INVESTIGATION OF THE PROCESS OF LIQUID EVAPORATION FROM A POROUS METAL UNDER VACUUM CONDITIONS

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The method and results of an investigation of the evaporation of
water into a vacuum are described. The steady-state regime of evap-
oration in a porous metal is examined. The dependence of the evap-
oration rate on the parameters of the porous metal and the heat flux
density has been experimentally determined. The temperature dis-
tribution in the porous plate is determined theoretically.

Porous evaporative cooling can often be used to good effect [1–4]. Liquid coolant is continuously supplied to the evaporation zone, where, as a result of phase transitions, heat is absorbed.

The intensity of porous cooling increases considerably if the porous element of the system is located in a gas medium at reduced pressure. Under certain conditions the liquid (water) forced through the element at a certain excess pressure converts to ice in the evaporation zone and heat is removed from the system by subliming into the low-pressure region.

To investigate this process we studied the evaporation of distilled water into a vacuum. The water evaporated from porous metal plates fabricated by a powder metallurgy technique. Our object was to determine the dependence of the evaporation rate on the characteristics of the porous metal and the thermal head. The plate served as a membrane separating the water from the low-pressure gas, so that the water evaporated not from a free surface but in the pores of the material. Consequently, the pressure in the evaporation zone was nonzero owing to the sharp increase in the specific volume of the evaporating substance and the hydrodynamic resistance of the pore channels. This pressure is a function of the temperature in the evaporation zone, the diameter and length of the pores, and the thermophysical characteristics of the evaporating liquid.

The porous metal plate also transferred heat to the evaporation zone. We note that for metals, with rather high thermal conductivity (on the order of tens of W/m · deg), the thermal resistance of the skeletal material plays a secondary role as compared with the resistance at the contacts between individual grains. The effective thermal conductivity of porous titanium is 2–5 W/m · deg; that of porous aluminum is 11–25 W/m · deg.

The vacuum apparatus is shown schematically in Fig. 1. Its principal elements are: vacuum chamber 1, balance 2, pan 3 with electric heater and test specimen reservoir 4, and cylinder 5.

Pressure in the vacuum chamber was maintained by means of a vacuum pump and a needle valve. The pressure in the vacuum chamber, temperature, and evaporation rate were measured with a VT-2 vacuum gauge, copper–constantan thermocouples with a wire diameter of 0.1 mm, and a VTK–500 balance graduated in 0.1 g, respectively.
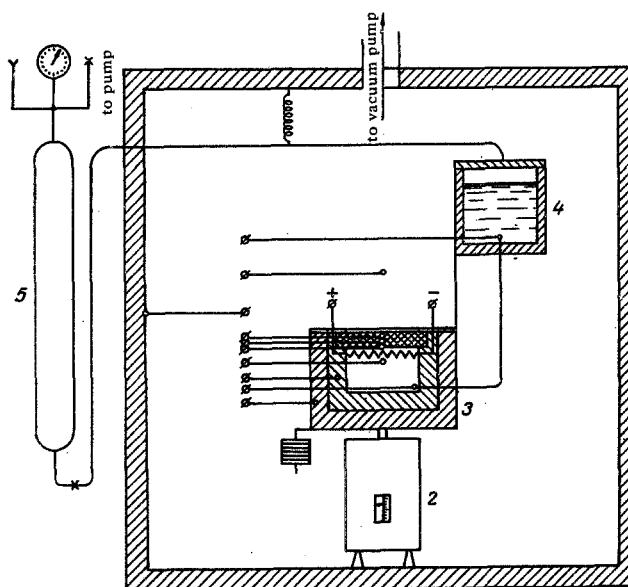

Fig. 1. Schematic representation of experimental apparatus.

The surface of the porous plate was observed visually through an inspection port, during the runs.

We studied specimens of porous titanium and aluminum in the form of disks 90 mm in diameter and 3 mm thick. The disk was clamped to plexiglas pan 3 (see Fig. 1). The part of the pan beneath the plate contained distilled water for evaporation. Also in the pan, 2–3 mm below the bottom of the plate, was a flat electric heater. During the experiments the water evaporated through the plate; the loss was compensated for by admitting water from reservoir 4.

