HEAT-TRANSFER INTENSITY AND TEMPERATURE PROFILES IN BOILING LIQUID AND IN A FOAM LAYER

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The hydrodynamics and heat transfer in boiling water with the formation of a foam layer in a vertical rectangular channel with omnidirectional heating are experimentally investigated.

Boiling processes, which are widely used in various fields of engineering, are constantly under investigation. Most often, these processes are studied with forced motion in channels or with natural convection in "large-volume" conditions. This ignores the region of mixed convection, where the influence of forced and free motion and their interaction appear. Studying the hydrodynamics and intensity of heat and mass transfer in boiling in these conditions, for which very low rates of forced motion and relatively large channel dimensions are characteristic, may give important information on the mechanism of the process.

Hydrogas dynamics, the temperature profiles in the boundary layer, and the heat-transfer intensity have been considered in bubble boiling in a vertical homogeneous heated channel in conditions of very low velocities of forced motion. In addition, the features of heat transfer and hydrodynamics in the foam layer arising in the experimental section in the formation of the level were investigated.

A vertical channel of rectangular cross section (40 × 100 mm) and height 400 mm with a floor, a roof, and lateral connecting pipes of diameter 10/8 mm for water input and output was used as the experimental section. Two glass windows (dimensions 30 × 60 mm) were positioned on opposite lateral faces of the section. A vertical narrow face of the channel was heated on a section of length 350 mm by an electric heater made in the form of spirals (diameter 13 mm) of Nichrome wire (diameter 1.2 mm) on a ceramic yoke. The heater power was regulated using an RNO-5/250 autotransformer. The section was included in a hydrodynamic stand supplied from a water pipeline in a closed scheme.

The investigation procedure was intended for measuring the temperature of the heat-transfer surface t_w in 14 cross sections distributed uniformly over the channel height and the temperature profiles in three cross sections by means of extensible probes with thermocouples and also by visual observation of the hydrodynamics in the section.

The operational parameters were varied over the following ranges: water pressure P = 0.1-0.18 MPa, mean water velocity W = 0.2-13 mm/sec, water temperature at outlet t_{out} up to 102°C, heat-flux density q = 20-130 kW/m^2.

The experimental results with mixed single-phase water convection in this section in [1] show that free motion has a determining influence, leading to the development of circulation over a closed loop and consequently to intensification of the heat transfer. This influence was expected to be stronger in boiling at the heat-transfer surface.

The boiling was pinpointed visually, and also from the condition that the wall temperature t_w exceeds the saturation temperature $t_s(p)$ [2]. For the cross section of onset of surface boiling, the degree to which the water heating in the central zone of the section is below saturation temperature $\Delta t_{und} = t_s - t_w$ was determined. As is evident from Fig. 1, this underheating is reduced with increase in q and decrease in W, in contrast to the experimental results of [3] for water flow in a tube of diameter 20 mm, $W\rho$ < 200 kg/m^2·sec. The dependence obtained is of unusual form, because of the intensification of single-phase convective heat transfer in the given conditions under the influence of the development of circulation with high values of the mixed-convection parameter $Ra/Re^2 = 10^2$-10^4 and asymmetry of the flow.

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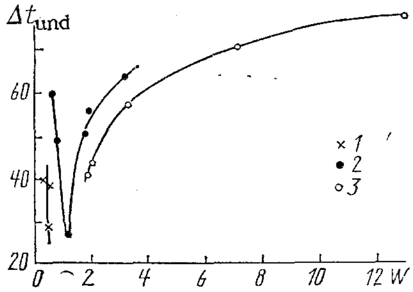

Fig. 1

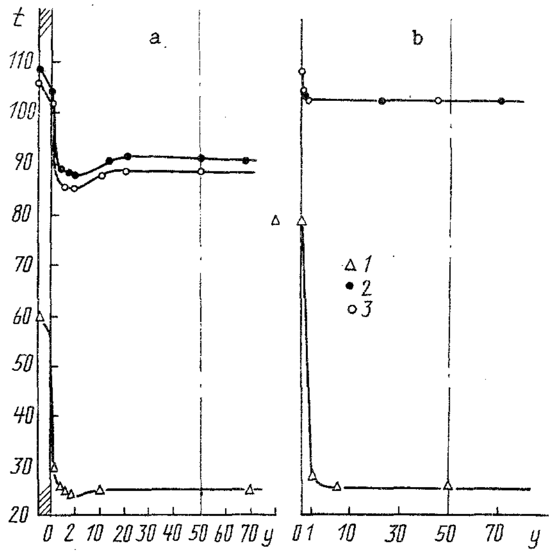

Fig. 2

Fig. 1. Degree to which water heating is below the saturation temperature in the cross section of onset of surface boiling: 1) q = 37-56 kW/m²; 2) 75; 3) 127; Δt_{und}, °C; W, mm/sec.

