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HEAT EXCHANGE IN POROUS MEDIA

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Results are offered from an experimental study of the heat-exchange process in porous materials and the effect on this process of high-frequency oscillations in the heat-exchange fluid flow.

The study of heat exchange and its intensification in porous materials is a problem of practical interest, inasmuch as such materials are finding ever wider use as construction materials for surfaces drafted by high-temperature flows. Both stationary and nonstationary heat-exchange processes in porous materials are of great interest.

The goal of the present study is to examine the heat-exchange process in porous materials and the effect upon this process of high-frequency oscillations in the heat-exchange fluid flow.

The one-dimensional method of describing the heat-exchange process in porous structures will be used. For a plate of porosity Π with internal heat source q_V the problem reduces to solution of the thermal balance equations

$$\frac{d^2T_w}{dx^2}-\frac{GC_p}{F_\Sigma\lambda_M}\frac{dT_f}{dx}+\frac{q_v(1-\Pi)}{\lambda_M}=0, \tag{1}$$

$$\frac{GC_p}{F_\Sigma}\frac{dT_f}{dx}=\alpha_v(T_w-T_f), \tag{2}$$

where Π is the porosity, defined as the ratio of the pore volume to the total plate volume, and α_V is the heat-liberation coefficient, characterizing the heat produced per unit volume of the porous material.

Up to the present most authors have studied the mean heat liberation coefficient without consideration of the effect of heat exchange on the specimen boundaries. Results as to the effect of boundary conditions on the heat-liberation coefficient in porous structures α_V are extremely contradictory. According to [1-3], heat exchange at the boundaries of a porous specimen does not affect heat exchange within the pores, while according to [4] and [5], this effect can be significant. In connection with this fact, in the present study the effect of heat exchange at the specimen boundaries on heat exchange within the porous structure was considered in addition to a study without consideration of heat exchange at the boundaries. For this purpose two variants of boundary conditions were considered for Eqs. (1), (2).

In the first variant heat exchange at the specimen boundaries was ignored, and the boundary conditions had the following form:

$$\left.\begin{aligned} x=0,\ \lambda_M\frac{dT_w}{dx}=0,\\ x=\delta,\ \lambda_M\frac{dT_w}{dx}=0.\end{aligned}\right\} \tag{3}$$

In the second variant heat exchange at the input (the boundary where coolant enters) boundary was considered, and the boundary conditions had the form

$$x=0,\ \lambda_M=\frac{dT_w}{dx}=\alpha_{in}(T_{w_1}-T_{f_0}), \tag{4}$$

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$$x=\delta,\ \lambda_{\text{M}}=\frac{dT_w}{dx}=0,$$

where α_{in} is a heat-liberation coefficient characterizing the thermal interaction between the surface of the porous wall and the coolant at the boundary $x = 0$. The interaction of the coolant with the porous material boundary at the exit ($x = \delta$) will not be considered in the present study, since it can be assumed that upon approach to the outer boundary of the porous material $x = \delta$ the coolant temperature practically reaches the temperature of the material, and in the region $\delta < x < \infty$ intensity of the heat exchange between coolant and porous wall surface is negligibly small.

In order to define critical equations for the heat-exchange process, the following model of the phenomenon will be used. In contrast to previous studies in which the porous structure was modeled by parallel capillary channels or in the form of alternating constrictions and expansions in the pore channel, it was assumed here that the coolant flow in the porous structure is analogous to flow in rough channels, in which the dimensions of the roughness, or in the present case the mean statistical particle diameter, is comparable to the channel diameter. On the basis of studies of flow in porous media [6-8] and a study of flow around bodies of various geometries, in particular, a sphere, the coolant flow in such a channel will be accompanied by formation of microvortices, the magnitude of which is determined by the size of the particles flowed over. The heat-exchange intensity in this case will depend not only on Reynolds number, but also on the geometric parameter $\varkappa$, whose value indicates the ratio of the equivalent porous channel diameter to the porous material particle diameter. In this case the critical equation for the heat-exchange process in porous materials for stationary coolant flow can be represented in the form

$$\mathrm{Nu}_s=F(\mathrm{Re}_{d_e};\ \varkappa), \tag{5}$$

where

$$\mathrm{Re}_{d_e}=\frac{Gd_e}{F_\Sigma\mu};\ \mathrm{Nu}_s=\frac{\alpha_s d_e}{\lambda_f};\ \varkappa=\frac{d_e}{d_p};$$

α_s is the heat-liberation coefficient per unit of heat-exchange surface.

