AN EXPERIMENTAL STUDY OF HEAT TRANSFER IN THE BOILING OF FREON-12 IN A FIELD ACTED ON BY CENTRIFUGAL FORCES

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Results are described for an investigation into the boiling of Freon-12 in the following range of inertial accelerations: $a/g = 1$–$5250$.

Increasing interest has recently been expressed in studies of the processes involved in the boiling of liquids under conditions of inertial acceleration. When using evaporation cooling in high-speed rotation equipment (for example, in turbine generators, and in specialized electrical equipment, etc.) the coefficients of acceleration may reach as high as 5000 and more. However, the majority of published studies [7–14] encompass only a relatively small range of variations in the gravitation-inertial accelerations (0–100g). In references [4, 5] only the critical heat flows in the boiling of water and alcohol [5] were investigated for accelerations to $a/g = 200$ and 2050, respectively.

Figure 1 shows the diagram of an experimental installation used in the experiments. A detailed description of the installation, its rotating parts, and the measurement circuit has been published in [2].

Here we are investigating the process of boiling Freon-12 at the inside surface of the test elements which are made up of copper cylinders with a diameter of d = 30 mm and a length of 40 mm.

Visual observations and stroboscopic photography revealed the possibility of establishing the nature of the processes involved in the boiling of Freon-12 in a field acted on by centrifugal forces, as well as the possibility of confirming the validity of earlier concepts regarding the phenomena taking place in this case [1, 2, 10]. The test liquid was thrown to the peripheral portion of the test element under the action of the centrifugal forces and it filled a portion of the test-element cross section (Fig. 2) in accordance with the specified regime of the test. In Fig. 2 we see the zones of the most intense vapor formation within the liquid layer. In the case of small heat flows these zones are situated near the phase-separation surface. With an increase in the heat flow (with a constant acceleration factor $a/g$ and a constant thickness $\delta$ for the layer) the region of intense boiling is extended into the depth of the liquid, i.e., into the region of elevated pressure. This fact can be treated as an experimental confirmation of the existence of various saturation temperatures within the thickness of the liquid layer in the case of inertial accelerations. This can also be concluded from the "displacement" of the boiling foci toward the free surface with an increase in the acceleration factor in the case of a constant heat flow. The light region in the

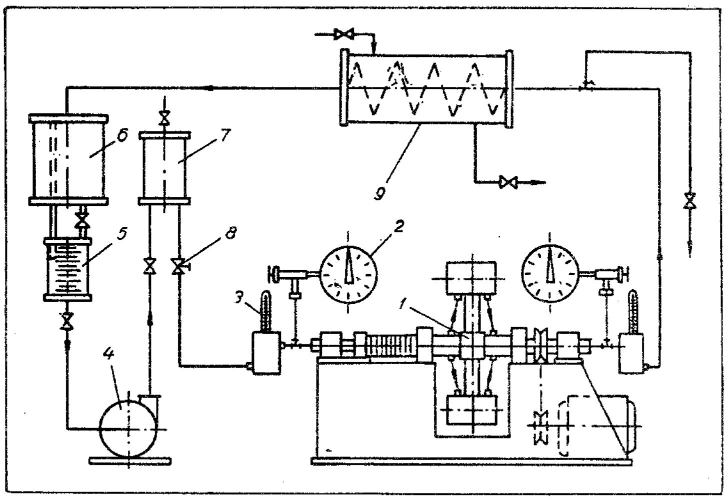


Fig. 1. Experimental installation: 1) rotating model; 2) manometer; 3) thermometer; 4) pump; 5) measuring tank; 6) condensate collector; 7) receiver; 8) needle valve; 9) condenser.

