HEAT TRANSFER IN PARALLEL-FLOW STAGGERED BANKS OF TUBES WITH VARIOUS RELATIVE SPACINGS

É. K. Kalinin, G. A. Dreitser, and A. K. Kozlov

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Heat-transfer data for parallel-flow staggered banks of tubes with s/d ratios of 1.16, 1.2, 1.34, 1.4, and 1.5 are examined. The experiments were performed with water at Reynolds numbers ranging from $2 \cdot 10^3$ to $7 \cdot 10^4$.

Interest in heat-transfer studies with parallel-flow banks of tubes has substantially increased during the past few years [1-11]. When processed with respect to the equivalent diameter, the results of most papers differ appreciably from the data for one tube, the heat transfer in banks of tubes increasing with increasing relative spacing s/d. Agreement with data for a tube exists only within a narrow range of s/d values (for staggered banks, at values on the order of 1.1 to 1.2). For s/d > 1.2, the heat transfer in parallel-flow staggered banks is substantially higher than in a tube. Experimental heat-transfer data [1, 3-5, 11] for staggered banks with s/d = 1.4 to 1.5 exceed by 40 to 50% the values calculated on the basis of the equivalent diameter or from the formulas for a tube.

In the literature, however, one finds numerous recommendations of calculating heat transfer for parallel-flow banks of tubes on the basis of data for a tube, where the equivalent diameter is taken as the characteristic dimension. Similar recommendations continue to appear during the past few years, as in [12, 13], for example. In [14], the obtained agreement between experimental heat-transfer data for a staggered bank with s/d = 1.2 and the values calculated from the formula for a tube is seen to be sufficient to make the unjustified conclusion of the applicability of the formula for a tube to the calculation of heat transfer in parallel-flow banks of tubes with s/d > 1.2, which in [14] were not investigated.

No reliable computational recombinations can be made without compiling further heat-transfer data for parallel-flow banks of tubes. The present paper presents experimental heat-transfer data obtained for parallel-flow staggered banks of tubes with s/d = 1.16, 1.2, 1.34, 1.4, and 1.5.

The usable lengths (Fig. 1) consisted of 7 tubes combined to a bank by means of two tube sheets. The tubes, made from 1Kh18N9T steel, had a wall thickness of 1 mm. The tube sheets were spaced 1400 mm apart. For banks with s/d = 1.16, 1.2, and 1.5, the hexagonal shell was made of Plexiglas. For banks with s/d = 1.34 and 1.4, a steel shell with an inside diameter of 49.9 mm was employed. The geometrical dimensions of the banks investigated are tabulated.

The water was led in and out of the system perpendicularly to the tubes: uniform flow in the usable length was achieved with the aid of inlet and outlet circular water boxes connected by a system of holes with the space between the tubes.

Only the central tube was heated in the experiments. Alternating low-voltage current from a TPO-252 step-down transformer was passed directly through the tube. The busbars were fitted to the ends of the central tube which were made to protrude, through packing glands, from the shell. Moving in them, the tube could freely expand during heating. The tube was stretched at each end by springs to avoid bending during heating.

In [3], [8, 15], [1], using banks with s/d = 1.4, 1.22, and 1.46, respectively, and in our preliminary tests it was shown that in the turbulent region in parallel-flow banks of tubes, heating of all tubes can be

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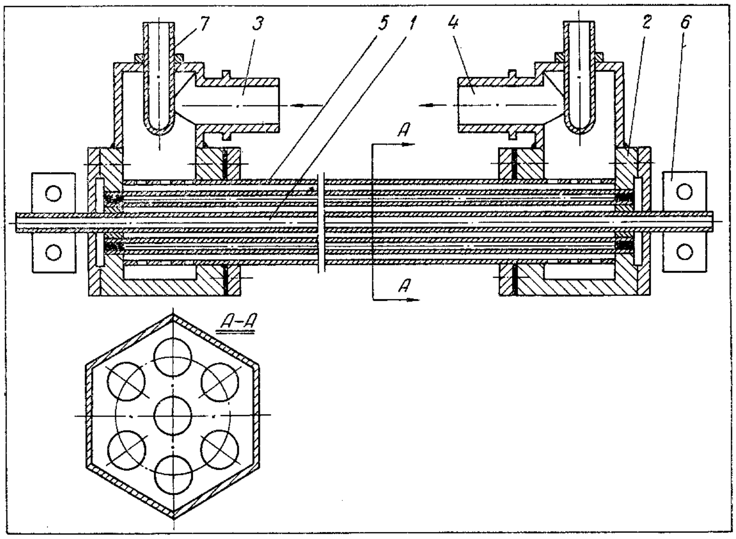

Fig. 1. The usable length: 1) heated tube; 2) tube sheet; 3) inlet; 4) outlet; 5) shell; 6) busbars; 7) thermometer casing.

