$Pr = \nu/a$, Prandtl number; $Re = Ud_e/\nu$, Reynolds number; $Gr = g\beta d_e^3 (T_w - T_0)/\nu^2$, Grashof number; $B_1 = Gr/(16d_e/L)$, parameter in momentum equation; α_0, local heat-transfer coefficient calculated from initial temperature head $(T_w - T_0)$; α_m, local heat-transfer coefficient calculated from local temperature head $(T_w - T_m)$; $Nu_0 = \alpha_0 d_e/\lambda$, $Nu_m = \alpha_m d_e/\lambda$, local Nusselt numbers; $Nu_{av} = \int_0^1 Nu_0 dx$; Nusselt number averaged over length of channel; Nu_{0t}, Nu_{0e}, calculated and experimental local Nusselt numbers.

LITERATURE CITED

1. J. Madejski, Bull. Acad. Pol. Sci., Ser. Sci. Tech., 9, 633, 61.
2. R. L. Pigford, Chem. Eng. Progress, Sympos. Ser., 51, No. 17, 79-93 (1955).
3. N. Mitsuishi, O. Miatake, and M. Tanaka, Mem. Fac. Eng. Kyushu Univ., 28, 207 (1969).
4. K. Murakawa, Bull. JSME, 4, 347 (1961).
5. F. Beck, "Wärmeübergang und Druckverlust in konzentrischen und exzentrischen Ringspalten bei erzwunger Strömung und freier Konvektion," Diss., Fak. Maschinenwes. Techn. Hochschule, Karlsruhe (1960).
6. J. R. Bodoia and J. F. Osterle, Trans. ASME, Ser. C., 84, 40 (1962).
7. P. H. G. Allen and A. H. Finn, Trans. ASME, Ser. C, 92, 299 (1970).
8. V. D. Vilenskii, B. S. Petukhov, and B. E. Kharin, Teplofiz. Vys. Temp., 7, No. 5 (1969).
9. W. J. Marner and H. K. McMillan, Trans. ASME, Ser. C, 92, 559 (1970).
10. B. Zeldin and F. W. Schmidt, Trans. ASME, Ser. C, 94, 211 (1972).
11. B. S. Petukov, Heat Transfer and Resistance in the Laminar Flow of a Liquid in Tubes [in Russian], Énergiya, Moscow (1967).
12. S. N. Kanev, Énergomashinostroenie, No. 9, 36 (1975).

INTENSIFICATION OF HEAT EXCHANGE IN BUNDLES OF RODS LONGITUDINALLY BATHED BY GAS

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Functions for the calculation of heat exchange and hydraulic resistance in bundles of rods with artificial roughness are presented and the optimum dimensions and shape of the roughness are indicated.

The creation of high-intensity thermoenergetic devices requires the intensification of the processes of heat exchange. This problem is particularly urgent for devices having gas cooling, since gas coolants, while having a number of advantages over others (safety in operation, the possibility of use in atomic gas-turbine installations, etc.), have drawbacks connected with the low density, small heat capacity, and low coefficient of thermal conductivity.

Various means exist for the intensification of gaseous heat exchange. One of the simplest means is an increase in the velocity of the coolant, but the possibilities of this means are limited, since with an increase in the velocity of the coolant the resistance and the power for pumping the coolant grow simultaneously with the increase in the heat-transfer coefficient. Another means of intensification of heat exchange is based on the use of more

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ideal constants, consisting of two-phase systems (liquid—gas, liquid—vapor, gas—solid particles) or gas mixtures, since the heat-transfer properties of high-molecular-weight gases (CO_2, C_6F_6) can be improved through their mixing with highly thermally conducting helium or hydrogen in small amounts, such as $0.1He + 0.9CO_2$. The use of mixtures of nonreactive gases allows one to improve a number of indices of gas-turbine installations.

One of the promising means of intensification of heat exchange in bundles of rods longitudinally bathed by gas is the use of artificial roughness on the surfaces of the rods. The increase in the heat-transfer coefficient obtained in this case is also accompanied by an increase in friction, since turbulent gas flow on rough walls differs considerably from flow in smooth pipes. As is known, the change in the coefficients of resistance of rough pipes is determined by the stream velocity. If the Reynolds numbers are relatively low, then the thickness of the laminar layer turns out to be greater than the height of the roughness protuberances, and therefore the roughness does not affect the flow. With an increase in the Reynolds numbers and a decrease in the laminar sublayer the roughness begins to play the main role in the formation of the regime. The quantitative change in the hydraulic resistance in pipes in the presence of roughness proposed by M. D. Millionshchikov comes down to the application of the principle of superposition for the total viscosity in the boundary layer

