used in classical heat pipes were investigated.

Finally, the last experiment step consisted of determining the best configuration (geometry) of the selected types of capillary structures for the different evaporation schemes. For the classical scheme, optimization was limited to study of the effect of capillary-structure thickness on the efficiency of evaporation. But optimization for the evaporation scheme with heat and mass counterflows consisted, in addition to study of the effect of CS thickness on the thermodynamic efficiency of the evaporator, of determination of the effect of the number and dimensions of the vapor-discharge channels.

Experiment Results and Discussion. The large quantity of experimental material makes the careful discussion of all results impossible. We shall therefore examine only the data required for direct solution of the problem formulated here.

According to the results of the second step of the experiment, the following types of capillary structures were selected for further study (see Table 1):

specimen No. 1 for porous powdered materials;
specimen No. 7 for high-porosity cellular materials;
specimen No. 10 for porous screen materials.

To optimize the geometry of the evaporation zone, the different schemes were studied separately. The following results were obtained for the classical scheme.

For the porous powdered materials, as which fine-pore titanium capillary structures were studied, the specimen with h = 1 mm was selected for further comparison as the thermodynamically most efficient (specimens with h = 6, 5, 4, 3, 2, and 1 mm were examined; it was not possible to produce a thinner specimen with the technology used to make the PPM specimens) [11].

Of the HPCM specimens (specimen No. 7 in Table 1), the specimen with thickness h = 10 mm was found to be most efficient (specimens with thicknesses h = 2, 3, 4, 6, 8, 12, and 14 mm were also studied). This is obviously a result of the presence of a second type of pore, in addition to the main pores (d_p = 600 μm). These pores (d_p = 10 μm) supply the liquid to a greater specimen thickness, which greatly increases the evaporation area and,

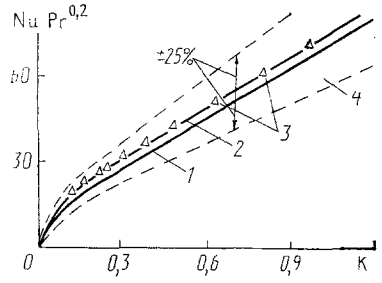

Fig. 3. Generalization of experimental data in the form of dimensionless curves for boiling of distilled water on free surface of heater: 1) curve calculated by Eq. (2); 2) by Eq. (1); 3) experiment; 4) region of spread of experimental data of other authors [9].

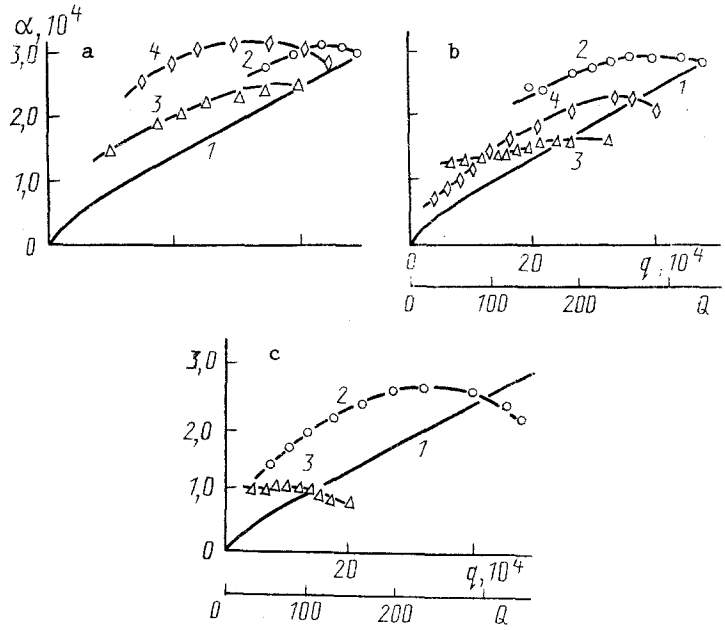

Fig. 4. Heat-transfer coefficient versus supplied thermal power. 1) Boiling on free surface of heater; 2) evaporation organized by scheme with heat and mass counter flows; 3, 4) evaporation on free surface of capillary structure. a) HPCM, water; 3) δ = 4.5 mm, 4) 10; b) PSM, water; 3) δ = 4.5 mm, 4) 0.5; c) PPM, water. α in W/(m²·K); q in W/m²; and Q in W.

therefore, improves the thermodynamic efficiency over that of the thinner specimens. However, since the creation of classical heat pipes with a CS thickness on the order of 10 mm is disadvantageous in most cases, owing to increased size and weight, the specimen with thickness h = 4.5 mm was selected for comparison along with the thermodynamically most efficient specimen.