Since in the present study we deal with media composed of spherical particles (sintered metal powders and poured charges of spherical particles), the value of the parameter $\varkappa$ was estimated from the condition that the porous structure consists of densely packed spherical particles. In this case the parameter is directly related to the porosity, being equal to $\varkappa = {}^2/_3\Pi/(1-\Pi)$. However, use of this parameter in the form of Eq. (5) is preferable, since it is the geometric dimensions of the particles, and not the porosity, which determines the character of coolant flow in such a structure. Equation (5) assumes a quantitative estimate of the heat liberation in the porous material, referenced to unit heat-exchange surface. Therefore, in calculating the similarity criteria one must use as the characteristic dimension the equivalent pore channel diameter. The size of the internal heat-exchange surface can be estimated from the condition that the porous material is composed of spherical particles with a diameter equal to the mean statistical particle diameter.

A physical picture of the effect of high-frequency oscillations on the heat-exchange process in porous structures can be formed in analogy to the smooth channels of [10]. The relative heat-liberation coefficient, which is the ratio of the heat-liberation coefficient with oscillations to the stationary heat-liberation coefficient, is dependent on the value of the relative amplitude of mass velocity oscillations ε and the ratio of the viscous layer thickness δ_0 to the oscillating layer thickness $\delta_{osc} = \sqrt{2\nu/\omega}$. The quantity δ_0 is then a characteristic parameter, defining the magnitude of turbulent vortices near the surface. Flow over the porous structure particle by an oscillating flux will also be accompanied by formation of vortices in analogy to passage of an oscillating flow over curvilinear surfaces (cylinder and sphere), which is accompanied by formation of secondary stationary vortices, whose dimensions are determined by the thickness of the oscillating layer δ_{osc}. Since in the porous structure the size of the stationary vortices is proportional to the particle diameter, in contrast to flow in smooth channels here we must use not the viscous layer thickness, but the particle diameter as the characteristic, so that the criterial equation for heat exchange with coolant flow oscillations takes on the form

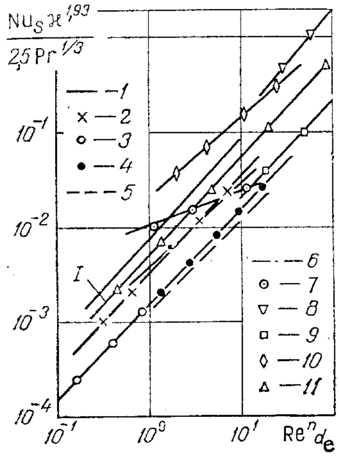

Fig. 1

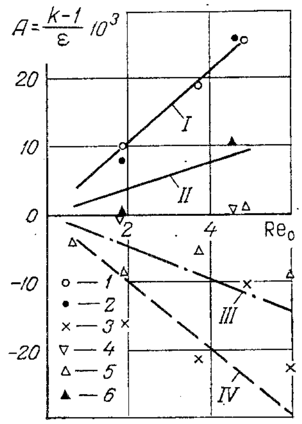

Fig. 2

Fig. 1. Results of experiments on heat exchange with stationary coolant flow in porous materials and spherical particle layers: 1) specimen No. 1; 2) No. 2; 3) [3], Re = 0.06-1.5; 4) [3], Re_{de} = 1.5-80; 5) [1]; 6) [13], Π = 0.212; 7) [13], Π = 0.318; 8) [12], data of M. É. Aérov; 9) [12] (data of V. N. Timofeev); 10) [14]; 11) by Eq. (12).