To reduce heat losses the outside of the pan was thermally insulated with plastic foam 25 mm thick. The pan, together with the test plate and vessel 4, was mounted on a balance. To reduce the weighing error introduced by the thermocouple wires, the rubber tubes, and the heater leads, the last two were supported on compensating springs. The balance readings were checked before each experiment. The absolute error in weight did not exceed 0.1–0.2 g per 50 g of liquid.

We measured the following quantities: pressure in the vacuum chamber, water pressure in the system (pan–reservoir), temperature of water entering the

pan, temperature of the sample (interior of specimen, outer and inner surface), temperature of chamber walls, ambient medium, and thermal insulation. The
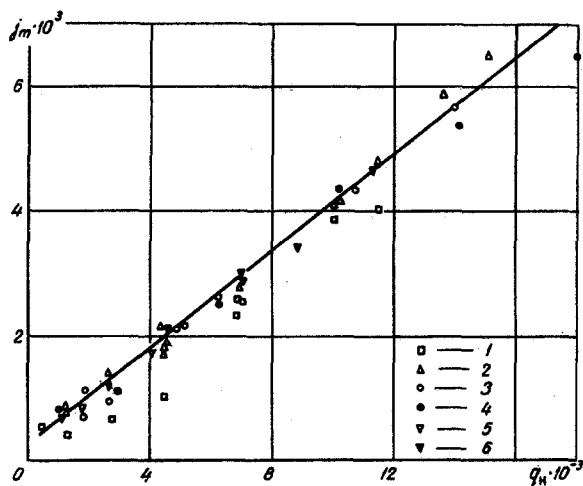

Fig. 2. Evaporation rate as a function of heat-flux density for porous plates: 1–4) titanium plates, P = 41, 23, 30, and 20%, respectively; 5 and 6) aluminum plates, P = 22 and 16%.

location of the temperature sensors is indicated schematically in Fig. 1.

We worked in the range of vacuum-chamber pressures from 5 to 400 N/m² and water pressures from $5 \cdot 10^3$ to $1 \cdot 10^5$ N/m². The specific heat flow from the electric heater varied from 400 to 18 000 W/m². These fluxes corresponded to a water temperature beneath the test specimen of 274–293° K and to evaporation rates from $4 \cdot 10^{-4}$ to $7 \cdot 10^{-3}$ kg/m² · sec.

These evaporation rates are several orders smaller than the rate of water suction through the plate corresponding to the above-mentioned water pressure in the system. Hence we conclude that plugs of ice formed in the pores, preventing the water from flowing out, i.e., the water entered the gaseous state as a result of a double phase transition (liquid-ice, ice-vapor). This effect took place at moderate heat fluxes. At large fluxes water was ejected into the vacuum chamber, visible as a snowlike deposit on the surface of the plate. The balance readings also indicated a sharp decrease in weight at the instant of ejection.

The critical heat flux at which ejection was first observed decreased with an increase in pore diameter and an increase in the water pressure in the pan.

The idea of a double phase transition is consistent with the measured values of the temperature at the outer surface of the plate, which during evaporation lay in the range 272.5–273° K. This is not an exact value for the temperature in the phase transition region, since the temperature drop over the thickness of the thermocouple junction was on the order of several tenths of a degree. However, we can assume that the true temperature in the phase transition zone is several tenths of a degree less than the value obtained experimentally, i.e., less than 273° K.

The relations obtained from the experimental data are presented in Figs. 2–4. All the data refer to

steady-state evaporation in the pores of the material.

In Fig. 2 the evaporation rate, expressed in kilograms per square meter per second, is shown as a function of the specific heat flow from the heater, expressed in watts per square meter. The experimental points corresponding to plates of different material with various porosities and fractional compositions lie in a narrow region about the straight line

$$j_m = kq_{\text{н}} + b, \qquad (1)$$

where k = $0.4 \cdot 10^{-6}$ kg/W · sec, and b = $0.2 \cdot 10^{-3}$ kg/ /m² · sec.

The numerical value of b/k determines the order of magnitude of the quantity of heat entering the evaporation zone from the chamber, and not from the heater. An estimate of the external heat flow shows that at chamber-wall temperatures on the order of 293° K it does not exceed 200–300 W/m².

Data on the temperature of the plate surface in contact with the liquid are presented in Fig. 3 for various heat-flux densities. The experimental points for each plate are well approximated with straight lines.

