HYDRODYNAMIC AND THERMAL CHARACTERISTICS OF SYSTEMS OF POROUS COOLING IN THE PRESENCE OF SMALL PERIODIC PERTURBATIONS

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Results of experimental investigations indicate the susceptibility of systems of porous cooling to small periodic perturbations, namely, wall vibrations and gas-coolant pressure pulsations. In this case, the filtration properties of porous materials and heat and mass transfer characteristics can change substantially.

Systems of porous cooling are finding increasing use in technology: porous fuel elements of nuclear reactors, various compact steam generators, evaporative systems, filters, and elements of the thermal shielding of aerospace vehicles [1–3]. As a rule, the operation of such systems is accompanied by small perturbations: vibrations of walls, pressure pulsations, acoustic vibrations, and turbulent noises. Depending on the type of perturbations and the frequency and amplitude of vibrations, the filtrational and thermal characteristics of porous materials can undergo distortions. Moreover, the vibrations of porous walls alter the aerodynamic properties of bodies and the wave resistance coefficient C_x.

The present work concerns an experimental investigation of the process of heat and mass transfer of blunt bodies immersed in high-temperature sub- and supersonic flows in the presence of gas-coolant injection through a permeable portion and successive exposure of the systems of porous cooling to harmonic vibrations of the wall along the flow and gas-coolant pulsations.

Experiments were conducted in the jets from the ohmic gas heater of an ÉDP-104A/50 plasmotron in sub- and supersonic wind tunnels. The flow parameters varied within the ranges: $\mathrm{Re}_D = 260-3200$, $M = 0.01-2.1$, $T_e = (300-4900)$ K.

The basic scheme of the experiments is presented in Fig. 1. Plates 1 made of porous materials (stainless steel and Ni-Cr alloy) were pressed hermetically into the small base of truncated cone 2. Arrows 3 and 4 denote injected gas (air, nitrogen) and external air flow. Gas-coolant pressure pulsations and linear (with respect to the symmetry axis) vibration models were generated by means of electric motor shaft 5 and worm 6. The difference between the vibrations of the model and the gas-coolant pulsations consisted in the means of the action of the worm on the model: for vibrations it acted directly on the wall, and for pulsations it acted on the gas-coolant in the gas-supply mains. The frequency of the perturbations f and the amplitude A were regulated by the rate of rotation of the electric motor shaft w and geometric dimensions of the worm. The frequency and the amplitude of the vibrations varied within the limits $f = (0-20)$ Hz, $A = (0-5)\cdot 10^{-3}$ m.

The geometric characteristics of the plates made of porous materials (the production method was knitted fabric [11]) are $D = (15-30)\cdot 10^{-3}$ m, $h = (1.0-2.0)\cdot 10^{-3}$ m, $\gamma = (20-50)\cdot 10^{-6}$ m, $\Pi = 0.347-0.67$, where γ is the diameter of a fiber.

The filtration characteristics, i.e., the viscosity α-coefficient and the inertial β-coefficient in the Darcy quadratic law of filtration [4, 5]

$$\frac{\partial P}{\partial y} = \alpha\mu v + \beta\rho v\,|v| \tag{1}$$

were determined by the approach suggested in [6]. Assuming the validity of the replacement of the ρ and v parameters by the equivalent gas parameters near the wall [7] $\rho v\Pi = (\rho v)_w = 4G_w/\pi D^2$ and the conditions of gas isothermality over the plate thickness, Eq. (1) can be easily integrated:

