GENERALIZATION OF TEST DATA PERTAINING TO THE MEAN HEAT TRANSFER COEFFICIENT FOR TUBE BUNDLES IN A TURBULENT LONGITUDINAL STREAM OF AIR AND SUPERHEATED STEAM

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A method has been developed for the generalization of test data on heat transfer which takes into account the effects of the temperature factor and of the bundle geometry. From a compilation and an analysis of the test data have come out new and reliable recommendations for the thermal design of heat exchangers.

The technique of placing tube bundles in a longitudinal stream has found many engineering applications. Despite the many studies dealing with this subject, not in one has the effect of geometrical factors been considered and, for this reason, the various results obtained differ by as much as 20% or more. No attempt has been made in published reports to refer all available data to the same set of conditions and to evaluate them on the basis of one common criterion. This explains why so many experiments have been performed with the same tube spacing, with smooth tubes, with wire-wound tubes, or with finned tubes in an attempt to improve the heat transfer.

In [4, 8, 11] this author has established a linear relation between the Nusselt number and the bundle geometry, the latter defined in terms of a correction factor $(S_1S_2/d^2)^{0.18}$ to Mikheev's formula. The governing dimension is the overall hydraulic diameter of the channel formed by tubes with a diameter d and distances S_1, S_2 between axes.

In generalizing the already published data pertaining to air and superheated steam, the author aimed at the simplest formulas convenient for practical applications and yet sufficiently accurate for engineering design, with consideration given to the wide circulation of nomograms as an aid in computations by the normative method [7] or the Mikheev method [21].

The test data on heat transfer were compared with formulas for circular pipes:

according to B. S. Petukhov for Re = 10^4–10^6 and Pr = 0.7–200

$$\mathrm{Nu}_\mathrm{T} = \frac{\xi \mathrm{RePr}\,(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.11}}{36\sqrt{\xi}\,(\mathrm{Pr}^{2/3}-1)+8.56}, \tag{1}$$

according to M. A. Mikheev for Re = 10^4–$2\cdot10^6$ and Pr = 0.7–200

$$\mathrm{Nu}_\mathrm{T} = 0.021\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.43}\,(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.25}, \tag{2}$$

according to S. S. Kutateladze for Re = 10^4–$2\cdot10^6$ and Pr = 0.5–25

$$\mathrm{Nu}_\mathrm{T} = 0.023\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}. \tag{3}$$

Values of the Nusselt number according to Petukhov, Mikheev, and Kutateladze are compared in Table 1. The calculations have shown that formulas (1) and (3) yield almost identical results, within the accuracy of the generalization (Pr = 1). For air C = 0.020 according to (3) and C = 0.018 according to (2).

The correction $\mathrm{Pr}_f/\mathrm{Pr}_w \simeq \mu_f/\mu_w$ to the power 0.11 or 0.25 is not applicable to gases. The reduction in the heat transfer rate is taken into account by adding on the right-hand side of Eqs. (1), (2), and (3) the

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TABLE 1. Compilation of Values for the Nusselt Number Nu According to Formulas (1), (2) (3) for Stabilized Flow through a Straight Smooth Pipe with Pr = 1 (Deviation in % from the values according to the formula based on the hydrodynamic theory of heat transfer)

Formula	Re		
	10^4	10^5	10^6
Friction coefficient ξ for isothermal flow through smooth pipes (Tables 1-5, p. 247 in [20]).	0,0316	0,0177	0,0115
$Nu=0,125\xi PrRe$	39,5	221	1440
W. H. McAdams (1942), S. S. Kutateladze (1970)	36,5 —7,6	230 +4,0	1450 +0,9
B. S. Petukhov (1958)	37 —6,3	210 —5,0	1368 —5,0
M. A. Mikheev (1959)	33,3 —15,7	210 —5,0	1320 —8,3

temperature factor $(T_w/T_f)^{-m} = \psi_t$, where T_f and T_w denote the mean absolute temperature of the gas and of the tube wall, respectively (in °K). According to [20], we may let m = 0.5 without a large error. Therefore, for heating air ($1 < T_w/T_f < 4$) we have $Nu_B = N_T/\psi_t$ and for heating superheated steam we have Nu_B $Nu_B(Pr_f/Pr_w)^{0.11}$.

