DRAG IN A HEAT EXCHANGER WITH A TWISTED FLOW

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A study was made of the effect of different parameters on the drag coefficient in longitudinal flow about bundles of twisted tubes of different profiles.

Heat-exchanging apparatus with longitudinal flow about bundles of twisted tubes have been examined in several works [1-4]. In [1-3], criterional relations were obtained for calculating the drag and heat-transfer coefficients in the space between the tubes and inside twisted tubes of oval profile, and the efficiency of such heat exchangers was demonstrated. In [4], results were presented from a study of the temperature and velocity fields in the intertube space of a heat exchanger with longitudinal flow about bundles of twisted tubes of oval and three-lobed profiles. The form of these profiles was described in detail in [5]. It was shown in [4] that if, along with the Reynolds number

$$\mathrm{Re}_f = \rho u_{av} d_e/\mu \tag{1}$$

we took as the determining criterion a criterion characterizing the twisting of the flow in the bundle and having either the form

$$\mathrm{Fr}_c = u_{av}^2/g_c d_e, \tag{2}$$

where

$$g_c = 2u_\tau/d; \tag{3}$$

$$u_\tau = \pi d u_{av}/s, \tag{4}$$

or the form

$$\mathrm{Fr}_m = s^2/dd_e, \tag{5}$$

then the velocity field in the boundary layer can be generalized with a power law for bundles of twisted tubes of different profile in the absence of geometric similitude, having taken the local thickness of the boundary layer as the characteristic dimension. If we consider the existence of a connection between the velocity field and the drag coefficient [6], then we can assume that the relations obtained for calculating this coefficient in bundles of twisted tubes of oval profile will also be valid for bundles of tubes of three-lobed profile. To check this hypothesis and determine the effect of the number $M = u/a$ and the temperature factor on the drag in bundles of twisted tubes of different profile, as well as to evaluate the length of the section of hydrodynamic stabilization of the flow, we conducted a study of the adiabatic and nonisothermal flow of air.

Drag was studied by a generally accepted method on the experimental units described in [1, 4]. Air flow rate was measured with a critical ring calibrated on a gas holder. The pressure drops on the control sections were measured with liquid differential monometers and DDFM induction-type transducers, while the pressure was measured with standard manometers and capacitance-type transducers. The tests embraced the following range of parameter values: $s/d = 6.5$–35; $\mathrm{Fr}_m = 64$–2000; $\mathrm{Re}_f = 3 \cdot 10^3$–$5 \cdot 10^4$; $T_w/T_f = 1.0$–1.42; $T_f = 287$–467°K; $T_w \leq 621$°K; $M = 0.03$–0.27; $N = 2.3$. The limiting error in the determination of the drag coefficient was 9% using a functional relation of the form

$$\xi = \xi(\mathrm{Re}, \mathrm{Fr}_m, M, x_i/d_e, T_w/T_f). \tag{6}$$

The coefficient ξ was determined from the momentum equation for a liquid flow with a variable density, with allowance for the fact that a bundle of twisted tubes is a channel with a lengthwise-constant cross-sectional area, for which $\rho u_{av} = \mathrm{const}(x)$:

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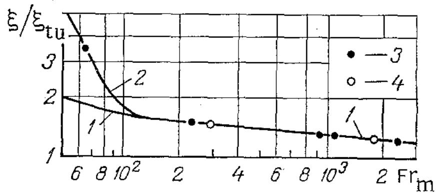

Fig. 1

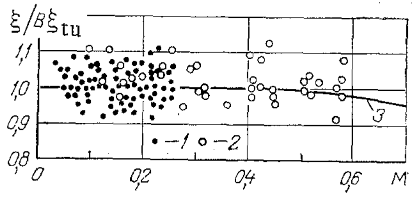

Fig. 2

Fig. 1. Dependence of drag coefficient of bundle of twisted tubes, referred to the friction coefficient in a circular tube, on a criterion characterizing the effect of twisting of the flow: 1) Eq. (13); 2) Eq. (16); 3) test data [1, 2] for tubes of oval profile; 4) test data for three-lobed tubes.

