BOILING HEAT TRANSFER AT REDUCED PRESSURES WITH WATER SUPPLIED TO THE HEAT-TRANSFER SURFACE THROUGH A CAPILLARY-POROUS BODY

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The boiling of water supplied to the heating surface through a capillary-porous body has been experimentally investigated at pressures from 6 to 310 mbar.

In certain modern heat-exchange equipment, water boils at low temperatures from 1 to 50° C. This temperature region corresponds to saturation pressures of from 6 to 80 mbar.

As has recently been shown [1–4, 8, 9], in this pressure range boiling is impeded owing to the increased size of the viable vapor nucleus. For example, in [1], total "degeneration" of nucleate boiling was observed even at the relative high pressure of 0.184 bar.

Clearly, the use of capillary-porous bodies to control liquid boiling considerably facilitates the creation of vaporization centers at the boundary with the heating surface and eliminates the above-mentioned difficulty. Moreover, a capillary-porous body can be used to facilitate phase separation under conditions of weightlessness.

We have carried out two series of experiments. The first series was conducted on apparatus No. 1 with the object of studying boiling when liquid is supplied to a heating surface located above the surface of the liquid. In the second series of experiments, conducted on apparatus No. 2, we studied boiling at a heating surface immersed in a porous body saturated with liquid.

The test specimen, an electrically heated brass rod whose polished end served as the boiling surface, was supported in a pressure chamber, where a constant pressure, measured with an MChR-3 manometer, was maintained (correct to 0.2 mbar). The diameter of the rod was 60 mm. Fifteen chromel-alumel thermocouples were imbedded at different depths in four sections along the height of the rod. The section nearest to the end had six thermocouples; the rest, three. The distance between sections was 10 mm, and the first section was 2.5 mm away from the end of the rod. This method of arranging the thermocouples made it possible to determine the surface temperature and the specific heat flow from the end of the rod easily and accurately. In view of the high rate of heat transfer from the end of the rod ($\alpha \approx 1000$ W/m$^2 \cdot$°C), in comparison with the heat transfer from its lateral surface ($\alpha \approx 5$–10 W/m$^2 \cdot$°C), it may be assumed that the problem to be solved is the one-dimensional problem of heat conduction for a rod, together with the determination of q from the temperature values in four sections, and the value of T_W by linear extrapolation.

We turned the heating surface downward; i.e., we investigated the worst case, that in which the separation of the vapor bubbles is obstructed. Beneath the heating surface we placed a vessel containing water, in which the level was constantly maintained at a distance of 20 mm from the heating surface. The capillary-porous body, a rod of technical polyvinylformal TPVF-1 60 mm in diameter, was mounted on a support in the vessel containing the water Polymer. TPVF-1 is a foamed alcohol polycondensation product, a porous

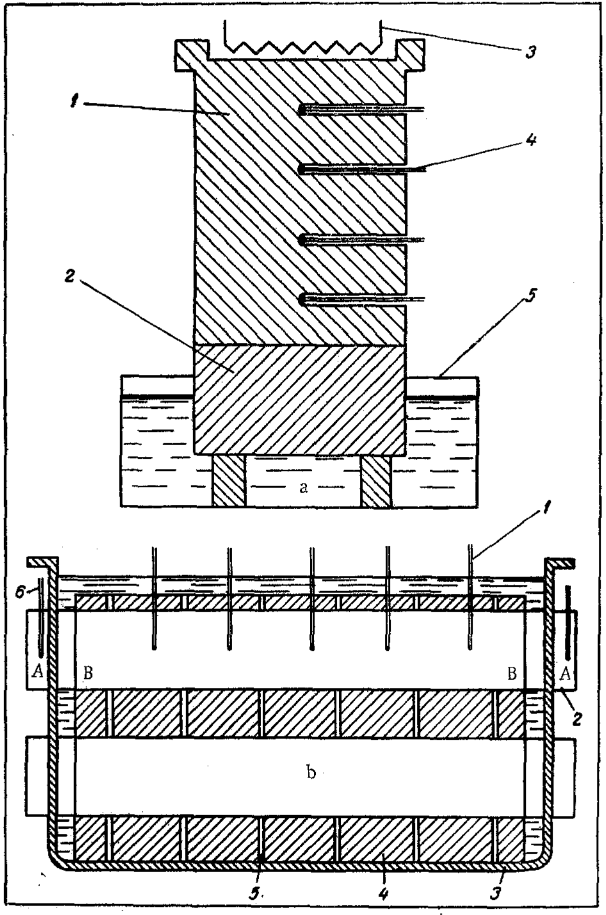

Fig. 1. Diagrams of the two experimental apparatuses. (a) Apparatus No. 1: 1) brass rod; 2) TPVF-1 rod; 3) electric heater; 4) thermocouples; 5) vessel containing water. (b) Apparatus No. 2: 1) thermocouples for measuring tube wall temperatures; 2) flat tube; 3) container; 4) capillary-porous body; 5) vapor channels; 6) thermocouples for measuring end losses.

material with a specific weight of 0.08–0.15 g/cm^3, porosity to 0.92–0.98, with open pores of various sizes—from fractions of a micron to a millimeter. Polymer TPVF-1 is a good absorber of moisture. In the dry state it is quite hard; when wetted with water it softens and swells.

