HEAT TRANSFER IN PLUG REGIME OF FILM BOILING OF HYDROGEN IN PIPES

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The article presents the results of the experimental investigation of heat transfer. It contains generalizing dependences that take the real values of phase slip and heating of liquids into account.

Heat transfer in film boiling of hydrogen was investigated in [1, 2]. Graham et al. [1] generalized the experimental data with the aid of a homogeneous model of two-phase flow disregarding the real structure of this flow. Yarkho [2] studied the disperse regime of film boiling. The plug regime of film boiling was practically not investigated with hydrogen in any of the known works. The few experimental data from [1] that, as a result of analysis, could be associated with the plug regime of flow are not a sufficient basis for obtaining generalizing dependences on heat transfer with film boiling of hydrogen in plug regime of flow. If we compare these data with the results of calculation of heat transfer according to the model of plug regime of flow adopted in [3, 4], we find a substantial difference between experiment and calculation. Regardless of the fact that the generalization of the experimental data obtained for nitrogen was carried out in [3, 4] using a univariate physical model of the process and running parameters of two-pahse flow, the difference in heat transfer between the experiment with hydrogen [1] and the calculation [3, 4] attained 200-600%. An analysis revealed that such a difference had the following causes.

1. In generalizing the experimental data in [3, 4] using the univariate model described by the equation of conservation, it was taken that the speeds of vapor and liquid in any cross section are equal, i.e., that the magnitude of slip $u_v/u_l = 1$.

2. In the distribution of the heat removed from the pipe wall, it was assumed that the heat flow for heating the liquid phase (if $T_l < T_s$) can be determined by the formula describing the heat transfer to the liquid with flow in the pipe, or by a formula obtained from a series of experiments. This series contained experiments in which high mass velocities and high subcooling ($T_s - T_l$) of the liquid nitrogen were attained. It was assumed that with the above conditions, the bulk of the heat is used for heating the liquid, and that the heat flow used for evaporation may be neglected.

The formula obtained with such an assumption was then used for calculating the heat transfer in all regimes.

With a view to this and to obtain substantiated dependences regarding heat transfer to hydrogen in plug regime, we carried out experiments involving the cooling of a pipe with 70 mm diameter made of steel 1Kh18N9T. In the process of cooling we recorded the following parameters: flow rate, pressure, temperature of the pipe wall in ten cross sections, temperature of the core of the flow in four cross sections. A radioisotope method was used to determine the volumetric vapor content in two cross sections of the pipe, and this enabled us, in finding the true parameters of two-phase flow, to abandon the hypothesis of the constant magnitude of slip. At the inlet to the pipe we ensured zero initial conditions for volumetric and mass vapor content by thermostating the supply branch pipe, the cutoff valve, the stabilizing section, and the turbine flow meters. To reduce the heat inflow from the environment, the cooled pipe was placed in a vacuum jacket whose inner cavity was continuously evacuated by a jet pump during the experiment.

All parameters were recorded on magnetic tape with subsequent computer decoding.

The primary processing of the experimental data, i.e., calculation of the temperature of

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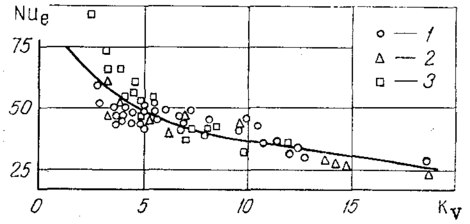

Fig. 1. Dependence of the dimensionless heat-transfer coefficient on the dimensionless temperature head in boiling under conditions of natural convection: 1) d = 12 mm; 2) d = 20 mm; 3) plate [6].

the inner pipe wall from the results of measuring the temperature of the outer wall and calculation of the heat flows removed from the inner wall, was carried out by known equations [4].

In studying heat transfer under conditions of forced flow, it is generally accepted, e.g., [5], to generalize the experimental data in the form of the dependence of the ratio of the dimensionless heat-transfer coefficient and the heat-transfer coefficient realized under conditions of natural convection. However, in the literature there are known to be only data concerning film boiling of hydrogen on a plate [6]. The generalizing dependences obtained for other liquids, when used in calculations, yield results that differ substantially from each other. Therefore, to obtain a basic dependence that would make it possible to calculate heat transfer in natural convection, we carried out experiments to investigate film boiling of hydrogen on vertical pipes made of steel Kh18N10T under conditions of natural convection. The installation made it possible to heat the pipes with electric current and to measure the pressure, the temperature of the liquid in the tank in three cross sections at different height, and the wall gemperature of the pipe in four cross sections. The experiments were carried out with pipes of 20 and 12 mm diameter, 570 mm long.

