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HYDRAULIC RESISTANCE AND HEAT TRANSFER IN LONGITUDINAL FLOW PAST STAGGERED TUBE BUNDLES WITH WALLS OF DIVERGENT-CONVERGENT PROFILE

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The results of an experimental investigation into the effect of the characteristic dimensions of staggered convergent-divergent tube bundles on hydraulic resistance and heat transfer in a longitudinal flow are presented.

As before, the enhancement of heat transfer in different-purpose tubular heat exchangers remains an urgent applied problem in the theory of heat and mass transfer [1-5].

In the case of the single-phase heat carrier transverse flow past tube bundles, the problem of heat transfer enhancement is usually unambiguously solved by applying finned tubes, because the use of different types of turbulizers, just as the change in the shape of the walls, does not give desirable positive results as compared with smooth-wall tube bundles [3-5].

By the present time, the most developed and verified method for the enhancement of heat transfers is during the turbulent flow of various heat carriers in tubes with different types of turbulizers or with the shape and geometric dimensions of the cross section varying along the flow path. The relevant literature contains voluminous information relating to the thermo-hydraulic characteristics of heat-exchanging tubes with annular knurling [1], helical tubes [2, 6], tubes with a divergent-convergent profile [7-10], etc. This has made it possible to start a wide practical incorporation of tubes with heat transfer amplifiers into real heat exchangers with a substantial improvement in their characteristics.

As a rule, the application of turbulization mechanisms to the internal walls of tubes leads inevitably to the deformation of the outer walls. The question then naturally arises as to the estimation of the extent of heat transfer intensification on these walls in the case of longitudinal flow past tube bundles. This attempt was made in a comprehensive study [2] where the characteristics of the ovaly-coiled tubes that ensured heat transfer enhancement both in tube and intertube space were investigated. However, the gain in energy with the use of ovaly-coiled tubes is relatively small, while they are very difficult to manufacture and have insufficiently reliable strength characteristics.

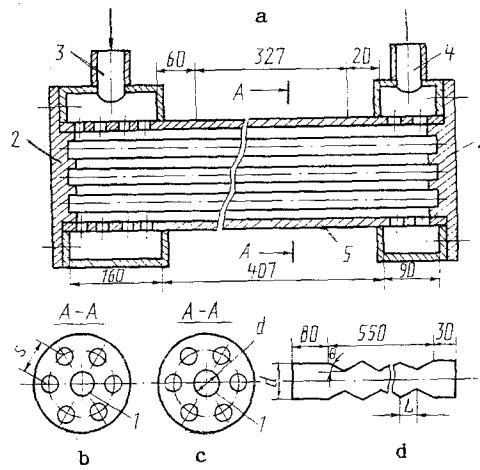


Fig. 1. Schematic of a test section with a staggered tube bundle (a), without a displacement of tubes (b), with a displacement of tubes (c), divergent-convergent tube (d); (1), heated tube; (2), tube plates; (3), (4), inlet and outlet branch pipes; and (5), casing.

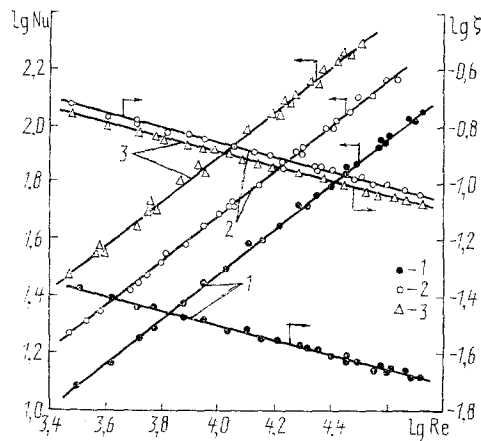


Fig. 2. Heat transfer and hydraulic resistance of longitudinal bundles of smooth-wall and corrugated tubes: (1) a smooth-wall tube bundle; (2) a symmetric bundle of corrugated tubes, $s/d = 1.114$; (3) a bundle of corrugated tubes with a mutual displacement by one diffuser or converger module, $s/d = 1.114$.

In our opinion, based on the available information, the use of divergent-convergent tubes arranged in longitudinal tube bundles seems to be an alternative variant to ovaly-coiled ones [2]. For the first time this problem was considered in [4] in application to the air heaters of boilers. Therefore, the task has been set to investigate experimentally the hydraulic resistance and heat transfer in longitudinal flow past staggered bundles of tubes of divergent-convergent profile in comparison with other familiar variants. In this paper the experimental data are presented on the characteristics of twelve staggered smooth-wall and divergent-convergent tube bundles with relative spacings $s/d = 1.114, 1.22, 1.432, \text{ and } 1.64$ immersed in a longitudinal flow.

The investigations were carried out on a rig the schematic diagram of which is similar to that given in [11] and which is depicted in Fig. 1. The method of local simulation was used, which is also described in detail in [11] in application to longitudinal flow past tube bundles.