The bottom of the rod rested in the water, while the top was in contact with the heating surface. Two mutually perpendicular slots measuring 3 × 3 mm and passing through the center of the rod were formed in the surface facing the heater to facilitate the removal of vapor.

Apparatus No. 1 is shown schematically in Fig. 1a.

The experiments were conducted in accordance with the steady heat flux method at a constant power supply to the electric heater. Three or four temperature measurements were made at constant pressure and heat flux. Runup from "cold" took 1.5–2 hr; transition from one flux to another required 30–40 min.

On this apparatus we conducted experiments at five different pressures: 310, 167, 107, 64, and 26.2 mbar. The minimum heat fluxes were selected so that the difference between the heating surface temperature and saturation temperature was not less than 1.5–2° C.

The graphs in Fig. 2 present in logarithmic coordinates the results of the investigations on this apparatus in the form $q = f(\Delta T)$.

The experiments to investigate boiling at a heating surface immersed in a porous body saturated with liquid were conducted on apparatus No. 2, which is shown schematically in Fig. 1b.

The investigations were carried out on an experimental heat exchanger-evaporator intended for cooling and drying gases. The heat exchanger consisted of 2 corridor bundles of flat aluminum-alloy tubes 1 mm thick, inside cross section 2 × 38 mm, length 210 mm, welded into a honeycomb with a 12-mm spacing. Normally, cooled fluid moves through the tubes. The space between the tubes was occupied by sheets of TPVF-1 7 mm thick, so that, on swelling, these sheets tightly pressed themselves against the flat surface of the tubes. To facilitate vapor removal, rectangular channels measuring 2 × 3 mm were formed in the part of the TPVF-1 sheets turned toward the surface of the tubes.

For the purpose of investigating boiling heat transfer, the heat exchanger was placed in a pressure chamber; measurements were made on one of the tubes into which a nichrome heater wound onto a mica plate was inserted.

To measure the surface temperature of the tube, we used five copper-constantan thermocouples, whose hot junctions were imbedded in the wall. The temperature was measured correct to 0.1° C. The pressure was measured correct to 0.2 mbar.

The ends of the experimental tube (sections AB in Fig. 1b), 20 mm long, were insulated from the water by a layer of heat-resistant crude rubber; losses through the latter were negligibly small. The heater inserted into the tube occupied exactly the section BB in contact with the water-saturated porous body. This section determined the active heat-transfer area from which the heat-transfer coefficient was calculated.

The heat flux was determined from the power consumption measured with class 0.1 and 0.2 instruments with allowance for the end losses estimated from the readings of thermocouple 6 (Fig. 1b); the error did not exceed 3%. The measurements were made in the steady-state boiling regime.

On this apparatus we investigated boiling heat transfer at pressures above the surface of the water of 6.5, 13, 20, and 33 mbar and heat fluxes from $0.5 \cdot 10^3$ W/m^2 to $5 \cdot 10^4$ W/m^2 for various vapor-channel densities.

The results are presented in Fig. 3 in the form of $\alpha = f(q)$ curves.

In analyzing the data presented in Fig. 3, the saturation temperature was taken in accordance with the pres-

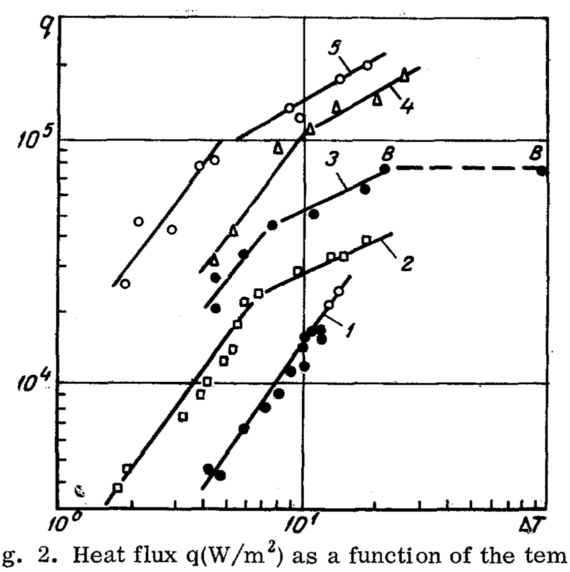

Fig. 2. Heat flux q(W/m^2) as a function of the temperature head ΔT (°C) according to data obtained on apparatus No. 1: 1) p = 26.2 mb; 2) 64.0; 3) 107; 4) 167; 5) 310.

