HEAT TRANSFER IN THE SUPERCRITICAL REGION AT LOW PRESSURE AND MASS VELOCITY

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Results are presented from experiments on heat transfer in the supercritical region for pressures p = 0.05-0.3 MPa and low mass velocities ρw = 16.9-168.9 kg/m^2·sec, i.e., for conditions characteristic of emergency and post-emergency regimes of operation of steam-generating equipment in nuclear power plants.

1. Film boiling in channels and assemblies of heat-emitting elements is usually studied at pressures $p \geq 7$ MPa and high mass velocities $\rho w = 10^3$ kg/m^2·sec, which are typical of normal operating conditions for the equipment of atomic power plants. However, in transitional and emergency regimes of operation of the nuclear reactor and steam generator, the steam-generating surfaces are cooled by steam or a steam-water mixture at low mass velocities and reduced pressures. There is an insufficient amount of data on these cases.

The study of heat transfer in the supercritical region has become particular important recently as a result of the development of reliable methods of calculating the thermal-hydraulic characteristics of a steam-generating channel during post-emergency cooling, when the surface temperature reaches 800-1200°K.

Until secondary wetting begins (this period may be tens of seconds for an RBMK and even hundreds of seconds for a water-moderated-water-cooled power reactor [1]), the active zone is cooled by a vapor-drop flow with a higher vapor content $x_0 > 0.8$ and a low mass velocity ρw = 10-100 kg/m^2·sec.

Present methods of calculating post-emergency cooling for this region use heat-transfer coefficients obtained on the basis of relations such as in [2] or [3], where only heat transfer to the vapor component is considered. Correcting for the presence of the drops in essence amounts to allowing for a reduction in the consumption of the vapor component as part of the total consumption of the two-phase mixture. It has also been proposed that the relations obtained by Hsu and Graham [4], Berenson [5], and others for film and transitional boiling regimes in a large volume be used for the region before the beginning of wetting (pre-rewetting zone). This approach has also not been adequately substantiated.

Experimental studies of heat transfer in the supercritical region at low mass velocities were conducted at the Moscow Energy Institute [6, 7], but present knowledge of the processes under discussion and the volume of empirical data accumulated should be considered insufficient.

2. Experimental Unit and Method of Investigation. Figure 1a shows the experimental unit. The main elements of the unit are an electrically heated steam generator 1 of 200-kW capacity, a working section 2, and a condenser 3. The condensate is delivered by a feed pump 4 to the generator. The generator produces and delivers steam of the required temperature T_0 or degree of dryness x_0 to the inlet of the working section. Passing through the working section, the steam enters the condenser. Negative pressure is developed in the latter by a vacuum pump 5. The electric power of the generator is controlled by an RNTO-600 voltage regulator and provides for degrees of vapor dryness x_0 from 0 to 1 and vapor temperatures from 20 to 250°C. Pressure in the working section varies from 0.05 to 0.3 MPa.

Figure 1b shows the working section. A flow of steam from a pipe enters a cylindrical receiver 1 and proceeds through a subsonic nozzle 2 installed in the center of the receiver into a rectangular channel 3. The width of the channel b = 0.035 m, while its length L =

