IMPROVING THE HEAT TRANSFER AT VARIOUSLY PITCHED TUBE BUNDLES IN A PARALLEL STREAM

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The results are shown of an experimental study concerning the improvement of heat transfer by means of circumferential grooves in stranded tube bundles in a parallel stream of water, with the pitch ratio S/D = 1.16, 1.20, 1.34, 1.40, and 1.50.

In [1] we have presented the results of an experimental study on increasing the heat-transfer rate by means of uniformly spaced circumferential crimps around a close-packed bundle of tubes placed lengthwise in an air stream (stranded bundle with S/D = 1.2). The crimps were corrugated. The depressions thus formed on the outside surface of the tubes contributed to turbulence in the boundary layer with a resulting higher rate of heat transfer. As has been shown in [1], this method of increasing the rate of heat transfer makes it feasible to reduce the weight and the volume of tubular heat exchangers by a factor of approximately 1.5, without increasing the hydraulic resistances. The studies in [1] were limited, however, to one particular pitch of tube spacing in the bundle.

In this article, which is an extension of [1], we present the results of a study on improving the heat transfer by that method in the case of stranded tube bundles placed lengthwise in a water stream with the pitch ratio S/D = 1.16, 1.20, 1.34, 1.40, and 1.50.

The test segments (Fig. 1) consisted each of 7 tubes held in a bundle by means of two boards. The wall thickness of these grade 1Kh18N9T steel tubes was 1 mm. The boards were spaced 1400 mm apart. The frame was hexagonal for the bundles with the tube pitch S/D = 1.16, 1.20, and 1.50. The bundles with the tube pitch S/D = 1.34 and 1.40 were mounted in a round steel frame with an inside diameter 49.9 mm. In the experiment one center tube was heated electrically by passing a low-voltage alternating current directly through it. Before the improvement in the heat transfer was evaluated here, a prior study had been made of the heat transfer [2] and the hydraulic resistance with smooth tubes. The geometrical dimensions of the tested bundles as well as the test procedure and the method of data processing were given in [2]. The data on the heat transfer agree closely with the results of other studies pertaining to the turbulence region in the case of stranded buncles. Owing to the small number of tubes in a bundle, the hydraulic resistances of the tested bundles depended considerably on the frame geometry, but our data agree with the relation obtained in [3] for stranded bundles with a large number of tubes in a parallel stream, if a correction is added for the effect of the peripheral cells according to [4]. It is for this reason that this method was used for studying the improvement in the heat transfer.

Since the heat transfer was examined at the center tube of a bundle, the equivalent diameter $d_{e\infty}$ defined on the basis of an infinite number of tubes in a bundle, i.e., defined for the center cells was used as the critical dimension in the data processing. The critical temperature here was the average temperature of the boundary layer t_f, equal to one half the sum of the mean wall temperature t_w and the calorimeter-mean temperature of the stream in the center units of the bundle $\bar{t}_{cc}$. For the definition of the hydraulic resistance factor ξ we used the equivalent bundle diameter d_e as the critical dimension and the calorimeter-mean temperature of the stream through the interstitial space as the critical temperature. The presence of grooves was not taken into account in determining the equivalent diameter, the wet perimeter, the area of the intermediate tube cross section, and the mean velocity of the stream. The heat-transfer coefficient and the thermal flux were referred to the surface area of a smooth tube.

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TABLE 1. Geometrical Dimensions of the Tested Tube Bundles with Circumferential Crimps

S/D	D, mm	$d_{e\infty}$, mm	d_e, mm	d/D	t/D	$h/d_{e\infty}$	$t/d_{e\infty}$
1,5	12,1	17,9	10,62	0,93	0,83	0,0236	0,559
1,4	11,4	13,21	12,18	0,95	0,44	0,0216	0,379
				0,95	0,88	0,0216	0,758
				0,97	0,44	0,0129	0,379
1,34	11,87	11,22	12,32	0,90	0,42	0,0528	0,445
				0,93	0,42	0,037	0,445
				0,95	0,42	0,0216	0,445
				0,97	0,42	0,0158	0,445
				0,90	0,84	0,0528	0,890
				0,93	0,84	0,037	0,890
				0,97	0,84	0,0158	0,890
1,2	12,0	7,05	4,95	0,90	0,417	0,0850	0,710
				0,93	0,417	0,0595	0,710
				0,95	0,417	0,0426	0,710
				0,93	0,834	0,0595	1,42
1,16	11,4	5,51	3,34	0,95	0,88	0,0516	1,82
				0,95	0,44	0,0516	0,91

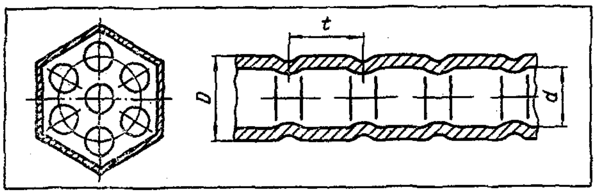

Fig. 1. Transverse cross section of the test segment and longitudinal cross section of a tube with crimps.