Thus, a final generalization and analysis of test data for 21 arrays of bundles has been based on formulas:

TABLE 2. Characteristics of Staggered (S) and Aligned (A) Experimental Tube Bundles with Regard to Generalization of the Heating of Air and Superheated Steam

Published source	Point No. in Fig. 1	$\frac{S_1}{d}$	$\frac{S_2}{d}$	$\frac{S_1 S_2}{d^2}$	ψ_s	z	d, mm	d_H, mm	l, m	$\frac{l}{d_H}$	d_g	$Re\cdot 10^{-3}$
M. A. Mikheev [1] (1931), S	3	1,50	1,30	1,95	1,35	19	19	30,6	1,0	32	21,8	6—25
	4	2,00	1,74	3,48	1,30	19	19	60,2	1,0	16	54,0	8—24
D. M. Ioffe [2] (1950), A	6	1,69	1,64	2,77	1,36	42	8,6	16,5	0,5	30	21,6	3—25
	7	2,00	2,00	4,00	1,25	30	8,6	22,0	0,5	23	35,0	5—35
	8	2,45	2,36	5,78	1,20	20	8,6	33,0	0,5	15	55,0	8—45
I. V. Kipriyanov [3] (1952), A	5	1,50	1,50	2,25	1,36	4	12	19	1,9	100	22,0	4—45
Readings taken at the YTI, GT-10 (1953), A	20	1,50	1,50	2,25	1,40	36	19	27,3	2,0	73	28,0	6—35
A. P. Salikov [5] (1954), S	11	2,37	2,06	4,86	1,22	39	19	73,3	2,0	27	82,5	5—52
	12	2,05	1,78	3,65	1,28	39	19	54,5	2,0	37	57,0	5—50
	13	1,76	1,52	2,68	1,37	39	19	37,9	2,0	53	37,0	5—45
T. Hobler [22] (1958), S	18	1,47	1,28	1,88	1,37	7	30	45	3,0	66	41,7	5—30
D. Palmer [24] (1961), S	19	1,015	1,015	1,03	1,00	7	168	23,7	0,76	32	19,0	10—60
H. Hoffman [25] (1961), S	21	1,71	1,48	2,54	1,42	7	20	27	1,018	38	36,0	75
R. M. Higgins [26] (1962), S	22	1,33	1,12	1,46	1,16	7	19	15,6	0,76	49	11,6	10—70
N. Kattchee [27] (1963), S	23	1,28	1,11	1,42	1,17	19	6	5,7	0,71	125	3,3	10—40
G. I. Kemel'man [9] (1964), S	9											
	10	1,56	1,35	2,10	1,42	7	32	39,7	4,40	110	61,7	40—220
K. Koziol [23] (1965), S	14	1,25	1,08	1,35	1,15	19	30	29,8	2,06	69	14,5	2—40
	15	1,50	1,30	1,95	1,40	13	30	48,0	2,06	43	34,0	2—50
	16	1,50	1,30	1,95	1,38	19	30	41,7	2,49	60	34,0	3—30
W. Sutherland [28] (1966), S	17	1,25	1,08	1,35	1,13	—	—	—	—	—	—	7—200
G. A. Dreitser [12] (1966), S: heating	1	1,20	1,04	1,25	1,10	19	11	6,15	0,8	130	4,1	10—80
cooling	2	1,20	1,04	1,25	1,08	19	11	6,15	0,8	130	4,1	10—120

