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SPECIAL FEATURES OF FLOW AND HEAT TRANSFER IN STAGGERED BUNDLES OF TRANSVERSELY FINNED TUBES

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The special features of flow and heat transfer in staggered bundles of tubes with external circular fins are investigated in a wide range of their geometrical characteristics.

The flow in bundles of transversely bathed tubes with external circular fins is distinguished by considerable complexity, because of which it is very difficult to investigate its characteristics. In this situation, the acquisition of even qualitative results is justified from the standpoint of a deeper physical understanding of the process and the possibility of explaining features of heat transfer in the system under consideration.

The present paper continues the investigation of [1], carried out using a method of visualization of the air stream with a kerosene-soot suspension. In the case under consideration, we posed the problem of obtaining flow patterns in the bundle as a whole, for which the image was recorded on the surfaces, polished and coated with white nitroenamel, of the lower tube plates of bundles with different cross-sectional sizes and longitudinal spacings. The experimental procedure is similar to that described in [1]. In Figs. 1 and 2 we show the results of experiments with tubes having the following geometrical characteristics: $d = 21$ mm, $h = 30$ mm, $t = 4.0$ mm, $\delta = 1.2$ mm, $\psi = 38.3$.

Because the flow patterns were recorded near the wall of the working section rather than in the stream core, they were obviously affected by processes related to the development of the intrinsic boundary layer of the wind tunnel. This effect can be assumed to be slight, however, particularly because the boundary layer at the wall was broken up by the tube array of the bundle, and the flow conditions in the gaps between the end fins and the tube plates were similar to the conditions in the gaps between fins, since these gaps were made equal in constructing the tubes and the working section of the stand. This is confirmed by the fact that the images on sections of the tube plates corresponding to fin boundaries, which are denoted on the photographs by increased stream velocities (with respect to the velocities in the gaps between tubes) in the channels between fins, agree fairly completely with the images obtained directly on the fins [1].

A study of the photographs reveals certain qualitative regularities in the bathing of tubes in bundles. The main source of disturbances in the stream bathing the tubes in a bundle is the cylinder carrying the fins. The large-scale, three-dimensional vortex formations originating from the separation from it of swirled shear layers [1], immediately after emerging beyond the channels between fins, break up into a small-vortex structure, forming a clearly defined, dark turbulent wake behind a finned tube. The dimensions of this wake and the nature of its interaction with tubes lying downstream, for constant fin parameters, depend on the spacings S_1 and S_2 of the bundle and are taken into account fairly well by the parameter S_1/S_2.

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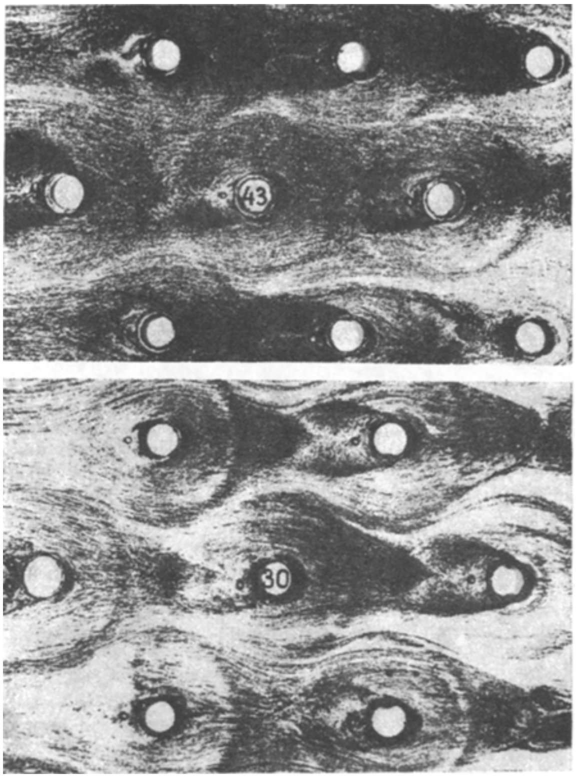

Fig. 1. Visualization of flow in bundles with equal transverse spacing S_1 and different longtitudinal spacing S_2 between finned tubes for Re = $5.4 \cdot 10^4$: fragment 43: S_1 = 138 mm, S_2 = 46 mm, S_1/S_2 = 3.00; fragment 30: S_1 = 138 mm, S_2 = 57 mm, S_1/S_2 = 2.42.

