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DRAG OF CRISSCROSS BUNDLES OF FINNED TUBES IN TRANSVERSE STREAM OF FLUID

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Experimental data are presented on the form resistance and the hydraulic drag of crisscross bundles of finned tubes, whereupon the Berman number and the Chen number are calculated for the near zone of the wake behind a finned tube.

In modern power engineering equipment one often uses heat exchangers with finned heating surfaces.

A survey of published studies reveals that no basic research has been done on bundles of finned tubes in a transverse air stream and that no detail data are available on the heat transfer at such bundles and their drag in a stream of viscous fluid.

In one study [1], the pressure distribution was determined around a tube with three different types of transverse finning, in the fourth row of a 1.64×1.42 crisscross bundle at $Re = 2.93 \cdot 10^4$ and at $Re = 3.01 \cdot 10^4$; in another study [2], such tubes, all with the same type of finning, were used in each row of a 1.93×1.69 crisscross bundle and in a 1.93×1.87 corridor bundle for determination of the pressure distribution at $Re = 3.1 \cdot 10^4$.

The experimental data obtained by various authors pertaining to hydraulic drag of bundles of tubes have been subsequently generalized [3, 4].

The objects of this study were eight crisscross bundles in various configurations, with the effective tube finning parameter h/s varied over approximately the 0.3-1.6 range. Typical data have been obtained on the local hydrodynamic characteristics of compact finned tube bundles, much attention now being devoted to the study of such bundles. In these studies one simultaneously determines the optimum geometrical characteristics for heat transfer and for streamlining by high-viscosity fluids, with the Prandtl number varying from 80 to 5000 and the Reynolds number varying from 20 to $2 \cdot 10^6$.

Method of Study. The study was made using a circulation system with transformer oil [5].

The dependence of the drag on the tube finning parameters was studied with tubes in a bundle spaced longitudinally and transversely in uniform steps (bundles No. 6-8 in Table 1). An "infinite" bundle was composed of half-tubes. The experimental tubes (Fig. 1), furnished with pressure-head gauges, had the following dimensions: $L = 150$, $\beta - 2\text{-}4°$, and d, h, δ_1, δ_2 either 45, 15, 1.29, 9 or 15, 5, 0.43, 3, respectively.

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TABLE 1. Characteristics of Bundles

Bundle No.	a	b	d	s	h	No. of tubes in a row	ε	δ_2/δ_1
1	1,13	1,06	15	3	1	8	1,43	4,1
2	1,33	1,23	15	3	2,5	7	2,66	4,95
3	1,66	1,53	15	3	5	6	5,13	6,95
4	1,13	1,06	45	9	3	2	1,43	4,1
5	1,33	1,23	45	9	7,5	2	2,66	4,95
6	1,66	1,53	45	9	15	2	5,13	6,95
7	1,66	1,53	45	9	7,5	2	2,66	4,95
8	1,66	1,53	45	9	3	2	1,43	4,1

Note. z = 7.

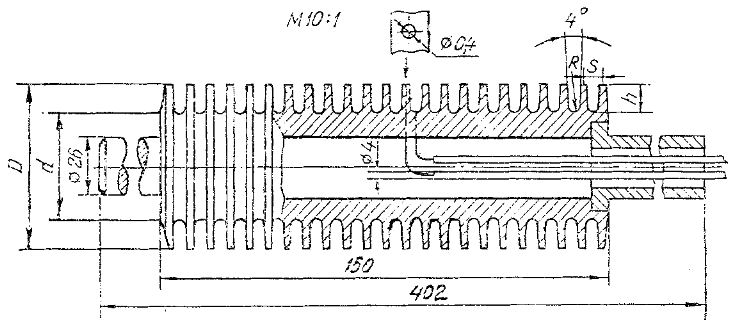

Fig. 1. Tube for hydrodynamic studies.

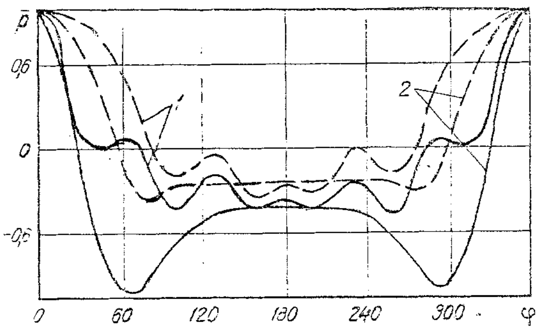

Fig. 2. Variation of $\bar{p}$ over the surface of a finned tube in first row (dashed line) and in fifth row (solid line) of bundle: 1) at tip of fin; 2) at root of fin.

