HEAT TRANSFER IN THE FLOW OF A SINGLE-PHASE AND BOILING COOLANT IN A CHANNEL WITH A POROUS INSERT

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The results of an experimental investigation of heat transfer, hydraulic resistance, and critical heat fluxes for the flow of coolant in a channel with a square cross section and porcelain inserts consisting of highly porous permeable cellular materials, are presented.

A great deal of attention has been devoted in the last few years to heat and mass transfer in porous media. One way to intensity heat transfer and increase the critical heat-flux density is to realize flows of boiling liquids in porous structures. Quite a large number of experimental investigations of heat transfer accompanying boiling of coolant in channels and pipes with porous inserts has been published [1-3]. In the overwhelming majority of these works, cellular, fibrous, brush-like, sintered, and powdered structures were studied.

Using a coolant in different heat-exchange systems with phase transformations appears to be very promising. This includes cooling systems for elements of high-power optics [4], highly porous permeable cellular materials (HPCM) [5], which have the highest permeability of known porous materials (the coefficient of permeability is equal to 10^{-8}-10^{-9} m^2). HPCM have a characteristic three-dimensional reticular-cellular structure with high cell repetition. All of the framework material is concentrated in the rib-crosspieces which bound a separate structural cell.

When a boiling coolant flows in a channel with a porous insert, energy transfer from the heated wall deep into the channel occurs as a result of heat conduction in the framework material, phase transformations, and convective transverse mixing of the flow.

In this paper we study a situation in which heat transfer is determined completely by the hydrodynamics of vapor-liquid flow in pores, and the contribution of the thermal conductivity of the framework to heat transfer is negligibly small.

The inserts* consisted of porcelain HPCM (0.03 × 0.01 × 0.01 m) with low thermal conductivity. The thermal conductivity of the framework of the HPCM is described [6] by the formula $\lambda_{fw} = 0.2\lambda_0(1 - \varepsilon)$, where $\lambda_0 = 1.4$ W/(m·K) is the thermal conductivity of porcelain. The main characteristics of the inserts are presented in Table 1.

A diagram of the experimental apparatus employed in this investigation is presented in Fig. 1. The working liquid (tap water, which is heated to a temperature below the saturation point (underheated), $\Delta T_{un} = 86$°C) was fed into the working section 3 through a fine filter 1 and a floating flowmeter 2. The working section is a square channel with cross section 0.01 × 0.01 m and 0.05 m long and has quartz-glass walls. In the experiments a nichrome element 4 of the wall was heated on one side by means of a resistance heating. The outer side of the heated wall was thermally insulated. An insert made of HPCM was pressed against the heating element. The use of resistance heating did not lead to volume heating in the porous insert because the framework material does not conduct electricity.

* The inserts were prepared at the Republics Engineering-Technical Center for Powder Metallurgy (in the city of Perm').

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TABLE 1. Characteristics of Inserts made of HPCM

No. of insert	ε	$d_p\cdot 10^3$, m	$d\cdot 10^3$, m	$\gamma\cdot 10^{-7}$, m^{-2}	$\beta\cdot 10^{-3}$, m^{-1}	$\lambda_{fw}\cdot 10^2$, W/(m·K)
1	0,83	1,75	0,4	5,3	2,3	4,8
2	0,92	1,1	0,2	8,1	1,2	2,2
3	0,86	3	0,6	1,5	0,86	4

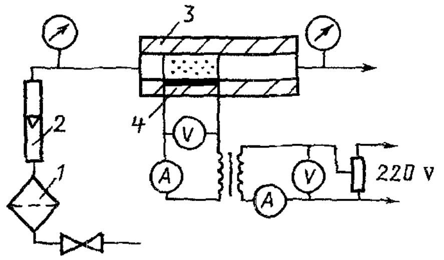

Fig. 1

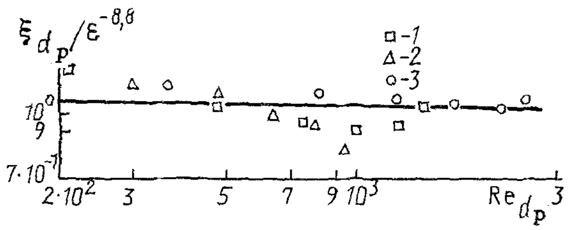

Fig. 2

Fig. 1. Diagram of the experimental apparatus.