For large values of the parameter S_1/S_2 (Fig. 1, fragment 43), the width of the turbulent wake is greatest because of the weak compressing action of the pipes in the adjacent transverse row, due to the large value of the spacing S_1. Because of the small spacing S_2, the wake, characterized, according to the data of [2, 3], by a high turbulence level at a small distance from the cylinder producing it (Fig. 3), covers a considerable part of the surface of the pipe located downstream in the same longitudinal row of the bundle. The surfaces of pipes in interior rows of bundles having large values of the parameter S_1/S_2 are thus almost completely bathed by a strongly disturbed stream: in the rear part this is its own wake behind the supporting cylinder (the rear vortex zone), and in the front part is the near wake from the upstream tube.

If the parameter S_1/S_2 is reduced by decreasing the spacing S_1, the vortex wake is compressed and its dimensions decrease, as a consequence of which the fraction of the surface of a finned tube bathed by the highly turbulent stream decreases, in both the front and rear parts. This can be seen by comparing Fig. 1 (fragment 30) and Fig. 2 (fragments 32 and 33) for bundles with similar spacings S_2 and considerably different spacings S_1. For $S_1 = S_{1min}$ (Fig. 2, fragment 33), the fraction of such surface becomes smallest, since the size of the rear vortex zone is reduced to a minimum, and it covers practically none of the surface of the downstream tube.

Reducing the parameter S_1/S_2 by increasing the spacing S_2 also leads to a decrease in the fraction of surface bathed by the highly turbulent stream (see Fig. 1), since in this case it bathes only the rear part of each tube in an interior row, while the remaining surface is covered by a section of wake in which the turbulence level is the lower, the greater the distance from the source of disturbance (Fig. 3), i.e., the larger the double spacing S_2.

Thus, reducing the parameter S_1/S_2 both by decreasing the spacing S_1 and by increasing the spacing S_2 leads to a decrease in the level of disturbance of the stream bathing the tubes in interior rows.

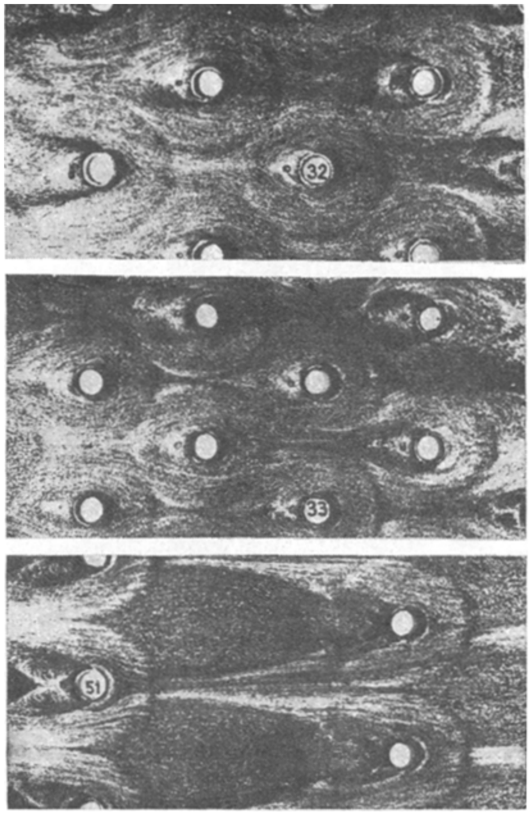

Fig. 2. Visualization of flow in bundles with variation of the transverse spacing S_1 and longitudinal spacing S_2 between finned tubes for $Re = 5.4 \cdot 10^4$:
fragment 32: $S_1 = 103.5$ mm, $S_2 = 65$ mm, $S_1/S_2 = 1.59$;
fragment 33: $S_1 = 83.5$ mm, $S_2 = 72$ mm, $S_1/S_2 = 1.16$;
fragment 51: $S_1 = 83.5$ mm, $S_2 = 200$ mm, $S_1/S_2 = 0.42$.

