HEAT TRANSFER IN TUBES WITH DISCRETE ANNULAR ROUGHNESS ELEMENTS

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The results are presented of an experimental study concerning the heat transfer and the hydraulics in tubes with roughness elements comprising a system of diaphragms. The conditions of optimum heat transfer are determined for this case and generalized formulas are derived.

The problem of heat transfer in rough tubes is of practical importance from the point of view of improving the heat transfer in all sorts of power apparatus.

In studies already published on this subject [1-5] the conditions for the extremum (maximum) heat transfer have not been determined. In [1, 3] it has been established experimentally that the optimum relative pitch between roughness elements (protuberances) is $t/h \simeq 10$, with t (m) denoting the actual pitch and h (m) denoting the height of the roughness elements, which ensures the maximum rate of heat transfer at a given roughness height. The effect of the relative pitch in the case of discrete annular roughness elements, in terms of a maximum rate of heat transfer, has not been studied to a great extent.

An analysis has shown that roughness in other forms (sandiness or, for example, dense triangular or other shape asperities) results in a lower rate of heat transfer than roughness in the form of discrete protuberances.

In this study we have experimentally analyzed the effect of the relative pitch of discrete protuberances (diaphragms) on the heat transfer in a circular tube, with the further aim of obtaining data with large relative pitches as well as of generalizing all available data.*

The geometrical dimensions of the tested tubes are given in Table 1.

The tests were performed under atmospheric conditions, with a circular tube 20 mm in diameter ($l/d_0 = 40$) as the active element and diaphragms installed inside ($d/d_0 = 0.2$-0.7, d (m) denoting the inside diameter of a diaphragm and d_0 (m) denoting the inside diameter of a tube). In order to eliminate the undetermined thermal contact resistance between a protuberance base and a tube wall, the tubes were assembled sectionally and each diaphragm was soldered to a tube wall, which ensured a reliable thermal contact. The tubes and the diaphragms were made of brass $\delta = 1$ mm thick, which yielded a thermal efficiency of finning $E \sim 1$ under our test conditions. This made it possible, unlike in the Nunner and Koch experiments [2, 3], to eliminate the error due to the indeterminacy of the thermal contact at a protuberance base. Unlike in [2-4], we performed tests with larger values of the h/R_0 ratio (R_0 denoting the tube radius).

The test apparatus and the test procedure have been described in detail in [1].

The test segment was heated up with boiling water and the small thermal resistance between water and tube wall was then accounted for. The physical constants used for the data processing were based on the mean temperature of the stream. As the reference velocity, we used the velocity in a tube without diaphragms (protuberances). The tests were performed at Reynolds numbers ranging from 2000 to 30,000 and $\psi = t_{wall}/t_{stream} = 1.1$.

*The tests were performed with the assistance of V. Maklyak.

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TABLE 1. Geometrical Dimensions of Tested Tubes

Table No.	Crimp height h, mm	Distance between crimps t, mm	h/R_0	t/h	$\frac{F_{tube}}{F_{tot}}$	l/d_0
1	4	147	0,4	37	0,96	40
2	4	74	0,4	18,5	0,924	40
3	4	48	0,4	12,2	0,885	40
4	4	16	0,4	4	0,8	40
5	5	150	0,5	30	0,95	40
6	5	74	0,5	14,8	0,91	40
7	5	49	0,5	9,8	0,87	40
8	6	60	0,6	10	0,88	40
9	6	36	0,6	6	0,81	40
10	3	30	0,3	10	0,855	40

The test data on the heat transfer are shown in Fig. 1. At $Re \gtrsim 4000$ the test points fit the equation $Nu = A_1Re^{0.8}$ corresponding to a turbulent flow, while at $Re \lesssim 4000$ the equation is $Nu = A_2Re$ for a transitional flow. The test points for a smooth tube fit the Mikheev equation $Nu_0 = 0.018Re^{0.8}$ [6]. The roughness geometry was defined in terms of two parameter groups: the relative roughness h/R_0 and the relative protuberance pitch t/h.