Fig. 2. Effect of the number M on the drag coefficient of a bundle of twisted tubes: 1) test data for bundles of twisted tubes with the numbers Fr_m = 290-2000; 2) test data for a circular tube; 3) equation from [7].

$$-dP = \xi \frac{\rho u_{av}^2}{2}\frac{dx}{d_e} + \rho u_{av} du_{av}. \tag{7}$$

For a section of finite length l

$$\Delta P = \frac{\rho_1 u_{av.1}}{2d_e}\int_0^l \xi u_{av} dx + \rho_1 u_{av\,1}(u_{av\,2} - u_{av\,1}). \tag{8}$$

For the section of stabilized flow, we may take ξ = const(x). Then, after making the substitutions $\int_0^l u_{av}dx = (u_{av.1} + u_{av.2})l_t/2$ and $\rho u_{av} = G/F_f$, we obtain the expression

$$\xi = \frac{\Delta P - (G/F_f)^2[(1/\rho_2) - (1/\rho_1)]}{(l_t/d_e)(G/F_f)^2[1/(2\rho_{av})]}, \tag{9}$$

where $\rho_{av} = P_{av}/RT_{av}$, $T_{av} = (T_1 + T_2)/2$, $P_{av} = (P_1 + P_2)/2$. Since the air density and flow velocity in the tests deviated relatively little from a linear relationship with respect to the length l_t, we determined the coefficient ξ from Eq. (9).

In analyzing the test results for an adiabatic air flow, we determined the parameters of the flow along its length using the method of successive approximation in conjunction with tables of gasdynamic functions. The resulting calculated velocity

$$\lambda = u/a_{cr}, \tag{10}$$

where

$$a_{cr} = 18{,}3\sqrt{T^*}, \tag{11}$$

was used to determine the thermodynamic temperature of the flow at a given cross section of the bundle $T = \tau(T^*)$, the air density $\rho = P/RT$, and the velocity $u = G/\rho F_f$, where $\tau = \tau(\lambda)$. The stagnation temperature

$$T^* = T + u^2/2c_p \tag{12}$$

was measured in a receiver located in front of the twisted tubes, where $u \approx 0$.

The results of the study of coefficient ξ for bundles of tubes of three-lobed profile are shown in Fig. 1. They are generalized by the relation [2]

$$\xi = B\xi_{tu}, \tag{13}$$

where

$$\xi_{tu} = 0{,}3164\,\mathrm{Re}_f^{-0{,}25}, \tag{14}$$

$$B = 1 + 3{,}6\,\mathrm{Fr}_m^{-0{,}357}. \tag{15}$$

Also shown for comparison is test data on the coefficient ξ in bundles of twisted tubes of oval profile [1, 2]. It can be seen from Fig. 1 that the empirical values of ξ for tubes of

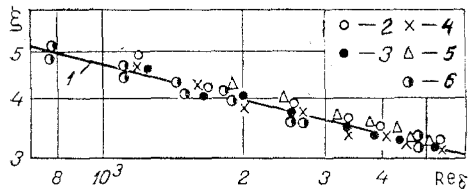

Fig. 3. Effect of the length of the section of hydrodynamic stabilization of the flow on the drag coefficient of a bundle of twisted tubes with the number $Fr_m = 924$: 1) Eq. (17); 2-6) test data for $x_i/d_e = 11.85$, $l_t/d_e = 47.5$, $x_i/d_e = 36.2$, $l_t/d_e = 23.1$; $x_i/d_e = 3.75$, $l_t/d_e = 55.5$; $x_i/d_e = 11.85$; $l_t/d_e = 8.12$; $x_i/d_e = 3.75$, $l_t/d_e = 86.7$, respectively.

three-lobed profile agree with Eq. (13), which is valid at numbers $Fr_m \geqslant 100$. At $Fr_m < 100$ there is a sharp increase in ξ in the bundle due to separation of the flow from the spiral surfaces of the tubes. Here, the coefficient is described by the formula

$$\xi = 0.3164\, Re_f^{-0.25} (1 + 3.1 \cdot 10^6 Fr_m^{-3.58}). \tag{16}$$

As already noted, the fact that the number of lobes on the twisted tubes will not affect ξ can be predicted on the basis of results of the study in [4]. To illustrate this more clearly, we obtain a power law of 1/7 for the velocity distribution in the boundary layer of a bundle of twisted tubes from the drag law (13), which can be reduced to the form

$$\xi = 0.266 Re_{\delta f}^{-0.25}, \tag{17}$$

if we introduce the quantity δ, determined as follows, as an integral geometric characteristic of the bundle

$$\delta = 0.5\,(1 + 3.6\, Fr_m^{-0.357})^{-4} d_e, \tag{18}$$

and determine the Reynolds number from δ:

$$Re_{\delta f} = \rho u_{av} \delta/\mu. \tag{19}$$

Then, regarding the equilibrium of an element of liquid being acted upon by shear stresses and pressure forces:

$$|dP/dx| = 2\tau_0 (1 + 3.6\, Fr_m^{-0.357})^{-4}/\delta, \tag{20}$$

determining the coefficient ξ by the formula

$$|dP/dx| = \xi (\rho u_{av}^2/4)\, [(1 + 3.6\, Fr_m^{-0.357})^{-4}/\delta] \tag{21}$$

and introducing the dynamic velocity $v_* = \sqrt{\tau_0/\rho}$, we can obtain the relation in [6]

$$u_{av}/v_* = 6.99\, (\rho v_* \delta/\mu)^{1/7}, \tag{22}$$

if we equate Eqs. (20) and (21) and perform simple transformations.

