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# ON THE OPTIMIZATION OF HEAT AND MASS TRANSFER IN FINE-POROSITY CAPILLARY STRUCTURES OF CONTOUR HEAT PIPES

A. A. Belyaev, Yu. A. Buevich, and
V. M. Kiseev

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*The problems of optimizing the evaporators of contour heat pipes with respect to the thickness of the capillary structure barrier wall, geometry of the vapor generation zone, as well as the dimensions and disposition of the system of vapor-discharge channels are discussed.*

The optimal solution of the problem of organizing highly efficient transmission of heat over a considerable distance can be obtained by applying contour heat pipes (CHP). The scheme of vapor generation used in them makes it possible to improve substantially their thermodynamic efficiency with a considerable increase in the length of heat transfer [1]. However, further improvement of these heat pipes is greatly inhibited by insufficient knowledge of the processes of heat and mass transfer in fine-porosity capillary structures (FCS) used as wicks of the evaporators.

The heat and mass transfer in the evaporator of a contour heat pipe [2-4] is influenced by the thickness of the barrier wall of the FCS, the design and dimensions of the vapor generation zone, the effective thermal conductivity, and the permeability of FCS [2-4]. It is precisely these factors that determine the magnitude of both the removed heat flux and the heat flux penetrating through the barrier wall of the FCS into the compensating cavity of the CHP.

Since the effects of the two latter factors are interrelated and their separate investigation is very difficult, here we shall limit our discussion to the study of the effect of only the first two factors on heat and mass transfer through the FCS. Below, we present the results of experiments obtained on a specially designed cell the schematic diagram of which is given in Fig. 1, as well as the discussion of these results in relation to the well-known computational methods used for optimizing the FCS with respect to different parameters.

Apart from maximum approximation of the conditions of heat and mass transfer to the operational conditions of an actual CHP, the design of the cell allowed for the fast replacement of wicks, the reproducibility of their mounting, and the prevention of vapor leakage through imperfect contacts of the FCS with the evaporator body. The cell is in the form of a closed loop consisting of evaporator 1 and condenser 2 connected by vapor pipe 3 and condensate pipe 4. Capillary structure 5 with vapor discharge channels on it is located in the zone of heat supply on copper heater 6. A portion of the evaporator free of the FCS forms compensating cavity 7 (CC). To improve the contact of the FCS with the heater, the surface of the latter was carefully polished.

To prevent the passage of vapor from evaporator 1 into compensating cavity 7, a featheredge seal was applied which is widely used in vacuum technology. The tooth of this seal pressed into the capillary structure prevented the leakage of vapor into the CC while not distorting substantially the transport and structural characteristics of the FCS. Moreover, the pressing of the featheredge into the FCS simultaneously provided its good compression to the heater.

The entire system was hermetically sealed. Through a vacuum valve, the compensating cavity was connected to a tank with a heat carrier. This connection made it possible to select the optimal charging of the

Ural State University, Scientific-Research Institute of Physics and Applied Mathematics, Ekaterinburg.
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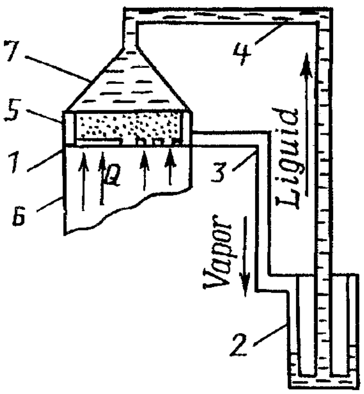


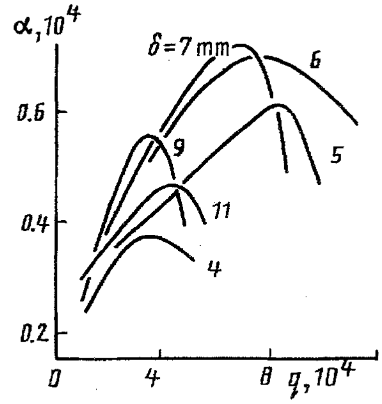


Fig. 1. Schematic diagram of the test cell.