The maximum error in determining the heat-transfer coefficient and the hydraulic resistance is not more than ±(10-12)%.

The test parameters were varied within the following ranges: the water inlet temperature from 7.7 to 66.2°C, the mean wall temperature from 44.2 to 90.2°C, the thermal flux density from $4.7 \cdot 10^4$ to $46.5 \cdot 10^4$ W/m^2, the Reynolds number Re_f = 2000 to 60,000, and the Prandtl number Pr = 3.5 to 11.2.

The improvement in the heat transfer was examined at 17 various tube bundles. The geometrical dimensions (Fig. 1) of the tested tubes are given in Table 1. The tubes had been crimped by the batch-production corrugating. The width of a crimp was about 1 mm. The groove pitch-to-depth ratio was varied from 8.2 to 66.8.

The test data for some of the variants are shown in Fig. 2. The effect of free convection on the heat transfer becomes noticeable at low Reynolds numbers and, therefore, the test result in this range can be evaluated qualitatively only. It is evident here that the circumferential crimps become effective in close-packed bundles and that the heat-transfer rate increases less as the S/D ratio becomes higher. Within the tested range of d/D and t/D values in a turbulent stream the heat-transfer rate S/D = 1.16 and 1.20 was at most 50-60% higher than with smooth tubes accompanied by an about equally higher hydraulic resistance, while with S/D = 1.34 and 1.40 it was 25-30% higher. With S/D = 1.50 the heat-transfer rate had increased by still less.

An analysis of these tests data has shown that it is possible to generalize them into single relations for different S/D ratios by the introduction of dimensionless groove parameters $h/d_{e\infty}$. For Nu/Nu_{sm} and ξ/ξ_{sm}, moreover, we used the relations from [1] for the bundle with S/D = 1.20.

Since during laminar flow the crimps are filled with stationary liquid and affect neither the heat transfer nor the hydraulic resistance, the heat-transfer rate begins to become higher than for smooth-tube bundles only when $Re > Re_1$, and it increases as the Re number further increases. Gradually, with an increasing Re number, the turbulent flow eventually stabilizes and penetrates deeper into the narrow cells while eddies extend over the entire circumference of the crimps. Within the stable range ($Re > Re_2$), apparently, the intensity of eddies and their buildup at the upper of turbulent pulsations coincide with the development of natural turbulence in the stream at a wall as the Re number increases. Therefore, the Nu/Nu_{sm} ratio becomes approximately constant while the ξ/ξ_{sm} continues to increase. For t/D = 0.54, S/D = 1.20, and d/D = 0.90-0.97 [1] Nu/Nu_{sm} begins to increase at $Re_1 = (3.60-33.7h/d_{e\infty}) \cdot 10^4$ and stabilizes at $Re_2 = (4.70-18.85h/d_{e\infty}) \cdot 10^4$. These relations are approximately valid also for tube bundles with S/D = 1.16-1.50.

The heat transfer and the hydraulic resistance of crimped tube bundles at S/D = 1.16-1.50, $h/d_{e\infty}$ = 0-0.1, and $t/d_{e\infty}$ = 0.25-2.0 can be generalized, with a maximum deviation of ±(10-12)%, as follows:

