The average friction coefficient on an impermeable surface at the section L was calculated from the friction law of [2] $C_{f0} = 0.0256\ \mathrm{Re}^{**-0.25}$ using the formula

$$C_{f0} = \frac{0.0256\,(\mathrm{Re}_2^{**0.75} - \mathrm{Re}_1^{**0.75})}{0.75\,(\mathrm{Re}_2^{**} - \mathrm{Re}_1^{**})}.$$

The solid lines in Fig. 3 show the results of calculations from Eq. (11) for various values of $j_+/j_-$. It can be seen from Fig. 3 that there is satisfactory agreement between the relation obtained and the test data for simultaneous mass blowing and suction at the wall.

## NOTATION

$\bar{j}_+$, $\bar{j}_-$, relative mass velocity of blowing and suction gas, respectively; $\delta_1^{**}$, $\delta_2^{**}$, momentum loss thickness at the start and the end of the perforated section, m; $\omega = u/u_0$, relative longitudinal flow velocity; $C_{f0}$, friction coefficient on an impermeable surface; $C_{f\pm}$, $C_{f\Sigma\pm}$, friction coefficient on a flat plate with simultaneous equal and unequal mass blowing and suction; $\Psi_\pm = C_{f\pm}/C_{f0}$, $\Psi_{\Sigma\pm} = C_{f\Sigma\pm}/C_{f0}$, relative friction law with equal and unequal mass blowing and suction; $\tau_+$, $\tau_-$, turbulent friction in a boundary layer with separate mass blowing and suction, respectively; $\mathrm{Re}_1^{**} = u_0\delta_1^{**}/\nu$, $\mathrm{Re}_2^{**} = u_0\delta_2^{**}/\nu$, Reynolds number at the start and the end of the perforated section; $b_+ = 2\bar{j}_+/C_{f0}$, $b_- = 2\bar{j}_-/C_{f0}$, $b_\pm = 2\bar{j}_\pm/C_{f0}$, permeability parameters.

## LITERATURE CITED


1. A. I. Leont'ev, V. G. Puzach, V. P. Komarov, et al., "Friction on the surface of a plate with simultaneous blowing and suction of gas," Inzh.-Fiz. Zh., 33, No. 2, 204-209 (1977).
2. S. S. Kutateladze and A. I. Leont'ev, Heat Transfer and Friction in the Turbulent Boundary Layer [in Russian], Énergiya, Moscow (1972).


# EXPERIMENTAL INVESTIGATION OF HEAT EXCHANGE IN A POROUS ANNULAR CHANNEL


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The problems connected with the intensification of heat exchange in an annular channel with a porous inner wall are considered. A method for calculating a "tube-in-tube" heat exchanger with a porous inner tube is described.


Various methods can be used for increasing the heat transfer coefficient and thereby intensifying the heat exchange. For instance, transverse flow of the heat-transfer agent through a porous wall can be provided. The transverse flow of the agent mass can be directed from the heated heat-transfer agent to the heating agent, and vice versa. For this, it is necessary to maintain a suitable pressure drop in the heat-transfer agents. In this case, the medium is drawn off continuously along the porous wall immersed in the flow on the part of one heat-transfer agent, while injection occurs on the part of the other agent. By

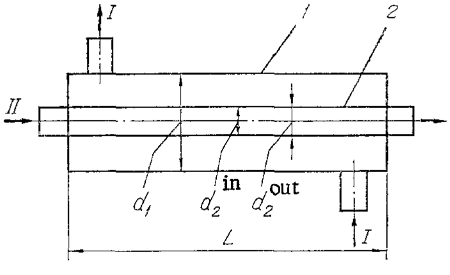


Fig. 1. Schematic diagram of the heat exchanger with an inner porous tube (I and II are the heated and the heating heat-transfer agents, respectively). 1) Outer tube; 2) inner porous tube.