Fig. 2. Dependence of the heat transfer coefficient on the transmitted heat power for different thicknesses of the barrier wall of fine-porosity capillary structure. $\alpha$, $W/(m^2 \cdot K)$; $q$, $W/m^2$.

measuring cell with the heat carrier, as well as to carry out experiments both with a constant pressure in the CC and under conditions of complete independence of the facility.

The experimental technique consisted in recording the dependence of the heat transfer coefficient on the magnitude of the removed thermal power for an evaporator with geometrically different samples of capillary structure.

The dependence of the heat transfer coefficient on the the FCS barrier wall thickness is a very complex problem [2-5]. The simplest method of its optimization showing promise for engineering calculations is described in [3] where it is suggested to determine the optimal version by the maximum of the Peclet number

$$\mathrm{Pe} = \frac{w_0 \delta}{a_{\mathrm{liq}}},$$

corresponding to the prescribed heat carrier flow. This makes it possible to eliminate the complexities involved in the calculation of the pressure and temperature balances [2, 4].

The Peclet number can be transformed in the following way [6]:

$$\mathrm{Pe} = \frac{w_0 \delta}{a_{\mathrm{liq}}} = \frac{w_0 \delta C_p \rho}{\lambda_{\mathrm{liq}}} = \frac{\rho C_p w_0 (T - T_0)}{\lambda_{\mathrm{liq}}/\delta \, (T - T_0)}. \tag{1}$$

Thus, this number characterizes the ratio of heat transferred by liquid flow through the capillary structure to the opposite heat flux conditioned by the thermal conductivity of the liquid.

For the above-described experimental cell the Peclet number can be written in the form

$$\mathrm{Pe} = \mathrm{Pr}\,\mathrm{Re} = \frac{C_p \eta_{\mathrm{liq}}}{\lambda_{\mathrm{liq}}} \frac{G}{2\pi \eta_{\mathrm{liq}}} \delta \frac{(\pi R^2 - 2\pi R \delta)}{\pi R^2}. \tag{2}$$

Solving the equation $d\mathrm{Pe}/d\delta = 0$ and determining the sign of the second derivative at this point, we obtain that the function $\mathrm{Pe} = f(\delta)$ has the sole maximum at the point $\delta = R/4$. For the samples investigated in this work the optimum, thickness amounts to 4 mm.

To determine experimentally the effect of the barrier wall thickness on the operation of the CHP, several sets of experiments were performed. In the first test series, nine samples of nickel capillary structure were investigated; these were made by hydraulic pressing at a pressure of $10^7$ $N/m^2$ and subsequent sintering at a temperature of 600°C. The diameter of the sample was equal to 30 mm, and the barrier wall thickness varied stepwise from 1 to 11 mm. The vapor discharge channels were produced mechanically on the surface of the FCS contacting the heater. They were made in the form of a system of equally spaced eight radial and four concentric channels of rectangular cross section (whose thickness and depth amounted to 1 mm). In the test series the heat carrier was acetone.

The resulting experimental data in the form of the dependence of the heat transfer coefficients on the heat flux transmitted are presented in Fig. 2 for different thicknesses of the barrier wall. The temperature at the place of contact of the FCS with the heat carrier in the compensating cavity was maintained constant and equal to 40°C due to the change in the mode of condenser cooling and, consequently, to the change in the heat carrier temperature at the inlet of the compensating cavity. The choice of precisely these relations to characterize the processes of vapor generation permits one to introduce the notion of the "region of the optimal operation" of an evaporator, i.e., of the region with a changing power in which the heat transfer coefficient has the maximum value.

As is seen from Fig. 2, the region of optimal operation of the evaporator is displaced to the side of large powers with a decrease in the barrier wall thickness to the value $\delta = 5.5$ mm. With further decrease in the barrier wall thickness of the FCS, a gradual fall is observed in both the maximum heat transfer coefficient and the corresponding heat power removed. Thus, for the FCS used in the present series of experiments the thickness $\delta = 5.5$ mm is optimal.

For geometrically similar titanium fine-porosity capillary structures made at the same packing pressure of $10^7$ N/m$^2$ an optimal thickness of $\delta = 5$ mm was obtained when the FCS was filled with acetone. Just as in the series of experiments in Fig. 2, there is a displacement of the evaporator optimal operation region to the side of smaller heat flux densities. Moreover, a decrease in the maximum heat flux removed is observed. These experiments are described in more detail in [5, 7].