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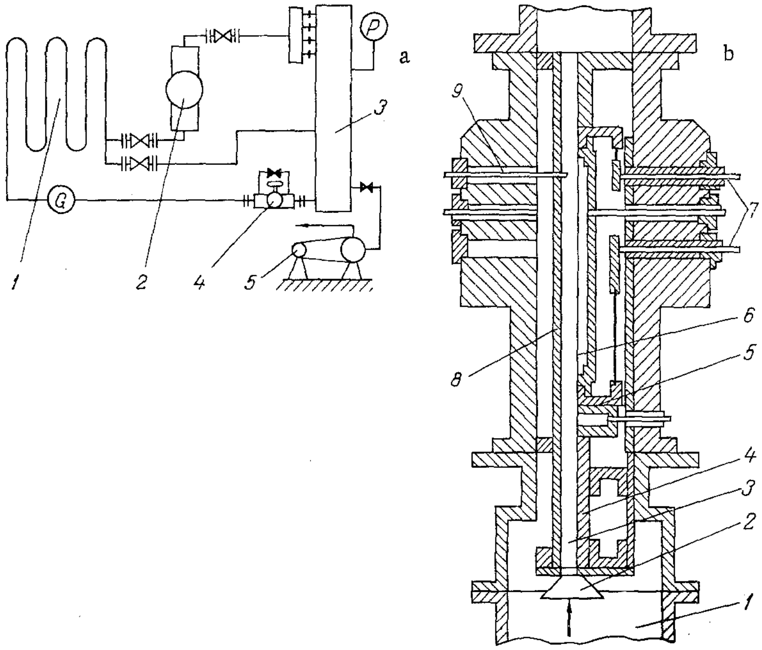

Fig. 1. Experimental unit: a) set-up; b) cutaway of working section.

0.72 m. Tests were conducted with three channel heights h = 0.020, 0.008, and 0.005 m to obtain a variable mass velocity ρw and relative working-section length L/D_h. An unheated section 4 for hydrodynamic stabilization of the flow, of length L_{unht} = 0.287 m, was located at the inlet of the working section. A chamber 5 for slit withdrawal of the film was located at the inlet of the heated section. The Nichrome plate 6 heated by direct current was 0.372 m long. The plate width was less than the channel width to avoid corner effects. The temperature of the Nichrome plate was measured at 25 stations along its length with Chromel–Alumel thermocouples soldered flush with the working surface with a silver solder. The plate was assembled together with copper current leads 7 in a ring made of an insulating material. The opposing, heat-conducting wall 8 contained four drainage holes to measure static pressure, as well as inlets for the thermoprobe 9 and a drop-diameter measuring instrument. Optical windows were located in the side walls.

The test procedure was as follows. We established steady-state conditions for a fixed value of consumption (flow rate) at a steam temperature T_0 at the working-section inlet several degrees higher than the saturation temperature. Wall temperature and static pressure were then measured for several values of heat flux. Then, reducing the power of the steam generator, we established a certain degree of dryness x_0 at the channel inlet. The film formed in the hydrodynamic stabilization section was removed through the slit, so that the heated plate was in the supercritical heat-transfer regime along its entire length. With x_0 = const, we measured all readings for six values of heat flux q_{wa} in the range from 0 to 326 kW/m². The dryness x_0 was reduced until visual observation and wall-temperature readings indicated the presence of wetted zones, in the form of individual rivulets, films, etc. on the heated plate, i.e., until the change from film to transitional boiling occurred. A futher reduction in x_0 led to rewetting along the entire plate length. After wetting of the plate for ρw = const, this series of tests was ended. In all, we conducted twenty series with ρw = 16.9; 23.6; 28.5; 33.8; 42.8; 57.1; 67.6; 71.4; 85.7; 101.4; 135.1; 168.9 kg/(m²·sec). The range of dryness was x_0 = 1.03–0.7. The range of Reynolds numbers, determined according to the hydraulic diameter $D_h = 2bh/(h + b)$, was $(1.07\text{–}16)\cdot 10^4$.

3. Experiments with Superheated Steam. Experiments with superheated steam were conducted to choose, from among the large number of relations for the Nusselt number in a single-phase flow, a formula which agreed the closest without empirical data. We also needed to determine the effect of unidirectional heating.

Figure 2a shows experimental points for heat transfer in the superheated steam for a stabilized flow ($L/D_h > 15$). The physical constants were chosen for the average temperature between the wall temperature and the temperature of the vapor $T_f = (T_{wa} + T_v)/2$.