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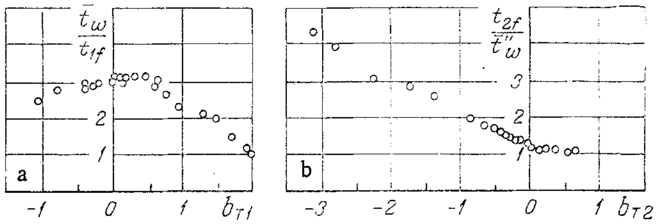


Fig. 2. Relationships $\bar{t}_w/t_{1f} = f_1(b_{T1})$ (a) and $t_{2f}/\bar{t}''_w = f_1(b_{T2})$ (b).

ensuring suction of the heat-transfer agent from the side of the wall with a lower value of the heat transfer coefficient and injection of a portion of this agent into the medium moving along the surface of the same wall with a higher value of the heat transfer coefficient, we can increase the lower, and reduce the higher, heat transfer coefficient and secure a higher overall heat transfer coefficient without an appreciable additional expenditure of energy.

We investigated earlier [1] a heat exchanger of the tube-in-tube type (Fig. 1). The heat exchanger consists of an outer tube 1, whose inside diameter is equal to $d_1 = 106$ mm, and an inner porous tube 2, with the inside diameter $d_2^{in} = 34$ mm and the outside diameter $d_2^{out} = 40$ mm. The length of the heat exchanger is equal to $L = 1017$ mm. Tube 2 is made of ten equal porous cylinders with a length of 100 mm, separated by means of thermal insulation rings. The permeability of the cylinders in their middle sections is higher by approximately 20% than at their ends. The porous tube is made of granulated powder of Kh18N10 stainless steel, 0.063-mm fraction, while the outer tube is made of steel (st. 20) and is covered on the outside with a thermal insulation layer. Air serves as the heat-transfer agent. The air discharge values at the inlet and outlet of the heat exchanger with respect to the heating as well as the heated agents were determined by means of chamber-type measuring diaphragms. The temperature of the walls and the heat-transfer agents was recorded by means of Chromel—Alumel thermocouples. Moreover, we tested a similar heat exchanger, which had an inner tube made of ordinary steel (st. 20).

The operation of the heat exchangers is based on the counterflow principle: The heating heat-transfer agent moves along tube 2, while the heated agent moves along tube 1. The following quantities were varied in the experiments: $Re_{1f} = (5.6\text{–}57)\cdot 10^3$ or $(Re_{1f})_L = (8.7\text{–}88)\cdot 10^4$, $t_1 = 16\text{–}23$°C; $Re_{2f} = (1.6\text{–}5.2)\cdot 10^4$ or $(Re_{2f})_L = (4.8\text{–}15.6)\cdot 10^5$, $t_2 = 120\text{–}150$°C.

The present article describes a method of determining the quantity of heat transferred in a heat exchanger with a porous inner tube. The heat balance equation is used in processing the experimental data:

$$Q_{po} = c'_{p1}G'_1 t'_1 - c_{p1}G_1 t_1 + Q_{loss} = c_{p2}G_2 t_2 - c'_{p2}G'_2 t'_2 . \tag{1}$$

The mean values of the heat transfer coefficients $K_{L1po}$ and $K_{L2po}$ are determined with respect to the quantity of heat transferred in the heat exchanger $Q_{po}$:

$$K_{L1po} = \frac{Q_{po} \mp c_{pw} G_w \bar{t}_w}{\Delta t L \pi}, \tag{2}$$

$$K_{L2po} = \frac{Q_{po} \mp c''_{pw} G_w \bar{t}''_w}{\Delta t L \pi}. \tag{3}$$

The mean temperature drop $\Delta t$ is determined from the relationship

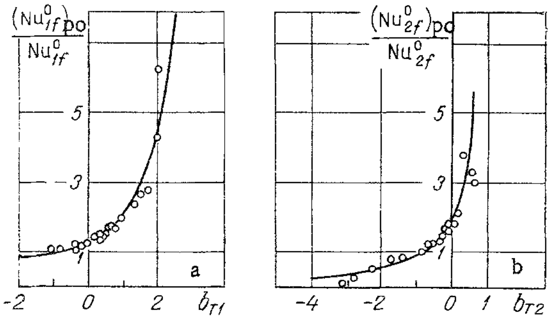


Fig. 3. Relationships $(Nu^o_{1f})_{po}/Nu^o_{1f} = f_2(b_{T1})$ (a) and $(Nu^o_{2f})_{po}/Nu^o_{2f} = f_2(b_{T2})$ (b).

