STUDY OF HYDRAULIC RESISTANCE AND HEAT TRANSFER IN PERFORATED-PLATE HEAT EXCHANGERS

S. A. Shevyakova and V. K. Orlov

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An experimental study is made of the hydraulic resistance and heat transfer in heat exchangers made of perforated plates with different internal geometries. Generalizing theoretical relations are obtained.

The improvement of industrial heat exchangers is directed toward increasing the intensity of heat transfer and making the heat-exchanging surface more compact.

Recently, along with study of ribbed-plate heat exchangers, considerable attention has been given to investigation of structures with a heat-exchange surface with a structure which is discontinuous in the direction of motion of the working flow. The principal elements of such surfaces are metal grids and perforated plates made of materials, such as copper and aluminum, with a high thermal conductivity.

The surface of the heat exchanger is formed by alternating perforated plates and intervening spacers assembled into a packet and oriented normal to the flow of heat carrier. The large number of geometric parameters of a heat-exchanging surface made of perforated plates — such as the thickness and spacing of the plates, the shape and dimensions of the holes, and the spacing of the holes — has led to a variety of designs being developed for such heat exchangers. Heat-exchanger surfaces made of perforated plates which have been studied to date embrace a wide range of geometry of the elements. Circular, square, and slit perforations were used in the experimental heat exchangers in [1-5]. The thickness of the perforated plates was 0.155-1.6 mm, the distance between them was 0.11-3.12 mm, and the compactness of the surface of the heat exchanger was located in the range 426-6320 m^2/m^3.

Analysis of the results of studies of hydraulic resistance and heat exchange showed that the data obtained by different authors for these geometrical ranges differed significantly in a quantitative sense and, in several instances, in a qualitative sense. This is a consequence of the different combinations of geometric parameters used. Comparisons among these combinations are often made difficult by the absence of complete information on the geometry of the heat-exchange surfaces and the experimental conditions. Thus, the application of these results to heat exchangers with a geometry different from those studied is invalid.

The effect of the geometric parameters of heat-exchange surfaces on hydraulic resistance and heat exchange is best studied on models made by a single method and studied under identical conditions.

The present investigation studies the thermohydraulic characteristics in heat exchangers with perforated plates with a circular perforation. Here we varied the diameter of the holes and the distance between the plates.

The experimental heat exchangers were made of perforated copper and aluminum (alloy AD1) plates 0.47-0.5 mm thick separated by glued intervening plates. The perforation consisted of circular holes of 0.625, 0.9, and 1.65 mm diameter located at the corners of squares. The plates were assembled into a packet, with the holes of adjacent plates orientd randomly relative to each other. The placement of 1, 2, or 3 intermediate plates in different models allowed us to vary the distance between the perforated plates within the range 0.4-1.6 mm. The geometry of the experimental models was detailed in [2]. Figure 1 illustrates the basic design of the packet.

The study was performed on an experimental unit which ensured a heat-balance agreement

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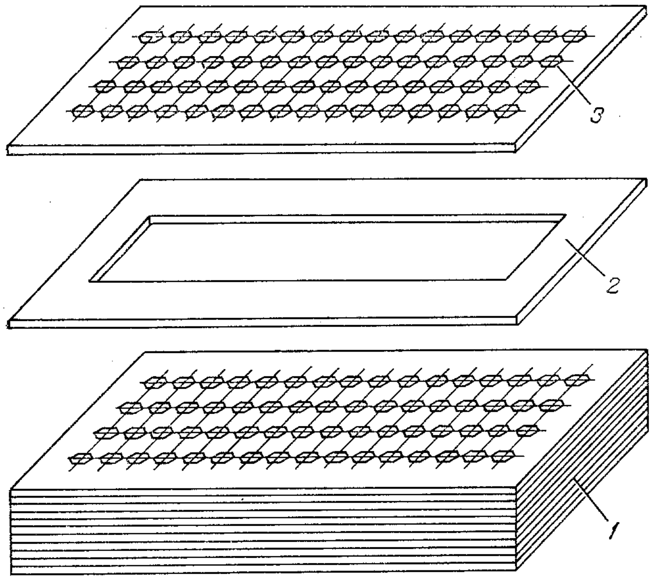

Fig. 1. Design of experimental heat exchanger made of perforated plates: 1) packet; 2) spacer plate; 3) perforated plate.

to within ±5%. The experimental section and experimental method were described in detail in [1]. The tests for hydraulic resistance were conducted under isothermal conditions, while the heat-exchange tests were conducted with condensation heating by water vapor. The working medium was air with parameters close to the ambient values.