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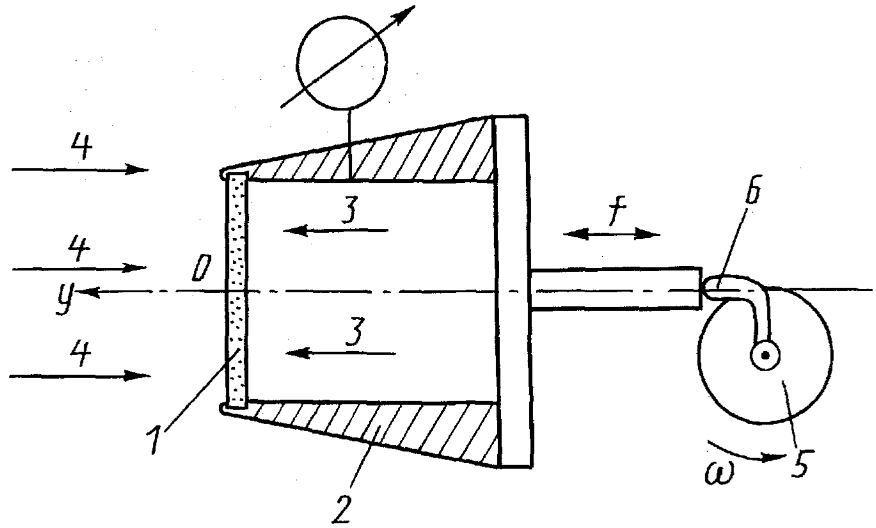


Fig. 1. Scheme of conducting the experiment.

$$\beta RT_0 h\,(\rho v)_{\rm w}^2 + \alpha\mu RT_0 h\,(\rho v)_{\rm w} = \frac{\Pi\,(P_k^2 - P_{\rm e}^2)}{2}\,. \qquad (2)$$

The system of quadratic algebraic equations (2) was used to find α and β. The values of P_k and $P_{\rm e} \approx P_{\rm w}$ were monitored by means of an MO-7-type gauge (see Fig. 1) or an MMN-240 micromanometer, the injected gas flow rate $G_{\rm w}$ was measured by means of rotameters of type RS and RM. Moreover, using a pneumatic gauge and a thermoanemometer [8], the values of the velocity v and the velocity fluctuations v' near the front stagnation point on the outer and inner sides of the plate were recorded. The degree of turbulence of the injected gas on the inner side of the plate did not exceed 0.03.

The filtration Reynolds number $\mathrm{Re_f} = v\sqrt{k}/\nu$, where $k = 1/\alpha$, changed within the range $\mathrm{Re_f} = (0-14.2)$, i.e., both laminar and turbulent modes of gas-coolant filtration were realized.

The resultant errors in the determination of the gas parameters were $\delta P \le 3.8\%$, $\delta G_{\rm w} \le 5.1\%$, $\delta v \le 5.3\%$. The confidence intervals were calculated with a 0.95 probability from the results of 3–5 observations.

The heat and mass transfer characteristics of the porous materials, such as the wall temperature $T_{\rm w}$ in the vicinity of the front stagnation point 0 and the heat flux density $q_{\rm w}$, were determined by measuring the brightness temperature (with subsequent conversion of the brightness temperature values in the real temperature [1]), by a Chromel-Alumel (ChA) thermocouple, by an exponential technique [1], and by solving the Abel equation [9]

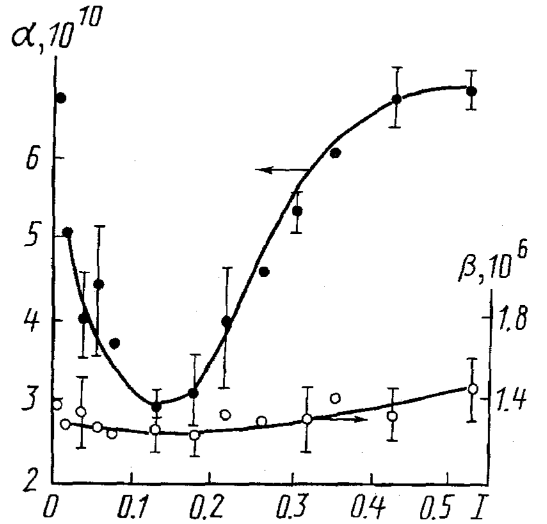
$$q_{\rm w} = (\lambda c_p \rho_c)^{-0.5} \int_0^t (dT_{\rm w}/dt)\,[\pi\,(t-\tau)]^{-0.5}\,d\tau\,.$$

The $T_{\rm w}$ values were approximated by a power polynomial or a cubic spline.