Fig. 2. Correlation of the experimental data on the hydraulic resistance of a channel with inserts made of HPCM under conditions of a developed flow: 1) insert No. 1; 2) No. 2; 3) No. 3.

The construction of the experimental apparatus permitted smooth regulation of the heat flux over the rnage $q = (0.08\text{-}2)\cdot 10^7$ W/m^2. The mass flow rate of the coolant varied in the interval $G = (14\text{-}100)\cdot 10^{-3}$ kg/sec.

When the experiments were performed the flow rate of the coolant, the parameters of the electric current, the temperature of the liquid at the inlet and outlet, and the pressure drop on the insert were measured. The temperature of the heated section of the wall was measured in three sections along the channel on the outer surface on the side of the thermal insulation. The temperature on the inner heating surface was determined based on the solution of the one-dimensional stationary problem of heat conduction in the presence of an internal heat source.

The hydraulic resistance of the channels with the inserts was calculated from the pressure drop Δp on the insert and the flow rate of the liquid and was determined in the form of the coefficient of the hydraulic resistance [7]

$$\xi_{d_p} = 2\frac{\Delta p}{L}\frac{d_p\varepsilon^2}{\rho v_f^2}. \tag{1}$$

The experimental data on ξ_{d_p} (Fig. 2) agree with the criterional dependence

$$\xi_{d_p} = \frac{40}{Re_{d_p}}(1 + 2,5\cdot 10^{-2}\varepsilon^{-8,8}\, Re_{d_p}), \tag{2}$$

which describes the relation between the coefficient ξ_{d_p} and Reynolds number with transitional and turbulent flow of liquid in a channel with an insert in the range Re_{d_p} = 200-3000.

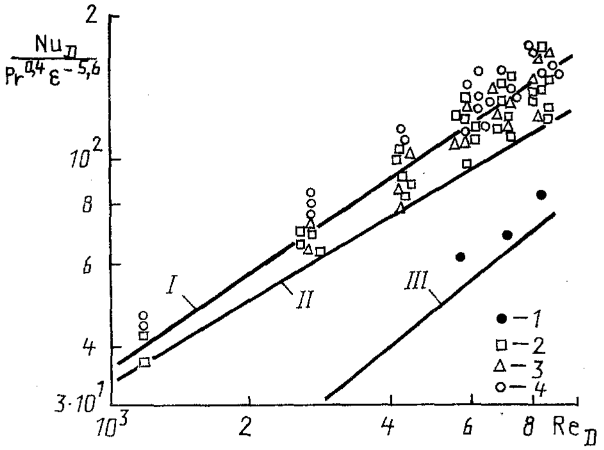

Fig. 3. Experimental data on convective heat transfer accompanying flow in a channel with porous inserts: 1) channel without inserts; 2) insert No. 1; 3) No. 2; 4) No. 3; I) calculation using Eq. (3); II) calculation using Eq. (4); III) empty channel, calculation using the relation $Nu_D = 0.023Re_D^{0.8}Pr^{0.4}$.