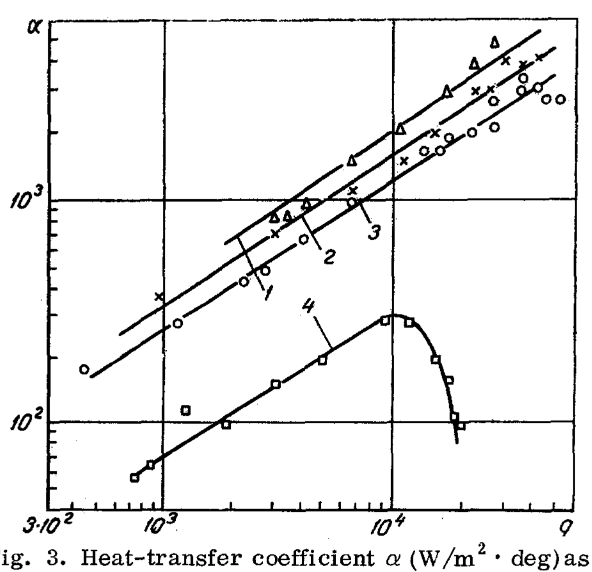

Fig. 3. Heat-transfer coefficient α (W/m^2 · deg) as a function of the heat flux q (W/m^2) from data obtained on apparatus No. 2: 1) p = 36.2 mb; 2) 23; 3) 9.6; 4) 9.6 (1–3—at s = 0.14; 4—at s = 0).

sure at the wall temperature measuring point (on the center line of the tube); i. e., the hydrostatic head of the liquid column was taken into account. Despite the fact that the liquid surface was maintained only 5–8 mm above the top of the heated tube, the correction was important, since at the low pressures measured, which were very close to the triple point of water, the $T_S = f(p_S)$ curve is very steep.

In addition to the experiments performed on apparatuses Nos. 1 and 2, we tested the heat exchanger-evaporator in the air and water cooling regimes to make a comparative analysis of the heat-transfer coefficients and determine the entrainment of liquid at various orientations of the vapor channels and the outlet header with respect to the earth's gravitational field.

Discussion of the results. In the experiments performed on apparatus No. 1 with the porous body incompletely saturated, we did not observe any visible entrainment of liquid with the vapor at any value of the heat flux.

In the experiments on apparatus No. 2, entrainment of liquid was observed, and at certain heat fluxes droplets of water were "shot" out of the channels together with the vapor. The entrainment diminished when the vapor velocity was lowered by increasing the area of the vapor channels or decreasing the heat flux. The studies of entrainment on apparatuses Nos. 1 and 2 were qualitative in character and purely visual.

In the experiments with the heat exchanger-evaporator, after the optimum frequency and arrangement of the vapor channels had been selected, no entrainment of moisture, determined from the heat and mass balances, was observed within the limits of experimental error (5%). In this case, the velocity of the vapor in the vapor channels was up to 30–50 m/sec, the boiling pressure 6–8 mbar, and the heat fluxes from $0.5 \cdot 10^3$ to $5 \cdot 10^3$ W/m^2. There was no entrainment at any orientation of the evaporator relative to the earth's gravitational field.

The heat-transfer coefficient for water boiling at low pressures (6–30 mbar) is small—on the order of 10^2–10^3 W/m$^2 \cdot$°C. This pressure region is often employed in cooling applications, and, since the rate of heat transfer on the boiling side is comparable with the rate of heat transfer on the side of the cooled liquid and gas, the study of boiling at these pressures is especially important from the standpoint of minimizing the size and weight of the heat-exchange equipment.

In all the regimes tested on apparatus No. 2 (Fig. 3) up to $q = 5 \cdot 10^2$ W/m^2, boiling was clearly observable together with intense motion of the vapor bubbles, which at the moment of separation were on the order of 1–5 mm in size. The presence of developed boiling at low heat fluxes and pressures is attributable to the fact that the capillary-porous body creates a large number of additional vaporization centers, including both the pores in the material itself and very fine air bubbles not displaced from the pores by the water.

The relations presented in Fig. 3 confirm previous conclusions [5, 6] to the effect that the principal influence on the boiling heat transfer of a liquid transported by a capillary-porous body is exerted by the removal of vapor from the heating surface.

In particular, this is indicated by the decreasing slope of the $q = f(\Delta T)$ curves in the region of large heat fluxes in Fig. 2.

Whereas in the region of small heat fluxes the usual equation $q = C\Delta t^3$ is replaced by the relation $q = C_1\Delta T^{1,4}$ (C and C_1 are numerical constants), at large heat fluxes the exponent of ΔT is reduced to 0.5. This is because of the increased vapor content of the contact layer and because under these conditions there is no free-convection component.