It must be noted, however, that the level of stream disturbance should decrease only down to a certain fairly small value of S_1/S_2, below which it should remain constant in a certain range of S_1/S_2. This conclusion follows from the fact that it is possible to obtain small values of S_1/S_2 in the actual ranges of variation of the spacings S_1 and S_2 only by increasing the longtitudinal spacing S_2, and for large S_2 the compressing effect of the tubes in adjacent rows gradually disappears, the size of the rear vortex zone increases, the width of the turbulent wake increases accordingly, and it starts to interact with the pipes in adjacent longitudinal rows (see Fig. 2, fragment 51). Here the distance between the interacting pipes is shortened to $L = S_2$ (instead of $L = 2S_2$ in the cases considered above). All this compensates for the natural decrease in the level of disturbance in the wake with increasing distance L to the turbulizer, which also occurs far more slowly for $L/d > 7$ than for $L/d < 7$, judging from Fig. 3*.

It we examine the nature of the variation of the degree of stream turbulence at small distances from the turbulizer (Fig. 3), we arrive at a similar conclusion about the opposite limiting tendency of the parameter S_1/S_2: the level of stream disturbance should increase only up to some fairly high value of the ratio S_1/S_2, above which it should remain practically constant.

The above analysis, together with literature data [4, 5] on local heat transfer, provides a physical basis for the results of studies of the average heat transfer of staggered multirow bundles of finned tubes [6], on the basis of which the author suggests the generalizing equations

*After the stabilization section should come a section of decay to the level of disturbance of the stream incident on the bundle.

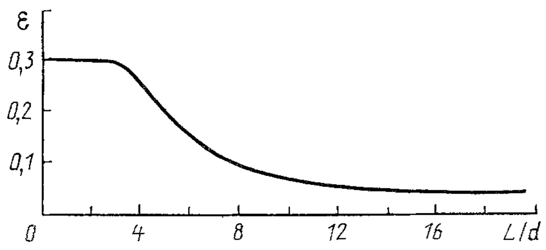

Fig. 3. Degree of turbulence ε as a function of the relative distance L/d to the turbulizer, based on data of [2].

$$\mathrm{Nu} = 1.13C_q \mathrm{Re}^m \mathrm{Pr}^{0.33}; \tag{1}$$

$$m = 0.7 + 0.008\,\mathrm{th}\left(\frac{S_1}{S_2} - \frac{1,26}{\psi} - 2\right) + 0.005\psi; \tag{2}$$

$$C_q = \left[1.36 - \mathrm{th}\left(\frac{S_1}{S_2} - \frac{1,26}{\psi} - 2\right)\right]\left(\frac{1,1}{\psi + 8} - 0.014\right), \tag{3}$$

recommended for use in the ranges $\psi = 1.2\text{-}39.0$, $S_1/S_2 = 0.3\text{-}5.2$, and $\mathrm{Re} = 5 \cdot 10^3\text{-}2 \cdot 10^5$.

The special features of [6] that make it possible to generalize experimental data in such wide ranges of geometrical and operating characteristics include, in particular, the fact that in it the dependence of the exponent m to the Reynolds number in Eq. (1) on the bundle spacing characteristics S_1/S_2 was revealed and investigated [Eq. (2), Fig. 4]. If we consider that an investigation of local heat transfer for smooth and finned cylinders [4, 5] indicates the existence of a direct relationship between the level of stream disturbance and the exponent m, then all that was said above about the dependence of the level of stream disturbance on S_1/S_2 may pertain to the dependence of m on S_1/S_2; i.e., the very fact of a relationship between m and S_1/S_2, as well as its asymptotic nature (Fig. 4), follow from the development of the hydrodynamic pattern of the process with variation of the spacing characteristics of the bundle.