TABLE 1. Geometrical Dimensions of the Banks of Tubes Studied

s/d	1,16	1,2	1,34	1,4	1,5
d, mm	11,4	12,0	11,87	11,4	12,1
$d_{e\infty}$, mm	5,51	7,05	11,22	13,21	17,9
F, mm^2	309,7	520,7	1181	1240	1152
U, mm	370,9	420,8	417	407	432,7

with a satisfactory accuracy replaced by the heating of the central tube alone, i.e., the method of local modeling can be applied. This method, where the hydrodynamic flow pattern results from all 7 tubes of the bank while heat transfer is studied only at the heated central tube, was used in our experiments. For laminar flow, the method is applicable only at small thermal loads.

The heat-transfer and hydraulic-resistance coefficients were measured in a length containing stabilized longitudinal flow. This length was predetermined experimentally. Its length was l = 1000 mm (l/d_e = 94 to 300). It began 200 mm from the inlet.

The flow rate was measured by a water-measuring tank. The temperature of the water at the inlet and outlet of the usable length was measured with Chromel–Copel thermocouples, whose hot junctions were soldered to the casings, and was checked by means of standard mercury thermometers. The temperature of the inside surface of the central-tube was measured with a mobile Chromel–Copel thermocouple with sliding contact. The hot junction of the thermocouple was pressed against the inside wall of the tube by a flat spring. The spring was fastened to an ebonite body that could move inside the tube. Owing to the absence of any noticeable temperature gradients in the tube and to the close fit between the junction and the wall, the wall temperature could be measured reliably in any cross section of the tube. Prior to the experiment, the mobile thermocouple was calibrated by standard thermometers at the inlet and outlet in isothermal flow of water of various temperatures. The reliability of the recordings of the mobile thermocouple was checked in preliminary tests also by means of thermocouples soldered directly to the tube wall. The thermocouple wires were led out of the tube through the end face. The emf of all thermocouples was measured with a semiautomatic P 2/1 potentiometer. The wall temperature was measured at 11 points.

The temperature profile along the tube perimeter was measured in preliminary tests by reversing the mobile thermocouple. No temperature nonuniformities along the tube perimeter were observed.

The electric power dissipated at the central tube was determined from measurements of the current strength and from the voltage drop at the useful length. The amount of heat released at the useful length was determined from the electric power. The discrepancy with the amount of heat determined from the changes in the heat content of the water was less than ±5%.

The maximum error involved in the determination of the heat-transfer coefficients did not exceed ±10%.

The temperature of the heat-transfer surface was determined with allowance for the temperature drop in the tube wall.

The mean heat-transfer coefficient for the central tube in the stabilized region of the flow

$$\bar{\alpha} = \frac{q}{\bar{t}_m - \bar{t}_{fc}} \quad (1)$$

was determined in the experiments. The mean temperature of the flow in the central regions of the bank, $\bar{t}_{fc}$, was determined under the assumption that no mixing occurs between the central regions of the flow. Then, the flow temperature at the end of the central regions is

$$t_{2c} = t_1 + (t_2 - t_1)\ \frac{F}{6F_1}\ . \quad (2)$$

In [3] and [8], for banks with s/d = 1.4 and 1.2, respectively, it was confirmed that the flow temperature at the end of the central regions, determined in this way, is in excellent agreement with the experiment. On the basis of experiments on the redistribution of foreign matter concentrations in water flowing through a bank model, it was shown in [17] that turbulence has only a slight equalizing effect on the temperature field across the shell and that for a bank with s/d = 1.4, the actual maximal temperature difference of the flow in the outlet cross section constitutes 87% of the difference that would take place if mixing were to be completely absent. For closer spaced banks, the equilibrization of the temperature field would be even less.