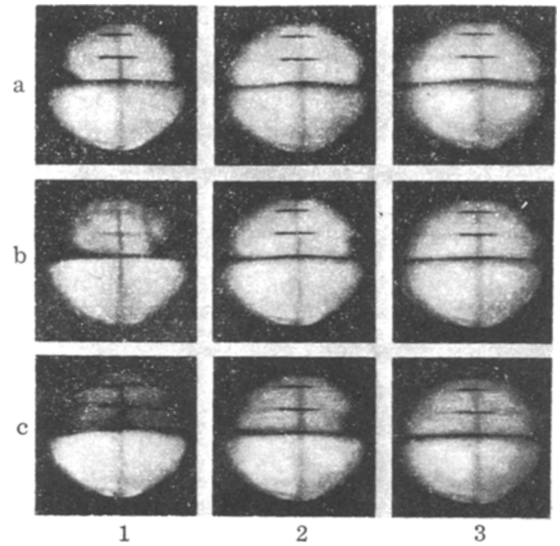

Fig. 2. Freon-12 boiling in the region of centrifugal forces: 1) $a/g = 204$; 2) 460; 3) 820; a) $q = 53\ 200\ W/m^2$; b) 106 400; c) 160 000

central portion of the liquid layer indicates the absence of vapor bubbles in this zone. There is reason to assume that the liquid—with the temperature prevailing at the phase-separation surface—is shifted into the depth of the layer under the action of centrifugal forces, thereby participating in the so-called "free" convection. As follows from Fig. 2, the role of the "free" convection increases in this case with an increase in the acceleration factor.

Experiments to study the transfer of heat in the boiling of Freon-12 were carried out with the following values for the parameters being investigated: $a/g = 1, 204, 460, 820, 1300, 1900, 2570, 3360, 4270, 5250$, $q = 26\ 600, 53\ 200, 79\ 800, 106\ 400, 133\ 000, 160\ 000, 200\ 000\ W/m^2$, $\delta = 3, 5, 10, 15$ mm.

In processing the experimental data we assumed that all of the heat supplied to the test elements is removed through the surface flushed exclusively by the liquid (the removal of heat by the vapor phase is not considered). The composition of the thermal balance whose maximum divergence amounts to 8% confirmed the validity of this assumption. Thus, the specific heat flow is determined from the formula

$$q = Q/2r \arccos \frac{r-\delta}{r}\, l. \tag{1}$$

With consideration of the error in the measurement of the liquid-layer thickness and of the geometric dimensions of the test element, the most probable error in the determination of q amounted to 10%.

The tests established that the temperature of the heating surface—because of the high thermal conductivity of copper—is virtually identical at all points at which it comes into contact with the liquid. The saturation temperature for the liquid varied over the surface of the test element as a result of a pressure gradient which was developed by the centrifugal forces. Thus, in connection with the change in the temperature head over the surface of the test element the local values of the heat-transfer coefficients will also be changed. To obtain the average heat-transfer coefficients we determined the mean-integral value for the liquid saturation temperature over the flushed surface:

$$t_{l_{av}} = \frac{\int_s t_l\, ds}{s}, \tag{2}$$

where

$$t_l = t_1 + \int_{p_1}^{p} \frac{\partial T}{\partial p}\, dp_x. \tag{3}$$

Assuming $\partial T/\partial p \cong \Delta T/\Delta p = \text{const}$ in the range of change in pressure over the thickness of the liquid layer, after integration of (2) with consideration of (3), we obtain

$$t_{l_{av}} = t_1 + \frac{\Delta T}{\Delta p} \rho\omega^2 R \left[ (\delta - r) + \frac{r\sqrt{1-\left(\frac{r-\delta}{r}\right)^2}}{\arccos \frac{r-\delta}{r}} \right], \tag{4}$$

where $\Delta p$ is the pressure difference across the thickness of the liquid layer:

$$\Delta p = \rho\omega^2 R\, \delta, \tag{5}$$

and $\Delta T$ is the change in the liquid saturation temperature, corresponding to $\Delta p$ for the given $p_1$.