$$\Sigma \nu_i = \nu + a(y - \delta_0) v_* + a(k - \delta_0)^* v_*, \tag{1}$$

where ν is the molecular viscosity; $a(y - \delta_0)v_*$ is the turbulent viscosity determined by the distance to the wall; $a(k - \delta_0)^* v_*$ is the turbulent viscosity determined by the roughness protuberances; k is the height of the roughness protuberances; v_* is the scale of the pulsation velocity; a is a constant; and δ is the dimensionless thickness of the laminar sublayer. An asterisk signifies the necessity of allowing for the diffuse nature of the roughness—laminar-sublayer boundary, a consequence of the mean-square deviation from the average quantity $\bar{k}$.

For two-dimensional roughness the coefficient of resistance can be regulated by the shape of the protrusions, their spacing-to-height ratio, and the degree of roughness of the rods. A study conducted by Pavlovskii [3] on the effect of the configuration of turbulizers on the thermal efficiency of a heating surface in a flat channel showed that the effect of the turbulizer configuration (turbulizers of rectangular, triangular, semicircular, and drop-shaped profiles were studied) has little effect on the heat-exchange intensity in a channel but a very important effect on the hydraulic resistance; in this case it was established that a heating surface with turbulizers of drop-shaped profile, characterized by the smallest coefficient of frontal resistance, is the best with respect to the power coefficient.

According to the data of Kandelaki [4] and Migai [5], the optimum ratio of the spacing of the protrusions to the height equals 13 for Re up to $3 \cdot 10^5$ and Pr = 3-225, while, according to the data of [6] it equals 10 for air. On the basis of experimental studies of [7], other experimental studies conducted for Re = 10^5-$3 \cdot 10^5$ (of interest for cooling by helium), and the optimum roughness geometry, one can propose the following simple empirical ratio of the friction factors for rough and smooth rods:

$$\frac{f_r}{f_s} = \frac{\left(\frac{St_r}{St_s}\right)^3}{1 + \frac{5}{3}\left(\frac{St_r}{St_s} - 1\right)} \approx \left(\frac{St_r}{St_s}\right)^{1.55}, \tag{2}$$

where f_r and f_s are the coefficients of friction for rough and smooth rods, respectively; St_r and St_s are the Stanton numbers for rough and smooth rods. Thus, it is determined that for the indicated conditions the doubling or tripling of the Stanton number increases the friction factor by 3 or 6.24 times, respectively (Fig. 1). At values of Re $\cong 10^6$ we have

$$\frac{f_r}{f_s} \approx \left(\frac{St_r}{St_s}\right)^2 \quad \text{for} \quad 1 \leqslant \frac{St_r}{St_s} \leqslant 3. \tag{3}$$

In many cases, in order to decrease the frictional losses it is desirable to make the surface of the rods partially rough at places where it is necessary to reduce the maximum temperature of the envelope. Since the maximum surface temperature of smooth rods T_{m2} (Fig.

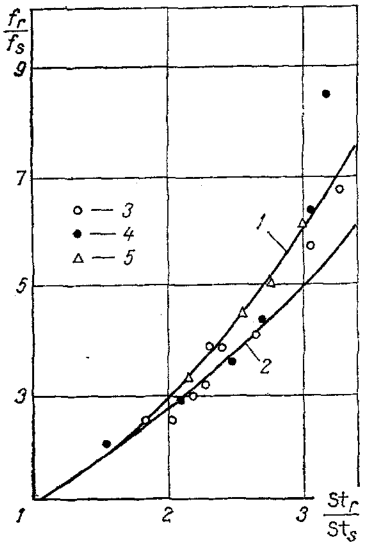

Fig. 1

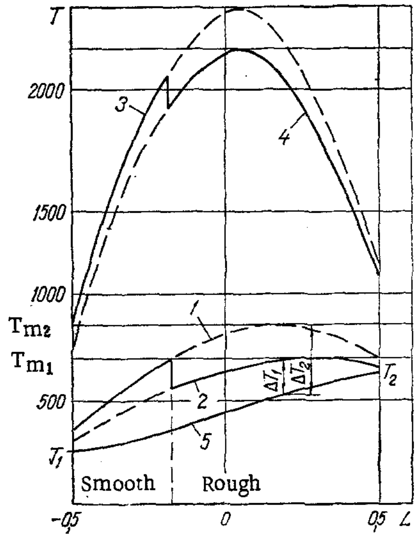

Fig. 2

Fig. 1. Properties of optimum surface roughness: 1) $\frac{f_r}{f_s}=\frac{\left(\frac{St_r}{St_s}\right)^3}{1+\frac{5}{3}\left(\frac{St_r}{St_s}-1\right)}$; 2) $\left(\frac{St_r}{St_s}\right)^{\frac{3}{2}}$; 3) Re = 10^5; 4) $2\cdot10^5$; 5) $3\cdot10^5$.