In our experiments, however, heating of the water in the turbulent region of the usable length was quite insignificant ($t_2 - t_1$ = 1.5 to 2°C): for a mean tempeature head, $\bar{t}_m - \bar{t}_{fc}$ = 12 to 30°C. Because of this,

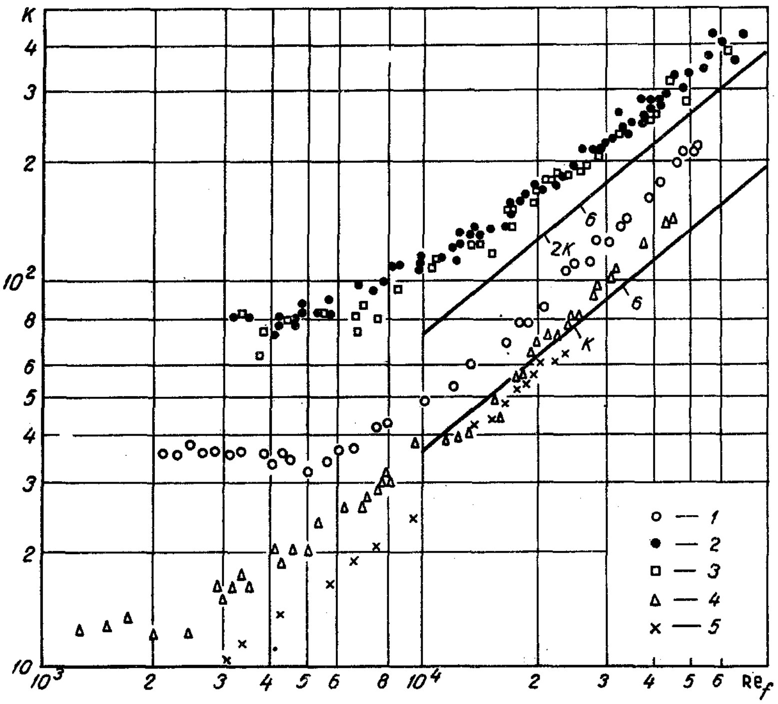

Fig. 2. Heat transfer in the banks of tubes investigated: 1) s/d = 1.5; 2) s/d = 1.4; 3) s/d = 1.34; 4) s/d = 1.2; 5) s/d = 1.16; 6) from formula $Nu_f = 0.023\ Re_f^{0.8} Pr_f^{1/3}$ for a tube [18]. 2K ($K = Nu_f/Pr_f^{1.3}$) are plotted in the figure for banks with s/d = 1.34 and 1.4.

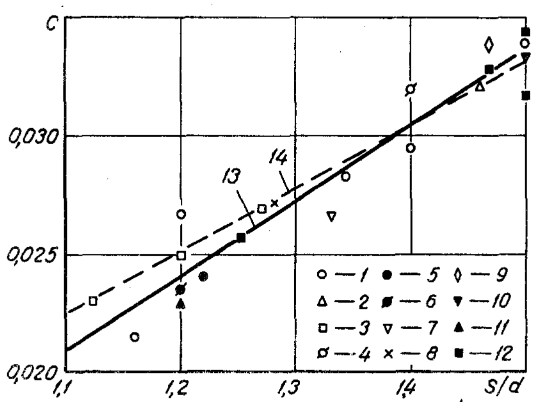

Fig. 3. Coefficient $C = Nu_f/Re_f^{0.8}Pr_f^{1/3}$ vs relative spacing s/d for staggered banks of tubes: 1) from data of the present investigation; 2) from data [1] for s/d = 1.46; 3) from [2] for s/d = 1.12, 1.2, and 1.27; 4) from [3] for s/d = 1.4; 5) from [8] for s/d = 1.22; 6) from [9] for s/d = 1.2; 7) from [6] for s/d = 1.33; 8) from [7] for s/d = 1.28; 9) from [4] for s/d = 1.47; 10) from [5] for s/d = 1.5; 11) from [10] for s/d = 1.2; 12) from [11] for s/d = 1.5 (two versions), 1.468 and 1.25; 13) from formula (5); 14) from formula (6).

the temperatures of the water in the central and peripheral regions differed only slightly, and even the assumption of ideal mixing in the bank would not affect the experimental heat-transfer coefficient by more than 4 to 7%. The actual error in the determination of $\bar{\alpha}$ owing to neglect of mixing is appreciably smaller. The mean temperature of the flow in the central regions is

$$\bar{t}_{fc} = \frac{t_1 + t_{fc}}{2} = t_1 + (t_2 - t_1)\,\frac{F}{12F_1}\,. \qquad (3)$$

In analyzing the data on heat transfer in critical form, as defining parameter we take the equivalent diameter based on the assumption that the number of tubes in the group is infinity, i.e., for central cells

$$d_{e\infty} = [1.102\,(s/d)^2 - 1]\,d. \qquad (4)$$

The mean boundary layer temperature, t_f, equal to the half-sum of the mean temperatures of the wall, $\bar{t}_m$, and the flow, $\bar{t}_{fc}$, was taken as the characteristic temperature.