The ChA thermocouples (with a junction diameter of $100 \cdot 10^{-6}$ m) extended on the inner side of the porous plate between the 2nd and 3rd rows of the strands of fibers in the wall. In this case, a certain distortion in the structure of the porous material was observed. To eliminate the systematic error in the measurements of $T_{\rm w}$ occurring because of the distortion of the material structure, the results of measurements of $T_{\rm w}$ by a ChA thermocouple were compared with those obtained by the brightness method. The difference did not exceed 4.3%. The resultant errors in the determination of the parameters were $\delta T_{\rm w} \le 7.8\%$, $\delta q_{\rm w} \le 9.2\%$.

The wave resistance coefficient C_x was determined as the sum of the resistance coefficients for the small base of the cone and the side surface. It was calculated in two ways: from the results of measurements of the pressure P through eight drainage holes by means of LKh-415-type probes and from the formulas [10]

$$C_x = \int_s \overline{P} \sin\beta_{\rm cone} dS/S_{\rm m}\,, \qquad (3)$$



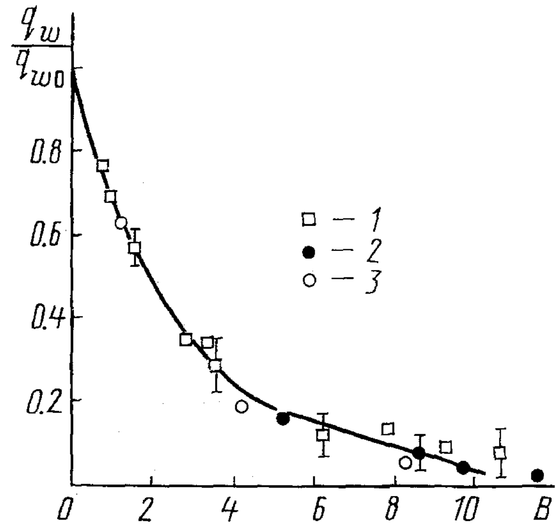


Fig. 2. Dependence of filtrational characteristics on the intensity of perturbations. α, m^{-2}; I, kg/sec^3; β, m^{-1}.

Fig. 3. Dependence of the relative heat flux on the injection parameter.

$$C_x = \overline{P}_0 \left[\sin^3 \beta_{cone} \frac{x_{cone}}{r_m^2} (2r_m - x_{cone} \sin \beta_{cone}) + 0.915 \right], \tag{4}$$

where $\overline{P} = 2(P - P_e)/(\rho_e v_e^2)$ is the pressure coefficient. The difference in the values of C_x calculated from formulas (3) and (4) did not exceed 3.7%. In calibrating the pressure probes a correction coefficient was introduced that took into account distortions appearing during vibration of the model.

In Fig. 2 results of calculations of α and β are presented as functions of the gas-coolant vibration intensity: $I = \rho c u^2/2$, $u = \omega A$, $\omega = 2\pi f$, $\Pi = 0.67$, $h = 2 \cdot 10^{-3}$ m. The solid lines represent approximations by the least-squares method. Experiments were conducted in an wind tunnel MT-324.

Attention is drawn to the decrease in the viscous filtration coefficient at relatively low intensities of vibrations. The inertial filtration coefficient remains virtually unchanged. Such curves were obtained for porous materials with large values of volumetric porosity $\Pi \geq 0.49$. For materials with $\Pi < 0.49$ the values of α and β are independent of I.

When $Re_f > 6.3$, an appreciable spread in the results was observed that was caused by the appearance of deformations in the structure of porous materials. This explains the appearance of rather large confidence intervals in Fig. 2.

