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HEAT EXCHANGE AND RESISTANCE IN CHANNELS CONTAINING A POROUS FILLER

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The heat exchange and friction in pumping air through channels containing a porous filler (PF) consisting of pellets or wire pieces with different dimensions and thermal conductivity values are investigated in a wide range of porosities.

Heat exchange in channels is intensified considerably if the channels are obstructed with a porous, permeable heat-conducting filler, where perfect contact exists between the filler elements and between the latter and the channel walls. The heat from the channel walls is transmitted to the PF by the thermal conductivity λ_f of the porous filler body and is dissipated in the coolant over the entire transverse cross-section of the channel as a result of the intensive interstitial, convective transfer h from the heat-exchange surface that has been magnified by the factor F. For perfect contact between the PF and the channel walls, the analytical solution [1] indicates that the relationships between the thermal conductivity of the filler λ_f and of the coolant λ play the determining role in heat exchange intensification, while the heat exchange between the coolant and the boundary surface in the channel depends on λ_f/λ, the flow conditions Re_D, the extent X/D of the channel measured from the PF inlet, and the convective transport intensity h. Measurements [2, 3] in channels filled with steel wire PF, sintered and soldered to the channel walls, have shown that heat transfer to a single-phase flow increases by more than one order of magnitude for a PF porosity of less than 80%. At the same time, it was found in [2] that there is a more significant increase in the hydraulic resistance. The results of heat transfer measurements [3], while agreeing qualitatively with the solution obtained in [1], differ from it by 30-60% and reveal an additional effect of porosity. In [4], the increase of hydraulic friction in PF is related to inertial ($\beta\rho V^2$) and viscosity ($\gamma\mu V$) effects in porous channels; it is adequately generalized by the Darcy solution. The empirical quantities λ_f, h, β, and γ, which constitute individual characteristics of PF's [4, 5], are determined with considerable errors, which complicates the use of the solutions from [1, 4] for engineering predictions.

In many cases in practice, fastening of the PF elements to each other or to the channel walls is precluded by the specific features of the operating process or by technological constraints. In such channels, transverse heat transfer to the flow core directly from the channel walls by thermal conductivity of the PF either is attenuated or does not occur at all. However, it should be expected that, in spite of the absence of perfect contact among the PF elements or with the channel walls, there would be considerable intensification of heat ex-

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TABLE 1. Characteristics of Tested Simulators

Simulator No.	PF type	Π, %	$d_w \cdot 10^3$, m	$d \cdot 10^3$, m	$D \cdot 10^3$, m	$L \cdot 10^3$, m	$\bar{F}$
1	RM-S	83,2	0,2	0,99	20	36	17,8
2	RM-S	84,6	0,3	1,648	10,2	29	6,29
3	RM-S	85,3	0,2	1,161	10,2	29	8,5
4	RM-S	84	0,09	0,473	10,2	27	19,2
5	RM-S	70,9	0,2	0,487	10	48	15,8
6	RM-S	59,1	0,2	0,289	10	45	21,5
7	RM-S	46,1	0,2	0,171	10	31	28
8	RM-S	37,4	0,2	0,12	10	18	32,3
9	RM-S	69,3	0,3	0,677	10	48	11,2
10	RM-S	63,8	0,15	0,265	10	41	25,1
11	RM-S	71,4	0,05	0,125	10	31	58,2
12	RM-S	94,3	0,26	4,3	9,6	74	3,1
13	RM-S	96,1	0,09	2,218	9,6	50	5,2
14	RM-C	72,2	0,09	0,233	10	50	31,9
15	RM-C	59	0,09	0,13	10	29	46,6
16	RM-C	51,6	0,09	0,096	10	48	51,8
17	RM-C	44	0,09	0,071	10	30	63,2
18	PEF-S	50	3	2	10	49	3,5
19	PEF-G	50	3	2	10	49	3,5
20	—	100	—	8—20	8—20	300	1

change. As a result of the finite extent of the heat-conducting PF elements in the direction across the channel and the interstitial, volumetric convective heat exchange between the coolant and the PF elements, intensive transverse heat transfer from the boundary layers near the walls to the main part of the flow should remain at a high level. The literature does not provide data on the heat exchange in channels with PF's of this design.

We provide here the results of experimental investigations of the heat exchange and friction in channels containing porous fillers where contact between the filler elements and between the elements and the boundary surface of the channel is ensured only by the coolant pressure and the inherent PF elasticity. Steel pellets, glass beads, and rubber-metal (RM), a nonwoven, elastic material consisting of steel and copper wire, were used as the PF. Industrial technology [6] makes it possible to obtain from RM fillers with different porosities and configurations by pressing tangled, stretched spirals of wire with different diameters and thermal conductivity values.