In the experiments, the temperature of the water at the outlet of the bank was varied from 7.7 to 66.2°C, the mean temperature of wall was varied from 44.2 to 90.2°C, the specific heat flux from $4.7 \cdot 10^4$ to $4.65 \cdot 10^5$ W/m², Re_f from $2 \cdot 10^3$ to $7 \cdot 10^4$, and Pr_f from 3.5 to 11.2.

Experimental heat-transfer data from the banks investigated, processed with respect to t_f and $d_{e\infty}$, are shown in Fig. 2. The influence of free convection on heat transfer becomes apparent at small Reynolds numbers, and therefore the experimental data for this range can be evaluated only qualitatively. In the region of laminar–turbulent transition, heat transfer varied smoothly for each of the banks investigated. It is characteristic that transition to fully developed turbulent flow in banks is delayed as compared to the flow in a tube, the delay increasing values of s/d. Thus, for banks with s/d = 1.16, 1.2, and 1.5, transition occurs at Re_f numbers on the order of $1.3 \cdot 10^4$, $2 \cdot 10^4$, and $3 \cdot 10^4$, respectively.

In the turbulent region, heat transfer increases as s/d increases. The experimental heat-transfer data for a bank with s/d = 1.16 are on the average by 7% smaller than the values calculated from the formula for a tube in [18]. The heat transfer in all of the remaining banks was greater than in a tube. The experimental data for a bank with s/d = 1.2 are on the average by 17% greater than in a tube. For banks with s/d = 1.34 and 1.4, this difference is 23 and 30%, respectively. Data for a bank with s/d = 1.5 are on the average by 50% greater than the values calculated from the formula for a tube.

It can be seen from Fig. 2 that at small Re_f numbers, the increase in heat transfer with s/d is more pronounced than in the turbulent region. This may, in part, be attributed to the fact that natural convection can develop more freely at large values of s/d.

Average relationships for the turbulent region ($Re_f > 1.3 \cdot 10^4$ for s/d = 1.16, $Re_f > 2 \cdot 10^4$ for s/d = 1.2, 1.34, 1.4, and $Re_f > 3 \cdot 10^4$ for s/d = 1.5) for the banks investigated are given in Fig. 3 in the form of plots of the coefficient $C = Nu_f/Re_f^{0.8}Pr_f^{1/3}$ vs s/d. In addition the figure shows the results of papers [1-11] for staggered banks of tubes, which are in satisfactory agreement with the experimental data under consideration.

All the experimental heat-transfer data in Fig. 3 for parallel-flow staggered banks of tubes can be generalized, with a maximum error on the order of ±10%, by the empirical relation

$$Nu_f = (0.032\,s/d - 0.0144)\,Re_f^{0.8}\,Pr_f^{1/3}\,, \qquad (5)$$

which holds for staggered banks with $1.1 \leq s/d \leq 1.5$ and for Re_f numbers corresponding to the turbulent region.

The generalized formula for staggered banks

$$\mathrm{Nu}_f = (0.026\, s/d - 0.006)\, \mathrm{Re}_f^{0.8}\, \mathrm{Pr}_f^{1/3}, \quad (6)$$

obtained in [19] by generalizing the experimental data [1, 2], conforms roughly with (5) for s/d ratios ranging from 1.3 to 1.5, but yields overestimated values for s/d ratios between 1.1 and 1.2.

NOTATION

q	is the heat flow;
$\bar{t}_m$	is the mean outside-surface temperature of the central tube;
t_1 and t_2	are the flow temperature at the inlet and outlet of a bank;
F	is the cross-sectional area of the bank;
$F_1 = \pi d^2/8[1.102(s/d)^2 - 1]$	is the area of a central region;
d	is the outside diameter of tubes;
s	is the spacing between tubes in a bank;
U	is the perimeter surrounded by the flow;
$d_e = 4F/U$	is the equivalent diameter determined from the total U.

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