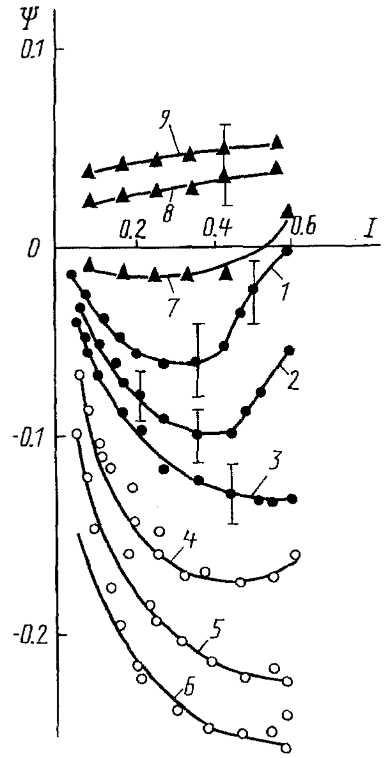
The values of α measured for a porous material without gas-coolant fluctuations were compared with α_{calc} calculated from the formula $\alpha_{calc} = 0.851 \cdot 10^{10} \Pi^{-4.73}$ [11] obtained for knitted fabric nets: $\alpha_{calc} = 5.58 \cdot 10^{10}$ $1/m^2$; $\alpha = 6.95 \cdot 10^{10}$ $1/m^2$. The difference amounts to 19.7%.

Figure 3 presents the dependence of the dimensionless heat flux density on the injection parameter without periodic perturbations: points 1 denote results obtained in a plasma jet, 2 in an ohmic heater, 3 in a supersonic wind tunnel. The measurement data agree satisfactorily with the results of calculations from the formula $q_w/q_{w0} = \exp[-0.37(M_w/M_e)^{0.7} B]$ [1] for $B < 4.8$ (the solid line in Fig. 3) and from the formula

$$q_w/q_{w0} = 0.28 - 0.024\,B \tag{5}$$

obtained for $10.4 \geq B \geq 4.8$ in the present work. The error of the approximation of formula (5) did not exceed 13.2%.

Figure 4 illustrates the dependence of the relative heat transfer function $\Psi = (q_w^+ - q_w^-)/q_w^-$ on the intensity of vibrations, where the superscripts + and − correspond to the parameters with and without perturbations. Curves 1–6 (approximations by the least squares method) were obtained for porous materials around which a jet



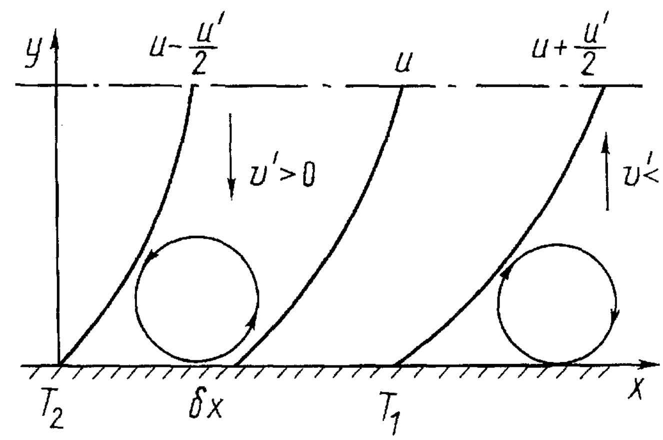


Fig. 4. Dependence of the relative heat transfer function on the intensity of perturbations and the injection parameter.

Fig. 5. Basic scheme of gas flow in pores.

of heated air flowed; dark symbols denote the presence of vibrations of the wall, and light symbols denote pulsations of the gas-coolant; curves 7–9 were obtained for materials around which a plasma jet flowed. For curves 3, 6, 7 the injection parameter was equal to $B = 5.3$; for curves 2, 5, 8 $B = 8.5$; for curves 1, 4, 9 $B = 11.7$.

From the results presented in Fig. 4 it follows that pulsations of the gas-coolant in the ranges of values I studied attenuate the process of heat transfer between the flow and the wall. Wall vibrations may both attenuate heat transfer and enhance it. The degree of the effect of perturbations on the heat transfer process is higher for gas pulsations than for vibrations of the wall and depends substantially on the coolant flow rate. Regimes with $\Psi < 0$ arise for materials with a porosity $\Pi \geq 0.49$. For materials with $\Pi < 0.49$, gas pulsations do not influence Ψ, while wall vibrations lead to $\Psi > 0$, i.e., only enhancement of the process of heat transfer is observed.

Table 1 presents values of Ψ and C_x for models in a supersonic ($M = 1.9$) heated air flow ($T = 382$ K is the stagnation temperature) in the presence of vibrations of the walls.