Examination of the physical pattern of the formation of the flow in a channel comprised of a packet of perforated plates, using the theory of free submerged jets [6] and employing the Euler number referred to a single plate of the packet as the hydraulic characteristic, allowed us to generalize the test data on hydraulic resistance for different perforations [2].

As was shown in [2], the hydraulic characteristic of the surface of a perforated-plate heat exchanger is independent of the geometry of the plate — the diameter of the holes and plate spacing — for a certain range of values.

For distances between plates of 0.5 mm or less, the maximum pressure loss corresponds to the case of location of the last plate in the plane of the minimum cross section of the jet. For the perforations studied, this case corresponds to an experimental packet with holes 0.625 mm in diameter and a distance of 0.48 mm between plates. The data on hydraulic resistance could not be generalized for this packet.

An increase in the distance between the plates leads to a reduction in the pressure drop on the single plate due to recovery of pressure in the jets. Until the plate spacing reaches values at which the jets of adjacent holes begin to interact with each other, the pressure losses do not change appreciably — within 10%. The spacing at which jet interaction begins depends on the geometry of the perforated plate. For packets with holes of 0.9 and 1.65 mm diameter, the last plate is located in the recovery zone of the jet. It was suggested in [7] that the resistance of single perforated plates be calculated by using a relation for the similitude region of the Reynolds number of the flow. Employing the Euler number, we write this relation as follows:

$$\mathrm{Eu}_{\mathrm{sim}} = \frac{(1.707 - \rho_{\mathrm{pl}})^2}{2}. \qquad (1)$$

Comparison of the test data we obtained on hydraulic resistance with the data calculated from Eq. (1) showed that the mean deviation of the empirical points for the simulitude region of the Reynolds number is ±15%. Figure 2a compares the calculation and experiment.

Simultaneous with generalization of the tests in the similitude region of the Reynolds number, we analyzed the test data for regimes corresponding to $\mathrm{Re} < \mathrm{Re}_{\mathrm{sim}}$. The following formula was obtained for the Euler number:

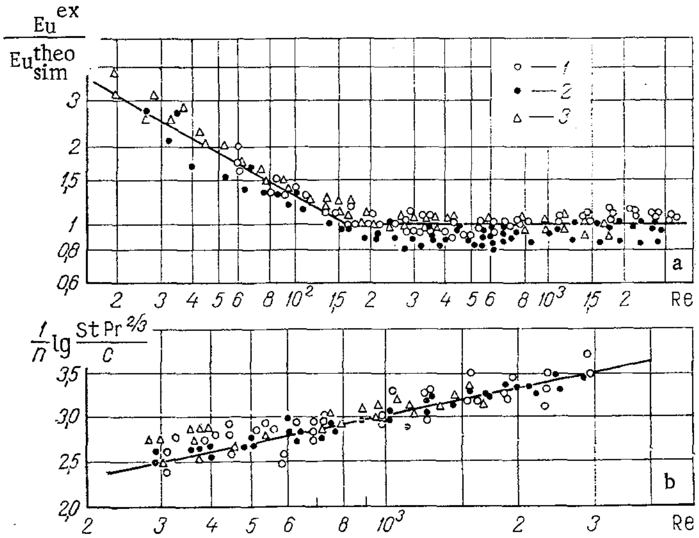

Fig. 2. Test data on hydraulic resistance (a) and heat exchange (b) in packets of perforated plates with holes of different diameter: 1) 1.65; 2) 0.9; 3) 0.625 mm.

$$\mathrm{Eu} = 16.34\mathrm{Re}^{-0.55}\,\mathrm{Eu}_{\mathrm{sim}}. \tag{2}$$

Equation (2) is valid for the region Re < 160. The mean deviation of the calculated values of the Euler number from the empirical values is ±15%. For using the test data on heat transfer under conditions different from the experimental conditions employed here, it is analyzed in the form of relations containing the criterional numbers of Stanton, Nusselt, Reynolds, Prandtl, etc., characterizing features of the process. In the case where the geometric characteristics of the heat-exchange surface exert an effect, the appropriate parameters should be introduced into the theoretical relation.