The studies of flow in bundles also provided a physical basis for the general dependence of the heat transfer intensity for finned bundles on the tube spacing parameter S_1/S_2. In Fig. 5a we show experimental data on $\mathrm{Nu} = f(S_1/S_2)$ at Re = const obtained by the author for four series of bundles with different finning coefficients ψ, and curves constructed from Eqs. (1)-(3) to approximate these data. The heat transfer varies by 30-50% in the investigated range of $S_1/S_2 = 0.3\text{-}5.2$. The extremal nature of these functions should be noted; the position of the extremum is not fixed but shifts toward larger S_1/S_2 with decreasing finning coefficients. For $\psi = 38.32$, for example, the extremum corresponds to $S_1/S_2 \approx 2.0$, while for $\psi = 1.46$ it corresponds to $S_1/S_2 \approx 3.0$, i.e., for each finning coefficient there is a tube spacing that yields the maximum heat transfer intensity.

An analysis showed that the coordinate $(S_1/S_2)_{max}$ of the heat transfer maximum can be determined using the relation

$$\left(\frac{S_1}{S_2}\right)_{\max} = \frac{1.26}{\psi} + 2 + \xi, \tag{4}$$

where ξ is a function of the Reynolds number,

$$\xi = \frac{1}{2} \ln \frac{0.189 \ln \mathrm{Re} - 1}{1 - 0.029 \ln \mathrm{Re}}. \tag{5}$$

For Re = 9800, we have $\xi = 0$. With increasing Reynolds number, ξ increases, and $(S_1/S_2)_{max}$ with it. But $(S_1/S_2)_{max}$ depends very weakly on Re: as the Reynolds number increases from $1 \cdot 10^4$ to $6 \cdot 10^4$, the coordinate of the maximum shifts by only 0.23. Considering this, the dependence of $(S_1/S_2)_{max}$ on the Reynolds number can be ignored in practical calculations.

The nature of the variation of heat transfer as a function of the parameter S_1/S_2 that was revealed can be explained by analyzing the corresponding variation of the two main hydrodynamic factors that determine heat transfer intensity in bundles of pipes for Re = const:

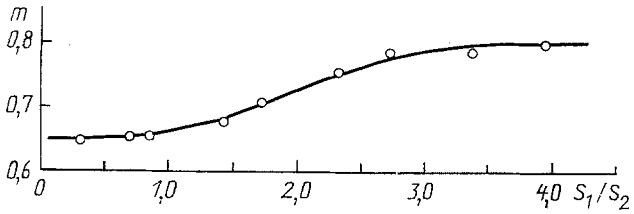

Fig. 4. Dependence of the exponent m to the Reynolds number on S_1/S_2 based on Eq. (2) with ψ = const ($\psi = 6.4$).

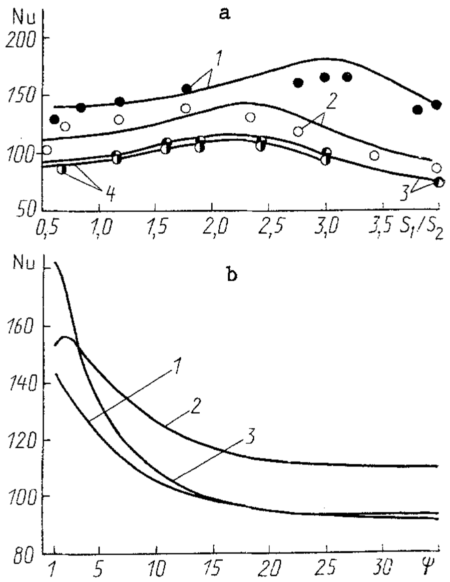

Fig. 5. Influence of the geometrical characteristics of bundles of finned tubes on the heat transfer intensity ($Re = 2.5 \cdot 10^4$): a) influence of the tube spacing parameter S_1/S_2; 1) $\psi = 1.46$; 2) 6.40; 3) 18.25; 4) 38.32; b) influence of the finning coefficient ψ; 1) $S_1/S_2 = 1.0$; 2) 2.0; 3) 3.0.