The measurements results were generalized on the basis of the empirical model, matched with the solutions given in [1, 4],

$$\mathrm{Nu},\ f = \Phi\left[\mathrm{Re}_D,\ \mathrm{Pr},\ \Pi,\ \frac{X}{D},\ \frac{X}{d},\ \bar{F},\ \frac{\lambda_w}{\lambda}\right]. \tag{1}$$

Channel simulators (Fig. 1) consisting of steel pipe sections were used to estimate the effect of the parameters in (1) in the following ranges (see Table 1): $\mathrm{Re}_D = 10^3$-10^5; $\Pi = 37.4$-96.1%; $X/D \leq 7.6$; $X/d \leq 500$; $\bar{F} = 3.1$-63.2; $\lambda_w/\lambda = 30$-1200; $D/d = 2.2$-141.

Dry air at $P_{in} = 6 \cdot 10^5$ N/m² and $T_{in} = 300$-350°K.

The heat exchange was investigated experimentally by performing calorimetric measurements on the simulators and blowing air through them while they were immersed in a crystallizing melt of pure zinc [7]. The specific thermal fluxes q_X and the heat exchange coefficients α_X between the coolant and the boundary surface at the control sections X of the channel were determined by means of the equations

$$q_X = 0{,}25\,\frac{\rho^* r}{\tau}\,\frac{D_X^2 - D_1^2}{D},$$

$$\frac{T^* - T_X}{q_X} = \frac{1}{\alpha_X} + \frac{D}{2\lambda^*}\left[\ln\frac{D_X}{D_1} + \frac{\lambda^*}{\lambda_{wa}}\ln\frac{D_1}{D}\right]$$

with respect to the measured parameters T_{in} and G of the coolant and the parameters D_1, D_X, and M_X of the zinc crust built up on the channel's outside surface during the blowing-through time τ.

The evaluation measurements of heat exchange in smooth channels (No. 20 simulators in Table 1), which completed the adjustment of the stand, showed satisfactory agreement with

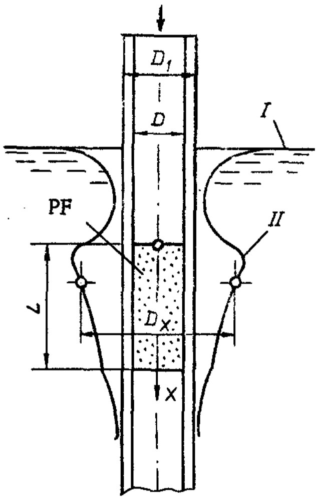

Fig. 1

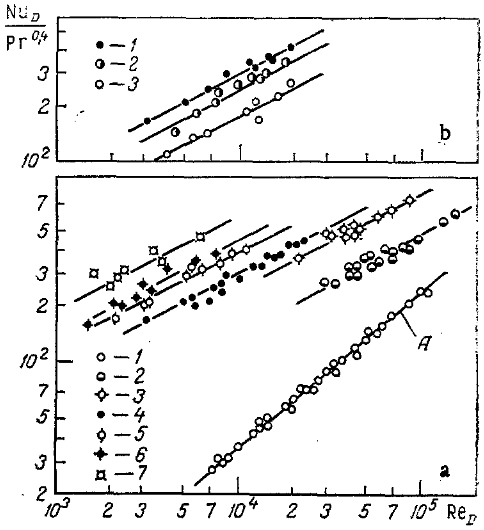

Fig. 2

Fig. 1. Segment of the channel simulator with PF in a zinc melt. I) Zinc melt level; II) crystallizing zinc crust.

Fig. 2. Effect of the porosoity of PF (a) and of the X/D ratio (b) on the heat exchange for Re_D = idem. a) 1) Smooth channel, No. 20 simulators, for X/D = 1.2-1.9; 2) simulators Nos. 12 and 13; 3) simulators Nos. 2-4; 4) simulators Nos. 5, 9, and 14; 5) simulators Nos. 6 and 15; 6) simulators Nos. 7 and 16; 7) simulator No. 8; A) $Nu_D = 0.023\ Re_D^{0.8}Pr^{0.4}$; b) simulators Nos. 5, 11, and 14: 1) X/D = 1.6-1.9; 2) 3.1-3.6; 3) 5.1-5.5.

the known similar measurements (see Fig. 2a, curve A) and corroborated the recommendations given in [7], confirming the suitability of the method and of the stand for investigating convective heat exchange in channels.

