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CRITICAL TWO-PHASE FLOW IN LONG CHANNELS

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Results are presented from experimental studies of the critical efflux of a two-phase flow through long channels.

It was established earlier [1] that the effect of compressibility proves decisive in the critical efflux of a single-phase liquid in the region of Reynolds number similitude. In the case of transonic velocity of the flow, the effect of dissipative forces becomes vanishingly small due to turbulence degeneration. This was also confirmed for a two-phase liquid in [2]. It has been shown in several studies devoted to determining the pressure loss in the two-phase flow of a liquid in channels with $l/d \leqslant 800$ that this loss is considerably greater than in a single-phase liquid [3, 4]. At the same time, the experimental data shows that an increase in flow velocity is accompanied by a decrease in the friction coefficient of a two-phase flow [4-6]. This suggests that the dependence of the pressure loss in a channel on l/d in two-phase critical flow is nonlinear in the hydraulic flow regime and that the loss in sufficiently long channels becomes less than the loss in this regime.

An experimental unit was developed to physically model the process of the critical discharge of a saturated liquid through long channels and obtain pressure and flow-rate values along the channel.

Water from pumps was delivered along a pipe to a heat exchanger where it was heated by steam to the appropriate temperature. The liquid proceeded from the heat exchanger along a feed pipe with $\varnothing 57 \times 50$ mm to the working section, which was a tube with $\varnothing 10 \times 8$ mm made of stainless steel. After the working section, the liquid entered a discharge pipe with $\varnothing 57 \times 50$ mm and thence a boiler, where it was cooled and removed from the system. The cross sections of the feed and discharge pipes were 37 times greater than that of the working section, which created conditions whereby the liquid flowed out of and into large volumes.

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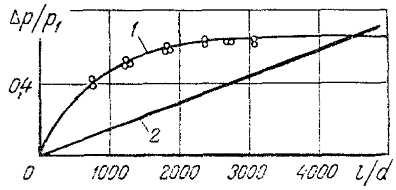

Fig. 1. Dependence of relative pressure loss on relative channel length: 1) two-phase flow; 2) hydraulic flow regime; Δp, static pressure loss; p_1, static pressure at beginning of working section; l, length of channel; d, channel diameter.

The length of the working section was increased by replacing part of the discharge pipe with the 10 × 8 mm diameter working tube. The flow rate of the liquid was measured with a diaphragm, while pressure was measured with standard manometers installed at the end of the feed pipe and at the beginning and end and intermediate points of the working section. Temperature was measured at the end of the feed pipe. The working section was well insulated with asbestos cord to ensure that the process was adiabatic.

The tests were conducted as follows. By regulating the rate of steam flow, we established a certain liquid temperature at the inlet of the working section. Then, accelerating the water flow with a valve, we brought the pressure at the beginning of the working section up to a value corresponding to the saturation pressure at the established temperature, i.e., we fed water close to the saturation state into the working section. The critical flow rate and static pressure along the working section were determined for different l/d. The pressure loss was determined at the same liquid flow rate in the hydraulic regime as in the critical regime. The tests were conducted at different temperatures of the saturated water at the working-section inlet.

The tests conducted on the unit revealed a similitudinous range of unit liquid flow rate and pressure loss with respect to the relative length of the tube. Quantitative characteristics of these quantities were determined. Figure 1 shows experimental data with a temperature of the saturated water at the channel inlet (t_0) of 160, 140, and 120°C. It is apparent from the figure that the near-similitude region begins at $l/d > 2500$ (curve 1), i.e., the character of the dependence of relative pressure loss in the channel on relative length of the tube $\Delta p/p_1 = f(l/d)$ is basically different from the relation for the hydraulic regime (curve 2). At $l/d \leqslant 2000$, the resistance of the two-phase flow is greater than that of the hydraulic flow, exceeding the latter by a factor of two or three at certain sites. This agrees with the data in [2, 3] for x = 0.02. However, the difference decreases with an increase in l/d and at $l/d > 4500$ the pressure loss in the two-phase flow becomes less than in the hydraulic regime. Thus, the advantage of having a two-phase flow increases with an increase in l/d.

