ARTIFICIAL FLOW TURBULENCE IN A TUBE BUNDLE

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The results are presented of an experimental investigation of the heat transfer and hydraulics of flow in a staggered tube bundle (s_1 = 1.48, s_2 = 1.6) with turbulence generators in the form of wire mesh.

One method of intensifying convective heat transfer is to render the boundary layer turbulent by creating artificial roughness. Investigations carried out with individual rough cylinders in a transverse stream of air [1], as well as with tube bundles, where the turbulence generators used were wires bent into sawtooth form [2], have indicated a substantial enhancement of heat transfer. The ordinary method of making the boundary layer turbulent is an arrangement of wires or other roughness elements washed by cross flow, which disturb, and in a number of cases cause separation of the boundary layer. A separated boundary layer has an increased level of turbulence, which increases the heat transfer when the flow "reattaches" to the surface behind the obstacle. A new method of artificially induced boundary layer turbulence is described in [3].

The proposal is to use a system of longitudinal wires parallel to the direction of the oncoming stream. The turbulence is caused by diffusion in the transverse direction of perturbations created by the wires. The pitch of the wires is established by calculation, so that they span the perturbation region and produce the required turbulence. The results of tests conducted on aerodynamic models have shown that this is an effective method of causing turbulence, and produces practically no changes in the external flow; it causes no appreciable increase in the hydraulic losses, such as occurs when obstacles washed by cross flow are used. It is therefore natural to use this method to intensify heat transfer. It is apparent that tubes washed by cross flow, having a spiral coil of wire or fine spiral knurling, satisfy the conditions mentioned. A cover in the form of ordinary metal wire mesh combines both methods of turbulence generation, since part of the wire is washed by parallel flow, and part by cross flow. Ordinary steel mesh is, moreover, a relatively inexpensive material and one that is widely used in industry.

Tests were conducted in a nine-row staggered bundle of three different steel meshes with different wire dimensions [mesh No. 1—wire diameter d = = 0.3 mm and cell size s = 1.4 · 1.4 mm; No. 2—0.8 and 6 · 6; No. 3—1.0 and 10 · 10. The bundle was made up of tubes with D = 25 mm and pitches s_1 = 37/25 = = 1.48 and s_2 = 40/25 = 1.6 (Fig. 1)].

Air was drawn by a fan through an aerodynamically profiled collector, downstream of which was mounted a Prandtl tube for measurement of velocity. The uniform velocity field at this location (thin boundary layer) allowed determination of the mean velocity from the one measurement. The density of the whole system was checked by measurements of mass flow rate in the supply line. The flow rate was controlled by a gate valve. The flow proceeded through the stabilizing channel to the test bundle, upstream of which measurements showed the velocity field to be uniform under all conditions. The bundle had a front aspect of five or four tubes with two half-tubes fastened to the side walls. The tubes were braced by special blind nuts. Measurement of static pressure ahead of and behind the bundle was made via taps (six in each section). Measurement of static pressure in the flow by means of probes gave readings identical with those at the taps. The heat transfer was investigated by the local modeling method. The central tube of the sixth row (the measuring tube) was made of thin-walled copper (Fig. 1b), in contrast to the remaining tubes, which were of steel. Grooves were milled to a depth of 1 mm in the surface of the tube, along the generators, and in the grooves thermocouples were laid in twin porcelain tubes of d = 0.8 mm. The grooves were then filled with solder. The thermocouples were brought in from the ends of the measuring tube. Inside the tube there was a three-section heater, mounted on an insulator tube along which ran spiral grooves for the heater wires.

