NOTATION

U_m, velocity of the mixture, m/sec; d, inside diameter of the tube, m; α, angle of inclination of the tube to the horizontal, degrees; g, acceleration due to gravity, m/sec^2; μ, absolute viscosity, N·sec/m^2; β_2, discharge gas content; $\beta_1 = 1 - \beta_2$, discharge liquid content; φ_2, true volumetric gas content; $\varphi_1 = 1 - \varphi_2$, true volumetric liquid content; λ, resistance coefficient; Fr_m, Froude number of the mixture; Fr_1, Froude number calculated from the corrected liquid velocity; Fr_{pr}, Froude number calculated from the velocity of no-head flow of the liquid over the entire cross section of the given tube; Fr*, Froude number corresponding to the transition from piston to stratified flow; $\chi = Fr_1/Fr_{pr}$, parameter; Re, Reynolds number.

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INTENSIFICATION OF HEAT EXCHANGE IN TUBE BUNDLES IN A TRANSVERSE FLOW

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UDC 536.244

Data are presented from an experimental study of heat exchange involving reticular bundles of smooth and wavy tubes. A formula obtained for a modified Reynolds analogy generalizes published data for rough tubes and tubes with intensifiers.

Tube bundles located transversely to the flow are widely used in modern heat exchangers, and intensifying heat exchange in such bundles is an important problem. The relatively high level of turbulence caused by periodic disruptions of the flow make this problem very difficult to solve. The flow is two-dimensional in the case of flow over individual tubes in conventional staggered and corridor bundles, and turbulence is generated as a result of the nonuniformity of the velocity field in planes normal to the tube axis. If we create an additional nonuniformity in the direction of the tube axis, heat exchange is additionally intensified due to an increase in the turbulence level. Such additional nonuniformity is seen with the use of reticular bundles, i.e., bundles with a latticelike arrangement of the tubes. There has not been sufficient study of such bundles [1], and below we present the results of a study conducted on an experimental air unit of one type of reticular corridor bundle composed of straight and wavy tubes. It should be noted that, on the whole, there has not yet been any study of reticular bundles of wavy tubes.

The bundle (Fig. 1) was composed of tubes with an outside diameter d = 15 × 1 and served as the heating surface of a two-pass (air and gas) air heater. The spacings of the tubes across the bundle width $S_1 = 22$ mm, $S_1/d = 1.47$. The spacings through the depth (two intervals) $S_2' = 17$ mm, $S_2'' = 44$ mm, $\bar{S}_2 = 30.5$ mm, $\bar{S}_2/d = 2.03$. The presence of two intervals depthwise was decided upon the basis of considerations related to the design of the heat exchanger. Cold air was heated by air (gas) heated in an electric heater. The temperature of the air and gas was determined by averaging the readings of eight thermocouples installed in each case

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 45, No. 2, pp. 221-226, August, 1983. Original article submitted March 29, 1982.

0022-0841/83/4502-0868$07.50

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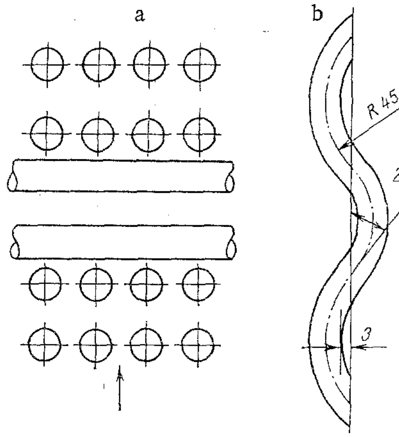

Fig. 1

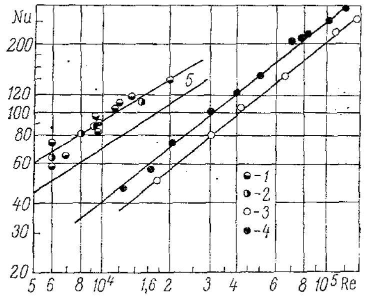

Fig. 2

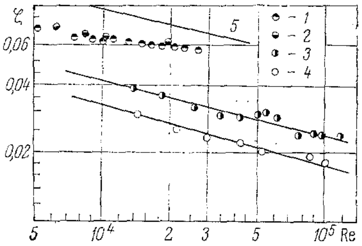

Fig. 3

Fig. 1. Types of heat-exchange surfaces: a) reticular corridor bundle; b) wavy tube.