The unit mass flow rate j in critical outflow, depending on the channel length (l/d), first decreases and then — when $l/d > 2500$ — increases in proportion to l/d (Fig. 2a). The value of j depends considerably on the saturated water temperature t_s at the channel inlet and, thus, the saturateion pressure p_s (Fig. 2b). The region corresponding to mass vapor contents x from 0.01 (1) to 0.02 (2), occurring at the channel outlet, is hatched in Fig. 2b.

It is interesting to see the change in pressure along the channel, especially on long channels, where similitude is observed. Figure 3 shows the change in pressure in channels with a relative length l/d equal to 3060 (curve 1), 2740 (2), 2350 (3), and 1825 (4), corresponding to the region of the beginning of similitude with respect to l/d at t_s = 160°C. Similar curves were obtained for t_s = 140 and 120°C. It is apparent from the figure that, given the same total pressure loss in the channel, the character of change in pressure along the channel is different and depends on the channel length.

An increase in channel length is accompanied by a decrease in the zone of effect of the critical (outlet) cross section and an increase in the zone where pressure decreases nearly linearly.

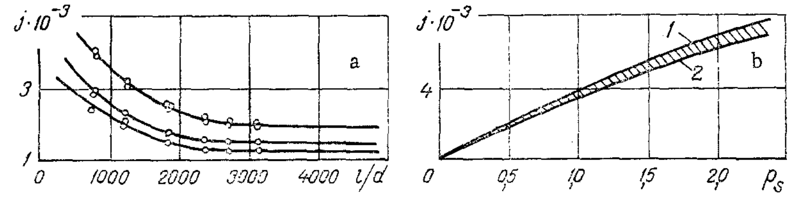

Fig. 2. Dependence of critical unit flow rate j (kg/(m^2·sec)) of liquid on relative length of channel at a temperature t_0 equal to 160 (top curve), 140, and 120°C (a) and on the pressure of the liquid P_s (b). p, MPa.

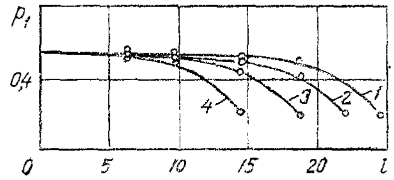

Fig. 3. Change in static pressure p_1 (MPa) along channel length l (m) with different relative channel lengths.

It may be suggested that on the initial section — when friction initiates vaporization of the saturated liquid — the liquid flows with a low mass vapor content in the wall region, and the speed of sound at this vapor content is two orders lower than in the liquid phase. Here, the velocity of the flow approaches the local speed of sound, i.e., motion with transonic velocity takes place in the wall region. At these transonic speeds, the drag is a function not only of the viscosity of the flow, but also of its compressibility — determined by the Mach number ($M = w/a$). At the same time, the speed of sound in the core of the flow is much higher and the average Mach number for the flow is close to zero.

The thickness of the boundary layer increases with further motion along the channel, the total vapor content of the flow increases, and there is a slow increase in the vapor content in the wall region (since most of the vapor formed has a gradient directed toward the core of the flow). The Mach number approaches or even reaches unity. Thus, the friction coefficient decreases, approaching zero [6], and pressure decreases nearly linearly. However, the velocity of the flow increases. Despite the outer similarity to the character of change in pressure in the first zone, with a hydraulic regime, the mechanisms of the process are basically different.

In the second zone, where there is a fairly sharp drop in static pressure, there is an increase in the pressure loss due to flow acceleration Δp_{acc}. The homogeneity of the flow was conserved.

The mechanism of the process requires additional proper experimental investigation focusing on study of the structure of the flow in the boundary layer on long channels and ensuring stability of the process. However, the results already obtained are of interest insofar as studying and realizing several production processes are concerned.