The heater was centered by end plugs made of insulating material. The annular space between the heater and the tube was filled with sifted quartz sand. The central section of the heater was the measuring section, the side heaters being used to compensate for end losses. The power of these heaters was set so as to equalize the readings of the two central thermocouples and of the corresponding thermocouples mounted on the same generators at the places where the end heaters finished. It is known that for the ordinary bundle with pitches of the order of s_1 = 1.5, s_2 = = 1.5, the local modeling method does not introduce appreciable error. When meshes are mounted in the tubes, mixing of the flow is intensified, and therefore the conditions for use of the local modeling method will be improved. Each tube of the bundle was enveloped by one mesh or another in such a way that the joint (the twisted wires) was in the part of the tube downstream with regard to the flow. In installing the meshes, a careful check was made to see that the longitudinal wires were parallel to the generators of the cylinder, as was necessary for balancing the sections of the heater, this being done from the read-

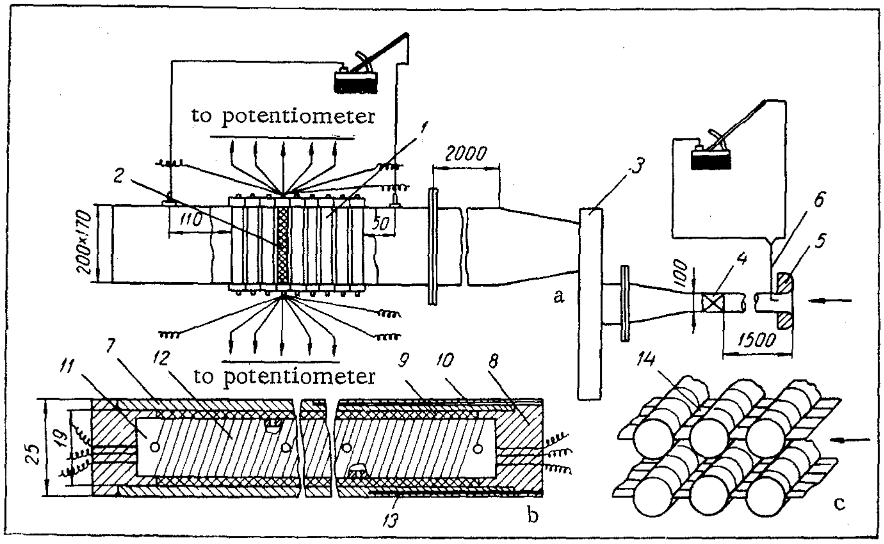

Fig. 1. Experimental set-up. a) Measurement system: 1) tubes of the bundle; 2) measuring tube; 3) fan; 4) gate valve; 5) collector; 6) Prandtl tube. b) Measuring tube: 7) copper cylinder; 8) end plugs; 9) quartz sand; 10) central thermocouple; 11) heater rod; 12) heater; 13) end thermocouple. c) Finned lattice construction: 14) fins.

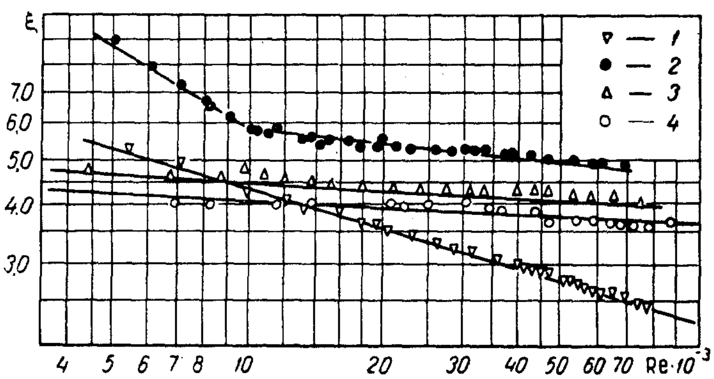

Fig. 2. Hydraulic resistance of the bundle with turbulence generators [$\xi = (\Delta p/\rho)(w_1^2/2)$], $Re = w_1 D/\gamma$: 1) smooth; 2, 3, and 4) with meshes Nos. 1, 2, and 3, respectively.

ings of the thermocouples located in pairs on a single generator at the front part of the tube. The central part of the tube had eight thermocouples around its circumference, their readings being averaged. For accurate measurement of wall temperature in the case when the meshes were installed, their wires then causing local variations of wall temperature, the meshes were oriented in two and three different arrangements in the tube, and thus the mean surface temperature was determined from 16 and 24 points, respectively. The thick-walled copper tube produced considerable spreading of the heat, which in turn brought the surface temperature closer to the mean value.

The results of the tests are shown in Figs. 2 and 3.