Fig. 2. Effect of time-averaged Reynolds number on maximum and minimum heat-exchange intensity: 1, 3, 5) specimen No. 1; 2, 4, 6) specimen No. 2; 2) Re_ω = 0.525; 3, 4) 0.125; 5, 6) 0.925; I, from formula $A_{max} = (5.3\, Re_0 + 0.15)\cdot 10^{-3}$; II, $A_{min} = (1.84\cdot Re_0 - 1)\cdot 10^{-3}$; III, $(-2.37\, Re_0 - 0.25)\cdot 10^{-3}$; IV, $(-4.94\, Re_0 - 0.22)\cdot 10^{-3}$.

$$K = Nu/Nu_0 = F(Re_0;\ Re_\omega;\ \varepsilon),$$

where

$$Re_\omega = 2\,\pi f d_p^2 / \nu = \omega d_p^2 / \nu. \tag{5a}$$

Just as in smooth channels [10], in porous structures one can expect development of resonant phenomena, i.e., when the dimensions of stationary vortices generated by flow over the particles of the structure are comparable to the dimensions of the secondary vortices generated by the flow oscillations. Thus, intensification of turbulent motion and the heat-transfer process itself can be expected.

Thus, the task of experimental study of the heat-exchange process reduces to the following:

a) with stationary coolant flow, the critical dependence of Eq. (5) must be determined;

b) with oscillations in the coolant flow, the criterial dependence of Eq. (5a) must be determined.

In the present study the heat-liberation coefficient within the pores was defined as the ratio of thermal flux to the mean integral (over thickness) temperature head.

Experimental study of the heat-liberation coefficient at the input boundary of the porous specimen raises insurmountable difficulties with measurement of the coolant temperature and thermal flux at the boundary. Thus, an indirect method was developed for determination of these quantities, based upon the one-dimensional model of the heat-exchange process. The heat-liberation coefficients α_{in} and α_s were determined from the experimentally measured values of temperature T_{f_0}, $T_w(0)$, $T_w(\delta)$, heat liberation q_v, and coolant flow rate given by solution of the reverse problem of heat exchange in porous structures described by Eqs. (1), (2). The method of successive approximations was used. The parameters were measured with experimental apparatus described in detail in [11]. The major characteristics of the specimens studied are presented in Table 1. In performing the experiments, parameters were varied over the following ranges:

$$Re_0 = 0.6 - 6.0;\ q_v = 10^6 - 10^8\ W/m^3;\ f = 20 - 400\ Hz;\ \varepsilon = 5 \div 150.$$

TABLE 1. Geometrical Characteristics of Experimental Specimens

No.	Material	d_{out}, mm	d_{in}, mm	L, mm	$d_{p\,av}$, mm	Π
1	Stainless steel	40	34	90	80	0,583
2	Nickel	40	34	70	120	0,634

Study of the heat-exchange process with stationary coolant flow without consideration of heat exchange at the input boundary of the porous specimen produced the following empirical relationships:

for specimen No. 1

$$Nu_s/Pr^{1/3} = 0{,}0198\,Re_{d_e}^{1{,}5}, \tag{6}$$

for specimen No. 2

$$Nu_s/Pr^{1/3} = 0{,}0082\,Re_{d_e}^{1{,}3}. \tag{7}$$

Comparison of Eqs. (6), (7) with data of other authors, expressed in the form of functions $Nu_s/Pr^{1/3} = f(Re_{d_e})$ revealed that the functions obtained here agree qualitatively with known principles of the heat-exchange process in porous materials and layers of spherical particles, although the experimental data diverge greatly (by two orders of magnitude). In order to establish the effect of the parameter $\varkappa$ on the heat-exchange process, the available experimental data [1-3, 12-14] (Fig. 1) was processed and reduced to functions of the form of Eq. (5). Analysis of the experimental data established that the parameter $\varkappa$ determines the value of the exponent of the Reynolds number Re_{d_e} as follows:

$$n = \sqrt{1{,}907\,(\varkappa + 0{,}0277)}. \tag{8}$$

With consideration of Eq. (8) the experimental data were represented in the form

$$Nu_s/Pr^{1/3} = F\,(Re_{d_e}^{n}), \tag{9}$$

from which it follows that at specified values of the number $Re_{d_e}^{n}$ the value of the dimensionless complex $Nu_s/Pr^{1/3}$ is related to the value of the parameter $\varkappa$ as follows:

