A STUDY OF BOILING AND CONDENSATION IN A HEAT TRANSFER ELEMENT

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Test data are shown on the heat transfer during simultaneous boiling and condensation in a closed channel, whereupon recommendations are made for the design of heat transfer elements.

Boiling and condensation were studied on a flat heat transfer element 1 m long and 100 × 8 mm in cross section, the heat flowing in and out on one side. The heat transfer element with phase transformations of the carrier medium [1, 2] constituted a closed channel partly filled with liquid and containing, besides, an evacuated internal cavity. The condensed heat carrier was driven toward the heat input segment by the force of gravity. Distilled water, ethanol, and methanol were used as heat carriers. The element was filled with heat carrier liquid up to the top surface of the heating segment, the length of which was varied from 200 to 50 mm. The 200 mm long condenser segment was located 600-675 mm away from the heating segment and cooled with running water. After installation of the heat carrier, air was removed from the cavity by means of a vacuum pump.

The tests were performed over a temperature range from 10 to 70°C with the element in three different positions: vertical, 45°, and 5° from the horizontal. The thermal flux density at the heating segment was varied from 2000 to 100,000 W/m^2. Measured were the power input, the flow rate, the temperature and the temperature rise of the cooling water, the temperature of vapor in the center of the element, the temperature of liquid in the heating segment, and the wall temperature over the length of the heat transfer element. In addition, the boiling and the condensation were visually observed and photographically recorded.

The mean coefficient of heat transfer during boiling within a confined volume was defined as

$$\bar{\alpha}_b = \frac{q_b}{\bar{t}_b - t_s},$$

with $\bar{t}_b$ denoting the temperature of the heating surface at the center section of the evaporator segment.

The maximum relative error in determining $\bar{\alpha}_b$ was 10% for the heater with a 0.02 m^2 surface area and 17% for the heater with a 0.005 m^2 surface area.

The author studied the effect of pressure and thermal flux density on the mean coefficient of heat transfer during boiling. Inasmuch as the heat transfer tests were performed within the 10-70°C temperature range, the saturation pressure in the system was 0.01-0.3 bar for water and 0.04-0.5 bar for the alcohols.

The boiling process within this pressure range features certain singularities noted in [3-5]: the process is rather unsteady and unstable, the bubbles are large ($d_0 > 10$ mm), the vapor nucleation centers are few, there is a characteristic crackle, the liquid is highly superheated, and the temperature of the heating surface as well as the temperature of the liquid fluctuate widely during bubble formation. To these features one should add an appreciable effect of the hydrostatic pressure exerted by the liquid column above the evaporator segment, which becomes very significant at lower pressure levels and causes a variation in the temperature of the heat transfer along the height of the evaporator segment.

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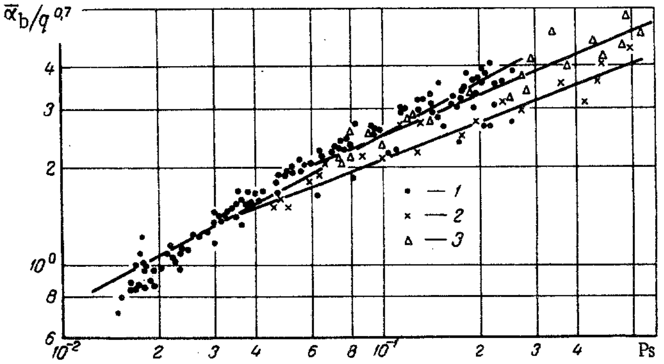

Fig. 1. Effect of pressure (P_s, bar) on the heat transfer during boiling of water (1), ethanol (2), and methanol (3).