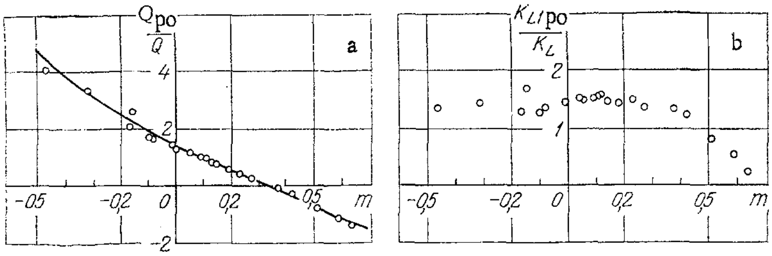

Fig. 4. Relationships $Q_{po}/Q = f_1(m)$ (a) and $K_{L1po}/K_L = f_2(m)$ (b).

$$\Delta t = \frac{t_2 + t_2'}{2} - \frac{t_1 + t_1'}{2} = t_{2f} - t_{1f}. \tag{4}$$

The experimentally determined temperature distributions over the inner and outer surfaces of the tube are used to find the mean temperatures of the porous wall on these surfaces by means of the equations

$$\bar{t}_w = \frac{1}{F_w} \int_0^{F_w} t_w dF, \tag{5}$$

$$\bar{t}_w'' = \frac{1}{F_w''} \int_0^{F_w''} t_w'' \, dF. \tag{6}$$

The mean discharge of transverse air flow $G_w$, averaged over the surface of the porous tube, is calculated as the difference between the heat-transfer agent discharges at the inlet and outlet of the heat exchanger.

The mean values of the heat transfer coefficients $\alpha_{1po}$ and $\alpha_{2po}$ are determined from the expressions

$$\alpha_{1po} = \frac{Q_{po} \mp c_{pw} G_w \bar{t}_w}{F_w \Delta t_{w1}}, \tag{7}$$

$$\alpha_{2po} = \frac{Q_{po} \mp c_{pw} G_w t_w''}{F_w'' \Delta t_{w2}}. \tag{8}$$

The mean temperature drops $\Delta t_{w1}$ and $\Delta t_{w2}$ are found by means of the expressions

$$\Delta t_{w1} = \bar{t}_w - t_{1f}, \tag{9}$$

$$\Delta t_{w2} = t_{2f} - \bar{t}_w''. \tag{10}$$

The experimental data are processed in the form of the relationships

$$Q_{po}/Q = f_1(m), \tag{11}$$

$$K_{L1po}/K_L = f_2(m), \tag{12}$$

$$\frac{\bar{t}_w}{t_{1f}} = f_1(b_{T1}); \quad \frac{t_{2f}}{\bar{t}_w''} = f_1(b_{T2});$$

$$\frac{(\mathrm{Nu}_{1f}^0)_{po}}{\mathrm{Nu}_{1f}^0} = f_2(b_{T1}); \quad \frac{(\mathrm{Nu}_{2f}^0)_{po}}{\mathrm{Nu}_{2f}^0} = f_2(b_{T2}),$$

where

$$m = n_f \left(\frac{\mathrm{Re}_{1f}}{\mathrm{Re}_{2f}} + 1\right); \quad n_f = \frac{2G_w}{G_1 + G_1'};$$

$$b_{T1} = \frac{J_{1f}}{\mathrm{Nu}_{1f}^0} \mathrm{Re}_{1f}\mathrm{Pr}_{1f}; \quad b_{T2} = \frac{J_{2f}}{\mathrm{Nu}_{2f}^0} \mathrm{Re}_{2f}\mathrm{Pr}_{2f};$$

$$J_{1f} = \frac{\rho_{wf} v_{wf}}{\rho_{1f} u_{1f}}; \quad J_{2f} = \frac{\rho_{wf}'' v_{wf}''}{\rho_{2f} u_{2f}};$$

$$u_{1f}=\frac{G_1+G_1'}{2F_1\rho_{1f}};\ u_{2f}=\frac{G_2+G_2'}{2F_2\rho_{2f}};\ v_{wf}=\frac{G_w}{\rho_{wf}F_w};$$

$$v''_{wf}=\frac{G_w}{\rho''_{wf}F''_w};\ \mathrm{Re}_{1f}=\frac{(G_1+G_1')\,d_{\text{equiv}}}{2F_1\mu_{1f}};$$

$$d_{\text{equiv}}=d_1-d_2^{\text{out}};\ \mathrm{Re}_{2f}=\frac{(G_2+G_2')\,d_2^{\text{in}}}{2F_2\mu_{2f}};$$