The difference between the results of calculation by the technique suggested in [3] and those obtained experimentally is associated with the not very accurate determination of the correction for the problem geometry. In particular, it is not clear whether or not one should identify the evaporation area with that of heat transfer or of vapor removing channels [2], or whether one should calculate it by a more rigorous method [4] that greatly complicates the solution. Moreover, in the estimates suggested in [3], for the purpose of simplification neither the parameters of the FCS nor the parameters of the other structural elements of the evaporator were taken into account. The effect of these parameters requires additional study.

In order to determine the effect of the heat carrier used in the experiment on the optimal thickness of the barrier wall of the FCS, additional experiments were undertaken for obtaining the optimal thickness of the wall for an FCS made of PNE-0.45 powder with a porosity of 73.5% with three heat carriers: acetone, ethyl alcohol, and distilled water. The experiments showed that a thickness of $\delta = 5$ mm was optimal for the more volatile acetone and alcohol, and $\delta = 6$ mm for distilled water.

As regard application, it is also of interest to find the optimal dimensions and number of vapor discharge channels. The method of finding the optimal dimensions and number of such channels by minimizing the function corresponding to the sum of hydraulic losses during the motion of liquid and vapor in the zone of vapor generation was suggested in [2] where the results of [8] were used.

For solving the problem, a number of assumptions are made. We assume that

1) the heat flux is supplied over the entire surface of heat supply;

2) the maximum capillary pressure exceeds the sum of pressures for the most intense portion, i.e., over the portion of heat supply; this means that the liquid-vapor interface is located on the surface of the channels and does not extend into the capillary structure;

3) evaporation is possible only from the side walls of vapor discharge channels;

4) the intensity of evaporation or the density of mass flux through the evaporating surface varies continuously along the evaporating faces;

5) there is a near-wall boundary layer whose thickness is $\delta_T = \lambda_{eff}/\alpha$ determined by the heat transfer coefficient $\alpha = q/(T_1 - T_v)$ provided that $\Delta T = 1$°C.

Based on the above assumptions, for a plane construction considered in the present work the vapor losses for rectangular channels with sides $a$ and $b$, assuming that the flow rate along the channels varies monotonically from zero to a maximum value, are

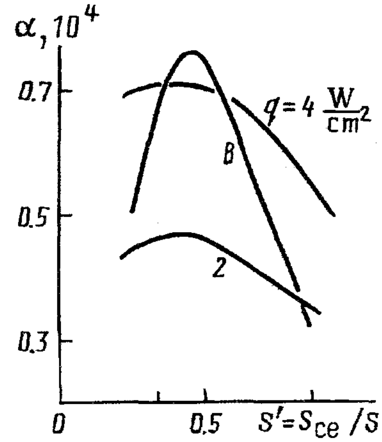


Fig. 3. Dependences of the heat transfer coefficient on the specific area of vapor discharge channels $S'$ for different heat fluxes transmitted.

$$\Delta P_{\mathrm{v}} = \frac{AG v_{\mathrm{v}} L_{\mathrm{se}}}{4D_{\mathrm{eq}}\, abn}. \tag{3}$$

The values of the resistance coefficient $A$ and of the equivalent diameter $D_{\mathrm{eq}}$ for different ratios between $a$ and $b$ are given in [7].

The mean pressure losses during the motion of liquid in the zone of vapor generation for a plane evaporator are

$$\Delta P_{\mathrm{liq}} = \frac{\nu_{\mathrm{liq}}}{K} \frac{(b/\delta_{\mathrm{T}} + 1)\, G}{(b/\delta_{\mathrm{T}} + 2)\, 2L_{\mathrm{se}}\, nb} \left[ b + \frac{1}{2} \left( \frac{2\pi R}{n} + a \right) \right] \frac{\pi R}{\pi R - 2na}. \tag{4}$$

To obtain the optimal vapor discharge channels, we must minimize the function

$$F = F(a\,,\ b\,,\ n) = \Delta P_{\mathrm{v}} + \Delta P_{\mathrm{liq}}. \tag{5}$$

Determining the minimum of this function $F$, we obtain that for rectangular channels the ratio $b/a = 10$ is optimal. In this case the specific area of the vapor discharge channels related to the entire area of heat removal should lie within the range 0.55-0.60.