Effect of the Filter Porosity and of the Re Number. Tests of two simulator groups (group 1 comprises simulators Nos. 3, 5, 6, 7, 8, and 12 containing RM-S filler and group 2 comprising simulators Nos. 14-16 containing RM-C filler), where the fillers in either group had equal characteristics D, d_w, and λ_w, but different porosities (from 37.4 to 96.1%), confirmed the considerable intensification of heat transfer in channels contianing a PF without perfect contact between its elements or between it and the boundary surface. The heat exchange intensifies with a reduction in porosity. The measurement results given in Fig. 2a for channel sections located at distances X/D = 1.2-1.9 from the PF inlet show that, in comparison with the heat transfer α_0 in smooth channels without PF, the heat transfer α_X for $Re_D = 10^5$ increases by factors of 2.1, 3.5, and 4.2 for Π values of 96, 85, and 71%, respectively; for $Re_D = 7 \cdot 10^3$, it increases by factors of 9.6, 12.2, 14.3, and 21 for Π values of 71, 59, 46, and 37.4%, respectively. It was found that heat exchange intensification increases with a reduction in Re_D. For instance, for simulators Nos. 5 and 14 and Π = 70.9-72.2%, a reduction in Re_D from 10^5 to $7 \cdot 10^3$ raised the intensification ψ_f from 4.2 to 9.6. It has been established that the heat exchange in channels containing PF's is proportional to $Re_D^{0.5}$.

Effect of the X/D Ratio. Careful measurements in channels with PF have established that, for X/D < 7.6, the heat exchange between the coolant and the boundary surface, determined with respect to the mean temperature T_X of the flow at the cross section in question diminishes as the distance between the control section and the filler inlet increases. For instance, the measurement results for simulators Nos. 5 and 14 (Π = 70.9-72.2%) shown in Fig. 2b indicate that, for Re_D = idem, the heat exchange at sections X/D = 1.6-1.9 exceeds the heat exchange in sections X/D = 3.1-3.6 and X/D = 5.1-5.55 by factors of 1.3 and 1.75, respectively. It was found that the heat exchange in a channel contianing PF is proportional to $(X/D)^{-0.5}$ for constant values of Re_D and Π.

Effect of the Relative Surface Magnification $\bar{F}$. Tests of three groups of simulators with RM-S fillers have shown that an increase in the heat-exchange surface area, related to the surface area of the PF elements, is not the cause of heat exchange intensification in

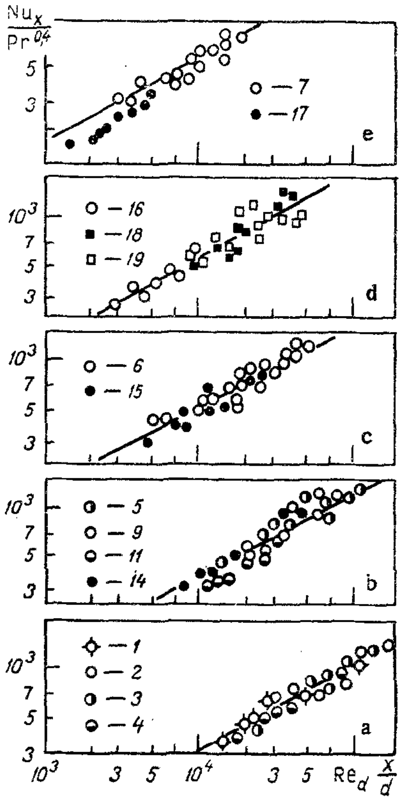

Fig. 3. Effect of the PF surface area (a, b, c, and d), the thermal conductivity (b, c, d, and e), and the type of PF elements (d) on the heat exchange. The numbers next to the symbols correspond to the simulator numbers in Table 1.

channels containing a PF without perfect contact between its elements and between the elements and the walls (group 1: simulators Nos. 5, 9, and 11 with equal values of Π = 69.3-71.4% and D = 10^{-2} m, but with different values of the wire diameter d_w, from $5 \cdot 10^{-5}$ m to $3 \cdot 10^{-4}$ m, and thus, with different values of the increase $\bar{F}$ in the heat-exchange surface area — from 11.23 to 58.2; group 2: simulators Nos. 2-4 with Π = 84-85.3% and D = 10^{-2} m, but with $d_w = 9 \cdot 10^{-5}$ m-$3 \cdot 10^{-4}$ m and, thus, characterized by F = 6.3-19.2; group 3: simulators Nos. 1 and 3 with Π = 83.2-85.3% and $d_w = 2 \cdot 10^{-4}$ m, but with different values of the channel diameter D, from 10^{-2} m to $2 \cdot 10^{-2}$ m, and thus, characterized by $\bar{F}$ = 8.5-17.8). The results of measurements performed on the above simulators that are shown in Fig. 3, a and b, indicate that, in channels with equal PF porosity values, a change in $\bar{F}$ from 6.3 to 58.2 does not alter the intensity of heat transfer through the boundary surface.