The values of C_x at $I = 0$ were compared with results of calculations from [12, 13] for a truncated cone with the equation for the generatrix of the surface in a cylindrical coordinate system $x^{10} + r^{10} = 1$. The calculated values of $C_{x\text{calc}}$ are close to those obtained experimentally: for $B = 1.3$, $C_{x\text{calc}} = 1.51$; for $B = 4.1$, $C_{x\text{calc}} = 1.5$; for $B = 8.2$, $C_{x\text{calc}} = 1.37$. The value of $C_{x\text{calc}}$ for a cone with the generatrix $r = z^{0.125}$ at $M = 2.0$, $B = 0$, $I = 0$ also agrees satisfactorily with the experimental results, i.e., $C_{x\text{calc}} = 1.571$.

From Table 1 it is seen that vibrations of a porous wall in a supersonic flow also influence Ψ, with the functions $\Psi(I, B)$ being similar to the functions $\Psi(I, B)$ in Fig. 4. In this case C_x decreases.

Comparison of results on the filtrational and thermal characteristics of porous materials in the presence of pulsational and vibrational perturbations (see Figs. 2 and 4) attests to relative similarity between the dependences of the viscosity term in the filtration law and of the relative heat transfer function on the intensity of vibrations and to the hydrodynamic nature of the process of heat transfer attenuation. The studied ranges of the change in the intensities of perturbations correspond to the parameter $H = \sqrt{\omega/2\nu}\, d \ll 1$, which characterizes the ratio of the "depth of penetration" of viscous waves to the body dimensions [14]. The condition $H \ll 1$ means the penetration of viscous forces far into the gas-coolant during its filtration through the pores.

TABLE 1. Dependence of the Relative Heat Transfer Function and the Wave Resistance Coefficient on the Intensity of Perturbations and the Injection Parameter

B	0	1.3				4.1				8.2			
I, kg/sec^3	0	0	0.2	0.36	0.6	0	0.2	0.36	0.6	0	0.2	0.36	0.6
Ψ	1	1	−0.05	−0.04	−0.02	1	−0.06	−0.05	−0.03	1	−0.07	−0.08	−0.04
C_x	1.52	1.51	1.50	1.46	1.42	1.49	1.43	1.41	1.40	1.37	1.32	1.29	1.28

Suppose the gas velocity in the pores (see Fig. 5) can be represented as the sum $u + u'$, where $u' = \delta x\omega \cos \omega t/2\pi$ is the pulsational component. Gas pulsations lead to the appearance of additional normalfriction stresses $\sigma = \rho \overline{u'^2}$ in the direction of the $0x$ axis, and tangential friction stresses $\tau = -\rho\overline{u'v'}$, [15]. Normal friction stresses are responsible for the appearance of an additional "pulsational" pressure P_{puls}. However, calculations show an insignificant value $P_{\text{puls}} = \rho c\omega\delta x \sim 10^{-3}$ Pa.

The estimates

$$\tau = -\rho\overline{u'v'} = -\rho\overline{\left(u_0 \cos \omega t l \frac{d\bar{u}}{dy}\right)} \simeq -\rho\,(\text{const}\;\omega\delta x^2)\frac{d\bar{u}}{dy}$$

indicate the appearance of an additional "pulsational" coefficient of kinematic viscosity $\nu_k = \text{const}\cdot\omega\delta x^2$; here $l = \text{const}\cdot\delta x$, and const < 1 is an analog of the Prandtl mixing length. Introducing the dimensionless resistance coefficient λ_{res} [15]: $\Delta P/L = \lambda_{\text{res}}\rho\overline{u}^2/2d$, where L, d are the length and diameter of a pore, and comparing the obtained results, for example, with the Hagen-Poiseuille resistance law $\lambda_{\text{res}} = 64/\text{Re}$, we find that gas-coolant pulsations lead to a stronger decrease in λ ($\text{Re}_a > \text{Re}$, where $\text{Re}_a = \overline{u}^2/\omega\nu$) with an increase in Re_a. Consequently, gas pulsations must decrease α (see Fig. 2). Transient and turbulent modes of gas filtration probably arise at vibration intensities $I > 0.14$ ($\text{Re}_a \gtrsim 2000$, $\text{Re}_f > 2.3$) with the penetration characteristics being impaired. The arrows in Fig. 5 illustrate possible mechanisms underlying the appearance of vorticities.