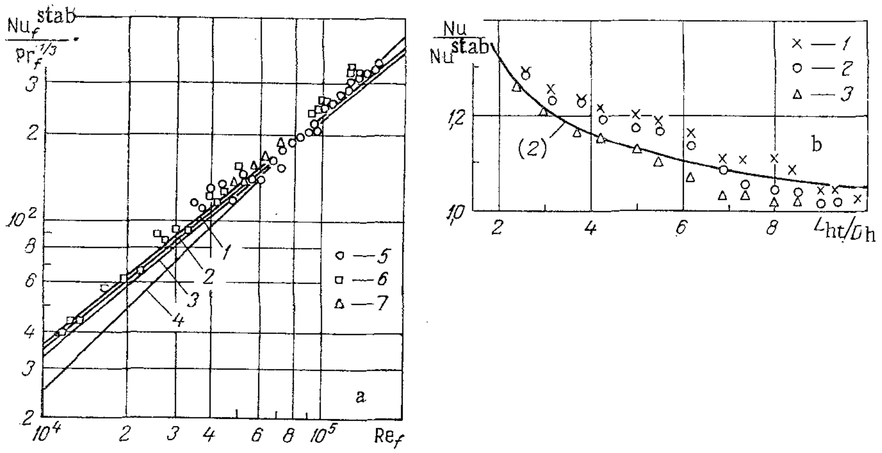

Fig. 2. Experimental data on heat transfer to superheated steam: a) comparison of empirical data with formulas: 1) Colburn [3]; 2) Petukhov [8]; 3) Heineman [9]; 4) Collier and Lacey [10]; 5) h = 0.020 m; 6) 0.008; 7) 0.005 m; b) change in Nu/Nu^{stab} for the thermal initial section; 1) ρw = 28.5 kg/(m²·sec); 2) 57.1; 3) 85.7.

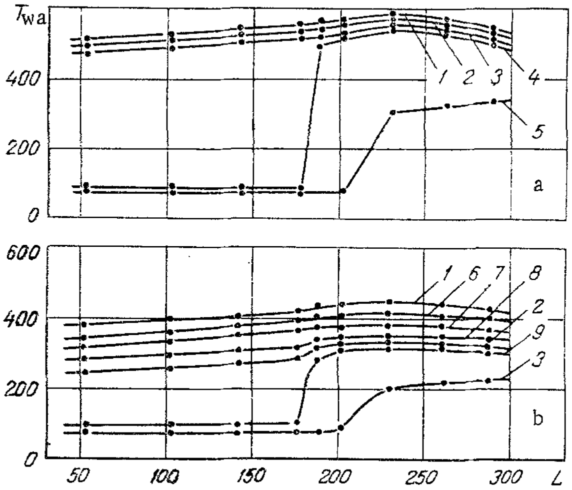

Fig. 3. Results of measurement of wall temperature with $q_{wa} = 123 \cdot 10^3$ W/m² and a variable degree of dryness x_0: a) for ρw = 28.5 kg/(m²·sec); b) for ρw = 71.4 kg/(m²·sec); 1) x_0 = 1.03; 2) 0.93; 3) 0.88; 4) 0.85; 5) 0.816; 6) 0.99; 7) 0.97; 8) 0.96; 9) 0.90. T_{wa}, °C; L, mm.

The experimental points, with a spread of ±22%, are grouped about the theoretical curve of Petukhov [8]

$$Nu_f^{stab} = \frac{\frac{\xi}{8}\mathrm{RePr}}{K_1 + K_2\sqrt{\frac{\xi}{8}}\,(\mathrm{Pr}^{2/3} - 1)}, \tag{1}$$

where $\xi = (1.82 \lg \mathrm{Re} - 1.64)^{-2}$; $K_1 = 1 + \frac{900}{\mathrm{Re}}$ and $K_2 = 12.7$.

The empirical points are described with the same accuracy by the correlations of Heineman [9] and Colburn [3] and with somewhat less accuracy by the formula of Collier and Lacey [10].