Effect of the Thermal Conductivity λ_w of PF Elements. Testing of four simulator pairs (1, simulators Nos. 5 and 14; 2, simulators Nos. 6 and 15; 3, simulators Nos. 7 and 17; 4, simulators Nos. 18 and 19) with equal filler characteristics Π and D in each pair, but with PF elements made of materials with thermal conductivity values λ_w differing from each other by factors of 17-23 (see Table 1), we established that considerable variation of the relative thermal conductivity λ_w/λ of the initial PF material, from 3 to 600 for a PEF filler and from 600 to 12,000 for an RM filler, does not affect appreciably the heat exchange at the boundary surface for the same filler porosity. A comparison of the measurements of the local heat transfer α_x at the control sections of the above simulator pairs is given in Fig. 3, b, c, d, and e.

Effect of the Type of PF Elements. Figure 3d provides a comparison between the results of local heat exchange measurements in a group of three simulators (Nos. 16, 18, and 19) with roughly the same PF porosities (50-51.6%) and the shame channel diameter D (10^{-2} m), but with substantially different types of filler elements (see Table 1). The comparison indicates that, in channels filled with a PF whose elements are not in perfect contact with each other and with the wall, the structural characteristics of the filler do not affect appreciably the heat transfer through the channel's boundary surface.

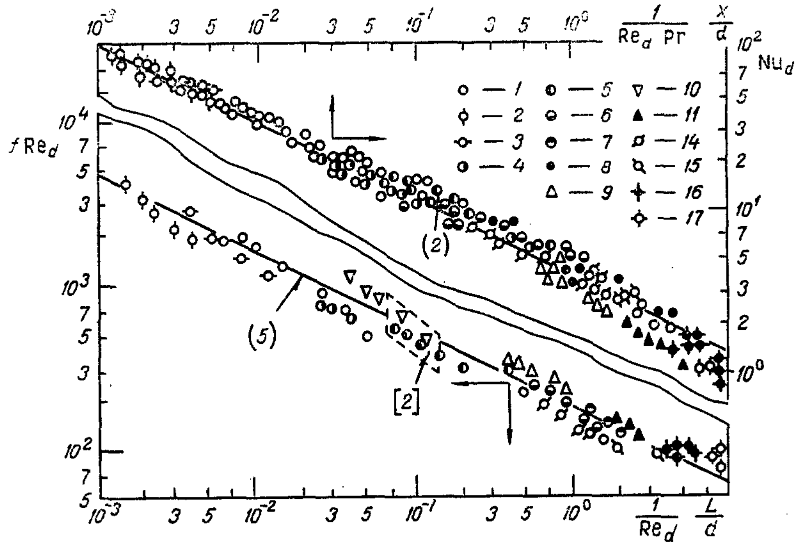

Fig. 4. Generalization of the measurements of heat exchange (2) and friction (5) in channels containing a PF. The figures next to the symbols correspond to simulator numbers in Table 1.

Figure 4 provides the results obtained in measuring the local heat exchange in channel simulators with porous fillers. The measurements for the entire investigated range of Re_D, Π, X/D, X/d, $\overline{F}$, λ_w/λ, and D/d values are generalized by a single relationship,

$$Nu_d = 4\left[Re_d\,Pr\,\frac{d}{X}\right]^{0,5} \text{ or } \qquad (2)$$
$$Nu_D = 4\,[Re_D Pr]^{0,5}\left[\Pi\,\frac{X}{D}\right]^{-0,5}.$$

The measurements indicate that the heat exchange between a single-phase coolant and the boundary surface of a channel containing a filler whose elements are not in perfect contact with each other and with the walls depends only on the mass velocity of flow through the porous channels and the relative distance from the PF inlet. It has been found that the intensification of local heat exchange (ψ_f) at the transverse cross-section X and the intensification of the mean heat exchange ($\overline{\psi}_f$) over the segment X from the inlet of a PF-containing channel for simulators Nos. 1-19 in comparison with the heat exchange in smooth channels (No. 20 simulators) are determined by the relationship