$$(\mathrm{Nu}^0_{1f})_{\text{po}}=\frac{\mathrm{Nu}^{\text{po}}_{1f}}{\overline{T}^{0,5}_{w1}}, \tag{13}$$

$$(\mathrm{Nu}^0_{2f})_{\text{po}}=\frac{\mathrm{Nu}^{\text{po}}_{2f}}{\overline{T}^{0,5}_{w2}}, \tag{14}$$

$$\mathrm{Nu}^{\text{po}}_{1f}=\frac{\alpha_{1\text{po}}d_{\text{equiv}}}{\lambda_{1f}}, \tag{15}$$

$$\mathrm{Nu}^{\text{po}}_{2f}=\frac{\alpha_{2\text{po}}d_2^{\text{in}}}{\lambda_{2f}}. \tag{16}$$

Figure 2 shows the behavior of the relative wall temperature for the heated agent (a) and its reciprocal value for the heating agent (b), respectively, as a function of the parameters $b_{T1}$ and $b_{T2}$. The parameters $b_{T1}$ and $b_{T2}$ have positive values when a portion of the heat-transfer agent is drawn off through the porous wall, and negative values when injection takes place. It is evident that, for large positive values of $b_{T1}$ (Fig. 2a), i.e., when the heated air reaches the heating air through the porous wall, and suction occurs on the part of the heated heat-transfer agent, while injection takes place on the part of the heating agent, the mean wall temperature $\bar{t}_w$ is virtually equal to the mean temperature of the heated air $t_{1f}$. This case corresponds to a quasi-uniform temperature distribution over the cross section of the heated-agent flow. In the case of suction of the heating agent, when it enters the heated agent, which corresponds to positive values of $b_{T2}$ (Fig. 2b), the mean wall temperature $\bar{t}''_w$ approaches the mean flow temperature with an increase in $b_{T2}$, i.e., as in the preceding case, the temperature distribution over the cross section of the heating-agent flow becomes more uniform. For negative values of $b_{T1}$ and $b_{T2}$, i.e., in the case of injection, the temperature distribution over the cross section of the heating flow becomes less uniform with a reduction in $b_{T2}$ (i.e., an increase in its absolute value) and is slightly equalized over the cross section of the heated flow with a reduction in $b_{T1}$. This difference between the variations in the temperature distributions for the heating and the heated heat-transfer agents with a decrease in $b_{T2}$ and $b_{T1}$ in the range of their negative values can be explained by the different rates of variation of $\bar{t}''_w$, $t_{2f}$ and $t_{1f}$, $\bar{t}_w$ with changes in $b_{T2}$ and $b_{T1}$.

The experimental data for the device with smooth walls are adequately described by the following relationships:

$$\mathrm{Nu}^0_{1f}=0.72\,\mathrm{Re}^{0.56}_{1f}, \tag{17}$$

$$\mathrm{Nu}^0_{2f}=0.178\,\mathrm{Re}^{0.8}_{2f}. \tag{18}$$

Here $\mathrm{Nu}^0_{1f}$ and $\mathrm{Nu}^0_{2f}$ are the mean values of the Nusselt number. The experimental data were processed by means of (17) and (18) for the $\mathrm{Re}_{1f}$ and $\mathrm{Re}_{2f}$ values used in experiments on the porous-tube heat exchanger. The results of this processing are given in Fig. 3. It is evident from the diagrams that suction of the heat-transfer agent promotes more intensive heat exchange, while the blowing-out of the agent tends to reduce the heat exchange, although the Nusselt number varies to different degrees for the first and the second heat exchanger, depending on the dimensionless parameter of the transverse air flow. The values of the Nusselt number in the absence of transverse air flow, but in the presence of a porous wall, are larger than the Nusselt numbers for the same Reynolds numbers in the presence of a solid wall. The experimental data for a heat exchanger with a porous wall are adequately approximated by equations expressing the mean Nusselt numbers for the first and the second heat-transfer agents:

$$(\mathrm{Nu}^0_{1f})_{\text{po}}=\mathrm{Nu}^0_{1f}\,(1.1^{b_{T1}}+3.5^{b_{T1}-1}), \tag{19}$$

$$(Nu^0_{2f})_{po} = Nu^0_{2f}(1.3^{bT2} + 10^{bT2}). \tag{20}$$

The measurement data for relationships (11), (12), and (19), (20), are processed in the following manner: Using the $Re_{1f}$ and $Re_{2f}$ values known from experiments on the heat exchanger with a porous wall, we determine by means of Eqs. (17) and (18) the values of $Nu^0_{1f}$ and $Nu^0_{2f}$. With a correction for the temperature factor, we obtain

$$Nu_{1f} = Nu^0_{1f}\overline{T}_{w1}^{0.5} \text{ and } Nu_{2f} = Nu^0_{2f}\overline{T}_{w2}^{0.5}.$$