The heat transfer data in the case when the meshes were used are shown by averaged curves without the experimental points. Flow blockage was taken into account in installing the meshes. The hydraulic resistance of the smooth bundle was 7.5% larger than according to [4]. The Nu number agreed to within 5% with Grimmison's data [5] for a bundle of the same orientation. The maximum hydraulic resistance proved to be for mesh No. 1, and the minimum for No. 3. Mesh No. 1 was characterized, moreover, by the lowest heat transfer. As Nunner [6] has shown, in intensifying convective heat transfer in a tube by annular turbulence generator inserts, the distance between the rings is important. The flow, separated by the obstacle, must "reattach" to the smooth surface in such a way that the increased turbulence and the conditions of the initial section of the boundary layer are effectively utilized. The parameter l/h, together with the parameter h/D, is decisive in these phenomena. For meshes Nos. 1, 2, and 3 the parameter l/d had values 4.67, 7.5, and 10, respectively. The high resistance and the low heat transfer for mesh No. 1 are evidently explained by the low value of the parameter l/h. In fact, the value of l/h will be even lower in this case, since in the places where the wires intersect the mesh has an obstacle size h = 2d, but the small value of l for this mesh will result in this dimension being the dominant one. The other meshes have larger cell size, and the places where the wires cross will be affected to a lesser extent.

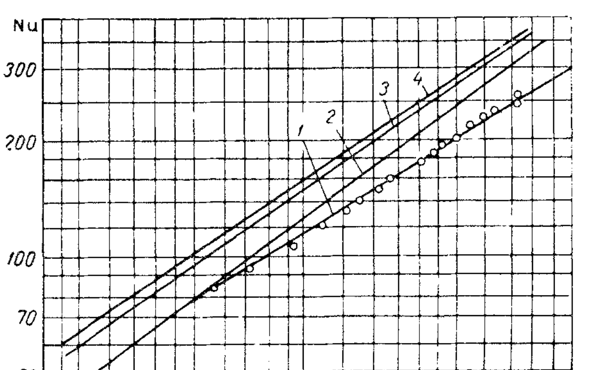

Fig. 3. Heat transfer for the bundle with turbulence generators: 1) smooth; 2, 3, and 4) with meshes Nos. 1, 2, and 3, respectively.

According to the data of [7], the laminar flow on the forward part of the cylinder will undergo transition to turbulent flow in the immediate vicinity of the wire, if $wh/\nu \geq 900$. In the cases examined, $wh/\nu \gg 900$ for all the meshes. It may be supposed that not only do the meshes make the boundary layer turbulent, but they also cause local flow separation from the individual wires.

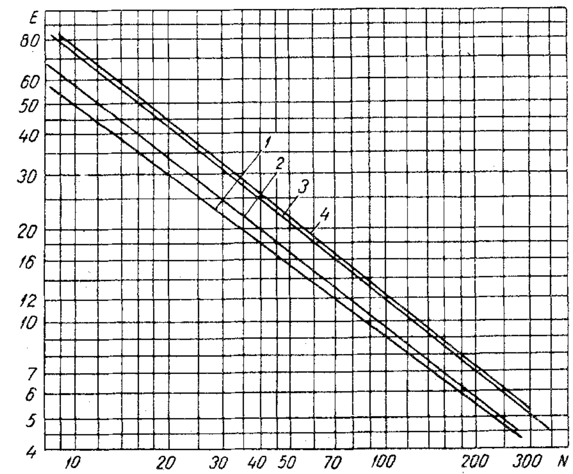

Fig. 4. Removal of heat from the bundles; 1) with mesh No. 1; 2) smooth; 3 and 4) with meshes Nos. 2 and 3.

In the case of mesh No. 1 the separated boundary layer evidently does not reattach to the smooth surface (the wires "interfere" with one another), and, as follows from Fig. 2, complete separation of the flow sets in, at low Re numbers, earlier than for the smooth surface (the wires do not behave as ordinary turbulence generators in the way that Prandtl rings do). These considerations may explain the fact that the mesh in question, with the least wire diameter, offers the largest hydraulic resistance.

Tests with turbulence generators on cylinders [7] have shown that the resistance increases with increase of parameter h/D at large Re numbers. Figure 2 shows the reverse tendency. The lack of agreement is evidently due to the fact that in the given case the influence of the parameter l/h is the decisive factor, exceeding the influence of parameter h/D.