$$\psi_f = 174\,Re_D^{-0,3}\left[\frac{X}{D}\,\Pi\right]^{-0,5}, \qquad (3)$$

$$\overline{\psi}_f = 2\psi_f. \qquad (4)$$

The hydraulic resistance of channels with PF's was determined with respect to the discharge and the paressure head loss $\Delta P_{in} = P_{in} - P_{out}$ of the coolant in the channel segment containing the filler during isothermal air blowing through simulators Nos. 1-19. The mean hydraulic friction coefficient f over the extent L of filler was determined with respect to the known values of ΔP, G, L, d, and Π from the expression for the total pressure loss in a channel:

$$\Delta P = \left[0{,}5\,(1-\Pi) + (\Pi^{-1}-1) + 4f\,\frac{L}{d}\right]\frac{1}{2\rho}\left(\frac{G}{f_f}\right)^2.$$

Figure 4 shows the results of all our measurements of the hydraulic friction f in the abovementioned investigation ranges regarding Re_D, Π, and L/d. The measurements are generalized by a single relationship,

$$fRe_d = 150\left[\frac{1}{Re_d}\,\frac{L}{d}\right]^{-0,5} \text{ or } f = 150\left[Re_D\,\frac{L}{D}\,\frac{1}{\Pi}\right]^{-0,5}, \qquad (5)$$

they are in agreement with the similar measurements performed in [2], and they indicate that the hydraulic friction in channels with a PF depends only on the mass velocity of flow in such channels and the relative extent of the filler.

NOTATION

PF, permeable, porous filler in the channel; RM, rubber-metal wire material; PEF, pellet filler; RM-C, copper wire filler; RM-S, steel wire filler: PEF-S, steel pellet filler; PEF-G, glass bead filler; Π and L, PF porosity and axial extent in the channel, respectively; X, coordinate in the direction of flow, measured from the PF inlet; D_1 and D, outside and inside diameters of the channel, respectively; $f_0 = \pi D^2/4$ and $f_f = \Pi f_0$, free cross-sectional area inside channels without and with a PF, respectively; $F_0 = \pi DL$, surface area of the channel's boundary surface over the segment L; F_f total surface area of the PF elements (wire or pellets); d_w, diameter of wire or pellets in the PF; d, hydraulic diameter of a channel containing a PF; RM, $d = \Pi(1 - \Pi)^{-1}d_w$; PEF $d = 0.666\ \Pi \times (1 - \Pi)^{-1}d_w$; $\bar{F} = 1 + F_f/F_0$ is the coefficient of surface area magnification in a channel with PF: RM, $\bar{F} = 1 + (1 - \Pi)D/d_w$ and PEF $\bar{F} = 1 + 1.5\ (1 - \Pi) \times D/d_w$; λ_w, λ_f, λ, and λ_{wa} are the thermal conductivities of the material in the PF elements, the porous filler body, the coolant, and the channel wall, respectively; G, P, T, V, ρ, c, and μ are the mass discharge, the pressure, the temperature, the velocity, the density, the specific heat at constant pressure, and the dynamic viscosity of the coolant in the channel respectively; q is the specific thermal flux through the boundary surface (W/m²); α_X and $\bar{\alpha}_X$ are the coefficients of heat exchange between the coolant and the boundary surface, the local value at the control section X and the mean value over the segment from the PF inlet to X, respectively (W/(m²·deg K)); f is the coefficient of hydraulic friction in the PF; γ, β, and h are the coefficients of viscous interstitial drag (m²), inertial drag (m⁻¹), and volumetric convective heat exchange (W/(m³·deg K)), borrowed from [4]; T^*, r, λ^*, and ρ^* are the crystallization temperature and heat, the thermal conductivity, and the density of zinc, respectively; τ is the blowing-through duration; D_X is the outside diameter of the zinc crust built up at the control section during the time τ; M_X is the mass of the zinc crust built up on the simulator walls from its inlet to the control section during the time τ; $Re_D = GD/\mu f_0$, $Re_d = Gd/\mu f_f$, $Nu_D = \alpha D/\lambda$, $Nu_d = \alpha d/\lambda$, $\underline{Nu}_X = \underline{\alpha}X/\underline{\lambda}$, and Pr are the Reynolds, Nusselt, and Prandtl numbers, respectively; $\psi_f = \alpha_X/\alpha_0$, $\bar{\psi}_f = \bar{\alpha}_X/\alpha_0$ are the heat exchange intensification factors in a channel with a PF, the local value and the mean value, in relation to an infinite, smooth channel for Re_D = idem, respectively. Subscripts: in, out, and X, parameters at the inlet, outlet, and control sections of a channel with PF, respectively; w, for a PF element; f, for a channel with PF; 0, for a smooth channel without PF.

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