These results coincide with the data obtained in [2] for a cylindrical geometry in which, for an evaporator with diameter $d = 30$ mm and heat supply length $L_{\mathrm{ce}} = 80$ mm, the following parameters of the optimal evaporator were obtained:

a) for distilled water as the heat carrier $b = 5$ mm, $a = 0.5$ mm, $n = 111$, $S_{\mathrm{ce}}/S = 0.59$;

b) for acetone as the heat carrier $b = 2$ mm, $a = 0.2$ mm, $n = 261$, $S_{\mathrm{ce}}/S = 0.55$.

In order to verify the above-described technique of calculation, experiments were carried out on the same cell (see Fig. 1). For this purpose, the operating characteristics of the measuring cell were recorded for capillary structure samples in which the zone of vapor generation was organized differently. All of the experimental data were represented in the form of the curves $\alpha = \alpha(S_{\mathrm{ce}}/S)$, since these relations make it possible to fully study the processes occurring in the evaporator.

The first experimental cycle was devoted to the determination of the optimal number of vapor discharge channels. Experiments were conducted on capillary structures made from PNE powder with mean diameter of particles 0.9 $\mu$m at a hydraulic pressure of pressing $10^7$ N/m$^2$. The samples were made by mechanical treatment of capillary structures of height $h = 6$ mm and diameter $d = 30$ mm. This diameter was selected to reduce to a minimum pressure losses for the motion of vapor through vapor discharge channels with simultaneous simulation of the system of vapor collection. The vapor discharge channels were cut in the sample in a strictly rectangular shape and had width and depth of 1 mm.

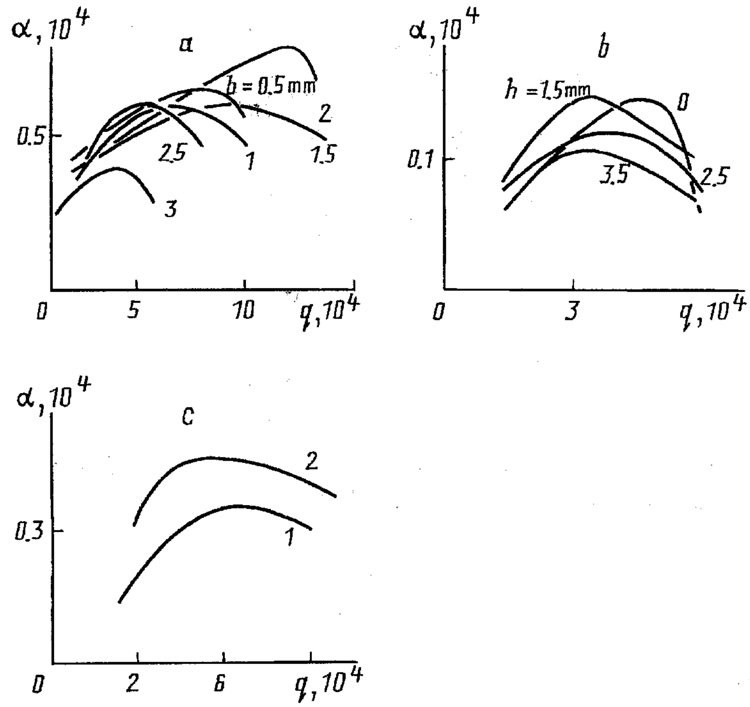


Fig. 4. Dependence of the heat transfer coefficient on the tramsitted heat flux for different schemes of vapor removal in the FCS: a) comparison of experimental data for evaporators with the same width ($a$ = 1 mm) and varying depth of vapor discharge channels (acetone, $S'$ = 0.45); b) comparison of experimental data for evaporators whose vapor discharge channels are located in the bulk of the FCS at a different distance from the heat supplying wall (acetone, $S'$ = 0.12); c) comparison of the efficiency of location of vapor discharge channels in the FCS (1) and in the heat supplying wall (2) (acetone, $S'$ = 0.42).