In the tests described in [7], individual wires were used. The hydraulic resistance of the bundle falls as the parameter l/h increases. At small Re, as is borne out by Fig. 2, the bundle with mesh of larger l/h values has less resistance than the smooth bundle, which is due to its action, like that of a "Prandtl ring" in making a laminar boundary layer turbulent, resulting in separation being displaced downstream. Figure 3 shows evidence of the fact that the heat transfer increases with increase of the parameter l/h, due to the hydrodynamic causes mentioned.

To a certain extent, in all these phenomena, the influence is also apparent of turbulence generation by the wires of the mesh that are washed by parallel flow, in agreement with [3], but it is impossible to isolate this influence in the limited tests performed here.

The area of contact of the meshes examined with the smooth surface is very small, and the effect of the fins does not appear here to a significant degree. The curves of Figs. 2 and 3 satisfy equations of the type

$$\mathrm{Nu} = A_1 \mathrm{Re}^{n_1}, \ \xi = A_2 \mathrm{Re}^{n_2}.$$

Here ξ, in contrast to Fig. 2, is referred to a single row. The physical constants are referred to the flow temperature.

A_1, A_2, n_1, n_2 for the different variants have the following values:

	A_1	A_2	n_1	n_2
Smooth bundle	0.3	5.74	0.6	− 0.27
Mesh, № 1	0.122	1.54	0.7	− 0.0955
№ 2	0.255	0.0775	0.645	− 0.0425
№ 3	0.266	0.0693	0.644	− 0.04554

For the bundle with mesh No. 3, the heat transfer intensity is 40% larger than for the smooth bundle.

In order to compare the various meshes, the data of Figs. 2 and 3 are presented in Fig. 4 in coordinates $E = \alpha/N, N$.

As may be seen from the figures, the bundle with mesh No. 1 is inferior to the smooth bundle as regards heat removal at equal value of the power required to overcome resistance, while the bundle with mesh No. 3 surpasses it by 30%. In spite of the increase in α and ξ not being proportional (ξ increased more), the gain in heat removal at equal N arises from the fact that $N \sim \xi w^3$, while $\alpha \sim w^{0.6}$, and thus the quantity α has a stronger influence than ξ.

Thus, the use of turbulence-generating meshes may be an appreciable economic factor. They may be used, of course, only in conditions of uncontaminated streams. Mounting of the meshes in the bundles in the way that was used in the given tests (the mesh was wrapped around the tube, and the joint was formed in the downstream region by twisting the individual wires) is complicated and laborious. A unique finned construction may be suggested, where the meshes play the part of the fins (Fig. 1, c). In this case a row of tubes is covered above and below by meshes of large size, which are sewn together with special equipment, or welded, in the spaces between tubes. These operations should be carried out while the mesh is drawn tight on the tube.

Such a construction is shown in Fig. 1 for a corridor-type bundle; similar arrangements may be made with a staggered bundle.

NOTATION

w_1—velocity in narrow section of bundle, m/sec; q—amount of heat divided by surface area of smooth tube, W/m^2; l—distance between obstacles, m; h—height of obstacle, m; d—wire thickness, m; D—diameter of tubes of bundle, m; N—power required to overcome hydraulic resistance, referred to unit surface, W/m^2; E—Kirpichev number.

REFERENCES

1. E. A. Brun, G. Brunello, and M. Vernotte, Publs. scient. et techn. Ministere air N. T., no. 60, 1956.

2. F. Fournell, Chaleur industrie Mars, no. 368, 1956.

3. R. Miche and M. Sirieix, C. R. Acad. Sci., 243, no. 3, 1956.

4. Aerodynamic Calculation of Boiler Equipment [in Russian], GEI, 1961.

5. E. R. Eckert and R. M. Drake, Theory of Heat and Mass Transfer [Russian translation], GEI, 1961.

6. D. Nunner, "Wärmeübergang und Druckabfall in rauhen Rohren," VDI-Forschungsheft, 455, 1956.

7. H. Schlichting, The Origin of Turbulence [Russian translation], IL, 1962.

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