$$\frac{Nu_s}{Pr^{1/3}} = \frac{2.5}{\varkappa^{1.93}}. \tag{10}$$

With consideration of Eq. (10), the heat-exchange process in porous materials and spherical particle layers can be satisfactorily generalized by the unique criterial equation (Fig. 1):

$$Nu_s/Pr^{1/3} = 1.25\cdot 10^{-3}\,\varkappa^{-1.93}\,Re_{d_e}^{n} \tag{11}$$

(where n is defined by Eq. 8), which shows that the intensity of the heat-exchange process in porous structures depends not only on Reynolds number, but also on the parameter $\varkappa$, with the value of the latter affecting not only the value of the constant in the criterial equation, but also the power of Re_{d_e} over the range n = 0.37-1.48.

Results of studying the effect of coolant flow oscillations on heat exchange can be reduced to the following. With increase in the pressure oscillation frequency from 25 to 400 Hz, the change in relative heat-liberation coefficient of porous stainless steel occurs in accordance with the change in relative amplitude of the mass velocity oscillations. In the frequency range 220-225 Hz for all values of Re_0 there is a maximum increase in heat liberation from K = 1.5 at Re_0 = 0.6 to K = 4.3 at Re_0 = 5.0. A similar pattern was observed in experiments with a porous nickel specimen.

For fixed values of oscillation frequency it was found that the change in relative heat-liberation coefficient K was linearly related to the change in relative oscillation amplitude. The effect of the oscillatory Reynolds number Re_ω on heat-exchange intensity is such that over the ranges Re_ω = 0.125-0.4 and Re_ω = 0.75-0.925 there is a decrease in heat liberation, while for Re_ω = 0.4-0.75 heat liberation increases, with the maximum value being realized at Re_ω = 0.525, which corresponds to a ratio d_p/δ_{osc} = 0.55 and confirms the presence of resonant phenomena in the porous materials. Similar principles have been observed in smooth cylindrical channels [9].

In Fig. 2 the parameter A = (k − 1)/ε, which characterizes the intensity of heat exchange, is shown as a function of Reynolds number. It was found that the Reynolds number averaged over time affects heat exchange in a linear manner.

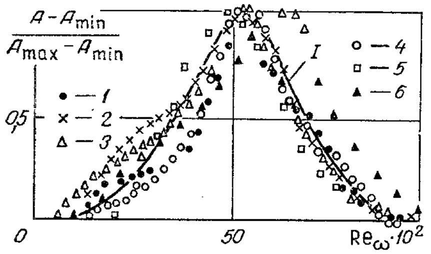

Fig. 3

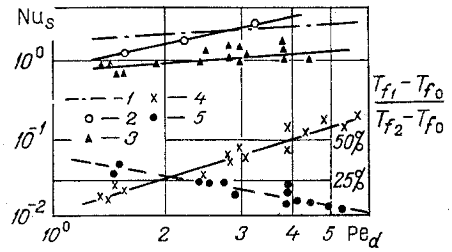

Fig. 4

Fig. 3. Generalization of experimental data on heat exchange in porous media with oscillatory coolant flow: I, by Eq. (15); 1-4, specimen No. 1; 5, 6, specimen No. 2; 1, $Re_0 = 0.631$; 2, 5, 1.86; 3, 3.71; 4, 4.86; 6, $Re_0 = 4.57$.

Fig. 4. Results of study of stationary heat exchange at entrance to porous specimen No. 1: 1, [14] (single sphere); 2, [5]; 3, by Eq. (13); 4, Eq. (14); 5, ratio of coolant heating before entry into specimen to total heating.

A generalization of the experimental data on oscillatory Reynolds number is shown in Fig. 3. Here the parameter $A_{max} = (k_{max} - 1)/\varepsilon_{max}$ characterizes the maximum intensity of heat liberation at $Re_\omega = 0.525$, while the parameter $A_{min} = (k_{min} - 1)/\varepsilon_{min}$ gives the minimum heat-liberation intensity at values of $Re_\omega = 0.125$ and 0.925.

The criterial function obtained (Fig. 3) defines the effect on heat exchange of the time-averaged Reynolds number, the oscillatory Reynolds number, and the relative amplitude of mass velocity oscillations.