The specimen of two layers of brass screen No. 10 (see Table 1) with total height h = 0.5 mm was the thermodynamically most efficient of the porous screen specimens. However, since this thickness is most often insufficient in practice for coolant transport in heat pipes, the specimen with h = 4.5 mm was selected for study.

It is very complicated to obtain the most efficient geometry (dimensions) for capillary structures with heat and mass counterflows. An attempt to solve the problem for porous powdered materials was undertaken earlier [12]. The results obtained then were used to prepare PPM specimens for comparison. The specimens had height h = 7 mm and channel depth a = b = 1 mm.

For the remaining capillary structures, optimization was performed by the procedures described earlier [12], except that a system of parallel vapor-discharge channels with width a = 2 mm at equal distances from one another (c = 3 mm) was selected initially as a basis for the present optimization. As a result of the experimental series, it was found that the most efficient PSM specimens (No. 10 in Table 1) were those with thickness h = 4 mm and vapor-discharge-channel depth b = 1.5 mm. The most efficient HPCM specimens (No. 7 in Table 1) were those with thickness h = 15 mm and vapor-discharge-channel depth b = 1.5 mm.

The results obtained for the selected capillary-structure specimens were compared with respect to the maximum thermal flux removed and the heat-transfer coefficient. Graphs of the heat-transfer coefficient as a function of the supplied thermal power obtained in the comparative experiments are shown in Fig. 4. It is apparent that for all types of capillary structures, the transition from the classical scheme to a scheme with heat and mass counterflows increases the maximum thermal flux removed and shifts the peak of the curve of the heat-transfer coefficient into the region of higher specific thermal fluxes. (The thermal power supplied was limited by shortcomings of the apparatus, which did not allow the temperature differential on the heat-measuring plate of the heater according to which the power was determined to be exceeded by more than ΔT = 40 K.) For large-pore capillary structures

of HPCM, the gain in maximum removable thermal power was only about 7%, which is obviously due to insufficiently accurate optimization of the geometry of the specimen with vapor-discharge channels. This gain was as much as 20-25% for the porous screen specimens and more than 130% for the fine-pore titanium structures (PPM). The heat-transfer coefficients were substantially higher for the screen and porous materials for the scheme with heat and mass counterflows and approximately the same for the HPCM specimens for both schemes. The results of the experiments demonstrate that the organization of evaporation in the examined types of capillary structures with heat and mass counterflows, which are used in contour heat pipes as well as inverted-meniscus heat pipes, is more efficient thermodynamically than those with a classical scheme. It must be noted again that the results on which this conclusion is based were obtained in experiments realized under conditions most favorable for the evaporation scheme – without additional capillary supply.

The physical causes of the noted differences are not completely clear at the present time. It is evidently necessary to construct models of heat exchange and evaporation in a capillary structure filled with a vapor–liquid mixture. Such work has actually been started [12], and the authors hope to continue it in the future. It is, however, already highly probable that the removal of vapor directly from a heat-transfer surface must to a considerable extent prevent the pores of the capillary structure from being filled with vapor, and this will increase the effective area of the liquid–solid phase-contact surface. Thus, practically all of the specific surface of the capillary structure must participate in the heat-exchange process.

NOTATION

α is the heat-transfer coefficient, δ is the thickness of the capillary structure, q is the specific thermal flux, Q is the thermal flux, a and b are the width and height of the vapor-discharge channels, Nu is the Nusselt number, Pr is the Prandtl number, and K is Jacob number.

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