3.1. Heat Transfer in the Thermal Initial Section of the Channel. Figure 2b shows the change in the relative Nusselt number Nu/Nu^{stab} in relation to the length of the heated sec-

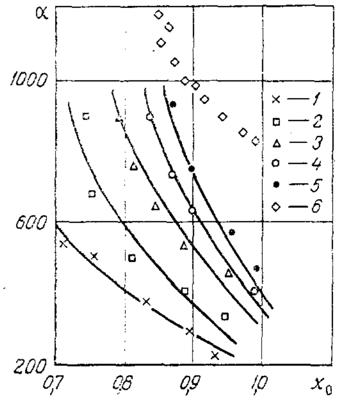

Fig. 4. Experimental data on heat-transfer coefficient for the region of reduced heat transfer: 1) $\rho w = 28.5$ kg/(m²·sec); 2) 42.8; 3) 57.1; 4) 71.4; 5) 85.7; 6) data from [7] for $\rho w = 278$ kg/(m²·sec); curves — calculated from Eq. (7). α, W(m²·°C).

tion L/D_h for a variable mass velocity ρw. It is apparent that the length of the thermal initial section increases with a decrease in ρw from 85.7 to 28.5. However, this increase is slight ($L/D_h = 9.8$ for $\rho w = 28.5$, while $L/D_h = 8.5$ for $\rho w = 85.7$). Figure 2b shows the curve of the equation obtained by Kurganov and Petukhov [11]:

$$\frac{\mathrm{Nu}_{f_0}}{\mathrm{Nu}_{f_0}^{\mathrm{stab}}} = 1 + 0.48\left(\frac{L}{D_\mathrm{h}}\right)^{-0.25}\left(1 + \frac{3600}{\mathrm{Re}\sqrt{\frac{L}{D_\mathrm{h}}}}\right)\exp\left(-0.17\frac{L}{D_\mathrm{h}}\right), \tag{2}$$

which generalizes the empirical points with good accuracy.

4. Experiments with Moist Vapor. Figure 3a shows measurements of wall temperature T_{wa} in the supercritical region on the heated plate with a constant heat flux $q_{wa} = 123 \cdot 10^3$ W/m². The measurements are shown as a function of degree of dryness x_0. It is apparent that an increase in the amount of drop moisture in the flow is accompanied by a monotonic decrease in the wall temperature at all stations along the plate. An increase in mass velocity leads to a reduction in T_{wa}. Meanwhile, whereas the stratification of the $T_{wa}(L)$ curves with an increase in drop concentration in the flow is relatively insignificant for a low mass velocity (Fig. 3a), for $\rho w = 71.4$ kg/(m²·sec) the wall temperature drop is considerably greater (Fig. 3b).

It follows from analysis of the wall temperature data that, given a fixed heat flux, an increase in drop moisture for $x_0 < 1$ leads to an increase in the total heat-transfer coefficient, defined as $\alpha_{tp} = q_{wa}/(T_{wa} - T_s)$.

Figure 4 shows results of experiments to determine the heat-transfer coefficient α_{tp} for the region of reduced heat transfer in the range of mass velocities from $\rho w = 28.5$ to 168.9 kg/(m²·sec). It is apparent that α_{tp} increases with ρw. An increase in the vapor content of the flow also increases the two-phase heat-transfer coefficient α_{tp}.

The increase in the heat-transfer coefficient in the moist vapor compared to the superheated vapor can be attributed to two causes. First, an increase in drop concentration in the flow is accompanied by an increase in the flow of liquid deposited in drop form on the hot plate, equal to

$$N_{\mathrm{wa}} = \varkappa \frac{6(1 - x_0)\rho w}{\rho_3 w_3 \pi d_3^3}. \tag{3}$$

As was shown in [12], even at high wall temperatures — up to $T_{wa} = 500$°C — drops come into contact with the surface, are heated, remain on the surface for tens of milliseconds, and cool the wall as they undergo explosive vaporization. After being reflected (drops are thrown back by the vapor generated at the wall), other drops are returned to the flow. Equation (3) can be used to evaluate the heat flux from the wall to the contacting drops

$$q_{wa_3} = r\varkappa\beta \frac{(1-x_0)\rho w}{w_3}, \tag{4}$$

where β is the drop mass defect during the contact process, ranging from 1.5 to 3% according to [12]. Estimates show that, for the conditions of the present experiments, the heat flux q_{wa_3} is roughly 15-20% of the total heat flux q_{wa}.