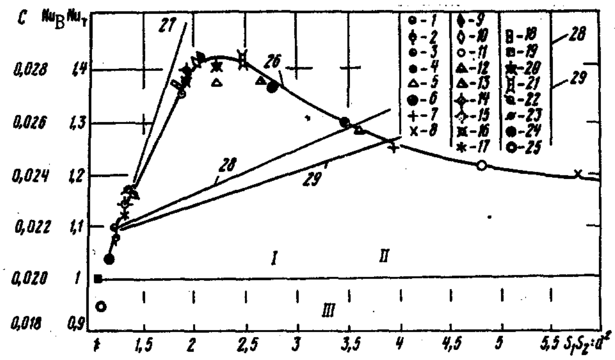

Fig. 1. Referred Nusselt number Nu_B/Nu_T and coefficient C as functions of the geometry factor S_1S_2/d^2 for both staggered and for aligned longitudinal bundles of tubes: 1) heating of air; 2) cooling of air as per [12] for $\Gamma = 1.25$; 3) as per [1] for $\Gamma = 3.48$; 4) as per [1] for $\Gamma = 1.95$; 5) as per [3] for $\Gamma = 2.25$; 6) as per [2] for $\Gamma = 2.77$; 7) as per [2] for $\Gamma = 4.00$; 8) as per [2] for $\Gamma = 5.78$; 9) with counterflow; 10) for heating of superheated steam with parallel flow as per [9] for $\Gamma = 2.10$; 11) for $\Gamma = 4.86$; 12) for $\Gamma = 3.65$; 13) for $\Gamma = 2.68$; 14) as per [23] for $\Gamma = 1.35$; 15) z = 13; 16) z = 19 as per [23] for $\Gamma = 1.95$; 17) as per [28] for $\Gamma = 1.35$; 18) as per [22] for $\Gamma = 1.88$; 19) as per [24] for $\Gamma = 1.03$; 20) as per readings taken at the VTI, GT-10 for $\Gamma = 2.25$; 21) as per [25] for $\Gamma = 2.54$; 22) as per [26] for $\Gamma = 1.46$; 23) as per [27] for $\Gamma = 1.42$; 24) as per [16] for $\Gamma = 1.25$; 25) as per [16] for $\Gamma = 1.14$. Straight lines 27, 28, and 29 represent the linear relations obtained as a result of generalization by Kalinin [15] for staggered bundles (1.1 < 1.5) by Ain and Puchkov [16] (1.1 < S/d < 2.4), and by Borishanskii [17], respectively [1) Petukhov's formula; II) Kutateladze's formula; III) Mikheev's formula].

for heating superheated steam

$$Nu_B = 0.023 Re^{0.8} Pr^{0.4} (Pr_f/Pr_w)^{0.11} (S_1S_2/d^2)^n \tag{4}$$

for heating air (Pr = 0.7)

$$Nu_B = 0.020 Re^{0.8} (T_w/T_f)^{-0.5} (S_1S_2/d^2)^n. \tag{5}$$

The purpose of this study was to establish relations which will describe the referred Nusselt number Nu_B/Nu_T as well as the coefficients for superheated steam $C = Nu/Re^{0.8} Pr^{0.4} (Pr_f/Pr_w)^{0.11}$ and for air $C = Nu/Re^{0.8} (T_w/T_f)^{-0.5}$ as functions of the bundle pattern factor $\psi_S = (S_1/S_2/d^2)^n$, with S_1 and S_2 denoting the distances between axes of tubes along a mutually perpendicular direction in the cross section plane through a bundle, and with the power exponent n of the geometry factor depending on the tube spacing.

The equivalent hydraulic diameter was defined according to the formula $d_H = 4F/u$, with F denoting the active section of the gas duct and u denoting the wet perimeter. For a gas duct filled with tubes

$$d_H = \frac{4ab - z\pi d^2/4}{2(a+b) + z\pi d}, \tag{6}$$

with a and b denoting the transverse inside dimensions of a gas duct, and with z denoting the number of tubes in a gas duct.

The characteristics of generalized tube bundles are given in Table 2. In [9] α was determined in terms of the heat transmission coefficient, in [27] the results were obtained by the method of transient heat transfer, and in the other studies they were obtained by the method of steady heat transfer. When inlet and outlet were at right angles, the mean heat transfer coefficient at the bundle was determined within the zone where the heat transfer had been stabilized. With the inlet exactly parallel to the stream, the mean heat transfer coefficient was determined for the entire length of the test zone [1, 3]. In [5] the heat transfer coefficient for the stabilization zone was converted to the mean heat transfer coefficient (q_w = const) according to [10, 18].