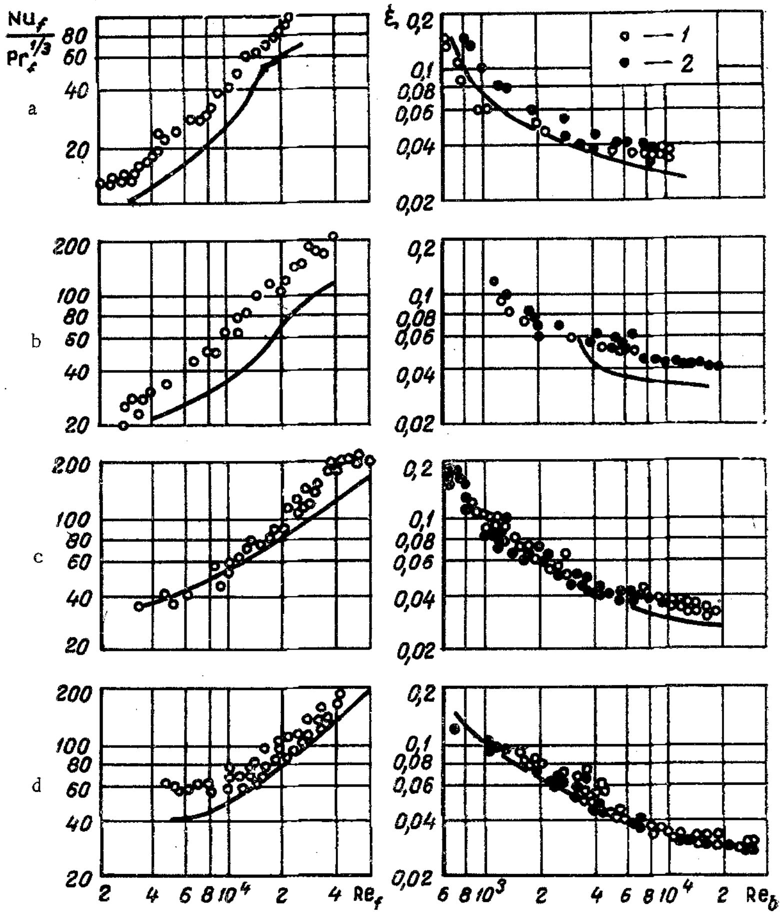

Fig. 2. Heat-transfer coefficients and hydraulic resistance factors of crimped tube bundles with various tube pitch ratios (solid lines apply to smooth tubes): 1) isothermal flow; 2) heating of water; a) S/D = 1.16; d/D = 0.95; t/D = 0.44; b) S /D = 1.2; d/D = 0.93; t/D = 0.417; c) S/D = 1.34; d/D = 0.90; t/D = 0.42; d) S/D = 1.4; d/D = 0.95; t/D = 0.44.

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{sm}}}=1+0.6\,\frac{\lg \mathrm{Re}-\lg \mathrm{Re}_1}{\lg \mathrm{Re}_2-\lg \mathrm{Re}_1}\left[1-\exp\left(-35.8\,\frac{h}{d_{e\infty}}\right)\right]\left(1-0.35\,\frac{t}{d_{e\infty}}\right)$$

for $\mathrm{Re}_1<\mathrm{Re}<\mathrm{Re}_2$;

$$\frac{\mathrm{Nu}}{\mathrm{Nu}_{\mathrm{sm}}}=1+0.6\left[1-\exp\left(-35.8\,\frac{h}{d_{e\infty}}\right)\right]\left(1-0.35\,\frac{t}{d_{e\infty}}\right)$$

for $\mathrm{Re}_2<\mathrm{Re}<10^5$ (Fig. 3);

$$\frac{\xi}{\xi_{\mathrm{sm}}}=1+\left\{7.55\,\frac{h}{d_{e\infty}}(\lg \mathrm{Re}-3.5)-0.035\sin\left[\left(1-22.44\,\frac{h}{d_{e\infty}}\right)\pi\right]\right\}\left(1.40-0.488\,\frac{t}{d_{e\infty}}\right)$$

for $\mathrm{Re}=3.1\cdot 10^3-2\cdot 10^4$;

$$\frac{\xi}{\xi_{\mathrm{sm}}}=1+\left\{3.21\,\frac{h}{d_{e\infty}}(\lg \mathrm{Re}-2.27)+0.09(\lg \mathrm{Re}-4.3)\sin\left[\left(1-22.44\,\frac{h}{d_{e\infty}}\right)\pi\right]\right\}\left(1.40-0{,}488\,\frac{t}{d_{e\infty}}\right)$$

for $\mathrm{Re}=2\cdot 10^4-10^5$.

