HEAT TRANSFER INSIDE A POROUS MATERIAL UNDER NONSTATIONARY CONDITIONS

V. N. Kharchenko

Inzhenerno-Fizicheskii Zhurnal, Vol. 15, No. 1, pp. 149–152, 1968

UDC 536.244

The internal heat transfer associated with the cooling of heated porous specimens by cold air has been investigated experimentally.

In experimental and theoretical investigations of cooling systems based on the injection of coolants into the boundary layer it is usually assumed that the temperature of the coolant as it leaves the porous wall is equal to the wall temperature. The published experimental data on heat transfer in porous materials under stationary conditions [1,2] indicate that, in reality, the coolant temperature may be much lower than the wall temperature. There are no published data on internal heat transfer under nonstationary conditions.

The model used in the experiments is shown schematically in Fig. 1. A disk of porous copper was mounted in a textolite housing. Chromel-copel thermocouples were installed inside and outside the model and also spot-welded to both surfaces of the disk. To reduce the inertia of the thermocouples, at the junctions the 0.3-mm wires were rolled to a thickness of about 0.05–0.1 mm. Two specimens of porous materials differing in the size of the fractions (d_1 = 0.1 mm; d_2 = 0.15 mm) were tested. The diameter of the disks was 8 mm, the thickness 3 mm, and the porosity 35%.

We first performed experiments to determine the permeability of the specimens

$$c = \mu\delta \frac{v}{\Delta p}. \tag{1}$$

It was found that for both specimens $c \approx 10^{-11}$ m^2.

In the heat transfer experiments the specimens were first heated in a jet of hot air to ~150° C. Then cold air was passed through them, and the variation of the temperature of the specimen and the air with time was recorded on the chart of a "Geofizika" oscillograph. The air was supplied to the model from a tank through an oxygen reducer, which maintained a constant pressure in the model. The consumption of air was determined from the pressure drop in the tank whose capacity was known.

As a result of the experiments it was established that the temperature of the gas on leaving the porous wall was lower than the wall temperature. In certain cases the difference amounted to 10–15°. Similar results were previously obtained in [1, 2] under stationary conditions. An analysis of the experimental data made it possible to obtain a simple method of calculating the internal heat transfer under the conditions investigated.

The quantity of heat per unit area that the air gives up to the wall in passing through the pores is given by the equation

$$q = c_p \rho v (t_1 - t_2). \tag{2}$$

The internal heat transfer coefficient per unit volume of porous material

$$\alpha = \frac{q}{\delta \Delta t_1}. \tag{3}$$

The mean logarithmic temperature head Δt_1 in Eq. (2) is determined as follows:

$$\Delta t_1 = \frac{(t_4 - t_2) - (t_3 - t_1)}{\ln \dfrac{t_4 - t_2}{t_3 - t_1}}. \tag{4}$$

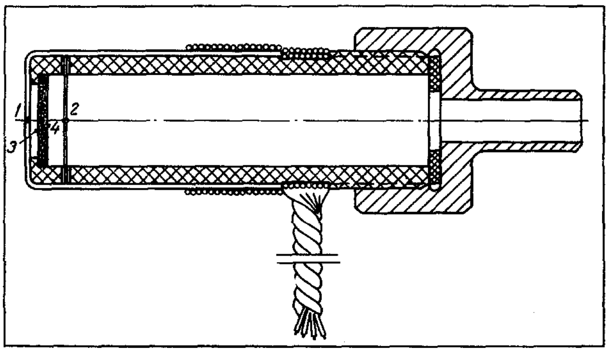

Fig. 1. Diagram of experimental model: 1) thermocouples for measuring the temperature of the gas on leaving the porous wall; 2) inside the model; 3) at outer and 4) inner surfaces of specimen.

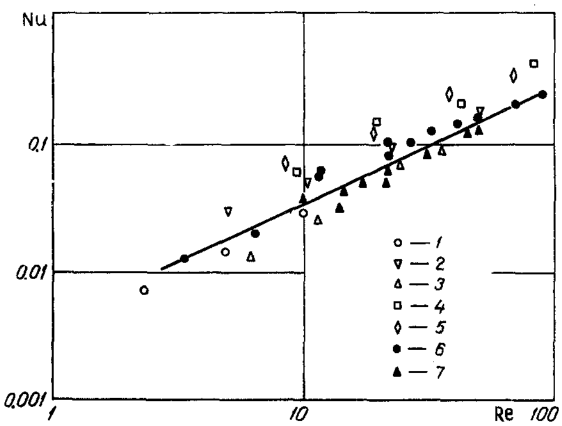

Fig. 2. The relation Nu = f(Re) for heat transfer in a porous material (1–5 according to [1], continuous line according to Eq. (8)).

	1	2	3	4	5	6	7
δ, mm	1.59	3.18	6.45	3.18	6.45	3.	3
d, mm	0.048	0.127	0.127	0.211	0.211	0.1	0.15
Π, %	33.3	39.2	36.4	35.6	39.5	35	35

However, it is difficult to use data analyzed in this way in experiments with porous cooling, since only t_2 and t_3 or t_4 are usually measured. In analyzing the experiments it is possible to employ the quantity

$$\Delta t_2 = \frac{t_4 - t_2}{2}. \qquad (5)$$

In fact, the quantities t_1, t_3, and t_4 are usually similar and much greater than t_2 (under our conditions t_1, t_3, and t_4 varied in the range 100–150° C, while $t_2 \approx$ 20–30° C; the quantities Δt_1 and Δt_2 differed by not more than 10%).

The results of the experiments were generalized in the form

$$\mathrm{Nu} = f(\mathrm{Re}).$$

The Nusselt number was determined as follows:

$$\mathrm{Nu} = \frac{\alpha d^2}{6(1-\Pi)\lambda}. \qquad (6)$$

The parameter $6(1 - \Pi)/\alpha$ determines the area of the surface over which the gas flows per unit volume of material and makes it possible to employ the volume heat transfer coefficient.

The Reynolds number was also determined with allowance for the particular characteristics of the flow in the porous medium:

$$\mathrm{Re} = \frac{\rho v d}{\mu \Pi}. \qquad (7)$$

The results of the experiments are presented in Fig. 2 (dots). The same figure includes the data of [1] obtained for stationary conditions (circles). Our experimental data are closely approximated by the relation

$$\mathrm{Nu} = 0.0042\,\mathrm{Re}^{0.9}. \qquad (8)$$

The results obtained can be used in the experimental determination of the heat transfer at a permeable surface under nonstationary conditions. This method is used in connection with intermittent wind tunnels. In this case the heat flow into the wall is expended in two ways: on heating the wall and on heating the injected gas. Knowing the flow rate of the injected gas, from Eq. (8) we can find the Nusselt number and from (6) and (3) determine the heat flux going toward heating the gas. The temperature of the injected gas as it leaves the wall can then be determined from Eq. (2).

NOTATION

t_1, t_2, t_3, and t_4 are the temperatures of the injected gas on leaving and before entering the porous wall, temperatures at inner and outer wall surfaces, respectively; λ, μ, c_p, ρ, and v are the thermal conductivity, viscosity, specific heat, density, and velocity of the injected gas; ΔP is the pressure drop over the thickness of the porous wall; δ is the thickness of the porous wall; d is the diameter of the sintered fraction; Π is the porosity of the material.

REFERENCES

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20 November 1967

Zhukovskii Central Aerohydrodynamics Institute, Moscow