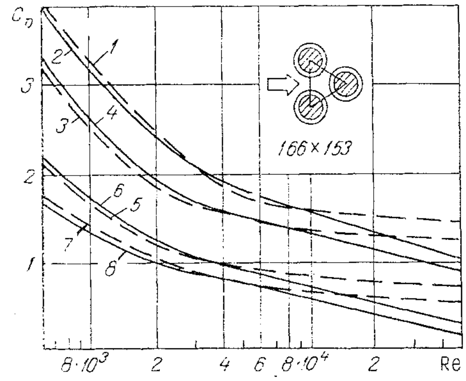

Fig. 3. Variation of c_D in first row (dashed lines) and in fifth row (solid lines) of bundles Nos. 3 and 6: 1, 2) at tip of fin with h = 15 mm; 3, 4) at tip of fin with h = 5 mm; 5, 6) at root of fin with h = 15 mm; 7, 8) at root of fin with h = 5 mm (1.66 × 1.53).

For processing the experimental data, the governing parameters used were the diameter d of a finned tube, the mean velocity w in the smallest active section of the bundle, and the temperature t of the oncoming stream.

Form Resistance. A study of the hydrodynamics of flow at finned surfaces should reveal the characteristics of local heat transfer at the tip and the root of a fin; it should also provide a better understanding of the heat transfer mechanism. Here experimental data will be presented on the pressure distribution around two typical finned tube bundles (Nos. 3 and 6) as well as on their shape stability. First of all, $p = f(\varphi)$ was determined at the tip and at the root of a fin.

The circumferential pressure distribution factor is, in dimensionless form,

$$\bar{p} = 1 - \frac{p_{\varphi=0} - p_{\varphi}}{1/2\rho w^2}. \quad (1)$$

In the study of bundles No. 3 and 6 (Table 1), $\bar{p} = f(\varphi)$ was measured in the first and fifth rows of tubes. The shape of the $\bar{p} = f(\varphi)$ diagram was found to depend on the depthwise location of the row and on the Reynolds number.

The streamlining in the first and fifth rows of bundle No. 6 at $Re = 4.9 \cdot 10^4$ was found to be symmetric, with the $\bar{p} = f(\varphi)$ diagram having a different shape at the tip and at the root of a fin (Fig. 2).

On the front side of a tube in the first row, $\bar{p}$ decreases. At the root of a fin $\bar{p}$ reaches its minimum at angles $\varphi = 80°$ and $280°$; on the back side of a tube $\bar{p}$ increases somewhat and remains uniform beginning at angle $\varphi = 100°$ ($260°$). (At the tip of a fin on the front side of a tube $\bar{p}$ does not reach its minimum.) On the back side of a tube there are two minima at $\varphi = 100°$ ($260°$) and $160°$ ($200°$), respectively, and two maxima at $\varphi = 130°$ ($230°$) and $180°$, respectively, owing to the presence of neighboring tubes.

In the fifth row $\bar{p}$ at the tip and at the root of a fin is correspondingly lower than in the first row. At the tip of a fin on the front side of a tube in the fifth row, $\bar{p}$ reaches its minimum at $\varphi \approx 45° (315°)$; no such minimum appears in the first row. At the root of a fin $\bar{p}$ reaches its minimum at $\varphi = 65°$ and $295°$, but then quickly increases and levels off at $\varphi \approx 140° (220°)$.

The pressure distribution pattern at various values of the Reynolds number has revealed that the recirculation zones become wider as Re increases. This causes a gradual shift of the extremum points toward the stagnation point on the front side. This phenomenon of extremum points shifting upon changes in the Reynolds number was also observed in the study of bundles of smooth tubes [6]. Let us examine the dependence of $\bar{p} = f(\varphi)$ on Re for tubes in the first and fifth rows of a bundle.

As Re increases from $6.2 \cdot 10^2$ to $4.9 \cdot 10^4$, the extrema of $\bar{p} = f(\varphi)$ at the tip of a fin on a tube in the first row do not change their location in the first row but the extrema of $\bar{p} = f(\varphi)$ at the root of a fin shift correspondingly from $\varphi = 110$ ($250°$) to $\varphi = 80$ ($280°$).

In the fifth row an increase of Re over the same range causes the extrema at the tip of a fin in the bundle to shift correspondingly from $\varphi = 50, 120, 170°$ to $\varphi = 40, 100, 160°$ and the extrema at the root of a fin to shift from $\varphi = 90°$ to $\varphi = 70°$.

The value of coefficient c_D

$$c_D = \frac{2P_D}{F\rho w^2} \quad (2)$$

was determined from experimental data.