Fig. 2. Temperature profiles: a) W = 0.5 mm/sec, q = 36 kW/m², t_{out} = 85°C, P = 1.12 bar; b) W = 0.75 mm/sec, q = 78 kW/m², t_{out} = 102°C, P = 1.16 bar; 1) cross section 2 (x = 50 mm); 2) cross section 12 (290 mm); 3) cross section 14 (330 mm). For the cross-section numbers, see Fig. 3; t, °C; y, mm.

Motion of the steam–water flow on boiling is clearly observed through the windows, from the trajectory of motion of steam bubbles and partly of air bubbles, especially with the appearance of a level in the section (a water–foam interface). The upward flux of water with steam bubbles of diameter 0.5-1 mm at a heated wall, without escaping completely into the upper part of the section, turns and descends to the bottom unheated surface, where it turns again toward the heating surface, forming a closed circulation loop.

The experimental data in Fig. 2a reflect this character of the hydrodynamics: the deformation (bending) of the temperature profiles observed experimentally is associated with the returning downward current of more heated water. This anomaly was not observed previously, even with asymmetric heating [4], since the hydrodynamic conditions were different ($W\rho$ = 1000 kg/m²·sec, tube diameter 10 mm) and the influence of natural convection does not appear.

Water-dissolved air cannot have a pronounced influence on the hydrodynamics and heat transfer in the experimental conditions because it is present in small amounts: on heating water from 20 to 110°C, 30 mg/kg of air (weight content 0.003) is liberated because of the reduction in its solubility. More than 2-3 orders of magnitude more steam is formed, for example, in the experiment with the results shown in Fig. 3, where the weight content of steam at the output is 1.7%.

With increase in q, the onset of surface boiling is shifted downward to cross sections 4-7, the motion of the steam–water mixture with the horizontal flows is intensified, and finally, with the boiling of saturated water, the closed circulation loop completely determines the hydrodynamics in the cross section, since the rate of forced convection is small (up to 1 mm/sec). The temperature profiles are compressed here; temperature variation occurs in a range 1-1.5 mm from the wall (Fig. 2b).

At t_{out} = 101-102°C, there appears a level above which bubbling foam is seen, while below the level there are powerful horizontal fluxes of steam–water mixture from the heated surface into the section and in the lower part of the window there are return fluxes to the heating surface. Inverse motion of the water–steam mixture to the heat-transfer surface is also noted at the unheated side walls.

The results of measuring the water and wall temperature over the height of the channel in one such experiment with P = 0.116 MPa (t_s = 102°C), W = 0.75 mm/sec, q = 78 kW/m², t_{out} = 102°C are shown in Fig. 3. As follows from Fig. 3, boiling appears in the area beginning

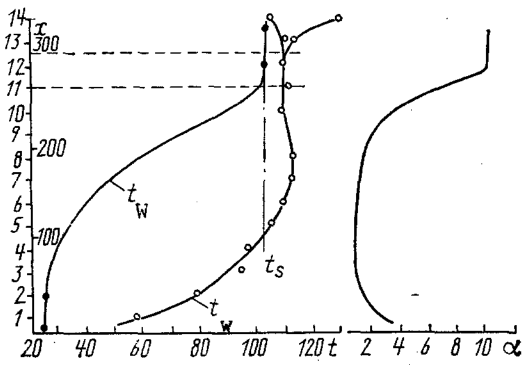

Fig. 3

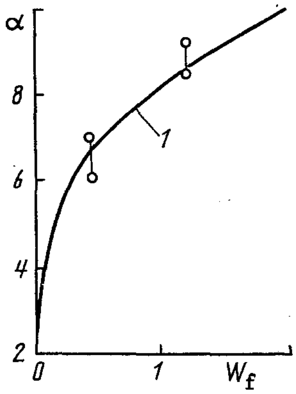

Fig. 4

Fig. 3. Distribution of t_W, t_w, and α over the height of the section with W = 0.75 mm/sec, q = 78 kW/m². The dashed lines show the water−foam level; the figures along the ordinate are the numbers of the cross sections; α, kW/m²·K; x, mm.

Fig. 4. Dependence of the heat-transfer intensity in the foam layer on the theoretical velocity of the steam: 1) data of [7]; W_f, m/sec.

with cross section 5 at Δt_{und} = 70°C, above which the surface temperature is practically unchanged even in the area above the foam level. Reduction in heat-transfer coefficient over the height of the plate in the initial area is associated with the formation of a boundary layer, while in developed boiling the heat-transfer coefficient satisfies the empirical dependence $\alpha = 3q^{0.7}p^{0.15}$ [5].