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HEAT EXCHANGE AND HYDRODYNAMICS DURING BOILING ON A HORIZONTAL TUBE BUNDLE IN A FLUIDIZED BED OF SOLID PARTICLES

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The results are given on the experimental study of heat transfer and hydrodynamics with boiling on a horizontal tube bundle located in a dispersed bed of solid particles and of this process hydrodynamics simulation by gas bubbling within the fluid filtration rate variation via the bed $(0\text{-}3.5)v_{l i.f}$. An analysis and correlation of experimental results are presented.

Arranging for boiling to occur under conditions of thermal fluidization [1] makes it possible to prevent the deposition of scale on the heating surfaces of tube bundles as a result of the collision of solid particles with these surfaces [2-4]. The cause of particle motion in this case is filtration through a dispersed layer of vapor generated on the tube surfaces. Since the amount of filtering vapor increases over the height of a bundle, scale removal will be adversely affected on the lower tubes compared to the overlying tubes [4].

One method of equalizing tube operating conditions, together with that proposed in [5], is arranging for boiling of a saturated liquid on the tube bundle under conditions of simultaneous thermal and hydrodynamic fluidization of a disperse bed of particles. The process of boiling on a horizontal tube located in a water-fluidized particle bed was studied in [6, 7]. High degrees of subheating in tests in [6, 7] caused condensation of the vapor phase in the boundary layer of the heating surface. The vapor phase generated here had no effect on hydrodynamic processes in the core of fluidized bed, as is typical for the case of boiling of a saturated liquid [3, 4]. The relations obtained for the conditions in [6, 7] cannot be used to calculate heat exchange at $T_l = T_{sat}$.

Presented below is an analysis and generalization of empirical data on heat exchange and the hydrodynamics of water boiling under the conditions $T_l = T_{sat}$ on a horizontal bundle of tubes located in a disperse bed of spherical particles. The tests were conducted in the range p = 0.05-1 bar and $\bar{q} = 10^4\text{-}1.5\cdot10^5$ W/m². The hydrodynamics of the process were also modeled using gas bubbling. The bundles consisted of a staggered arrangement of stainless steel tubes with ∅ 18/2 . The bundles were heated by condensing vapor. A bundle with a heating surface of 0.192 m² consisted of four vertical and five horizontal rows with relative spacings $S_{hor} = 3.4$ and $S_v = 2.2$. The 0.35-m-high disperse bed was made of particles of glass ($\rho_p = 2.5\cdot10^3$ kg/m³) and agalite ($\rho_p = 2.4\cdot10^3$ kg/m³) with mean diameters of 1.5, 2.75, and $4.75\cdot10^{-3}$ m. The circulating liquid, under conditions of simultaneous thermal and hydrodynamic fluidization, was fed into the bottom half of the bed. The filtration rate in the tests was variable: from 0 to 0.12 m/sec for the liquid, and from 0 to 0.4 m/sec for the vapor at the border of the bed.

The hydrodynamic model was a vertical cross section of a vessel with a tube bundle measuring 1.2 × 0.4 × 0.05 m. The vessel contained a transparent front glass window. The tube bundle was made up of tubes with ∅ 18/0.5 . Holes 0.7 mm in diameter were drilled in the sides of the tubes, 25 holes to each cm². The gas phase in the model was air, while the working liquid was water. The ranges of v_l and v_g in the bubbling tests were the same as in the boiling tests.

Preliminary tests showed that, as in [2-4], the relations $\alpha_i = f(q_i)$ under the conditions studied are the same for all tube bundles if the temperature head, in the determination of α_i, is reckoned from the temperature of the liquid at the level of the i-th horizontal row $\Delta T_{wa_i} = T_{wa_i} - T_{l_i}$, where $T_{l_i} = T_{sat}(p) + \Delta T_{h.d_i}$. It was shown in [3] that the value of $\Delta T_{h.d}$ can be

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