Fig. 2. Heat exchange in bundles and tubes: 1) reticular bundle of straight tubes, outside flow; 2) reticular bundle of wavy tubes, outside flow; 3) smooth tube; 4) wavy tube; 5) corridor bundle [2].

Fig. 3. Hydraulic resistance of bundles and tubes: 1) reticular bundle of straight tubes, outside flow; 2) reticular bundle of wavy tubes, outside flow; 3) wavy tube; 4) straight tube; 5) corridor bundle [3].

at the cold- and hot-air inlet and outlet. The mean heat-transfer and hydraulic-resistance coefficients were found on the basis of the usual balance relations with allowance for the heating scheme. We took the velocity in the narrowest constricted section as the characteristic velocity. The physical constants were referred to the temperature of the flow.

The test results for heat transfer and resistance are shown in Figs. 2 and 3 in the coordinates ζ, Nu, and Re. Also shown are the standard relations for conventional corridor bundles of the same orientation as in the tests in [2, 3]. It can be seen from the figures that the heat-transfer coefficient increased by 28% and hydraulic resistance decreased by 20%. These findings have to do with the additional agitation of the flow, which helps delay boundary-layer separation on the aft part of the tube (as a result of which the resistance decreases) and, conversely, helps increase the rate of heat transfer by increasing mixing in the boundary layer and reducing the size of the separation zone. Given equal heat-transfer coefficients inside the tube bundle, the heat-transfer coefficient increases by 12%. This translates into an appreciable savings in metal.

Additional intensification may be achieved in the reticular bundle by increasing the heat-transfer rate inside the tubes. For this, it is best to use bundles of wavy tubes (see Fig. 1), with the slope angle of the waves β being made relatively small ($\beta = 26°$). This results in a negligible increase in resistance inside the tubes and a marked increase in heat exchange. The plane of bending of the tubes coincides with the frontal plane of the bundle. To obtain data on heat transfer and resistance for an air flow inside a wavy tube, an 850-mm-long segment of the tube was studied on a unit heated on the outside with boiling water. The unit and test method are detailed in [4]. The tests were conducted with the assistance of P. G. Bystrov.

Preliminary calibration of the unit (the obtaining of results on heat transfer and resistance for a smooth, straight tube) showed good agreement with the canonical relations (see Figs. 2 and 3). The test results for a single wavy tube are shown in Figs. 2 and 3. The heat-transfer data lies significantly below the data from [1] for a tube of the same geometry. (Careful calibration of the unit ensured that the results obtained in our tests are reliable.)

To theoretically substantiate the results on heat transfer and resistance inside the tube, we will examine the question of use of the Reynolds analogy for flow in tubes with flow separation [5]. Analysis of the flow patterns in wavy tubes obtained in a flow channel shows that local, conservative separations of short length are seen. These separations help increase turbulence in the flow and renew the boundary layer, which leads to an intensification of heat exchange.

As is known, the formula for the Reynolds analogy $Nu = (\zeta/8)Re$ was derived for media with $Pr = 1$ in the absence of a pressure gradient and, thus, in the presence of similitude of the velocity and temperature fields. This analogy does not hold for tubes with flow separation, and it is necessary to introduce corrections which take these circumstances into account.

Let us examine the velocity and temperature fields in a turbulent flow. Drawing tangents to the velocity and temperature profiles at $y = 0$ until they intersect the lines $u = u_m$ and $T = T_m$, we obtain certain dynamic and thermal boundary layers which are considerably thinner than the normal layers. By definition

$$\delta_t = \frac{T_m - T_{wa}}{\left.\frac{\partial T}{\partial y}\right|_{wa}}, \quad \delta_t = \frac{u_m}{\left.\frac{\partial u}{\partial y}\right|_{wa}}. \tag{1}$$

Considering that $q = \lambda(T_m - T_{wa})/\delta_t$, $\tau = \mu\partial u/\partial y$, $\tau = \zeta\rho\bar{u}^2/8$ and introducing the notation $(T_m - T_{wa})/(\bar{T} - T_{wa}) = k_t$, $u_m/\bar{u} = k_d$, after some simple transformations we obtain the formula

$$Nu = \frac{\zeta_{tr}}{8} Re \frac{\delta_d}{\delta_t} \frac{k_t}{k_d}, \tag{2}$$

expressing the generalized relation of the Reynolds analogy.