Analyzing the data on heat transfer in the foam region, it may be noted that it is of high intensity. The temperature profiles under the water−foam level (in the saturated-water region), in the region of the foam layer, and above this layer (in the steam region) are the same (Fig. 2b). Since the processes of bubbling and boiling are analogous [6], data on the heat-transfer intensity in the foam region may be compared with the data of [7], where the foam in a column of dimensions 120 × 120 mm was produced by bubbling air through water and alcohol or glycerine solution. The results of the comparison (Fig. 4) show increase in α with increase in the theoretical velocity of the steam.

The temperature t_W in the upper cross sections only increases with reduction in foam level in the section. Thus, in the experiment with the results shown in Figs. 2b and 3, with a gravimetric steam content of 1.7%, t_W in cross section 14 oscillates from 104 to 135°C. As the level drops to the lower edge of the window, t_W in cross sections 13 and 14, which are 50-60 and 80-90 mm above the foam level, respectively, increases to 216 and 198°C, which indicates cooling of the heat-transfer surface by the weakly heated steam. The heat-transfer coefficients are reduced here to 610 and 730 W/m²·K with a mean steam velocity of ∿1 m/sec.

These results, relating to the bubble boiling of water in a vertical unidirectionally heated channel with theoretical velocities of < 10 mm/sec, demonstrate the determining influence of free motion on the hydrodynamics, suppressing the forced convection; this leads to the development of a closed circulation loop. This circulation in the section facilitates heating of the liquid in the volume and intensifies the heat and mass transfer in the zone of single-phase convection.

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EXPERIMENTAL STUDY OF THERMALLY INDUCED OSCILLATIONS AND HEAT TRANSFER IN AN ASCENDING FLOW OF SUPERCRITICAL HELIUM IN A VERTICAL TUBE

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An experimental study of the conditions for the onset of thermally induced oscillations and their influence on heat transfer in an ascending flow of supercritical helium in a heated vertical tube is reported.

One of the distinctive characteristics of heat transfer near the critical point is the existence of regimes with low- and high-frequency oscillations of the temperature and pressure together with the usual stable regimes. This fact is indicated by the results of numerous experiments with different liquids, including helium at supercritical pressure subjected to pure forced convection in tubes [1-3]. It has been established [2, 3] that the onset of low-frequency temperature and pressure oscillations is associated with the values of the heat input and flow rate and with the position of the pseudocritical temperature T_t within the limits of the heated section of the tube. Oscillations of the wall temperature T_w set in when it attains the value T_t, and pressure oscillations set in when the bulk temperature T_f of the fluid reaches the level T_t [3].

For the determination of the stability thresholds of the regimes, it has been proposed [2] that the parameters $R = (\rho_{in} - \rho_{out})/\rho_{in}$ and $\psi = (\Delta p_1 + \Delta p_{1t})/(\Delta p_2 + \Delta p_{2t})$ be used, where ρ_{in} and ρ_{out} are the densities of the fluid at the inlet and outlet of the heated tube, Δp_1 and Δp_2 are the pressure drops at valves located at the inlet and outlet of the tube, Δp_{1t} is the pressure drop along the section of the tube from the inlet valve to the cross section where the bulk temperature T_f of the fluid is equal to T_t, and Δp_{2t} is the pressure drop along the section of the tube from the cross section with $T_f = T_t$ to the outlet valve. The parameter R characterizes the quantity of heat admitted to the tube and the compressibility (elasticity) of the circulating fluid. The parameter ψ is an analog of the well-known criterion proposed by Petrov for two-phase flows [4].

Despite the considerable number of articles that have been published on the stability problem, it is still not clear how the flow stability is affected by thermogravitation or how the heat transfer is influenced by thermally induced oscillations in the mixed convection of a supercritical fluid. The solution of this problem is important for the design of various cryogenic energy devices. In the present article we report an experimental investigation of the influence of these factors on the thermohydraulic stability and heat transfer in supercritical helium.

For the analysis we used experimental data from [5], along with our own results obtained on an arrangement described in [3], in which certain modifications were introduced. The experiments were carried out in a cryostat filled with liquid helium. After reduction and cooling in a liquid-nitrogen cryostat to $\sim 80°K$, the direct flow of helium from the ramp was first directed into the main heat exchanger, where it was further cooled down to 8-10°K by heat transfer with the reverse helium flow, and then into a heat exchanger immersed in liquid helium, where it was cooled to 4.2-5.2°K. The direct flow at this temperature was admitted into the working section. The reverse helium flow from the working section, prior to

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