The velocities $\bar{u}$ and u_{av} in the bundle of twisted tubes are related as follows

$$u_{av} = \bar{u}\, [1 - (4\delta^*/d_e)] \tag{23}$$

or are connected by the following expression in the case of numbers $Fr_m = 240$–380, for which we can expect optimum intensification of transfer processes:

$$u_{av} \approx 0.965 \bar{u}. \tag{24}$$

Then, substituting (24) into (22), we obtain

$$\bar{u}/v_* = 7.25\, (\rho v_* \delta/\mu)^{1/7} \tag{25}$$

or

$$u/v_* = 7.25\, (\rho v_* y/\mu)^{1/7}. \tag{26}$$

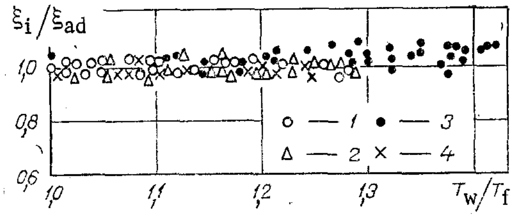

Fig. 4. Dependence of the drag coefficient of a bundle of twisted tubes on the temperature factor: 1-4) test data for the numbers Fr_m = 64, 232, 924, 1050, respectively.

These expressions differ from the analogous expressions for a circular tube by a constant multiplier. This multiplier is equal to 8.74 [6] for the tube, since $u_{av} = 0.8\bar{u}$ for it. The 1/7 power law for bundles of twisted tubes presented in [4]

$$u/\bar{u} = (y/\delta)^{1/7} \tag{27}$$

is obtained after Eq. (26) is divided by (25).

Figure 2 shows the results of study of the effect of the number M on the coefficient ξ for bundles of twisted tubes of three-lobed and oval profiles. Also shown are results of study of the effect of M on the friction coefficient in smooth circular tubes. We obtained these results at numbers Re = $2.1 \cdot 10^3$–$8.2 \cdot 10^4$ and λ_{av} = 0.09-0.63. This data on ξ was obtained by a single method, which facilitates comparison of the results. It follows from Fig. 2 that M does not affect ξ in the adiabatic flow of a compressible gas in bundles of twisted tubes and in a circular tube in the investigated range of M. This conclusion agrees for the most part with the data from studies of the flow of a compressible gas in tubes, such as [7, 8]. At the same time, when M > 0.5 [7], there is some reduction in ξ in the tubes (Fig. 2). This is confirmed by the results in [8]. However, the ranges of Re and M which we investigated, where M has no effect on ξ, are of the greatest interest for heat exchangers with a twisted flow.

The effect of the length of the section of hydrodynamic stabilization of a flow in a bundle of twisted tubes on the coefficient ξ was studied for the case of axisymmetric inlet of the flow from a large volume by calculating ξ from Eq. (9) for different control-section lengths l_t with x_i/d_e = 3.75, 11.85, and 36.2. The study results are shown in Fig. 3. It follows from the latter than for $x_i/d_e \geqslant 3.75$ or $x_i/2\delta \geqslant 12$, the mean value of ξ is nearly constant along the control section for all of the variants examined. Since ξ may be underestimated by at least 5% in a bundle of twisted tubes due to different conditions of flow about the static pressure transducers [9], then the value x_i/d_e = 3.75 should be regarded as a minimum estimate of the length of the section of hydrodynamic stabilization of the flow. The length of the thermal initial section for bundles of twisted tubes was determined in [10] and is equal to $(l_i/d_e)_h$ = 14, in contrast to $(l_i/d_e)_h$ = 50 for a circular tube [11]. The length of the section of hydrodynamic stabilization of the flow in a tube $(l_i/d_e)_y$ = 30 [11], i.e., it is roughly 70% less than the length $(l_i/d_e)_h$. Then, by analogy, it can be assumed that $(l_i/d_e)_y \approx 8$ for bundles of twisted tubes. The relative shortness of the sections of hydrodynamic and thermal flow stabilization in bundles of twisted tubes can be explained by the equilizing effect of the twisting of the flow, leading to significant expansion of the core of the flow along the entire bundle and to a thin boundary layer on the twisted tubes.