As pointed out above, the heat flow into the evaporation zone from the external medium did not exceed 200–300 W/m²; accordingly, at heat fluxes above 2000–3000 W/m² the external flux can be neglected flux can be neglected and we can write

$$T_1 = 273 + nq_{\text{н}}. \qquad (2)$$

The proportionality factor n increases for plates made of the same metal with an increase in porosity and depends on the skeletal material (it is greater for titanium than for aluminum) (Fig. 3). Since under all conditions the temperature in the evaporation zone is close to 273° K, for sufficiently high T_1 (several de-
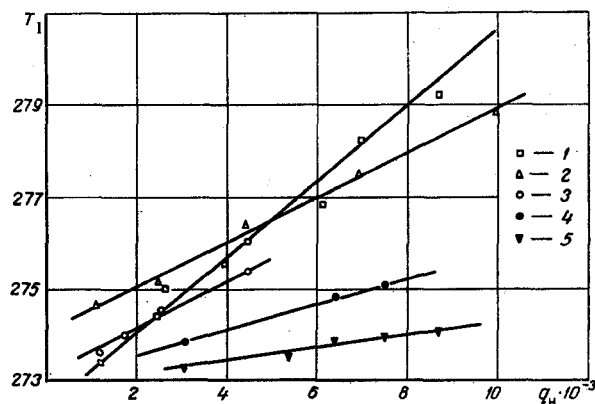

Fig. 3. Temperature of liquid beneath porous plate as a function of heat-flux density: 1–3) titanium plates, P = 41, 27, and 23%, respectively; 4 and 5) aluminum plates, P = 22 and 16%.

grees higher than the temperature in the evaporation zone) we can write the difference between the tem-

perature of the plate surface in contact with the liquid and the temperature in the evaporation zone as

$$\Delta T = T_1 - 273. \qquad (3)$$

Thus Eq. (2) becomes

$$q_{\text{H}} = k' \Delta T,$$

where k' = 1/n. This is analogous to the formula for steady-state heat transfer in a solid

$$q = -\frac{\lambda_{\text{eff}}}{\delta} \Delta T. \qquad (4)$$

We also determined the thermal conductivity of the porous plates. For titanium plates with 41% porosity, λ_{eff} = 3 W/m · deg, while λ_{eff} = 5.5 W/m · deg for titanium plates with 20% porosity, and 11 and 25 W/m· · deg for aluminum plates with 22 and 16% porosity, respectively. From these data and the slope of the straight lines in Fig. 3, we determined the value of δ. With a sufficient degree of accuracy, δ is equal to the thickness of the plate.

It follows from Fig. 3 that evaporation of the liquid takes place near the outer surface of the plate, and that the thermal conductivity depends on the porosity, which decreases as the porosity increases.

In Fig. 4 the evaporation rate is given as a function of the mean pore diameter. Clearly, the intensity of evaporation at constant temperature for the liquid beneath the plate, i.e., for an almost constant temperature gradient, decreases with an increase in pore diameter. This is evidently associated with the particular conditions of heat supply to the evaporation zone. Since the thermal conductivity of the skeletal material is greater than the thermal conductivity of the water, the heat is supplied to the evaporation zone primarily through the skeletal material. Hence, as the pore diameter decreases by a factor of i, the heat flow to the evaporation zone, referred to unit-pore cross section, increases by a factor of i. In other words, in these experiments the effective thermal conductivity λ_{eff} in Eq. (4) depends not only on the porosity but also on the pore diameter.

In investigating the porous cooling process, we attempted to evaluate the effect of liquid motion on the temperature distribution in the porous plate by solving the corresponding heat-transfer problem.
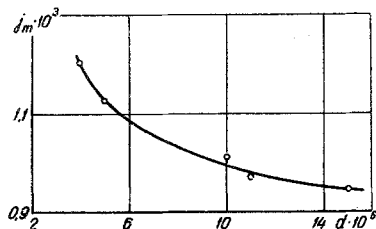
Fig. 4. Evaporation rate as a function of mean pore diameter.