the level of disturbance of the stream and the degree of its compression. The maximum effect from the action of each of these factors corresponds to a different value of the parameter S_1/S_2. For example, the degree of stream compression, which largely determines the ratio of flow rates of the medium in the channels between fins and between pipes, and hence the level of local velocities of bathing of the finned tubes reach a maximum at $S_1/S_2 = 2/\sqrt{3} = 1.16$ (see Fig. 2, fragment 33). In this case, each tube is in a kind of annular deflector formed by the six adjacent tubes. Practically all of the coolant then flows through the channels between fins, and the rear vortex zone is reduced to a minimum. As seen from Fig. 5a, however, the heat transfer maximum does not fall at this value of the spacing parameter S_1/S_2 in practically the entire range of variation of the finning parameter (ψ). The point is that a second factor, the level of stream disturbance in this case, is still fairly low, as noted above. The level of disturbance, and the fraction of the heat transfer intensity associated with it, increase with increasing S_1/S_2 up to $S_1/S_2 = 3.0$-4.0. The maximum of heat transfer intensity, which is determined by the combined action of the two hydrodynamic factors, is therefore shifted into the region $S_1/S_2 > 1.16$ and occurs where the degree of compression has not yet decreased considerably and the level of disturbance is already fairly high.

The shift of the coordinate of the heat transfer maximum along the S_1/S_2 axis in accordance with Eq. (4) with variation of the finning parameter is explained by the different sensitivity of the heat transfer system under consideration to each of the identified hydrodynamic factors for different degrees of finning ψ. In fact, the role of the degree of stream compression, which reaches a maximum at $S_1/S_2 = 1.16$, is greater for larger finning coefficients, i.e., when the effect of displacement of the stream from the channels between fins is more pronounced due to the increased thickness of the boundary layer on a fin. For larger ψ, the heat transfer maximum is therefore shifted to smaller S_1/S_2. With a decrease in ψ, the role of the degree of compression decreases, and the increase in the level of disturbance starts at higher values of S_1/S_2, which together result in a shift of the heat transfer maximum toward the right (Fig. 5a).

The nature of the variation of the heat transfer intensity of finned tubes with variation of the finning parameter for S_1/S_2 = const and Re = const is also interesting. The results of our analysis are shown in Fig. 5b. The heat transfer intensity decreases (by 40-80%) with increasing finning coefficient in almost the entire range of ψ investigated, which agrees with the data of [7, 8]. With allowance for the results of [1], however, we can state that this decrease is not related to the appearance and growth of stagnant zones in the rear parts of the fins. The causes of the decrease in the heat transfer intensity for bundles of tubes as their degree of finning increases must include:

a decrease in the fraction of transversely bathed surface of a finned tube, which is more efficient in heat transfer, and a corresponding increase in the fraction of the less efficient longitudinally bathed surface, i.e., the fin surface;

an increase in the thickness of the boundary layer on a fin with an increase in its height;

a decrease in the fraction of fin surface occupied by the rear detached circulation zone [1], within which the level of heat transfer is higher, on the whole, than in the average over the fin surface;

an increase in the relative size of the rear vortex zone, within which the level of heat transfer is lower, on the whole, than in an average over a fin.

At the same time, we must note the existence, at certain values of the spacing parameters, of an extremum of the function Nu = $f(\psi)$ in the range of low degrees of finning.

NOTATION

d, diameter of a finned tube; h, fin height; t, fin spacing; δ, fin thickness; ψ, coefficient (degree) of finning; S_1, transverse spacing of tubes in the bundle; S_2, longitudinal spacing of tubes in the bundle; ε, degree of stream turbulence; L, distance to the turbulizer; Nu, Nusselt number, calculated from the coefficient of convective heat transfer α averaged over the surface and from the diameter d; Re, Reynolds number, calculated from the velocity in the narrowest cross section of the bundle and from the diameter d; C_q, experimental coefficient in the equation of similarity of convective heat transfer; m, exponent to the Reynolds number; ξ, quantity allowing for the dependence of the coordinate $(S_1/S_2)_{max}$ of the heat transfer maximum on the Reynolds number; Pr, Prandtl number.

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