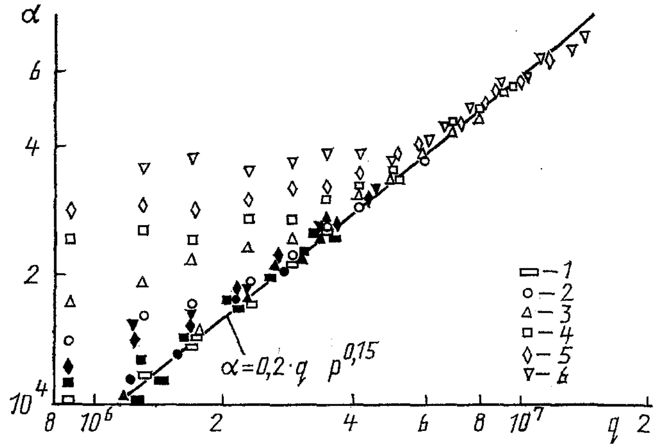

Fig. 4. Coefficients of heat transfer under conditions of boiling of water in a channel with an HPCM insert: white dots refer to the insert No. 2; black dots refer to a channel without inserts; the velocity is equal to 0.14 (1), 0.32 (2), 0.5 (3), 0.68 (4), 0.86 (5), and 1 (6) m/sec; α_{boil}, $W/(m^2 \cdot K)$; q, W/m^2.

The experimental data on heating under conditions of single-phase convection for the inserts investigated (Fig. 3) are generalized by the dependence

$$Nu_D = 0{,}35\, Re_D^{0,65}\, Pr^{0,4} \left(\frac{Pr}{Pr_{w\,l}}\right)^{0,14} \varepsilon^{-5,6}. \tag{3}$$

The physical properties of the liquid were determined at T_{ℓ_0}. The main parameters were varied in the following ranges: $Re_D = 10^3$-10^4; ε = 0,83-0.92; Pr/Pr_w = 1.4-8.4.

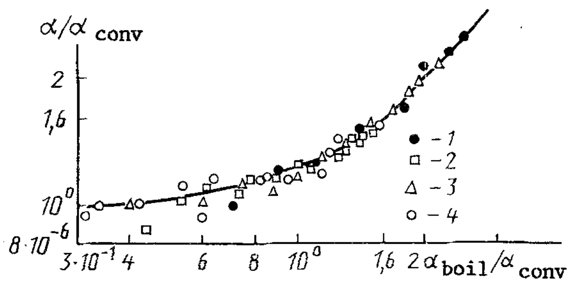

Fig. 5

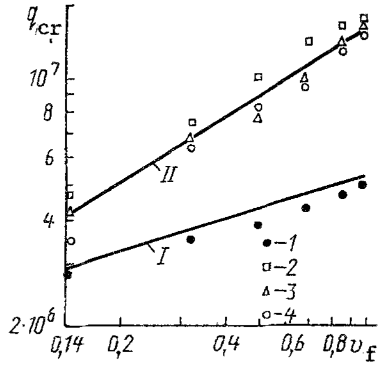

Fig. 6

Fig. 5. Comparison of the computed dependence (7) with the experimental data: 1) channel without insert; 2) insert No. 1; 3) No. 2; 4) No. 3.

Fig. 6. The effect of the filtration velocity on q_{cr}: 1) channel without insert; 2) insert No. 1; 3) No. 2; 4) No. 3; I) calculation using Eq. (8) for a channel without an insert; II) calculation using Eq. (8) for a channel with an insert. q_{cr}, W/m^2; v_f, m/sec.

In [8], the following dependence was obtained to calculate Nu_D analytically:

$$\mathrm{Nu}_D = \mathrm{Pe}^{0,5}\left(\frac{\lambda_e}{\lambda}\right)^{0,5}\left(\frac{D}{L}\right)^{0,5}. \tag{4}$$

Here λ_e is the effective thermal conductivity of the coolant, which was determined from the experimentally determined values of T_w, T_{l_0}, q, and v_f from the expression

$$\bar{T}_w - T_{l_0} = q\sqrt{\frac{2L}{\rho c_p v_f \lambda_e \pi}}, \tag{5}$$

obtained by integrating over the range (0, L) the solution of the equation of the one-dimensional stationary distribution of the temperatures of the framework and liquid with the corresponding boundary conditions.

Satisfactory agreement between the calculations using Eq. (3) and (4) for small numbers Re_D becomes somewhat worse as Re_D increases. This disagreement is apparently caused by the fact that as the number Re_D increases the velocity and temperature profiles become increasingly more filled, and their approximation with cubic polynomials [8] becomes more approximate.