It is apparent from the results of the study of the effect of the temperature factor on ξ that the following relation holds for all of the bundles of twisted tubes investigated in the range of T_w/T_f covered by the experiment

$$\xi_i/\xi_{ad} = \mathrm{const}\,(T_w/T_f), \tag{28}$$

which was noted in [1-3]. The coefficient ξ deviates within the limits of the experimental error only for the bundle with Fr_m = 924 in the range T_w/T_f = 1.2-1.42. This allows us to conclude that the variability of the physical properties of the liquid through the thickness of the boundary layer in a bundle of twisted tubes has almost no effect in the range of parameters embraced by the experiments. It should also be noted that, in pipe flow, the variability of the physical properties of the liquid through the boundary-layer thickness also affects friction to a significantly lesser degree than it does heat thransfer [12].

NOTATION

Re, Reynolds number; Fr_c, Fr_m, criteria characterizing features of the flow in a bundle of twisted tubes; ρ, density; u, velocity; d_e, equivalent diameter; d, maximum dimension of tube profile; s, pitch of twisting of tube profile; T, temperature; μ, viscosity, a, speed of sound; N, number of tube lobes; ξ, drag coefficient; P, pressure; ΔP, pressure drop; l_t, length of control section; x, longitudinal coordinate; G, mass flow rate of air; F_f, cross-sectional area of bundle; c_p, specific heat; δ, thickness of boundary layer; y, coordinate reckoned from tube wall; τ_0, shear stress on wall. Indices: f, flow; w, wall; c, centrifugal; τ, tangential; av, mass-average; i, initial; t, control; 1, inlet to section; 2, outlet from section; tu, tubes.

LITERATURE CITED

1. B. V. Dzyubenko and G. A. Dreitser, "Study of heat transfer and drag in a heat exchanger with a twisted flow," Izv. Akad. Nauk SSSR, Energ. Transp., No. 5, 163-171 (1979).
2. B. V. Dzyubenko and V. M. Ievlev, "Heat transfer and drag in the intertube space of a heat exchanger with a twisted flow," Izv. Akad. Nauk SSSR, Energ. Transp., No. 5, 117-125 (1980).
3. V. M. Ievlev, Yu. I. Danilov, B. V. Dzyubenko, et al., "Heat transfer and hydrodynamics of twisted flows in channels of complex shape," Heat and Mass Transfer — VI, Vol. 1, Pt. 1, ITMO AN BSSR (Institute of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR), Minsk (1980), pp. 88-99.
4. B. V. Dzyubenko, "Study of velocity and temperature fields in the intertube space of a heat exchanger with a swirled flow," in: International Handbook of Scientific Researches: Current Problems of Hydrodynamics and Heat Transfer in Power Plane Elements and Cryogenic Technology, Vol. 8, VZMI (All-Union Correspondence Institute of Mechanical Engineering), Moscow (1979), pp. 93-104.
5. Yu. V. Vilemas, B. A. Chesna, and V. Yu. Survila, Heat Transfer in Annular Gas-Cooled Channels [in Russian], Makslas, Vilnius (1977).
6. G. Schlichting, Boundary Layer Theory, McGraw-Hill (1966).
7. B. S. Petukhov, A. S. Sukomel, and V. S. Protopopov, "Study of drag and the wall-temperature recovery coefficient in the motion of gas in a circular pipe at a high subcritical velocity," Teploenergetika, No. 3, 31-37 (1957).
8. A. F. Gandel'sman, A. A. Gukhman, N. V. Il'yukhin, and L. N. Naurin, "Study of the drag coefficient in flow with near-sonic velocity," Zh. Tekh. Fiz., 24, No. 12, 2234-2249 (1954).
9. B. V. Dzyubenko, A. V. Sakalauskas, and Yu. V. Vilemas, "Distributions of velocity and static pressure in a heat exchanger with a twisted flow," Izv. Akad. Nauk SSSR, Energ. Transp., No. 4, 112-118 (1981).
10. B. V. Dzyubenko, "Heat transfer in the initial section in a heat exchanger with a twisted flow," Inzh.-Fiz. Zh., 42, No. 2, 230-235 (1982).
11. S. S. Kutateladze and V. M. Borishanskii, Heat-Transfer Handbook [in Russian], Gosénergoizdat, Moscow (1959).
12. D. M. McEligot, S. B. Smith, and C. A. Bankston, "Quasideveloped turbulent pipe flow with heat transfer," Paper ASME, WHT-8 (1970).