The obtained experimental data together with the data of [6] were generalized by the correlation

$$\mathrm{Nu_e} \equiv q l_{\mathrm{cr}}/(T_{\mathrm{c}} - T_{\mathrm{s}})\,\lambda_{\mathrm{v}} = 10\,[1 + 4.7\ \exp(-0{,}06\mathrm{K_v} - 0{,}013/\mathrm{K_v} + 1{,}9/\mathrm{K_v^2})]. \tag{1}$$

The dependence (1) is presented in Fig. 1, and it is correct in the following range of regime parameters: P/P_{cr} = 0.08-0.62, K_l = 0-0.20, K_v = 2-19. In generalizing the experimental data, we used as characteristic dimension the critical length of the capillary wave developed on the phase interface in film boiling. As determining temperature we took the arithmetic mean of the wall temperature and of the saturation temperature.

In the analysis of the experimental data on film boiling with forced flow we sought the dependence of the ratio of the experimental values of the dimensionless heat-transfer coefficients and the dimensionless heat-transfer coefficients calculated by formula (1) with the same pressures and dimensionless temperature heads. The true parameters of two-phase flow were determined from the solution of the univariate system of equations of conservation for each phase. The solution was carried out using the experimental data on the density of the heat flows, the heating of the liquid phase, and the volumetric vapor contents. The mass vapor content was determined by integrating by sections of the pipe using the theoretical values of its gradient which were calculated by the formula

$$\frac{dx}{dz} = \frac{\pi d q_{\mathrm{c}}}{G(r + i_l - i_{ls})} - \frac{(1-x)}{(r + i_l - i_{ls})}\,\frac{di_l}{dz}. \tag{2}$$

As a result of an analysis it was discovered that in the region of Reynolds numbers $Re_{lcr} \leqslant 1.59 \cdot 10^5$, the heat transfer in plug regime of film boiling of hydrogen in forced flow is self-similar with respect to all hydrodynamic factors and can be described by formula (1). This can be explained by the fact that the principal influence on heat transfer is exerted by the state of the superheated near-wall layer of vapor whose thickness is much smaller than the thickness of the vapor film. With low mass velocities of flow corresponding to $Re_{lcr} \leqslant 1.59 \cdot 10^5$, the processes occurring on the phase interface do not have a substantial effect on the state of the superheated near-wall vapor layer, and therefore the heat transfer within

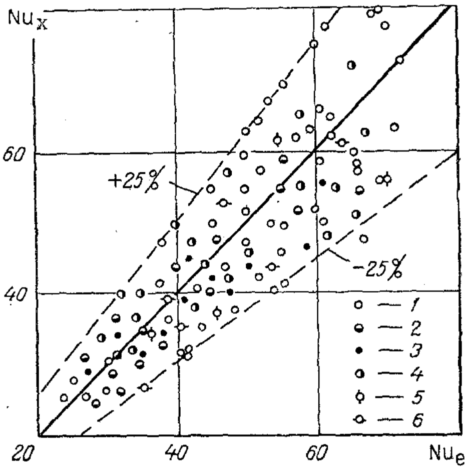

Fig. 2

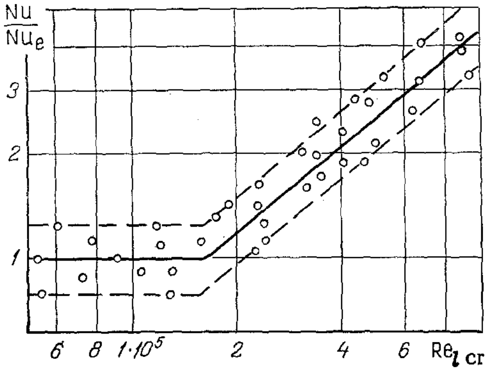

Fig. 3

Fig. 2. Ratio between heat transfer in forced flow with low mass velocities ($Re_{l cr} \leqslant 1.59 \cdot 10^5$) and heat transfer in boiling under conditions of natural convection: 1) d = 70 mm, upflow; 2) d = 70 mm, downflow; 3) d = 39 mm; 4) d = 18 mm; 5) d = 9.5 mm, downflow; 6) d = 12 mm, downflow.

