CRITICAL THERMAL LOADS IN VERTICAL PIPES WITH THE LOWER END SEALED

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The critical point in heat exchange for boiling water in vertical tubes with the lower end sealed is investigated. A computational formula is suggested for determining the critical heat loads.

When water boils in vertical tubes with a zero flow rate, the normal cooling by heat transfer through the surface is limited by the conditions of flooding, which appears for corresponding flow rates of the phases, pressures, and the channel geometry.

The relations for flooding in vertical tubes can be written as follows [1]:

$$(W_*'')^{1/2} + m(W_*')^{1/2} = C. \tag{1}$$

For turbulent flow, m = 1. The quantity C depends on how the ends of the tubes are constructed, as well as on how the gas and liquid are injected and removed. For tubes with sharp edges at the inlet the value C = 0.725 is recommended, and if the end effects can be neglected, then 0.88 < C < 1 [2, 3].

The dimensionless combinations entering into (1) and relating the flow rates with hydrostatic forces are written as follows:

$$W_*' = W' \sqrt{\rho'}\,[gD(\rho' - \rho'')]^{-1/2}, \tag{2}$$

$$W_*'' = W'' \sqrt{\rho''}\,[gD(\rho' - \rho'')]^{-1/2}. \tag{3}$$

The condition for deterioration of normal cooling of the wall of a heated tube in the presence of boiling in the flooding regime with the lower end of the tube sealed, while feeding is effectuated through the upper end of the tube from a large volume of liquid at the saturation temperature, is written in the form

$$W'\rho' = W''\rho''. \tag{4}$$

Substituting (2)-(4) into Eq. (1), we obtain

$$(W'')^{1/2}(\rho'')^{1/4}\left[1 + \left(\frac{\rho''}{\rho'}\right)^{1/4}\right] = C\,[gD(\rho' - \rho'')]^{1/4}. \tag{5}$$

The reduced velocity of the vapor W" in the upper section of a heated round tube can be computed according to the formula

$$W'' = \frac{4Q}{\pi r \rho'' D^2}. \tag{6}$$

Using Eqs. (5) and (6), we can obtain an expression for the maximum heat flux

$$Q_{\max} = \frac{\pi}{4}\, C^2 r D^{5/2}\, \frac{\sqrt{g\rho''(\rho' - \rho'')}}{\left[1 + \left(\frac{\rho''}{\rho'}\right)^{1/4}\right]^2}. \tag{7}$$

We carried out experiments in order to determine the critical heat loads with boiling water in vertical tubes with pressure varying in the range 0.5-8 MPa. In the experiments, we used tubes made of stainless steel with an inner diameter 9.3, 16, and 23 mm and length varying from 1 to 2.4 m. Heating was effectuated by alternating electrical current. The heat load was distributed uniformly along the height of the tube. The upper end of the tube

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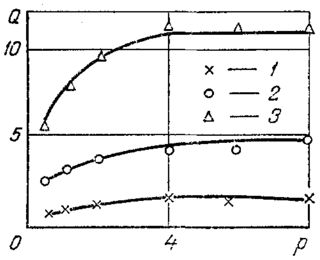

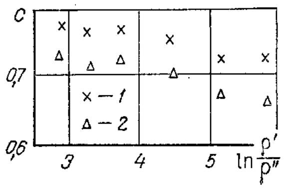

Fig. 1 Fig. 2

Fig. 1. Total critical heat flux Q as a function of the pressure p (solid line is a calculation using Eq. (7); Q in kW; p in MPa): 1) D = 9.3 mm; 2) 16; 3) 23.

Fig. 2. The coefficient C as a function of pressure: 1) D = 9.3 mm; 2) 23.

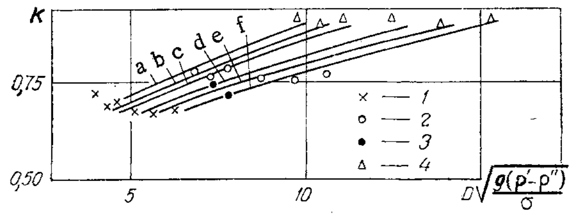

Fig. 3. The criterion K as a function of the parameter $D(g(\rho' - \sigma)^{1/2}$ (solid line is computed from Eq. (10); p in MPa; D in mm): a) p = 0.5; b) 1; c) 2; d) 4; e) 6; f) 8; 1) D = 9.3 mm; 2) 16, L/D = 146; 3) 16, L/D = 65; 4) 23.

was connected to a vessel with a very large diameter filled with water at the saturation temperature. The lower end of the tube was sealed, while the upper end had a sharp edge. The steam forming in the tube passed through the water in the vessel and entered into a condenser-cooler, where it was condensed and returned into the vessel.