In Fig. 2 we show the effect of the relative protuberance pitch on the heat transfer in a tube.

According to the graphs, the maximum rate of heat transfer corresponds to $t/h \sim 10$-12, and this applies to all tested values of the parameter h/R_0. Separation of the stream from the diaphragm edges and its subsequent adherence to a smooth surface ensure a fast rise of the heat-transfer rate (up to 3.8 times). According to [7], the maximum rate of heat transfer occurs at the merger with the boundary layer, while the structure of the boundary layer which builds up beyond the merger point (critical point) is not the same as in an ordinary turbulent downstream layer at a plate.

For a turbulent boundary layer at a plate, according to [8], we have $Nu_x = 0.032Re_x^{0.8}$ or, in terms of the diameter, $\overline{Nu}_{d_0} = 0.032Re_{d_0}^{0.8}(l/d_0)^{-0.2}$.

We assume that from the critical point on, the boundary layers develop downstream as well as upstream and that in the latter case the outer stream is, to the first approximation (conservatively), a backstream in the eddy region when $t/h = 10$ and $h/R_0 = 0.4$, which allows us to write $\overline{Nu} = 0.032Re^{0.8}$. We assume here also that the velocity at the outer edge of the backflowing boundary layer is equal to the mean velocity of the stream. Actually, it is much lower and the final values in our analysis will be on the high side. Thus, $\overline{Nu}/Nu_0 = 1.78$, while we obtained from the test $\overline{Nu}/Nu_0 = 3.8$.

The fast rise of the heat-transfer rate is explained by the beneficial effect of turbulence on the characteristics of the boundary layer, as long as the stream after separation is very turbulent. Studies of the local heat transfer behind a single diaphragm have shown [7] that the heat-transfer coefficient increases up to the merger point at a distance $l = (6$-$8)h$; it is maximum at the merger point and then again decreases along the stream. If we approximate the data in this study for $h/R_0 = 0.5$ by the equation $Nu_x/Nu_0 = 3.12(x/h)^{0.187}e^{-0.003(x/h)^2} + 1$ and define the mean value of the Nusselt number as

$$\frac{\overline{Nu}}{Nu_0} = \frac{h}{t}\int\limits_0^{\frac{t}{h}}\left[3.12\left(\frac{x}{h}\right)^{0.187} e^{-0.003\left(\frac{x}{h}\right)^2} + 1\right] d\left(\frac{x}{h}\right),$$

then we obtain $\overline{Nu}/Nu_0 = 3.64$, 4.24, 4.24, 3.98, and 3.71, respectively, for $t/h = 4$, 8, 12, 16, and 20.

In this way, the optimum value for t/h is 10-12, which corresponds to the data in Fig. 2. It must be noted that the absolute values of Nu for a single diaphragm are not the same as the values of Nu for a system of diaphragms, owing to the effect of initial conditions and to the different states of the thermal boundary layer as well as to the effect of the diaphragms on one another.

In Fig. 2 is also shown the effect of the relative roughness, based on the data of this study as well as on the data by other authors. Along the solid line the heat transfer is referred to the surface area of a smooth tube, along the dashed line it is referred to the total rough surface. As long as the fin efficiency remains $E = 1$, the dashed line represents actual values of convective heat-transfer coefficients. At $h/R_0 = 0.4$ we observe the maximum rate of heat transfer.