The quantity P_D was calculated according to the relation

$$P_D = \Delta s \sum_{i=1}^{n} p \cos\varphi, \quad (3)$$

with $\Delta s = \frac{\pi L d}{n}$, and with $\Delta s = \frac{\pi L D}{n}$.

According to the graph in Fig. 3, the value of coefficient c_D at the tip and at the root of a fin is higher in bundle No. 6 than in bundle No. 3 at the same Re. This is attributable

TABLE 2. Values of Constant k and Exponent r in Relation (11)

Bundle No.	Re	k	$-r$	Bundle No.	Re	k	$-r$
1	$1\cdot10^3$—$3\cdot10^3$	26,8	0,495	4	$4,0\cdot10^3$—$1,1\cdot10^5$	9,02	0,345
	$3\cdot10^3$—$3\cdot10^4$	6,07	0,31	5	$2,0\cdot10^3$—$6\cdot10^4$	6,99	0,31
2	$3,8\cdot10^2$—$2,3\cdot10^3$	38,4	0,555	6	$1,8\cdot10^3$—$5\cdot10^4$	7,15	0,31
	$2,3\cdot10^3$—$2,5\cdot10^4$	4,59	0,28	7	$8\cdot10^2$—$5\cdot10^4$	3,09	0,25
3	$2\cdot10^2$—$1,7\cdot10^3$	78,7	0,66	8	$5\cdot10^2$—$4,5\cdot10^4$	1,17	0,18
	$1,7\cdot10^3$—$2,5\cdot10^4$	4,75	0,283				

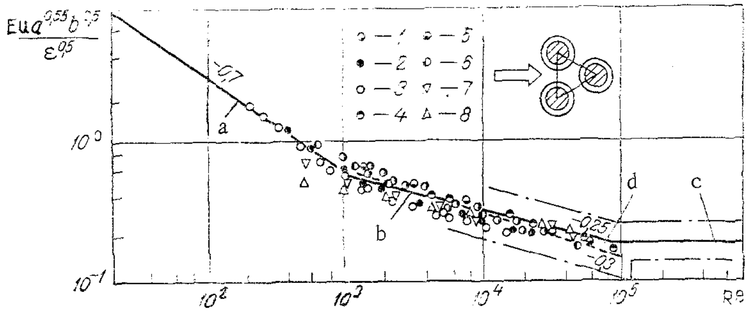

Fig. 4. Generalization of data on hydraulic drag of bundles (numbers next to dots refer to corresponding bundles). Fitting curves: a) according to data in this study and in study [11]; b, c) according to data in study [4]; d) according to this study.

to a finer vortex structure produced by streamlining of bundle No. 3. In both bundles the coefficient c_D has a higher value at the tip than at the root of a fin — in the first row 1.8 times higher at Re = $6\cdot10^2$ and 2.1 times higher at Re = $5\cdot10^4$, and in the fifth row 1.8 times higher at Re = $6\cdot10^2$ and 3.3 times higher at Re = $5\cdot10^4$.

At the same Re, the form resistance coefficient at the tip and at the root of a fin is approximately 1.2 times higher in bundle No. 6 than in bundle No. 3.

According to the graph in Fig. 3, the values of $c_D = f(\mathrm{Re})$ at the tip and at the root of a fin do not differ by more than 3% and 12%, respectively, between the first row and the fifth row at any Re up to 10^4. The difference between c_D in the first row and in the fifth row increases as Re increases above 10^4.

Determination of Berman Number Be and Chen Number Ch from Experimental Data. On the basis of these experimental data on the circumferential pressure distribution around a tube and on the form resistance coefficient, the hydrodynamic characteristics of a bundle of finned tubes were calculated, for the purpose of determining the Berman number Be = fh_S/w_S and the Chen number Ch = $fh_S{}^2/\Gamma$. The former is a more precise quantity than the Strouhal number, inasmuch as it contains characteristics of flow within the separation zone between boundary layer and tube surface. It increases only from 0.18 to 0.21, however, at large values of the Reynolds number (Re $\approx 2\cdot10^5$). At the same time, the Strouhal number increases approximately to 0.46 over the same range of streamlining characteristics. In the case of tube bundles, moreover, Sh depends on the configuration and the relative spacing $a \times b$ of tubes; its value here can become approximately 0.7 or even higher.

The more universal Chen number [7] has been introduced for a very wide range of Re, its value remaining 0.165 throughout. It is of scientific interest to establish how reliable the use of numbers Be and Ch will be for intricate flow patterns such as streamlining of developed heat exchanger surfaces, bundles of finned tubes being a typical example.