In deriving the equations, we made the following assumptions

1) the pores are uniformly distributed over the volume of the porous plate;

2) the physical parameters of the liquid do not depend on temperature;

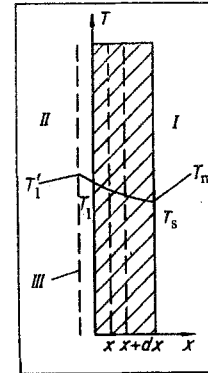3) the temperature field in the plate is one-dimensional, since the dimensions of the plate in the y- and

Fig. 5. Temperature distribution in porous plate: I) gas; II) liquid; III) heat source.

z-directions are much greater than the plate thickness, and the temperature of the solid skeleton and the cooling liquid are equal at any point in the porous material [5]; and

4) liquid at a rate of j_m kg/m² · sec passes through the plate and is evaporated at its surface.

Then the differential equations of heat transfer and the boundary conditions δ ≥ x ≥ 0 for the plate (Fig. 5) have the form

$$\frac{d^2 T}{dx^2} - \xi_{\text{T}} \frac{dT}{dx} = 0, \qquad (5)$$

$$x = \delta \quad \alpha_c (T_{\text{m}} - T_s) - \lambda_{\text{eff}} \frac{dT}{dx} = j_m r,$$

$$x = 0 \quad T = T_1, \qquad (6)$$

where $\xi_{\text{T}} = j_m c_{\text{pl}} / \lambda_{\text{eff}}$; λ_{eff} is the effective thermal conductivity by which we take into account the porosity of the material and the thermal conductivity of the liquid saturating the porous material. Boundary condition (6) assumes that all the heat transported through the plate by heat conduction and the heat transmitted by convection from the ambient medium is expended on evaporation. Moreover, evaporation, as established experimentally, proceeds from the surface, i.e., at x = δ.

The particular solution of Eq. (5) satisfying boundary conditions (6) has the form

$$T = T_1 + \frac{j_m r - \alpha_c (T_{\text{m}} - T_1)}{\alpha_c - (\alpha_c + j_m c_{p1}) \exp(\xi_{\text{T}} \delta)} \times$$

$$\times [\exp(\xi_{\text{T}} x) - 1]. \qquad (7)$$

In the same way we construct the equation for the region $-\infty \leq x \leq 0$:

$$\frac{d^2 T'}{dx^2} - \xi_1 \frac{dT'}{dx} = 0, \qquad (8)$$

where $\xi_1 = j_m c_{\text{pl}} / \lambda_1$.

At the liquid-porous plate interface, as pointed out above, there is a heat source of density q_H. Then the boundary conditions for the region $-\infty \leq x \leq 0$ have the following form

$$x = -\infty \quad T' = T'_1,$$

$$x = 0 \quad -\lambda_1 \frac{dT'}{dx} + q_\text{и} = -\lambda_\text{eff} \frac{dT}{dx}. \qquad (9)$$

The particular solution of Eq. (8) satisfying boundary conditions (9) may be written as

$$T' = T'_1 + \left[\frac{q_\text{и}}{j_m c_{p1}} + \right.$$

$$\left. + \frac{j_m r - \alpha_c (T_m - T_1)}{\alpha_c - (\alpha_c + j_m c_{p1}) \exp(\xi_\text{т} \delta)} \right] \exp(\xi_1 x). \qquad (10)$$

Eliminating the temperature T_1 from Eqs. (7) and (10), after mathematical transformations we obtain the final solution of Eq. (5)

$$T = T'_1 + \frac{q_\text{и}}{j_m c_{p1}} -$$

$$- \frac{\alpha_c (T'_1 - T_m) + \frac{q_\text{и} \alpha_c}{j_m c_{p1}} + j_m r}{\alpha_c + j_m c_{p1}} \times$$

$$\times \exp[\xi_\text{т}(x - \delta)]. \qquad (11)$$

Equation (11) includes the liquid temperature T'_1 and the temperature of the medium, which can be measured with high accuracy.

Equation (11) is in good agreement with the temperature distribution obtained experimentally.

NOTATION

j_m is the evaporation rate, kg/m$^2 \cdot$ sec; r is the specific heat of phase transition; T'_1, T_m, and T_s are the temperatures of the liquid, the medium, and the evaporation surface, respectively; T_1 is the temperature of the plate surface in contact with the liquid; q_H is the heat-flux density due to the heater; λ_eff is the effective thermal conductivity of the capillary-porous plate; λ_1 and c_{p1} are the thermal conductivity and specific heat of the liquid, respectively; δ is the distance from the wet surface of the plate to the evaporation zone; d is the mean pore diameter; P is the porosity of plate, %.

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