A significant role is also played by the reduction in the temperature heat ($T_{wa}-T_s$) due to the reduction in nonequilibrium superheating of the vapor resulting from drop vaporization. In fact, as can be seen from the heat-balance equation

$$\frac{c_p}{r}\frac{dT_1}{dz} = \frac{q_{wa}}{r\rho w h} - \frac{6(1-x_0)\alpha_{vap}(T_1-T_s)}{\rho_3 w_3 d_3 r}, \tag{5}$$

the right side of Eq. (5) contains a term which retards superheating of the vapor component of the flow.

The heat-transfer coefficient for a spherical vaporizing drop surrounded by vapor was determined from Frosling's formula [13]

$$\alpha_{vap} = \frac{\lambda}{d_3}[2 + 0.55\mathrm{Re}_{d_3}^{0.5}\mathrm{Pr}^{1/3}]. \tag{6}$$

For drops 100 μm in diameter $(1-x_0) = 0.2$, and $\rho w = 10^2$, the reduction in superheating along the heated plate is about 18°C compared to $(1-x_0) = 0$. Furthermore, if we assume that the rate of heat transfer from the wall to the vapor component α_{wa_1} remains roughly constant, then a reduction in the temperature of the vapor also means that there is a reduction in the temperature of the wall, i.e., a drop in the temperature head ($T_{wa}-T_s$) and an increase in α_{tp}. Thus, drop vaporization in the superheated vapor next to the wall (internal heat transfer), even without allowance for direct cooling of the wall by falling drops, leads to an increase in the total heat-transfer coefficient.

The experimental data for moist vapor was approximated by the relation

$$\mathrm{Nu}_f^{tp} = \mathrm{Nu}_{f_0}\exp\left[(1-x_0)\left(\frac{\mathrm{Re}_f - 770}{7000}\right)^{0.5} + 1.95\right], \tag{7}$$

which describes the representative empirical points to within 15%. Of course, this accuracy cannot be considered good, but it is significantly better than the accuracy of the approximations in the literature for low pressure and the investigated range of mass velocities.

NOTATION

p, pressure, MPa; ρw, mass velocity, kg/(m²·sec); x_0, degree of dryness at the channel inlet; T, temperature, °C; b, channel width, m; L, channel length, m; h, channel height, m; L/D_h, relative length; D_h, hydraulic diameter, m; T_v, mean vapor temperature, °C; T_f, average temperature between wall temperature and vapor temperature, °C; Nu_v and Nu_f, Nusselt numbers at the temperatures T_v and T_f; Re_v and Re_f, Reynolds numbers at the temperatures T_v and T_f; Pr_v and Pr_f, Prandtl numbers at T_v and T_f; q, heat flux, W/m²; ξ, drag coefficient; α_{tp}, heat-transfer coefficient in the two-phase flow, W/m²·deg; ρ_3, density of drops, kg/m³; w_3, velocity of drops, m/sec; d_3, diameter of drops, m; ϰ, deposition velocity, m/sec; β, drop mass defect; q_{wa_3}, heat flux from wall to contacting drops, W/m²; c_p, specific heat, J/kg·deg; r, specific heat of vaporization of liquid, kJ/kg; α_{vap}, heat-transfer coefficient for drops, W/m²·deg; Indices: unht, unheated; h, hydraulic; wa, wall; stab, stabilized; tp, two-phase; vap, vaporization.

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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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