In (1) and (2) $\mathrm{Nu}_{\mathrm{sm}}$ has been defined according to [2]; in (3) and (4) ξ_{sm} has been defined according to [4] for a large number of tubes in the bundle. For the derivation of (1)-(4) we also used the data in [5, 6] for channels of annular cross section with crimps in the inner tube. It can be seen in Fig. 3 that $\mathrm{Nu}/\mathrm{Nu}_{\mathrm{sm}}$ increases at a lower rate with increasing groove depth $h/d_{e\infty}$. Since the hydraulic resistance increases

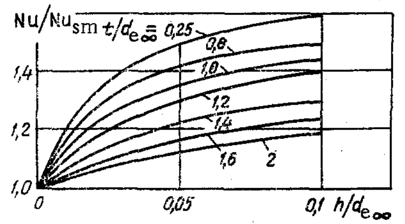

Fig. 3

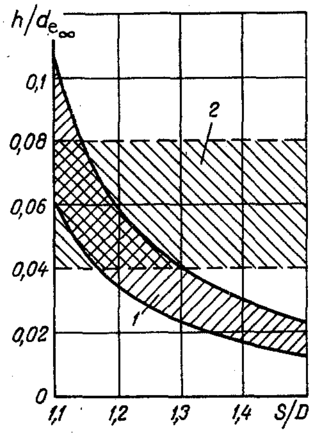

Fig. 4

Fig. 3. Effect of crimping depth $h/d_{e\infty}$ and crimping pitch $t/d_{e\infty}$ on the improvement of the heat transfer at tube bundles in a parallel stream at Re > Re_2.

Fig. 4. Ranges of the optimum depth of annular diaphragms inside tube bundles in a parallel stream, as a function of S/D: optimum range for improving the heat transfer inside (1) and outside (2) tubes of a bundle in a parallel stream.

with increasing $h/d_{e\infty}$ throughout the test range of $h/d_{e\infty}$ [1], the optimum crimp depth is within $h/d_{e\infty} = 0.04$-0.08, which ensures an appreciable increase in the heat-transfer rate at a moderately higher hydraulic resistance.

In optimizing the crimp parameters it is necessary to consider the increase in the rate of heat transfer inside the tubes as well, where this is achieved by means of annular diaphragms and where, according to [8], the optimum ratio of diaphragm width to inside tube diameter is 0.02-0.035. For the outside of a tube, moreover, this range is h/D = 0.02-0.035 or $h/d_{e\infty} = (0.02\text{-}0.035)D/d_{e\infty} = (0.02\text{-}0.035)/[1.102(S/D)^2 - 1]$. This range of $h/d_{e\infty}$ values is shown in Fig. 4 for various S/D ratios and is compared here with the range $h/d_{e\infty} = 0.04$-0.08 optimum for increasing the rate of heat transfer on the outside of tubes. These ranges overlap for S/D = 1.1-1.3, where this method of increasing the heat-transfer rate is most applicable. In bundles with S/D > 1.3 the heat transfer in the interstitial space is improved most by means of diaphragms inside the tube whose widths are greater than optimum and, consequently, at the expense of considerable pressure losses inside the tube. The maximum improvement of the heat transfer inside a tube yields an insignificant improvement outside the tube, as has also been revealed in these experiments. For this reason, transverse finning is more effective in relatively loose-packed bundles (S/D = 1.3-1.5) [7].

NOTATION

D	is the external diameter of tubes;
d	is the diameter of circumferential groove;
$d_e = 4F/U$	is the equivalent diameter of a bundle, based on the total wet perimeter;
$d_{e\infty} = [1.102(S/D)^2 - 1]D$	is the equivalent diameter of center cells in a bundle;
F	is the effective section area of a bundle;
h	is the depth of groove;
S	is the tube pitch in a bundle;
t	is the annular groove pitch;
U	is the wet perimeter;
ξ	is the hydraulic resistance factor.

Subscript

sm refers to smooth.

LITERATURE CITED

1. G. A. Dreitser and É. K. Kalinin, Inzh.-Fiz. Zh., 15, No. 3 (1968).
2. É. K. Kalinin, G. A. Dreitser, and A. K. Kozlov, Inzh.-Fiz. Zh., 16, No. 1 (1969).
3. A. I. Mikhailov, É. K. Kalinin, and G. A. Dreitser, Inzh.-Fiz. Zh., 7, No. 11 (1964).
4. M. Kh. Ibragimov, I. A. Isupov, L. L. Kobzar', and V. I. Subbotin, Atomnaya Énergiya, 23, No. 4 (1967).
5. H. Brauer, Atomkernenergie, 4, 152-161; 5, 207-211 (1961).
6. C. A. Kemeny and J. A. Cyphers, Trans. ASME, 83, Ser. C, No. 2 (1961).
7. É. K. Kalinin, G. A. Dreitser, and A. K. Kozlov, Izv. Akad. Nauk BSSR, Ser. Fiz.-Tekh. Nauk, No. 2 (1966).
8. É. K. Kalinin, G. A. Dreitser, and S. A. Yarkho, Papers Japan. SME Semi-Internat. Symp. September 4-8, 1967; Tokyo; Heat Transfer, 1 (1967).