Comparative analysis of data on heating in a channel with and without an insert leads to the conclusion that under conditions of single-phase convection in a channel with a porous thermally nonconducting insert the coefficients of heat transfer are on the average 2-2.3 times higher than those of an empty channel.

In the case of flow of a boiling coolant (Fig. 4), for small values of q and in each velocity regime the coefficients of heat transfer do not depend on the magnitude of the heat flux. These are regimes of single-phase convection, and here the intensity of the heat transfer can be calculated from the relations (3) and (4) recommended above. As the heat flux intensifies the effect of vapor formation on heat transfer increases, but the effect of the velocity remains. As q increases further there obtains a regime of developed bubble boiling on the wall, where the intensity of heat transfer is determined entirely by the vaporization process. As the filtration velocity increases, in order to achieve a regime of developed bubble boiling higher heat fluxes are required, since the intense mixing of the liquid must be compensated. Least-squares analysis of the experimental data for this region showed that the data can be described by a single dependence for all inserts studied:

$$\alpha_{\mathrm{boil}} = 0{,}2q^{0,8}p^{0,15}. \tag{6}$$

The following interpolation formula was proposed in [9] in order to generalize the experimental results on heat transfer to the case of the combined effect of the filtration velocity and heat-flux density:

$$\frac{\alpha_{\Sigma}}{\alpha_{\mathrm{conv}}} = \left(1 + \left(\frac{\alpha_{\mathrm{boil}}}{\alpha_{\mathrm{conv}}}\right)^2\right)^{1/2}, \tag{7}$$

where α_{Σ} is the coefficient of heat transfer to the forced flow of the boiling liquid; α_{conv} is the coefficient of heat transfer in the absence of boiling; α_{boil} is the coefficient of heat transfer in the case of developed bubble boiling, when the velocity has no effect.

When the experimental values of the coefficients of heat transfer are compared with the values calculated using Eq. (7) in the case of the combined effect of the filtration velocity of the coolant and the heat flux density good agreement obtains (Fig. 5) in the entire range of parameters indicated above; this shows that the interpolation used is legitimate. However it should be noted that the relation (7) gives satisfactory results when only an insignificant part of the liquid evaporates in the process of boiling and the volume vapor content of the flow does not exceed 70%. The measurements performed in this work by the method of electric conduction showed that for the realized underheatings of the coolant with respect to the saturation temperature and the quite short length of the heated section, the true volume vapor content of the flow ranged from 0 to 30%.

When the heat load is further increased and the critical heat flux density on the heated surface of the channel is reached the film regime of boiling is established abruptly. This regime was recorded in the experiments based on the sharp increase in the temperature of the wall and subsequent burning through of the wall. The results of the measurements show that the critical heat flux density is proportional to the filtration velocity of the coolant in the first step. The maximum critical heat flux density was obtained on the insert No. 1, which has the highest hydraulic resistance and the highest coefficients of heat transfer.

A series of control experiments in a channel without an insert were performed. The critical heat fluxes accompanying boiling with underheating in a channel with inserts is 2.5-3 times higher than the values achieved in an empty channel (Fig. 6).

The following well-known relation [9] was used to correlate the experimental data on the critical heat fluxes:

$$q_{\mathrm{cr}} = (k_1 r \sqrt{\rho_{\mathrm{v}}}\sqrt[4]{g\sigma(\rho_{\ell} - \rho_{\mathrm{v}})} + k_2 r \sqrt{\rho_{\ell}\rho_{\mathrm{v}}}v_{\mathrm{f}})\left(1 + 0{,}1\left(\frac{\rho_{\ell}}{\rho_{\mathrm{v}}}\right)^{0,75}\frac{c_{p\ell}\Delta T_{\mathrm{un}}}{r}\right), \tag{8}$$

where k_1 and k_2 are empirical coefficients, and in addition k_1 is the criterion of stability of two-phase boundary-layer flow under conditions of free convection in a large volume of boiling liquid and k_2 is the criterion of stability of the two-phase boundary layer under conditions of forced convection.