The frequency of vortex separations from bundle No. 6 was determined in accordance with established procedure [8-10], yielding Sh = 0.34 here. From the experimental data we now calculate w_S and Be = fh_S/w_S

$$w_S = k_0 w, \tag{4}$$

where $k_0 = \sqrt{1 - c_p}$, and $c_p = \frac{(p_S - p_{min})}{1/2\rho w^2}$.

The distance h_S between separated vortices is determined from experimental data on local heat transfer at the circumference of a tube, these data revealing the locations where vortices separate from the carrier segment of a tube, and also from visual observations and from photographs of the flow pattern. One can express the Berman number as

$$S_B = \frac{\mathrm{Sh}}{k_0} \frac{l}{d} \frac{h_S}{l}. \tag{5}$$

According to the experimental data for Re = $3 \cdot 10^3 - 5 \cdot 10^4$, the Berman number for bundle No. 6 does not differ from its proposed value [7] of 0.18 by more than 15%.

According to that study [7],

$$C = \frac{(h_S/d)(h_S/l)(fd/w)}{\Gamma/wl}. \tag{6}$$

The circulation Γ in expression (6) is determined from experimental data on c_D and h_S, namely

$$c_D = \frac{h_S}{d}\left[2\frac{\Gamma}{lw} - \frac{h_S}{l}\left(\frac{\Gamma}{lw}\right)^2\right]. \tag{7}$$

The constant Ch for the zone of flow separation from a tube is calculated according to relation (6). For Re ranging from 10^3 to $5 \cdot 10^4$ we have obtained Ch = 0.149, which differs by 9.4% from the value originally proposed [7]. It has thus been verified experimentally that more precise and universal separation criteria can be applied to compact bundles of tubes within the intermediate range $3 \cdot 10^3 - 5 \cdot 10^4$ of Re.

Knowing the circulation Γ, the vortex separation frequency, k_0, and w, we will now determine the energy expended on formation of vortices in finned bundles

$$\mathrm{E} = \frac{2\Gamma f}{k_0^2 w^2}. \tag{8}$$

The thus-obtained value E = 0.604 indicates that a large part of this energy contributes to formation of vortices within the flow separation zone. The values of Be, Ch, and E here refer to the nearest trail, i.e., the site where the boundary layer separates from the tube surface and vortices form.

Hydraulic Drag. The total hydraulic drag of the bundles (Table 1) was determined from the drop of static pressure across a bundle (difference between static pressures before and behind a bundle) and then generalized by the relation

$$\mathrm{Eu}_Z = \frac{\Delta p}{\rho w^2}. \tag{9}$$

The hydraulic drag of one transverse row in a bundle was calculated as the total drag divided by the number of rows

$$\mathrm{Eu} = \frac{\mathrm{Eu}_Z}{zc_z}. \tag{10}$$

The experimentally obtained values for bundles Nos. 1-3 differ very little over the entire range of Re. The Euler number is only 1.2 times larger for bundle No. 6 than for bundle No. 3. The experimental data on the hydraulic drag of the tube bundles (Table 1) were generalized by the relation

$$\mathrm{Eu} = k\,\mathrm{Re}^r. \tag{11}$$

The results of this generalization for one row of the bundles (Table 1) are given in Table 2.

For comparing the results of this study with those of studies made by other authors one must use the relation proposed in studies [3, 4]. The relation

$$\xi = 13.1(1 - s/d)^{1.8}(1 - h/d)^{-1.4} a^{-0.55} b^{-0.5} \mathrm{Re}^{-0.25} \tag{12}$$

proposed in study [3] is valid for Re < 10^5. A finning factor ε

$$\mathrm{Eu}/\varepsilon^m a^n b^q = k\,\mathrm{Re}^r \tag{13}$$

has been introduced in study [4] for generalization of experimental data over the 10^3-$1.4\cdot10^6$ range of Re.

In this study the values of the exponents at ε in expression (13) and at $(1 - h/d)$ in expression (12) were determined from experimental data on bundles Nos. 6-8, where the finning geometry had been varied with a and b held constant. Exponents m = 0.5 and −1.4 at $(1 - h/d)$ were obtained as a result.