The preliminary evaluation of boiling test data has shown that, for all practical purposes, the mean heat transfer coefficient depends neither on the size of the heat transfer surface nor on its inclination from the horizontal (90° and 45°). The relation $\bar{\alpha}_b = f(q_b)$ is a power-law relation with an exponent 0.7, just as at higher pressures. Some $\bar{\alpha}_b/q^{0.7} = f(P_s)$ curves are shown in Fig. 1 for water, ethanol, and methanol. The corresponding formulas, based on the linearization of test data in logarithmic coordinates, are

for water

$$\bar{\alpha}_b = 8.17 q_b^{0.7} P_s^{0.52}, \tag{1}$$

for ethanol

$$\bar{\alpha}_b = 4.97 q_b^{0.7} P_s^{0.37}, \tag{2}$$

for methanol

$$\bar{\alpha}_b = 6.25 q_b^{0.7} P_s^{0.4}. \tag{3}$$

Within the same pressure range, the heat transfer coefficient is a different function of pressure for the different heat carrier substances. The test data have been generalized by the method proposed by V. M. Vorishanskii [6] and based on the thermodynamic law for the respective states. An evaluation of the test data in the coordinates

$$\frac{\bar{\alpha}_b}{P_{cr}^{1/3} T_{cr}^{-5/6} M^{-1/6} q_b^{2/3}} = f(P/P_{cr})$$

in Fig. 2 has yielded the following universal formula for all three substances:

$$\bar{\alpha}_b = 750 \frac{P_{cr}^{1/3}}{T_{cr}^{5/6} M^{1/6}} q_b^{2/3} (P/P_{cr})^{0.023(P/P_{cr})^{-0.25}}. \tag{4}$$

Formula (4) is essentially a modification of the formula proposed in [6], extending the latter to the range of lower pressures and applicable for calculating the heat transfer coefficient at $(P/P_{cr}) \leq 4.5 \cdot 10^{-3}$.

In this way, if the entire range of boiling pressures is considered, the $\bar{\alpha}_b = f(P_s)$ curve passes through an inflection point characteristic of each particular liquid, inasmuch as the exponent of P increases with rising pressure above atmospheric. This confirms an earlier conclusion that the boiling process becomes qualitatively different at low pressures. Since the physical properties of the liquid and the vapor vary monotonically under varying pressure within the range of the respective phase, hence such a qualitative difference can be explained only by a change in the kinematic characteristics of the two-phase stream. This means that the spectrum of velocity fields within the volume of boiling liquid is determined not only by the thermal flux density but also by the dimensions, the number, and the frequency of bubbles — all of which vary with varying pressure much more in the low-pressure than in the high-pressure range. It is well known that at $P > 1$ bar the frequency and the number of vapor nucleation centers decrease with increasing pressure, but they also do when $P < 1$ bar; at $P \leq 0.2$ bar the frequency as well as the number of vapor nucleation centers become disproportionately lower than at $P = 1$ bar. Thus, decreasing the pressure by one order of magnitude (from 1 to 0.1 bar) results in a tenfold increase in the specific volume of vapor, while the increase in bubble dimension by one order of magnitude (and in volume by three orders of

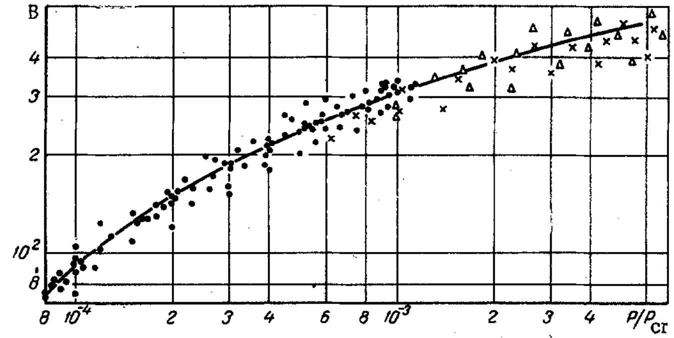

Fig. 2. Evaluation of boiling test data in universal coordinates (designations are the same as in Fig. 1): B $= \bar{\alpha}_b/q_b^{2/3}T_{cr}^{-5/6}P_{cr}^{1/3}M^{-1/6}$.

magnitude) indicates a decrease in the product (number) × (frequency) of vapor nucleation centers by a factor of 100.

Stable film condensation on the cooled segment begins only under heavy heat loads, when the quantity of condensate becomes large enough to form a continuous film. At $q_c < 10^4$ W/m^2 the condensate begins to flow off along liquid streaks.