With consideration of $\partial T/\partial p$ as a function of pressure, we obtained a refined formula for the determination of $t_{l_{av}}$ which, in view of its cumbersome nature, is not presented here. The divergence in the values of $t_{l_{av}}$, derived from Eq. (4) and from the refined formula, does not exceed 1%.

Thus, the average heat-transfer coefficients were determined from the formula

$$\alpha_{av} = \frac{q}{t_w - t_{l\,av}}. \tag{6}$$

Analysis of the errors demonstrated that the most probable error in the determination of $\alpha_{av}$ amounted to ±25%.

As a result of the adjustment tests carried out in the absence of machine rotation, we obtained data generalized by the empirical equation

$$\alpha = 7.96 q^{0.6}. \tag{7}$$

The divergence of these data from the results of reference [7] does not exceed 10%.

In carrying out the basic tests we established that for specific q, $a/g$, and $\delta$ the liquid does not boil over the entire flushed heating surface, i.e., the temperature of the wall underneath the liquid layer was lower than the saturation temperature for the corresponding pressure $p_1 + \Delta p$. This means that a specific value for the minimum heat flow $q_{min}$ corresponds to each value of $a/g$ and $\delta$; below this minimum value for the heat flow the liquid beneath the layer had not reached the saturation temperature. Here the heat flow was removed by the boiling of the liquid near the phase-separation surface and by "free" convection to the remaining segments of the heating surface. The data on $q_{min}$ over the entire studied range of $a/g$ and $\delta$ have been generalized by the empirical equation

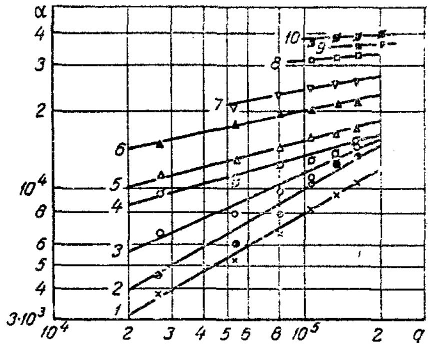

Fig. 3. Heat-transfer coefficients versus heat flux: 1) $a/g = 1$; 2) 204; 3) 460; 4) 820; 5) 1300; 6) 1900; 7) 2570; 8) 3360; 9) 4270; 10) 5250

$$q_{\min} = 4.1 \cdot 10^{-3} \left(\frac{a}{g}\right)^{1.8} \delta^{1.44}. \qquad (8)$$

The experimental data were processed according to formula (6) for $q > q_{\min}$. Here we found that the heat-transfer coefficients are independent of the liquid-layer thickness $\delta$.

Figure 3 shows the heat-transfer coefficients as a function of the specific heat flows for a liquid-layer thickness of $\delta = 5$ mm and various values of the acceleration factor (from 1 to 5250). We see from the graph that with an increase in the inertial accelerations the absolute values of the heat-transfer coefficients increase substantially. The relative influence of the heat flow on the intensity of heat transfer with an increase in $a/g$ diminshes. The experimental data for each value of the acceleration factor can be generalized by an equation of the following type:

$$\alpha = Cq^n, \qquad (9)$$

where the exponent n diminishes from 0.6 for $a/g = 1$ to 0.1 $a/g = 5250$.

Figure 4 shows the heat-transfer coefficients as a function of the accelerations for various values of the specific heat flows. On the whole, the nature of these functions is in accord with Kutateladze [3], which characterizes the combined action of the heat flow and of the organized motion of the liquid (forced or free motion) on the intensity of heat transfer. In this graph we can distinguish two regions. In the first of these ($a/g = 1-800$) the effect of the accelerations is substantial for relatively small heat flows. With an increase in the heat flow, the relative effect of the acceleration on $\alpha$ is diminished. In the heat-flow range 26 600–160 000 W/m$^2$ the degree of effect exerted by the accelerations on $\alpha$ diminishes from 0.56 to 0.04.