Our experimental data have been generalized according to relation (13) (Fig. 4). For calculating the hydraulic drag over the $5\cdot10^2$-10^5 range of Re (Fig. 4), we propose the expression

$$\mathrm{Eu} = 4.7\varepsilon^{0.5}\mathrm{Re}^{-0.3}c_z/a^{0.55}b^{0.5}. \tag{14}$$

According to the graph in Fig. 4, the spread of experimental values falls within the range of relation (13) (dash-dot lines). On the basis of these data, as well as the data in studies [4, 11], therefore, for practical calculation of the hydraulic drag of crisscross bundles of finned tubes we have established the relations:

for Re from 20 to 10^3,

$$\mathrm{Eu} = 67.6\varepsilon^{0.5}\mathrm{Re}^{-0.7}a^{-0.55}b^{-0.5}c_z; \tag{15}$$

for Re from 10^3 to 10^5, and

$$\mathrm{Eu} = 3.2\varepsilon^{0.5}\mathrm{Re}^{-0.25}a^{-0.55}b^{-0.5}c_z; \tag{16}$$

for Re from 10^5 to $1.4\cdot10^6$

$$\mathrm{Eu} = 0.18\varepsilon^{0.5}a^{-0.55}b^{-0.5}c_z. \tag{17}$$

Expression (15) is valid for ε from 2.66 to 9.12, a from 1.33 to 2, and b from 1.23 to 1.73. Expressions (16) and (17) are valid for ε from 1.9 to 16.3, a from 1.6 to 4.13, and b from 1.2 to 2.35.

The relative correction factor c_Z in expressions (10) and (14)-(17), for the drag of bundles with only a few rows of tubes, is determined according to the procedure in study [12].

NOTATION

$a = s_1/d$ is the transverse relative pitch of a bundle; $b = s_2/d$, longitudinal relative pitch of a bundle; d(m), diameter of a finned tube; D (m), diameter of a fin; h (m), height of a fin; h_S (m), distance between two vortex strings along a vortex trail; δ_1 (m), thickness of a fin at its tip; δ_2 (m), thickness of a fin at its root; $l = s_2$ (m), distance between two adjacent vortices in a string; L (m), length of a tube; s (m), finning pitch; Δs (m^2), element of a cylindrical surface bounded by two generatrices passing through the centers of two measured segments; F (m^2), area of the cross section of a tube at its midspan; t (°C), temperature; φ (deg), angle read from the stagnation point on the front side; β (deg), angle of attack of a fin; ρ (kg/m^3), density; ν (m^2/sec), kinematic viscosity; w (m/sec), mean velocity of the stream in the smallest active section of the bundle; w_S (m/sec), velocity of the stream at the point of separation between boundary layer and cylindrical surface; f (Hz), vortex separation frequency; Γ (m^2/sec), circulation of a vortex; p (N/m^2), pressure; p_S (N/m^2), pressure at the point of separation between boundary layer and cylindrical surface; p_{min} (N/m^2), pressure in the smallest active section of a bundle; $\bar{p}$, pressure distribution factor; c_D, form resistance coefficient of a tube; c_P, bottom pressure coefficient; ε, tube finning factor; k, proportionality factor; E, vortex formation factor characterizing the fraction of total energy of the separating layer expended on formation of vortices; c_Z, correction factor for drag of bundles with only a few rows; z, number of transverse rows of tubes in a bundle; Bĕ, Berman number; Ch, Chen number characterizing a vortex trail; Pr, Prandtl number; Re, Reynolds number; Sh, Strouhal number; and Eu, Euler number.

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EFFECT OF A CHANGE OF THE TURBULENT STRUCTURE OF A FLOW ON UNSTEADY HEAT EXCHANGE UPON HEATING OF GASES AND LIQUIDS IN PIPES

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The article presents the results of experimental investigations of the unsteady heat exchange upon heating of gases and liquids and of the change in heat liberation from the pipe walls, as well as of the generalization of the experimental data which confirm the model of the effect of a change in the turbulent structure of the flow on the heat exchange.

Earlier investigations of the heat exchange accompanying the heating of a gas in a channel and of the change of heat flow density on the wall vs time [1] showed that the effect of the thermal nonsteady state on the heat emission coefficient increases with decreasing Reynolds number. The results of [1] were obtained chiefly for $Re_b = 8 \cdot 10^4 - 5.8 \cdot 10^5$, and the pipes used had almost equal diameters (d = 5.56 and 5.39 mm). It is therefore of interest to investigate unsteady heat exchange accompanying turbulent flow in a pipe with smaller Reynolds numbers Re_b and to verify the previously obtained theoretical dependences in pipes with other diameters.

An analysis in [2] showed that the difference between the unsteady and the quasisteady heat emission coefficients is due to the change of turbulent structure of the flow when the wall temperature increases or decreases. The main part in this is played by the nonsteady state of the turbulent boundary conditions $\partial T_w / \partial \tau$ which, as was shown in [3, 4], is most conveniently taken into account by the parameter

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