Furthermore, we use the found values of $Nu_{1f}$ and $Nu_{2f}$ to determine the mean values of $\alpha_1$ and $\alpha_2$ by means of relationships (15) and (16). The value of $K_L$ is determined with respect to the known heat-transfer resistance of the wall, as for an ordinary tube-in-tube heat exchanger:

$$K_L = \frac{1}{\left(\frac{1}{2\lambda_w}\ln\frac{d_2^{out}}{d_2^{in}} + \frac{1}{\alpha_1 d_2^{out}} + \frac{1}{\alpha_2 d_2^{in}}\right)} = \frac{1}{E}. \tag{21}$$

The values of Q are determined from the expression

$$Q = K_L \Delta t L \pi.$$

Figure 4a illustrates the relationship $Q_{po}/Q = f_1(m)$. The experimental data are adequately described by the dependence

$$Q_{po} = Q(2\cdot 10^{-0,8m} - 0,55\cdot 10^{0,8m}). \tag{22}$$

The form of this relationship depends on the heat-transfer resistance of the wall, which figures as a component in the heat transfer coefficient in calculating Q. In the case of low heat-transfer resistance of the wall, Eq. (22) can be used for determining the quantity of heat transferred in a heat exchanger with transverse flow of a portion of one of the agents. Figure 4b shows the relationship $K_{L1po}/K_L = f_2(m)$.

Analyzing the data given in Fig. 4a and b, we note the following.

For small values of m (from 0.5 to −0.5), the heat transfer coefficient $K_{L1po}$ exceeds the value of the heat transfer coefficient for an ordinary heat exchanger. If a portion of the heated heat-transfer agent enters the heating agent, i.e., when suction takes place on the part of the heated agent, while injection takes place on the part of the heating agent, the transferred quantity of heat $Q_{po}$ is larger in a heat exchanger with an inner porous tube than in an ordinary heat exchanger if $m = 0$–$0.1$. This is explained by the fact that the velocity and temperature profiles in the transverse cross sections of the heat exchanger vary for both the heating and the heated agents. In this, the heat transfer coefficient on the part of the heated agent increases, while it decreases on the part of the heating agent, and the overall heat transfer coefficient increases, so that the quantity of transferred heat also increases. If $m > 0.12$, the quantity of heat transferred in a heat exchanger with a porous wall is smaller than that in a simple heat exchanger, while if $m > 0.36$, the $Q_{po}/Q$ ratio becomes negative, since, in this case, there is a quantity of heat transferred to the heating agent with the transverse flow of the heated air mass, such that the total enthalpy of the heating air is larger at the outlet of the heat exchanger than at its inlet, while the enthalpy of the heated air at the outlet of the heat exchanger is smaller than at the inlet.

If a portion of the heating heat-transfer agent enters the heated agent, i.e., when injection occurs on the part of the heated air, and suction takes place on the part of the heating air, the value of $Q_{po}$ increases with a reduction in m (with an increase in its absolute value). In this case, the heat transfer is greatly affected by the large quantity of heat transferred with the transverse flow to the heated agent. If all of the heating agent enters the heated agent, or, on the contrary, the heated agent enters the heating agent, we obtain an ordinary heat exchanger of the mixing type. It should be mentioned that also in the absence of transverse flow for $m = 0$, the heat exchange is more intensive in the presence of a porous wall, as is indicated by Fig. 4a. Thus, the heat transferred in a heat exchanger with a porous inner tube and a counter-flow arrangement can be calculated in the following manner: We determine $Re_{1f}$ and $Re_{2f}$ with respect to the known structural dimensions of the heat exchanger for the assigned parameters of the heat-transfer agents and the wall temperature and then the values of $Nu^0_{1f}$ and $Nu^0_{2f}$, by means of Eqs. (17) and (18), after which we find $(Nu^0_{1f})_{po}$ and $(Nu^0_{2f})_{po}$ by using relationships (19) and (20). We then determine the