TABLE 1. Values of the Coefficients A and C in Eqs. (1) and (2)

Coefficient	Tube bundles							
	without mutual displacement				with mutual displacement			
	relative step s/d							
	1,114	1,22	1,432	1,64	1,114	1,22	1,432	1,64
A	0,0304	0,0407	0,0546	0,0374	0,0391	0,0340	0,0392	0,0442
C	1,39	0,76	0,60	0,522	1,30	0,76	0,54	0,57

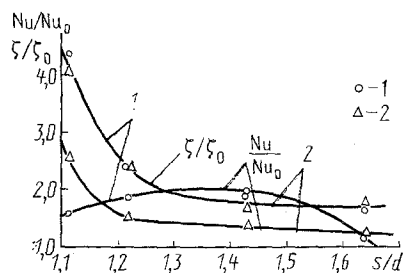


Fig. 3

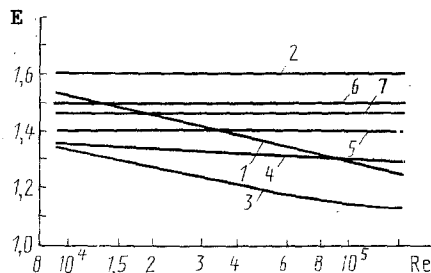


Fig. 4

Fig. 3. Comparative characteristics of heat transfer and hydraulic resistance of a staggered bundle of divergent-convergent tubes in axial flow: (1) symmetric and (2) nonsymmetric arrangement of tubes.

Fig. 4. Comparative thermohydrodynamic efficiency of heat exchanging bundles and tubes with walls of different profiles. Turbulent flow in tubes: (1) tubes with annular knurling [1]; (2) divergent-convergent tubes and channels [7]; (3) shaped coiled tubes [6]; and (4) ovaly-coiled tubes with $s/d = 6.2$ [2]. Turbulent axial flow past tube bundles: (5) ovaly-coiled tubes with $s/d = 4.15$ [2]; (6) a symmetric bundle of divergent-convergent tubes with $s/d = 1.432$; and (7) a bundle of mutually-displaced tubes with $s/d = 1.114$.

The experimental sections consisted of seven tubes fixed as a bundle in a cylindrical casing with the aid of two tube plates. The tubes were made of copper, and they had the maximum diameter of $d = 21.4$ mm and a wall thickness of 1.5 mm. The distance between the tube plates in all of the cases was 657 mm.

When a bundle was made up of corrugated tubes (Fig. 1b, c), the latter represented the surfaces in the form of alternating smooth and wavy sections with dimensions indicated in Fig. 1d. In this case, the same geometry of a tube, consisting of 20 alternating "diffuser-converger" cycles, with the relative length of the diffuser or converger module amounting to $L/d = 0.643$ and with the divergence half-angle to $\theta = 10^\circ$, was used for all of the bundles. The edges of the transitions of adjacent modules were rounded. According to [9], such tubes have almost the highest thermohydraulic efficiency among the channels with internal flows.

Two variants of longitudinal arrangement of divergent-convergent tubes were tested: with their symmetric arrangement (Fig. 1b) and with mutual longitudinal displacement of neighboring tubes by one diffuser or converger module (Fig. 1c). Comparison tests of divergent-convergent and corresponding smooth-wall tube bundles were conducted and the relationship between the heat transfer enhancement and the increase in the hydraulic resistance was estimated.

The heat carrier (air) was supplied to and drawn from the bundle transversely. For creating a uniform heat carrier flow, the test section is provided with inlet and outlet annular collectors connected by a system of holes with an intertube space. According to [11], the dimensions of the collectors ensured the flow stabilization and eliminated the influence of end-effects on the flow past the main part of the bundle of the heat exchanging tubes.

In the experiments, one central tube which served as a calorimeter was heated. For this purpose, an electric heater was placed inside the tube, while the space between the tube wall and the heater was filled with a turbine oil in order to in-

crease the uniformity of heating along the tube length and around the perimeter of the walls. The mean heat transfer and resistance coefficients were measured over the section with a stabilized longitudinal flow. Its length was 327 mm and its beginning was 60 mm from the inlet collector. In the bundle of corrugated tubes the stabilization was reached much earlier than in the smooth-wall tube bundle.

The air discharge was measured by metering disks. The inlet and outlet air temperatures in the test section were measured with standard mercury-in-glass thermometers. The outer wall temperature of the tube was measured by ten copper-Constantan thermocouples mounted flush with the surface at different corresponding points of different modules and then was determined as the mean arithmetic one. The error in the determination of the heat transfer and resistance coefficients did not exceed 10% at a confidence probability of 0.98.

For the rest, as regards the design of the rig and the technique of processing experimental data, all the necessary requirements were adhered to [11]. Investigations were carried out at $Re = (0.35-5.0)10^4$ and $M \leq 0.25$, which corresponded to a developed turbulent flow regime of the heat carrier. In addition, account was taken of the fact that the bundles of corrugated tubes were excellent turbulence stimulators.