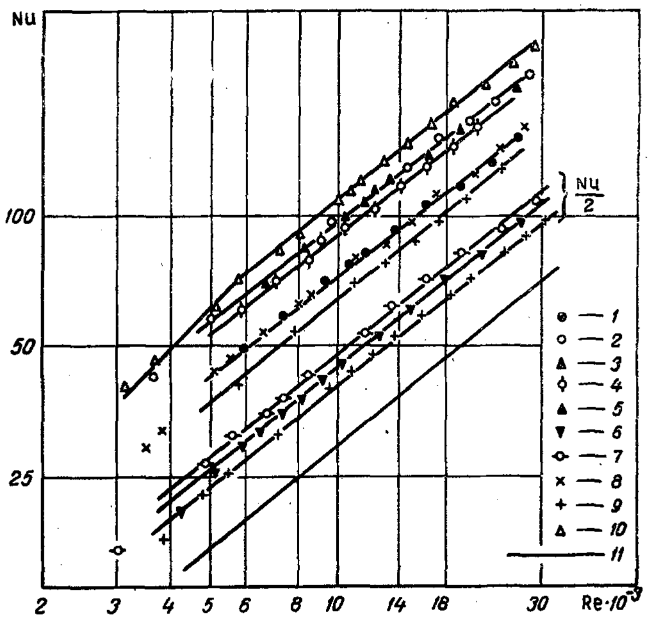

Fig. 1. Heat transfer in tubes with diaphragms: 1–10) according to Table 1; 11) according to $Nu_0 = 0.018Re^{0.8}$.

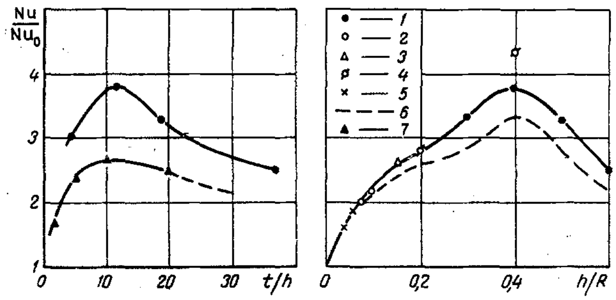

Fig. 2. Effect of parameters h/R_0 and t/h on the heat transfer in tubes with diaphragms: 1) according to the data of this study; 2) according to the data in [1]; 3) in [2]; 4) in [3]; 5) in [4]; 6) referred to total roughness; 7) according to the data in [2].

All this can be explained by the fact that, as the diaphragm height is increased, the intensity of eddies increases on the one hand and the variance from a smooth surface increases on the other hand, as a result of which the turbulence is damped and its effect on the merged boundary layer is weakened. In conclusion, at h/R_0 we have an optimum. Thus, $Nu/Nu_0 \sim 3.8$ corresponds to the maximum rate of heat transfer attainable in a tube by introducing an artificial roughness, inasmuch as a discrete annular roughness ensures a better heat transfer than any other form of roughness. This value $Nu/Nu_0 = 3.8$ is very close to the theoretical limit of heat transfer $Nu\ Nu_0|_{lim} = 4.5$ [5] in tubes with artificial roughness, being only 16% less than that. When referred to the total rough surface, this difference becomes 25%. The value of Nu/Nu_0 obtained by us for $h/R_0 = 0.4$ is 14% lower than according to Koch [3]. The explanation for this is that in the Koch experiment the diaphragms were held in place by means of rods parallel to be near the tube wall, which ensured additional turbulization. The data obtained in our study agree closely with those obtained by other authors. The data in [4] have been obtained for water, but a reevaluation based on the formulas in [5] shows that the effect of the Prandtl number within Pr = (0.7–3.0) on the magnitude of Nu/Nu_0 at the given values of the parameter h/R_0 is negligibly small.

The test data in Fig. 2, which yield the maximum rate of heat transfer (at $t/h \sim 10$) as a function of the relative roughness, may be approximated by the equation

$$Nu/Nu_0 = 1 + 11.41\varepsilon - 13.8\varepsilon^2 + 24.1\varepsilon^3 - 43.1\varepsilon^4,$$

where $\varepsilon = h/R_0$ and $0.6 > \varepsilon > 0.1$.