In all the experiments acetone was used as the heat carrier. The vapor discharge channels were made in the sample in such a way that the specific area occupied by these channels was the same over the entire surface of heat supply of the capillary structure. The relations obtained in the experiments are presented in Fig. 3. From the figure it is seen that almost 40% of the optimal operation zone of the evaporator corresponds to the specific area; but as the specific area increases, the heat transfer coefficient falls sharply. In the case considered, among the samples investigated the sample with four concentric vapor collecting channels spaced 2 mm from each other and four radial vapor discharge channels turned out to be thermodynamically optimal.

Of course, the conclusions concerning the optimal organization of vapor discharge channels relate only to the given dimensions of channels and zones of vapor generation. However, by minimizing the function $F(a, b, n)$ simultaneously with respect to three parameters we could obtain, just as in work [2], a large number (~100) of very fine channels. It is very different to fabricate samples with such a pattern of channels for vapor discharge. On the other hand, the specific area determined experimentally, which corresponds to the optimal number of channels, roughly coincides with that calculated by the method described in [2] for the given dimensions of the channels (at $a = b = 1$ mm we have $(S_{ce}/S)_{opt}$ = 0.40-0.45). The authors of [2] also carried out the experimental optimization of the ratio between the depth of the vapor discharge channel to its width, as well as investigated the effect of the location of the channels on heat and mass transfer during vapor generation in the FCS.

It is pointed out in [2] that in vapor discharge channels of optimal dimension with width-to-depth ratio $b/a = 10$ the appearance of a competing process is possible process, i.e., of recondensation at the depth of the channel. This increases the warming-up of the barrier wall. Moreover, when optimization is carried out with respect to the function $F(a, b, n)$, the process of decrease in the barrier wall thickness with increase in the depth of channels is

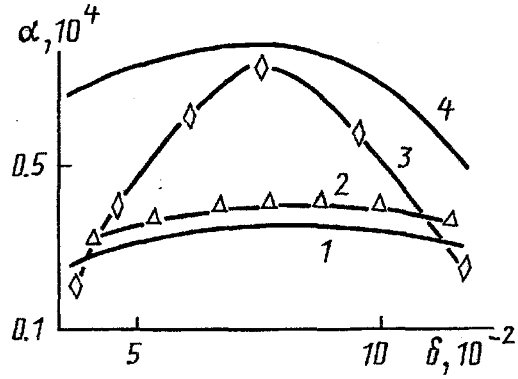


Fig. 5. Comparison of calculation data using the method of [4] with experiments on the determination of the effect of barrier wall thickness on heat transfer for different transmitted heat fluxes; calculated (1, 3) and experimental (2, 4) relations at $q = 3\cdot10^4$ W/m$^2$ (1, 2) and $q = 6\cdot10^4$ W/m$^2$ (3, 4). $\delta$, m.

taken into account with simultaneous preservation of the total thickness of the FCS. Therefore, according to [2], the channels with $b/a$ varying from 2 to 5 are most efficient.

To check this conclusion, a special series of experiments was undertaken. Since simultaneous optimization with respect to three parameters (as done in [2]) is impossible, we considered samples with the obtained optimal pattern of vapor discharge channels and varying dimensions.

In the first series of experiments performed with the same capillary structures we used samples with a constant width of vapor discharge channels of 1 mm and with the depth varying each 0.5 mm. The thickness of the barrier wall was held constant and equal to 5 mm. The results of these experiments are presented in Fig. 4a. As is seen from this figure, the operation of an evaporator with capillary structure having channels with $b = 2$ mm is most effective. When the ratio increases above $b/a = 2$, much worse results were obtained. Moreover, we carried out investigations on samples with vapor discharge channels of depth $b = 1$ mm and varying width. The capillary structure for these samples was the same as in the previous case. The results of these experiments showed that a sample with channel width $a = 0.5$ mm gives the best results, i.e., the ratio $b/a = 2$ is most efficient for these samples too.