Let us now estimate the additional transfer of heat in the case of periodic gas-coolant pulsations $q = -\rho c_p\overline{U'T'}$. With transition of a gas or a wall from the position T_1 to T_2, $T_1 > T_2$, the following heat flux is transferred into a porous wall in the time $\tau/2$

$$Q = \int_0^{\tau/2} \rho c_p u' T' \,dt + \int_0^{\tau/2} \lambda \frac{\partial T}{\partial x}\,dt. \tag{6}$$

Expression (6) is valid if the time of thermal relaxation $\tau_{\text{calc}} = d^2/a \ll 1$, where a is the thermal diffusivity coefficient. After averaging in the first term of Eq. (6), expansion in a Taylor series in the vicinity of the temperature T_2, $T_2 = T_1 + \partial T/\partial x \delta x$, and the obvious substitutions $\delta x = \int_0^{\tau/2} u' dt$, $\omega = 2\pi/\tau$, we obtain a formula for the heat flux density in a porous wall with gas-coolant pulsations (surface vibrations):

$$\int_0^{\tau/2} \rho c_p u' T' \,dt \cong \bar{\rho} c_p \overline{u'} \int_0^{\tau/2} T' \,dt \simeq \bar{\rho}\,\overline{u'}\,c_p \frac{\tau}{2}(T_1 - T_2),$$

$$q = -\left(\lambda + \frac{\pi c_p \bar{\rho}\,\overline{u'^2}}{\omega}\right)\frac{\partial T}{\partial x}.$$

Consequently, periodic perturbations lead to the appearance of an additional effective heat conduction coefficient and to heat transfer into a porous material. A similar formula for the "transport" of heat into a porous medium in the case of pressure pulsations was obtained in [16]. Calculations made in [16] demonstrate a possible fivefold increase in λ in the case of gas pressure pulsations. Therefore, the surface temperature of a porous material and the heat flux density decrease in the case of wall vibrations and gas-coolant pulsations (see Fig. 4).

The effect of gas-coolant pulsations and wall vibrations on the relative heat transfer function (Fig. 4) is not equivalent. Wall vibrations can turbulize a gas flow near the outer surface of a porous material; then heat transfer enhancement is observed (see curves 8 and 9 in Fig. 4). Measurements of the degree of gas turbulence ε near a porous wall immersed in a gas flow in an MT-324 wind tunnel confirm the fact of gas turbulization by surface vibrations: for $I = 0.20$, 0.38, and 0.51 kg/sec, $\varepsilon = 0.06$, 0.08, and 0.11, respectively.

Thus, the results obtained point to the susceptibility of systems of porous cooling to small periodic perturbations, namely, gas-coolant pulsations and wall vibrations. The susceptibility is probably attributable to the appearance of additional normal and tangential friction stresses during filtration of the gas-coolant through the pores and to additional heat transfer into the porous wall. It provides the possibility of regulating and controlling the hydrodynamic and thermal characteristics of such systems.

NOTATION

P, pressure; ρ, v, gas density and velocity; μ, coefficient of dynamic viscosity; Π, volumetric porosity; D, diameter of the permeable section; R, gas constant; T_0, gas-coolant temperature; ν, coefficient of kinematic viscosity; λ, c_p, ρ_{res}, coefficients of specific thermal conductivity and heat capacity at constant pressure, density of porous materials; h, wall thickness; c, speed of sound; $\tilde{p}_0$, pressure coefficient for the front stagnation point; r_{m}, S_{m}, radius and area of the midsection; x_{cone}, length of the cone generatrix; β_{cone}, angle between the generatrix and the symmetry axis of the cone; S, surface area; M_{w}, M_{e}, molecular weights of the injected gas and the main stream; M, Mach number; Re, Reynolds number; $B = (\rho v)_{\text{w}}/(\alpha/c_p)_0$, $\bar{\alpha}$, heat transfer coefficient. Subscripts: k, e, on the inner and outer sides of the porous material; 0, without coolant injection; w, on the wall; a, acoustic.

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