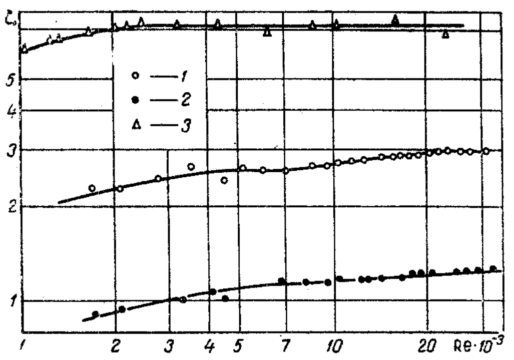

Fig. 3

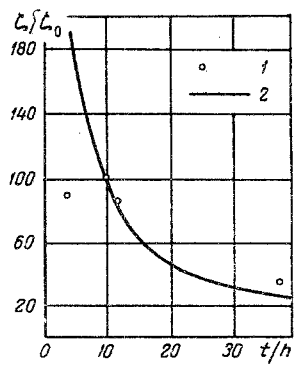

Fig. 4

Fig. 3. Hydraulic resistance in tubes with diaphragms: 1) tube No. 3; 2) tube No. 1; 3) $\zeta/10$; $h/R_0 = 0.7$ and $t/h = 7$.

Fig. 4. Effect of the parameter t/h on the hydraulic resistance in tubes with diaphragms (Re = 10,000): 1) test with $h/R_0 = 0.4$; 2) calculation according to the formula.

The effect of the relative pitch can be represented by the equation:

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\left(\frac{t}{h}\sim 10\right)}} = 1 - \frac{1}{e^{\frac{0{,}008}{\varepsilon}}}\left[0.00398\left(\frac{t}{h} - 12\right)^2 \exp\left(-0.0603\,\frac{t}{h}\right)\right].$$

Finally, for the heat transfer in tubes with a discrete annular roughness we have the formula

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_0} = (1 + 11.41\varepsilon - 13.8\varepsilon^2 + 24.1\varepsilon^3 - 43.1\varepsilon^4)\left\{1 - \frac{1}{e^{\frac{0{,}008}{\varepsilon}}}\left[0.00398\left(\frac{t}{h} - 12\right)^2 \exp\left(-0.0603\frac{t}{h}\right)\right]\right\}, \tag{1}$$

which is valid for $0.6 > \varepsilon > 0.1$ and $40 > t/h > 3$.

The test results pertaining to the hydraulic resistance (Fig. 3) indicate that ζ increases slightly as the Reynolds number becomes higher, especially in the low range, and this is explained by a restructuring of the eddy system.

In Fig. 4 we compare the calculated and the tested values of hydraulic resistance at $\varepsilon = 0.4$. Treating a diaphragm as a system of sudden contraction (with the entrance loss coefficient $\zeta_{\mathrm{ent}} = 0.5$) and sudden expansion, we have

$$\zeta = \frac{\Delta p}{\rho\,\frac{w^2}{2}\cdot\frac{l}{d_0}} = \zeta_0 + \frac{n}{\frac{l}{d_0}}\left(\frac{d_0}{d}\right)^4\left[0.5\left(1 - \frac{d^2}{d_0^2}\right) + \left(1 - \frac{d^2}{d_0^2}\right)^2\right], \tag{2}$$

where n is the number of diaphragms.

The comparison shows that at $t/h > 9$ there is a close agreement with tests, while at $t/h < 9$ the diaphragms begin to interfere with one another and the test points lie below the theoretical curve. This drop in the resistance is in this case explained by the fact that the system of eddies between the diaphragms becomes more symmetrical, as has been confirmed by visual observations, and completely fills the space between them thus reducing the pressure head [9]. We note that the maximum hydraulic resistance occurs at $t/h \sim 10$, when also the maximum rate of heat transfer occurs.

As has been shown in [1], from the energy standpoint (in terms of the ratio of heat-transfer rate to hydraulic resistance) most beneficial are small values of relative roughness ($\varepsilon = 0.07$–0.10). In all cases where a maximum rate of heat transfer is required and high losses are allowed, however, the data presented here will be useful. Furthermore, these data indicate the limits to which convective heat transfer in tubes can be improved.

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