Nusselt numbers by taking into account the temperature factor, using expressions (13) and (14), and then the mean values of the heat transfer coefficients $\alpha_{1po}$ and $\alpha_{2po}$ from expressions (15) and (16). Using the known heat-transfer resistance of the wall material, we find the mean value of the heat transfer coefficient $K_{L1po}$ or $K_{L2po}$ from the relationships

$$K_{L1po}=\frac{1}{E_{po}}\left(1\pm\frac{\Delta Q_w}{\Delta t L\pi\alpha_{2po}d_2^{in}}\right), \tag{23}$$

$$K_{L2po}=\frac{1}{E_{po}}\mp\frac{\Delta Q_w}{\Delta t L\pi}\left(1-\frac{1}{\alpha_{2po}d_2^{in}E_{po}}\right), \tag{24}$$

where

$$\Delta Q_w=c_{pw}G_w(\bar{t}_w''-\bar{t}_w); \tag{25}$$

$$E_{po}=\frac{1}{2\lambda_w}\ln\frac{d_2^{out}}{d_2^{in}}+\frac{1}{\alpha_{1po}d_2^{out}}+\frac{1}{\alpha_{2po}d_2^{in}}. \tag{26}$$

The quantity of heat transferred in the heat exchanger is determined by means of the equation

$$Q_{po}=K_{L1po}\Delta t L\pi\pm c_{pw}G_w\bar{t}_w \tag{27}$$

or

$$Q_{po}=K_{L2po}\Delta t L\pi\pm c_{pw}''G_w\bar{t}_w''. \tag{28}$$

In relationships (2), (3), (7), (8), (23), (24), (27), and (28), the upper sign holds when a portion of the heating agent is drawn off, so that it enters the heated agent, while the lower sign applies when the direction of the transverse flow is reversed, i.e., when the heated agent is drawn off.

Analysis of the obtained results indicates that an inner porous tube makes it possible to intensify considerably the heat exchange without, as is shown by experiments, appreciable energy expenditure. In the case of transverse flow, one of the heating surfaces is automatically cleansed from impurities when one of the agents from which mechanical impurities have been eliminated is injected into the contaminated heat-transfer agent. However, the above method of heat exchange intensification can be used only if technological conditions allow partial overflow of one heat-transfer agent into another.

## NOTATION

$c_p$, specific heat of the heat-transfer agents; F, heat exchange surface; $F_1$ and $F_2$, surface area of the annular cross section between $d_1$ and $d_2^{out}$ and the cross-sectional surface area of the porous tube with respect to the inside diameter, respectively; $F_W$ and $F_W''$, surface areas of the porous tube with respect to the diameters $d_2^{out}$ and $d_2^{in}$, respectively; G, mass discharge of the heat-transfer agents; J, relative transverse flow discharge of the heat-transfer agents; $K_L$, heat transfer coefficient; L, length of the porous tube; $n_f$, relative discharge of the agents through the wall; m and $b_T$, dimensionless parameters of the transverse gas flow; u, gas velocity; Q, quantity of transferred heat; $Q_{loss}$, heat loss to the ambient; T, temperature of the heat-transfer agents across the wall; $\bar{T}_W = T_f/T_W$, temperature factor; t, temperature of the heat-transfer agents in the heat exchanger's transverse sections; $v_W$, velocity of the transverse gas flow through the porous wall; $\alpha$, heat transfer coefficient on the part of the heat-transfer agents; $\lambda$, thermal conductivity coefficient; $\mu$, dynamic viscosity of the heat-transfer agents; Re, Reynolds number; Nu, Nusselt number; Pr, Prandtl number; $\rho$, density of the heat-transfer agents. Subscripts or superscripts: f, mean flow parameters; po, flow parameters in the heat exchanger with a porous inner tube; 0, flow parameters for a temperature factor equal to unity; w, parameters of the transverse flow through a porous wall; 1 and 2, heated and heating heat-transfer agents, respectively; the prime denotes the flow parameters at the outlet from the heat exchanger, while the double prime denotes the parameters of the flow through the surface of the porous tube $F_W''$.

## LITERATURE CITED

1. P. N. Romanenko, V. Z. Kitaev, and A. Ya. Voloshchuk, "Intensification of heat exchange in gas—gas heat exchangers," Khim. Promyshlennost', No. 10, 62-66 (1975).