Fig. 3. Dependence of the ratio of the dimensionless heat-transfer coefficients on the Reynolds number (X = 0.017-0.027; b < 0.07 in upflow in a pipe with d = 70 mm).

the limits of the accuracy of measurements does not differ from the heat transfer under conditions of natural convection. The ratio between the heat transfer in forced flow in the mentioned range of Reynolds numbers and the heat transfer under conditions of natural convection is shown in Fig. 2. With Reynolds numbers $Re_{l cr} > 1.59 \cdot 10^5$, the value of Nu/Nu_e increases, other conditions being equal, with the increase of the above criterion, and it is proportional to its values to the power of n = 0.8. The dependence of the ratio Nu/Nu_e on the Reynolds number is shown in Fig. 3.

The presented dependences can be explained in the following manner.

On the surface of the liquid plug, capillary waves form whose ridges may interact with the near-wall layer of the superheated vapor if the vapor film is very thin. However, when the mass velocity of the motion is low, the film is fairly thick and the wave ridges cannot interact with the near-wall layer whose state determines the heat transfer. In this case the dimensionless heat-transfer coefficients within the accuracy of the measurements do not depend on all the factors other than the dimensionless temperature head. With increasing mass velocity the thickness of the film decreases, and drops becoming detached from the surface of the liquid plug, which have large initial kinetic energy, come close to the pipe wall. With $Re_{l cr} > 1.59 \cdot 10^5$, the heat transfer begins to be substantially affected both by the interaction of the wave ridges with the near-wall layer of superheated vapor and by the bombardment of the near-wall layer by drops that become detached.

Analysis also showed that the heat transfer is substantially affected by the intensity of the evaporation from the surface of the liquid plug. For generalizing the experimental data, we used as the criterion X, characterizing this process, the magnitude of evaporation on the section of the liquid plug whose length is equal to the critical length of the capillary wave. Other conditions being equal, with increasing criterion $X = l_{crs} \cdot dx/dz$, the heat transfer increases. This fact can be explained by increased turbulence of the vapor film when the intensity of evaporation increases. In the range of high mass velocities ($Re_{l cr} > 1.59 \cdot 10^5$), the heat transfer is affected by the thickness of the vapor film. With increasing thickness of this film the heat transfer decreases because the thermal resistance of the film increases, and the probability of interaction of the wave ridges with the near-wall vapor layer decreases.

The results of the experiments carried out with a pipe of 70-mm diameter, and, as it turned out, including the measurements of the volumetric vapor content and heating of the liquid phase, and also the results of the investigations carried out by the present authors

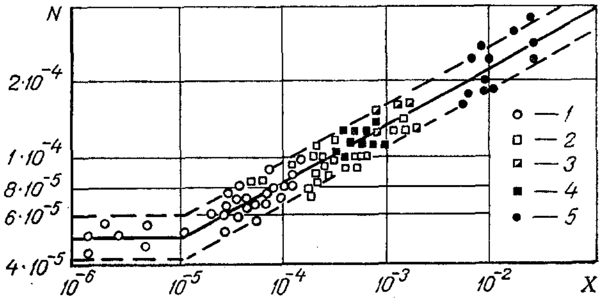

Fig. 4. Dependence of the heat transfer in forced flow with high mass velocities ($Re_{l cr} > 1.59\cdot10^5$) on the intensity of evaporation: 1) d = 70 mm, upflow; 2) d = 70 mm, downflow; 3) d = 39 mm; 4) d = 9.5 mm, downflow; 5) d = 7.92 mm [1].

$$X \equiv l_{crs}\cdot dx/dz;\ \mathrm{N} \equiv \frac{\mathrm{Nu}\exp[2{,}3\,(1-\sqrt{1-\varphi})]}{\mathrm{Nu_e}\,(\mathrm{Re}_{l\,cr})^{0,8}}.$$

with pipes of 39, 18, and 9.5 mm diameter without measurements of the volumetric vapor content and heating of the liquid phase, are generalized by the formula

$$\mathrm{Nu/Nu_e} = 6{,}03\cdot10^{-4}\mathrm{Re}_{l cr}^{0,8}\,X^{0,22}\exp[-2{,}3\,(1-\sqrt{1-\varphi})]. \tag{3}$$

This same formula also generalizes the experimental data taken from [1]. The scatter of all experimental points relative the the dependence (3) does not exceed ±23%.

To generalize the experimental data without direct measurements of the volumetric vapor content in solving the system of univariate equations as the closure dependence describing the phase shift, we used the formula

$$\bar{u} = A\,\frac{Gl_{crs}}{F\mu_{vs}}\,x^{0.75}, \tag{4}$$

where for upflow $A = 0.355\cdot10^{-4}$, for downflow $A = 0.245\cdot10^{-5}$. The heat transfer to the liquid phase was calculated by the formula

$$q_l = 0{,}49q_c\,[1 + 2{,}45\mathrm{K}_l - \exp(-80\mathrm{K}_l^{0,8})]. \tag{5}$$

Formulas (4) and (5) were obtained by generalizing the results of measurements of the volumetric vapor content and heating of the liquid phase in hydrogen flow through a pipe of 70-mm diameter.