This conclusion is confirmed by the results of the experiments on a submerged surface. At a vapor-channel density s = 0.14 (i. e., the ratio of the area of the heating surface facing the channels to the area of the heating surface in contact with the porous body), a uniform increase in α with increase in q is observed (Fig. 3), which is consistent with the formulas for free boiling. Obviously, for the heat fluxes investigated, this channel density creates satisfactory conditions for vapor removal.

The results of experiments with TPVF-1 from the same block, i. e., with the same characteristics, but without vapor channels are also presented in Fig. 3. At a certain value of the heat flux ($q = 1.8 \cdot 10^4$ W/m^2) ΔT begins to increase continuously. Since the heating surface is submerged, the chief obstacle to an increase in heat transfer is not the limited ability of the material to supply liquid to the heating surface by overcoming the force of gravity (as might have been assumed in the experiments on apparatus No. 1), but the excessive increase in the vapor content of the contact layer. By occupying more and more of the capillary channels, the vapor formed obstructs the access of water to the heating surface.

Thus, a sort of indirect boiling crisis is observed, in this case at $q = 1.6–1.9 \cdot 10^4$ W/m^2. A similar continuous temperature rise was also observed in one of the experiments on apparatus No. 1 at $q = 8.6 \cdot 10^4$ W/m^2 and at p = 107 mbar (interval BB in Fig. 2).

Clearly, in the presence of capillary-porous bodies a crisis in the usual sense is not possible, since although the vapor formed drives water away from the heating surface into the large capillaries, under the action of capillary forces, water continues to reach the heating surface through the network of small capillaries. The "crisis" is mildly expressed, and the "critical" heat flux can be raised by improving the vapor removal conditions.

The value of the channel density s = 0.14, investigated on apparatus No. 2, was selected to ensure that there would be no entrainment of liquid droplets at the relatively low heat flux $q = 10^3–5 \cdot 10^3$ W/m^2 in the presence of steady-state boiling. The value s = = 0.14 is the minimum channel density ensuring reliable vapor separation and the absence of liquid entrainment. As already pointed out, in this case the vapor velocity in the channels was 30–50 m/sec.

It is possible that this density is not optimum from the standpoint of the maximum rate of heat transfer. In Fig. 3 the data for boiling at $p_S = 9.6$ mbar in TPVF with channels s = 0.14 and in TPVF without channels shows that in the "subcritical" region the α differ sharply. Accordingly, taking into account the above-

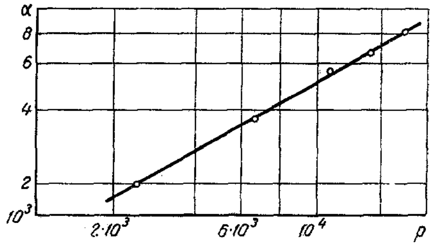

Fig. 4. Heat-transfer coefficient α ($W/m^2 \cdot deg$) as a function of the pressure p (N/m^2) at a heat flux $q = 25 \cdot 10^3$ W/m^2 from data obtained on apparatus No. 1.

mentioned small absolute values of the heat-transfer coefficient, it is important to find the optimum channel density which, while excluding the entrainment of liquid, at the same time ensures maximum heat transfer.

Since boiling is a very complicated process, the experimental data are always characterized by a very considerable scatter. This is particularly true of the case described, where the number of factors controlling heat transfer is increased owing to the presence of the capillary-porous body. The effect of these factors has not yet been adequately studied. Hence, it is difficult to propose any final formulas for calculating the heat transfer. However, it can be stated that, as in ordinary boiling, reduced pressure leads to a reduced rate of heat transfer. The relation $\alpha = f(p)$ for a constant heat flux $q = 25 \cdot 10^3$ W/m^2 is quite well represented by the expression $\alpha = \alpha_0 (p/p_0)^{0.5}$ (Fig. 4).

It should also be noted that in the case of boiling at a submerged surface, surrounded by a capillary-porous body, if the vapor removal conditions are sufficiently good ($s = 0.14$) in the "subcritical" region the slope of the $\alpha = f(q)$ curve for low-pressure boiling (6–30 mbar) is approximately the same as for free boiling at near-atmospheric pressures.

The experimental data presented in Fig. 3 are closely approximated by the equation $\alpha = 2.3\, q^{0.65}$ (for $s = 0.14$).

NOTATION

T_W is the temperature of heating surface, °C; T_S is the liquid saturation temperature, °C; $\Delta T = T_W - T_S$ is the temperature head, °C; p_S is the saturation pressure, mbar; q is the specific heat flux, W/m^2; α is the heat-transfer coefficient, $W/m^2 \cdot deg$; α_0 is the heat-transfer coefficient at atmospheric pressure p_0; s is the channel density (dimensionless).

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