In the second region, with acceleration-factor values of 1000 and higher, the relative effect of acceleration on the heat-transfer coefficient is virtually identical for all of the heat-flow values studied here and it is equal to 0.6. In the acceleration-factor range 1000–5250 the experimental data are described well by the empirical equation

$$\alpha = 14.2 \left(\frac{a}{g}\right)^{0.6} q^{0.23}. \qquad (10)$$

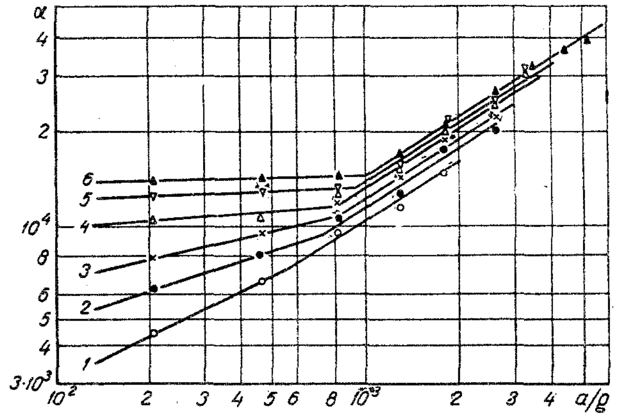

Fig. 4. Heat-transfer coefficients versus inertial acceleration: 1) q = 26 600 W/m$^2$; 2) 53 200; 3) 79 800; 4) 106 400; 5) 133 000; 6) 160 000

For the cases in which the boiling of the liquid did not take place over the entire flushed heating surface, i.e., when $q < q_{\min}$, the experimental data have been processed according to the formula

$$\alpha_{\text{av}} = \frac{q}{t_{\text{w}} - t_1} \qquad (11)$$

and with a scattering of ±15% have been generalized by the equation

$$\alpha = 130q^{0.4}. \qquad (12)$$

Thus, the heat-transfer coefficients in this region are virtually independent of the inertial accelerations. This circumstance can be explained by the fact that the intensification of heat transfer as a result of the increasing effect of free convection is offset by the reduction in the number of developed-boiling zones which, with an increase in the accelerations, are "displaced" toward the phase-separation surface.

The experimental data of this investigation have been obtained for a range of pressures between 7.5 and 9.5 bar over the free surface of the liquid. An increase in pressure in the vapor space of the test element, resulting from the rotation of the vapor column in the radial segment of the rotating contour, was determined from the change in the saturation temperature at the phase-separation surface, as well as from the readings of tensometric pressure gauges. We obtained excellent agreement between the experimental data and those calculated from the theoretical formula [1].

An increase in the heat-transfer coefficients with a rise in the inertial accelerations could also result from an increase in the saturation pressure within the test element. However, separate investigations of the function $\alpha = f(p)$ for $a/g$ = const showed that in this range of pressure variations the increase in $\alpha$ could amount to no more than 10%.

NOTATION

$\vec{a}$ is the inertial-gravitational acceleration, m/sec$^2$; $\omega$ is the angular velocity, 1/sec; R is the liquid-layer rotation radius, m; g is the gravitational acceleration, m/sec$^2$; Q is the heat flux, W; q is the specific heat flux, W/m$^2$; $t_w$ is the wall temperature of the experimental sample, °C; $t_l$ is the temperature of the liquid, °C; $t_i$ is the saturation temperature at the interface, °C; $p_i$ is the pressure at the interface, bar; $p_x$ is the pressure in liquid layer at a distance x from the free surface, bar; $\Delta p$ is the pressure drop across the thickness of the rotating liquid layer, bar; $\delta$ is the thickness of the liquid layer, m; r is the internal radius of the experimental sample, m; $l$ is the length of the experimental sample, m; $\rho_{l_{av}}$ is the density of the liquid, kg/m$^3$; $\alpha$ is the heat-transfer coefficient, W/m$^2$ · deg; s is the half-surface of the experimental sample in the flow, m$^2$.

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