Expression (3), which generalizes the experimental data on heat transfer, is correct for conditions of steady-state and non-steady-state cooling of pipes in downflow and upflow within the following range of changes of the regime parameters:

$$P/P_{cr} = 0{,}16-0{,}65;\ \mathrm{Re}_{l\,cr} = 1{,}59\cdot10^5-1{,}6\cdot10^6;\ \mathrm{K}_{l} = 0-0{,}21;\ b = 0{,}002-0{,}3;$$

$$\mathrm{K_v} = 2{,}0-8{,}6;\ z/d = 0{,}25-70.$$

It can be seen from Fig. 4 that with $X < 1.2\cdot10^{-5}$, heat transfer is self-similar with respect to this criterion. Analysis shows that the self-similarity with respect to X is realized with thin vapor films, $b \leqslant 0.03$. In this case, the formula describing heat transfer can be simplified and presented in the form

$$\mathrm{Nu/Nu_e} = 5\cdot10^{-5}\mathrm{Re}_{l\,cr}^{0,8}. \tag{6}$$

The obtained empirical expressions (3) and (6) on heat transfer in plug regime of film boiling of hydrogen make it possible to calculate the cooling of pipes by using a system of univariate equations of conservation, written for each phase separately, and the closure dependences (4) and (5) by phase shift and heat transfer to the liquid.

NOTATION

$b = 0.5\,(1-\sqrt{1-\varphi})$, dimensionless thickness of the vapor film; c_p, specific heat for p = const; d, diameter of the heat transferring pipe surface; i, specific enthalpy; $l_{cr} = 2\pi\sqrt{\sigma/g(\rho_l-\rho_v)}$, critical length of the capillary wave; $K_v = c_{pv}\,(T_c-T_s)/r$, dimensionless temperature head; $K_l = c_{pl}(T_s - T_l)/r$, dimensionless subcooling; q, density of heat flux; r, heat of evaporation; F, area; G, mass flow rate per second; $Re_{lcr} = \rho_l u_l l_{cr}/\mu_l$, Reynolds number of the liquid phase; T, temperature; u, velocity; $\bar{u} = u_v/u_l$, ratio of the mean mass velocities of the phases (slip); z, axial coordinate; δ, thickness of the vapor film; λ, heat conductivity; μ, dynamic viscosity; ρ, density; σ, surface tension; φ, volumetric vapor content; x, mass vapor content. Subscripts: e, under conditions of natural convection; l, liquid; v, vapor; s, saturation; c, wall; 0, in the initial cross section; x, experimental value.

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EFFECT OF POROUS COATINGS ON HEAT EXCHANGE IN FILM BOILING

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The work concerns the investigation of heat exchange in film and transient boiling of water and Freon-113 on a surface with sintered porous metal coatings.

Film boiling is widely used in engineering. The heated surfaces on which film boiling occurs may have a porous coating that was specially applied or originated as a result of deposits. Oxide films and badly heated conducting coatings with different structures lead to increased values of q_{cr_2} and ΔT_{cr_2} [1-3] and of the heat transfer coefficient [4-6].

Zhukov et al. [5] investigated the effect of a porous layer of Al_2O_3 on heat transfer in film boiling of Freon-113 on a horizontal copper disk at atmospheric pressure. It was established that aluminum oxide applied by gas-flame spray coating causes the heat-transfer coefficient to increase to twice its value. Nikolaev and Tokalov [6] obtained data on the effect of a nickel coating 0.3 mm thick, applied electrolytically, on the heat exchange upon boiling of carbon dioxide. It was discovered that at a pressure of 71 bar, a porous layer in film boiling causes α to increase by a factor of 1.5. However, the effect of coatings on heat transfer in film boiling in the range of ΔT higher than ΔT_{cr_2} requires further study because, according to [1, 3], it does not manifest itself, and according to the data of [4-6] it exists.

The object of the present work is to investigate the effect of sintered coatings on the heat exchange in transient and film boiling. The data on heat transfer were obtained in cooling heated spheres coated with layers of various metals. According to [7], when the rate of change of the mean temperature of the section is less than 200°C/sec, the boiling may be regarded as a quasisteady-state process. This condition was met during the present work.

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