Fig. 2. Temperature distributions in the case of partial roughness of rods: 1, 2) at rod surface; 3, 4) at center of rod; 5) of coolant (T_1 at inlet, T_2 at outlet); L) axial distance.

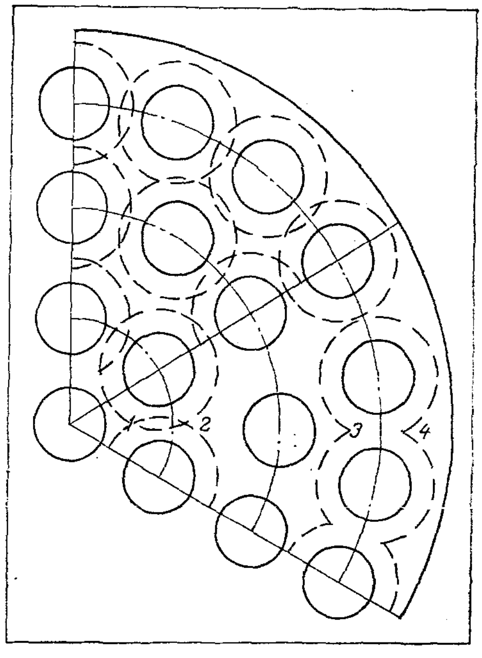

Fig. 3. Arrangement of bundle: 1, 2, 3, 4) zones; dashed lines) surfaces with zero shear stress.

2) is not allowable for technological reasons, a rod is made rough enough along the length so that the maximum surface temperature of the rod in this region is equal to the maximum allowable temperature T_{m1}. Curve 1 changes into curve 2 and curve 3 changes into curve 4 [8]. The optimum heat-transfer coefficients are obtained in the case when 64% of the surface of all the rods is made rough and 36% smooth; in this case the Stanton number for the rough part is doubled in comparison with that for the smooth part.

A section of a bundle with a characteristic arrangement of rods is shown in Fig. 3; the area of the stream is arbitrarily divided into "annular zones" by the surface with zero shear stress. The method of division of the area of the stream into two regions [7] is based on the assumption that they are independent of each other, which makes it possible to calculate them separately. In the given case the correction for the effect of roughness is made by multiplying the ordinary coefficient of friction $f_s = 0.046\ Re^{-0.2}$ for a smooth rod surface by a correction factor b, which according to some data [9] approaches unity (for the first zone, b = 1.119; for the second, b = 1.164; and for the fourth, b = 1.264), while according to other data b = 1.4.

The Stanton number in a bundle with $Re = 3.3\cdot10^5–6.8\cdot10^5$ varies over the zones in the following way: In the first zone (Fig. 3) $St = 0.0275\cdot Re^{-0.1242}$; in the second zone $0.380\cdot Re^{-0.1491}$; in the third zone $0.401\cdot Re^{-0.1549}$; in the fourth zone $0.0419\cdot Re^{-0.1672}$; for a single rod $0.0539\cdot Re^{-0.179}$.

The values of the zonal coefficients of friction and the Stanton numbers depend on the distribution of the coolant stream among the zones of the bundle. Tests showed [7] that in the fourth zone the values of the coefficients of friction and the Stanton numbers are lower than expected from the equation presented above. This disagreement may be caused by the effect of the shape of the channels, momentum transfer between adjacent zones of coolant, and distortion of the stream by the components which support the bundle.

LITERATURE CITED

1. M. D. Millionshchikov, in: Turbulent Flows [in Russian], Moscow (1974), p. 5.
2. M. D. Millionshchikov, V. I. Subbotin, M. Kh. Ibragimov, G. S. Taranov, and L. L. Kobzar', Atom Énerg., 34, Part 4 (1973).
3. V. G. Pavlovskii, Inzh.-Fiz. Zh., 17, No. 1 (1969).
4. R. D. Kandelaki, Author's Abstract of Candidate's Dissertation, Sukhumi (1969).
5. V. K. Migai, Author's Abstract of Doctoral Dissertation (1974).
6. W. A. Sutherland, Intern. J. Heat Mass Transfer, 10, 1589 (1967).
7. G. Melise-d'Hospital, Nucl. Sci. Eng., 3, 271 (1968).
8. E. R. Christensen, J. Nucl. Energy, 24, 11 (1970).
9. W. Walker, L. White, and P. Burnett, Intern. J. Heat Mass Transfer, 15, 403 (1972).