TABLE 3. Correction ψ_s to Pipe Formulas Accounting for the Bundle Geometry

$\frac{S_1S_2}{d^2}$	Correction	
	B. S. Petukhovs normative method's, S. S. Kutateladze's	M. A. Mikheev's
1,25	1,09	1,19
1,50	1,20	1,30
1,75	1,30	1,40
2,00	1,40	1,50
2,25	1,43	1,53
2,50	1,41	1,51
3,00	1,36	1,46
3,50	1,30	1,40
4,00	1,26	1,36
5,00	1,21	1,31
6,00	1,19	1,29

The referred Nusselt number and the coefficient C are shown in Fig. 1 as functions of the geometry factor, covering all published data on longitudinal bundles (USSR, USA, and Polish sources). On the basis of this graph and Table 3, we have derived a correction factor ψ_s to formulas (1), (2), (3) accounting for the bundle geometry.

Owing to the lack of tables of original test data, in [16, 22, 24, 27] the conversion according to the method used in Fig. 1 was made on the basis of the respective formulas and, therefore, those test points were not plotted in Fig. 2.

In Fig. 2 the test data have been generalized in the form $K_i = f(\mathrm{Re})$ referred to the mean heat transfer coefficient:

during heating of air (Pr = 0.7)

$$K_1 = \mathrm{Nu}/(T_w/T_f)^{-0.5}(S_1S_2/d^2)^n = 0.020\mathrm{Re}^{0.8},$$

during heating of superheated steam ($\mathrm{Pr} \simeq 1$)

$$K_0 = \mathrm{Nu}/(\mathrm{Pr}_f/\mathrm{Pr}_w)^{0.11}(S_1S_2/d^2)^n = 0.023\mathrm{Re}^{0.8}.$$

For convenience, the curves based on formulas (4) and (5), as well as the test points obtained by various authors, have been segregated into four groups with the K_i complex multiplied by 0.3, 0.5, 0.75, and 1.0, respectively (curves *a*, b, d, e) for air and one group (curve c) for superheated steam.

The test points clustered along curve *a* represent the results obtained at the Moscow Aviation Institute [12], where air was heated and cooled in a staggered bundle of 19 tubes with a hexahedral displacer piston. For the cooling of air ($0.5 < T_w/T_f < 1$), the correction $1.27-0.27\,(T_w/T_f)$ amounts to only 2-4%, i.e., remains within the generalization error. For the heating of air, ψ_t = 3-11%. The test conditions were as follows: temperature difference Δt = 17-100°C, thermal flux q = 200-31,000 kcal/m$^2\cdot$h, air velocity w = 20-370 m/sec, Reynolds number Re = 10,000-120,000, heat transfer coefficient α = 20-1150 kcal/m·h·deg, and Nusselt number Nu = 26-215.

The test points along curve b represent: 1) the results obtained by Mikheev [1], who heated and cooled air in a staggered bundle of 19 tubes with a cylindrical displacer, with natural and forced convection either aiding or opposing each other (Δt = 18-240, T_w/T_f = 1.13-1.74, ψ_t = 6-32%, q = 500-6000, w = 1-16, α = 8-60, Re = 3000-32,000, Nu = 13-83); 2) the results obtained by Ioffe [2] with an aligned bundle and a quadrilateral displacer piston in a longitudinal stream; and 3) the results obtained by Kipriyanov [3] with a duct containing four tubes inside (aligned bundle). Kipriyanov had studied the heat transfer in ducts very different from multitubular bundles and, for this reason, his data were used in our generalization for reference only.

The test points along curve c represent the results obtained at the All-Union Technical Institute (VTI) [9], where heat transfer with a bundle of seven tubes in a longitudinal stream of superheated steam (q = 10,000-70,000) was studied for the first time. The steam–steam heat exchanger for regulating the secondary superheat had been designed for long-term duty under the following conditions: primary steam 275 atm·abs and 500°C, secondary steam (passing through the intertubular space) 40 atm·abs and 400°C, corresponding to actual conditions in serially manufactured boiler aggregates for supercritical pressures in 200, 300, and 500 MW generating plants.