The processing of experimental data on both resistance and heat transfer in a power-law form, depending on the Reynolds number for the two variants of the tube arrays, has shown that the distribution curves have the same slope, closely coinciding with the case of flow past smooth-wall bundles [11]. In view of this, all of the obtained experimental data were approximated by the power-law relations: for heat transfer

$$Nu = ARe^{0.8} \tag{1}$$

and for hydraulic resistance

$$\zeta = CRe^{-0.25}, \tag{2}$$

where the coefficients A and C depend on the relative spacing between the tubes in a bundle. The values of these coefficients in Eqs. (1) and (2) for all the bundles are presented in Table 1. It should be noted that the resistance of smooth-wall tube bundles obeys the rules given by the Blasius formula, whereas their heat transfer behavior coincides with the data of [11].

As an example, Fig. 2 presents the changes in the Nusselt number and in the resistance coefficient with the Reynolds number, typical for all the surfaces, in smooth-wall and corrugated tube bundles with the relative spacing $s/d = 1.114$. It is seen that both the heat transfer rate and the resistance of corrugated bundles greatly exceed the data for the corresponding smooth-wall tube bundle. Thus, over the entire range of the Reynolds numbers, the heat transfer of a symmetric bundle exceeds that of the smooth-wall tube bundle by a factor of 1.6, while that with the displacement of tubes is higher by a factor of 2.6. A corresponding increase in the resistance was equal to 4.4 and 4.1, respectively.

To evaluate the degree of heat transfer enhancement and the associated increase in the hydraulic resistance by the technique of [1, 4], the relative coordinates Nu/Nu_0 and ζ/ζ_0 were used as functions of the dimensionless spacing between the tubes in a bundle, where Nu_0 and ζ_0 are the Nusselt number and the resistance coefficient of the corresponding smooth-wall bundles. In Fig. 3 the experimental data on heat transfer and resistance, processed in the proposed coordinates, are presented for all the bundles. It should be noted at the outset that since, according to Eqs. (1) and (2), the exponents for the Reynolds number do not change for all the bundles, the dimensionless quantities Nu/Nu_0 and ζ/ζ_0 will depend only on the relative spacing and will not depend on the Re number.

According to Fig. 3, in the case of the nonsymmetric arrangement of tubes, both the relative heat transfer coefficients and the resistance coefficients decrease with an increase in the relative spacing between the tubes in an array s/d . In bundles without a mutual displacement of tubes, the resistance coefficients also decrease with an increase in s/d , and the trends in this variation, within the error of the experimental investigation, coincide with those observed in the case of a nonsymmetric arrangement of tubes. However, the data on the heat transfer of these bundles display other trends. For the relative spacings varying from 1.10 to 1.40, the heat transfer rate increases, whereas for $s/d = 1.40-1.70$ it, conversely, decreases. The most efficient bundle, consisting of symmetrically arranged tubes, is a bundle with the spacing $s/d = 1.432$. To the version with a displacement of tubes there corresponds an optimum bundle with a smaller spacing equal to 1.114.

In the practice of designing of different-purpose heat exchangers, of greatest interest are the channels of divergent-convergent type, tubes with annular steps, spirally- and ovaly-coiled tubes, and curved channels. The comparative thermohydrodynamic efficiency of these heat transfer surfaces, and also of the bundles of ovaly coiled and divergent-convergent tubes in an axial flow, carried out according to the technique of [4, 7] is shown in Fig. 4, where, according to [4], the thermohydrodynamic efficiency parameter E is equal to

$$E = (Nu/Nu_0) / (\zeta/\zeta_0)^{0.4}.$$

The data for the flow in tubes with heat transfer surface of optimal versions have been fully borrowed from [10]; for the external flow past ovaly-coiled tubes the data were taken from [2]. As optimum bundles of the two versions, use was made of the arrangement of divergent-convergent tubes without a mutual displacement with the spacing $s/d = 1.432$ and with a mutual displacement with the spacing $s/d = 1.114$. As the analysis of the data of Fig. 2 showed, these arrays were optimal among all the studied bundles.

It follows from Fig. 4 that divergent-convergent channels with internal flow (curve 2, $E = 1.6$), bundles of such tubes without a displacement (curve 6, $s/d = 1.432$, $E = 1.5$), and bundles with a mutual displacement (curve 7, $s/d = 1.114$, $E = 1.46$) are most efficient. Tubes with annular knurling may compete with the channels of the "diffuser-converger" type only at small Reynolds numbers (curve 1, $E = 1.3-1.5$), but at high Re numbers they are much inferior. Other variants of heat transfer surfaces are obviously nonpromising and can be used only if there are considerable technico-economic advantages in the technology of their fabrication.

Thus, from the analysis of [10] and of the experimental data presented it follows that among the considered heat transfer surfaces with turbulent flow in tube and intertube spaces the most efficient are divergent-convergent tubes and their bundles.

NOTATION

d , maximum diameter of a tube; d_{eq} , equivalent diameter based on wetted perimeter, $4F/U$; s , spacing between tubes in a bundle; L , length of diffuser or converger module; E , parameter of thermohydrodynamic efficiency, $(Nu/Nu_0)/(\zeta/\zeta_0)^{0.4}$; θ , divergence half-angle for the diffuser; Re , Reynolds number based on equivalent diameter, wd_{eq}/ν ; Nu , Nusselt number, $\bar{\alpha}d_{eq}/\lambda$.

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