The mean coefficient of heat transfer during condensation is defined as

$$\bar{\alpha}_c = \frac{q_c}{t_s - \bar{t}_c}.$$

The temperature of the condenser wall $\bar{t}_c$ was determined as the average reading of nine thermocouples uniformly spaced on the inside condensation surface. The maximum relative error in determining $\bar{\alpha}_c$ during film condensation was 16%, but reached 24% at $q_c < 10^4$ W/m^2 because of low temperature drops during condensation.

We will analyze the data pertaining to film condensation with the test element inclined 90° and 45° respectively from the horizontal.

The film flow was laminar under all test conditions and in most cases subundulatory. The Reynolds number for the film in the water tests was not higher than Re = 5. In the alcohol tests waves were forming at a thermal flux density $q_c \geq 2 \cdot 10^4$ W/m^2. An explanation for this is that the alcohols have a lower heat of phase transformation than water and that the flow rate of their condensate is correspondingly higher under the same heat loads. The values of the heat transfer coefficient obtained here were 20-25% lower than those calculated by the Nusselt equation. This discrepancy between tested and calculated values is normal for the given system and due to the combined effect of higher moisture content in the vapor and of vapor–liquid counterflow.

In studying the heat transfer in a closed system it is desirable to use the same method of evaluating and generalizing the test data on the basis of the thermodynamic similarity theory. As has been shown in [7], the following expression may be set up for the coefficient of heat transfer during condensation:

$$\bar{\alpha}_c = q^{n_1} l^{n_2} d^{n_3} \left(\frac{M}{g}\right)^{m_1} T_{cr}^{m_2} P_{cr}^{m_3} R^{m_4} F(P/P_{cr}). \tag{5}$$

During film condensation of vapor at a vertical flat wall $\bar{\alpha}_c \sim q^{-1/3} l^{-1/4}$ which follows from an analytical solution of the Nusselt equation and is valid when t_w = const at a moderate vapor velocity. In this case the exponents m_1, m_2, m_3, and m_4 are, by dimensional analysis, found to be

$$m_1 = -2/3; \quad m_2 = -1/4; \quad m_3 = 5/4; \quad m_4 = 3/4,$$

and relation (5) becomes

$$\bar{\alpha}_c = \frac{A P_{cr}^{5/4} R^{3/4} F(P/P_{cr})}{T_{cr}^{1/4} \left(\frac{M}{g}\right)^{2/3} l^{1/4} q^{1/3}}. \tag{6}$$

The trends of function $F(P/P_{cr})$ and the numerical coefficient A were determined on the basis of test data (Fig. 3) in the coordinates of relation (6).

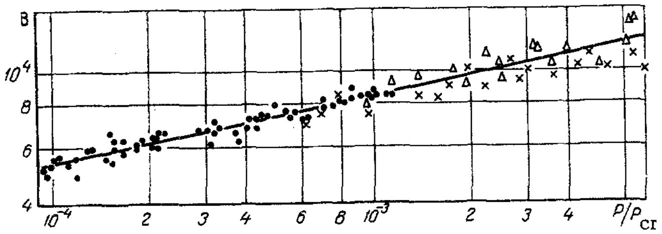

Fig. 3. Test data on condensation, in universal coordinates (designations are the same as in Fig. 1): B $= \bar{\alpha}_c/P_{cr}^{5/4}T_{cr}^{-1/4}M^{-2/3}q^{-1/3}l^{-1/4}$.

In this way, a universal equation has been obtained for the mean coefficient of heat transfer during condensation which is structurally analogous to Eq. (4):

$$\bar{\alpha}_c = 3.36 \cdot 10^4 \frac{P_{cr}^{5/4}}{T_{cr}^{1/4} M^{2/3}} q^{-1/3} l^{-1/4} (P/P_{cr})^{0.2}. \tag{7}$$