In our opinion, more interesting is the question of the optimal place for locating a vapor removal system. During experimental verification, the results of which are presented in Fig. 4b, we investigated samples with four prefabricated radial vapor discharge channels which were drilled in the capillary structure at different distances from the heat supplying wall. As expected, with approach of the system of vapor removal to the heat source the heat transfer coefficient grows significantly at the expense of a decrease in the thickness of the capillary structure layer which in the given case plays the function of heat supply to the vapor generation region. Since the thermal conductivity of the given capillary structure is not high ($\lambda_{eff}$ = 2-4 W/(m·K)), its thinning leads to a substantial increase in the efficiency of vapor generation.

In this case it is of interest to compare the efficiency of the zone of vapor removal within the thickness of the capillary structure and the zone near the plane of contact of the FCS with the heat supplying wall. Vapor removal at the boundary of the capillary structure is technologically simpler, but the location of this zone within the thickness of the FCS permits one to intensify the vapor generation by increasing the area of vapor escape from evaporating menisci in the pores to vapor discharge channels. However, the placing of an FCS layer between the vapor removal system, where the evaporation of heat carrier occurs, and the heat-supplying wall may lower the efficiency of operation due to the low thermal conductivity of the capillary structure. In fact, as is seen from Fig. 4b, location of the vapor discharge channels at the FCS surface with preservation of their quantity and equivalent diameter leads to the displacement of the heat transfer maximum to the zone of high heat fluxes.

Along with the above-considered problems, one other problem arises, i.e., in what way one should deploy vapor discharge channels on the plane of contact of the capillary structure with the heat supplying wall. When they are cut in the thickness of the FCS, then, during vapor generation, vapor enters a channel through all the three sides of the channel, which greatly increases the area of vapor generation. But when the vapor discharge channels are located in the heat-supplying wall (in practice, in the body of the heat pipe evaporator), escape of vapor into the channel is possible only through one side of the vapor discharge channel. On the other hand, it is necessary to take into account also the fact that, when the vapor discharge channel is located within a capillary structure, shift of the evaporation front into its bulk occurs, and this leads to the thickening of the FCS and to an increase in hydraulic losses for the transport of heat carrier. Moreover, in this case the distance between the heat-supplying wall and the evaporating menisci increases, and this also decreases the efficiency of operation of the vapor generation zone. What has been said above shows that vapor generation with vapor removal within the heat supplying wall is more optimal thermodynamically. This is also confirmed experimentally (see Fig. 4c).

The obtained data on the optimization of the contour heat pipe evaporator are of interest as control data for checking the efficiency of the existing techniques of calculation of a CHP [2-4]. For lack of space a detailed description of these techniques, and of the block diagram of the corresponding computer calculations is not given here. Qualitative agreement between the calculated and experimentally derived conclusions on the optimization of vapor generation in a CHP is observed in practice only in calculations using the technique of [4]. (As an example, Fig. 5 gives a comparison of such calculations with the data of experiments on the determination of the effect of the barrier wall thickness on heat transfer.) This leads to the problem of improving the techniques of theoretical optimization of the CHP which is naturally associated with the refinement of the existing models of vapor generation. The authors hope to return to this problem in one of their subsequent studies.

## NOTATION

$A$, coefficient; $a_{\text{liq}}$ thermal diffusivity of liquid; $a$, $b$, width and depth of vapor discharge channels; $\alpha$, heat transfer coefficient; $C_p$, heat capacity coefficient; $d$, diameter; $D_{\text{eq}}$, equivalent diameter; $\delta$, thickness of barrier wall of the capillary structure; $\delta_{\text{T}}$, boundary layer thickness; $G$, mass flow rate of the heat carrier; $K$, permeability factor; $L_{\text{se}}$, length of the vapor discharge channels; $L_{\text{se}}$, length of the portion of heat supply for the cylindrical evaporator; $\lambda_{\text{liq}}$, thermal conductivity of the liquid; $\lambda_{\text{eff}}$, effective thermal conductivity of the capillary structure; $n$, number of vapor discharge channels; $P$, pressure; $R$, radius; $\rho$, density; $S$, area of the section of heat supply; $S_{\text{ce}}$, area of the section of heat supply occupied by vapor discharge channels; T, temperature; $T_0$, temperature of heat carrier entering evaporator; $T_{\text{sin}}$, sintering tempearture; $T_{\text{v}}$, vapor temperature; $w_0$, liquid velocity; Pe, Peclet number; Re, Reynolds number; Pr, Prandtl number.

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