For a specific plate-perforation geometry with 0.9-mm-diameter holes, a relation was obtained for heat exchange in packets with different plate spacings without parameters which, in explicit form, would contain the geometry of the perforation [1]. Similar results were obtained for perforations with diameters of 0.625 and 1.65 mm. A change in the plate spacing within the range 0.4-1.6 mm does not affect the heat-exchange characteristic:

$$\mathrm{St} = C\mathrm{Re}^{n}\mathrm{Pr}^{-2/3}. \tag{3}$$

However, the test data for different-geometry perforations are generalized by independent curves with different coefficients C and n. Heat exchange in this case was calculated with allowance for the efficiency of the perforated plates in the direction of heat propagation, in accordance with [8].

The data obtained on heat transfer was used for a generalization which considered the effect of perforation geometry. As in the relation for hydraulic resistance, the quantity characterizing the effect of geometry was the porosity of the perforated plate. The empirical coefficients in Eq. (3) have the following expressions:

$$C = 3.6\cdot 10^{-4}\left[(1-\rho_{\mathrm{pl}})\rho_{\mathrm{pl}} - 0.2\right]^{-2.07}, \tag{4}$$

$$n = -4.36\cdot 10^{-2}\rho_{\mathrm{pl}}^{-2.34}. \tag{5}$$

Figure 2b shows data on heat exchange for the investigated perforations in generalized form. The maximum deviations of the empirical points from the theoretical relation does not exceed ±15%.

The theoretical relations obtained are suitable for designing heat exchangers made of perforated plates 0.5 mm thick with a perforated-plate porosity within 0.3-0.6, a plate spacing within 0.4-1.6 mm, and arbitrary location of the holes of adjacent plates relative to each other.

NOTATION

Eu = $\Delta p\rho/w^2$, Euler number; Δp, pressure drop on a single perforated plate, N/m²; ρ, density of heat carrier, kg/m^3; w, mass velocity of flow in the holes of the perforation, $kg/(m^2 \cdot sec)$; ρ_{pl}, porosity of the perforated plate; $Re = wd_0/\mu$, Reynolds number; d_0, diameter of hole of perforation, m; μ, absolute viscosity of heat carrier, $N \cdot sec/m^2$; $St = \alpha/(wc_p)$, Stanton number; α, heat-transfer coefficient, $W/(m^2 \cdot {}^\circ K)$; c_p, specific heat of heat carrier, $J/(kg \cdot {}^\circ K)$; $Pr = \mu c_p/\lambda$, Prandtl number; λ, thermal conductivity of heat carrier, $W/(m \cdot {}^\circ K)$; C, n, empirical coefficients.

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HEAT AND MASS TRANSFER IN A SUBMERGED AXISYMMETRIC TWISTED JET

Z. P. Shul'man, V. I. Korobko, and V. K. Shashmin

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The excess-temperature distribution in a twisted flow is obtained for numbers $Pr \neq 1$, and the distribution of the concentration of a gaseous impurity is investigated.

L. G. Loitsyanskii was the first to formulate and analytically solve the problem of the development of a laminar submerged axisymmetric twisted jet of a viscous incompressible fluid [1]. The problem was solved on the basis of the equations of a laminar boundary layer by the method of asymptotic expansions. Loitsyanskii found the first and second terms of the expansions of the component velocities in their final form, these velocities corresponding to slightly twisted jets. For jets with a moderate twist, characterized by a "dip" of the longitudinal velocity on the jet axis, the authors of [2, 3] found the succeeding terms of the velocity and pressure expansions. In [4], heat exchange in an axisymmetric nonsimilitudinous jet at Pr = 1 was examined. In [5], the distribution of excess temperatures in submerged axisymmetric jets was found for arbitrary Prandtl numbers, and results of experimental studies were presented.

The present work is a logical continuation of [5]. The excess-temperature distribution is obtained for twisted jets for a wide range of Prandtl numbers. Results are presented from experimental studies of the distribution of a gaseous impurity in axisymmetric turbulent twisted jets of air. These results are compared with the solution obtained.

A. V. Lykov Institute of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR, Minsk. Novolipetsk Polytechnic Institute.

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