The variation of $\bar{\alpha}_c$ depending on the inclination angle β can be accounted for by introducing the factor $(\sin\beta)^{1/4}$ in Eq. (7). However, Eq. (7) is valid only when the condenser is not wetted by water coming from the evaporator. At low inclination angles (5°), even under a minimal heat load the condenser surface is completely wetted by a vapor–liquid mixture. This leads to a change in the heat transfer pattern, inasmuch as heat is now transmitted not only directly through phase transformations but also through convection in the two-phase stream. The flow of the two-phase stream is quasiplugged in nature, i.e., somewhere between split apart to annular. This is also indicated by the relation between the thermal flux density and the heat transfer coefficient at the heat input segment. While for a split flow $\alpha \sim q^{0.6}$ and for an annular flow $\alpha \sim q^{0.2}$, the water test with $\beta = 5°$ yielded

$$\bar{\alpha}_b = 2.62\, q_b^{0.41}\, t_s^{0.68}. \tag{8}$$

The coefficient of heat transfer at the cooling segment is almost independent of the thermal flux density and is determined only by the saturation pressure of temperature:

$$\bar{\alpha}_c = 330\, t_s^{0.96}. \tag{9}$$

This means that convective heat transfer predominates at the heat input segment.

The basic problem in designing a heat transfer element is to ensure a minimum temperature difference between the heat input and the heat output segment at a given heat load and a given length of the element. As the performance criterion for a heat transfer element we may use its effective thermal conductivity

$$\lambda_{eff} = \frac{Ql}{F_L(\bar{t}_b - \bar{t}_c)}. \tag{10}$$

The necessary surfaces and the arrangement of evaporator and condenser segments are designed on the basis of formulas (7) and (4), aiming at the critical thermal flux density during boiling and the minimum flux density at which film condensation of vapor begins.

The transverse dimensions of a heat transfer element are determined on the basis of a hydrodynamic analysis of the vapor flow. Considering that the thickness of the liquid film flowing from the condenser is negligibly small as compared to the cross section area of the channel (in our tests it did not exceed 0.2-0.5 mm) and that its velocity is infinitesimal as compared to the velocity of the vapor stream ($u_V/u_L \geq 1000$), the flow of vapor in a stationary channel may be assumed steady. Under our test conditions the vapor flow was laminar in all modes. The solution to the Navier–Stokes equation

$$\frac{dP}{dx} = \mu \frac{d^2u}{dy^2} \tag{11}$$

leads to a relation between the heat load, the pressure drop from heat input segment to heat output segment, and the geometrical dimensions of the heat transfer element.

For a flat element

$$\frac{Q}{r\rho''} = \frac{a\delta^3}{2\mu}\left(-\frac{dP}{dx}\right), \tag{12}$$

where $-dP/dx = (P_b - P_c)/l = \text{const}$.

With the geometrical dimensions and the heat load known, the drop in vapor pressure between evaporator and condenser segment can be found from the relation

$$\Delta P = \frac{2Q\mu l}{r\rho'' a\delta^3}. \tag{13}$$

For a tubular element

$$\Delta P = \frac{128Q\mu l}{r\rho''\pi d^4}. \tag{14}$$

With the aid of Eq. (13) or (14), the dimensions of the vapor channel should be matched so that the drop in vapor pressure will not produce a significant drop in saturation temperature, because at low pressures this would lead to an additional increase in the temperature difference between the heating and the cooling segment.

In our tests the temperature difference between the heat input and the heat output segment was 7.5-25°C at a transmitted power of 600 W. The effective thermal conductivity was as high as 10^5 W/m · °C.

NOTATION

α	is the heat transfer coefficient, W/m^2 · °C;
q	is the thermal flux density;
T, t	are temperatures;
P	is the pressure;
ΔP	is the pressure drop;
A	is the constant coefficient;
M	is the molecular weight;
g	is the acceleration of gravity;
l	is the length;
d	is the diameter;
R	is the universal gas constant;
β	is the inclination angle;
Q	is the heat load, power;
F_L	is the cross section area;
λ_{eff}	is the effective thermal conductivity;
μ	is the viscosity coefficient;
u	is the velocity;
r	is the heat of phase transformation;
ρ''	is the vapor density;
2δ	is the gap width;
a	is the channel width.

Subscripts

c	refers to condensation;
b	refers to boiling;
cr	refers to critical values;
s	refers to saturation;
w	refers to wall.

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