For gradientless flow and $Pr = 1$, $\delta_d = \delta_t$, and $k_t = k_d$, we obtain the usual formula of the Reynolds analogy $Nu = (\zeta_{fr}/8)Re$.

Thus, in the general case it is necessary to know the coefficient of friction resistance ζ_{fr} — which is not always equal to the total coefficient ζ — the ratio of the thicknesses of the thermal and velocity layers, found from Eqs. (1), and the ratio of the space factors of these fields.

Let us determine these quantities. Jet flow is realized in tubes with flow separations. In fact, the separated flow is the boundary of the jet, and it is governed by all of the laws appropriate to this type of flow. As is known, the velocity and temperature fields in jet flow are determined by the formulas

$$\frac{u}{u_m} = \exp(-\eta^2/2), \quad \frac{v}{v_m} = \exp\left(-\eta^2 \frac{1}{Pr_t}\right), \tag{3}$$

where $\eta = y/b$, $Pr_t = \mu_t/a_t$. In accordance with [3], we assume that $Pr_t = 0.55$. It follows from (3) that $\left.\frac{\delta_d}{\delta_t}\right|_{jer} \approx 0{,}75$.

Assuming that the jet model is realized in the case of rough tubes being considered, we have

$$\left.\frac{\delta_d}{\delta_t}\right|_{jer} = \frac{\delta_d}{\delta_t} = 0{,}75. \tag{4}$$

The space factor of the velocity profile (for rough tubes as well, in this case) is determined by the formula $u_m/\bar{u} = 1.325\sqrt{\zeta} + 1$.

Analysis of the test data for rough tubes shows that while the velocity field changes abruptly in relation to the degree of roughness, the temperature field is more conservative — the degree of its filling is practically no different from the case of a smooth tube. This conclusion is consistent with the theoretical results in [6], where it was shown that the pressure gradient has a slight effect on temperature profiles. In accordance with the above, we have the following for the parameter k_t/k_d

$$\frac{k_t}{k_d} = \frac{1{,}325\sqrt{\zeta_0} + 1}{1{,}325\sqrt{\zeta} + 1}. \tag{5}$$

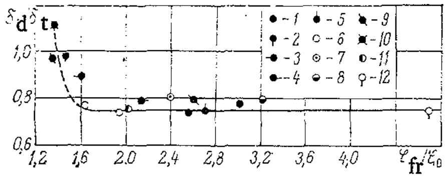

Fig. 4. Ratio of thicknesses of the thermal and dynamic boundary layers in rough tubes: 1) wavy plane channel; 2) tube with knurled projections, h/d = 0.0414, t/d = 8.8; 3) tube with knurled projections, h/d = 0.0527, t/d = 21.7; 4) tube with agitating rings, h/d = 0.04, t/d = 2; 5) tube with agitating rings, h/d = 0.04, t/d = 40; 6) tube with agitating rings, h/d = 0.04, t/d = 80; 7) tube with transverse projections, h/d = 0.036, t/d = 40; 8) coiled tube; 9) wavy tube; 10, 11) plane wavy channel; 12) tube with agitating rings, h/d = 0.04, t/d = 40 (h is the height of the projection, t is the distance between projections).

Figure 4 shows published data for rough tubes [7] and for the investigated wavy tube in the coordinates δ_d/δ_t, ζ_{fr}/ζ_0. The value of ζ_{fr} for the rough tubes was determined by the method described in [8], where the following formula was obtained for ζ_{fr}:

$$\zeta_{fr} = \frac{\zeta_{Re=5000}}{(Re/5000)^{0.25}}. \tag{6}$$

In the wavy tube investigated, the exponent of the Reynolds number in the formula for the resistance coefficient $\zeta = ARe^n$ is equal to $n = -0.27$, i.e., is close to the case of a smooth tube. This is evidence of the predominant effect of friction resistance (in the case of predominance of pressure resistance, the resistance coefficient is dynamically similar with respect to the Reynolds number). In connection with this, for the wavy tube, $\zeta_{fr} \approx \zeta$. It follows from Fig. 4 that at $\zeta/\zeta_0 > 1.5$ all of the test data corresponds well to the predicted value of the coefficient $\delta_d/\delta_t = 0.75$, i.e., the jet model is realized in this case. At $\zeta/\zeta_0 < 1.5$, the value of δ_d/δ_t is close to unity, and mixed flow is seen. The data for the wavy tube investigated is near this case ($\zeta/\zeta_0 = 1.35$, $\delta_d/\delta_t = 0.95$) and coincides with the results for rough tubes at similar values of ζ/ζ_0.