The experiments showed that for a channel without inserts $k_1 = 0.057$ and $k_2 = 0.01$; for a channel with inserts $k'_1 = 0.059$ and $k'_2 = 0.04$; for these values Eq. (8) describes well the results of direct measurements of q_{cr} (Fig. 6). Analysis of the obtained results shows that in view of the low thermal conductivity of the framework the HPCM studied do not affect appreciably heat transfer under conditions of free convection ($k_1 \approx k'_1$); on the contrary, they interfere strongly with the process of heat removal under conditions of forced convection ($k'_2 > k_2$) owing to effective mixing of the coolant flow in the porous framework.

Thus in this paper we have presented for the first time the direct and generalized results of an experimental investigation of heat transfer and hydraulic characteristics of thermally nonconducting HPCM.

NOTATION

ε, porosity; d_p, pore diameter; d, diameter of the bridges; γ and β, viscosity and inertial coefficients of hydraulic resistance; λ_{fw}, λ_0, and λ, thermal conductivities of the framework, the material of the bridges, and the coolant; λ_e, effective thermal conductivity of the coolant; q, heat flux density; G, mass flow rate of the coolant; T_{ℓ_0}, temperature of the liquid at the channel inlet; $\bar{T}_W = 1/L \int_0 T(\chi)d\chi$, wall temperature averaged over the length of the channel; $\alpha_{conv} = q/(\bar{T}_W - \bar{T}_{\ell_0})$, average coefficient of heat transfer D, equivalent diameter of a square channel; L, length of the insert; μ, c_p, ρ, v_f, p, r, and σ, dynamic viscosity, the specific heat capacity, the density, the filtration velocity, the pressure, the latent heat of vaporization, and the surface tension of the liquid; g is the acceleration of gravity; $Nu_D = \alpha_{conv}D/\lambda$, $P_r = \mu c_p/\lambda$, $Re_D = \rho v_f D/\mu$, $Re_{d_p} = \rho v_f d_p/\mu$, $Pe_D = Re_D Pr$, Nusselt, Prandtl, Reynolds, and Peclet numbers; ξ_{d_p}, coefficient of hydraulic resistance of the insert. The indices are as follows: w refers to the parameters determined using T_w, ℓ refers to the physical properties of the liquid at T_{ℓ_0}, v refers to the physical properties of the vapor on the saturation line, D and d_p refers to the parameters determined based on the characteristic dimension D and D_v, and ' denotes a channel with an insert.

LITERATURE CITED

1. Megerlin, Merfi, Bergles, Teploperedacha, No. 2, 30-38 (1974).
2. V. M. Polyaev, V. A. Maiorov, and L. L. Vasil'ev, Hydrodynamics and Heat Transfer in Porous Elements of Aircraft Structures [in Russian], Moscow (1988).
3. S. A. Kovalev and E. G. Shklover, Teplofiz. Vys. Temp., 26, No. 5, 918-922 (1988).
4. V. V. Apollonov, V. P. Gamarskii, V. I. Mankeev, et al., Pis'ma Zh. Tekh. Fiz., 14, No. 14, 236-241 (1988).
5. V. N. Antsiferov, V. D. Khramtsov, O. M. Pitirimov, et al., Poroshk. Metall., No. 12, 20-24 (1980).
6. V. N. Antsiferov, A. P. Kunevich, V. A. Basanov, et al., Poroshk. Metall., No. 8, 87-91 (1988).
7. Handbook for Heat Transfer Engineers [Russian translation], Moscow (1987), Vol. 1.
8. Yu. F. Gortyshov, S. R. Ashikhmin, and I. N. Nadyrov, Izv. Vyssh. Uchebn. Zaved., Aviatsion. Tekh., No. 4, 31-35 (1989).
9. S. S. Kutateladze, Fundamentals of the Theory of Heat Transfer [in Russian], Moscow (1979).