Results of the study of the effect of heat exchange at the entry surface of the porous specimen on heat exchange within the pores reduce to the following: for stationary coolant flow (Fig. 4), heat exchange at the entry of the porous specimen is close to the function defining heat exchange on a single sphere, and also close to the function recommended in [5], exceeding the intensity within the pores. The ratio of the amount of coolant heating before entry into the specimen to the total heating of the air upon passage through the specimen comprises 30% at Pe ≈ 1.2; upon increase in filtration velocity to Pe = 5.0 this ratio decreases to 3-4%. These results can be generalized by the equations

$$Nu_{in} = Pe^{0.235}, \tag{12}$$

$$Nu_s = 0.013\, Pe^{1.46}. \tag{13}$$

The effect of coolant flow oscillations on heat exchange in this case is similar to the effect of an oscillating flow in the case considered above. At the input surface of the specimen, just as within the pores, the maximum heat-exchange intensity is realized at an oscillatory Reynolds number value of $Re_\omega = 0.525$, when the values of the particle diameter and the oscillating layer thickness are comparable.

The effect of the flow time-averaged Reynolds number on maximum and minimum heat exchange intensity is also linear, but as compared to the previous case the maximum heat-exchange intensity within the pores with consideration of heat exchange at the entrance surface is 1.5 times lower than without consideration of this process over the entire range of Re_0 studied.

The following generalized expressions were obtained for this case:

at the input boundary

$$\frac{A_{in} - A_{in\,min}}{A_{in\,max} - A_{in\,min}} = \frac{1 - 6.25\,(Re_\omega - 0.525)^2}{1 + 17.97\,(Re_\omega - 0.525)^2}, \tag{14}$$

where

$$A_{in\,min} = (-Re_0 - 0.6)\cdot 10^{-3} \quad \text{for} \quad Re_\omega = 0.925;$$

$$A_{in\,min} = (-4.6\,Re_0 - 0.3)\cdot 10^{-3} \quad \text{for} \quad Re_\omega = 0.125;$$

$$A_{in\,max} = (0.2\,Re_0 + 4.5)\cdot 10^{-3} \quad \text{for} \quad Re_\omega = 0.525;$$

within the pores

$$\frac{A_s - A_{s\,\min}}{A_{s\,\max} - A_{s\,\min}} = \frac{1 - 7.11\,(\mathrm{Re}_\omega - 0.525)^2}{1 + 54.52\,(\mathrm{Re}_\omega - 0.525)^2}, \tag{15}$$

where

$$A_{s\,\min} = (-\mathrm{Re}_0 - 0.6)\cdot 10^{-3} \quad \text{for} \quad \mathrm{Re}_\omega = 0.925;$$

$$A_{s\,\min} = (-4.6\,\mathrm{Re}_0 - 0.3)\cdot 10^{-3} \quad \text{for} \quad \mathrm{Re}_\omega = 0.125;$$

$$A_{s\,\max} = (0.12\,\mathrm{Re}_0 + 0.7)\cdot 10^{-2} \quad \text{for} \quad \mathrm{Re}_\omega = 0.525.$$

These functions generalize the experimental data with a maximum error of ±40% over the range Re_0 = 0.125-0.925 for changes in relative mass velocity amplitude of ε = 10-150.

NOTATION

T, temperature, °K; x, coordinate; d, diameter; $K = \mathrm{Nu}/\mathrm{Nu}_0$, relative heat-liberation coefficient where Nu_0 is the Nusselt number for stationary coolant flow; $\delta_{\mathrm{osc}} = \sqrt{2\nu/\omega}$, thickness of oscillating layer; $\mathrm{Nu} = \alpha_V d_e^2/\lambda_f$, Nusselt number; $\mathrm{Re} = G d_p/F\mu$, Reynolds number; $\mathrm{Re}_\omega = 2\pi f d_e^2/\nu$, oscillatory Reynolds number; $\varepsilon = \Delta\rho u F_\Sigma/G$, relative amplitude of mass velocity oscillations; G/F_Σ, filtration rate or coolant mass flow rate, referred to total cross-sectional area of plate; f, frequency, Hz. Subscripts: w, wall; f, coolant; 0, in flow core; 1, at specimen input surface; 2, at specimen output surface; M, porous material; p, particle; s, surface; e, equivalent.

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