Thus, the completed analysis offers great confidence in the experimental data obtained on heat transfer and friction inside the tube, considering the contradictions present (between our results and the results in [1]).

To check the effectiveness of intensifying heat transfer both from outside the tube bundle and inside the tubes, we studied a reticular bundle composed of the above-described wavy tubes. The mean spacings of the tubes were kept the same as in the investigated reticular bundle of straight tubes. In determining the heat-transfer coefficient, heat exchange inside the wavy tubes was determined from Fig. 3.

The test results for the wavy reticular bundles are shown in Figs. 2 and 3. It is apparent that tube waviness has almost no effect on the heat-transfer rate from outside, since the flow is transverse to the tubes. The heat-transfer coefficient increases by a total of 23% compared to the usual corridor bundle of straight tubes. Thus, the use of a reticular bundle of wavy tubes significantly decreases tube consumption (by 23% for an air preheater and by 28% for an economizer). The technology of manufacture of the wavy tubes is not complicated.

NOTATION

d, outside diameter of tubes in the bundle; S_1, transverse spacing of the bundle; S_2, longitudinal spacing of the bundle; δ_t, thickness of the thermal boundary layer; δ_d, thickness of the dynamic boundary layer; q, heat flux; τ, friction stress on the wall; ζ, resistance coefficient; Nu, Nusselt number; Re, Reynolds number; b, jet width; h, height of projection; t, distance between projections; Pr_t, turbulent Prandtl number.

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FORCE ACTION OF A SUPERSONIC DUSTY GAS FLOW ON A BLUFF BODY

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UDC 532.5

Results of a numerical computation of the dusty gas flow around a spherical segment are discussed.

1. Gas deceleration and subsequent energy and momentum redistribution between the particles and the gas occur behind the bow shock in the supersonic flow of a dusty gas around a bluff body. A result is particle retardation and gas acceleration, leading to an increase in the force effect of the gas on the body and a diminution in the action of the particles. The intensity of the exchange depends substantially on the dispersion of the solid phase: For particles of the fine fraction this exchange is realized in a narrow zone adjoining the shock, and in the major part of the perturbed flow the particles and gas are in thermal and kinematic equilibrium; the particles of the coarse fraction are incident on the body surface almost without any change in the kinetic energy they possessed in the free stream. The computation of the force effect is simplified in both limit cases: In the first case, for equilibrium flow behind the shock, it is sufficient to perform the computation of the flow by a pure gas with certain effective parameters dependent on the mass fraction of the solid phase [1]; in the second case summation of the effects of the gas and the particles, determined without taking account of their interaction, is possible. In addition, results of computing the flow of a dusty gas around bluff bodies, obtained taking account of particle deceleration in the shock layer without the reverse influence of the particles on the gas motion ([2] and the bibliography in [3-5]) are presented in the literature. Analysis of a dusty gas flow in a shock layer with interaction between the gas and the particles taken into account is presented in [3-5] without revealing the general regularities of the phenomenon.

Results are discussed below of a numerical computation of the force effect of a dusty gas on a bluff body as a function of the mass fraction and disperseness of the solid phase. Data are presented on the influence of gas and particles separately on an integral force effect on a body, and of the total effect of a dusty gas which would permit estimation of the accuracy of the approximations described above in a specific example.

2. Let us consider the dusty gas flow around a sphere on the basis of a model of a two-velocity and two-temperature continuous medium [6]. Within the framework of this model, a monodisperse cloud of solid particles is considered as a gas deprived of intrinsic pressure.

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Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 45, No. 2, pp. 226-231, August, 1983. Original article submitted January 4, 1982.

0022-0841/83/4502-0872$07.50

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