The test points along curve d represent the results obtained by Salikov [5], who heated air in a staggered bundle of 39 tubes with a quadrilateral displacer piston. The test points along curve e represent the results obtained by Koziol [23] and Sutherland [28]; the dashed curve g represents the results of calculations according to the formula $\mathrm{Nu} = 0.06\,\mathrm{Re}^{0.75}\,\mathrm{Pr}^{0.4}$ in Hobler's study [23] of heating the air with saturated steam. In [23] the calculations were made very carefully and taking into account the radiative heat transfer, the latter are likely to reach a 20-25% level at large temperature differences (at low values of the Reynolds number).

It has been established (Figs. 1 and 2) that in such an evaluation better results are obtained if the dependence of the heat transfer coefficient on the bundle geometry is taken into account, and that then the overwhelming number of test points will spread by not more than ±4-6%.

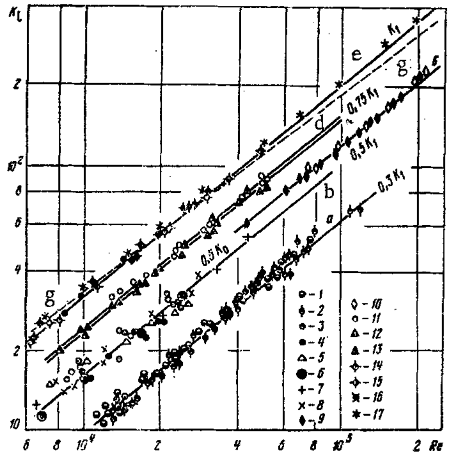

Fig. 2. Generalization of test data pertaining to the mean heat transfer coefficient for rod bundles in a stream of air (a, b, d, e) or superheated steam (c); legend is the same as for Fig. 1.

For comparison, in Fig. 1 are also shown straight lines representing the generalizations according to [15, 16, 17]. Evidently, our nonlinear relations agree with the test data very accurately.

The generalization of test data pertaining to the mean heat transfer coefficient for tube bundles in a turbulent longitudinal stream of water will, if made by the proposed method, yield results analogous to those obtained here.

Thus, the mean heat transfer coefficient for a tube bundle in a turbulent longitudinal stream of air or superheated steam ($Pr = 0.7\text{-}1.0$) can be calculated according to the formulas of Petukhov, McAdams, Kutateladze (normative method), and Mikheev for a pipe, with corrective temperature and geometry factors (Table 3) for the following ranges: $Re = 6000\text{-}220{,}000$; $S_1/d = 1.1\text{-}2.45$; $S_2/d = 1.1\text{-}2.36$; $S_1S_2/d^2 = 1.25\text{-}6$.

The results of this generalization can be extended to gases under moderate pressures and to liquids at high temperatures. In calculating the heat transfer for water and steam far from the critical region and under actual conditions prevailing in modern boilers ($1 < Pr < 2$), these formulas may be used without the temperature correction factor, inasmuch as the temperature differences are small.

Thus, thermal calculations for parallel rod bundles by the conventional method [13, 29] yield heating surface areas in heat exchangers which are larger than necessary and would result in a waste of material.

NOTATION

α	is the mean heat transfer coefficient;
$\Gamma = S_1S_2/d^2$	is the geometry factor;
$\psi_S = (S_1S_2/d^2)^n$	is the bundle pattern factor;
d	is the outside diameter of a tube;
d_H	is the governing dimension (hydraulic diameter of a complete bundle with sheath);
l	is the length of the test zone;
z	is the number of tubes in a bundle;
d_∞	is the thermal-equivalent diameter of the center cells in a bundle; arrangement of cylinders is: staggered S ($d_\infty = (1.102\,S_1S_2/d^2-1)d$) at corners of an equilateral triangle, or aligned A ($d_\infty = (1.272\,S_1S_2/d^2-1)d$) at corners of a square;
t_W	is the mean-over-the-surface temperature of a tube wall;
t_f	is the mean governing temperature of the fluid stream;
$\Delta t = t_W - t_f$	is the temperature difference;
w	is the velocity of air or superheated steam;
q	is the thermal flux density;
$\xi = (1.82\log Re_f - 1.64)^{-2}$	is the friction coefficient for an isothermal turbulent flow in smooth pipes, according to Filonenko's formula (1954);
Nu_T	is the Nusselt number for turbulent flow in pipes.

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