The results of the experiments are presented in Fig. 1 in the form of a function showing the behavior of the total critical heat flux as a function of the pressure.

Comparison of the experimental data with computations using formula (7) allowed for a determination of the value of the coefficient C. It was found that in the range of parameters studied the magnitude of C is not a constant, but varies with increasing diameter of the tube and increasing pressure (Fig. 2). For small pressures, the values of C obtained agreed with the data in [1], wherein for pressures close to atmospheric the value C = 0.725 is recommended. The numerical relation proposed can be represented in dimensionless form as

$$C = \left[D\sqrt{\frac{g(\rho' - \rho'')}{\sigma}}\right]^{-0.08}\left(\frac{\rho'}{\rho''}\right)^{-0.04}. \tag{8}$$

Calculations were performed using Eq. (7) taking into account (8) and the results are presented in Fig. 1.

The experimental data were analyzed and correlated on the basis of the fact that the critical point in the heat exchange with boiling of the liquid at the heat transfer surface (critical point of the first kind) and flooding with a counter-flowing liquid–gas flow are similar phenomena, which are accompanied by a basic change in the hydrodynamics of the process leading to a loss of stability in preceding the structure of a two-phase flow. The following system of dimensionless parameters was used in [4] in order to investigate the stability of the motion of gas–liquid mixtures in vertical tubes:

$$K = f\left(\frac{W'^2}{gD},\ D\sqrt{\frac{g(\rho' - \rho'')}{\sigma}},\ \frac{\rho'}{\rho''}\ \ldots\right). \tag{9}$$

Here, $K = W''\sqrt{\rho''}/\sqrt[4]{\sigma g(\rho' - \rho'')}$ is the criterion for stability of a two-phase layer [5].

We can obtain an expression for K by determining the reduced velocity of the steam W" from Eq. (5)

$$K=\left[\frac{C}{1+\left(\frac{\rho''}{\rho'}\right)^{1/4}}\right]^2\left[D\sqrt{\frac{g(\rho'-\rho'')}{\sigma}}\right]^{1/2}. \qquad (10)$$

Figure 3 shows the results of the analysis of the experimental data in terms of the criterion K as a function of the parameter $D\sqrt{g(\rho'-\rho'')/\sigma}$. The computed curves, obtained using Eq. (10) taking into account (8), are indicated here as well.

In the pressure range from 0.5 to 8 MPa, relation (10) agrees quite well with the experimental data and is recommended for use with calculations of critical heat loads in vertical tubes with the lower end sealed.

NOTATION

W', W", reduced velocities of water and steam; ρ', ρ", density of water and steam; B, diameter of the tube; g, acceleration of gravity; Q, total heat flux; r, heat of evaporation; σ, coefficient of surface tension.

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HYDRAULIC RESISTANCE FOR LIQUID FLOW WITH BOILING-UP IN A PARTIALLY HEATED CHANNEL

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An approximate analytic solution is obtained to the problem of calculating the hydraulic resistance. The hydraulic characteristics are analyzed and the region for static instability is determined.

At the present time, heat removal systems with long channels are constructed in cryogenic technology. These systems are characterized by a relatively low heat flux density, which allows for heat removal in the single-phase (economizing) region with subsequent boiling-up of the liquid along the adiabatic (unheated) section of the channel. Thus, for example, the low-temperature state of the heat-absorbing shields of thermal vacuum chambers is maintained in this manner with the help of a natural circulation loop with an external boiling zone [1], as is the cooling of cryogenic condensation pumps and cryogenic electrical devices with forced circulation of liquid nitrogen and helium in parallel channels.

Boiling-up of a liquid in this case is understood to be the process of evaporation caused by a decrease in pressure. References [3] and [3] examine primarily the boiling-up process as resulting from an increase in the flow rate due to a decrease in the channel cross section or a sudden change in pressure. In this case, the kinetic energy of the flow is low in comparison with the enthalpy, and for this reason, the change in the flow rate along the length of the channel can be neglected. However, the change in pressure along the channel,

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